



For more free Copeland literature please visit www.HVACRinfo.com

Application Engineering Bulletin

AE-1260-R1

Revised March 1, 1993

COMPRESSOR OVERHEATING

It is ironic that in an industry whose product is cooling and refrigeration, the most serious field problems arise from overheating the compressor. It is not only possible, but probable, that the great majority of compressor failures on low temperature systems originate in compressor overheating, and field experience indicates the great majority of those failures can be eliminated if discharge gas temperatures are reduced to a reasonable level.

Obviously it would be to everyone's benefit if it were possible to design a compressor that could live with the most demanding conditions with no concern about overheating. There are approaches in compressor design which can minimize the problems, such as good exterior heat transfer surface; direct metal to metal contact between the motor and the compressor housing; minimal heat transfer between the low pressure and high pressure areas of the compressor; low volumetric clearance; good motor design; and oil coolers. The fact remains that we are dealing with some physical laws of nature and at extreme operating conditions, the temperatures created may be beyond the capabilities of the best refrigeration compressor.

Why then do we continue to follow practices that make inevitable high failure rates? Certainly no responsible engineer in the industry knowingly designs for unsafe conditions. The reasons strangely enough appear to be as much psychological as physical.

Many engineers, servicemen, and users simply refuse to accept the fact that there are limits to a compressor's operation. They have been conditioned by past experience with conservative safety factors and product improvement to always expect that in some miraculous fashion a change of compressors will solve all their problems. They always cite the one compressor that survives, and ignore the five that have failed. These are probably the same people who refuse to accept the fact that the world's energy resources are reaching the bottom part of the barrel. Some people can cope with problems only by refusing to acknowledge them.

A second part of the problem may be that our educational and training programs are out of date. Twenty years ago liquid refrigerant slugging and liquid floodback were very serious problems. Compressor valves and bearings were more vulnerable, and system designs were not always the best. For twenty years every educational and training program has hammered at the dangers of liquid refrigerant and the need for proper superheat, and few words have been said about the dangers of high return gas temperatures. In effect we have tried to brainwash a whole generation of service personnel to worry only about liquid floodback.

Possibly because of the old fears of liquid refrigerant, we as compressor manufacturers and in general the industry have been slow in recognizing and identifying high temperature failures. At cylinder temperatures of 315°F to 325°F, the lubricating film is literally evaporated off the cylinder walls like water on a hot griddle, but modern refrigeration oils are so resistant to breakdown, carbon is not formed on the valve plate. As a result many high temperature failures are misdiagnosed as liquid failures because the analyst doesn't know the application.

Compressor rating data is a continuing source of confusion in the industry. In order to provide a common basis for comparison, compressors are rated and capacity data is published at common conditions. In the case of Copeland low temperature compressors, the rating data is presented with return gas temperature of 65°F. This temperature was selected many years ago, probably for convenience of testing as much as any other factor, and at the time might have represented an acceptable operating condition for most systems. Over the years the trend to larger horsepower compressors, the demand for lower evaporating temperatures, and in particular the increasing use of multiple compressors on a single system have tremendously increased the stresses on the compressor.

Users still interpret the published rating data with 65°F return gas as a recommendation that the compressor should be operated at that condition. In the case

of smaller, single compressor systems operating at more moderate suction pressures, 65°F may still be acceptable, but on larger, more sophisticated equipment, lower return gas temperatures are required if the compressor is to be maintained within acceptable temperature limits.

Refrigeration is not an exact science. System operation can be drastically affected by many variables in the field installation and operation, and since each party would like to believe that all problems are the fault of someone else, unfortunately there are widely varying opinions on every aspect of system performance. It is easy to understand the difficulty of the final user in intelligently evaluating and specifying the reliability that he might truly desire. It is easy to understand the difficulty of the design engineer in maintaining conservative design factors based on only his good judgement when the analysis of failures is cloudy, and he is subject to the tremendous cost pressures of a fiercely price competitive industry.

At a meeting of service managers from several large commercial refrigeration manufacturers, it was unanimously agreed that compressor overheating is the greatest single field problem existent today, and the one on which the least headway was being made in solving the problem at the user level. If we are to make any major reduction in failure rates due to overheating, it is essential that there be widespread understanding of the problem in the industry.

For those that refuse to accept the fact that there is a problem, little can be done. The encouraging side of the picture is the fact that for those that are willing to make the effort, low temperature overheating failures can be stopped dead.

Temperature Limits

Most refrigeration oils will start to break down or carbonize at temperatures of 350°F. Tests in a contaminant free atmosphere may indicate a reasonable tolerance for even higher temperatures, but the real world is inhabited by a multitude of systems that have varying degrees of contaminants such as air and moisture.

Extreme ring and piston wear can occur at cylinder temperatures of 310-330°F with little oil carbonization. There is growing evidence that modern refrigeration oils have been so highly refined to obtain good solubility and high breakdown temperatures that the oil is unable to maintain a lubricating film at high temperatures.

Field experience in general would indicate for good long life characteristics, piston, ring, and valve port temperatures should be maintained below 300°F. Normally discharge line temperatures within 6 inches of the compressor outlet will be from 50°F to 75°F cooler than cylinder and piston temperatures, depending on the compressor design and the refrigerant mass flow. Therefore as a general rule, 275°F discharge line temperatures represent a certain failure temperature condition, 250°F is usually a danger level, and 225°F and below are desirable for reasonable life expectancy.

There are different opinions in the industry on oil sump temperatures. The viscosity of the common refrigeration oils decreases rapidly as the temperature increases, and becomes dangerously low at temperatures of 200°F and above. At high temperatures the characteristics of the lubricant are critical, additives may be necessary, and bearings must be capable of withstanding the extreme temperatures. Lower temperatures are generally much more conducive to long life.

Low Temperature R-502 Applications

Few users realize the critical nature of single stage low temperature refrigeration. The supermarket manager demands brick hard -20°F ice cream; the blast freezer operator demands tunnel temperatures of -40°F for faster processing; the ice cream distributor demands -25°F truck body temperatures; each thinking only of satisfying his temperature demand, and none of them realizing their demands are an almost certain death sentence for the compressor. The normal low temperature evaporating limit for single stage compressors is -40°F, and although this can be extended to intermittent periods of operation down to -50°F, operation below -50°F (0 psig with R-502) creates discharge temperatures which can destroy the compressor.

In a similar fashion, the contractor or original equipment manufacturer who for reasons of convenience or expediency cycles the fan providing compressor cooling, or locates the compressor so that adequate cooling air cannot impinge directly on the compressor may unwittingly be creating the same critical temperature conditions. At low temperature operating conditions the decreasing density of the refrigerant vapor and the heating effect of high compression ratios combine to create high discharge temperatures which cannot be controlled by refrigerant cooling alone. The additional heat transfer gained from a direct air blast on the compressor is absolutely essential for compressor survival, and any decrease in the recommended airflow or loss of direct impingement on the compressor can lead directly to excessive cylinder temperatures.

At today's demanding temperature conditions, a low temperature compressor may be operating right on the edge of its survival limits. The lower the evaporating temperature and the higher the condensing temperature the more critical the discharge temperature becomes. The only way to insure reasonable discharge temperatures at extreme conditions is by means of very low return gas temperature.

Table 1 illustrates some typical internal temperatures on low temperature compressors with 65°F return gas temperature, a common field condition. The cylinder temperatures have been calculated based on a 75°F increase in the return gas temperature after entering the compressor and prior to entering the cylinder on the suction stroke. The 75°F temperature increase is typical of that found in laboratory testing, and is caused by heat transfer from the motor and compressor body. The final column in the table is the return gas temperature entering the compressor that would be necessary to maintain internal cylinder temperatures below 300°F.

Undoubtedly with operating wear, the temperature conditions become even more severe. Table 1 data is calculated based on laboratory testing, but individual installations vary, and the goal should be to maintain temperatures on the discharge line six inches from the compressor at 225°F or less.

Due to the danger of freezing in underground trenches and sweating in the machine room, low return gas temperatures cannot be achieved in a typical supermarket with uninsulated suction lines. The increasing incidence of low temperature failures from overheating indicates some major change in design approach is required to improve system reliability.

One large mid-western chain decided to attack this problem in the early 1970's with spectacular results. They set the expansion valves for low superheat control, completely insulated the suction lines to the compressor service valves, provided cool air ventilation for the machine room, with the result that compressor failures have been almost completely eliminated.

The need for completely insulating suction lines on low temperature systems seems clearly indicated, and is highly recommended to improve compressor life and reliability.

R-22 Applications

An equally critical temperature condition can occur on R-22 systems with evaporating temperatures below 10°F. Unfortunately many medium temperature R-22 systems designed for nominal evaporating temperatures of 5°F or 10°F wind up operating at suction pressures equivalent to evaporating temperatures of -10°F or lower, and such installations can develop severe problems. Note that it is the suction pressure at the compressor which is critical, not the case evaporating temperature, since in many cases the critical threat is caused by pressure drop between the case and the compressor.

Table 2 illustrates some typical internal temperatures on medium temperature applications with R-22. As in the case of R-502, the cylinder temperatures have been calculated based on a 75°F increase in the return gas temperature after entering the compressor and prior to entering the cylinder.

Demand Cooling

Overheating of R-22 systems operating at evaporator temperatures below 10°F can be avoided by direct liquid injection into the compressor. This "Demand Cooling®" system utilizes solid state electronics to inject liquid only when required thereby improving system efficiency compared to mechanical injection systems.

Desuperheating Expansion Valves

On existing systems with uninsulated lines where it may not be possible to change the system operating conditions, the only means of reducing discharge temperatures to an acceptable level may be with a desuperheating expansion valve.

Extensive field testing on problem installations revealed that discharge temperatures could be reduced almost degree for degree by reducing return gas temperatures. Repeatedly we found systems operating with 250°F to 260°F discharge line temperatures and 60°F return gas temperatures could be modified with a desuperheating expansion valve to obtain discharge line temperatures below 225°F with return gas temperatures of approximately 30°F.

The desuperheating valves were installed within three to six feet of the compressor suction valve with the suction line insulated from the expansion valve to the compressor inlet. Initially ton valves were tried, but better performance was obtained with nominal one ton valves with compressors in the 7 H.P. to 25 H.P. range. Several alternate expansion valve charges were

Table 1

Typical R-502 Cylinder Discharge Gas Temperatures

Saturated Evaporating Temperature At Compressor Suction Pressure	Saturated Condensing Temperature At Compressor Discharge Pressure	Typical Return Gas Temp.	Cylinder Discharge Temperature	Return Gas Temperature Necessary To Limit Discharge Temp. To 300 °F
-40°F	130°F	65°F	340°F	20°F
-40°F	120°F	65°F	335°F	35°F
-40°F	110°F	65°F	320°F	45°F
-25°F	130°F	65°F	320°F	45°F
-25°F	120°F	65°F	310°F	55°F
-25°F	110°F	65°F	300°F	65°F

tried but satisfactory performance was obtained only with the following;

- Sporlan GR-1-L1
- Alco LCL2E-IE.

If suction lines can be insulated, desuperheating valves are not necessary, but desuperheating valves do offer a means of reducing temperatures on systems where no other approach is possible.

Liquid to Suction Heat Exchangers

Liquid to suction heat exchangers in refrigeration systems are useful in raising the return gas temperature to prevent frosting or condensation, to evaporate any liquid droplets in the vapor stream, and to subcool the liquid to prevent flash gas in the liquid line.

However there appears to be some misunderstanding in the industry regarding any capacity increase as the result of a heat exchanger. To the extent that heat absorbed by the vapor from the liquid refrigerant displaces heat that might be picked up from the ambient or other non-refrigerated spaces there is a capacity increase, and therefore on systems with uninsulated suction lines there is normally an increase.

The mere transfer of heat from liquid to suction does not in itself add any significant capacity or efficiency to the system. While the enthalpy of the liquid will be decreased, thus increasing the enthalpy change per pound in the evaporator, the warmer vapor has a higher specific volume (cubic feet per pound) so the pumping

capacity of the compressor will be decreased. These two factors, the decreased enthalpy of the liquid and the higher specific volume of the vapor, largely cancel out the effect of each individually, so the net effect is small, and if the liquid line loses any temperature in transit, it is doubtful if any increase is measurable.

With insulated suction lines the heat exchanger not only loses its capacity benefit, but may actually be a threat if it raises the temperature of the return gas excessively. In particular, liquid to suction heat exchanger in the machine room at the compressor may actually contribute to compressor failure if they elevate the temperature of the return gas beyond an acceptable level. This is especially critical on R-22 low temperature applications.

Summary

Compressor overheating has become and remains a major field problem primarily because few people recognize or understand the pattern of failure. In today's large sophisticated systems, lower return gas temperatures are essential, and it appears the most direct way of accomplishing this is by insulating suction lines. While this will increase the initial system cost, field experience indicates the reduction in compressor failure rates can quickly repay the added investment.

On existing systems where insulating lines is not feasible, desuperheating expansion valves have been proven to be an effective means of obtaining lower discharge temperatures.

Table 2

Typical R-22 Cylinder Discharge Gas Temperatures

Saturated Evaporating Temperature At Compressor Suction Pressure	Saturated Condensing Temperature At Compressor Discharge Pressure	Typical Return Gas Temp.	Cylinder Discharge Temperature	Return Gas Temperature Necessary To Limit Discharge Temp. To 300 °F
-10°F	130°F	65°F	365°F	5°F
-10°F	120°F	65°F	350°F	25°F
-10°F	110°F	65°F	340°F	35°F
0°F	130°F	65°F	335°F	25°F
0°F	120°F	65°F	320°F	45°F
0°F	110°F	65°F	310°F	50°F
10°F	130°F	65°F	320°F	45°F
10°F	120°F	65°F	300°F	65°F
10°F	110°F	65°F	290°F	65°F