FRIGI-TECH INTERNATIONAL

Understanding Oil Return in Refrigeration Systems

Part I: Flooded Evaporators

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Introduction

With few exceptions, all compressors that are lubricated with oil will discharge oil into the gas stream. The rate of discharge can be as small as parts of oil per million parts of refrigerant for direct drive hermetic centrifugal compressors and as much as several percent for screw compressors. Oil discharge rates are usually expressed in terms of lbm of oil discharged per lbm of refrigerant compressed or in mass percent of oil in the discharge gas.

Oil in compressor discharge gas is in two forms: fine oil droplets (mist) in the gas stream; and liquid oil driven by the gas velocity, crawling along the tube walls. Oil flows from the compressor with the discharge gas through the oil separator (if equipped and always less than 100% efficient), and into the condenser. The liquid leaving the condenser consists mostly of refrigerant with some amount of dissolved oil (assuming that the oil is miscible in the refrigerant). The oil content in the liquid refrigerant at this point is the same as the oil discharge rate of the compressor/separator.

The liquid oil-containing refrigerant flows through the expansion valve and into the evaporator. In the evaporator, the refrigerant boils off delivering its refrigerating effect. The oil, however, does not evaporate as its boiling temperature is very high relative to the temperatures existing in the evaporator. In the absence of an oil return system, oil will continue to collect and concentrate in the evaporator which will lead to two negative consequences: heat transfer in the evaporator will be progressively degraded and the compressor will eventually run out of oil shutting it down. Hence, an effective oil return system is essential.

Refrigerant and Oil Mass Flow Balance in a Flooded Evaporator

Consider the evaporator of an operating water chiller. Oil is arriving at a certain rate, specifically: the oil discharge rate of the compressor less the removal rate of the oil separator, if equipped. For illustration purposes, assume the mass arrival rate in the evaporator to be 2 lb of oil along with 1000 lb of refrigerant liquid in one hour. The compressor/separator has an oil discharge rate of 0.2%, i.e. mass of oil per mass of refrigerant compressed expressed as a percent. This would be a good discharge rate for a screw compressor/separator.

Oil is also leaving the evaporator via the oil return system. The amount of oil leaving via the oil return system is a function of the liquid removal rate and the concentration of oil in that liquid. Let us assume that the oil return system draws 50 lbs of refrigerant/oil mixture from the evaporator per hour. If the concentration of oil in the evaporator liquid is say 2%, then the oil returned is 1 lb per hour. Since this leaving rate is less than the arrival rate, oil will further accumulate in the evaporator and the oil concentration will rise. Under the conditions stated above the oil concentration in the evaporator will rise to and stabilize at 4%.

Four percent is unacceptably high. There are two things we can do to reduce this concentration. The first is that we can increase the oil return liquid withdrawal rate. If we double the oil return flow rate to 100 lbs/hr and the oil concentration is 2%, the oil arrival and removal rates will be equal at 2 lbs/hr and the concentration will be stable at 2%. Or, we can decrease the concentration of oil in the liquid entering the evaporator (perhaps by installing a more efficient oil separator). These two possibilities also suggest the cause of unacceptably high oil concentrations in evaporators and of chiller shutdowns due to loss of oil. The first is a failure of the compressor (leaking o-rings, missing plugs, etc.) and/or of the oil separator that causes unusually and unacceptably high oil discharge rates. The second is a failure of the oil return system, such as plugged lines, inadequate capacity of a pump, or inadequate driving pressure difference for an eductor. Considering the above, it should be obvious that the more effective improvement to any oil return system is to reduce the oil arrival rate; i.e. reduce the compressor oil discharge rate and/or improve the efficiency of the oil separator.

Oil Inventory in the Evaporator

If you were to do an oil mass balance analysis on an operating flooded evaporator as described above, by measuring liquid line flow and concentration and oil return line flow and concentration, you might yet experimentally find more oil in the evaporator than you expect. The discussion which follows offers a possible explanation. The point of the discussion is that the design of the evaporator itself and the location of the oil return pickup can have a major impact on the success or failure of an oil recovery system. This is relevant because it can mean that replacing a poorly operating oil return system of one kind with another (e.g. pump with eductor) may not fix the problem, the real problem being that the oil return pickup point is poorly located.

Consider a one pass flooded evaporator. Warm water enters tubes at one end and exits as chilled water at the other end. Refrigerant liquid surrounds the tubes and is introduced by a pipe at the cold water end of the shell. Liquid refrigerant is withdrawn from the shell by the oil return system from the middle of the shell (or worse, from the cold end by the liquid inlet). As above, the refrigerant entering the evaporator contains 0.2% oil, and refrigerant is drawn by the oil return system at a rate of 100 lbs/hr and the concentration at the point of withdrawal is 2%. The arrival and removal rates are identical at 2 lbs per hour. If the evaporator refrigerant charge were 100 lbs, one would be tempted to conclude that the evaporator contained 2 lbs of oil. Yet, if you were to measure the oil concentration at the ends of the shell, you might find that the concentration was 10% at the warm end and 0.2% at the cold end. Why would this be? The answer is that most of the evaporation of liquid refrigerant takes place at the warm end of the shell where the temperature difference between water and refrigerant is the greatest. Gravity will see to it that this liquid is replaced with liquid from a higher elevation: liquid at the cold end of the shell which is evaporating, but slowly. Hence, there will be a slow axial flow of liquid refrigerant from the cold end of the shell to the warm end and it will take oil with it that will not return while the chiller operates. But that oil will not evaporate at the warm end nor will it be picked up by the oil return system which draws from the middle of the shell. Hence, oil will tend to concentrate in a place where the oil return system does not pick it up. And where the oil return system does pick up liquid, that liquid will not contain much oil. This will result in a “stored inventory” of oil in the evaporator which can be substantial. So it is important to know where in the evaporator the oil tends to concentrate and to draw return liquid from that point. That location varies by design of the evaporator and any associated internal liquid distribution system.

Part 2: Use of Eductors for Oil Return

By Ed Keuper - GEA Consulting

From time to time it is reported that on screw chillers using eductors for oil return, when operating at low load conditions, it appears that the eductor does not operate efficiently enough to return a sufficient amount of oil to the oil separator or sump to maintain its oil level, which then causes the chiller to shut down on low oil, a consequence of oil retained in the refrigerant charge in the evaporator.

For such a chiller which uses an eductor for oil return, the cause of the failure may not be low load, but rather low lift. In a comfort cooling environment, chiller load is responsive to outdoor temperature. That is, when it is hot outside, heat flows rapidly into the building and chiller load is high. Simultaneously, the chiller must reject its heat to a high ambient temperature. Hence the chiller operating at a high load condition is also operating at a high lift condition. Lift is defined as the difference between the suction and discharge saturation temperatures (or pressures).

When the outdoor temperature is cool, little heat needs to be removed from the conditioned space and so chiller load is low. The low load is accompanied by a low lift condition since the ambient temperature is down from its high value. The low lift is the cause of the loss of effectiveness of the eductor. The eductor is driven by the pressure difference between the condenser and the evaporator. When this pressure difference falls, the flow inducing capacity of the eductor is reduced. The flow inducing capacity of the eductor is approximately proportional to the square of the pressure difference. Hence, a pressure difference reduction to 50% of design will lead to an induced flow reduction to 25% of design.

Not all chillers serve the comfort cooling market. There are chillers applied to chemical processes, for example, that may have varying load but constant lift; i.e. constant suction and discharge temperatures. These chillers would not likely have oil loss problems related to load if served by an adequately sized eductor based oil return system.

Possible remedies for poor eductor performance in low lift applications include reducing the oil discharge rate of the compressor/separator and modifying the control system to increase the minimum lift of the system.

Liquid in the Compressor Suction

Ideally, any liquid entering the compressor suction will be rich enough in oil and lean enough in refrigerant that lubrication will be satisfactory. Yet, if any liquid ingested into a compressor has too low a concentration of oil, lubrication may be compromised and wear leading to compressor failure can ensue. All compressors are vulnerable to lack-of-lubrication failure, either from lack of oil or from too much refrigerant in the oil..

A second type of failure is the result of injecting too much liquid refrigerant/oil into a compressor that can damage or destroy the compressor by “liquid slugging”. Screw and scroll compressors are rather more tolerant of liquid in the suction stream than are reciprocating compressors. This is due to the differing nature of the compression processes.

In a reciprocating compressor designed for a three to one compression ratio, the gas may reach the discharge pressure when the piston is only at half stroke. At this point the discharge valve opens and gas is discharged as the piston continues to rise even though gas pressure in the cylinder no longer rises. The final clearance volume may be only one tenth of the total swept volume. This clearance volume is not discharged, but is re-expanded on the suction stroke. One might say at this point that the true compression ratio is ten to one considering a closed discharge valve (swept volume divided by swept volume plus clearance volume). If a volume of liquid of 110% of clearance volume is in the cylinder when compression begins, the piston will be compressing only liquid at the end of its stroke and the liquid may not be able to exit the discharge valve fast enough to avoid developing a very high pressure in the cylinder. This high pressure can cause failure of the connecting rod or failure of the head bolts. For a reciprocating compressor to be efficient, a small clearance volume is required. Yet, it is the small clearance volume that makes reciprocating compressors susceptible to liquid slugging damage. Allowable levels of liquid in the suction are determined by the ratio of clearance volume to swept volume.

In contrast, screw and scroll compressors designed for a three to one compression ratio capture a volume of suction gas (and some oil and maybe some liquid refrigerant) and reduce its volume to one third its original value. But the compression process is completed before the discharge port opens. Any liquid in the suction stream will cause the compression ratio to rise above the design value of three, but the rise is slower than in the reciprocating compressor. For example, assume that the suction stream for a screw compressor consists of 1 part liquid and 8 parts gas by volume. The compressor will reduce these 9 parts to 3 parts. At the completion of compression, one part will still be liquid and two parts will be gas. The pressure in the compressor when the discharge port opens will be four times suction pressure (8 parts gas going in divided by 2 parts gas going out). The one part of liquid remains one part because the liquid is essentially incompressible. Thus, the effect of liquid in the suction stream is to increase true compression ratio. But a four to one true compression ratio in a compressor designed for three to one is probably safe to operate. Allowable levels of liquid in the suction stream are determined by the design pressure ratio and the maximum pressure that can be tolerated in the compression chamber.

Part 4 - Lubricant Issues with Unitary Systems

By Dick Cawley - GEA Consulting

Ed Keuper, a GEA colleague, has presented three (3) very informative postings concerning oil return in refrigeration systems. His emphasis was on chillers and other refrigeration systems that use flooded evaporators and depend on oil separators at compressor discharge as well as a means for returning lubricant from the system to the compressor sump. His key points are (1) that the compressor must always have enough lubricant in the sump and (2) oil must not be present in the heat exchangers, mainly the evaporator, to the extent that heat transfer is significantly degraded.

This sequence deals with direct expansion unitary systems that are designed to move lubricant around the system and back to the compressor sump by momentum, in the case of refrigerant gas transport (Suction and discharge), and solubility where refrigerant is in liquid form. Another way of stating the design objective for this type of refrigeration system is that if, say, 1 % (oil in refrigerant) of lubricant is discharged from the compressor during stable operation, ***that ratio must be present anywhere in the system at any given time*** so that the 1 % is safely returned to the sump. One can view this arrangement as two (2) fluid streams – oil and refrigerant – traveling side by side from the compressor, through the condenser and evaporator, and back to the compressor sump after reaching equilibrium.

This series is presented as an overview at best. Many details for piping design to provide oil entrainment and circulation and to prevent oil drainage into places while a system is dormant can be found in the ***2010 ASHRAE Handbook of Refrigeration, Chapter 1 – Halocarbon Refrigeration Systems***.

It is important that a system as discussed here - or any system for that matter - be allowed sufficient run time to approach equilibrium after startup. What is equilibrium and how long to reach it? If a compressor is started, transients will occur for a period of time before the lubricant and refrigerant streams settle to a constant, or near constant pace.

To be certain that equilibrium is obtained, the very minimum run time after startup can be calculated by dividing the system refrigerant charge by the refrigerant flow rate. For example, if we have a system with refrigerant charge of 20 lb and a flow rate of 900 lb per hr (typical for 5 ton R22 system), the very minimum run time should be 20/900, or 0.022 hours (1.32 minutes). A safe minimum run time in this case might be 2 or 3 minutes.

Minimum run time is usually determined by the unitary system manufacturer and programmed into the control system; but, if not, this is one way of calculating it. Run time is usually ensured by a timed on control or some space thermostats. Note that the time required will be system dependent. Most unitary systems apply scroll compressors. Run time information and many other subjects are treated in Copeland Bulletin AE4-1331 R3.

Maintaining an Adequate Lubricant Level

Copeland Bulletin ***AE4-1331 R3*** covers the need to ensure that adequate lubricant always remains in the compressor sump. For instance, they recommend the ***addition of 1 oz oil for every five (5) lb of refrigerant over 20 lb of refrigerant charge***.

A scroll compressor is equipped with about 65 oz lubricant charge. One can quickly calculate the lubricant residing in a system having, say, and 1% lubricant in refrigerant. If the system has 20 lb of refrigerant, the oil in continuous circulation (away from the compressor) would be 20*.01 = 0.2 lb, or 3.2 oz. A system charged with, say, 50 lb refrigerant would have 8 oz lubricant in circulation. This is a substantial portion of the oil charge (12.3%) that cannot be in the sump. Six (6) oz. additional oil, as recommended by the compressor manufacturer, is needed in the system. Note, too, that these calculations are for steady state operation. Many times during transient operation, the circulation rate can be higher, which emphasizes the need for sufficient run time after the compressor is started.