Oil, just for lubrication! (part 1)

by:
Titus M.C. Bartholomeus,
Senior Development Engineer,
Grasso Products b.v.

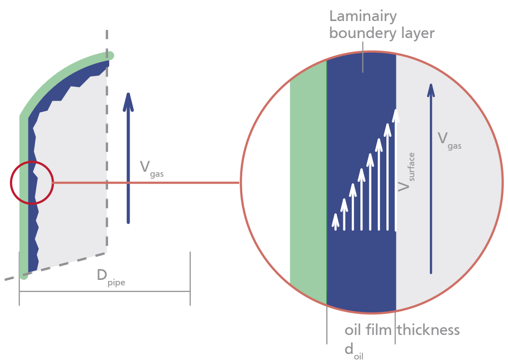
Introduction

Apart from the occasional special application, oil is used to lubricate all refrigeration compressors. An annoying side effect of this is that oil residues enter the plant with the compressed gas, in vapour and liquid form. This oil will contaminate the heat exchanger present, with a negative impact on performance and operation. This two-part article presents a few ideas on how simple installation and/or adjustment of components can significantly reduce the oil concentration and therefore increase the operation and performance of the refrigeration plant.

If oil gets into the heat exchangers, it will form a deposit there. The surface that is contaminated and the thickness of the layer depend on the local flow pattern.

- With an air-cooled or evaporative condenser, the liquid and therefore also the oil will flow over the bottom part of the condensor pipes. Because the oil is relatively warm here and its viscosity is therefore still acceptable, the thickness of the oil layer will remain limited. The larger part of the circumference can still be used for condensation, so that the liquid oil contamination does not play an important part. Any oil vapour present, practically only with ammonia, will condense on the free surface just like the refrigerant vapour, and fill in the roughness of the pipe wall. This brings the capacity loss to about 1.5%/µm.In other words, with the commonly selected condensers the oil vapour film results in an increase of condensation temperature of over 2K. That means a C.O.P. reduction of more than 4%, so it is important to keep the oil vapour content low.
- With evaporation in pipes, it is the pipe section moistened with evaporating refrigerant that mainly determines the total heat transfer. The cooler load is preferably chosen so that ring flow occurs.

Figure 1. oil film built up in riser



The following figure explains the theory of how the oil film builds up with ring flow, based on the boundary layer theory (see figure 1).

1. $v_{surface} = \delta_{oil} \cdot \dfrac{\tau}{\eta_{oil}}$

2. $v_{oil} = \dfrac{v_{surface}}{2}$

3. $\tau = \dfrac{\lambda_{pipe}}{4} \cdot \dfrac{\rho_{gas}}{2} \cdot \left(v_{gas}\right)^2$

4. $ppm_{oil} = \dfrac{m_{oil}}{m_{gas}} \cdot 10^6$

5. $m_{oil} = \pi \cdot d_{pipe} \cdot \delta_{oil} \cdot v_{oil} \cdot \rho_{oil}$

6. $m_{gas} = \dfrac{\pi}{4} \cdot \left(d_{pipe}\right)^2 \cdot v_{gas} \cdot \rho_{gas}$

δ_{oil}	oil film thickness (m)
η_{oil}	dynamic viscosity (Pa.s)
τ	Shear stress (N/m²)
d_{pipe}	pipe diameter (m)
v_{oil}	average oilvelocity in boundery layer(m/s)
λ_{pipe}	flow resistance values
ρ_{gas} , ρ_{oil}	specific weight of gas resp. oil (kg/m³)
v_{gas}	gas velocity (m/s)
ppm_{oil}	massa concentration of oil in gas
m_{oil} , m_{gas}	massa flow oil resp. gas (kg/s)

We can calculate the thickness of the oil film by reducing formulas 1 to 6:

$$\delta_{oil} = 2 \cdot \left(\frac{ppm \cdot \upsilon_{oil} \cdot d_{pipe}}{\lambda_{pipe} \cdot v_{gas}}\right)^{0,5}$$

δ_{oil}	oil film thickness (µm)
υ_{oil}	oil viscosity (cSt)

Example:

what is the average oil film thickness in a DX evaporator pipe, at an oil concentration of 100ppm? Internal pipe diameter 19 mm, λ_{pipe}=0.035 (typical for industrial pipes), evaporation temperature -10°C, refrigerant NH_3, viscosity of mineral 68 oil 1000cSt. Ring speed halfway, so at X=0.5 (using Flowchart Hbb4 from VDI Waermeatlas) it is 13 m/s. (Internal heat transfer about 3000W/m²K) This puts the calculated thickness of the oil film at 18µm and reduces the internal heat transfer by a good 25%.

If an evaporator is not laid out in accordance with the flowchart, a layered flow can occur. Because only the bottom pipe section is moistened in this case and the oil also settles there, the impact of oil contamination obviously becomes disproportionately large. Reducing the proportion of oil in the evaporator reduces the layer of oil on the lowest section of the pipe. The evaporating refrigerant flows in this part and is now offered more heat exchange surface. Now more refrigerant will evaporate, resulting in an increase in the mass flow speed. The higher speed will ensure that the thickness of the oil layer decreases even further, etc. A 50% reduction in oil concentration results in an oil film that is 30% thinner; however, the previously described phenomenon of the increase in the mass flow speed will make the film thickness smaller and therefore the influence of contamination decrease by more than 50%. With common evaporators this means an increase in evaporative temperature of at least 2K and along with that, a C.O.P increase of 8%. The open literature (including ASHRAE) mentions much higher C.O.P. losses due to oil concentration.

This problem also occurs in plate heat exchangers. With plate exchangers the higher load and the unsettled flow character in every channel do make the final oil layer thickness smaller, but it is precisely that higher load that makes the influence of the thinner oil layer on the total heat exchange at least equal to that of the other exchangers, and therefore not to be disregarded.

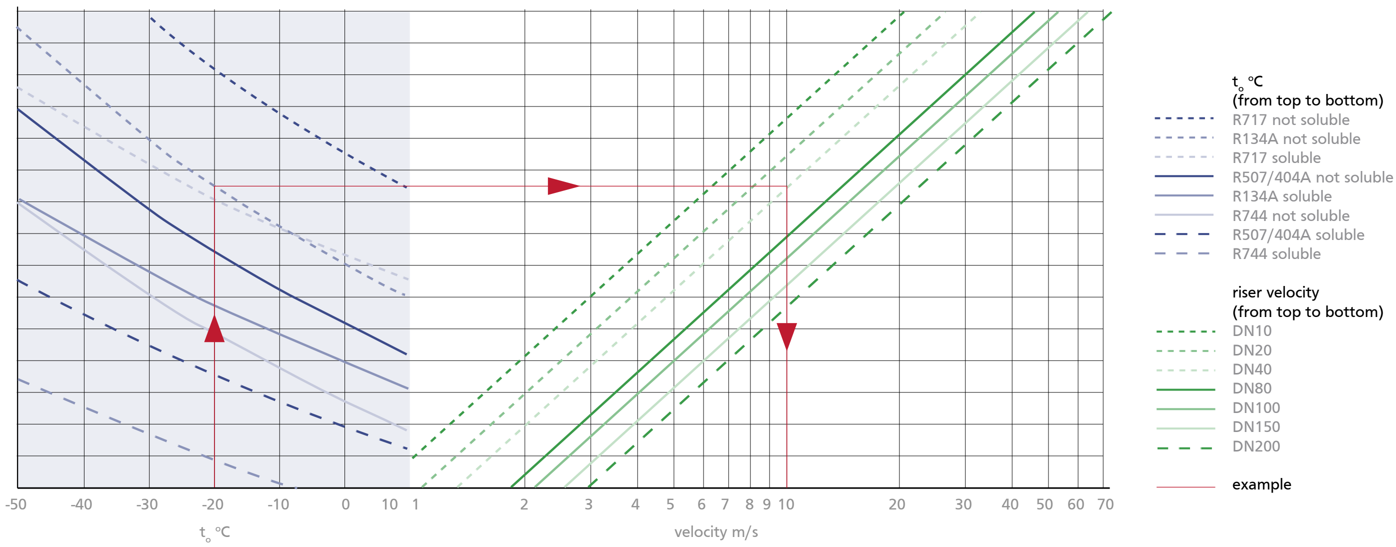
How can we reduce the contaminating influence of the lubricating oil in the refrigeration plant?

1. By choosing lubrication oil that dissolves totally in the refrigerant in combination with a properly functioning oil return.
2. And with insoluble oil:

- by choosing a compressor with low oil consumption*,
- by using a high-efficiency oil separator,
- by using a compressed gas cooler for the oil separator, at high oil vapour content and/or
- by using an oil with low vapour pressure and high mol mass*,
- by arranging the liquid tank so that oil can settle and be returned to the compressor(s); for 2-stage plants this also applies to the intermediate tank(s),
- by regularly draining off the oil, automatically or otherwise.

** See also article Oliemanagement voor zuigercompressoren [Oil management for piston compressors], Koude&Luchtbehandeling volume 95, no. 11 2002*

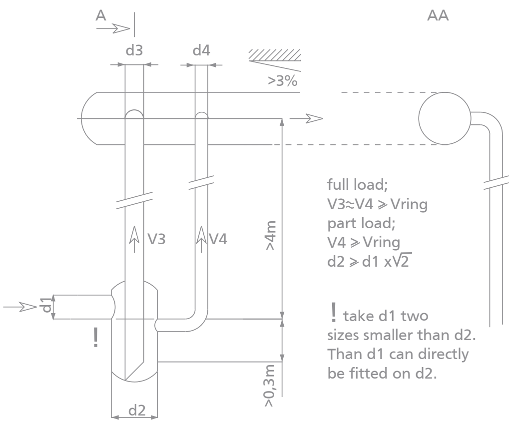
Figure 2.1 Minimum riser velocity



1. Oil return for oil soluble in refrigerant

The design of direct expansion (DX) systems is based on oil returning via the suction line(s). This works on the principle that the speed (impulse) of the gas is so high that the liquid present in a film along the pipe wall is dragged along as well. This is at the expense of pressure loss and therefore energy. If you step down far enough in part load it is not possible use this flow phenomenon without adverse effects, as it will result in unacceptably high losses with full load. That is why syphons and double riser pipes are used in these situations (see figure 2.2).

Figure 2.2 Double riser



At low gas speeds the syphon fills up, resulting in the passage above the oil bath decreasing and the speed increasing. The locally higher speed ensures that a gulp of oil is swept along as well. This principle will only work if the height to be covered is not too great, for instance at the exit of evaporators or at a U-pipe in knock-out drums. If greater heights must be tackled, you use double riser pipes whereby the filled syphon shuts off the thicker riser pipe. The speed in the thinner pipe can then increase sufficiently so that ring flow arises again. The gas stream will take along liquid refrigerant and fully soluble oil if:

- a full load pressure loss of 0.01K/m is selected,
- the horizontal pipes are laid sloping (>3%) and
- double riser pipes are provided for part load.

For partially mixable and non-mixable oils this will function above the evaporation temperatures of -5/-10^{o} C in the case of R134A and R717 and -20^{o} C in the case of R507/404A only if we allow a full load pressure loss of 0.02K/m.

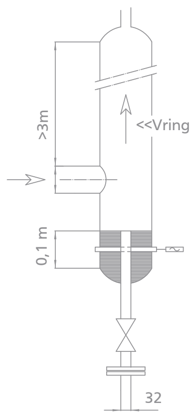
If the piping is not laid in an optimal design and the oil remains behind during part load, oil will often be refilled unthinkingly to keep the sumps at the correct level. At full load a great deal more oil will return than will fit in the sumps, resulting in enormous damage due to oil surge. Piping is not adapted because of time pressure or simply because it is too expensive, the plant is set to full load periodically for a certain time as a makeshift measure. This does not exclude overfilling the sumps and is undesirable for refrigeration processes that are sensitive to temperature. One solution is building in a high-efficiency oil separator.
The residual viscosity of the oil that comes out of the evaporator depends on the degree to which the oil in question has dissolved and the concentration of oil in the plant.

Example:

*Oil consumption compressor(s) 100 ppm. Vapour content after expansion valve X=0.2, at that level the oil concentration in the liquid increases to 100/(1-0.2)=125 ppm. There is usually still 1% liquid at the end of the evaporator, so its oil concentration is (0.8/0.01)*125 = 10,000 ppm = 1%. Even with a liquid concentration at the condenser exit of 1 percent, the residual viscosity of the mixture is still so low that the gas can easily take the mixture along.*

The ring flow graph (see figure 2.1) shows the minimum speed suitable for the refrigerant, evaporation temperature and pipe diameter for riser pipes for insoluble and soluble oils. Horizontal pipes must be laid with a fall of >3%. At low evaporation temperatures the pressure losses necessary to have the gas bring the oil back are too high, and other solutions will have to be found. One possibility is connecting all evaporator exits to a header pipe at a lower level and join it to an oversized riser pipe (see figure 3). The horizontal header pipe is lower so that oil is prevented from flowing back into disconnected coolers. By laying the first part (about 3 m) of this riser pipe at a full load speed that is well below the ring speed, all the oil will be collected in the dome. The oil return system mounted beneath it can then pump the oil evaporated out back to the compressor sump.

Figure 3
Liquid separating riser

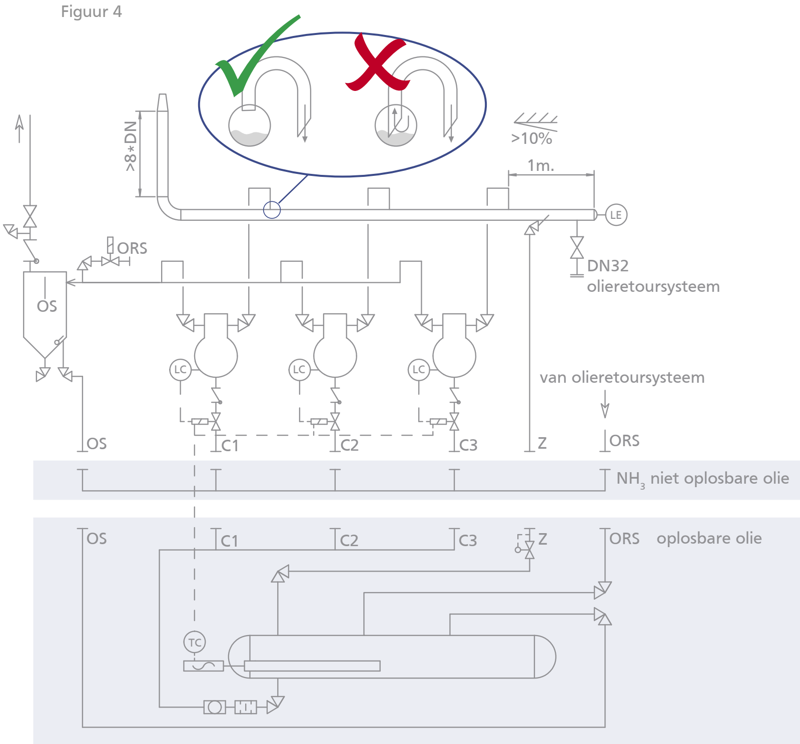


Like the oil return with DX via the suction system, with pump circulation and bath evaporative (FX) systems the exit of the oil return system (see figures 9.0 to 9.3 in part 2 of this article in the next issue) is also usually connected to the suction line.

In both systems the oil does not return in the same measured amount as ejected by the compressors, but in drops. These drops arise through the effects of accumulation in the plant components and pipes. If the compressor does not have a liquid separating suction section, the liquid will hit the suction valves. Valves are designed for high gas speeds of about 0.3xMach (>300km/h), which makes it clear how disastrous the influence of entering liquid can be. With hermetic compressors the pressure vessel is made as a knock-out drum. With semi-hermetic compressors the quickly turning motor flings the entering liquid out of the gas stream. With commercial and light industrial compressors, the suction section includes a knock-out drum, or it is sucked through the sump. Industrial compressors often don't have this kind of solution due to low oil consumption and optimal heat dissipation (ammonia), and the size of the suction section determines whether part of the liquid sucked in is still separated. With an ammonia compressor the heat dissipation and the oil consumption are so important that the compression part is kept as far as possible out of the housing and the suction chamber is kept as small as possible. Excellent heat dissipation therefore goes at the expense of sturdiness against the incidental entrainment of liquid. With this kind of compressor the oil evaporated out is returned to the sump. The gas released through the evaporation has to be returned via the suction and not via the sump, because that would result in the sump pressure increasing, with a significant increase in oil consumption as a consequence. How the oil should be returned for this type of compressor is shown in the sketches (see figure 4) for insoluble and soluble oils.

A buffer tank must always be used for soluble oils. This tank catches the variable quantities of oil in the plant and ensures that the oil can evaporate out of the oil separator. Latter is because the oil in the oil separator contains a lot of refrigerant due to the high pressure and relatively low temperature. The compressed gas temperature of the current "freons" is too low to heat the oil bath sufficiently. Electrical heating is not an option as the power would have to be disproportionately high to transfer even a little heat to the stationary oil bath and to compensate the heat dissipation of the relatively large oil separator. The suction header pipe is completely different to that customary with DX plants. It must be as large as possible so that a layered flow occurs (take the speed for the application in question from the vertical riser pipe). The incoming oil will deposit and due to the 10% slope, gather at the end. An oil return system built beneath will then send the oil back to the individual sumps on an as-need basis. To prevent oil surge if the oil rectifier malfunctions, a maximum level alarm must be installed at about 35 mm height from the pipe base, that will stop the compressors.

Figuur 4



Oil return for oil that is not soluble in refrigerant

As mentioned above, it is important with insoluble oils to choose a compressor or compressor/oil separator combination with a low oil consumption. Small systems like chillers combined with a compressor with low oil consumption can usually do without the oil separator. For larger systems, the contamination from the oil to be returned from the evaporative system is so great (fly rust, welding slag, blasting dust, burnt oil and other residues as well as water) that the compressors would be damaged and oil return is therefore not permissible. The oil consumption must be kept as small as possible in these systems. Grasso Products has designed a range of oil separators for this purpose that reduces the transmitted portion of liquid many times over. Normal performance of the oil separators on the market that are based on gravity (so without coalescing elements) is around 80%, while the current Grasso OS series has a performance of about 95% (see accompanying performance graphs figures 5.1 and 5.2). In other words, 20/5=4 times less oil is let through. This makes the exiting liquid portion so small that in fact only the amount of oil vapour is of any importance in the oil consumption. The selection is based on the mass flow density (kg/m²s) obtained over the entire internal diameter, corrected with a refrigerant-related correction factor Crefr. The graphs show that the lighter gases retain their high performance over a larger area. By improving the internal flow, the load can be increased greatly, bringing equal performance for the lighter gases and improved performance for the heavier gases.

Part 2 of this article will be published in the next issue.

Figure 5.1

efficiencies of momentairy OS and new development

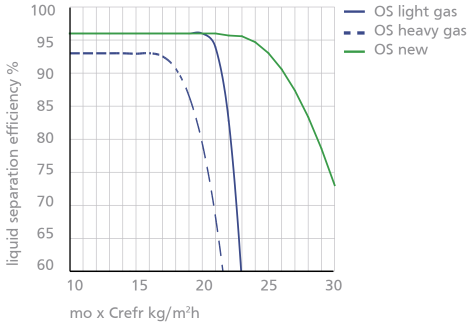
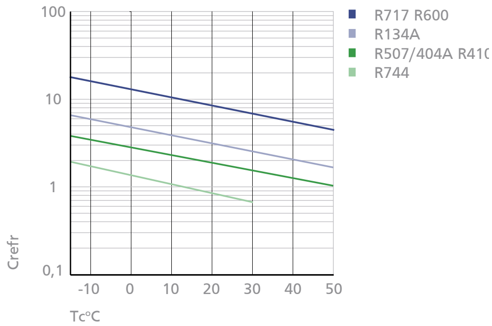


Figure 5.2



• Grasso Products b.v.
Jan-Pieter Habraken
Tel: 073 6203 845
jphabraken@grasso.nl

You can find all the articles on the Grasso website: www.grasso-global.com > News&Events > Latest News.

Grasso 1212(E) package