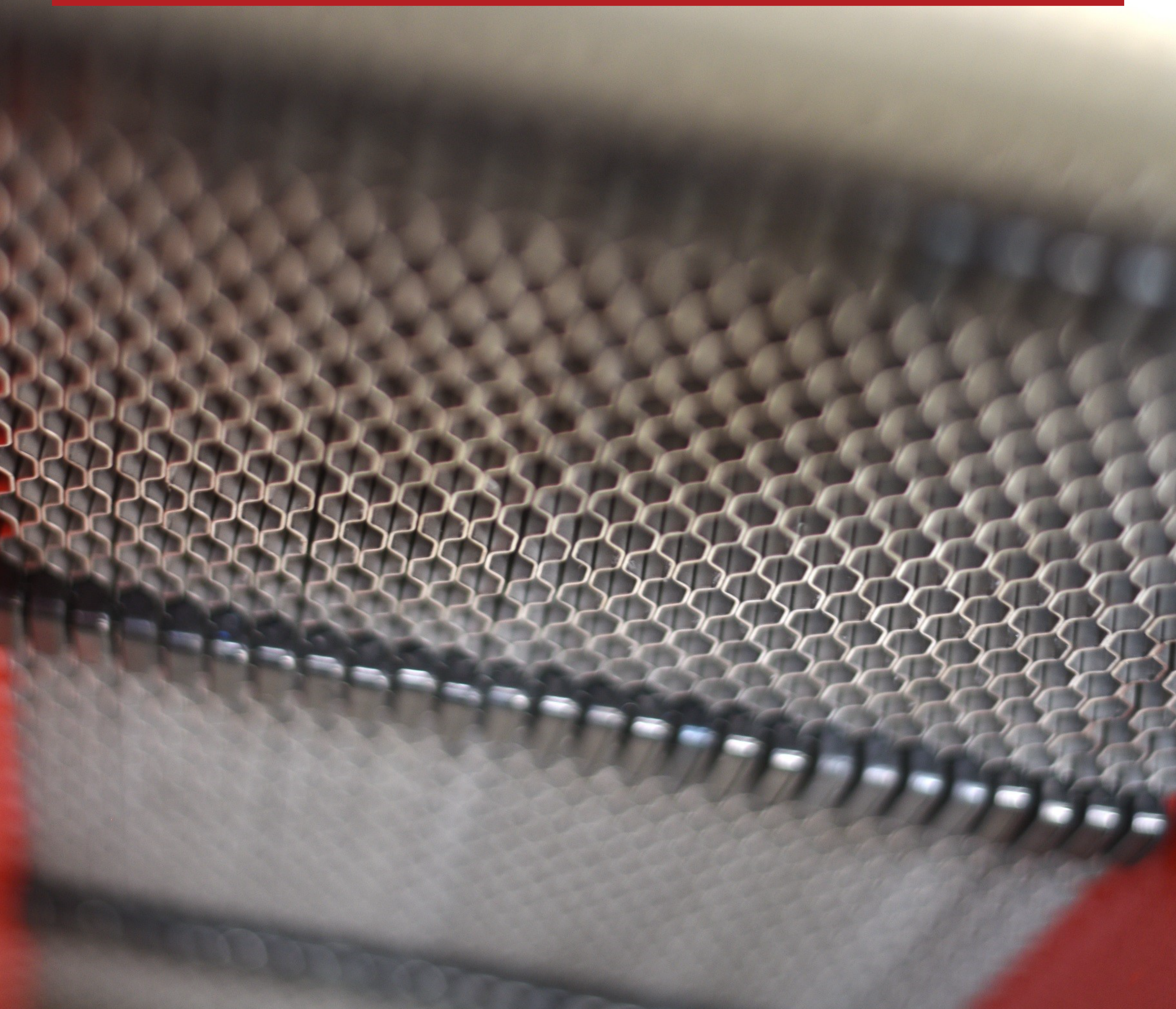
ENGINEERING
TOMORROW

Application Handbook
Industrial Refrigeration
Semi-welded plate heat exchanger



Contents

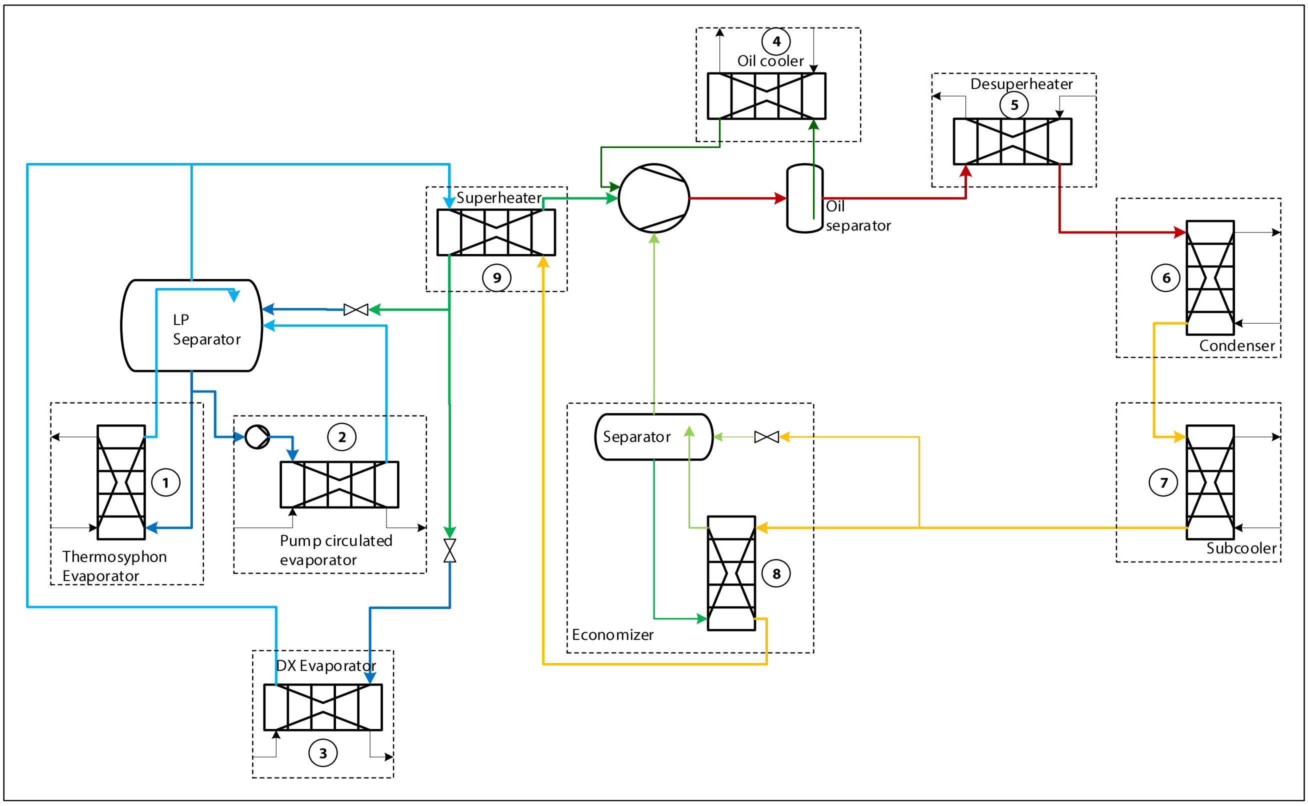
1. Heat exchangers

Heat exchangers perform the primary duties of the refrigeration/heat pump systems – heating and cooling - as well as several internal functions.

For efficient operation of heat exchangers, it is necessary that the system is both designed, constructed and controlled appropriately.

Figure 1.1 below shows the heat exchangers that is normally used in a single stage system. Note that not all are necessary and that the heat exchangers numbers (7) to (9) are not often all used at the same time. Likewise, it is not normal to use all types of evaporators – thermosyphon (1), pumped (2) and DX (3) – in the same system. These variations are included here to illustrate all possible variations. All valves except expansion are left out for clarity.

Figure 1.1: Overview of the most common heat exchangers in a refrigeration system



1. Thermosyphon evaporator: This type is a 'flooded evaporator' signifying that there more refrigerant is circulated than evaporated. Because of that the exit flow is a mixture of vapor and liquid and because of that it is returned to a separator to ensure a dry suction gas is fed to the compressor. The thermosyphon evaporator is fed with liquid through natural circulation.

2. Pumped evaporator: This type is also a 'flooded evaporator' but in comparison with the thermosyphon, it uses a pump to feed liquid to the evaporator. These are very often air coolers located in such a way that thermosyphon operation is not possible (e.g. far away from the compressor/separator, for instance in a cold store or a freezing tunnel). Plate freezers is another common pumped evaporator.

3. DX evaporator: DX is an abbreviation for 'Direct expansion' signifying that the evaporator is fed with high pressure liquid through an expansion valve. The control of the expansion valve ensures that the refrigerant return from the evaporator is fully evaporated, e.g. with no liquid in it, and as such it can be passed directly to the compressor.

4. Oil cooler: Screw compressors require oil cooling and the oil cooler perform this task. Oil coolers can be glycol/water cooled (as shown here) or thermosyphon oil coolers (not shown) that evaporates liquid from the condenser to cool the oil. Evaporated liquid is returned to the compressor discharge line before the condenser to be re-condensed. Water/brine cooled oil coolers are often part of a heat recovery system or heat pumps.

5. Desuperheater: The desuperheater cools discharge gas from the compressor before entry into the condenser, usually for use in a heat recovery system / heat pump. The higher temperatures in the desuperheater enables a higher temperature of the heat recovery than is possible in the condenser.

6. Condenser: The condenser cools the compressor discharge gas and condense it into liquid. Condensers can be evaporative condensers (not shown), air cooled condensers (not shown) or a glycol/water cooled heat exchanger as shown here. The heat delivered to the glycol/water circuit can either be dispersed by a cooling tower or used in a heat recovery system / heat pump.

7. Subcooler: The subcooler cools the condensed liquid further using water/glycol or possibly air. With a colder liquid being passed to the low-pressure side, the compressor capacity increase. Subcoolers are often part of a heat recovery / heat pump system.

8. Economizer: Screw compressors usually have an additional port – the economizer port - that allows refrigerant to be fed in after the compression has started. As such it can be used to compress additional refrigerant at a starting pressure above the evaporating pressure, without compromising the compressors capacity at evaporating pressure. Generally, two different economizer types exist – the 'open' economizer (not shown) and the 'closed' economizer. The open economizer is a simple vessel without heat exchangers, so it will not be covered here. The economizer cools the condensed liquid further, increasing the compressor capacity in the same way as a subcooler. The cooling is performed by flashing and evaporating some of the condenser liquid and this vapor is passed to the compressor's economizer port. In the closed economizer, a heat exchanger performs the cooling duty.

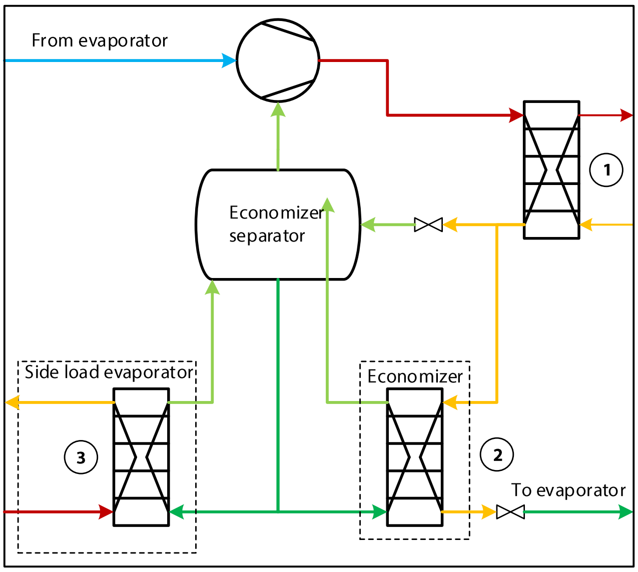
9. Superheater: Sometimes called a Suction Superheater. This heat exchanger heats the suction gas to the compressor to ensure that no liquid is fed to the compressor. As a heat source, the condenser liquid is used. Since the superheating of the compressors suction gas decrease the gas density, the compressor mass flow drops, but the subcooling of the condenser liquid compensate for this. Depending on the refrigerant and operating conditions it can result in a total change in capacity – either positive or negative – but usually it is very close to break-even.

Four note-worthy additions to the above diagram were not added to aid the clarity.

1.1.1 Side load

It is not unusual to add an evaporator to an economizer, allowing external cooling to be performed at an additional (higher than evaporating temperature) temperature level. This is often referred to as 'side load'.

Figure 1.2: Side load evaporator

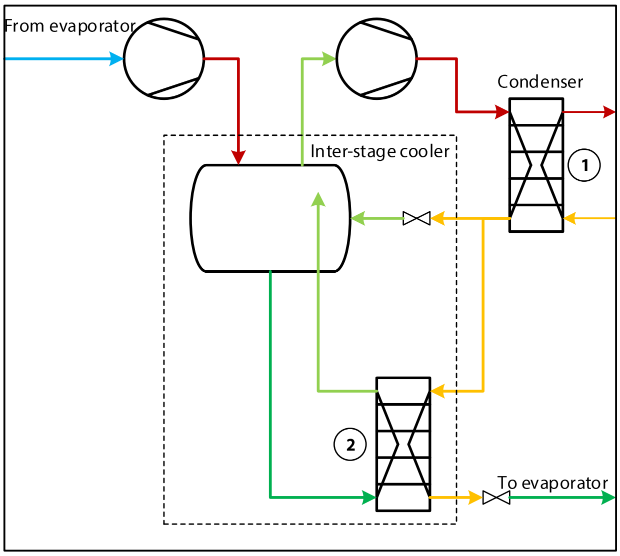


In the above sketch in Figure 1.2, a thermosyphon evaporator (3) has been added to the economizer (2) which here is mounted directly after the condenser (1). For all practical purposes, this evaporator can be treated as a normal thermosyphon evaporator. Pumped operation from the economizer is also an option.

1.1.2 Intercooler

Further, on two-stage systems an intercooler (also known as an interstage cooler) cools the discharge gas from the low-stage compressor before entry into the high stage compressor. If not cooled, the high temperature of the suction gas to the high-stage compressor will result in a (too) high discharge temperature of this compressor. The cooling of the gas is often performed by bubbling it through a liquid bath, thereby cooling it to (near) saturation. In much the same way as the economizers the intercooler can be 'open' or 'closed' where the open type does not involve a heat exchanger. A closed intercooler is shown in Figure 1.3 below.

Figure 1.3: Inter-stage cooler



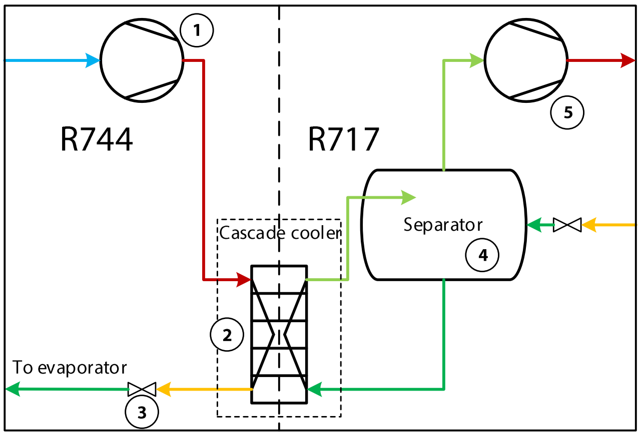
The intercooler takes refrigerant liquid from the condenser (1) and cools it further in the intercooler heat exchanger (2), using condenser liquid flashed to the vessel. Note that the flashed liquid is also used to desuperheat the low-stage compressor discharge gas, through bubbling it through a liquid bath. Other possible ways to desuperheat the discharge gas is injection of liquid into the low-pressure compressors discharge pipe or in a heat exchanger with external cooling. The former method must be controlled carefully to avoid having liquid in the suction of the high-stage compressor. Generally, the bubbling method is the most common.

In much the same way as on the economizer, a heat exchanger can be added to provide a refrigeration duty at the interstage pressure (not shown), either through thermosyphon operation or pumped operation. Alternatively, a separator can be added in parallel with the intercooler.

1.1.3 Cascade cooler

A cascade cooler is the connection between the two refrigerant circuits in a cascade system. Typically, the low temperature refrigerant will be R744 (CO_2) while the high temperature refrigerant is R717. The cascade cooler condenses R744, the cooling for this process is provided by evaporating R717 at a temperature below the R744 condensing temperature.

Figure 1.4: Cascade cooler



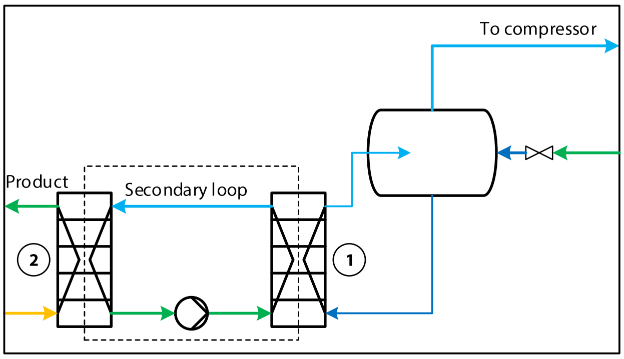
In the Figure 1.4 above, the R744 compressor (1) discharge gas is condensed in the cascade cooler (2) and flashed back to evaporating temperature, by the R744 expansion valve (3) (possibly after storage in a receiver). On the R717 side the cascade cooler is used as a thermosyphon evaporator with a liquid separator (4). From the separator, the vapor created is compressed by the R717 compressor (5).

Typically, R744 is condensed at -10°C and R717 evaporates at -15°C. The temperature difference in the cascade cooler can be considered an efficiency loss, so it is recommended to examine the change in operating cost (compressor power) versus the installation cost of a larger cascade cooler that operates at for instance 3°C temperature difference.

1.1.4 Intermediate circuit

Finally, it is not uncommon to have an intermediate circuit between the evaporator and a product.

Figure 1.5: Secondary loop/brine loop



In case of a leak in the evaporator (1), refrigerant can contaminate the secondary side. If the secondary side is a product such as beer or milk, the product will be destroyed. Thus, the product is cooled in a secondary heat exchanger (2), that is cooled by the water/glycol cooled by the evaporator (1). In this way, product safety is ensured.

1.2 Heat transfer fundamentals

Heat exchangers utilizes differences in temperature to move heat. Two fluid flows are brought in to close contact on either side of a wall or a pipe, allowing energy – in the form of heat – to travel through the wall or pipe from the hot side to the cold side. The heat exchanged (gain or loss) leads to changes in the media on either side. In the case of a single phase (liquid or vapor) it leads to a temperature change. In evaporators, the heat exchanged leads to refrigerant liquid to be evaporated and in condensers, the heat exchanged leads to refrigerant vapor being condensed.

Generally, the capacity of a heat exchanger can be expressed by:

$$Q = U \cdot A \cdot \Delta T$$

Where:

- Q is the capacity [W]
- U is the overall heat transfer coefficient [W/(m^{2}K)]
- A is the heat transfer area [m^2]
- ΔT is the temperature difference [K or °C]

1.2.1 Heat transfer types in heat exchangers

In heat exchangers for refrigeration systems, heat is typically transferred from gas/vapor or air to a liquid refrigerant or phase-changing refrigerant, or the other way around. Two types of heat transfer are experienced in this case:

- Conduction: When heat is transferred through a solid material, e.g. through the wall of a metal pipe, or when heat is transferred between two solid materials through contact, e.g. in plate freezers.
- Convection: When heat is transferred between a liquid, gas, or phase-changing refrigerant and a solid material, e.g. when evaporating refrigerant flow through the evaporator pipes and heat is transferred from the pipe wall to the evaporating refrigerant.

1.2.2 Heat transfer coefficient - U

The overall heat transfer coefficient, U, describes how well heat is transferred in the heat exchanger. The overall heat transfer coefficient depends on the flow properties for the heated media and the cooled media in the heat exchanger, the wall/pipe size and its thermal conductivity (k-value) as well as factors like fouling (dirty surfaces).

For single-phase flows (water, glycol, vapor or liquid refrigerant) the heat transfer coefficient depends on the fluid properties, which are usually a given factor, and the velocity of the fluid in the heat exchanger. A higher velocity gives a higher heat transfer coefficient, but it also gives a higher-pressure loss. Liquid flows usually have the highest heat transfer coefficients.

For evaporation, the heat transfer coefficient depends on the properties of the refrigerant as well as the heat flux (W/m2) and operation method - flooded, DX and various parameters such as circulation rate or superheat. For condensation, it is much the same, albeit with different parameters. Often, there is not much a designer of heat exchangers can do about evaporation and condensation heat transfer coefficients.

Fouling is a value given by the designer that express the dirt, oil etc. that adds resistance to the heat transfer. This value is very much an experience-based value with the actual value depending on the type of fluid, such as clean water versus river or sewage water – the latter requiring a much higher fouling than the former. Oil contamination of refrigerants is also the source of fouling. Fouling is usually stated for each side of the heat exchanger.

Finally, the properties of the wall between the two sides are usually given by the actual heat exchanger through the materials and the geometry selected.

All of these factors combine into the total heat transfer coefficient. It is note-worthy that it is the side with the lowest heat transfer coefficient, that is the largest resistance to heat transfer, which dominates the total heat transfer coefficient. For instance, an evaporator with very low speed on the water/glycol side will have a low heat transfer coefficient on that side, so the evaporative side can basically be very high without any significant difference to the overall heat transfer coefficient.

1.2.3 Heat transfer surface - A

The heat transfer area is not always as simple as it sounds. In many heat exchangers the surface available to the two sides are not the same. For instance, in shell-and-tube heat exchangers, the pipe surface inside and outside is not the same due to the difference in diameter (wall thickness). This is a factor that affect the overall heat transfer coefficient and it is important (when programming the calculation tool) to ensure that the area used is in accordance with the correction to the overall heat transfer coefficient. In cases such as plate heat exchangers, the area is the same on both sides and thus present few problems, calculation wise.

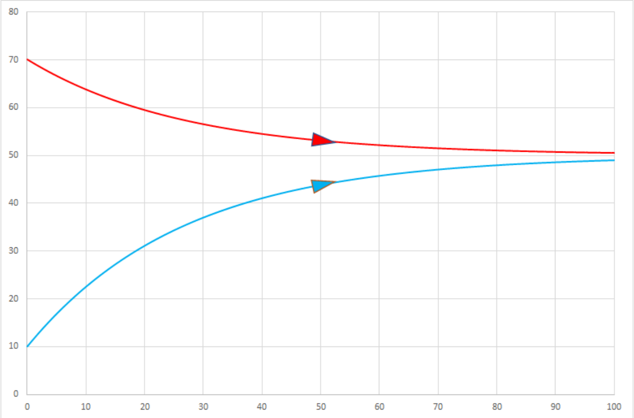
Typically, a total heat transfer rate, U*A given in [W/K], is calculated based on the individual contribution from each heat transfer type's U-value and corresponding heat transfer area, A.

1.2.4 Temperature difference - ΔT

The temperature difference is the driving force for a heat exchanger, as the heat flows from the hot side to the cold side. However, as the temperatures of the flows change, it is not always apparent what the temperature difference is at a given point.

An example: Two water flows enter a heat exchanger from the same side with the temperatures 70°C and 10°C respectively. The hot water flow has twice the volume flow rate of the cold-water flow. The figure below shows the temperature on the y-axis (vertical axis) and the relative path travelled by the flow through the heat exchanger in %, thus 0% is the inlet, 50% is half-way and 100% is the outlet.

Figure 1.6: Temperature profile for co-current heat exchanger



Since the hot water flow rate is twice the flow rate of the cold water, an equilibrium temperature of 50°C would be achieved if the flows were mixed. It can be seen from the figure #, that the flows approach 50°C at the outlet. The rate of temperature change for the flows through the heat exchanger should be noted. At the inlet, the heat transfer is high, as the temperature changes much. As the flows gets heated/cooled, the temperature difference decreases, and the heat transfer is reduced. At the outlet the flows are almost at the same temperature, but the temperatures change very little from about 80% until the outlet, due to the very low heat transfer at the small temperature difference. If the heat exchanger was infinitively large (A = ∞), the temperatures of the flows would reach 50°C.

This figure above illustrates the problem of determining what the actual temperature difference is since the difference varies from 60°C at the inlet to 1.5°C at the outlet. The solution is to use a 'logarithmic mean temperature difference' (LMTD) that compensate for the changes in temperature. The LMTD is calculated as follows:

$$LMTD = \frac{(\Delta T_{inlet}) - (\Delta T_{outlet})}{\ln\left(\frac{\Delta T_{inlet}}{\Delta T_{outlet}}\right)}$$

Where ΔT's represent the temperature differences on either end of the heat exchanger. Using this, the LMTD is calculated to be 15.86°C.

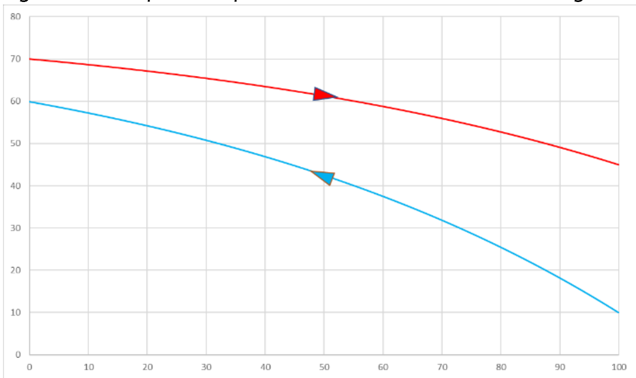
1.2.5 Co-current vs counter-current

Co-current and counter-current are terms that refer to the direction of the flows in a heat exchanger relative to one another.

- Co-current: A co-current flow heat exchanger has the hot and cold flow to flow in the same direction. The figure above is a co-current heat exchanger. By having the flows in the same direction, a high heat transfer is achieved around the inlet of the heat exchanger. However, for a co-current heat exchanger, the hot side flow cannot be cooled below the outlet temperature of the cold side flow and vice versa, the cold side flow cannot be heated above the outlet temperature of the hot side flow.
- Counter-current: A counter-current flow heat exchanger has the hot and cold flow to flow in opposite directions, such that the cold side inlet is placed at the hot side outlet. A counter-current heat exchanger can cool the hot side below the cold side outlet, since the hot side outlet transfers heat to the cold side inlet and vice versa.

The figure below shows the same example as before but with a counter-current heat exchanger instead of a co-current. It is seen that the temperature drop for the hot side is 25°C, whereas it was 20°C for the co-current. This means that more heat has been transferred in this situation meaning that the counter-current heat exchanger is more efficient than the co-current, and thus the counter-current has the highest capacity.

Figure 1.7: Temperature profile counter-current heat exchanger



In this case, the cold stream can be heated to 60°C and the hot stream can be cooled to 45°C. All other parameters are the same. The difference is that the LMTD is higher at 20.0°C, some 25% which is directly reflected in the capacity as well. As a conclusion, the counter-flow heat exchanger is much more efficient than a co-current.

Visually the difference in LMTD can be seen as the area between the two curves. Although the co-current starts out with a high temperature difference, it drops quickly to near zero, while the counter-current maintain a large difference throughout the whole heat exchanger.

1.2.6 Pinch point

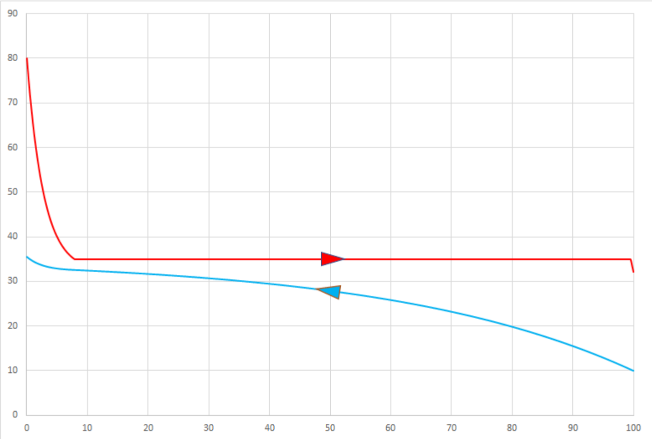
The pinch point is the point in the heat exchanger where the temperature difference is at the minimum. This means that the pinch point is the limiting factor to the heat exchanger regarding capacity. For the two examples shown above, the co-current heat exchanger in figure # has a pinch point temperature difference of 1.5°C, while for the counter-current heat exchanger it is 10°C. A too high pinch point temperature difference is not wanted, since this means that the heat exchanger is too large, and its capacity is not utilized optimally.

The pinch point temperature difference is parameter that is typically looked into when dimensioning a heat exchanger to optimize the performance. Typically, it is a trade-off between efficiency, cost and capacity.

1.2.7 Condensers

In a heat exchanger with phase change it is a little more complex. During the phase change the refrigerant will remain at a constant temperature, or constant temperature glide (slope) for refrigerant mixtures, when the phase change takes place. Below is a sketch of condensing ammonia in a heat exchanger.

Figure 1.8: Temperature profile for condenser

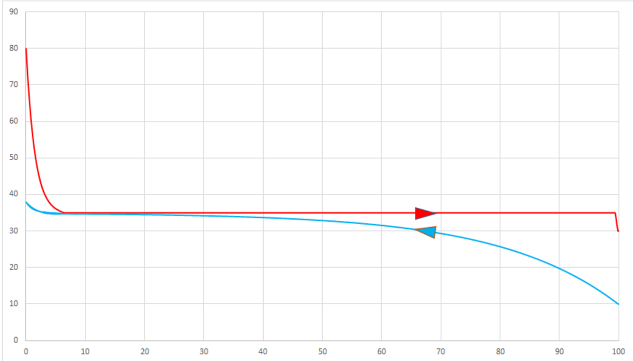


The ammonia enters the heat exchanger at the compressors discharge temperature, which has been set to 80°C in this example. The gas is cooled until it reaches the saturation temperature – here 35°C – where condensing begins. As long as condensation takes place, the temperature remains at the saturation temperature. At the end, all is condensed and a little subcooling takes place. On the water side, the water enters at 10°C and is heated to near the condensing temperature with a gradually decreasing capacity as the temperature difference decrease. Before the end, the water enters the area where the hot discharge gas is cooled, and the water gets a little boost from the increasing temperature difference.

But the sketch illustrates a key point in condenser design. Looking at the refrigerant inlet temperature (80°C) it would be easy to hope for a water exit temperature of perhaps 60°C, which would be very nice in heat recovery systems or heat pumps. However, at the start of the condensation process – at around 8% on the sketch – the water temperature is very close to the condensing temperature. This is the pinch point where the temperatures in the heat exchanger limits the performance.

The water temperature increases approximately 25°C in the condenser sample, but if it was wanted to have a 60°C water temperature out, e.g. a 50°C increase, it would be intuitive to lower the water flow to half. However, that results in the below temperature profile shown in Figure 1.9:

Figure 1.9: Temperature profile for condenser at half water flow



Clearly the water temperature approaches the condensing temperature more rapidly, but it is unable to heat further since the temperature difference drops to nearly zero.

Since the temperature difference between 8% and 30% of the path is approximately 0, there is effectively no heat transfer from the ammonia to the water in that part of the heat exchanger.

As the water stream enters the desuperheating zone, it heats to around 38°C. Hotter than before, but with half flow. As a result, the condensing capacity is reduced to around 55% of the previous example. The pinch point has limited the heat transfer in a large part of the condenser – which is essentially wasted.

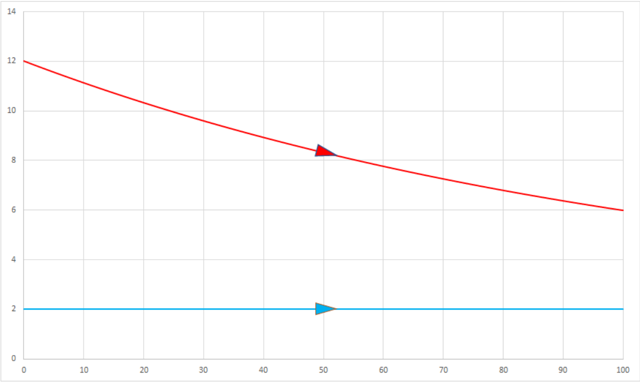
The conclusion is that the overall LMTD – which focus on inlet and outlet temperature differences – is not sufficient to describe the actual situation. Using the inlet and outlet temperatures the LMTD is calculated to 29.76°C (for the first condensing example). However, if the heat exchanger is split into sections according to the 3 different situations on the condensing side, a more precise LMTD can be calculated using the LMTD for each section. In the desuperheating section the LMTD is 7.77°C, in the condensing section 4.98°C and in the subcooling section it is 22.09°C. Since the desuperheating accounts for 11.3% of the total capacity, the condensing 86.4% and the subcooling 2.4% a weighted LMTD scaled according to this is 5.70°C which is approximately 20% of the overall calculation.

It can be noted that the desuperheating duty – which is 11% of the total capacity in this example – requires around 8% of the total area. This is due to the higher LMTD in this area. In this example, that focus on temperatures, the heat transfer coefficient has been set to a constant value. In real life, the heat transfer coefficient of the desuperheating – which is simple gas cooling – is much less than in the condensing and subcooling zones, so the area used for desuperheating would need to be somewhat larger.

1.2.8 Evaporators

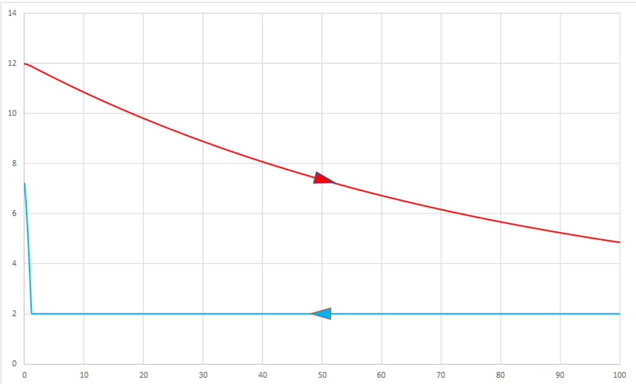
In a flooded evaporator, the evaporating side temperature is constant throughout the heat exchanger. With this simple behavior of both sides, the overall LMTD gives a sufficiently good picture of the temperature difference. Note that from a temperature perspective it does not matter if the refrigerant and brine flows in the same or opposite directions, however it is the norm to have the flows in the same direction to 'kick-start' the evaporation with the largest possible temperature difference.

Figure 1.10: Temperature profile for flooded evaporator



However, in a direct expansion (DX) evaporator the evaporating temperature remains constant until all refrigerant is evaporated and then the gas is superheated to the outlet temperature.

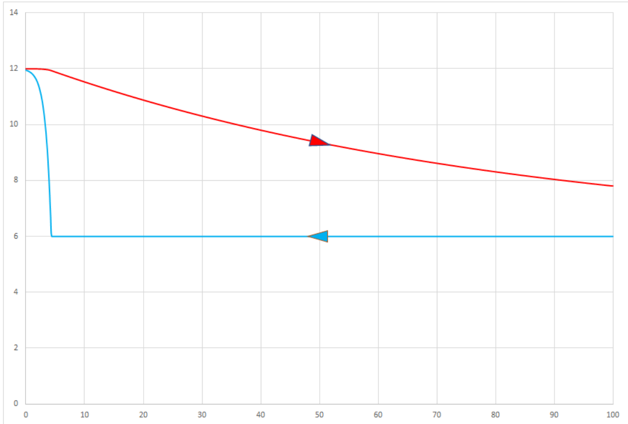
Figure 1.11: Temperature profile for DX evaporator



Again, a simple LMTD consideration from the in/outlet temperatures do not show the real situation. Using these temperatures, the LMTD is 3,75°C, while doing a split similar to the condenser example, the weighted LMTD is 5.70°C – approximately 50% higher.

Similar to the 'pinch-point' problem in the condenser, the creation of superheat in the evaporator can be a problem if the temperatures are too close. It is clear that the hot side outlet temperature cannot go below the evaporating temperature, but it is also necessary to have sufficient temperature difference in the inlet end to create superheat. A sufficiently large superheat is essential for securing that the refrigerant flow to the compressor is always without liquid. Normally superheat is in the range 5 to 10°C. In the temperature profile shown below in Figure 1.12, the DX evaporator from the previous example has had its evaporating temperature raised from 2°C to 6°C.

Figure 1.12: Temperature profile for DX evaporator with higher evaporation temperature



Clearly the evaporator has difficulties in creating superheat and as such, is not safe.

These are the basic operations of heat exchangers, but many variations exist. Cross flow is where the two fluids are neither co- or counter-current but rather perpendicular to each other. Similarly, heat exchangers can have several 'passes' on one or both sides, where the flows change direction. These variations require a careful application of LMTD functions to adequately express the real temperature difference, however this is not covered here.

1.2.9 Limitations to heat exchangers

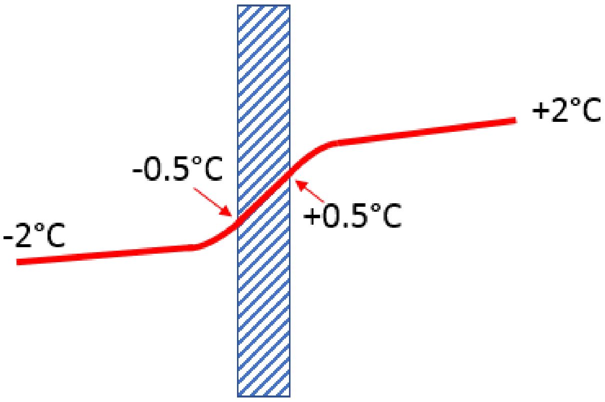
As mentioned, the heat transfer is composed of 3 parts – heat transfer on the refrigerant side (left), heat transfer through the wall (center) and heat transfer on the water side (right). Fouling is ignored in this example.

Apart from the above temperature difference related limitations, a couple of other issues needs to be addressed to ensure a safe and efficient operation of a heat exchanger.

Freezing of the brine/water is the biggest risk as it has the potential to destroy the heat exchanger. As water expand when it freezes, the forces released from this is sufficient to break the metal of the heat exchanger.

Considering an application where it is desired to cool water to +2°C using a refrigerant that evaporates at -2°C, the temperatures involved could as an example be as shown in the Figure 1.13 below.

Figure 1.13: Temperature profile across wall in heat exchanger with 4°C temperature difference

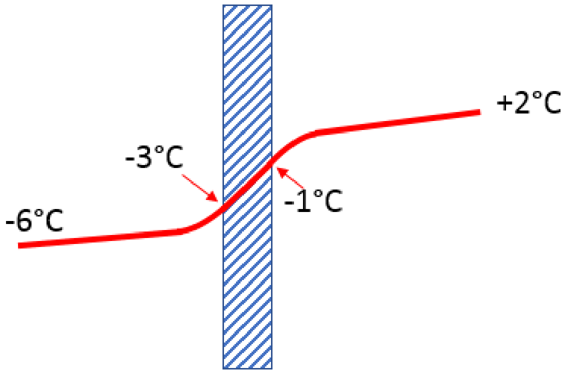


Any heat transfer needs a temperature difference as a driving force, so the wall temperature on the water side is lower than the water bulk temperature, the wall temperature on the refrigerant side is lower than on the water side and finally the refrigerant (evaporating) temperature is lower than the wall temperature on the refrigerant side. The actual temperature differences can be calculated from the heat transfer coefficient on the two fluid sides and the heat conductivity and wall thickness in the wall. The values given here are just to provide an example.

The all-important value is the wall temperature on the water side, which here is at +0.5°C – above freezing, but close.

It would be tempting to run with at lower evaporating temperature since the larger temperature difference would mean a smaller – and cheaper – heat exchanger. However, the lower evaporating temperature shifts the wall temperatures on both sides.

Figure 1.14: Temperature profile across wall in heat exchanger with 8°C temperature difference



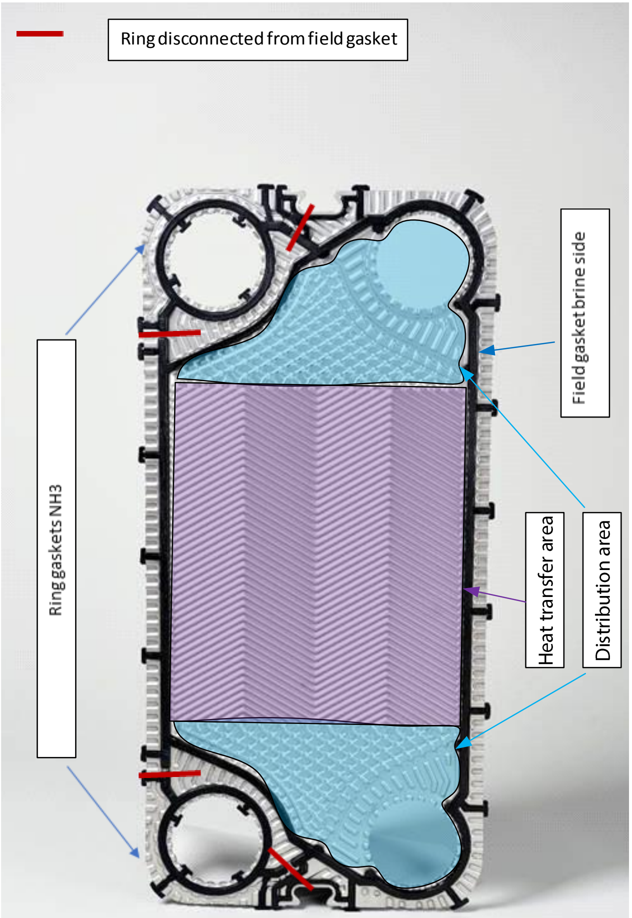
In the Figure 1.14 above, the temperature difference from evaporating to water outlet temperature has been doubled from 4°C to 8°C. With the assumption that the heat transfer coefficients are constant, this effectively doubles all temperature differences. Consequently, the wall temperature on the water side is now -1°C and the heat exchanger will freeze for sure.

1.3 Plate heat exchangers

Plate heat exchangers are built up by layers of metal plates. The metal plates are designed as corrugated plates to create small channels between each plate, which make plate heat exchangers very efficient compared to other heat exchanger types, since a larger surface area for heat transfer is obtained.

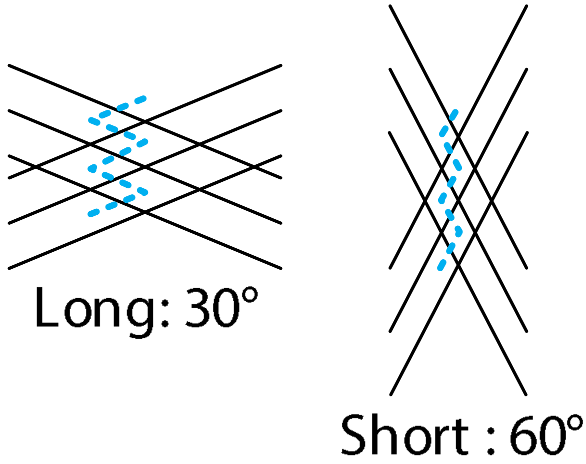
The channels on a plate is typically formed as fishbone shapes. The angle of the fishbone, the chevron angle or corrugation angle, is an important design parameter for a plate along with the depth of the corrugations. They determine the length of the path inside a channel that the fluid must travel. The channel and path lengths determine the pressure drop and the heat transfer capacity of the plate heat exchanger. A fish-bone plate is shown in Figure 1.15.

Figure 1.15: A fishbone plate for a plate heat exchanger



If the corrugation angle is small, the plate is referred to as a long plate, since the flow path is long, and a large corrugation angle yields a short flow path, and the plate is referred as a short plate. The flow path for a long and a short plate is shown in Figure 1.16.

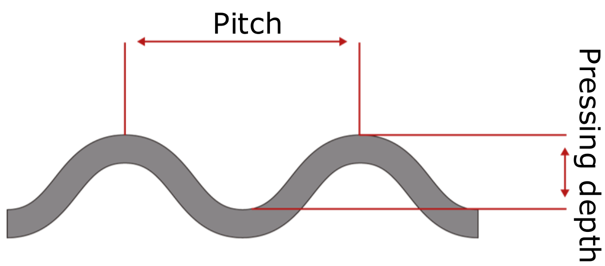
Figure 1.16: Flow path for a long and short plate



A long flow path on the long plate gives a high heat transfer rate with a high penalty of a high pressure drop due to the length of the path and changes of flow direction. A short flow path on a short plate gives a lower pressure drop but also a lower heat transfer rate.

The corrugations in the plate are made by cold pressing the plates. The pitch depends on the pressing depth. A high number of pitches will give narrower channels in the plate and a high turbulence. Wider channels will yield a lower turbulence. A plate corrugation principle sketch is shown in Figure 1.17.

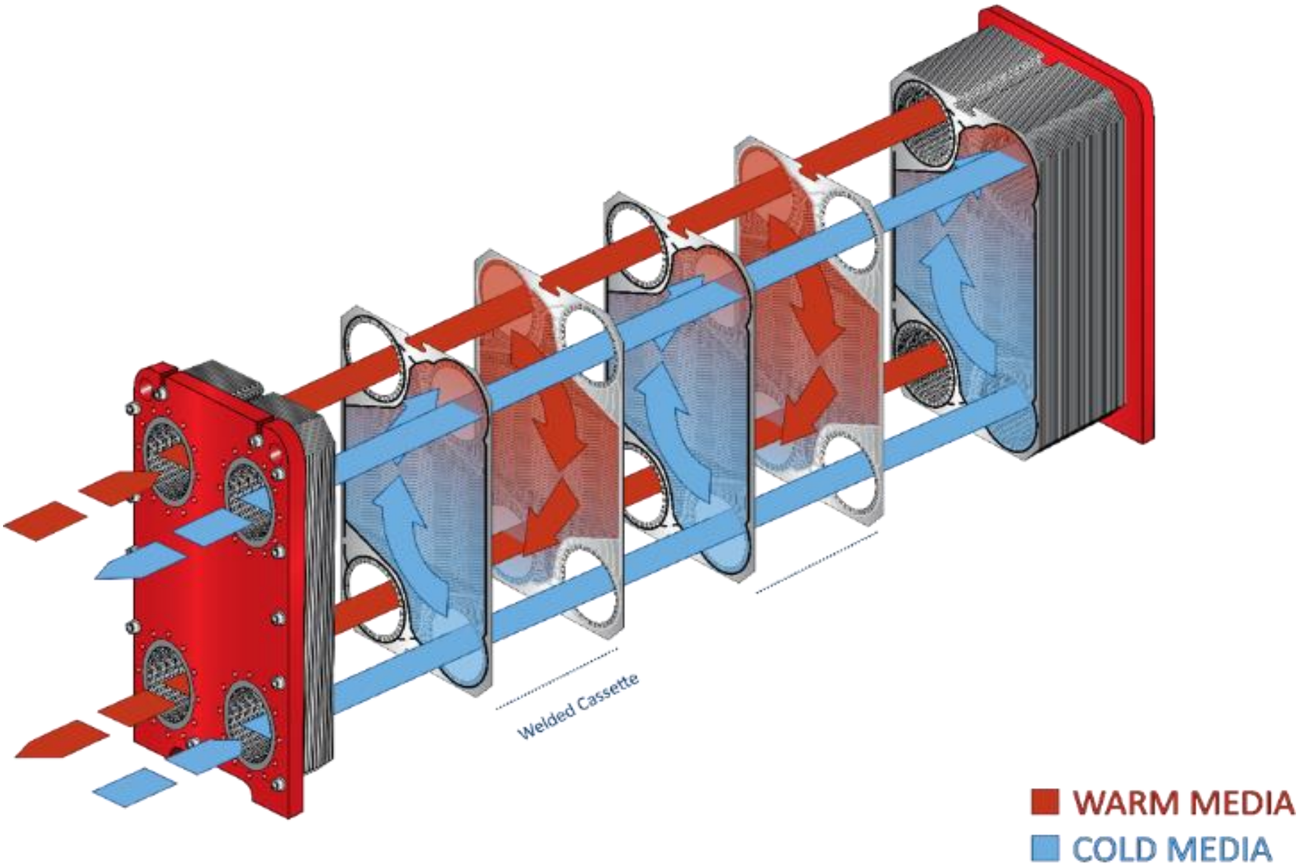
Figure 1.17: Plate corrugation



The plates have gaskets mounted at the outer rim to make the space for the channels between the plates and gaskets mounted on the plate to keep the fluids from mixing.

The plates are kept together in a rigid frame to form the flow channels with alternating hot and cold fluids as shown in Figure 1.18. The thick red front plate in the frame is called a 'Head' and the thick back plate is called a 'Follower'.

Figure 1.18: Working principle of a plate heat exchanger



The semi-welded plate heat exchanger shown in Figure 1.18 could be a condenser, as the high-pressure hot fluid is passed through the welded cassette.

The cassettes in a semi-welded plate heat exchanger can have various properties depending on which type of plates that are combined into the cassette. The combination of cassette types that are used in the plate heat exchanger will determine the properties of the PHE. Three types of cassettes are used in semi-welded plate heat exchangers:

- Long: Long cassettes consist of two long plates that are welded together, giving a high heat transfer rate and a high pressure loss.
- Mixed: Mixed cassettes consist of a long plate and a short plate that are welded together.
- Short: Short cassettes consist of two short plates that are welded together, giving a low pressure drop but also a low heat transfer rate.

It is possible to change the flow direction of a media in the plate heat exchanger by using a multi-pass solution. Changing the flow direction can be beneficial, e.g. for evaporators when running co-current in an evaporation section to kickstart the evaporation with a large temperature difference, while running counter-current in the superheating section.

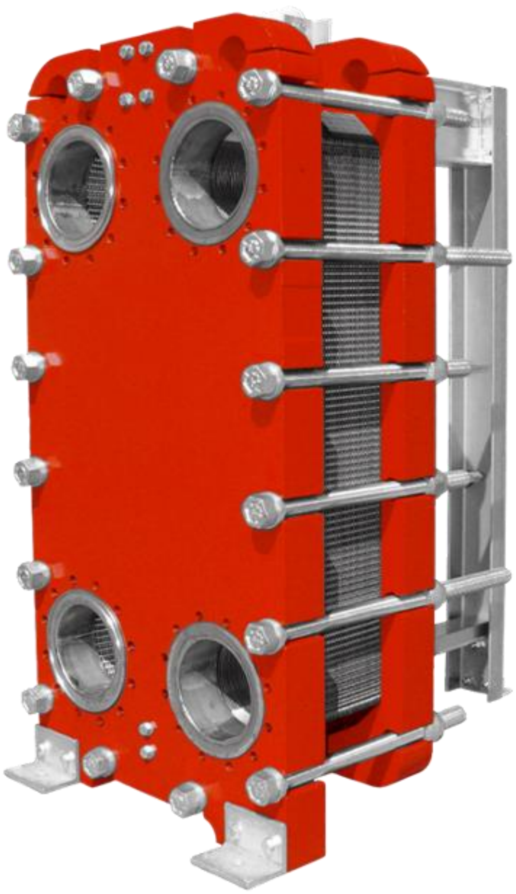
The plate materials for semi-welded plates are typically stainless steel or titanium, which supports ammonia, HFC's and brines.

The plate thicknesses available are 0.5 mm, 0.6 mm and 0.7 mm. The thickness of the plates determines the strength of the plate heat exchanger and the maximum operating pressure.

The last plate in the plate rack, the end plate, does not usually have connection ports where fluid runs through as the other plates. When doing evacuation of the system before start-up, one should be aware of the vacuum pressure that is applied to the plate heat exchanger. If the connection ports are large, a too low vacuum pressure can cause the area of the connection ports on the end plate to deform permanently. This affects the strength of the plate heat exchanger and can lead to unwanted and unsafe operation of the heat exchanger. One should pay attention to the vacuum pressure that is used to evacuate the system, when plate heat exchangers are installed. Typically, the thickness of the end plate will be as thick as possible, e.g. 0.7 mm.

An assembled semi-welded plate heat exchanger is shown in Figure 1.19. The plates are kept together by tie bolts that are bolted to the 'Head' plate and the 'Follower' plate. Further the plates are supported by a guiding bar in the bottom and a carrying bar at the top. The guiding bar and the carrying bar are bolted to the 'Head' plate and a support column behind the 'Follower' plate. This can all be seen in Figure 1.19.

Figure 1.19: Assembly of a semi-welded plate heat exchanger



Danfoss can provide plate heat exchangers for industrial refrigeration applications, including plate heat exchangers for evaporators and condensers.

▪ Fully gasketed plate heat exchangers: All plates are mounted with gaskets to enclose the heat transfer area on the plates. Fully gasketed PHE's are typically used with water or brine on both the hot and cold side. Pressure levels are typically below 25 bar and temperatures below 150°C.

▪ Semi-welded plate heat exchangers: Plates are made in cassettes. Two flow plates are laser-welded together as a cassette to create a sealed flow channel. On the other side of the plate, gaskets are mounted to seal the flow channel between each cassette. Semi-welded PHE's can usually handle higher pressures than fully gasketed PHE's – up to 40 bar (on the welded side) and temperatures below 180°C. Semi-welded PHE's are typically used for heat transfer between refrigerant and brine or refrigerant and refrigerant.

▪ Fully welded plate heat exchangers: All plates are welded together to seal the flow channels between them. Among fully welded plate heat exchangers there also exist brazed plate PHE, which is sealed by copper or nickel melted under vacuum and fusion welded PHE which is made of stainless-steel plates that are welded together by fusion welding.

1.4 Installation of heat exchangers

It is important to install and use the systems heat exchangers in a way that does not compromise their efficient and safe operation.

1.4.1 Evaporators

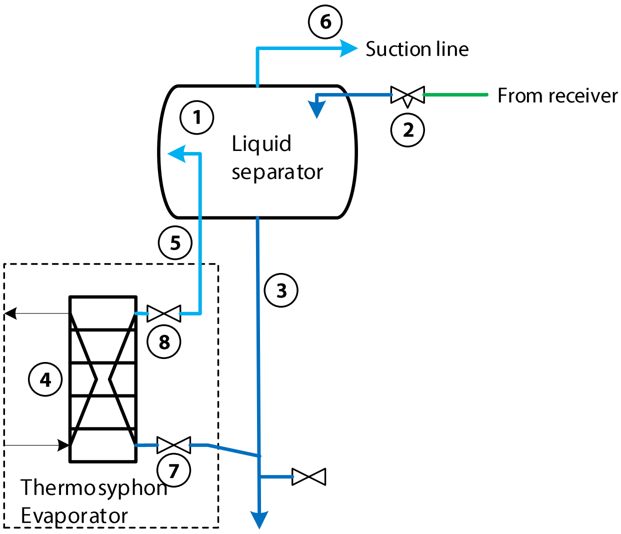
It is crucial that evaporators are installed in a correct way, so that pressure loss is minimized, liquid is fed properly and oil in the evaporator is handled to avoid fouling and loss of cooling capacity.

1.4.1.1 Thermosyphon Evaporators

Thermosyphon evaporators are the most critical component with regards to installation. Thermosyphon evaporators rely on a natural circulation to maintain a sufficient flow of refrigerant, and if the system does not support this, the performance will suffer.

A thermosyphon evaporator is shown in Figure 1.20 below.

Figure 1.20: Thermosyphon evaporator



High-pressure liquid is fed to the separator (1) through an expansion valve (2). Liquid settles at the bottom of the separator and is led to the evaporator (4) through the 'drop leg' (3). From the evaporator, the two-phase return flow is led through the 'wet return' (5) back to the separator where the liquid part is separated. After separation, the dry (liquid free) gas is led to the compressor through the suction nozzle (6). The evaporator has two SVA service valves (7) and (8).

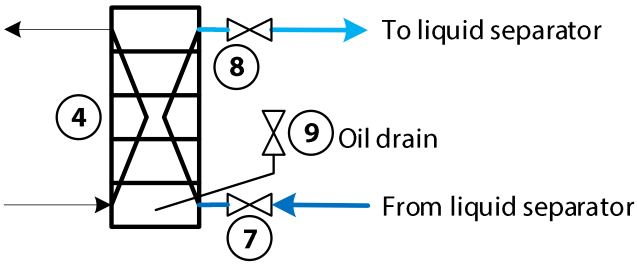
Oil drain

An oil drain/return valve is located at the bottom of the drop leg. It is important to ensure that oil does stay in the drop leg and does not enter the heat exchanger. If oil is present in the heat exchanger, it creates fouling which reduce the heat transfer or in the worst case, partially fill the heat exchanger with oil. The line from the drop leg to the evaporator must be with a slope up towards the heat exchanger and it needs to be designed to have a low speed to allow oil to drain back into the bottom of the drop pipe where it can be collected.

In some cases, an external vessel is attached at the bottom of the drop leg to collect oil. If automatic oil return is wanted, this vessel can be isolated from the separator by solenoid valves and the oil moved back to the compressor through the use of the compressors discharge pressure.

An alternative solution often seen is to provide a small pipe that collect oil from the bottom of the heat exchanger

Figure 1.21: Oil drain from evaporator



While this will drain oil from the evaporator, it is not recommended because it means that the oil is already in the evaporator. In the 'port' – the channel created by the holes in the plates – the flow can be rather violent, and the oil might not be available for collection before a large amount is present in the evaporator.

Circulation

The height difference between the level in the separator and the evaporator inlet is the driving force for the thermosyphon evaporator. The pressure difference between the separator and evaporator inlet is thus:

$$\Delta P = \rho * g * \Delta H$$

Where

- ΔP is the driving pressure [Pa]
- ρ is the liquid density [kg/m^3]
- g is the gravitational constant [m/s^2]
- ΔH is the height difference [m]

This driving pressure needs to be larger than the combined pressure loss in the evaporator, the wet return and the valves in the circuit.

It is recommended that the circuit is designed for a circulation rate of 3 since this provide a safety margin. The system will perform satisfactorily if the circulation rate drops to 2.5 because of uncertainties in the calculation, while if it is calculated at 1.2 and uncertainties give a circulation rate less than 1, performance will not be as expected.

The pressure loss in the evaporator is given from the design of the heat exchanger and is listed in the calculation results.

1.4.1.2 Pumped evaporators

For pumped evaporators – very often air coolers or plate freezers –some of the same considerations for thermosyphon evaporators should also be taken for pumped evaporators. The driving pressure, however, is secured by a pump. The design of the wet return still needs to be designed according to the riser rules, but rather than hindering circulation, the primary problem of a riser in a pumped system is the pressure loss. An excessive pressure loss will increase the evaporating temperature in the evaporator and thus reduce capacity.

1.4.1.3 DX evaporators

DX, or direct expansion, signifies that the expansion valve feeds the evaporator directly from the high-pressure side. The amount of refrigerant expanded to an evaporator is traditionally controlled by superheat. Usually a superheat of 5 to 10K is wanted to ensure that all refrigerant is evaporated, the failure to do so can result in compressor damage from either liquid hammer (attempt at compressing liquid) or bad lubrication from refrigerant in the oil.

Typically, DX evaporators are used in CO2 and HFC systems, primarily in air coolers, but also in chillers. In R717 systems DX has not been used to a great extent due to difficulties with control of the injection of refrigerant. However, in recent years the problem is being worked at and a solution is perhaps within reach.

The refrigerant injected into a DX evaporator is partially evaporated from the flashing process, so the speed in the evaporator channels is high from the start as opposed to a flooded evaporator where it is 100% liquid and the speed is low. That results in higher heat transfer in the first part of the evaporator, however since the evaporator needs to superheat the evaporated refrigerant it has another zone with low heat transfer (and lower temperature difference). All together the DX evaporator often needs a higher media to evaporating temperature difference to cope with the superheating zone which makes it less efficient than a flooded evaporator.

A special challenge in DX evaporators is to distribute the refrigerant fed to the evaporators since it is a mixture of gas and liquid. Depending on the type of evaporator, different types of distributors are available, but in general they have a relatively high-pressure loss, which in itself is not a problem as it does not affect the performance of the evaporator. However, the pressure loss reduces the pressure difference available to the expansion device and it needs to be accounted for in selection of valves.

1.4.2 Condensers

Condenser installation depend on the way injection to the low-pressure side is controlled. Note that the terms 'high-pressure float valve operation' and 'low-pressure float valve operation' refer to the mode of operation, not specifically the use of a float valve. Both operation modes can be achieved by using level switches/transmitters and a normal expansion valve.

1.4.2.1 High-pressure float valve operation

High-pressure float valve operation is expansion of the liquid immediately after the condenser. Any variation in the charge volume due to variations in capacity must be handled at the low-pressure side, e.g. in a liquid separator. Since the flow after the expansion valve is two-phase, it is not suitable for distribution to more than one location and thus it is primarily used in systems with only one low-pressure separator such as a chiller unit.

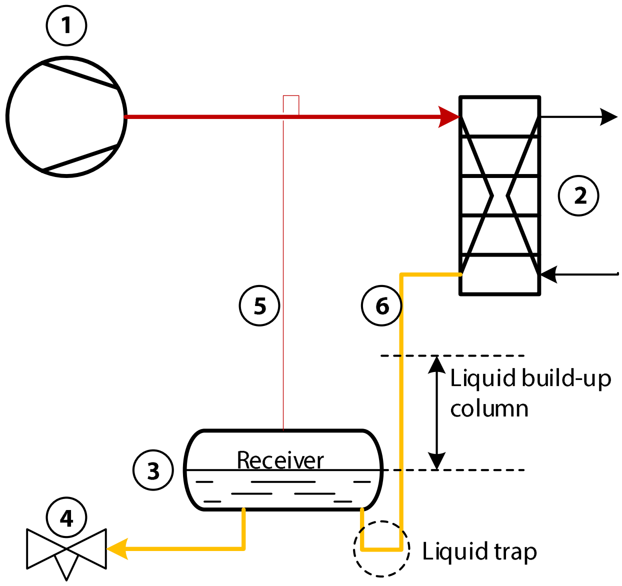
High-pressure float valves are usually mounted immediately after the condenser and as such pose no special problems with regards to condenser installation.

1.4.2.2 Low-pressure float valve operation

Low-pressure float valve operation controls the expansion of the condensed liquid to keep a given level in one or more low-pressure separators. Any variation of the charge volume due to variations in capacity must be handled on the high-pressure side, e.g. in a receiver.

To ensure a proper function of a condenser in a low-pressure float valve system, the installation must be correct.

Figure 1.22: Low-pressure float valve operation with connection to bottom of receiver

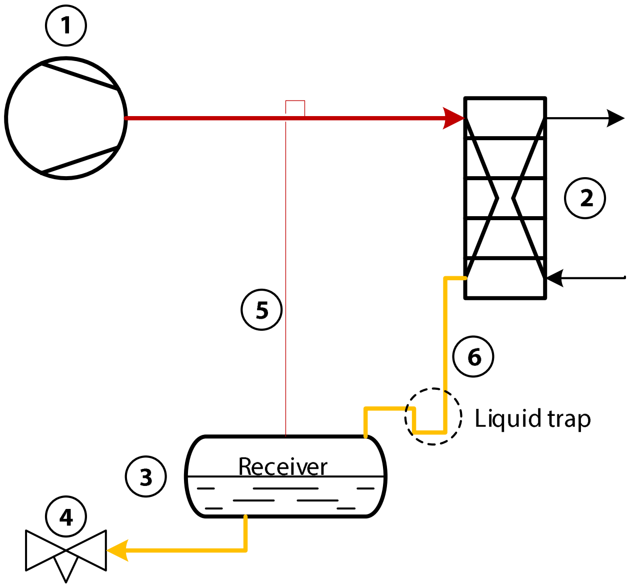


The compressor (1) deliver discharge gas to the condenser (2) that condense the gas into liquid, which is stored in the receiver (3) until a demand from the low-pressure side require re-filling through the expansion valve (4).

During times where the expansion valve is closed, the condenser will still produce liquid that needs to flow into the receiver. However, adding a certain volume of liquid requires the removal of the same volume of gas from the receiver. If not, the pressure in the condenser will increase to compress the gas in the receiver or the liquid will accumulate in the condenser. Both situations lead to an undesirable operation of the condenser (and compressor). To remove gas from the receiver, an equalization line (5) is added, allowing gas to pass back to the condenser and securing free flow of liquid into the receiver.

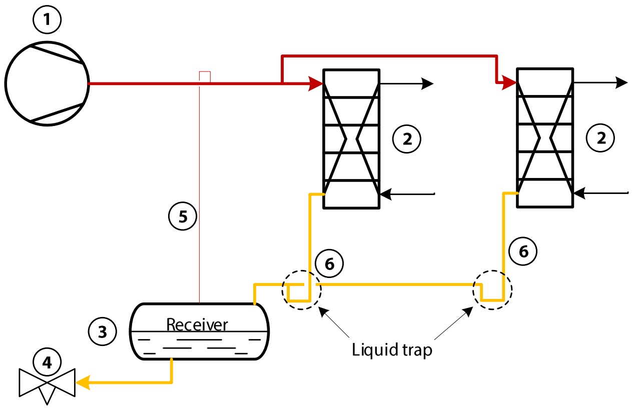
However, the installation of the equalization line short-circuits the condenser. The condenser will have a pressure loss and therefore, the pressure in the outlet of the condenser will be lower than the pressure in the receiver, restricting the flow to the receiver. To counter this, a drop leg (6) on the outlet of the condenser must be used. The drop leg must have a liquid trap at the bottom to allow liquid to build up in the drop leg. This liquid build-up will provide a positive pressure that counters the pressure drop in the condenser. The height of the drop leg must be larger than the pressure loss in the condenser, expressed in meters of liquid. In the sketch above in Figure 1.22, the liquid trap is secured through entering the receiver from the bottom, using the liquid level in the receiver as the top level. An alternative solution is to enter the receiver through the top while having an external liquid trap, as sketched below.

Figure 1.23: Low-pressure float valve operation with connection to top of receiver



Especially in the case of multiple condensers or multiple circuits in a single condenser, it is essential that these guide-lines are followed.

Figure 1.24: Low-pressure float valve operation with parallel connected condensers



Note that the condenser outlets are not collected until after the liquid trap. This allows the condensers to even out their pressure loss individually. If the two condensers are of a different size/construction or one is more fouled than the other, they will have a different pressure loss. Connecting these at outlet level will result in liquid accumulating in the one with the lowest pressure loss, reducing capacity.

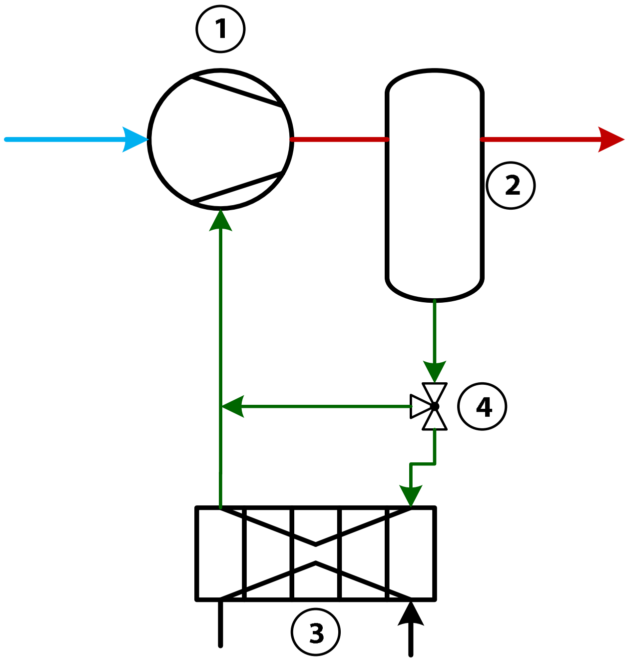
Note that the use of individual drop legs is also necessary for high-pressure float valve operation in the case of multiple condensers/circuits.

1.4.3 Oil coolers

The oil coolers described in this section are either brine/water or refrigerant cooled. Air cooled oil coolers are shown in chapter # (OIL SYSTEMS).

Brine/water cooled oil coolers often deliver the oil cooling load to a heat recovery system, but it is also common to disperse the oil cooling load in a separate air-cooled cooling tower. Brine/water cooled oil coolers pose no special problems to install.

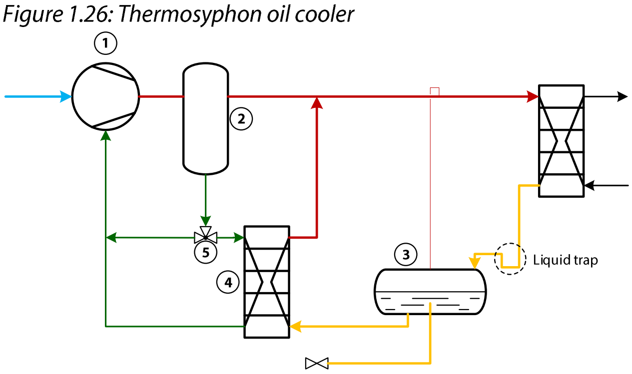
Figure 1.25: Brine/water cooled oil cooler



The (screw) compressors (1) discharge gas contains oil, which is separated in the oil separator (2). This oil is led to the oil cooler (3) where it is cooled by a water/glycol flow. A 3-way valve (4) regulate the temperature of the oil passed to the compressor as the capacity of the oil cooler might be too high with varying compressor loads or condensing temperatures.

Refrigerant cooled oil coolers are often referred to as 'thermosyphon oil coolers' as the evaporative circuit employs the thermosyphon principle.

Figure 1.26: Thermosyphon oil cooler



The oil circuit is the same as for the brine/water cooled oil coolers. The cooler is cooled by evaporation of liquid from the condenser. This liquid is evaporated at condensing pressure/temperature and passed back to the compressors discharge line to be re-condensed in the condenser. Alternatively, the oil cooler can return the partly evaporated refrigerant to the receiver, however this requires the pressure equalization line between the receiver and the compressors discharge line to be dimensioned to take the gas flow.

The evaporating side of these thermosyphon oil coolers need the same considerations as a thermosyphon evaporator. The driving force is a height difference from the liquid level in the receiver and the pressure losses are again the pressure loss in the evaporator (oil cooler) and the wet return line. As before the wet return line needs to be calculated as a riser.

Note that it is necessary to provide the oil cooler with a 'priority'. If the low-pressure system is demanding liquid and no priority is supplied, then the oil cooler might run dry which results in no oil cooling. No oil cooling is naturally a problem for the compressor. Usually a priority volume of refrigerant liquid is calculated from what the oil cooler evaporates in 5 to 7 minutes. The priority can be secured by having the receiver outlet for the low-pressure system raised above the bottom of the receiver, or by adding a separate vessel that is filled by the condenser before the receiver gets any liquid as seen in Figure 1.26.

Thermosyphon oil coolers are rarely seen on high-pressure float valve systems since there is no receiver to supply liquid. It is possible to mount a priority vessel between the condenser and the expansion valve, however careful considerations need to be made to counter the pressure loss in the condenser, since the oil cooler needs to return the evaporated refrigerant back to the discharge line, which is at a higher pressure (corresponding to the pressure loss in the condenser) than the liquid before the expansion valve. A drop leg similar to the low-pressure float valve installation is necessary and on top of that there must be a driving height to drive the thermosyphon function of the oil cooler.

1.4.4 Desuperheaters

The primary consideration of a desuperheater is to have as little pressure loss as possible, as this increase the compressors discharge pressure which result in an increased power consumption.

A secondary consideration is to avoid condensation in the desuperheater. If the water/brine heated in the desuperheater is colder than the condensing temperature it is possible to condense some of the refrigerant in the desuperheater. It is recommended to avoid running with such low water temperatures. If it is not possible, then the desuperheater should be designed in such a way that condensed liquid can drain from the heat exchanger and not be built up inside. Further the piping from the desuperheater should be sloping down towards the condenser, hindering the condensed refrigerant from settling in undesirable places.

1.4.5 Subcoolers

Similar to the desuperheater, a subcooler is not critical with regards to installation. The primary consideration is to avoid having too large a pressure loss on the refrigerant side. If the pressure loss (in °C) exceeds the subcooling (in °C) provided by the subcooler, the refrigerant will 'flash' causing problems for expansion valves downstream.

1.4.6 Economizers

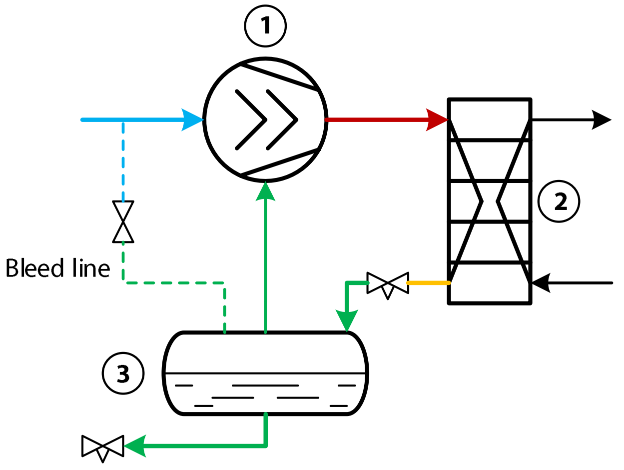
Economizers can be of the 'open' or 'closed' type. For both types the pressure in these are a balance between the capacity of the economizer and how much the compressors economizer port can absorb.

The economizer port is located physically in the screw compressor after the compression has (just) started. As such the pressure between the screw compressors rotors is higher than suction pressure and to be able to feed additional gas in between the rotors, the pressure in the economizer port needs to be higher still. The higher the economizer port pressure, the more gas can be absorbed by the port. On the other hand, the port pressure equates to a saturation (evaporating) temperature in the economizer, which again with a higher pressure mean a lower capacity.

In many cases the economizer pressure is controlled by a suction pressure regulation to be above what the economizer port dictate. It is naturally not possible to go below what the economizer port dictate.

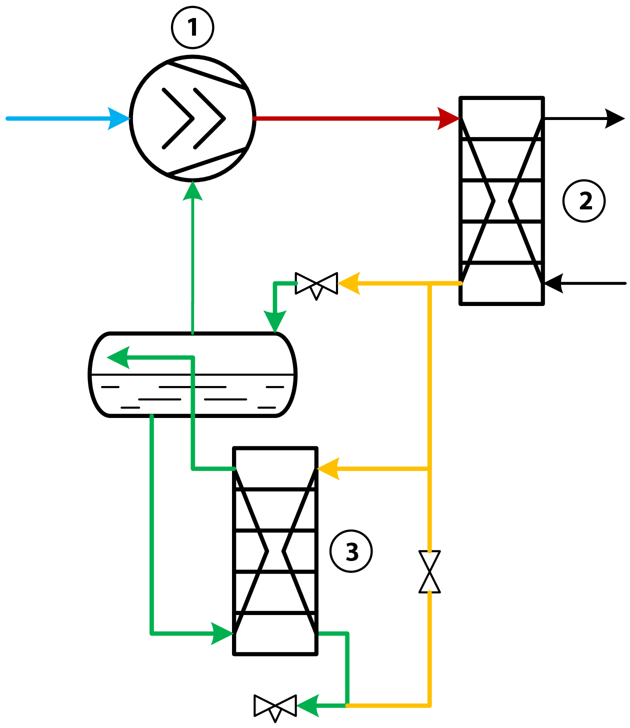
The simplest form of economizer is the 'open' economizer, which is a vessel (3), whereto the condenser (2) liquid is flashed from condensing pressure to economizer pressure. The gas developed by this flashing is passed to the economizer port. The liquid, at economizer temperature, is flashed another time to the low-pressure side. The benefit of this type is that it is a very simple vessel, the temperature of the liquid to the low-pressure side is the lowest possible, but the down side is that the liquid to the low-pressure side is at a lower pressure, requiring a larger expansion valve.

Figure 1.27: Open economizer with bleed line



In contrast, the 'closed' economizer is a heat exchanger (3) that cools the condenser (2) liquid at condensing pressure. To perform the cooling duty, a part of the condenser liquid is flashed to economizer pressure, evaporated and passed to the compressors (1) economizer port. In the below sketch, the economizer heat exchanger is a thermosyphon evaporator, but a DX version can be used if suitable.

Figure 1.28: Closed economizer



The benefit of the closed economizer is that the liquid to the low-pressure side is at condensing pressure and thus the expansion devices can be smaller than with an open economizer. The down side is that the liquid temperature cannot reach the temperatures achieved with an open economizer, since a temperature difference needs to exist in the heat exchanger. Because of this, the closed economizer is a little less efficient than an open type.

One important thing to know about screw compressor economizer ports is that if the compressor is capacity regulated using slide valves, the economizer port will be exposed to the suction side very shortly after the compressor regulate down from 100%. This has two important implications. The economizer pressure will drop to suction pressure, which is very bad for the open economizer, since the pressure difference between the economizer vessel and the suction side is zero. This means that no flow will pass through the expansion devices from the economizer to the low-pressure side. The closed economizer will have an increased capacity. Further the economizer will load the suction port, meaning that there is less flow available for actual evaporation on the low-pressure side.

Naturally these problems don't occur if the compressor is VFD (variable frequency drive) controlled only. Usually economizers are fitted with valves in the economizer suction line to control the pressure, to start/stop the economizer and to avoid the above problems. In some instances, it is not desirable to use all the capacity available on the economizer port and thus a simple suction pressure regulation is employed. With open economizers on slide valve-controlled systems, it is necessary to control the minimum pressure in the economizer to ensure a minimum pressure difference for the expansion valves. This can be implementing through a minimum pressure, assuming suction pressure does not rise above a certain value, or a pressure difference to the suction pressure.

Finally, it can be desirable to start and stop the economizer according to the operation of the system. The closed economizer is easily stopped with a solenoid in the economizer suction line. It is necessary to consider the pressure loss in the heat exchanger compared to the (anticipated) subcooling from the condenser to avoid flash gas before the expansion valve.

A simple solenoid in the economizer suction line is not appropriate for an open economizer. The entire refrigerant flow is flashed to the economizer vessel and when the pressure is below condensing pressure, the gas developed during the flashing will increase the pressure until it is at or just below the condensing pressure. This makes it impossible for the expansion valve between the condenser and economizer to deliver enough flow. Therefore, a separate bleed line is needed to dispense of the flash gas to the suction line. The bleed line must contain valves that secure an appropriate economizer pressure for both expansion valves to function. Compared to not having an economizer, the total impact of this installation on the compressor will be zero. The amount of gas flashed to the suction line is countered by a lower temperature of the liquid passed to the evaporators.

However, if a side load is run in parallel with the economizer, it is either not possible or ineffective to stop the economizer. Stopping a closed economizer will stop the side load as well, while with an open economizer, stopping it will move the side load to the suction side, which is much less efficient.

The screw compressors rotor 'lobes' pass the economizer port while shifting from one 'channel' to another within the compressor. This results in oscillations in the economizer pressure, which can result in undesirable effects such a valve oscillation. To counter this a damper can be fitted, that reduce these oscillations and thereby reduce the strain put on the economizer system. This is described in chapter (COMPRESSOR CONTROLS) in section (ECO DAMPER) regarding economizer line pulsation damping units.

1.4.7 Superheaters

Superheaters use cooling of the condenser liquid to heat the suction gas for the compressor to provide security of dry suction gas.

Like subcoolers and desuperheaters, the pressure loss in these heat exchangers are important for system efficiency, but apart from that the installation pose few problems.

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