System Conditions Harmful to Compressors Their Diagnosis and Prevention

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A refrigerating system should be so designed that it delivers refrigerant to the compressor at acceptable conditions and allows discharge at a pressure suitable for the compressor limitations when related to the refrigerant entering condition. Compression ratio and refrigerant characteristics are important factors but, the conditions of liquid with the suction gas and too high suction gas temperature are all that can be considered in this discussion.

There are indications that some liquid refrigerant and relatively high temperature vapor are not detrimental to screw type compressors. Thus some of the comments as to the effect on positive displacement compressors may apply only to reciprocating or rotary types.

It is well known that the return of liquid in considerable proportions (in a form referred to as slugs) may cause immediate rupture of some of the compressor parts. There seems to be much less appreciation of the harm resulting from a return of a portion of the refrigerant in finely divided drops, or a mist, of liquid. This dilutes the lubricant on the cylinders and reduces the useful life of the compressor. For example, it has not been considered unusual for a 1750 RPM, about 3" stroke compressor to require new liners and rings after 15,000 hours operation. When such a compressor was used with a system which returned liquid free, nearly saturated vapor, there was no apparent change in performance after 45,000 recorded hours of operation.

Most of the present reciprocating compressors discharge a large portion, or nearly all, of the friction heat to the refrigerant vapor. (With hermetic machines the waste heat of the motor is usually added.)

The smaller compressors may transfer a considerable part of the heat to the surrounding air without excessive temperature rise because the ratio of the surface of the unit to its rate of heat output, due to friction, is high. As the compressor increases in size, the proportion of friction heat lost to the surroundings is very small. Most of this heat, which is about 100 BTU/hr for each CFM of compressor displacement, must be removed by the refrigerant vapor before it enters the compressor cylinder. The weight rate and temperature of the suction vapor must be such that it will remove this heat without excessive compressor temperature. This is important because the relatively light oil necessary for low temperatures in the evaporator must not be at too high a temperature to have the necessary lubricating value.

CAUSES OF LIQUID IN SUCTION GAS

1. A sudden increase of compressor capacity as occurs when an additional cylinder, or cylinders, become effective. This reduces suction pressure, causing the superheat control type of expansion valve to move toward the open position (a 4 psi decrease changes most valves from closed to fully open) and pass refrigerant at a greater rate than required. Cooling of the expansion valve bulb will correct this condition but this cooling takes some seconds while the pressure change is nearly instantaneous.

2. Opening of a liquid line solenoid of a circuit in a multicircuit system in which operation was already taking place with one or more of the circuits in use at normal evaporating temperature. The bulb of the expansion valve, to which liquid is now supplied, is warm resulting in the valve being wide open. Even though the load on this circuit may be low, the valve will pass liquid at its highest rate until the temperature of the bulb has been reduced to that corresponding to the pressure plus the superheat setting. Some tests showed periods as
long as 85 to 120 seconds before the valve started to control.

3. A sudden increase in load such as caused by warmer liquid reaching a chiller or opening the face dampers of a face and bypass arrangement. This will increase the boiling rate and may cause boil over into the suction line. If the compressor effective displacement remains constant at this time, the pressure will rise and move the expansion valve toward a closed position. This will reduce the rate of liquid feed to the coil and help terminate the boil over.

4. Boiling over of a flooded chiller because of overloading during pulldown, or from some other cause, or due to a poorly designed or otherwise inadequate surge drum.

Of course there are other causes including defective expansion valves or other controls.

**CASES OF LIQUID WITH SUCTION GAS**

One fallacy has confused many. This is the belief that if the temperature of the gas is above that corresponding to saturation at the existing pressure (indicating superheated gas) there can be no liquid present. This concept results from the knowledge that liquid and superheated gas cannot exist in equilibrium, together. However, there is no assurance that a condition of equilibrium exist at a point in a suction line. With a velocity of probably 10 to 60 ft/sec, the mixture is in the suction line for a very short time to expect equilibrium.

Several cases will illustrate the condition.

1. Probably one of the most striking cases of liquid with superheated gas was in the 5" or 6" steel pipe between the first and second stages of a large horizontal ammonia compressor. In the header through which this hot discharge gas passed, at about 20 psig (6F saturation equivalent) liquid was injected with the intent of desuperheating the gas. Unfortunately, the design was such that the liquid was not in a fine mist which would have given reasonable results. Instead it ran out of some holes in a perforated pipe inside of the header. This liquid ran along the bottom of the header and down one side of the pipe. One side of the pipe was too hot to touch and the other was frosted.

2. Another case involved a direct expansion brine chiller using R-22 to low-

er the brine to -20F with evaporation at about -30F. The compressor was a six cylinder with three groups of two cylinders each. One group was vertical and the others at a 60 angle on opposite sides. Both suction and discharge valves were in the cylinder heads with a dividing partition in the center of the head between discharge and suction. The suction side of the two side groups was the lower side.

The particular path for the gas to each pair of cylinders of this compressor is such that more of the compressor friction heat is added to the gas to the two side pairs than to the gas going to the vertical pair. With all gas entering the suction of this machine, the suction side of the head of the vertical pair would be slightly (but sensibly) cooler than that of the two side heads. Any liquid, because of its inertia, would go more to the side pairs.

The suction gas passed through a shell and coil heat exchanger before reaching the compressor and the suction line was about 40 to 50F as it entered the compressor. The pressure, at the time, was about 10 psig (-20F sat. equiv.).

Applying a hand to the suction side of each pair of cylinders showed the side pairs to friction heat is added to the gas to the two side pairs than to the gas going to the vertical pair. With all gas entering the suction of this machine, the suction side of the head of the vertical pair would be slightly (but sensibly) cooler than that of the two side heads. Any liquid, because of its inertia, would go more to the side pairs.

Their doubt was short lived because the head bolts of the suction side of one side pair of cylinders were so loose that a leak started. Liquid refrigerant ran out on the floor where it boiled.

3. In another case, there were two duplex eight cylinder compressors (a total of 4 compressors). Each duplex was the common arrangement with a double shaft motor between two compressors.

The suction line to each duplex unit came from one end, dropped about a foot in the space between them and divided. See Fig. 1. At the dividing point, one line continued in the same general direction as the common line before it dropped to a compressor. The other went in the reverse direction to connect to the other compressor. At various times, during operation, the
branch which ran in the same direction as the main was found to be at least 10°F cooler than the one which went in the opposite direction.

Certainly, if there were only gas in the suction line, it would be impossible to have the division of this type resulting in the temperature of the two streams so different.

FIGURE 1

4. A two stage refrigerating system cooled liquid to about -15°F in one chiller and water to +40°F in another. Both chillers were flooded type using R-22, with -20°F and +35°F evaporating respectively.

The first stage compressor took the gas from the -20°F evaporator and discharged to the suction of the second stage compressor. The second stage compressor took this first stage discharge and the gas from the +35°F evaporator and discharged to about 105°F condensing temperature.

With proper flooded chiller design and operation, the gas should leave the -20°F evaporator at about -20°F and free of liquid. With such a condition entering the first stage compressor, its discharge would be about 90-100°F. However, this temperature was +35°F (corresponding to saturation). With the temperature indicating saturation there would be no evidence as to whether or not liquid was present in this discharge gas. The condition of the discharge from the second stage at about 150°F showed that little or no liquid reached the second stage suction.

Using the pressure-enthalpy chart it is found that at least 8%, by weight, of the refrigerant entering the first stage suction was in liquid form.

Operation, even with this liquid return rate, appeared perfectly satisfactory.

There was no sound from the compressor to suggest anything wrong. Unfortunately, after about 700 hours of operation, the compressor required new liners and rings.

CHECKING FOR LIQUID WITH SUCTION GAS

When liquid in small drops is being carried along by the gas, it will usually not be apparent from the outside of the pipe or a temperature reading. There are various methods of checking for its presence. Some of these are based on its separation, or partial separation. Another is based on temperature and pressure measurements of the suction and discharge gases and comparison with the values which would apply if there is no liquid with the suction gas. It is also often possible to discover the presence of liquid by a restriction (such as a partially open valve) in the suction line.

LIQUID SEPARATION OR PARTIAL SEPARATION

The design of the compressor already mentioned, when it is the six cylinder model, had gas paths which resulted in partial separation and made it possible to sense the resulting temperature difference, with the hand, before compression. Since this often resulted in temperatures like 10°F in ranges from 20°F to 50°F, there was little doubt as to which head was warmer than the other. This type of check is almost never possible with present compressors because the suction gas seldom, if ever, passes through the head. Although the same conditions which resulted in a lower suction temperature also result in a lower discharge temperature, it is rather difficult to sense, with the hand, differences like 10°F when both temperatures are around 140°F or higher.

When there are different flow paths to two or more compressors (such as shown in Fig. 1) where the inertia would tend to deliver the liquid selectively, temperature differences can show the presence of liquid.

If an adequate and properly design suction trap is installed in the suction line, any liquid will be collected in it. There it can be identified by sight glass, by draining off a little or by other means.

This method was used to convince the designer that liquid was returning with the suction. After the compressor had been damaged by what the compressor manufacturer was convinced was liquid, it was agreed that the owner would have the recommended trap installed. After installing the trap, the system would be operated in a normal manner with starts and stops and capacity changes. The compressor manufacturer agreed that, if no liquid were caught
in the trap during an hour of such operation, he would pay the cost of the trap and its installation. In the cases where this was done, liquid was collected.

TOO LOW DISCHARGE TEMPERATURE

It is possible to learn much from a calculation of what the discharge temperature should be and comparing this value with the actual discharge temperature.

The calculation can be made from capacity and power data for the particular compressor at the actual suction and discharge pressures. It can also be made from the displacement, volumetric efficiency and friction power data for the compressor.

As an example of the first method, suppose a system is operating with refrigerant 22. At the entrance to the compressor, the pressure is 69 psig (corresponding to 40F saturated) and the gas temperature is 65F. The discharge pressure is 213 psig (corresponding to 105F saturated). The liquid refrigerant, at the entrance to the expansion valve, is at 100F. The rating for the compressor, at these conditions, is given as 82.1 tons, requiring 79.0 HP. (If the rating is not given for the actual operating condition, it can be obtained by interpolation and corrected for differences of liquid and suction conditions.)

Refer to the skeleton pressure enthalpy chart for R-22 in Fig. 2. At 83.7 psia (69+14.7) and 65F, the enthalpy is 112.5 BTU/lb. From the complete chart or an R-22 table, the enthalpy of 100F liquid is 39.3 BTU/lb. Each pound of refrigerant gains 112.5-39.3 = 73.2 BTU/lb, so that the compressor handles 82.1x200/73.2 = 224 lbs/min.

The 79.0 HP corresponds to 79.0x2.4 = 3350 BTU/min. With a compressor of this size, nearly all of this energy is added to the refrigerant. The surface of the compressor and the temperature difference between the compressor surfaces and its surroundings could hardly account for more than about 3% of the 3350 BTU/min. Thus the enthalpy of the refrigerant vapor leaving the compressor should be about 112.5+3350/224 = 127.0 BTU/lb. Referring to Fig. 2, it will be noted that, at 228 psia and 127.0 BTU/lb, the temperature is slightly over 170F.

If the discharge temperature is much less than 170F when other conditions are as described, it is evident that some liquid is present in the gas entering the compressor.

Of course, if an oil cooler, water jacket or other means is used to remove heat from the compressor, the value of the heat so removed must be deducted in calculating the discharge temperature.

Temperature and pressures must be taken with reasonable accuracy. Experience with use of this method has shown the calculated discharge temperature always within 10F of the actual, when no liquid enters the compressor.

Incidentally, it might be mentioned here, that it is well worth while to check this discharge temperature occasionally since it can give indications of other undesirable conditions. When this was done for one compressor, it was found that the discharge temperature was 80 to 90F higher than it should be for the suction conditions. A calculation showed that the enthalpy rise was about twice as large as appropriate, thus indicating that the energy was being added to about half of the normal gas flow. Evidently the compressor was compressing all of the gas about twice and a leak between discharge and suction existed. An examination showed the relief valve (between discharge and suction chambers of the compressor) was defective.

During operation, with the defective valve the compressor was only producing about half capacity but this had not been noticed because, at the particular time of the year, no more was required.

REDUCTION IN SUCTION LINE PRESSURE

If a restriction (such as a partially closed valve) can be imposed in the suction line, the resulting temperature drop can usually indicate the presence of existing liquid. For example, assume refrigerant returning at 60 psig and about 45F indicated temperature. A valve is closed so that the pressure after the valve is 45 psig.

This reduction in suction pressure will
reduce capacity 20-25%. If the compressor had been operating at part capacity, conditions on the suction side may be kept unchanged, if coincidentally with imposing the restriction, the compressor capacity is increased by 25-33% to maintain the same pressure before the valve. Suppose this change has been made, and the pressure is 60 psig before the valve with 45 psig after. If there is no liquid with the gas, the refrigerant temperature will change from 45°F before the valve to about 40°F. If liquid is present, the temperature after the valve will be less than 40°F and may be as low as 22°F. (About 3% liquid, by weight, would result in the 22°F.)

It may be that the only valve in the suction line is the compressor service valve. This will serve the purpose when, as is usually the case, the temperature of the gas can be measured after the valve but before heat is added by the compressor.

When an evaporator pressure regulating valve is used, the restriction already exists. Temperatures and pressures at the inlet and outlet will provide what is needed to show the presence or absence of liquid.

When a restriction is to be imposed in the suction line, and no compensating increase in compressor capacity is possible, the check for liquid is still valid but, of course, it is made at different low side conditions. The lower heat removal rate from the evaporator might eliminate liquid carryover which existed at the higher rate.

**DESIGN TO ELIMINATE LIQUID TO COMRESSOR**

In spite of all that is known about the danger of liquid to a compressor as well as how to design to assure that no liquid will reach the compressor, systems which return liquid to the compressor periodically or almost continuously, are still being installed.

In many cases, liquid traps, or separators, installed in suction lines are too small or otherwise inadequate to separate the liquid from the gas and keep the liquid in the trap. Two recent installations had so-called "suction line accumulators" (at least 3 or 4 on each installation). Each accumulator on one system was the vertical type with the gas line entering the side, fairly high, with a 90° elbow turning down at the center. The gas leaves by a 90° elbow, turned up, very near the top. On the other system, each accumulator is a horizontal type of a fairly well known make.

When liquid enters any of these vessels, it leaves instantly, with little or no accumulation in the vessel. The reason for this is that, for the design, the vessels are all too small. The tendency of the vapor to carry liquid with it varies with the square of the vapor velocity times the ratio of vapor density to liquid density. When separation is by gravity, it has been found that: - (vapor velocity in ft/min)² times ratio of vapor to liquid density, must be less than about 85 for reasonable certainty of separation. When the gas must go down and back up, as in the vertical one described, it may be necessary to provide a larger cross section to accommodate the two directional flows.

It is difficult to understand why one would make a trap like the two just described when a vertical cylindrical drum with tangential entrance (see Fig. 3) can be less than half the diameter required by the other type to be effective.

An experience with an ammonia system gives rather conclusive evidence of the superiority of the tangential entrance type over the one with the 90° downturned elbow on the entering line and the 90° upturned elbow, above it, on the leaving line.

The system was about 60 tons capacity at about 24 psig or +10°F evaporating. The suction line was 2" and the temperature of this pipe was about 60°F. The compressors were the relatively slow speed vertical single acting type. The trap was 8" diameter with 2" in and out.

The refrigeration contractor reported that after less than three hundred hours operation, it was necessary to rebuild the compressor. One compressor was dismantled at the time so that the type of wear of pistons, rings and cylinders could be seen. From this it was clear that liquid was causing the damage by removing the lubrication from the cylinders.

The first observation, when this diagnosis is proposed is how can liquid be present with suction gas at 40 or 50°F superheat. It happened that, in this case, occasional spots of frost would appear and disappear on the sides and top of the uninsulated suction trap. Of course, this was rather conclusive evidence of liquid.

The next question was as to why the suction trap did not separate it. A quick calculation indicated that a trap of that type should be about 20" diameter and preferably have larger connections. However, if the tangential entrance type trap were used, the 8" diameter would effectively separate the liquid.

Because of the urgency, the change was made by closing the existing trap entrance and making a new 2" tangential entrance about 4" below the old one. As soon as the system was again operated, liquid collected in the bottom of the trap from where it
was evaporated by the liquid coil already in the trap. The compressor problem was eliminated.

![Diagram](image)

FIGURE 3

The tangential entrance trap has another advantage when used with refrigerants at conditions where superheat (even resulting from useful refrigeration) is thermodynamically undesirable. The vapor enters the trap and moves toward the outlet. There is practically no tendency for vapor to circulate in the lower part of the trap. This is very evident from the higher temperature of the lower part of such a trap when uninsulated and surrounded by air at 30°F or so above the refrigerant vapor, if no liquid is entering. This is noted even on traps with no warm liquid coil in the bottom. Only liquid goes to the bottom and any heat entering the lower part, through its surface or from a liquid coil, serves to evaporate liquid without heating the vapor appreciably.

Of course, the vertical cylindrical tangential entrance vessel is also the lowest cost effective design for the surge drum of a flooded evaporator.

If there is any possibility of liquid leaving the evaporator with the refrigerant vapor, a properly designed trap of adequate size, should be installed in the suction line. Means must be provided to evaporate the refrigerant liquid at the average rate it is collected and also return the oil, which collects, after proper filtering and removal of refrigerant. If the vapor goes to more than one compressor, the oil must be returned to the compressors in the proportion required.

**DESIGN FOR VAPOR RATE AND TEMPERATURE REQUIRED**

Compressor oil temperatures should usually be kept from exceeding about 150°F. A limit of 170°F has been given by at least one manufacturer. It is wise to design for less because minor variations from expected conditions often result in increases above design.

If no oil cooler is used, practically all of the compressor friction heat must be removed by the refrigerant vapor before it enters the cylinders. If an oil cooler is used, its heat removal capacity should be known. If the friction heat exceeds the oil cooler capacity, the excess must be removed by the vapor before it enters the cylinder. The vapor weight rate and temperature must be adequate to absorb the heat at a temperature which does not result in too high oil temperature.

It is impossible in a discussion so limited as this, to give any detailed description as to the design for proper temperature of the compressor or acceptable limitation of discharge temperature. Many factors influence the result, including the refrigerant characteristics, the temperature range in which it is used, the compression ratio in the cylinder, the temperature of the suction gas and the heat to be removed from the compressor.

**CONCLUSION**

No matter how good a compressor is, it must operate within the limits for which it is designed. If the system is so designed as to force the compressor to operate outside its limits, results will be unsatisfactory. The life of the compressor will be reduced, possibly to a small fraction of what should be expected. Unfortunately, the user usually blames the compressor.

Everything necessary for design of a system so that the compressor will operate within any specified limits, is known. For user satisfaction and less undeserved costs for the compressor manufacturer, more effort should be made to have the system designed by one who really understands all the factors which apply.

If a system gives trouble, replacing a damaged compressor, without finding the cause and making the necessary corrections is likely to result in more user dissatisfaction and further cost to the compressor manufacturer. Correctly diagnosing the cause and a complete explanation to the user, will reduce cost for everyone and result in more user satisfaction.