Plate heat exchangers for refrigeration applications

Technical reference manual

A Technical Reference Manual for Plate Heat Exchangers in Refrigeration & Air conditioning Applications

by

Dr. Claes Stenhede/Alfa Laval AB

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Alfa Laval AB

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Foreword

The brazed plate heat exchanger was developed by Alfa Laval in the late seventies. It has increasingly proven useful in applications where compact, rugged heat exchangers with high heat transfer coefficients and high heat recovery are needed.

One of the most important applications is refrigeration system, the topic of this manual.

This is the fifth edition of the manual. Since the first edition, there has been an intense debate about refrigerants, both new, with and without glide and natural, such as propane and ammonia. This debate has influenced the development of equipment including plate heat exchangers.

One of the effects on PHE's has been that the old boundary between the small BPHE for halogenated hydrocarbons in direct expansion systems and the large semi-welded PHE for ammonia in flooded systems has disappeared. Ammonia is increasingly used in direct expansion systems in nickel brazed PHE's and there are large brazed PHE's suitable for flooded systems.

Hence, this new edition deals not only with brazed PHE's but also with the larger welded and semi-welded PHE's. After all, the thermal and hydraulic properties are the same for all PHE's, from the smallest brazed to the largest gasketed.

The purpose of the manual is to provide an understanding of the properties of the brazed and semi-welded plate heat exchanger in order to design, install, operate, and troubleshoot them correctly

This Technical Reference Manual is divided into two sections. The first - the present booklet - contains general information about BPHE's in refrigeration systems. It comprises eight chapters and four appendices. Although all the chapters can be studied individually, some cross-references are made.

The second part - the product catalogue - contains specific data about the product range.

For the content of this manual, I have had the invaluable help of colleagues within Alfa Laval, in both Italy and abroad, but especially Bernard Pasquier, Loris Sartori, Mats Stromblad and Alvaro Zorzin to whom I express my thanks.

Finally, I thank an understanding wife and son for their patience during all the working evenings. Especially Eric thinks now that a father comes with an integrated laptop.

Alonte, Italy, February 2nd, 2004.

Claes Stenhede

1. Applications. The Basic Refrigeration cycles and Applications.

1. The pressure-enthalpy diagram.

The thermodynamic properties of a refrigerant are often represented in a pressure-enthalpy diagram. In this, the logarithm of the pressure is plotted as a function of the enthalpy, with the various properties as parameters. See figure 01. The main components are:

- The solid line represents saturated liquid and the dotted line saturated vapour. Instead of pressure, the saturation temperature could be used. The two lines meet at the critical point marked by a circle. The difference in enthalpy between the two lines for a given pressure represents the latent heat.

  The area to the left of the black line represents subcooled liquid and the area to the right of the black double line superheated vapour. In between is a mixture of saturated liquid and vapour.

- Isoconcentration lines show the lines for equal vapour fractions of the liquid-vapour mixture.

- Isotherms represent the temperatures of the subcooled liquid and the superheated vapour

- Isentrops represent changes with no heat exchange to the surroundings, e.g. compression of the refrigerant.

- Isochors (constant volumes) are sometimes included.

2. The basic cycle.

Figure 1 shows the basic refrigeration cycle, both in an enthalpy-pressure diagram and as the physical components. To study the cycle we can work it through, starting at any point but a good starting point is usually the slightly subcooled liquid refrigerant, 35 °C at a pressure of 15.33 bar, i.e. a saturation temperature of 40 °C. This is point A in the figure.

This starting point is suitable because it normally changes very little, regardless of the modification of the basic cycles, which are described later.

A - B. The liquid expands in the expansion valve. No energy - thermal or mechanical - is exchanged with the surroundings; the expansion is isenthalpic. It is represented in Fig. 01 with a straight perpendicular change of state.

When the pressure starts to decrease, at first nothing happens; the temperature remains (almost) constant. The liquid reaches its saturation curve, though. A further decrease in pressure means that the temperature has to decrease as well; otherwise, the liquid would be superheated, which is a thermodynamically unstable state. The liquid is thus cooled down and the energy released evaporates a part of the liquid or, expressed differently, the evaporating liquid cools down the remaining liquid. The lower the pressure, the more liquid evaporates.

B. The liquid has reached the final pressure and the fraction evaporated can be found from the lines for constant vapour fraction. In the example, the refrigerant has expanded to 1.63 bar/-30 °C with 33.9 % vaporized.

B - D. The partly vaporized refrigerant enters the evaporator. Here the remaining liquid refrigerant evaporates, thereby producing the required cooling effect. The refrigerant at first reaches point C, as 100 % saturated vapour, but usually leaves slightly superheated at D.

D. The vapour leaves the evaporator at 1.63 bar/-30 °C and superheated to -25 °C.

D-E. The vapour is compressed to the condensation pressure in the compressor. The compression should if possible be ideal, i.e. mechanical but no thermal energy is applied to the vapour, until the pressure has reached the required level, in the example 15.3 bar/40 °C.

If so, the vapour should move along the isentrop D - E'. Note the difference to the expansion A - B. This has no energy exchange with the surroundings, thus it moves along the isenthalp. Here there is an addition of mechanical, but not thermal, energy; thus, the vapour moves along the isentrop. With compression, the temperature increases, as the diagram shows. The increase of the temperature is ahead of the pressure increase, i.e. the refrigerant not only remains as a vapour but also superheats.

However, the compression is not ideal. There is internal friction between the moving parts of the vapour, friction energy in the lubricating oil, and compressed vapour moves back to the entrance, etc. All this means an extra addition of heat to the vapour. The vapour thus does not move along the isentrop D - E' but along an undefined path to the final higher temperature at E. This added energy depends on the compressor efficiency h. Thus:

$H_E - H_D = (H_{E'} - H_D) / h$ (The actual compressor power)

With knowledge of h (from the manufacturer), $H_{E'}$ & H_D (diagram), H_E is calculated and, together with the end pressure, the actual exit temperature is found (diagram).

E - F. The superheated vapour leaves the compressor at a fairly high temperature. This vapour represents energy, which is too valuable to waste. Thus, the vapour can be desuperheated in a special heat exchanger and the heat used for hot water production or room heating.

F - A. The vapour enters the final condenser, probably a little superheated (a little to the right of point F), and condenses. The condensate normally does not leave exactly saturated, but a little subcooled and we are back to the starting point A, 15.33 bar/40 °C, subcooled to 35 °C.

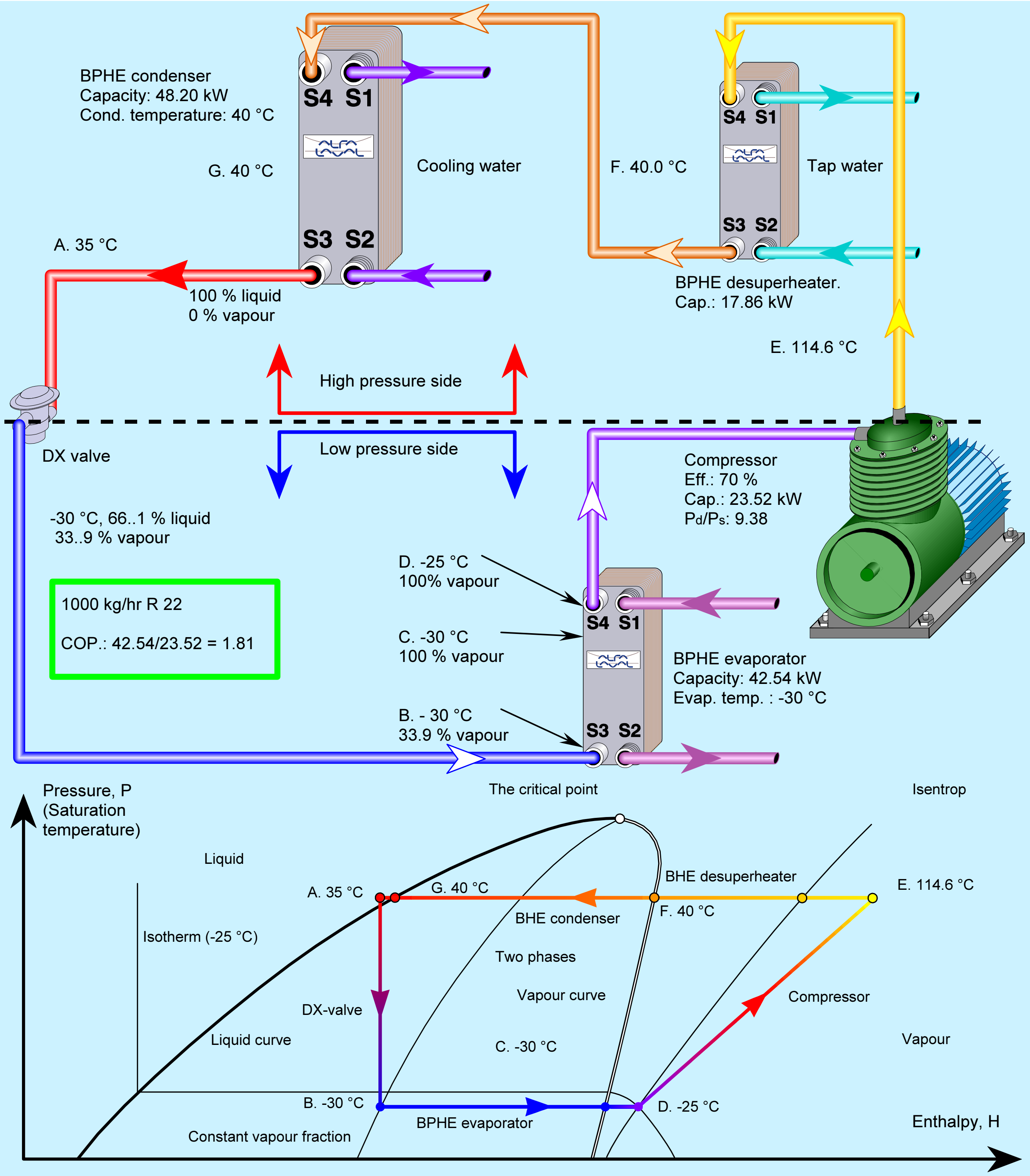


Fig. 01. The basic refrigeration cycle.

The function of the refrigeration plant is to remove heat from a process fluid or air at a low temperature and dump it in recipient that can be water or air.

The figure shows a schematic cooling plant, composed of an evaporator, a compressor, a condenser, an expansion device and connecting pipes. These are the minimum components necessary in the basic compression refrigeration cycle.

The pressure is shown as a function of the enthalpies of liquid and vapour. To the left of the liquid line is liquid and to the right of the vapour line, vapour. Between the two lines is a two-phase area. The lines meet at the critical point. Other properties can then be plotted as parameters, e.g. isotherms, lines of constant temperature. In the figure, the -25 °C isotherm is shown. It is almost vertical in the liquid area, mirroring the fact that the liquid specific heat is little pressure dependent. In the vapour area it is curved and inclined, i.e. the vapour specific heat is strongly pressure (and temperature) dependent.

The figure also shows an isentrop, a line expressing a change of state, but where no heat energy is exchanged between the fluid and the surroundings. An ideal compression would follow this line (D - E´), but because of the in evitably released friction energy, a real compression is (D - E), i.e. to a higher final temperature.

3. Intercoolers & subcoolers.

Very often a plant has access to well water. This water is normally too expensive to use directly but can be used as make up for the cooling tower water. As the well water is normally much cooler than the cooling tower water, it is a waste of cooling capacity to use it directly. Instead, it could be used for minor cooling duties, but where as low a temperature as possible is required.

Two possibilities are shown in figure 02. In order to compare this system with the basic refrigeration cycle, the condensing and evaporating temperatures, as well as the circulating amount of refrigerant, are the same.

This system uses a DX evaporator (**4. Evaporators and separators, §5.**) but a thermosiphon evaporator (**4. Evaporators and separators, §3.**) could be used as well. The thermodynamic is the same with one exception; the vapour emerges normally saturated from a thermosiphon not superheated. To be able to compare the cycles we have to assume that the vapour in both cases superheats 5 K on the way to the compressor and that this heat comes from the liquid to be cooled.

In order for the expansion valve to operate properly, the LR pressure must be controlled, **5. Condensers & Liquid Receivers.** figure 10.

Note that in case of a thermosiphon system there should be a system to recover oil. Such a system is discussed in **8. Oils & Refrigerants**.

This cycle is equipped withy a condensate subcooler and an intermediate vapour desuperheater, both cooled by well water.

The effect of these two measures is a reduction of the exit temperature to about 86 °C, well below the danger of oil breakdown, but still interesting as a hot water source, albeit of a lower capacity and at a lower exit temperature. Furthermore, the cooling factor is increased by about one third from 1.81 to 2.40. This is due to three effects:

- The lower temperature to the expansion valve means that less refrigerant has to evaporate in order to reach the evaporation temperature of -30°C.

- Consequently, more liquid refrigerant is available in the evaporator, and its capacity increases. The increased efficiency of the compressors means that less compressor power is necessary.

- The colder vapour to the second stage further reduces the compressor load.

In the examples with a circulating amount of 1000 kg/hr R 22, the cooling capacity increases from 42.54 kW to 48.53 at the same time as the total power consumption decreases from 23.52 kW to 20.2 kW.
These thermodynamic effects can be used either to keep:

- the evaporator capacity but lower the compressor load,

- the compressor load and increase the cooling capacity

The actual performance of both the basic cycle and of any of the improved cycles obviously depends on the total required capacity, the efficiency of the compressors for the actual operating conditions, type of condensers and evaporators, availability of water, available space, etc.

4. The components.

4.1. The expansion process & devices.

In all types of the refrigeration cycle there is a step where liquid refrigerant is expanded from the condensing pressure to the evaporating pressure, from A to B in Fig.01. Refrigerant evaporates and simultaneously cools down to the evaporation temperature.

The expansion device could be any type of restriction which exerts a pressure drop on the fluid which is equal to the difference between the condensation and evaporation pressure for the actual refrigerant flow rate.

Fluid dynamic theory teaches us that the pressure drop

increases with:	Increasing	flow length
		surface roughness
		direction changes
		de/accelerations
	Decreasing	hydraulic diameter.
		Cross-section area.

Theoretically, all these parameters could be used to control the pressure drop but it is difficult to imagine a device, which uses the surface roughness. Similarly, the number of directional changes and the flow length are closely interrelated as are the hydraulic diameter, the cross section area and the de/acceleration.

There are three methods to control the pressure drop:

- ***A valve.*** A multitude of designs exists, but the basic parameter, which is changed, is the cross section and thus the velocity. Simultaneously the hydraulic diameter changes, the flow changes direction a little and the necessary acceleration energy is not fully recovered when the flow finally decelerates.

  Flow length and roughness have little importance.

- ***An orifice***. The same principle as a valve but fixed.

- ***A capillary tube***, which is a thin long tube usually spiral wound to save space. All the parameters above are important and have to be balanced with each other.

  Compared to an orifice, which is a hole in a wall, the cross section can be made larger as the flow length plays an important part for the pressure drop. This is important for small capacity devices, where an orifice would require such a small hole that it could easily be clogged by wear and tear products.

  It is usually a fixed design used for small systems such as refrigerators.

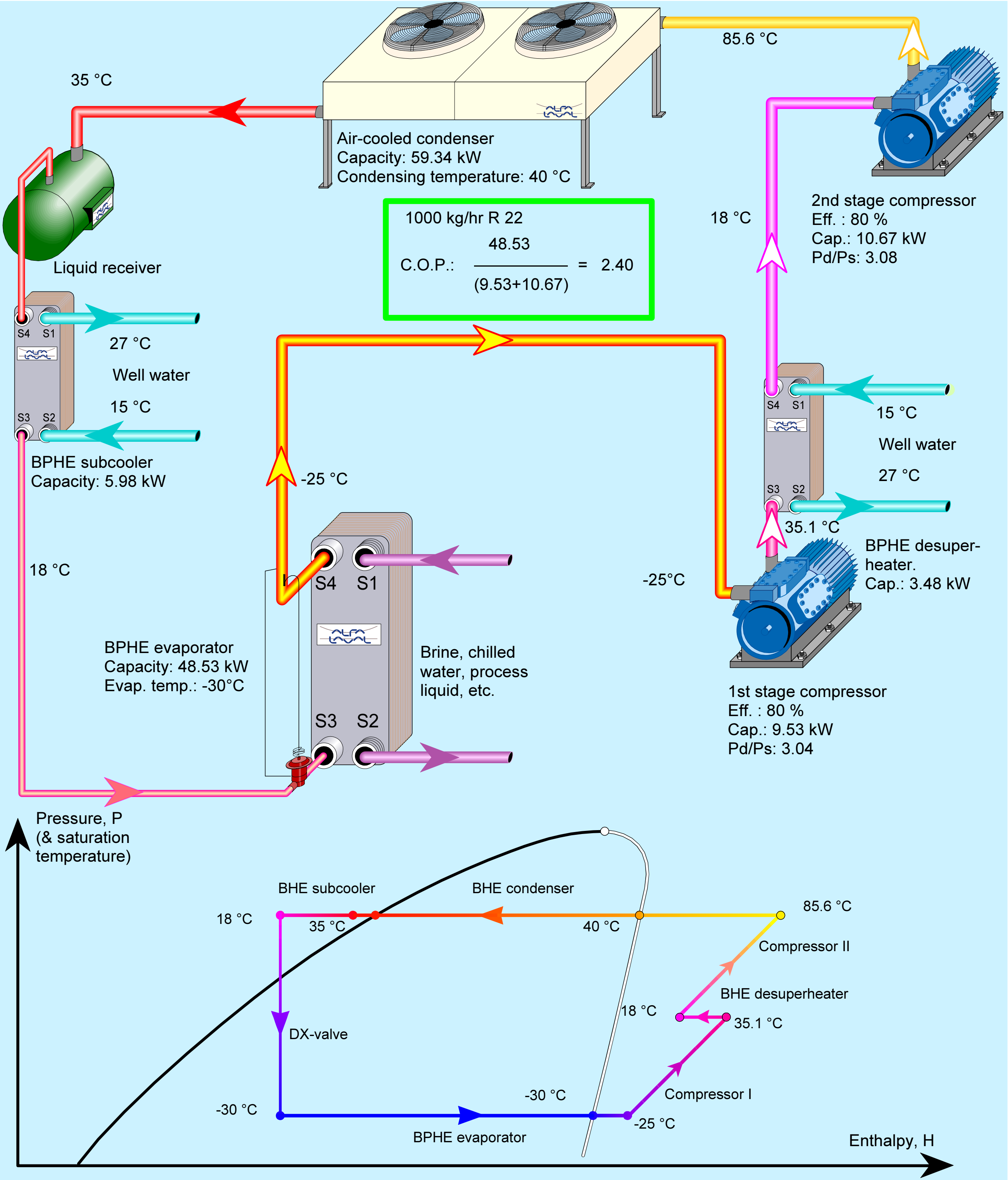


Fig. 02. The intercooler & subcooler refrigeration cycle.

This is principally the basic refrigeration cycle, but with a subcooler added to the main condenser and an intercooler between the compressor stages. The addition of these external coolers increases the cooling factor to the second largest of the discussed cycles and gives a high cooling capacity for a given amount of refrigerant.

An air-cooled condenser is used to desuperheat and condense the refrigerant. The refrigerant leaves subcooled to 35 °C. It is then further subcooled to 18 °C with well water of 15 °C in the BPHE.

A further improvement is the use of a two-stage compression, either with two compressors in series or in a two-stage compressor. The intercooler between the compressor stages is only possible for a compressor where the first-stage vapour leaves the first stage and then enters the second stage possibly together with additional intermediate pressure vapour, i.e. not for a compressor with only an intermediate vapour inlet.

An optimum intermediate pressure is normally the geometric mean of the end pressures, here around 0 °C.

The first stage vapour leaves at about 35 °C and is cooled in the BPHE to about 18 °C, also with well water. The higher vapour density increases the amount of refrigerant in the second stage, i.e. a higher capacity. The higher vapour density decreases the refrigerant volume in the second stage, i.e. a smaller compressor.

From the expansion device, a two-phase liquid-vapour mixture emerges. The liquid part has to evaporate in the evaporator, where it supplies the cooling effect. There are two ways to do this:

- ***A direct expansion evaporator.*** All the mixture enters the evaporator, the liquid part evaporates and a pure vapour leaves the evaporator and enters the compressor. To ensure that no liquid droplets enters the compressor, where they could cause liquid hammering and damage, the vapour has to be somewhat superheated. The superheat is also an excellent variable for the control of the expansion valve.

  Of special importance is the thermostatic expansion valve, which is controlled by the vapour superheat from the evaporator. As this is of prime importance for the function of a direct expansion evaporator, its function is treated in the evaporator chapter.

  The electronic expansion valve is increasingly common. The advantages, flexible control parameters and remote control, make it is very popular in supermarkets with numerous air conditioning units and refrigeration duties at different temperature levels. The most dependable valves are equipped with a heat motor, but pulse modulated and step motors are also used.

- ***Flooded flow.*** The vapour and liquid are separated, usually in a vessel large enough for the liquid droplets to settle by gravity or, sometimes, by centrifugal force.

  The liquid together with recirculated liquid enters the evaporator, partly evaporates, and the emerging vapour-liquid mixture re-enters the separator, where the vapour and liquid separate.

  The circulation can either be natural - ***thermosiphon*** - or by a pump - ***forced circulation***.

  The vapour joins the expansion vapour and leaves for the compressor. The liquid mixes with the expansion liquid and re-enters the evaporator.

  The vapour from the evaporator is neither dry or superheated. Thus, there is no superheat to be used as a control variable. Expansion is controlled by the level in the separator, instead. If the level decreases, the expansion valve opens and more refrigerant enters the separator-evaporator loop. See figure 02

4.2. The compressor.

The vapour from the evaporator is compressed from the evaporation pressure to the condensing pressure. The compressor is mechanically complicated but its function is simple and easy to monitor and it is usually dependable. Normally the worst thing that can happen is liquid hammering, equivalent to knocking in a motor.

Different compressor types respond differently to liquid hammering, the most sensitive being the piston and the scroll compressor. The least sensitive is the screw compressors, possibly because of the large oil content.

4.3. Evaporators, condensers and interchangers.

This manual deals exclusively with plate heat exchangers and especially brazed plate heat exchangers, where one fluid is a one or two-phase refrigerant and the other is a liquid for cooling or heating or a refrigerant.

The three plate heat exchanger types used in refrigeration, brazed, welded and semiwelded, have similar thermal and hydraulic properties and what is said in this manual is basically valid for all three types.

The differences are mainly in the corrosive properties of the materials, different qualities of stainless steel, titanium and rubber for the SWPHE, stainless steel or titanium for the AWPHE and stainless steel and copper or nickel for the BPHE. Moreover, the liquid side of a SWPHE can be opened for inspection and cleaning while this not possible for AWPHE and BPHE.

4.4. The interconnecting pipe work.

The interconnecting pipe work with its various fittings and valves is often overlooked when designing a cooling system. This is unfortunate, since an ill-considered layout can spoil the most intricate design.

The pipe between the TEV and the evaporator and the pipe between the condenser and the liquid receiver stand out as potential trouble spots and these are covered in the chapter on evaporators and condensers.

Similarly, ill-conceived shut-off valves can be sources of trouble in thermo siphon system

5. The flash economizer cycle.

In the basic cycle, the refrigerant can be divided (internally) into two parts:

1) One is expanded and evaporated from the condensation to the evaporation pressures, here from pressures corresponding to 40 °C and -30 °C.

2) The other part remains liquid and cools from 35 °C (it is already cooled 5 K in the condenser) to -30 °C by the evaporated part.

The evaporated part 1), -35 °C, thus cools the liquid part 2) all the way from the high 35 °C to the low -30 °C.

Thermodynamically it is a waste of energy to cool the high temperature part with a low temperature vapour, as this vapour must subsequently be compressed again from a low -35 °C to a high 40 °C. It is better to flash a part of the refrigerant 1) to an intermediate pressure, say 0 °C, and use this to cool the upper part of the refrigerant 2).

Instead of compressing all the vapour in 1) from -35 °C to 40 °C, a part now has to be compressed from only 0 °C to 40 °C, a save of energy. The cooling factor increases and the discharge temperature from the second compressor stage decreases. See figure 03.

This first cooling - with refrigerant at 0 °C - is made in the economizer. This can be either an internal cooling - the flash economizer, figure 3 - or in a separate heat exchanger - the evaporator economizer cycle, figure 4.

The somewhat wet vapour from the economiser, figure 3, enters the intermediate stage of a two-stage compressor, where it mixes with and cools the vapour from the first stage.

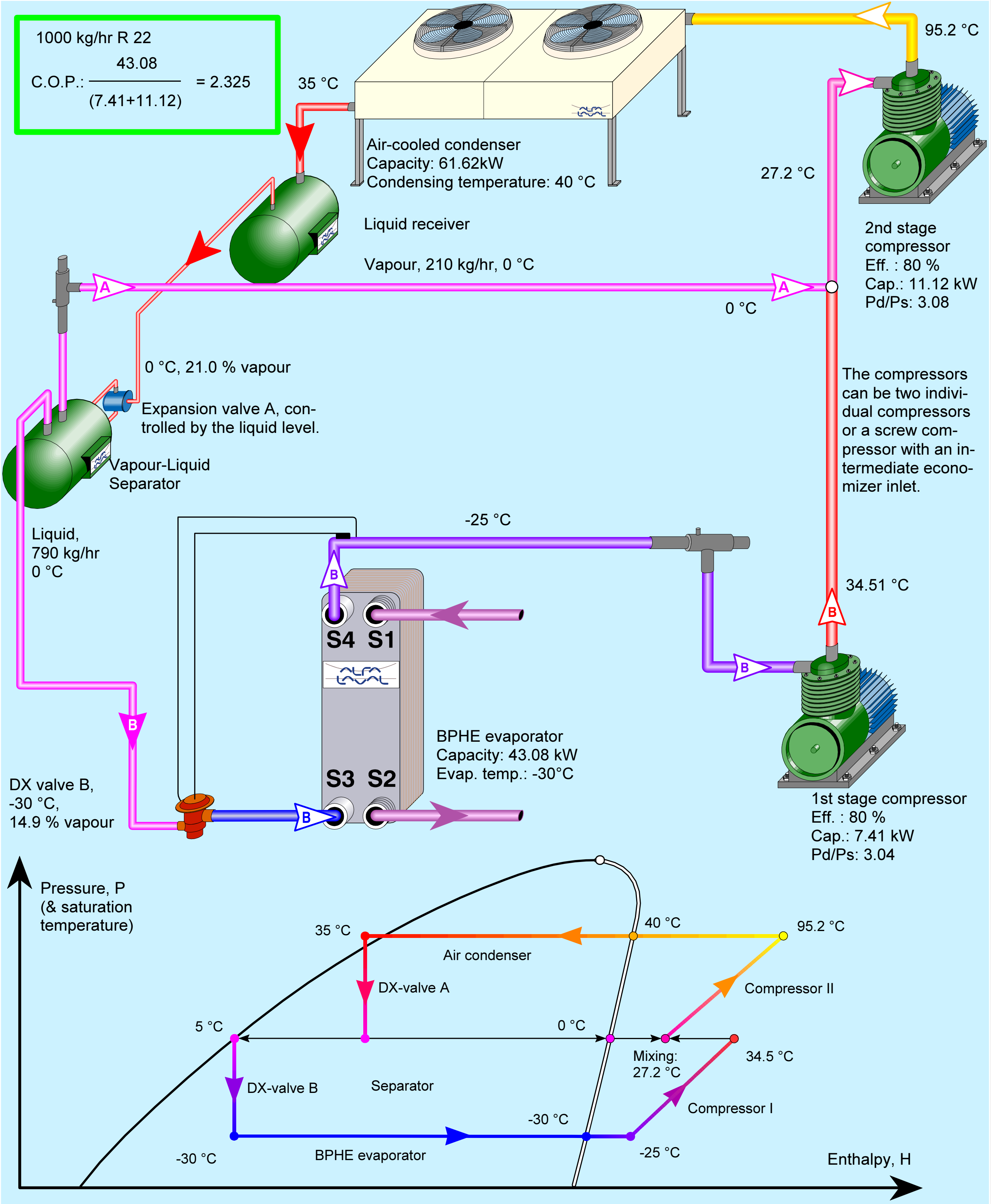


Fig. 03. The flash economizer refrigeration cycle.

The entire refrigerant is flashed to an intermediate pressure and the vapour/liquid is separated in the economizer separator. The vapour from the economizer then mixes with discharge vapour from the first stage, but as the HP vapour is dry and thus has less cooling capacity, the mixing temperature is higher.

The major difference to the heat exchanger economizer cycle - figure 04, p. 9 - is that flashing in a separator rather than sensible cooling in a heat exchanger, thus the cost of a separator versus a heat exchanger, cools the liquid here.

This separator tends to be rather bulky and it seems to be a trends towards the more compact evaporator economizer. The PHE, compact and efficient, has its place here as shown in figure 04.

Thermodynamically both operate with the same principle. It is better to do the upper part of the condensate cooling with a high-pressure vapour, which is less costly to compress again.

See explanation in the text, **§5. The flash economizer cycle**, p. 6.

6. The evaporator economizer cycle.

See figure 4.

7. The real versus the ideal refrigeration cycle.

The flow sheets described here show ideal refrigeration cycles, with the exception of the compression where normally the actual shaft power is included.

In a real plant there are, however, pressure drops in the various parts of the system. The expansion is to a slightly higher temperature than -30 °C, say -29 °C. The refrigerant then enters the evaporator at this temperature, expands and leaves at a pressure corresponding to -30 °C.

Likewise, there are pressure drops in the condenser, pipes, valves, fittings etc., which decreases the real performance.

These pressure drops can be minimized or have no importance, e.g. it does not matter if the expansion is entirely in the expansion valve or a part of it occurs in the evaporator (before the entrance to the heating area).

8. Definition of capacity and coefficient of performance.

When defining capacities and efficiencies it is important to differentiate between the ideal cycle performance (actually semi-ideal as the actual shaft input power to the compressor is normally used) and an actual plant with its various pressure drops, energy losses, electricity demands of various auxiliary equipment, etc.

Normally the power demands of pumps such as for glycol and cooling water are not included as these depends on external factors, irrelevant to the evaluation of the performance of the plant. However, at least the internal power consumption (in the condenser/evaporator and piping) should be included.

When comparing efficiencies it is important to know exactly what is included and under what conditions the unit is operating. Obviously, an air-conditioning unit installed outside, unprotected from the sun, in Saudi Arabia could give very different result from a protected unit in Japan, even though both the chilled and the cooling water have the same temperature levels.

Refrigeration (Heating) capacity: The heat removal (addition) from a medium, e.g. a glycol solution.

Refrigeration (Heating) effect: The heat transfer to (from) the refrigerant itself. The difference to the above might be losses to the surroundings, additional heat input from auxiliary equipment, etc.

Coefficient of performance, COP. This is divided into various subdefinitions:

COP, (semi-ideal): Refrigeration effect of the evaporator or heating effect of the condenser divided by the shaft power to the compressor (in hermetic compressors sometimes the electricity input to the motor) for given conditions. The given conditions include suction & discharge pressures, superheating, subcooling, compressor efficiency, cycle type.

The semi-ideal COPs are used by designers to evaluate a new refrigerant, the effect on the performance by addition of a condensate subcooler, an economizer, etc. The cycles presented in this chapter use this COP. It is, thus, easy to see what the addition of an economizer or a condensate subcooler would mean.

COP, plant: Refrigeration or heating effect of the plant divided with the total energy input to the plant, in consistent units for specific conditions.

The plant COPs are used by the operators of a plant. A potential customer of a residential heat pump, comparing various machines, is probably only interested of how much heat a unit delivers at a given temperature, for a given heat source and electricity consumption.

Whether a heat pump achieves a certain COP by addition of an economizer or a separate subcooler or a larger evaporator is of little importance to the final customer.

Compression efficiency. The ideal adiabatic work (as found from the diagrams) of the compression of a volume of vapour to the actual work delivered by the piston or rotors of the compressor.

Mechanical efficiency. The ratio of the work defined above to the actual shaft input work.

9. Improvements of the cycles and the use of PHE in these.

The exit temperature in the basic refrigeration cycle above is about 115 °C. This temperature, though excellent for heat recovery, is the one at which oil starts to break down. The cooling factor - the ratio of the evaporator capacity to the compressor capacity - as well as the heating factor - the ratio of the condenser capacity to the compressor capacity - are also low.

Another consideration is the oil flow in the system. Oil leaves the compressor with the discharge vapour and has to be recovered, and frequently it has to be cooled.

A third consideration is compressor protection. The compressor needs a certain superheat of the suction vapour and should not discharge vapour that is too superheated.

The basic cycle can be improved by various methods. We have already seen what the simple addition of a condensate subcooler and vapour desuperheater means. The duties where BPHE can be used include:

- ♦ Evaporators, direct expansion or flooded
- ♦ Intermediate evaporators, which cools a brine used for cooling air in a unit cooler,
- ♦ Condensers,
- ♦ Intermediate condensers, which are cooled by circulating water cooled by a liquid cooler, the real heat dump.
- ♦ Oil coolers & evaporators,
- ♦ Desuperheaters & subcoolers.

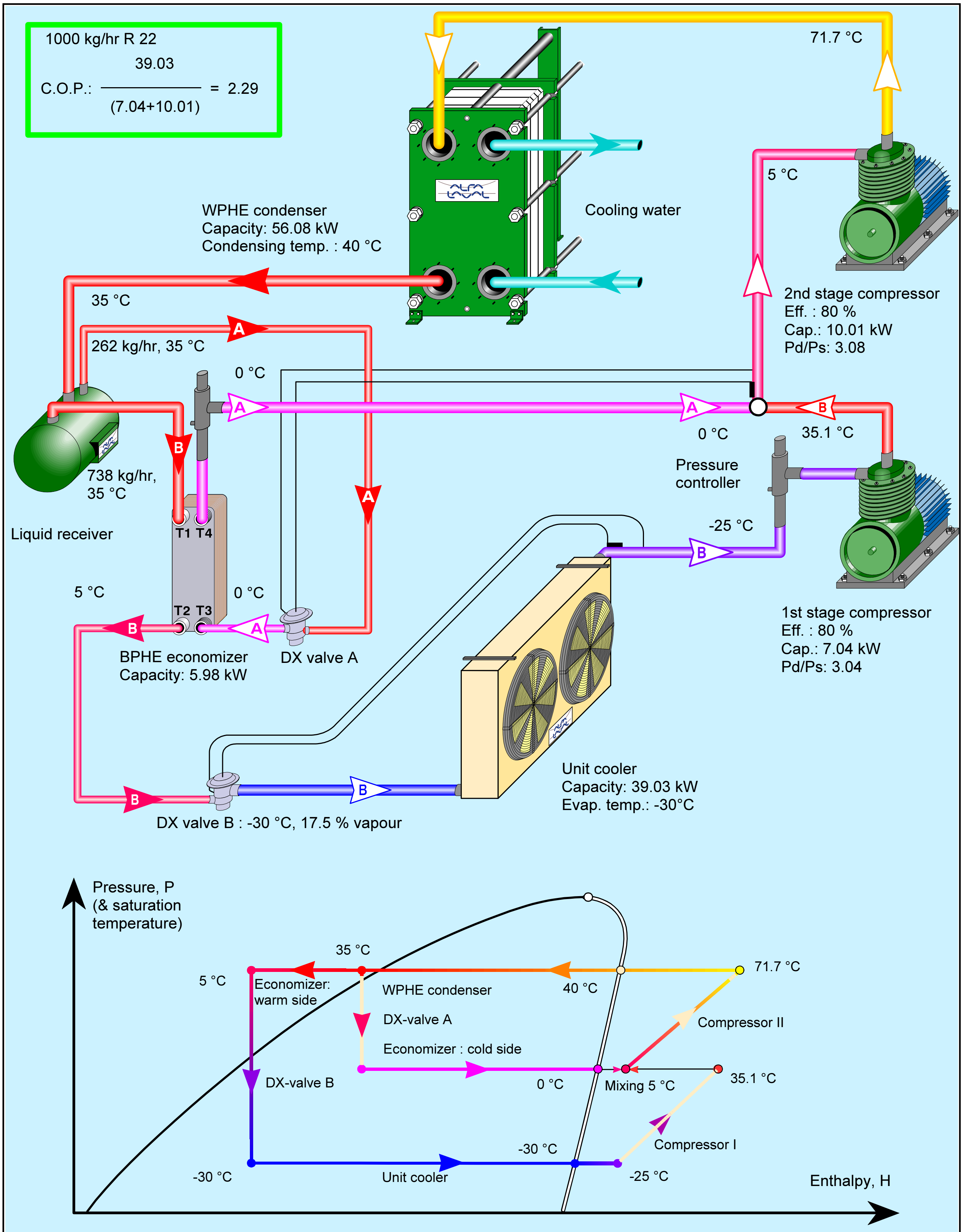


Fig. 04. The evaporator economizer cycle.

A part of the condensed refrigerant is flashed and used to cool the bulk of the refrigerant in an economizer. The evaporating refrigerant leaves the economizer slightly wet. When mixed with refrigerant from the first compressor stage, the remaining droplets evaporate at the same time as the first stage vapour is cooled. The result is a lower compressor discharge temperature from the final stage and a better cooling factor.

The improvement is due both to a thermodynamically more effective process and to improved compressor efficiencies. The TEV A can of course only be controlled by the temperature after the mixing if the compressor arrangement allows it. It has a similar effect as water-cooled desuperheater between the stages, but with the advantage that the plant is not dependent on an external water source.

- Subcoolers/superheaters in low temperature systems,
- Economizers,
- Condensers/evaporators in cascade systems,
- Heat recovery in general,
- as well as numerous applications not directly connected to the refrigeration plant.

10. The subcooler/superheater cycle.

What happens when the superheat after an evaporator and thus the suction temperature increases?

- The vapour density decreases and for a compressor with a fixed swept volume, i.e. most compressor types, the refrigerant mass flow decreases, consequently with a decreased capacity.
- If all the increased superheat enters the evaporator, the capacity increases.

Which of the two effects takes overhand depends on the latent and specific heats of the refrigerant:

- For a refrigerant with a high vapour specific heat, the decrease of the circulating flow is more than offset by the increased capacity due to the higher superheat. The sensible heat increases and the capacity thus increases with the increasing suction temperature.
- For a refrigerant with high latent and low specific heats, the increased superheat cannot make up for the loss of capacity due to the decreased refrigerant flow. The latent heat part decreases and the capacity thus decreases with increasing suction temperature.

At evaporation temperatures around 0°C,

- Ammonia looses capacity.
- R22 maintains the capacity.
- Propane, Propene, Butane, R134a, R410a, and R407c give some improvement.
- R404a and R507a show fairly large improvements.

Compressor manufacturers sometimes give the capacity at 25 K superheat. This higher capacity is then only valid if ***all the superheat is provided by the evaporator***. Superheat after the evaporator in the suction pipe, is a pure loss. However, is 25 K superheat in the evaporator in most cases feasible? There is another possibility, though, see figure 05.

Instead of superheating the vapour in the evaporator, the vapour superheats in a separate exchanger by subcooling the condensate. The colder condensate then does not have to evaporate to the same extent in the expansion valve. Thermodynamically the two methods are equal but there are some important differences:

- The evaporator now operates with a normal superheat of some 5 to 10 K.
- An extra heat exchanger is necessary. The pressure drop in this can obstruct the positive effect of the superheating/subcooling.
- The evaporator operates with a lower inlet vapour fraction and a lower mass flow, which can decrease the K-value.
- It is possible to control the evaporator with the superheat ***after*** the interchanger. The evaporator can then operate with a much lower superheat or even wet without danger of liquid hammering. The MTD and the K-value increases as well. This should be done with an electronic expansion valve, though. Normal TEVs are apt to be unstable at this use.
- A BPHE, with its close temperature approach, is a good choice for the interchanger. ***If only a few degrees superheat is necessary, it will be too large.***

Common for both methods are:

- The circulating refrigerant flow decreases, important in cases of long refrigerant lines.
- The compressor operates at a considerably higher temperature than the evaporator. This eliminates the risk in a two-stage system that oil suitable for the high-pressure stage will become too sticky and lose its lubricating ability at the temperatures of the low-pressure stage. Moreover, at very low temperatures, the compressor materials could become brittle.
- Oil droplets are effectively stripped of its refrigerant content. Oil containing too much refrigerant could otherwise impair the compressor lubrification. This is mainly a problem for refrigerants with dissolves oil well, such as butane, propane, propene and the fully chlorinated hydrocarbons.

The discharge temperature from the compressor increases. This could be an advantage in a heat pump, which heats tap water by desuperheating the discharge vapour but it could also lead to unwanted high temperatures. The vapour in the intermediate stage then has to be cooled, either by water, condensate injection or a combination, see figure 5.

Example. A compressor, with a swept suction volume of 1 m^3/s, compresses propene of a saturation temperature of 3.5 °C and a superheat of 5 K. The condensation temperature is 43.5 °C and the condensate leaves at 40.5 °C. The vapour density is 13.291 kg/m^3. The enthalpy difference between the evaporator inlet & exit is 283.08 kJ/kg giving a capacity of 13.291*283.08 = 3762.4 kW.

The evaporator is changed and superheats the vapour to 23.5 °C. The vapour density decreases to 12.322 kg/m^3. If the swept volume is the same, the refrigerant flow decreases to 12.322/13.291 = 92.7 % of before. The refrigerant heats up 15 K more which means an increase of the enthalpy difference with 308.33 kJ/kg, i.e. with 8.9 %. Despite the lower vapour flow, the capacity increases from 3762.4 to 12.322 * 308.33 = 3799.2 kW.

As an evaporator cannot normally manage 20 K superheat then try the superheater/ subcooler. Here the vapour superheats 8.5 to 23.5 °C equal to 25.25 kJ/kg. The condensate enters the TEV containing 25.25 kJ/kg less enthalpy and thus has to expand less and enters the evaporator containing 0.0678 kg/kg more liquid refrigerant.

The net result is thus the same increased cooling capacity, not by heating the vapour further, from 8.5 to 23.5 °C but by evaporating 0.0678 kg/kg more refrigerant.

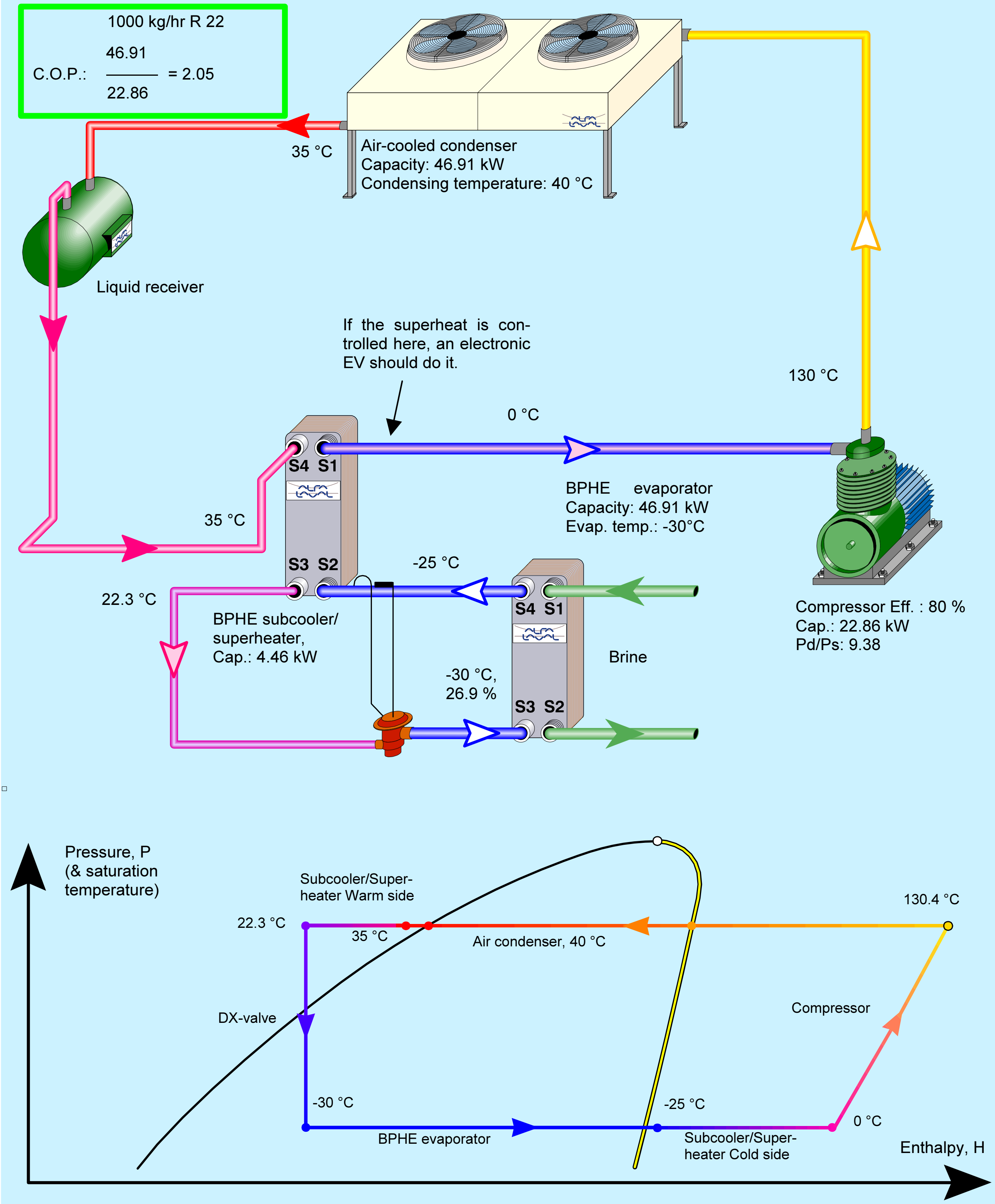


Fig. 05. The subcooler/superheater cycle.

- An effective way of upgrading an old system is to install a condensate subcooling unit. In this way capacity increases of to up to about 20 % can be obtained, presumed that the cooling comes from an external source, e.g. cooling tower make-up water.
- In the figure above, the vapour leaving the evaporator, which then superheats, makes the subcooling. Depending on the refrigerant, this can increase or decrease the capacity, see the text.
- A large vapour superheating can mean an unnecessary high discharge temperature. The discharge temperature can be lowered by a water cooled interchanger or by liquid refrigerant injection.
- A subcooler/superheater system is prone to hunting. When the capacity of the evaporator drops, the vapour flow drops too. The condensate temperature remains for some time and the subcooler/superheater will now be far too large and the leaving vapour is still more superheated. Especially, if the TEV is controlled by the superheat after the subcooler/superheater, it senses this larger superheat and opens, contrary to what it should do. An electronic valve should be used here.

11. The cascade cycle.

Multistage refrigeration cycles can be made thermodynamically effective but they have some drawbacks.

- ♦ At high temperatures, the refrigerant pressure could be very high, necessitating expensive equipment.
- ♦ At low temperatures, the pressure could be low, possibly vacuum. As there will always be some leakage though small, air could enter the system and reduce condenser performance. The vapour volumes will be large, again requiring expensive equipment.
- ♦ Oil tends to distribute unevenly between the stages, especially in systems, where the oil is soluble in the refrigerant at high temperatures but not at low. This could lead to difficulties in lubricating the compressors, or possibly require an elaborate and expensive oil management system.

The cascade cycle overcomes these problems by insulating the two stages in separate circuits. Here the HT stage evaporator condenses the LT stage refrigerant. A BPHE, with its capacity for a close temperature approach is an excellent choice here. See figure 06.

The refrigerants need not be the same in the stages, but can be optimized to the conditions in each stage.

The efficiency of the system increases with decreasing temperature difference between the media in the condenser/evaporator, and the BPHE is excellent here.

In figure 06, the duties correspond exactly in the two stages, but this is not necessary. The HT stage normally has a larger capacity and serves various cooling duties, including the LT condenser/evaporator.

There is one problem with the condenser/evaporator. There can be a pretty large temperature difference between the two sides. If the startup of the unit is too rapid, there could be thermal tensions. This could lead to ruptures of the brazing between the plates (and loosening of the tubes in a S&THE). See also **4. Evaporators & Separators**, § 6.7 for some comments on the control system.

12. The absorption cycles.

12.1. Theory.

In the refrigeration cycle, there is a step where the refrigerant vapour leaving the evaporator at a low temperature & pressure is transformed to refrigerant vapour of a high temperature & pressure, thus enabling the use of a higher temperature cooling water.

In the vapour compression cycle, this step is made by a compressor. Compression of a refrigerant vapour is expensive. Operation, installation, control equipment, the compressor itself; all are expensive.

There are other methods, though. In the absorption cycle, the vapour leaving the evaporator is absorbed (dissolved) in an auxiliary liquid fluid - the absorbent. The pressure of the liquid solution is then increased by a pump. Pressure increase of a liquid by a pump is a simpler and cheaper operation than compression of a vapour. Another advantage of the absorption cycle is the silence of operation. A pump makes considerably less noise than a compressor.

At the higher pressure, the refrigerant and the absorbent are separated by distillation or a simple evaporation.

The result is, just as in the vapour compression cycle, a high-pressure refrigerant vapour, which subsequently condenses and then expands, i.e. the normal cycle.

At distillation, low-grade heat is added in a generator (reboiler). This heat replaces the mechanical energy at the vapour compression. Apart from a high-pressure refrigerant vapour, more or less regenerated absorbent is obtained, which then is recirculated to the absorber.

There are mainly two types of absorption systems, ammonia/water and water/lithium bromide.

Ammonia/water is a high-pressure system (~24 bar), suitable for compact installations. Ammonia is the refrigerant and water the absorbent. There are no particular corrosion problems except that copper and zinc may not be used. Mild/stainless steels are normally used.

Water/lithium bromide is a low-pressure system (vacuum - 1 bar). Water is the refrigerant and LiBr the absorbent. The concentrated LiBr solution is potential corrosive against common construction material and special precautions have to be taken. Due to the large vapour volumes, the system tends to be bulky.

12.2. Applications.

The relative economy of the two cycles depends on the available energy. If low cost electricity is available, the vapour compression cycle might be the more economical. If low-grade waste heat is available at little or no cost, the absorption cycle might be the more economical solution.

There are basically three types of application for absorption cycles, where plate heat exchangers can be used. Typically for most are that the heat for the generator has to be virtually free of charge and that the heat exchangers play a crucial part in the economy of the plant.

- ♦ ***Cogeneration***. A dairy, a slaughterhouse, a fish processing plant, etc. have demands of electricity, steam/heat and chilled water/refrigeration.

  The electricity is generated in diesel engine or a gas turbine and the resulting combustion gas produces steam. A part of the steam or possibly the combustion gas is then used for the absorption plant.

  Plate heat exchangers of the industrial types (all welded or semi welded) have been used for all positions in the absorption plant.
- ♦ ***Chemical industries*** at times produce large amount of excess heat, which can be used for production of chilled water. All- or semiwelded PHEs are used.

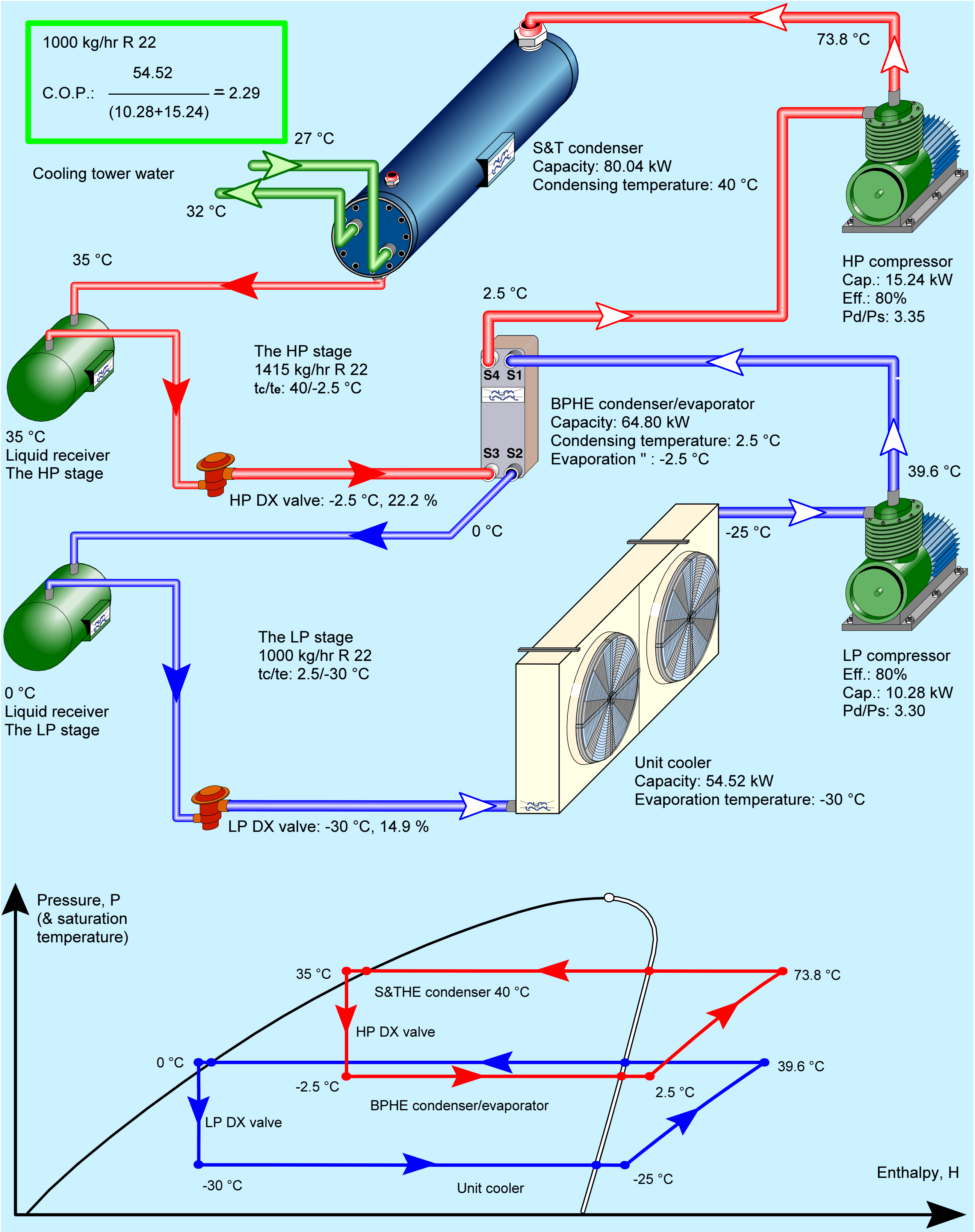


Fig. 06. The cascade refrigeration cycle.

The condenser-evaporator is the critical part of a cascade system. If the capacity of the LP cycle is larger than the actual need, the Co/Ev will be switched on & off, probably only on the LP side. Cold refrigerant then enters the warm condenser and the result could be thermal fatigue. All welded or brazed (PHEs or S&THEs) heat are susceptible, semiwelded PHE are not. Direct expansion is more dangerous than flooded flow.

Try the same methods as for oil cooling, figure 08, **8. Oil & Refrigerants**, slow start-ups or a small continuous flow of the cold side refrigerant, i.e. avoid sudden surges of cold refrigerant.

A discussion is also made in **4. Evaporators & Separators**, § 6.7. See also the system in **8. Oil & Refrigerants**, figure 16.

- ***Residential air conditioning/heat pumps.*** This is a new development. In many countries, especially in East Asia, there is a demand for residential heating during the winter and air conditioning during the summer.

  In contrast to more northern countries, which have an electricity consumption peak during the winter, the electricity demand is high throughout the year. The natural gas network has a consumption peak during the winter.

  The increase in air-conditioning units of the last few years has lead to an overload of the electricity grid during the summer while the natural gas net is underused. As in all residential A/C units, there is a need for silent units.

  Absorption A/C-heat pumps are thus an excellent alternative. However, units based on S&THEs will simply be too heavy (~ 250 kg) to be of practical use. These are sold as white wares, like refrigerators, stoves, etc. and have to be delivered on a pick-up truck by two men.

  The soltion could be heat pumps/air conditioners based on nickel (or other ammonia resistant materials) brazed heat exchangers. The weight can then be halved. More development of these systems is necessary, though.

12.3. The ammonia/water system.

Ammonia is the refrigerant and water the absorbent. To the left of the red rectangle in Fig. 6 is a refrigeration cycle with a condensate subcooler/vapour superheater.

It sends evaporated, low-pressure refrigerant in this case ammonia to the system in the red rectangle and receives back a high-pressure ammonia vapour.

If the items in the rectangle are replaced with a compressor (and a vapour desuperheater), the cycle becomes the normal vapour compression cycle.

Instead of using mechanical energy to increase the pressure, the absorption system mainly uses heat.

Ammonia is extremely soluble in water and thus readily dissolves in the absorber. As the process is exothermic, the absorber has to be cooled.

After the absorber, a pump increases the liquid pressure to the condensation pressure. At this higher pressure, ammonia vapour and liquid water are recovered. This is complicated as both ammonia and water are volatile.

A distillation is therefore necessary and any chemical engineer recognizes the equipment as a distillation tower with a reboiler, a top condenser and a feed/bottoms interchanger, a set up very common in the chemical process industries.

In this way, it is possible to obtain ammonia vapour with a purity of more than 99.5 %. Because of the affinity of ammonia for water it is not practically possible, nor is it necessary, to obtain pure water as bottom product. The bottom product - the weak aqua - contains about 20 - 40 % ammonia and after the absorption, the strong aqua contains 5 - 30 % more ammonia, about 40 to 50 %.

Finally, the strong aqua should be heated to the distillation temperature and the weak aqua cooled to facilitate the absorption. This is done in the feed preheater.

All duties in an ammonia absorption plant can be carried out by PHEs, ranging from Ni-brazed units in small domestic air conditioning units to welded or semiwelded PHEs in large industrial or commercial plants.

The thermal problems are much the same regardless of size. Most positions are uncritical and can be treated as normal one or two-phase heat exchangers but some merit special consideration.

- **The rectifying condenser.** In the figure, a rectifying condenser is installed. This condenses a part of the vapour and returns it as a reflux to the column. A reflux is necessary to provide the part above the feed inlet - the rectifier - with liquid.

  The vapour ascends in the condenser and meets the descending condensate flow, i.e. in counter current. This requires a condenser with a very low pressure drop. The plate heat exchanger can be used here but only if the velocity/pressure drop of the vapour is low, otherwise there is a danger of flooding the condenser.

  A low pressure drop is almost automatically ensured if the condenser is designed with high heat recovery, i.e. only a few degrees temperature difference.

  If the condenser is designed for cooling water, which normally means a larger temperature difference, the number of plates decreases and the pressure drop increases, creating conditions for flooding.

  It is difficult to give exact design condition. A calculated pressure drop of 0.05 - 0.1 kPa/(m flow length), corresponding to a flow of 20 kg/(m, hr) of 20 bar ammonia vapour in a 2 mm channel seems to be safe.

- It is also possible to operate the rectifier with downwards flow of both vapour and condensate. The position of it then becomes critical as a liquid column is necessary to overcome the pressure drop and force the condensate back into the column or an extra pump must be used. Thermodynamically this is less efficient than upwards flow as this separates ammonia and water more efficiently.

- **The ammonia evaporator.** It operates as a normal pure ammonia evaporator. However, in some cases, especially if a thermosiphon evaporator is used, the water content of the evaporated vapour is less than in the incoming condensate. Water then concentrates in the evaporator-separator and the evaporation temperature increases.

  A special additional evaporator - similar to the oil evaporator in a freon thermosiphon, see §16 or **8. Oils & Refrigerants** - is then necessary.

A DX-evaporator is then a better choice, especially if it can be arranged so that all liquid droplets leaving the evaporator move directly to the absorber, without encountering any pockets, where they can collect. The best would be to have liquid injection at the top of the NBPHE but up to now all designs of evaporator with liquid injection at the top have shown a clear reduction in performance of the order of 25 to 30 %.

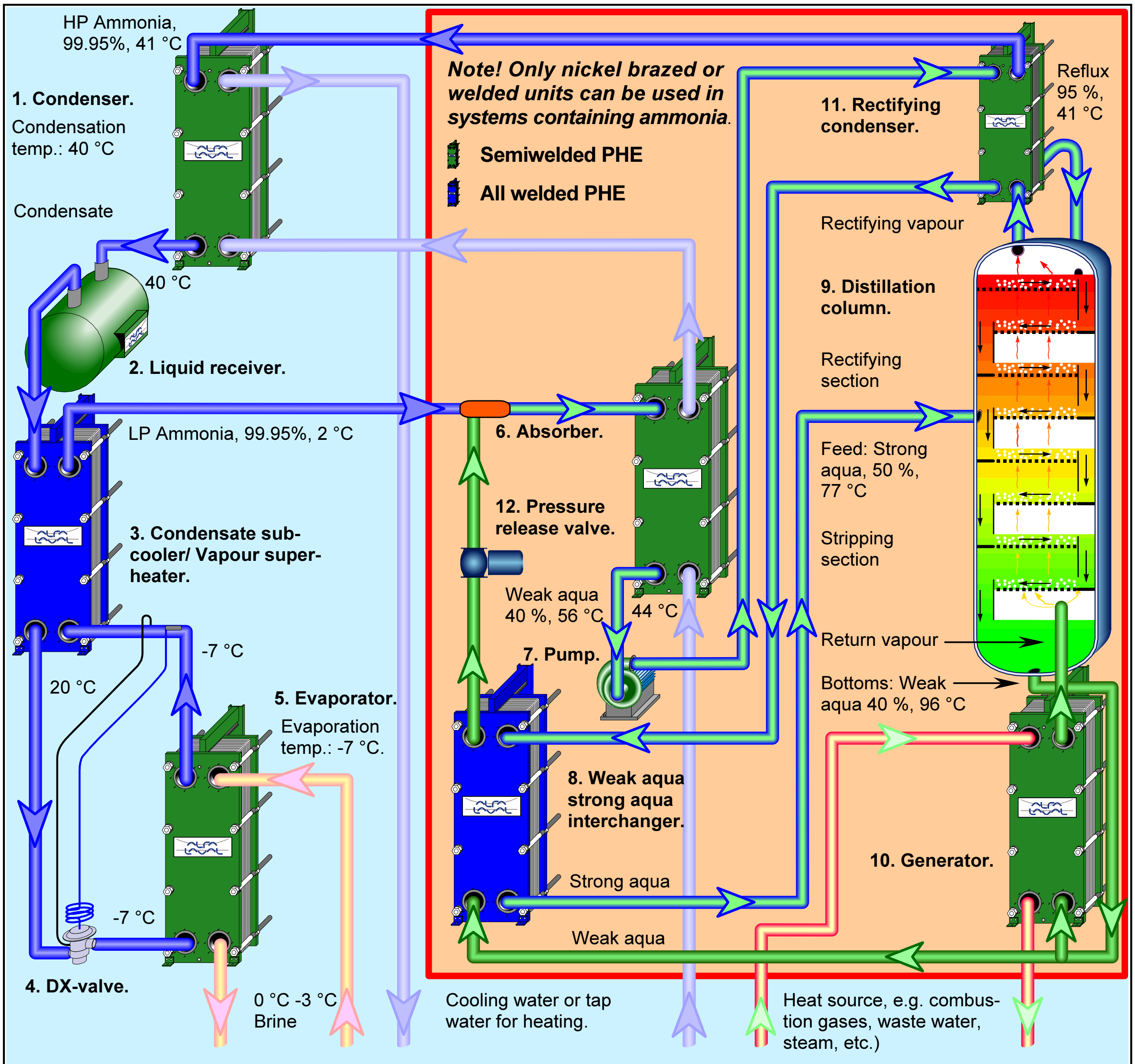


Fig. 07. The ammonia-water absorption refrigeration cycle.

The refrigeration part:

1. The condenser condenses the ammonia vapour.

2. The liquid receiver equalizes variations in the effective refrigerant filling.

3. The interchanger subcools the condensate with refrigerant vapour, thereby increasing the cycle efficiency.

4. The expansion valve, a thermostatic DX-valve.

5. The evaporator here is a normal DX evaporator.

The absorption part:

6. The absorber is composed of two parts, the injection stage, where the cooled weak aqua is injected into the ammonia vapour and the subsequent cooler. The ammonia readily dissolves in the weak aqua helped by the intense turbulence in the cooled, corrugated channels. It leaves the absorber as strong aqua, and enters:

7. The pump raises the pressure from the evaporation to the condensation pressure, and further to:

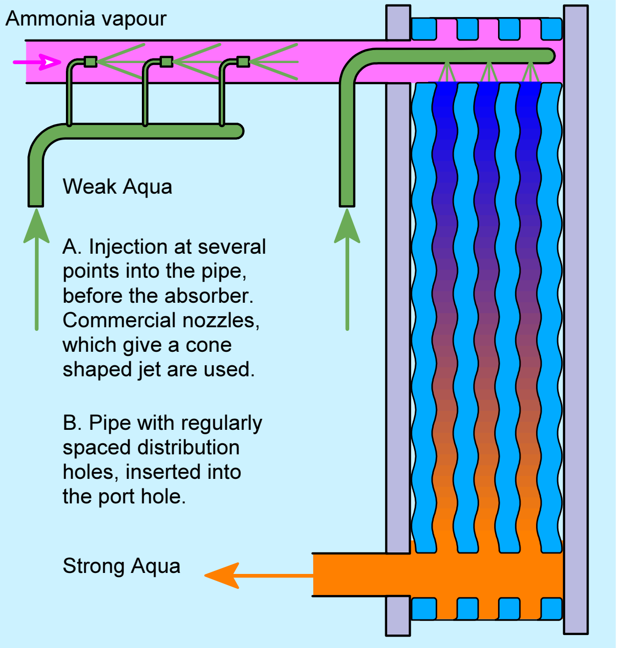
8. The feed preheater. The strong aqua is preheated to the distillation temperature by the weak aqua, which cools. A cool, weak aqua facilitates the absorption in 1.

9. The distillation column can be simple or more elaborate, as shown here. The strong aqua descends and meets an ascending vapour flow. The high boiling component of the vapour - water - condenses, and the low boiling component of the liquid - ammonia - evaporates. The result is a liquid that is gradually stripped of its ammonia content from top to bottom and a vapour, which is gradually enriched in ammonia from bottom to top. The part below the inlet is used to recover the volatile component from the liquid - stripping. The part above is used to concentrate it in the vapour - rectifying.

10. The generator provides the column with vapour.

11. The rectifier provides the column with reflux.

12. The valve reduces the strong aqua pressure.



A. Injection at several points into the pipe, before the absorber. Commercial nozzles, which give a cone shaped jet are used.

B. Pipe with regularly spaced distribution holes, inserted into the port hole.

Fig. 08. Injection system for the ammonia absorber.

- **The absorber.** This is the heart of an absorption plant. A PHE can make an excellent absorber, due to its ability to mix fluids while simultaneously cooling them.

  An absorber is composed of two sections, injection of the absorbing liquid into the ammonia vapour and the subsequent absorption and cooling of the mixture.

  The problem lies in the distribution of the mixture to the channels. Each channel should be fed with its share of vapour and liquid. Unfortunately, it can happen that the vapour and liquid separate after the injection and the liquid then normally enters predominantly the first channels, while the vapour enters the last channels.

  The problem is similar to the distribution of a two-phase mixture coming from a TEV to an evaporator.

  Various methods have been proposed to obtain good distribution, most of them proprietary. Some general rules can be given:

  There are no entirely reliable design methods, but a PHE can be designed as a condenser, with a portion already condensed at the inlet. The ammonia-water sys tem is a refrigerant with a very large glide. The heat release when the two vapours cocondense is due not only to the latent heat but also to a high mixing heat.

  Avoid bends (i.e. centrifugal forces) and large distances between the injection point and the entrance to the BPHE. The liquid then settles and separates.

  The weak aqua should be injected into the ammonia vapour pipe. Multiple injection points along the inlet pipe have been tried with good result for larger pipes (≥100 mm) See Fig. 08 A,

  For smaller pipes a distributor pipe as shown in figure 8B has been tried with good result.

  Injection, type ejector pump, is untried but might be a good idea. The high velocity in the nozzle breaks down the liquid in fine droplets.

  It is an unsolved question whether the inlet should be mounted from the top or from the bottom. Most installations are from the top. It is however easier to ensure a good distribution, especially of the liquid, from below, but the flow may be unstable at least at low capacities.

- **The generator.** The actual design of this depends on the type of heat source available. In industrial plants, where steam is available, a welded PHE is a good choice. From an operational point of view, it is uncritical. It performs as a normal steam generator.

  Domestic types of heat pump/air conditioner usually have natural gas as a heat source and the generator is integrated with the burner.

- **The subcooler/superheater.** In a compression cycle this is a questionable unit, see §10, p.10, due to the decrease of the circulating amount of ammonia.

  An absorber is not that much affected by the vapour density, the little more energy necessary to be cooled off can easily be accommodated in the absorber cooling part, possibly with a small increase of the cooling surface and/or amount of cooling water.

  Above all, the increased superheat of the ammonia helps to evaporate the last water in the liquid. Water leads to a substantial increase of the dew point, see **8. Oils & refrigerants**, Table 3..

12.4. Water/Lithium bromide system.

For the general principle see figure 09.

The basic principles are similar fo both the ammonium and lithium bromide absorption systems. While the ammonia absorption system has a high pressure but few corrosion problems - as far as copper is not used - a lithium bromide system has a low pressure but is corrosive.

A pure lithium bromide solution is corrosive. It has to be passivated with molybdate or chromate solutions and the pH should be kept as high as possible. The oxygen and chlorine content should be as low as possible.

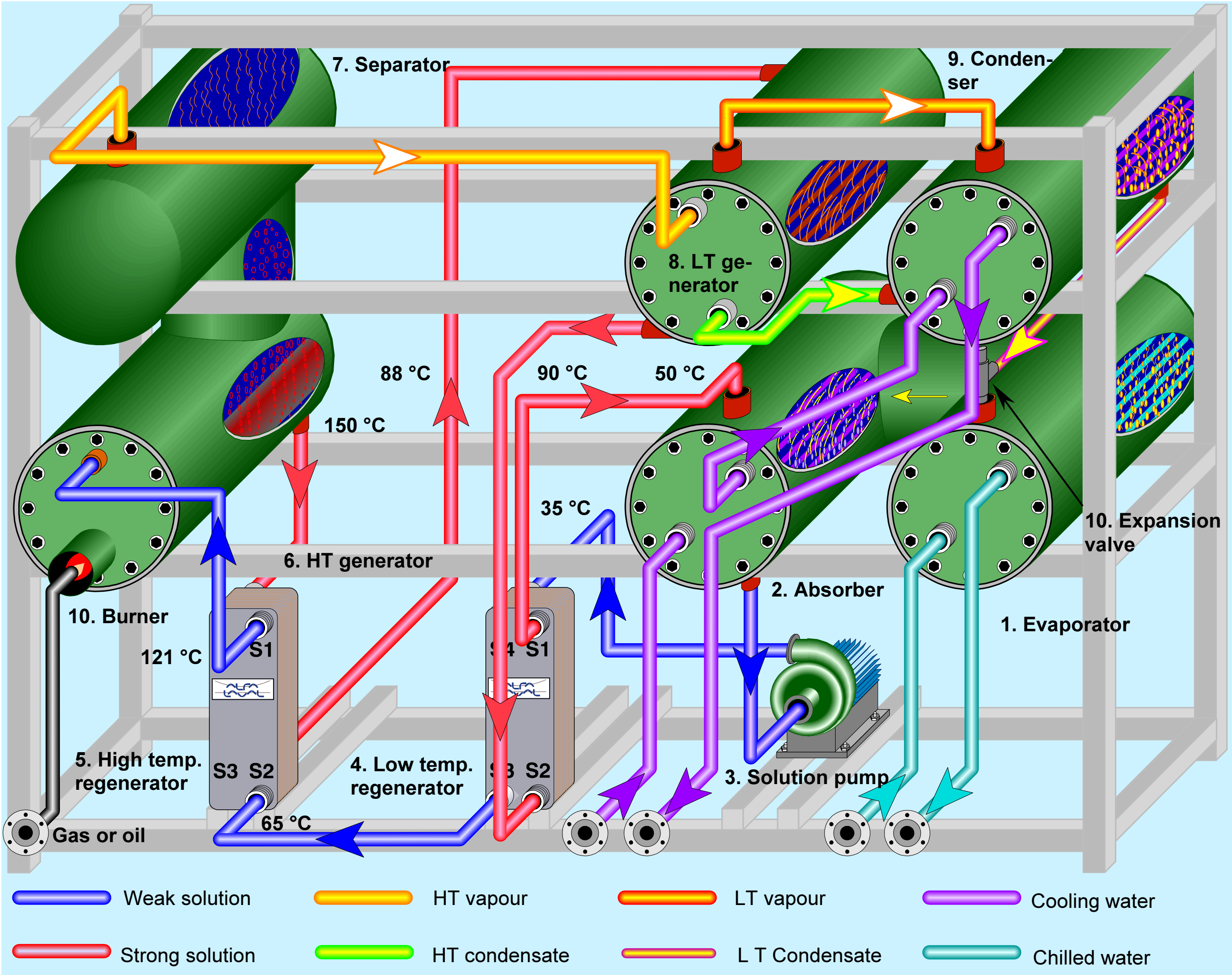
Due to the corrosive nature of LiBr, tests should be made with the particular composition before it is used in a copper/ stainless steel heat exchanger.

The corrosive mechanism involved is pitting, crevice and stress corrosion. These are all interrelated. As the names imply, not only the materials are important but also the design and execution of the unit. See also the chapter on corrosion about different types of corrosion.

The stainless steel plates used for the BPHE are smooth almost polished, which reduces the pitting danger. The copper effectively fills out all crevices, thus limiting danger of crevice corrosion. The brazing is an effective stress relieving operation thereby eliminating stress corrosion, at least the part depending on residual stresses. Thus, most conditions are fulfilled to prevent corrosion

Finally, to improve the wetting of the surface - and thus to increase the effective heat transfer area - a surface-active compound, type detergent is added. This could be octyl alcohol or something similar.

It is interesting to know that lithium hydroxide, used to increase the pH, is a better absorber than LiBr, but is not used partly because the tradition of a well-established technology partly because of still worse corrosion problems.



1. ***The evaporator.*** Cold water evaporates under vacuum and cools the chilled water. At these temperatures, water vapour has a very low density and requires special types of heat exchangers. PHEs - brazed or gasketed - are usually not suitable for handling low-density vapours.

After the evaporation, the water vapour moves to:

2. ***The absorber***, which absorbs the water vapour in a strong solution of LiBr.

3. ***The pump,*** moves the resulting weak solution via:

4 & 5. ***The LT & HT regenerators,*** which heat the weak solution to the vicinity of the boiling point. A high regenerative efficiency is crucial to the economy of the process. The long temperature program make these duties excellent for PHEs - most likely in two-pass.

After the regenerative, the weak solution enters

6. ***The HT generator***. A part of the water boils off, usually in a gas fired boiler. The resulting strong solution releases its heat in the HT & LT regeneratives, more water evaporates in the intermediate placed LT generator.

The HT vapour continues to

7.***The separator,*** to be used as a heating medium in

8.***The LT generator*** boils off water from the rich solution at a lower temperature/pressure than in the HT stage.

The use of vapour from a HT stage to heat a LT stage is commonly used in evaporation plants to improve the economy. The conditions are similar here.

9. ***The condenser***. Both the direct vapour (HT stage) and the LT vapour condenses here. Here too, the vapour volume is too large for PHEs. The resulting condensate, expands to the evaporation temperature in

10. ***The expansion valve*** and then to the evaporator.

Fig. 09. The Water-Lithium bromide absorption chiller.

Lithium bromide is the absorbent and water doubles as refrigerant and solvent for the absorbent.

The separation of the water and lithium bromide is simple. As lithium bromide is a solid, there is no need for a distillation tower to separate the water and the lithium bromide.

The water is simply boiled off, taking care to leave enough water to keep the lithium bromide in solution and avoid crystallisation.

The low temperature limit is set by the freezing risk of the water and crystallization of the lithium bromide.

Air cannot be cooled directly in a unit cooler nor can an air condenser be used to condense the water vapour. The volumes of both the water vapour and the air are simply too large.

This impedes its use in small residential A/C-heat pumps. It is thus basically a water chiller for medium capacities.

13. Heat exchanger applications in a large supermarket.

See figure 10. This is an example of how a modern supermarket could look. However, the plant is not complete. A large supermarket would have even more cooling duties at different temperature levels.

Since the principal purpose is to show the heat exchanger applications, other equipment, valves, pumps, controls, pipes are only partly shown

Characteristic for the plant are:

- The entire ammonia circuit is kept in the machine room, thus there is no ammonia in spaces frequented by unauthorized staff and supermarket customers.
- The ammonia evaporator cools brine. The brine then cools the various unit coolers. Two semiwelded PHEs are used here, both connected as thermosiphons to a common vapour-liquid separator.

  The evaporator-brine tank loop is separate from the brine tank-UC loop. Each has its separate pump. The brine mixes in the tank, though. This is in order to control the evaporator and the UC independently.

  The brine inlet temperature, -8 °C, is equal to all the UC, but the return temperature, -4 °C, is a mixture of the different UC.
- The picture shows one brine circuit serving various UC. If the air temperatures of the various stores are very different, several brine circuits (thus several evaporators) with different temperatures might be necessary.
- Similarly, the ammonia condenser is cooled by circulating water, in turn cooled in a liquid cooler, situated on a rooftop. Here it is shown in a semiwelded design, with a separate heat recovery section. See below.
- Instead of a liquid cooler, a cooling tower, direct water or a secondary water-cooling system can be used.

  The last system is very popular in the Far East. High humidity makes cooling towers less interesting and fresh water is increasingly scarce.

  A circulating fresh water system cooled by sea water in a GPHE with titanium plates has been used extensively in HK, Singapore and other places to dump the heat from large A/C installations.
- It is possible to use brine to cool a deep-freezing space but it becomes increasingly inefficient, the lower the temperature drop is.

  Shown here is cascade connected R404a system for deep freeze storage. Evaporating ammonia from the main system cools the R404a condenser. This is a duty suitable for a small semiwelded unit or as here, a nickel brazed unit with a frame.
- If the deep freeze store is remote from the machine room, the R404a lines might be too long. It might then be better to place the refrigeration unit close to the store and cool the condenser with brine from the main unit. Another alternative is a heat pipe with carbon dioxide as heat (refrigeration) carrier. A copper brazed PHE could then be used as the condenser.
- Compression from the evaporation temperature -12 °C to the condensation temperature 45 °C gives a low C.O.P. More importantly, ammonia gives very high discharge temperatures, in this case around 160 °C.

  This is far too high, leading to oil break down and seizure of the compressor.
- The plant is therefore equipped with two compressors connected in series (or a two-stage compressor with intermediate refrigerant inlet) and an economizer (see Fig. 3); here a nickel brazed unit. This has the double function of increasing the C.O.P. and decreasing the discharge temperature to a more reasonable 96 °C.
- A discharge temperature of 96 °C is still high enough to recover the heat for hot water production. Water of a temperature of 80 to 85 °C can easily be produced in a PHE. The condenser is hence equipped with a desuperheating section for production of hot water. 20 to 25 % of the cooling energy can be recovered as hot water.
- The separator is composed of two parts. The horizontal part mainly serves as a vapour liquid separator.

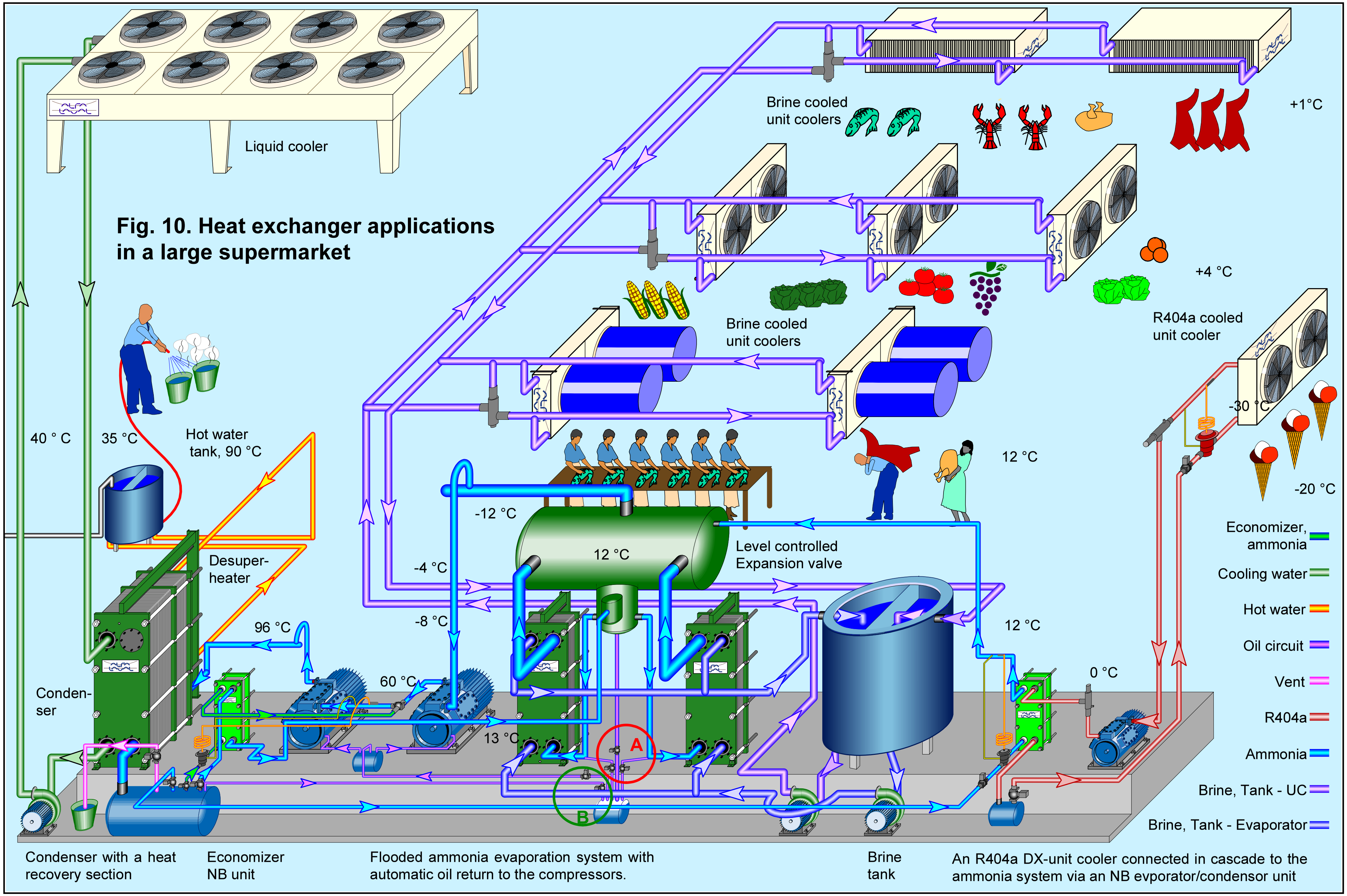
  The ammonia liquid level is maintained in the vertical vessel. Because of the comparatively small cross section of this as compared with the horizontal vessel, the ammonia content can be kept small.
- The liquid level in a separator is not quiet. To get a stable reading for the float, which controls the expansion valve, the float and valve is placed in a separate vessel, which communicates with main vessel.
- Such a valve-vessel combinations are sold as integrated units. Some are especially made for placement at the exit of a semiwelded plate condenser,
- Oil is insoluble in and heavier than ammonia. Oil from the compressors collects at the lowest point of the thermosiphon loop. From these points, the oil drains to the oil tank through the pipes with the valves A.

  There will inevitably be some ammonia entering the oil tank. The ammonia evaporates and the outside of the tank will be covered by ice. When the tank is full of oil ammonia cannot enter, the temperature increases and the ice melts.

  This serves as a signal, visual or by a thermometer, that the oil tank is full. The valves A are then closed and the valves B are opened. Through one of the valves B, high-pressure ammonia pushes the oil out of the tank through the other valve B and to the secondary oil tank. This feeds oil to the compressors.

  If the ambient temperature is not high enough to evaporate the ammonia, an electric heater or a coil with condensate can be installed to improve the evaporation. A pump might be used instead of HP ammonia.

Fig. 10. Heat exchanger applications in a large supermarket



14. Reversible systems.

14.1. Applications.

Certain systems can reverse the refrigerant direction, apart from the compressor, which always operates in one direction. A special four-way valve is then used to reverse the flow from the compressor to the other components.

Figure 13 shows the basic principle of such as system. A pilot operated four-way valve is used. The previous condenser then becomes the evaporator and the evaporator becomes the condenser.

PHEs and TEVs need some consideration when installed in such systems especially if one of the components is an air-to-liquid heat exchanger. Only systems with at least one PHE are considered here.

14.1.1. Cold stores.

The condenser is a BPHE and the evaporator a UC with an evaporation temperature less than 0 °C. In such a UC frost forms and it has to be defrosted regularly.

Simply shutting off the evaporator and letting the fan run, an electric heater or injecting hot discharge gas from the compressor to the UC could do this.

Flow reversal is very effective defrosting method. The release of the latent heat when the UC operates as a condenser rapidly thaws the ice on the fins.

14.1.2. Heat pumps.

In heat pumps with exterior air as a heat source, air evaporates the refrigerant in an outdoor coil and the condenser heats water, which then is distributed to various room heaters. Figure 13 could also be an example of this. In winter when the ambient air drops, the evaporating temperature could drop below zero and frost form, exactly as above. Here too an effective way of defrosting the coils is to reverse the flow.

The design criteria differ between an A/C unit and a heat pump, see table 1. See also table 6, **8. Oils & Refrigerants** for refrigerant properties, important for HPs.

14.1.3. Reversible A/C-Heat pumps units.

A. Air as heat source/sink, water room BPHE. This is similar to 2 above, figures 14 - 16 shows some other examples. The reversed cycle is now not only for defrosting but also as A/C during the summer.

B. Water as heat source/sink, room air coil. The water source is used both as a heat source and a heat dump in a BPHE and the air is heated/cooled in a coil. Figure 13 applies here as well but possibly with reversed water flow direction, see below. Note, defrosting is not necessary.

C. Water/Water heat exchangers. As no frost formation from the air is involved, defrosting is not necessary.

14.2. Design considerations.

14.2.1. Co- vs. counter current flow.

The evaporator and the condenser work with forwards and reversed refrigerant directions but normally with constant water direction. The BPHE operates with alternatively counter- and cocurrent flow. The effect on the mean temperature difference could be dramatic in a heat exchanger. such as the PHE where both sides are in parallel, i.e. close to true counter- or cocurrent flow

An air coil normally has the two in cross flow, i.e. there will very little change of the MTD when the refrigerant flow is reversed, though there could be some effect on the heat transfer coefficient.

The conditions in the BPHE thus determine if the normal operation shall be in co- or counter current. Normally the BPHE is laid out for the most effective heat transfer, i.e. largest MTD, which usually but not always means counter current flow. Reversed flow then means cocurrent flow.

At reverse flow the duties normally change, e.g. the reversed condenser duty will not be the same as the forward duty. Normally both the temperature programs and the capacities change during the reverse cycle.

Thus general rules for the flow direction in the BPHE cannot be given, but it has to be decided from case to case.

- ***Flow reverse only for defrosting.*** As this is for a limited time, the main cycle sets the system layout.

  Ex. A UC cooling air of -5 °C and with an evaporation temperature of -10 °C and a BPHE with a condensing temperature of 45 °C cooled by water 27 to 32 °C.

  At the defrosting cycle, the UC will be charged by condensing refrigerant with an initial condensing temperature of 45 °C and the BPHE will be charged by evaporating refrigerant of initially -10 °C.

  The refrigerant temperatures changes but there is ample temperature difference to accommodate any flow direction in the BPHE.

  Note what is written about freezing below.

- ***Reversible HP|A/C.*** It is here impossible to define which is the main cycle. In a hot climate only A/C units are of interest. In progressively colder climate, A/C with some HP function, then HP with some A/C function and finally only HP. A HP unit always needs a defroster but the reversed cycle competes with other methods.

  In figures 11 A & B the typical temperature programs and duties are shown for a split HP|A/C unit and for both R22 and R407c. The figures show clearly that an evaporator cannot operate in cocurrent flow for the normal air conditioning temperature program, especially not a refrigerant with glide as R407c. Thus, the flow by default has to be cocurrent when it operates as condenser in the heating cycle. Read also more about refrigerants with glide in the **8. Oils & Refrigerants**.

  The conclusion is therefore: ***When the BPHE operates as an evaporator for the normal A/C temperature program (12/7/2°C) it has to be in counter current.***

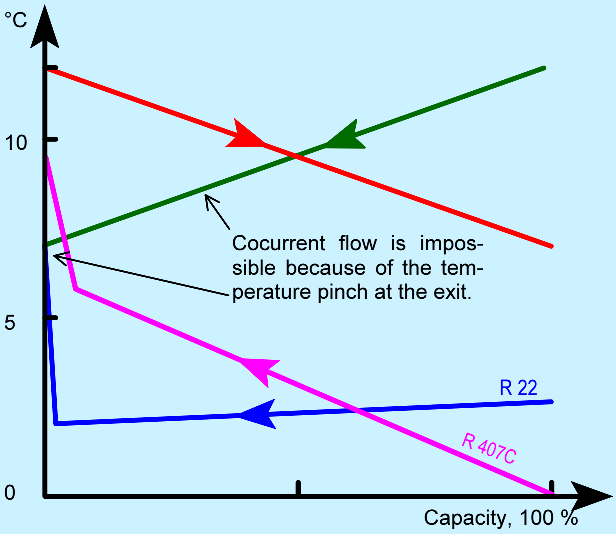
  This is valid for all types of split system regardless of whether the forward flow is a heating or a cooling cycle.

  Note that in e.g. a cold climate where the A/C cycle is less important, it might be possible to increase the chilled water temperature. The BPHE could then operate as evaporator in cocurrent during the A/C cycle and as condenser in counter current during the HP cycle when maximum efficiency is necessary.

- ***Two-pass refrigerant HP|A/C.*** An interesting new development is shown in figure 12.

Component	Air conditioning units	Heat pumps
Best compressor Worst compressor	Open, installed outdoors Hermetic, indoors or outdoors	Hermetic, indoors or outdoors Open, outdoors
Refrigerants	See B below,	←
TEV & Evaporator:	As low superheat as possible	←
Best evaporator Evaporation temp.	Direct refrigerant As high as possible	Water or brine better than air ←
Condenser Best condenser	As large as possible Water cooled with well water	As large as possible Floor heating, direct would be the best but not feasible.
Condensate temp.	As low as possible through Economizer Water condensate subcooler Subcooler/Superheater	Kept at sufficient level through a pressure controller, especially important in the winter. This level has to be as low as possible (large condenser) to obtain a high COP. Note that the various condensate subcooling methods are normally not justified.
COP	Depending on ratio AC/HP Important in south US, APR Not so important in Europe	← Important in Europe Not so important in US, APR

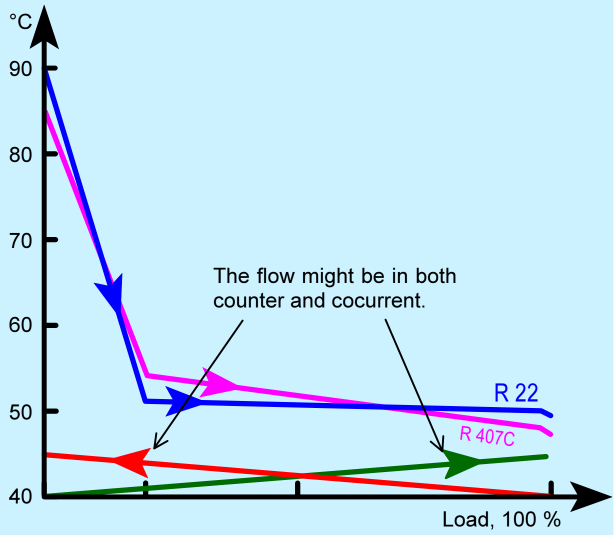
Table 1. Design of A/C units vs. heat pumps.



Cocurrent flow is impossible because of the temperature pinch at the exit.

BPHE (evaporator) 20.6 kW	Air coil (condenser) 25.9 kW
Water 12 → 7 °C	R22 45°C + 35 K → 42 °C
R22 2°C + 5 K ← 2 °C	Air 35 °C ← 30 °C

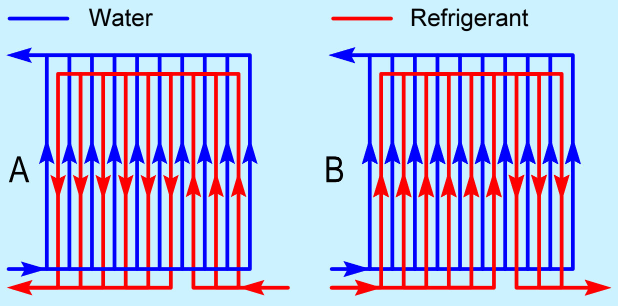
Fig. 11A. The temperature program in the BPHE at the cooling cycle.



The flow might be in both counter and cocurrent.

BPHE (evaporator) 23.1 kW	Air coil (condenser) 25.9 kW
R22 50°C + 39 K → 47 °C	Air 10 °C → 7 °C
Water 45 ← 40 °C	R22 0°C + 5 K ← 0 °C

Fig. 11B. The temperature program in the BPHE at the heating cycle.



The refrigerant is connected in such a way that the last pass is always in counter current to the water.

A.***Condenser.*** The vapour enters from below and desuperheats somewhat. It then enters the last pass and condenses in downwards flow in counter current.

B.***Evaporator.*** The flow reverses and the evaporation is in cocurrent. The critical superheating in the last pass is now in counter current flow.

- The advantage of this is set-up is especially noticeable for refrigerants with a glide, notably R407c. Due to the glide, there might be a temperature pinch in cocurrent flow, which thus has to be avoided.
- The water is always in upwards flow, which secures the water filling even at low pressure drops.

Figure 12. HP|A/C with the refrigerant in a two-pass flow arrangement.

14.2.2. Freezing risk at the flow reverse.

At the flow reversal, the condenser pressure drops from some 15 bars to the evaporator pressure, maybe 5 bars. The evaporator pressure increases correspondingly. At some intermediate point, the pressures in the two units become close, then equal, again close but reversed.

The evaporator pressure suddenly becomes much higher than normal. In a given compressor, the necessary shaft power is proportional to the suction pressure. The electrical motor cannot normally handle such an increase of the load and burns out.

To prevent this, most TEVs in reversible systems are of the MOP type. These close when the pressure exceeds a certain value regardless of the superheat. During this time, when the pressure difference over the TEV - the driving force - is small or zero and the MOP valve is closed, none or little refrigerant passes the valve. The compressor capacity then becomes much larger than the valve capacity leading to a drop of the (new) evaporator pressure. The temperature might then drop below zero and freezing occurs in the evaporator water circuit.

Compounding the problem is the amount of condensate in the condenser becoming an evaporator, i.e. between the BPHE and the TEV in figure 13. If this line is very long, it could contain, together with the condensate in the BPHE condenser, a sizable amount of liquid refrigerant. This refrigerant has to evaporate when the BPHE becomes an evaporator. The effect will be similar to a pump-down and the result will be freezing temperatures.

This liquid body, if very large and in a system with mismatched components, could enter the compressor and liquid hammering might then damage the compressor. Moreover, this liquid refrigerant could decrease the superheat or prevent the vapour being superheated, thus closing the TEV and reducing the flow still more.

- ♦ ***Freezing refrigerant temperatures*** should normally not be a problem, if not too low and lasting too long, ***as the water temperature is high, giving the BPHE a substantial heat capacity reserve.*** Should a channel be blocked or another dead space formed by fouling, mechanical damage or the like, then the enclosed body of water might rupture the BPHE.

- ♦ ***The use of MOP valves*** means that to protect the compressor, the evaporator is put at risk, a somewhat questionable engineering practice. They should open well above zero degrees but there might be a time delay and the damage is done. A better solution is:

- ♦ ***A suction pressure regulator***, which solves the problem as this is placed after the compressor. It throttles the flow until the suction pressure falls below the set point. Cost prevents the use in many systems, though. It should not be used together with a MOP-valve.

- ♦ ***Frost formation on the BPHE*** should normally not occur due to the high water temperatures. If a spot on the outside is frosted, that could hint at an obstruction of the water flow and formation of an enclosed body of water most likely due to fouling.

- ♦ ***The liquid receiver*** should not be placed between the TEV and an evaporator, see below.

- ♦ ***Long flow lines*** for liquid and two-phase refrigerant might mean a large body of refrigerant to evaporate.

- ♦ ***The four-way valve*** could be stuck by ice, mechanical damage or fouling and prolong the freezing time.

- ♦ ***One TEV systems.*** The TEV - see below - in such a system has a lower capacity in the reverse direction, which might prolong the freezing period, figure 13.

- ♦ ***Two TEV systems*** as in figure 14 on the other hand contain more pipe work and extra non return valves, i.e. more liquid refrigerant.

- ♦ ***TEV installation.*** This theme is discussed later in this chapter and in detail in the evaporator chapter.

- ♦ ***By-pass***. One possibility would be to use a small solenoid valve parallel to the TEV. This opens for a certain time during the switch over and supplies the evaporator-compressor with refrigerant while the pressure difference is low or the TEV is closed.

  There are TEVs, which gives a refrigerant flow at zero superheat. Basically they are TEVs with an internal bypass (“bleed port”). These could possibly be used.

14.2.3. Suction line separators.

See figure in the evaporator chapter. If the vapour from the evaporator contains sudden surges of liquid droplets, which might cause liquid hammering (see above) a liquid separator should be installed in the suction line.

Note that a liquid separator in a DX-system should not operate as an accumulator. This function should be reserved for the liquid receiver. This makes better use of the liquid as prevention against a liquid starved TEV. There is also the danger that the liquid accumulated in the separator is flushed out and into the compressor therefore with damages.

14.2.4. Condensate subcoolers.

When the liquid refrigerant leaves the condenser, it is usually subcooled some degrees. This is normally enough but in some cases a lower exit temperature is needed:

- ♦ The condensate is reheated in the LR. The pressure in the LR then increases which pushes condensate back into the condenser. See Liquid Receivers below.

- ♦ The expansion valve operates at a lower pressure than the LR and the refrigerant vaporizes in the line.

A ***lower condensate temperature*** can be obtained by

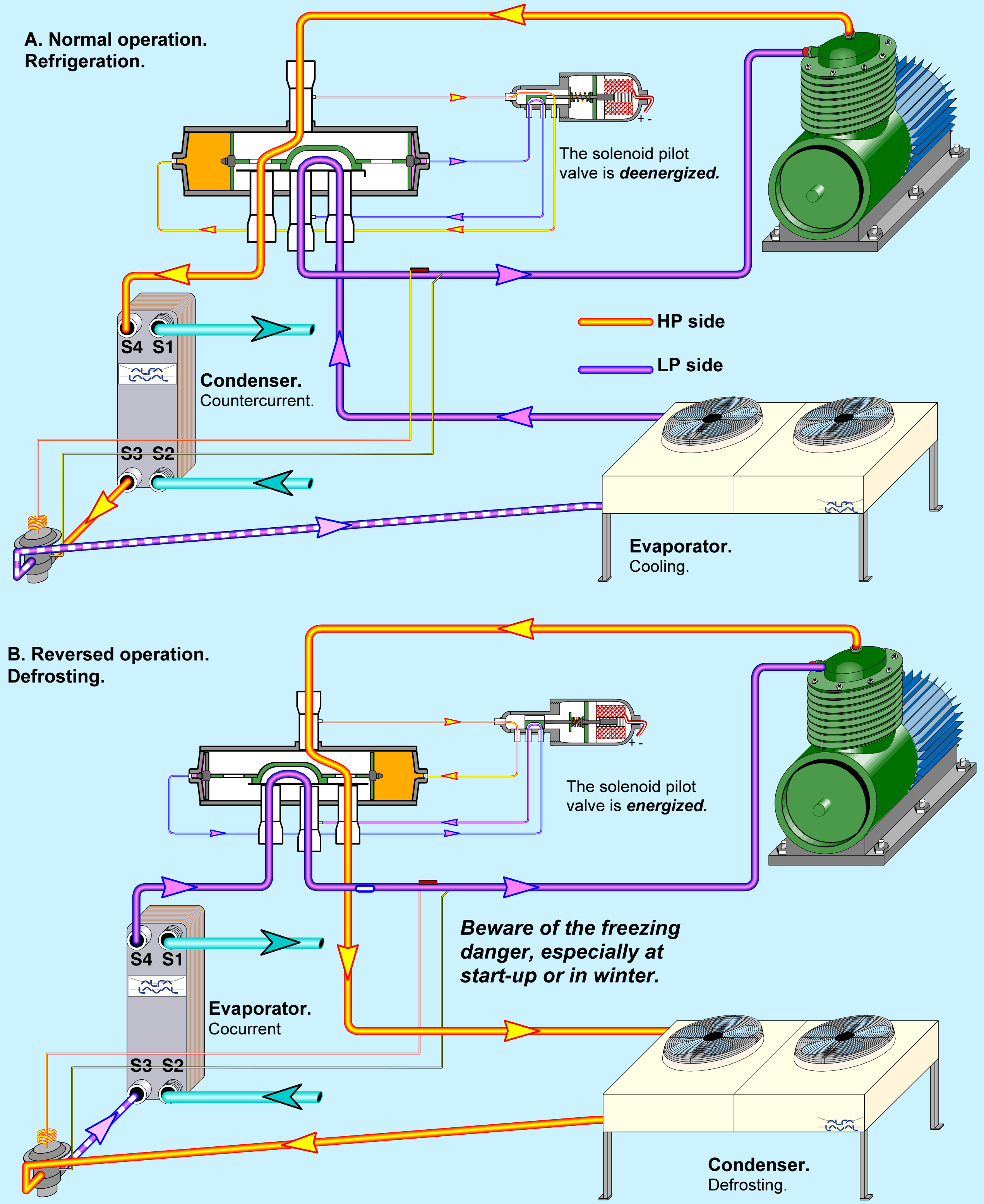
- ♦ An economizer, figure 03 & 04 in this chapter.

- ♦ A water cooled subcooler, figure 02 in this chapter.

- ♦ A suction vapour subcooler. See figures 5 and 15.

14.2.5. Oil separators, sight glasses and filter/driers.

A possible oil separator should be installed immediately after the compressor in line L3 in figures 14 & 15. A sight glass or better two - one to look through, one to light through - could be installed in the L1-L3. Filter/driers exist both for vapour and liquid, one or bi-directional flow. This too should be installed in one of the lines L1 - L3.



A. Normal operation. Refrigeration.

The solenoid pilot valve is *deenergized.*

B. Reversed operation. Defrosting.

The solenoid pilot valve is *energized.*

Beware of the freezing danger, especially at start-up or in winter.

Fig. 13. A pilot operated, four-way valve used in a reversible cooling unit.

The design and function of the pilot and main valves is similar. The pilot valve is actuated by a solenoid, the main valve moves by pistons operated by the discharge and suction pressures and controlled by the pilot valve.

When the solenoid valve is deactivated, the pilot valve connects the left cylinder in the main valve to the compressor discharge pipe and the right cylinder to the suction pipe.

The cylinder-slide assembly (green in the figure) then moves to the right.

When the solenoid valve is activated, the pilot valve connects the right cylinder in the main valve to the compressor discharge pipe and the left cylinder to the suction pipe. The cylinder-slide assembly (green in the figure) then moves to the left and reverses the refrigerant flow.

14.2.6. Liquid receivers in reversible systems.

- ***Types.*** There are two types, the through and the surge. We will deal here exclusively with the through LR.

- ***Refrigerant storage.*** During service on the components, the refrigerant is stored in the LR. This is especially the case for large systems. In small systems, the service engineer might temporarily store the refrigerant in an external vessel.

- ***Damping device***. It evens out disturbances in the refrigerant flow.

- ***Refrigerant storage at the cycle reverse***. An HE contains different amount of refrigerant in condenser and evaporator modes. Similarly, the pipe work holds different amount of refrigerant depending on whether they contain vapour, liquid or a two-phase mixture.

  An air unit, installed on a rooftop remote from the compressor and the condenser can have a substantial difference of refrigerant content in evaporator and condenser modes. A BPHE close to the compressor will have little difference. The LR will store this difference.

- ***Feed of the TEV*** is ensured by the body of liquid refrigerant in the LR. This liquid body evens out variations in liquid refrigerant during the operation and ensures that there is always liquid before the TEV.

14.2.7. Installation of the liquid receiver.

- Solenoids valve might be needed to shut off various parts of the system but they are not shown here.

- The LR should be installed lower than the condenser and the interconnecting pipe should not have any major flow obstructions. See e.g. figures 14 & 15.

- The LR should not be heated up by installation in a warm room or close to a heat-producing machine. If warmed up, liquid evaporates and the vapour will push backwards through the condensate line to the condenser exit. Here it can block channels and impede the condensation. The problems can be solved by an equalization line from the LR to the condenser inlet, but other problems may arise, see below.

- An equalization line could be placed between the LR and the compressor discharge pipe to return vaporized refrigerant from the LR back into the condenser. To keep the pressure equilibrium, condensate has to rise in the condensate exit pipe, back to the condenser. If the condenser is not placed high enough, it can be flooded and difficult to vent.

  The recommendation is therefore: ***Do not install an equalization line in a reversible system.***

- ***The LR should not be installed between the TEV and an evaporator at a flow reverse.*** A large body of liquid refrigerant will then have to be evaporated with freezing danger as a consequence or if not evaporated, it will enter the compressor resulting in liquid hammering and possibly damages. See the preceding chapter about freezing.

- The two condensate pipes in figure 14 & 15 can be joined before being connected to the LR. This simplifies the installation as standard LRs can be used.

By carefully balancing the various components, pipe lengths and diameters, it might be possible to design a system, at least a small one, where the refrigerant difference between the cycle modes is small and can be stored in the port hole of the BPHE and interconnecting pipes and a LR is thus not necessary, see figure 16.

14.2.8. One or two TEV systems.

The basic requirements of a ***TEV*** installation for PHE evaporators - whether brazed, welded or semiwelded - are:

- A straight pipe between the evaporator and the valve.

- No bends, fittings, valves, vessels etc., which can cause phase separation. A common mistake is to install some pipe bends between valve and evaporator.

- The pipe diameter should not be larger than the exit diameter of the TEV.

- An intermediate valve is permitted if there are no bends at the valve exit. An inline valve, e.g. a check-valve could even have a beneficial effect by remixing a separated vapour-liquid mixture.

- The HP side has always to be filled with liquid.

14.2.9. Systems with a liquid receiver.

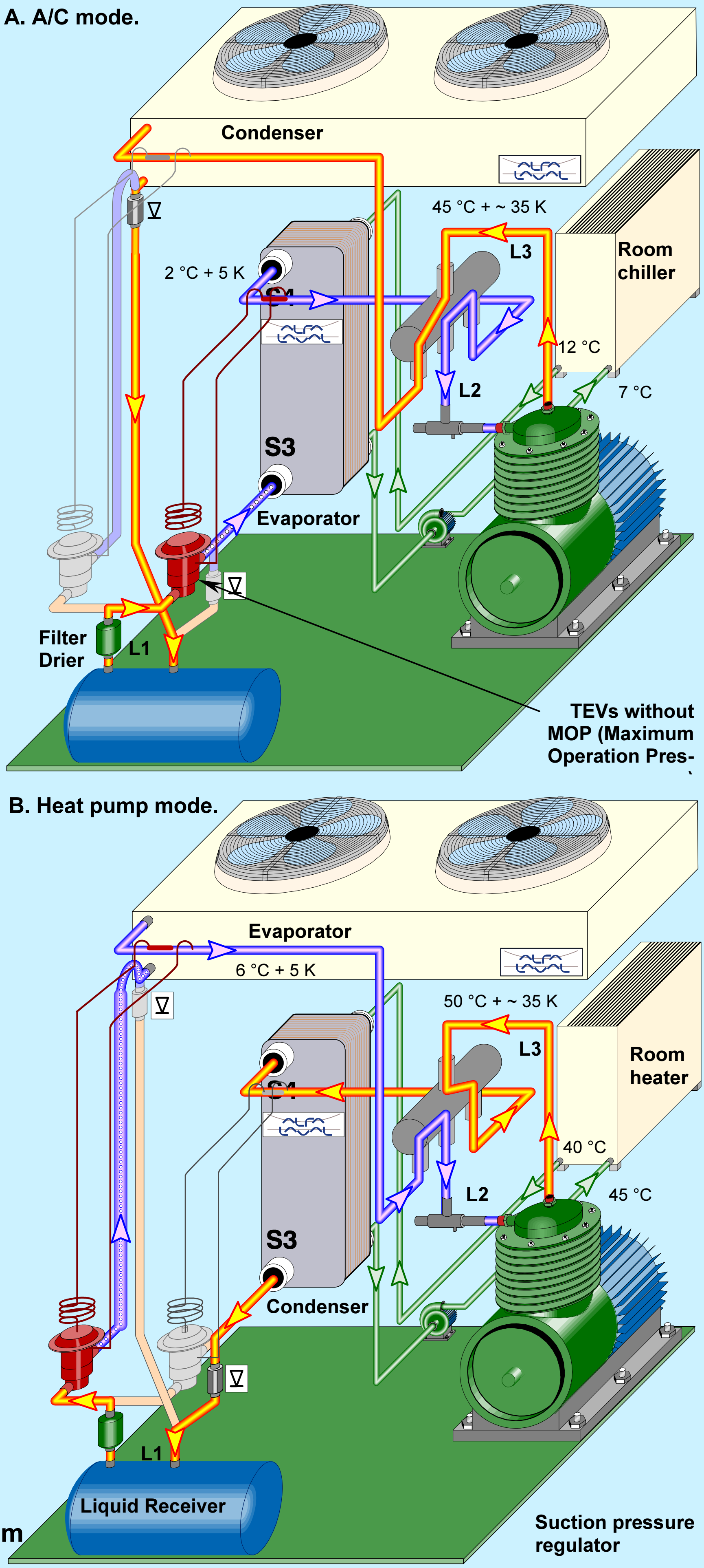
To avoid the LR receiver becoming positioned between the TEV and the evaporator in one of the flow modes either two TEVs and two check (one-way) valves, figure 14 or one TEV and four check valves, figure 15 can be used.

Two valves systems. Figure 14 shows a system with two BPHEs, which fulfils all the requirements above. Note:

- Two TEVs mean that each can be selected and installed for the appropriate duty and evaporator type.

- In reverse flow, the condensate has to be routed around the forward TEV. This is done by a parallel mounted check valve, which opens in the reverse direction. When the flow changes to forward direction again, the reverse TEV closes and the flow pass through a forward mounted check valve.

- An inactive TEV have the bulb and equalization lines placed at what now is a condenser inlet. The superheat signal is then large and the valve tries to open. The valve has thus to be of the MOP type, which closes above a certain pressure or another check valve has to be installed in the DX line.



The set-up fulfils the most important requirements for a reliable installation, notably that there is ***a straight pipe between the TEV and the BPHE*** and that there is ***a direct uninterrupted passage from the condenser to the LR***.

Moreover, the pipe from the TEV to the air coil enters the condensate line from above. It is thus not filled with condensate when the air coil operates as condenser.

The corresponding pipe from the valve to the BPHE is short and enters horizontally. The little condensate, which collects here, is unimportant.

The pipe parts L1 to L3 have constant flow direction and can be used to install equipment, which shall be used in both forward and reverse modes.

♦ A filter-drier is here installed at L1. In this case, it is designed for bi-directional flow as it is installed with upward flow.

♦ A sight glass could be installed here as well.

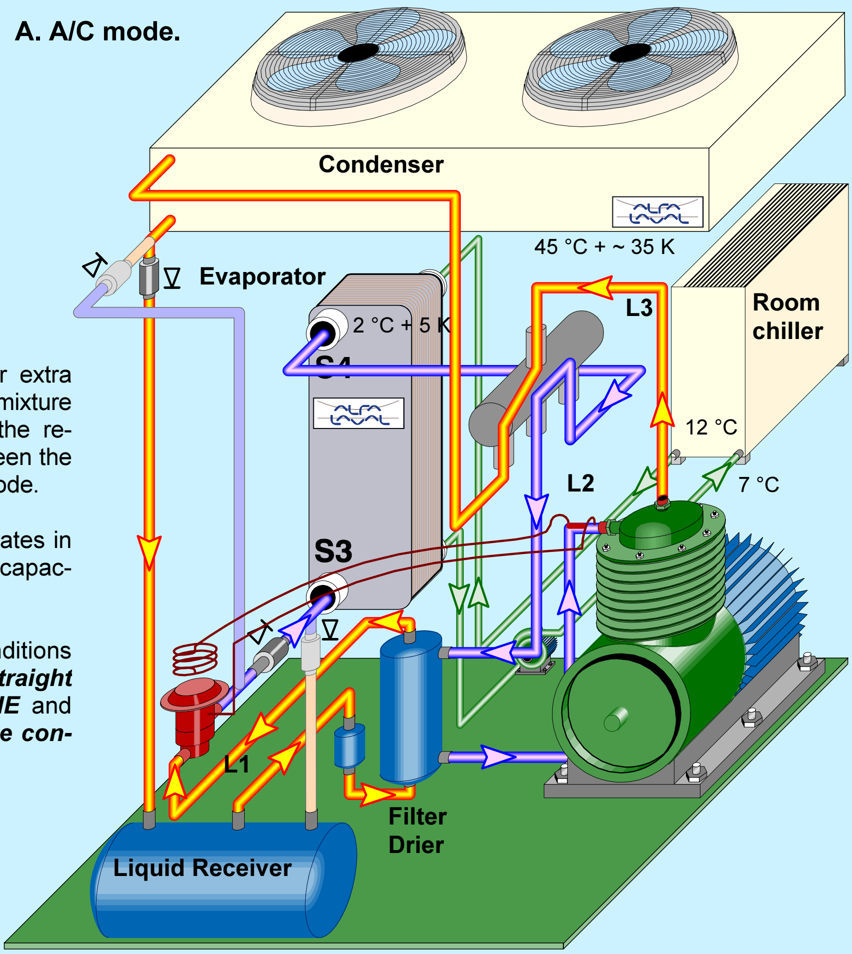
♦ An oil separator could be installed at L3.

♦ A suction separator at L2.

♦ A condensate cooler using the flows at L1 and L2.

Fig. 14. A double TEV system with a liquid receiver.

A. A/C mode.

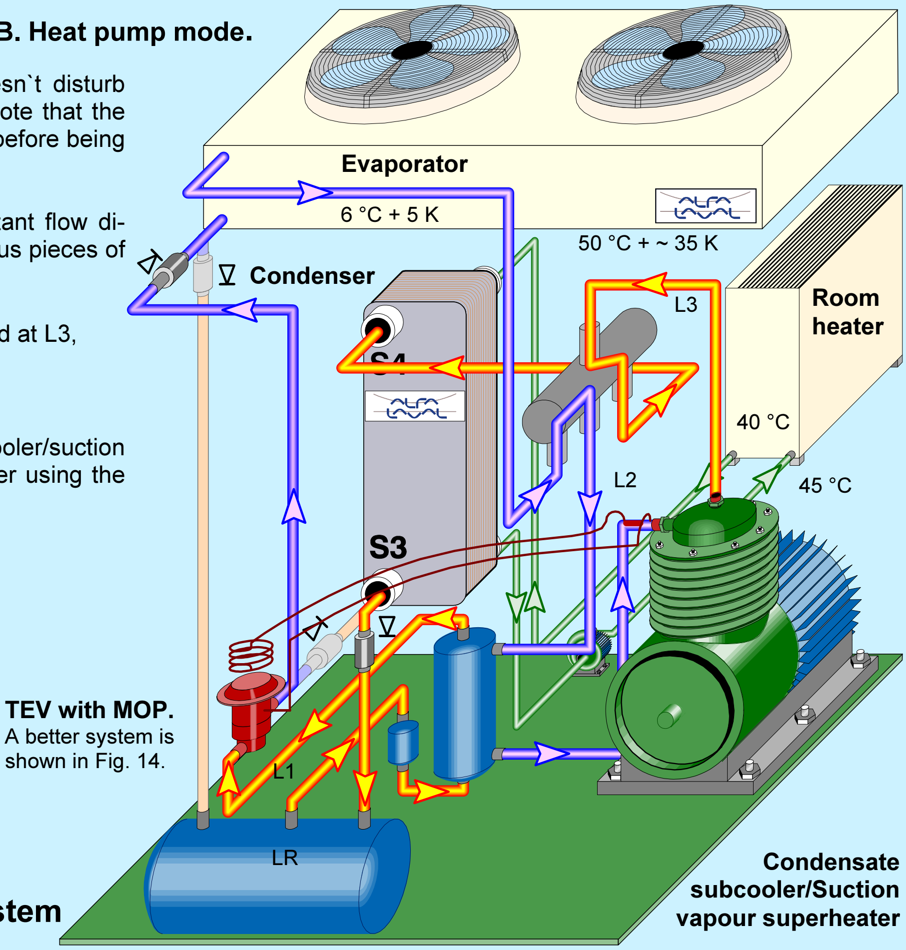


This installation has one TEV and four extra one-way valves to route the evaporating mixture around the LR. Without these valves the receiver would have been positioned between the TEV and an evaporator in at least one mode.

It also means that the TEV always operates in one flow direction thus maintaining full capacity in both forward and reverse mode.

The set-up fulfils the most important conditions for a reliable installation, notably ***the straight pipe between the TEV and the BPHE*** and ***the direct unbroken passage from the condenser to the LR.***

B. Heat pump mode.

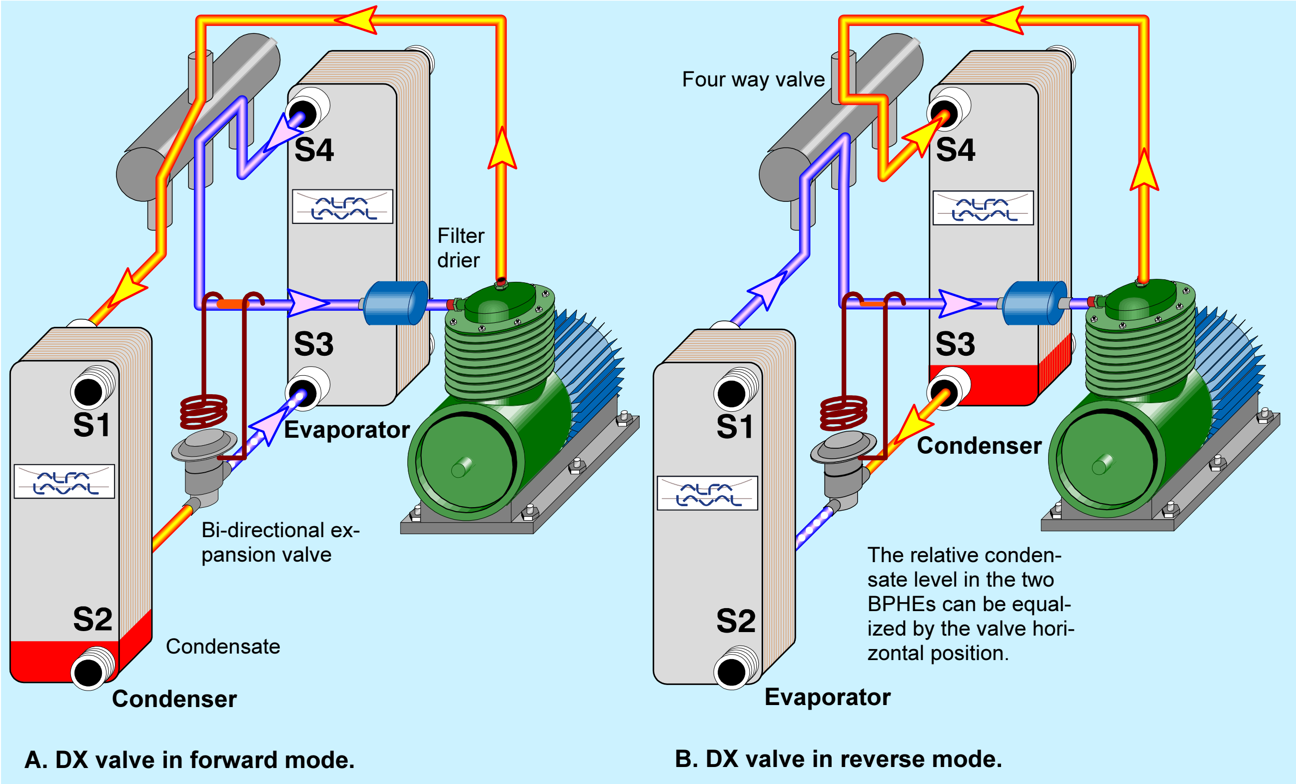


The intermediate check-valve doesn`t disturb the flow, it can even improve it. Note that the condensate pipes could be joined before being connected to the LR as in Fig. 14.

The pipe parts L1- L3 have constant flow direction and could be used for various pieces of equipment.

- An oil separator could be installed at L3,
- A suction separator at L2 and
- Here is installed condensate cooler/suction gas super heater and a filter/drier using the flows at L1 and L2.

Fig. 15. A single-TEV system with a liquid receiver.



The relative condensate level in the two BPHEs can be equalized by the valve horizontal position.

A. DX valve in forward mode.

B. DX valve in reverse mode.

Fig. 16. A single TEV system without liquid receiver.

One valve systems. Instead of two TEVs, two more check valves can be used to direct the flow. The trick is not to introduce too many bends between the TEV and the BPHE evaporator. Figure 15 shows a possible solution. Note that the TEV has a constant flow direction.

14.2.10. Systems without a liquid receiver.

Two valve systems. If the condensate level in the LR in figures 14 and 15 remains constant, the LR can be omitted and replaced by connecting pipes and we have a system with one (or two) TEVs.

One valve systems. Figure 16 shows a system with two BPHEs, which fulfils all the requirements above. Note:

- ♦ The refrigerant volume is small in a BPHE, both as condenser and evaporator. If the two BPHEs are roughly the same size the difference in volume in forward and reverse is small and can easily be adjusted by the horizontal position of the valve.

- ♦ The only liquid line is the short pipe between the condenser and the valve. The total filling is thus small.

- ♦ A slight bend or better, installation of the BPHEs at different height is necessary if the valve is not of the inline type. Note! An inline valve might have a much larger bend, which is not desirable.

The TEV has normally a somewhat smaller capacity in the reverse direction than in the forward. The reason for this is that in the forward direction, the flow helps open the valve; in reverse, it helps close it.

The valve should thus be installed in the forward direction for the principal mode, whether cooling or heating.

- ♦ The filling of the system and positioning of the valve has to be done carefully in order for the valve to have sufficient liquid head in both directions. This is facilitated by the almost perfectly symmetrical design.

Figure 13 shows another one-valve system, this too without LR. This system consists of a BPHE and an air unit. This is at times placed far from the TEV. Long pipes, bends and fittings can then not be avoided. Air units are fortunately less sensitive to the TEV installation.

- ♦ The BPHE is located according to the suggestions above. When it operates as a condenser, the liquid should fill the porthole entirely. The line from the valve to the air evaporator is filled by two-phase mixture.

- ♦ At reverse flow, the single-phase liquid volume increases but this is easily taken up by the pipe between the valve and the air condenser.

Note! A system with two very different units and very asymmetrical, i.e. different amount of liquid lines in forward and reverse mode, needs careful balancing and trimming in order to operate without liquid receiver and only one TEV, yet with a faultlessly operating BPHE.

15. Sea water cooled systems.

Figure 17 shows a system where sea water cools the condenser cooling water in a PHE, a SECOOL system. The cooled fresh water is then piped to various A/C units.

In figure 18 sea water, directly cools the PHE condenser in a central A/C plant, which delivers chilled water, directly or via pressure breakers to room fan coils.

This section discusses the advantages and disadvantages of indirect SW system vs. direct SW systems.

There is also a short description of the equipment and use of sea, brackish or river water as cooling media in either system. The term SW is here used to denominate all types of water, if nothing specifically indicates the type. In the optimization chapter, the two systems are studied from the perspective of how to optimize the total temperature difference between the room air and the sea water as well as SECOOL optimization.

15.1. What is a SECOOL system?

The name derives from SEcondary COOLing, a term coined by Alfa Laval to describe the cooling of a primary fresh water circuit with secondary water, usually sea or river water. The primary water is then used for cooling in industries such as chemical process industries, power stations, steelworks, refineries, onboard ships, etc.

Of special interest to the refrigeration engineer is cooling of water in water-cooled refrigerant condensers. In many places, especially in Hong Kong, Singapore, Shanghai, Japan and other Far East locations the summer is hot and humid and fresh water is scarce, and, therefore, conditions are not suitable for cooling towers. Sea water is often readily available, making a suitable heat sink. Figures 17 & 18 show typical installations in a high-rise building.

The two gasketet PHE´s with titanium plates cools circulating fresh water, which is then used to cool the refrigerant condensers in the A/C units.

The advantages of the system are many. No fresh water is consumed as in a cooling tower. The fresh water or more commonly, demineralised water is kept in a closed circuit, free from oxygen and other corrosive products. The plant equipment only makes contact with the noncorrosive and nonfouling fresh water. The corrosive and fouling sea water is contained in the SECOOL GPHE.

The only disadvantage with the system is the availability of sea or river water. If available, the system is viable; a kilometre from the water source, it is probably not.

15.2. Direct vs. indirect systems.

In a high-rise building, the pressure in a water column from the basement to the roof can be large, far too large to safely operate the fan coils. The fan coils in a group of floors are supplied with circulating chilled water from a substation on each service floor. The service floors are placed at regular intervals, typically starting with the 13th (which doesn't show up in the lifts in e.g. a hotel). There are many ways to arrange such systems.

- The simplest and the most effective use of the available temperature difference between the room air and the sea water would be to have a SW cooled condenser in an A/C plants placed midway in the building. One FW circuit would feed the upper part of the building, another the lower. The drawbacks are that the corrosive and fouling SW enters the machine room and there is limit on the height of the building.

- In the system shown in figure 17, circulating FW is cooled by SW. The FW cools the condensers in the small, local A/C plants, placed in the service floors.

  The FW circuit has to stand the full pressure from the top of the building to the basement. There is also a loss of temperature difference as the condensers are fed with water some 3 to 7 K higher than the SW.

- An A/C unit for the whole building is placed in the basement. A FW circuit - from the basement to the top - delivers chilled water to the fan coils via a PHE ("pressure breaker"). A pressure breaker is a small installation and could be installed in a service room on each floor or grouped together for a number of floors.

  The advantage of the system is obvious, all the machinery is kept in one place in the basement and only chilled water is delivered to the building.

- The disadvantage is that the A/C unit has to deliver chilled water some 3 to 7 °C colder than the previous in order for the fan coils to operate with the same temperature program but on the other hand, the condenser has a higher water temperature.

15.3. Components.

Water intake. In figure 17, the water is lead through a short canal to the basements where the sea water pumps are situated. The advantage of such a system is the low civil engineering cost.

The disadvantage is the sometime excessive amount of debris found close to shore. Water can then be taken from an intake tower some distance from the shore and some meters from the bottom and piped to the pumping station. The water at such a point is usually much cleaner than either at the surface, the bottom or the shore.

Moreover, it can be piped to a pumping station some distance from the shore and the system then doesn't occupy a possibly useful shore area.

Strainer or grate. A grate or a self-cleaning strainer prevents large debris from entering the pumping station.

The sea water pumps. In figure 17, the pumps are placed lower than the water surface, thus with a positive inlet pressure and no priming system is necessary.

It is also possible to place them somewhat higher than the surface but then a priming system is necessary.

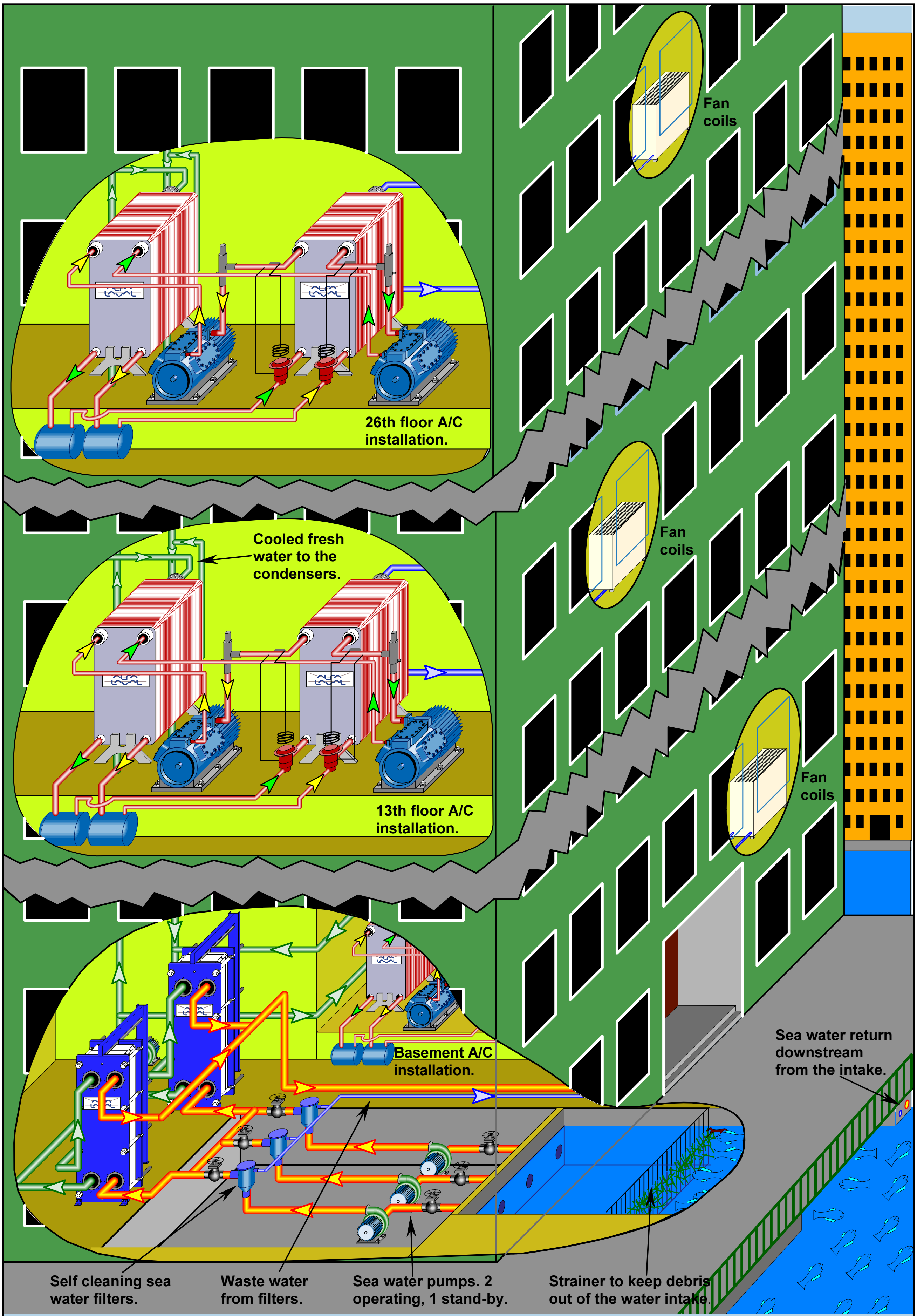


Fig. 17 A secondary cooling (SECOOL) system to cool circulating fresh water with sea water. The fresh water is used to cool condensers in a high rise build-

SW pumps are usually made with bronze impeller, stainless steel shaft and bronze or cast iron casing. Large pumps should have double suction inlet, i.e. the SW enters the impeller from both sides, fig. 18 shows a single.

Self-cleaning filters. See figure 17. Various types exist. Alfa Laval has developed a filter especially for use with PHEs and for sea or river water, see special documentation. The casing is rubber clad mild steel and the mesh of stainless steel. The smallest size has a connection of 100 mm. It is a stationary filter, where the cleaning action comes partly from flow reverse, partly from shearing forces when the water is made to flow along the sieve surface. The cleaning takes place automatically at predetermined intervals or controlled by the pressure drop. The absence of moving parts (except at the flow reverse) makes the filter very dependable.

At the flow reverse the debris is flushed out in a small waste water pipe, which preferably connects to the main SW line after the SECOOL PHE.

Rotary types exist which cleans continually, but some types have a tendency to be stuck by the debris.

Back flushing. Small debris, seaweed, fish, etc., which have passed the water intake can, in the absence of a filter, tend to be stuck at the PHE inlet. A simple way of getting rid of the debris is to reverse the flow in the PHE and flush them out, see figure 18. This is done automatically, usually four to six times a day, 15 to 30 minutes each time.

Note 1. At the flow reverse, the flow changes from normally counter current to cocurrent. There might thus be a temporary decrease in capacity.

Note 2. Except in very special cases, it is not necessary to install both a filter and a back flushing system.

Note 3. A back flushing system has no particular waste water pipe, which simplifies the installation, not the least from a legislative viewpoint.

Note 4. A direct cooled condenser should have a better antifouling system than an indirect as an opening involves the refrigerant side, which is costly and time consuming.

Chlorination. Sea and brackish water contain a multitude of marine life, algae, fungi, barnacles, bacteria, etc., fresh water somewhat less. The larger organisms are caught in the filter but smaller pass. Barnacles and mussel larvae can pass almost any type of sieve, settle on the walls, grow and finally detach and clog the condenser. HE of mild steel cooled by fresh water can harbour iron-eating bacteria. Especially in winter, the condenser is a warm cosy place to hibernate in.

An effective way of controlling unwanted marine growth - where legislation permits - is chlorination, especially shock chlorination at irregular intervals as e.g. barnacles tend to memorize regular intervals and then close. The extent of chlorination is difficult to predict, marine activity varies from site to site, season, time of the day, etc. 15 minutes, four to six times a day to a chlorine content of 1 - 2 ppm ***after*** the PHE is a suitable starting value. The injection point should be immediately before the PHE. If far away, all the microorganisms in the pipe between the injection point and the PHE will detach and clog the PHE.

There are three principal chlorination systems in use:

- Direct injection of liquid chlorine, the preferred system in chemical industries, where knowledge exists on how to handle liquid chlorine, (but probably nowhere else).
- Injection of a water solution of calcium or sodium hypochlorite. It is an easy and safe but somewhat messy method to chlorinate, as the solution is highly corrosive.
- The most elegant method is electrolysis of sea water. Electrolysis of a NaCl solution yields chlorine and hydrogen, right where it should be, in the water. It is expensive, but the advantage is that there is no storage of a corrosive and/or dangerous liquid.

Chemical cleaning. SW contains not only microorganisms but also pollutants, such as oil, smoke, and dead organic matter. All of these can attach to the surface and have to be removed. Cleaning must also be used if chlorination for one reason or other cannot be used.

In a CIP (Cleaning In Place) system, the SW flow is shut off and cleaning solutions are circulated through the PHE. Alfa Laval deliver two types of cleaning solutions especially developed for PHE, see special documentation.

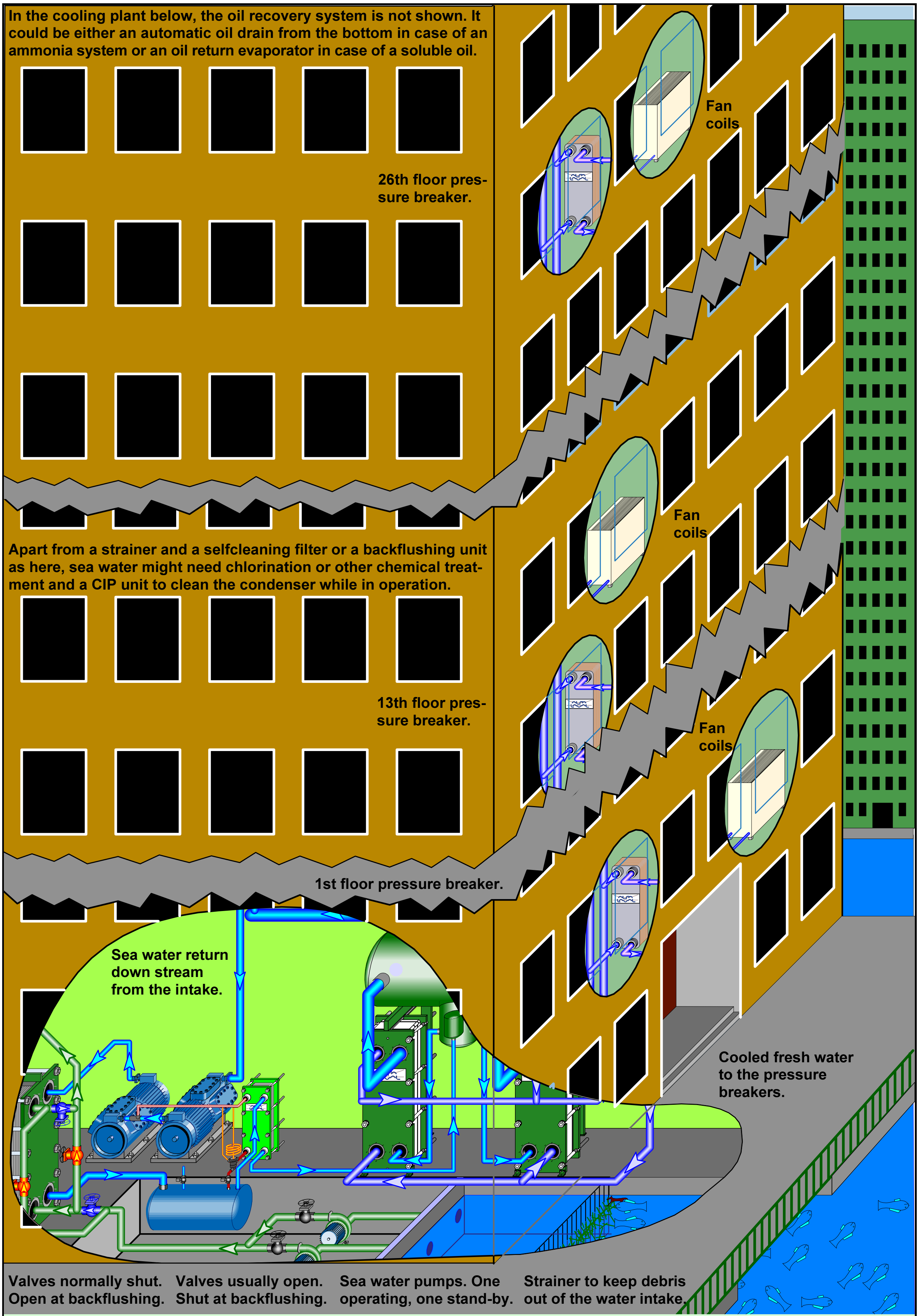
One is a type of industrial detergent, which is effective against various types of organic fouling. The other is an acid solution, which dissolves calcium or magnesium carbonate scaling, the most common types in cooling water.

Manual cleaning. Usually it is advisable to open a GPHE once a year for manual inspection and cleaning. The intervals can vary though, from several times a year to once every ten years. If an annual cleaning, it should be done just before the warmest season, i.e. when the load of the condenser is the largest, it is recently cleaned.

The PHEs. Only GPHE & WPHE are used for SW. The material in a SW PHE, whether a condenser (WPHE) or a SECOOL PHE (GPHE), should be titanium as this is fully resistant to sea water and to most other chemicals. The exceptions are fluorine compounds either in solution or as compounds in the gaskets, e.g. fluoro rubber - Viton - that never should be used in titanium PHEs. There are some high quality stainless steels, e.g. Avesta 254 (and 654) SMO, which are resistant to brackish water but ***not in combination with chlorination.*** See the section on **6. Fouling & Corrosion.**

Note 1! A SWPHE for R22 cannot operate with a higher ***continuous*** vapour temperature than about 90 °C. For ammonia the continuous temperature is above 120 °C.

Note 2! ***CuNi is not resistant to polluted sea water*** found in harbors and industrial areas. It can be used on ships, which spend most of the time at open seas.



In the cooling plant below, the oil recovery system is not shown. It could be either an automatic oil drain from the bottom in case of an ammonia system or an oil return evaporator in case of a soluble oil.

Apart from a strainer and a selfcleaning filter or a backflushing unit as here, sea water might need chlorination or other chemical treatment and a CIP unit to clean the condenser while in operation.

Fig. 18. A sea water cooled condenser in an A/C system. Chilled water is supplied to the pressure breakers, which serve the fan coils in a floor or a group of floors.

16. Soluble oil in flooded systems.

16.1. The oil evaporator.

The vapour-liquid mixture leaving the evaporator enters the separator, where the vapour is separated from the liquid and possibly further dried in a demister. This separation not only separates the liquid refrigerant from the vapour, but also the oil from the vapour.

Only the pure dry vapour is then returned to the compressor and thus oil entering the evaporator-separator loop will effectively be trapped there. If no special measure is taken, the oil concentration will gradually increase in the evaporator, depleting the oil content in the compressor.

The first step to be taken is installation of an effective oil separator after the compressor, figure 19.

However, this is not enough. Even a highly effective oil separator cannot capture all the oil; some will inevitably leave the separator and ultimately the oil is concentrated at the evaporator-separator loop.

The second step is then installation of an oil return evaporator; figure 19.

Here some liquid refrigerant is drawn off from the evaporator-separator loop and evaporated as completely as possible. Left in the vapour leaving the oil return evaporator are then only oil droplets containing some refrigerant, but in quantities small enough not to cause liquid hammering.

HP condensate is normally used as heating medium.

The refrigerant drawn off has to be balanced so that all the oil entering the separator also leaves it. This inevitably means that the oil concentration in the evaporator-separator loop is many times higher than in the feed.

Ex. Suppose that 5000 kg/hr of an R 22-oil solution containing 5 kg/hr oil, i.e. 0.1 %, enters the evaporator-separator loop from the condenser/liquid receiver.

What will the oil concentration in the separator loop be?

Entering the loop is 5000 * 0.001 = 5 kg/hr oil. All this oil has to leave with the 1000 kg/hr evaporating R 22-oil solution. Thus the concentration is:

100 * 5/1000 = 0.5 %

Of the 1000 kg/hr liquid entering, 99.5 % evaporates, i.e. the remaining 0.5 % is the oil. Thus, we can say that the oil concentration in the evaporator-separator loop is about five times higher than in the liquid entering from the condenser/liquid receiver.

The oil return evaporator resembles the direct expansion evaporator in that all the refrigerant should be evaporated at the exit. The oil concentration is much higher and the available pressure drop is normally much smaller.

This means some difficulties in designing the evaporator, as the physical properties vary from those of almost pure refrigerant to almost pure oil. Especially the viscosity shows a very large variation. The evaporation should be divided into various zones from inlet to outlet, each designed with its proper physical properties. This is unfortunately a very tedious task.

However, when 95 % of the liquid has evaporated, the oil concentration has increased to 10 %. Up to about this point, the increase in viscosity is reasonably small.

The remaining part, when the oil concentration increases from 10 to almost 100 %, is the difficult one. Fortunately, the amount of liquid is small here, and the evaporation process is liquid droplets in a gas, i.e. the liquid physical properties have little influence. In practice, it is possible to design the unit with mean physical properties or as a pure refrigerant with a large margin (25 - 50%).

The design of an oil return evaporator is not very critical. Suppose that the unit is underdesigned by 50 %. This would mean an oil concentration of 0.75 %, still a manageable value for the main evaporator.

Finally, of the 5000 kg/hr mixture entering the evaporator-separator loop, 1000 kg/hr is drawn of for the oil return evaporator. Will this decrease the plant capacity?

No, as the heat needed to boil of the refrigerant is taken from the condensate, which then comes in cooler and contributes to the cooling load.

Figure 19 shows the oil return evaporator. A flow is drawn off from the separator, evaporated, and an oil mist returns to the compressor. Thus, the main properties of the evaporator are:

- An as complete evaporation as possible of the oil-refrigerant solution.
- Return to the suction line of an oil mist only.
- The driving force for the flow can be the static head as in a normal thermosiphon.
- The static head might not be enough to force the oil mist into the suction line. If the line from the oil evaporator enters the suction line after a pressure drop producing valve, the same pressure drop has to be added to the oil evaporator as driving force.
- The entrance to the suction line could be as an ejector, producing some extra driving force.
- The inlet to the oil evaporator could be made from the discharge pipe of the pump in forced flow system.
- The flow is controlled by the superheat but without flashing of the refrigerant as in a normal DX system. The control valve simply regulates the flow so that no more enters than what can evaporate. The allowable pressure drop is low, i.e. distributor BPHEs can probably not be used. Similar components as in a DX system can be used, though.

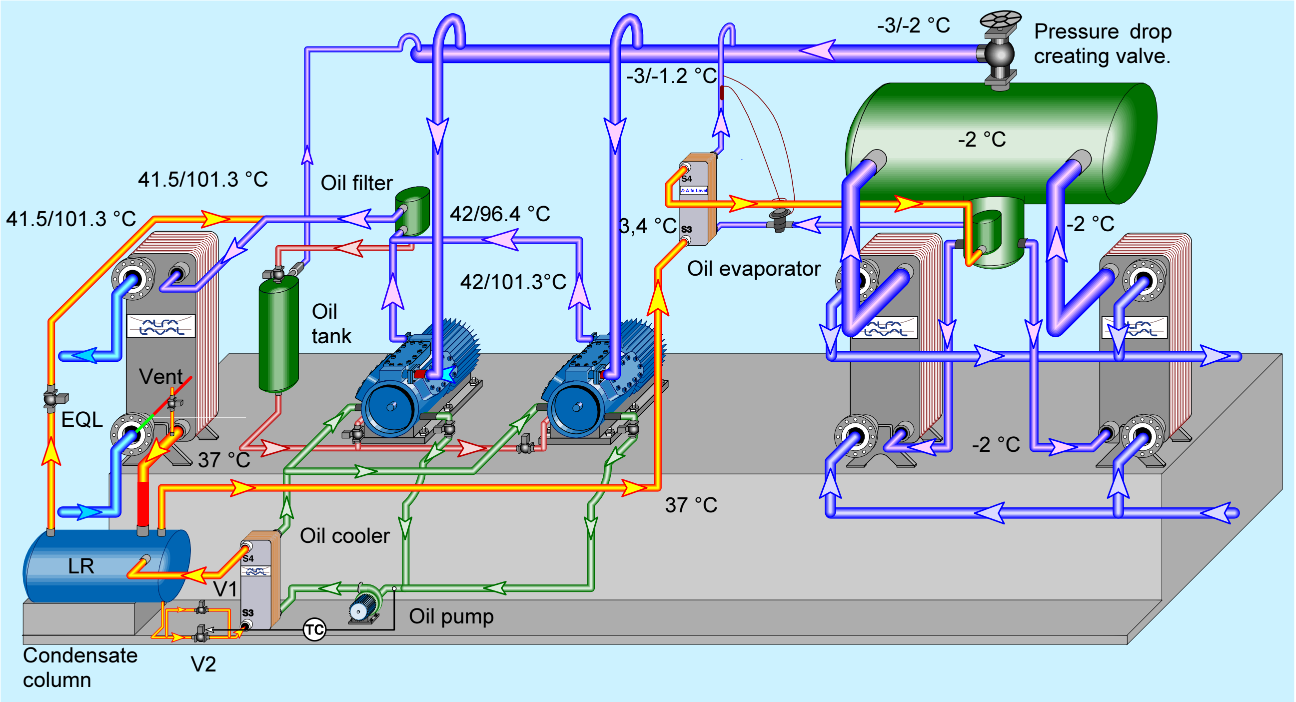


Fig. 19. Oil cooler and oil return evaporator in flooded system with soluble oil.

16.2. The oil cooler.

The system is the same for both a soluble and an insoluble oil. The oil is cooled with evaporating warm condensate. The other possibilities - cold refrigerant or water - give larger thermal shocks.

After the evaporation, the vapour is condensed in the condenser again. It returns to condenser via either:

A) The LR and the equalization line (EQL), figure 19.

B) Directly to the condenser inlet.

C) Back through the condensate pipe to the condenser exit and into the channels from below.

In A) & B) this means that the refrigerant moves in a circle. For this to be possible in a condenser with a fairly high pressure drop, the following conditions must be met:

♦ The LR pressure has to be higher than the condenser inlet pressure, otherwise vapour cannot flow back. This is easy; the condensate is heated in the oil cooler until the vapour pressure is large enough to press the vapour back to the condenser inlet.

♦ The vapour must not enter the condensate pipe. The condensate pipe in the LR thus has to be drawn to below the lowest condensate level encountered.

♦ To over come the pressure drop in the condenser there must be a sufficient gap between the liquid level in the LR and the bottom of the condenser, to allow a liquid column to build up.

What happens at the start of the oil cooler? Suppose that the EQL is closed.

♦ When the EQL opens, the condenser inlet pressure transfers to the LR. The pressure pushes the condensate into the condensate pipe. When the static height in the column equals the condenser pressure drop, equilibrium is reached and there is no flow in the EQL. The pressure is equal in the LR and at the condenser inlet.

♦ The refrigerant valve to the oil coolers opens and refrigerant evaporates. The pressure increases enough for the vapour to be pushed through the EQL to the inlet. The liquid column increases correspondingly.

♦ If the refrigerant contains inert, it has to be vented at the upper part of condensate exit pipe, not at the LR.

In case C) no EQL is necessary and the condensate pipe discharges above the condensate level in the LR. Vapour moves in counter current to the condensate and enters the channels.

♦ ***The liquid column, necessary to overcome the pressure drop in the channels, then forms in the channels. Parts of the channels are then submerged and are rendered useless as a condensing surface.***

♦ ***It is difficult or impossible to vent the condenser as the venting point is inside the channels.***

To minimize the danger of thermal shocks, the oil is running all the time. A small stream of refrigerant is constantly flowing through the cooler through the loop V1, ensuring the minimum cooling requirement. When more cooling is needed (controlled by TC), the parallel-connected valve (V2) opens and lets in enough refrigerant to meet the maximum cooling demand. A complete on/off system would cause more stresses on the system.

2. Optimization of Plate Heat Exchangers in Refrigeration Systems.

1. What is optimization?

Picture a simple refrigeration system. Brine enters at a given temperature and leaves at a lower temperature while cooling water enters at a low temperature and leaves at higher temperature. The system has a specified capacity. Although we have some freedom in changing the brine inlet temperature, the cooling water exit temperature and the flow rates, three parameters are normally fixed - the brine exit temperature, the available cooling water temperature and the capacity.

Our aim is to design an optimal system within these limits. The question is what constitutes an optimal system and whether we can obtain this in practice. Unfortunately, the term optimal is not a strictly defined quantity. Below is a somewhat arbitrary definition.

- ***Total optimization.*** This might be defined as a system with as low an operating cost as possible for a given equipment cost. The sum of the annual investment cost and the operating cost should then be minimized. Usually, such an analysis gives an optimum for a high capital cost and a low running cost. In practice, however, plant operators usually prefer to decrease the capital cost to the detriment of the running cost.

  A total optimization is costly and time consuming to do. Normally there is not enough data available on how a change in one parameter affects the others, which makes an exact optimization close to impossible.
- ***Cycle optimization.*** The designer decides if to install an economizer, a two-stage compressor, a subcooler, type of refrigerant, etc.
- ***Equipment optimization.*** Does it pay, to make the evaporator a little smaller but the condenser larger? Can a smaller compressor be used but with a larger TEV and evaporator? Can the evaporator be made smaller if the coolant pressure drop is increased? These questions can often be answered without too much trouble and constitute the theme for this chapter.

1.1. Equipment optimization.

Optimization means changing the parameters of one component to fit better with another or to the whole system. In order to make an exact analysis of a system, good design programs for all the equipment, heat exchangers, compressors, piping, valves, etc. are needed.

The question is then which components have to be constant and which have to change. Of the major components, compressors, valves & other control equipment, pumps, and pipes & fittings heat exchangers we find that:

- ***Compressors.*** These are made in fixed sizes. The jump in cost from one size to another might be considerable.

  Taken all available compressors from all manufacturers together, the change in capacity might be considered continuous, but usually the selection is restricted. A given compressor can operate under various conditions. However, a compressor has an efficiency maximum, which occurs when the required discharge pressure is the same as the one the compressor actually delivers and thus no throttling takes place.

  Therefore, the compressor selection has to be restricted to a few nominal capacities and to limited design conditions. It is virtually impossible to say what an incremental change in size would mean for the plant.
- ***Control equipment.*** Once the plant capacity and layout are decided upon, the cost of the control equipment does not change for minor changes in the operating condition, i.e. a pressure control valve does not change when the pressure changes.
- ***Pumps, pipes and fittings.*** The same is valid here. A small change of flow normally only means a change of pump impeller and the pressure has to change considerably before a pipe diameter is changed.
- ***Heat exchangers.*** A small change in the evaporation pressure might not mean much for the pipe, TEV or the pressure controller, but it could have a dramatic effect on the evaporator size.

  Fortunately, PHEs lend themselves very well to optimization. Particularly suitable are the SWPHEs, which can be increased in size with one plate at a time, i.e. a practically continuous change. Moreover, this change can be made after installation should the need arise for an increase in capacity. AWPHEs and BPHEs come in fixed sizes, but it is possible to tailor them individually, with a change of one plate at a time.

1.2 . Conclusion.

Valves & other control equipment, pumps, and pipes & fittings do not change cost for minor changes in the operating parameters and are thus not optimized. The compressors come with fairly large steps between the nominal capacities. PHE’s can be almost infinitely changed and should thus be adjusted to the compressor and not the contrary. The study is divided into three parts:

- The thermal and hydraulic properties of PHEs, which can said to be the optimization of the PHE without regard to other parts of the system including other PHEs. This forms the basis for the next steps.
- Secool optimization from the application for which this concept was originally developed.
- The interaction of PHEs with other parts of the system notably other heat exchangers, which can be said to be optimization proper.

2. The thermal & hydraulic properties of PHEs.

2.1. Purpose of the study.

To understand how a PHE can be used to optimum efficiency under given conditions, it is important to know its thermal and hydraulic behaviour. It is obviously of no use to allocate a higher pressure drop to a PHE if it cannot use it, i.e. be reduced in size or give a higher capacity.

An excellent way to illustrate the properties of a PHE is to study how the total surface varies, when water flow rate moves from zero to infinity as illustrated in the example.

2.2. The thermal duty.

The exact duty, margin or pressure drop is not important, but it is easier to discuss a problem with real figures than with symbols. Although, we have used a water duty, it could, as easily be a condenser, an evaporator, glycol,etc.

Q = 156.2 kW, Margin, $M \geq 5$ %.
1: Water, 10 kg/s 12 -> 7 °C, ΔP: Free
2: Water G_2 kg/s ? <- 2 °C, $\Delta P \leq$ 45 kPa

Nominal 10 kg/s

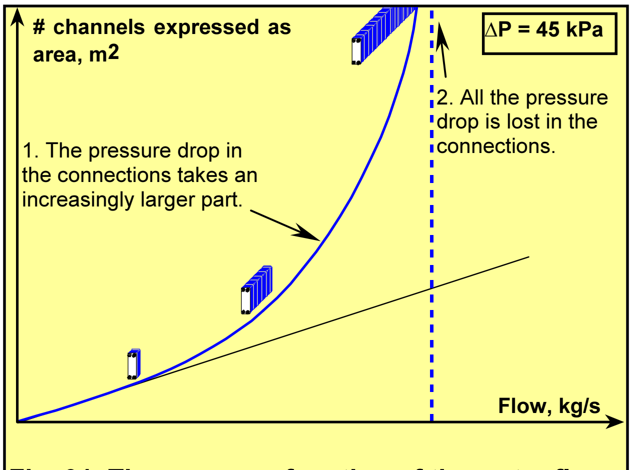


Fig. 01. The area as a function of the water flow.

2.3. The optimally designed PHE.

This occurs when:

- The margin, M, is exactly the required 5 %, i.e. the installed area is 5 % larger than that calculated.
- The pressure drop should be fully utilized, i.e. equal to the required 45 kPa.

We will see how and if these conditions can be fulfilled. This is the best heat exchanger for the given conditions. The conditions might not be optimal for the plant, though. We will study how to optimize the conditions later.

2.4. The water flow changes.

Then how does the total surface change when the water flow rate, X, moves from zero to infinity. We will study the behaviour from two aspects; either the requirement on the pressure drop or on the margin is fulfilled.

2.5. The pressure drop.

This shall not exceed 45 kPA when the water changes from zero to infinity. There is no requirement on the thermal performance. See figure 1. This is simple. When the water flow is zero, the number of plates - and the area - is zero. When the flow increases, plates - or more correctly, channels - are added. At the beginning, the area is thus roughly a liner function of the flow. Roughly, as the increase of surface is of course in definite steps, one channel at a time.

The curve would look like a staircase but for the purposes of this study, we consider the curve continuous.

With increasing flow, another effect enters: the pressure drop in the connections. This reduces the pressure drop available for the nozzles and proportionally more channels have to be added; the curve deviates upward from the straight line.

At a certain flow rate, the entire available pressure drop is lost in the connections, thus none is left for the channels, i.e. an infinite number of channels is necessary as indicated by the asymptote.

However, long before this happens, a second unit has probably been added. The addition of another unit decreases the pressure drop in the connections and more is available for the channels. The number of channels then suddenly drops, see figure 2.

The flow continues to increase and a third unit is added, causing a drop in the number of channels. This is repeated with the addition of a fourth, fifth unit. The curve gradually gets smoother and smoother and increasingly linear as the flow increases and units are added.

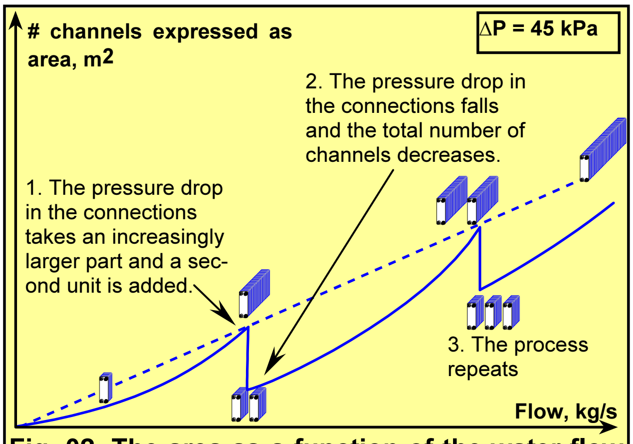


Fig. 02. The area as a function of the water flow for parallel PHEs.

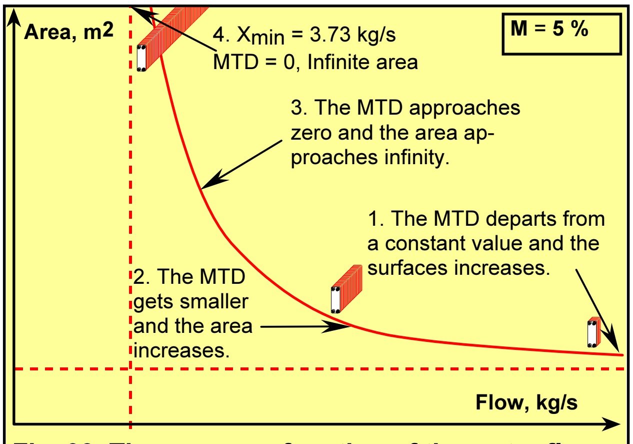


Fig. 03. The area as a function of the water flow.

Note! The warm side intentionally is not taken into consideration at this stage; this will be done later.

2.6. The margin.

This must not be lower than the required 5 %. There is no requirement on the pressure drop. See figure 3.

It is easier to see what happens, if we start with an infinite water flow and let it decrease.

Note! In the previous study, we had to add ***channels*** in order to keep the pressure drop. Here we have to add ***area*** to keep the thermal duty.

For an infinitely large flow, the exit temperature is equal to the inlet and the MTD has its maximum. This means a small area, large channel velocities and a high K-value.

A decreasing flow has two effects and both mean an increasing area:

- The MTD decreases, at first slowly then rapidly.
- The channel flow rate and thus the K-value decrease.

It is easy to see that there is minimal water flow. A still smaller flow would mean a water exit temperature higher than the warm side inlet temperature. How large is it?

With an infinitely large heat exchanger, the water can be heated to 12 °C, i.e. a temperature increase of 10 K. This means a flow of

$X = 156.2 / (4.186 * 10) = 3.73$ kg/s

In the pressure drop case we could reduce the area by adding units. Can we do something similar here?

The main reason for the large area is the increasingly small MTD. We cannot improve the MTD, for given flow rates and temperatures. On the contrary, a heat exchanger could impair the true counter current MTD, even though PHEs are good in this respect.

However, the other reason for the increasing area is the low K-value, caused by the decreasing channel velocity. By splitting the area between two units and connecting them in series, the channel velocity is doubled, resulting in an increased K-value and thus smaller area. See Fig 4.

For still lower flow rates the area can be split between three, four, ... units in series. This reduces the rate of the increase of area somewhat but when the temperature difference approaches zero, the area approaches infinity.

2. 7. The complete thermal and hydraulic programs.

Obviously, we cannot design a unit by only taking the pressure drop or margin into consideration, both have to be satisfied simultaneously.

To do this, we plot the pressure drop curve from figure 2 together with the area curve from figure 4. See figure 5.

At the intersections of the curves, the pressure drop is equal to the required 45 kPa and the margin is 5 %. The minimum area - the optimal PHE - occurs here. Valid heat exchanger configurations are then marked by the full drawn curves. The greyed out parts of the curves are not valid if both requirements - a pressure drop $\leq$ 45 kPa and a margin $\geq$ 5% - shall be fulfilled simultaneously. We will now study the curve in detail. A calculation of brazed units for these duties gives:

a) The nominal duty:

Water, 10 kg/s 12 -> 7 °C, ΔP_1: Free
Water 10 kg/s 7 <- 2 °C, ΔP_2: $\leq$ 45 kPa
Two (CB76-64H, 6.2 m2), M = 54%, $\Delta P_{1/2}$ = 41/43 kPa
The margin here is far higher than necessary.

b) A reduced chilled water flow duty:

Water, 10 kg/s 12 -> 7 °C, ΔP_1: Free
Water, 6.25 kg/s 10 <- 2 °C, $\Delta P_2 \leq$ 45kPa
Two(CB76-96H, 9.4 m2), M = 5 %, $\Delta P_{1/2} \leq$ 20/9 kPa
The pressure drop is much lower than allowed.

c) An optimal design. This requires some trial and error:

Water, 10 kg/s 12 -> 7 °C, ΔP_1: Free
Water 7.7 kg/s 8.5 <- 2 °C, $\Delta P_2 \leq$ 45 kPa
Two (CB76-48H, 4.6 m2), M = 6 %, $\Delta P_{1/2} \leq$ 68/45 kPa
The margin and the pressure drop are as required.

2.8. Discussion of the results.

See figure 6. It is basically the same as figure 5, but the water flow on side two is now expressed as a fraction of the flow on side one. This makes the curve a little more general. The three cases a) to c) are marked on the curve. We can draw the following conclusions:

- At the right part of the curve - the blue - the pressure drop is the specified 45 kPa, while the margin is larger than specified. Case a) occurs here. Duties with very large flow rates and/or low pressure drops on one or both sides commonly fall on the right part.

  The area, actually the number of channels, is determined by the pressure drop requirement.

- At the left part of the curve - the red - the margin is the specified one, 5 % in this case. The pressure drop is smaller than the specified 45 kPa. Case b) occurs here.

  Heat transfer duties with very large heat recovery or limited amount of water, i.e. close temperature difference, fall on the left part of the curve.

  The heat transfer determines the necessary area.

- In one point - case c) - the pressure drop as well as the margin is equal to the specified values.

  The area has a minimum when both conditions are fulfilled.

- In the actual calculations a) to c), the values are close to the required 5% and 45 kPa, but not exactly.

  The addition of area is not continuous but occurs in discrete steps, e.g., when a plate is added, the margin might jump from less than 5% to more than 5%.

- The minimum area does not necessarily occur for equal flow rates on the warm and cold sides.

  It is a common misconception, that a PHE should have both flows equal, in order to be optimal.

2.9. Further properties of the curve.

The previous case is valid for practically all types of heat transfer duties, but there are some questions:

- What effect has the pressure drop on the warm side?
- What effects have the pressure drop and margin?
- How does the physical properties affect the curve?
- How do the temperature difference affect the curve.
- How do the plate properties affect the curve?

2.10. Pressure drop limit on side 1.

Lets study our example, see Fig. 7. Suppose there is pressure drop limit of 20 kPa. This is what case b) gives. Thus, we need 2 * 96 plates (or rather 2 * 48 warm water channels) to keep the warm side pressure drop. When we increase the cooling water flow, the required area decreases but we still need the 2 * 48 channels on the warm side and case a) & c) are thus equal to b). The result is a constant area, indicated by the black horizontal line, which goes from b) to well beyond case a), where we are obliged to switch over to three units.

2.11. The pressure drop varies.

Fig. 8 shows how a decrease of the cold side pressure drop changes the necessary area and the optimal flow. A pressure drop increase has of course the opposite effect. Thus, we recalculate our study:

Water, 10 kg/s 12 -> 7 °C, ΔP_1= Free
Water ***7.24*** kg/s 8.9 <- 2 °C, $\Delta P_2 \leq$ 30 kPa
Two (CB76-***56***H, ***5.4*** m2), M = 6 %, $\Delta P_{1/2}$ = 52/***30*** kPa

A change of the pressure drop on warm side has no direct effect on the optimal flow, see Fig.7. In this example, we have permitted the warm side pressure drop to be free to study the effects of the cold side pressure drop.

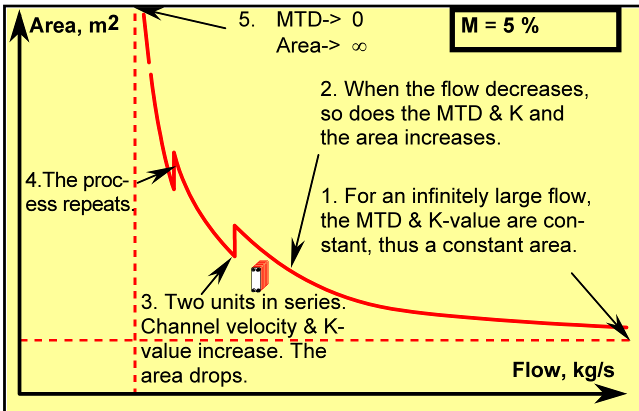


Fig. 04. The area vs. water flow for PHEs in series.

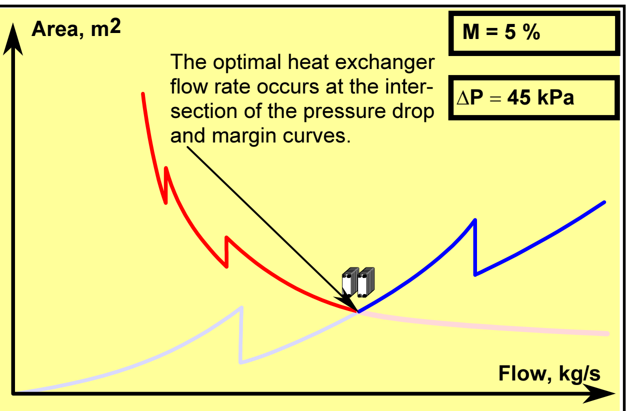


Fig. 05. Both the thermal and hydraulic data are in accordance with the requirements.

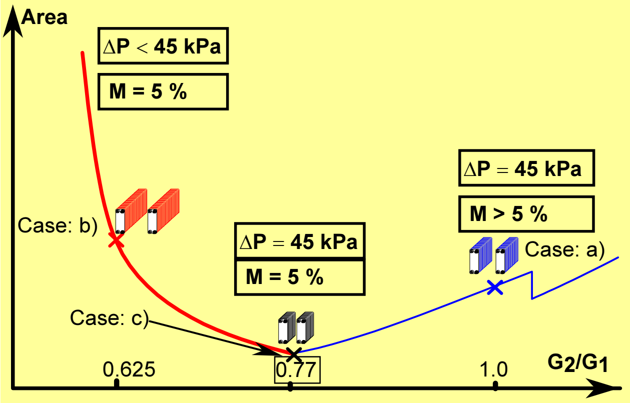


Fig. 06. The general aspect of the area-flow curve.

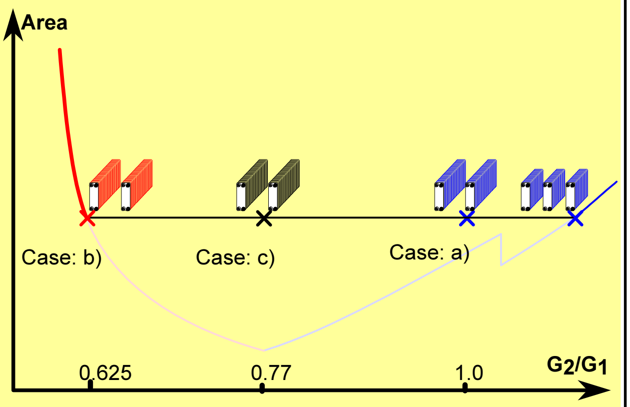


Fig. 07. Pressure drop limit on the warm side.

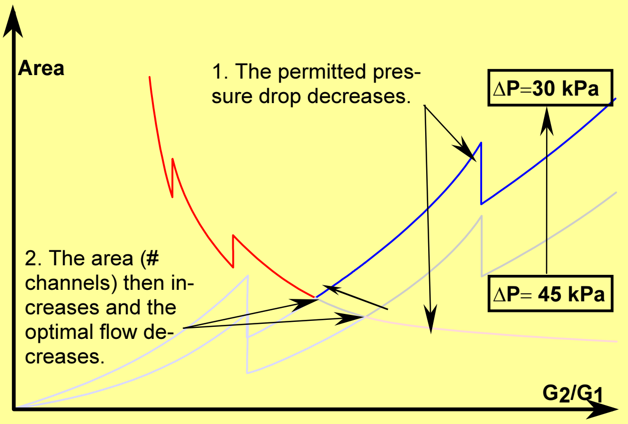


Fig. 08. The effect on the optimal flow, by a decrease of the cold side pressure drop.

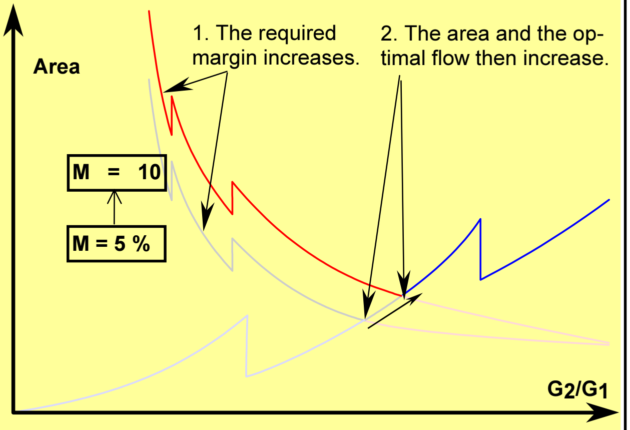


Fig. 09. The effect on the optimal flow, by an increase of the margin.

2.12. The margin varies.

Figure 9 shows how an increase of the margin changes the necessary area and the optimal flow. The increased margin means more plates and a larger flow - and a larger optimal flow - can be accommodated. A decrease of the margin will have the opposite effect.

Water, 10 kg/s 12 -> 7 °C, ΔP_1= Free
Water ***7.93*** kg/s 8.9 <- 2 °C, $\Delta P_2 \leq$ 45 kPa
Two (CB76-***50***H, ***4.8*** m2), M = ***10*** %, $\Delta P_{1/2}$ = 63/45 kPa

2.13. Change in physical properties.

Strictly speaking, the physical properties do not enter the optimization procedure. We cannot optimize a heat exchanger by changing the viscosity, as this is a fluid property. Still, it is sometimes important to know how a heat exchanger reacts to a change, e.g. from water to glycol.

The heat transfer coefficient decreases with increasing viscosity and decreasing thermal conductivity. The dependence on the specific heat and density is less clear and in any case less strong. The heat transfer coefficient increases with increases in both but then less fluid - less channel flow rate - might be necessary.

Anyhow, if we switch to a fluid with higher viscosity and lower thermal conductivity, the area - the number of channels - increases. That would normally mean that the flow could be increased, increasing the MTD and overall heat transfer coefficient, again lowering the area and the number of channels until equilibrium is reached.

If the increase in viscosity was very large, the pressure drop increases and this might force a reduction in flow. The area will increase but the optimal flow rate can occur for a higher or lower flow rate than for the first fluid.

As an example, we will see what happens when we switch to 20% propylene glycol in our case study. The area for the water optimal temperature program is (figure 10):

Water	10 kg/s	12 -> 7 °C,	ΔP_1=	Free
20 % PG	***8*** kg/s	8.5 <- 2 °C,	ΔP_1=	45 kPa

Two (CB76-***70***H, ***6.8*** m^2), M = 5 %, $\Delta P_{1/2}$ = 35/26 kPa

The area - the number of channels - has increased and the pressure drop has dropped to 26 kPa. The margin is the required. It is firmly on the left side on the glycol curve in Fig. 10. An optimization of the duty gives:

Water	10 kg/s	12 -> 7 °C,	ΔP_1=	Free
20 % PG	***8.82*** kg/s	8.5 <- 2 °C,	ΔP_1=	45 kPa

Two (CB76-***56***H, ***5.4*** m^2), M = 5 %, $\Delta P_{1/2}$ = 52/45 kPa

This final heat exchanger is not very much larger than the original water heat exchanger but the optimum occurs for a much larger flow rate.

As a conclusion, we can say that if the physical properties tend to make the PHE larger, then the optimum tends towards a larger flow rate and a larger unit.

2.14. The temperature difference.

How does the temperature program affect the optimal flow? In the example, water was cooled from 12 to 7 °C with chilled water of 2 °C. If we increase the chilled water temperature to 4 °C, the area will probably increase. How will the optimal flow, which gives the minimum area, change? If we change the cooled water inlet temperature from 12 to 10 °C, what will the optimal flow be?

To answer this question we have to normalize the temperature program. It is obviously not the same to cool water from 12 to 7 °C as from 8 to 7 °C, even though the chilled water temperature is the same 2 °C.

We therefore introduce the concept of thermal length. This is the change in temperature in degree Kelvin of one fluid, normally the warm side, per degree Kelvin MTD. Without going into computational details, this makes sense. If we assume a constant overall heat transfer coefficient, then cooling of water from 12 to 7 °C with water from 2 to 7°C is the same as cooling of water 120 to 70 °C with water from 20 to 70 °C.

In one case we cool the water 5 K for a MTD of 5 K, in the other 50 K for a MTD of 50 K. In both cases we get 1.0 K/K. The thermal length, normal called Θ (theta), is here simply 1.0 (dimensionless). The problem here is that to calculate MTD we need all four end temperatures, but here we do not know the cold water exit temperature; it depends on the amount of water used, but this quantity is yet to be decided.

We introduce the concept of Terminal Thermal Length (TTL), figure 11. It is simply the temperature change of the warm water divided with the temperature approach, i.e. the temperature difference between warm side out and cold side in.

The figure shows two different temperature programs, which look rather different, but a close study reveals that the TTL is equal. The exit water temperature varies in both cases between the extreme, but never approached values set by the minimum water flow and infinite flow.

Now, if we change the TTL, how does the optimal water flow change? What other parameters could have an influence? The pressure drop? We calculate some examples:

a) We change the cold-water inlet temperature in our case study to 4 °C, and try to find the optimal flow:

Water,	10 kg/s	12 -> 7 °C,	ΔP_1	= Free
Water	10.9 kg/s	9 <- 4 °C,	ΔP_2	$\leq$ 45 kPa

Two (CB76-***70***H, ***6.8*** m2), M = ***5***%, $\Delta P_{1/2}$ = 35/43 kPa

For TTL = 5/3 = 1.67 the flow increases to G_2/G_1 = 1.09.

b) In the second example, we decrease the TTL to 0.5 by changing the inlet water temperature to 9.5 °C:

Water,	10 kg/s	9.5 -> 7 °C,	ΔP_1	= Free
Water	4.54 kg/s	7.5 <- 2 °C,	ΔP_2	$\leq$ 45 kPa

Two (CB76-***30***H, ***2.8*** m2), M = ***10***%, $\Delta P_{1/2}$ = 164/42 kPa

Comment: Side 1 pressure drop is very high, i.e. there will probably a line optimum as in figure 7. The optimal flow rate for TTL = 2.5/5 = 0.5, is G_2/G_1 = 0.454.

Thus, we can make a table:

TTL	0	0.5	1.0	1.67	∞
G_2/G_1	0	0.454	0.77	1.09	∞
Optimal	Deduced				Deduced

See also figure 12, which shows the optimal flow as a function of the TTL with the margin and the pressure drop as parameters. The direction of the parameters could be deduced from the previous results.

The above results are in line with sound engineering. For a very close temperature approach and/or a large temperature change - a high TTL - case a), common sense says that a large temperature difference would be advantageous, thus a low exit temperature and a large flow.

For a small temperature change and/or a large approach - a low TTL, case b) - there is a comparatively small area for large flows and the number of channels is determined by the allowable pressure drop. The heat exchanger is probably thermally over dimensioned.

Thus, we reduce the flow rate and the number of plates is reduced. As the heat exchanger is thermally over dimensioned this does not matter.

2.15. Change of plate properties.

2.15.1. Heat transfer in a channel.

Up to now, we have seen that the selected heat exchanger might not fit the required temperature program. We have then tried to alter the duty, by changing the flow, by accepting a higher than required thermal overdimensioning or by a lower than permitted pressure drop.

The other solution is to change the heat exchanger. We then need to know the properties of the PHE.

We then study our example again, cooling of water 12 °C with water of 2 °C. In this case, we will see what happens in one channel pair of cold and warm water for an increasing flow rate. The temperature changes in the cold and warm channels are equal. Only the warm side is shown, the cold side being similar. We are not concerned with the heat exchanger, only the “bare” channels. Table 1 shows some data for CB76.

Theta is the thermal length, i.e. the temperature drop per mean temperature difference. Note that an absolute temperature change is meaningless to discuss; it always has to be related to the driving force, the mean temperature difference.

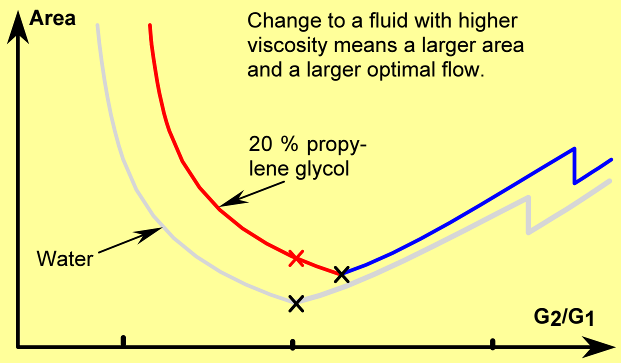


Fig. 10. Change of fluid.

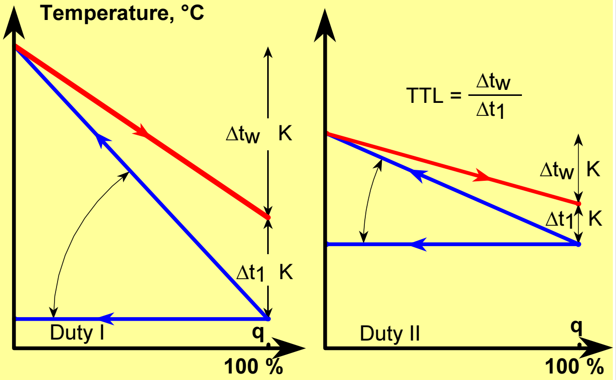


Fig. 11. Definition of the Terminal Thermal Length (TTL).

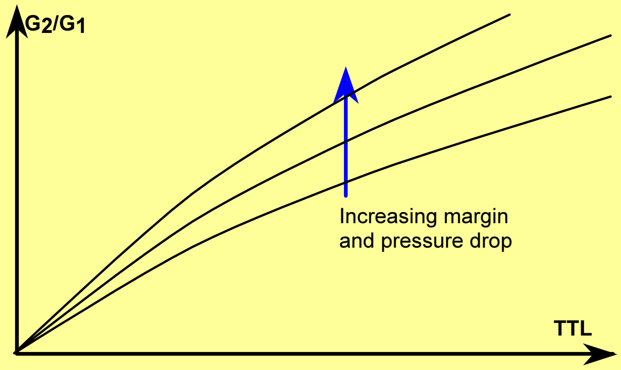


Fig. 12. The optimal flow rate as a function of the Terminal Thermal Length.

Water flow kg/(hr, ch)	Temperatures °C in	out	Δt	MTD K	Theta (Θ)	ΔP Kpa
2000	12 -	8.14 =	3.86	6.14	0.629	129.
1500	12 -	7.86 =	4.14	5.86	0.707	74.8
1000	12 -	7.48 =	4.52	5.48	0.826	34.7
500	12 -	6.86 =	5.14	4.86	1.056	9.75
250	12 -	6.30 =	5.70	4.30	1.326	3.03
203.5	12 -	6.14 =	5.86	4.14	1.414	2.14

Table 1. The theta & ΔP of CB76 M-channels.

a) We want to cool 16000 kg/hr water of 12 °C to 8.2 °C with an equal amount of 2 °C water.

2000 kg/(hr,ch) gives 8.14 °C, close enough. 16000/2000 = 8 channels (double) will do the duty. The pressure drop might be somewhat on the high side, 129 kPa, though.

What if only 10 kPa? 500 kg/(hr, ch) gives 9.75 KPa and an exit temperature of 6.86 °C, but we need 32 channels. We are firmly on the right side in the curves of figures 5 to 10 and pressure drop only determines the number of channels.

b) Suppose that we want to cool 10000 kg/hr water of 16 °C to 8.8 °C with equal amount of 2 °C water. Suppose that the K-value does not change with the change of the temperature from 12 to 16°C. ΔP permitted is 10 kPa.

The theta of this duty is (16 -8.8) / (8.8 -2) = 1.059. The table gives a theta of 1.056 for 500 kg/(hr, ch). That means an exit temperature of 8.81 °C and a ΔP = 9.75 kPa, which is OK. Thus we need 10000/500 = 20 channels.

c) We want to cool 20 350 kg/hr water from 12 to 6.1 °C with a maximum ΔP of 70 kPa.

The table shows that a flow of 203.5 kg/(hr, ch) gives a temperature of 6.14 °C. Thus 20350/203.5 = 100 channels are needed. The pressure drop is only 2.14 kPa and we are on the left side of the curve in figures 5 to 10.

d) There is another way. Suppose that we connect two channels in series and send 1500 kg/(hr, ch) through them. Each channel would have a theta value of 0.707 and we would get the following temperature program:

		Channel 1		Channel 2	
12	->	9.07	->	6.14	°C
7.86	<-	4.93	<-	2	°C

This is close to what we want but now we only need: 2 * 20350 / 1500 = 27.1 ~ 28 channels.

Unfortunately, the total pressure drop will be twice the channel pressure drop, i.e. 2 * 74.8 = 150 kPa. In order to get to the required pressure drop of 70 kPa, we can allow 35 kPa/channel, i.e. what 1000 kg/(hr,ch) give, thus:

2 * 20350/1000 = 40.7 ~ 41 channels, which is still better than the original 100 channels. As an extra bonus, the exit temperature will be about 5.8 °C instead of the required 6.1 °C. The operating point is now to the right.

2.15.2. Different channels.

The preceding paragraph was basically a repetition of the first part of this section. We will now see how we can change the channels to better fit the duty.

In case a) we needed 8 channels to obtain the required temperature 8.2 °C but 16 channels to get down to the required pressure drop of 35 kPa. The heat exchanger is then thermally overdimensioned with an exit temperature of 7.48 °C.

What we need is a plate channel, with a lower pressure drop and we can then accept a lower heat transfer. The simplest way to obtain such a plate would be to increase the channel depth. This will lower the pressure drop and the heat transfer.

Similarly, in case c), the pressure drop is very low. By decreasing the channel depth, the pressure drop would increase but also the heat transfer, expressed as the theta value, i.e. the temperature change for a given temperature difference would increase.

At a certain channel depth, the heat transfer would be the required for the correct pressure drop and with a reduced number of plates.

Another possibility is shown in d). The heat exchanger, by being connected in two passes in series, is simply doubled in length. The result is a doubling of the theta value, i.e. the thermal length is connected to the physical length and we can simply change the plate length to obtain the required properties

Thus, we can use the channel depth and the length to adjust the thermal and hydraulic properties of a PHE.

Increased length	=>	Higher theta	& higher ΔP
Decreased depth	=>	" "	& " "

2.15.3. Cross-corrugated plates.

The two basic methods, which we can use to adjust the thermal properties, depth and length, are theoretically very effective but have some practical drawbacks:

- Length. Plates of different lengths do not fit in the same frame of a gasketed plate heat exchanger. For brazed PHE's, different lengths means different manufacturing tools, which makes production expensive? Still, change of the length is a very cheap method to quickly design a new PHE with different thermal properties, which is why it is much used.
- Depth. In order to make a strong plate, the depth has to be as small as possible. As a high pressure is always required, the designer uses the lowest possible pressure depth. This means that the room for variation is very limited. The depth can be increased but to the detriment of pressure resistance.

Enter the cross corrugated or the herring bone plate. Herring bone because it resembles a well-eaten herring.

See figure 13 for the ***plate*** types and figure 14 for the resulting ***channel*** types. The two plate types H & L, form three channels: L (two L-plates), H (two H-plates) & M (one L- and one H-plate).

The L-channel has a low DP but also a low heat transfer - low theta - for a given channel flow, see Table 2.

The H-channel has a high ΔP but also a high heat transfer - high theta - for a given channel flow, see Table 3.

The M-channel has intermediate ΔP and theta for a given channel flow, see Table1.

2.15.4.Properties of the channel types.

In the previous examples, we used with more or less success the CB76 M-channels. We shall now see if the H- and L-channels can do better.

e) Try another channel for the a) duty, i.e. cooling of 16000 kg/hr water from 12 to 8.2 °C, ΔP max. 10 kPa.

In a) we found that 8 M-channels were enough for the thermal duty but 32 channels are necessary to keep the pressure drop. We thus try the L-channels, table 2.

By interpolation, we find that 611 kg/(hr,ch) is cooled to 8.2 °C for a pressure drop of 8.3 kPa and we need:

16000/611 = 26.2 = 26 channels

8.3 kPa is close enough to the permitted 10 kPa.

Note! H-channels would require some 64 channels and the number of channels is only determined by the ΔP.

f) Example b) would require about 57 L-channels (175.5 kg/(hr,ch)) or 35 H-channels (233 kg/(hr,ch)).

g) Example c) would require about 151 L-channels (134.5 kg/(hr,ch)) or 29 H-channels (705 kg/(hr,ch)) with a pressure drop of 58 kPa, close to the required 75.

Table 4 shows the result, which can be summarized as :

a) is a low theta duty. Thus, a low theta channel fits this duty. Higher theta channels - M & H - need more plates to maintain the required pressure drop.

b) is a medium theta duty and an M-channel is the best. Both L- & H-channels would require more plates to give the required thermal and hydraulic performance.

c) is a high theta duty and the H-channels are the best. The next best are M-channels in series.

2.15.5. The area vs. flow for H, M & L channels.

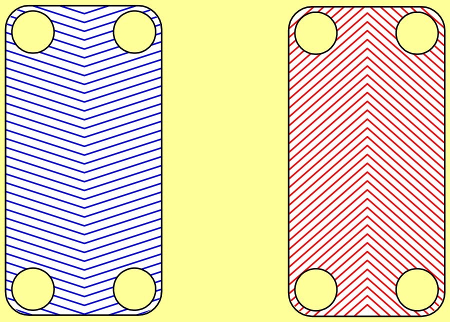
It is interesting to study the appearances of the three channels in an area-flow diagram as done previously, see figure 15. The characteristics are:

- Each channel has its double - margin & ΔP - curve.
- The three curves approach infinity for the same minimum flow.
- Each curve has its minimum for a given flow rate. The H-channel has the lowest, then the M-channel and the L-channel has the highest optimal flow rate.
- For very low flow rates, the H-channel gives the lowest area, then the M-channels and finally the L-channels.
- For very large flows, the order is reversed.
- The smallest area for a given flow rate is marked by the full coloured lines. The greyed out lines are valid solutions, but which give larger areas.
- Each of the three minima, L, M & H can be the lowest. This depends on the thermal duty, the TTL as defined previously. See also examples a -g.
- In certain cases - both sides one-phase media - the channels can be mixed in the same heat exchanger. The minimum area then follows the black line.

2.16. Summary of PHE designs.

We can now summarize the design and calculation of PHEs, whether brazed, semi-welded, welded or gasketed.

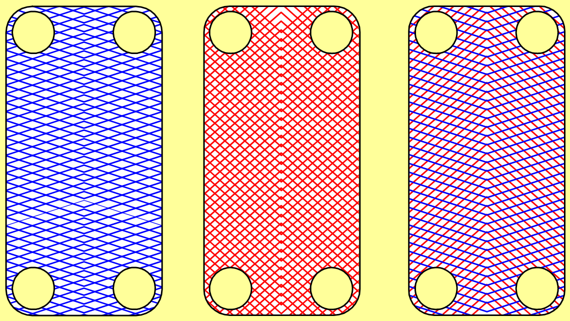
- Each thermal duty has a certain thermal length. This length is at times fixed; at times, it can be changed by varying flow rates, temperatures or pressure drop.
- Each channel type has a certain thermal length. The thermal length of a channel cannot normally be changed, but a PHE can have a series of ***plates*** from which ***channels*** of various thermal lengths are formed.
- In an optimal heat exchanger, the thermal length of the duty matches that of the channel.
- In the general case - non-matching lengths - the number of plates in a PHE is either determined by the pressure drop or by the required thermal duty.



H-Plate. Obtuse corrugation angle.

L-Plate. Acute corrugation angle.

Fig. 13. The *plate* types.



H-channel

L-channel

M-channel

Fig. 14. The *channel* combinations.

Flow kg/(hr, ch)	Temperatures °C in out Δt	MTD K	Theta (Θ)	ΔP Kpa
2000	12 - 9.08 = 2.92	7.08	0.413	62.3
1500	12 - 8.86 = 3.14	6.86	0.458	36.7
1000	12 - 8.56 = 3.44	6.56	0.524	17.9
500	12 - 8.05 = 3.95	6.05	0.653	5.84
250	12 - 7.46 = 4.54	5.46	0.833	1.91
203.5	12 - 7.15 = 4.85	5.15	0.942	1.37

Table 2. The theta & ΔP of CB76 L-channels.

Flow kg/(hr, ch)	Temperatures °C in out Δt	MTD K	Theta (Θ)	ΔP Kpa
2000	12 - 7.35 = 4.65	5.35	0.871	424.3
1500	12 - 7.00 = 5.00	5.00	1.000	245.4
1000	12 - 6.51 = 5.49	4.51	1.217	114.1
500	12 - 5.77 = 6.23	3.77	1.653	30.33
250	12 - 5.14 = 6.86	3.14	2.185	8.218
203.5	12 - 4.97 = 7.03	2.97	2.367	5.584

Table 3. The theta & ΔP of CB76 H-channels.

Duty	Θ	L	M	H
a) & e) 16000 kg/hr 12 -> 8.2 °C 5.8 <- 2	**0.613**	**26**	**32**	**64**
b) & f) 10000 kg/hr 16 -> 8.8 °C 9.2 <- 2	**1.059**	**57**	**20**	**35**
c) & g) 20350 kg/hr 12 -> 6.1 °C 7.9 <- 2	**1.439**	**151**	**100(41)**	**29**

######	**Channel # minimum.**
######	**Channel # determined by ΔP.**
######	**Channel # determined by heat transfer.**

Table 4. Summary of the results for a - g.

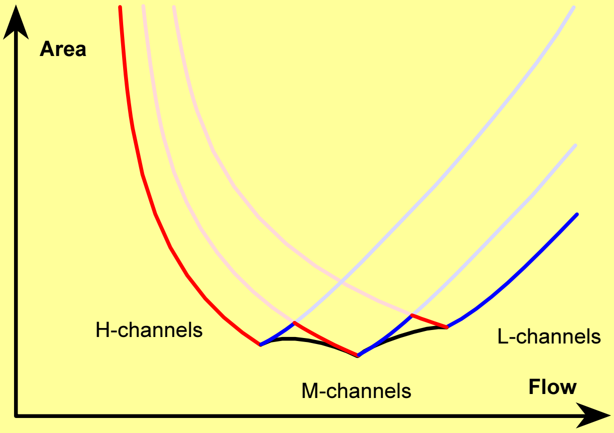


Fig. 15. The general aspect of the area-flow curve for of three, different theta, channels.

- ♦ If the permitted pressure drop solely determines the number of plates, the heat exchanger is normally thermally overdimensioned.

- ♦ If the number of plates is determined by the thermal duty, the permitted pressure drop is normally not used. To match the thermal lengths, either the thermal duty or the channel type can be changed.

- ♦ Change of the thermal duty cannot normally be done without affecting the entire system.

- ♦ Change of the PHE type does not normally affect the system. Some possibilities are shown in figure 16.

- ♦ In the case of a PHE with multiple channels, especially if these can be mixed, the optimal point (Fig. 10) is replaced by an area (figure 15). This permits a PHE to utilize the pressure drop for a range of thermal lengths.

It is sometimes necessary to use a PHE outside its normal range of thermal lengths. The reasons could be material demands, pressure resistance; no better type is available etc. It might thus appear to be very large, thus "bad". It might not be the case; it is simply not optimal for the given duty but could be it for another.

Example. A PHE optimized for high heat recovery, i.e. a high thermal length, will be very large if used to cool a large amount of water only a few degrees.

2.17. Fouling factors and margins.

The fouling process is described in **7. Fouling & Corrosion**. Sufficient here to say that either the system is properly designed and operated and the PHEs then operate without problems or fouling occurs and they might rapidly clog. Even in a well-managed system, there is some fouling and the heat exchangers have to be designed with a certain initial excessive capacity. A PHE can be designed with this initial capacity in three ways:

1) ***A fouling factor*** (R_f) is added to the K-value. The term factor is a misnomer as this term indicates a multiplication and R_f is an additive term.

The idea behind the fouling factor is that fouling occurs as growth of the fouling component on the surface. After some time - the cleaning interval - the growth has reached a certain thickness. With knowledge of the thermal conductivity, it easy to calculate the thermal resistance of the layer and that together with the calculated clean K-value (U-value) gives the design K-value.

Various attempts (ARI, TEMA, etc.) have been made to recommend fouling factors for different type of fluids. Unfortunately, the fluid of most interest to the refrigeration engineer, water, is a highly varied fluid and the same type of water can vary from site to site, from month to month, see **7. Fouling & Corrosion**..

Not only the type of water, but other factors, mainly type of heat exchanger, shear stresses and surface temperatures are also important for the fouling tendencies. Moreover, the fouling in water, except scaling, is often not the steady growth of a layer on the surface.

2) Surface margin, which is numerically equal to the margin on the K-value is another approach. The actually installed surface is a certain percentage larger than the calculated necessary clean surface.

The reasoning is that the PHE should basically not operate under fouling conditions. The required oversurfacing contains not only the inevitable small fouling due to debris, slime, oil, wear & tear, etc. but also various uncertainties such as lack of exact physical properties of the media and uncertain design or operating condition.

The advantage of this is that various components can be compared, type, “If the condenser is 30 % overdimensioned, then zero margin on the evaporator is OK”.

3) Extra capacity. From the designers point of view, this is the worst way of expressing what is basically the same as 2), because it is ambiguous. How is the extra capacity obtained? By increasing the flow rates? If so what shall the Δp be? Correspondingly higher or lower because of a lower pump head. Alternatively, shall the temperature program be changed? The new temperatures could result in a temperature pinch (figure 17) and the requested heat exchanger might be infinitely large.

Without a careful specification of the condition, which the extra capacity shall be designed for, this type of fouling specification should not be used.

Are methods 1) and 2) comparable? The fouling factor is 0.0001 (m^2,°C/W) and the K-value is 5000 (W/m^2,°C.) in one case and 1000 in the other, typical water/water values for PHE and S&THE in heat recovery applications. In the first case, the overall heat transfer coefficient is:

K_{Design} = 1/ (0.0001 + 1/5000) = 3333 => 50 % margin

and in the other:

K_{Design} = 1/ (0.0001 + 1/1000) = 909 => 10 % margin

Conclusion: The higher the K-value is, the higher the margin becomes for a given R_f. The problem is, that published R_fs are almost exclusively developed for S&THE. If applied on PHE, which usually have much higher K-values, the margins become large, exceedingly large sometimes. Some points to consider:

a) ***Control problems.*** A heat exchanger does not operate with a margin. It adjusts the temperatures until the margin (and the fouling factor) become zero. The resulting large margins, if S&THE R_fs are applied to PHE’s, could result in freezing or overheating a fluid if it is throttled to decrease the MTD, and possibly instable operation.

b) ***Decrease of the shear stresses.*** One possibility to increase the margin in a PHE is to add more plates in parallel. The pressure drop and thus the shear forces then decrease very rapidly. The result could be the contrary to the desired, increased fouling.

c) ***Plates in series.*** If another PHE is added in series to the first, the Δp doubles. To keep the Δp, plates are added in parallel to both PHEs. The resulting area margin will probably be much larger than the required at the same time as the shear stresses have decreased and the effect will be a combination of a) and b).

d) ***Temperature difference.*** Suppose that PHEs are designed for each duty below, both with a zero margin.

42 -> 37	49 -> 44
40 <- 35	25 <- 20
MTD = 2 K	MTD = 24 K

The nominal K-value decreases with 20 % due to fouling. How much must the warm side temperature change to keep the nominal duty?

If the K-values decreases with 20 %, the MTD has to increase with 1/0.8 => 25 % and the new temperatures are:

42.5 -> 37.5	55 -> 50
40 <- 35	25 <- 20
MTD = 2.5 K	MTD = 30 K

A temperature increase of 0.5 K might be acceptable but 5 K is certainly not.

Conclusion. It is important to consider the effect of the margin on the temperature program.

e) ***Plant type.*** If an A/C plant cannot keep the design air temperature during an extraordinary heat wave, the result might be an inconvenience. If the same happens in a hospital or a pharmaceutical industry, the result could be a catastrophe. Obviously, the layout of the two plants, including the margins for the PHE, must be different.

f) ***Cleaning*** should be done just before the warmest season starts. The PHEs then meet the most severe conditions with the highest K-value.

g) ***Plant type.*** The margin could be less in e.g. an office, which could be shut down during off hours as compared with a continuous operating plant such as in a hospital.

Summary: PHEs should not be overdimensioned. Depending on the type of duty and other conditions, the margin should not be more than 0 - 15 %, rarely 15 to 25.

A. We want to cool water, the blue line, from 12 °C to 6 °C. The selected heat exchanger channel is too short, the water it delivers at the exit is only 8 °C. What can we do?

B. One possibility is to add more parallel channels. This adds more surface and the channels cool better. The ΔP and the K-value decrease though, making the added channels increasingly ineffective. The permissible ΔP might not be fully utilized.

C. Another possibility is to rearrange the channels from all in parallel to two groups in series, each group containing. half the number of parallel channels. This doubles the channel velocity and thus increases the K-value. The DP increases about ***seven times*** though, which might necessitate more parallel channels. The HE will then be thermally overdimensioned.

D. Still another possibility is to use a ***physically*** longer channel. This cools better but the ΔP increases. Channels might have to be added in order to keep a ΔP limit.

E. A similar effect can be obtained with a ***thermally*** longer channel. In this case, the corrugations are tighter which creates an increased turbulence, thus a higher K-value but also a higher ΔP.

F. Another method to obtain a ***thermally*** longer channel is to decrease the channel depth. In this case, the depth is less which creates increased turbulence thus higher K-value and but also higher ΔP.

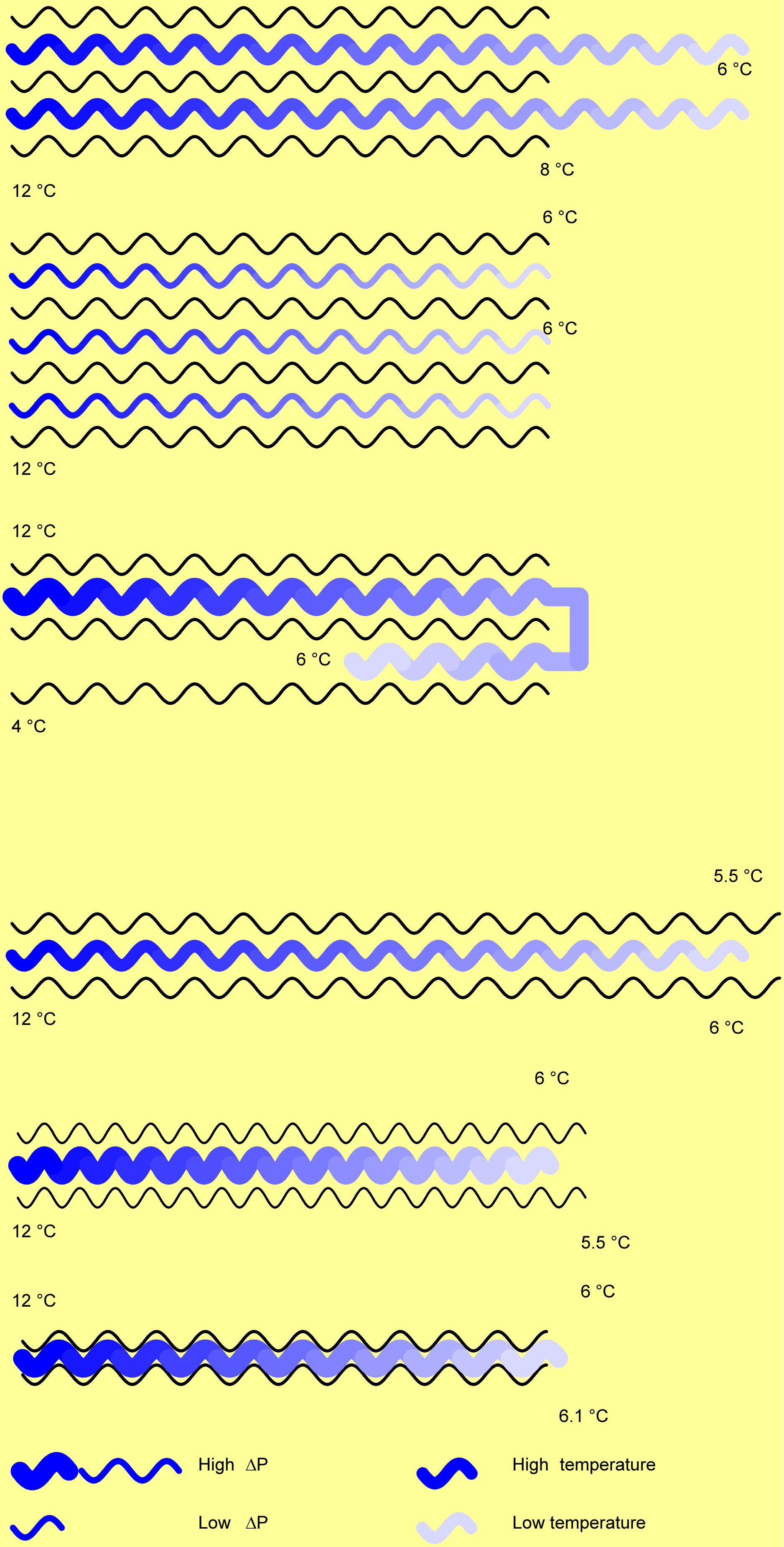


Fig. 16. Summary of methods on how to adapt the PHE to the duty.

2.18. Evaporators and condensers.

In the previous chapter, we have discussed the behaviour of PHEs for one-phase flow, but what happens if we try to extend the reasoning to cases where one side is a two-phase refrigerant flow. Below we will study some special aspects when optimizing evaporators and condensers.

Note that the studies here are theoretical and based on how the flows and the temperature programme affect the size of the heat exchangers. There are a number of other considerations, which affect the design of condensers and evaporators. These considerations are discussed in the appropriate chapters on condensers and evaporators.

2.18.1. Evaporators.

The refrigerant flow cannot be changed, only the warm side flow. It would be possible to change the water flow by keeping the inlet temperature constant and change the exit temperature. That poses some problems, though, as the exit temperature is an important value in a refrigeration system, which cannot readily be changed. Change of the entrance temperature while maintaining the exit temperature is meaningless. The duty cannot be changed in an air-conditioning system from 1 l/s water from 12 to 7 °C to 0.1 l/s water 57 to 7 °C.

Thus, we have to take another approach. In most systems, the water leaving the evaporator cools air in some type of unit cooler. In a unit cooler, the flow is essentially in cross flow and the temperature difference between the air and the water is fairly large. In such systems, the most important value is the mean temperature of the water.

The unit cooler will more or less perform in the same way, regardless of whether the water temperatures change 7 -> 12 °C, 8 -> 11 °C or 6 -> 13 °C .

Fig. 17 shows the temperature program. In the normal air conditioning temperature program, 12 to 7 °C, the mean temperature is 9.5 °C. The minimum flow occurs for a temperature program of 17 to 2 °C, still with a mean temperature of 9.5 °C, in an infinitely large evaporator. The maximum (infinite) flow, also gives an infinite large evaporator. The inlet and the exit temperatures are then equal to the mean temperature 9.5 °C. The area-flow curve, when water changes from the minimum flow to infinity, looks very much the same as for a condenser or a liquid-liquid heat exchanger, see Fig. 16.

Note! If the water is used to cool another liquid in a counter current HE, the exit temperature from the evaporator is the most important parameter. See later.

2.18.2. Condensers.

Condensers behave very much like one-phase PHEs, with some crucial differences, see Fig. 18.

The figure shows the temperature programme in a condenser. Two cooling water temperatures are entered, one for an infinite flow and the other with a temperature pinch.

- It is only meaningful to study variation of the cooling fluid. The refrigerant flow cannot normally be changed.
- In a one-phase heat exchanger, there is a minimum flow, obtained when the exit temperature of the cold fluid approaches the inlet temperature of the warm.

  In the same way, there is a minimum water flow in a condenser. However, this minimum flow is not determined by the end temperature, but by the temperature pinch, which occurs when the vapour has been desuperheated and condensation starts.

Unfortunately, the temperature pinch is more difficult to calculate than the end temperature difference. It might happen that a condenser is laid out with a pinch, i.e. the cold side is equal to or higher than the warm side, an impossible programme. Apart from the difficulty of determining the minimum flow, the area-flow looks about the same as for a liquid-liquid HE, e.g. as shown in Fig. 16.

3. SECOOL optimization.

3.1. What is SECOOL optimization.

A typical SECOOL system as described in § 15 and figures 17 & 18 of **1. Applications** and the simplified system shown in figure 19 in this chapter consists of pumps, pipes, valves, filters, fittings, etc. and the PHEs.

We will here discuss how to design an optimal system, because the technique of this demonstrates the properties of PHEs and can be used not only for typical SECOOL systems but also for all system where a circulating liquid is cooled or heated in a PHE.

The more spent on the system (PHE size, larger pipe & valve diameters, etc.) the lower and cheaper the pumping energy will be. A total optimization usually shows that the lowest total annual cost (investment and operating) occurs for high investment and low operating costs.

The usual approach when making an investment is normally to keep the initial costs down, i.e. high running costs and low investment costs. SECOOL optimization circumvents the question of capital vs. operating costs and instead tries ***to minimize the investment cost, notably of the PHEs, for a given operating cost,*** whatever this operating cost might be.

Consider the system in figure 19. Suppose that everything is decided - pipe length & diameter, filters, valves etc. - except the sea water flow and thus the PHEs size. The larger the flow, the larger the MTD and thus probably the smaller the PHE will be.

The primary side flow, whether it is a fresh water flow as in figure 19 here and figure 17 or a condensing vapour figure 18 both in **1. Applications,** is usually considered to be fixed but sometimes a pressure drop limit is imposed.

One more component has to be decided, the sea water pumps. We will consider two cases:

- The pumps already exist and we have the pump characteristic curve. Thus, we have to select a point on the pump curve.
- Only the pumping cost is decided. This depends on the flow times the total head, but these are not yet specified and we are free chose any combination of flow and total head as long as the pumping energy is constant.

Moreover, we assume that the sea water is free of charge and that the only cost is the pumping cost. All other costs might be considered constant are thus not applicable.

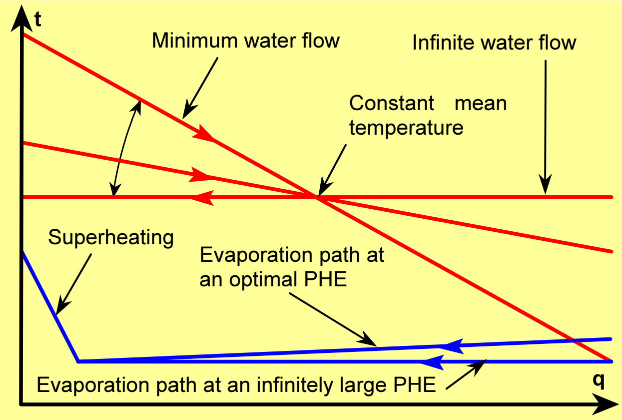


Fig. 17. Changing water flow around a constant mean temperature in an evaporator.

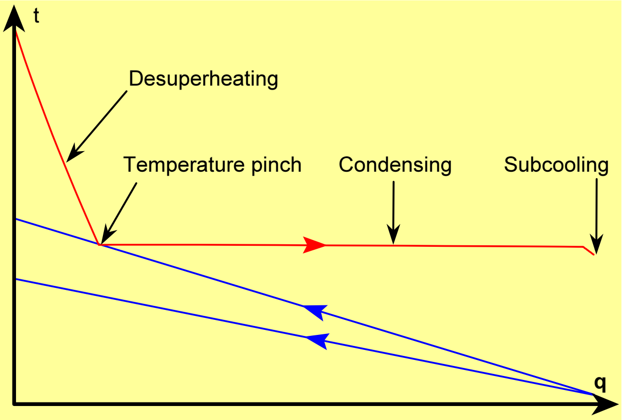


Fig. 18. Changing water flow in a condenser.

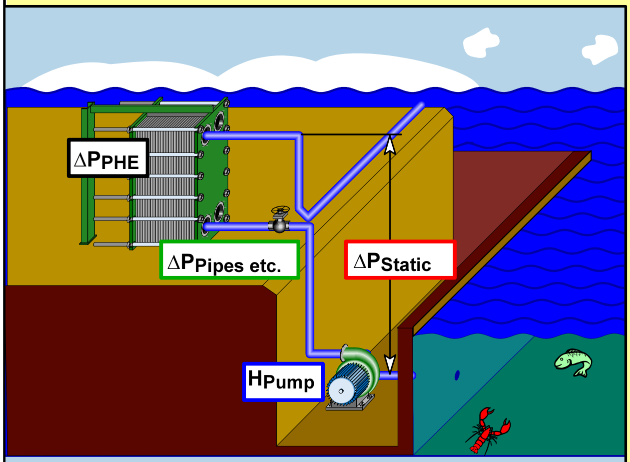


Fig. 19. The main elements in a SE-COOL system.

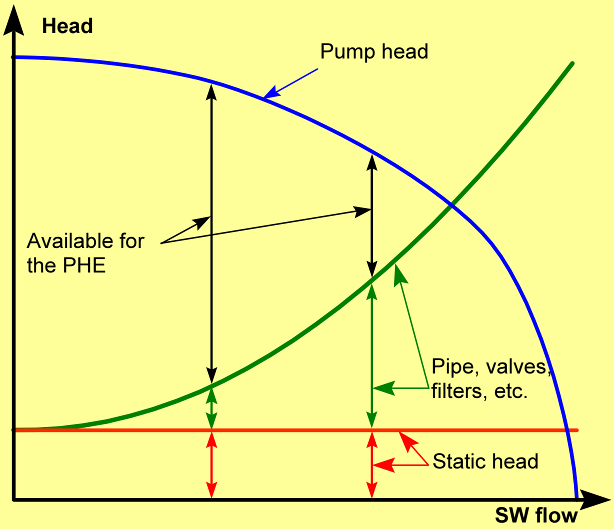


Fig. 20. Pressure drop distribution.

3.2. Existing SW pumps.

The only equipment not yet decided is the PHE and the only operating parameters not yet decided are the SW flow and the PHE pressure drop. In order to obtain the best PHE we make a number of calculations for different SW flows. For each SW flow the pump gives a total lifting head, which can be found from figure 20. Here the various pressure drops in the system are also shown. Note in consistent units, either as height or pressure difference.

- ***The pump head.*** This is the driving force. The total head usually decreases with increasing flow rate. Somewhere in the middle of the curve is the point of maximum efficiency for the pump, i.e. lowest electricity consumption. Changing the impeller diameter and/or the rotational speed changes the characteristic curve.
- ***The static head***. This is the total lift of the water from the pump inlet to the summit (Δp in Fig. 19). This does not change with the flow rate.
- ***Pipes, valves, filters, etc.*** The pressure drop for all parts of the system except the PHE. It normally varies as a little less than the square of the flow rate.
- ***The PHE.*** The PHE gets what remains when the static head and the pipe work pressure drop have been subtracted from the total pump head. Values for two SW flows are shown in the figure. They vary from a high value at zero flow to zero, when all the pump head is consumed by the pipe work and the static lift.

A calculation of the PHE area vs. the sea water flow with the corresponding available pressure drops gives an area/flow curve, which is similar to the one in figure 6.

The curve is superimposed on the hydraulic curves and shown in figure 21. The area approaches infinite values for a low and a high SW flow rate. The temperature programme as described in § 2.6 in this chapter sets the minimum. The higher flow rate occurs when no pressure drop remains for the PHE. It differs thus from figure 6, where the area approaches infinity for infinite flow.

Example 1:

Capacity 1000 kW,
10 % margin on the K-values
R22 superheated to 73 °C, condensing at 40 °C
Water 70 kg/s 32 to 35.4 °C.

From the pump curve (typical):

Pump head at	0 kg/s:	320 kPa
" "	40 kg/s:	300 kPa
" "	200 kg/s:	0 kPa

At nominal flow, 70 kg/s

Head (Calculated)	270 k pa
Static lift:	100 kPa
Pipes	135 kPa
Left for the PHE:	35 kPa

A design for this duty gives:
A15BWFD 182 plates, 135 m2
DP_1/DP_2 = 0.73/34.6 kPa
Margin: 10 + 20 %

What are the minimum and maximum flows?

The minimum flow occurs when the water curve just touches the vapour curve just where superheating is finished and condensation starts, see figure 17. From thermodynamic data, we note that the latent heat in this case is about 85 % of the total load. The same is then valid for the water side. At the temperature pinch at 40 °C, 82 % of the water heat is transferred. Thus

Total temperature change: (40 - 32) / 0.82 = 9.76 K

Minimum flow rate: 70 * (3.4 / 9.76) = 24.4 kg/s

When the flow rate increases, so does the pressure drop in the pipe work, roughly to the power of 1.8. The pump head decreases at the same time.

At a flow rate of 77.3 kg/s we get:

Pump head (interpolated)	261.5 kPa.
Pipe work: 135 * $(77.3/70)^{1.8}$ =	161.5 kPa
Static head:	100.0 kPa

Total static & pipe pressure drop 161.4 kPa = Pump head

For these two flow rates the surface approaches infinity but for different reasons, at the lower because there is no temperature difference left, at the higher because there is no pressure drop left, see figure 21. A number of trials is needed to obtain the minimum area, see table 5.

Conclusion: By changing the operating point we have managed to reduce the size of the PHE. The water flow has decreased and the allowable pressure drop increased, but the important result is that this is done using the same pump, ***therefore requiring no change in the pumping cost***.

Note that this is not the only possibility for optimization. At the nominal flow, the condenser is thermally overdimensioned. That might be used to decrease the condensing temperature and thus the compression cost.

However, a change in the compressor involves the whole system and might thus be more difficult to carry out.

3.3. Fixed pumping cost.

In the previous example, the pump operating point was changed from 70 kg/s & 270 kPa to 43. kg/s & 297.6 kPa.

The pumping power is proportional to flow times head and in the first case this is 18900 and in the second only 12797. That indicates that the optimal point falls on a point with a very low efficiency.

Suppose that the pump has not yet been specified, but only the pumping power. In the previous case, this was proportional to 18900 and the pump was presumably selected because it had a high efficiency at this point. Thus, we could change the flow and the head simultaneously, calculate the condenser as before and when the new combination of flow and head is determined try to get a pump which is as effective as possible at this point.

See figure 22, which is similar to figure 20, but doesn't show a pump characteristic curve. It shows a curve for constant pumping power, a hyperbola.

Example 2.

We calculate the various pressure drops as before. The result is shown in table 6.

The operating point has thus moved from 70 kg/s & 270 kPa to 48 kg/s and 393.7 kPa. Considering all the pumps available, it should be possible to find one pump with efficiency as high at 48/393.7 as at 70/270.

3.4. Summary.

Note 1! The pressure drop on the condensing side increases when the number of plates is reduced from 182 to 102 (or 116 as in the previous example).

In this case, it does not enter into the optimization as the exit pressure from the condenser has a very little effect on the function of the expansion valve. This will hardly notice that the inlet pressure has decreased from 15.4 to 15.2 bar. The decrease of the condensation temperature and the effect on the mean temperature difference is taken into consideration by the PHE design program.

Note 2! In case of a liquid on side one, the pressure drop of should be taken into consideration when optimizing. This is best done by limiting the pressure drop, if necessary some optimizations with different levels.

Note 3! The result above is astonishing. Few engineers would object to a design specification for heat exchangers, which says 70 kg/s and 35 kPa pressure drop. Most would object to the high pressure drop in a design specification 48 kg/s and 225 kPa even though the total pumping power is the same in the two cases.

Note 4! If the same optimization were made for a S&THE, the result would be radically different, especially for liquid/liquid duties where close temperature approaches are important. In a S&THE of the most common type - 2/1 - it is difficult to heat the cold fluid to above the exit temperature of the warm fluid; the efficiency drops rapidly, with increasing temperatures.

On the other hand a S&THE can be designed to accommodate large volumes at comparatively low pressure drop.

Temperature programs like

1.	FW	50 - 40 °C	2.	FW	50 - 35 °C
	SW	40 - 30 °C		SW	45 - 30 °C

would be no problem for a PHE but a S&THE will get into difficulties, especially the second which would be close to impossible to obtain it in one S&THE; it has to be many connected in series. If we try to optimize the duties above according to the principles, we find that:

- A S&THE would benefit from as large a temperature difference as possible, especially important is to get the exit temperature of the cold media below the exit temperature of the warm, hence large flows but low pressure drops.
- A PHE has normally no problem with large heat recovery, but has problem keeping the pressure drop down for large volume flows, thus small flows and large pressure drops are optimal.

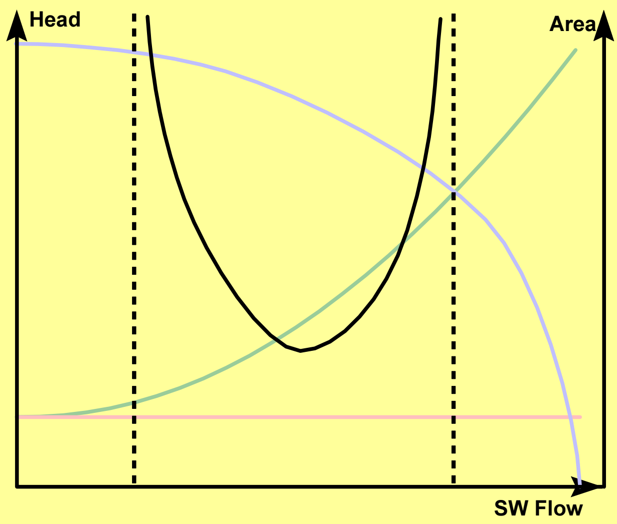


Fig. 21. The area-flow curve.

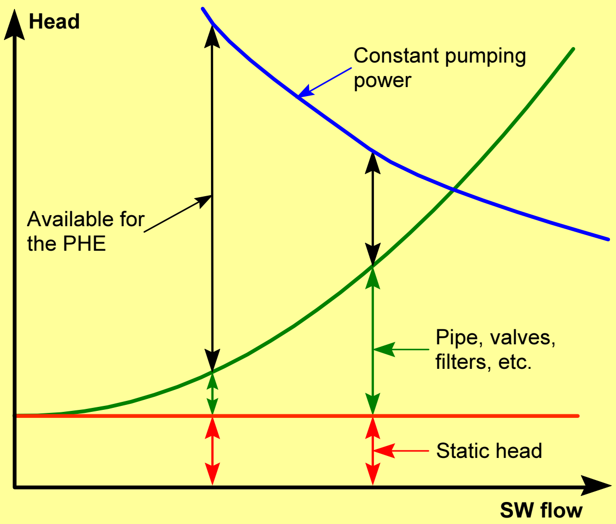


Fig. 22. Pressure drop distribution.

Flow	Static head	Pipe work	Pump head	Left for the PHE	A15BW with # plates
24.4	100	21.6	310.4	190.1	∞
30	100	29.4	307.1	177.7	1128
40	100	49.3	300	150.7	132
50	100	73.7	291.6	117.9	132
60	100	102.3	281.8	79.5	136
70	100	135.0	270	35	182 Nominal
75	100	152.9	264.5	11.6	328
77.3	100	161.5	261.5	0	∞

The minimum seems to be somewhere between 40 and 50 kg/s. A closer calculation gives the minimum.

43 100 56.2 297.6 141.4 116 Optimal

Table 5. The number of plates vs. flow.

Flow	Static head	Pipe work	Pump head	Left for the PHE	A15BW with# plates
24.4	100	20.3	774.6	654.3	∞
30	100	29.4	630	500.6	1128
40	100	49.3	472.5	323.2	132
50	100	73.7	378	204.3	132
60	100	102.3	281.8	79.5	136
70	100	135.0	270	35	182 Nominal
75	100	152	252	0	∞

The minimum seems to be somewhere between 40 and 50 kg/s. A closer calculation gives the minimum.

48 100 68.4 393.7 225.3 102 Optimal

Table 6. The number of plate vs. flow.

4. Optimization of PHEs in a system.

4.1. An intermediate circuit in a system.

See chapter **1. Applications,** §15 and the figures 17 & 18. These two systems differ in the use of sea water. In figure 17, the condenser is cooled by circulating fresh water, which is cooled by sea water in a SECOOL heat exchanger, normally gasketed titanium PHE.

In figure 18, sea water cools the condenser directly but on the other hand, a circulating chilled fresh water circuit is necessary to cool the chilled water for the fan coils.

Thus in both circuits an intermediate circulating fresh water circuit is installed, in the first system at the condenser, in the second at the evaporator. The big difference between the two systems is that the first has localized A/C plants, the second a central A/C plant.

In order to compare the heat exchangers in the two systems more easily, we make a modification and assume that both have central systems. Note that the choice between the two systems is a major one and the difference between the heat exchangers is only a small part of all the factors influencing the decision. Data in § 3.2 - existing sea water pumps- have been used in the examples.

4.1.1. Indirect condenser cooling.

Ind. A. SECOOL PHE.
1005 kW, MTD = 5.2 K.
FW 42 -> 37 °C, 76 kPa M10B, 32.6 m^2
SW 36 5 <- 32 °C, 107 kPa Titanium

Ind. B. Condenser.
1005 kW, MTD = 5.3 K.
R22 78 -> 45 °C A15BW, 99 m^2
SW 42 <- 37 °C, 52 kPa Stainless steel

Ind. C. Evaporator.
809 kW, MTD = 4.5 K.
Water 12 -> 7 °C, 55 kPa A15BW 78 m^2
R22 7 <- 2 °C Stainless steel

4.1.2. Direct condenser cooling.

Dir. A. Condenser.
1000 kW, MTD = 4.7 K.
R22 73 -> 40 °C A15BW 85.5 m^2
SW 37.6 <- 32 °C, 138 kPa Titanium

Dir. B. Evaporator.
809 kW, MTD = 4.4 K.
Glycol 7 -> 2 °C, 45 kPa A15BW 87 m^2
R22 2 <- -3 °C Stainless steel

Dir. C. Water chiller.
809 kW, MTD = 5.0 K.
FW 12 -> 7 °C, 47 kPa M10B, 39.6 m^2
Glycol 7 <- 2 °C, 45 kPa Stainless steel

4.1.3. Comparison of the systems.

The total surfaces in the two cases are practically equal:

Ind. Ti: 32.6 m^2	SS: 177 m^2	Total: 209.6 m^2
Dir. Ti: 85.5 m^2	SS: 126.6 m^2	Total: 212.1 m^2

The first has less, expensive Ti surface than the second. This and the fact that the small water/water M10B is considerably easier to open than the large A15BW condenser, strongly favour an indirect condenser system.

4.1.4. Optimization of the systems.

Only two positions are optimized. Ind. A and Dir. A are SECOOL optimized. What more can be optimized?

1. ***Condensing/evaporation temperatures.*** The only fixed temperatures are the sea water at 32 °C and the chilled water for the fan coils, which has to be cooled from 12 to 7 °C. Thus, the condenser/ evaporator temperatures of 40/-3 °C in the second case could be increased with e.g. 3 K to 43/0 °C. The only requirement should be that the compressor load is the same.

2. **Condenser loop.** The fresh water circuit 42 to 37 °C in Ind. A & B could be changed, both flow rates and temperatures with constant pumping power.

3. ***Evaporator loop***. The glycol circuit 2 to 7 °C in Dir. B & C could be changed, both flow rates and temperatures, again with constant pumping power.

4. ***Chilled water for the fan coils.*** These circuits could be optimized as well, but as the fan coils are standardized equipment, it s better to keep the nominal temperature program 12 to 7 °C.

These optimizations have to be made simultaneously, a tedious task, which is only justified for the very largest plants. The design engineer has to look at the duties and be content to optimize some very obvious unoptimized positions. There are some points to consider:

A. Margins. If one position has a very large margin, say the 1B condenser, the FW temperature level could be increased and thus a decrease of position 1A.

B. Pressure drops. If an intermediate water circuit has not utilized the available pressure drop at one position, this could possibly be utilized at the other position, to reduce the pipe work or to increase the total flow rate thereby increasing the temperature difference for one and/or both positions.

C. Circulation. The best utilization of the available temperature difference between two positions occurs for as small a temperature difference as possible of the circulating water. That means a large circulating flow, which the available pressure drop could prevent.

D. Rule of thumb. The MTD of each heat exchanger in a train of connected heat exchangers should be proportional to the incremental costs. For a PHE only the plate area, not the frame is counted.

4.1.5. Reoptimization of § 4.1.2, direct condenser cooling.

The rule of thumb gives some interesting points to consider. In Ind. A - C, the area divided with the MTD is 6.27, 18.68 and 17.33.

Ind. A is very small but as this is a Ti heat exchanger, the cost is much higher than SS. If we assume that a titanium square meter is twice as expensive as an SS square meter, the relative cost of Ind. A changes to 12.54 .

If the temperature level of the intermediate water circuit is lowered a little, Ind. A increases and Ind. B decreases and all values become comparable. However overall, this layout is acceptable.

Dir. A - C give values of 36.38 (18.19 * 2), 19.78 and 7.92. There is thus a very large imbalance between all duties. What could be done?

If the condensing and evaporating temperatures and the temperature level of the Dir. B-C water circuit increase by the same amount, Dir. A would decrease, Dir. B stay the same and Dir. C increase. The optimization is then made in the steps below. All the rules A - D in § 4.1.4 are used:

- The temperature programme is increased 3 K upward. The first effect is that the evaporation temperature increases to 0 °C and we can switch over to water instead of glycol, a very important change.
- Dir. A is once more recalculated and SECOOL optimized. We use the existing pump model § 3.2.
- The water circuit Dir. B - C is optimized as well. We have here assumed that flow times (pressure drop in Dir. B + C) shall be constant, i.e. the pipe work pressure drop and static lift is zero. From Example 1 in § 3 we get a value of 38.22 * (45 + 45) = 3440. The positions Dir. B & C are optimized when

  Flow * ($DP_{\text{Dir. B}}$ + $DP_{\text{Dir. C}}$) = 3440.
  Margins are the specified (5% for 2B, 0% for Dir. C).
  $(\text{Area/MTD})_{\text{Dir. B}} = (\text{Area/MTD})_{\text{Dir. C}}$

A number of trials are necessary.

Result:

1000 kW, MTD = 7.87 K, (Area/MTD) * 2.0 = 12.46

R22	76	->	43 °C	A15BW 49.5 m^2
SW	37.8	<-	32 °C, 144 kPa	Titanium

Dir. B. Evaporator.

809 kW, MTD = 4.06 K, Area/MTD = 24.4

Glycol	9.2	->	4.3 °C, 38 kPa	A15BW 99 m^2
R22	5	<-	0 °C	Stainless steel

Dir. C. Water chiller.

809 kW, MTD = 2.75 K, Area/MTD = 24.9

FW	12	->	7 °C, 44 kPa	M10B, 68.64 m^2
Glycol	9.2	<	4.3 °C, 48 kPa	Stainless steel

The area/MTD are now close to each other for the two last designs. The first is now too small, indicating that a temperature increase of 3 K was too much. We get the following relative costs before and after the optimization.

Before:	2.0 * 85.5 + 87 + 39.6 =	297.6
After:	2.0 * 49.5 + 99 + 68.64 =	266.64

We could decrease the total cost further by increasing the temperature level a little less than 3 K. On the other hand, as it is such a large advantage to use pure water instead of glycol we accept the cost. In any case, the total cost is lower than before the change.

4.2. Heat exchanger duties in a supermarket.

Figure 23 shows a typical application, which lends it self to optimization. The plant illustrated is a small supermarket (for a large super market see figure 10 in the application chapter) but the application is not important, it could have been an ice rink, a slaughter house, a chemical industry, an office and warehouse, etc.

The following duties can be optimized:

- ***The evaporation and condensation temperatures.*** If the evaporation temperature is increased with 1 K, the condensation temperature should be changed, probably increased as well with a certain amount. To make the comparison useful, we have to assume that the compressor load remains constant.

  The amount of change in the condensation temperature depends on the compressor; it could move to a more efficient operation, making the change large or vice versa.

  The evaporator will get a smaller MTD and will probably be larger and the condenser a larger MTD and will probably get smaller. The sum of the cost of the two items determines if the change is justified or not.
- ***The condenser-liquid cooler circuit***, figure 23 A. This is very similar to the SECOOL system optimized in the previous chapter. The condenser is not cooled directly by the ultimate heat sink, which in the SECOOL case is raw water, and in this air but by a circulating fresh water circuit. The temperature in the water loop as well as the flow rate could be freely adjusted between the air temperature and the evaporation temperature. At some point, the equipment cost will be minimized.
- ***The evaporator- unit cooler circuits***, figure 23 B1-3. Again, this is similar to the previous case, the fresh water circuit between the evaporator and the fan coils. Actually there are four circuits here: an evaporator-tank, and three Tank-UC loops. Note that the exit temperature from the three UCs can be different. As before, the temperatures and flow rate of the brine can be varied freely to get the lowest equipment cost.

4.2.1. Optimization of the liquid cooler - condenser circuit.

The techniques used above are all valid and the best way to apply them is though an example:

Example 3:

A 215 kW R22 condenser has to be cooled by air of 25 °C via a closed loop liquid cooler. The circulated water contains 30 % ethylene glycol. The condensing temperature is 45 °C and the entrance temperature is 78 °C. The circulation pump gives a head of 19.5 m at a flow of 40 m3/h (E.glycol has a density of 1035 kg/m3).

1. The following steps are followed:

1) Recalculate the pump data to kg/h and kPa to get consistent data, i.e. 41 400 kg/h and 198 kPa.
2) This head is distributed as:

LC:	60 kPa
Condenser:	60 kPa
Pipe work:	60 kPa
Spare:	18 kPa

3) Select a liquid cooler. Alfa Laval has a model LCS 219, which gives 219 kW at standard conditions (15 K difference between the air and the liquid inlet temperatures and a temperature drop of the liquid of 5 K). In our case, that means 41140 kg/h 40 to 35 °C (215 kW). This gives a pressure drop of about 42 kPa. This fits fine, thus we use that.
4) The actual pipe work (and valves, etc.) ΔP = 55 kPa.
5) A calculation of a BPHE gives a:

CB300-50M, 14. 44 m2, Margin = 0 %.

R22,	78 ->	45 °C	
Glycol	40 <-	35 °C,	$\Delta P2$ = 31 kPa

2. Comments: The design is acceptable. There is no margin - not necessary because of clean water - on the BPHE, some on the LC. The only way this design is not optimized is that the available pump head is not utilized. We have three ways of optimizing this circuit.

1) The pump head could be lowered, e.g. by changing the impeller, rotational speed, or another pump.
2) The pipe work could be changed, reduced diameter, smaller valves or the like.
3) We could play with the rate and temperature level of the circulating water and change either or both of the LC and the CB. See fig. 24 for an optimization curve.
4) The total pressure drop determines the flow rate we intend to spend on the circuit. The pressure drop has to be redistributed between the unit until it is fully utilized, which means some trials and errors.

Note that the optimum usually follows the rule of thumb.

3. Recalculation: We will here see if we can get a smaller BPHE or a smaller LC. The fact that there are no large excessive margins and that the pressure drops are underutilised too by about the same degree, except in the pipe work, indicates that the design is close to optimal. There is no point in starting a series of designs from a water temperature level close to the LC to close to the CB.

We will then study two cases

1) We keep the CB and use a smaller LC and see what it means.

or

2) We keep the LC and use a smaller CB and see what it means.

4. Change of the LC. The next smaller model LCS 190 has a nominal capacity of 190 kW.

1) To squeeze out 215 kW we need to increase the MTD about two degrees.
2) That on the other hand means a much smaller MTD in the CB, possible but with the use of H-plates.
3) H-plates have a higher pressure drop than M-plates. We then reduce the flow rate by accepting 6 K temperature change instead of 5 K.

4) The lower flow rate reduces the pressure drop in the pipe work and LC and increases the head.

5) The LC has a correction table, which corrects the nominal capacity to other temperature programs. The correction factor should be F = 190/215 = 0.884. To get this correction factor for a temperature difference of 6 K, we need a temperature difference between the inlet air and inlet liquid of about 17.15 K (interpolated).

6) The circulation rate then decreases to 34283 kg/h. That means a recalculation of all the pressure drops.

    Pump head increases to: 231 kPa (Pump curve)
    Pipe work decreases to: $55 * (5/6)^{1.8} = 41$ kPa
    LC decreases to: 37 kPa
    Spare, 10 % of total: 23 kPa
    Left for the CB: 130 kPa

7) A calculation of a CB300 for the new duty gives:

    CB300-50H, 14. 44 m2, Margin = 0 %, MTD = 5.4 °C
    R22, 78 -> 45 °C
    Glycol 42.15 <- 36.15 °C, ΔP2 = 129 kPa

5. Comments. It fits more or less exactly to the duty and all the available head has been utilized. The pump operates at a somewhat changed point on the characteristic curve.

In the next section, we will see if we can decrease the CB by keeping the LCS 219.

6. Change of the CB.

To decrease the area of the CB300 we need to:

1) Change to H-plates to increase the K-value.

2) Reduce the flow otherwise; the pressure drop will be far too large. We need to reduce the flow more than in the previous recalculation.

3) The correction factor has to be 219/215 = 1.0186. Try with an 8 K temperature difference. We obtain this correction factor for a temperature difference of 16.13

4) The circulation rate then decreases to 25712 kg/h. That means a recalculation of all the pressure drops.

    Pump head increases to: 258 kPa (Pump curve)
    Pipe work decreases to: $55 * (5/8)^{1.8} = 24$ kPa
    LC decreases to: 36 kPa
    Spare, 10 % of total: 26 kPa
    Left for the CB: 172 kPa

5) A calculation of a CB300 for the new duty gives:

    CB300-34H, 8.96 m2, Margin = 0 %, MTD = 6.7 °C
    R22, 78 -> 45 °C
    Glycol 41.27 <- 33.27 °C, ΔP2 = 159 kPa

7. Comments. Once more, the optimum for a PHE occurs for a fairly low flow rate and high pressure drop. A comparison of the results gives.

	LC	BPHE	Pump
Before	LCS 219	CB300-50	41140 kg/h 198/128 kPa
LC Opt.	LCS 190	CB300-50	34283 kg/h 231/207 kPa
CB Opt.	LCS 219	CB300-34	25712 kg/h 258/219 kPa

Normally it is difficult to get hold of thermal data for the LC such as exact temperature difference and the application of the rule of thumb is then more tedious. Moreover, the rule of thumb works best when there is an almost continuous change in size as in a PHE. A LC comes in definite sizes and the difference could be considerably between a change upward and downward.

Fortunately, that makes the choice easier. The next lower LC size is LCS 166, which requires almost 20 K temperature difference, i.e. there is no place for a water circuit. A decrease of the CB too much lower than 34 plates means a very low flow rate and/or a high pressure drop.

The recalculated variants are both better than the original. Which is the better depends on the actual prices. As the LC optimization utilizes the total pump head better, at least from a purely technical point this is the best.

4.2.2. Optimization constraints.

There is another aspect of optimization, which might impede the design of a fully optimized system. In many cases, there are restrictions, which have to be complied with. As examples can be mentioned:

a) A limitation on the total amount of cooling water, which can be drawn from a water source. The optimization might indicate a very large flow, but that might simply not be available.

b) A restriction of the sea or river water temperature increase in order not to unduly disturbs the marine flora and fauna. Thus, even though only a small amount of water is necessary, more has to be pumped.

c) Danger of scaling if water temperature is too high.

d) Penalties on peak load electricity consumption, might affect the ratio between capital and operating costs.

e) Noise restrictions could impede large pumps or fans.

f) Freezing if the water temperature is too low.

g) A calculation might give an optimum for a low water flow rate and a high pressure drop, but for a particular application, there might only be available low head pumps of various capacities. Circulation pumps for A/C duties are often of this type.

h) Frost formation on the fins of a unit cooler could restrict the lowest glycol or refrigerant temperature.

i) The larger the temperature difference between the air and the glycol or refrigerant temperature in a UC is, the dryer the air will be. This could be both an advantage and a disadvantage, which it mostly is.

Thus, application of the rule of thumb has to be modified for this. A unit cooler might appear unnecessarily large and with a very small temperature difference. It thus sticks out as prime object for optimization, but the small temperature might be necessary in order not to frost the fins or dry out the air.

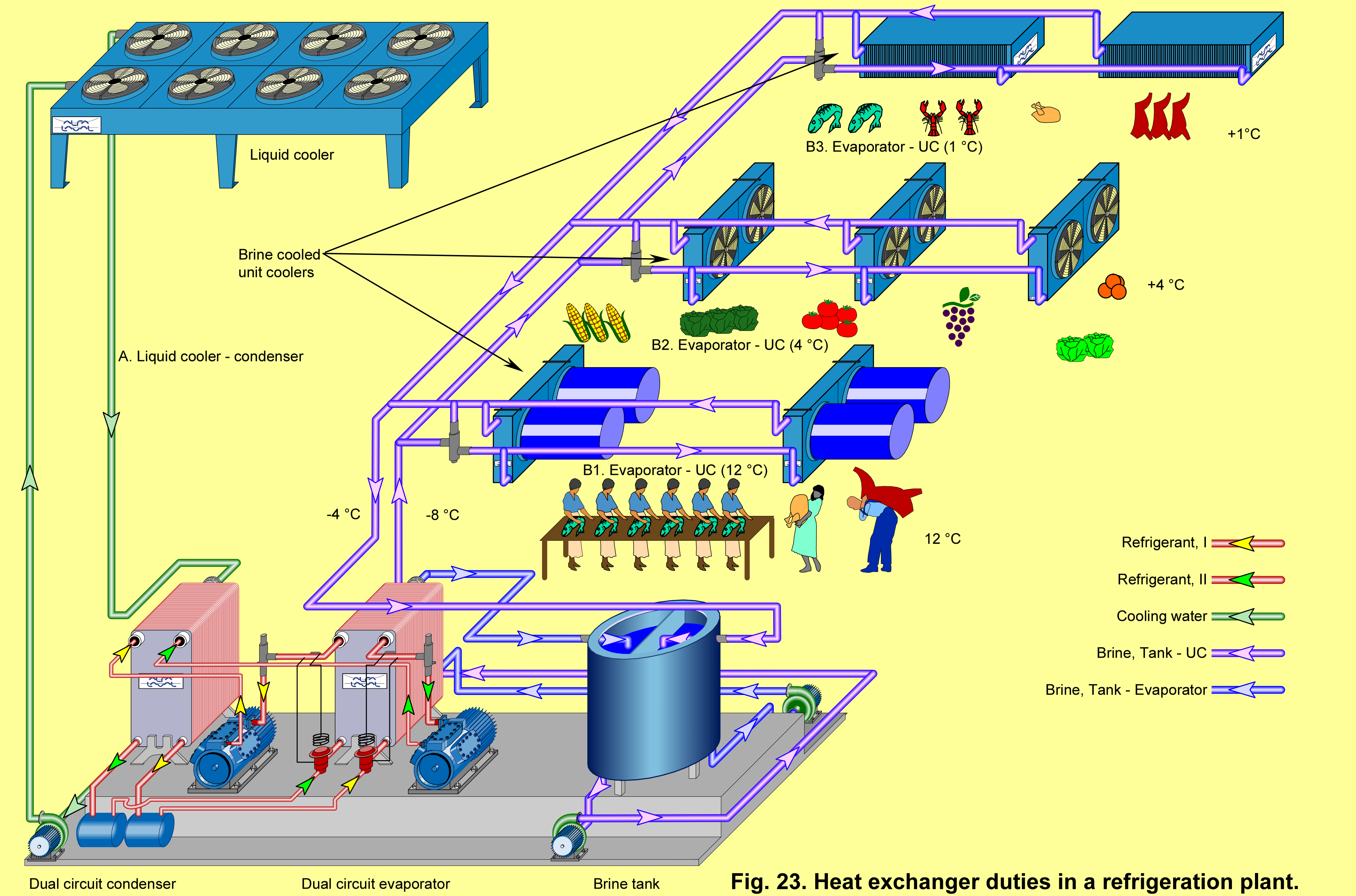


Fig. 23. Heat exchanger duties in a refrigeration plant.

4.2.3. Optimization of the unit cooler - evaporator circuits.

In the system shown in figure 23 there is one tank-evaporator loop feeding three tank-UC loops but apart from the large number of trials and errors necessary, the techniques used in the previous sections can be used here too.

We will here consider the effect of the two last constraints in the previous paragraph, frost formation and too dry air.

Note that the layout of a system of unit coolers for proper treatment and storage of especially alimentary products at various temperature levels is a complicated subject which is not covered by this manual. Here only some aspects of optimization of the PHEs are covered. We use the plant in figure 23 as an example.

Example 4:

Set-up.

a) Meat & fish:	+ 2 °C/80 % Humi.	71 kW
b) Fruits & vegetables:	+ 4 °C/80 % Humi.	47 kW
c) Preparation:	+12 °C/70 % Humi.	38 kW
d) Totally available		160 kW

The unit coolers for all three spaces are supplied with 30 % ethylene glycol of - 8 °C. The return temperature to the tank is assumed to be -4 °C.

This is also the temperature program for the tank-evaporator loop, but as this has an individual pump, the flow rate and the temperatures can be adjusted.

Frost formation. The glycol enters well below zero. Inevitably, frost will form on the fins. The preparation room coolers could conceivably be fed with glycol at above zero, the others not, but that would mean a double evaporator system, which is expensive. Thus, a system of defrosting has to be installed.

Humidity. In a cooled, sealed off space the long-term humidity is set by ratio of water vapour pressure at the lowest temperature the air encounters and the vapour pressure at the bulk temperature. External factors have an influence, e.g. if there is a continuous influx of moist air, the relative humidity increases.

In most cases, there is thus a limitation on the largest temperature difference possible in order to keep the humidity on a certain level. This looks something like:

Temp. diff. °C	5	6	7	8	9	10	11
Humidity %	93	89	85	82	79	76	73

In the case of a liquid cooled UC, there is an extra complication that the temperature is not constant but varies from the inlet to the exit. In this example, we will assume that the mean temperature of the glycol can be assumed.

In the above examples we have assumed at least 80 % humidity in a) and b) and 70 in c), corresponding to temperature differences of 8, 8 and 12 K. The temperature programs thus looks like:

a) Glycol 3*5599 l/hr	-8 to -4 °C,	3* IRL 555,	3*24	kW
b) Glycol 1*5520 l/hr	-8 to 0 °C,	1*IRL 657,	1*46	kW
c) Glycol 2*1916 l/hr	-8 to 8 °C,	2*IRL 451,	2*19	kW
d) Glycol 26149 l/hr	-8 to -1.4 °C		156	kW

Note 1. There is no possibility to obtain a humidity of 80 % or more in c) as this would require a mean temperature of +4 °C and a temperature program of -8 to +16 °C.

Note 2. The capacities of the UC differ slightly from the required but as the available capacity, 160 kW, is higher than the requested, 156 kW, it has no importance.

Can we optimize anything?

If the mean temperatures and the entrance temperature are fixed, the flow rates are also decided.

It would be possible to change the entrance temperature, but a reduction would mean still more difficulties in maintaining the humidity in c) (if that is required) and increased frost formation. Increasing the temperature might be possible, but if the mean temperature of a) must be kept it means a very rapid increase of the circulation rate.

The tank-UC loop is thus set to -8 °C -> -1.9 °C. What about the tank-evaporator loop -4 °C -> -8 °C? This temperature programme assumes that the circulating amount of glycol is larger than the tank-UC loop. A part of the cooled glycol then mixes with the returning glycol from the UC and form the entering -4 °C glycol.

A design of a CB gives:

CB300 62M, 16.80 m2. Margin = 5 %, MTD = 5.2 K
E. glycol 39215 kg/hr -4 -> -8 °C. ΔP = 25 kPa
R22 -10 <- -15 °C.

A design for various glycol flow rates and corresponding inlet temperatures reveals that the area-flow curve is extremely flat, i.e. the present design is close to an optimum. A check for the loop temperature programme gives

CB300 58M, 15.68 m2. Margin = 5 %, MTD = 6.1 K
E. glycol 23762 kg/hr -1.4 -> -8 °C. ΔP = 12.3 kPa
R22 -8 <- -15 °C.

There will be a saving in the pumping costs. However, it is advisable to keep the pressure drop fairly high, otherwise, there might be maldistribution of glycol.

4.2.4.Change in condensation and evaporation temperatures.

The optimized condenser and evaporator are thus:

Pos.	Unit	MTD K	Plates/MTD
Cond:	CB300-34H,	6.7 K.	5.07
Evap:	CB300-62M,	5.2 K	11.9

A decrease of the evaporation temperature to -16 °C and a decrease of the condensation temperature to 43.7 °C

Cond:	CB300-94H,	4.4 K.	21.4
Evap:	CB300-56M,	5.7K	9.82

There is thus a very sharp increase of the condensing surface with decreasing condensing temperature. The optimum probably lies very close to the first trial.

Note! The rule of thumb has to be used with judgement as two-phase duties can behave somewhat erratic for high pressure drop, especially at evaporation and when approaching the temperature pinch (Fig.17) at condensation.

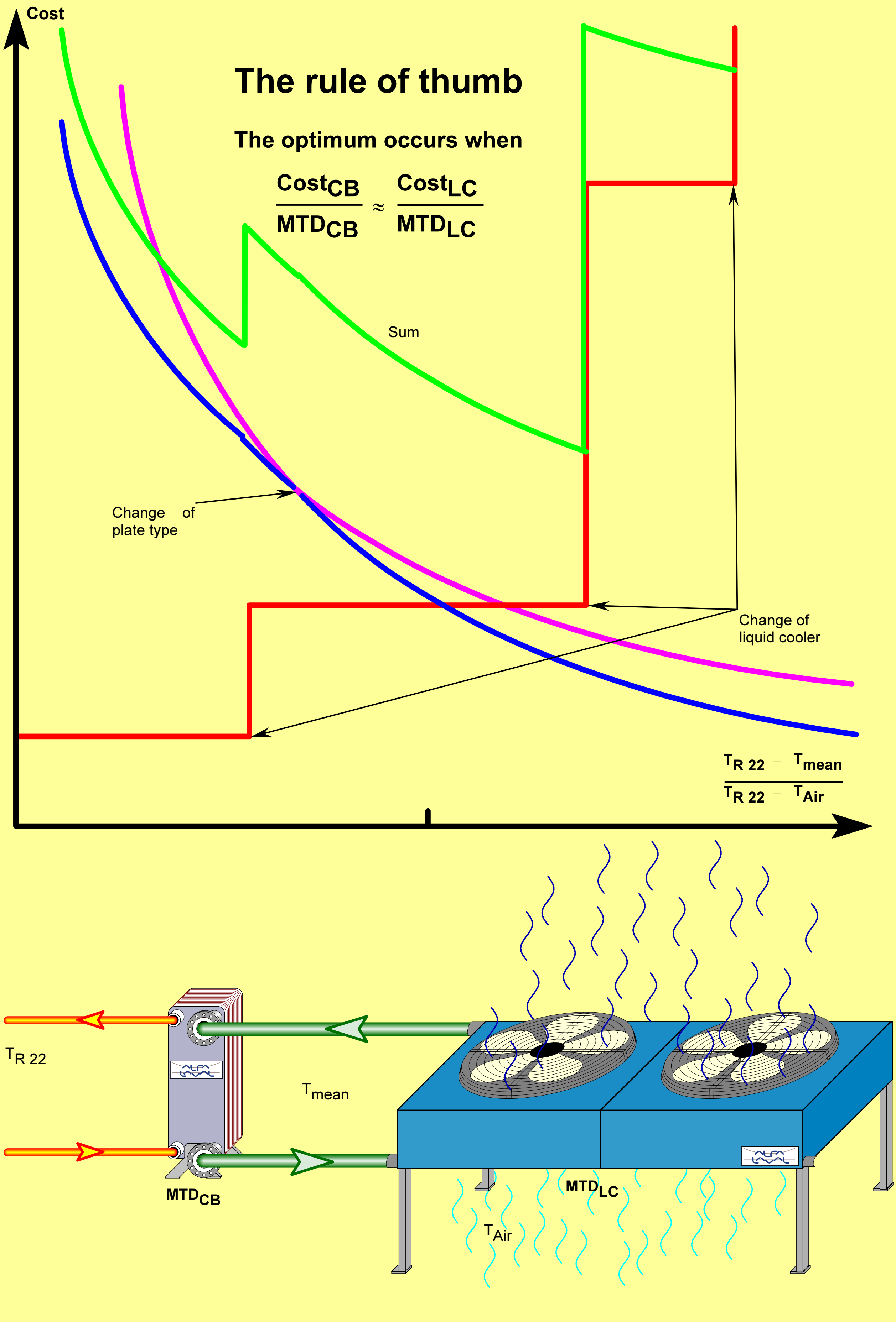


Fig. 24. Optimization of two heat exchangers interconnected by a circulating liquid stream.

3. Design & Installation.

1. Design.

1.1. Design and material.

The BPHE is a variant of the traditional Gasketed Plate Heat Exchanger (GPHE). It is composed of a number of "herringbone" corrugated, 316 type stainless steel plates, assembled in a plate pack, much as in an ordinary PHE.

The angle and depth of the corrugations together with the dimensions of the plates determine the thermal and hydraulic properties of the BPHE. The corrugation angles are in opposite directions for adjacent plates. The plate pack is then supported by a number of contact points, formed where the ridges of the corrugations meet.

Instead of a gasket groove, the plate edge is folded down, making contact with the adjacent plate. Between the plates, thin foils of copper - with the same shape as the plates - are placed. The plate pack is clamped together between two thicker, non-corrugated end plates, where the nozzles are attached and the assembly is brazed in a vacuum oven. See figure1.

1.2. Plate design.

A channel formed by two plates with an acute corrugation angle and a large channel height, has a low pressure drop and a low heat transfer coefficient for a given flow.

When the corrugation angle increases and/or when the channel height decreases, the pressure drop and the heat transfer coefficient increase accordingly.

Increasing the length of the plate has very much the same effect as decreasing the channel height or increasing the angle. The pressure drop increases because of the longer flow length. The heat transfer increases as well, not because of a higher heat transfer coefficient, but because of a larger heat transfer area.

- A plate with an acute angle is of the L(ow) type.
- A plate with an obtuse angle is of the H(igh) type.
- A *channel* formed by an L-plate and an H-plate is of the M(edium) type; an M-*plate* does not exist.

Depending on the thermal duty, either channel type can be optimal for the duty in question. In general, duties with high volume flows and low heat transfer (with a low specific heat or a low temperature change) require L-channels. Air at ambient pressure is a good example, and requires such extreme L-channels that it is impractical to use a PHE.

Duties with small flow but high heat transfer (high specific heat, phase change or large temperature change) favour H-channels. A good example of this is phase changes of refrigerants. Consequently, almost all refrigeration heat transfer duties require H-channels, which is the standard plate type for refrigeration BPHEs.

Theoretically, it is possible to mix channel types in one PHE, i.e. a number of H-channels followed by a number of M-channels. This would give a heat exchanger with a thermal performance between H and M channels. This is a very common method of changing the thermal performance in the GPHE.

In case of heat transfer with phase change, this mixing will lead to a serious maldistribution of the media between the first H-channel and the last M-channel. It is therefore not used in refrigeration BPHEs.

1.3. Plate denomination and arrangement.

See figure 2.

- A refrigeration BPHE has always all the refrigerant channels surrounded by water channels, i.e. the number of water channels is one more than the number of refrigerant channels, thus the outermost channels on each side of the plate pack are water channels.

- The number of heat transfer (channel) plates is then even. This is a clear indication of a refrigeration BPHE.

- The space between the first channel plate and the front plate is open on older models. These have to be installed with the angle pointing upwards: otherwise, water can enter, collect and freeze.

- The corrugation on the last channel plate of refrigeration units points downward. If water enters between this and the end cover plate, it can collect and freeze.

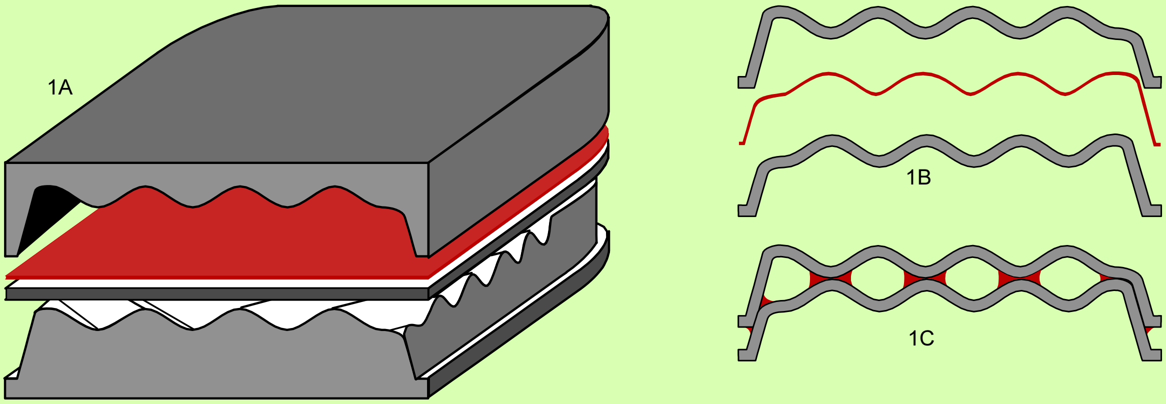
  To prevent this, some old models have an extra plate, with an upward pointing angle. The new channel formed is a refrigeration channel, which now is the outermost. It is then sealed off from the remaining channels by the addition of washers at the portholes. See figure 5.

- The space between the last channel plate and the end cover plate is effectively sealed off on current models by the folded down edges of the last plate; consequently, an extra plate is no longer necessary.

- All current models have an additional uncorrugated seal plate between the front cover plate and first channel plate. This plate is only there to seal off the space.

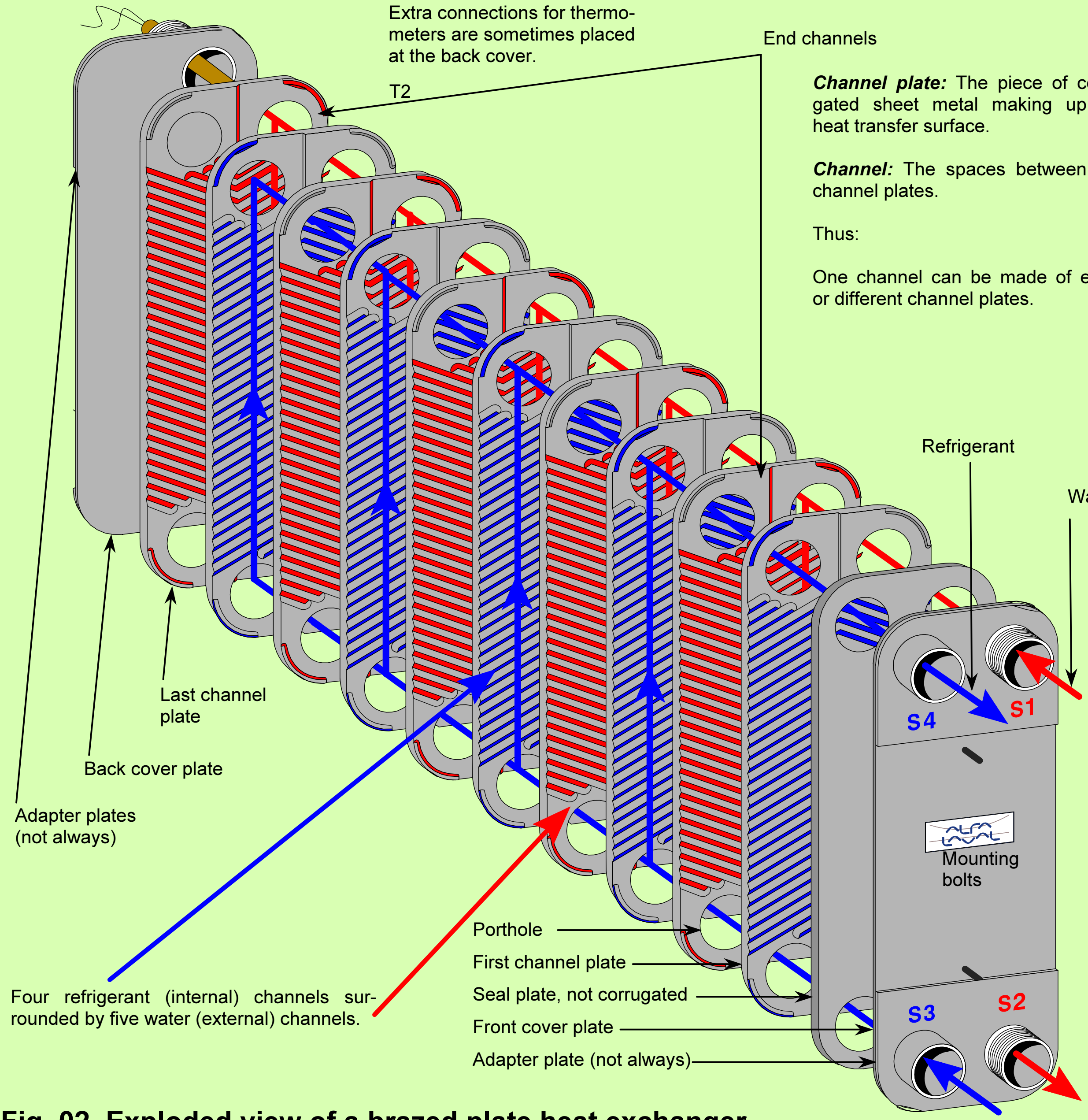
All current models are symmetrical, i.e. the media can enter either end of the channel.

Exception. Some BPHEs are specifically designed as evaporators. These have to be installed with refrigerant connections at specified positions.



A stainless steel sheet and a copper foil are pressed together in a hydraulic press (1A). A number of plates with their foils are then stacked with the corrugation angles in opposite directions (1B & C). The plate pack is then placed in a vacuum oven and heated to the melting point of the copper. Because of the surface tension, the liquid copper collects at the edges and the contact points, effectively forming sealed off channels.

Fig. 01. The manufacturing of the brazed plate heat exchanger.



Extra connections for thermometers are sometimes placed at the back cover.

Channel plate: The piece of corrugated sheet metal making up the heat transfer surface.

Channel: The spaces between two channel plates.

Thus:

One channel can be made of equal or different channel plates.

Four refrigerant (internal) channels surrounded by five water (external) channels.

Fig. 02. Exploded view of a brazed plate heat exchanger.

1.4. Nozzle directions.

Figure 3 shows the denominations of the connection positions. In the case of a misplaced or unreadable label, base the identification on the position of the folded down edge of the channel plates. This is folded away from the front cover plate. The refrigerant nozzles are normally for soldering and the BPHE should then be positioned so that these nozzles are to the left.

While it is possible to arrange a BPHE with the refrigerant nozzles to the right, at S1 & S2, this is not recommended as it could lead to confusion if the unit is later used for other duties, not to mention that a non-standard unit will be more expensive and involve longer delivery time. If the refrigerant nozzles have to be to the right, simply turn the unit upside down. Figure 4.

1.5. Identification of an unknown unit.

Figure 5 gives some hints on how to determine if an unmarked BPHE is a refrigeration unit and the refrigerant side. Note, however:

- Do not count the seal plate, if any, when counting the number of plates.

- For strength reasons some plates are double on some models. Two plates are placed together, with the corrugation angle pointing in the same direction and a copper foil in between. The plates are in metallic contact over the entire surface and the plate distance as seen from the outside is maybe 0.1 instead of >2.4 mm. This double plate obviously counts as one.

  The most difficult BPHE to identify is the model with an extra blocked channel (figures 2 and 5C). This has an uneven number of plates, like a non-refrigeration unit. If the unit has nozzles at the end cover plate, it might be possible to see the double washer, which blocks the channel; otherwise, only a careful measurement and check for a single washer at the S1,2 reveals the type.

2. Properties.

2.1. Flow regime.

The channels formed by the "herring bone" corrugation will force the liquid in a helix like flow path - a type of forced turbulence. Actually there is no sharp transformation between laminar and turbulent flow. This gives the PHE much higher heat transfer coefficients than normal S&THEs, especially for viscous flow.

2.2. Control.

The practical effect of the flow characteristics is that thermal performance does not exhibit the typical jumps or hysteresis effects which can occur when the flow changes between laminar and turbulent flow in a tubular HE, making control difficult or impossible.

2.3. Compactness.

The BPHE weighs about 25 % less and occupies less than 25 % of the volume of its closest rival, the coaxial.

To be fair to the coaxial, it has to be said that sometimes the space inside the helix wound shell can be used, e.g. as a liquid receiver.

2.4. Liquid volume.

The volume on the refrigerant side of a BPHE is about 2 l/(m2 heating area), i.e. about ten times lower than a multi-tubular HE. The practical implications of this are:

- Low filling volume.
- Rapid response to capacity changes.

2.5 Pressure and temperature limits.

♦ Temperature	-160 °C to +150 °C
♦ Pressure:	30 or 40 bar

The burst pressure of a BPHE is more than 150 bar, and most pressure vessel codes require a burst pressure of five times the design pressure. The above limits are general, for specific data see the product catalogue.

The design pressure for nickel-brazed units is lower.

Non-refrigerant BPHEs are sometimes used for high temperature, low-pressure service and are accordingly rated for a higher temperature, but a lower pressure.

2.6. Thermal efficiency.

The induced turbulence together with the small hydraulic diameter means very large heat transfer coefficients - up to three times those found in tubular equipment.

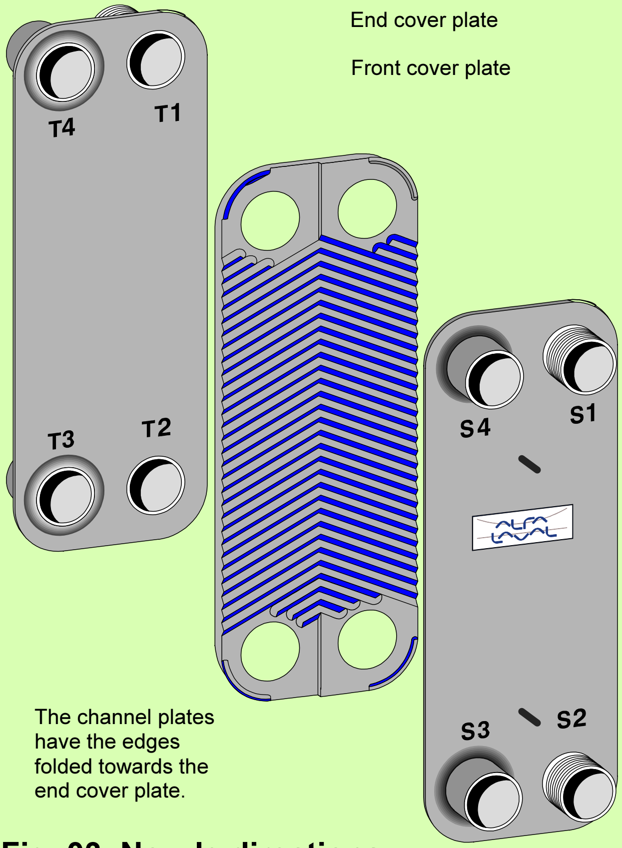
The almost true counter current flow means that the practical temperature difference between the media is very close to the ideal.

The only other HE type which can rival or even surpass the BPHE is the coaxial HE, which however has a much lower transfer coefficient. This also tends to be very long or requires many in series to achieve a close temperature approach.

The true counter current flow also means that high heat recovery is possible; more than 90 % is no problem as compared to less than 50 % in a multi-tubular HE.

2.7. Fouling.

This is a complex subject and is treated in detail in the section about fouling and corrosion. Suffice to say here that the absence of dead spots and high shear forces makes the BPHE very resistant to scaling. It is somewhat sensitive to clogging by fibre-like impurities in the water, though, but less so to sand like impurities.



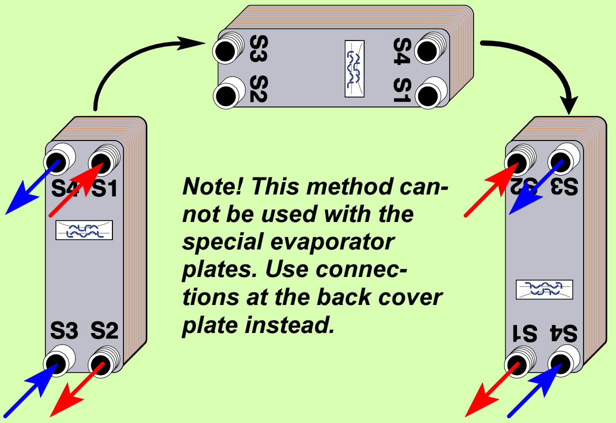
The channel plates have the edges folded towards the end cover plate.

Fig. 03. Nozzle directions.

The name plate is placed on the front cover plate. The two nozzles to the right, S1 & S2, are normally used for the liquid and the two to the left, S3 & S4, for the refrigerant. The corresponding nozzles, T1 - T4, on the back cover plate are not always used. The refrigerant nozzles are normally made for soldering and the liquid nozzles for threaded connections.

Observe that each fluid will remain on its side regardless of the internal arrangement of the channels, i.e. water cannot enter at S1 and leave at T3.

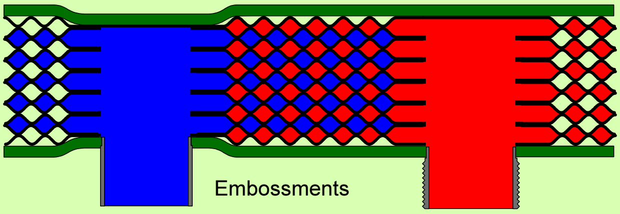
Exception. The BPHE type CB12 (replaced by CB14) is the only BPHE with diagonal flow. The nozzles are S1,3 (or T1, 3) for one side and S2, 4 (or T2,4) for the other.



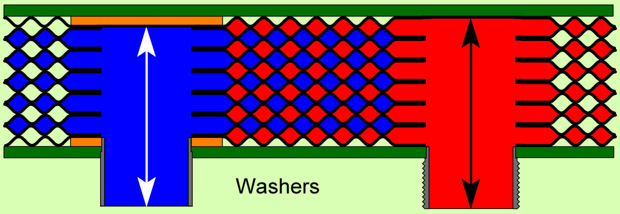
Note! This method cannot be used with the special evaporator plates. Use connections at the back cover plate instead.

Fig. 04. Reversed connection placement.

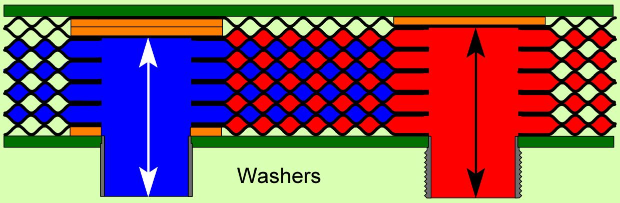
The unit should be placed upside down.



A. Design with embossment on the end plate.



B. Design with washers.



C. Heat exchanger with a blocked last channel.

Fig. 05. Identification of the refrigerant side.

A refrigeration BPHE always has all the refrigerant channels surrounded by liquid channels. The question might arise if an unknown unit has a side with one more channel and if so which side. Here are some guidelines:

The number of heat transfer plates has to be even. A complication is that there is a sealing plate (Fig. A.) and sometimes the first or the last plate is double. These extra plates should not be counted. The channel plates are well separated, with a pitch from 2.4 to 2.9 mm.

1. The second channel from either cover plate has to be connected to the cover plate through embossments on the cover plate or washers. The embossments, easily seen on the cover plates, identify the type of unit.

Embossments at positions S3,4 and T3,4 indicate a refrigeration unit with the refrigerant side at position 3 & 4.

Embossments at S1,2 & T3,4 or S3,4 & T1,2, indicate a normal unit and equal number of channels on each side.

2. In most cases it is impossible to see the location of the washers from the outside. The only possibility then is a careful measurement of the porthole depth.

3. Some old models have closed end channels and thus an even number of plates. If the side with the shorter porthole has washers at the inlets, it is a refrigeration unit.

3. Flow arrangement.

3.1. The one-pass BPHE.

3.1.1. Evaporator.

- ♦ The evaporating fluid is normally flows upwards and the heating media flows downwards.

- ♦ Downwards evaporation is not impossible but needs a comparably high channel and low nozzle pressure drop to distribute the liquid properly.

  A high nozzle pressure drop means that the fluid will have a maldistribution from the first to the last channel. A low channel pressure drop means that the liquid will not distribute properly over the channel width.

  Downwards evaporation in BPHE has been little studied and tests should be made before an installation is made. Expect a fairly large capacity reduction compared to upwards evaporation.

- ♦ A direct expansion evaporator with an entrance nozzle that is too large, resulting in low velocity, could cause the liquid and vapour to separate and make the evaporator difficult to operate. This is especially severe in downward flow.

- ♦ An evaporator with a restriction at the channel inlet distributes the refrigerant more evenly, both over the plate pack length and the channel width.

- ♦ A DX evaporator has the flow normally in counter current as this gives the highest MTD.

- ♦ A flooded evaporator normally flows in cocurrent. The reason for this is the resulting large temperature difference at the refrigerant inlet, which facilitates the start of the boiling.

  Note! In case of a large (> 10 K) temperature differences at the refrigerant inlet, there might be too much vaporization at the very first stage of the heating surface. Especially at the startup, when the evaporator and the exit pipe are filled with liquid, the pressure drop for this vapour is larger for the flow path through the evaporator to the separator than for the backwards path through the feed pipe to the separator. The flow might then reverse or oscillate.

- ♦ See also the section on evaporation.

3.1.2. Condenser.

- ♦ The condensing fluid normally flows downwards and the cooling media upwards.

- ♦ Upwards condensing needs a comparably high channel pressure drop to drive the liquid upwards.

  A low pressure drop can cause flooding of the condensate, which will cause an irregular flow and capacity loss. The lowest pressure drop, when the flow is on its lower control limit, should in this case be at least as great as the liquid static head.

- ♦ If the nozzle pressure drop is too large a part of the total pressure drop (> 25 %), the flow will distribute unevenly from the first to the last channel, causing a capacity loss. The solution is a BPHE model with larger nozzles or double inlet nozzles.

- ♦ If the pressure drop is a large part of the total pressure, there is a danger of sonic booms. This is, however, seldom the case in a refrigeration condenser.

- ♦ Counter current flow is normal.

3.1.3. Double nozzles.

The single pass arrangement is the only one, which permits the use of double inlet and/or exit nozzles. These are used to reduce the nozzle velocity and thus decrease the maldistribution of fluid along the plate pack.

A double nozzle is normally placed at the vapour entrance of a condenser and at the vapour exit of an evaporator.

Double inlets & exits are used for the liquid side as well, to reduce the nozzle velocity. If double nozzles on the liquid side are necessary, install them on the refrigerant side as well. See figure 6.

3.1.4. Back end nozzles.

An extra pair of nozzles can be placed at the back end cover of the water side to be used for thermometer connections (figure 2) or as normal nozzles.

Nozzles at the back end cover are an advantage if the BPHE is to be installed in a package unit. The backside with the water nozzles is then placed towards the outside for easy connection of the water pipes and the refrigerant nozzles are then directed into the unit for easy connection to the cooling circuit inside. See figure 13A.

Avoid this arrangement if the nozzle velocities are high, i.e. if the nozzle pressure drop is a large part of the total pressure drop. The resulting maldistribution of the fluids will be in different directions. See figure 6.

3.1.5. Vents and drains.

The one-pass BPHE can be vented and drained on both sides through the normal connections. See figure 7.

Note that a vent for is located at the end of the heating surface, normally at the bottom of a PHE. This is especially important for water vapour and ammonia, the only two vapours easier than air. In case of other vapours in condensers with very low pressure drops, there might be a need for venting at the top. See the chapter about condensation for a further discussion.

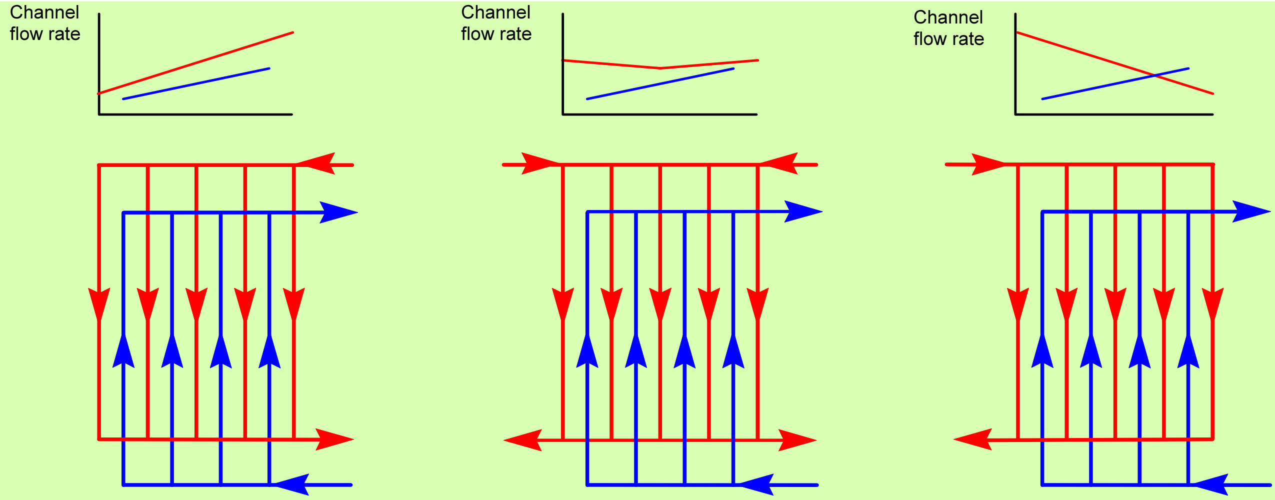


Fig. 06. Maldistribution.

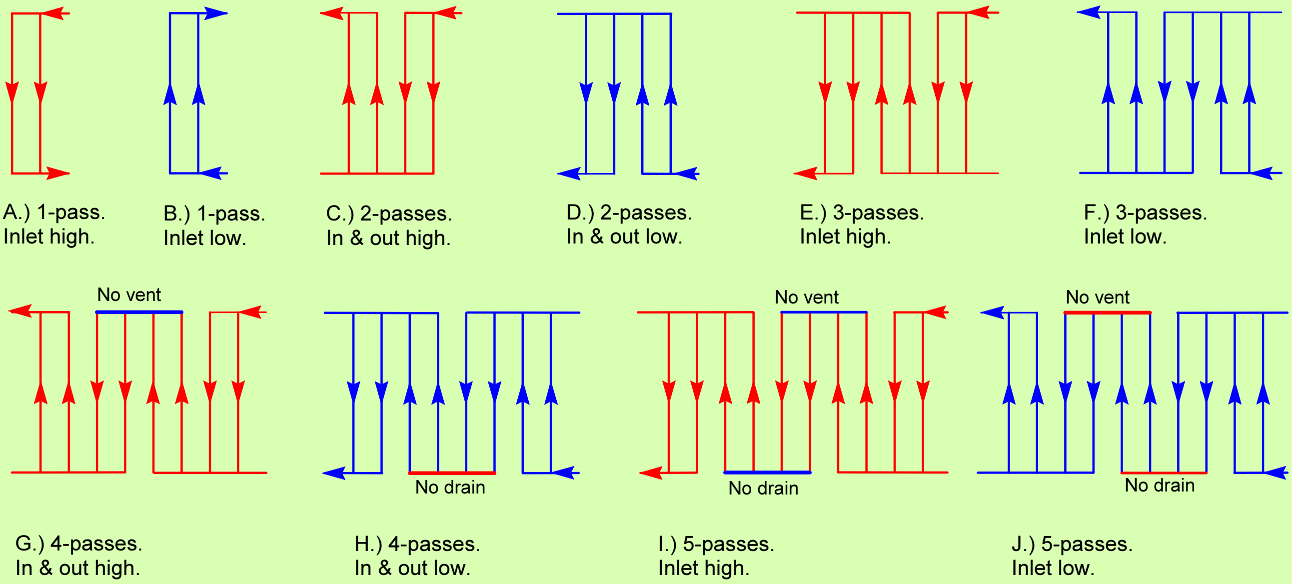
The figure shows the effect of various nozzle arrangements on the channel distribution in the case of ***large*** nozzle pressure drops. The diagram above the channel arrangement indicates the channel flow rate.

A. All nozzles at the front end cover. The channel flow rates decreases from the first to the last channel, i.e. a channel with low refrigerant flow rate is surrounded by water channels with low flow rates and vice versa.

This means that the refrigerant leaving the channels has close superheats, but there is a loss of capacity.

B. The water side has nozzles at the back end cover as well. A refrigerant channel with a low flow rate may face a water channel with a high flow rate, i.e. the state of the refrigerant emerging from each channel varies more than above. The overall result is a decrease in capacity.

C. The water connections are here at the back cover. Here a water channel with low flow rate faces a refrigerant channel with high flow rate and vice versa. The refrigerant leaving the channels has different degrees of superheat from the inlet to the exit. In the worst case liquid refrigerant could emerge from the first channel.



A.) 1-pass.
Inlet high.

B.) 1-pass.
Inlet low.

C.) 2-passes.
In & out high.

D.) 2-passes.
In & out low.

E.) 3-passes.
Inlet high.

F.) 3-passes.
Inlet low.

G.) 4-passes.
In & out high.

H.) 4-passes.
In & out low.

I.) 5-passes.
Inlet high.

J.) 5-passes.
Inlet low.

Fig. 07. Venting & draining of a BPHE.

Only one side is shown. The other side is horizontally reversed, i.e. the draining and venting are equal. The number of passes on the two sides does not have be equal.

A, B. Vents & drains through the normal connections.

C, D. On each side an extra vent or an extra drain is necessary on the front ***or*** back end cover.

E, F. On each side an extra drain ***and*** an extra vent are necessary on the front ***and*** back end cover.

G, H. One section is impossible to drain ***or*** to vent despite nozzles at all possible positions.

I, J. One section is impossible to drain ***and*** to vent despite nozzles at all possible positions.

3.2. The multi-pass BPHE.

3.2.1. General.

The BPHE arranged in multi-pass either as condenser or evaporator has been studied and tested less than the single pass BPHE, Therefore, a careful evaluation and, if possible, tests should be carried out before a major installation.

The normal flow direction is upwards evaporation and downward condensation. In a multi-pass BPHE the flows are reversed in one or more passes. This can lead to:

- ♦ ***A condenser*** can normally operate from 100 % to 0 % capacity. This is however not the case if the flow is upwards; flooding may then occur. The major effect of this is irregular flow at low capacities, which can lead to control problems.

  A condenser should therefore be designed with the last pass in downwards flow, at least if the condenser has to operate at a very low fraction of the design capacity.

- ♦ ***An evaporator*** cannot normally operate at very low fractions of the normal capacity (flooding and oil retention); consequently, downwards flow should improve the low capacity operation. However there is a danger of phase separation at a low downward velocity - the liquid enters the first channels and the vapour the last.

  To minimize the danger, the first pass - where the lowest velocity occurs - should then be upwards.

This arrangement works very well for a chiller/heater. The refrigerant flow is in reverse when changing from condenser to evaporator, and the above requirement is maintained in both cases.

3.2.2. Properties of multi-pass.

See figures 8 A, for the channel arrangement of a two-pass arrangement on the water side and one-pass on the refrigerant side. A multi-pass arrangement has the following general properties (see also fig. 10):

- ♦ Only one inlet and exit nozzle is permitted.
- ♦ The two sides are independent. It is possible to have a different number of passes on the two sides. However, not all arrangements make sense. See below.
- ♦ The total number of water channels is one more than the total number of refrigerant channels, i.e. each refrigerant channel is surrounded by a water channel.
- ♦ In exceptional cases - usually for close temperature differences between the media - each pass will be separated from the others by an empty refrigerant channel. The water side then has, as many more channels as there are passes.
- ♦ In case of asymmetrical grouping - an unequal number of passes on each side - there will be groups of channels in counter current and other groups in cocurrent flow, regardless of how the inlets are arranged.
- ♦ In case of symmetrical grouping - an equal number of passes on each side - there will be a plate per pass with counter current flow in the adjacent channels. This could be a problem with few channels per pass and a small temperature difference.
- ♦ The passes form U or inverted U. This means that there is a danger of water remaining in a U at shutdown or gas in an inverted U at startup.

  In order to completely drain or vent a BPHE, extra vents can be added. See figure 7 for the possibilities of venting and draining. As more than three passes are very uncommon in refrigeration, venting & draining is no problem.

  In normal operation, letting the liquid have a pressure drop per pass as least as large as the static height difference prevents formation of gas bubbles.

3.2.3. Applications.

The main reasons for using a multi-pass BPHE are:

The nozzle positions. A two-pass arrangement permits an inline connection to the pipes; space in a chiller unit only permits certain nozzle positions, etc.

Antifreeze or antifouling measures. If a BPHE changes - i.e. a new one - from one to two passes on the water side, the channel flow rate doubles. The number of passes on the refrigerant side remains constant. This has three effects:

- ♦ The wall temperature on the water side in an evaporator increases (for a more detailed study, see chapter **7. FREEZE**), i.e. decreased freezing danger.

  (The wall temperature increases if the number of refrigerant passes increases, as refrigerant heat transfer is less velocity-dependent than that of water.)

  However, there could be problems of maldistribution of the evaporating refrigerant; see figure 9 B3.

- ♦ The wall shear stresses increase, i.e. fouling compounds have more difficulty adhering to the heating surface and the unit stays cleaner.
- ♦ The pressure drops increase about seven times. (The flow length and the channel velocity are now double.)

Accommodate a long thermal duty. This is probably the major reason. An example will show why and how. Compare the two following duties:

1.	Water	3 kg/s	50 -> 40 °C or	2.	1 kg/s	60 -> 30 °C
	Water	3 kg/s	40 <- 30 °C or		1 kg/s	50 <- 20 °C

Both the duties have the same capacity, the same temperature difference and the same fluids, thus the required area should be the same.

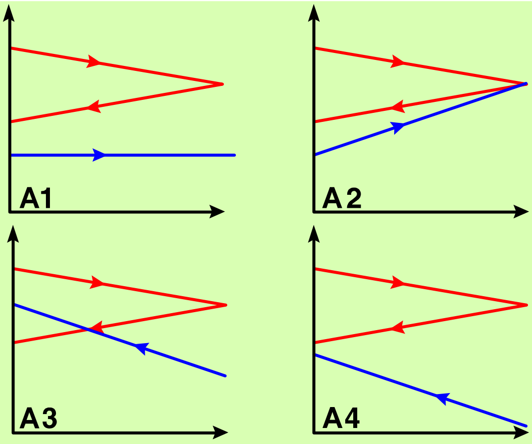
Suppose we design a unit for the first duty. A-CB26-60H with an area 1.45 m^2 and a pressure drop of 0.80 bar will do fine. However, when we try to use this for the second duty, problems arise.

The flow rate is three times less, consequently a lower overall heat transfer coefficient and a larger heat transfer area is necessary.

The only way we can achieve this is to increase the number of plates. This of course means a further reduction in channel velocity; a lower heat transfer coefficient and we end up with chasing our tails. The final result is two CB26 120H with 2 * 2.95 m2 area.

Is there any possibility to reduce the area? There are two:

- ♦ We can switch over to a CB52. This has the same width as the CB26 but is twice as long. Thus, we can reduce the number of plates, which is fine for the lower flow rate. A CB52-50 with 2.4 m2 area is then required.
- ♦ A CB26 in two-pass. Effectively two plates are then connected in series and the result is similar to a CB50. A CB26-90, grouped as 23+22/2*22, i.e. two groups of 22 channels in parallel connected in series and with an area of 2.2 m2, is then necessary.



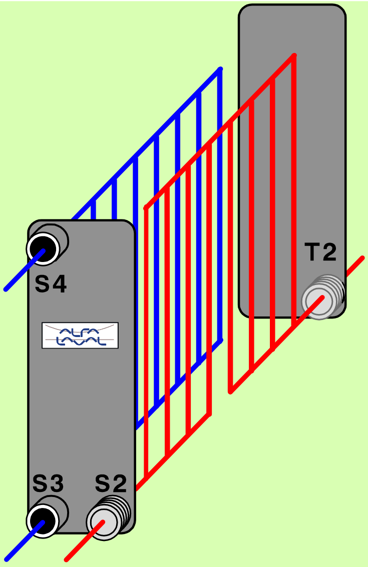


Fig. 08A. One pass/Two-pass.

The refrigerant side is in one-pass and the water side in two-pass. This is an excellent way of cooling water from a high temperature to close to 0 °C in a flooded flow WPHE. The water has a high velocity and thus a high K-value and the wall temperature is moved upward.

The temperature for this unit is shown in figure A1. Because of the almost constant refrigerant temperature no special problems are encountered.

In case of liquid on both sides, the temperature program might become a little problematic. In A2 there is a temperature pinch in the turning point of the two-pass flow.

This might become worse (or better) if the flow direction is changed on one of the sides as in A3.

If the temperature difference increases, the pinch disappears, A4. Note that the flow direction is important.

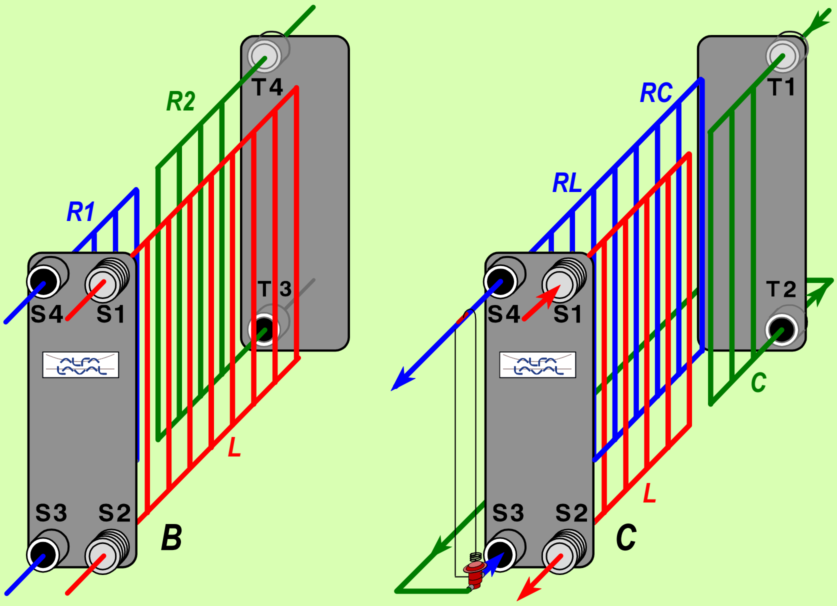


Fig. 08 B - C . Two circuits/One circuit.

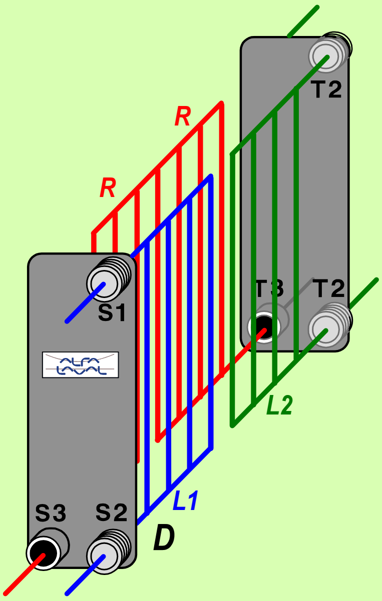


Fig. 08D. Two water/One 2-pass refrigerant circuit.

B. Two refrigerant circuits and one water circuit is a common way of arranging a two circuit unit, but with some drawbacks, see § 3.3.3. The arrangement is symmetrical i.e. both circuits have the same nominal capacity.

It can be also used as a double circuit condenser.

C. The condensate is subcooled in one circuit ($\mathbf{R_C/C}$) with part of the evaporating refrigerant, which superheats very much due to the large temperature difference.

The subcooled condensate then expands, evaporates and cools the water in the other section ($\mathbf{R_L/L}$).

The total superheat is controlled after the evaporator. Due to the high super heat in the ($\mathbf{R_C/C}$) section, the superheat in ($\mathbf{R_L/L}$) is low. This means a very efficient evaporator. Thus, where the heat transfer coefficient is low because of the high superheat ($\mathbf{R_C/C}$) the temperature difference is large and contrary. Depending on the refrigerant type and the temperatures the arrangement can gain some 5 to 10 % in COP.

As the leaving vapour is a mixture between high & low superheat vapours, a good mixing is necessary. An example of this is shown in **4. Evaporators and separators**, figure 21, p. 95. See also figure 05, p. 11 in **1. Applications,** regarding subcooling/superheating**.**

D. This is a common arrangement for a desuperheater and a condenser. The refrigerants desuperheats in the first sections and heats tap water, then the condensing heat is dumped into cooling water.

The disadvantage of two passes (8A & D) is the high pressure drop.. Compared to a single pass section with the same total number of plates this arrangement has about seven times higher pressure drop.

The results are similar, which is what could be expected. What happens if we try to use a CB50 or a CB26 in two passes for the first duty?

We then have to increase the number of plates in order to reduce the pressure drop to an acceptable level, and we end up with either two CB26-69, 2 * 1.675 m2, connected as 2*17/2*17, or a CB50-70, 3.4 m2, connected as 35/34. Both are thermally overdimensioned, about 120 %. In the example above we have used water, but the result is similar if we use refrigerant, either condensing or evaporating, on one or both sides.

By connecting the BPHE in two passes we obtain a HE which is twice as long as the one-pass HE.

Better utilization of the available pressure drop. This is another way of expressing the previous paragraph.

3.2.4. Temperature difference vs. pressure drop in a condenser or evaporator.

When the flow is switched from single-pass to two-pass, the following happens.

- The flow per channel increases. This means a higher heat transfer and a cleaner heating surface.
- The number of channels per pass decreases. This together with the increasing channel flow reduces the danger of maldistribution of the refrigerant.
- ***Warning.*** In an evaporator, there might be phase separation and a maldistribution in the second pass. An evaporator should always be tested before being installed in multi-pass, as there is little practical reference material.
- The pressure drop increases. This unfortunately means that the inlet temperature has to increase if the evaporation temperature is to be kept constant. The available temperature difference is thus reduced.
- In a condenser, the exit temperature is decreased, again with a decreased temperature difference as a result.

The design will be a balance between increasing the heat transfer coefficient and decreasing the temperature difference. Unfortunately, it is impossible to give a general rule where the equilibrium will be; an exact design calculation is necessary.

3.2.5. Channel arrangements.

The symmetrical grouping - the same number of passes on each side - is probably the most common arrangement. As this arrangement is used to accommodate small flows at close temperature differences (high heat recovery) it practically implies counter current flow as well.

Beware of false counter current as opposed to true counter current flow, see figure 9 A.

The asymmetrical grouping is seldom used if there are substantial temperature changes of both the media. As there are a number of channels in cocurrent flow, the effective temperature difference will rapidly decrease the closer the temperatures of the two media are.

- Figure 8A shows the temperature program for two passes on the warm side and one on the cold side. Temperature programs A2 & A3 are impossible here as the cold side temperature approaches or surpasses the warm side temperature leading to an infinitely large heat exchanger, but it is entirely possible for the same number of passes on both sides.
- If, however, one flow has a small temperature change (approaching Fig. 8A1), it matters less if it is symmetrical or asymmetrical; e.g. for an evaporating or condensing liquid, asymmetrical flow could be interesting.

  Figure 9 B shows some possible ways of arranging the passes. Observe that it is not necessary to have the same number of plates per pass.
- The liquid side in multi-pass and the refrigerant in single pass are seldom used; see figure 9 B3.
- An unequal number of passes should only be used in two-phase flow, in order to accommodate a changing vapour fraction. An unequal number of channels on the liquid side are normally very uneconomical.

Counter current vs. cocurrent flow. Thermal effectiveness prescribes counter current flow. The exceptions are:

- In flooded evaporators; see § 3.1.1.
- In case of a very high pressure drop on the evaporating side, cocurrent flow could give a higher temperature difference. See figure 9 C.

The practical consequences of the above are:

- All passes equal on both sides.
- One pass on the liquid side and unequal number of channels per pass on the refrigerant side.
- In some rare cases, e.g. refrigerant/refrigerant, there could be unequal number of channel per pass on both sides or one pass on the refrigerant and multi-pass on the water side (freezing danger).

3.3. The multiple circuit BPHE.

3.3.1. Applications.

In many applications, there is a need for two independent circuits on one side, either the water or the refrigerant side. There are mainly two different applications:

a) One fluid has to be heated or cooled in two steps with different media on the other side in each step.

   A condensate subcooler/vapour superheater and an evaporator is one example. **1. Applications**, figure 5, shows an example where a double circuit unit, as in figure 8 D, could be used instead of two separate units. Refrigerant evaporates in the evaporator section, and then moves to the superheater/subcooler section.

   Another example is a desuperheater/condenser. In the desuperheater section, the refrigerant vapour gives up its sensible heat to a tap water circuit, in the condensing section it is condensed with cooling water. See **1. Application**, figure 10 or Figure 8D in this chapter.

b) In order to control the capacity, especially for evaporators, the refrigerant circuit is split into two, nominally identical circuits. At full capacity both circuits are in operation, at low capacity one is shut down.

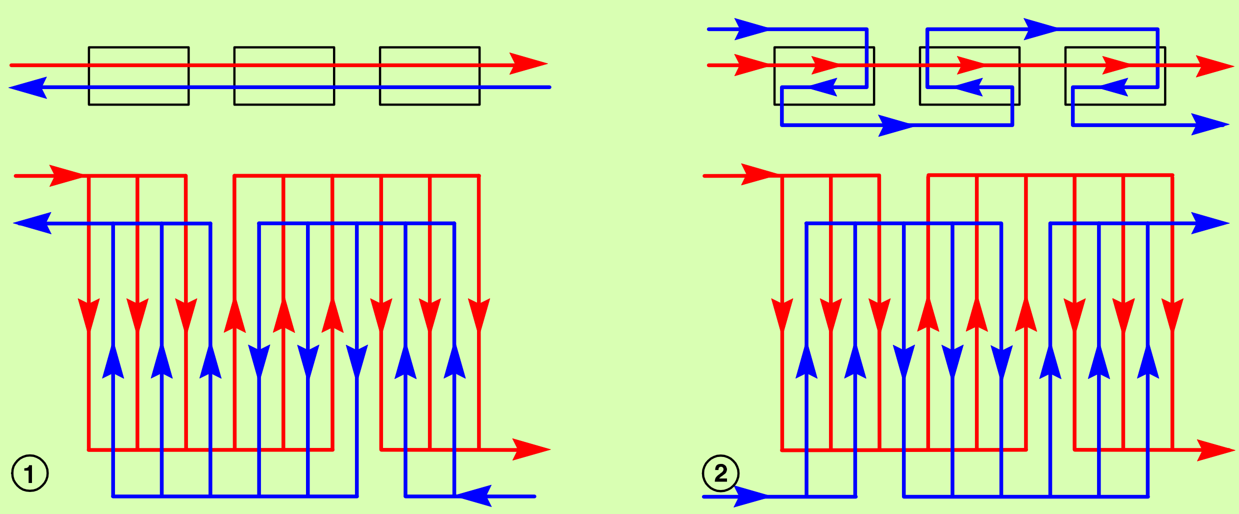


Fig. 09A. Countercurrent flow.

1. True counter current flow. The flow in each pass as well as the passes are connected in counter current. The exception is the plate separating the passes, where two adjacent channels are in cocurrent flow.

2. False counter current flow. The flow in each pass is in counter current, but the passes are in cocurrent flow. This arrangement has a smaller effective temperature difference than 1.

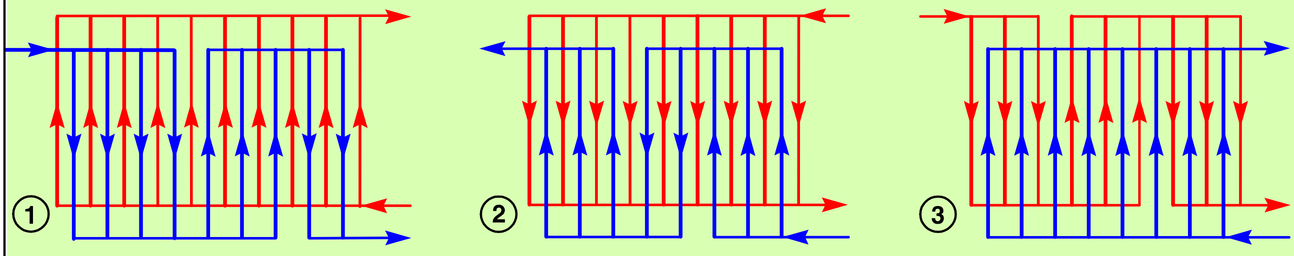


Fig. 09B. Multipass arrangement.

1. The number of channels per pass in this condenser decreases from the first to the last pass, thus adaptingto the decreasing vapour flow and minimizing the pressure drop. Note that the last pass is in downwards flow. An evaporator should have a reversed flow.

2. The number of channels in cocurrent flow - in the middle pass - is small in this evaporator, which maximizes the total temperature difference. The first pass is in upwards flow. A condenser should have a reversed flow.

3. The entering refrigerant meets water of different temperatures in the three passes. This could lead to uneven evaporation and control problems. Use this arrange-ment only if it is necessary to keep the wall temperature as high as possible. If possible make tests.

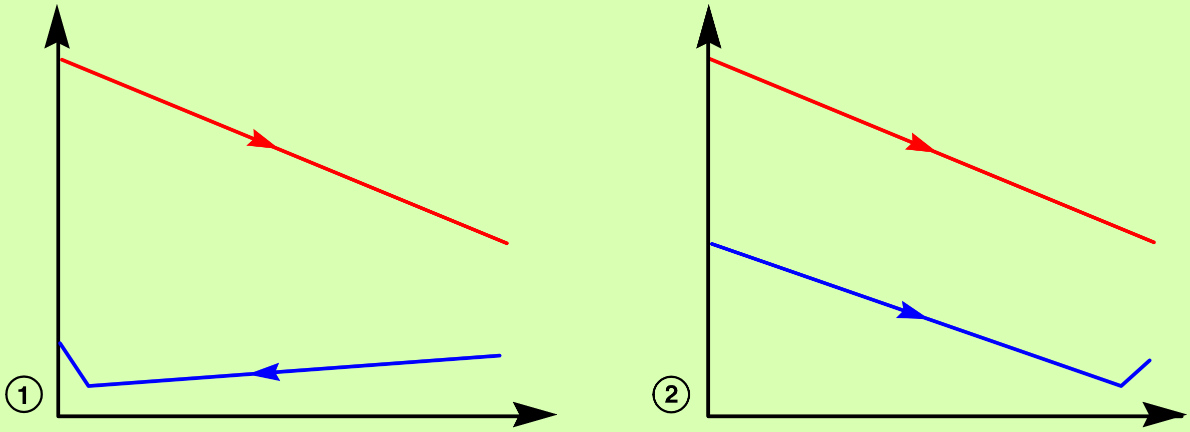


Fig. 09C. Cocurrent vs. countercurrent flow.

1. Countercurrent is normally the best choice, especially for low refrigerant pressure drops.

2. If the pressure drop is large on the refrigerant side, then cocurrent flow could be better.

3.3.2. Types.

There are some possibilities to make a PHE with two circuits on one side, three with any type of PHE and one with a special designed BPHE.

1) ***Normal, one-pass.*** Figure 8B shows a BPHE with two refrigerant circuits and a one-pass water circuit.
2) ***Normal, two-pass***. Figure 10 shows a BPHE with two refrigerant circuits and a two-pass water circuit. Figure 8D is similarly arranged but reversed sides.
3) ***Dual circuit BPHE.*** Figure 11 shows a new design.
4) ***Mixed design,*** where there is refrigerant on both sides.

3.3.3. Normal, one-pass.

This is a normal BPHE with connections both at the front and back end plates of one side. By leaving at least one plate without portholes, this side is separated into two independent circuits. The separation can be at any platesbut the most common are 50/50 and 33/67 partitions. The other side is in one pass and a part of the one-pass side fluid is in contact with one circuit and the other part with the second circuit, see figure 8B.

In case of a double refrigerant circuit evaporator, this design could lead to potential freezing problems. When one circuit closes, the water passes uncooled through this section and the exit water is a mixture of cooled and uncooled water. If this mixed water temperature controls e.g. a pressure controller, LT switch or the like, it could happen that the controller pushes the cooled water well below zero, i.e. freezing, even though the mixture temperature might still be well above zero. Neither should it be used for cooling viscous fluids as the viscosity difference between the cooled and uncooled fluid causes severe maldistribution of the fluid.

It should therefore not be used as a method for capacity control, but only where all circuits are in operation simultaneously. One example is a thermosiphon evaporator, where the refrigerant is on the one section side and cools two different liquid circuits. Even if one circuits closes, the remaining will operate without trouble.

3.3.4. Normal, two-pass.

This is a very general design; figure 10, which can be used for both applications in § 3.3.1. Here too, one side has connections at both the front and the back end plates. By leaving at least one plate without portholes on this side, two circuits are created. The other side is connected in two pass in such a way that one pass corresponds to one circuit on the other side.

- Maximal two independent circuits can be installed.
- The two circuits/passes do not need to contain an equal number of channels.
- Each circuit has to be in one-pass flow and with one inlet and one exit.
- The sides are independent of each other.

The thermal properties in an application as in a) are straightforward. It is two units, where the exit of one fluid in the first unit directly enters the second. It has therefore to be designed as two separate (but identical models) heat exchangers.

The application in § 3.3.1.b) needs some clarification. It is mainly used in evaporators for capacity control by closing one or the other refrigerant circuit. The water side is in two pass and the refrigerant circuits each in one pass.

This arrangement means that the water is counter current in one circuit and cocurrent in the other and the capacities are then not equal, see figure 10. This is not necessarily a bad thing as it gives three capacity levels (Side 1 open, Side 2 open & both open) instead of only two.

All the water always passes an active circuit, regardless of which is closed down, as opposed to the previous design. This is important to decrease the freezing danger.

The design works well but it has one very important disadvantage; the pressure drop on the water side tends to be very high, thus only a few BPHE models can be used.

3.3.5. Dual circuit BPHE.

This design, figure 11, eliminates the high pressure drop or the freezing problems found in the previous designs.

- Channel # 1, 3, 5, 7,9,11 are water channels.
  Channels # 2, 6, 10, ...are refrigerant circuit 1.
  Channels # 4, 8, 12, ...are refrigerant circuit 2.
- The first and the last channels are water circuits.
- The flow is diagonal, both in relation to the water circuit and to the other refrigerant circuit.
- Multipass is possible on the water side only.
- The two refrigerant circuits have identical properties.
- It can be used as an evaporator or a total condenser but neither as partial condenser nor as a desuperheater.
- The water is always in contact with an active circuit.

3.3.6. Mixed design.

The channel arrangement of this design is the same as in **§3.3.3. Normal, one-pass**, but the two sibling circuits are now very unequal, see figure 8C. One, usually the larger, is used for water to be cooled and in the other the condensate is cooled.

All the evaporating refrigerant channels are in parallel as in a normal evaporator.

In the section furthermost from the refrigerant inlet, condensate is cooled. The temperature difference here is usually very large and despite the low heat transfer coefficient, the superheat can be large. These furthermost channels often receive more than their share of refrigerant and the high temperature difference helps to evaporate it.

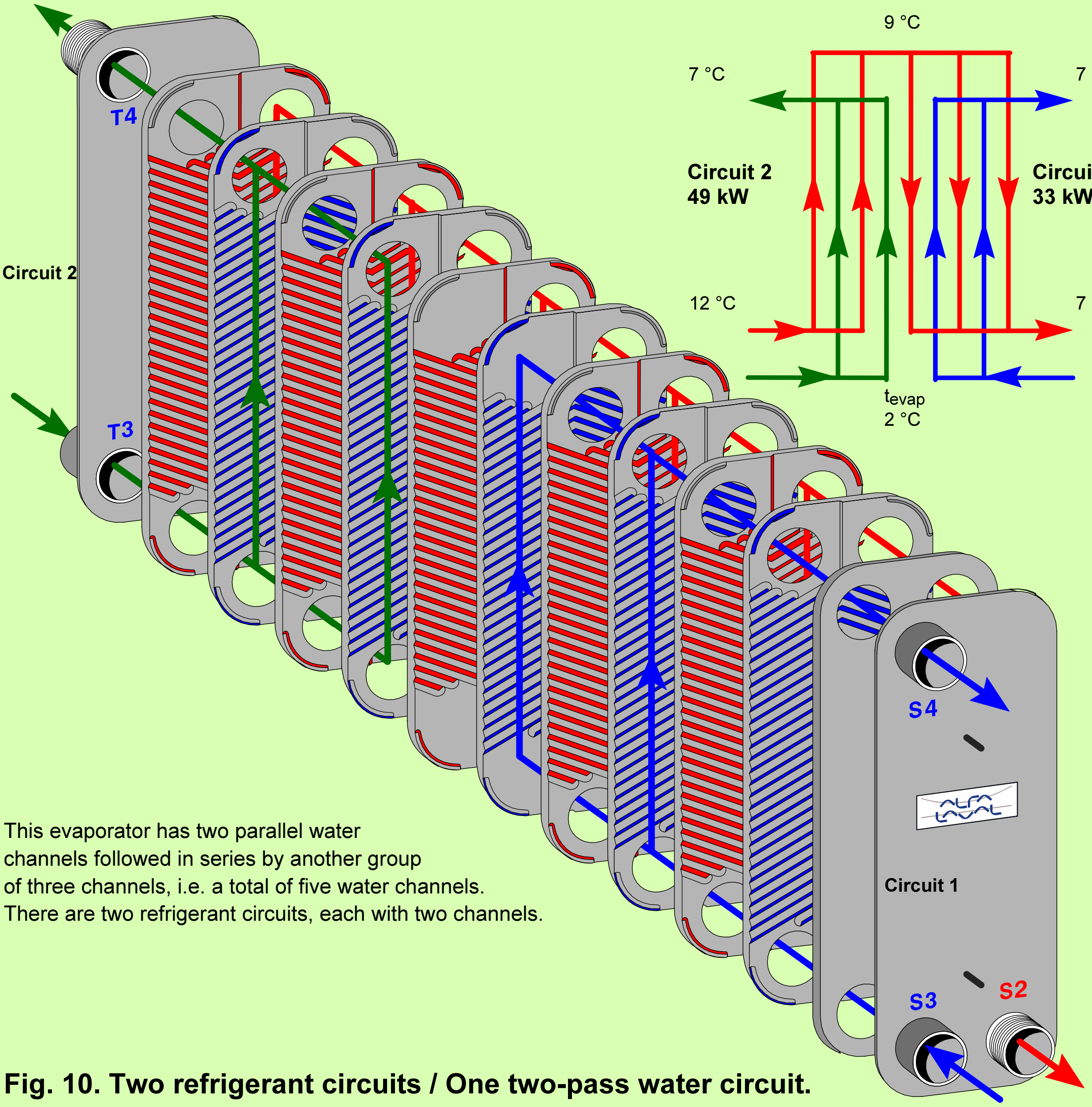
After the subcooling, the refrigerant expands and enters the evaporation side, where the water is cooled in the section close to the inlet.

The vapour from the two sections has to be thoroughly mixed before the superheat is measured. One example on how this can be done is shown in figure 21, **4. Evaporators and separators**.

The superheat in the water evaporation circuit can then be proportionally less than the controlling superheat as superheat in the other section is higher.A low superheat means a high heat transfer coefficient, which is an advantage as the temperature difference usually is low here.

Thermodynamically nothing is changed from a normal evaporator without subcooler. The advantage of the arrangement is to use a high temperature difference where the heat transfer is low and vice versa. See also figure 05, **1. Applications,** regarding subcooling.

The effectiveness of the arrangement depends on the difference in condensing and evaporating temperatures. If these are far apart, the arrangement can be ustfied, if close it is more questionable.



This evaporator has two parallel water channels followed in series by another group of three channels, i.e. a total of five water channels. There are two refrigerant circuits, each with two channels.

Fig. 10. Two refrigerant circuits / One two-pass water circuit.

Evaporators.

The figure shows an evaporator with two independent refrigerant circuits and one water circuit. The water circuit is arranged with two groups in series. Note the difference in performance from the arrangement in figure 11.

The water, here in two-pass, is connected to circuit 1 in counter current flow and to circuit 2 in cocurrent flow.

The duty is here asymmetrically divided in 33 & 49 kW. The difference from figure 11 is that the total water flow passes through each circuit and there is no mixing of cooled and uncooled water when one circuit closes. Thus no false temperature reading is possible.

Note that a thermometer cannot be placed in one of the extra nozzles as this would measure either an intermediate temperature or an inlet or exit temperature depending on which circuit is closed down.

At part load there is a choice of 33 kW, 9 -> 7 °C or 49 kW, 12 -> 9°C, assuming no change of the water inlet temperature and compressors are matched to the duties.

The two circuits should each have their own expansion valves and pressure controllers.

Regardless if there is one or two refrigerant circuits, the following is valid (see also fig. 8 A&B):

Direct expansion:

If the liquid flow forms an inverted U as above, the leaving superheated vapour meats the same liquid temperature, 9 °C above, in the two sections. This tends to equalize the superheat in the two sections. Thus:

Liquid in/out at T2/S2 Refrigerant in/out at S3/S4

Flooded flow:

If the liquid flow forms a U, the entering subcooled liquid meats the same liquid temperature in the two sections. This tends to equalize the nucleate boiling and the boiling starts simultaneously in the two sections. Thus:

Liquid in/out at S1/T1 Refrigerant in at S3 (T3)

Condensers.

The arrangement can be used for condensers as well, but with reversed flow directions on the refrigerant side. The water side connections can be either S1 & T1 or S2 & T2, depending on which gives the best temperature difference. It is not very common, though.

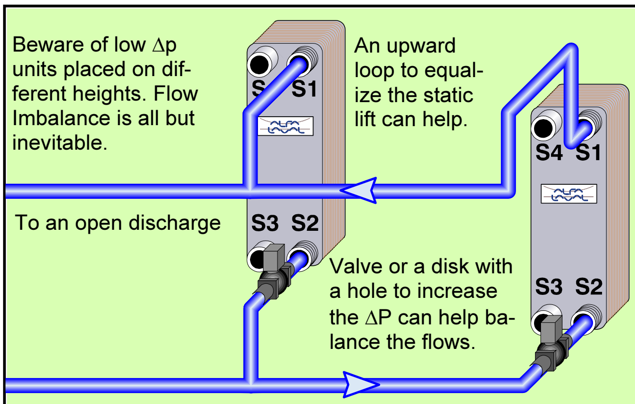


Fig. 11 A. Balancing of parallel BPHEs.

A simple manual valve or an upward loop can help distribution of the water flow between parallel units of different Δps or mounted on different heights. Note! In a closed circuit the Δp but not the height difference is important for the water distribution.

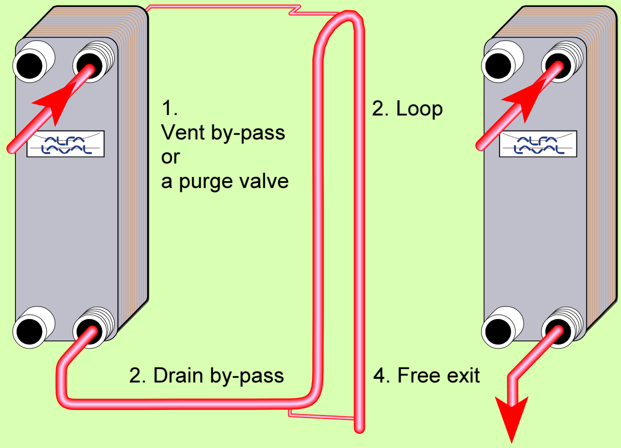


Fig. 11 B. Upward loop with by-pass.

1) A small pipe as by-pass vents the BPHE.
2) The loop keeps the BPHE constantly filled with water.
3) A small pipe as by-pass drains the BPHE at shut off.
4) The BPHE drains automatically at a shut-off.

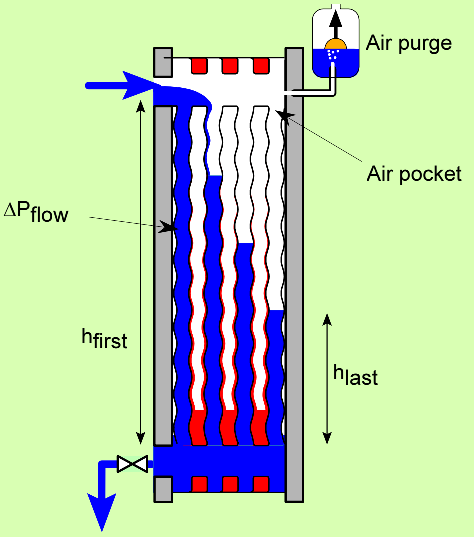


Fig. 11 C. Air pockets.

If the static pressure recovery due to the height difference in a channel is larger than the dynamic ΔP, there is a risk of formation of a stable air pocket, i.e.:

$$h_{first} - \Delta P_{flow} = h_{last}$$

In other words, the resistance to the water flow is so small that the water immediately enters and leaves from the first channels and doesn't fill the last channels. This usually happens only for very low pressure drops.

Increasing the pressure dop would decrease h_{last} to a point lower than the channel exit, i.e. the air starts to leave the condenser

The air pocket can be vented by the air purge. When the air has left and water enters the purge, the float rises and closes the vent. To facilitate the filling, there should be a restriction at the exit, symbolized by the valve.

If the water is self draining, i.e. a free exit to the outside, the valve opens when the water leaves the purge, air enters and the draining is facilitated.

4. Water balancing, draining & purging.

Two or more identical units installed in parallel should theoretically receive the same amount of cooling water. However, especially if the pressure drop is small, there will probably be an imbalance between the units. The units foul differently, there might be extra bends and fittings, an imbalance on the hot side could cause imbalance on the cold side and vice versa.

The imbalance is accentuated if the units are placed on different levels, are different sizes or types or connected by pipes with different length or diameter.

Balancing a system of heat exchanger fed from a common water source is a general problem valid for all heat exchangers and is the subject of an entire engineering discipline and is not treated here. However, installation of a valve before each unit or sometimes just a disk with a hole in to increase the pressure drop, could improve the distribution between parallel-connected units with too low pressure drops. See figures 11 A & B2.

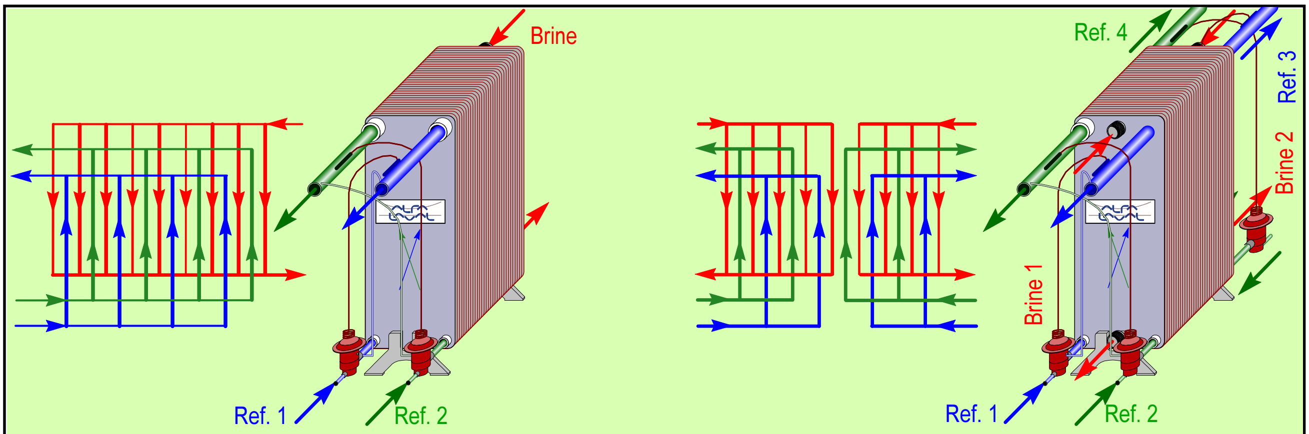
Any heat exchanger should be vented during startup and drained during a prolonged shut off.

If the water does not fill the channels as described in figure 11 C, a loop as in 11 B2 might help. The BPHE is then not self-draining at off periods, but remains water filled. A small by pass line, from the exit connection to the pipe beyond the loop, allows for an automatic drain. There will be a continuous by-pass but that can be kept small.

There might be an air cushion at the inlet port, which needs to be vented. Usually air exits with the water, but in the event of a very low flow rate, it might collect in the upper port. From this air can be vented either by:

- A purge valve at T1 (figure 11 C). This needs a certain pressure difference to the atmosphere to operate.
- A small by-pass line from T1 to the top of the loop. This does not actually vent the water side, it just moves the air further away, but the advantage is that it operates independently of the pressure difference to the atmosphere. The air can then be vented at a more suitable location or simply leave the system with the water.

When the PHE has been vented, water bypasses the channels and enters the loop directly but this by-pass can be kept small. See figure 11 B1.

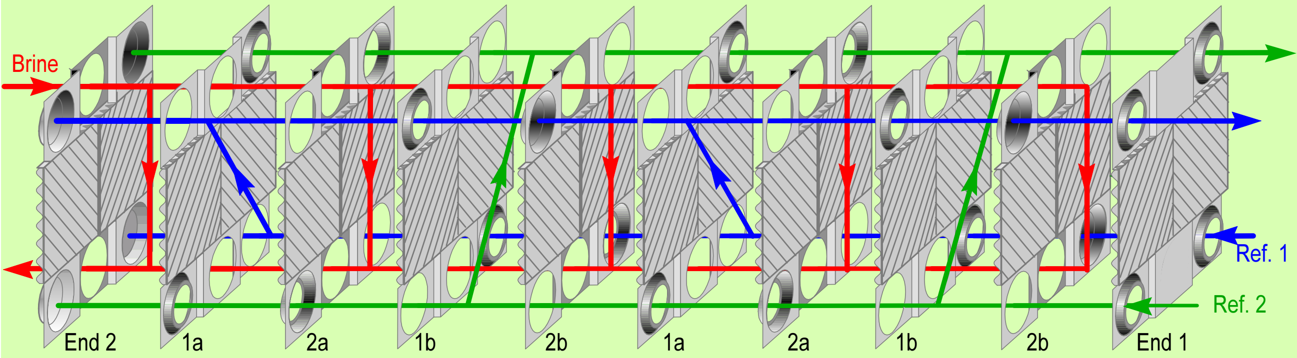


A. The basic dual circuit unit.

All the brine channels, except the first and the last, are always in contact with an active refrigerant circuit.

B. Two water and four refrigerant circuits.

All channels of either of the two the brine cicuits are always in contact with an active refrigerant circuit.



C. The plate arrangement.

There are two basic plates (corrugation angle up, 1, or down, 2), each with holes cut in two ways, a & b, totally four different plates. This creates a plate arrangement with diagonal flow of the refrigerant. This is important, as a double-circuit unit tends to be comparatively wide. Diagonal flow gives here a closer temperature approach than parallel flow. This effect is important in short and wide units, not so important in tall and slender.

To understand how the channels are arranged, follow the entrance of Ref. 2

The first plate pair the refrigerant encounter is 2b/End 1. A plate pair 2/1 is closed to the refrigerant in all four ports but open for the brine. The refrigerant thus passes 2b/End 1.

Plate 2 & 3 are type 1b/2b. A plate pair 1/2 as two diagonally placed ports entirely cut out and two semi-cuts. A & b are cut-out on different diagonals.

The refrigerant enters the pair 1b/2b, through two entirely cut-out holes and finds the entrance to the channel free and enters the channel.

It passes a pair 2a/1b, which has the channel entrance closed regardless of the whole cutting. This is thus a brine channel.

The next pair is 1a/2a. These have the holes semi-cut but the other diagonal compared to b. The refrigerant can pass the hole but the channel entrance is closed. It is thus a Ref. 2 channel.

The remaining channels/plates are similarly arranged. The other refrigerant circuit has alternatively cut holes, permitting it to enter every other second refrigerant channel.

Regardless of the hole cutting, the brine channels are always closed at the refrigerant ports and vice versa. There is thus no danger of mixing the refrigerants and the brine.

Fig. 12. Dual circuit BPHEs.

5. Instruments.

Thermometers. These can be placed in the extra connections on the water side. This has a drawback. The temperature of the water emerging from different channels can vary a little. Depending on the depth of the probe, the measured temperature varies somewhat.

This location is acceptable for control purposes, where the error in the reading is taken into account when calibrating the system, but not for exact performance measurements of a plant. The thermometer should be located some distance from the BPHE, in insulated pipes.

Refrigerant temperatures are often measured by the corresponding pressure. Refrigerant is often subcooled or superheated, and if a performance measurement has to be made, the "true" temperatures have to be measured.

Manometers. These are sensitive to turbulence. Manometers should if possible be placed downwards inside a straight pipe, at least 15 - 20 diameters long. This can be difficult to achieve in a refrigerant circuit, and errors of both pressure and temperature (see above) could then be introduced. It is important to be aware of this when evaluating the performance of a condenser or an evaporator.

If possible, use differential manometers for pressure drop measurements as a small difference between two large absolute pressure readings could give large errors. Another possibility is to switch the placement of the manometers and take a second reading.

6. Installation.

6.1. Position.

A BPHE should generally be installed vertically. Figure 13. If a horizontal installation is necessary, there are some starting points to consider (figure 14):

- ♦ A horizontal installation should only be made after a thorough testing and evaluation, as the thermal and hydraulic properties of an inclined heat exchanger are less well documented and can vary considerably.
- ♦ Never install the unit on its back with the plates in a horizontal position (figure 14A).
- ♦ A horizontal installation as in Fig. 14B is preferable. Expect a capacity loss for an evaporator of about 25%, condenser unknown but probably less.
- ♦ A slightly inclined, 5 - 10 °, unit (figures 14C-D) reduces the losses considerably.
- ♦ A condenser ***must be*** installed with the refrigerant nozzles on the lower part of the BPHE (figure 14C); otherwise, condensate will block the heating surface. This means that the water side cannot be drained.
- ♦ An evaporator could be installed with the refrigerant nozzles on the upper part. The water side can now be drained (figure 14C, but reversed connections).

6.2. Mounting.

See figure 13. Ensure that neither vibrations nor thermal expansions in the pipes are transferred to the BPHE. This could be prevented as follows:

- ♦ Use some rubber strip or Armaflex between the support and the BPHE.
- ♦ Use vibration dampener for the compressor.
- ♦ In case of long straight pipes, use bellows or some other expansion absorbing devices in the pipes.

6.3. Insulation.

A BPHE has a small exterior surface and can sometimes be installed without insulation. Should insulation be required, 5 - 15 mm Armaflex or similar is suitable. Pre-cut Armaflex sheets are available for most BPHE models.

Units - both condensers and evaporators - installed outside in freezing ambient temperatures need insulation as well, see the chapter **7. Freeze.**

WPHE should be fitted with removable insulating sheets to allow easy dismantling for inspection and servicing of the plates and gaskets.

6.4. Connections.

Water side. These are normally threaded, figure 15 A.

- ♦ Keep the threads clean.
- ♦ Do not over tighten. (Recommended values are found in the product catalogue.
- ♦ A gasket between the pipe and nozzle seats and not at the threads should make the seal.
- ♦ Use Teflon or Molycote or similar products to protect and lubricate the threads.
- ♦ Threaded connections are acceptable if the connections are easily accessible from the outside. If the PHE is installed inside a package and maybe insulated, the PHE is better connected to the outside via brazed connection on the PHE and threaded connections on the pipes at an easily accessible place – figure 13 C - for connection to an external water circuit. A possible leakage at the threads is then easy to detect and repair.
- ♦ Beware of liquid hammering. Note that valves, which close against the flow produce less hammering than the normally used valves, which close with the flow.

Refrigerant side. These are normally made for soldering. In some cases, notably for nickel brazed units, welding might be necessary, figure 15C.

- ♦ Follow the local pressure vessel codes, safety regulations and refrigeration standards.
- ♦ Thoroughly clean and degrease the surfaces to be brazed with a solvent preferably a commercial solvent with instructions for disposal otherwise the flux does not wet the surface when brazing.
- ♦ Degreasing is normally not necessary when welding except for a highly corrosive atmosphere or medium.

  Degreasing prevents the increase of the carbon content in stainless steel. Carbon could then form carbide with the chromium in the steel during the welding, usually in a defined zone on each side of the weld. The chromium-depleted steel is then no longer a stainless steel and is susceptible to corrosion.
- ♦ To prevent oxidation and to cool the unit, blow nitrogen through the pipe to be brazed or welded.
- ♦ Keep water running through the water side. The water should be running before the brazing or the welding starts and continues until the unit can be touched.
- ♦ Water-cooling is not strictly necessary when brazing, but it cools the unit faster and cheaper than nitrogen.
- ♦ Depending on the welding method, welding could release substantially more heat than brazing. TIG welding does not normally need any more protection than letting the water run. If more heat needs dissipating, the connection base should be protected by a wet rag or by flushing the body with water.
- ♦ It is easier to centre the connecting pipe and the brazing/welding is faster if the BPHE is place horizontally.
- ♦ Use a solder of at least 45 % Ag and braze at less than 650 °C. Under no circumstance should the brazed part exceed 800 °C.
- ♦ TIG welding and a protection gas is the least heat releasing welding method and should be used if possible.

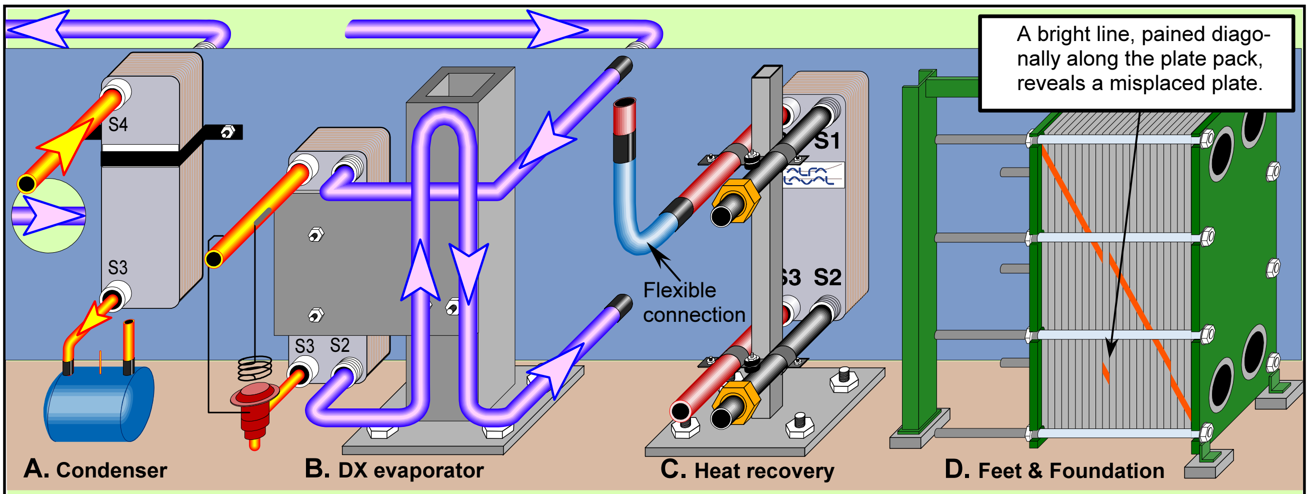
7. Thermal guarantees.

Alfa Laval can not take any responsibility for proving that a heat exchanger gives the capacity if there are no possibilities for proper measurements of temperatures, pressures, flows, etc.

Thus in case of a claim, ***the customer has to prove that it does not give the capacity and not that Alfa Laval has to prove that it gives the capacity.***

The reason for this is simple. If the proof lies on the designer of the plant, he will equip it with sufficient measurement devices, a fairly cheap operation if foreseen at the design stage.

Fitting measurement devices to an operating plant can be expensive, possible including shutdowns, emptying vessels, removing insulations, welding and subsequent controls, etc. Thus in the interest of the designer, operator and equipment vendor, a plant should be equipped with proper measurement devices at the beginning.



A bright line, pained diagonally along the plate pack, reveals a misplaced plate.

Fig. 13. Mounting of a BPHE.

A. The unit is attached with brackets (or a bar) to a wall. The water connections point outward for direct attachment of the water pipes. The condensate should have drain freely to the LR.

B. If the BPHE is equipped with pin bolts, use these for clamping it to a column, a wall, etc. The water draws the water pipe in an upward loop for secure filling. This is especially important at low pressure drops.

C. Smaller units could be supported by the connections if these are firmly attached to a column or the like.

D. The unit rests on the floor, a shelf etc. Do not place it directly on the floor as water and dirt could damage it. Use a concrete foundation or the like. See also Fig. 11, which illustrates feets on a BPHE.

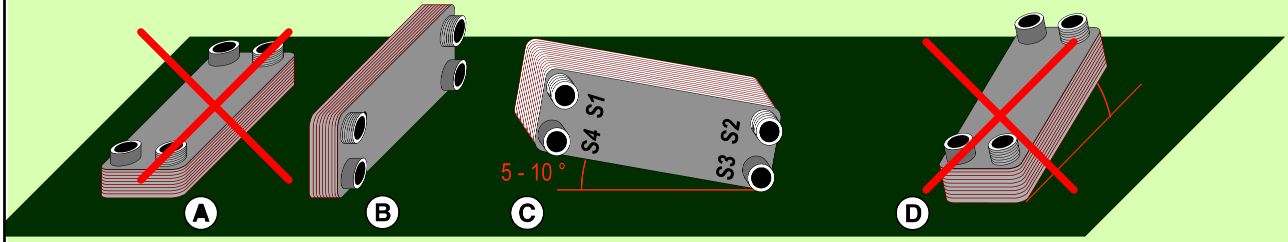


Fig. 14. Mounting positions to be avoided for a BPHE.

A. Positions with horizontal plates should never be used.

B. Horizontal with vertical plates is better. Refrigerant in the lower nozzle side. The water side cannot be drained.

- *Evaporator*: Expect a capacity (> 40 - 50 %) decrease.
- *Condenser*: No capacity decrease, but draining, important for condensers, can be difficult.

C. A slightly inclined unit with the refrigerant in the lower nozzle side gives:

- *Evaporator*: A moderate capacity decrease (15 - 25 %).
- *Condenser*: No capacity decrease; improved draining.

D. Don't incline a unit like this. Condensers but probably also evaporators give large capacity decreases.

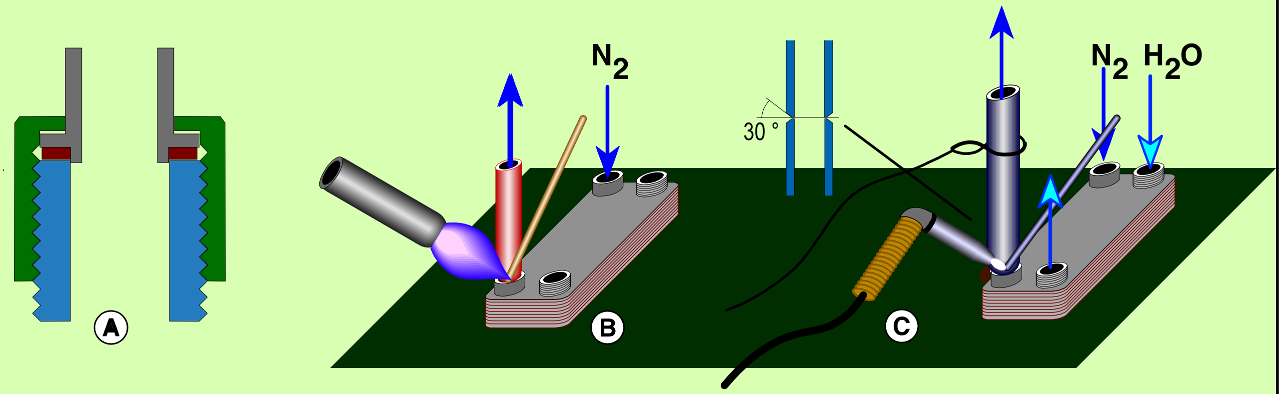


Fig. 15. Connections.

A. The tightening in a threaded connection should be made by the gasket, not by the thread.

B. Brazing should preferably be done in a horizontal position. N2 prevents corrosion and assists cooling.

C. Protect the unit not only with a nitrogen flow, but also by water running through the water side. In addition, put a wet rag around the nozzle base or flush it with water.

TIG welding with protection gas is recommended.

4. Evaporators & Separators.

1. The evaporation process.

1.1. What is boiling?

- Boiling is the change of phase from a liquid to a vapour, when heat is transferred to the liquid at a temperature at or above the boiling point at the actual pressure. There are two basic types of boiling processes - each with its own boiling mechanism:

  Pool boiling.

  Flow or convective boiling.

- The change of phase, which occurs when the pressure of a saturated liquid, thermally insulated from the surroundings, decreases, is not deemed to be boiling proper, even though it has many factors in common with the real boiling process. The vaporisation that occurs in the thermostatic expansion valve is thus not a boiling process.

1.2. Boiling types.

Pool boiling. This is the type of boiling, which occurs when a heat transfer surface is submerged in a relatively large body of quiescent liquid. The only agitation is that due to natural circulation and bubble motion. An example is water boiling in an ordinary household saucepan or a hot wire in a liquid, a set-up generally used to study pool boiling.

Figure 1 A shows how the heat flux from such a set-up varies with increasing temperature difference between the wire and the liquid. Thus, a heat exchanger should be designed close to point 4, the maximum heat flux. This point is normally called burnout. The major heat transfer mechanism here is the nucleate boiling.

Bubble formation will not start in the liquid bulk but at nucleation sites on the surface. The reason for this is that the superheat required to support an embryo bubble against the surface tension, which tries to annihilate it, is larger for a free bubble than for a bubble attached to a nucleation site. The surface tension on the bubble surface tries to increase the pressure in the bubble, i.e. the bubble condenses and collapses.

A nucleation site may be a high-energy molecular group, a cavity formed by a foreign substance, dirt, local stresses & strains, etc. Probably the best nucleation site is a cone-shaped hole on the surface.

The number of nucleation sites as well as the bubble rate from a given site is a strong function of surface properties, heat flux, pressure and liquid physical properties. In general a rough surface - i.e. corroded iron - gives much higher heat transfer coefficients than a smooth surface - i.e. glass or the extreme, Teflon. Tests have been made with artificially prepared surfaces in order to increase the number of nucleation sites and thus the heat transfer coefficients. Sand blasting of stainless steel plates is one method. Another is to spray molten copper or other metals on the stainless steel surface.

The effect can initially be large, but with time, there is usually a decrease of the performance, probably depending on fouling which inactivates the nucleation sites. Variation over time of an evaporator’s performance could sometimes be explained by changes to the surface. A thin oil film can smoothen the surface and thus decrease the heat transfer coefficients (and at the same time increase the fouling).

Deposits resulting from mechanical wear and tear in the system could increase the number of nucleation sites and thus improve the heat transfer coefficients.

In a refrigerant that dissolves oil, the effect of the oil could be an increase of the heat transfer coefficients. Up to a concentration of about 3 - 5 % oil, the heat transfer coefficient increases with increasing oil concentration, and then it decreases again. The effect could be explained with lowered surface tension of the mixture, which makes more nucleation sites active. For higher concentration the negative effect of the oil - higher viscosity- prevails.

Unfortunately, it is very difficult to predict nucleation heat transfer coefficients. The best correlations can give errors of several hundred percent. It is also difficult to determine exactly the surface conditions. Minor variations in the manufacturing process of the material used in the heating surface can give slightly different surface structures, different degrees of oxidisation; storage under different conditions can influence the corrosion, etc.

Fortunately - from the design engineer's point of view -this boiling mechanism normally plays a minor part in industrial, particularly in refrigeration evaporators.

Flow boiling. When a two-phase vapour-liquid mixture flows in a channel, there may be a number of different flow regimes. The number and name of them vary according to the investigator. Figure 2 B shows some models (with increasing vapour fraction) for flow in straight channels. The nucleate boiling coefficient normally plays a minor part in the total heat transfer, but is important to start the evaporation in a thermosiphon evaporator.

The major part of the heat transfer mechanism is two-phase convective. This is of the same nature as the normal turbulent heat exchange between a liquid and a wall, except that the Reynolds number is much larger, i.e. cold particles of liquid are heated at the wall and then transferred to the colder liquid where the heat is released.

This type of heat transfer is not dependent on the condition of the heat exchanger surface and is thus more easily correlated than the nucleate boiling coefficients. Fortunately, this is the type of heat transfer mechanism usually found in refrigeration evaporators.

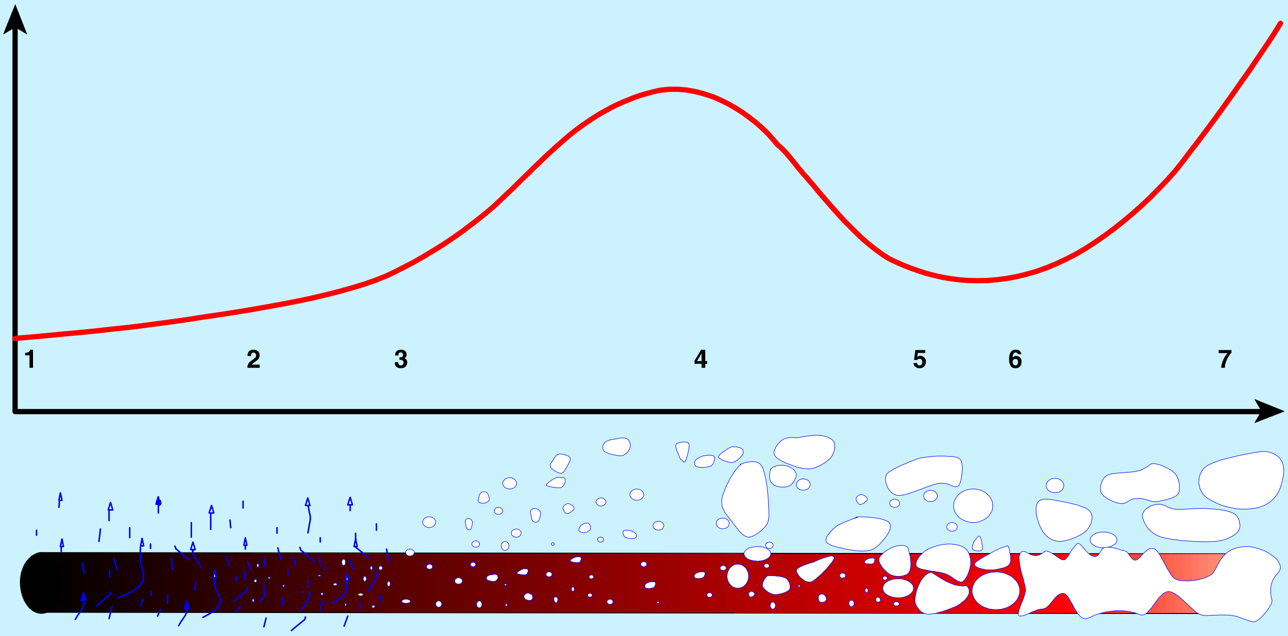


Fig. 01 A. Pool boiling heat transfer at a heated wire.

1 - 2. *Natural convection.* No bubbles are formed here.

2 - 3. *Subcooled boiling.* Bubbles are formed at the wire surface, but as soon as they enter the colder liquid bulk, they collapse. The heat transfer coefficient increases rapidly here as the nucleate boiling is added to the relatively low heat transfer due to natural convection.

3- 4. *Saturated nucleate boiling.* There is a net vapour production from the wire. The heat transfer has its maximum (burn-out) here.

4 - 5. *Partial film boiling.* The heat flux is large enough that part of the surface is blanketed by vapour. This however, means a decreasing heat transfer due to the insulating properties of the gas.

5 - 6. Stable film boiling. All the surface is blanketed by a vapour film. The heat transfer has a minimum here.

6 - 7. Stable film boiling with radiation. The temperature difference is large enough for thermal radiation to be an increasingly important part of the total heat flux.

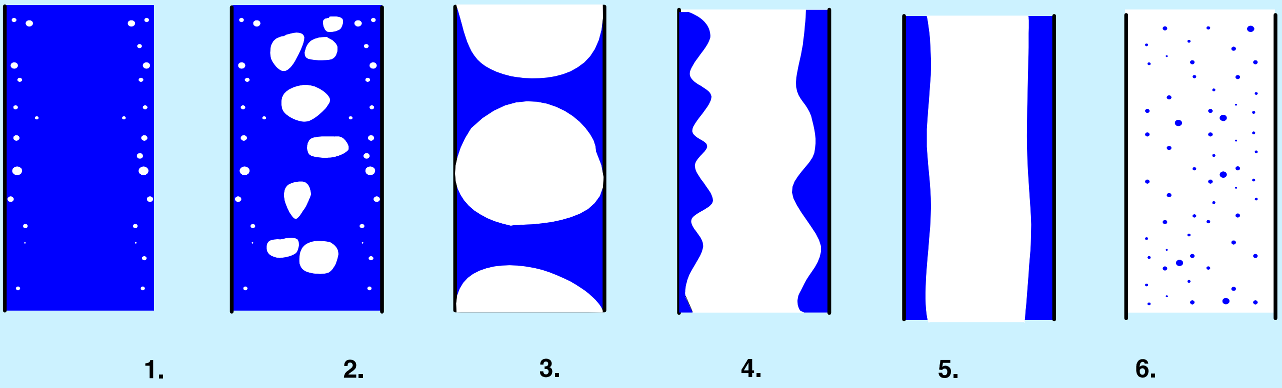


Fig. 01 B. Flow regimes for vertical, upward two-phase flow in a circular channel.

1. Subcooled boiling. The bubbles collapse when they enter the liquid bulk.

2. Bubble flow. A net vapour production is obtained.

3. Slug flow. The bubbles have merged into larger vapour slugs, which fill the channel cross section.

4. Churn flow. The liquid now flows along the walls, but the vapour has difficulty in carrying the liquid upward. The flow is unstable.

5. Annular flow. The vapour velocity is sufficient large to carry the liquid annulus upward. The heat transfer coefficient is here very large and the flow is stable.

6. Mist flow. The flow is droplets in a gas, not bubbles in a liquid, with the low heat transfer of a gas.

1.3. Flow in corrugated plate channels.

The flow in plate channels has been studied less than in straight pipes. Tests made by Alfa Laval have shown that the flow - not surprisingly - is a more homogenous mixture of vapour and liquid than what figure 1 B indicates.

The channel corrugation forces a violent mixture of the flow from the two channel halves. Half a channel can be said to belong to one plate, the other half to the adjacent.

A fluid particle, flowing along a groove in a channel, cannot continue for a very long time before it is forced to move over to the other channel half. Here it changes direction and moves along for a while before it has to move back again and thus continues in a sort of helix like flow. There is a very efficient mixing of the flow.

2. Evaporator classifications.

2.1. Pool boiling units.

The heat transfer area - i.e. a number of channels, with the heating media flowing on the inside - is submerged in the liquid to be evaporated. The liquid is quiescent. The heat transfer is mainly nucleate boiling. The vapour separates from the liquid above the channels in an enlarged shell. Sometimes a vapour separator is fitted immediately above the evaporator. This type is mainly used in large industrial refrigeration plants. The advantage is stable operation for varying operating conditions.

As PHE are not used as pool boiling units, this type is not discussed here.

2.2. Flow boiling units.

- ***Once-through evaporation.*** The liquid to be vaporised enters the heat exchanger and is totally or partially vaporised. After the evaporator, liquid and vapour separate and continue to further processing.

  This evaporator type has a short residence time of the fluid in the evaporator. This means rapid responses to changes in the operating conditions.

- ***Circulating units.*** After vaporisation, the vapour and liquid are separated and the liquid - mixed with newly entering liquid - is circulated back to the inlet to be evaporated again. The circulation can be by a pump, through natural convection - the thermosiphon principle - or by an ejector. The response time is slower than for a once-through evaporator.

- ***Intermediate types.*** In a falling film, evaporator there is a constant flow of the fluid, but the heat transfer mechanism is mainly nucleate boiling. This type is not used much in refrigeration plants but has been used in the technology related Organic Rankine Cycle. The advantage is the ability to evaporate at an almost constant temperature, making it suitable for operating at very small temperature differences.

It can be mentioned that the refrigeration terminology is not very consequent. In general, industrial terminology the word evaporator is used, when the purpose is to separate a solute from a solvent by means of evaporation of the solvent. An example is sugar evaporators, found in sugar refineries.

The word vaporiser is used, when the purpose is to vaporise a pure liquid, i.e. a condensed gas. Thus, the term vaporiser covers the function of a refrigeration evaporator better than the term evaporator.

2.3. Refrigeration evaporators.

The refrigerant leaves the condenser as a slightly subcooled liquid at a fairly high temperature and pressure. Before it enters the evaporator it has to be brought down to the evaporating pressure and temperature, by expanding the refrigerant, whereby a part of the refrigerant vaporises. The energy released by the cooling is absorbed by the evaporating refrigerant.

Refrigerant evaporators are classified according to how this expansion is arranged:

- ***The circulation evaporator*** can be a thermosiphon, pump or ejector; figures 02 B - D.

  The two-phase mixture, that leaves the expansion valve, separates into vapour and liquid in a separator. The liquid mixes with circulating liquid from the evaporator and once more enters the evaporator. The vapour mixes with the vapour from the evaporator and leaves for the compressor.

  This type of evaporator always operates with much less than 100 % evaporation. The heating surface is thus always wetted by the refrigerant. The heat transfer coefficient is high, thus requiring only a small heat transfer area, but a separator is necessary.

  The flooded evaporator needs a special oil recovery system as the liquid containing oil enters the evaporator, but only vapour leaves the system.

  In case of ammonia - insoluble oil, heavier than the ammonia - the oil is drained from the bottom of the separator-evaporator loop (**1. Applications**, figure 10). In case of oil soluble in the refrigerant, an oil evaporator is necessary (**1. Applications** Figure 19).

- ***The direct expansion evaporator****.* The two-phase mixture, that leaves the expansion valve, directly enters the evaporator. There it evaporates completely and the vapour leaves slightly superheated. The superheat is used to control the expansion valve; figure 2 A.

  At the end of the evaporation when the vapour content approaches 100 %, the heat transfer mode is gas-phase convective, i.e. low. The relative area is thus larger than in the circulation evaporator, but a special separator is not necessary.

  The oil follows the refrigerant back to the compressor.

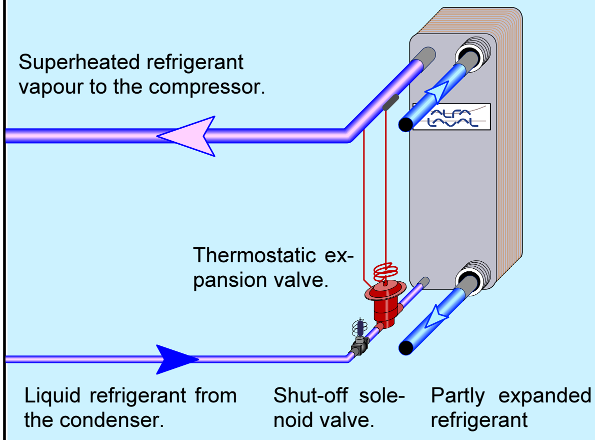


Fig. 02A. Direct expansion evaporator.

The refrigerant enters the thermostatic expansion valve, where it partly vaporizes and thus cools down. The valve is controlled by the superheat at the evaporator exit. The two-phase refrigerant mixture then enters the evaporator, where the refrigerant evaporates completely and then superheats.

The oil is left as a liquid phase, usually a mist or fine droplets which follows the vapour to the compressor.

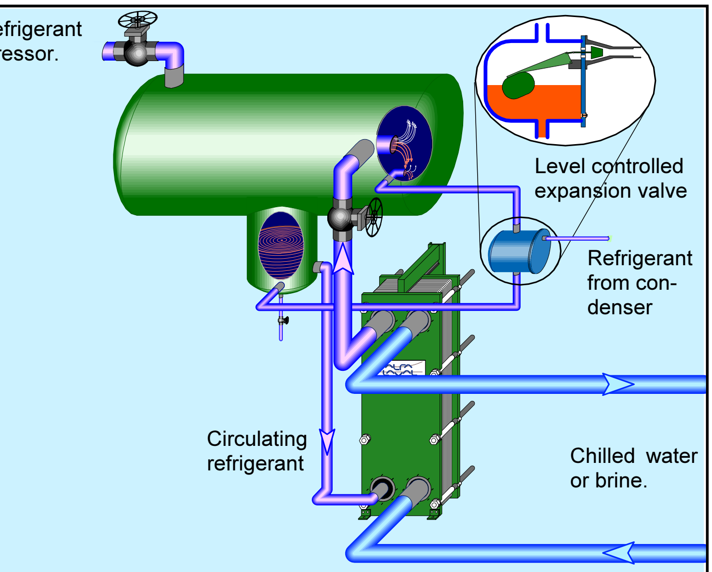
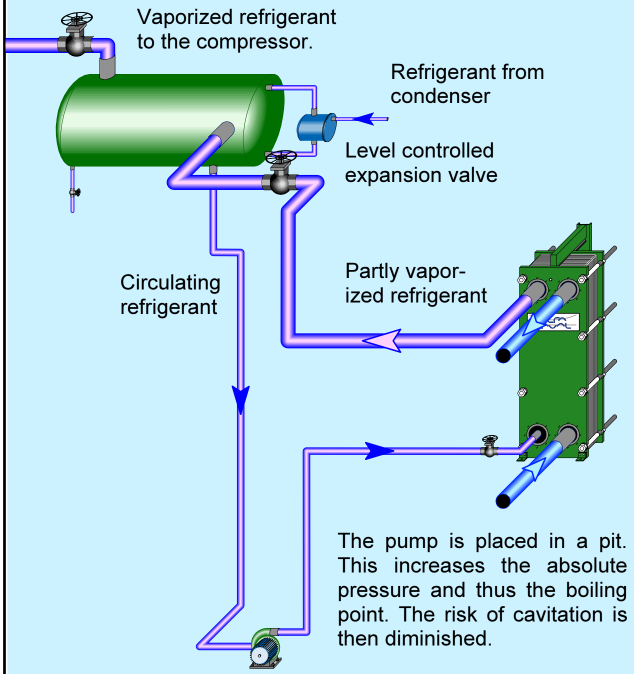


Fig. 02B. Thermosiphon evaporator.

The liquid refrigerant vaporizes in the expansion valve before entering the vapour-liquid separator. Liquid refrigerant flows from the separator to the evaporator and partially vaporises. The liquid-vapour mixture separates in the separator. The vapour then leaves for the compressor, while the liquid circulates back to the evaporator.

The system above is valid for both ammonia and freons, only the oil recovery system (not shown) differs.



The pump is placed in a pit. This increases the absolute pressure and thus the boiling point. The risk of cavitation is then diminished.

Fig. 02C. Pump circulation evaporator.

This is basically the same design as the thermosiphon evaporator, but the circulation is here maintained by a pump.

This arrangement means a greater freedom for the separator location. The vertical distance can be increased and valves & bends can be added.

It is also suitable if sudden a surge in capacity is expected as it back flow is impossible.

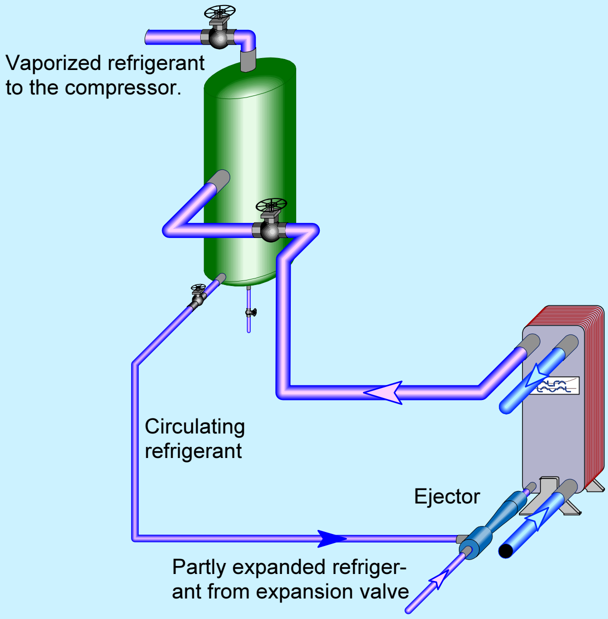


Fig. 02D. Injection circulation evaporator.

This is a compound type between the thermosiphon and the DX evaporator. The expanding refrigerant acts as an ejector pump, which helps maintain the circulation.

The system is tricky to get to operate correctly. The ejector has to be correctly designed and should operate continuously, otherwise there is a danger of reverse flow. It also needs some type of vapour-liquid distributor at the inlet.

It should only be used with soluble oil. Insoluble oil atomizes in the ejector and a part enters the channels as it has not time to settle. In the channels, it contaminates the heating surface and thus decreases the heat transfer.

3. The flooded flow or circulation evaporator.

3.1 General principles.

Figure 03 shows the general principle of a circulation evaporator. The evaporator is connected to a vapour-liquid separator by two pipes, a lower leg, which feeds the evaporator with liquid, and an upper leg, which returns the partially evaporated liquid.

The separator always maintains a liquid level; usually well above the top of the evaporator. The evaporator is thus always filled with liquid and the type is usually called the flooded flow evaporator.

Depending on the driving force for the circulation, flooded flow evaporators are classified as ***thermosiphon evaporators***, where the driving force is the natural density differences between the two legs of the system separator-evaporator, or ***forced flow evaporators***, where the driving force is a pump or an ejector.

The circulation - the ratio of the total refrigerant amount entering the evaporator and the amount evaporated - can vary from about 5 to 10 for a S&THE to about 1.2 in a PHE, of either the brazed or semiwelded type. A smaller circulation rate eans smaller pipe work & separator and a reduction of the total refrigerant content of the plant.

The heating surface is thus always wetted by the refrigerant. This is important, as the heat transfer is then two-phase convective of a vapour in a liquid, i.e. high as compared with the direct expansion evaporator, where the heat transfer mode, at the end of the evaporation and the superheating of the vapour, is gas heat transfer, i.e. low.

3.2. Applications.

The relatively small size of the flooded flow evaporator compared to the direct expansion evaporator favours this type at large capacities. Here the price of the evaporator is more important than the extra cost of a separator.

The smaller BPHEs are thus normally not used as flooded flow evaporators. They are sometimes used as auxiliary coolers of oil, water or refrigerant in large plants. Larger BPHEs can be used as flooded flow evaporators for non-ammonia refrigerants.

Ammonia, for reasons later described, favours the use of flooded flow evaporators and the recent development of nickel-brazed PHEs could increase their use here.

3.3. Thermosiphons.

This is the most economical solution for a flooded flow evaporator as the cost for a pump and its operation is saved. The circulation rate is, however, very dependent on the heat transfer and pressure drops in the various parts of the system, which in their turn are dependent on the circulation, i.e. there is an interaction and interdependence between pressure drop, circulation rate and heat transfer in the circulating system.

By definition thermosiphon means circulation owing to density differences between the fluids in two connecting legs, one hot and one cold. Assume that the unit in figure 03 B is not operating but filled with liquid refrigerant. Both the valves are open. The refrigerant level in the separator is the same as in the evaporator.

When liquid enters the evaporator on the other side, the refrigerant heats up, slowly bubbles start to form and the channels are partly filled with ascending bubbles. Thus, the mean density in the leg formed by the evaporator is much lower than in the leg formed by the separator and the descending pipe.

The two legs are thus not in balance and the refrigerant gradually enters the evaporator from the separator via the descending pipe. At the top of the evaporator, a two-phase mixture is pushed into the separator and the liquid and vapour separate. At the bottom, the entering refrigerant heats up and finally starts to boil. Thus, a two-phase mixture is always maintained in the channels.

As the circulation rate increases, the various pressure drops increase and finally the driving force is balanced by the retarding forces. See figure 03. The system is now in balance and a constant refrigerant flow enters the evaporator while a certain fraction of the flow evaporates.

When the liquid-vapour mixture enters the separator, it is saturated. The liquid separates and the refrigerant enters the evaporator again but it is now not saturated. The temperature is the same as in the separator, but the pressure is higher, increased with the static head from the liquid level to the inlet, i.e. the refrigerant is subcooled.

This means that at the first part of the heat exchanger there will be no boiling, just a temperature increase. However, as the refrigerant rises, it will decrease in pressure, reducing the subcooling.

These two effects-increasing temperature and decreasing pressure - mean that after a while boiling point is reached and the refrigerant starts to boil, albeit at a higher temperature than at the exit. The pressure continues to decrease because of the changing height and the pressure drop, and the refrigerant, now saturated, will continue ascending with decreasing temperature until it reaches the separator and the loop is closed.

Figure 03 B shows how the temperature changes from the inlet to the exit of the evaporator. Observe that there is a small temperature drop of the refrigerant, because of the pressure drop in the exit piping. This temperature drop is not accompanied by a heat transfer as this is an adiabatic (i.e. no heat exchange with the surroundings) expansion of the refrigerant mixture. Because of this temperature drop, the temperature at the exit of the heat exchanger is slightly higher than at the inlet.

At the lowest point in an ammonia system there is an oil drain. See **8. Oils & Refrigerants** for further descriptions.

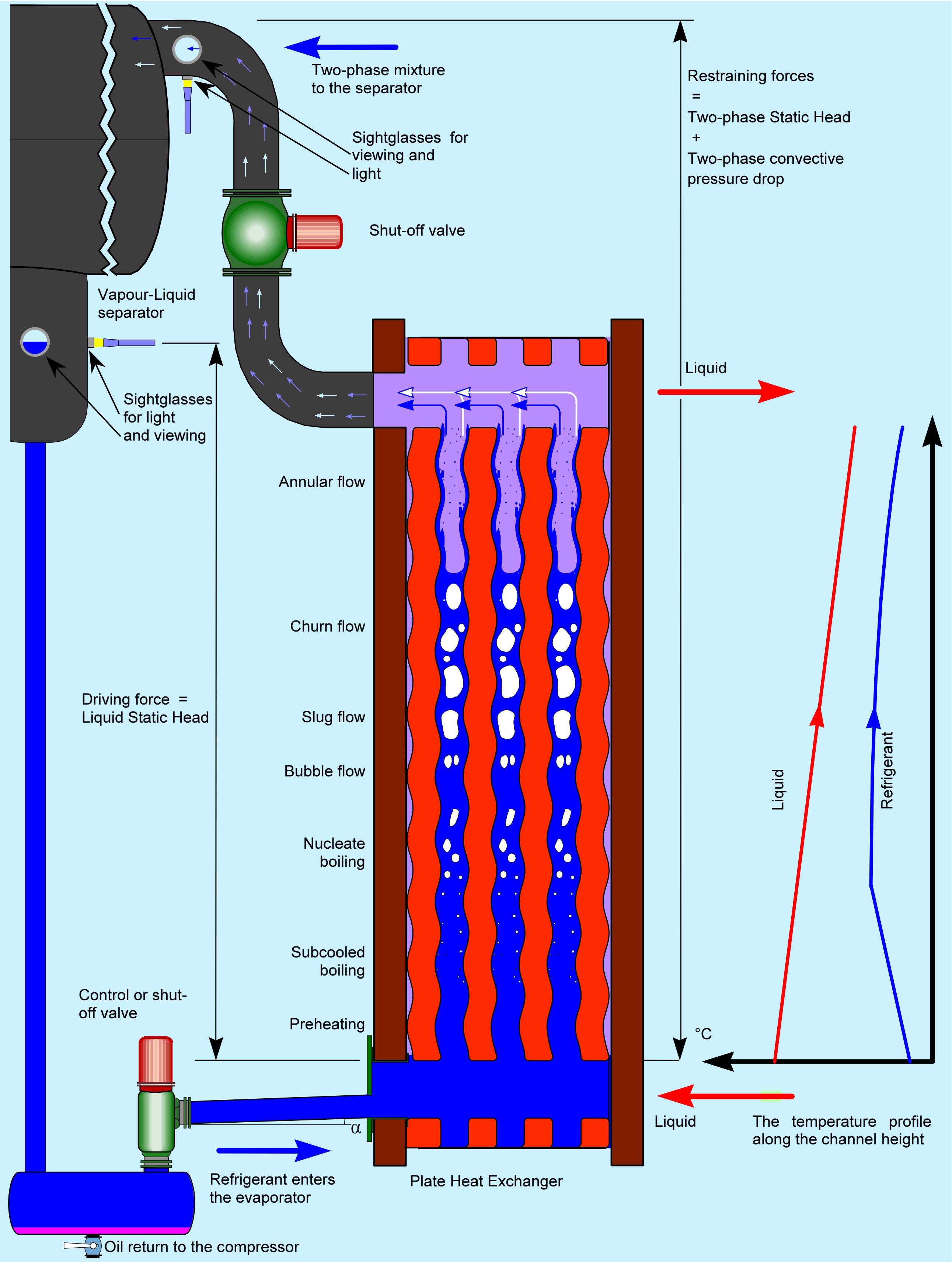


Fig. 03. The Plate Heat Exchanger as a thermosiphon evaporator.

The water enters at the bottom, i.e. in cocurrent flow. The reason for this arrangement is that the initial temperature difference, important to start the boiling, is larger than in countercurrent flow. The exit pipe shall be as short as possible and free from unnecessary bends. If a shut-off valve is necessary, this shall be of a type and size with an as low resistance as possible. The figure also shows the collection of oil in case of an insoluble oil heavier than the refrigerant. In practice this means oil/ammonia (in a semiwelded or nickel BPHE).

3.4. Design considerations for thermosiphons.

3.4.1. Pressure drop.

- Suppose that the system contains some equipment with larger than normal pressure drops in the circulation loop, either the evaporator, pipes, valves or bends. It is then tempting to increase the driving head by placing the separator well above the evaporator (Figure 02 C & 04 A2) and thus increase the driving force. However there can be some problems:

- A large static head increases the actual subcooling of the refrigerant at the evaporator inlet as compared with a "normal" installation (Figures 02 B & 04 A1).

- The preheating part will then be larger. As this has a low heat transfer coefficient, the net result might be an enlarged evaporator.

- Because of the larger subcooling, the refrigerant is heated to a higher temperature before the boiling starts, i.e. the effective temperature difference decreases; see figure 04 A2. The net result could be crossing temperatures, i.e. impossible and a non-working evaporator.

3.4.2. Evaporation temperature.

When the evaporation temperature decreases, the subcooling increases. Ex.:

- R 22 at 0 °C is subcooled by 3 K, if the pressure is 0.5 bar over the saturation temperature.

- R 22 at -30 °C is subcooled by 7 K, if the pressure is 0.5 bar over the saturation temperature.

The net result will be as § 3.4.1, i.e. a decreased heat transfer coefficient and a reduced MTD. At very low temperature R 22 should be replaced by a refrigerant with a steeper vapour-pressure curve.

3.4.3. Flow distribution and double exits.

With a large number of plates, there could be a maldistribution of refrigerants in the channels. In this case, it is recommended to use double exits on the refrigerant side. Note that the pipe arrangement from the two sides should be as symmetrical as possible. The same is valid if two evaporators are connected to one separator, see figure 05.

3.4.4. Minimum temperature difference.

In order for boiling to start, the minimum temperature difference between the wall and the entering refrigerant saturation temperature should be in the order of 2 - 3 K. This value is very dependent on the refrigerant type and the surface condition of the plate.

3.4.5. Cocurrent vs. counter current flow.

Counter current flow normally gives the largest temperature difference, thus the smallest evaporator.

However, if the temperature difference is small (< 10 °C), it might be better to use cocurrent flow, as this will give the highest wall temperature facing the entering refrigerant. A high wall temperature is important to start the boiling; see figure 02, 1&2.

If the ratio between the entrance and the exit temperature differences becomes too large though (more than double), it might be better to maintain counter current, as there might be a risk of instabilities, see § 3.5.

3.4.6. Exit vapour fraction.

A BPHE operates with considerably lower circulation values than a S&THE. Normally the vapour content at the exit is about 70 to 90 %, i.e. a circulation of 1.4 to 1.1.

3.4.7. Oil drain.

The basic principle for draining of an insoluble oil, with a density higher than the refrigerant is that:

- The oil should be allowed to settle as soon as possible after the expansion of the refrigerant before it can enter the channels.

- The drain should be at the lowest point.

See figures 6 - 8 here and **8. Oil & Refrigerants.**

Considering this, the use of an ejector to boost the circulation - see figure 2D - is questionable. The oil emerges from the expansion of the refrigerant in the ejector thoroughly dispersed in the refrigerant and then enters the channels. Once there, a large part of it remains.

3.5. Troubleshooting thermosiphons.

3.5.1. Instabilities.

The thermosiphon evaporator, being part of a loop with more or less elaborate control system, is susceptible to various instabilities. These can depend on the setting of the control system, components not being matched to each other, etc. There are however two types of instability which directly concern the circulation loop, and are thus specific to the thermosiphon evaporator.

Instability 1. Suppose that the static head is the major pressure drop in the evaporator and the two-phase friction pressure drop is negligible. This usually means a small mean vapour fraction. A slight increase in the temperature difference causes a decrease in the two-phase density and a decrease of the static head and thus an increase in the circulation. The normally restraining friction pressure drop is too small to influence the circulation rate. The increased circulation improves the heat transfer, which leads to further vaporisation, still lower static head, higher circulation etc.

Thus, a small increase of the temperature difference leads to a large increase of the circulation with accompanying control problems.

As the vapour fraction is normally large in a PHE, thus a large two-phase friction pressure drop, this type of problem is seldom found in a PHE.

- ***Instability 2***. In the second case, there is large vapour production and a large temperature difference. If there are excessive pressure drops at the evaporator exit -too small and too long pipes, too many bends, valves with high pressure drop etc. - the available driving head may not be sufficient to force the two-phase mixture through these restrictions, and choke flow is reached. The refrigerant is pushed backwards through the inlet.

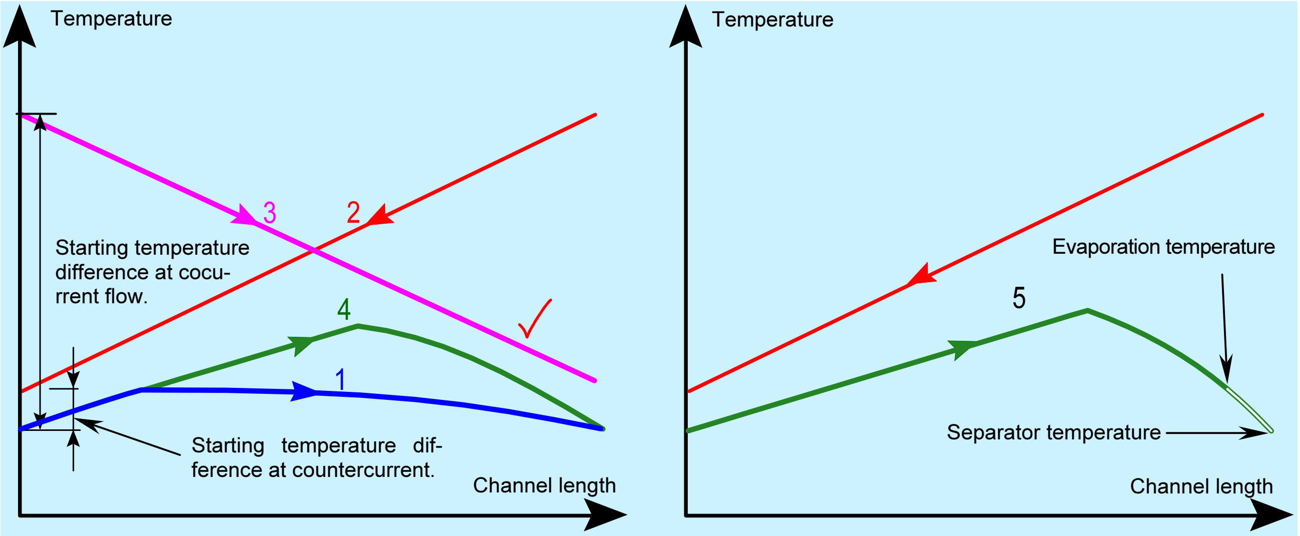


Fig. 04. The temperature program in a thermosiphon evaporator.

1. Refrigerant, normal operation.

2. Fluids in countercurrent flow. The temperature difference at the refrigerant inlet is small, thus the boiling starts late and with a large area as a consequence.

3. Heating fluid in cocurrent flow. There is a large temperature difference at the refrigerant inlet, which is important in order to start the boiling.

However, at very high inlet temperature differences, cocurrent flow could provoke instabilities, see § 3.5.1. Then use counter current flow if the resulting inlet temperature difference is sufficient to start the boiling.

4. Vacuum operation. The temperature change for a given change of the saturation pressure becomes increasingly larger the lower the pressure is. The refrigerant has to be heated appreciably before the boiling starts and the temperature begins to decrease again. This means a large preheating zone and a reduced MTD.

In cocurrent flow there could be a temperature pinch as indicated by the tick. Vacuum operation should have an as low liquid level as possible in the separator in order to keep the preheating zone small.

5. If the pressure drop in the evaporator is large, and consequently the static head has to be large, i.e. the separator has to be placed high above the evaporator exit or a pump has to be used (figure 02C).

Then, there will be an appreciable pressure drop and temperature difference between the exit of the evaporator and the separator.

The evaporation temperature - defined at the evaporator exit - is then higher than the separator temperature.

The difference becomes larger the lower the pressure is. The effect and the consequence for co- or counter current flow is similar to vacuum operation.

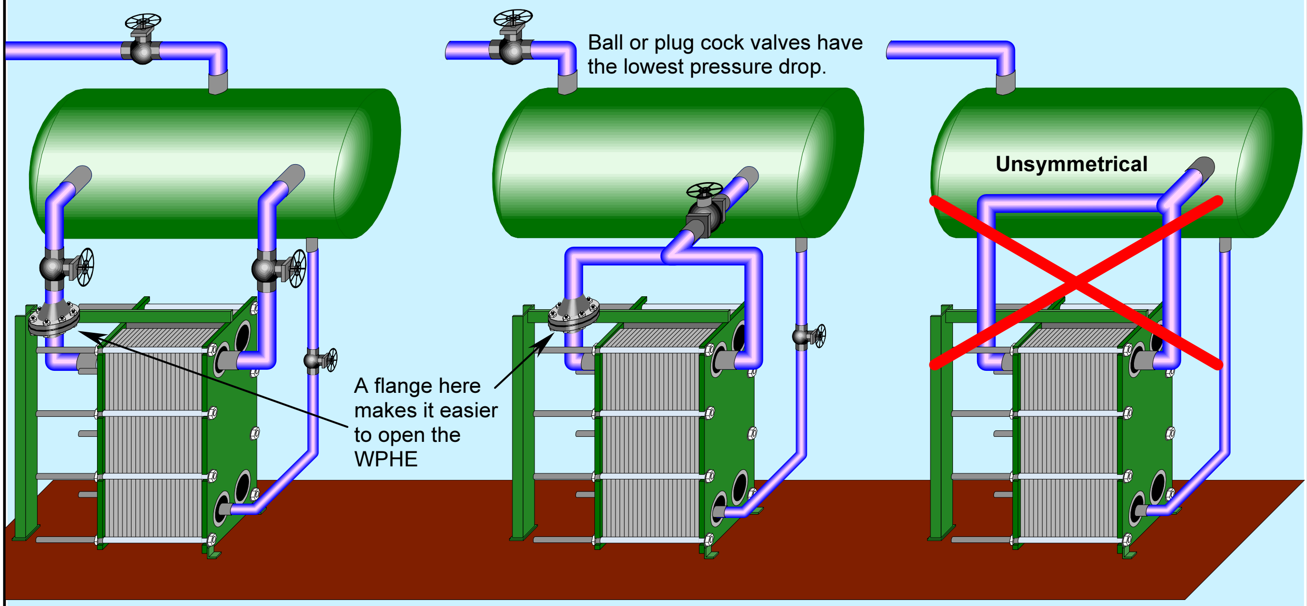


Fig. 05. Piping for a PHE with double exits.

The exit piping has a large influence on distribution of the refrigerants in the channels. In the case of two exits, which reduces the port velocity and thus improves the distribution - the exit piping should be as symmetrical as possible. The inlet piping is less critical as flow velocity is much smaller.

This causes a decrease in vaporisation and pressure drop and the two-phase mixture can leave through the exit. Refrigerant may again enter the evaporator and the cycle is repeated. This problem is more serious than the first as it is oscillatory. Apart from causing control problems, the capacity decreases, as the evaporator is ineffective part of the time. The problem can be controlled by a valve at the refrigerant inlet.

The problem is compounded if the inlet temperature difference is very large and the exit temperature difference small. This can happen in cocurrent flow (see figure 04). The vaporisation rate can then be larger at the first part of the channels than at the end and the vapour finds it easier to escape backward. In such a case, counter current could be better. Backward flow can be prevented by a control valve as above.

3.5.2. Thermal performance.

a) Check all flow rates, temperatures, pressure drops etc. Do the pressure drops hint at anything abnormal? An obstruction of the water flow, too much oil?

b) Interchange thermometers between various locations. Small temperature differences can easily be masked by incorrect thermometers.

Ex**.** The thermometer at the warm side exit has an error of +0.5 K and the one at the cold side exit (cocurrent) -0.5 K and the measured difference is 4 K. Switching thermometers gives 2 K difference. The mean, 3 K should correspond to a better value.

c) Double-check the heat transfer, by using the various combinations of temperatures and flows.

d) Inspect the evaporator for temperature variations on the outside. Large differences could hint at maldistribution on one side or the other.

e) Inspect the heating fluid. If glycol or similar, check for concentration and/or viscosity. A too high concentration will impair the heat transfer and a too low could mean ice formation.

f) Check for ice formation. Ice impairs the heat transfer and the exit temperature could actually increase.

g) Inspect the evaporator for signs of fouling on the water side & excessive oil on the refrigerant side.

h) Are the condenser & compressor capacities correct? If so, the evaporator capacity should be correct too. If not, there must be some irregularities somewhere.

i) If the refrigerant (freons only) contains water, ice could form in the expansion valve, thus obstructing the flow.

j) Check the refrigerant feed to the separator. It cannot evaporate more refrigerant than what enters, thus if this is too low, the capacity decreases. See below.

k) Are there instabilities? These decrease the capacity.

l) Inspect the refrigeration loop. Too large a difference between the liquid surface and the evaporator inlet could mean a temperature pinch (figure 04 A.2). If too low, the upper part of the evaporator could run dry.

m) Check the refrigerant exit temperature. Does it correspond to the pressure? If the vapour superheats, the heat transfer coefficient drops dramatically.

n) Does an ammonia system have efficient oil recovery? (**1. Applications, fig. 10.**) Otherwise, oil could cover the heating surface and decrease the heat transfer.

o) Does a system with soluble oil have an oil evaporator? (**1. Applications, fig. 19**). If so oil will not cover the surface but the physical properties will be impaired (higher viscosity) and the heat transfer decreases.

p) Measure the temperature (if possible the real temperature, and the corresponding pressure) at the evaporator exit, the separator exit and at the suction pressure. Too large differences indicate some flow restriction.

q) Beware of refrigerant trapped in a downward loop.

r) Is there water in an ammonia system? For a given pressure, addition of water to the ammonia increases the evaporation temperature. As the control system is mainly based on pressure, there could be a hidden decrease of the temperature difference.

s) Inspect the refrigerant loop for flow obstructions. Are there excessive numbers of valves, valves with to high ΔP, bends, etc.? Especially dangerous are obstructions at the vapour exit. How does the vapour inlet to the separator look? Obstructions?

t) A superheated vapour could hint at:

- Water in an ammonia evaporator, see r) above.
- A restriction in the refrigerant flow to the evaporator, see s) above, means that the evaporation capacity is not fully utilised and the vapour superheats.

Note that a superheated vapour means an underutilised evaporator, i.e. more refrigerant could vaporise if more enters. This is similar to a too cold condense from a condenser. In both cases, the problem is to be found outside the PHE.

u) If the operation of the evaporator-separator system is very erratic, difficult to control or shows other irregularities, check the float valve. A float valve works best if moving with a certain velocity. A slow moving valve, especially after some time operation, can get stuck. An additional problem is that the float can implode, if the outside pressure becomes larger than the pressure inside the float.

3.6. Oil separation.

- ***Soluble oil*** entering the loop gradually concentrates in the refrigerant as pure refrigerant vapour returns to the compressor from the separator. A small stream has to be withdrawn, the refrigerant evaporated and the oil returned to the suction line. See § 16 in **1. Applications**.
- ***Insoluble oil, heavier than the refrigerant,*** i.e. the oil-ammonia system. More or less pure oil collects in the lowest part of the system and can be drained from there. See figures 03 & 07 here and figure 10 and § 13 in **1. Applications**, which show an oil recovery system.
- ***Insoluble oil, lighter than the refrigerant,*** is a not very common system, which should be avoided. The varying liquid level where the oil collects, makes a oil drain difficult.
- ***Note*** that the oil has to be drawn off ***from*** the evaporator-separator loop. If oil is drawn of ***before*** the loop, the inevetable oil rests in the refrigerant will ultimately collect and concentrate in the loop. See § 16 in **1. Applications**

4. Vapour-liquid separators.

4.1. The purpose of this section.

This section describes the function of the vapour liquid separator for a flooded - either pump or natural circulation - evaporator system. It is valid for:

- Refrigerants with insoluble oil, ***heavier*** than the refrigerant, i.e. in practice the ammonia-oil system.
- All refrigerants with soluble oils.

The main stress is laid on the interaction of the separator with Plate Heat Exchangers (PHE). The description is by no mean complete. Security & control equipment, the mechanical design of the vessels, etc., are not covered. These depend on local refrigeration & pressure vessel codes, type, size & placement of the refrigeration plant, etc., and are thus beyond the scope of this manual.

In (**Appendix II. Design of vapour liquid separators.**) is a review of some design methods for separators.

4.2. The function of the separator.

The separator (or the low pressure receiver) in a flooded system has the following functions:

- To separate the vapour from the liquid.
- To accumulate the refrigerant content of the system during a shutdown.
- To even out changes of volume in the system during load variations.
- Under certain conditions, the refrigerant might foam and space has to be provided to accommodate this.
- To provide a static liquid level which then provides the driving force for the circulation or the suction head for a circulation pump.
- This liquid level is also used to control the expansion valve. This is sometimes done in the high-pressure receiver, sometimes in the low-pressure receiver.
- To act as an oil trap/separator.

4. 3. Operation.

The separation of the vapour and the liquid is obtained by gravitational forces, sometimes assisted by centrifugal forces, which allow the heavier liquid droplets to settle.

Thus, liquid droplets small enough to be kept in suspension by molecular movements (Brownian movements) do not separate. In practice, liquid droplets sometimes much larger than Brownian droplets do no separate either, but there are additional methods to separate these.

4. 4. Equipment of a separator.

A separator should be equipped with (see figures 06 - 08):

- Demister (for scroll & piston compressors).
- Sight glasses (one for viewing, one for a light) at the inlet and exit of the separator (from/to the evaporator). This is especially important for monitoring the operation of an injection recirculation evaporator.
- Connections for a thermometer and a manometer.

4.5. Separators for S&THEs and PHEs.

In a S&THE, the circulation rate is 4 to 5, corresponding to a vapour fraction of 0.25 to 0.20. Thus, in a STHE, evaporating 1000 kg/hr refrigerant, up to 4000 kg/hr liquid enters the separator together with 1000 kg/hr vapour.

A PHE in general has much lower circulation rates, about 1.25 or lower corresponding to an evaporation fraction of 0.8 or higher. In a WPHE, also evaporating 1000 kg/hr a, as little as 176 kg/hr liquid enters the separator together with the 1000 kg/hr vapour (0.85 evaporation). This does not affects the separator size as such, as the size of this mainly depends on the vapour flow, but the pipelines, notably the feed pipe to the PHE & circulation pumps can be greatly reduced.

A side effect of the low circulation rate is the low refrigerant hold-up in the PHE. This reduces the separator size as variation of the total refrigerant volume is less.

4.6. Separator types.

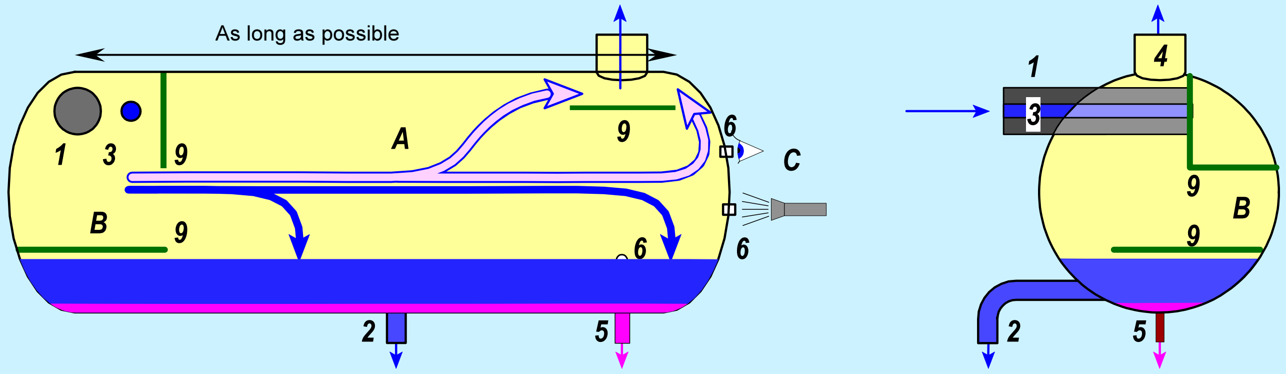
There are basically two types of separator, the horizontal and the vertical and a hybrid type.

The horizontal separator (Fig. 2C) has the properties:

- The flow is horizontal. If the residence time is sufficiently long, the droplets separate regardless the velocity.
- The hold up time and the height of the separation space, not the velocity, determine the efficiency.
- When the liquid level increases, the cross section decreases, the velocity increases and the hold up time decreases, i.e. a decreased separation.
- It is easy to connect two or more evaporators or an evaporator with double exits. Double entrances halve the velocity but also the separation distance preserving the efficiency.

The vertical separator (Fig. 2D) has the properties:

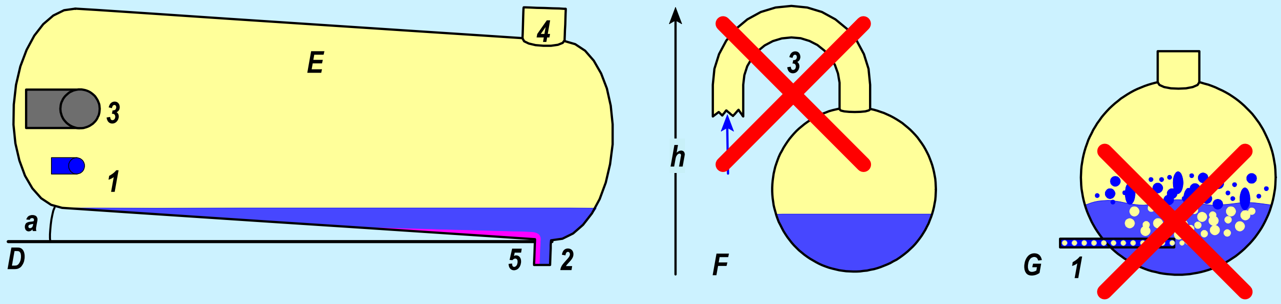
- The vapour flow is mainly upward. If the velocity is less than the separation velocity (see later) the droplets separate. Variation of the liquid content causes a correspondingly large variation of the liquid level.
- The liquid level does not affect the vapour velocity.
- The liquid body is easily agitated, providing a variable signal to the TEV and a difficult oil separation.
- It occupies little floor space, but more headspace.
- Do not confuse up the vertical separator with the cyclone. The cyclone separates particles or droplets by entering the vapour tangentially at high velocity, thereby creating a strong ***centrifugal force***, which effects the actual separation. Separation is very effective, but the pressure drop is also very high.
- It is difficult to connect two or more evaporators or an evaporator with double exits.



A. Utilize the total length as much as possible for the separation.

B. Splash protecting plates (9) help keeping the liquid body calm, especially at tangential inlets.

C. Two sight glasses or even better three (6), for a light and for viewing, provides a cheap method to check the operation of the separator.

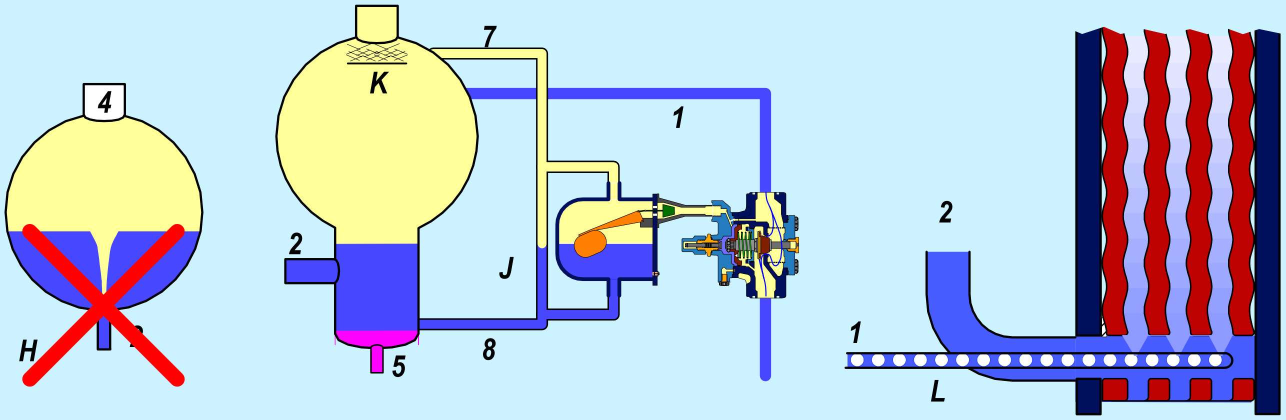


D. An inclined vessel helps drain the oil in an ammonia system. ***However, note the danger of a vortex*** at the exit.

E. The oil & evaporator return exits should be placed as far as possible from the vapour inlets. Fine oil droplets entering at 1 then have time to settle. See also H.

F. Avoid a high inlet as the static liquid column - h - and the pressure drop might be too high.

G. Avoid injection of the flashing refrigerant into the liquid. It reduces mist and fine droplets but the agitated liquid gives an unsteady signal to the level controller.



H. Avoid vertical liquid refrigerant exits as the formation of vortexes is inevitable. If vapour is drawn into the evaporator the capacity might decrease and the pump in a forced circulation system might be destroyed.

J. The upper and lower exits, 6 & 7, for the level controller (shown here as a pilot valve) should be spaced as far apart as possible. This facilitates changing of the operating liquid level should the need arise.

K. A demister catches mist and fine droplets.

L. Injection of the flashing liquid in the circulating loop is difficult. It could be made via an ejector pump (see figure 2 D) or via a distributor (a pipe with holes, one for each channel) into the PHE evaporator.

In both cases, the expansion should be continuous (as in a TEV), not on-off as is normal in a flooded system.

Fig. 06. Horizontal separator designs.

1. Liquid/vapour from the expansion valve.
2. Circulating liquid to the PHE.
3. Returning liquid/vapour from PHE.
4. To compressor.
5. Oil drain.
6. Sight glass.
7. Upper liquid level indicator pipe.
8. Lower " " " "
9. Splash protecting plates.

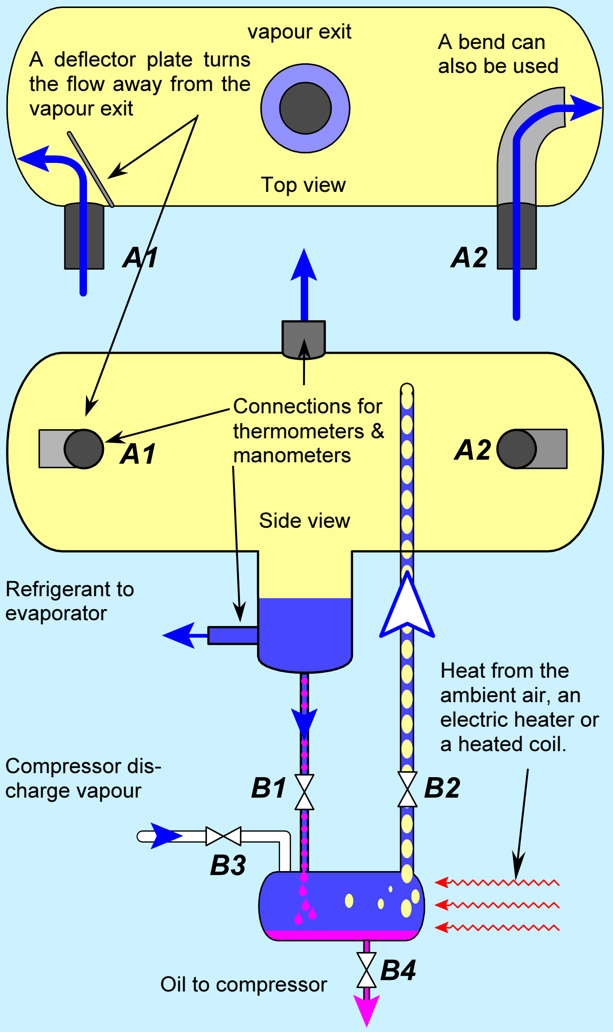


Fig. 07. The hybrid separator for ammonia with an oil recovery system.

Oil comes from the separator via the line & valve B1 and from the lowest point in the loop (see also Applications figures 03 & 10) and collects in an oil pot.

The ammonia accompanying the oil into the pot evaporates and returns to the evaporator via the line & valve B2. The heat for this comes from the surroundings, an electric heat or discharge vapour.

When the pot is full of oil, i.e. no more ammonia evaporates, the temperature increases. In case of an evaporation temperature below zero, this is clearly visible as the frost on the outside of the pot disappears.

Valve B1 is then closed for a while and the last traces of ammonia evaporate. Finally, valve B2 closes and B3 & B4 open. High pressure gas then enters through B3 and forces the oil back to the compressor oil system.

The hybrid separator (Fig. 2B) has the properties:

- It is basically a horizontal separator with a vertical vessel attached to the bottom.
- The liquid level is maintained in this vessel.

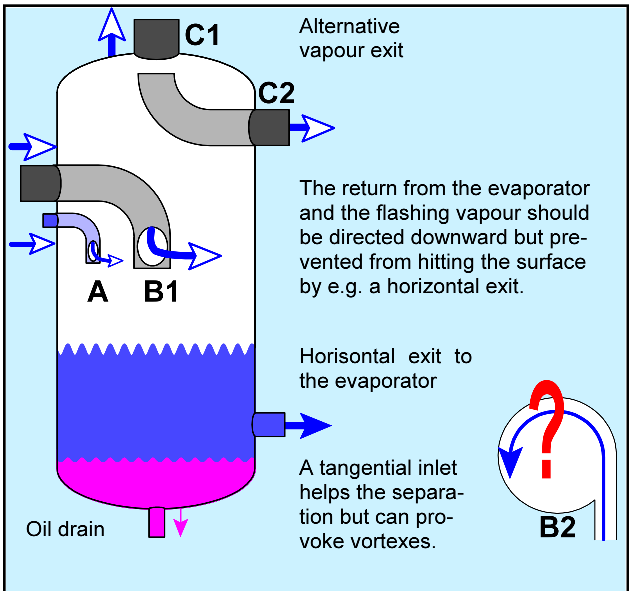


Fig. 08. The vertical separator.

The liquid body is more agitated than in the horizontal, which impairs the oil drain.

Apart from problem with the oil draini this might lead to control problems as the liquid level is not stable. In such a case the refrigerant expansion could be controlled by the level in the HP receiver.

- Separation is occurs in the horizontal part.
- The velocity is independent of the liquid level.
- The total refrigerant filling is less than the horizontal.
- The liquid level is less affected by the flashing refrigerant or the circulating vapour-liquid mixture.

4.7. The horizontal & hybrid separators.

- The direction of the vapour and especially the flashing vapour should not be directed towards the liquid surface. The resulting waves when the expansion valves opens and closes could give a feed back to the float controller and a steady state oscillation could result.

  The entrance should be directed towards the shell, away from the settling space. A tangential inlet (1 & 3) as in figures 06B or 07A1, assisted by protecting plates (8) is one solution, figures 02 B3 and 07 A2 is another. The first has less pressure drop than the other.
- All inlet and exits should be at opposites ends. Oil entering with the flashing vapour then has a chance to settle before it leaves the separator with the liquid. Otherwise, finely dispersed oil fouls the evaporator.
- Sight glasses (6 figure 06), one for a light and for viewing is an excellent method to check the operation of the separator. Note that one sight glass is close to worthless, especially at freezing temperatures.
- Also useful are connections for thermometers & manometers at the evaporator inlet and exit.
- The vapour-liquid return from the PHE to the separator should be as short as possible without unnecessary bends, valves, and other restrictions. If a shut-off valve is necessary at the separator inlet, this should be of a type which causes as little pressure drop as possible.

The best type is probably the ball valve, which causes practically no extra loss, followed by the plug cock & gate valves, butterfly valves, and finally the ones with the largest pressure drop, glove & angle valves. The latter are extensively used as control valves.

- If control of the circulation is necessary, control valves should be installed in the liquid line (figure 2B).

  It is normally not necessary to control the flow, except possibly for very high placed separators.

- Heavy foaming can occur if the system has been cleaned with kerosene, white spirit or the like, some of which remains. Other likely foaming agents are oil decomposition products.
- A longitudinal standing wave can form in the horizontal separator. This can then give an incorrect level reading or expose a connection.

4.8. Vertical separators.

The vertical separator, figure 08, has a more forceful agitated liquid body than the horizontal but it is less sensitive to longitudinal waves. The design principles discussed for the horizontal separator can be used here as well.

4.9. Summary of separator design.

The best design is usually the simplest one. A separator full of splash plates, direction vanes, distributor pipes, nozzles, etc. might work very well if manufactured as designed, but if not, troubleshooting might be impossible. It is better to use the money on size instead. Do not forget two sight glasses and some connections for manometers and/or thermometers.

5. The thermostatic expansion valve and the direct expansion evaporator.

5.1. The thermostatic expansion valve.

The refrigerant enters the evaporator partially vaporised. Typically for R22 is about 25 %. The expansion is usually made in the thermostatic expansion valve (TEV).

One controlling variable of the TEV could be the temperature. As the temperature as such is difficult to use, the corresponding vapour pressure is used instead. A bulb containing a two-phase mixture of the refrigerant (or more likely, for reasons later explained, a slightly different mixture) is placed in metallic contact with the evaporator exit.

The bulb is connected to the compartment A in the valve through a thin tube (figure 09 C). For the moment we assume that compartment B is open to the air.

The pressure in compartment A exerts a force F_A through the movable membrane to the push pin and further on to the needle. The adjustable spring D exerts a force F_D on the needle in the opposite direction. How does it work?

When the temperature increases, at the evaporator exit, the bulb and diaphragm pressure increase as well and the needle is pushed downwards, increasing the slot size and letting through more refrigerant. The spring can be tightened by the screw, thus allowing for more or less refrigerant throughput and/or superheat.

This works fine, but on one condition: a fixed evaporation temperature. The bulb alone only monitors the temperature, superheated or not. Conceivably, exchangeable sets of orifices/needles or springs, one set for each evaporation temperature, could be used.

To extend the working range of the valve, the controlling variable is changed to the superheat. We already have the superheated temperature converted to a pressure. From this we have to subtract the evaporation pressure. This could be found where the temperature is monitored - external pressure equalisation - or immediately after the expansion - internal pressure equalisation.

- ***Port forces***. Before the equalization methods are considered, another set of forces has to be discussed. When the liquid passes the port, it exerts a pressure, F_F, on the spindle. The direction of the force is in the flow direction. In figure 09 it tends to close the valve. If the flow is reversed it tends to open it. In other words, the valve capacity depends on the flow direction. This might not be a drawback; a reversible A/C-HP have usually different capacities in the two modes.

  However, it is difficult to position the spindle exactly as the flow forces vary, sometimes irregularly. Precision of the spindle positioning is especially important at low capacities of a large valve. The spindle is then close to the seat and even small movements cause fluctuations.

  One way to solve the problem is a double spindle-seat assembly (port), mounted on the same axis, with liquid entrances in opposite directions, see figure 09E. The flow forces F_{FA} and F_{FB} now largely cancel each other and the spindle position depends only on the forces F_A, F_B & F_C.

  The various manufacturers each have their own designs.

- ***External pressure equalisation.*** It is obviously most correct to monitor the pressure and temperature at the same point. However, an external pressure equalisation line is required.

  Downstream the bulb, a small pipe is attached to the exit. Through this pipe, the pressure enters compartment B and exerts a force on the underside of the diaphragm, in opposite direction to the bulb pressure. To see how it works, consider the following example for two evaporation temperatures 0 and -20 °C, each with 5 K superheat.

  Evap. temp.: 0 °C. Vapour pressure: 4.95 bar

  Bulb ": 5 °C. Bulb ": 5.81 bar

  Force on needle: $(F_A\text{-}F_B) \sim (5.81\text{-}4.95) = 0.86$ bar

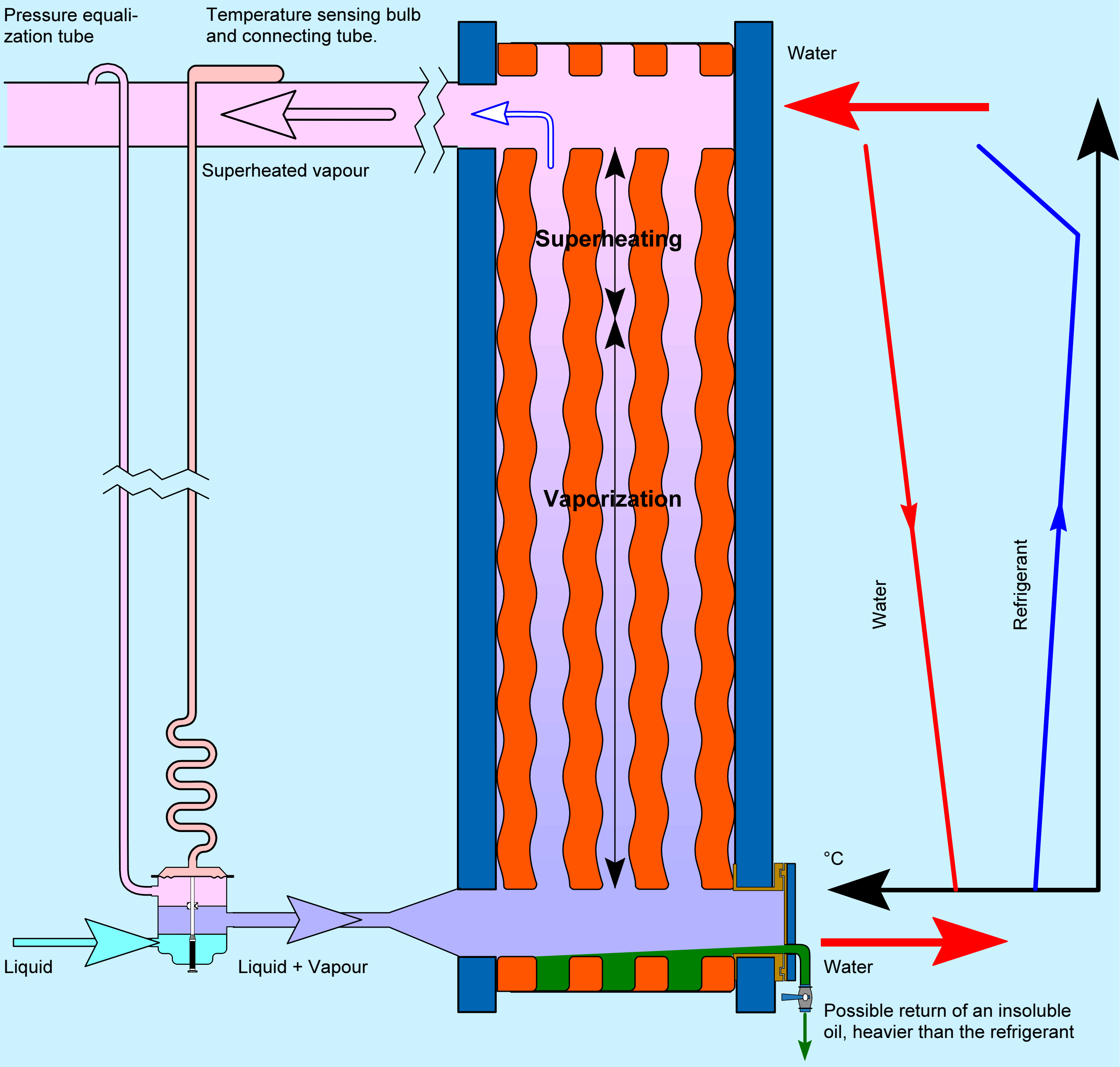
  Evap. temp.: -20 °C. Vapour pressure: 2.44 bar

  Bulb " : -15 °C. Bulb " : 2.94 bar

  Force on needle: $(F_A\text{-}F_B) \sim (2.94\text{-}2.44) = 0.50$ bar

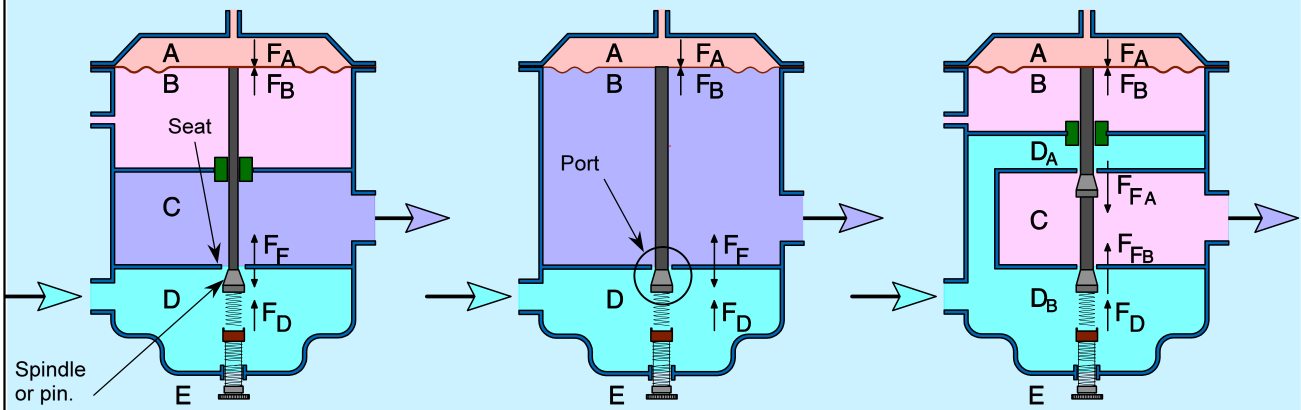
  A valve without pressure equalisation thus faces forces ranging from 2.44 to 4.95 bar, i.e. a ratio of 2.03:1, while a valve with pressure equalisation faces forces ranging from 0.5 to 0.86, i.e. a ratio of 1.72:1.

  There is a decrease in the range of the forces, but not enough to justify the pressure equalisation. It now becomes clear why the bulb filling might not be the same as the refrigerant. If the bulb pressure at - 15 °C remains the same 2.94 bar, the bulb pressure at 5°C should be 5.45 bar and the force from the membrane in both cases should be proportional to 0.50 bar. If so, the settings of the superheat adjustment screw in both cases can be equal for a given superheat.



A. A direct expansion evaporator with a thermal expansion valve & external pressure equalization.

B. The temperature profile in a direct expansion evaporator.



C. The thermal expansion valve with external pressure equalization.

D. The thermal expansion valve with internal pressure equalization.

E. The thermal expansion valve with balanced ports.

Fig. 09. The thermostatic expansion valve.

- ***Internal pressure equalisation.*** A simpler design is the thermal expansion valve with internal pressure equalisation, figure 09 D. The wall between compartment C & D is simply removed and the pressure after the expansion is directly exerted on the underside of the membrane. The temperature is therefore monitored after, and the pressure before, the evaporator. The thermostatic expansion valve with internal pressure equalisation should thus be used for:

  A. Evaporators with a low pressure drop, i.e. exit pressure equal to the inlet pressure.

  B. Evaporators with constant pressure drop.

  As the BPHE has a relatively high pressure drop, thermostatic expansion valves with internal pressure equalisation should be avoided.

- ***Limitations.*** A thermostatic expansion valve is thus used to control the refrigerant flow and hence the evaporator capacity providing that the vapour leaves the evaporator superheated. It cannot be used to provide a pre-determined two-phase flow at the evaporator exit, nor can it be used to control the evaporation pressure.

5.2. The purpose of the superheat.

- ***Compressor protection.*** If refrigerant droplets enter the compressor, the result may be liquid hammering. This is the sudden pressure shock, which arise when the liquid refrigerant evaporates - due to the temperature increase at the compression - in the compressor.

  Due to this extra addition of vapour, the pressure increases more than if the normal amount of vapour alone were compressed. The compressor becomes overloaded and could be destroyed. Liquid hammering is not the result of the liquid being incompressible as is sometimes stated. The vapour to the compressor sometimes contains fairly high - 5 % -"incompressible" oil droplets, without harming it.

  Refrigerant droplets could also wash away the oil and the compressor could then seize.

  The compressor types most sensitive to liquid hammering are non-hermetic piston and scroll while the least susceptible are screw & turbo compressors.

  The vapour is superheated to ensure that any remaining liquid will evaporate before the vapour enters the compressor. How much liquid can evaporate?

  Ex. R22 with a saturation temperature of 0 °C and superheated to +5 K can contain 1.8 % liquid droplets. These will then evaporate and the result will be 100 % saturated vapour at 0 °C.

  Ammonia vapour containing 1.8 % liquid has to be superheated to almost 11 K in order to evaporate all the liquid, thus considerably decreasing the effective temperature difference to the water.

  This is one of the reasons why ammonia based plants normally use flooded evaporators, which do not superheat the vapour.

- ***Control of the thermostatic expansion valve.*** The valve lets through a determined amount of refrigerant controlled by the superheat after the evaporator. Why?

  To answer the question, imagine what would happen if the exit from the evaporator were to be saturated or even contain some residual liquid, say 1 %. What would then be the controlling variable to assure the 1% exit liquid?

  Unfortunately, it is difficult to find such a variable. Pressure and/or temperature could not be used, as these are the same for a mixture of 1 or 99 % liquid. The mean density could be used, but in practice, it would be extremely difficult or impossible to monitor this.

  The light dispersion or absorption could be used to detect liquid droplets, but then a certain amount of liquid refrigerants needs to be present.

  The sound of the liquid hammering monitored by a microphone in the compressor could also be used to control the expansion valve.

  When the vapour starts to superheat, the situation changes. The temperature and pressure then move independently, providing a set of two control variables. Rising superheat, i.e. the evaporator operates below capacity, the valve opens and more refrigerant passes.

  Sinking superheat, i.e. the evaporator operates above capacity, the valve closes and less refrigerant passes

5.3. The direct expansion evaporator.

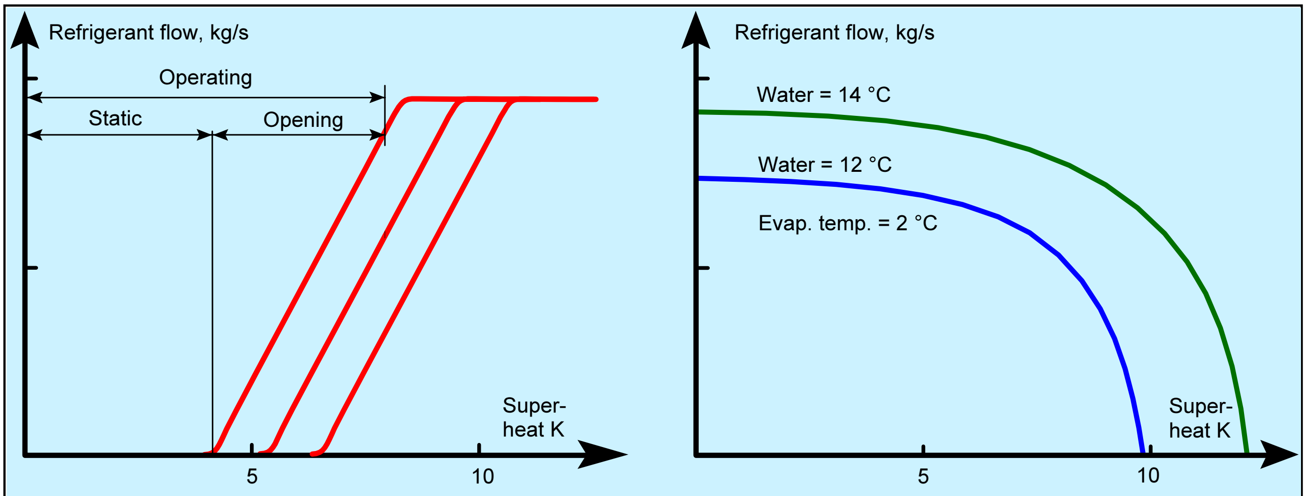
When the saturated mixture rises through the evaporator, the pressure decreases. This is due to a decrease in the pressure drop and to a less extent, an increase in the static height.

Because of this decrease in pressure and because of the temperature-pressure relationship, the temperature will decrease from the inlet to the point, where the entire refrigerant has vaporised. Here the vapour - freed from the restraints of the vapour pressure curve - will start to superheat. The extent of superheat varies, but a typical figure for R22 is 5 K. See Fig. 09 B for the temperature profile. Notice the difference in profile from a thermosiphon evaporator (Figure 03 B).

5.4. The valve-evaporator system.

5.4.1. The bulb filling.

To give the appropriate control for the evaporator a number of different bulb charges have been developed. These control the behaviour of the valve at different evaporation temperatures, at startup or shutdown, etc. There are three main types:



A. The valve characteristic curve.
The valve remains closed until a certain superheat - the static superheat - is reached. It then opens and the refrigerant throughput (capacity) increases until the valve is fully open. The figure shows the curves for various settings of the static superheat. The through- put is here a linear function of the superheat.

B. The evaporator characteristic curve.
When the refrigerant flow (the capacity) is maximum the superheat is zero. When the flow decreases, the superheat increases until a maximum, set by the hot side inlet temperature, is approached for zero flow. The figure shows the curves for two different inlet temperatures - 10 and 12 °C - of the warm side.

Fig. 10. The valve and evaporator characteristic curves.

- ***Liquid charge.*** The bulb contains both vapour and liquid over the entire working temperature range. The other parts (line and valve diaphragm chamber) may contain liquid.

  If the bulb temperature is higher than the other parts, refrigerant migrates to the other parts and can eventually fill them entirely with liquid. The system pressure is set by the two-phase part, i.e. the bulb.

  If the bulb temperature is lower than the other parts, liquid in these evaporates and only vapour remains. The system pressure is still set by the two-phase part, i.e. the bulb.

- ***Cross charge.*** The slope of vapour pressure of the bulb filling is less than that of the refrigerant, i.e. the vapour pressures cross at some temperature and the superheat increases then with increasing temperature.

  This is an advantage at low temperature operations. When the system starts, usually at a much higher temperature and thus capacity, too much for the evaporator, the required high superheat then restricts the refrigerant flow and impedes liquid refrigerant from entering the compressor.

- ***Gas charge.*** The charge is mainly vapour in the entire system, except for a small quantity in the bulb. Above a certain evaporation temperature, all the liquid has evaporated. The bulb pressure then increases very slowly, while the evaporation, pressure still increases exponentially. The result is that the valve closes above a certain temperature/pressure.

  This MOP (Maximum Operating Pressure) feature relieves the load on compressors with low torque motors at startup. An excessive starting pressure could burn out the motor. A disadvantage is that the bulb must always be the coldest part of the bulb-valve system; otherwise, liquid migrates to the coldest part, which then takes over the control. The valve has a rapid response to temperature changes.

- ***Absorption charges.*** The system is charged with a gas and the bulb with an absorbent. The colder the bulb, the more gas it absorbs. It will only react on the bulb temperature and it will not provide MOP.

5.4.2. The valve characteristic curve.

Even when idling, the spring exerts a force on the needle, i.e. the valve remains closed as long as the force from the membrane is lower or equal to the spring force. The needle opens when a force corresponding to a certain superheat - the static superheat -is reached.

In order for the valve to give the required refrigerant flow a further superheat - the opening superheat - is required. The sum of the two is called the operating superheat and has to be equal to the superheat from the evaporator for the required capacity. Figure 10 A shows the characteristic curve for a typical valve. The valve is usually factory set for a certain static superheat. This can be changed with adjusting screw E. This means a parallel displacement of the curve. The superheat transmitted to the valve is transformed to the corresponding valve opening.

The throughput of the valve measured in refrigerant flow per time unit or in capacity depends not only on the valve opening but also on the pressure difference between inlet and outlet, the absolute pressure and the type of refrigerant. Moreover, the valve needle can have different forms, i.e. the opening is not necessarily a linear function of the superheat. See below for possible other curves.

5.4.3.The evaporator characteristic curve.

The evaporator characteristic curve can be plotted in a similar way. The independent variable here is the refrigerant flow and the dependent the resulting superheat. In order to facilitate comparison with the valve, the axes are oriented as for the valve, i.e. the flow as a function of the superheat. Figure 10 B shows the curve for a CB52-50H.

The nominal capacity is 29.7 kW corresponding to a flow rate of 694 kg/hr R22 at an evaporation temperature of 2 °C and a superheat of 5 K. The nominal water inlet temperature is 12 °C and the flow is constant 5086.5 kg/hr. The green line shows the characteristic curve when the water inlet temperature has increased to 14 °C.

When the R22 flow rate changes, the superheat changes according to the curve. For a flow of about 736 kg/hr, the resulting superheat is zero and when the flow approaches zero the resulting superheat approaches its maximum value, 10 K (12 K for a water temperature of 14 °C).

The evaporator curve is more complex than the valve curve. The amount, temperature and type of heating media will affect the superheat. It is also fairly difficult to calculate correctly the thermal characteristics of an evaporator because there are many parameters, e.g. surface finish, which are difficult to correlate, the flow behaviour is not well understood, etc.

5.5. Matching the valve to the evaporator.

5.5.1. Operating point.

When matching a thermostatic expansion valve to an evaporator, two parameters at the actual operating point have to be evaluated:

- ♦ ***The required capacity*** has to be obtained.
- ♦ ***A stable system*** is required.

It is easy to obtain the operating point of a valve-evaporator system; simply superimpose the evaporator curve on the valve curve. The intersection between the two curves is the actual operating point. At the operating point, the given refrigerant flow is superheated to a value for which the expansion valve gives this flow. See figure 11.

5.5.2. Capacity of the system valve-evaporator.

The capacity is thus easily found for the system in figure 11 A. However, the capacity is only 27 kW, less than the nominal 29.7 kW. What can be done in order to achieve the nominal capacity?

One possibility would be to increase the evaporator size. The green line in figure 11 A shows the curve for a CB50-60H. The intersection of this curve with the valve curve lies at about 29.7 kW. The superheat has however increased to almost 8 K and this might not be permitted.

Another possibility is to use a larger valve. Figure 11 B shows this. The intersection of the valve and evaporator curves is at about 29.7 kW for a superheat of 5 K. The inclination of the curve is larger than the one above, which could mean stability problems; see below.

Figure 11 B shows an inadequate small valve, the magenta line. The intersection of the valve and evaporator curve is here in a region where the valve is fully open. The valve has here effectively ceased to control the superheat, apart from being too small for the required duty.

It should be kept in mind that the TEV is a proportional control device, i.e. it can only control the superheat for a determined capacity.

Example. Assume a condenser/evaporator at a cascade system. The TEV has a capacity of 60 kW for a superheat of 6 K, which exactly matches the capacity of the three compressors feeding the condensing side. What happens when only one compressor operates?

The capacity decreases to a third, 20 kW. The evaporator/separator solves this by reducing the MTD to 1/3, i.e. a decrease of the condensing temperature. In order for the valve to give this capacity, the ***superheat has to decrease*** to only 4 K, assuming a static superheat of 3 K. At this low superheat - only 1 K operating superheat - the TEV is very vulnerable to disturbances. (See also § 11, **1. Applications** and § 6.7, here).

Conclusion. It is important that the valve and evaporator match each other, i.e. that the valve gives the required capacity for the superheat delivered by the evaporator at this capacity. The operating point should allow the valve a control range in both directions. Normally a valve can be used to ~ 20 % above its rated capacity, which should allow a sufficient range at the rated capacity.

5.5.3. Stability of a valve-evaporator circuit.

In addition to giving the correct capacity, the system valve-evaporator has to operate free from oscillations and unsteady state behaviours. The phenomenon of the valve moving from near closed to near open is commonly referred to as "hunting" in refrigeration terminology.

There are various sources of oscillations and unsteady state behaviour in a refrigeration system:

- ♦ ***Noise***. We will here adapt the terminology from control technology. Noise is the random occurrence of irregularities in a variable; the most common example and the origin of the term are heard in an ill-tuned radio.
- ♦ ***Feedback oscillations***. This is the shrill sound that can be heard from a public address system, composed of a microphone, an amplifier and a speaker, when the microphone gets too close to the speaker or when the amplification in the circuit is too large. It occurs in circuits where an output signal from a device is fed back to the input and thus affects itself.

  In a refrigeration system, there are numerous examples of feedback systems. One example is the valve-evaporator loop. Actually, the whole refrigeration system is one feedback system. In such a system, oscillations can occur even though all the individual components operate without apparent noise or other instabilities.
- ♦ ***Constant change of the operating parameters,*** especially conflicting ones. Increasing superheat would cause a TEV to open while increasing condensation pressure would cause more refrigerant to pass, the effect of this would be to close the valve.
- ♦ ***Unstable system.*** Sometimes the system is inherent unstable, e.g. the subcooler/superheater cycle (§10, fig. 5, **1. Applications.**) especially if the TEV is placed after the subcooler/superheater.

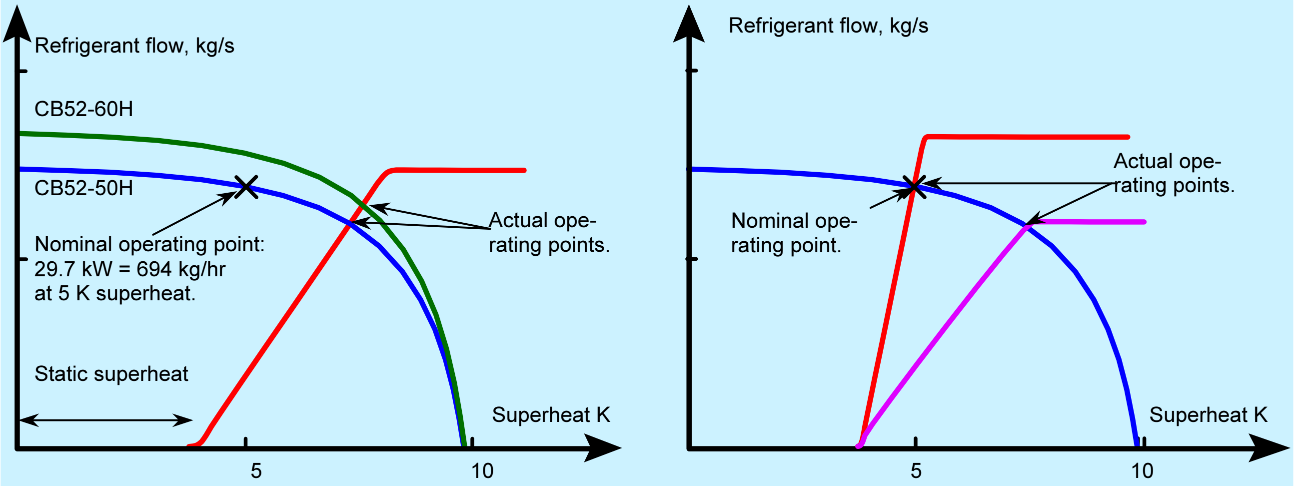
5.5.4. Noise in an evaporator.

In this paragraph, the term noise is used in the sense described in the preceeding paragraph i.e. as a source of disturbances and fluctuations in a control circuit, not in the normal acoustic sense.

Consider an evaporator, which is fed a steady flow of refrigerant at constant conditions. The conditions at the warm side are constant as well. The output from this evaporator - the superheat - should then be constant as well.

Unfortunately, this is not always the case. The vapour state can sometimes vary from humid to superheated. The variations are most obvious for low channel flow rates (water & refrigerant), i.e. low channel pressure drops and low rated superheats. The reasons are:

- ♦ The random nature of turbulent flow, especially of a two-phase medium. A channel can suddenly receive more than its share of vapour, backflow can occur, etc.



A. The operating point of a thermostatic expansion valve for two evaporator sizes.

The operating point is found by superimposing the two characteristics curves from the valve and the evapora- tor respectively. The crossing point between the curves ves gives the actual operating conditions, about 7.6 K at a flow of 654 kg/hr refrigerant for the small size and 7.9 K at a flow of 694 kg/hr for the large one.

B. The operating point of an evaporator and two sizes of a thermostatic expansion valve.

Assume that the required operating point of the system is 5 K and 696 kg/hr. Obviously, the valve in C is too small. The solution is a larger valve, one that gives 694 kg/hr for an opening superheat of just 1 K instead of for 3.9 K. The smaller valve shown operates fully open, i.e. far above its control area.

Fig. 11. Matching the valve and the evaporator.

The frequency of the resulting oscillations is lower, the lower the Reynolds number (~ channel pressure drop) is. High frequency oscillations might be absorbed by the inertia of the system, while low frequency, oscillations continue to the exit.

- ♦ A low pressure drop on the liquid side and, especially for downward flow, there might be backflow, induced by temperature-dependent density differences. 1 K temperature difference between the nominal and real superheat is obviously more important the lower the nominal superheat is, i.e. the valve reacts relatively stronger at a lower superheat.
- ♦ Large amounts of oil - especially insoluble types -entering the evaporator can provoke fluctuations in the heat transfer and thus be a source of noise.
- ♦ The largest source of noise - actually it is not noise - is constant change in the operating parameters. The refrigerant flow, temperature & pressure and the water flow and temperature change as result of varying operating conditions. All these change the superheat at the evaporator exit and the TEV has to respond to this.

5.5.5. The evaporator stability curve.

Attempts have been made to correlate the necessary superheat to avoid instability with the capacity. The capacity is then plotted as a function of the superheat and the valve characteristic curve is superimposed. The valve curve should then touch but not cross the stability curve.

Unfortunately, the stability curve; see figure 12 (note, it is not an evaporator characteristic curve) is a slightly nebulous quantity.

It cannot be calculated, only measured, which makes it impractical or impossible for practical use. It is also difficult to specify test conditions, which are the same as the design conditions.

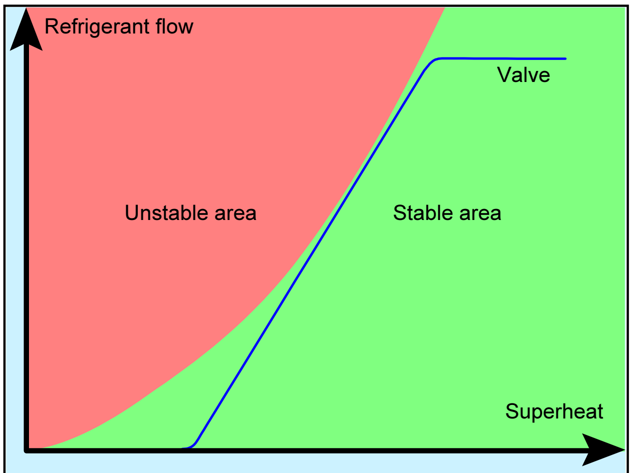


Fig. 12. Stability curve of an evaporator.

The red zone to the left shows the unstable area of the valve. The valve characteristic curve should ideally fall within the green, stable zone.

For given design conditions, a specified heat exchanger gives a certain superheat, figure 10B.

In order to vary the superheat for a constant capacity, the warm side inlet temperature, the evaporation temperature or the evaporator size has to change. This means that the test conditions might not be the same as the design conditions, rendering the result questionable.

Moreover, oscillations from an unstable component in a well-designed circuit (see below) could attenuate, rendering the use of the stability curve still more doubtful.

5.5.6. Feedback oscillations in a valve-evaporator loop.

A system with a feedback like the TEV-evaporator loop, figure 13 A, is prone to oscillations (hunting). In order to study the dynamic behaviour of the system, the loop is interrupted at some point. This can be anywhere, but in this case, a convenient point is where the superheat signal from the evaporator is fed to the TEV, figure13 B.

A sine wave formed superheat signal is then applied to the valve inlet. The valve needle position and the refrigerant flow changes and the superheat response from the evaporator is studied. Both signals should be measured in the same units, here K. The criterion of oscillatory (the Nyqvist diagram) flow can be simply stated as:

- ***The response has to be in phase with the inlet.***
- ***The response has to be amplified.***

Figure 13C shows the various possibilities. For a given system, the phase and amplification change with the frequency, and for some frequencies the oscillation criteria above could exist. This frequency depends on system parameters such as the valve and evaporator characteristics (figure 11), placement of the valve and the bulb as well as the more difficult to define dynamic characteristics. Suppose that we close the circuit again. What happens?

- ***Response out phase, but amplified***. A new cycle starts and most likely, the oscillations will die out.
- ***Response in phase but attenuated.*** If subsequently fed to the valve and once more returning, it is still more attenuated, i.e. the oscillations will die out.
- ***Response in phase and amplified.*** When fed to the valve, the response is still more amplified, resulting in a self-sustaining oscillation, i.e. hunting.

What is then the risk that a sine formed disturbance of the correct frequency enters the loop? If such a frequency exists, it always amplifies. Any irregular disorder, which enters the loop, separates into a number of sine waves (Fourier analysis), and the one with the resonance frequency then amplifies. Note the similarity to audio systems.

5.5.7. Response of a valve-evaporator system to a sudden temperature change.

The most frequent source of disturbances in a refrigeration circuit is probably a change of the refrigeration load. The incoming water temperature and thus the superheat then change, provoking a reaction from the valve. A study of the valve-evaporator curve in figure 11 shows that one parameter - the relative inclination of the two curves in the operating point - could be of importance.

The exact dynamic behaviour is very difficult but not impossible to analyse mathematically. It is however outside the scope of this manual. We will here give a simplified analysis for two cases; the valve curve is steeper than the evaporator curve and the evaporator curve is steeper than the valve curve. See figure 14.

The analysis given in figure 14 is not actually a dynamic one, but a prediction of the dynamic behaviour based on the static properties of the system. Unfortunately, there are very few studies of the dynamic behaviour of PHE evaporators with and without TEVs.

5.5.8. Factors affecting the valve-evaporator behaviour.

Thus, it seems that if the TEV curve is steeper than the evaporator curve there is a danger of hunting. However, in order for hunting to occur, the response must not only be amplified but also in phase, see figure 13.

- The superheat signal might be attenuated, e.g. by placing the bulb further away from the evaporator. Back mixing of the vapour due to the turbulence decreases the amplitude of the changes.
- A change of the bulb position means that a signal may no longer be in phase but signals of another frequency might then be in phase.
- The valve responds very rapidly if the superheat changes because of a pressure change. A pressure change transmits almost instantly to the valve, especially one with internal pressure equalisation. See figure 09. Conceivably, the response could be delayed in a valve with external pressure equalisation by placing an orifice or a small valve in the equalisation line.
- The valve reacts relatively slowly to a change in temperature as the wall of the connection has to be heated, heat has to be conducted through the wall and to the bulb and the bulb-filling has to change phase.
- The hunting risk could thus be different for a superheat change due to a pressure or a temperature change.
- If the static superheat decreases, i.e. the valve curve moves left and becomes relatively steeper, the hunting danger increases. See figure 11A.
- Two valves of different sizes see figure 11B. The smaller one might not give the capacity but gives less danger of hunting than the larger valve.
- The total mass of refrigerant and metal can vary from one system to the other and this affects the response.
- A valve can have the characteristic curve as in figure 15. This can be obtained by by-pass, either internal in the valve or external via a parallel-connected valve.

5.5.9. Stability criterion of the valve-evaporator system.

If the inclination of the valve curve at the operating point is larger than that of the evaporator curve, there is a danger of instability. See figure 16.

5.5.10. Choosing & installing an expansion valve.

- There are a number of TEVs on the market with different characteristics of the bulb filling, response time, shutdown features, etc. Select a valve in co-operation with the valve manufacturer. As a rule, at the desired operating point ***the valve and the evaporator must have the same capacities and superheats***. The valve could be somewhat larger, though.
- In case of two different, superficially equal, valves, select the one with the lowest inclination of the characteristic curve. See figure 11 B.
- It is difficult to give any absolute rules for the inclinations of the valve and the evaporator curves at the operating point. If the inclination of the evaporator curve is larger than the valve curve, there should be no danger of instabilities, but this is probably too strict a rule.
- MOP valves are sometimes used for pressure control. Note however, that they can only act as pressure controller during normal operation, not at startup.
- Balanced port valves are probably good choices, especially at small capacities of a large valve. The spindle is then close to closing and even very small movements cause large variations. See figure 09E.
- Is the valve correctly installed? A TEV with adsorbtion or liquid charge can be in any position while gas charge needs a vertical valve.

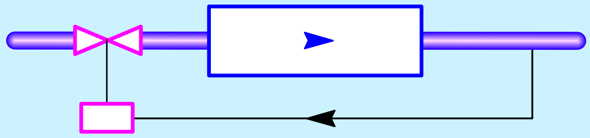


Fig. 13 A. The TEV - evaporator loop.

The evaporator and the valve form a closed loop, where the superheat is fed back to the valve as an input signal.

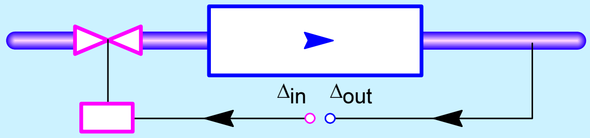


Fig. 13 B. The open TEV-evaporator loop.

The loop is interrupted at the superheat signal. The purpose is to tudy the effect a change in Δ_{in} has on Δ_{out}.

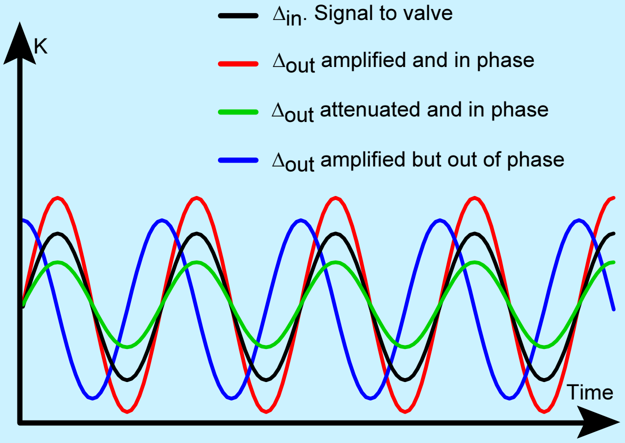


Fig 13C. The dynamic behaviour.

The figure shows the various possible responses of Δ_{out}, when a sine wave is applied to Δ_{in}.

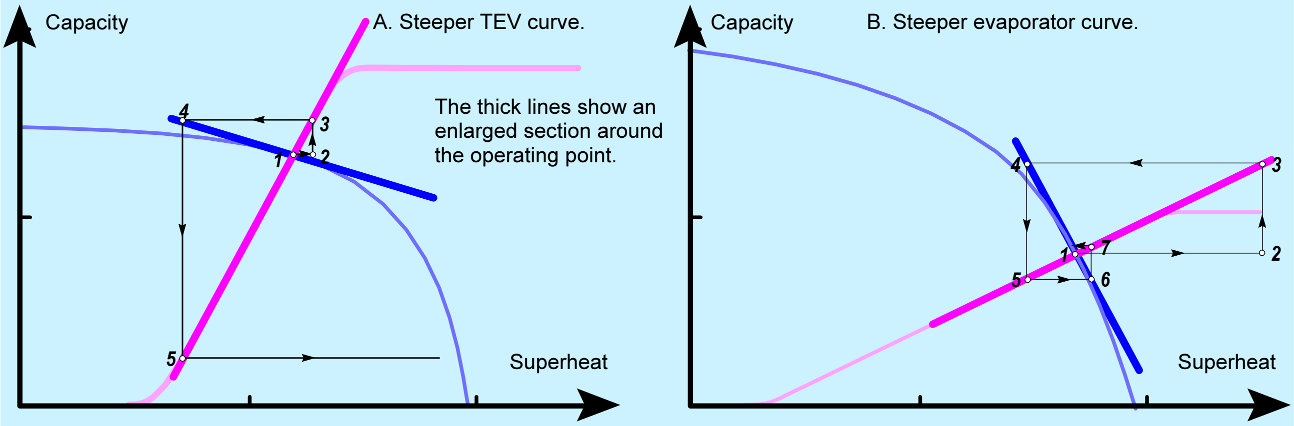


Fig. 14. The response to a sudden change of the superheat.

The valve-TEV operates in a steady state at the operating point 1.

Then, the superheat suddenly changes, e.g. as a response to a load change, to a new value at 2.

This new value corresponds to a new capacity at 3.

The new capacity corresponds to a new superheat at 4.

The capacity then changes to 5 and so forth.

It is easy to see that in case A, the operating point spirals outward and with a risk of hunting. In case B it spirals inward, back to the stable operating point.

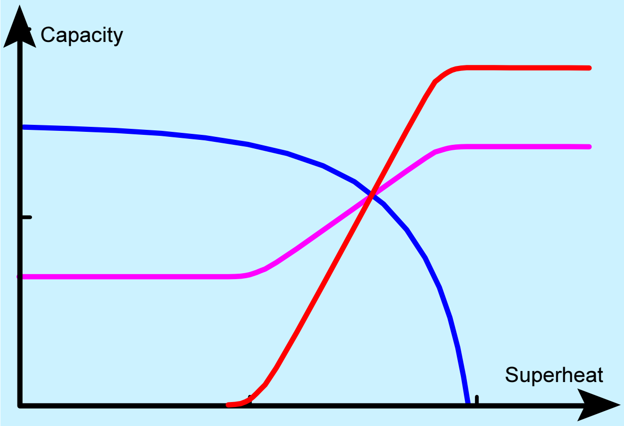


Fig. 15. Valve with a by-pass.

A by-pass, either internal or external, decreases the inclination at the operating point and makes the assembly more stable than a single large valve.

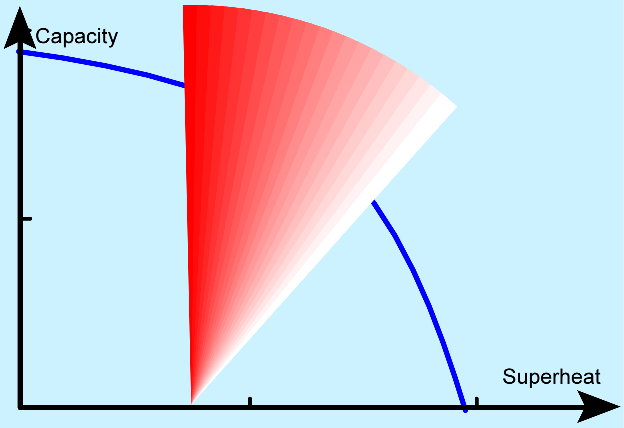


Fig. 16. Inclination of the valve curve.

The risk of hunting is small or non existent for a valve curve of the same or less inclination than the evaporator curve. The risk increases with increasing inclination.

- The evaporator should, if instabilities are anticipated, be designed with a larger superheat than the normal 5 K. Due to the normal overdesigning because of fouling, this is actually the case. This increases the inclination, but with some loss of capacity.
- The bulb of a MOP valve should never be installed on a vertically pipe with the capillary tube pointing down nor should it be warmer than the valve. This can facilitate the transfer of the filling from the bulb to the valve, especially if the bulb get warmer than the valve. Heat the valve to move back the filling to the bulb.
- Select a valve with a maximum capacity less than the capacity for which the evaporator gives zero superheat. Should hunting occur there is no danger that liquid refrigerant enters the compressor.
- Figure 17 shows some rules for installing a thermostatic expansion valve. The bulb is here uninsulated, but should usually be insulated.
- Refrigerant could be leaking through the packing box of the expansion valve and thus liquid refrigerant enters the vapour to the compressor.

  Normal design instructions say that the bulb should be installed upstream of the pressure gauge to prevent false bulb readings. However, this means that the actual superheat entering the compressor is not correct.

  However, if there is sufficient distance between the evaporator and the compressor to place the bulb some 400 -600 mm downstream of the pressure gauge, the bulb could be placed there. The liquid refrigerant from the pressure gauge then has time to evaporate and the bulb will monitor the correct superheat. See also figure 21.
- When the pressure and pressure difference decrease, so does the force the diaphragm exerts on the valve pin. A sufficient large force is important to obtain a stable operation. The low force can be compensated by a large diaphragm surface. Differently expressed, of otherwise equal valves chose the one with the largest diaphragm diameter, especially important at low temperature operation.
- Avoid valves with internal parts of plastic as the lifetime can be reduced.
- In case of suction line separators in a DX installation, consider installating the temperature bulb after the separator. See **§ 6. Suction line separators** here.
- Parallel connected evaporators should each have its own TEV as it is difficult to evenly distribute a two-phase mixture between the evaporators. See also fig. 11a, **3. Design & Installation**..
- A step motor valve does not close at a loss of power (nor does it change position). Even though it normally closes tight when the plant is shut down, it should therefore be preceeded by a solenoid valve.
- Avoid pulse width modulated valves, i.e. where the valve stays open for a certain period and then closes, see **§ 6.2. The electronic expansion valve,** as they can damage the evaporator.

5.5.11. Evaporators with distributors.

At low nozzle velocities in a BPHE, there is a risk that the vapour and liquid, emerging from the expansion valve, separate and enter different channels. On the other hand, if the nozzle velocity is too high and the resulting pressure drop is high in comparison with the one in the channels, there is a maldistribution of refrigerant from channel to channel. This is a general problem of distribution between parallel channels regardless of the channel type.

Various solutions have been proposed; figure 18, but all have some drawbacks. Good distributors as in figure 18 A and B have fairly high pressure drops and can thus not be used in the condenser mode in a reversible cycle.

The system shown in figure 18 C is composed of a premixer and the distributor pipe, which is pressed into the plates. It gives a good distribution over the plate pack without excessive pressure drop. It can be used as a condenser, but only total condensation. Partial condensation as well as one-phase vapours and liquid cannot be accommodated.

The pressure drop in the distributor is less than 0.1 bar at condensing, i.e. some tenths degree Kelvin. Refrigerant flashing is thus not a problem as the condensate is usually more than three degrees subcooled.

The expansion valve has to be somewhat larger than normal to make up for the lower effective driving force.

5.6. Troubleshooting the system valve-evaporator.

5.6.1. Liquid hammering.

If the pressure of a vapour bubble suddenly increases, the bubble collapses. If the energy of the surrounding liquid is taken up by e.g. a metall surface, this can be damaged. This is what happens in cavitation.

A similar phenomena might happen in refrigeration plant. Suppose a DX evaporator as in figure 02A. The solenoid valve is closed and the system is pumped down. If the TEV is tight, but not entirely, the liquid evaporates between the solenoid and the TEV. If the solenoid suddenly opens, high pressure liquid enters. The vapour column implodes and a high speed liquid column hits the the TEV with possible damages as consequence. Such imploding liquid bubbles can be the origin of the bangs which sometimes are heard in refrigeration plants.

5.6.2. Hunting.

If, despite all precautions, hunting occurs:

- Experiment with changing the static superheat. Depending on the shape of the valve curve this can affect the stability of the system.
- Move the bulb further away from the evaporator. This will attenuate oscillations. The bulb should not be too close to the compressor though, otherwise it has insufficient time to react to prevent unevaporated refrigerant entering the compressor.
- Try to open the valve completely and to keep it open. Are there still oscillations in the system? Try to find out from where they originate.
- Does hunting only occur at low capacities? At very low channel flow rate the operation is sometimes erratically.
- Is the flow from the condenser or liquid receiver constant and with constant properties?
- Is there a hot gas by-pass control or freeze protection (§5, **7. FREEZE**)? Is that the origin of the hunting?
- Try to increase the temperature difference between the media in the evaporator. This gives some leeway for moving the valve curve to an area where the inclination is less than the evaporator curve. (Figure 11A).
- Check the refrigerant filling of the system. If insufficient, the liquid receiver runs dry and the expansion valve receives an irregular flow of refrigerant. This affects both the stability and the capacity of the evaporator.
- MOP valves have normally less tendencies for hunting.

The figure shows a BPHE with the water connections at the back cover plate.

1. The bulb must be installed as the figure shows, never at the bottom of the tube, where oil can give errors.

2. The pressure probe must be installed downward from the bulb. See comments in § 5.4.9.

3. The bulb and the pressure probe should preferably be placed after a bend, on a horizontal section of the pipe. The bend acts as a vapour-liquid separator, to prevent liquid refrigerant and oil from interfering with the measurements.

4. The pipe from the thermostatic expansion valve to the evaporator inlet should be straight and of the same diameter as the valve exit.

5. If the distance between the compressor and the bulb & the pressure probe is too short, liquid refrigerant could enter the compressor, as the valve does not have time to react to load changes.

6. The solenoid valve (not shown here) should be placed as close as possible before the thermostatic expansion valve.

7. Chose a valve with an as straight exit as possible. Be observant on possible turns and bends of the vapour-liquid mixture inside the valve as the resulting centrifugal forces can lead to phase separation.

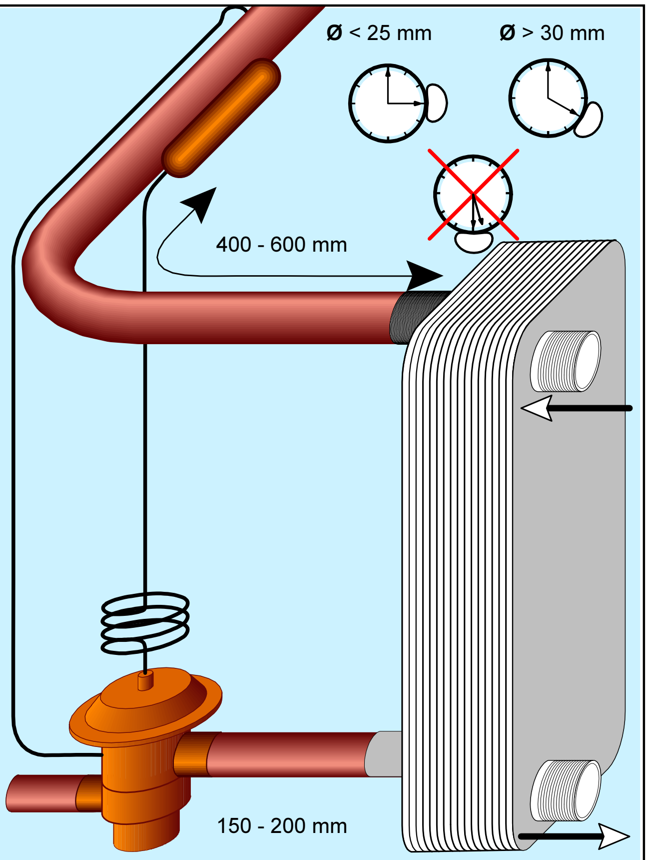
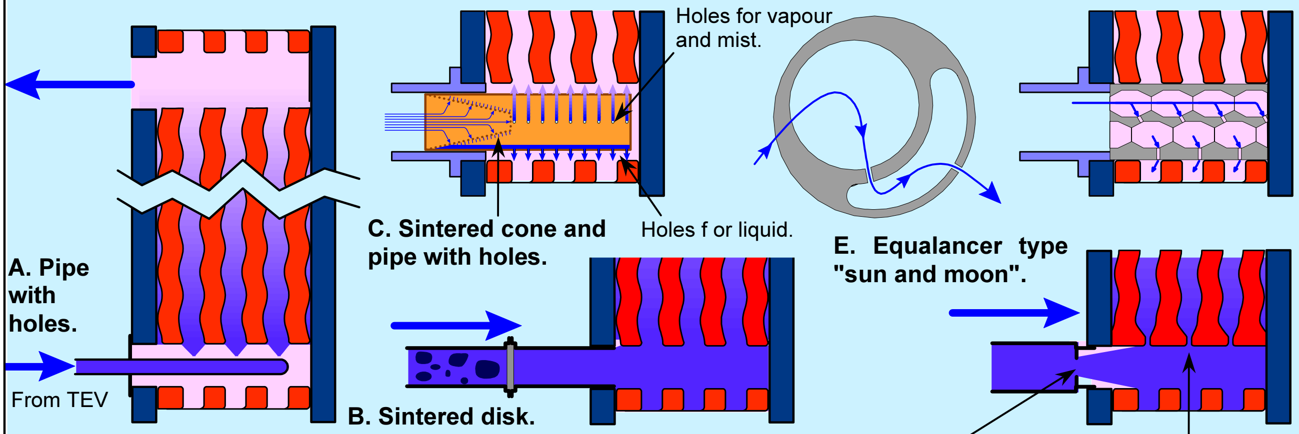


Fig. 17. Installation of the thermostatic expansion valve.



A. Pipe with holes.

B. Sintered disk.

C. Sintered cone and pipe with holes.

D. Premixer and integrated distribution holes pressed into the plates.

E. Equalancer type "sun and moon".

Fig. 18. Refrigerant distributor.

Various methods have been tried to improve the distribution of refrigerant from channel to channel. All offer a mixture of Improvements and drawbacks.

A. A pipe with drilled holes, usually one for every second channel, is inserted in the port hole. The hole diameter increases with the distance from the entrance. This is an effective method to improve the distribution in semi-welded heat exchangers, both for ammonia and freons but it can also be used in brazed heat exchangers.

B. A sintered disk with micropores is inserted in the pipe after the TEV. It breaks up large liquid slugs and thus improves the distribution. It is susceptible to fouling and it cannot be used as a condenser in a reversible system.

C. Similar to B but with a sintered cone instead of a disk. After the cone the mixture enters a pipe with two rows of holes. The lower row discharges mainly liquid and vapour the upper cone vapour and mist. The liquid at the bottom has now time to redistribute longitudinally.

D. The distributors pipes are pressed into the plates and the connection is equipped with a premixer. Various distribution systems have been tried with good results. The design can also be used for total (but not partial) condensation,and is therefore suitable in reversible units.

E. The equalancer system (patented). The liquid-vapour mixture enters the first port hole, which, from a topological point of view, is placed on the liquid side and separated from this by the brazing between the plates. From these port holes it enters the second set of port holes which are placed normally on the refrigerant side and from here it enters the channel. The design gives a good longitudinal mixing but the disadvantage is that it can not be used for single phase flow.

5.6.3. Insufficient capacity.

- Check all flow rates, temperatures, pressure drops, etc. Does the pressure drop hint at anything abnormal? An obstruction of the water flow, too much oil, etc.?
- Is one TEV feeding several evaporators? It is difficult to distribute a two-phase mixture between parallel evaporators.
- Interchange thermometers between various locations. Small temperature differences can easily be masked by incorrect thermometers.

  Ex. The thermometer at the warm side exit has an error of +0.5 K and the one at the cold side inlet (counter current flow) -0.5 K. The measured difference is 4 K. Switching the thermometers gives a difference of 2 K. The mean, 3 K corresponds to a truer value.
- Double-check the heat transfer, by using various combinations of temperature and flow.
- Inspect the evaporator for temperature variations on the outside. Large differences could hint at maldistribution on one side or the other.
- Inspect the heating fluid. If glycol or similar, check for concentration and/or viscosity. A too-high concentration impairs the heat transfer and a too-low means freezing.
- Check for ice. Ice impairs heat transfer and the water exit temperature could actually increase.
- Inspect the heat exchanger for signs of fouling on the liquid side and oil on the refrigerant side. Oil, especially at low temperatures, could impair the function of the TEV and block part of the evaporator.
- Are the condenser and compressor capacities correct? If so, the evaporator capacity should be correct too. If not, there must be some irregularity.
- If there is water in the refrigerant, ice could form in the expansion valve, thus obstructing the flow.
- Insoluble oil, e.g. in the system oil-ammonia could clog the valve, especially at low temperatures.
- Does the TEV show signs of mechanical damages?
- Is the TEV installed according to the manufacturers recommendations?
- Is the equalisation line properly connected? If it is blocked or capped, the pressure in compartment B in figure 09 C will be equal to that in C, especially for an old or worn out valve. This pressure is higher than the correct equalisation pressure. The valve then assumes that the superheat is too small and closes.
- Is the bulb correctly installed and making good contact with the pipe wall?
- Refrigerant is steadily leaking through a faulty hot gas bypass valve, which decreases the capacity.
- Is the valve used for the correct refrigerant?

  Example: A system for R 12 is refitted with R 134a. The TEV settings are a superheat of 5 K and an evaporation temperature of 0 °C. The valve expects a pressure from the equalisation line of 3.084 bar (R 12 at 0 °C). If R 134a is set to 0 °C (2.929 bar), the valves assumes that the superheat is 5 °C - (-1.6 °C) (R 12 at 2.929 bar) = 6.6 K, i.e. much larger than the setting and it opens. In this case, good for the capacity, potentially disastrous for the compressor.

  Note! Refrigerants with glide such as R 407c can give rise to confusion in the settings. See **8. Oil & Refrigerants**. Be careful to follow the valve manufacturers' recommendation when changing refrigerant.
- Water in ammonia has the same effect. See the appendix for the rise in boiling point due water pollution.
- Detach the bulb from the exit pipe and let it warm up. The increased temperature in the bulb forces the valve to its maximum capacity. Does the capacity increase?
- Certain bulb fillings demand that the bulb is the coldest place in the valve-bulb system. If the valve is installed outdoors, the reverse might be true in winter.
- If there is a leak at the pushrod, refrigerant could cool down the membrane. The valve then stops working or works erratically.
- Is there vapour or air in the liquid line? These increase the pressure drop both in the TEV and in the lines and less refrigerant passes.
- Check the condenser pressure. If this is too low, there might not be enough pressure to force the required amount of refrigerant through the valve.
- Are there instabilities that can lower the capacity?
- ***Check the superheat from the evaporator. If the superheat is larger than the design value, this indicates that the evaporator should be able to evaporate more refrigerant than what it actually does, i.e. increase the capacity***.

  The problem is then to be found outside the evaporator. In one way or other, the refrigerant flow to the evaporator is restricted. This could be a too small valve, an obstruction in the pipe, dirt in the filter, ice, the liquid receiver running dry, etc. More refrigerant cannot evaporate than what enters, and if this is insufficient, the capacity is insufficient as well.
- If the valve cannot provide the required capacity for the measured superheat and the static superheat is set to a minimum, there is no way the system could deliver more capacity. See figure 11. A small hand controlled valve in parallel could both increase the capacity and prevent hunting. See figure 15.
- Check the entrance temperature at the expansion valve. If it is substantially subcooled, e.g. by an economiser, less vapour has to evaporate than if it enters at the condensation temperature. The lower vapour content lowers the heat transfer coefficient, reducing the capacity.
- Check if liquid refrigerant leaves the evaporator. The cause of this could be a faulty distributor, a not correct cassette type, blockage of the brine side, etc. Liquid refrigerant then emerges from some channels, leaves the evaporator and causes the bulb to report a too low superheat. The valve then closes.
- The above is especially dangerous if the bulb is placed too close to the evaporator.

5.6.4. Too low suction pressure.

A too low suction pressure manifests itself as:

- A too low evaporation temperature with freezing danger as a consequence.
- In severe cases, the LP guard cuts out the compressor.

The reason for a too low pressure is incompatibilities between the compressor and evaporator capacities, the compressor sucks more vapour than the evaporator produces. This might depend on:

- A too large compressor or a compressor, which has a defect of its capacity control mechanism, e.g. on the slide of screw compressor.
- Severe maldistribution on either side of an evaporator.

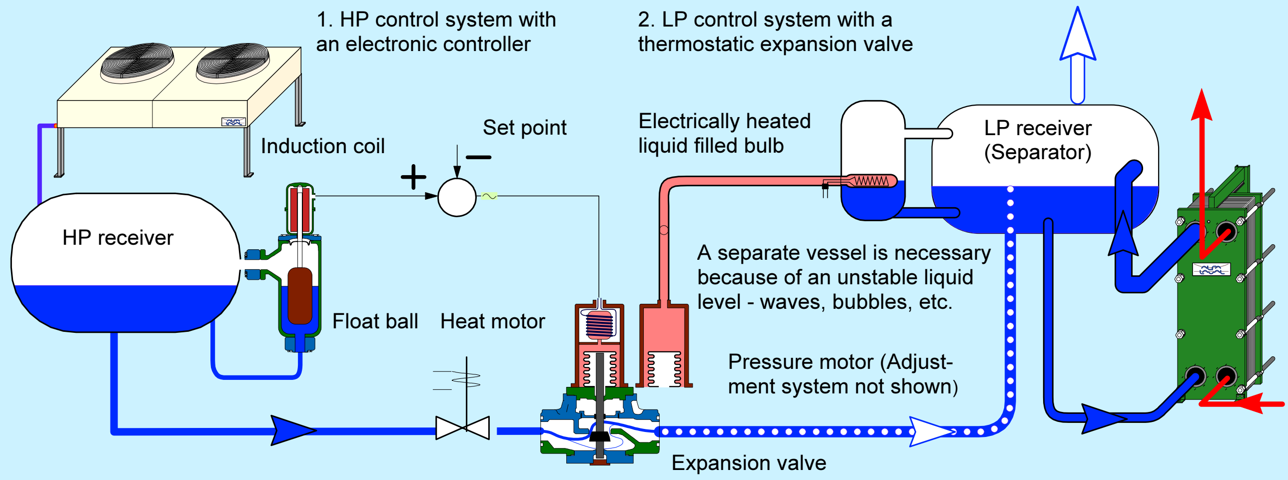


Fig. 19. Control of the expansion valve in a flooded evaporator.

1. When the liquid ascends in the HP receiver, a float ball in the attached controller ascends too and pushes an iron rod into the induction coil. The electrical property of this then changes and the electronic controller sends pulses to the heat motor. The pressure in the liquid filling increases and the valve opens, draining the receiver.

A HP controller has the advantage of operating with a relatively calm liquid level. It is sensitive to the liquid filling in the system though. If too low, the level might not be high enough to give the required capacity; if too high it floods the LP receiver with too much liquid, which then might enter the evaporator.

2. When the liquid level ascends above the bulb, there is a net heat flux from the bulb and the pressure in the bulb drops, closing the valve. When it descends, the heat flux is to the bulb; the pressure increases, opening the valve. It is essentially a modified thermostatic expansion valve.

The LP controller is less sensitive to liquid filling than the HP controller. However, when the plant stops, the liquid level drops as the evaporator is then empty of vapour and liquid enters it. The valve opens and fills up the separator. At startup, liquid might enter the compressor. A solenoid valve, which closes at compressor shut- down, prevents refrigerant from entering the separator.

- ♦ A dirty, damaged, frozen or too small TEV.
- ♦ Far too small condensing pressure.
- ♦ Vapour in the condensate line, before the TEV.
- ♦ A TEV with a leakage through the equalization line.

Example. The LP guard cut out the compressor in a DX-evaporation plant after about 90 seconds operation. A thorough inspection did not reveal any anomalies.

A thermal camera showed that the temperature decreased after the expansion valve and at the evaporator inlet the first seconds after startup. No temperature decrease was detected at the evaporator exit, but curious enough, in the suction line far away from the evaporator.

Finally, a picture taken of the equalization line inlet to the suction line, revealed the cause. The inlet was vertical, to the top of the horizontal suction line.

At startup, the image was a low contrast orange, indicating an even temperature. After a while, the equalization line inlet became blue, below zero, followed by the bottom of the suction line, just below the inlet.

A study of the TEV showed that in this particular valve, the high pressure condensate was on the other side of the stuffing box for the push rod. (As it would be, with reversed flow, in compartment C in fig. 09, p. 83.)

A leakage in the stuffing box would lead the high pressure to the underside of the diaphragm and close the valve. At the equalization inlet to the suction line there was a restriction, i.e. the pressure didn't decrease until there. A change of the valve solved the problem.

- ♦ A too small, a too fouled or an evaporator with a too high pressure drop.
- ♦ Lack of brine. Obviously, if there is no brine entering the evaporator no vapour is produced.

5.7. Non-refrigerant BPHEs.

These should not be used as evaporators because they:

- ♦ usually have an equal number of channels on each side (unequal number of plates), i.e. one channel on each side will be at either extreme of the plate pack, adjacent to the cover.

  This channel has a heat transferring plate on only one side. Then, not all the refrigerant will evaporate, and most likely liquid refrigerant will leave from this channel and the total superheat will decrease. This is especially pronounced for a BPHE with less than 30 plates and if the "half" end channel is the one close to the exit nozzle. See however § 6.2.

  Decreased superheat means that the expansion valve will close, letting through less refrigerant. The capacity regulator then tries to decrease the evaporation temperature and thus presenting a ***freezing risk.***

- ♦ sometimes has an open space between the first corrugated plate and the end plate. If the corrugation is pointed downward, water collects here and freezes; Freeze figure 01C.
- ♦ have four threaded connections. These are more difficult to braze as the larger mass of a threaded connection means a danger of overheating.

6. Control.

6.1. Refrigerant expansion in a flooded evaporator.

The expansion of the refrigerant is seldom controlled by the superheat as in the direct expansion evaporator. There is simply no superheat or temperature difference between exit and inlet to use as control variable.

The expansion valve is instead controlled by the liquid level in either the HP or LP liquid receiver. The HP controller should open the valve for increasing liquid level and the LP controller should close it. The level control could be mechanical, thermal, light absorption, etc. See figures 2B, 06K and 19 for some conceptual designs.

A HP controller is normally used if there is only one evaporator and LP controllers if there are many parallel evaporators (each one with its own controller).

In contrast to the direct expansion evaporator, a flooded evaporator could be installed with a pulse width modulated expansion valve. The separator acts as a giant dampening device, which will even out valve fluctuations.

Flooded flow is often used with SWPHEs, which in any case are more insensitive to pulsation.

6.2. The electronic expansion valve.

The simple thermostatic expansion valve as described in § 5 of this section has some drawbacks.

One is that it is a proportional control device, i.e. the output is proportional to the input. In practice, it means that the controlled variable, the superheat, can only be maintained for one capacity. When this changes, so does the superheat. The superheat thus has to be large enough for the compressor to tolerate variations of it; up or down.

An evaporator operates better, however, with a smaller superheat, because the heat transfer coefficient is low in the superheating zone. This has led to the development of electronic control systems, which give more control over the expansion valve.

- One example is the PI (Proportional Integral) control system in which the superheat is measured and compared with a set value. Sometimes it is not the true superheat, which is measured, but the difference between the outlet and the inlet temperatures. See figure 20.
- The valve can be driven by either a pneumatic, hydraulic, thermal, magnetic, or step motor.

  Avoid pulse width modulated valves, i.e. where the valve stays open for a certain period and then closes. The width - time - of the opening determines the capacity. The combination of thermal and mechanical shocks could destroy a brazed or an all-welded evaporator. They can however, be used for the more shock resistant semiwelded and gasketed plate heat exchangers.
- The valve opening has no direct relationship to the temperature difference. This makes it possible to design a valve which moves to the operating point (see figure 14) as rapidly as possibly. Once there, it adjusts (lower) the response to avoid hunting (see figure 16).
- As the saturation temperature corresponding to the exit pressure is difficult to obtain, electronic valves do not usually use the superheat but the total temperature difference between the inlet and the exit.

  The electronic controller in figure 20 has the difference, 1 K, as the control variable. The TEV would use the true superheat, 5 K. The difference between the two is the evaporator pressure drop.

  Suppose that the load decreases to 25 % of the nominal capacity. The pressure drop then decreases to a little more than 0.5 K. The refrigerant thus enters at -3.5 °C and leaves saturated at -4 °C.

  The electronic controller will adjust the valve until a difference of 1 K between exit and inlet is obtained and the refrigerant leaves at -2.5 °C, superheated to only 1.5 K instead of 5 K. This might be a too small difference to assure a vapour completely free of liquid refrigerant.
- In such a case, it is better to use the true superheat as the control parameter for the controller. Instead of the inlet temperature to the evaporator, the exit saturation pressure can be used and converted to a temperature.
- The use of the true temperatures, i.e. not converted from the pressure, to control the valve has one specific advantage: The valve will be refrigerant independent. The question is how to measure the saturation temperature at the same point as the superheat temperature, usually at the exit. Figure 21 gives one idea.

6.3. Matching the evaporator and compressor capacities.

The normal refrigeration compressor is of the positive displacement type, i.e. vapour enters a confined space in the compressor. This space is gradually reduced and the vapour is discharged at a lower volume but at a higher pressure (and temperature).

The compressed volume is independent of the state of the refrigerant vapour; if the compressor sucks 1 litre/s at 3 bar and discharges it at 12 bar, then it still sucks 1 litre/s at 2 bar and discharges it at 8 bar. As the vapour density decreases with decreasing pressure, the mass flow refrigerant decreases with decreasing pressure.

The cooling capacity is not dependent on the amount of vapour sucked into the compressor but on the amount of liquid refrigerant remaining after the expansion. More refrigerant has to expand, thus less liquid remains, the larger the difference between the condensation and evaporation pressures is. The two effects - the decreasing mass flow and the decreasing liquid content - mean that the capacity decreases with decreasing suction and increasing discharge pressure of the compressor.

The capacity of an evaporator increases with increasing temperature difference between the media, i.e. the capacity increases with decreasing evaporation temperature.

So, what happens when we connect an evaporator, operating at a certain evaporation temperature, to a compressor with a larger capacity at this temperature?

Since the compressor has a larger, capacity than the evaporator it can handle more vapour than the evaporator can deliver, i.e. the compressor starts to evacuate the evaporator and the pressure decreases.

Decreasing pressure has opposite effects on the compressor and the evaporator:

- The evaporator evaporates more refrigerant, i.e. the capacity increases and the vapour density decreases. These two effects increase the total vapour volume.

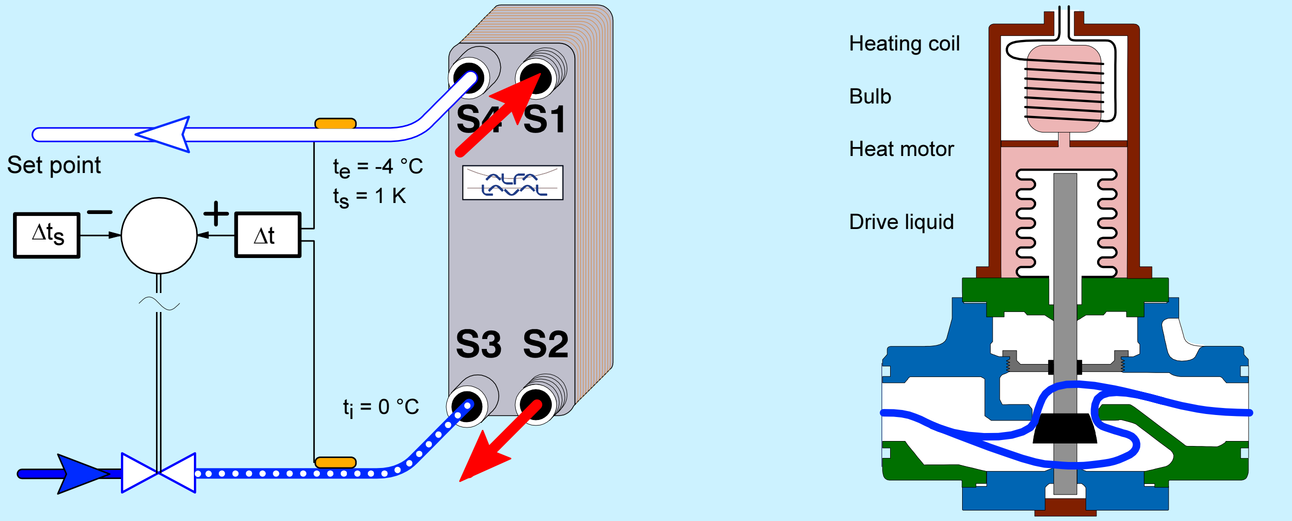


Fig. 20. The electronic expansion valve.

Saturated refrigerant at 0 °C enters the evaporator. Here the temperature falls 4 K due to the pressure loss, then superheats 5 K and finally, the refrigerant leaves at 1 °C, at a saturation temperature of -4 °C.

The controller measures the temperature difference between the exit and inlet, Δt = 1 - 0 = 1 K, and compares this with the set point. As long as there is a difference. the controller sends signals to the valve to open or close. In case of the thermal motor, these signals are current pulses of various length to the heating coil. The liquid in the bulb heats up or cools down, resulting in pressure changes, which are transformed to valve movements.

The design here is conceptual. A real valve would have a circuit for controlling the temperature of the bulb liquid.

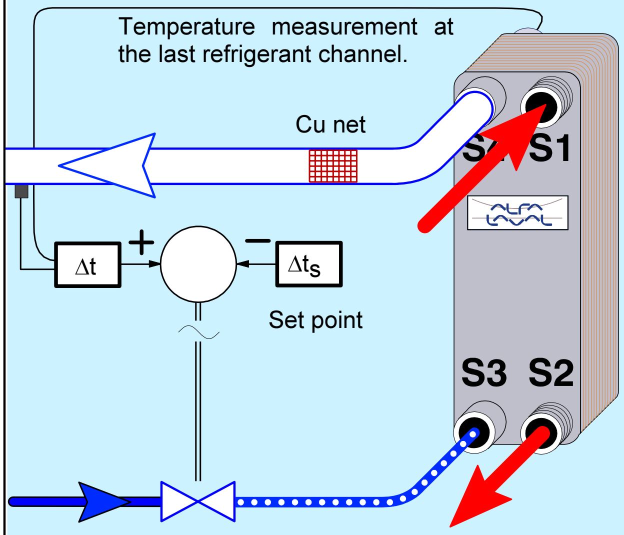


Fig. 21. Temperature measurements.

The last channel in this evaporator is on the refrigerant side. Some unevaporated refrigerant (see § 5.7) then leaves the channel. At the channel exit, the refrigerant is thus saturated and provides the saturation temperature.

There is a small pressure drop from this point to the superheated temperature point on the exit tube, although this is usually a matter of tenths of a degree.

Unevaporated refrigerant can be separated in a suction line separator or as shown above, when it passes the hot wires in the roll of a Cu net.

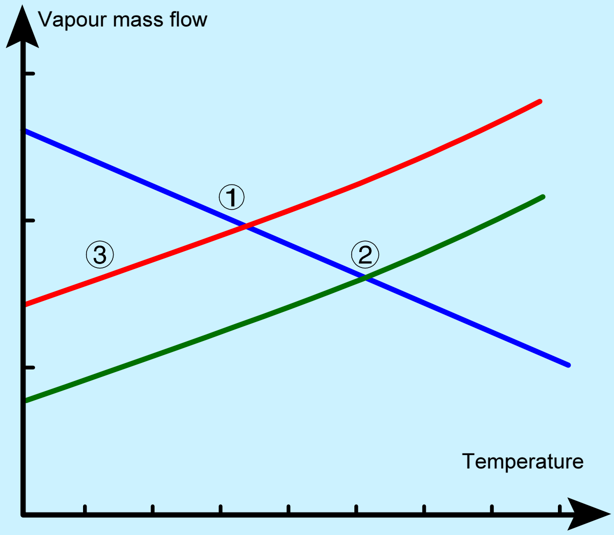


Fig. 22. Matching the evaporator & compressor capacities.

In the absence of a pressure controller, the operating point falls at (1). If, at this point there is a danger of freezing, the pressure has to be kept at a higher level.

A pressure controller (figure 23), maintains a pressure (2) higher than the freezing pressure in the evaporator.

The vapour flow from the evaporator at this point (2), is what the compressor gives at (3), and the pressure stabilizes at this point in the suction line.

- The compressor sucks a decreasing amount of refrigerant in kg/s, i.e. a decreasing capacity, due to the lower vapour density at the suction line.

At a certain pressure, there is a match between the vapour volume leaving the evaporator and the space compressed by the compressor, i.e. a match between the compressor and evaporator capacities. See figure 22.

If the compressor capacity increases relative to the evaporator capacity, temperature and pressure drop and vice versa. The equilibrium falls at the match between the compressor and evaporator capacities.

6.4. Capacity control.

The evaporator capacity can be controlled by:

1) The area. The only practical proposition is to (dis)connect a circuit in a multi-circuit evaporator. The area can thus only be changed in relatively large steps.
2) The temperature difference. By changing the flow rate of the cooled fluid, the exit temperature and thus the temperature difference can be changed. However, the exit temperature of an evaporator is usually fixed; it is the controlled, not the controlling, variable.
3) The evaporation temperature gives a better option to change the MTD. There might be a danger that the evaporation temperature decreases below the freezing point of the liquid to be cooled.
4) The heat transfer coefficient. In principle it would be possible to change the circulation rate in a thermosiphon evaporator and the superheat in a direct expansion evaporator and thus to obtain a change of the over-all heat transfer coefficient.

Both methods have some drawbacks, though. The heat transfer coefficient is not a clear function of the circulation rate and changing the superheat would interfere with the operation of the expansion valve or could lead to a compressor discharge temperature that is too high.

The compressor capacity can be controlled by:

5) On/off. The advantage of this is that the evaporator operates with the nominal (maximum) flow rate when operating. This is important to keep oil moving through the evaporator. See **8. Oils & Refrigerants, § 3.3**.
6) Change of the compressed volume. This is easily done in a screw compressor. In a piston compressor it can be achieved by lifting the cylinder head (valve plate).
7) Changing the number of revolutions, i.e. through a thyristor controlled motor or, even better, a D/C motor.

Others. There are also some very important methods which are neither evaporator nor compressor controls.

8) Changing the vapour volume from the evaporator, by a backpressure controller. The evaporator pressure can exceed but not be lower than the set pressure. The suction pressure is undefined, but matched to the capacity of the compressor. See figures 22 & 24.
9) Recirculating vapour from the compressor discharge line to the suction line or to before the evaporator. A decrease of the vapour volume from the evaporator is balanced by the additional injection of refrigerant.

   Numerous variations on this theme exist. Gas from the discharge line can be injected into the suction line or before the evaporator, vapour from the HP receiver, or liquid refrigerant from the HP receiver together with vapour can be used, etc.

- 8) & 9) are expensive, though. The compressor operates against a constant low pressure and no power saving is made. They are thus used mainly in small DX systems.
- 6) & 7) are more expensive to install but provide opportunities for power saving. These are the most common in large flooded systems.
- The most interesting method for good evaporator performance, is injection of vapour between the expansion valve and the evaporator, 10) and on-off 5). This keeps the vapour velocity more or less constant in the evaporator, which is important for a secure oil flow through the evaporator, figures 23.

6.5. Pressure control.

Pressure control is intimately connected to capacity control as in both the evaporator and the compressor the capacity is related to the pressure. The capacity control methods in § 6.4 are then also pressure control methods

The direct expansion evaporator. Matching the refrigerant volume from the evaporator to the compressor by a pressure regulator as described in § 6.4 mainly controls the pressure. 2) above and figures 24 & 25

The flooded evaporator. The methods described in § 6.4 above can all be used, but with some preference for changing the compressor capacity as a method for controlling the pressure. The simple pressure controller in figure 24 can be used for lower capacities. For larger capacities, pilot valves are used.

6.6. Liquid temperature control.

The exit temperature of the cooled liquid is controlled by :

- ***The evaporation pressure*** is controlled by the liquid exit temperature. If this decreases, indicating a decreased cooling demand, the evaporation temperature increases, in order to reduce the evaporator capacity.

  See figures 25 and **8. Oils & Refrigerants,** figure 09.
- ***Sending the liquid through a bypass***, figure 26. Cooled and uncooled liquid then mix to the correct temperature.

  If the exit temperature starts to decrease, indicating a reduced cooling demand, the bypass valve switches liquid from the evaporator to the bypass, to obtain the correct exit temperature. The temperature from the evaporator decreases and so does the temperature difference, and less refrigerant evaporates.

  This, of course, means an imbalance between the compressor and the evaporator capacities and the refrigerant pressure and temperature start to drop, i.e. a danger of freezing arises if none of the pressure/capacity control measures, described above, are applied.
- ***On-off control.*** In many cases, the liquid temperature is allowed to vary a few degrees. If a cooled liquid is sent to various recipients, there might be an appreciable amount of liquid and thus inertia in the system.

  The simplest system here is to close the plant when a given temperature has been reached and start it again when the temperature has increased somewhat. As in all on-off systems, care should be taken to avoid pressure and thermal shocks.

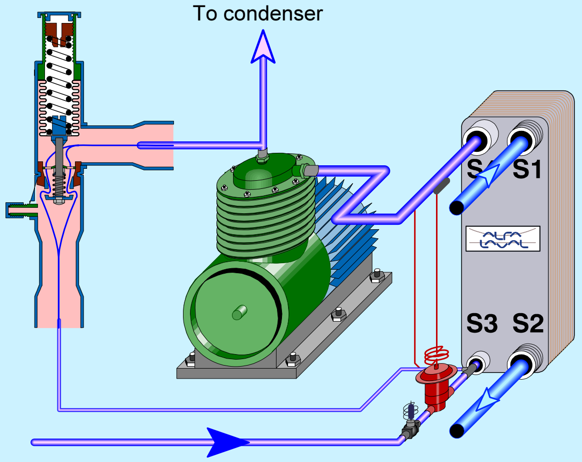
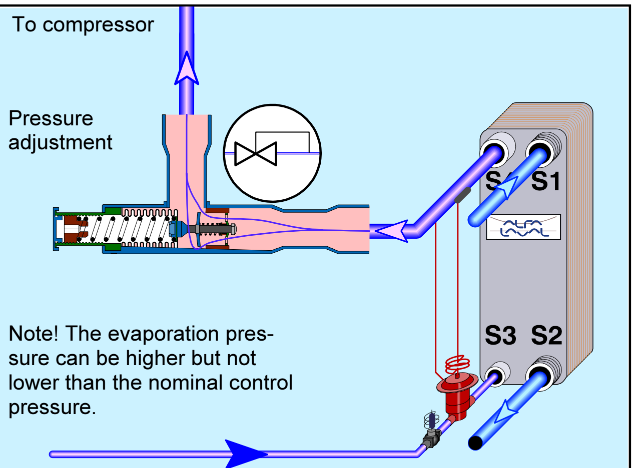


Fig. 23. Hot gas bypass control of a direct expansion evaporator.

Decreasing pressure at the compressor entrance (or at the evaporator exit) signifies that the evaporator load is decreasing. This means a decreasing vapour velocity as well, i.e. a danger of oil retention in the evaporator.

The decreasing pressure actuates the bypass valve. This opens and injects hot gas between the expansion valve and evaporator, i.e. the lower vapour production is offset by the vapour injection and the channel vapour velocity remains more or less constant.

Use a proper hot gas by-pass injector otherwise maldistribution of the refrigerant might occur.



Note! The evaporation pressure can be higher but not lower than the nominal control pressure.

Fig. 24. The evaporation pressure controller.

When the evaporator pressure drops, the valve starts to close. The valve opening depends only on the inlet pressure. The valve throttles the vapour from the evaporator pressure to the compressor pressure, which then can operate on a lower level than the evaporator.

The compressor pressure is uncontrolled, though.

Put differently, the vapour from the evaporator is expanded to the volume needed by the compressor.

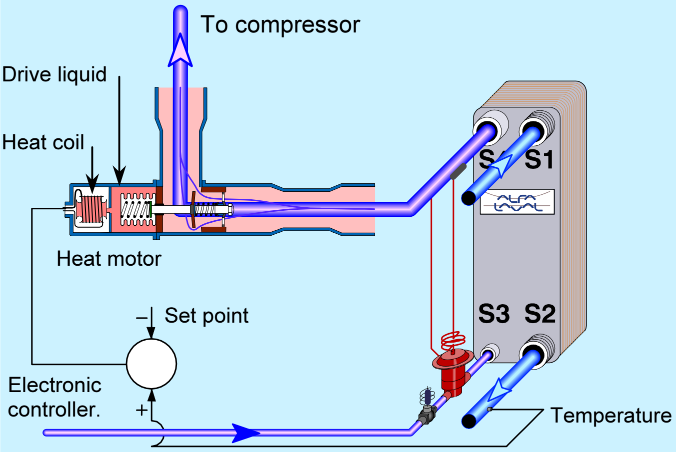


Fig. 25. The evaporation pressure controller with an electronically controlled set point.

This is basically the same as in figure 23 but instead of manual control of the pressure, the pressure is controlled by the exit temperature of the cooled fluid.

Another option is to control the temperature difference between the inlet temperature of the cooled fluid and the evaporation temperature in order to avoid excessive thermal shocks, e.g. at oil cooling.

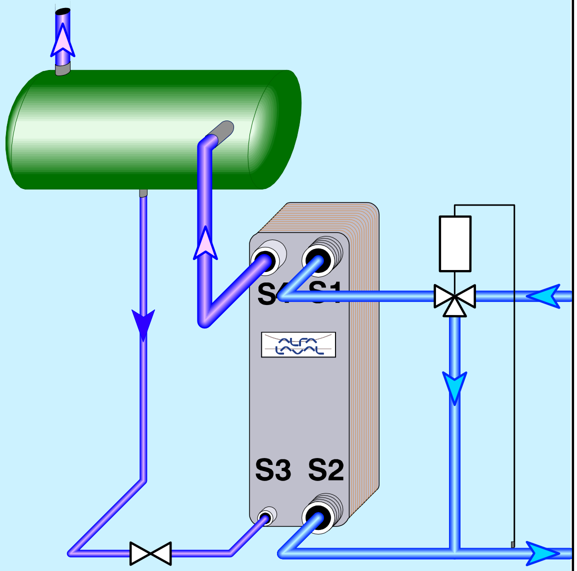


Fig. 26. Temperature control in a thermosiphon evaporator.

The evaporation temperature is controlled by any of the methods described in the text, i.e. by matching the evaporator capacity to the compressor capacity.

The temperature of the cooled fluid is then controlled by by-passing part of the cooled fluid. A decreasing inlet temperature means that less cooling is necessary and the valve then switches more liquid through the bypass. The liquid in the evaporator then cools down more, and the effective temperature difference decreases.

- ***Control through a valve.*** The system is a hybrid between a flooded and a DX evaporator, see figure 28.

  A control valve is installed in the thermosiphon loop. Obviously, the refrigerant cannot be flashed to any great extent, as the pressure difference over the valve corresponds to the maximum static height.

  The valve can either be controlled by the superheat as in a normal direct expansion evaporator or as shown in the figure, by the exit temperature of the cooled liquid.

  The system is sometimes used for auxiliary evaporators such as oil coolers (**8. Oils & Refrigerants,** figure 8), oil evaporators (**1. Applications**, figure 19), etc.

  The advantage is that the vapour emerges dry, ready to enter the compressor, and there is still no need for the large pressure difference required for a normal thermostatic expansion valve.

  The simplest system is a solenoid (on-off) valve.

  A variant of this is a pulse width modulated valve. These should normally not be used together with BPHE's as expansion valves as the pressure shocks can damage the BPHE. In this case, there are almost no shocks. If the temperature difference between the refrigerant and the cooled fluid is not too large (< 30 °C) the thermal shocks will be negligible.

6.7. Condenser/Evaporator.

In a BPHE with condensing refrigerant on one side and evaporating refrigerant on the other, there is a danger of rapid temperature changes because of the excellent heat transfer, due to the phase changes, on both sides. The danger is especially large if there are compressors in parallel on either side. To minimize the risk of damage due to thermal shocks, the control system should, especially if there are large temperature differences between the sides, be laid out as below and in figure 27:

a) At startup, the condensing side should, if possible, be put into operation at the same time or after, but not before, the refrigeration side. See b) below.

   If the vapour line from the LT compressor is sufficient long there will automatically be a delay before the hot vapour enters the E/C.

   If the condensing refrigerant enters first, the BPHE heats up to the superheat temperature and the hot BPHE then receives the cold, evaporating refrigerant. Contrarily, a BPHE of the evaporation temperature will effectively receive refrigerant of only the condensation temperature, i.e. a much lower temperature difference.

b) A suction line liquid separator (see § 7.) could be installed to handle an initial liquid refrigerant surge from the evaporator if an arrangement as in a) is made.

c) An evaporation pressure controller should be installed, especially in a multi-compressor installation. If the C/E (Condenser/Evaporator) is one of a number of parallel evaporators, e.g. unit coolers, the temperature setting should be somewhat higher than for the rest of the evaporators.

   The purpose is to insulate the C/E from rapid pressure changes, which can occur when a compressor cuts in or out. If the C/E then operates at a low capacity, the superheat is likely to be low. If a compressor on the evaporating side suddenly cuts in, the pressure drops. The lower pressure is immediately transmitted to the TEV while a possible temperature change takes longer time. The valve then senses a higher superheat and opens, hunting maybe starts, and at the exit, the superheat disappears and liquid refrigerant hits the relative hot walls causing thermal fatigue. See also § 5.5.2 - 3.

d) If a condensing pressure controller is used, it should be installed before the C/E, never after. Installation after could cause condensate to back-up in the C/E. See figure 10 A & B, p. 109 (**5. Condensation and Liquid Receivers**) and e) below.

e) The condensing side should be drained properly, i.e. condensate is not allow to back-up in the E/C. See d) above and figures 05 & 06, p. 103ff (**5. Condensation and Liquid Receivers**).

f) The above also means that condensate subcooling in the E/C is not allowed.

g) When only one of many LT compressors operates, the E/C capacity will be only a part of full capacity. If only one TEV is used, this will then operate with a very low superheat and is thus susceptible to disturbances. See also § 5.5.2, p. 86.

   The arrangement in figure 27 could be used to secure an inoffensive small flow of refrigerant at part load. (Compare with **8. Oils & Refrigerants**, 8 D).

h) Mufflers and oil separators tend to even out pressure variations from the LT-compressors.

i) If possible, use slow moving solenoid valves, especially if the condensing side is started up first.

j) A flooded evaporator, which remains filled with refrigerant during a shutdown, gives fewer temperature shocks than a DX evaporator.

7. Suction line separators.

Despite all precautions to prevent liquid refrigerant from entering the compressor, there might be occasions when the evaporator simply cannot cope with a sudden surge of refrigerant and liquid droplets leave the evaporator.

The liquid droplets can either be separated from the vapour, evaporated, or the methods can be combined.

- There are many types of liquid separators on the market. One type, designed and manufactured by Alfa Laval, is shown in figure 29 A & B.

  During a sudden surge of refrigerant, the liquid droplets leaving the evaporator collect in the separator. Here a system of pipes ensures that the liquid leaves in a steady even flow and then evaporates before the refrigerant enters the compressor.

- After the evaporator, the refrigerant passes a heat exchanger, where the vapour is heated. The liquid droplets then evaporate. Heating media is normally condensate. A BPHE could be used for this. A disadvantage is that the refrigerant could leave the compressor too superheated, i.e. a danger of oil breakdown.

The two methods could be combined, e.g. with either separate heat exchanger and separator or combined as is shown in figure 29 C.

The heat exchanger should be sufficient large to evaporate all the refrigerant droplets. In one large industrial system, a heating coil was underdimensioned with the result that liquid remained in the separator. Eventually, the system was drained of refrigerant. Compounding the problem was the fact that the TEV temperature bulb was placed immediately after the evaporator. It would have been better to install it after the separator.

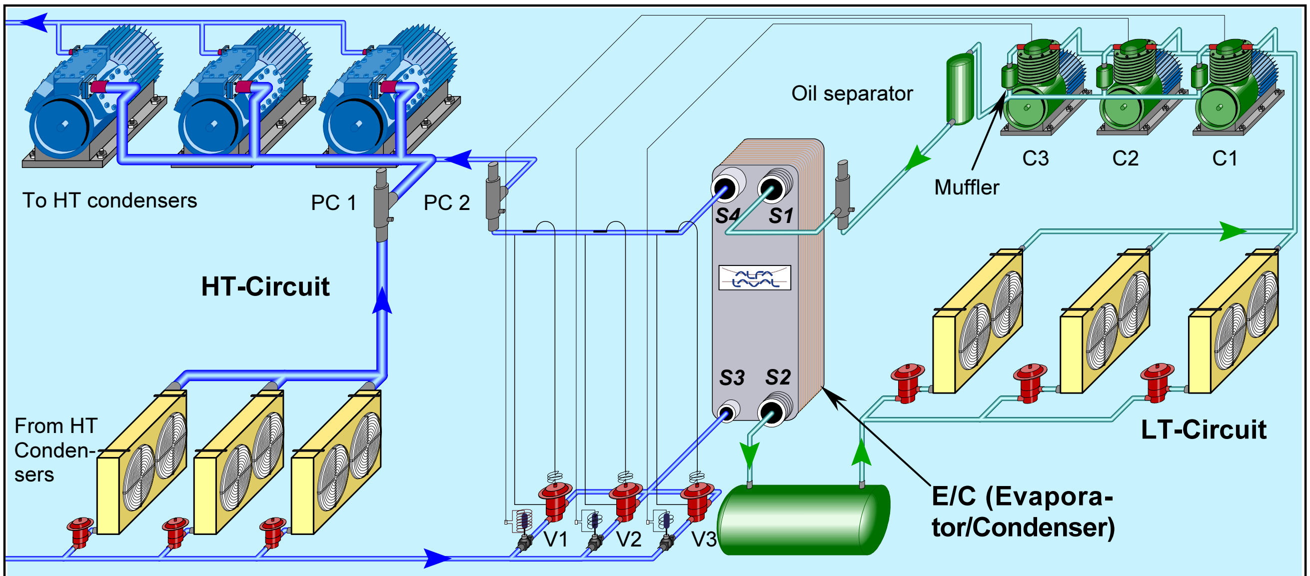


Fig. 27. Control of the evaporation in a condenser/evaporator at a cascade system.

The above system is common in supermarkets. The LT vapour from three compressors condenses in a E/C cooled by liquid refrigerant from the HT circuit. The E/C usually operates in parallel with the HT unit coolers.

When only one LT compressor operates, a single TEV would be susceptible to pressure fluctuations when a HP compressor cuts in or out. To minimize fluctuations the following system could be used:

- Instead of one TEV, three 1/3 capacity TEVs, each controlled by its own compressor, are installed.
- The condensing side should be properly drained.
- An alternative is one 1/3 TEV and one 2/3 capacity TEV. Depending on the number of compressors operating, one or both TEVs are active.
- In stead of a small TEV a fixed valve, which gives about 1/3 capacity could conceivably be used
- The pressure setting of the evaporation pressure controller PC2 should be higher than PC1 Or than the actual HT suction pressure).
- A possible condensing pressure controller should be installed before not after the E/C.
- Subcooling is not allowed in the E/C

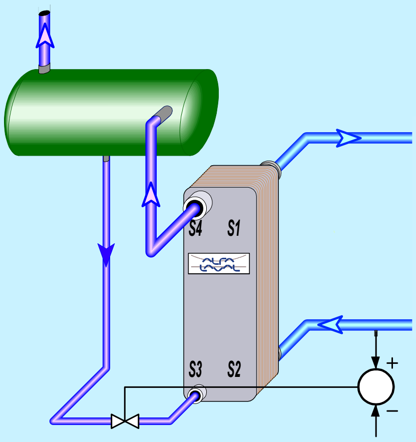
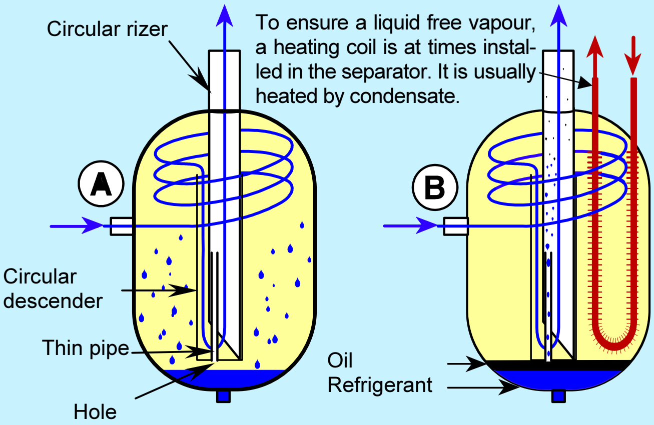


Fig. 28. Temperature control with control valve in a flooded evaporator.

Here the temperature is controlled by throttling a valve, similar to a thermostatic expansion valve, controlled by the exit temperature of the cooled liquid.

The difference from a thermostatic expansion valve is that there is almost no expansion in the valve. It simply lets through enough liquid to vaporize in the evaporator. The control action is either:

- Continuous and depends on a combination of a reduction in the heat transfer coefficient and in the area.
- On/off control. This should not be used for large temperature differences (> 30 K) between liquid and refrigerant.



To ensure a liquid free vapour, a heating coil is at times installed in the separator. It is usually heated by condensate.

Fig. 29. Suction line liquid separator.

A. Vapour, charged with droplets of refrigerant and oil, enters the separator. The droplets separate and collect at the bottom of the separator. The vapour then enters the descending pipe, turns at the bottom of it and leaves the separator from the rising tube.

B. When the levels of oil and refrigerant have reached the bottom of the descender, they enter the thin pipe through the small hole. The cross sections of the various pipes are calculated to give a certain pressure drop between the liquid level and the mouth of the thin pipe.

This pressure then forces the liquid into the exit pipe, where the liquid refrigerant flashes and vaporizes and the oil is dispersed and entrained by the vapour.

5. Condensation & Liquid receivers.

1. The condensation process.

When a superheated vapour cools in a heat exchanger, the following steps occur (figure 01):

- ***Vapour desuperheating.*** This is normal gas cooling.
- ***Condensation on the wall.*** A certain subcooling of the vapour is actually necessary. The condensation can only start if some suitable nucleus is present. This can be a heat exchanger surface, a dust particle, etc. Note the similarity to boiling.
- ***Subcooling of the condensate.***

The condensation is classified according to the type:

- ***Drop wise condensation*** occurs when the liquid does not wet the surface. The condensate forms droplets of - liquid on the heat exchanger surface. When large enough they coalesce into larger agglomerates, which will flow off the surface.
- ***Film condensation***. When the liquid wets the surface, a continuous descending liquid film forms. The film flow can be either laminar or turbulent. The heat transfer coefficient for laminar film condensation decreases with increasing flow until turbulence is reached, when the heat transfer coefficients start to increase again.

Drop wise condensation has a much higher heat transfer coefficient than film condensation. Unfortunately, it is difficult to maintain a steady drop wise condensation, so it is difficult to design a condenser with this condensation mode. In certain cases - e.g. condensation on Teflon - it is possible to achieve a steady drop wise condensation. Conceivably, condensation of a vapour on an oily surface, which is not wetted by the condensate, could obtain drop wise condensation, e.g., an ammonia condenser in a refrigeration plant with insoluble oil.

2. Condenser arrangement.

The selection of condenser type, cooling media, temperature level, control system, etc., depends very much on the purpose of the plant and is beyond the scope of this manual. Below are some indications:

A) If the condenser is installed in an A/C or refrigeration plant without heat recovery, the only purpose of the condenser is to dump the heat in a suitable receptacle. The condenser type then depends on this receptacle and has to be as small as possible for the given data. For a water-cooled PHE, the techniques in the chapter **2. Optimization** can be used to minimize the surface and the cooling water flow.

B) The correct condensing temperature or better the pressure in A) depends on the type of evaporator:

   A ***direct expansion*** system needs a minimum pressure to provide the TEV with an ample driving force.

   The driving force in a ***flooded system*** is less critical and the condensing pressure might be left to vary with the cooling media temperature, thus saving energy.

C) In a heat pump, condensation is the primary objective. Condensation might be split in a desuperheater for tap water heating and a condenser for space heating. The control system is probably arranged to maintain a specified minimum temperature level.

D) An A/C can be used to recover heat from the condenser, usually the high level heat from the superheated vapour. A special desuperheater, a water-cooled heat exchanger, is then installed. The control system might be arranged to maintain a minimum refrigerant temperature to give the required water temperature.

3. Design considerations.

3.1. Temperature profile.

In a condenser, the refrigerant enters superheated, with temperatures ranging from about 60 to 120 °C. Higher temperatures than 120 °C, though possible, are seldom encountered as oil in the compressor then starts to decompose and the system is, consequently, designed to keep the discharge temperature below this value.

As in all heat exchangers, the cold side can only approach but never entirely reach the hot side temperature. When the exit water temperature increases by decreasing the water flow rate, the water temperature curve approaches the vapour curve, not at the end temperature but at the point where the vapour starts to condense.

The water exit temperature is still far from the vapour entrance temperature, see figure 1, which shows the desuperheating and condensing of 1 kg/s R22 vapour entering at 87 °C, condensing at 45 °C and leaving subcooled at 40 °C. 3.28 kg/s water is heated 25 to 40 °C.

3.2. Heat recovery.

The total capacity in figure 1 is about 206 kW. It is tempting to recover this heat, say for tap water heating. Unfortunately, only a part of the total condenser load is available for heating of the water to temperatures nearing the vapour inlet temperature.

In order for the hot water to become suitable for use as tap water, it has to be heated to around 80 °C. In the above example, the water is heated to 40 °C. What happens if we try to lower the water flow, in order to increase the exit temperature?

Figure 1 shows clearly that the exit temperature is limited by the temperature pinch point. What is then the highest exit temperature? The following relationships is valid:

$$\frac{\Delta t_{cond.}}{\Delta t_{total}} = \frac{q_{cond.}}{Q_{total}}$$

Δt_{cond} = 45 - 25 = 20 K (The temperature increase in the condensing section.)

$q_{cond.}$ = 167 kW (The thermodynamic design.)

q_{total} = 206 kW (The thermodynamic design.)

thus, Δt_{total} = 24.7 K and an exit temperature of 49.7 °C, corresponding to a flow rate of 3.28 * 15 / 24.7 = 2.0 kg/s.

Note that this is the limit for an infinitely large condenser. An even lower flow rate would mean a higher temperature of the water, which is clearly impossible.

The nominal water flow, 3.28 kg/s, gives a temperature of 37.5 °C at the pinch point, which is somewhat below the practical limit for a heat exchanger.

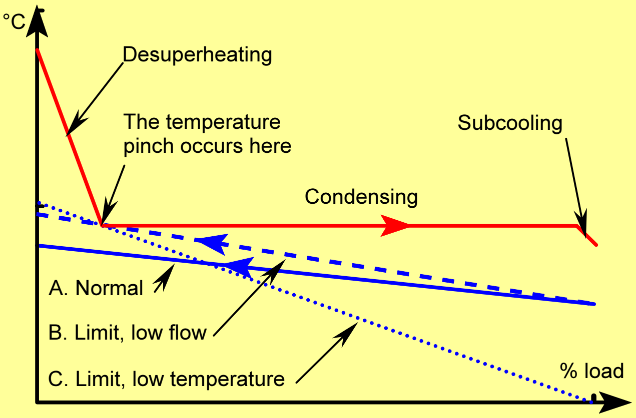
In practice, a decrease of the water flow means that the condenser will not condense everything. That means moving the pinch point to the right to clear the water curve. This applies regardless of the size of heat exchanger.

Fig. 01. The condensing temperature profile.

A. A “normal” temperature programme. The temperature pinch should not be closer than a couple of degrees.

B. When the water flow decreases, the exit temperature increases, but long before it approaches the vapour temperature, there is a temperature pinch at the point where the vapour starts to condense.

C. It is somewhat ironic that the colder the water inlet temperature is, the higher the exit temperature can be.



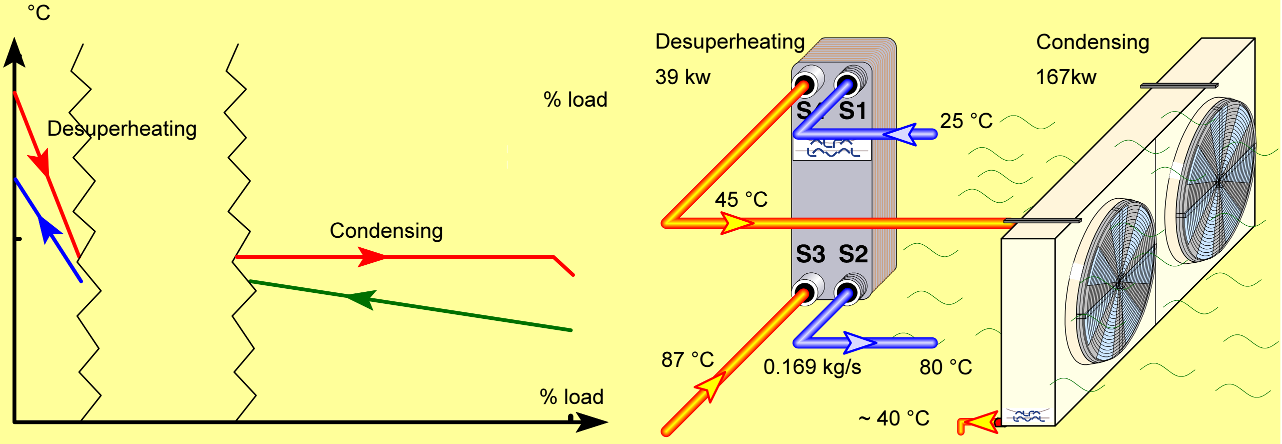


Fig. 02. Desuperheater & condenser.

One way to obtain higher temperatures, necessary in e.g. tap water heating, is to split the desuperheating and condensing duties.

The desuperheater heats a smaller amount of water to some 80 °C in a PHE and the now saturated vapour is condensed in a separate condenser.

An air unit is used here, but a PHE could be used as well depending on how (and if) the latent is used.

Desuperheaters and condensers of the same PHE model - either brazed, welded or semiwelded - could be integrated as shown in **1. Applications**, figure 10 or in **3. Design & Installation**, figure 8D.

If a higher water temperature is necessary, there are two possibilities, independent or in combination.

- ♦ If water flow of the order of 3.28 kg/s is necessary, the condenser could be used to preheat the water. In winter, when the water temperature approaches zero, that could mean a substantial energy saving.
- ♦ A separate desuperheater is installed, figure 2. A BPHE is excellent for this application. Here a small water flow is heated to 80 °C and the remaining latent heat is cooled in either a water or air cooled condenser.

In a PHE the pressure drop is normally, large enough to allow for upward vapour flow. The inevitable condensation means that droplets (liquid refrigerant and oil) have to move upward and a certain minimum pressure drop is necessary, in the order of 1 - 3 kPa/m flow length.

3.3. Inerts.

When a vapour containing an inert gas condenses in a condenser the following phenomena occurs, figure 03

a) The volume flow of the vapour-gas mixture decreases and the heat transfer coefficient decreases.

b) The vapour layer close to the wall is progressively enriched with gas as condensation proceeds. This gas film acts as a barrier to the vapour. The vapour has no direct access to the cold surface to condense; it first has to diffuse through the inert gas layer.

c) The partial pressure of the vapour decreases and the vapour-gas mixture has to be cooled down to remain saturated and for condensation to proceed.

d) If the inert is not vented, the pressure will increase until the HP pressostat cuts out the compressor.

In practice this means that even small amounts of noncondensable gases (inerts) could reduce the heat transfer coefficient and the MTD considerably or worse, impede the operation of the plant altogether.

Although noncondensable gases should not normally be found in a refrigeration circuit, it can sometimes happen.

1) Breakdown of the refrigerant, especially ammonia or oil.
2) A faulty vacuum pump.
3) The plant not being properly evacuated at startup.

Example I: In an ammonia condenser at a refrigeration plant for a skating rink, the pressure increased and finally the compressor was shut off whenever the second of two screw compressors was started.

A check showed that the condensate left - within a tenth of a degree – at the same temperature as the cooling water. Such an extreme subcooling – about 13 K - could only mean either a condensate back-up (see next paragraph) or an inert gas plug in the condenser. Both decrease the effective area for condensing and cause the condensing pressure to increase.

According to the plant manager, the expansion valve was amply sized and the plant thoroughly vented.

In the absence of sightglasses it can be difficult to determine which of the two was the culprit.

However, the suction and discharge pressure manometers, vibrated heavily. Especially the later was almost impossible to read. The third manometer, the oil pressure was completely stable. All were of the same type, had the same pressure range and were mounted side by side, close to the compressor. The stable oil manometer ruled out the cause as vibrations from the compressor.

A vibrating discharge manometer hints of inerts.

A check of the condensate filling would mean a lot of work with the piping while the simpler check with venteing could be done immediately without stoping the plant.

The condenser was vented from the subsequent liquid receiver by simple connecting one end of a hose to the vent and the other into a bucket of water with cold water. Ammonia dissolves readily in water while the air bubbles rise to the surface.

There was an initial very large air flow, which after about half an hour decreased to a stable value, which then continued for about five hours.

After the initial surge of air, the manometers became stable and after the five hours both compressors could be connected. The subcooling was then normal, about 3 K.

4) Leakage into an evaporator operating at vacuum.
5) In once-through installations, see example below.

Example II: A WPHE condensing propane was troubled by frequents cutouts of the HP pressostat. Inerts were suspected but an analyser revealed only 100 % propane.

A closer investigation revealed that this was not a true refrigeration plant. The source of the propane vapour was the inevitable vaporization of propane stored in a commercial tank farm. This propane vapour is compressed, liquefied and returned to the tank storage.

When the tank is emptied, the space above the propane is filled with an inert gas, normally nitrogen. The propane vapour to be compressed thus contains nitrogen.

Furthermore, the analyser was actually for butane but the reading was converted to propane according to the formula X % butane = Y % propane. Unfortunately, it is impossible in this way to distinguish between 100 % or say 99.9 % propane. (A nitrogen analyser would have been better; the difference between 0 % and 0.1 % is large.)

Installation of a venting system solved the problem.

Inerts can have a dramatic effect on capacity. The problem is especially acute in a condenser with condensate level control (see below). Here, the inerts are effectively trapped in the condenser, see figure 04, p. 103.

3.4. Condensate subcooling.

The condensate in a condenser is normally subcooled a few degrees. If more subcooling is required, this should be done in a separate subcooler, not in the condenser as the only way to cool the condensate substantially below the saturation temperature in the condenser is to operate it with a condensate level, see figure 04. It is not impossible to cool the condensate in the condenser, especially with an upwards loop as shown in the figure but:

- Inerts, insoluble light oil, decomposition products could be trapped at the vapour-liquid interface. See figure 04.
- The condensate temperature depends on the condensate level, controlled by a valve at the exit but the surface available for condensation changes simultaneously. As the condensing heat transfer coefficient is much larger than the subcooling coefficient, a small change of the subcooling could mean a large change in the condensing part. It is thus impossible to control the temperature by a simple level control without affecting the capacity.
- The level could be kept constant and the condensate temperature controlled by the water flow but this too would affect the condensing part.
- It is impossible to control the capacity by changing the condensate level or the water flow without also affecting the temperature but this is usually less serious.
- The dynamic response when decreasing the condensate level is more rapid than when increasing the level. Decreasing, i.e. draining is almost immediate, while increasing could be slower, as sufficient refrigerant has to condense in order to build up the required condensate level. This could cause control problems.
- The condensate level might be different during summer and winter operation and the refrigerant difference has to be stored in the liquid receiver. This increases the total filling of refrigerant, which is expensive.
- The valve used to control the condensate level has to be installed after the condenser, but as close to it as possible. Otherwise, the volume in the pipe from the condenser to the valve unnecessarily increases the total refrigerant content while at the same time causing a delay when the condensate level is increased.
- ***Condensate cooling in the condenser should never be made for ammonia and water.*** The high latent and specific heats, together with the high thermal conductivity and low viscosity lead to very high heat transfer and rapid temperature changes, which could cause thermal fatigue. In general, ammonia and water condenser should be drained as completely and rapidly as possible. Note the similarities to oil cooling, see **8. Oil & Refrigerants**, §2, p 138.
- To minimize the danger of thermal fatigue, do not use condensate subcooling (condensate should not be allowed to collect in the condenser) if the temperature difference between vapour and water inlet temperatures is more than 40 - 50 K for HCs, HFCs & HCFCs.

Conclusion: Condensate temperature control should be avoided in the condenser. Capacity control, with the considerations and restrictions above, could be done if the exit temperature is accepted as it is.

3.5. Parallel connected condensers.

Figure 05 shows four parallel-connected condensers. Note that nominally identical condensers can be very different in practice. Vapour distributes differently, entrained oil mostly likely enters in the first condenser, the cooling water might contain solids with uneven distribution, the water flow rate might be larger in the first than in the last condenser, etc. The resulting different pressure drops have to be compensated:

- As figure 5 shows by creating liquid columns at the exit. A large pressure drop is then recovered when descending a large column.
- By valves, which introduce extra pressure drops in the low pressure drop condensers. It is difficult to balance properly the various pressure drops. The effective temperature difference might also be reduced, thus increasing the surface. The beauty of the system above is that it is self-adjusting.
- Liquid columns can be created in the channels. This has to be avoided as, it reduces the condensing surface and is probably very unstable.

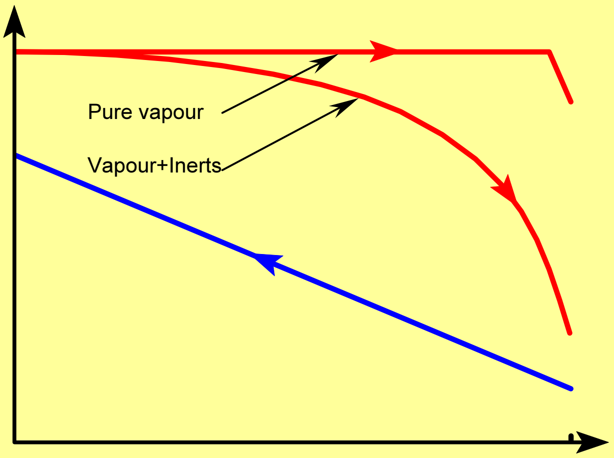
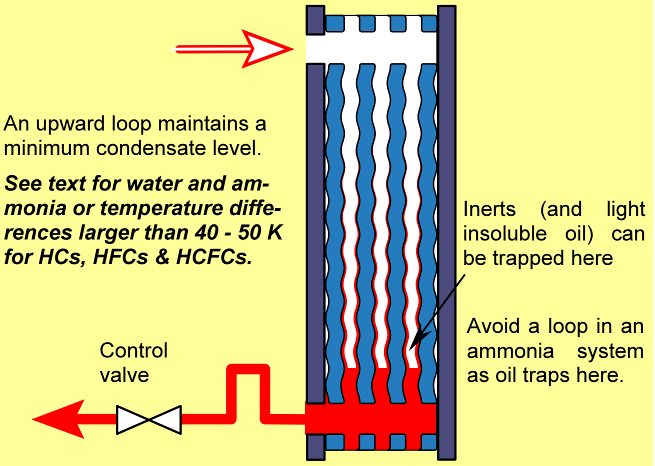


Fig. 03. Vapour with inerts.

The saturation temperature of the vapour-gas mixture as well as the heat transfer coefficient drops rapidly as the condensation proceeds.


An upward loop maintains a minimum condensate level.

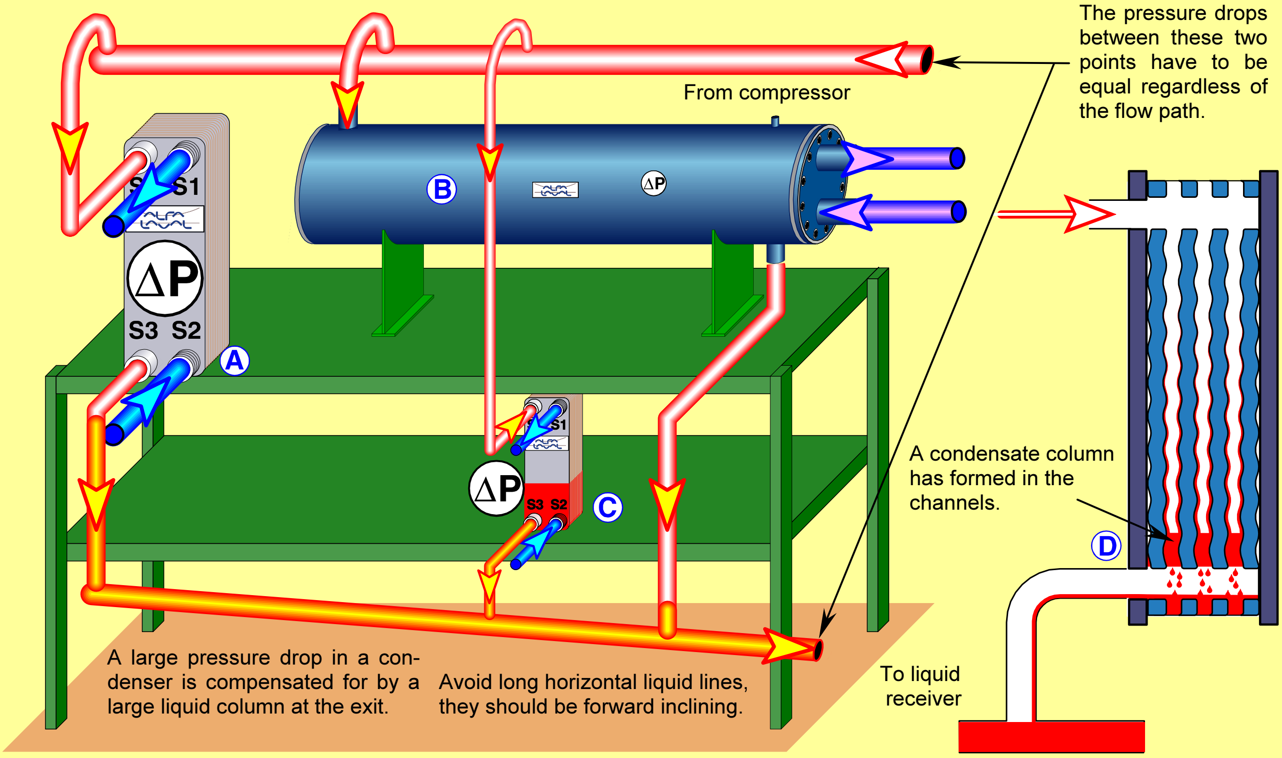
See text for water and ammonia or temperature differences larger than 40 - 50 K for HCs, HFCs & HCFCs.

Inerts (and light insoluble oil) can be trapped here

Avoid a loop in an ammonia system as oil traps here.

Fig. 04. Condensate subcooling.

Inerts and insoluble decomposition products accumulate above the condensate surface and are easily trapped there with a resulting decrease of the K-value.



The pressure drops between these two points have to be equal regardless of the flow path.

A condensate column has formed in the channels.

A large pressure drop in a condenser is compensated for by a large liquid column at the exit.

Avoid long horizontal liquid lines, they should be forward inclining.

Fig. 05. Parallel-connected condensers

The three condensers have different ΔPs. However, the total ΔP has to be equal, whichever path we follow from the compressor discharge line to the liquid receiver.

At start-up, the vapour distributes automatically between the condensers and the ΔPs become equal. To achieve this some of the vapour from A & C diverts to B .

If B can handle the increased load, equilibrium is reached. Chances are though, that vapour leaves uncondensed and enters the receiver. The increased pressure in this blocks the condensate flow from A & C, and liquid columns build up in the condensate lines. These columns help to evacuate the liquid from A & C and more vapour enters into these. Finally a new equilibrium is reached, where the higher ΔPs in A & C is offset by the suction action from the condensate column.

If the height difference between a condenser and the liquid column is not great enough, the condensate can block a part of the heating surface, with a loss of capacity as a consequence. This is what happens with C.

If the condensate line is too large to maintain a stable liquid column, the column forms in the channels, see D. This is an unstable situation, a column forms, suddenly drains, forms again, etc., and in any case there is a loss of capacity as condensate blocks part of the surface.

3.6. Pressure drop limitations in PHEs.

A PHE condensing is often pressure drop limited, either on the water or on the vapour side. The number of plates is only determined by the need to keep a certain pressure drop. Pressure drop limitations on the vapour side usually occurs for low-pressure vapour, type vacuum steam. On the water side, it usually occurs for very high MTD. The typical refrigeration condenser is thus not concerned.

Example: Water is heated from 50 to 90 ºC by saturated steam of 150 ºC. The condenser turns out to be 500 % overdimensioned with the required pressure drop on the water side. As a heat exchanger always operate with 0 % margin, the evaporator adjust itself by either of:

- The total vapour pressure drops until a sufficient low MTD is reached, which might occur at vacuum. The condenser is then impossible to drain and vent. When enough condensate or inerts have collected, the pressure increases again but the operation can be erratic.
- Throttling the water flow decreases the MTD, leading to unwanted boiling of the water.

It is entirely possible to use a PHE as steam condenser but a careful analysis of the operation is necessary. See also figure 04 and p. 102 for condensate draining.

4. Liquid receivers.

4.1. Types and purposes.

The condensate leaving the condenser in larger systems is normally collected in a vessel, the liquid receiver (LR) See figures 06. It is sometimes called the HP (High Pressure) receiver to distinguish it from the vapour-liquid separator in a flooded system, which then is a LP (Low Pressure) receiver. There are mainly two types of LRs:

A) ***The surge liquid receiver***. The condensate normally bypasses this. It handles normal load variations in the plant. It is usually connected to the condenser inlet through a pressure equalization (EQ) line, see below.

B) ***The through liquid receiver.*** The condensate always flows through this. It is better for dumping refrigerant contents from other parts of the plant. It should not normally be equipped with an EQ line.

The liquid receiver serves the following functions:

- The LR has to absorb or release refrigerant during load variations as the refrigerant contents of the various components in the system then change.
- If a component is shut down, e.g. for service, its refrigerant content can be dumped in the receiver.
- Maintains a supply of liquid refrigerant at constant pressure to the expansion valve.
- In a flooded evaporator, the liquid level in the separator normally controls the expansion. The liquid receiver can be used as it has a more stable liquid surface. The drawback is that only one expansion valve (evaporator) can be controlled.
- Act as a vapour-liquid separator for cooling of the hot compressor oil in a small BPHE evaporator. The BPHE is subjected to less thermal shocks if using warm HP refrigerant rather than LP refrigerant.

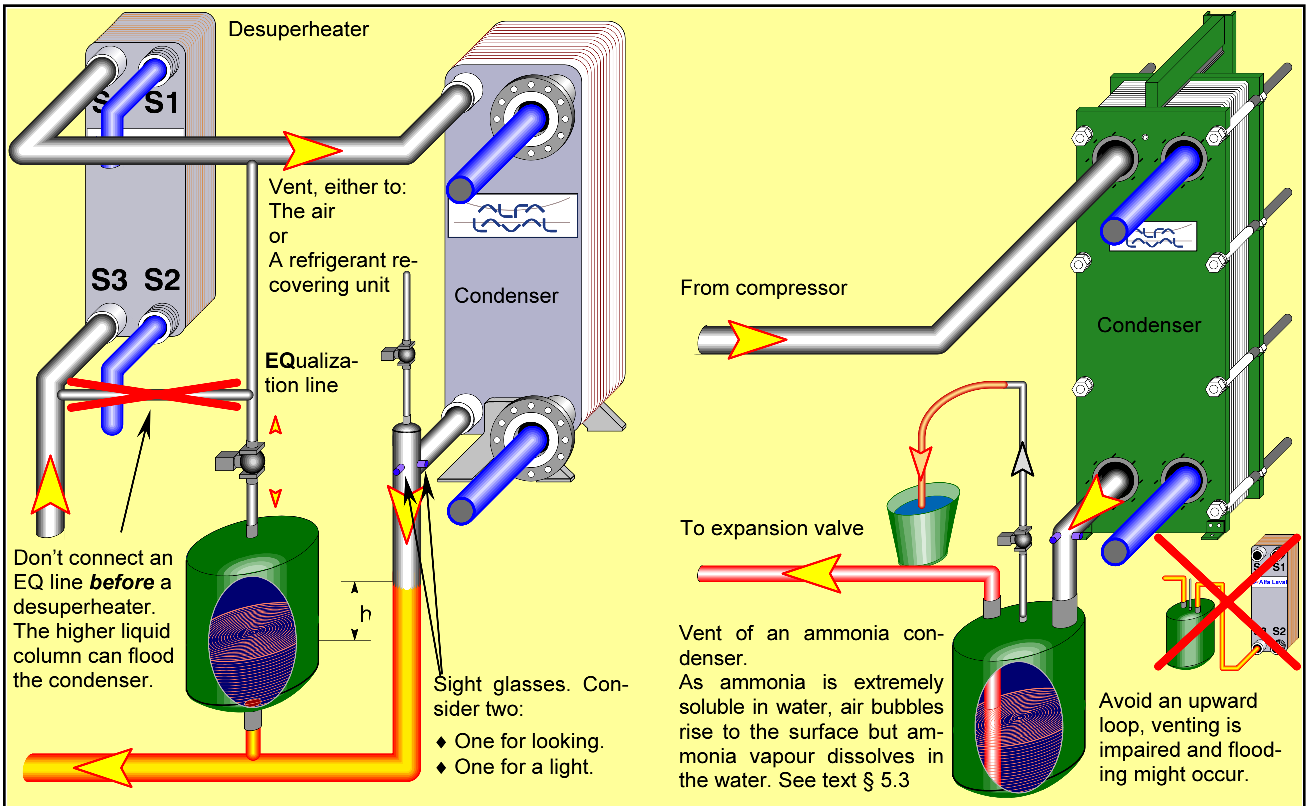
4.2. The pressure equalization line.

- A pressure equalization (EQ) line is used to direct vapour formed in the LR back to the condenser. It is connected between the condenser inlet and the LR. Figure 6 shows the function and installation of an EQ line. An EQ line is usually found only in large systems with flooded evaporators, not in small or DX based system.
- A ΔP of 10 kPa in the condenser, a ΔP easily found in a PHE, corresponds to a liquid column of about 0.7 m for most fluorinated refrigerants and 1.5 m for ammonia. The plant thus has to be of a certain size in order to accommodate the necessary height.
- If the LR serves a direct expansion evaporator, the pressure in the LR has to be maintained at a minimum for faultless operation of the thermostatic expansion valve. The control valve necessary for this precludes the simultaneous use of an EQ line. See §5.

4.3. Operation.

- The main use of the EQ line is to direct vapour back to the condenser if the refrigerant in the receiver is heated and evaporates. In winter, the opposite could happen. A very cold condensate produces a low temperature, vapour flows from the inlet to the receiver and heats up the contents. As a warm condensate means loss of capacity, the EQ line should be closed.
- If the refrigerant evaporates in a through receiver, vapour flows back into the condensate line, enters the channels from below and condenses in the lower part of the condenser. Small amounts of vapour can be accommodated in this way but larger amounts could create a condensate plug in the lower part of the channels. If the condenser is properly designed, i.e. the extra vapour from the LR is added to the flow from the compressor, the condensation might work.

  The condensate plug is unstable, the more so if inerts are present even in very small amounts. It could selectively build up in some channels, then suddenly drain, rebuild in others, resulting in a fluctuating flow and loss of capacity. The design is recommended for S&THEs.
- A similar condition to the above could happen if the condensate subcools very much and subsequently reheats somewhat, by vapour or by the water if this is in cocurrent. The lowest pressure then occurs a bit inside the channels and a condensate column might be “hanging” in the channels. This too is unstable, though it should not give problems with insufficient capacity.
- Installation of the through LR above the condenser is not impossible but should be avoided as it can lead to irregular flow and control problems, notably in the case of parallel condensers. Care has then to be taken that the pressure in the condenser is higher than in the LR. Certain expansion valves for flooded operation are integrated with the LR. A small pipe from the vapour space of the LR connects to the line after the expansion. This creates the necessary lower pressure and these can be mounted above the condenser exit.
- A downward U-bend between the condenser and a through LR transforms this to a hybrid LR. It cannot be vented but it can be equipped with an EQ line as a condensate column now can form.



Vent, either to: The air or A refrigerant recovering unit

Don't connect an EQ line ***before*** a desuperheater. The higher liquid column can flood the condenser.

Sight glasses. Consider two:

- ♦ One for looking.
- ♦ One for a light.

Vent of an ammonia condenser. As ammonia is extremely soluble in water, air bubbles rise to the surface but ammonia vapour dissolves in the water. See text § 5.3

Avoid an upward loop, venting is impaired and flooding might occur.

Fig. 06. The liquid receiver and the pressure equalization line.

A. The surge liquid receiver.

1) The condensate enters and leaves from the bottom. As the condensate line is always liquid filled, a liquid column can be created in the condensate pipe.

2) Inerts cannot follow the condensate to the receiver and be vented from this. Use the position shown.

3) It should never be installed above the condenser (liquid level below condenser exit) as the condensate might flow back and flood the condenser.

Troubleshooting a flooded condenser can be very difficult.

4) A pressure equalization line can be installed. The purpose of this is to carry vapour formed in the receiver back to the condenser inlet for recondensation. To overcome the higher pressure at the inlet, a condensate column - height h - forms in the condensate pipe. This is easily done as it is always liquid filled.

5) Sources of vapour in the receiver can be:

- ♦ Heat from the outside in a warm machine room. As the refrigerant content is more stationary here than in the through type there is a greater danger of heating from the outside.
- ♦ An oil cooler, cooled by evaporating condensate. The vapour could return to the LR and then through the EQ line to the condenser inlet or directly to the condenser inlet.

6) Note that the EQ line has to be open if the receiver is heated, either by "mistake" or design.

B. The through liquid receiver.

1) The condensate enters from the top through the "empty" condensate line. A liquid column can thus not be created and maintained in the condensate pipe.

2) Inerts follow the condensate to the receiver and can be vented from this.

3) It can be installed above the condenser as the condensate cannot flow back to the condenser. The venting might be impeded, though and there is a danger of backflow and it is better to avoid this installation. See the text.

4) A pressure equalization line should not be installed. The higher pressure at the condenser inlet must be compensated by a liquid column. As this doesn't form or remain in the condensate line, it forms in the channels, where it blocks a part of the surface and decreases the capacity. Moreover, it is unstable.

5) This should not be used in a vapour generating environment (Figure 6. A 5) as the vapour then flows back where it shouldn't, into the condensate pipe. A small vapour flow is acceptable in a PHE, though.

Note that this is contrary to advices for S&THEs.

Note also that the presence of inert gases impedes the creation of a liquid column in the channels as the condensate subcools without a pressure decrease.

If an oil cooler must be installed in a through receiver, feed it by a pump or possibly by a liquid column from the receiver to the oil cooler and return the vapour to the condenser inlet, not to the liquid receiver.

5. Venting.

5.1. Parameters affecting the venting.

Inerts can be suspected and venting should be done if:

- ♦ The capacity is decreased and fouling is excluded.
- ♦ The exit temperature is very much lower than the condensing temperature of the vapour.
- ♦ The manometer needle vibrates rapidly. (Ex. I, § 3.3)

See § 3.3 for the effect of inerts (noncondensable gases) on the condensation process.

Note that a decreased capacity and a low exit temperature can depend on a condensate level in the condenser as well. The exit piping from the condensor to the expansion valve has to be checked for obstructions. See also §3.4 & 3.3.

5.2. Location of the vent.

A correct positioning of the vents depends on:

- ~ ***The flow path.*** The concentration of the inerts is highest in the coldest part of the condenser, normally at the end of the condensation process. The inerts follow the condensing vapour to the exit. Here, there is at least a partial separation of the liquid condensate and the remaining vapour/inerts mixture. If separated, they can leave from different exits, if not they leave together for separation later.
- ~ ***The density difference.*** An inert gas, in contact with a heavier refrigerant vapour, rises above the refrigerant and should be vented from the highest position in a vessel. If heavier than the refrigerant, the venting position should be as low as possible.

The venting can thus be made at the lower port hole - from the frame plate or the pressure plate - or from the upper port hole - the one opposite the vapour entrance (it make no sense of venting from the entrance). There is a difference between ammonia and all other refrigerants.

- ♦ ***Ammonia.*** There are only three gases - hydrogen, helium and methane - lighter than ammonia but none is likely to be found in a refrigeration plant. If ammonia enters the ambient air, it rises and eventually mixes with the air. The mixture is always lighter than pure air at equal temperatures. (For pure ammonia to be heavier than air, it has to be more than 100 K colder than the air).

  The venting position is thus clear, the lower port in a PHE.

- ♦ ***Other refrigerants*** are heavier than air A body of air in contact with a vapour ascends to the highest point in the system. In a mostly horizontal condensation, such as in a S&THE, the venting should be done at upper part of the shell as far away from the inlet as possible.

  In a PHE, the situation is more complex as it is a vertical configuration with an inlet at the top and exit at the bottom. In practice, the flow path has the greatest importance. The vapour/gas mixture passes the channels and leaves at the bottom. The pressure drop prevents the lighter mixture at the end from rising in the condenser.

  When the condenser shuts off there is a risk that the lighter mixture at the bottom rises to the top, but as a corrugated channel is a good mixing device, the result will probably be a fairly uniform gas/vapour mixture. Even though an inert gas pocket could form at the upper porthole, it will leave with the condensing vapour when this re-enters. Only in the rare cases of a extremely low pressure drop there could be a remaining air pocket. In such a case a vent at the upper port could be useful. See §4 in **3. Design & Installation.**

Conclusion. Venting should be from the lower port.

Venting must not be done from the port itself. It can be done from the succeeding piping or the liquid receiver.

Figure 06 shows one method of venting for both LR types. This method for the surge receiver can be used for the through receiver as well and ***should be used*** if there is any danger of condensate blocking the exit pipe.

5.3. Venting methods.

- ♦ In case of ***ammonia,*** the venting is normally to the air but not directly because of the smell and security. The inert gas outlet is lead in to a bucket with water. As ammonia is very soluble in water, it is absorbed in the water and if bubbles rise to the surface, the ammonia contains air.

  To decrease the vapour concentration, the mixture can be cooled with LP refrigerant in a vent condenser, fig. 07A.

  The design is simple; a coil cooled by expanded refrigerant in a vessel of a certain volume. The vapour-inert mixture enters, refrigerant condenses and flows back to the LR through the line & valve V3. Finally, the vessel is full of inerts and no more vapour enters. The condensate flow back to the LR then stops, which can be monitored by e.g. the temperature. The normally open valves V1 - V3 then close and the normally closed valve V4 opens. The valve V4 vents the condenser into a bucket of cold water, which dissolves the remaining ammonia.

  As ammonia is so easy to vent, the vent bypass found on some float expansion valves (see fig. 07B and below) is not necessary, it is better to make a “real” vent.

- ♦ ***Halogenated refrigerants*** are usually not allowed to be vented to the air or the amount has to be kept to a minimum. There are specially refrigerant recovery units, which collects the refrigerant. Usually the first stage of this is cooling of the refrigerant to a low temperature, a refinement of the recovery unit in fig. 07A.

  Another method of venting is shown in figure 07B. A small pipe connects the vapour inert space with the LP side of the expansion valve and the inerts leave through this. It is not exactly venting as the inerts continue to circulate in the system but they are at least distributed more evenly.

- ♦ A ***hydrocarbons*** can be vented to the air but great care has to be taken that it does not collect in a low place, where it can be a fire hazard. The danger can be minimized with some type vent condenser, figure 07A.
- ♦ ***Carbon dioxide*** poses a special problem as it change phase when it passes the triple point of 5.18 bar. The resulting solid “dry ice” can block the piping. The reducing valve (and emergency valves) should thus have a direct exit or through a very large diameter pipe to the air.

5.4. End of the venting.

Ammonia is easy, when no more air bubbles emerges from the bucket with water, the air has disappeared.

Other refrigerants are more difficult. The best is to monitor the exit temperature. As long as this remains colder than the dew point of the vapour or at least as long as it continues to decrease, there are inerts in the vapour.

When a pure vapour condenses, the exit temperature is the same as the entrance temperature (minus the effect of the pressure drop). In practice it can be difficult to measure the exit temperature exactly as cold condensate could wet the thermometer or the wall on which an outside thermometer is placed. An outside thermometer should thus always be placed on the upper part of a pipe.

Because of the difficulties in determine the content of the noncondensables, the “false” vent in figure 07B is useful.

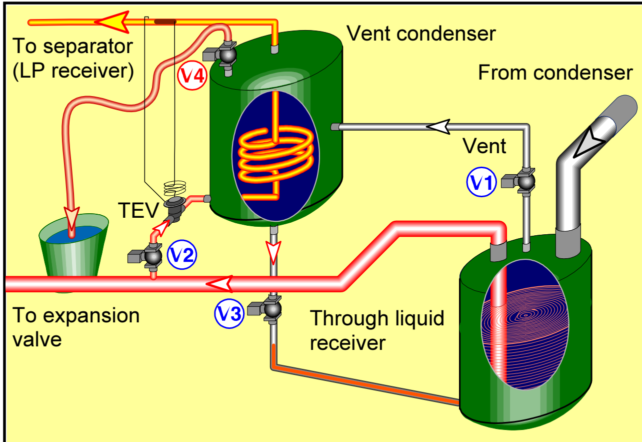
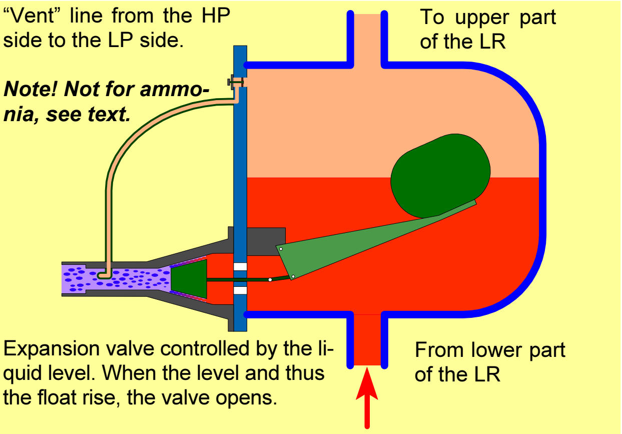


Fig. 07A. An ammonia recovery unit.

The vent is not directly to the air but to a recovery unit. This design is for ammonia. If bubbles arise to the surface in the bucket with water, then there is air in the ammonia.

The design can also – but without the water bucket – be used for other refrigerants. See also the text.



Note! Not for ammonia, see text.

Expansion valve controlled by the liquid level. When the level and thus the float rise, the valve opens.

Fig. 07B. A level controlled expansion valve with a vent for an HP receiver.

The design is conceptual. Different manufacturers have various designs for manual control, pilot, venting, etc.

Turn it upside down and it becomes an LP receiver, i.e. closing at rising level.

6. Control of condensers.

6.1. Condenser control requirements.

The control system depends on the type of system:

a) ***Cooling plant***, with flooded evaporators, either refrigeration or A/C. No control is necessary except that water could be saved at low cooling demands.

b) ***Cooling plant***, with DX evaporators. The condensing pressure has to be kept to a minimum, to give the TEVs sufficient driving force.

c) ***Cooling plant***, with heat recovery. The pressure has to be kept to a minimum to recover useful heat but the primary function of the plant, the cooling, must not be jeopardized. Thus, either hot water of a certain temperature or more water of a lower temperature might be produced but probably not both simultaneously.

d) ***Heat pump.*** The heat output is the primary. If possible both requirements in c) should be complied with.

In all systems, there are additional control requirements, which can be said to be "security controls":

e) The condensation pressure must not be so high that it jeopardizes the compressor. This a part of the compressor control and is not discussed here.

f) The water must not freeze. This is not usually a problem in a condenser, but in a reversible HP/AC unit the condenser becomes evaporator and vice versa or a condenser could be installed outdoors in winter. As freeze control is very important, an entire chapter, **7. Freeze** and a part of **1. Applications,** is dedicated to this.

g) The water temperature must not be so high that the cooling water scales. This is not only a control problem but also a water quality problem.

h) The steady forward flow of the refrigerant has to be assured. There are thus a number of check-valves, solenoid valves, and level controllers etc. to assure that the refrigerant doesn't flow to nonoperating equipment.

The design of a control system for a refrigeration plant is not a part of this manual, but some comments on the particularities of PHE will be given.

6.2. Change in a condensing parameter.

When a parameter in the condensation process changes, the control system has to make an adequate response ranging from nothing to a major change in another parameter. Below are some changes and responses:

♦ The capacity, i.e. the vapour amount decreases but the condenser continues to condense at the old capacity. The pressure and the temperature start to decrease. The resulting lower density means a smaller mass flow from the compressor, which finally matches the lower condensing capacity due to the lower MTD.

a) None or possibly a hand regulation of the water flow to save water and/or pumping cost during winter.

b-c) A pressure control is necessary. This could be done either with a pressure controller or by controlling the water flow, thus the MTD. Another method would be to control the compressor output but except for d), compressor control is usually reserved for the evaporator.

d) The temperature has to be maintained even though the refrigeration capacity will suffer.

♦ The water entrance temperature decreases, the MTD increases, and the condenser capacity surpass the compressor capacity.

a) Basically no control is necessary or a control of the water flow, manual if slow variations, and automatic if more frequent changes of the water temperature.

b-c) As above.

d) If a certain water temperature has to be kept, a decrease of the inlet temperature means a capacity increases. The water flow has then to decrease or the condensing pressure has to increase.

♦ There are thus basically only a few control methods:

◊ The water flow as a function of the condensing pressure or as a function of the water exit temperature.

◊ Maintaining a certain condensation pressure.

◊ The pressure as a function of the water temperature.

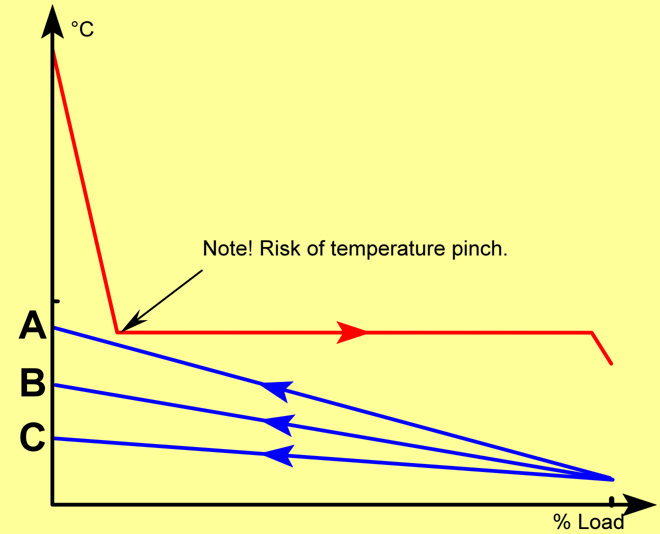


Fig. 08. The correct water flow .

A) A too small MTD means an erratic response, as it changes rapidly for a small change of the water flow.

B) A small change of a large flow has little effect on the MTD and thus on the performance.

C) A correct flow. The MTD responds well to a change of the water flow rate.

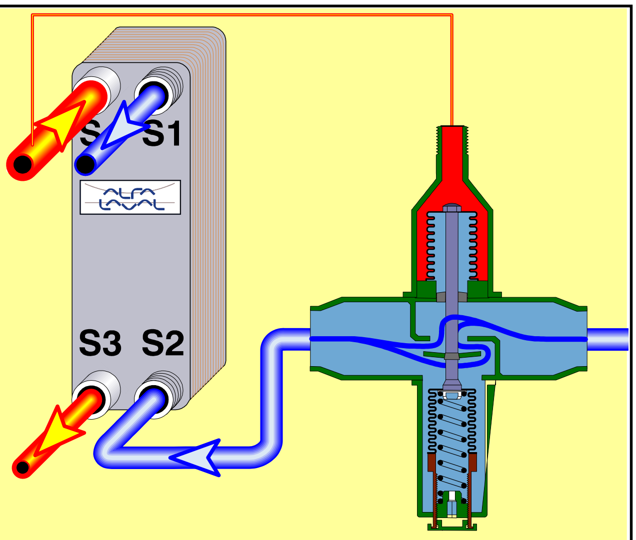


Fig. 09. A vapour pressure controlled water valve.

When the refrigerant pressure increases, the pressure transmits to the valve. The valve seat is then pressed against the spring force and opens, more water enters and the increasing temperature difference makes more refrigerant condense and a new equilibrium is attained. The opposite happens at a pressure decrease.

6.3. Control on the water side.

A change in the water flow must give an apt change of the MTD, neither too small, nor too large, see figure 08.

Figure 09 shows a typical valve design. Such a valve (proportional) can maintain the set pressure at only one given capacity. This is usually no serious problem, though. It is important to design the heat exchanger and the valve to give an appropriate response at a change.

If the water temperature comes very close to the refrigerant temperature (figure 8A), a small change in the water flow causes a large change of the MTD, with instabilities the consequence. It is by no means impossible to heat water to a close temperature approach, but then the control method has to be changed. The water exit temperature could be the controlling variable, for instance.

6.4. Control on the refrigerant side.

See figures 10 A & B. The pressure is controlled by "itself", i.e. it forces a valve to open or to close when the pressure changes, thus changing the refrigerant flow.

The set point - the pressure - can be adjusted by changing the spring force, i.e. manually, but it can also be set by some other variable, the temperature or pressure at some other point, in a design similar to A above.

The control of the condenser depends very much on the type of evaporator used in the system.

System with DX evaporator and no heat recovery. The important variable to control is not the condenser but the LR pressure, as this feeds the thermostatic expansion valve. The design in either figure 10 A or B can be used.

Valve 2 in figure 10 B has to be placed as close as possible to the condenser to minimize the refrigerant filling and response time. If the condenser is placed outside, the valve must also be placed outside.

In such cases, the arrangement in B is better, as all the control, elements can be placed in the machine room.

System with DX evaporator and heat recovery. The valve 2 in figure 10 A maintains a set condensing pressure as well as a LR pressure. The condensing pressure might be set high enough to enable heating of water in either a desuperheater or a condenser to a desired level.

The system in figure 10 B leaves the condensing pressure undefined and only maintains a LR pressure, which makes it suitable for a TEV but not for heat recovery.

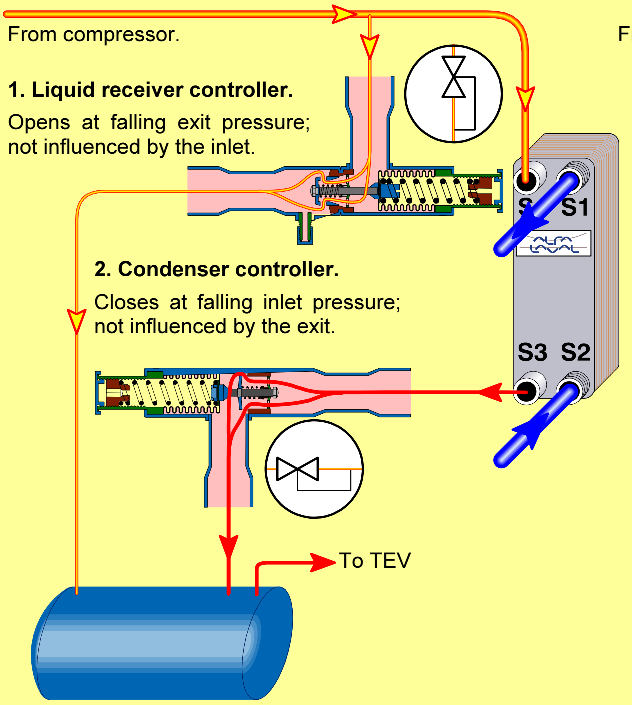
A flooded evaporator does not get the refrigerant directly from the TEV, but from the LP receiver. A lower LR pressure would mean a slightly lower liquid level, which has little effect on the capacity.

The pressure can be left free or it can be controlled by the cooling water flow.

7. Desuperheating.

Desuperheating might occur in:

- The main condenser and is treated as a part of this.
- In a special desuperheater, usually for recovery of high temperature energy. Sometimes refrigerant condenses in the desuperheater and has to be drained from this. This desuperheater could be a separate heat exchanger figures 11A, 13 & 14 or integrated with the main condenser (11B).
- In pipe or a vessel in order to decrease the inlet temperature to the condenser, mainly SWPHE and with ammonia as refrigerant. The purpose is to decrease the thermal load on the ring gaskets.



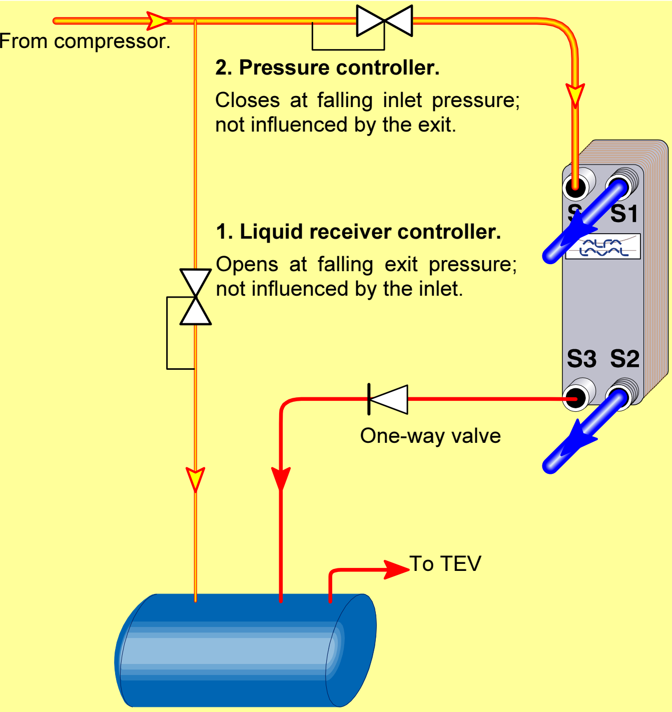
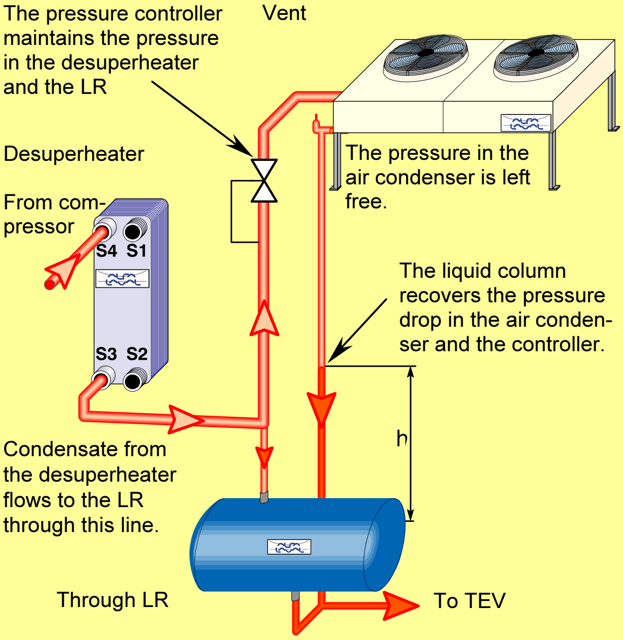


Fig 10A. Refrigerant pressure control I.

Valve 2 closes at decreasing inlet pressure. The condensate level then ascends and blocks the heating surface. The resulting decreased condensation forces the pressure to stabilize at a preset level. Valve 1 opens when the LR pressure falls and admits hot vapour, which then increases and stabilizes the LR pressure.

Fig. 10B. Refrigerant pressure control II.

The equipment is similar to that in figure C, but valve 2 is now placed before the condenser. It closes at falling discharge pressure and thus tries to maintain this. Valve 1 uses this pressure to control the LR pressure. The condenser pressure is uncontrolled. A check valve prevents liquid from flowing back into the condenser.



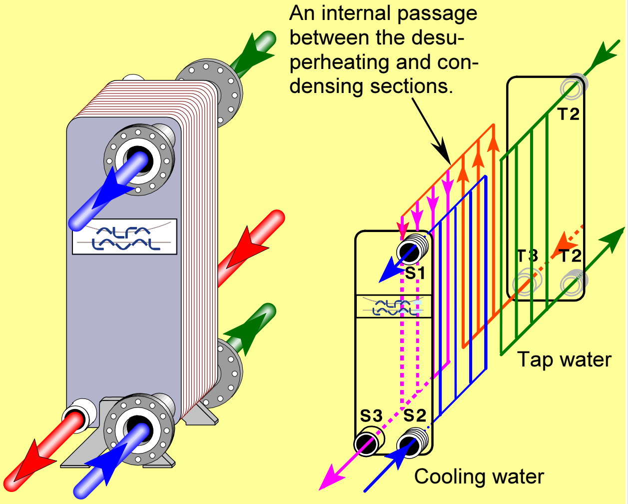


Fig. 11A. Draining of a desuperheater.

The condensate line serves both as a EQ line and to transmit the pressure before the controller to the LR. A special LR pressure controller is thus not necessary.

Note that there must be sufficient height difference for a liquid column to form which shall balance the pressure drop in the pressure controller and air condenser.

Fig. 11B. A condenser with an integrated desuperheater as a water heater.

The desuperheater and the condenser are integrated into one unit. The condensing vapour normally flows downwards, a natural direction as this is a two-phase flow.

The design can be used for all types of plate heat exchangers.

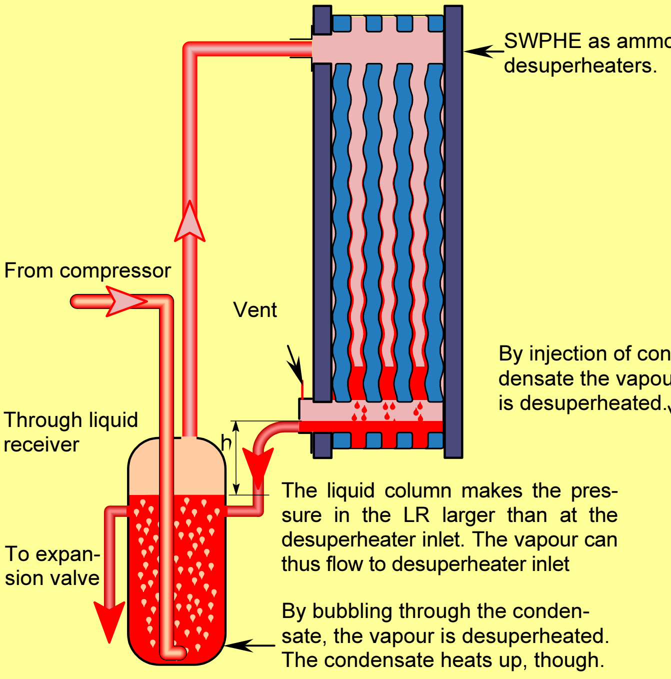


Fig. 12A. A bubble through desuperheater.

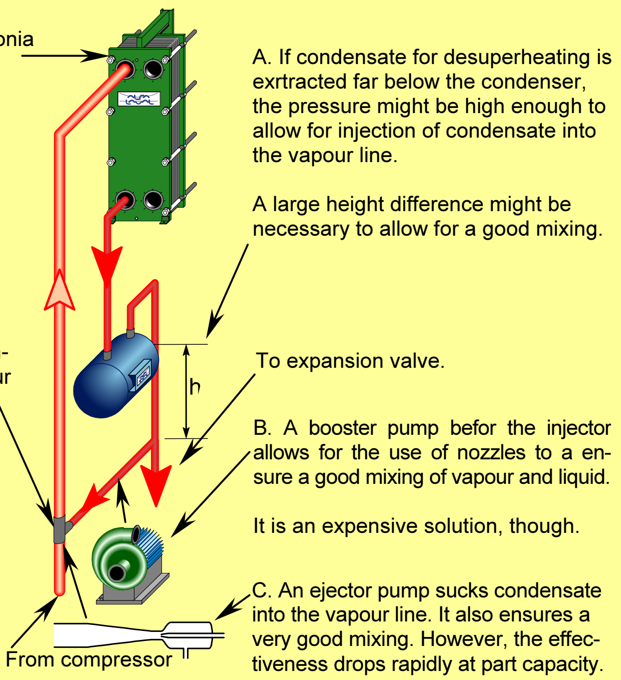


Fig. 12B. An injection desuperheater.

7.1. Draining of a desuperheater.

The ideal desuperheater would have the exit connected to the inlet of the main condenser by a short downward inclining straight pipe. See figures 13B & 14. If some refrigerant condenses in the desuperheater, the condensate moves along with the vapour down to the main condenser.

In many cases, it is not possible to install the desuperheater above the condenser. The main condenser might be an air cooler placed on a rooftop, while the desuperheater is placed in a machine room in the basement.

If only small amounts condense it might not be a problem especially not if the desuperheater is a PHE. The condensate from a PHE desuperheater normally leaves as well dispersed droplets, which the vapour can carry with it.

If larger amounts condense in the desuperheater the design in figure 11A could be used.

7.2 . Desuperheating of a vapour.

If no heat recovery is necessary but the vapour temperature has to be reduced in order to save the thermal stress on the ring gaskets in SWPHEs it is sufficient to inject condensate to the discharge vapour, see figure 12.

Note that the total system capacity is not reduced as the condensate used for desuperheating is recirculated.

7.3. Control for heat recovery.

Heat can be reclaimed at the condensation either by recovering all the latent heat to heat water (or air) to a relatively low temperature in the condenser, or by heating a small amount of water to a temperature close (< 3 K in a BPHE) to the actual refrigerant inlet temperature in a special desuperheater.

Furthermore, the heat recovery can be the main purpose - a heat pump - of the plant or it can be an extra use of a refrigeration plant. From a control point of view, a single purpose plant - refrigeration or heating - is simple. A double purpose plant needs some extra consideration. See also figures 13 & 14.

Heat recovery in a refrigeration plant. The main purpose of the plant is refrigeration and the heat recovery should never interfere with this primary function. In order for the refrigeration and the heat recovery to peacefully coexist, the following points should be considered:

- The condensing pressure should never be so high that the necessary refrigeration capacity suffers.
- A plant operating for both heat recovery and refrigeration could be economically designed for a higher condensing temperature than for refrigeration alone.

  Ex. A plant with a cooling demand of 100 kW (te = 2°C, tc = 35 °C) yields totally 120 kW energy. However, energy at a temperature of 35 °C is close to worthless. The superheat, at about 60 °C and only a little more than 10 % of the total, is too low for tap water heating.

  Increasing the condensation temperature increases the compressor power from about 20 to 34 kW, but yields totally 134 kW, whereof 20 % is superheat at 90 °C, excellent for tap water heating in a BPHE, and the remaining can be used for space heating.

  Systems can be designed for two condensation pressures; the higher one is then used when heat recovery is necessary. See figure 14.
- The condensing pressure should on the other hand not fall so low that the DX valve does not work properly.
- The liquid receiver (or at least the line from it) should never run empty.
- When heating is not needed, sufficient extra desuperheating and condensing capacity should be available.

  Ex. Heat is reclaimed by desuperheating the refrigerant vapour, which then is condensed in an air condenser. When hot water is not necessary, the air condenser both desuperheats and condenses the refrigerant.
- Another source of heating should be provided when the refrigeration capacity is not needed.

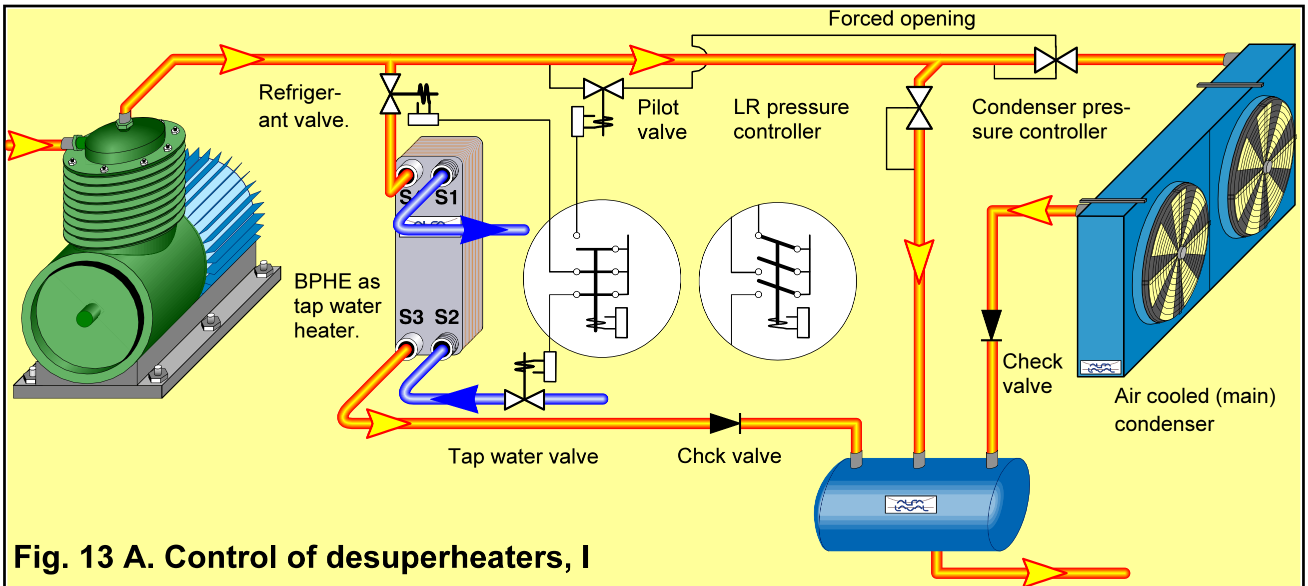


Fig. 13 A. Control of desuperheaters, I

No heat recovery. The refrigerant and the tap water valves are closed and the condenser pressure controller is forced wide open by the opened solenoid valve.

The condenser pressure is thus not controlled when no heat recovery is necessary as no specific temperature level has to be maintained. In winter the pressure could then fall below the required LR pressure if precautions are not taken, e.g. a liquid column from the condenser to the LR, using the system only in warm areas or controlling the pressure by some other mean, e.g. the fans.

Heat recovery. The refrigerant and the water valves open and the solenoid valve closes, thereby putting the pilot valve into operation. The pressure is then kept to a level suitable for heat recovery.

A part of the refrigerant flow diverts to the BPHE and heats the water. Higher heat load in the BPHE means lower discharge pressure and the condenser controller closes, forcing more refrigerant to the BPHE. Lower heat load means higher discharge pressure and the controller opens, forcing more refrigerant to the air condenser.

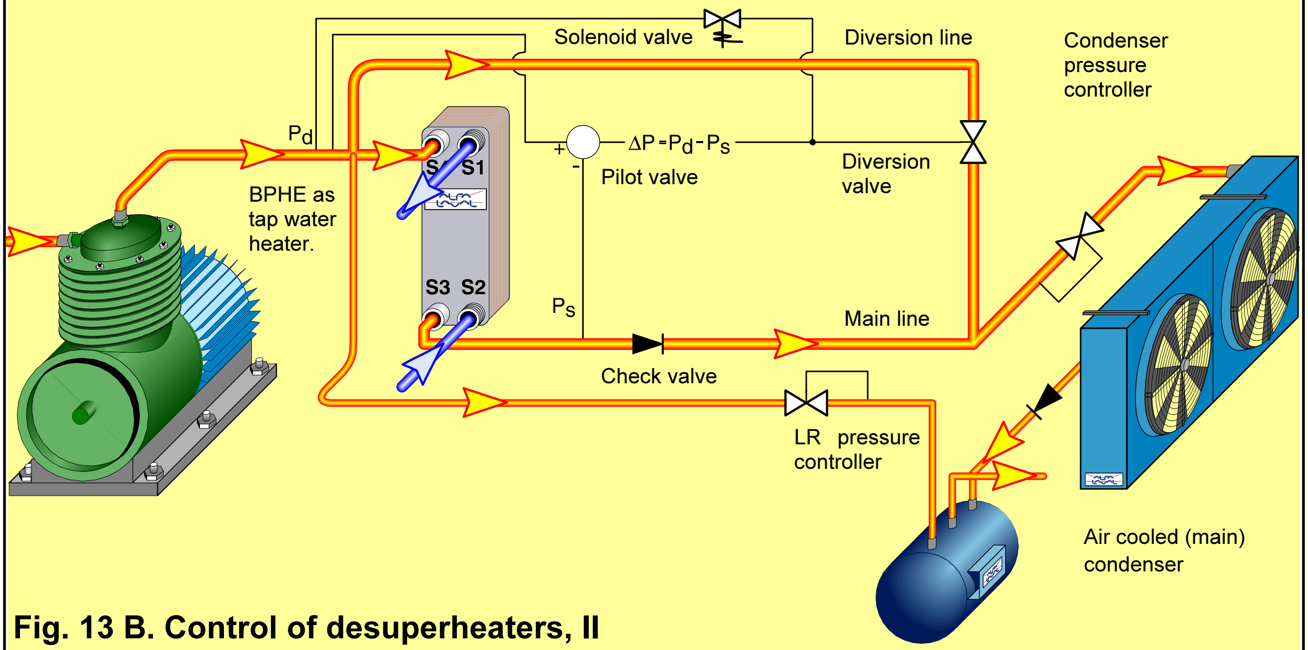


Fig. 13 B. Control of desuperheaters, II

No heat recovery. The refrigerant enters both the diversion line and the BPHE. The pressure is controlled by the condenser pressure controller as in figure 11 A.

Vapour thus enters the BPHE continuously and heats the cooling water even if no heat is needed. The system has to be designed for this. It can be fitted with valves on the water and refrigerant lines as in A above. The system as shown is thus mostly suitable in a plant, where there is continuous demand for hot water.

Heat recovery. When the water inlet temperature decreases (or the water is switched on) vapour starts to condense in the BPHE. The pilot valve senses the ΔP over the BPHE. When heat requirement increases, indicated by a lower water inlet temperature, more vapour condenses. The ΔP then falls and the diversion valve closes and more vapour enters the BPHE.

The solenoid valve is used to open the diversion valve during prolonged periods of no heat requirement.

8. Troubleshooting.

8.1. Insufficient capacity.

- Check all flow rates, temperatures, pressure drops, etc. Does the pressure drop suggest anything abnormal? An obstruction of the water flow, flooding from the LR, an unusual noise, etc.
- Interchange thermometers between various locations. Small temperature differences can easily be masked by incorrect thermometers.
- Inspect the condenser for temperature variations on the outside. Large differences suggest an inert gas pocket or a blocked water or refrigerant flow.
- Is a filter installed in the water inlet port? If this is partly blocked, the resulting unequal flow distribution decreases the capacity.
- Double-check the heat transfer, by using various combinations of temperatures and flows.
- Inspect the cooling fluid if it is not pure water. If glycol or similar, check for concentration and/or viscosity. A too high concentration will impair the heat transfer.
- Inspect the condenser for signs of fouling on the liquid side and oil on the refrigerant side.
- What are the signs of the insufficient capacity? If the condenser is used to recover heat and the exit water temperature is too low, the reason could be that the water temperature cannot pass the temperature pinch on the condensation curve, see figure 01.
- Are the evaporator and the compressor compatible? If so, the condenser capacity should be correct too. If not, there must be an irregularity somewhere.
- Check the compressor. Does it give sufficient refrigerant at the rated pressure? More refrigerant cannot condense than what enters. Is the compressor worn out, have internal leakages? Are the speed and electricity consumption compatible with the capacity?
- Are there instabilities? These can lower the capacity.
- A low capacity, but a very large condensate subcooling suggests a too high condensate level, blocking the heating surface for condensing. This could be due to a too large refrigerant filling in the system or too short condensate lines in a system of parallel condensers, thus causing one HE to flood.
- Another reason for low capacity/large subcooling could be inerts, which are not properly vented. In the absence of sight glasses it can be difficult to verify if this is due to inerts or condensate blocking. ***Inerts can manifest themselves by a heavily vibrating discharge pressure manometer***, see example I, p. 101
- Note that the capacity can be controlled by regulating the condensate level in the condenser if the condensate exit temperature is left free.

  The ***condensate temperature cannot be controlled by varying the condensate level*** as this would mean that the condensing capacity has to be left free, which normally is not possible. A too little capacity could thus be caused by a mistaken control of the condensate temperature.

  If condensate level control is used in the various condensers in a system, the necessary amount of refrigerant is larger during the cold season as a part of the refrigerant is used to block the heating surface.

  During the warm season, this extra refrigerant is dumped in the liquid receiver. If this is too small, it will block a heating surface somewhere.
- A leaking liquid receiver pressure controller bypasses refrigerant vapour around the condenser into the receiver and the appearance could be as if not all the refrigerant condenses.
- Check the EQ line. If this is open and the LR is of the through type, condensate could block the channels.
- Contrary, if the LR is of the surge type and is heated (warm machine room, oil cooler, etc.) and there is no EQ line, condensate could ascend to the condenser and flood the channels and impede venting.
- ***Check the installation according to the recommendations in figures 05 & 06. After fouling on the water side, badly installed and operated EQ lines and vents are the principal sources of condenser failures.***

8.2. Instabilities.

A refrigeration plant is a system of interconnected feedback loops, with the inherent risk of oscillation. See section Evaporation for a discussion on oscillations.

- Check the temperature difference. If the water temperature at some point is too close to the refrigerant temperature, a small change of the water flow rate could cause a large change of the temperature difference therefore with instabilities.
- Carefully choose the location of the control valve as in figures 10A 11. A large distance between valve and condenser means that condensate has to fill up the line before the condensate level increases, i.e. a long response time but a short evacuation time.
- Check the mechanical functioning of the various valves. Some valve types, notably expansion valves, have very small flow cross-sections. Debris such as metal chips from wear and tear are easily stuck there, impair the flow, and destroy the valve.

  If the flow decreases because of an obstruction, the capacity decreases as well. Such a valve could be revealed through an irregular control function and/or an unusual noise.
- If condensate blocks the channels e.g. because of a too high LR pressure, see figure 6, it could suddenly drain and cause instabilities.

8.3. Compressor HP cutout.

An increase of the discharge pressure followed by a cutout by the HP pressostat can be caused by:

- Condensate blockage of the condenser caused by too small an evaporator, too small an expansion valve, an obstruction in the condensate line, followed by a decrease of the condenser capacity.
- Decrease of the condenser capacity due to fouling; too warm cooling water or insufficient cooling water flow.
- The most insidious cause of a pressure increase is the presence of inerts; see the example in § 3.3, p. 101, 102.

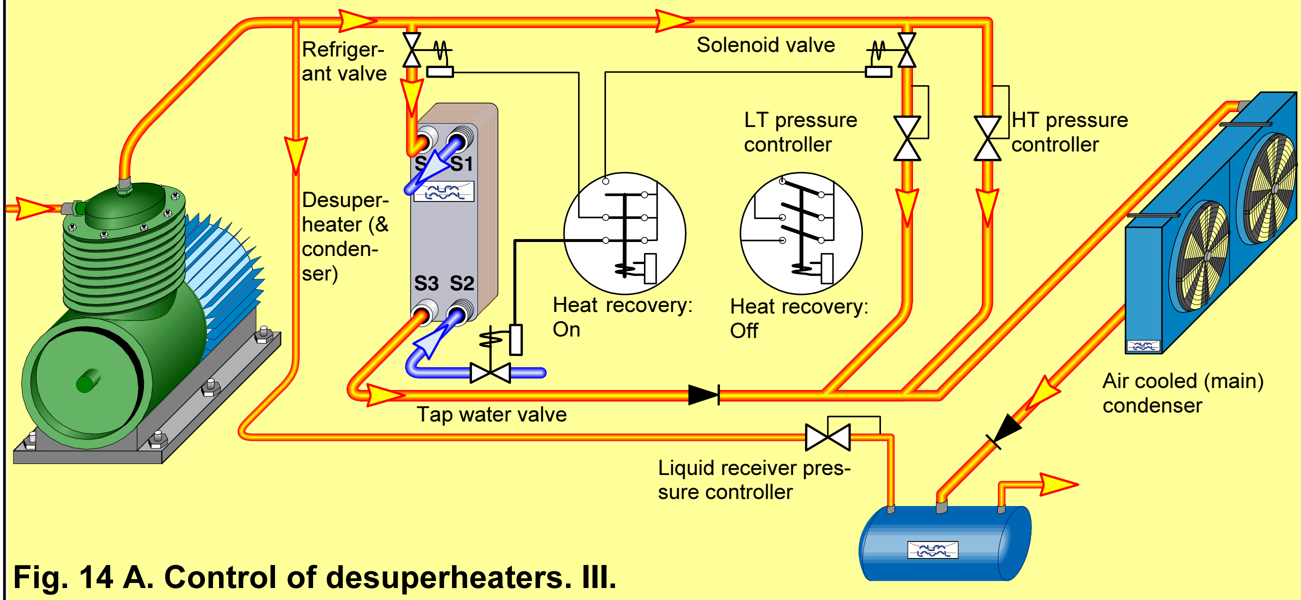


Fig. 14 A. Control of desuperheaters. III.

No heat recovery. The tap water and the refrigerant valves are closed, while the LT solenoid valve is open. The LT pressure controller maintains a low discharge pressure. The HP controller is set to a higher pressure level and is thus closed. If the LR pressure falls, the LR pressure controller opens. The system operates as in figure 10 B.

Heat recovery. The tap water and refrigerant valves open and the LT pressure solenoid valve closes. As the LT pressure controller is now shut off, the pressure is now set by the HT controller to a temperature level more suitable for heat recovery. A part of the refrigerant enters the BPHE, where it desuperheats and possibly condenses. The remaining refrigerant condenses in the main condenser or is used for space heating.

If the heat load drops in the BPHE, the discharge pressure increases and the HT controller opens and diverts more refrigerant directly to the condenser. If the load increases, the discharge pressure decreases, and the HT controller closes and more refrigerant enters the BPHE.

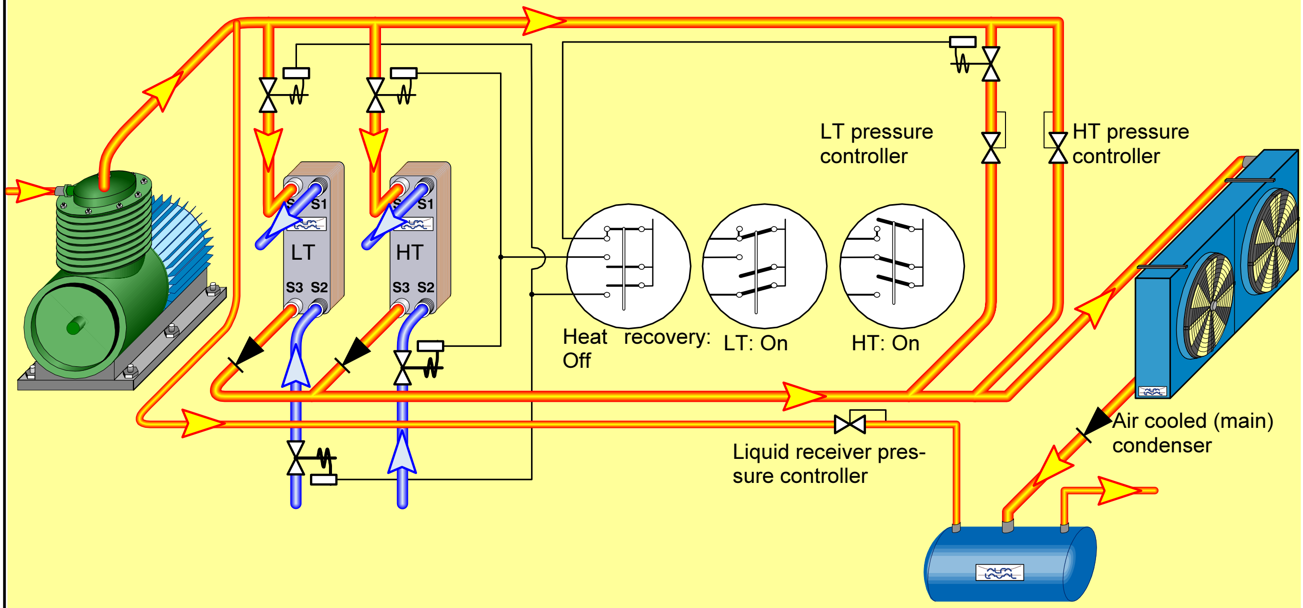


Fig. 14B. Control of desuperheaters, IV.

No heat recovery. Both the refrigerant valves (and the water valves) are closed. The low temperature controller is open, maintaining an economical pressure. The high temperature controller is automatically closed as the discharge pressure is below the set point.

The liquid receiver pressure maintains the LR pressure at a high enough pressure to satisfactorily operate the thermostatic expansion valve.

The system operates as in figure 10 B.

LT heat recovery. When heat is needed, the LT controller opens and a part of the refrigerant enters the LT BPHE. Changing heat demand causes the LT controller to open or close. The check valve at the HT BPHE prevents refrigerant from reenter by the exit connection.

HT heat recovery. When the HT BPHE is connected, the LT controller closes and the HT controller now controls the discharge pressure. The HT BPHE thus operates at a higher temperature level, more economical for heat recovery. The system operates as in A above.

6. Fouling & Corrosion.

1. Refrigerant side fouling.

1.1. Source of fouling.

A. General. Fouling on the refrigerant side - except from oil - does not normally occur to any great extent in a cooling circuit. When it happens, however, the consequences could be severe; especially for the expansion valve and the compressor.

B. Oil. The refrigerant will carry oil - either dissolved or as an emulsion - into the various components of the plant. Although the amount can be reduced, it can rarely be prevented entirely. The effect of the oil depends on whether the oil is dissolved in the refrigerant or not.

Oil will not only have an effect as fouling substance, but it will also affect the heat transfer coefficients of both condensers and evaporators in other ways. See also to the chapters on evaporation and condensation.

C. Soluble oil. The refrigerant keeps the oil in solution preventing formation of an oil film. The exception to this rule is the direct expansion evaporator, where the refrigerant evaporates and leaves oil droplets in the vapour. These stick to the walls and form an insulating oil film.

Fortunately, the heat transfer coefficient is small in this section of the evaporator, so the effect of the oil film is fairly small. The major concern here is to keep the velocity high enough to prevent oil droplets forming in the evaporator, and prevent loss of efficiency as a consequence.

D. Insoluble oil. This is the case for ammonia (all temperatures) and R22 (< ~ 0 °C). The effect depends very much on whether it is a condenser or an evaporator.

- ***Condenser***. As R22 dissolves oil at normal condensing temperatures, oil will not pose any great problem here. The effect for insoluble oil will be less than in an evaporator. The temperature is much higher, the oil is less viscous, and the gravitational forces and shear forces both try to move the refrigerant (ammonia) and the oil in the same direction. Consequently, there is no danger of oil build up in the condenser.
- ***Evaporator***. Oil suspended in the refrigerant will form a film on the heating surface both for dry and wetted evaporation. If the vapour velocity is large enough it will ultimately carry out the oil.

  Oil film which forms on the wetted part of the heating surface, where the heat transfer coefficient is large, will also have a measurable effect on performance.

E. Oil decomposition products. The oil can be heated in the compressor to above its decomposition temperature. The result could be a spectrum of various decomposition products from carbon to tar-like products. These can ultimately find their way into the heat exchangers.

The evaporator, the coldest component and the place where all liquids are at their most viscous, - has a particular tendency to foul.

The newer synthetic oils are especially problematic. These are often polarized compounds, which dissolve inorganic products - oxides from soldering, metal fines from the compressor, etc. - better than mineral oils. Some oils - largely polyester oils, polyether oils less so - also hydrolyse in the presence of water under formation of both fouling and corrosive products.

The problem is more severe for a refrigerant that does not dissolve oil. If the refrigerant dissolves oil, it washes away the oil and most likely its decomposition products as well.

F. Wear and tear. Solid particles resulting from wear and tear of the compressor can find their way into the heat exchangers. In the evaporator, the effect might not be entirely negative since they can form nucleation sites and thus increase the nucleate boiling coefficient.

G. Driers. Silica gel is a common substance used to dry the refrigerant. It is delivered as crystals packed into the dryer. If the packing is not tight and if the filter is not correctly installed, the crystals will move around and wear against each other, resulting in the release of solids.

H. Water. In theory, water should only be in ppm quantities on the refrigerant side. In practice, it can be different. Certain compressor types - piston - react immediately to even small amounts of water. Others - screw and turbo - can tolerate appreciable amounts of water.

Water, oil and decomposition products can form a sludge that can settle on the heating surface - especially in the evaporator, since this is the coldest place in the system. It can also cause the refrigerant to foam, resulting in reduced heat transfer. Water can enter the system by the hygroscopic synthetic oils, which pick up air moisture.

1.2. Cleaning and prevention.

A. Cleaning. Normal practice is not to clean a heat exchanger on the refrigerant side unless a complete overhaul of the system is necessary. The most likely type of fouling will be oil and its decomposition products and a suitable industrial detergent can be used.

B. Oil separators. Oil should be trapped in a separator, especially at low temperature applications, purified from possible refrigerant and returned to the compressor.

C. Filters & Dryers are commonly combined into a unit, installed vertically with the refrigerant flow downward usually between the condenser and the TEV. This minimizes the risk of unwanted movements and thus wear and tear of the silica gel crystals. There are also special models for bi-directional flow. Decomposition and metal fines, water, etc. are also trapped in the filter/drier.

D. Design. In order to ensure a good entrainment of the oil in the evaporator, the vapour velocity, or more correctly the shear stresses should be as high as possible. The shear stresses are proportional to the pressure drop per meter flow length. Observe that other considerations, e.g. temperature difference, could limit the pressure drop. A pressure drop of about 5 kPa/meter is usually enough.

2. Fouling in a closed circuit.

The liquid side of an evaporator usually contains the liquid in a closed loop, either water or water containing a solute which lowers its freezing point. As it is a closed loop, the liquid can be kept pure, thus causing little trouble. The most common impurities are oil and other greasy products. At higher concentrations, a glycol solution can dissolve or at least disperse oil. Consequently, it provides a self-cleaning effect.

Where cleaning is required, the best solution is an industrial detergent, preferably in a CIP (Cleaning In Place) system. This can be very simple - a container for the cleaning solution, a pump and connecting hoses - or very sophisticated, with tanks for various cleaning solutions, heating equipment, valves, interconnecting pipes & hoses all mounted on a skid.

If the liquid to be cooled is a process solution, it is difficult to give general advice. Normally the final owner/operator of the cooling plant knows the properties of the liquid and the fouling problems it can cause.

In case of raw water, see below. The same water used in both a condenser and an evaporator usually gives fewer problems in the evaporator. Neither scaling nor biological growth is usually a problem in cooling plants.

Note that some brine or glycol solutions corrode mild steel or zinc if not properly inhibited. The corrosion product might then foul the heat exchanger.

Calcium chloride in an evaporator is somewhat difficult brine. A pH that is too low makes it corrosive, one that is too high results in scaling.

3. Raw water fouling.

3.1. Water types.

Water can be classified into the following types:

A. City water. Usually excellent. Cost normally inhibits its use in all but the smallest condensers. It is used as make-up water in cooling towers.

B. Well water. This usually has a low biological content, but the concentration of scale forming and/or corrosive salts (calcium & magnesium sulphate & carbonates) can sometimes be very high. The solid content can be high as well. Depending on the quality, pre-treatment might be necessary, ranging from a simple screening to more elaborate methods.

As it is usually fairly cold and normally available in lesser quantities, the temperature is allowed to increase more than for cooling tower water, 10 - 15 °C with a corresponding lower flow rate.

C. Cooling tower water. The water circulates in an open circuit between the condenser and a cooling tower. It is normally 15 - 20 °C warmer than well water from the same region. The salt content can be ten times as high as the make-up water, normally city water (very clean) or well water. In heavily polluted areas, it can pick up dust and corrosive gases. (See later on corrosion.).

The net effect could be a fairly fouling and corrosive brew, which needs various treatments before being allowed into the more sensitive parts of a plant. Fortunately, as it is a closed loop, this is easy to do.

A cooling tower is normally designed to cool water by about 5 °C, thus specifying the available temperature increase in the condenser.

D. Surface water from rivers and lakes. The concentration of scale-forming salts is usually fairly low. However, It can contain appreciable amounts of solids, ranging from sand & earth to leaves and fish. Its microbiological activity (algae, bacteria & fungi) can be high, especially in water from agricultural areas, which contains fertilizers. Usually some type of pre-treatment is necessary, especially to control the biological activity. The temperature is usually between that of well water and cooling tower water. Normally the temperature should not be allowed to increase more than 10 °C for environmental reasons.

E. Municipal wastewater. In certain locations, the only water available for large consumers of industrial cooling water such as steel works, refineries, chemical process industries, etc. is municipal wastewater, usually treated, which can be more or less crude. It contains large amounts of natural fertilizers, especially free ammonia. It is sometimes aerated to dissipate the ammonia.

The suitability of this water is questionable as a cooling medium for a BPHE condenser, because of the risk of corrosion and fouling.

F. Brackish & sea water should not be used in current BPHE condensers (stainless steel and copper construction) because of the corrosive action of the Cl^- ion.

Note! Nickel brazed BPHE with plates of Avesta 254 SMO could be used with brackish water.

Semiwelded heat exchangers with titanium plates are fully resistant to all types of sea water, even the most polluted.

3.2. Fouling types.

Fouling can be classified into two major types, pressure drop and surface fouling. A third type, settling of solids such as dust and sand, can affect both pressure drop and heat transfer simultaneously. Below is a description of the most common types of fouling, how to clean the heat exchanger and some advice about what to factors to consider when designing BPHE condensers.

3.2.1. Pressure drop fouling.

Particles in the water, fibres, leaves and wood chips from the cooling tower are stuck in the inlet of the PHE. After some time the pressure drop increases but the thermal performance of the BPHE is unaffected as long as the pressure drop increase does not affect the flow.

Normally particles such as sand will pass through the BPHE if their diameter is less than about 1 millimetre. However, particles such as fibres, have a tendency to get stuck inside regardless of size and should not be allowed into the cooling water.

3.2.2. Prevention and cleaning.

There are basically two methods of preventing & cleaning pressure drop fouling:

Back flushing. The flow direction of the cooling water is reversed. If the condenser is one of many in parallel, the others should be closed, if possible, to increase the flow rate through the one to be flushed. The boosted reverse flow will flush out the debris at the inlet. This method is acceptable if the water does not contain too many fine particles, which can be stuck in the interior of the BPHE.

Filter/Strainer. A strainer with a mesh size of 0.5 to 1.5 mm is installed in the cooling water circuit, preferably before the pumps, which also need protection from debris. The mesh size depends on the water quality and the degree of protection required.

The finest mesh size need not necessarily be the best solution. Although a very fine mesh will keep the heat exchanger clean the trouble might simply be transferred to the strainer, necessitating frequent opening & cleaning. On the other hand, a mesh that is too coarse will shift the problem to the heat exchangers instead. Consequently, if the strainer is easy to open & accessible and it is vital that the condensers operate 24 hr/day, use a fine mesh. If the condensers can be on stand-by for long periods and the strainer is hard to open, use a coarse mesh.

The BPHE, which cannot be opened, should be equipped with the finer mesh size while the normal PHE should have the larger mesh size.

3.2.3. Surface fouling, scaling.

This occurs when the heat exchange surface is gradually covered by a layer of fouling products. The thermal performance of the heat exchanger deteriorates but the pressure drop is not affected, at least not initially. The various types of fouling to be found in normal cooling water can be classified into the following groups.

Sticky products. Oil and grease can enter the cooling water circuit and stick to internal surfaces.

Scaling. Certain inorganic salts, most notably $CaSO_4$ calcium sulphate, have an inverted solubility curve, i.e. the solubility in the water decreases with increasing temperature. Thus when the cold water makes contact with the warm condenser surface, these salts are deposited on the surface. Pure calcium sulphate is very difficult to dissolve, making cleaning difficult or impossible. Fortunately another component of the scaling, $CaCO_3$, calcium carbonate, is easily dissolved by diluted acids.

The composition & concentration of the salts, pH and temperature are the factors that affect the scaling rate. Scaling is seldom found where wall temperatures are below 45 °C, thus making it important only in certain cases (heat pumps, desuperheaters, etc.). See figure 01.

Algae. At certain times of the year, mainly in the summer, water of type, D & E can contain large amounts of algae. If large, they can be trapped by a strainer; if fine, they can pass through the condenser without causing any harm. However, in many cases they can cause trouble. Apart from the decrease in thermal performance, deposits of algae can act as starting points for corrosion (pitting).

If cooling tower water is not treated with an algaecide, the open structure of the tower provides an excellent breeding ground for algae. Algae are not found in closed circuits, as they need light to grow.

Thus fouling from algae always enters from outside the condenser, making it at least theoretically open to interception and treatment.

Bacteria, unlike algae, can grow anywhere and cannot be trapped by filters or strainers, making it impossible to prevent bacterial growth, if favourable conditions exist.

Some bacteria can feed on sulphates in the water and convert them into sulphuric acid, which causes corrosion. Others feed on iron, either dissolved or oxides, forming a coat of a viscous mass. Still others feed on nitrogen or phosphor compounds forming slime on heat exchanger surfaces. As the condenser is the warmest place in a system, bacteria find its surface a warm, cosy place to settle on. Water can look innocent when inspected but still contain bacterial growths, making this type of fouling insidious and hard to detect.

Fungi. Produce similar effects to bacteria but are usually harder to eliminate. The biggest problem with fungi is not the effect on the heat exchanger and other process equipment, but the way they attack the wooden structure of cooling towers.

Deposits of sand, etc. Sand, dust, etc. can settle in the BPHE inlet, but usually only at very low water velocities.

Growth of barnacles, mussels, larvae, etc. is normally only a problem in salt or brackish water.

3.2.4. Prevention and cleaning.

Water treatment is a difficult subject, especially the control of biological growths. Some of the products used are toxic, for humans or for marine life, and are thus subject to governmental control and regulation, and water treatment specialists should be consulted. Cleaning liquids can be corrosive or in other ways hazardous. Thus, it is preferable to use commercial products instead of the single chemicals mentioned here or elsewhere. Alfa-Laval has a series of descaling products and detergents.

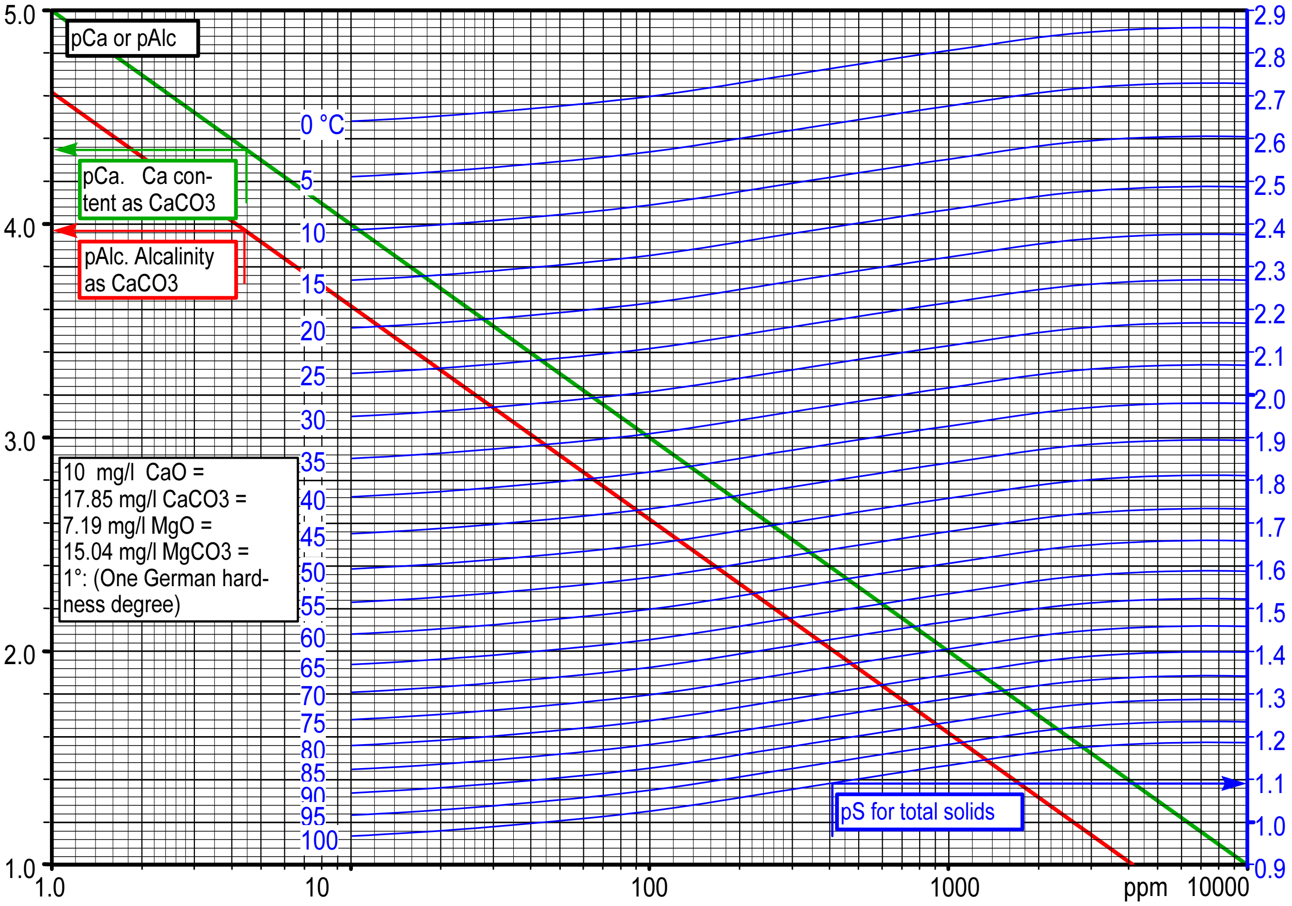
Commercial products usually contain inhibitants to prevent or reduce corrosion, are mixtures of various individual compounds to cover a broader spectrum of microorganisms, are equipped with appropriate warnings, safety precautions and instructions and sometimes a warranty.

A. Normal commercial detergents can be used for cleaning a BPHE condenser. These are effective against oil & grease, bacterial slime, deposits of algae, etc. A CIP system as described under evaporators could be used. In emergency normal household detergents could do. A suitable commercial product is P3 T6560 from Henkel.

Cleaning with detergents is best made in a closed system, as foaming could be a problem in an open system.

Warning 1!! Certain detergents contain ammonia. Because of the copper, these are prohibited for CBPHE.

B. In case of resistive organic compounds, such as deposits of fat and proteins, cleaning with a warm 10 % sodium hydroxide solution could be effective. Do this after a possible acid wash.



Is the water scaling or corrosive?

Langlier's Index:

< 0 Tendency corrosive
pH - pS - pAlc - pCa = 0 Stable
> 0 Tendency scaling

Scaling or corrosive? The two indices above indicate if the water will precipitate calcium carbonate (scaling) or dissolve it (corrosive). The corrosive action is valid for carbon steel and to a less extent, copper but *not for 316 stainless steel*. For this, the chloride ion concentration is the more likely culprit of corrosion. See §5.2.

pH. The measurement of the pH is straightforward and is done routinely. The only problem is that it can vary with the season and the climatic conditions, so some different measurements should be made.

Total solids, pS. The relative proportions of the various salts are fairly constant in naturally occurring water. The ionic strength is thus proportional to the total amount of dissolved solids expressed in ppm.

Treated water, e.g. cooling tower water, process water, waste water, etc., can exhibit totally different behaviour and the simple total dissolved solids cannot be used as a measure of ionic strength. Also excluded are nondissociated, dissolved solids. As remarked before, sugar belongs to this group.

The diagram gives pS as a function of ppm solids.

For orientation only; not for warranty!

Ryznar Stability Index:

= 4 - 5 Large scaling tendency
2*(pS+pAlc+pCa)-pH = 6 - 7 Stable
= 8 - 9 Large corrosion tendency

Alcalinity, pAlc. The bicarbonate alkalinity is determined by titration of the water with an acid and with methyl orange as indicator. The result is expressed in various ways. Here we use the equivalent content of calcium carbonate, $CaCO_3$. Get pAlc from the diagram.

Calcium content, pCa. This is normally expressed as either the concentration of calcium as calcium carbonate, $CaCO_3$ or as calcium ion Ca^{++}. The relationship between the two is:

100 g $CaCO_3$ <=> 40 g Ca^{++}

Ex. An analysis gives the calcium content as 92 ppm Ca^{++}, which corresponds to 230 ppm $CaCO_3$. With this value, get pCa from the diagram above.

Example. Water: pH = 10, 2300 ppm total solids, alcalinity = 100 ppm $CaCO_3$, Calcium = 500 ppm $CaCO_3$. Is it scaling in contact with a surface of 55 °C?

pH = 10, pS = 1.72, pAlc = 2.6, pCa = 2.3

L.I. = 10 - 1.72 - 2.6 - 2.3 = 3.38
R.S.I.= 2*(1.72-2.6-2.3) - 10 = 3.24

Both show large scaling tendencies.

Fig. 01. Scaling and corrosion tendencies of carbon steel.

Warning 2!! Sodium hydroxide dissolves proteins. These not only include bacterial deposits in the heat exchanger but also your skin and especially your eyes.

Warning 3!! Sodium hydroxide will corrode aluminium very rapidly. Apart from destroying the aluminium, hydrogen will develop, which means a danger of explosion.

C. Scaling is best cleaned with commercial products, which usually contain additives to enhance the effect and/or to prevent corrosion. An example of a suitable product is P3 T288 from Henkel.

In an emergency, weak acids such as formic, citric, acetic, oxalic or phosphoric acid can be used. A 5 % solution in water is normally sufficient. After use, do not forget to rinse the heat exchanger carefully with clean water. A solution of 1 - 2 % sodium hydroxide (NaOH) or sodium bicarbonate ($NaHCO_3$) before the last rinse ensures that all acid is neutralised.

Warning 4!! Concentrated acids are corrosive and should be handled carefully. Acetic acid, though innocent on the dinner table as vinegar, causes skin blisters. The author of this manual once was able to remove the skin of his left hand, almost like a glove, after only a few minutes contact with concentrated acetic acid.

Warning 5!! Under no circumstances use strong inorganic acids such as hydrochloric, nitric or sulphuric acid. Under the right conditions, hydrochloric acid can corrode stainless steel in minutes, nitric acid corrodes copper and all, but especially concentrated sulphuric acid, are extremely hazardous.

Warning 6!! When producing a diluted solution of an acid, lye, a detergent or whatever, always pour the compound to be diluted into the water, never the contrary. A little water poured onto a large amount of sulphuric acid will start to boil and splash acid. A little acid poured into water will sink to the bottom and cool off.

D. Fouling from biological growths can be prevented through an appropriate treatment of the cooling water. This is easier in a closed circuit like cooling tower water than for once-through cooling water, but it is sometimes done for this as well. Biological water treatment should always be made in cooperation with a water treatment specialist. Below are some examples of chemicals that can be used for prevention of organic growths.

- ***Chlorophenols***. Effective against a large spectrum of microorganisms, although certain microorganisms can develop resistance.
- ***Potassium permanganate.*** Being a strong oxidant it attacks and decomposes organic material in general, including microorganisms.
- ***Sodium dichromate or sodium chromate.*** These are used mainly as corrosion inhibitors, but being oxidants, they decompose microorganisms as well.
- ***Copper sulphate.*** This should not be used where there are components of mild steel, as the electrolytic deposit of copper from the copper sulphate could cause severe corrosion. See figure 04.
- ***Quaternary ammonia compounds.*** These are organic compounds where one or more hydrogen atoms in ammonia have been exchanged for an organic group. Being closely related to ammonia they could have similar corrosive action on copper. Use only proprietary brands with an assurance from the manufacturer that they are safe to use in equipment containing copper.
- ***Chlorine.*** Chlorine is extremely effective against all type of microorganisms, and resistance cannot develop. It attacks organic compounds in general and some inorganic compounds too. As it is toxic, care should be taken (consult a specialist) when using it.

  Chlorine is a gas at normal pressure and temperature (BP -34.7 °C). It is normally supplied as a liquid in pressurized vessels. As it is heavier than air it can collect on the floor in confined spaces, machine rooms, cellars, ships, etc. Being a strong irritant, its presence will be revealed long before toxic level is attained.

  The first effects of chlorine poisoning are pains in the chest and difficulties in breathing. It is useful to know that breathing some ammonia (e.g. from a bottle of ammonia solution in water) will relieve the pains and facilitate breathing.

  Due to the toxic properties, the use of liquid chlorine is usually limited to large chemical plants. Smaller plants use either calcium hypochlorite or sodium hypochlorite. When dissolved in water the active chlorine compound is the same as for free chlorine dissolved in water.

  The most elegant way to produce chlorine is with sea water, especially on board ships. Electrolysis of NaCl - common salt - produces chlorine on one electrode and hydrogen and sodium hydroxide on the other.

  Chlorine is used not only in cooling water but also in drinking and swimming pool water. In cooling water systems chlorine (or chlorine salts) are used either for continuous or shock chlorination.
- ***Continuous chlorination***. A concentration of 0.2 - 0.3 ppm is sufficient. This is consistent with stainless steel and copper used in CB heat exchangers.
- ***Shock chlorination***. The period of chlorination and the frequency has to be established through experience, as the properties of the cooling water vary greatly from place to place and with the season. Usually a period of about half an hour for 4 - 6 times a day, at a chlorine concentration of 1 - 2 ppm after the heat exchanger, is a good starting value. The total chlorine consumption is less for shock than for continuous dosing, an important environmental factor. A disadvantage of shock chlorination is that certain types of marine organisms - e.g. barnacles - can learn to anticipate the chlorine shock and close their shells, rendering chlorination ineffective. In such cases, chlorination should be carried out at unequal time intervals.

  Warning 7!! Free chlorine in combination with the chlorine ion can be extremely corrosive on all types of stainless steel and some other high quality alloys, e.g., Hastelloy C. Recommendations for the use of chlorine in stainless steel heat exchangers can be found later in this chapter, in the section on corrosion. d copper used in CB heat exchangers.
- ***Ozone*** - chemical formula O_3 - is a gas, which in higher concentrations is toxic, in lower irritating. It is the smell, which is felt close to large electric motors and during thundering. Like chlorine, it is highly reactive to organic material and thus kills all types of microorganisms. Resistance to ozone cannot be developed.

  The advantage over chlorine is that the decomposition products are more innocent than the organic chlorine compounds resulting from the use of chlorine. It is, however, more expensive to produce and difficult to transport.

  The efficiency in water and seawater seems to increase, the colder the water is.

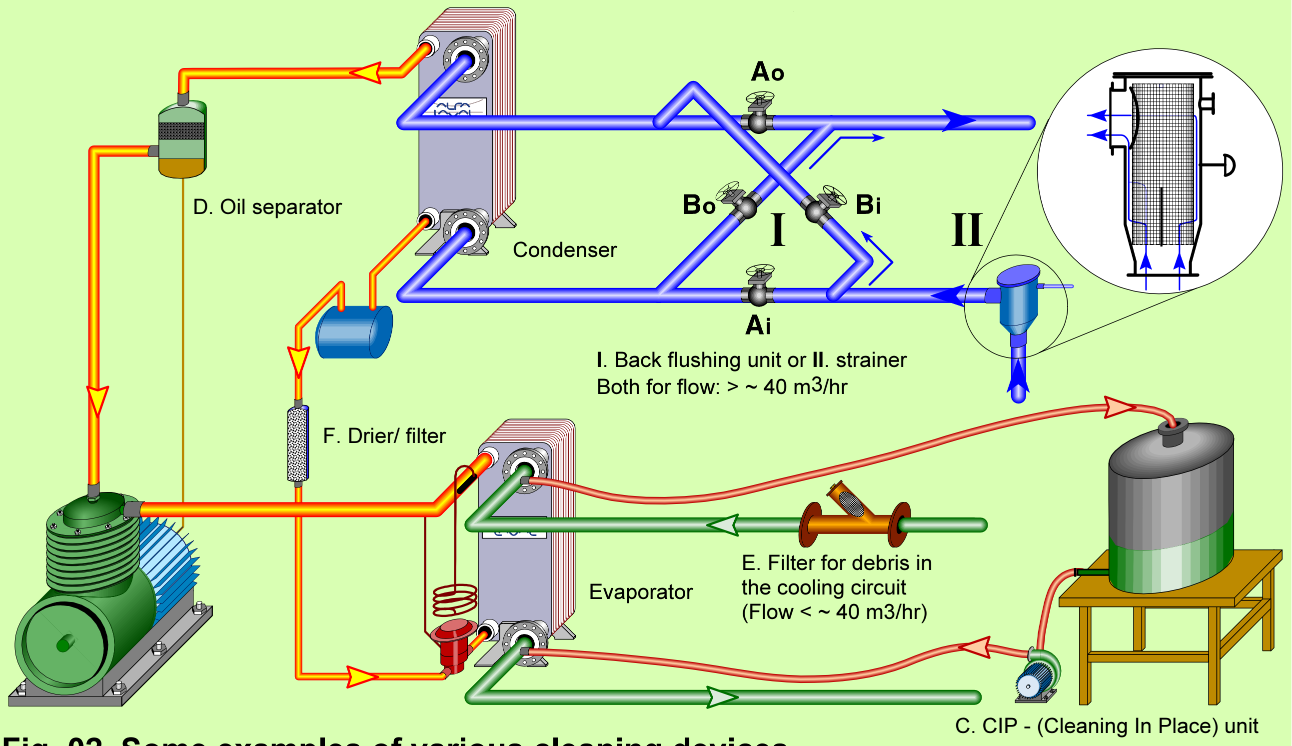


Fig. 02. Some examples of various cleaning devices.

A. Backflushing unit.

Normal: Valves Ai & Ao open, Bi & Bo closed.
Back flushing: Valves Bi & Bo open, Ai & Ao closed.

The dashed arrows show the flow direction during the back flushing.

B. Alfa Laval strainer. The small side stream is the waste water flow during the filter regeneration period.

C. CIP (Cleaning in Place) unit. The hoses from the CIP unit are connected to the second pair of nozzles. The cleaning liquid should have reverse flow direction compared to the normal flow direction.

D. Oil filter.

E,. Filter for debris in the cooling circuit.

F. Filter/Drier for the refrigerant.

4. Design recommendations.

The PHE heat exchanger provides good resistance to ***surface fouling*** due to:

- The plate corrugation causes a very high turbulence, even at low flow rates. Turbulent flow is important to keep suspended particles from settling which they do more readily with laminar flow.

  Ex. The shear forces are proportional to the pressure drop per meter flow length. A coaxial HE and a BPHE both have a pressure drop of 0.5 bar for given equal water flow rate. The coaxial, however, has a total pipe length of 6 m as compared to the brazed 0.6 m. Therefore; the BPHE generates shear forces ten times as great as the coaxial.
- No dead zones where fouling compounds can settle.
- A high surface finish prevents particles from attaching to the heat exchanger surface.
- The copper in a copper brazed unit creates an unfriendly environment for microorganisms.
- It is, however, susceptible to fouling by sticky products such as certain types of clay, slime, and mud and to fibrous substances.
- The PHE is susceptible to ***pressure drop fouling.*** It can, however, be easily controlled by the two methods previously mentioned; back flushing or filter/strainer.

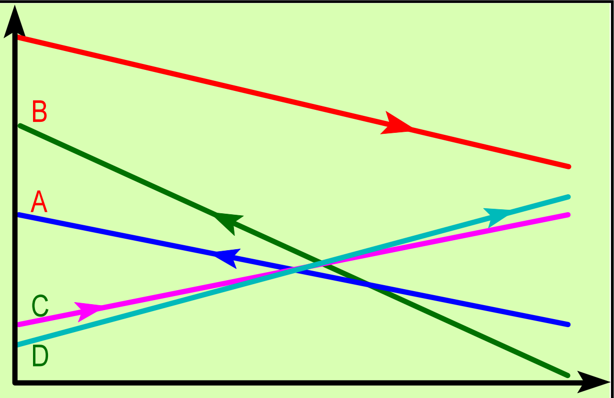


Fig. 03. Design to minimize scaling.

A. In counter current, the warmest part of the water meets the warmest part of the hot fluid resulting in a high wall temperature and a scaling risk.

B. This risk is increased when throttling the water flow at a low entrance temperature.

C. Cocurrent, reverses the condition in A.

D. The highest temperature of the water is lower than the exit temperature of the warm fluid.

A PHE operating with potentially fouling fluids should be oversized to accommodate the capacity decrease due to the fouling. Unfortunately, it is difficult to say how much, as fouling tends to be either everything or nothing.

If the fouling consists of debris, these should be prevented from entering the PHE. If so the PHE could operate infinitely, if not, the PHE could clog in some days.

It is impossible to give a general recommendation of a suitable addition of surface to accommodate the fouling.

Fouling due to scaling or due to growth of microorganisms such as slime can be kept in control by previous mentioned methods. Here too general recommendations of surface additions are impossible to give. See also § 4 in chapter **2. Optimization**, for the effect of surface margins under various conditions.

Recommendation 1: Use as high a water pressure drop (i.e. shear stresses) as possible.

Note! If the main source of fouling is debris, it might be questionable to increase the shear stresses by increasing the water flow rate. A larger amount of water, with potentially more debris then enters and the result could be more clogging. In case of scaling and slime formation though, large shear stresses are always beneficial.

Recommendation 2: For a heat exchanger with a very high temperature (> 70 °C) on the hot side and/or very hard cooling water (and danger of scaling), increase the pressure drop as much as possible on the cooling water side and reduce it on the hot side.

This reduces the wall temperature of the cooling water side and increases the shear stresses, thus making it more difficult for the scaling compounds to adhere.

Recommendation 3: Consider the use of cocurrent flow instead of counter current. The warmest part of the hot side, the entrance, will then face the coldest part of the cold side. This will normally lower the maximum wall temperature on the cooling water side and automatically limit the exit water temperature. See figure 03.

Recommendation 4: The normal practice is to let the cooling water enter the lower nozzle. This arrangement should be used whenever possible since cooling water entering through the upper nozzle could encourage debris to enter the channels.

Recommendation 5: In certain cases it might be advantageous to place a condenser horizontally or almost horizontally (with the individual plates vertically). There is little or no decrease of the condensing capacity.

However, sand and other particles settle more easily in such an arrangement and clog the heat exchanger. Ideally, keep the normal vertical position for water containing particles that settle or increase the pressure drop.

5. Corrosion.

Corrosion is the nature's revenge on the engineer, who transforms metals from their chemically stable combined state as ore, into their elemental forms, which are usually less stable once they make contact with air or water.

Corrosion in a refrigerant plant is largely confined to the liquid side of the heat exchangers, mainly to the condensers, which have the highest temperatures and the worst water quality. However, corrosion on the refrigerant side and in the evaporator can occur as well.

5.1. Refrigerant side corrosion.

5.1.1. Refrigerants.

No refrigerant, whether hydrocarbons (including halogenated ones) or ammonia, corrodes stainless steel or copper (apart from ammonia). The exception might be in the ***production*** of halogenated hydrocarbons as the product then could contain small amounts of acids.

Copper and copper containing alloys corrode in ammonia if water is present in more than some ppm. See also section **5.3.7. Water containing hydrazine.**

5.1.2. Decomposition of refrigerants and oils.

Under normal circumstances, refrigerants and oils do not corrode the equipment. In certain conditions, however, the HCFC will decompose and the chlorine and fluorine found in most refrigerants together with hydrogen will form highly corrosive hydrochloric and hydrofluoric acids.

The HCFCs are probably worst in this respect. What makes them environmentally acceptable is that they, unlike the "old" refrigerants (CFCs), will break down more readily in the atmosphere. The reason for this is that the HCFC molecule contains not only carbon, fluorine and chlorine but also the hydrogen that is necessary to form the acids mentioned previously, i.e. HCFCs contain this in the molecule, the CFCs do not. Consequently, HCFCs will probably decompose more easily in a refrigeration plant as well. Decomposition occurs more readily in the presence of oxygen.

A second condition necessary for corrosion is the presence of water. Completely dry hydrochloric and hydrofluoric acids are not very corrosive but in a water solution, they form some of the strongest acids known.

A third condition is high temperature. In the temperature range found in most refrigeration plants (< 100 °C) the danger is small, but in the presence of catalysing components, the decomposition rate could accelerate. Oxides of nickel, chromium, molybdenum, vanadium, etc. could act as catalysts. These oxides can be formed by welding or soldering of stainless steel.

Oxidation caused by welding or soldering should always be prevented on the refrigerant side of an exchanger.

A second source of corrosive compounds is formation of organic acids during the decomposition of oil. This too is greater in the presence of water. Mineral oils are usually very stable and give few problems in this respect. Some of the newer synthetic oils contain molecules with double bonds. Under some conditions, these double bonds can be very reactive and with water (and oxygen) among other things form organic acids. Polyolester oils seem especially sensitive. Some newer polyether oils seem to be more stable. Decomposition products of oil could also reduce the lubricating properties of the oil.

A third source of corrosive products is the flux from soldering, which has accidentally entered the refrigerant circuit and been overlooked. The properties of fluxing compounds - removing oxides from the metal surface - make them very potent corrosive agents.

Ammonia is a special case. Polyolester (POE) oils react vigorously with ammonia while the generally more stable mineral oils and the polyalkylene glycols (PAG) do not. The thermodynamic properties of ammonia mean a high exit temperature from the compressor, i.e. danger of oil decomposition. This might cause problems with lubrication and encourage the formation of fouling sludge but not corrosion. The ammonia effectively neutralizes acids formed during the decomposition of the oil.

5.1.3. Prevention of decomposition products.

- Check driers regularly.
- Limit the exit temperature from the compressor.
- Check filters. If unusually clogged, it might be a sign that oil decomposition has occurred.
- Use nitrogen protection to prevent oxidisation when brazing or welding on the refrigerant circuit. The easiest method is simple to blow nitrogen inside the pipes and equipment being welded.
- Take care when soldering and be sure to inspect and clean the refrigerant side.

5.2. Corrosion in water solutions.

There are three aspects of corrosion when using copper brazed stainless steel units:

- Corrosion of the stainless steel type 316 (2.5 % Mo).
- Corrosion of the copper.
- Interaction of the copper with connecting equipment.

5.2.1. Corrosion of stainless steel.

Passivation. The corrosion resistance of stainless steel depends on the formation of a thin oxide film on the steel surface. This film - mainly chromium oxide - forms very easily in an oxidising environment, e.g. air, and protects the underlying metal surface from corroding. A metal protected in this way is said to be passivated.

It is important that the oxide film is unbroken, free from pores and crevices, etc.; otherwise, corrosion could occur on the unprotected spots. The surface should be free from scaling, slag remnants, etc. If the film is damaged and conditions do not favour the formation of a new oxide film, the surface remains active and unprotected and corrosion can occur.

Reducing components - sulphurous acid & hydrogen sulphide - impair or destroy the Passivation, while dissolved oxygen, nitric acid, Cu^{2+} and Fe^{3+} improve it. This oxide film offers very good resistance against oxidising acids, such as nitric, chromic, weak sulphuric acid, etc. On the other hand, resistance against nonoxidising acids, e.g. hydrochloric acid, is very low.

General corrosion. This is uniform corrosion over the entire surface and generally occurs in acids or bases. This corrosion is predictable, i.e. if certain equipment has corroded 0.5 mm in one year; it is likely to corrode 1 mm in two years, thus making it easy to deal with. This type of corrosion should not occur in refrigeration plants.

Galvanic corrosion. This type of corrosion occurs when two metals of different electrochemical potential are in contact in an electrically conductive solution, e.g. an aluminium boat in sea water, repaired with a copper plate (which happened in the early days of aluminium boats).

A galvanic element ("a battery") then forms, where the less noble metal will form the anode and the other the cathode. The anode dissolves, i.e. corrodes. In principle

This is valid for copper on stainless steel where, being the less noble should corrode, but in practice, this is seldom a problem in water solutions. In condition of poor water quality, the more severe pitting or crevice corrosion will probably occur first. Air coolers or air-cooled condensers are often made of copper tubes & aluminium fins, and in a marine atmosphere, corrosion could occur. In these circumstances, copper fins are the better option.

Pitting and crevice corrosion are basically the same phenomena. Conditions that favour this corrosion are:

- An oxidising environment, e.g. presence of oxygen, nitric acid or the Cu^{2+}, or the Fe^{3+} ions.
- Presence of chlorine ions, even in low concentration.
- A weak spot on the passive surface film where the chlorine ion can penetrate. This could be a crevice between the gasket and the tightening surface of a flange, a badly executed weld or an overlap joint. . Pitting occurs on an open surface, below a spot covered by slag remnants, scaling, or a bubble.
- The penetrated spot has to be low in the oxidizing component.
- Low pH. At pH>12 the danger of localised corrosion is negligible (instead the danger of scaling will be high).
- Stagnant liquid. This is an important requirement. It would for instance be possible to use stainless steel in sea water, if a constant water stream always swept all surfaces in contact with the sea water. Normally there is always some stagnant spot, and when the heat exchanger is closed down all the surfaces would be susceptible to pitting.
- High temperature. In an open system (atmospheric pressure) and if oxygen is the only oxidising component, a high temperature (> 80 °C) reduces the oxygen content, thus reducing the risk.

If these conditions occur, then a local galvanic element forms with the penetrated area as an anode and the surrounding passive area as the cathode. As the cathode is large and the anode small the current density will be high in the anode, which then dissolves rapidly - several mm/day.

As many different factors can contribute to crevice corrosion, it is very unpredictable and potentially dangerous. Since the right conditions are more likely to exist in a crevice than on an open surface.

Stress corrosion is similar to pitting & crevice corrosion. A surface under tensile stress in a chloride solution forms the anode. The temperature has to be higher than that normally found in refrigeration plants or heat pumps.

5.2.2. Prevention of corrosion on stainless steel.

- Stress corrosion and crevice corrosion are seldom found in BPHE heat exchangers. Brazing effectively relieves stress in the material and the melting copper flows into any crevices.
- Pitting will not normally occur in an operating heat exchanger. In case of a prolonged shutdown of a condenser operating with poor quality water, the unit should be rinsed with clean water and either filled with water or drained and dried. In a unit where some water remains, this could evaporate and increase the chloride concentration until conditions for pitting occur.
- Be careful when using poor quality cooling water. Contaminants, such as hydrogen sulphide and sulphur dioxide, will enhance the effect of the chlorine.
- The pH should normally be between 7 and 9. A too high pH could cause scaling.
- Rinse carefully after chemical cleaning.
- Keep the surface clean. Fouling & scaling might contain corrosive compounds and create crevices and other surface imperfections, where corrosion can start.

5.2.3. Corrosion of copper.

Copper belongs to the noble metals and does not corrode in contact with most other common fluids. Nevertheless, most types of corrosion can occur on copper as well.

General corrosion I occurs mainly in acids, e.g. nitric acid, which simply dissolves the copper. This corrosion should not occur in a refrigeration circuit for the simple reason that acids do not belong to a refrigeration circuit.

General corrosion II occurs in water containg oxygen, i.e. in most natural water. The copper oxidizes and after a couple of months is covered by a protective layer of black cuprous oxide (Cu_2O), which prevents further corrosion. Depending on the water quality, the cuprous oxide is then covered by a layer of usually green copper carbonate (attaches well, thus prefered) or sulphate (loose).

Warm water pitting, occurs in soft water with a low pH, at temperatures above 60 °C. The pitting is usually covered by amorphous copper sulphate but there can be large variations or even absence of covering. This type of pitting is not very common.

Cold water pitting, occurs mainly in oxygen rich water at temperatures below 30 °C. The pitting is covered by Cu_2O and below this also white CuCl. A voluminous layer of green copper carbonate usually covers the pitting area.

Erosion/corrosion. For a large water velocity, the protective Cu_2O layer errodes and the bare surface corrodes.

5.2.4. Prevention of corrosion on copper.

- Low oxygen concentration. If less than 0.1 ppm, pitting does not occur.
- High content of HCO_3^- & Cl^- ions ***decreases*** the danger of corrosion.
- High content of SO_4^{2-} & NO_3^- ions ***increases*** the danger of corrosion.
- $6.5 < pH < 9.5$.
- $CO_2 \leq 44$ ppm (1 mol/m^3).
- Surface water instead of well water. The organic compounds in surface water act as natural inhibitors.
- Organic material containing ammonia and sulphur compounds ***increases*** the corrosion danger.
- Magnetite (Fe_3O_4) particles increases the danger.
- No crevices or surface irregularities, no stagnant water.
- A carbon film (e.g. from decomposition of oil after soldering) on the surface ***increases*** the corrosion danger.

5.2.5. Interaction of copper with the connecting equipment.

Galvanic corrosion. Being a noble metal, copper can cause corrosion on less noble metal such as unalloyed iron and galvanised iron. For this to happen the copper has to be in electric contact with the other metal.

Micro galvanic corrosion. At normal conditions, copper in a BPHE corrodes a little. The corrosion rate is small enough to be unimportant for the lifetime of the BPHE. , However, the dissolved copper precipitates as metallic copper on unalloyed or galvanised iron. This is the well-known phenomenon of coating an iron nail with copper when it is dipped in a copper salt solution.

These areas of metallic copper on iron or zinc will act as a galvanic element as described above. The precipitation will cover a large area and can cause rapid corrosion on the iron or, more important, the zinc layer of the galvanisation. This can occur even though the copper containing equipment is remote from the iron.

5.2.6. Prevention of corrosion on mild steel and zinc by copper.

- Galvanic corrosion can be prevented by electrically insulating the BPHE from surrounding equipment and/or ***galvanised iron or carbon steel should never be installed downstream of the BPHE unit.***
- Micro galvanic corrosion cannot be prevented by electrically insulating the equipment. Here the concentration of corrosive compounds has to be within the limits specified in table 02. However, it does not happen on equipment upstream of the BPHE unit.

Warning 8!! In a closed loop system, the concepts of upstream and downstream have of course no meaning, i.e. galvanic corrosion can occur on equipment placed immediately before the BPHE unit. However, there must be electrical contact for corrosion to occur.

5.3. Water types.

5.3.1. Chlorinated (swimming pool) water.

A particular problem is the heating of chlorinated water as in swimming pools in the condenser of a heat pump. It is especially a problem for stainless steel, copper is considerably less sensitive to chlorine corrosion.

As chlorine is added and transformed to chloride ions, the concentration of chloride ions gradually increases. After some time, the chloride ion concentration could be high enough for the chloride ions together with the free chlorine to cause pitting, see figure 05 for limits.

It is easier to supervise the chlorination in large public pools than in small private as public pools usually have a trained maintenance engineer, which private pools seldom have. The larger the pool is, the more constant the conditions are and it more difficult to accidentally overcharge the pool with pellets.

Note that indirect systems - the pool water is heated by another water circuit, not directly by the refrigerant - are inherently safer than direct systems, especially for small, difficult to control pools.

However, there are installations of PHEs in public pools running without opening for more than ten years.

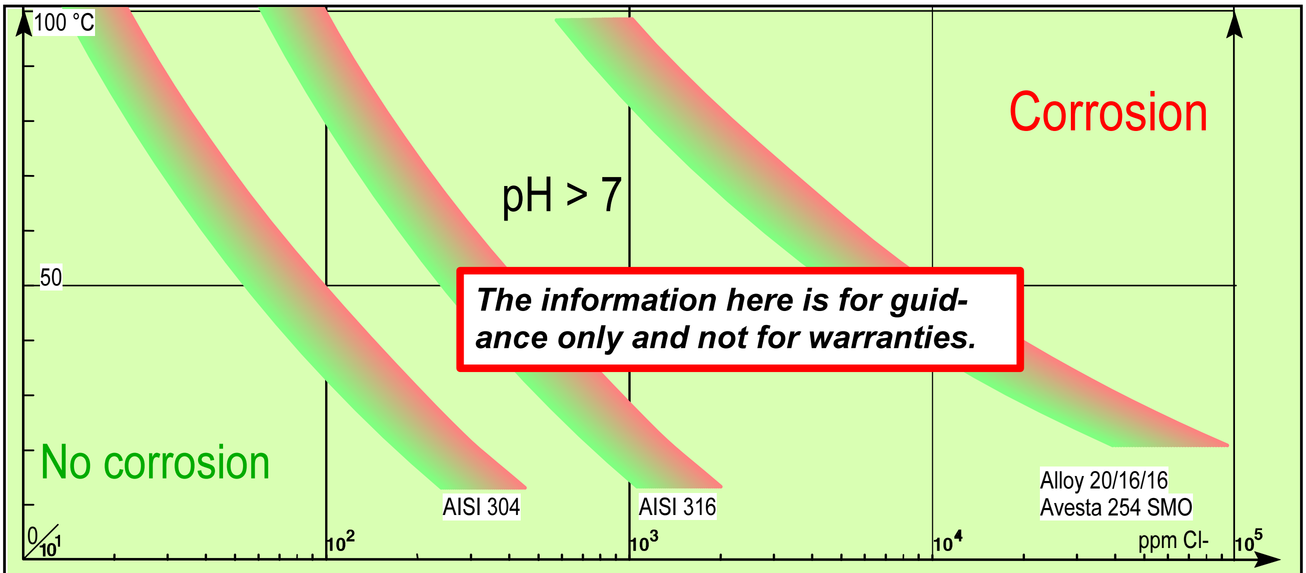
Warning 9! Higher quality stainless steels are only marginally more resistant to Cl_2/Cl^-.

5.3.2. Calcium chloride and lithium bromide.

Concentrated calcium chloride solutions at high pH and low temperature (< ~0°C) are not corrosive to stainless steel. If the solution is properly inhibited, e.g. with bichromate or molybdate, the same is valid for copper. Pitting can happen, when the plant stops and the temperature increases and especially if the pH of the solution is allowed to decrease, e.g. after an improper rinsing with water. Lithium bromide has similar, but less aggressive, properties. Pitting is a rapid process and the consequences of pitting in an evaporator could be disastrous.

Warning 10! Take extreme care when using calcium chloride or lithium bromide solutions and use only inhibited commercial solutions, which are expressly stated to be compatible with copper and stainless steel.

- Keep the pH between 8.5 and 11.5, but not more in case of calcium chloride, otherwise scaling might occur.
- Consider the addition of potassium bichromate or molybdate as inhibitors.
- Use it only in a closed, air free environment.



The information here is for guidance only and not for warranties.

Fig. 04. The effect of chloride ion concentration and temperature on pitting and crevice corrosion tendencies on stainless steel.

Other factors which influence these corrosion types are:

- ♦ pH value.
- ♦ Water velocity.
- ♦ Fouling, both scaling and biological.
- ♦ Impurities, such as CO2, H2S, SO2, native iron etc.
- ♦ Surface condition.
- ♦ Redox potential of the solution, which depends on oxidizing compounds, i.e. oxygen, hypochlorous acids, sodium bichromate, etc.

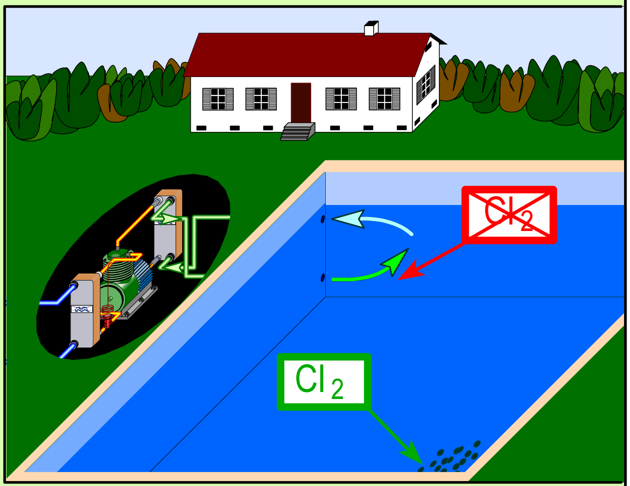
- ♦ The ratio $HCO_3^-/SO_4^{2-} > 1$, otherwise risk of pitting.
- ♦ pH should be 7.5 - 9.0
- ♦ Oxygen content as low as possible, < 0.1 ppm.
- ♦ Ammonia < 0.5 ppm, If oxygen present
- ♦ Ammonia < 30 ppm If no oxygen present
- ♦ Compounds such as hydrogen sulphide corrode copper and impairs the corrosion resistance of stainless steels.

  See also Table 2 for micro galvanic corrosion. Note that microgalvanic corrosion cannot be prevented by electrically insulating the various parts of the circuit

Table 1. Corrosion limit on copper.

pH	7.5 - 9.0			
SO4--	<	100	ppm	
HCO3-/SO4--	>	1.0	ppm	
Total hardness	4.0 - 8.5		dH	
Cl-	<	50	ppm	
PO43-	<	2.0	ppm	
NH3	<	0.5	ppm	
Free Chlorine	<	0.5	ppm	
Fe+++	<	0.5	ppm	
Mn++	<	0.05	ppm	
CO2	<	10	ppm	
H2S	<	50	***ppb***	***Note!***
Temperature	<	65	°C	
Oxygen content	<	0.1	ppm	

Table 02. Concentration limits of impurities in water to prevent microgalvanic corrosion of unalloyed or galvanized iron in the *adjacent piping.*



- ♦ Never put the chlorination point just before the heat exchanger; it should be as distant as possible.
- ♦ The water intake should be close to the surface and the exit at the bottom. This improves the mixing of the heated water and above all, it prevents entrance of chlorine pellets or concentrated solution into the HE.
- ♦ ***Warning 11 !*** Unfortunately, in practice the pellets are sometimes put in immediately before the HE intake. This improves the chlorination but with potentially disastrous result for the HE.
- ♦ Corrosion of stainless steel can occur for chloride concentrations far below that which figure 04 indicates.
- ♦ Keep the pH as high as possible; at least > 7.5.
- ♦ Cl_2 < 0.5 ppm (continuous) at the BPHE entrance.
  < 2 ppm (shock).
- ♦ Cl^- <150ppm, if the water is heated to 50-60°C.
  <100ppm, 70-80°C.

Fig. 05. Swimming pool chlorination.

5.3.3. Demineralised, distilled or soft water.

"Can distilled water be used?" is a common question. Copper in microscopic amounts is always released into the water, regardless of how clean it is. This has no bearing on the lifetime and function of the BPHE but it is important to know why the water is demineralised:

- Sometimes a closed loop water circuit contains demineralised water just to prevent fouling and corrosion on equipment. In such a case, the use of copper in equipment presents no problems.
- In certain applications, notably some types of laser cooling units, even microscopic amount of copper can disturb the functioning of the laser.
- Conceivably other types of application, such as pharmaceutical and electrochemical, could be affected by the presence of copper.

Corrosive properties of ion free water. Water with a low content of dissolved minerals usually has a very low buffering capacity, i.e. the pH changes rapidly.

A droplet of concentrated hydrochloric acid, added to a litre of demineralised water of pH 7, decreases the pH to about 3 or 4, i.e. to a very acid and corrosive water.

The same acid droplet, added to a concentrated calcium chloride solution, again of pH 7, hardly changes the pH.

Warning 12! It is important, especially after cleaning of a unit with acid, to carefully rinse it and check the pH. This is always valid but the more so the less dissolved salts and minerals the water contains.

5.3.4. Softened water.

If the water is hard (§ 5), i.e. contains appreciable amounts of calcium and magnesium ions, it has to be softened; otherwise, there is a danger of scaling.

Softening of water is a sort of "half way" demineralisation; only the scaling ions, calcium and magnesium are removed. This is a standard treatment for tap water, cooling tower make-up water and cooling water. Such treated water makes a good cooling water for all types of PHEs.

5.3.5. Surface, well and cooling tower water.

- Practical experience with copper brazed heat exchangers shows that most types of surface water, such as tap water, river or lake water are not corrosive.
- Most cases of corrosion have occurred with well water and of course brackish water. A careful examination and possibly corrosion tests should be made if no previous experience from that particular water exists.
- Cooling tower water, with bad make-up water can be corrosive. The make up water for the cooling tower water is concentrated about five times. A well water containing some chloride ions might not be corrosive but with a five times higher concentration it might be.

  Warning 13! It is important to clarify if a water analysis is made for the make-up water or the circulating water.

5.3.6. Water containing hydrazine.

Hydrazine - chemical formula N_2H_4 - is sometimes added to circulating water (district heating, air conditioning, power stations, etc.) as a corrosion inhibitor for iron. Its main action is the reduction of dissolved oxygen. It reacts with oxygen according to the formula:

$N_2H_4 + O2 \rightarrow 2H_2O + N_2$

This is not the only action of hydrazine. It also reduces the soluble cupric oxide (CuO) to the insoluble cuprous oxide (Cu_2O), which forms a fairly stable passivating film.

$N_2H_4 + 4CuO \rightarrow 2Cu_2O + 2H_2O + N_2$

Conclusion 1. The reactions above contribute to the very low corrosion - close to zero - of copper in water containing hydrazine. Practically no copper ions are released into the water providing that there is sufficient hydrazine in the water to reduce all the oxygen and cupric oxide (CuO).

A possible negative effect of hydrazine in water is the danger of formation of ammonia (NH_3) with subsequent corrosion of equipment containing copper. Hydrazine decomposes above temperatures of 200 - 250 °C, i.e. not at temperatures normally occurring in refrigeration circuits.

$3\ N_2H_4 \rightarrow N_2 + 4\ NH_3$

Fortunately, an ammonia water solution (NH_4OH) only dissolves (as a complex) cupric oxide (CuO), neither metallic copper nor cuprous oxide (Cu_2O).

$CuO + 4NH_4OH \rightarrow Cu(NH_3)_4(OH)_2 + 3H_2O$

Cupric oxide (CuO) is formed by oxidation of copper or cuprous oxide (Cu_2O) by oxygen.

$2Cu + O_2 \rightarrow 2CuO$ and $2Cu_2O + O_2 \rightarrow 4CuO$.

Because cupric oxide (CuO) is reduced by hydrazine to cuprous oxide (Cu_2O) there will be no corrosion even at ammonia concentrations as high as 20 to 30 ppm as long as the oxygen content is zero. If the oxygen content exceeds zero (no excess hydrazine) the ammonia concentration has to be limited to 0.5 ppm. Ammonia only corrodes copper in the presence of both water and oxygen.

5.3.7. Solutions with inhibitors.

Water solutions of secondary refrigerants - glycols, calcium chlorides, acetates, formates, etc., contains inhibitors. It is important to use them at their intended concentration and not dilute them, otherwise the inhibitor concentration could fall below the effective range and corrosion occur. ***Follow the manufacturers recommendations.*** See also §3.10 in **8. Oils and refrigerants.**

5.3.8. Water analysis.

A water analysis to determine if a particular water is suitable for a certain material or material combination should contain the concentration of at least:

♦ pH	(Acidity)
♦ Cl^-	(Chlorides)
♦ S^{2-}	(Sulphides)
♦ NH_3/ NH_4^+	(Ammonia compounds
♦ SO_3^{2-}/SO_4^{2-}	(Sulphur oxides)
♦ Fe^{3+}, Fe^{2+}, Fe	(Iron compounds)
♦ O_2	(Oxygen & oxidizing components)
♦ CO_3^{2-}	(Carbonate)
♦ HCO_3^{1-}	(Bicarbonate)
♦ Ca^{2+}, Mg^{2+}	(Calcium & Magnesium)
♦ Total solids	Content, size, aspect (fibres, grainy, sticky, slime, etc.)

Unfortunately, water is a living substance and the concentration of corrosive substances might change with season, weather (rain storms, dry spells), flooding etc.

Several water analyses at different seasons and different meteorological conditions are necessary.

The value of a water analysis made after a suspected corrosion leakage has occurred is also doubtful, especially if the unit has been in operation for many years. A single cleaning with a corrosive liquid could have corroded the unit but did not actually do it. Several years later, a further cleaning, leading to a temporarily lower pH and/or higher oxygen content or the like, could start the corrosion again but now the surface is penetrated.

That the concentrations limits have been exceeded do not show up in a water analysis several years later.

6. Leakage in a BPHE.

6.1. Causes of leaks.

- ***Corrosion.*** This can occur on the entire surface, but tends to predominate on the lower part, close to the porthole. This is because some liquid can remain in the porthole of a closed-down unit and corrosive compounds then concentrate here. Corrosion in a BPHE is usually pitting/crevice corrosion on the stainless steel.

- ***Freezing.*** This occurs on the water side at a point where the wall temperature is lower than the freezing point. The point with the lowest temperature in a DX evaporator is normally some distance from the refrigerant inlet, but freeze damage normally occurs on the water side close to the refrigerant inlet, as the conditions for forming a closed-off space are greater here.

Excessive torque and/or forces on the connections. The connections are kept in place by the double action of being expanded into the frame plate and brazed to it. However, excessive force can cause ruptures.

- ***Vibrations from the surrounding equipment.*** This can cause fatigue in the material, therefore with rupture of the plates and/or brazing. The problem will most likely occur in combination, e.g. with excessive forces on the connections.

- ***Water hammer or other pressure shocks.*** The amount of water flowing in a circuit represents a considerable kinetic energy. If the movement is suddenly stopped, e.g. by a fast-closing valve, the energy has to be captured somewhere. This may result in deformation and rupture of the confining surfaces.

  In a BPHE, the most likely damage is a deformation and, after repeated shocks, rupture of the end plate opposite the water inlet nozzle. Thus, a valve in the water pipe after the HE must never close rapidly. There is less danger on the refrigerant side since the vapour phase is compressible and absorbs the shocks.

- ***Thermal shocks.*** If cold water suddenly enters a very hot BPHE or vice versa, the thermal stresses between the first channel plate and the end plate might cause ruptures, as the thin channel plates expand/contract more rapidly than the end plate.

- ***Material defects.*** Most defects are discovered during pressure or helium tests of the BPHE, but some, e.g. slag inclusion in the plate, might turn up later.

- ***Manufacturing defects.*** As with material defects, most are detected at the testing of the BPHE. Sometimes a crack caused in the pressing might be covered with copper during the brazing and show up later, mostly in a corrosive environment. The brazing might be incomplete and, most likely in combination with some other factor, might be the cause of leakage.

- ***Exceeding the test pressure and/or the design temperature.*** This is rare though, as the BPHE will stand a pressure of at least five times the design pressure and the melting point of copper is 1083 °C.

6.2. Leak seeking.

6.2.1. General.

Most leaks in a BPHE are on the heating surface because of one of the factors described above. This means that the leakage will most likely be internal, i.e. one fluid will leak into the other. Apart from possibly causing damage, the exact place of an internal leak is difficult to pinpoint. It is however important to know exactly where a leak is situated, in order to discover the cause.

For this to be done successfully, knowledge of the operating environment and an investigation of the BPHE is necessary.

6.2.2. Inspection of the system.

- Check the pump-down procedure & temperature in an evaporator. Is the pressure controlled in the condenser? A decreased condensing pressure, most likely in the winter, could force the evaporation temperature to decrease. Is there a separate pressure controller for the liquid receiver?

- Check the shutdown and startup procedure and the temperature changes, if thermal shocks are suspected. Is cold fluid suddenly entering a hot BPHE or vice versa? Which is switched on/off first; the cold, the hot side or both simultaneously? Will a shut down BPHE reach the entrance temperature of one of the media?

- Check for vibrations from other equipment. How are the pipes attached to the BPHE? Are there any bends or bellows, which can take up forces or vibrations?

- In case of parallel-connected compressors or BPHEs, sudden pressure and temperature surges could occur when one machine is suddenly started or closed down. Do all the BPHEs have their own pressure controller?

- Are motor valves or solenoid valves used on the water side? Beware of solenoid valves after the BPHE. These could cause liquid hammering.

- Are valves, which achieve semi-continuous operation by modulating the opening time, used? Such valves could open for one second, close for five seconds, change to opening for five, and closing for one second, and so on. These could cause pressure and temperature transients.

- Does the water contain excessive amounts of chlorine ions or other corrosive compounds? Try to get a water sample. Be aware that water in alimentary installations is often chlorinated.

  As the chlorine is consumed, more is added, and soon there is an excessive amount of chlorine ions. This could lead to pitting.

6.2.3. External inspection.

Strip the BPHE of insulation and inspect the exterior.

- Is there a bulge on the cover plate, opposite the water inlet nozzle? This could indicate liquid hammering.
- Is there any hint of deformations on the sides?
- Are the nozzles firmly attached?
- Check both sides for fouling and traces of corrosion.
- Check for transport or installation damages.

6.2.4. Locating the leak.

The simplest method is to fill the BPHE with water on one side and connect a pressurized air supply to the other. The bubbles emerging from one or two of the nozzles will reveal a leakage. In case of an external leakage, the BPHE has to be immersed in water, or a soap solution could be applied at the suspected location.

If the leak is close to a porthole, (1 in figure 06 B) it might be easy to locate. The exact location of other leaks (2 in figure 06 B) is harder to determine.

Figure 06 shows a method, which could be used. This method, however, assumes that the leak is large, i.e. the pressure drop over the leak should be minimal. More sophisticated methods use helium or another tracer gas to detect a leakage.

6.2.5. Cutting the BPHE.

When the leakage has been located, the BPHE has to be cut into pieces for inspection. Before that is done, the position of the leak has to be pinpointed.

Once a piece has been cut out, it might be difficult or impossible to establish if there is a leak at all, let alone its position. This is particularly true if a leak needs a certain pressure for the bubbles to emerge. This could happen if the edges of a crack have to be bent apart by the air pressure before the crack opens and air can leave.

Start the cutting as indicated in figure 06 B in the order A, B, (& C, if the leakage is close to the port). Use an endless saw and keep the piece cooled and lubricated. Reversing saws are less suitable as they deform the plates.

The ideal situation would be to be able to cut out a piece, which clearly shows the type of damage. This could happen in case of freezing, which normally deforms a number of channels in addition to rupturing one of them.

Sometimes the leak is more elusive, but normally it is possible to cut out a piece where the leak is situated. If the cut out piece clearly shows leaks or deformations, then proceed to § 6.3; otherwise the brazing of the piece has to be dissolved in nitric acid; see below.

6.2.6. Dissolving the BPHE in nitric acid.

In order to dissolve the copper brazing, the piece is immersed in nitric acid. Nitric acid readily dissolves copper (and silver) but leaves the steel unaffected. There are some precautions to be taken.

- The cut out piece should be as small as possible. This is not always easy, but the BPHE should at least be cut as section A - A' in figure 06 B. This is because the copper brazing around the ports is extremely hard, not to say impossible to dissolve.

  If the BPHE is not cut, the result of the process will be that all the brazing is dissolved, except around the ports. As each plate is attached to the preceding plate at the ports of one side and to the succeeding plate at the ports of the other side, the result is a plate pack kept together at the ports. If cut along A-A', the plates separate into pairs, the plates of each pair brazed to each other at the left or the right ports, but each plate available for inspection.

  The difficulty of dissolving the port brazing makes it virtually impossible for internal leakage to occur through corrosion of the copper. The port brazing is the only place where brazing separates the fluids, but before this is corroded; the brazing at contact points and the edges has long gone.

- The brazing at the contact points will dissolve in a few hours. The edge brazing will dissolve in about two days. Thus, cut or grind away the edges if possible.
- Stir the nitric acid every now and then to get fresh acid to difficult-to-reach places.

Observe local regulations for the use and disposal of nitric acid. If there are none, use the recommendations in figure 06 C.

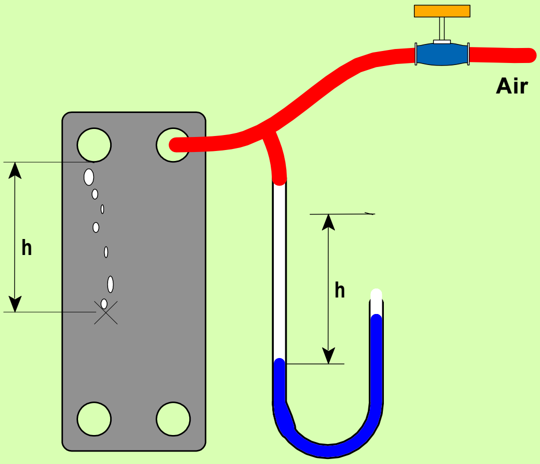
6.3. Examining the result.

See Figure 06 D for some typical damages. Note that:

- Corrosion damages show seldom any deformations of the plates.
- Freezing can rupture the plates, so that the edges of the rupture are displaced.
- Channels, which are bulging outward, are normally a sign of freezing.
- Thermal stresses usually shear of the plates at the contact points.

The most difficult cases of damages to examine are combination damages. The copper corrodes and the weakened contact points cannot stand the pressure.

A similar effect can happen for thermal stresses or fatigue from vibrations. The contact points are weakened or destroyed and cannot hold the pressure.

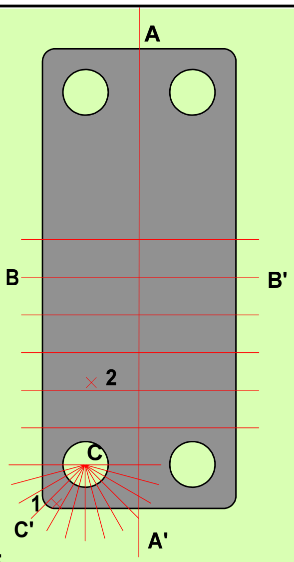


A. Determining the height of a leak.

One side is filled with water and the other connected to a supply of pressurized air. The air flow has to be controlled carefully. The air pressure has to be measured precisely, e.g. by a glass U-tube as shown.

When bubbles start to emerge, the level difference in the U-tube is equal to the depth of the leakage. To account for a possible hysteresis effect, the level at which the bubble formation starts and stops should be noted.

Proceed in the same way with the BPHE lying on the side and on the back.



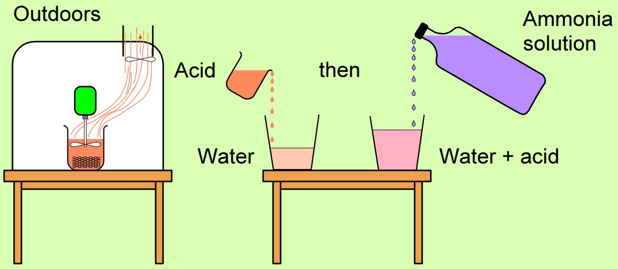
B. Cutting a BPHE.

Once the position of the leak has been determined, the BPHE has to be cut up for a closer examination.

The first cut should be well away from the suspected place of the leakage, usually A-A' or the lowest B-B'.

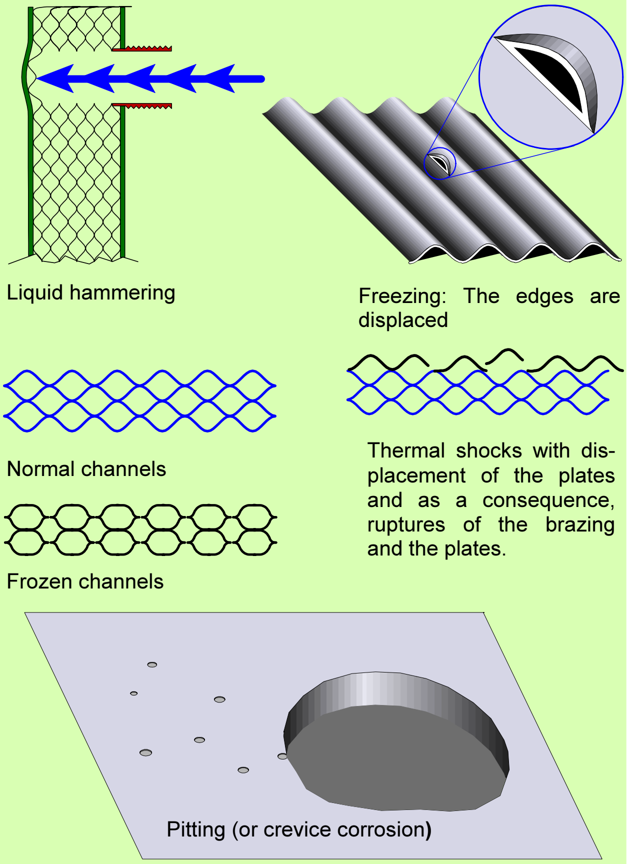
Finding a leakage by cutting is difficult. A too close cut could destroy the leak, a too wide (> ~10 mm) makes it impossible to see the leak.

Check the cuts for damages. Freezing gives large de- formations, other types of damages maybe none.



C. Dissolving a BPHE in nitric acid.

- Nitric acid dissolves copper under the development of NO_2. This is a brown irritating gas, heavier than air, and the vessel (glass, stainless steel or polyethylene) should be placed in a well ventilated space or outside.
- Do not dispose of nitric acid in the wastewater drain. Neutralize it first with a basic compound. In the absence of this, calcium carbonate stone, e.g. marble, is an excellent substitute. Ammonia solution is also suitable. See the next point, though.
- Before neutralizing the acid, dilute it with water. Pour the acid into the water, not the contrary. This makes the subsequent neutralization less violent. Neutralize with ammonia till the solution smells faintly ammoniac.
- Use protective gloves and glasses. Nitric acid leaves yellow stains on the skin and makes it leathery. After some days the affected skin will peel off.


Liquid hammering

Freezing: The edges are displaced

Normal channels

Thermal shocks with displacement of the plates and as a consequence, ruptures of the brazing and the plates.

Frozen channels

Pitting (or crevice corrosion)

D. Different type of damages.

Fig. 06. Leak seeking in a BPHE.

7. Freeze. Freeze protection of brazed plate heat exchangers.

1. Freeze control in a BPHE.

1.1. The freezing process in a PHE.

In order for ice to form, the wall temperature on the water side has to be lower than zero. An ice film then forms on the surface. Ice will not form if the temperature is exactly zero; some subcooling is necessary. If the bulk water temperature is well above zero, this ice film will be thin or even non-existent. If the water bulk temperature is close to zero, the ice film grows and can block the channels.

Under normal stable conditions, the evaporation temperature can be far below zero before the wall temperature falls below zero. Exactly where this happens - at the water exit or where facing the lowest refrigerant temperature - is difficult to determine. It depends on the temperature program, the pressure drops, etc.

Unfortunately, there is no such thing as a stable operating condition. The nature of turbulent flow means that the flow on both sides is fluctuating; there might be some maldistribution - natural or due to fouling - on either side. Ice could start to form in a channel. If this happens, the flow passage is obstructed and the water flow is reduced. A reduced flow means cooling to a lower temperature; and a lower wall temperature, more ice forms, still lower water flow - and suddenly the channel is blocked by ice.

Warning 1. Ice formation, with possible damage to the BPHE, can occur whenever the evaporation temperature falls below zero.

1.2. Freezing of water solutions.

In order to lower the freezing point, glycols or salts are added to the water. The decrease of the freezing point is roughly proportionate to the concentration. When such a solution freezes, the ice crystals formed are composed of pure water. These of course have a melting point of 0 °C.

What then could happen is that the temperature drops shortly and a little ice forms. Then the temperature increases, but instead of melting, which would happen with pure water, the ice remains. The temperature drops again, more ice forms etc.

Warning 2. Because of this hysteresis effect, there could be an ice build-up in the evaporator.

Fortunately, ice formed from solutions contains inclusions of glycol or salt and it will more likely be like a slurry or porous and not a hard block of ice, as from pure water.

1.3. Prevention of freezing.

The BPHE is about half the weight and contains about half the amount of refrigerant and water of a comparable coaxial evaporator. Compared with the tubular heat exchanger, the ratio is maybe one third. The inertia for a change of the process condition is less in a BPHE than in other types of heat exchanger. This puts special demands on the control equipment, which needs a faster response time than normal. While this sometimes could be an advantage, and the low refrigerant amount certainly is, this means that a BPHE puts up less resistance to freezing than other types of heat exchangers.

The precautions can be divided into the following groups:

- ♦ Design of BPHE for refrigeration duties.
- ♦ Installation.
- ♦ Thermal and hydraulic calculation of the BPHE.
- ♦ Control system and operational praxis.

Another classification is if freezing occurs because of subzero temperatures of the refrigerant or if the ambient temperature is below zero. In the above classification b) mainly concerns protection against low ambient temperature and the rest against low refrigerant temperature.

2. Anti-freeze designs.

Non-refrigerant BPHEs. Because of the corrugations, there will be a space between the channel plate and the front cover. On non-refrigerant BPHEs, obsolete refrigeration BPHEs and some nickel BPHEs this space is open to the atmosphere. If the corrugation angle points downwards, water can collect in the space, freeze and possibly destroy the BPHE. See figures 01C and **3. Design & Installation** , figure 02.

Refrigerant BPHEs. A flat seal plate with a folded down edge is inserted between the front cover and the first plate to seal off the open space, see D&I, and figure 01. Note that a seal plate is not needed at the end plate; the folded down edge of the plate effectively seals off the space.

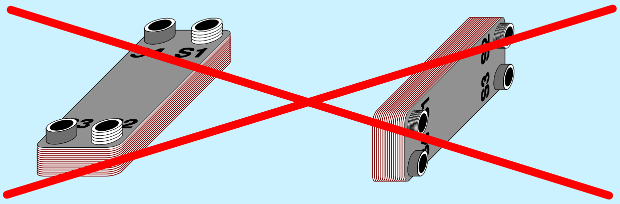
Gasketed PHEs or the ***gasket side of SWPHEs,*** are resistant to damages due to freezing. The gaskets expand and the worst damage is some blown out gaskets.

Tests have been made with a SWPHE where the evaporation temperature was reduced to -20 °C with the water running. When the unit was entirely frozen, with no water flow, the evaporation temperature was increased until the full water flow was restored. No damage was detected.

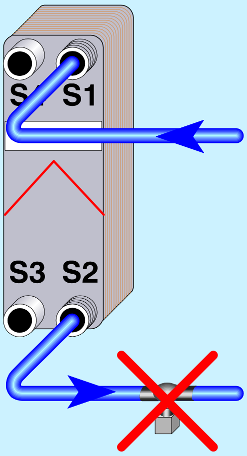
AWPHEs have basically the same freezing properties as BPHEs and the same advices apply.

3. Anti-freeze installations.

Figures 01 A - F contain some suggestions. A BPHE installed outdoors is subjected to the ambient winter temperatures, which might be below zero. If the BPHE is not used during this period or used only on & off, it should be drained during the off periods. The easiest way of doing this is to enter the water at the upper connection and let it leave at free flow from the lower connection, see figure 11 in **3. Design & Installation**. Providing, of course that the thermal duty permits such an arrangement.



A. Avoid positions for which water cannot be drained.



Note! Never install a valve with the drive mechanism at the bottom. Dirt and debris might collect there and impede the spindle movements.

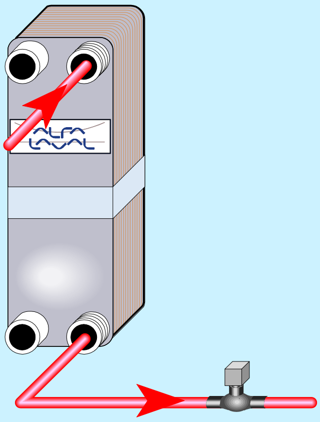
C. BPHE without seal plate.

For the following BPHE:

- ♦ CB14
- ♦ Old models
- ♦ Non refrigeration BPHEs
- ♦ Some nickel BPHEs

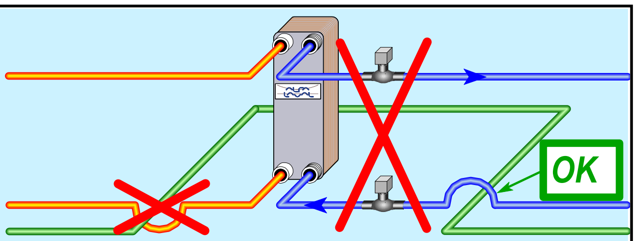
do either

- ◊ Corrugations of the first plate should point upward.
  or
- ◊ Seal the open space with silicon rubber.



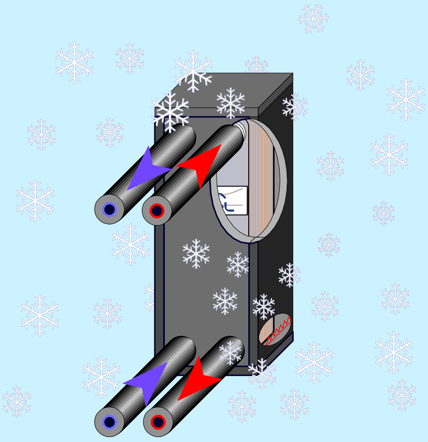
E. Ice plugs.

An ice plug and a valve can form a confined space. Note that an ice plug will be very solidly stuck in a BPHE.



B. Neither trap water in the BPHE between closed valves nor use downward loops.

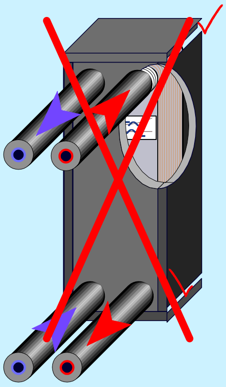
Except for the specially design oil traps in some refrigerant lines (see §1. O & R.), neither refrigerant nor liquid lines should have downwards pointing loops.



D. Low ambient temperatures.

HEs installed outdoors, e.g. in heat pump should be protected from low ambient temperatures by:

- ♦ Insulation of the unit.
- ♦ Installation of a small electric heater.
- ♦ Draining of the unit when not in operation.
- ♦ Keeping the water always running.



F. Faulty insulation.

Damaged insulation or absence of insulation could provoke freezing. Especially dangerous is freezing on two different locations as the two ice plugs might create an enclosed water body. This will certainly rupture the unit as the freezing proceeds.

Fig. 01. Design & installation of refrigeration BPHEs.

Moreover, there might be an air cushion at the entrance port. This air cushion causes maldistribution of the water and creates channels with low water flow or even channels with no running water at all. If the evaporation temperature is even slightly below zero, the water in these channels might freeze. This air pocket has to be vented. See figure 11 in **3. Design & Installation** for some ideas.

In evaporators operating close to or below freezing point, the water flow should be well balanced, both between the channels in a unit and between parallel units. See figure 11 in **3. Design & Installation**.

4. Thermal & hydraulic design.

4.1. Wall temperatures and shear forces.

Keep the water wall temperature as high as possible. The wall temperature moves upward for:

- A high minimum water temperature.
- A high evaporation temperature.
- A high water channel heat transfer coefficient.
- A low refrigerant channel heat transfer coefficient, i.e. a low pressure drop.

Warning 3! Too low a pressure drop could lead to distribution and oil flow problems as well as unstable flow.

Keep the shear forces on the water channel wall as high as possible. The shear forces are proportional to the pressure drop per meter flow length. The pressure drop increases for:

- A high water flow rate.
- A channel with as high a theta value as possible.
- A large MTD decreases the number of channels.
- Multi-pass.

Sometimes the effects above are in the same direction, sometimes opposite.

4.2. Over- & underdimensioning.

Do not overdimension the BPHE if the evaporation temperature is set to a fixed value and the water exit temperature is let free. Then the overdimensioned BPHE cools the water to below its nominal exit temperature, and possible danger of freezing.

Do not underdimension the BPHE if the water exit temperature is controlled by changing the evaporation pressure. To compensate for the inadequate area, the controller might lower the evaporation temperature, below zero. A larger BPHE could work with a lower temperature difference, thus a higher evaporation temperature and less danger of freezing.

4.3. Co- vs. counter current flow.

In counter current flow, the cooled medium exit - the lowest temperature - faces the cooling medium inlet - also the lowest temperature, thus this point has the highest freezing danger.

If cocurrent can be used - it is normally less advantageous - then the lowest temperature on cooled side faces the highest temperature on the cooling side, with less freezing risk as a result.

4.4. Multi-pass.

Multi-pass on the water side is an effective way of increasing the water side wall temperatures. It is mainly used for a very long temperature drop of the water, which has to be cooled close to the evaporation temperature. A typical temperature program might look like:

Water	28	->	1	°C
Refr.	-1	<-	-1	°C

Such a temperature programme is close to impossible in a single pass unit. The water pressure drop will be very low, too low, leading to unstable flow and oscillations. The low K-value and the low shear forces push the water wall temperature very close to the evaporation temperature.

As the main use of this arrangement is to cool water close to zero with refrigerant which is sometimes far below zero it will be dealt with here even though it can be used for any temperature level. The water flow can be arranged as a U or an inverted U.

U-arrangement, figure 03A. The temperature program in the passes would look something like:

Pass	1, counter current				2, cocurrent.			
Water	25	->	6	°C	6	->	1	°C
Refr.	-1	<-	-1	°C	-1	->	-1	°C

The effect on the evaporator depends on if this is a direct expansion or a flooded flow evaporator.

DX-evaporator. There is simply no room for any superheat in the second pass. The exit vapour is a mixture of much superheated vapour from the first pass and wet from the second, a not very suitable arrangement.

Flooded flow. The entering refrigerant meets the same temperature difference in the two cases, which is an advantage as the boiling then starts simultaneously. The exit quality can vary very much but this has less importance in flooded flow.

Inverted U-arrangement, figure 03B. The temperature program in the passes would look something like:

Pass	1, cocurrent				2, counter current.			
Water	25	->	5	°C	5	->	1	°C
Refr.	-1	->	4	°C	4	<-	-1	°C

DX-evaporator. There is enough temperature difference in both passes at the end of the evaporation to secure a sufficient superheat. The middle water temperature, 5 °C, defines the maximum superheat in both passes and there can be no large variations, e.g. at capacity changes and a DX evaporator is suitable. The entrance temperature difference is a little low at the second pass but as the refrigerant is already partly vaporized, it does not matter.

Flooded evaporator. The low inlet temperature difference at the second pass might be too low to start the boiling, which makes this design unsuitable for a flooded evaporator.

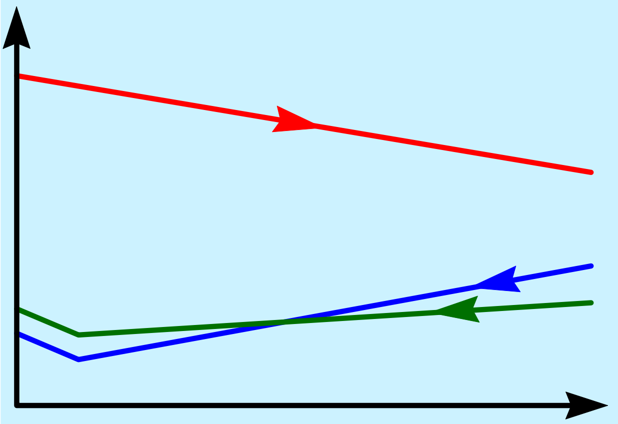


Fig. 02. Low refrigerant pressure drop.

Certain plates have asymmetrical channels, i.e. one side might have an option of two channels with different pressure drop. This could be used to vary the refrigerant pressure drop.

A reduction of the refrigerant pressure drop means a reduction of the temperature drop as well (from the blue to the green line). If the temperature difference is to remain constant, the evaporation temperature could increase, thus increasing the wall temperature and reducing the freezing risk.

As the refrigerant heat transfer coefficient decreases, the wall temperatures are pushed further upward.

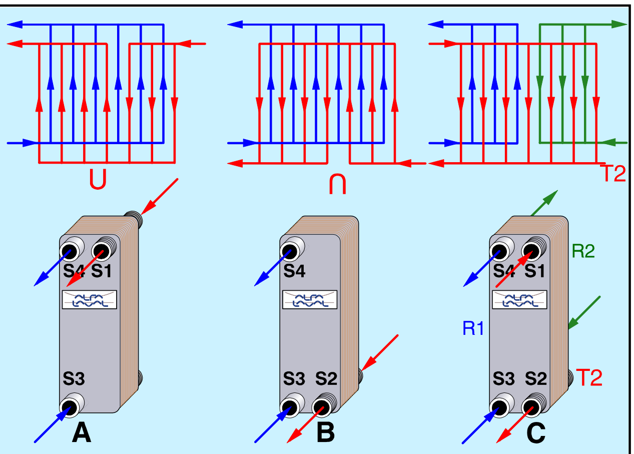


Fig. 03. Plate arrangement.

A. The water (red) is connected in two passes at the lower connections, substantially increasing the water velocity and reducing the freezing danger. The entering refrigerant (blue), meets two different water temperatures, but leaves with one.

B. Two passes as above, but upper connections. The entering water meets one water temperature, but leaves with two. An extra drain connection is placed at T2.

C. Two refrigerant circuits (blue and green). The furthermost circuit - the green - should always be operating and a low temperature guard should be installed at T2. See discussion in the text.

Case study. A chicken slaughter plant in Portugal needed cool fresh water to wash and cool the slaughtered chickens. The entrance temperature of the water was a high 28 °C in the summer, which had to be cooled to 1 °C. As refrigerant, ammonia of -7 °C was available from a LP receiver. This temperature could not be changed as many other evaporators were connected to the LR.

A study showed that a flooded, single-pass WPHE would do the duty, but as the water pressure drop was very low, the water wall temperature was only slightly above -6 °C. Though freezing would not damage the WPHE, it would act as a type of fouling with obstructed channels and therefore with a possible capacity decrease.

The solution was a flooded WPHE in two passes on the water side, connected as in 4A. This is effectively two heat exchangers, in series on the water side, parallel on the ammonia side. The design criteria were:

- The pressure drop on the ammonia side should be the same in both sections.
- The K-value margins should be the same as well.

The result was am M10BW with H-channels in the first section and L-channels in the second, a somewhat unusual combination. (The normal combinations are L, L & M, M & H and H.)

It performed as expected with no operating problems except an occasional cleaning on the water side.

Note! The SWPHE is probably the only type of heat exchanger, which can be used for such an application. The low ammonia temperature means that freezing occurs if the water flow stops due to unforeseen circumstances.

4.5. Dual refrigerant circuits.

The chapter Design and Installation describes the three possibilities of arranging the refrigerant in two separate circuits. One of these, figure 03C above, has the disadvantage that the exit water is a mixture of cooled and uncooled water when one circuit is shut off. If the temperature of the mixed water is controlled by e.g. the evaporation temperature, this could push down the cooled water temperature to below zero, the mixture still being above zero.

The advantage of this design is that it can be adapted to any type of PHE, brazed or gasketed. In order to use this the following measures should be taken:

- Install a LT thermostat at T2, which shuts off the compressor at a too low water temperature. This monitors the water temperature leaving the R2 circuit.
- Circuit R1 should always be running, i.e. the capacity control is made by (dis)connecting the R2 circuit.

The result was am M10BW with H-channels in the first section and L-channels in the second, a somewhat unusual combination. (The normal combinations are L, L & M, M & H and H.)

Note that it is very difficult to monitor the water temperature from R1 as the water is immediately mixed with the R2 water as soon as it leaves the channels. Control by disconnecting R1 would then be impossible.

5. The control system and the operational practice.

Figure 04 contains some points to consider when designing a freeze protection system. ***They are not all needed or should not necessarily be used simultaneously.***

A system which cools down water close to zero, obviously needs to be more sophisticated than one where the evaporation temperature is always well above zero.

5.1. Pump-down at evaporators.

If liquid refrigerant remains in the evaporator during a shut down of the plant, there is a danger of liquid refrigerant entering the compressor at the subsequent startup, with compressor damage as a consequence. The evaporator is therefore emptied - pumped down - of liquid refrigerant at shut-down.

The danger with pump-down is that the refrigerant temperature could decrease below the freezing point of the cooling liquid, therefore with freeze damage.

The sequence at pump-down.

1. The solenoid valve before the expansion valve closes. The signal to close the valve is normally a decreased load, i.e. a reduced water temperature. See figure 04 F.

2. The compressor continues to operate and evacuates the refrigerant in the closed-off evaporator, thereby decreasing the pressure and the temperature.

The relationship between the temperature and the liquid content in the evaporator is dependent on whether there is a heat transfer to the refrigerant during the pump down.

♦ ***No heat transfer.*** The pressure and temperature can theoretically reduce to absolute zero with liquid refrigerant still present. This means that the pump-down can be made to any given temperature and there might still be a risk that liquid refrigerant is present.

♦ ***Heat transfer***. The temperature might decrease, remain constant or increase depending on the rate of the heat transfer to the suction power of the compressor.

Thus, if the heat transfer is large enough, the pump-down could be made with an increase of the temperature without a liquid phase being present.

In praxis there is always a heat transfer to the refrigerant, and the liquid phase has disappeared at some point not too far below the starting temperature, but there is no temperature at which it is possible to be certain if liquid refrigerant is present or not.

3. When the pressure has reached the low set point, the LP pressostat stops the compressor, followed a little later by the water pump, see figure 04 G

The closing criterion.

The compressor is normally shut down when a given pressure is reached. As seen from the above, this is no absolute proof that a liquid phase is not present.

Far better would be to use the superheat at the lowest point or the point where liquid refrigerant is likely to remain the longest. As soon as there is a superheat there can be no liquid phase present and this should be the correct criterion for shutting down the compressor.

Recommendations at pump-down.

♦ The water should always be running.

♦ The compressors should run with as low a capacity as possible, i.e. the pump-down time is maximised.

♦ The solenoid & DX valves should be as close to the evaporator as possible, to minimize the liquid content.

♦ If possible, stop the compressors when a superheat is reached. This can be done automatically or manually. Record the superheat as function of the pump-down pressure and use the pressure at which there is a noticeable superheat as closing pressure.

Alternatives to pump-down.

♦ Pump-down is not recommended or necessary at BPHEs. The refrigerant volume is small enough that the liquid will evaporate by the heat remaining in the BPHE for a temperature decrease of only 1 to 2 K.

♦ Install a bypass line between the compressor exit and the evaporator inlet. This is the same arrangement used to prevent too low temperatures at startup; see figures 04 L& M.

♦ The solenoid valve in the bypass opens when the refrigerant feed solenoid valve closes. The hot vapour from the compressor evaporates the refrigerant in the evaporator and maintains the temperature.

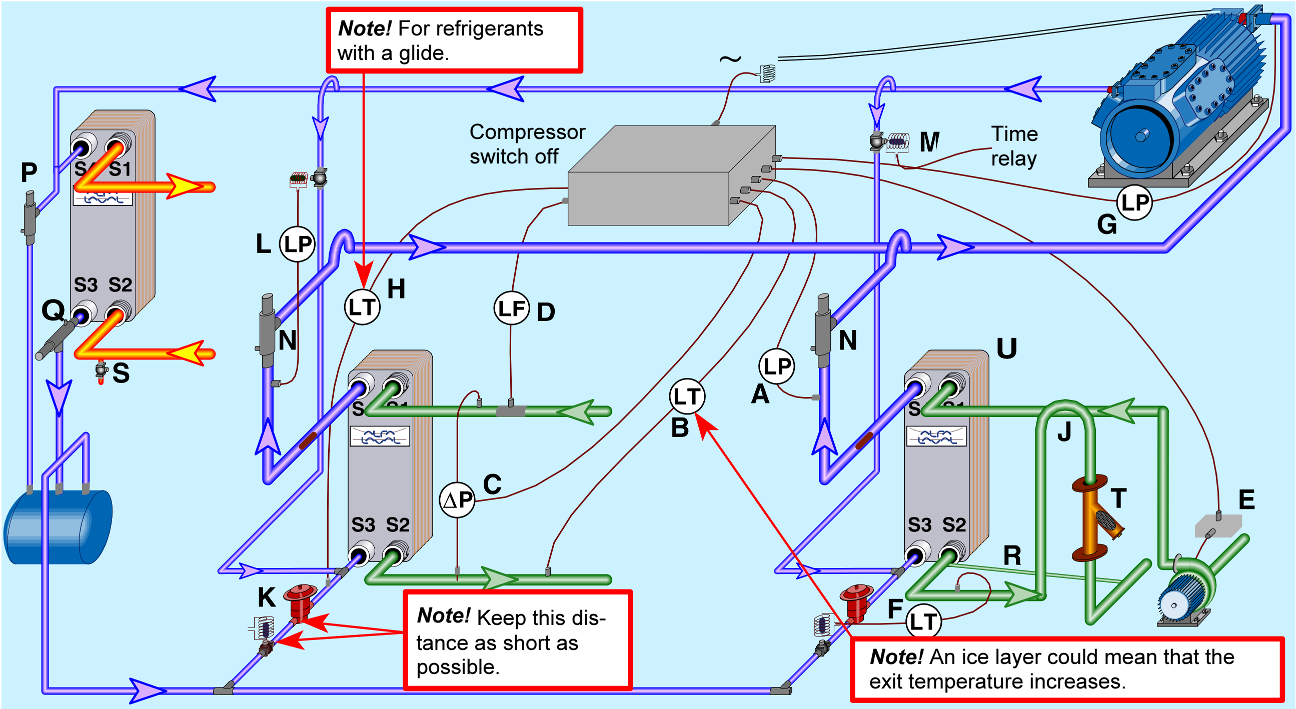
♦ The compressor runs for a determined number of seconds after the solenoid valve has closed, i.e. a sort of "semi pump-down".

5.2. Reversible systems.

When the refrigerant changes direction and the condenser becomes an evaporator and vice versa, there is a period when the pressure could drop far below zero. This case is described extensively in the section Applications.

5.3. Refrigerants with glide.

See figure 04H and **7. Oils & Refrigerants**. Note that the real ***temperature*** not the ***pressure*** should be used for freeze control.



Note! For refrigerants with a glide.

Note! Keep this distance as short as possible.

Note! An ice layer could mean that the exit temperature increases.

Fig. 04. Freeze control in a refrigeration plant.

The compressor is shut-off by various combinations of:

A. A low-pressure switch at the refrigerant exit. The lowest system pressure occurs here. The temperature cannot be monitored directly, as it is superheated, thus the corresponding pressure is used.

B. A low temperature switch at the water exit.

C. High pressure drop on the water side.

D. Low flow rate on the water side.

E. Water has to circulate before the compressor is started and for some time after it has been shut off.

F. Pump down, see text.

G. The compressor has LP pressostat, which shuts off the compressor at a e suction pressure.

H. In a refrigerant as R 407c, which has a fairly large glide of about 6 °C, the inlet temperature could be several degrees below the saturation temperature at the exit. Thus, monitoring the temperature at the exit or even worse, the pressure, could lead to large errors.

Freeze protection for a large glide refrigerant should be made by monitoring the temperature at the inlet.

Furthermore, the following measures should be taken:

J. The exit water passes through a loop, which reaches above the entrance. This assures that the evaporator is always filled with water, even at low flow rates.

K. Keep the solenoid - expansion valve line as short as possible to minimize refrigerant buildup at a shutdown.

L & M. At low pressure and a time controlled hot gas by-pass. These can be used to flush out unevaporated refrigerant at shut-off and maintain the evaporation temperature during start-up, shut off or a pump down.

N. Each evaporator needs it own pressure controller.

P & Q. LR (R) and condenser (S) pressure controller.

A cooling plant is normally laid out for summer conditions. When the autumn arrives, the air and cooling water temperatures decrease. The condensation temperature decreases as well. The compressor capacity increases as the capacity increases with reduced compression ratio. The decreased condensing pressure means lower driving force for the expansion valve. Insufficient refrigerant then passes the valve at the same time as the compressor sucks with increasing force.

The compressor is sucking a larger vapour volume than is produced by the evaporator. The pressure decreases; more refrigerant passes the valve and the volume increases, until a match is reached, which could occur at an inadequate evaporation temperature. To prevent this the liquid receiver pressure should be controlled.

R. & S. A small by-pass (T) to automatically drain a unit with inlet from above or a normal drain (U). Drain a unit at freezing ambient temperatures when not operating.

T. A filter on the water sides should be installed for all units operating with potentially fouling liquids.

U. A filter in the inlet port should be used with caution. Debris stuck in the filter or in the channels could cause uneven distribution of water from channel to channel and possibly freezing. A separate filter as in L is probably the best solution.

8. Oils & Refrigerants.

1. Oil in refrigeration systems.

1.1. Oil in the compressor.

The primary duty of oil in a refrigeration system is to lubricate the moving parts of the compressors. An important secondary duty is to assist in the sealing between the moving parts, thereby preventing refrigerant from leaking from the high to the low pressure parts of the compressor. Ideally, the oil should be confined to the compressor. However, some oil always leaves with the hot vapour from the compressor discharge line and enters the various components of the refrigeration circuit.

If too much oil leaves the compressor, there will be a double negative effect:

- The oil will infringe on other components and diminish their capacity or in other ways impede their function.

- If too much oil leaves the compressor there might not be enough left to ensure sufficient lubrication and the compressor might seize.

Thus, some type of oil management is necessary to process the oil/refrigerant mixture leaving the compressor and return the oil to it.

1.2. The oil-refrigerant system.

Solubility. It ranges from insoluble to completely soluble. An important intermediate state is the formation of a two-phase mixture, “oil dissolved in refrigerant” and “refrigerant dissolved in oil”. See figure 01.

Distribution of the refrigerants in the oil. The solubility of a refrigerant in an oil varies with the properties of both the oil and the refrigerant. A particular oil could dissolve very much of one refrigerant and very little of another. The oil in contact with both would then contain the two refrigerants in different concentrations. This is a largely unstudied effect of special importance for zeotrops, i.e. refrigerants with a glide, figure 02.

Vapour migration. If refrigerant vapour is in contact with pure soluble oil, refrigerant dissolves in the oil. If there is an unlimited supply of oil, the oil will be gradually diluted, see figure 03.

Chemical reactivity. The long hydrocarbon chains in mineral oils are chemically very inert. By contrast, the various ester-, ether, alcohol-, oxide-, and other groups attached to the carbon chain in synthetic oils make them more reactive when water and oxygen are present. Compounding the problem is that some metals and metal oxides - produced by wear and tear & welding - are very good catalysators. The possible decomposition products range from hydrogen, hydrocarbons, acids, tar & asphalt, i.e. from problems with inert gases in condensers to fouling in evaporators.

The refrigerant can also react with ill chosen oil. The best example is probably ammonia, which reacts readily with polyolester oils (POE). Another example is R 134A, which is unstable in contact with mineral oils, or residual chlorine compounds, which can be found in the oil after a switch from R12.

Vapour pressure. Soluble oil decreases the vapour pressure of the oil. The effect is small except at the very end at the evaporation of soluble oil in direct expansion evaporation. In order to evaporate the refrigerant completely, the mixture has to be heated to far above the inlet temperature of the warm fluid. As this is not possible, the oil drops thus contain some unevaporated refrigerants. The effect on the size of the evaporator is small, though.

Density differences. In case of insoluble oil, the density difference to the refrigerant is important. An insoluble oil, heavier than the refrigerant collect at the bottom and is drained from there.

An insoluble oil, lighter than the refrigerant collect at the surface and can be drained from there in principle. As the level in most cases is not constant, it might be difficult to find a good location for the drain.

Lubrication power. Chlorinated hydrocarbons together with oil have superior lubrication actions, especially with machine parts of aluminium. Unfortunately, in this respect only, they are soon to be phased out,

Retrofitting with a new refrigerant could lead to disaster for the compressor if the oil is not carefully selected in strict accordance with the compressor manufacturer’s recommendations.

Hygroscopic properties. Synthetic oils are sometimes hygroscopic and care has to be taken when charging the system as the oil might pickup moisture from the atmosphere.

An advantage of the hygroscopic property could be less ice formation in the TEV.

The large disadvantage is that hygroscopic oils better dissolves inorganic compounds, such as from wear and tear. Fouling on the refrigerant side in the evaporator will then probably increase.

Alkyd-based oils are especially hygroscopic, while the newer ester- or polyolester-based oils are better.

1.3. Oil in evaporators.

There are two problems with oil in an evaporator:

- The refrigerant flow is usually upward. Oil could then be retained in the channels.

- The temperature is low and the viscosity high and the oil tends to stick to the surface.

Below and in figure 05 are some points to consider:

- Regardless of the density, the oil droplets move along the channels under constant changes of direction and impinge on the surface. The channels act as a sort of de-emulsifier and an oil film forms.

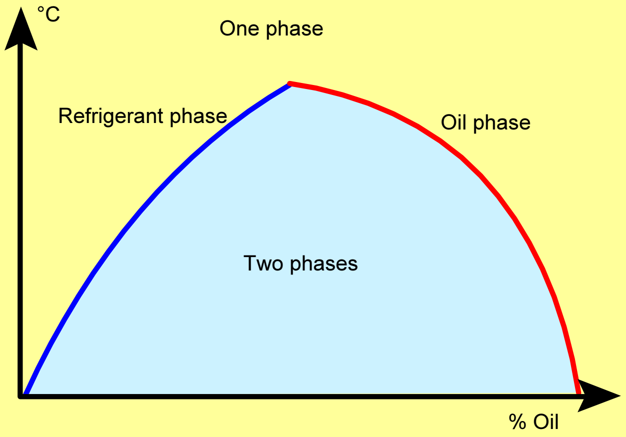


Fig. 01. Distribution of the oil between two phases.

A semisoluble oil distributes between two phase, an oil rich, the oil phase and a refrigerant rich, the refrigerant phase. The composition of each phase at a given temperature can be found from the solution curve.

Above a certain temperature, where the curves meet, there is only one phase regardless of the oil content.

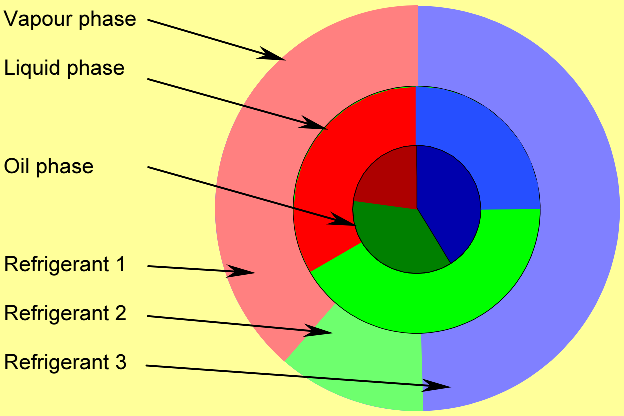


Fig. 02. Phase distribution of refrigerants.

The three components of this zeotropic refrigerant have different solubilities in the oil as well as different volatilities. The three components are initially of the same concentration, but refrigerant 3 has a larger concentration in both the oil phase and the vapour phase.

If the amount of oil is substantial, the composition and thus the vapour pressure of this zeotrop might change.

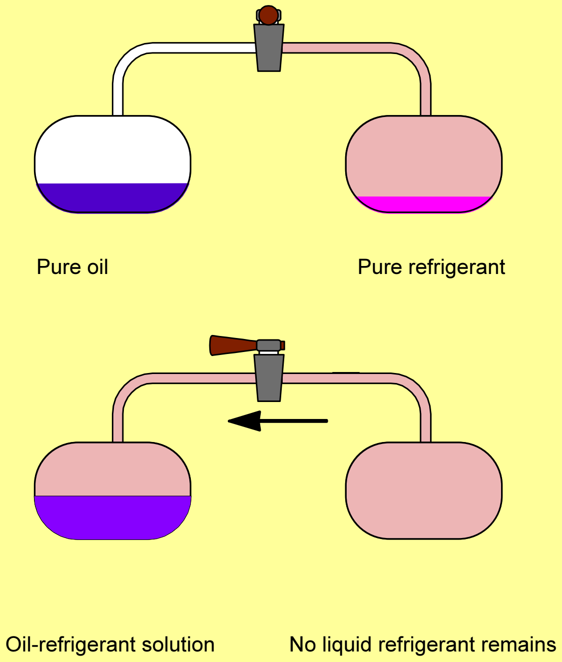
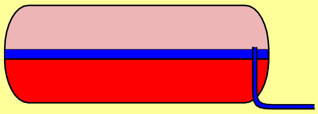
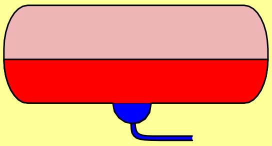


Fig. 03. Refrigerant migration.

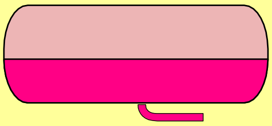
When the valve opens, refrigerant dissolves in the oil. The migration of the refrigerant from the right to the left vessel continues until there is no more liquid in the refrigerant vessel or until the oil becomes saturated with the refrigerant. Migration like this could happen if a shut-off compressor is connected to e.g. an evaporator with liquid refrigerant. The oil then loses the lubrication ability.



A. The oil is lighter than the refrigerant (R 502). Draining from the top, close to the interliquid surface.



B. The oil is heavier than the refrigerant (ammonia). Draining from the bottom, preferably from a low point.



C. Completely dissolved oil. Draining is not possible (or necessary). Some liquid is sometimes taken out for evaporation of the refrigerant, thus concentration of the oil.

Fig. 04. Density differences.

Traditionally, thermosiphon systems, especially ammonia have favoured oil-refrigerant system, type B, as the insoluble, heavy oil is easy to drain.

Thermosihons are not well adapted to systems A or C. A because of the difficulty of finding a stable drain point, C because of the necessity of an oil evaporator.

- The behaviour of the film reflects the oil viscosity. A low viscosity oil flows at times as a mist in the channels, at times as a liquid layer on the wall. High viscosity oil might stick to the surface as a reasonably quiescent oil layer. An important but largely unstudied factor is the effect the surface tension has on oil fouling.

- Insoluble oil fouls the entire heating surface, in a DX or a flooded system.

- Soluble oil fouls only the superheating part of a DX evaporator.

- The only system that is not fouled by oil is a flooded system with soluble oil. If severe oil problems are expected, this might be an option, despite the need for an oil evaporator, see **1. Applications**, § 16.

- At low channel flow rates in a DX evaporator, the shear forces might not be large enough to carry the oil droplets upward. The oil - soluble or not - then collects in the evaporator. It might stick onto the surface or remain as an oil slug in the evaporator.

- On-off control maintains the vapour velocity.

- Consider hot gas by-pass control as a way of controlling the capacity in DX evaporators. **4. Evaporators & Separators**, figure 24 shows one example. There are numerous variants and the valve manufacturers have specialized valves for this.

- Hot gas by-pass control has one advantage. When the refrigerant flow decreases because of decreased capacity, the missing refrigerant flow is replaced by compressor discharge gas. The refrigerant flow rate remains reasonably constant.

- A second effect of hot gas by-pass control is the increased temperature. This helps flush away the oil.

- A simpler variant of the above is an injection of hot discharge gas by a solenoid valve. This can be used as an alternative to a pump-down as well. See **7. Freeze**, figure 04, L & M. Note this is not a capacity control, only a way to flush out oil and refrigerant.

- A third possibility is to heat up the oil with the warm side fluid, thus lowering the viscosity.

- The best way is not to let the oil enter the evaporator. Primarily a good oil separator is necessary if oil problems are expected.

- Secondly, a thermosiphon evaporator needs an oil drain system, see **1. Applications**, figure 10 and **4. Evaporators & Separators**, figure 07.

- A DX-evaporator normally does not need an oil drain.

- The exception is ammonia and insoluble oil otherwise, oil slugs could cause problems in the TEV. The drain should be in the HP receiver, i.e. before the oil enters the TEV. It could be arranged as an LP oil drain.

- The parameter most important to carry the oil through the evaporator is the wall shear stress or the more easily calculated pressure drop per meter flow length. The simple vapour velocity is misleading, as the hydraulic diameter and channel shape as well as the physical properties, specially the viscosity; of the oil and refrigerant have very significant effects.

- A plate heat exchanger can carry oil with it for a velocity, which is about ten times lower than the velocity in the vertical connecting pipe. There is no absolute lower limit of the pressure drop per meter, but try to obtain a value larger than 5 kPa per meter channel length (port centre to port centre).

  Observe that a very large pressure drop means a high inlet temperature, and this decreases the effective temperature difference.

- In case of expected oil problems, try to make the temperature difference as large as possible, as this decreases the number of plates (channels) and thus increases the pressure drop on both sides.

- A high pressure drop on the refrigerant side means a high pressure drop on the liquid side as well. If this is not acceptable, a bypass on the liquid side could help.

- The BPHE with distributors improves the oil flow at DX evaporation, **4. Evaporators & Separators,** figure 18.

- WPHE needs a distributor pipe with a number of holes with increasing diameter that is inserted into the port, figure 06.

- Heavy oil is drained from the inlet port of WPHE at DX-evaporation, see figure 06.

1.4. Oil in condensers.

In figure 05 is a list of the effect of various combinations of density and solubility of the oil on condensers and evaporators. Condensers are in general uncritical equipment; the only thing that can normally happen is fouling of an oil film in ammonia systems.

1.5. Oil in refrigerant lines.

Figure 07 show some pipe drawing practices. Moreover:

- The vapour velocity should be large enough to carry oil and possibly liquid refrigerant with it.

- The pressure drop in the lines should be as low as possible. This contradicts the above, though.

1.6. Oil separators.

The best method is to intercept the oil, just when it has left the compressor, in an oil separator. Usually larger droplets are separated in a system of baffles, while the finer droplets are separated in a demister.

Observe that a demister is not a sieve. This prevents particles larger than the holes in the sieve from passing. Both baffle systems and demisters operate according to the same principle. The vapour passes baffles or wires under frequent direction changes.

Droplets cannot change direction that rapidly and are caught by the wires or the baffles. Once on a surface, they coalesce to larger drops, which flow down to the bottom of the demister or separator by gravity forces.

The oil ultimately collects in a tank at the bottom of the separator. Sometimes the oil is heated, by hot vapour or electrical, in order to evaporate refrigerant.

	Insoluble oil	Soluble oil
Oil lighter than the refrigerant	***General:*** There is practically only one system of this type, the soon to be phased out R 502/mineral oil. (R 22 at low temperatures has one phase “oil in refrigerant” and one phase “refrigerant in oil”.) An oil film could develop and fouling due to the oil might be a problem especially at very low temperatures. It could improve condensing heat transfer if dropwise condensation occurs. See **5. Condensers & Liquid Receivers**, § 1.2. The film could block the nucleation sites and impair the boiling heat transfer. See **4. Evaporators & Separators**, § 1.2. ***DX-evaporator:*** As the oil flows above the refrigerant, it enters the channels and leaves with the refrigerant. Large amount of high viscosity oil (low temperature) could impede the proper function of the TEV. Then an oil separator and occasional flushing with hot gas might help (**4. Evaporators & Separators**, Fig. 23). ***Flooded flow*** is not recommended as oil flows on the surface and is difficult to recover. In case of R22, an effective oil evaporator (**1. Applications**, § 16) impedes formation of the light oil rich phase and the exit to the oil evaporator could then be from anywhere. ***Condenser:*** No particular problems if the condensate drains from the condenser. The oil then moves with the condensate. However, if there is a condensate level as in **5. Condensers & Liquid Receivers**, figures 04, 05 & 11A, oil collects on the top of the condensate and impedes the heat transfer.	***General:*** The oil impairs the physical properties of the refrigerant-oil solution, notably the viscosity and thus lowers the heat transfer coefficient. A little oil (< 1 %) could even improve the boiling heat transfer. An oil film cannot develop and fouling due to the oil is therefore no problem in the liquid phase. ***DX-evaporator:*** At the end of the evaporation, when all refrigerant has evaporated, oil might get stuck on the heating surface: This is normally only a problem at low temperatures (< -20 °C) and for high viscosity oil, especially for R22, as the oil is only partly soluble at low tempertures. Oil might then collect at the exit and flow back into the evaporator. The newer BPHEs with distributors improve the oil flow, see **4. Evaporators & Separators**, Fig. 18., for WPHEs see also Fig. 06 here. Use a good oil separator and maybe hot gas by-pass control, see **4. Evaporators & Separators**, Fig. 23 or **7. Freeze** Fig. 04, L & M. ***Flooded flow:*** As the liquid refrigerant washes all the surfaces, oil fouling is no problem, but if no precaution is taken, oil concentrates in the system and drains the oil from the compressor. An oil evaporator is thus necessary, see **1. Applications**, § 16. ***Condenser:*** No particular problems. The oil moves with the condensate.
Oil heavier than the refrigerant	***General***: There is only one commercial system, but an important one, ammonia/mineral based oils or synthetic polyalfa-olefin (PAO) oils. See above on the effect of an oil film. ***DX-evaporator***: The oil might collect at the bottom of the inlet port and drain from there Fig. 06. The oil could also be drained from the HP receiver and only the remaining small amount from the inlet port. A good oil separator is necessary and maybe flushing by hot gas as described in **4. Evaporators & Separators**, Fig. 23, Hot gas by-pass .... ***Flooded flow:*** With a good oil recovery system, where oil drains from the lowest point of the evaporator-separator loop, fouling due to oil will not be a problem. **1. Applications,** Fig. 10 and **4. Evaporators & Separators**, Fig. 03 & 07 and **§3.4.7 & §3.6**. Note! Some oils (polyalfaolefins) could be difficult to drain from the channels. Further experience is needed. ***Condenser:*** No special problems. The oil moves with the condensate. Condensate level should not pose any problems as the heavier oil drains from the bottom.	***General:*** See above. The two commercial systems are ammonia / synthetic oils type polyalkylene glycol (PAG) and hydrocarbons (propane/propene/butane) / mineral oils. Note! The in ammonia soluble polyol ester oils (POE) react vigorously with ammonia. ***DX-evaporator:*** The difference to HCF systems is that ammonia is normally used in larger evaporators, such as welded and semi-welded PHE’s. The larger port holes in these can cause a maldistribution of the ammonia over the channels. A specially design distributor pipe is then installed in the porthole, see figure 06. Hydrocarbon/mineral oils are used in the same BPHE as the HFCs and have the same problems, see above. ***Flooded flow:*** See above. In the case of ammonia it might be better to use an insoluble oil. ***Condenser:*** See above.

Fig. 05. The effect of the oil on evaporators and condensers.

When the oil level is high enough, a float opens a valve and the discharge pressure forces the oil to return to the compressor inlet.

Some oil separators can return more than 99 % of the oil to the compressor, but normally it is in the order of 80 %.

An oil separator should be used whenever there are problems with oil retention in the evaporator. This will most likely be more severe the lower the temperature is. Apart from removing oil, an oil separator also acts as an efficient muffler, thus reducing noise.

Take care if an oil separator is installed in an old system. The separated oil, which was previously distributed in the entire system, could flood the compressor.

2. Oil cooling.

2.1. Temperature shocks at on-off cooling.

If the compression ratio is too high, the exit temperature, including the oil temperature, might be too high as well. Some compressor types, notably screw compressors, heat the oil excessively. Certain refrigerants, e.g. ammonia, superheat more than others during the compression. A too high oil temperature could lead to a breakdown of the oil. Thus, oil cooling is necessary.

This oil cooling can be done in a BPHE. However, if the heat exchanger is filled with hot oil and the cooling fluid - water or refrigerant - suddenly enters, e.g. when an on -off controller is actuated, this could lead to thermal shocks and ruptures, usually between the end plate and the first heat transfer plate. In case of a S&THE, tubes could loosen at the tube plate.

As BPHEs are effective oil coolers compared to other types of HEs, thus economically advantageous, some precautions have to be taken to avoid thermal shocks. Consider a BPHE working with hot oil on one side and cooling media on the other. The system has been brought gradually to the steady operating state, i.e. no thermal shocks have occurred.

The viscosity of the oil entering the oil cooler is usually less than specified on the manufacturers data sheet as it contains dissolved oil. This can lead to an oversized oil-cooler if the design is based on the nominal oil properties.

An oversized oil cooler rapidly cools the oil and in case of an on-off control system it will probably be operating in many short cycles instead of a few longer ones. This lead to more temperature cycles and thermal fatigue.

Beware of oversizing an oil cooler.

What gives the largest shock? A sudden change of the oil temperature or a sudden change of the cooling media (Water or refrigerant) temperature?

Study the two theoretical extreme cases: On-off control of the oil side (figure 08 B) or of the cooling medium side (figure 08 E).

When the oil temperature has reached its upper limit, either oil or cooling fluid suddenly enters the BPHE. What is the effect on the BPHE when the oil or cooling fluid has just replaced the previous content of the channels? There are two effects to be considered, the change of energy and the effect on the heat transfer coefficients:

- ***Energy changes in the BPHE.*** Oil has about half the specific heat of, and a lower density than, water, i.e. the relative change of energy is lower if oil enters than if water enters. In case of an evaporating refrigerant, the relative energy change might be still larger as there is a phase change and thus a transfer of latent heat.
- ***Heat transfer***. Oil normally has several times higher viscosity than the cooling medium, either refrigerant or water. Furthermore, the heat conductivity is much higher for water than for oil. Oil has about the same heat conductivity as refrigerants, but for the latter there is the additional heat transfer of the evaporation.

Thus, there is not only a larger energy change for a sudden change of the cooling medium temperature, but it is also more easily transferred to the plates, i.e. with danger of rupture of the plates. The conclusion is thus that sudden temperature shocks such as from on-off control should be avoided, especially on the cooling fluid side.

2.2. Temperature shocks at parallel connected compressors.

Another source of rapid temperature changes is startups or shutdowns of a compressor in a plant with compressors connected in parallel. This could lead to fluctuations of the pressure and thus temperature changes, either directly or indirectly, if an on-off control circuit is triggered.

A DX system with its small refrigerant hold-up is especially sensitive to changes. A flooded system has a larger inertia and is thus less sensitive to changes. See fig 09 and **4. Evaporators and Separators,** § 6.7.

2.3. Control of oil coolers.

Some examples of on-off controls are shown in figure 08. The on-off valves in figure 08 B&E can be changed to control valves, with a continuous regulation of the by-pass stream. Instead of on-off control, the liquid - oil or cooling medium - is gradually varied and continuously flowing, thus keeping the oil temperature constant.

Sometimes, evaporating refrigerant is used as cooling fluid. See figure o8 A and 09. To even out the pressure fluctuations in case of multiple compressors, each oil cooler should have its own pressure controller. The oil exit temperature should control the set points of these either manually or automatically. See figure 09.

2.4. Cooling media at oil cooling.

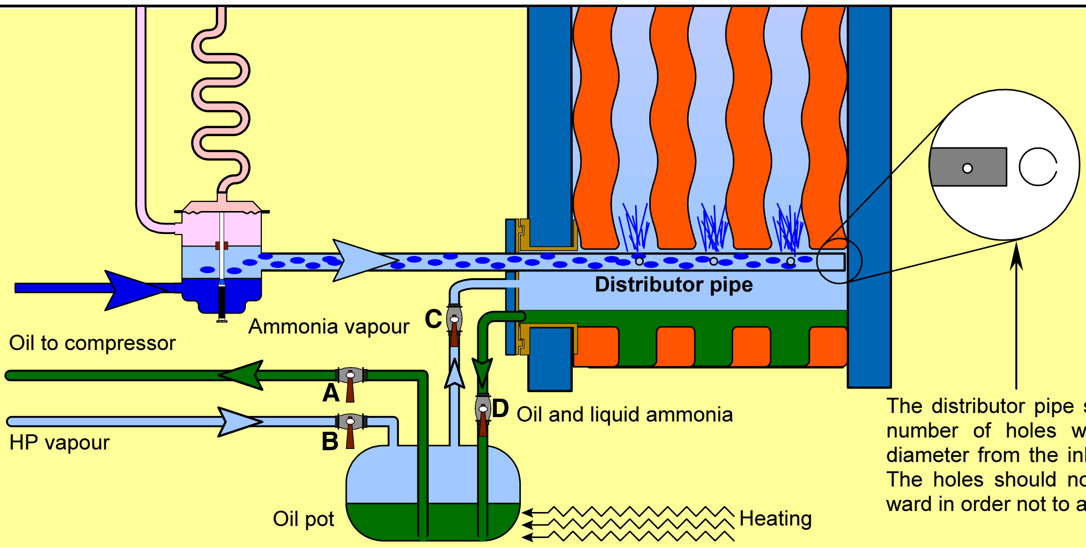
Water has the following pros and cons:

- Water is normally cheap and available.
- Water might not be at hand at the compressor site and/or the quality might be questionable.
- If the flow stops the compressor could be damaged.
- The colder it is, the larger the risk for thermal shocks.

Refrigerant has the following pros and cons:

- It is available at the compressor site and the flow only stops when the compressor stops.

Running and installation are more complicated.



The distributor pipe should have a number of holes with increasing diameter from the inlet to the end. The holes should not point downward in order not to agitate the oil.

Fig. 06. Draining of an insoluble oil from an ammonia DX-evaporator.

1.The oil drain shall be from the lowest part of the inlet.

2. Oil and ammonia flows to the oil pot through line D. Ammonia in the pot evaporates, either by the ambient temperature, by an electric heater, or a coil with HP refrigerant.

3a. Vaporized ammonia flows back to the inlet through line C

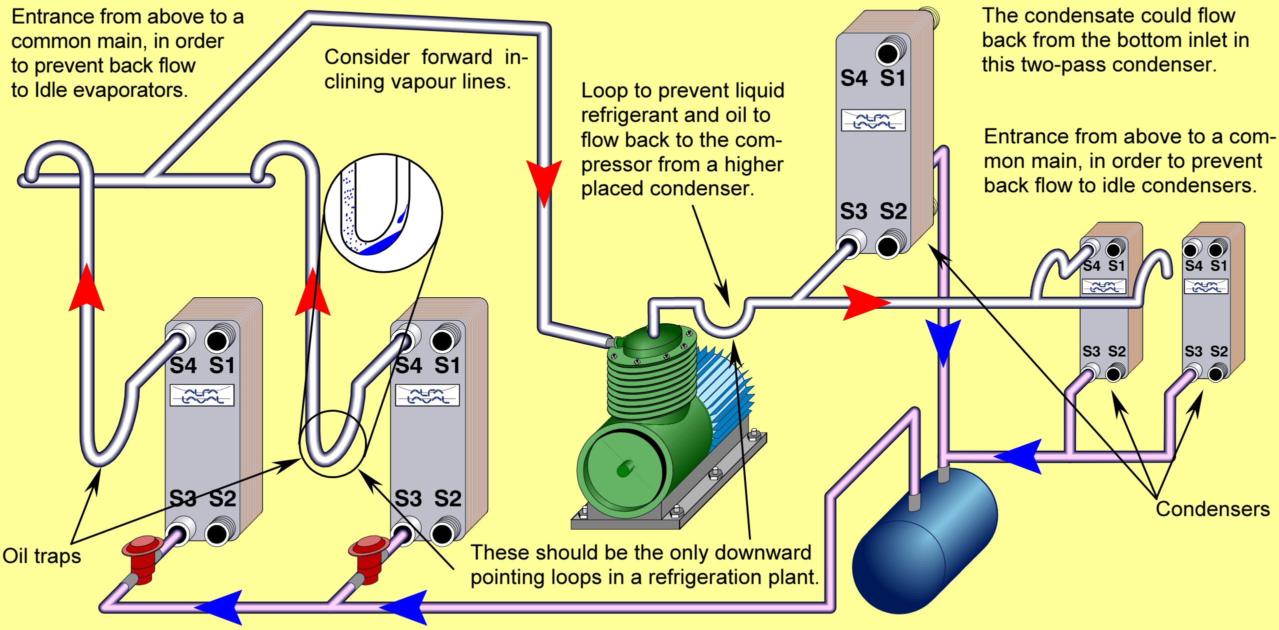
3b. An alternative would be to leave to the exit, but that needs a careful adjustment of valve C.

4. When the oil pot is full of oil, no further ammonia enters and the temperature increases (no frost formation on the oil pot, if subzero evaporation).

5. The normally openvalves C & D then close and the normally closed valves A & B open. The HP vapour then pushes the oil back to the compressor.

The valve D should close some time before C to let residual ammonia evaporate.

6. If return to the exit (3b), step 5 can be omitted.



Entrance from above to a common main, in order to prevent back flow to Idle evaporators.

Consider forward inclining vapour lines.

Loop to prevent liquid refrigerant and oil to flow back to the compressor from a higher placed condenser.

The condensate could flow back from the bottom inlet in this two-pass condenser.

Entrance from above to a common main, in order to prevent back flow to idle condensers.

Condensers

Oil traps

These should be the only downward pointing loops in a refrigeration plant.

Fig. 07. The vapour lines Evaporator - Compressor - Condenser.

The vapour/oil mixture from the evaporator enters the oil trap. Oil droplets too heavy to be carried by the vapour in the vertical line collect in the trap. The vapour velocity increases and breaks up the oil into smaller droplets, which can be carried by the vapour.

To prevent back flow of oil and liquid refrigerant to an idle evaporator, mains should be entered from above.

If there is an appreciable vertical distance between the compressor and the condenser, the refrigerant might condense in the line at a shutdown of the plant. Liquid refrigerant and oil then enter the compressor. To prevent this, the line from the compressor should make a downward loop before it rises again.

As before, lines should enter a common main from above.

Refrigerant temperature level. From where shall the liquid refrigerant be drawn; the HP or the LP receiver?

A flooded evaporator as oil cooler could be added to the LP separator. However, the refrigerant could be very cold here, leading to large thermal shocks. The large temperature difference most likely makes the HE thermally overdimensioned and the oil will be cooled too much.

Refrigerant from the HP receiver is usually of a temperature of 30 to 45 °C, suitable for the oil, which normally is 70 to 90 °C. If a direct expansion system is used (figure 09) and the evaporation temperature is close to the receiver temperature, there is no flashing of the refrigerant in the expansion valve; it simply ensures that no more refrigerant enters in the oil cooler than what can evaporate.

A flooded evaporator could be connected to the HP receiver. The vapour produced is normally recondensed in the condenser. See the discussion in **5. Condensers and Liquid Receivers**, figure 06, of the layout of the HP receivers.

As a HP receiver is not a separator, the refrigerant should be evaporated as much as possible, e.g. controlled by the superheat through an expansion valve (but no flashing).

2.5. Conclusion.

- ♦ Rapid changes especially of the cooling medium temperature have to be avoided. Especially dangerous is on-off control, see figure 8 A & B.
- ♦ Try always to have continuous flows, however small, on both sides or attenuate the temperature changes.
- ♦ The range between the high and low temperature settings at on-off control should be as small as possible. This eliminates large temperature variations.
- ♦ If solenoid valves with slow opening and closing times are available, temperature shocks could be reduced; figure 08 C. Motor valves are excellent but expensive.
- ♦ Do not oversize the oil cooler. This is easily done if the pure oil viscosity is used instead of the mixture viscosity, which is considerably lower.

Avoid valves which control a variable by changing the opening time, e.g. open 19 seconds, closed 1, changing this to open 1, closed 19 seconds as the load varies. This causes both thermal and mechanical shocks.

3. Refrigerants.

3.1. General.

Over recent years, the number of generally used refrigerants has increased from less than ten to multiples of ten.

A number of different refrigerants together with the appropriate oils are thus under evaluation. Some of these combinations will prove to be unsuitable for various reasons and the large majority of these will never be commercially viable. In some years there will be a new set of environmentally friendly refrigerants, probably only between five and ten.

Fortunately, plate heat exchangers are uncritical. In contrast to most other equipment in a cooling or refrigeration plant, PHEs benefit from a wealth of design and operating experience in refineries, chemical process industries, and in mechanical, food and marine application as well as for all types of duties such as condensation, evaporation, single-phase, viscous media, vapour with inert gases, etc.

3.2. The differences between the “new” and the “old” refrigerants.

All freon type refrigerants, are halogenated hydrocarbons. The halogenes are mainly chlorine and fluorine; for some special applications bromine. What then, are the differences between the “new” and the “old” refrigerants?

- ♦ The “old” refrigerants are mainly fully halogenated, i.e. the molecules contain only carbon and halogenes. They are denominated CFCs (chlorine, fluorine, carbon) and are in general:
  - ∗ Good solvents, especially to mineral oils.
  - ∗ Incombustible.
  - ∗ Chemically stable, which is a bonus for the operating properties, a drawback for the environment.
  - ∗ The chlorine gives good lubricating properties, especially to aluminium.
  - ∗ Not hygroscopic.
  - ∗ Inert to generally used construction materials.
  - ∗ The chlorine reacts with atmospheric ozone, leading to the ozone depletion.
  - ∗ Contribute to the greenhouse effect.
- ♦ The addition of a hydrogen atom converts the CFCs to HCFCs and changes the properties:
  - ∗ Breaks downs easier, which decreases the ozone depletion potential considerably.
  - ∗ Many hydrogen atoms in the molecule increase the flammability.
- ♦ Removal of the chlorine atom converts the HCFCs to HFCs and changes the properties:
  - ∗ No more ozone depletion potential.
  - ∗ Decreased greenhouse effect.
  - ∗ Decreased solubility of mineral oils.
  - ∗ Decreased lubricating properties.
  - ∗ In many cases more hygroscopic.
  - ∗ Special oils are necessary which brings about new problems, see §2 and 8. Fouling & Corrosion.
- ♦ Removal of the fluorine atom converts the HFCs to normal hydrocarbons, HCs, actually old refrigerants:
  - ∗ No more greenhouse effect, nor ozone depletion.
  - ∗ Flammable.
  - ∗ Good solubility to mineral oils.
  - ∗ Not hygroscopic.
- ♦ The system R134a/R12 could serve as an example what happens when a CFC is replaced by a HFC.
  - ∗ R134A dissolves nitrogen 10 times better than R12. If R134A has been in contact with air, it might contain nitrogen, which could make trouble in the condenser.

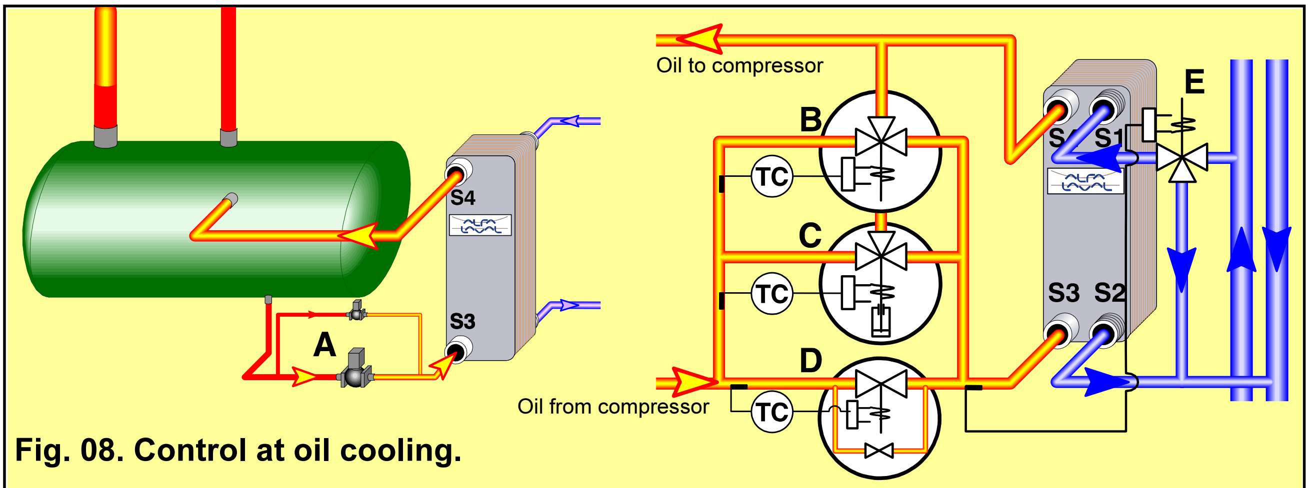


Fig. 08. Control at oil cooling.

On-off control.

A. A small refrigerant stream, corresponding to the minimum cooling requirement, always flows through the oil cooler. When more cooling is needed, the main refrigerant valve opens.

B. The oil normally flows through the bypass. When the oil temperature reaches its high set-point, the three way solenoid valve switches the oil from the by-pass to the BPHE. If the set point is not too high above the water temperature and the oil is fairly viscous, the tempera- ture shock might be moderate.

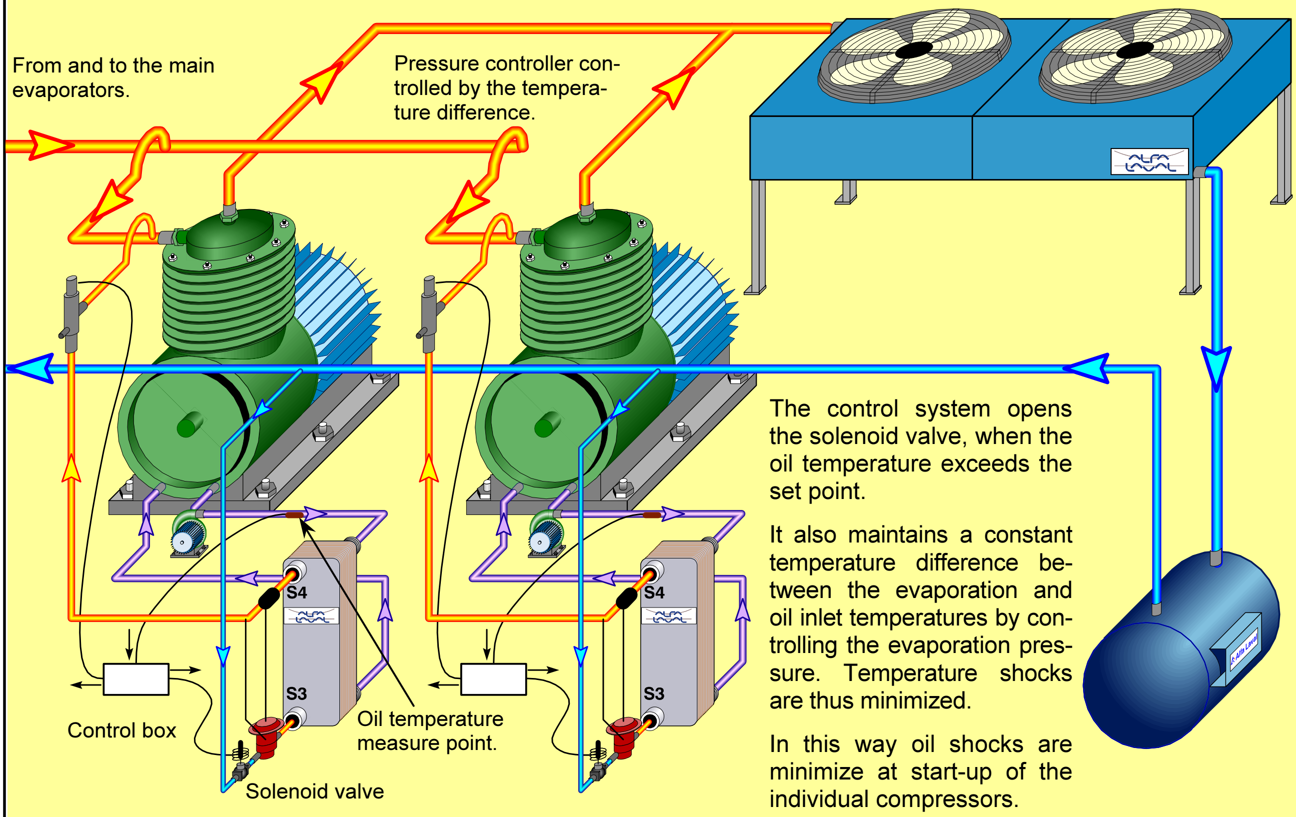
C. A solenoid valve with a damper, therefore slow moving, would give a gradual change of oil temperature.

D. Similar to A, but here the oil is in by-pass. The small valve opens at compressor start-up. When more cooling is needed, the solenoid valve opens and sends an oil flow large enough that the combined flows could handle the largest cooling requirement.

E. The system is equal to B, but on the water side. As explained in the text, severe thermal shocks could arise, leading to unequal dilatation of adjacent plates, resulting in rupture of the brazing. ***Avoid this system!***

Continuous.

B & E. These valves could be replaced with three-way valves, which adjust the flow rates continuously. The large thermal shocks are then avoided, especially at B.



The control system opens the solenoid valve, when the oil temperature exceeds the set point.

It also maintains a constant temperature difference between the evaporation and oil inlet temperatures by controlling the evaporation pressure. Temperature shocks are thus minimized.

In this way oil shocks are minimize at start-up of the individual compressors.

Fig. 09. DX-oil coolers with parallel compressors.

- R134A itself is more hygroscopic than R12. R134A in combination with water can lead to micro galvanic corrosion with copper, see **6. Fouling and Corrosion.** Dry R134A is safe.

- As R134A is a smaller molecule than R12, desiccators (molecular sieves) for R12 might not work.

- ***If R 12 remains after a switch to R 134A, R 134A and R 12 form a high-pressure azeotrop, which has a high potential for damaging the plant.***

3.3. PHEs and the new refrigerants.

In general, it can be said that if a PHE works for an old refrigerant it will work for a new as well, albeit with a change of performance, depending on the physical properties of the refrigerant. Nevertheless, there are some points to consider when evaluating a new refrigerant and/or a new oil.

3.3.1. Fouling.

As described in § 1 and in the chapter **6. Fouling & Corrosion** there might be a danger of fouling on the refrigerant side in the evaporator, most likely of tar like products and wear and tear products from the compressor.

This type of fouling will most likely be a fairly slow process, probably over many years, which also means that there is little experience on maintenance and cleaning methods. Possible fouling will be still more detrimental to compressors and control equipment than to the heat exchangers, which are why, unsuitable combinations will be phased out.

3.3.2. Inerts.

If the performance of a condenser gradually diminishes, one reason could be blocking by inert gases. Thermal decomposition of synthetic oils could result in hydrogen and carbon dioxide. Some refrigerant and oils could also dissolve ambient air. Venting every now and then helps.

3.3.3. Vapour pressure curve.

Temperature measurements in refrigeration plants are often indirect; the pressure is measured and converted to a temperature with the help of a vapour pressure curve as in a manometer or in a TEV. This method of obtaining the evaporation & condensation temperatures is acceptable for controlling the compressor and safety devices, where after all the pressure is the important parameter. In general, it is not acceptable for control of heat exchangers, which for the correct operation are dependent on the exact temperatures or rather the temperature differences.

There are some cases where a pressure controller or a TEV could be in error.

A) ***Drop-in replacement of an old refrigerant with a new.*** The vapour pressure curve, however well chosen the replacement is, differs between the old refrigerant and the replacement.

B) ***Faulty equipment.*** The bulb filling could not be the correct one in a TEV, the manometer could be mechanically blocked or the like. This could happen with any refrigerant, new or old.

C) ***Uncertain physical properties.*** There are sometimes deviations in the stated physical properties between different manufacturers (and from reality), especially for refrigerants with glide (see below). The deviations are usually small but in some cases, 2 °C deviations in the saturation temperature have been observed.

D) ***Composition changes.*** The most common example is probably water in ammonia. Water decreases the vapour pressure of ammonia. Examples that are more recent are composition changes in refrigerants with glides (see below) and composition changes because of an incorrectly executed drop-in replacement, resulting in a mixture of the old and new refrigerant. The deviation between the ideal and the real vapour pressure curves affects both the condensation and evaporation pressure and the superheat at an evaporator.

E) ***Temperature glide.*** There is no constant boiling or condensation temperature, but the two-phase change occurs over a temperature range, the glide. This is treated extensively below.

3.4. Temperature changes.

The change in temperatures resulting from the effects described in § 3.3.3 is the parameter of most interest for the heat exchangers. They can be grouped into:

♦ The vapour pressure of the actual refrigerant differs form the one the control system is designed for, cases A - C above. A study of a retrofit (A) is made in § 3.5.

♦ The evaporation and the condensation do not occur for constant temperatures, cases D & E above. Both refrigerants with a glide, case E, and the unintentional formation of a glide, which happens when water is dissolved in ammonia, case D, will be studied.

3.5. Retrofit with a new refrigerant.

A) ***The evaporation pressure*** controller in an R 12 plant is set to 2 °C. What happens at a retrofit with R 134a? See figure 10, for the vapour pressure curves.

The R 12 controller regulates the pressure to 3.3 bar, the R 12 saturation pressure at 2 °C. This corresponds to 3.38 °C for R134a. To get 2 °C for 134a, i.e. 3.145 bar, the pressure controller has to be set to 0.5 °C.

B) ***The condensation temperature*** is 40 °C. How shall the condensing pressure controller be adjusted.

The corresponding pressure for R12 is 9.63 bar. This corresponds to a 38 °C for R134a. The R12 controller has to be set to 42.2 °C, which is 10.16 bar. This corresponds to 40 °C for R134a.

C) ***Superheat.*** A DX evaporator for R12 with an evaporation temperature of 2 °C and a superheat of 5 K. is retrofitted with R134a. The pressure controller (the one for R 12) is adjusted to 0.5 °C, corresponding to 3.15 bar, i.e. 2 °C saturation temperature for R 134a.

The TEV gets a temperature signal from the bulb of 7 °C and a pressure signal from the equalization line of 0.5 °C, i.e. a superheat of 6.5 K. This is larger than the required of 5 K and the valve opens.

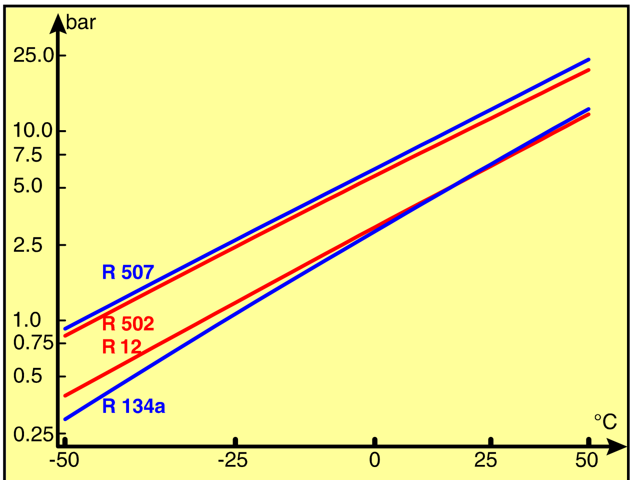


Fig. 10. Vapour pressures of the pairs R 12/R 134a and R 502/R 507.

D) ***An evaporator*** where R 502 is replaced with R 507 has a pressure of 6.10 bar (R 502 at 2 °C). At this pressure, R 507 has a temperature of -0.52 °C. To get 2 °C (6.60 bar) for R 507, the R 502 has to be set to 5.2 °C, which corresponds to 6.60 bar.

Comments:

1) If the controller in the R 502/R 507 cases is not adjusted, the evaporation temperature decreases, the MTD and thus the capacity of the evaporator increase, and i.e. there is no incentive to change the setting. The evaporation temperature in this case is now below zero, i.e. ***freezing danger.***

2) The settings change with temperatures and refrigerant system. R 502/R 507 is a well-behaved pair; the change in the setting is practically constant over the entire temperature range.

3) R 12/R 134a need no change around 18 °C, below that the temperature setting is increasingly lower and above, in the condensing range, increasingly higher.

4) The old pressure controller might be used but the control range has to be sufficient large.

5) Adjustment of a TEV is more complicated than shown. In the example, the valve gets a superheat signal of 6.5 K, when expected was 5 K. It then tries to close the valve. However, the pressure difference has increased from 9.63 - 3.3 = 6.33 bar to 10.16 - 3.15 = 7.01 bar. That would force more refrigerant through the TEV and decrease the superheat.

   The compression ratio for R12 is about 2.92. Depending on the type of compressor, the compression ratio might have to remain. The condensing pressure for R 134a is then 3.145 * 2.92 = 9.18 bar => 37.8 °C.

   This lowers the capacity of both the TEV and the condenser.

   The capacity of a TEV is not only dependent on the pressure difference and absolute pressure, but also on the thermodynamic and transport properties of the refrigerant.

   A TEV with R 134a seems to give about 20 % more capacity than the same valve with R12, while R507 gives basically the same capacity as R 502.

   The valve manufacturer's recommendation should be followed when adjusting valves, especially TEVs, at a retrofit of a plant.

3.6. Refrigerants with glide.

R 502, R 12 and R 22 are the three dominant refrigerants, which have to be phased out. There are a number of suitable substitutes for R 502 and R 12. It has been more difficult to find a replacement for R22 and at the same time being environmentally acceptable.

The search has led to several, more or less acceptable refrigerant mixtures. Refrigerant mixtures have always been used but then in the form of azeotrops, e.g. R 502.

3.6.1. Properties of refrigerant mixtures.

Assume that we mix two miscible liquids with different boiling points and heat them until the solution starts to boil. Observe that the liquids have to be miscible (soluble) over the entire concentration range. What happens and what will the difference to the two individual components be? We assume atmospheric pressure.

Fig. 11 demonstrates the two cases, A, when the liquids are kept separate and B, the actual case. The evaporation temperature, equal to the condensation temperature for a pure fluid has been replaced with a boiling interval. The boiling starts at the bubble point and ends at the dew point. In figure 11, the temperature is plotted against fraction vaporized. This is a practical way of describing a liquid mixture, as it is directly applicable on e.g. an evaporator. The refrigerant enters partly vaporized and leaves superheated. The boiling curve then gives the temperature profile for the two-phase part of the duty. It is also easy to get experimentally even for very complicated mixtures. Hydrocarbon mixtures, such as petrol, are often defined by boiling or distillation curves.

To explain the different types of two-component mixtures, it is better to plot the temperature against the concentration of one component in both the liquid and the vapour phase, figure 12. The result is an equilibrium curve. The boiling curve can be calculated from this, but not the contrary. Figure 12 shows also the difference between a zeotrop and the two types of azeotrops.

Azeotrops also exist for three or more components but it is then difficult graphically illustrate the relationships.

Azeotrops act for all practical purposes like pure fluids. The composition might change a little with the pressure, giving a theoretical possibility of separation, but it seems to be the case only for azeotrops of close boiling components, where the glide in any case will be negligible.

A mixture of two components of an azeotrop but with a nonazeotropic composition behaves like two fluids, the azeotrop and a pure fluid. A well-known example of this is alcohol and water, which upon a distillation can be separated into 95.4% alcohol and almost pure water.

Zeotrops are the rule rather than the exception in the two-phase world; most fluids do not form azeotrops. Below is a summary of some properties.

- Zeotrops can be characterized by the boiling curve as in figure 11, but for heat exchanger calculations, additional information is necessary. For a given pressure, the temperature, enthalpy and the physical properties of the vapour & liquid are necessary in a number of points. In refrigerant with a small glide or no glide, one point is necessary, in a refrigerant with a large glide; rarely more than three (start, middle, end) are needed.
- When the liquid mixture starts to boil, the vapour is enriched in the Low Boiling Component (LBC), but also contains the HBC. Similarly, when the vapour starts to condense, the liquid becomes enriched in the HBC but also contains the LBC. The LBC doesn't evaporate alone followed by the HBC; both occur simultaneously, albeit with different rates.

- Contrary to what is generally believed, the vaporization does not follow an exact defined path, from the bubble to the dew point. It does, under assumption that the liquid and vapour is intensely mixed and moves at the same velocity. In other words, a certain quantity of liquid has to be present and mixed during the entire process. The same is valid for condensation.

  The above condition might not be true, especially not for some condenser types. If the liquid is removed as soon as it condenses, it does not participate in the equilibrium at the condensation any more. When the very last vapour condenses, it is in equilibrium with the condensate just formed and not with the entire condensate amount. This just formed condensate contains relatively more of the LBCs than the total amount of condensate and the effect is that the mixture has to be cooled down further before everything is condensed.
- When a boiling zeotrop is in contact with the heating surface in an evaporator, mainly the LBC evaporates at the liquid close to the wall. The temperature in this layer then increases and as a result, the temperature difference is less than the theoretically calculated. The practical effect is that the K-value seems to be lower.
- The same happens in condensers but the effect is less.

These three last effects above are less pronounced in a PHE with its intense turbulence and good mixing.

3.6.2. The evaporation and condensation temperatures for refrigerants with glides.

If both the condensation and evaporation occur over a temperature interval, then the question arises of the definition of evaporation and condensation temperatures.

- ***The evaporation temperature*** is the mean value between the inlet temperature and the dew point at the exit pressure. As the refrigerant usually enters partly vaporized, the glide in the evaporator is less than the difference between the dew and the bubble points.
- ***The condensation*** temperature is the mean value between the dew and the bubble points.
- **The evaporation superheat** is the difference between the exit temperature and the exit pressure dew point.
- ***The condensate subcooling*** is the difference between the exit temperature and the exit pressure bubble point.

There are some problems with these definitions (as are with alternate definitions). See also figure 13.

- The exit temperature is not the sum of the evaporation temperature and the superheat. In figure 13, it is 9.6 °C for R407c, not the sum of 2 °C and 5 K as for R 22. It is easy to make the mistake of using a too low water inlet temperature, especially for TEV with large superheats.
- An advantage with the glide is the possibility to cool the water a little more. If the approach of a certain HE is 3 K an R22 evaporator operating with 2 °C evaporation temperature cools the water to about 5 °C while an R407c could cool the water to 3 °C. Note the above, though.
- The inlet temperature to an evaporator might vary as the temperature to the TEV varies. Despite a constant pressure, the mean temperature will then vary as well. This will make comparisons to other systems difficult.
- Despite a positive evaporation temperature, the inlet temperature might be negative, i.e. ***freezing danger***. It is easy to forget, that at a retrofit of R22 to R407c the inlet temperature might be below zero for a 2 °C.

3.6.3. Utilization of the glide.

There have been many speculations if the glide could be used to operate a heat exchanger with less MTD. As figure 13 shows, there is no effect on a normal A/C duty.

If an evaporator could be made to operate without superheat, there is a possibility to work with a closer approach, but it has to be judged from case to case.

3.6.4. Utilization of zeotrops for thermosiphons.

Refrigerant with glides, at least the ones with large glides should only be used in direct expansion evaporators. As the refrigerant only partly vaporizes in a flooded flow evaporator, there will be composition changes, thus difficult to evaluate temperature changes in the various parts of the system.

3.6.5. Leakages.

A leakage of a zeotrop from a vessel with similar amounts of both vapour and liquid, thus with different compositions, means that the original composition changes. The temperature programmes in the evaporators and condensers then change with possible malfunction. The majority of the tests, which have been made, seems to agree that the system continues to perform as designed if the refrigerant is “topped-up” with the original zeotrop.

3.7. Troubleshooting for refrigerants with glide.

Up to now, most systems are designed for pure fluids or azeotrops. Pressure, converted to temperature can then be used as a control variable for a heat exchanger. In the near future, many systems will be converted to the new refrigerants. A lot of equipment will then operate with other refrigerants than they were designed for, with refrigerants with changed compositions due to leaks or internal mass imbalances as in thermosiphons, unadjusted instruments, inadvertent mixing of refrigerants etc.

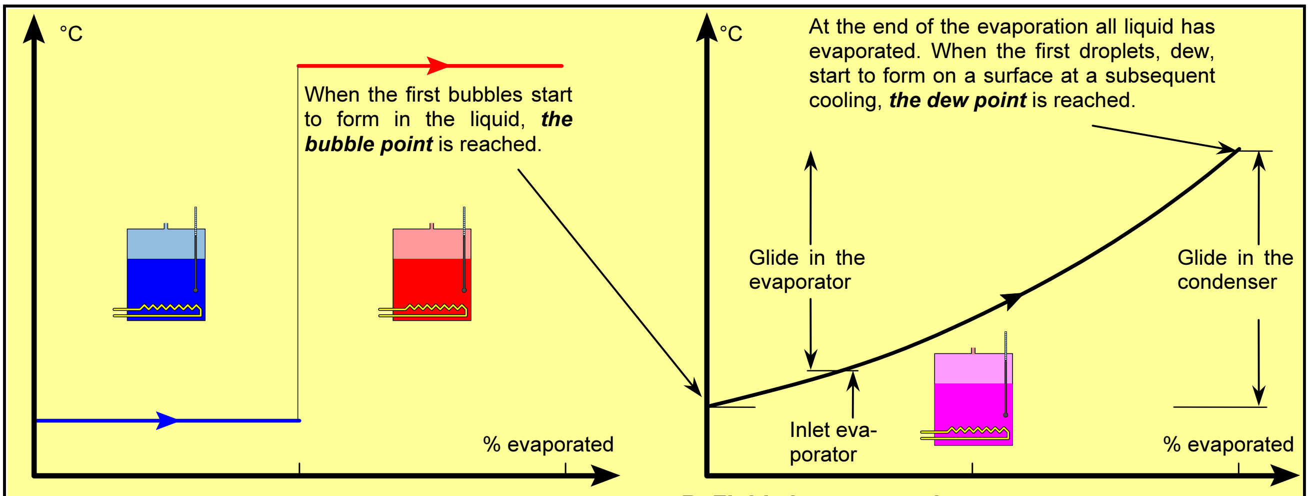
Leakage. A leakage of liquid from a two-phase part of the plant causes the temperatures to drop throughout the plant. A leakage of vapour raises the temperature level. A leakage from a single phase part, e.g. from between the evaporator and compressor or just for the TEV will not change composition and thus not the temperatures.

Capacity. By troubleshooting a heat exchanger, especially if there is a capacity claim, ***double check all temperatures***, which originate from various types of pressures readings, ***with thermometers.*** Heat exchangers need ***temperatures***, not pressures as control variables.

Freezing. Check the inlet temperature with a thermometer. Do not trust a converted pressure reading.

Erratic behaviour of a TEV. Remember that the superheat is no longer the difference between the exit temperature and the evaporation temperature. For the same nominal temperature, the superheat will be much smaller for R 407c than for R 22. A TEV, which have worked perfectly for R 22, might not give the required capacity or might be sensitive to hunting after a switch over to R 407c. Follow the valve manufacturers recommendations.***A condenser*** gives usually less troubles than an evaporator. Notice that subcooling is not simply the difference between the exit and the condensing temperatures.

A condensate could thus be mistakenly supposed to be sufficient subcooled to allow for a certain pressure drop without starting to boil.



A. Fluids in separate vessels.

B. Fluids in one vessel.

Fig. 11. Evaporation of two miscible liquids.

The two liquids are each contained in its vessel (A). The temperature increases simultaneously in both vessels. When the lower boiling fluid, the blue, reaches the boiling point, it starts to evaporate at a constant temperature. When everything has evaporated, the temperature increases until the high boiling fluid, the red, starts to evaporate. It then evaporates at constant temperature.

The boiling curve - the temperature vs. fraction evaporated -consists here of two separate horizontal lines. Note! This is also the boiling curve for two imiscible liquids.

When the two fluids are mixed in one vessel (B), the boiling starts at a little higher temperature than for the low boiling fluid but not as much as for the high boiling fluid. The first vapour emerging is relatively richer in the low boiling fluid and continues to be that during the evaporation. As the evaporation proceeds the concentration of the high boiling fluid in the liquid increases. The boiling temperature then has to increase and the temperature increases until all liquid has evaporated, usually at a temperature a little lower than for the high boiling fluid.

The ***evaporation*** starts at the **bubble point** and ends at the **dew point**. The ***glide*** is the difference between the dew point and the entrance to the evaporator, i.e. in a DX evaporator already partly vaporized mixture.

The ***condensation*** starts at the **dew point** and ends at the **bubble point**. The difference is the ***glide***.

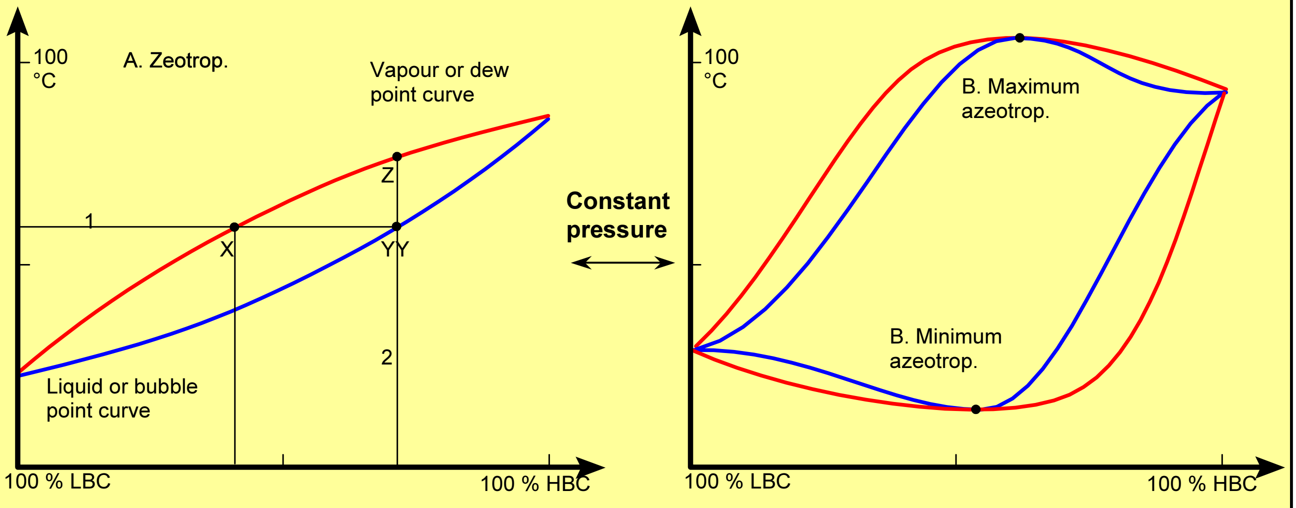


Fig. 12. The equilibrium curve for a zeotrop and the two types of azeotrops.

The concentration of the high boiling component in a ***zeotrop*** is always lower in the vapour than in the liquid.

At 60 °C (line 1), the vapour concentration is given by (X), 40 %, and the liquid concentration by (Y), 70 %. A liquid of 70 % has thus a bubble point of 60 °C. The dew point of this mixture (line 2) is at Z), about 75 °C.

The components of a zeotrop can be separated by a series of evaporations and condensations, a disitillation or a fractionation.

The bubble and dew points curves of an ***azeotrop*** touch each other at one point. In this point the vapour and liquid compositions are equal. The boiling point is either higher than for the individual components or lower.

For an azeotrop we can talk of a boiling point as the bubble and dew points are the same. The components in an azeotrop can not be separate by an evaporation and a subsequent condensation as the vapour concentration is the same as the liquid concentration. An azeotrop thus behaves like a single component liquid.

Concentration, ppm	Effect
5	Some people pick up the smell.
20	Most “ “ “ “ “
50	Very distinct smell.
100	Unpleasant, inexperienced persons get anxious.
300	Experienced people leave inexperienced panic.
400-700	Strong irritations in eyes, nose & lungs. Impossible to stay.
1700	Coughs, cramps. 0.5 hr exposure can lead to severe injuries.
2000 - 5000	May be lethal in less than 0.5 hr.
7000	Lethal within minutes.

Table 1. Physiological effects of ammonia

Hydrogen group	Water group	Air group	Refrigerants and others
H_2=2	CH_4=16	C_2H_2=26, HCN = 27	C_3H_6=42
He=4	NH_3=17	B_2H_6,N_2,CO,C_2H_4=28	CO_2,C_3H_8=44
	H_2O=18	Air= 29	R32=52
	HF= 20	NO, C_2H_6=30	R125=120
	Ne = 20	N_2H_4, O_2=32	Cl_2=71

The gas density is proportional to the molecular weight. There are only three gases lighter than ammonia and seven lighter than air. Six more are lighter but so close that they hardly separate when the gas enters an air body. The lightest refrigerant, propene C_3H_6, is considerably heavier than air. The HFC's are between R32 and R125.

Table 2. Molecular weights of some gases.

Pressure-Temperature readings. Remember that published data differ. A difference of up to 2 K has been observed between data from different manufacturers. Therefore, double check always temperatures based on pressure readings.

Temperature readings. Evaporators and condensers are often evaluated with the help of the compressor instruments, normally manometers, usually situated some distance from the heat exchangers, i.e. with pressure drops. The important parameters for compressors are the pressures but for heat exchangers the temperatures. A pressure reading converted to a temperature from a manometer should basically be mistrusted, the more so if the manometer is not situated immediately in the vicinity of the heat exchanger.

3.8. Ammonia.

Water and ammonia have heat transfer properties, both for single and two-phase heat transfer, far superior to all other fluids. The combination of low viscosities, high thermal conductivities and high specific & latent heats makes these fluids to the best possible heat transfer fluids. Ammonia could have a somewhat higher density to make it a perfect refrigerant, but this is a minor deficiency. There are some be tter fluids, but condensed hydrogen and helium are not liquids in the normal sense.

Ammonia has optimal properties in the important temperature range of -50 to +10 °C. Ammonia has one drawback, it is perceived as dangerous. No doubt, it is toxic and flammable but these properties alone do not make it dangerous, provided it is handled in a correct way. In certain respects, it is less dangerous than other refrigerants.

General. Ammonia, NH_3, is a gas at normal temperatures. It dissolves easily in water forming a complex compound simplified as NH_4OH. This is a strong base, i.e. it neutralizes acids, usually under a strong heat release.

It is a polar compound, i.e. the molecule has a positively and a negatively charged part, but is electrically neutral. It is a good solvent, especially for other polar compounds and salts.

At temperatures over 450 °C it disassociates and reverts to nitrogen and hydrogen. This is used when brazing, as the mixture is reducing and hence removes oxides.

Occurrence. Ammonia occurs naturally with all living materia because of decay of nitrogen containing organic material. A cow barn smells strongly of ammonia. It is as intrinsically connected to the circulation of nitrogen as carbon oxide is to the circulation of carbon.

Production. Ammonia was first isolated and produced from condensation of ammonia containing gases from decaying organic material. It is now mainly produced from hydrogen (from natural gas), nitrogen and a catalyst, the reverse of the disassociation.

Environment. Being a natural product, ammonia is the ultimate environmentally friendly refrigerant. It has no ozone depleting, nor any greenhouse effects. Even if it had, it would not matter; the strong affinity to water means that it will not reach very high or remain in the atmosphere.

Corrosive properties. Ammonia, in a joint attack with water and oxygen, corrodes copper and zinc.

Physiological effects. Even though it is a natural product ammonia is poisonous in too high concentrations. It is also irritating to the eyes, nose and lungs. See table 1. Ful recovery can be expected even after fairly severe exposure.

Flammable concentration in air is 15 to 28 %. It can burn by itself only in a confined space, for outdoor combustion a supporting flame is necessary.

Note that ammonia ***burns,*** it does not explode. It is possible to follow the flame front with the eyes. It has not the violent force of a natural gas explosion, which can completely destroy a building. In a building with a brick wall and no escape routes for the combustion gases, the wall would collapse. In a more solid structure, with escapes routes in form of e.g. roof windows, the window would shatter but the building remain. However, more research is needed on the type and size of escape routes.

Note also that millions of absorptions refrigerators, are installed all over the world, e.g. in hotel rooms, without large number of serious accidents. These refrigerators contain a high pressure mixture of ammonia and hydrogen and are sometimes driven by an open gas flame.

Vapour density. See table 2 and figures 14 & 15. This is a most important quantity. In case of a leakage, ammonia vapour will rise above the air.

Leakage of ammonia. The seriousness of a leakage depends on where the leak is situated.

- A leak from a vapour part of a plant would cause the vapour to leave. When the pressure decreases, the liquid evaporates. Some 10 to 15 % would evaporate and escape until a temperature of - 33.4 °C, the boiling point, is reached. As ammonia vessels are normally insulated, the remaining liquid ammonia evaporates very slowly. The high latent heat of ammonia contributes also to the slow evaporation. The ammonia vapour then rises upwards.
- A leak from a liquid containing part would cause all the liquid above the leak to escape plus some 10 to 15 %. The remaining liquid evaporates very slowly. The most severe consequence of a liquid leak is the:

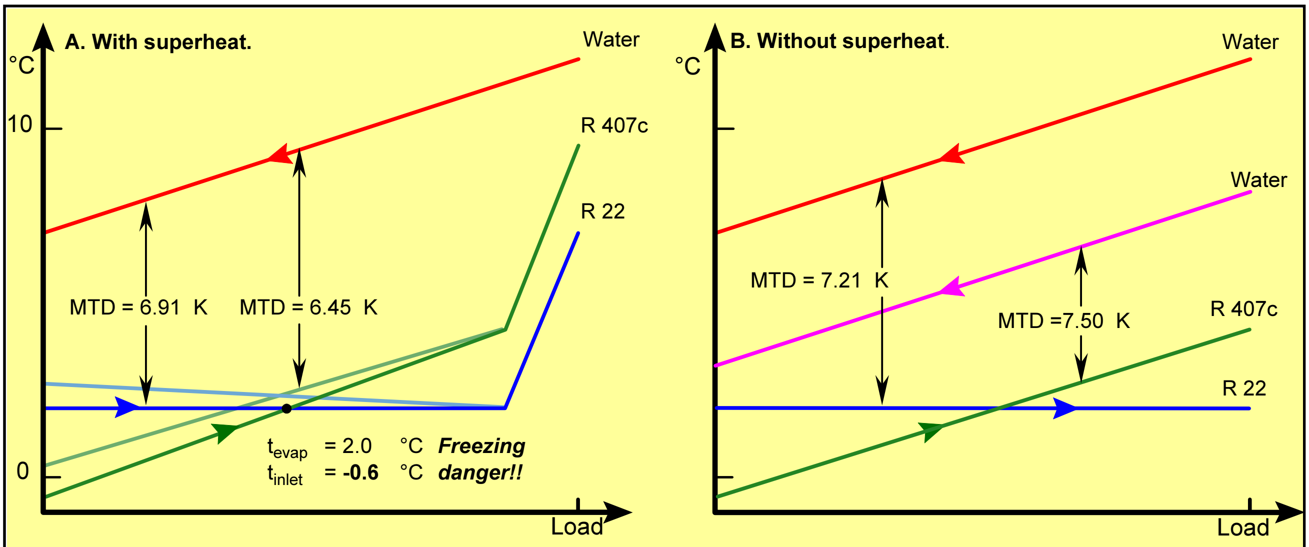


Fig. 13. MTDs for R 22 and R 407c.

The full lines show the temperature programmes without the pressure drops. Both R 22 & R 407c have the nominal evaporation temperature 2 °C. R 22 as real, R 407c as the mean between the inlet and the dew temperature.

Despite the evaporation temperature 2 °C, the inlet temperature of R 407c is below zero, i.e. ***freezing danger.***

The weak lines show the temperature programmes for a HE with pressure drops. The inlet temperature is then higher and there is less danger of freezing. In this case MTD_{R22} is slightly higher than MTD_{R407c}.

If we try to lower the water temperatures, there is rapidly a temperature pinch for R 407c as the inlet temperature of the water approaches the exit of R 407c, 9.6 °C.

Evaporation without the superheat as in B changes the situation. MTD_{R407c} is now higher than MTD_{R22}.

In this case we can lower the water temperatures (or increase the evaporation temperature) substantially more for R 407c than for R22.

However in the unlikely case of cocurrent flow it is easy to see that there will be a temperature pinch for R 407c.

Formation of an aerosol. Under certain circumstances, ammonia can form an aerosol, ammonia droplets in vapour. This aerosol can be heavier than air and therefore considerable more dangerous than an ammonia vapour. The formation of the aerosol is easier the higher the ammonia pressure, i.e. a leak from an HP receiver is more severe than from an LP receiver. The most severe effect happens if the leak is situated in a weak airflow, which carries away the aerosol.

In a totally quiescent air, as in a confined space, the aerosol remains, the droplets coalesce, the temperature drops and most of the liquid remains as liquid. A too strong airflow dilutes the aerosol and the droplets evaporate.

As the aerosol mixes with air or heats up, the droplets evaporate and as the gas mixture is lighter than air it moves upward. Note that ***comparisons cannot be made with chlorine and sulphur dioxide***, two common industrial compounds. Both are four times as heavy as ammonia and remains also as vapours close to the ground.

Leaks on pressure vessels. The ammonia in a refrigeration system is not contained in one large vessel, but spread out in interconnected vessels and pipes. This limits the leakage rate and gives time to close valves, which further limits the leakage.

The most likely leak in a system will be a faulty packing box, a gasket, a valve seat, etc. A pipe will not rupture in two parts but more likely a longitudinal crack will develop, corrosion does not make a vessel disappear suddenly, but starts with a small hole. All this limits the leakage rate and gives possibility to close valves, evacuate machine rooms, and activate alarms and other security measures.

Note that modern pressure vessels, manufactured and controlled according to the valid codes are high quality equipments and are very unlikely to suddenly rupture.

A liquid ammonia spill on a floor evaporates slowly, considerably slower than a freon spill. This is due to the large latent heat of ammonia. Note, never flush liquid ammonia with water, see also Warning 6, § 3.2.4. **6. Fouling & Corrosion.** Ammonia vapour can safely be scrubbed with water, though.

Noncondensables. Despite precautions, a refrigeration system can and will contain noncondensables (inerts). The chief effect of these will be in a decreased condenser capacity. The presence of inerts are sometimes difficult to establish and the venting can be difficult.

Ammonia is different from all other refrigerants. As it is readily soluble in water, it is enough to let the pipe/hose with the inerts discharge into a bucket with water. If bubbles emerges to the surface it is air, ammonia dissolves. See **5. Condensation and Liquid Receivers,** § 5. Venting, p. 106.

The large danger of ammonia does not come from ammonia in refrigeration plants, which is about 1 % of the world production. Ammonia is used as fertilizer and as raw material for the chemical industry and is transported by road, railway and ships and pumped between various vessels in very large quantities. Despite these risky operations, there are relatively few severe accidents.

Water in ammonia. The ammonia is now transformed to a refrigerant with glide, see table 3 and § 3.6.

Conclusion. Ammonia can be risky, but correctly handled it is in many respect safer than other refrigerants.

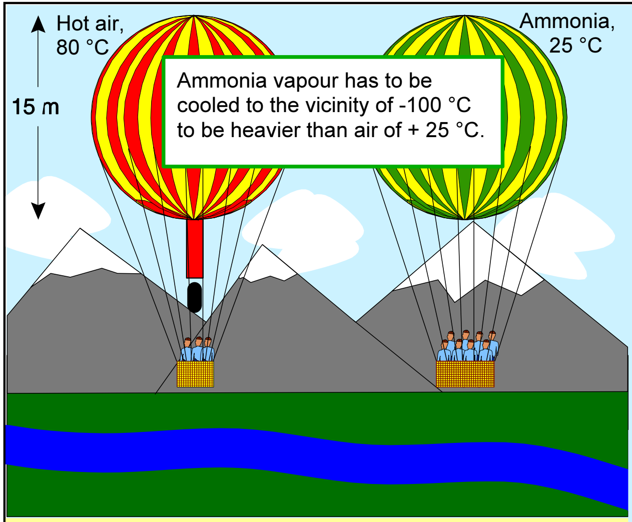


Fig. 14. Density of ammonia.

Ammonia is a very light gas. A balloon filled with ammonia would lift some five persons more than a comparatively sized hot air balloon.

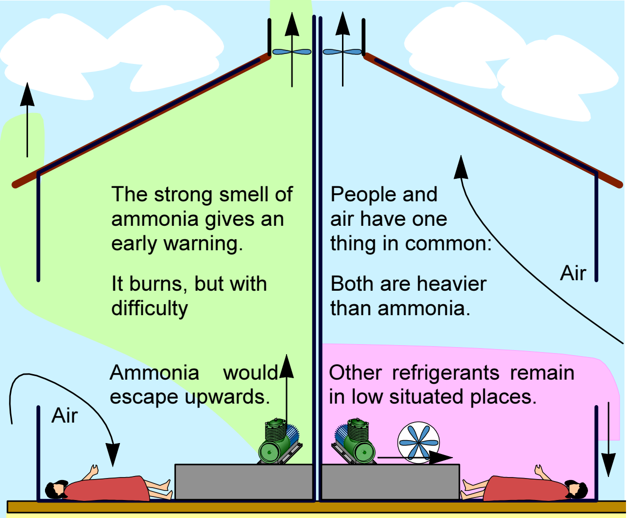


Fig. 15. Escape of refrigerants.

Air would remain close to the floor, while ammonia escapes upward, contrary to all other refrigerants. There are some exceptions, see the text.

3.9. Carbon dioxide.

Carbon dioxide rivals ammonia of being the oldest commercial refrigerant. While ammonia continues to be used in larger industrial installations and increasingly in other applications as well, carbon dioxide has in the last few years virtually ceased to be used. The reason is simple, the very low critical temperature, 30.9 °C needs a very cold cooling media for the condensers.

Physical properties. The vapour is about 52 % heavier than air and thus collects in low places. At a pressure above 5.18 bar there can be a solid, a solid-liquid, a liquid-vapour or a vapour phase depending on the temperature. At -56.6 °C/5.18 bar – the triple point – the three phase coexist.

Below 5.18 bar there is no liquid phase. At the boiling point of –78.2 °C, the carbon dioxide evaporates from a solid to a vapour, it sublimates. The resulting dry ice is used in transport refrigeration.

Hydraulic properties. The sublimation has important implications for the lay-out of a system and choice of equipment, especially valves. A security or relief valve is usually vented to the atmosphere via a pipe. When the pressure is released from a storage pressure of say 25 bar to 1 bar, the carbon dioxide flashes but not to a vapour-liquid mixture but a vapour-solid mixture. The solid can then easily clog a pipe and prevent the pressure relief. The exit from a safety valve should thus be directly to the atmosphere or maybe to a very large diameter pipe. See also **5. Condensation & Liquid Receivers,** §5. Venting, p.106.

Solid carbon dioxide can also form in a liquid at around the melting point. It forms rapidly, a valve or a pump can easily be clogged by “dry ice”.

For this reason care should be taken with the choice of valves. Best are ball and plug cock valves followed by gate and butterfly valves and worst globe valves.

Compared to halocarbons and hydrocarbons, carbon dioxide contains virtually no oil. Valves and pumps are thus deprived of the little lubricant they need and usually get from the dissolved oil and thus need the ability to operate “dry”.

Chemical reactivity. Dry carbon dioxide is virtually inert, thus not corrosive, nor does it burn or react with oils.

A “new” application is condensing of carbon dioxide with evaporating ammonia, see below. In this condenser/evaporator the carbon dioxide pressure is considerably higher than the ammonia pressure. In case of a leak on the heating surface carbon dioxide enters the ammonia side.

Carbon dioxide and ammonia will then immediately form ammonium carbamate, which is corrosive and could destroy a compressor. It is readily dissolved in water.

Check with the compressor manufacturer if the vapour-liquid separator in a thermosiphon system should be equipped with a demister as extra protection.

Note that leakage between the two side is a very rare phenomena in a PHE.

Physiological effects. Low concentrations stimulates breathing - the refreshing effect from carbonated beverages but in higher concentrations - ~ 20 % - it causes death, mainly by suffocation. Symptoms of carbon dioxide poisoning are nausea, vertigo, dizziness, headache, sight & hearing difficulties, thus general and insidious. However. the allowable concentration is in the order of 5000 ppm, several times higher than most HCFCs.

Environmental effects. Zero ozone depletion potential but a fairly low green house effect, see below though.

Manufacturing. Carbon dioxide is, contrary to all other refrigerants except water, not manufactured, it is recovered, mainly from flue gases, fermentation processes and limekilns. The environmental effects are thus more realistically described as zero. Carbon dioxide escaping from a refrigeration plant is recovered elsewhere and thus the net effect to the atmosphere is zero.

Refrigeration properties. Carbon oxide is cheap enough to be used in once-through refrigeration. It is liquefied (solidified), evaporates at the refrigeration and the vapour is discarded to the atmosphere. The liberated carbon dioxide has the important secondary effect that it protects food from attacks by bacteria and fungi.

Heat exchangers. No particular problems are expected. Insoluble oil could pose difficulties at low temperatures.

Water dissolved in the carbon dioxide can freeze and block components. A good filter/dryer is mandatory.

Saturation temp. °C	Bubble point 100% condensed 0% evaporated			10% condensed 90% evaporated			1% condensed 99% evaporated			Dew point 0% condensed 100% evaporated		
	0.01	0.1	1.0 % H_2O	0.01	0.1	1.0 % H_2O	0.01	0.1	1.0 % H_2O	0.01	0.1	1.0 % H_2O
-40	0.0	0.1	0.3	0.0	0.3	2.0	0.3	2.0	43.1	15.2	32.2	52.7
-35	0.0	0.1	0.3	0.0	0.3	2.1	0.3	2.1	42.3	13.8	30.7	52.0
-30	0.0	0.1	0.3	0.0	0.3	2.2	0.3	2.2	41.6	12.1	29.0	51.4
-25	0.0	0.1	0.3	0.0	0.3	2.3	0.3	2.3	40.7	10.5	27.3	50.4
-20	0.0	0.1	0.3	0.0	0.3	2.5	0.3	2.5	39.8	8.7	25.4	49.4
-15	0.0	0.1	0.3	0.0	0.3	2.7	0.3	2.6	38.9	6.9	23.4	48.3
-10	0.0	0.1	0.3	0.0	0.3	2.8	0.3	2.8	38.0	5.2	21.4	47.1
-5	0.0	0.1	0.3	0.0	0.3	3.0	0.4	2.9	37.0	3.9	19.5	45.8
0	0.0	0.1	0.4	0.0	0.4	3.2	0.4	3.0	35.9	3.0	17.5	44.4
5	0.0	0.1	0.4	0.0	0.4	3.4	0.4	3.1	34.8	2.5	15.6	43.0
10	0.0	0.1	0.4	0.0	0.4	3.6	0.4	3.1	33.7	2.2	13.8	41.5
15	0.0	0.1	0.4	0.0	0.4	3.7	0.4	3.3	32.4	2.0	12.1	39.9
20	0.0	0.1	0.4	0.0	0.4	3.9	0.4	3.3	31.1	1.9	10.5	38.2
25	0.0	0.1	0.5	0.0	0.5	4.0	0.5	3.3	29.6	1.8	9.1	36.6
30	0.0	0.1	0.5	0.1	0.5	4.2	0.5	3.3	28.1	1.8	7.9	34.9
35	0.0	0.1	0.5	0.1	0.5	4.3	0.5	3.3	26.5	1.7	6.8	33.2
40	0.0	0.1	0.5	0.1	0.5	4.4	0.5	3.3	25.0	1.7	5.8	31.5
45	0.0	0.1	0.5	0.1	0.5	4.5	0.5	3.3	23.4	1.6	5.1	29.8
50	0.0	0.1	0.6	0.1	0.6	4.6	0.5	3.3	22.1	1.5	4.6	28.2

Table 3. The saturation temperatures of the system ammonia-water.

The table gives the bubble & dew point increases in ° K for 0.01, 0.1 and 1.0 % water in ammonia, together with the intermediate 90% and 99% vapour points.

Ammonia in commercial refrigeration systems contains some water, usually in ppm quantities, occasionally more. The evaporation then starts at the bubble point and ends at 100 % vaporized, the dew point. The condensation starts at the dew point and ends at the bubble point.

Ex. Ammonia with 0.1 % water evaporates in a DX evaporator at nominally 5 °C , by water from 15 to 10 °C.

The evaporation starts at 5 + (0.1) = 5.1 °C. The first part occurs in the expansion valve and the mixture enters a few tenths degrees higher. At 5 + (+0.4) = 5.4 °C, 90 % has evaporated. However, the temperature has to increase to 5 + (+3.1) = 8.1 °C to evaporate 99 %.

A TEV set to a superheat of 5 + 5 = 10 K would see a vapour, which is not superheated but containing somewhat less than one percent liquid.

The NH_3 has to be heated to 5 + 15.6 = 20.6 °C for all the liquid to evaporate. A DX system is here doubtful.

Thermal properties. As can be seen from table 5, carbon dioxide is in general a good refrigerant, secondary only to ammonia. The outstanding property is, however, the high vapour volumetric refrigeration capacity and the related high vapour pressure.

New applications. Environmental concerns have meant an increasing interest for ammonia. To reduce the danger of accidental leaks, especially in confined spaces and spaces frequented by the general public, type supermarkets, indirect systems are increasingly used.

The drawback of indirect systems is a loss of temperature difference and the pumping cost for high viscosity brine.

A supermarket can serve as an example. In addition to the air conditioning requirements, refrigerants of two temperature levels, e.g. -10 and -35 °C, are needed.

The lower temperature level would merit a cascade system. An optimized cascade system should use a refrigerant with a high vapour pressure in the LT stage to increase the volumetric cooling capacity. The drawback of this is the necessary very high design pressure.

- A system, as in figure 16, with a central ammonia refrigeration plant supplying all coils & unit coolers with liquid carbon oxide solves all of these problems.
- The low viscosity and high volumetric refrigeration capacity gives no or small pumping costs and small pipe diameters. The central cooling station can be far away from the end applications.
- Ammonia evaporates at -15 °C and condenses carbon dioxide at -10 °C.
- Part of the liquid carbon dioxide is transported, by a pump or natural circulation, to the HT applications. It evaporates here at -10 °C. This part of the system is actually a heat pipe.
- Because of the virtual isothermal operation, the total temperature difference between the evaporating ammonia and the cooled air is maximized. There is no loss due to a temperature increase as the case of a brine.
- The other part of the liquid is expanded to -35 °C and used for the LT applications and finally compressed again to -10 °C, i.e. a normal refrigeration cycle.
- In case of an emergency shutdown, e.g. a power failure, the carbon oxide can simply be vented to the atmosphere. The design pressure can then be limited to a more usual 26 bar. See **Hydraulic properties**, p. 148.

  There are many refrigerants, which can be used in a cascade system, but only carbon oxide can be vented without any concern for the cost or environment.

3.10. Secondary refrigerants.

Water is an excellent single-phase heat transfer fluid but the relatively high freezing point means that an antifreeze compound has to be added.

Most antifreeze compounds are corrosive to zinc and iron, some to stainless steel as well, why corrosion inhibitors are needed.

		Dens.*Spec.Heat kJ/°C,m^3		Viscosity cP		Thermal cond. W/m,°C		Comments
Freezing point	°C:	-30	-50	-30	-50	-30	-50	
Data at	°C:	-20	-40	-20	-40	-20	-40	
Water at 0°C:		4200		1.3		0.59		Excellent, but limited to above 0 °C.
Ammonia (l):		3001	3046	0.21	0.28	0.62	0.69	Irritant, toxic, high pressure, flammable.
Ammonium hydroxide (40%):		4257	4224	3.5	6.6	0.53	0.52	Irritant. Not flammable, nor high pressure.
Calcium chloride:		3589	3465	8.4	33	0.51	0.48	Corrosive.
Sodium chloride:		3877		5		0.48		Corrosive, limited to -20 °C
Ethanol-water:		3500	2853	22	35	0.35	0.30	Low boiling point, flammable.
Methanol-water:		3707	3370	9	20	0.41	0.35	Low boiling point, flammable. Toxic
Ethylene glycol:		3484	3035	19	140	0.36	0.30	Somewhat toxic.
Propylene glycol:		3601	3265	63	824	0.34	0.29	Less toxic but high viscosity
Potassium formate:		3542	3165	5.5	44	0.48	0.48	Long time corrosive properties?
Potassium acetate:		3830	3715	30	90	0.29	0.27	Corrosive for zinc.
Silicon oil (typical):		1512	1500	3.1	5.2	0.12	0.12	Expensive & ineffective, used at low temp.
		Excellent ########		Medium ########		Bad ########		

Table 4. Comparison of secondary refrigerants.

A good secondary refrigerant should:

- Not be corrosive.
- Has a high volumetric heat capacity (ρ * Cp) to minimize the temperature change in a heat exchanger.
- Has a low viscosity and high thermal conductivity.
- Not be toxic.
- Be environmental acceptable.
- Be economical.

See table 4 for a list of some secondary refrigerants.

Ammonia rivals water as a heat transfer fluid par excellence, but the low boiling point and toxicity limit its use.

Ammonium hydroxide is ammonia dissolved in water. This is an excellent heat transfer fluid. Many of the drawbacks of ammonia are eliminated or reduced. In case of a leakage, very little vapour is released into the air, as the partial pressure of ammonia in water is very low. Irritating but hardly toxic concentrations would be the result.

Calcium & Sodium chlorides are cheap and efficient brines, which are acceptable in the food industry. The draw back is the chlorine ions, which can cause corrosion, see **6. Fouling & Corrosion**, §5.3.2. Calcium chloride is inhibited with potassium bichromate, which is environmentally questionable, better is potassium molybdate. This decreases corrosion on copper but not on steels.

Calcium chloride brine has a minimum freezing point of - 55 °C at 29.87 w%. At lower concentrations the liquid freezes, at higher $CaCl_2$ precipitates. The minimum is very sharp, at 29.0 w% it freezes at - 45.2 °C, at 30.0 w%, the precipitation starts at - 46 °C (data from ASHRAE). A strict control of the concentration is necessary, especially for evaporation temperatures below - 40 °C as otherwise the heating surface could easily be clogged.

Ex. In a plant operating with an evaporation temperature of -49 °C, the heat transfer was too low. A check revealed a concentration of more than 30 % and a wall temperature of less than - 45 °C. $CaCl_2$ precipitation was the problem.

Ethylene glycol is the standard antifreeze fluid. It is slightly toxic and is thus not used in the food industry.

Propy*lene glycol* is less toxic and is the standard antifreeze in the food industry. The drawback is the very high viscosity at high concentrations and at low temperatures (< - 30 °C). This can cause problems with the distribution of both brine and refrigerant from channel to channel in a PHE, especially if the brine pressure drop is low.

Alcohol solutions are good secondary refrigerants but the low boiling points means that the alcohol could evaporate in case of open systems at elevated temperatures.

At elevated temperature, they become flammable as well. All commonly used metals and gaskets are resistant.

Potassium acetate (Tyfoxit, GS4) and ***potassium formate*** (Freezium, Hycool) - the salts of acetic and formic acids - or mixtures (Temper, Pekasol) are recent compounds. Some properties:

- Excellent thermal properties. Potassium formate has ten times lower viscosity at low temperatures than propylene glycol. Acetate is somewhat less advantageous.
- Environmentally acceptable as both are naturally occurring compounds. Acetic acid comes from oxidation of alcohol and formic acid was first isolated by distillation of ants. At higher concentrations, both are toxic but the toxic level for e.g. potassium formate is higher than for sodium chloride, table salt. They are thus good replacements for propylene glycol.
- Both are corrosive, formates somewhat more, and have to be inhibited. Various manufacturers have different proprietary solutions, such as borax, metasilicates, carboxylic acids, etc. Long time experience is necessary, but zinc and high zinc brass (> 15%) are not allowed. l.
- Some formate/acetate mixtures have a high pH, which can be irritating for the skin, while other can be foaming.
- A thorough cleaning of the system is necessary before the brine is charged. The oxygen should be evacuated by e.g. a nitrogen pressure test. The manufacturers recommendations should be strictly followed, otherwise corrosion can occur on copper as well.
- Plastic pipes should be avoided. Air diffuse through these and the system can then be difficult to deareate.

Silicon oils are advantageous at low temperatures due low viscosity. They are chemically inert, thus not corrosive, neither for metals nor gaskets. They do not have, apart from the low viscosity, very good thermal properties, but for temperatures lower than -50 °C, they are sometimes the only choice.

Thermal oils are mainly used at high temperatures but some are used at low temperatures as well. They are not aggresive to metals (some are towards gaskets). They have poor thermal properties but their advantage is the very large temperature span they cover.

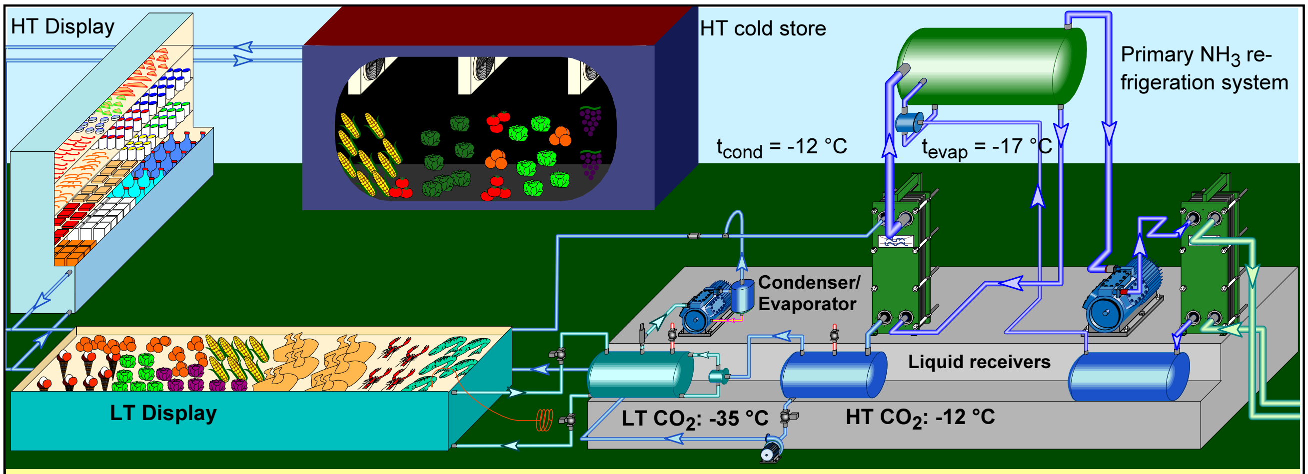


Figure 16. Ammonia as the primary and carbon dioxide as the secondary refrigerant.

Carbon dioxide condenses in the ammonia evaporator at -12 °C. The liquid carbon dioxide is used

either directly in the HT displays

or expanded to -35 °C for the LT displays. The flows to the displays are normally controlled by the display temperature through an on-off thermostat.

Depending on the distance the carbon dioxide is pumped or flows by natural circulation. Natural circulation in an HT unit is essentially a heat pipe, i.e. condensation and evaporation at the same pressure.

Heat exchangers for carbone dioxide should be:

- able to withstand the HT pressure. For carbon dioxide of -12 °C it is 25 bar = 24 barg. A safety valve releases at higher pressure than 25 barg. A lot of equipment is designed for 25 bar, thus 25 bar is a suitable pressure.
- equipped with an oil filter, as oil at low temperatures can block part of the heating surface.
- designed to avoid thermal shock in case of S&THEs or CBs in especially DX systems. See 1. **Applications**, §11, p. 12. WPHEs are normally uncritical.

Refrigerant		**R717**	**CO$_2$**	**R1270**	**R22**	**R404a**	**R410a**	**R507a**	**R134a**
Volumetric latent	Liq. kJ/m^3	839463	267590	212890	274065	198828	278502	195548	262606
coolin capacity	Vap. kJ/m^3	1030	8137	1445	1220	1413	1926	1483	677
COP		6.78	6.29	6.96	6.95	6.82	6.87	7.01	7.01
Liquid (ρ *C$_p$)	kJ/(K, m^3)	3037	2197	1308	1528	1605	1796	1605	1773
Viscosity	cP	0.262	0.184	0.182	0.323	0.280	0.254	0.277	0.438
Thermal conductivity	W/m, K	0.671	0.142	0.125	0.111	0.093	0.135	0.081	0.108
Temperature at 25 bar,	°C	58.2	-12.0	59.4	61.4	53.5	41.4	52.5	77.6

Common data:

Evaporation at:	-35 °C,	#######	Excellent
Condensation at:	-10 °C,	#######	Medium
Compressor efficiency: 80%.		#######	Bad

Table 5. Thermal properties of some refrigerants at -35 °C.

Property	**Units**	**R22**	**R717**	**R134a**	**R404a**	**R407c***	**R410a**	**R1270**	**R290**
Condensation temperature.	°C	40	40	40	40	40	40	40	40
Condensation pressure,	bar	15.3	15.5	10.1	18.3	16.5	24.1	16.6	13.7
Compressor efficiency,	%	87.3	87.6	87.7	87.1	87.6	87.1	86.4	86.7
Discharge temperature	°C	66.6	105.4	51.4	52.0	62.7	68.0	57.5	51.9
Condensation capacity	kW	4.29	4.73	2.73	4.47	4.34	6.26	4.46	3.65
Superheat capacity.	%	12.4	14.5	7.4	9.6	10.4	15.8	11.8	7.7
COP	Heating	6.28	6.44	6.35	5.79	6.14	5.86	6.12	6.19
Condensation temperature.	°C	60	60	60	60	60	60	60	60
Condensation pressure,	bar	24.3	26.4	16.8	29.0	26.6	38.4	25.4	21.2
Compressor efficiency,	%	85.0	82.2	83.0	85.3	83.8	85.2	86.3	86.0
Discharge temperature	°C	96.9	***161.7*** **	75.6	76.8	91.3	99.2	82.0	74.2
Condensation capacity	kW	4.03	4.83	2.49	3.62	3.86	5.4	24.00	3.27
Superheat capacity.	%	20.9	23.4	12.8	22.8	20.2	34.0	20.2	12.2
COP	Heating	3.93	4.08	3.41	3.24	3.63	3.39	3.82	3.84

Common data: Evaporation temperature: 2 °C, Superheat: 5 K, Subcooling: 3 K

Comments: * Entrance temperature below zero. ** Far too high.

Table 6. Refrigerant properties in heat pump applications.

1. Equipment.

HE:	A heat exchanger in general.
S&THE:	A shell and tube heat exchanger.
SHE:	A spiral heat exchanger.
UC:	A unit cooler. Used to cool air.
LC:	A liquid cooler. Used to cool a liquid by air.
ACC:	An air cooled condenser.
PHE:	A plate heat exchanger in general.
GPHE:	A gasketed PHE.
SWPHE:	A semiwelded PHE. A PHE where two plates are welded together, forming a cassette. The refrigerant flows inside the cassette, the water in the gasketed space between the cassettes.
AWPHE:	An all welded PHE.
WPHE:	A SWPHE or a AWPHE.
BPHE:	A brazed PHE in general.
CB:	A copper brazed PHE.
NB:	A nickel brazed PHE.
LR:	A liquid receiver.
EV:	A valve used to expand and vaporize the liquid in direct expansion system.
TEV:	A thermostatic expansion valve.
EEV:	An electronic expansion valve.
DX:	Direct expansion.

2. Plates and channels.

A PHE is composed of a number of plates stacked together. The space between two ***plates*** form a ***channel.***

- Two ***H(igh)-plates*** form one ***H-channel.***
- Two ***L(ow)-plates*** form one ***L-channel.***
- One ***H-*** and one ***L-plate*** form a ***M(edium)-channel.***

Note 1! There no such thing as an ***M-plate.***

Note 2! These are the most common channel and plates types and are used in examples in this book. A PHE could have a number of different plates, miscible or not.

Single channels:	L, M or H channels only.
Mixed ” :	L & M or M & H mixed. Mixed arrangements are normally not used for two-phase fluids. L & H are normally not mixed as channels.
Asymmetrical channels:	The channels of the two sides have unequal thermal and hydraulic prop erties.
Multi-circuit unit: (Dual circuit unit)	One side is divided into two or more independent circuits.
Parallel channels:	All the channels in a PHE are arranged in parallel.
Channels in series:	A groups of channels arranged in parallel, a pass. The passes are arranged in series. A multi-pass unit.

Note 3! The two sides in a PHE are independent, i.e. one side could have a parallel arrangement and the other two groups in series.

Parallel flow:	The inlet and exit of the flows are on the same side of the PHE for each flow.
Diagonal flow:	The inlet and exit of the flows are diagonally placed on the PHE for each flow.

3. Thermal and hydraulics.

Side 1:	Normally the warm side .
Side 2:	” ” cold ”
A, m^2:	Heat transfer area.
ΔP:	Pressure drop, Pa or lb_f/ft^2
ΔP_1, kPa:	Pressure drop on side 1 (warm).
ΔP_2, kPa:	Pressure drop on side 2 (cold).
Δt_1, K:	Temperature change on side 1 (warm).
Δt_2, K:	Temperature change on side 2 (cold).
Δt:	The temperature difference between the media in a given point.
Δt_w, K:	The warm end temperature difference.
Δt_c, K:	The cold end temperature difference.
K-value:	(U-value). The overall heat transfer coefficient. Measured in W/m^2, °C.
K_s, K_i	Coefficients used in Appendix II.
MTD, Δt_m K:	Mean Temperature Difference.
LMTD, Δt_{lm},K:	Logarithmic Mean Temperature Difference. See below for a closer definition of MTD, LMTD and K-value
Θ or Theta:	Thermal length. Δt_1 / MTD or Δt_2 / MTD. Θ gives the temperature change of a fluid pro degree temperature difference.
TTL:	Terminal Thermal Length. $\Delta t_1/\Delta t_w$, $\Delta t_2/\Delta t_w$, $\Delta t_1/\Delta t_c$ or $\Delta t_2/\Delta t_c$.
q, kW:	The heat transferred up to a given point.
Q, kW:	Total heat transferred, duty or capacity.
t, °C:	Temperature.
T, K:	Absolute temperature. T = t + 273.16.
T_c,K:	Critical temperature.
t_b, °C, T_b, K:	Normal boiling point.
T_r:	Reduced temperature = T / T_c
p, bar:	Gauge(over) pressure.
P, bar:	Absolute pressure.
P_c, bara:	Critical pressure.
P_r:	Reduced pressure.

Note 4! In refrigeration terminology, ***temperatures*** are usually expressed in °C, ***temperature differences*** in K.

4. Units and conversion factors.

g:	9.8057 m/s^2 = 32.17 ft/s2
1 RT	3.504 kW. One Refrigeration Ton is the necessary heat to melt 2000 lbs (1 short ton = 907.18 kg) of ice during 24 hrs = 79.71 * 4.186 * 907.18 / 86400 = 3.504 kW.

Note 5! Although a power unit, comparable to kW, horse power or kcal/hr, RT is customarily used only for cooling. A 400 RT condenser probably means a condenser serving in a system where the evaporator gives 400 RT and not a condensing capacity of 400 * 3.504 = 1401.6 kW. Note the danger of misunderstandings.

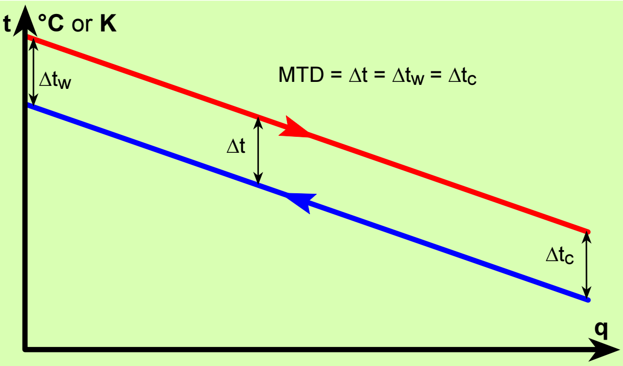


Fig. 01. Constant temperature difference .

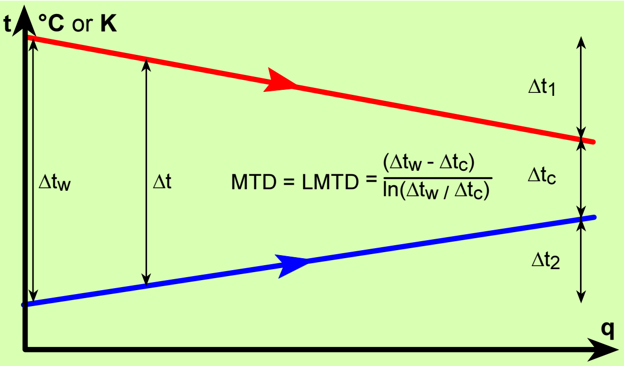


Fig. 02. Changing temperature difference.

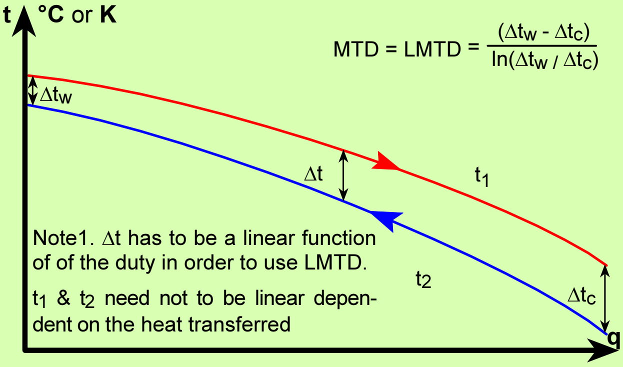


Fig. 03. A non linear temperature change.

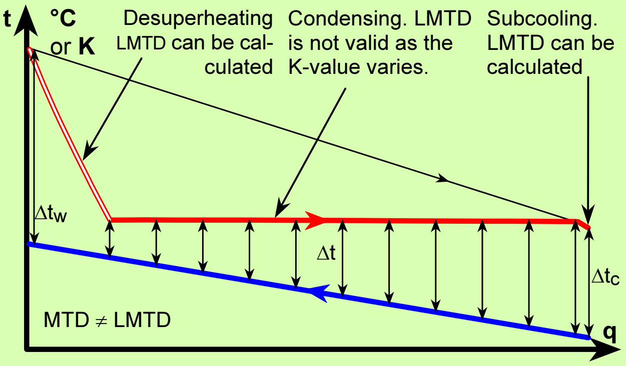


Fig. 04. Nonlinear changes in K-value and ΔT.

5. MTD, LMTD and K-value.

In the general heat transfer equation:

$$Q = K * A * \Delta T \qquad \text{(Eq. 1)}$$

the quantities Q (duty) and A (Area) are fairly unambiguously defined.

In case of a duty with no phase change, K (the overall heat transfer coefficient) and ΔT (the temperature difference), are also easily defined. Especially in Fig.1 , where the temperature difference is constant from inlet to exit, ΔT is simply equal to this temperature difference.

If the temperature change varies from the cold side to the warm side, Fig. 2, the situation gets more complicated. The simple temperature difference ΔT has to be replaced with some mean temperature difference, MTD. The question is how MTD shall be calculated.

If the duty complies with the following requirements, then LMTD, as defined in Fig. 2, can be used as a suitable mean value in the calculations of HEs:

- The K-value has to be constant.
- Δt has to be a linear function of the heat transferred.

Fig. 3 shows another valid case. The specific heat of the media varies with the temperatures. This makes the temperature vs. heat transferred curved, but in this case the ΔT is still linear and LMTD can still be used.

Figure 4 shows condensation of a superheated vapour as in the following example:

R 22 vapour enters at 73, condenses at 40 and subcools to 37 °C. Water enters at 27 and leaves at 37 °C.

As can be seen from the figure, ΔT varies from the inlet to the outlet in a highly irregular way. So does the K-value. In the beginning it is a gas-liquid duty, i.e., a low K-value, then changes to condensing-liquid with a high K-value and finally subcooling with a medium K-value. None of the requirements above are fulfilled and the true mean temperature difference, MTD, has to be calculated.

In such a case, the duty has to be separated into three subduties, the desuperheating, the condensing and the subcooling duties. The Δt_m and the K-value have to be calculated separately for the three duties.

At the subcooling and superheating duties, LMTD and K-value can usually be used to calculate the areas. However, at the condensing duty, the K-value varies and the duty has to be divided into zones and Δt and K-value calculated for each zone. The area for each zone can then be calculated.

MTD is then calculated as a mean of the Δt in the desuperheating, subcooling and the zones in the condensing section. Calculated in this way, the MTD for the above duty is 6.9 K. The LMTD is in this case 20.3 K. It implies that the refrigerant temperature follows the black line in Fig. 4, which clearly is not the case.

The total area, A, is simply found by summarizing all the individually areas. The K-value is then finally found from:

$$K = Q / (MTD * A) \qquad \text{(Eq. 2)}$$

The K-value is thus a final result of, not a parameter used in, the calculation of the area.

1. $F_g = (\rho_l - \rho_g) * g * (D_p^3 * \pi / 6)$
2. $F_f = C_d * (D_p^2 * \pi / 4) * \rho_g * v_t^2 / 2$
3. At equilibrium: $F_g = F_f$
4. $v_t^2 = 4 * g * D_p * (\rho_l - \rho_g) / (3 * C_d * \rho_g)$
5. $C_d = f(Re)$ from figure 02.
6. In order to calculate v_t, C_d has to be known. However, C_d is a function of v_t and a trial and error procedure is needed. Usually only a few trials are necessary.
7. The nominal design velocity is taken as a fraction of the settling velocity.
8. $v = (0.75 \text{ to } 0.9) * v_t$
9. $D = (4 * V / (\pi * v))^{0.5}$

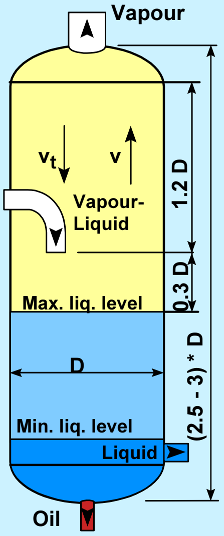


Fig. 01. Vertical separator design.

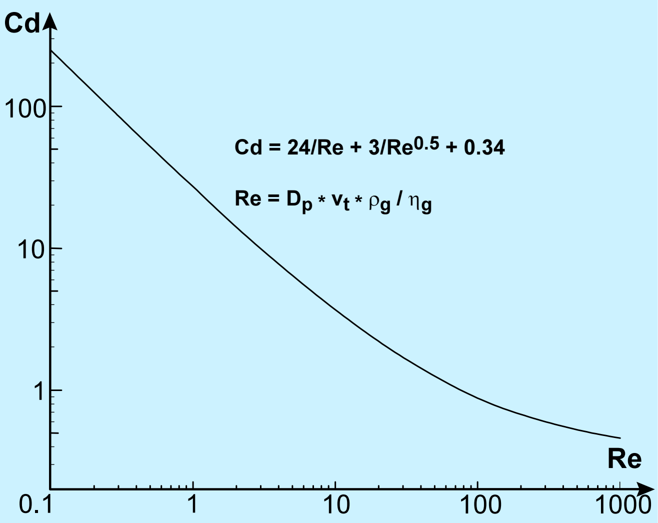


Fig. 02. The drag coefficient Cd vs. Re.

1. The state of separation design.

This appendix covers the quantitative aspects of vapour-liquid separators and the thermosiphon loop. In **8. Evaporators and Separators,** §4 is a discussion of separators.

Separators cannot be designed with only the knowledge of the quantity and quality of the inlet refrigeration vapour. Separators act not only to separate liquid from vapour but also serves as recipients for refrigerant at load variations or shutdowns. The type and number of condensers and evaporators in the system determine how large these variations are. One compressor type can be more resistant to residual liquid refrigerant than others. One and the same refrigerant can behave different what regards foaming from plant to plant, probably depending on the cleaning procedures. The control system can have different demand on the reaction time from plant to plant.

All these factors influence the separator design. Fortunately, separators designed to operate with PHEs do not normally need to be designed for large volume changes at load variations as PHEs have small refrigerant fillings and are normally placed close to the separator.

Below is a review of some published design methods. All methods refer to earlier published papers, which in turn refer to earlier works. This chain can go back more than 50 years. It is obvious that errors can be (and are) introduced somewhere in such a chain, especially if unit conversions are involved. In more severe cases, the original source should be consulted before the design method is used. One such a case is discussed later.

There are large variations in the results as separators are not equal. Inlet and exit arrangements, baffles, demisters, vessel shapes, type of fluids, process conditions etc. influence the separation efficiency and are responsible for the spread in the results. A manufacturer, which has successfully used a design method for his particular separator type, should obviously continue to use this method.

Conclusion. The design methods reviewed here could be used to check a design and for guidance.

P_{nom}:	Nominal pressure = 1 bar = 14.504 psi		
T_{nom}:	Nominal temperature 273.16 K = 491.69 °R		
ρ_n:	Nominal density, 1 =kg/m^3 = 62.43 * 10-3 lb/ft^3.		
F_g, F_f:	The gravity and friction drag forces on a droplet.		
V, V_l, V_g:	Total, liq. & vap.	**volume flow**,	m^3/s or ft^3/s.
v, v_l,v_g:	“ “ “	**linear velocity**,	m/s or ft/s.
M, M_l, M_g:	“ “ “	**mass flow**,	kg/s or lb/s.
m, m_l,m_g:	“ “ “	**mass velocity**	kg/m^2,s or lb/ft^2,s.
y:	Vapour fraction.		
D_p:	Droplet Ø, use 152 * 10^{-6} m = 499 * 10^{-6} ft.		
v_t:	Terminal velocity of the droplet of diameter D_p, m/s or ft/s.		
v_{re}:	Re-entrainment velocity, m/s or ft/s.		
ρ_l, ρ_g:	Liquid & vapour densities, kg/m^3 or lb/ft^3.		
η_l, η_g:	Liquid & vapour dynamic viscosities, kg/m,s or lb/ft,s.		
σ:	The liquid-vapour surface tension, N/m or lb/s^2.		
ϕ_g2	The ratio two-phase to vapour pressure drops.		

General Indices

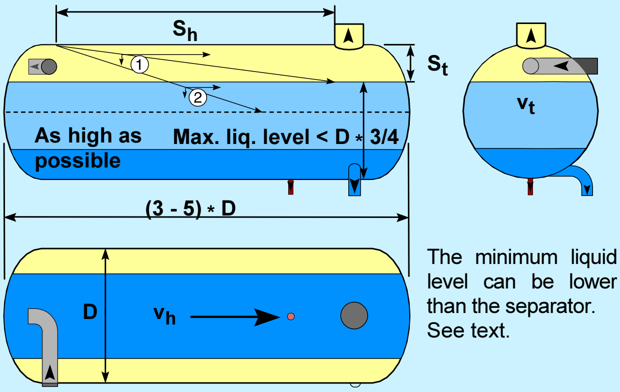
b	Bend.
v	Valve or vertical.
r	Sudden reduction.
e	Sudden expansion.
p:	The PHE inlet or exit port.
tp	Two-phase flow with slip.
h	Homogeneous two-phase flow or horizontal.
s	Static.
f	Friction.
m	Momentum.

Tab. 1. Symbols & constants in appendix III.

However, without a written agreement between Alfa Laval and a second part for a specific case, they cannot constitute the base for a warranty as Alfa Laval normally does not sell separators and thus cannot take any responsibility for a design.

2. Vertical separators.

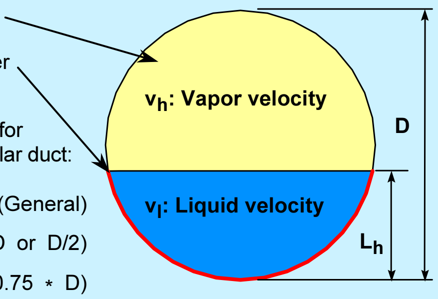
The principle of the vertical separator is simple. The vapour-liquid mixture enters the separator and the vapour flows upwards with a certain velocity v, see figure 01.



The minimum liquid level can be lower than the separator. See text.

1. v_t, the settling velocity, is obtained from figures 01 & 2.
2. Try a horizontal vapour velocity $v_h = (1 \text{ to } 5) * v_t$
   Check with figure 4, if this is low enough.
3. Assume a maximum liquid level: $L_h \leq 0.75 * D$.
   A suggested value is: $L_h = 0.5 * D$.
4. Calculate the diameter from $D = (8 * V / (\pi * v_h))^{0.5}$
   Note! The suggested value of L_h from 3 is used.
5. The settling height S_t is then: $S_t = D - L_h = 0.5 * D$.
6. Calculate the separation distance: $S_h = M_a * S_t * v_h / v_t$
   M_a is a margin which should be in the order of 1.10 to 1.25.
   Try to get a holding time, sec: $3 < S_h / v_h > S_t / v_t$.

Fig. 03. Horizontal separator design.



Hydraulic diameter for liquid flow in a circular duct:

$d_{hl} = 4 * A / P$ (General)

$d_{hl} = D$ ($L_h = D$ or $D/2$)

$d_{hl} \approx 1.2D$ ($L_h = 0.75 * D$)

$d_{hl} \approx 0.6D$ ($L_h = 0.25 * D$)

A horizontal circular duct or pipe

Liquid Reynold #.: $Re_l = (d_{hl} * v_l) / (\eta_l / \rho_l)$

Density factor: $R\rho = (\rho_l / \rho_g)^{0.5}$

Interfacial viscosity #: $N = \eta_l * (\rho_l * \sigma)^{-0.5} * \left[(\rho_l - \rho_g) * g/\sigma\right]^{0.25}$

Eq.	Re_l	N	Re-entrainment velocity v_{re}
A.	<160	-	$>1.5 * (\sigma/\eta_l) * R\rho * Re_l^{-0.5}$
B.	$160 \leq Re_l \leq 1635$	≤ 0.0667	$>11.78 * (\sigma/\eta_l) * R\rho * N^{0.8} * Re_l^{-1/3}$
C.	$160 \leq Re_l \leq 1635$	>0.0667	$>1.35 * (\sigma/\eta_l) * R\rho * Re_l^{-1/3}$
D.	>1635	≤ 0.0667	$>(\sigma/\eta_l) * R\rho * N^{0.8}$
E.	>1635	>0.0667	$>0.1146 * (\sigma/\eta_l) * R\rho$

Maximal values of v_h to prevent re-entrainment. Adapted from (3). Note that higher turbulence gives lower velocities, thus use equations D & E, which are not dependent on the geometry.

Fig. 04. Reentrainment of liquid droplets.

The bulk of the liquid moves directly downward but smaller droplets are suspended in the vapour for a while. The droplets settle with different velocities, slower the smaller the droplet. To separate all droplets with a diameter larger than D_p and the corresponding terminal velocity v_t, it is sufficient to keep vertical vapour velocity v_v less than v_t, normally with a certain margin, around 25 %.

Once v_v is known and with the given vapour volume flow (including flash vapour) V, the diameter D and the other dimensions of the separator can be calculated.

The drop diameter should be 100 to 200 µ. 152 gives a vapour velocity of 0.5 m/s for ammonia of -10 °C, a velocity is used in many commercial installations, see below.

Figures 01 & 02 contain all necessary information for the design and they are valid for all refrigerants and can be used with any consistent set of units whether SI, Metric or British. See also table 5 for settling velocities and **4. Evaporators & Separators**, § 4

3. Horizontal separators.

Horizontal separators are more complicated than vertical. The basic operation principle is simple, even though it differs from vertical separators. As in vertical separators, droplets larger than a certain diameter D_p, with the corresponding terminal velocity v_t, have to be separated. The vapour-liquid mixture enters in one end of the separator, travels horizontally along the separator and leaves at the other end. A droplet therefore moves in two directions:

- Horizontally with the vapour velocity v_h the distance S_h.
- Vertically with the settling (terminal) velocity v_t the maximum distance S_t.

If the separation distance S_h is long enough, the droplets have time to settle, regardless of the vapour velocity v_h.

The design criterion is now easy to formulate. The critical droplet is a droplet with the diameter D_p starting the horizontal path at the maximum height S_t, i.e. just below the upper part of the shell and has to touch the liquid surface at the end of the separation distance S_h, see figure 03.

In other words, during the time it takes the droplet to settle the height S_t with the velocity v_t, it must not travel longer than the horizontal distance S_h with the velocity v_h.

However, there are many combinations of shell diameters and lengths, which solve this design criterion. A fat, short separator has a large separation distance but the horizontal vapour velocity is low. When the diameter reduces, the length has to be increased in order to compensate the larger velocity with a larger flow length.

The design and properties of the horizontal separator can thus be summarized as in figure 03. This is a general design method valid for all types of refrigerants. Further:

- As long and slender shells are in general cheaper, this might be advantageous.
- Unfortunately, another effect prevents too long and slender shells. The increasing vapour velocity causes re-entrainment, i.e. liquid already separated is ripped off the surface by the vapour. The vapour velocity v_h therefore has to be below a certain limit, the re-entrainment velocity v_{re}, which depends both of the geometry and the refrigerant properties, see figure 04.

  For most LP separators, the velocity v_{re} is usually much less than what figure 04 predicts.

The re-entrainment velocity depends also on the surface tension of the liquid. Values of the surface tension can sometimes be hard to find. There is however a surprisingly easy correlation for the surface tension σ, as a function of the absolute temperature T, using only the easily available critical properties T_c and P_c and the normal boiling point temperature T_b.

$$T_{br} = T_b / T_c \qquad \text{(Eq. 1)}$$

$$Q = (0.12 * (1 + T_{br} * \ln(P_c) / (1 - T_{br})) - 0.28) / 1000 \qquad \text{(Eq. 2)}$$

$$\sigma = Q * P_c^{2/3} * Tc^{1/3} * (1 - T / T_c)^{11/9} \text{ kg/s}^2 \qquad \text{(Eq. 3)}$$

See Table 5 of values for the re-entrainment velocity v_{re} for some common refrigerants.

- The separation efficiency also depends on the liquid height in the separator as this effects both the separation height and the vapour velocity, see figure 03.Suppose that the separator is designed for maximum liquid filling, which should not be more than about 80 % of the cross section. The separation distance is then about 25 % of the diameter. The limiting droplet moves as 1) in figure 03. If the filling is reduced to half the cross section, the vapour velocity is reduced 2.5 times (the cross section changes from 20 % to 50%). The separation distance only doubles and the limiting droplet now flows like 2).

  The maximum liquid level sets the design limit if the maximum capacity occurs simultaneously, but as this can change, it has to be checked from case to case.

- In a separator with two vapour inlets, e.g. as in figure 05, (**4. Evaporators & Separators)**, the total vapour flow is split into two streams. The vapour velocity is therefore reduced to half of that in the one inlet separator. However, the separating distance is reduced to half as well, which means that the separating efficiency remains. The large advantage of splitting the vapour flow is that the reduced velocity means less re-entrainment and the shell can then be made longer and slender, i.e. normally reduced cost.
- In general can be said that the number of exits does not affect the separation efficiency, as the decreased velocity is balanced by reduced separation distances.
- The design of a multi inlet/exit separator should be based on the most loaded section of the separator. The injection points of the flashing refrigerant probably do not correspond to the vapour inlet/exits and the loads on the different sections are then unsymmetrical.
- Note that both the minimum and maximum liquid level can be lower than the separator if a vertical container as in figure 02B, (**4. Evaporators & Separators**) is added to the horizontal part. The separating distance is in this case the diameter of the horizontal part.

4. Common points.

- The minimum liquid level shall be high enough that there is no danger of vapour entering the liquid line. This is especially important for pump circulation.
- The separator volume shall be large enough in order to at least contain the LP side refrigerant.
- In a system, where the PHE is placed close to the separator, a rough rule to calculate the operating refrigerant content is to use the PHE plus the inlet pipe volume. When the PHE starts, part of the PHE volume is displaced by vapour, but on the other hand, the exit pipe now contains some liquid.
- The vertical cross section of the separator has to be tuned to the expansion valve, which is probably a float type. If this operates intermittently, i.e. opens at a low set point and closes at a high set point, the level variations shall be so designed that the valve opens & closes at a reasonable frequency. This should be checked with the valve manufacturer.

5. The droplet settling velocity.

The major problem is to find the design velocity v_d and/or the settling velocity v_t for the chosen droplet diameter. The method in figure 01 & 06 calculate the gravity force Fg of the droplet and the friction force F_f. The settling velocity v_t makes the two forces equal.

To calculate the friction force, a drag coefficient C_d is necessary. This is plotted in figure 02 as a function of the Reynold number of the droplet. Three correlations are given. Number 1 is from (1), 2 from (2) and 3 is an interpolation between the two in the commercially important flow regime 1 < Re < 30. Note that any correlation for Cd can be used, presumed that Re is in the validity range.

Note that it is impossible to calculate the mass fraction separated. In order to do this, an exact size spectrum is necessary and this is close to impossible to find.

Therefore, there is a need for a simple correlation, based on actual separators. A number of correlations have been proposed. Most of them try to correlate the settling velocity and/or the design velocity based on the formula:

$$v_t = K_s * ((\rho_l - \rho_g)/\rho_g)^{0.5} \qquad \text{(Eq. 4)}$$

The K_s value is either taken as a constant or depends on one or more variables. In table 2 there is compilation of values from different sources. As can be seen, there is a large spread between the recommended values.

6. Correlations for the settling (terminal) velocity.

A) The method in figures 01 & 02 is used. A droplet size of 0.152 mm gives a settling velocity of 0.5 m/s for ammonia at -10°C is, a value used in many commercial installations (see 14, 16, 17). This diameter is then used to recalculate to other conditions.

This is the recommended method.

A comparison for R12, R22 & R717 between -40 °C and 0 °C, gave K_s-values to be used with (eq. 4) of 0.021 and 0.075 m/s (0.068 and 0.24 ft/s). See table 5 of settling velocities for some common refrigerants.

B) Use (eq. 4) directly. Most sources gives K_s to 0.03 m/s = 0.1 ft/s, i.e. 0.5 m/s for ammonia vapour at -10 °C.

C) Combine (eq. 4) and equation 10 in figure 01. As the vapour viscosities change very little and the gravity still less, they can be included in the constant. Therefore:

K_s m/s	ft/s	Comments	Ref.
0.03	0.1	Vertical	4
0.08	0.263	Vertical	5
0.03	0.1	Vertical	6
0.069/0.01	0.227/0.034	With/without demister	7
0.04/0.08	0.13/0.26	V/H with demister	8
		Various	
$0.245 * (F-50)^{0.4}$		F= Decontamination factor	15
$f(M_l/M_g)(\rho_g/\rho_l)0.5$		= 0.02 to0.4 (British)	9
f(Decontamination factor)		= 0.014 to 0.11 (SI)	10
The same as 9, but computerized			11
Vapour velocity as a function of vertical distance			12
The source of the above (12) but better			13
Velocities between 0.5 and 1.0 m/s for ammonia			14

Tab. 2. Comparison of K_S-values in eq. (4).

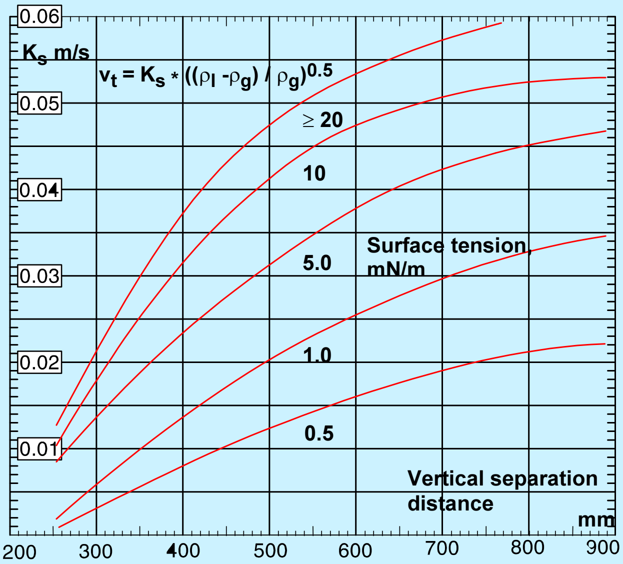


Fig. 05. The velocity constant K_S.

1	K.E. Arnold, C.T. Sikes	pp 60-64	Oil & Gas Journal	July 21, 1986
2	F.H. Wu	pp 74-80	Chemical Engineering	April 2, 1984
3	J.C. Viles	pp. 405 - 409	JPT (Society of Petroleum Engineers)	May, 1993
4	A. H. Younger	pp. 201-202	Chemical Engineering	May, 1955
5	E.R. Niemeyer	pp. 155-156	Hydrocarbon Proc. and Petroleum. Ref.	June 1961, Vol 40, No 6
6	ASHRAE	pp. 1.7 -1.9	ASHRAE Handbook. Refrigeration	1994
7	A. Gerunda	pp. 81-84	Chemical Engineering	May 4, 1981
8	B. Sigales	pp. 157-156	Chemical Engineering	March 3, 1975
9	R.N. Watkins	pp. 253-256	Hydrocarbon Proc. and Petroleum. Ref.	Nov. 1967, Vol 46, No 11
10	S.S. Alam, S.C. Agarwal	pp. 511-515	CHEMICAL AGE OF INDIA	June 1969, Vol 20, No 6
11	A.K. Coker	pp. 55-62	Oil & Gas Journal	May 10, 1993
12	D.K. Miller	pp.	Proceedings I.I.R	1971
13	G. G. Brown and ass.	pp. 346-349	Unit Operations, John Wiley & Sons Inc.	1950
14	G. Lorentzen	pp. 89-97	Kältetechnik - Klimatisierung 18 Jahrgang,	Heft 3/1966
15		pp. 11-3,36	Perry’s Chemical Engineer’s Handbook	5th Ed. 1973
16	C.F. Müller Verlag	pp. 104-107	Ammoniakverdichterkälteanlagen,	Juli, 1992
17	Maake-Eckert	pp. 287-295	Pohlmann, Tachenbuch der Kältetechnik	17 Auflage, C.F. Müller
18	HTRI	Design manual	Heat Transfer Research Inc.	1995
19	M. Souders, M., C.G. Brown	Design of fractionating columns	Ind. & Eng. Chemistry,	1934

Table 3. Papers on vapour liquid separations.

$$K_s = k' * ((\rho_l - \rho_g) * \rho_g)^{0.328} \quad \text{(Eq. 5)}$$

$k' = 0.00272$ (SI)

$k' = 0.00893$ (British)

D) Use a demister. This can increase the allowable design velocity more than six times (7). Then follow the demister manufacturers recommendations.

E) ASHRAE (6) gives two methods of calculating v_t. One is (eq. 4) with K_s = 0.1 or 0.2 ft/s. (Note that the 1994 editions uses SI units, but have forgotten to convert K_s from British units, i.e. K_s is still reported as 0.1/0.2).

The principal method refers to (12) which refers to (13) which in turn refers to (19). This is a method used to calculate the spacing in distillation columns. In a distillation column figure 07, (**1. Applications**), the vapour passes upward through the liquid with a velocity higher than the superficial velocity, i.e. probably more severe entrainment. The velocity depends also on the spacing. It probably gives a too low velocity but is then safe.

The main problem is that the spacings used by (12), 12” & 24” & 36” (12” is incorrectly reported by ASHRAE as 10”) are close to impossible to interpolate between. Thus, it is better to use the curves from (13), converted to SI units in figure 05. It is valid for vertical separators.

7. Example.

Vapour from a 1000 kW ammonia evaporator, together with the flash vapour, about 0.92 kg/s at 0 °C, shall be separated in a horizontal separator. Calculate v_t from (eq. 5).

$\rho_l = 638.6$ kg/m^3, $\rho_g = 3.457$ kg/m^3, $\eta_l = 1872 * 10^{-6}$ kg/m,s

$\sigma = 26.76$ mN/m. The volume is = 0.266 m^3/s.

$v_t = 0.00272 * ((\rho_l - \rho_g) * \rho_g)^{0.328} * ((\rho_l - \rho_g)/\rho_g)^{0.5} = 0.46$ m/s

The more stringent calculation in table 5 (A and Fig 1 & 2) gives 0.45 m/s.

Try a horizontal velocity of 3 * 0.47 = 1.41 m/s and a half full separator. The diameter is then:

$D = (8 * 0.266/\pi * 1.41)^{0.5} = 0.693$ m, chose 700 mm. The min. length S_h (S_t = 0.7/2) is 0.35*1.41/0.47= 1.05 m.

This is a too small volume; the residence time is also too small. Increase the length to 3.5 m. The residence time is now 3.5/1.41 = 2.5 sec. Small but on the other hand, the length is much longer than the calculated necessary.

Will there be a danger of re-entrainment? Figure 04 gives N = 0.000995. Use equation D. Equation D gives $V_{max.}$ = 7.70 m/s, i.e. OK. See also table 5.

8. Pressure drops in a thermosiphon loop.

See figure 06 and **4. Evaporators & Separators**, fig. 03 for schematic views of the various pressure drops. These can be separated into:

8.1. Separator - evaporator. One-phase pressure drops.

- ***The static pressure recovery***, the driving head, is given by the height difference between the liquid level in the separator and the evaporator inlet

  $\Delta p_{sl} = \rho_l * g * \Delta H_l$ (Eq. 6)

- ***Friction pressure drop.*** Formulas for this can be found in most engineering handbooks. One is:

  $m = M / (\pi * D^2 / 4);$ (Eq. 7)

  $Re_l = m * D / \eta_l = v_l * D / (\eta_l / \rho_l)$ (Eq. 8)

  $f_l = 0.0055 / Re_l^{0.2}$ (Eq. 9)

  $\Delta p_{fl} = 4 * f_l * (L / D) * m^2/(2 * \rho_l)$ (Eq. 10)

  The friction pressure drop has to be evaluated for each pipe diameter D, length L and the corresponding liquid mass flow, m or velocity, v_l. The length shall include the passage through bends, valves, fittings, etc.

- ***Local pressure drops*** in valves, fittings, bends, etc. each for its corresponding liquid mass velocity, m:

  $\Delta p_i = \Sigma k_i * m^2 / (2 * \rho_l) = \Sigma k_i * \rho_l * v_l^2 / 2$ (Eq. 11)

  k_i can be found in most engineering handbooks. Some examples are given in figure 06.

- ***Momentum change.*** Every time the velocity changes, by a change in the vapour fraction or in the cross section, there is a change in momentum, expressed as a change of the pressure.

  - At a velocity increase, the pressure decreases, i.e. a pressure drop.
  - At a velocity decrease, the pressure increases, i.e. a pressure recovery.

  In a single-phase flow when the velocity changes from state 1 to 2, the calculation is straightforward:

  $\Delta p_{ml} = (m^2 / \rho_l)_2 - (m^2 / \rho_l)_1$ (Eq. 12)

  Note that the pressure drop in e.g. a sudden contraction is of two types:

  - Loss as given by Eq. 11.
  - Momentum change as given by Eq. 12.

  Thus if a sudden contraction is followed by an equal large sudden expansion, there is a pressure loss due to the momentum change in the contraction but this is recovered in the expansion. The contraction and the expansion then have their unrecoverable losses as given by the coefficients k in figure 06 and table 5. In a closed single-phase loop, all contractions and expansions equalize each other and the net result is no change in momentum.

8.2. Evaporator. One- and two-phase pressure drops.

The calculation of the pressure drop in heat exchangers with phase changes is extremely complicated and is beyond the scope of this manual.

The stated pressure drop for an evaporator (or condenser) always includes both the static, the momentum and the friction pressure drops. This is contrary to a single-phase heat exchanger, where normally only the friction pressure drop is given.

8.3. Evaporator - separator.

The designer sails between Scylla and Charybdis at the design of the pipe from the evaporator to the separator.

If a too large pipe diameter, the flow might be unsteady and the evaporator suffers from control problems.

If too small or too many bends and valves, the total pressure drop will be too high and the circulation rate cannot be maintained. A to small exit pipe can lead to choke flow, see § 3.5 in **4. Evaporators & Separators.**

The vapour fraction has also a large influence. A large vapour fraction decreases both the various pressure drops and the minimum velocity but it also decreases the thermal performance in the evaporator. In praxis vapour, fractions from 0.7 to 0.9 are used in PHEs.

8.3.1. Two-phase flow models.

When a two-phase vapour-liquid mixture flows in a duct, a pipe, a PHE-channel a HE tube or the like, the flow can be classified in flow regimes, see figure 01. B. in **4. Evaporators & Separators**. The actual flow regime depends on the pipe diameter, vapour fraction, the relative densities and viscosities of the vapour and liquid.

The flow can also be classified as gravity controlled or shear controlled flow regimes.

- In the ***gravity*** controlled flow regime, the direction of the flow, horizontal, vertical up or down is important, as the vapour and liquid tend to separate not only by gravity forces but also by centrifugal forces.
- In the ***shear*** controlled flow regimes, the turbulence and thus the shear forces are high. The behaviour of the vapour-liquid mixture tends to be less influenced by the direction and more of the flow parameters.

An important classification, notably for design calculation, is in separated or homogenous flow.

- In ***separated flow***, the vapour and liquid move with different velocities but with equal pressure drop. The two-phase density and velocities cannot be easily calculated but have to be correlated experimentally.

  The separated model is useful for calculating pressure drops in pipes at mass flows and vapour fractions normally encountered in thermosiphon exit piping.

  Separated flow is mostly gravity controlled.

- In ***homogeneous flow***, the vapour and liquid move with the same velocity. The two-phase density and thus the velocity can easily be calculated from the mass fractions and densities. It is also called no slip flow.

The homogeneous model is suitable to correlate local pressure drops in valves, bends, fittings, etc. and pipes for very large mass velocities and high vapour fractions. Homogenous flow is normally shear controlled.

In this appendix, index h indicates homogeneous flow and tp a flow with slip, e.g. ρ_h and ρ_{tp}, the homogeneous and slip two-phase densities.

All pressure drop correlations show large differences from each other, as it is very difficult to exactly correlate a two-phase mixture.

- There is e.g. a large difference if the liquid and vapour enters a pipe well separated or intimately mixed.
- The surface tension, both liquid-liquid and liquid-wall, has a little understood effect.
- Entrance effects to a pipe play an important part.
- Correlations developed for a small pipe diameter do not necessarily scale up to a larger diameter.
- A standing sound wave can form and disturb the flow.
- Certain parameters are difficult to determine experimentally, which affects the quality of the correlations.
- Two-phase flow is correlated to single-phase flow data. There can be large deviations in these, depending on source, mechanical execution, methods, etc.
- Impurities such as oil, water, remains of detergents, etc. change the physical properties.

After a few bends, valves, fittings, changes of directions etc, the flow appearance might have changed notably and the pressure drop in the last meter of a pipe might differ from the first despite the same flow parameters.

8.3.2. Two-phase pressure drops.

Table 5 shows the physical properties, the separator velocities, and the minimum flow in a vertical pipe and the different two-phase pressure drops for some refrigerants.

- ***Static pressure drop*** in vertical upward flow. The liquid phase flows slower than the vapour phase - separated flow - and a given cross section thus contains proportionally more liquid, i.e. the slip two-phase density is higher than the homogenous density. Extensive work has been done to correlate the two-phase slip density ρ_{tp} with physical properties and operating parameters but the phenomena are still not fully understood and quantified.

  The static pressure drop is then correlated as:

  $$\Delta p_{stp} = \rho_{tp} * g * \Delta H_{tp} \qquad \text{(Eq. 13)}$$

  Pressure gain can occur in downwards flow, but is usually not taken into consideration. The reason is that in practice it can be difficult to obtain.

- ***Friction pressure drop.*** The most popular correlations correlates the two-phase pressure drop as the single-phase pressure drop, i.e. as if the vapour was flowing alone in the pipe. The vapour pressure drop, Δp_g, is then multiplied with a correction factor ϕ_g^2.

  $$\Delta p_{ftp} = \Delta p_g * \phi_g^2 \qquad \text{(Eq. 14)}$$

  As in the single-phase design, the calculation is made for each pipe section with lengths L and the corresponding diameters D.

  Note that the two-phase correction factor is surprisingly constant for a given vapour fraction. Estimate of the two-phase pressure drop by multiplying the vapour pressure drop (note, as if vapour were flowing alone) with the factors below:

  All hydrocarbons: $\phi_g^2 = 2.3$ (Eq. 15)
  Ammonia: $\phi_g^2 = 2.0$ (Eq. 16)

  Table 5B gives the multiplication factor ϕ_g^2 and the friction pressure drop for an one meter high, vertical two-phase column, a vapour fraction of 85 %, a diameter of 0.1 m and for the corresponding minimum velocity

  Note that the static pressure drop is dependent on the velocity and diameter. It can easily be recalculated to other velocities and diameters as shown in table 5.

  Horizontal and downward flow give less friction pressure drop than predicted for vertical flow.

- ***Local pressure drops in valves, fittings, bends, etc.*** Experimental correlations give varied results. The homogenous two-phase density, ρ_h, used with the single phase pressure drop coefficients, usually gives a conservative result and is used in table 5B:

  $$\rho_h = 1 / ((1 - y) / \rho_l + y / \rho_g) \qquad \text{(Eq. 17)}$$

  $$\Delta p_{hi} = \Sigma k_i * \rho_h * v_h^2 / 2 = \Sigma k_i * m^2 / (2 * \rho_h) \qquad \text{(Eq. 18)}$$

- ***Momentum change*** in a two-phase flow is complicated, as the two-phase density has to be correlated.

  In general, the homogenous density correlates data in sudden changes better than the separated flow density and is probably conservative.

  There is a large change in momentum from the inlet of the evaporator to the exit as the fluid changes from liquid alone to two-phase with a large increase in volume. This change in momentum is included in the pressure drop for the evaporator and is not dealt with here.

  In the circulating loop, there is normally no change in the vapour fraction and change in momentum thus occurs only in contractions and expansions.

  $$\Delta p_{mtp} = (m^2 / \rho_h)_2 - (m^2 / \rho_h)_1 \qquad \text{(Eq. 19)}$$

  There is thus a large recovery when the vapour liquid mixture enters the separator (and a loss due to the k). However, this pressure recovery is normally not taken into account. There might be some reasons for this. The vapour probably turns a couple of times in the separator and this is normally not accounted for. There might also be guiding vanes, baffles, etc.

8.4. Pressure drop correlations.

Note! Be careful when using pressure drop correlations from different sources. The resistance coefficient for flow in pipes, as defined in Europe, is four time as large as the Fanning friction factor used in America and here (thus the factor 4 * f in. eq. 10).

Likewise, check the cross section for a localized pressure drop (entrance or exit cross section of the connection, minimum cross section of the flow passage or some other definition). It is important how this is defined.

8.5. Limitations & Restrictions.

See also **Evaporators and Separators** § 3.4 pp. 76 - 77.

8.5.1. The minimum two-phase flow.

The minimum mass velocity m_{min} and the minimum homogenous two-phase velocity v_{hmin} for annular vertical upwards flow are found in table 5B.

Churn flow has to be accepted sometimes, but unsteady behaviour might be encountered. The lower limit of slug flow is about 30 to 50 % of the churn flow limit.

Avoid flow regimes below churn flow - slug and wave - because of the instabilities, which can occur here.

8.5.2. Back flow.

If the flow resistance is too large at the evaporator exit, the vapour might simply not leave sufficient rapidly and the liquid is pushed backwards into the inlet leg.

Few reliable investigations exist on the allowable exit pressure drop, but a limit of about 25 % of the total driving head is recommended. In a PHE, this should include the exit port pressure drop. This is generally found in the specification sheet for the evaporator. If not, the following approximate formula can be used. It is based on the homogeneous port velocity v_h:

$$\Delta p_{hp} = 1.5 * \rho_h * v_h^2 / 2 = 1.5 * m^2 / (2 * \rho_h) \qquad \text{(Eq. 20)}$$

Note! Never put a control valve between the evaporator exit and the separator. If a valve is necessary, it should be placed before the evaporator.

8.5.3. Increase of the evaporation temperature.

The temperature in the evaporator will be higher than the separator temperature, corresponding to the pressure drop between the separator and the evaporator. Table 5A gives the value of the saturation pressure slope, which can be used to estimate the temperature increase.

Ex. Ammonia has a vapour pressure slope of 6.23 K/bar at 0 °C. A pressure drop in the exit piping of 3.32 kPa gives an evaporation temperature about 0.0332 * 6.23 = 0.21 K higher than the separator temperature.

The problem is mainly acute for large level differences at low pressures but it can seriously decrease the MTD. There is no rule how much can be allowed but the effect on the evaporator has to be taken into account.

8.5.4. Pressure drop margin.

The driving head should be in the order of 20 to 50 % larger than the total pressure drop.

8.5.5. Maldistribution in the evaporator.

A flooded flow evaporator can tolerate a higher degree of maldistribution over the plate pack length than a DX-evaporator. Maldistribution is the unequal channel flow rate from the first to the last channel due to a too high port pressure drop.

The subject and limitations are treated further in the design manuals for evaporators.

8.6. Pressure drop balancing.

The thermosiphon loop is in equilibrium, when the driving head is equal to the sum of various pressure drops, thus:

♦ Driving head:	+	Eq. 6.
♦ Inlet leg (liquid):	-	Eqs. 10, 11 & 12.
♦ Evaporator:	-	From the specification.
♦ Exit leg:	-	Eqs.13-19 & table 5.

under the restrictions below:

♦ Limitations:	§8.4 (Table 5B).

By definition, this balance is always valid for a thermosiphon loop in operation, but what happens if there is a difference between the nominal driving force and the sum of the nominal pressure drops?

8.6.1. Too large driving force.

- In an operating system, the circulation increases, i.e. more liquid enters the evaporator and both the single and two-phase pressure drops increase until the system gets in balance. The heat transfer in the evaporator might also improve and the increased pressure from the higher vapour flow helps balance the system.
- The evaporator could also be made smaller by using more of the available driving force.
- The separator could be placed lower.
- The extra driving force is taken up by a control valve before the evaporator.
- The size of the pipe work could be reduced.

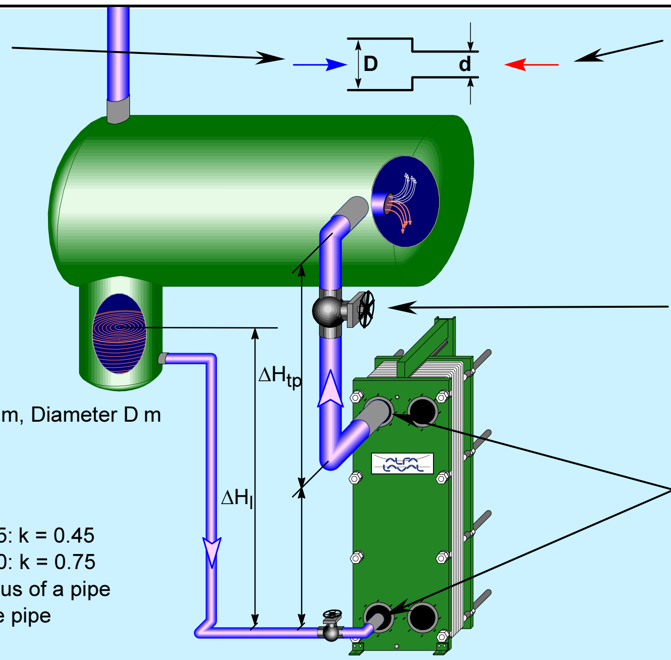
8.6.2. Too small driving force.

- The circulation decreases and thus all the pressure drops decrease. The heat transfer in the evaporator might impair and the decreased vapour flow helps to get the system in balance. Note that this means a reduced capacity of the evaporator.
- The designer might also opt to increase the size of the pipes, change the valves, etc.
- Increase the driving head, i.e. increase the height of the separator. Note that in low pressure operation this might impair the evaporation, see §3.4, p. 76.
- The evaporator will larger by using less of the available driving force, but this is another method to balance the pressure drops. There is an optimum between reduced pipe work and larger evaporator.

9. Pump circulation.

The above described design methods could be used by forced circulation as well. The driving head is now mainly the pump head. Excessive head is released in a valve between the pump and the evaporator.

Note that to avoid cavitations, a pump is usually placed appreciably lower than the separator and the heat exchanger, which increases the pressure and thus the liquid subcooling at the pump. For the same reasons, there should not be any larger local pressure drops, e.g. a valve, before the pump.



Sudden reduction

d/D =	k =
0.0	0.5
0.2	0.48
0.4	0.42
0.6	0.32
0.8	0.18
1.0	0.00

Velocity based on d

Pipe section: Length L m, Diameter D m

Long bend R / D = 1.5: k = 0.45
Short bend R / D = 1.0: k = 0.75
R: Centre curvature radius of a pipe
D: Inside diameter of the pipe

Sudden expansion:

d/D =	k =
0.0	1.00
0.2	0.92
0.4	0.71
0.6	0.41
0.8	0.13
1.0	0.00

Velocity based on d

Open angle valve: k = 3
Open ball valve: k = 0
Open gate valve: k = 0.2
Open globe valve: k = 6 -9

The PHE pressure drop, ΔP_{PHE}, is defined between these two points.

Fig. 06. The pressure drops in a thermosiphon circuit.

Item	Unit	Value	Pressure drop or driving head calculation for the item	Net
Total mass flow, M:	kg/s	0.9556	(Note the difference to the separator calculation)	kPa
Vapour fraction, y:		0.80		
Inlet: Driving head,	m	2.5	Δp_{sl} = 638 * 9.8054 * 2.5 = 15639 Pa =	**+15.64**
Diameter, D:	m		D = 0.05	
Velocity	m/s		v_l = 0.7947	
1) Velocity head	Kpa		(Δp_{vhl} = 638 * 0.7947^2 / 2 = 201 Pa = 0.201 kPa)	
1) Reynolds No.			(Re_l = 0.795 * 0.05 / (0.000187 / 638 =135566)	
1) Pipe length, L:	m	2.5	Δp_{fl} = 4 * (0.055 / 135566$^{0.2}$) * (2.5 / 0.05) * 0.201 =	***-0.21***
1) # short bends:		4	Δp_{bl} = 4 * 0.75 * 0.201 =	***-0.30***
1) # globe valves :		1	Δp_{vl} = 1 * 6 * 0.201 =	***-1.21***
1) # sudden red.;		1	Δp_{rl} = 1 * 0.45 * 0.201 =	***-0.09***
1) # sudden exp.:		1	Δp_{el} = 1 * 1.1 * 0.201 =	***-0.22***
Exit: Diameter, D	m	0.15		
Velocity	m/s	13.6	v_h	
2) Two-phase head,	m	1.96	Δp_{stp} = 1.96 * 0.454 =	***-0.89***
2) Pipe length, L:	m	4	Δp_{ftp} = 4 * 0.0162 * (13.6/4.88)$^{1.8}$/(0.1/0.15)$^{1.2}$ =	***-0.25***
2) # long bends:		3	Δp_{btp} = 3 * 0.0232 * (13.6/4.88)^2 =	***-0.54***
2) # angle valves:		1	Δp_{vtp} = 1 * 0.1544 * (13.6/4.88)^2 =	***-1.20***
2) # sudden reduction:		0		
2) # sudden expansion:		1	Δp_{etp} = 1 * 0.0566 * (13.6/4.88)^2 =	***-0.44***
Head in the MK15BW:	m	1.04	ΔP_{PHE}= 6.61 (channel) + 0.69 (port)	***-7.30***
Net driving head:				**+2.99**
2) Min. vel. required	m/s	5.98	(v_{hmin} = 4.88 * (0.15 / 0.1)$^{0.5}$ = 5.98)	**OK**
Back flow: (0.89 + 0.25 + 0.54 + 1.2 + 0.44 + 0.69) = 4.01 kPa which is 25.6 % of the total head				**OK**
Temperature increase in the evaporator: 4.01 * 6.23 /100 = 0.25 K. Check with the evaporator.				**OK**
Head margin: The driving head, 15.64 kPa is 24 % larger than the total Δp = 15.64 - 2.99 = 12.65 kPa				**OK**

1) Calculated by the use of the equations 6 to 11.

- The separator is connected to an MK15BW-35LG/36LW, 32.2 m^2, a semiwelded unit.
- If the driving force becomes too small, consider the use of ball valves instead of angular and glove valves.

2) Calculated from table 5B.

- The driving force can be increased by lowering the evaporator. One meter lower evaporator means a higher head by 6.25 kPa but only an increase of the static and friction pressure drop by 0.46 & 0.06 kPa. The evaporation temperature increases, though.

Table 4. Example 2. Data from Ex. 1.

Refrigerant & general data	Temperature °C	Pressure bar 1)	Slope °K/m liq. column 1)	Latent Heat kJ/kg 1)	Densities kg/m^3 1) Liquid ρ_l	1) Vapour ρ_v	2, 3) Hom. $\rho_{Hom.}$	Viscosities kg/s,m*1000 (cP) 1) Liquid η_l	1) Vapour η_v	Separator velocities m/s Settling 152 µ drops v_t	Reentrainment v_{Re}
NH3	-40	0.717	1.76	1389	690.2	0.6438	0.8046	0.2812	0.00786	0.718	21.2
	-30	1.194	1.14	1360	677.8	1.037	1.296	0.2441	0.00815	0.633	16.5
Mw = 17.03	-20	1.901	0.77	1329	665.1	1.603	2.003	0.2144	0.00845	0.559	13.1
Tc = 132.3 °C	-10	2.907	0.54	1297	652.1	2.391	2.986	0.1902	0.00875	0.493	10.5
Pc = 113.3 bar	0	4.294	0.39	1262	638.6	3.457	4.315	0.1701	0.00906	0.435	8.53
NBP = -33.3 °C	10	6.150	0.29	1226	624.6	4.868	6.073	0.1530	0.00936	0.385	6.99
	20	8.575	0.22	1186	610.2	6.703	8.356	0.1383	0.00968	0.341	5.76
R134a	-30	0.844	3.40	219.5	1388	4.426	5.528	0.4010	0.00964	0.693	6.01
	-20	1.327	2.32	212.9	1358	6.784	8.469	0.3476	0.01000	0.589	4.77
Mw = 102.03	-10	2.006	1.64	206.0	1327	10.04	12.53	0.3035	0.01036	0.502	3.83
Tc = 101.2 °C	0	2.928	1.20	198.6	1295	14.43	17.99	0.2665	0.01073	0.431	3.10
Pc = 40.67 bar	10	4.146	0.89	190.7	1261	20.23	25.19	0.2349	0.01110	0.371	2.52
NBP = -26.7 °C	20	5.717	0.68	182.3	1225	27.78	34.53	0.2074	0.01149	0.320	2.07
R23	-90	0.624	3.63	245.8	1476	2.962	3.701	0.3490	0.00903	0.838	8.31
	-80	1.137	2.20	237.7	1438	5.195	6.488	0.2953	0.00958	0.677	6.13
Mw = 70.01	-70	1.937	1.41	229.2	1399	8.582	10.71	0.2530	0.01012	0.553	4.64
Tc = 25.95 °C	-60	3.119	0.95	220.0	1358	13.50	16.83	0.2187	0.01067	0.456	3.57
Pc = 49.0 bar	-50	4.789	0.66	210.0	1315	20.43	25.44	0.1903	0.01122	0.379	2.76
NBP = -82.03 °C	-40	7.065	0.48	199.0	1270	29.97	37.24	0.1660	0.01180	0.316	2.16
	-30	10.070	0.35	186.8	1221	42.95	53.22	0.1449	0.01242	0.264	1.68
R404a	-50	0.810	3.17	202.7	1325	4.442	5.548	0.3881	0.01001	0.660	5.59
	-40	1.310	2.12	196.2	1293	6.985	8.719	0.3265	0.01050	0.554	4.42
Mw = 97.6	-30	2.022	1.48	189.4	1259	10.55	13.16	0.2778	0.01100	0.467	3.54
Tc = 73.0 °C	-20	3.002	1.06	182.1	1225	15.41	19.20	0.2386	0.01152	0.397	2.86
Pc = 37.8 bar	-10	4.308	0.78	174.3	1189	21.90	27.25	0.2066	0.01205	0.338	2.32
NBP = -47.0 °C	0	6.003	0.59	165.7	1151	30.47	37.84	0.1800	0.01262	0.289	1.88
	10	8.158	0.45	156.4	1111	41.66	51.59	0.1574	0.01322	0.247	1.52
R507	-50	0.864	3.02	199.1	1327	4.808	6.005	0.3818	0.01004	0.644	5.32
	-40	1.387	2.03	192.5	1294	7.509	9.373	0.3213	0.01053	0.541	4.22
Mw = 98.9	-30	2.128	1.42	185.8	1261	11.28	14.07	0.2734	0.01104	0.457	3.39
Tc = 71.0 °C	-20	3.144	1.03	178.4	1227	16.41	20.44	0.2349	0.01156	0.388	2.74
Pc = 37.9 bar	-10	4.493	0.76	170.6	1191	23.25	28.92	0.2034	0.01211	0.331	2.23
NBP = -47.0 °C	0	6.240	0.57	162.1	1153	32.25	40.03	0.1772	0.01270	0.283	1.81
	10	8.453	0.44	152.6	1112	44.03	54.50	0.1550	0.01333	0.242	1.46
R290	-50	0.705	1.72	434.0	590.5	1.725	2.155	0.2172	0.00611	0.596	8.61
	-40	1.110	1.18	423.6	579.0	2.632	3.286	0.1935	0.00636	0.515	6.85
Mw = 44.10	-30	1.677	0.84	412.6	567.2	3.867	4.826	0.1731	0.00662	0.446	5.53
Tc = 96.7 °C	-20	2.445	0.62	400.8	554.9	5.508	6.868	0.1552	0.00689	0.389	4.51
Pc = 42.48 bar	-10	3.453	0.46	388.3	542.1	7.639	9.515	0.1396	0.00716	0.340	3.72
NBP = -42.09 °C	0	4.746	0.36	374.7	528.7	10.36	12.89	0.1257	0.00745	0.298	3.08
	10	6.368	0.28	360.0	514.7	13.80	17.14	0.1133	0.00775	0.261	2.55
	20	8.368	0.22	343.9	499.9	18.11	22.43	0.1021	0.00809	0.229	2.12
R1270	-50	0.911	0.71	441.7	611.9	2.136	2.668	0.2011	0.00629	0.570	7.98
	-40	1.419	0.98	430.5	599.4	3.225	4.026	0.1799	0.00657	0.491	6.36
Mw = 42.08	-30	2.123	0.70	418.7	586.5	4.699	5.862	0.1617	0.00686	0.425	5.14
Tc = 92.4 °C	-20	3.066	0.52	406.1	573.1	6.642	8.279	0.1459	0.00717	0.369	4.19
Pc = 46.65 bar	-10	4.295	0.39	392.5	559.2	9.154	11.40	0.1321	0.00749	0.322	3.45
NBP= -47.7 °C	0	5.859	0.31	377.8	544.6	12.35	15.35	0.1199	0.00784	0.281	2.85
	10	7.809	0.24	361.8	529.3	16.39	20.33	0.1090	0.00823	0.246	2.35
	20	10.200	0.19	344.3	513.0	21.44	26.52	0.0991	0.00867	0.214	1.94
R600a	-20	0.722	1.94	373.4	603.2	2.061	2.574	0.2526	0.00638	0.566	7.41
Mw = 58.12	-10	1.081	1.39	364.4	592.0	3.004	3.750	0.2233	0.00662	0.495	6.06
Tc = 134.7 °C	0	1.566	1.02	355.0	580.5	4.250	5.303	0.1984	0.00686	0.435	5.00
Pc = 36.31 bar	10	2.203	0.77	345.1	568.6	5.862	7.309	0.1770	0.00711	0.383	4.16
NBP = -11.61 bar	20	3.020	0.60	334.6	556.2	7.910	9.852	0.1585	0.00736	0.339	3.49
CO2	-50	6.823	0.41	339.7	1155	17.92	22.31	0.2293	0.01131	0.364	3.11
Mw = 44.0	-40	10.040	0.30	322.4	1116	26.12	32.46	0.1938	0.01187	0.306	2.46
Tc = 31.06 °C	-30	14.280	0.22	303.5	1076	37.10	45.98	0.1642	0.01246	0.257	1.95
Pc = 73.84 bar	-20	19.700	0.17	282.4	1032	51.70	63.83	0.1393	0.01312	0.217	1.53
NBP= -78.4 °C	-10	26.490	0.13	258.6	982.9	71.18	87.39	0.1180	0.01386	0.181	1.19
	0	34.81	0.10	230.9	928.1	97.32	118.5	0.1054	0.01431	0.151	0.878

1) Reported properties show sizable variations depending on source, especially for refrigerants with glide. The data here are mainly based on NIST.

2) y = 0.80.

3) See eq.17

4) y = 0.80, D = 0.1 m

5) Independent of D & v_h

Table 5A. Physical properties, the settling velocity for 152 µ droplets and the reentrainment velocity in a horizontal separator for some refrigerants.

Refrig-erant	**Tem-pera-ture**	**Minimum flow**		**Two-phase ΔP**			**Local ΔP (homogenous)**						
		Mass flow 4), 6) M_{min}	Linear velocity 4), 7) v_{hmin}	Static, 2), 5), 11) ΔP_{stp}	Vapour multi-plier 2), 5) ϕv^2	Friction 4), 9), 10) ΔP_{ftp}	Velocity height k = 1 2), 8), 9) Δp_{vhh}	Angle valve k = 3.0 ← Δp_{avh}	Gate valve k = 0.2 ← Δp_{gvh}	Long 90° bend k = 0.45 ← Δp_{lbh}	Short 90° bend k = 0.75 ← Δp_{sbh}	Sudden reduc. k=0.45 ← Δp_{srh}	Sudden expan. k = 1.1 ← Δp_{seh}
	°C	kg/hr	m/s	kPa	←	←	←	←	←	←	←	←	←
NH3	-40	170	7.48	0.070	1.80	0.0079	0.0225	0.0674	0.0045	0.0101	0.0169	0.0101	0.0247
	-30	220	5.99	0.125	1.86	0.0081	0.0233	0.0698	0.0047	0.0105	0.0175	0.0105	0.0256
	-20	280	4.95	0.210	1.92	0.0085	0.0245	0.0736	0.0049	0.0110	0.0184	0.0110	0.0270
	-10	355	4.21	0.321	1.97	0.0091	0.0264	0.0792	0.0053	0.0119	0.0198	0.0119	0.0290
	0	447	3.66	0.454	2.03	0.0100	0.0289	0.0868	0.0058	0.0130	0.0217	0.0130	0.0318
	10	560	3.26	0.606	2.08	0.0111	0.0323	0.0969	0.0065	0.0145	0.0242	0.0145	0.0355
	20	700	2.96	0.772	2.13	0.0126	0.0367	0.1100	0.0073	0.0165	0.0275	0.0165	0.0403
R134a	-30	693	4.43	0.600	2.12	0.0187	0.0543	0.1630	0.0109	0.0245	0.0408	0.0245	0.0598
	-20	901	3.76	0.908	2.19	0.0207	0.0600	0.1800	0.0120	0.0270	0.0450	0.0270	0.0660
	-10	1166	3.29	1.268	2.26	0.0235	0.0678	0.2035	0.0136	0.0305	0.0509	0.0305	0.0746
	0	1501	2.95	1.667	2.32	0.0271	0.0783	0.2350	0.0157	0.0353	0.0588	0.0353	0.0862
	10	1925	2.70	2.092	2.37	0.0299	0.0920	0.2761	0.0184	0.0414	0.0690	0.0414	0.1013
	20	2461	2.52	2.530	2.42	0.0339	0.1097	0.3291	0.0219	0.0494	0.0823	0.0494	0.1207
R23	-90	559	5.34	0.381	1.98	0.0173	0.0527	0.1582	0.0105	0.0237	0.0396	0.0237	0.0580
	-80	776	4.23	0.703	2.07	0.0191	0.0580	0.1740	0.0116	0.0261	0.0435	0.0261	0.0638
	-70	1064	3.51	1.118	2.15	0.0219	0.0661	0.1984	0.0132	0.0298	0.0496	0.0298	0.0728
	-60	1448	3.04	1.602	2.22	0.0253	0.0779	0.2338	0.0156	0.0351	0.0584	0.0351	0.0857
	-50	1959	2.72	2.130	2.28	0.0290	0.0943	0.2830	0.0189	0.0424	0.0707	0.0424	0.1038
	-40	2635	2.50	2.680	2.34	0.0339	0.1166	0.3497	0.0233	0.0525	0.0874	0.0525	0.1282
	-30	3532	2.35	3.235	2.38	0.0405	0.1466	0.4398	0.0293	0.0660	0.1100	0.0660	0.1613
R404A	-50	682	4.35	0.603	2.12	0.0181	0.0525	0.1574	0.0105	0.0236	0.0394	0.0236	0.0577
	-40	904	3.67	0.926	2.18	0.0202	0.0586	0.1757	0.0117	0.0264	0.0439	0.0264	0.0644
	-30	1188	3.19	1.304	2.24	0.0231	0.0671	0.2013	0.0134	0.0302	0.0503	0.0302	0.0738
	-20	1554	2.86	1.718	2.29	0.0265	0.0786	0.2359	0.0157	0.0354	0.0590	0.0354	0.0865
	-10	2022	2.62	2.155	2.34	0.0298	0.0939	0.2816	0.0188	0.0422	0.0704	0.0422	0.1033
	-0	2624	2.45	2.602	2.38	0.0342	0.1138	0.3414	0.0228	0.0512	0.0854	0.0512	0.1252
	10	3394	2.33	3.048	2.41	0.0399	0.1397	0.4190	0.0279	0.0629	0.1048	0.0629	0.1536
R507A	-50	717	4.22	0.653	2.13	0.0185	0.0536	0.1607	0.0107	0.0241	0.0402	0.0241	0.0589
	-40	948	3.58	0.988	2.20	0.0207	0.0600	0.1801	0.0120	0.0270	0.0450	0.0270	0.0660
	-30	1246	3.13	1.375	2.25	0.0238	0.0691	0.2072	0.0138	0.0311	0.0518	0.0311	0.0760
	-20	1630	2.82	1.797	2.30	0.0271	0.0813	0.2438	0.0163	0.0366	0.0609	0.0366	0.0894
	-10	2121	2.59	2.239	2.35	0.0306	0.0973	0.2919	0.0195	0.0438	0.0730	0.0438	0.1070
	0	2751	2.43	2.690	2.38	0.0353	0.1182	0.3547	0.0236	0.0532	0.0887	0.0532	0.1301
	10	3560	2.31	3.140	2.41	0.0412	0.1454	0.4363	0.0291	0.0655	0.1091	0.0655	0.1600
R290	-50	279	4.59	0.232	2.05	0.0080	0.0227	0.0680	0.0045	0.0102	0.0170	0.0102	0.0249
	-40	361	3.89	0.354	2.13	0.0088	0.0249	0.0746	0.0050	0.0112	0.0186	0.0112	0.0273
	-30	464	3.40	0.496	2.19	0.0099	0.0279	0.0836	0.0056	0.0125	0.0209	0.0125	0.0306
	-20	591	3.04	0.655	2.25	0.0113	0.0318	0.0955	0.0064	0.0143	0.0239	0.0143	0.0350
	-10	749	2.78	0.823	2.30	0.0131	0.0369	0.1107	0.0074	0.0166	0.0277	0.0166	0.0406
	0	945	2.59	0.997	2.35	0.0149	0.0433	0.1299	0.0087	0.0195	0.0325	0.0195	0.0476
	10	1186	2.45	1.172	2.39	0.0169	0.0513	0.1540	0.0103	0.0231	0.0385	0.0231	0.0565
	20	1483	2.34	1.347	2.42	0.0192	0.0613	0.1839	0.0123	0.0276	0.0460	0.0276	0.0674
R1270	-50	323	4.28	0.289	2.07	0.0086	0.0245	0.0734	0.0049	0.0110	0.0184	0.0110	0.0269
	-40	418	3.67	0.427	2.14	0.0095	0.0271	0.0813	0.0054	0.0122	0.0203	0.0122	0.0298
	-30	536	3.24	0.585	2.20	0.0108	0.0307	0.0920	0.0061	0.0138	0.0230	0.0138	0.0337
	-20	684	2.92	0.758	2.26	0.0124	0.0353	0.1060	0.0071	0.0159	0.0265	0.0159	0.0389
	-10	867	2.69	0.938	2.31	0.0144	0.0413	0.1238	0.0083	0.0186	0.0310	0.0186	0.0454
	0	1094	2.52	1.123	2.36	0.0163	0.0488	0.1463	0.0098	0.0219	0.0366	0.0219	0.0536
	10	1375	2.39	1.308	2.39	0.0185	0.0581	0.1744	0.0116	0.0262	0.0436	0.0262	0.0640
	20	1721	2.29	1.490	2.42	0.0213	0.0698	0.2095	0.0140	0.0314	0.0524	0.0314	0.0768
R600a	-20	314	4.32	0.280	2.13	0.0087	0.0240	0.0720	0.0048	0.0108	0.0180	0.0108	0.0264
	-10	397	3.74	0.401	2.19	0.0095	0.0263	0.0788	0.0053	0.0118	0.0197	0.0118	0.0289
	0	498	3.32	0.540	2.25	0.0106	0.0293	0.0879	0.0059	0.0132	0.0220	0.0132	0.0322
	10	623	3.01	0.691	2.30	0.0120	0.0332	0.0995	0.0066	0.0149	0.0249	0.0149	0.0365
	20	774	2.78	0.851	2.34	0.0137	0.0380	0.1141	0.0076	0.0171	0.0285	0.0171	0.0418
CO2	-50	1718	2.72	1.873	2.35	0.0275	0.0828	0.2483	0.0166	0.0372	0.0621	0.0372	0.0910
	-40	2300	2.51	2.347	2.39	0.0317	0.1019	0.3058	0.0204	0.0459	0.0764	0.0459	0.1121
	-30	3060	2.35	2.823	2.42	0.0373	0.1274	0.3822	0.0255	0.0573	0.0955	0.0573	0.1401
	-20	4055	2.25	3.288	2.44	0.0444	0.1611	0.4834	0.0322	0.0725	0.1209	0.0725	0.1773
	-10	5368	2.17	3.735	2.43	0.0536	0.2062	0.6187	0.0412	0.0928	0.1547	0.0928	0.2268
	0	7112	2.12	4.156	2.44	0.0656	0.2669	0.8007	0.0534	0.1201	0.2002	0.1201	0.2936

6) Other D: $M_h = M_{hmin} * (D/0.1)^{2.5}$

7) Other D: $v_h = v_{hmin} * (D/0.1)^{0.5}$

8) Other v_h: $\Delta p = \Delta p * (v_h/v_{hmin})^2$

9) For the minimum flow v_{hmin}.

10) 1.0 m pipe. Other D & v_h: $\Delta p = \Delta p_{ftp} * (v_h/v_{hmin})^{1.8}/(0.1/D)^{1.2}$

11) 1 m vertical pipe.

Table 5B. A. Minimum flows in a vertical pipe, two-phase static, friction and local pressure drops.

Name	Formula	Synonym	NBP °C	T_{26bar} °C	T_{crit} °C	P_{crit} bar	Mw	ODP	GWP
R11	$CFCl_3$	Trichlorofluoromethane	23.76	162.7	198.05	44.67	137.37	1	4000
R12	CF_2Cl_2	Dichlorodifluoromethane	-29.76	86.4	111.80	41.80	120.91	1	8500
R12b1	$CBrClF_2$	Bromochlorodifluoromethane	-3.54	126.3	154.60	41.24	165.35	3	
R13	CF_3Cl	Chlorotrifluoromethane	-81.48	11.3	28.86	38.70	104.46	1	11700
R13b1	CF_3Br	Bromotrifluoromethane	-57.83	46.3	67.05	40.17	148.91	10	5600
R14	CF_4	Tetrafluoromethane	-127.98	-57.8	-45.65	37.95	88.00	0	5500
R21	$CHFCl_2$	Dichlorofluoromethane	8.85	137.6	178.5	51.70	102.93	0.04	
R22	CHF_2Cl	Chlorodifluoromethane	-40.86	63.2	96.15	50.54	86.47	0.05	1700
R23	CHF_3	Trifluoromethane	-82.03	1.2	25.95	49.0	70.01	0	12100
R30	CH_2Cl_2	Methylenchloride	39.64	174.7	245	60.9	84.93	0	9
R31	CH_2ClF	Chlorofluoromethane	-9.1	111.7	151.7	51.3	68.48	0.02	
R32	CH_2F_2	Difluoromethane	-51.66	42.0	78.2	57.95	52.02	0	580
R40	CH_3Cl	Methylchloride	-23.38	89.2	143.1	66.8	50.49	0	
R41	CH_3F	Fluoromethane	-78.12	9.1	44.13	58.97	34.03	0	
R50	CH_4	Methane	-161.5	-101.9	-82.5	46.4	16.04	0	24.5
R113	$CFCl_2$-$CClF_2$	1,1,2-Trichlorotrifluoroethane	47.65	196.1	214.35	34.56	187.38	0.8	5000
R114	$CCl\ F_2$-$CClF_2$	1,2-Dichloro-1,1,2,2-tetrafluoroethane	3.59	132.5	145.65	32.48	170.92	1	9300
R114a	CCl_2F-CF_3	1,1-Dichloro-1,2,2,2-tetrafluoroethane	3.0	131.5	145.50	33.03	170.92	1	9300
R114B2	$CBrF_2$-$CBrF_2$	1,2-Dibromo-1,1,2,2-tetrafluoroethane	47.1	197.0	214.5	33.72	259.83	>0	
R115	$CCl\ F_2$-CF_3	Chloropentafluoroethane	-38.94	70.6	79.9	31.53	154.47	0.6	9300
R116	$CF_3\ CF_3$	Hexafluoroethane	-78.20	13.6	19.65	29.79	138.02	0	9200
R123	$CHCl_2$-CF_3	1,1-Dichloro-2,2,2-Trifluoroethane	27.83	162.6	183.68	36.63	152.93	0.02	93
R123a	$CCl\ F_2$-$CHClF$	1,2-Dichloro-1,1,2-Trifluoroethane	30.01	166.3	187.95	37.41	152.93	>0	
R124	$CHFCl$-CF_3	1-Chloro-1,2,2,2-Tetrafluoroethane	-11.94	104.6	122.47	36.37	136.48	0.03	480
R125	C_2HF_5	Pentafluoroethane	-48.14	51.1	66.18	36.29	120.01	0	3200
R132a	$CHCl_2$-CHF_2	Dichlorodifluoroethane					134.91		
R132b	CH_2Cl-$CClF_2$	Dichlorodifluoroethane(Est. values)	93.0	111.7	142.4	72.3	134.91	0.005-0.008	
R133a	CH_2Cl-CF_3	1-Chloro-3,3,3-Trifluoroethane	6.1	127.3	152.5	40.58	118.49	>0	
R134	CHF_2-CHF_2	1,1,2,2-Tetrafluoroethane	-19.81	89.4	118.95	45.62	102.03	0	
R134a	CF_3-CH_2F	1,1,1,2-Tetrafluoroethane	-26.07	79.4	101.15	40.67	102.03	0	1300
R141b	$CFCl_2$-CH_3	1-Fluoro-1,1-Dichloroethane	32.15	172.3	204.15	41.20	116.94	0.11	1800
R142b	CF_2Cl-CH_3	1-Chloro-1,1-Difluoroethane	-9.14	111.3	137.15	41.20	100.49	0.065	2000
R143	CF_2H-CFH_2	1,1,2-Trifluoroethane	4.01	118.8	156.75	45.2	84.04	0	
R143a	CF_3-CH_3	1,1,1-Trifluoroethane	-47.23	55.4	73.60	38.32	84.04	0	3800
R152a	CHF_2-CH_3	1,1-Difluoroethane	-24.02	85.0	113.26	45.17	66.05	0	140
R160	CH_3-CH_2Cl	Ethylchloride	14.0	141.6	187.0	50.8	64.52	0	
R216	$CClF_2$-CF_2-CF_2Cl	1,1,2,2,3,3-Hexafluoro-1,3-Dichloropropane	35.7	176.7	179.99	27.54	220.03	>0	
R218	C_3F_8	Perfluoropropane	-36.75	71.6	71.95	26.81	188.03	0	7000
R290	C_3H_8	Propane	-42.09	70.3	96.70	42.48	44.10	0	3
R227ea	CF_3-CHF-CF_3	1,1,1,2,3,3,3-Heptafluoropropane	-15.61	96.1	102.80	29.80	170.04	0	3300
R236ea	CF_3-CHF-CHF_2	1,1,1,2,3,3-Hexafluoropropane	6.19	125.1	139.29	35.33	152.05	0	
R236fa	CF_3-CH_2-CF_3	1,1,1,3,3,3-Hexafluoropropane	-1.42	114.3	124.92	32.0	152.04	0	8000
R245ca	CHF_2-CF_2-CH_2F	1,1,2,2,3-Pentafluoropropane	25.22	150.9	174.42	39.25	134.05	0	560
R245cb	CF_3-CF_2-CH_3	1,1,1,2,2-Pentafluoropropane	-17.59	97.1	106.93	31.37	134.06	0	
R245fa	CF_3-CH_2-CHF_2	1,1,1,3,3-Pentafluoropropane	14.90	135.7	154.05	36.40	134.06	0	
R600	C_4H_{10}	N-Butane	-0.54	129.3	152.01	37.96	58.12	0	<10
R600a	$(CH_3)_2$-CH-CH_3	Isobutane	-11.61	115.0	134.70	36.31	58.12	0	3
R717	NH_3	Ammonia	-33.33	59.8	132.25	113.33	17.03	0	0
R744	CO_2	Carbondioxide (Subli.)	-57.00	-10.6	30.89	73.80	44.01	0	1
R1270	CH_3-CH-CH_2	Propene(Propylene)	-47.70	61.3	92.40	46.65	42.08	0	3
RC270	C_3H_6	Cyclopropane	-32.90	81.6	125.15	55.80	42.08	0	3
RC318	C_4F_8	Perfluorocyclobutane	-5.99	111.9	115.22	27.78	200.04	0	8700
RE134	CHF_2-O-CHF_2	Bis(Difluoromethyl)Ether	6.17	124.1	147.10	42.28	118.03	0	
RE245	CF_3-CH_2-O-CHF_2	2-Difluoromethoxy-1,1,1-Trifluoroethane	29.20	156.9	170.88	34.20	150.05	0	
RE170	CH_3-O-CH_3	Dimethylether	-24.84	89.3	126.95	53.70	46.07	0	
R400	R12/R114	60/40 % Zeotrop, Glide 6.4 K	-23.20	101.3	125.4	39.9	136.9	>0	
R401a	R22/R152a/R124	53/13/34 % Zeotrop, Glide 6.4 K	-33.05	75.9	107.8	46.06	94.4	0.03	1080
R401b	R22/R152a/R124	61/11/28 % Zeotrop, Glide 6.0 K	-35.70	73.5	103.5	46.82	92.8	0.035	1190
R401c	R22/R152a/R124	33/15/52 % Zeotrop, Glide 6.7 K	-30.5	82.6	109.8	45.76	101.03	0.03	830
R402a	R125/R290/R22	60/2/38 % Zeotrop, Glide 2.0 K	-49.00	53.0	75.0	42.4	101.6	0.02	2570
R402b	R125/R290/R22	38/2/60 % Zeotrop, Glide 2.3 K	-47.00	56.1	83.0	44.5	94.71	0.03	2240
R403a	R290/R22/R218	5/75/20 % Zeotrop, Glide 2.4 K	-50.00	59.6	93.0	~45	89.8	0.04	2670
R403b	R290/R22/R218	5/56/39 % Zeotrop, Glide 1.2 K	-51.00	58.2	90.0	43.4	102.6	0.03	3680
R404a	R125/R143a/R134a	44/52/4 % Zeotrop, Glide 0.7 K	-47.00	55.4	73.0	37.8	97.6	0	3750
R405a	RC318/142b/152a/22	42.5/5.5/7/45 Zeotrop, Glide 8.4 K	-28.67	80.4	106	42.92	111.9	0.028	4582
R406a	R600a/R142b/R22	4/41/55 %, Zeotrop, Glide 9.2 K	-30.00	82.86	116.5	48.8	89.85	0.057	1760
R407a	R32/R125/R134a	20/40/40% Zeotrop, Glide 6.6 K	-42.30	56.9	83.0	45.4	90.1	0	1920
R407b	R32/R125/R134a	10/70/20% Zeotrop, Glide 4.4 K	-45.20	54.0	76.0	44.5	102.9	0	2560
R407c	R32/R125/R134a	23/25/52% Zeotrop, Glide 7.1 K	-40.00	59.24	86.0	46.5	86.2	0	1610
R407d	R32/R125/R134a	15/15/70% Zeotrop, Glide 6.7 K	-36.05	65.77	91.56	44.8	90.96	0	
R407e	R32/R125/R134a	25/15/60% Zeotrop, Glide 7.3 K	-39.15	60.7	88.76	47.34	83.78	0	
R408a	R125/R134a/R22	7/46/47% Zeotrop, Glide 0.6 K	-44.00	57.6	83.0	40.9	87.0	0.026	3050
R409a	R22/R124/R142b	60/25/15% Zeotrop, Glide 8.1 K	-34.00	76.4	107.0	46.1	97.45	0.05	1440
R409b	R22/R124/R142b	65/25/10% Zeotrop Glide 7.2 K	-35.00	73.9	104.4	47.1	96.67	0.05	1425
R410a	R32/R125	50/50% Zeotrop, Glide 0.2 K	-51.00	43.1	72.0	49.5	72.6	0	1890
R410b	R32/R125	45/55% Zeotrop, Glide 0.2 K	-51.47	43.4	69.46	46.65	75.57	0	1890
R411a	R1270/R22/R152a	1.5/87.5/11% Zeotrop, Glide 2.4 K	-38.46	66.4	98.06	49.5	82.36	0.078	1503
R411b	R1270/R22/R152a	3/94/3% Zeotrop, Glide 1.3 K	-40.52	63.4	95.95	49.5	83.07	0.052	1602
R411c	R1270/R22/R152a	3/95.5/1.5% Zeotrop, Glide 1.0K	-41.34	62.9	94.75	49.51	83.44	>0	
R412a	R22/R142b/R218	70/25/5% Zeotrop, Glide 8.1 K	-36.06	73.8	116.6	~45	92.17	0.055	2040
R413a	R134a/R218/R600a	88/9/3% Zeotrop, Glide 6.9 K	-35.00	75.4	101.0	~46	103.95	0	1770
R414a	R22/124/142b/600a	51/28.5/16.5/4 Zeotrop, Glide 8.3 K	-29.9	79.7	110.6	46.96	96.93	>0	
R414b	R22/124/142b/600a	50/39/9.5/1.5 Zeotrop, Glide 8.3 K	-30.27	78.5	108.0	45.9	101.6	>0	
R415a	R22/R23/R152a	80/5/15% Zeotrop, Glide 10.3 K	-41.9	63.5	97.14	50.37	81.72	>0	
R417a	R125/R134a/R600	46.6/50/3.4% Zeotrop, Glide 5.5 K	-41.80	64.5	90.5	38.6	106.6	0	1950
R500	R12/R152a	73.8/26.2% Azeotrop	-33.5	77.5	105.5	44.23	99.31	0.74	6310
R501	R12/R22	25/75% Azeotrop	-40.41	65.3	96.2	47.64	93.10	0.29	3400
R502	R22/R115	48.8/51.2% Azeotrop	-45.50	59.46	80.7	40.0	111.7	0.33	5590
R503	R13/R23	59.9/40.1% Azeotrop	-88.7	-1.6	19.5	41.82	87.5	0.6	11860
R504	R32/R115	48.2/51.8 Azeotrop	-57.2	38.32	62.14	44.39	79.2	>0	
R505	R12/R31	78/22% Azeotrop	-29.9				103.5	>0	
R506	R31/R114	55.1/44.9 Azeotrop	-12.3	103.6	142.06	51.55	93.7	>0	
R507a	R125/143a	50/50% Azeotrop	-47.00	54.3	71.0	37.9	98.9	0	3800
R508a	R23/R116	39/61% Azeotrop	-87.40	-2.5	11.0	37.0	100.1	0	10175
R508b	R23/R116	46/54% Azeotrop	-86.00	-2.9	12.1	38.3	95.39	0	10350
R509a	R22/R218	44/56% Azeotrop	-47.50	57.0	83.2	~45	123.97	0.025	4668
R723	NH3/Dimethylether	60/40% Azeotrop	-36.6	58.7	131.0	110	23.0	0	0

1. Gasket properties

Gaskets can be of a wide variety of materials but in PHEs and SWPHE used in refrigeration applications, practically only elastomers - rubber - are used.

Polar/Nonpolar. In the classification of gaskets, the polarity is an important property. A polar compound, despite being electrically neutral, has one end negatively charged and the other positive. A nonpolar compound has no such unequal distribution of electrical charges. Polar compounds have usually a group or atoms such as -OH, -HCO, -COOH, -CL, or NH_2^- attached to the molecule.

As a general rule, a polar solvent dissolves a polar compound but not a nonpolar compound and vice versa or differently expressed, the smaller the difference in polarity is, the more soluble is the compound

Another condition for solubility is that the compounds are chemically similar.

Components in a gaskets. An elastomer gasket is made of several components:

♦ ***The elastomer*** is a long chain of carbon molecules to which other atoms are attached. It is basically of three types:

- Pure hydrocarbon - the chain is composed of only carbon and hydrogen - polymers, which are non-polar. Typical gaskets are EPDM & RCB.
- Hydrocarbon with some hydrogen atoms substituted to other atoms than carbon/hydrogen or to a polar group. A polar polymer is then created. Typical gaskets are nitril rubber, chloroprene and fluorocarbons
- The main chain is made of silicon atoms instead of carbon. Silicon rubber is an example.

♦ ***Curing agents***. The raw polymer chains have to be cross-linked forming a three dimensional gitter, before becoming an elastomer proper. This is made by agents such as sulphur, resins, peroxides, etc.

♦ ***Fillers.*** Carbon black - the most common - increases hardness, tensile & shear strengths.

♦ ***Antidegradants***, (such as anti-oxidants/ozonants) improve the resistant against various forms of attacks.

♦ ***Processing aids***, e.g. softeners & mould release agents improve the manufacturing process.

Gasket properties that characterize an elastomer are:

♦ ***Hardness.*** A hard gasket resists swelling better than a soft, but if too hard, the sealing properties might suffer.

♦ ***Tensile & tear strengths and elongation to brake.***

♦ ***Compression set*** is the residual deformation after a pressure has been applied under specified conditions and then released.

♦ ***Relaxation*** is the rate which a compressed gasket looses its compression force. This is an important property to evaluate the sealing ability of a gasket.

Gaskets for PHEs. The gasket in a PHE is subjected to much more severe conditions than e.g. an O-ring in a flange. The practical significance of this is that temperature and pressure limits issued by rubber manufactures are not always pertinent to PHEs.

♦ ***Sealing.*** A PHE gasket is normally not self-sealing, contrary to a normal O-ring.

♦ ***Shape.*** The straight part of a field gasket - the gasket that surrounds the heating surface - have better sealing abilities than e.g. the corners.

♦ ***Length.*** The sheer length of field gaskets means that a relative better sealing ability is required than in a short flange O-ring.

Nonglued gaskets should not be used:

♦ when frequent opening is necessary,

♦ for gaskets with large swelling,

♦ in oxidizing brines;

♦ when the gasket sticks to the matching plate.

2. Gasket corrosion

Attacks on the gasket are:

♦ ***Swelling,*** which is a similar phenomena to dissolution of a compound in a solvent and the same rules apply, e.g. a nonpolar solvent, oil, swells a nonpolar gasket, EPDM, but not a polar gasket, NBR.

Swelling is a physical phenomena, the gasket is not destroyed by it, though properties can change if the gasket contains soluble additives. The gasket might then ***shrink***.

♦ ***Oxygen, ozone***, ***light and high temperatures*** attack and destroy the chain or the cross-links. The rubber becomes hard and brittle and looses its elasticity.

♦ ***Chemicals*** can have the same effects as above, but the resistance is very dependent on gasket and chemical. There are gaskets resistant to hot concentrate sulphuric acid but less to acetones.

Chemicals can also attack the other components in the gasket and destroy or dissolve them.

♦ ***Fluoroelastomers*** (FPMs) have some or all of the hydrogen atoms in the chain exchanged to fluorine. The result is an elastomer, which is very resistant to all types of attacks but with one very important exception. It is readily disintegrated by ammonia, ammonia compounds, hydroxyles and amines, which then never should be used with FPM.

This is especially important for by titanium, which is corroded by even minute amounts of fluorine.

Diffusion of a fluid is closely related to swelling. If the gasket swells in a fluid, the fluid diffuses well through the gasket. It is temperature and pressure dependent.

Cold leakage is a high temperature phenomena occurring when a PHE operating at high temperatures is shut down and started up a number of times. The cold gasket then obtains a permanent set causing leakage. Both the set and leakage disappears when the gasket heats up. EPDM is susceptible to cold leakage.

Storage. Long-term storage of gaskets should be done in a dark place, preferably stored in e.g. plastic bag filled with an inert gas such as nitrogen.

Organic compounds	Gasket	Organic compounds	Gasket
Acids (R-COOH)	EPDM, FPM	Peroxides (R-O-O-R)	-
Alcohols, Glycols, Polyols (R-OH)	EPDM, FPM	Phenols (R-OH)	(EPDM)
Aldehydes (R-CHO)	(EPDM)	Sulphides (R-S-R)	-
Amins ($R-NH_2$)	EPDM	Sulphonic acids ($R-SO_3H$)	FPM
Aromats (incl. alkylated)	FPM		
Esters (R-COO-R)	EPDM	**Inorganic compounds**	
Ethers (R-O-R)	(EPDM)		
Halocarbons (R-F,Cl, Br, J)	FPM, Typ G	Acids, oxidizing. (HNO_3, $HClO_3$, H_2SO_4(>75%), CrO_3)	(FPM, Type G)
Hydroxyl amines	EPDM, HNBR	“ non “ (H_3PO_4, HCl, HB_3, H_2SO_4(<75%), H_2S)	EPDM
Ketones (R-CO-R)	(EPDM)	Alkalies (NaOH, NH_4OH, KOH, LiOH	EPDM
Naphtenes	NBR, HNBR, FPM	Hot pressurized air	FPM
Nitrates ($R-NO_2$)	EPDM	Salt solutions, oxidizing ($K_2Cr_2O_7$, $KMnO_4$, NaOCl)	(EPDM)
Oximes (R=N-OH)	EPDM	“ " non ” (NaCl,Na_2SO_4), K_2CO_3, $CuSO_4$)	EPDM
Paraffins & Olefins	NBR, HNBR, FPM	Water, all types	HNBR,NBR,EPDM

Gasket selection for various fluids.

3. Refrigerant gaskets.

General. A refrigerant gasket has to stand not only the refrigerant but also the oil. The problem is that a refrigerant such as ammonia is a polar compound but a mineral oil is nonpolar. The effect of the combination oil-refrigerant on gaskets is complex and depends on refrigerant, oil, temperature, pressure, PHE type, etc.

The discussion below is a general treatment. In an actual case, Alfa Laval should be consulted especially for more unusual combinations of oil and refrigerants. Alfa Laval has a very large bank of data on gasket usage.

The porthole gasket. This gasket is circular and is the only gasket in contact with refrigerant in a SWPHE. The small amount of this gasket means that it can be made of more expensive materials. The simple shape also means that various protective devices can be fitted to the gasket and in general, it can withstand harder operating conditions than a field gasket. Below are some types.

- ***Cloroprene (CR)*** is a polar gasket but despite this, it has good resistance to both ammonia & R22 and oil.
- ***LT-NBR*** is a soft gasket, which keeps its sealing properties down -45 °C. At high pressures (temperatures), the diffusion is too large.
- ***Hydrated nitril rubber (HNBR)*** is a special nitril, which can be used up to 120 °C for ammonia but not R22.
- ***Fluoroelastomers (FPM)*** have very good resistance to a wide variety of chemicals, see table 1. It is expensive though and is normally not required for refrigerants.

The new refrigerants & oils. R22 is the last of the “old” refrigerants. It contains a chlorine atom, which makes it a good solvent, but it also swells gaskets. The new, chlorine free refrigerants are less good solvents and they will accordingly swell gaskets less.

Some of the new synthetic oils, especially PAG & polyesters but not polyolefines are polar compounds. This makes them interesting for use together with ammonia and EPDM-gaskets, a non-polar gasket, as EPDM has very good general properties, especially at high temperatures.

4. Brine gaskets.

General. The field gaskets on the brine/liquid side are available in two commonly used materials and one on special request. The table below shows the general properties of these gaskets in different fluids.

Nitril rubber (NBR) is the most common field gasket compatible with most brines and glycols. At very low temperatures ***LT-NBR*** to be used.

Ethylene propylene dien monomer (EPDM) is used when polar compounds such as acids are used. It has good high temperature properties.

Fluoroelastomers (FPM) have excellent resistance to a number of chemicals, e.g. hot concentrated sulphuric acid. It is however, a very expensive gasket and usually a minimum quantity is required. It should never be used with titanium se §3. It is thus seldom justified except in very special cases.

Fluids, which can be treated in a PHE includes practically all common brines and glycols as well as most chemicals and alimentary fluids. There are however some types of fluid which should be avoided or discussed with Alfa Laval before being used together with gaskets. Some of them might be used with FPM gaskets but the price normally excludes this.

Alimentary fluids, e.g. tap water, beer & soft drinks ***can*** be treated in a PHE. Contact Alfa Laval if formal FDA (the commonly accepted US standard) approval is needed.

Heat transfer oils, except pure mineral and silicon oils, should in general be avoided

Chlorinated (except R22) ***& many unsaturated*** (with double or triple bonds between the carbon atoms) ***compounds,*** **aldehydes, ethers, ketones** ***& aromatics*** (benzene rings) should in general be avoided together with gaskets. This is particularly true for compounds with low molecular weights. A large molecule with only one offensive group can be less aggressive. Contact Alfa Laval before use.

Contact Alfa Laval before using ***strong oxidizing acids & salts***, e.g. potassium bichromate.

Note, that gaskets are in general resistant to nonoxidising acids and salts, e.g. hydrochloric acid, but it might be difficult to find a corrosion resistant plate material.

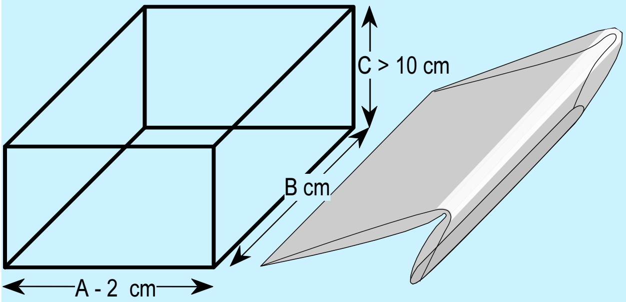
Swedish marinated salmon, "Gravlax".

1. Origin

"Gravlax" or "gravad lax" is an old method of preserving salmon, originating in Sweden. The Swedish word "lax" means salmon, "grav"signifies grave, pit, moat or trench, which gives an idea how the method probably was developed.

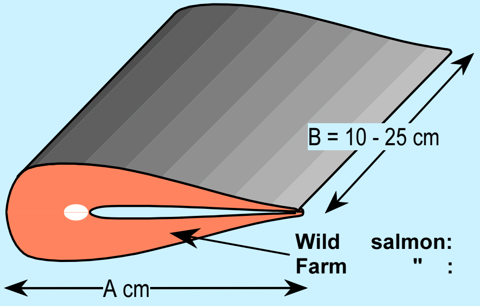
2. Equipment:

A plastic or Al box and a plastic bag.



2. The fish.

The middle part of a salmon.



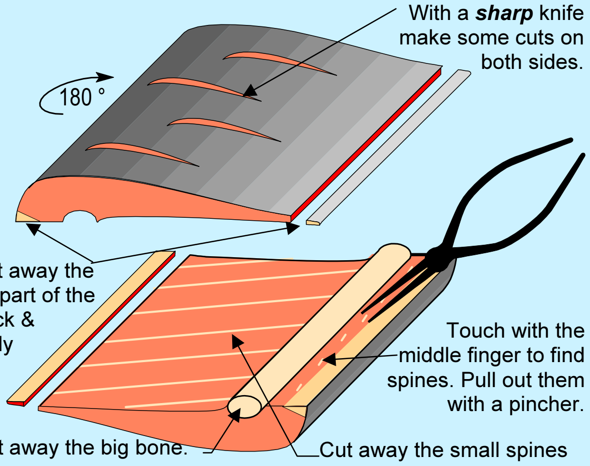
Wild	**salmon:**	Grey -pink
Farm	**"** **:**	Orange -pink

The fish should be raw, fresh - neither smell nor red eyes - or frozen. If frozen, it should be as an entire fish, not cut-out parts.

The best salmon is the wild, as found in the rivers of Scandinavia and British isles. However, it has sometimes difficulty in competing with the farmed salmon due to the more vivid colour of the later. It is hard to find, though. Try to get a middle part. Do not wash it.

4. Preparing the salmon.

Cut and trim the salmon as shown. One of the halves should be turned 180 ° as shown in the figure.



With a ***sharp*** knife make some cuts on both sides.

Cut away the fat part of the back & belly

Touch with the middle finger to find spines. Pull out them with a pincher.

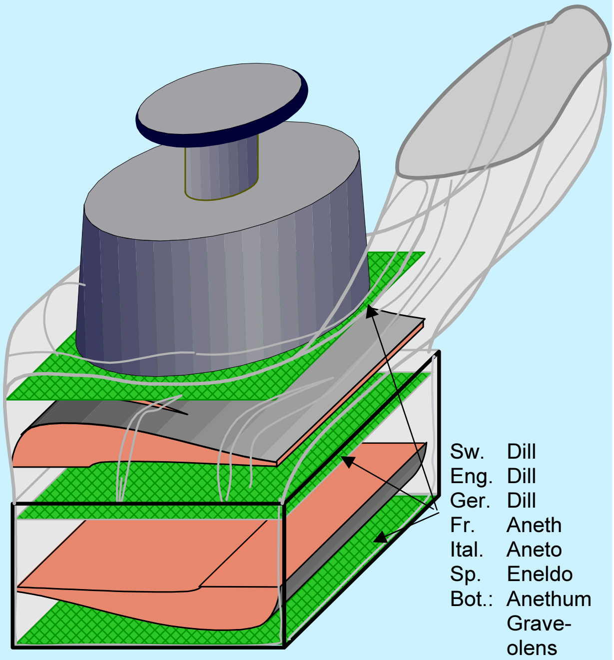
Cut away the big bone.

Cut away the small spines

5. The ingredients.

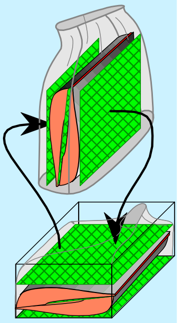
Prepare:

Salt:	~ 100 g
Sugar:	~ 70 - 100 g
Black pepper:	~ 10 g (ground)
Dill:	~ 100 g (Freeze 20 g for the sauce)



Sw.	Dill
Eng.	Dill
Ger.	Dill
Fr.	Aneth
Ital.	Aneto
Sp.	Eneldo
Bot.:	Anethum Graveolens

6. The marination.



Mix all the ingredients except the dill in a large flat bowl. (The sugar is necessary. It does not impart any sweet taste but prevents the salmon from picking up too much salt.)

Put in the salmon halves and be sure that they are well covered with the mixture on the surface. Leave the mixture which remains on the salmon, but not more.

Put the two salmon halves as shown in the plastic bag, intermixed with three layers of dill and put a weight on the top.

Turn it around 1 - 3 times/day. It needs a curing time of at least two full days at a temperature of about 5 to 10 °C

Addition of some good cognac and/or tarragon is a variant worth trying even though it is not the traditional recipe.

5. The sauce (Hovmästarsås).

Salt & pepper:	According to taste
Sugar:	1 table spoon (to taste)
Dill:	~20 g (incl. in the amount above)
Mustard:	2 table spoons
Eggs:	1 egg yolk
Oil:	100 ml
Vinegar:	2 table spoons

All ingredients should have the same temperature. Mix sugar, mustard, vinegar & egg yolk. Add oil while stirring vigorously. Add salt & pepper.

Take the deep frozen dill (dill freezes well), it is brittle. It is then easy to draw off the finer parts of the leaves between the thumb and the index finger, leaving the naked stem. If necessary, chop the leaves. Mix the leaves with the sauce.

6. Serve with:

• As entry:	Toast or avocado, lemon and the sauce above.
• As main:	Stewed or boiled potatoes & stewed spinach.
• Drink:	Dry white wine or beer & absolut vodka.

K-Å Pihlsgård, C. Stenhede & M. Strömblad.

Alfa Laval in brief

Alfa Laval is a leading global provider of specialized products and engineered solutions.

Our equipment, systems and services are dedicated to helping customers to optimize the performance of their processes. Time and time again.

We help our customers to heat, cool, separate and transport products such as oil, water, chemicals, beverages, foodstuffs, starch and pharmaceuticals.

Our worldwide organization works closely with customers in almost 100 countries to help them stay ahead.

How to contact Alfa Laval

Up-to-date Alfa Laval contact details for all countries are always available on our website at www.alfalaval.com

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