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Monitoring to Prevent Problems

Refrigeration System Malfunctions

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Commercial and industrial refrigeration systems usually consist of separate components from different manufacturers. A typical refrigeration system consists of heat exchangers, such as evaporator(s), condenser(s) and liquid suction; compressor(s); refrigerant feed control devices, such as expansion valves; refrigerant filters; and vessels (high-side receivers and low-side accumulators).

In some systems, a secondary fluid such as ethylene glycol-water also may be used to avoid long runs of refrigerant piping that would increase system refrigerant inventory. The components for a system must be selected and integrated by the design engineer based on information from component manufacturers that includes catalog equipment performance and application engineering recommendations. Guidelines for the process of component selection and integration are provided by ASHRAE.[1,2] However, there are a number of details in this process than can lead to non-optimal system configurations and operation.

A case study was performed on a refrigeration system serving an indoor skating arena in Madison, Wis., that has 25,000 ft^2 (2323 m^2) of ice. The HCFC-22 based system, which was installed in the winter of 1996, consists of six semi-hermetic reciprocating compressors having a total capacity of 103 tons (355 kW). The shell-and-tube evaporator has two separate tube-side direct-expansion refrigerant circuits. A parallel rack of three reciprocating compressors serves each refrigerant circuit. (The two independent refrigeration loops are designated as "odd" and "even" throughout this article.)

On the shell side of the evaporator, a secondary fluid (ethylene glycol-water solution) is cooled to the design supply temperature of 14°F (–10°C) prior to being pumped remotely for cooling two separate skating rinks. Heat is rejected from the refrigeration system by a two-circuit induced draft single speed evaporative condenser. As originally designed, the evaporative condenser fan cycles on and off as necessary to maintain the saturated condensing pressure between 220 and 250 psia (1517 to 1725 kPa). A line diagram of the system is shown in *Figure 1*.

The original objectives of this project were to compare the field performance of a large refrigeration system with design predictions from a computer model constructed from readily available catalog data and to identify opportunities for reducing energy costs. The motivation for selecting this particular site resulted from owner concerns that system energy costs were higher than expected.

The components in the system were instrumented to measure refrigerant pressures and temperatures at numerous points throughout both the odd and even loops as well as the secondary fluid flow rate, supply, and return temperatures. These data were recorded with a data acquisition system at two-second time intervals over several hours and a range of ambient conditions.

During the course of this investigation, the lack of agreement between the model (based on component manufacturer performance data) and actual system data resulted in the discovery of a number of operational problems. These operational problems were not expected because the system is new, the operators at the site check the equipment daily and it functions adequately to meet the refrigeration loads. This article describes methods implemented to detect and resolve these operational problems.

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In addition, the performance of the system was found to be significantly improved by implementing simple control strategy changes. The energy cost impacts of the operational problems and modified control strategy are presented. This article presents results for a specific refrigeration system. However, the problems discovered, their resolution, and control strategy changes at this site are applicable to many large refrigeration systems and thus the conclusions should be of general interest.

Unless specified otherwise, all calculations presented in this article are based on the actual refrigeration load of 53 tons (186 kW), a fixed average head pressure of 235 psia (1620 kPa) and an average electrical cost of $0.047/kWh with no demand charges. The refrigeration load needed to maintain the ice sheets does not vary significantly with time of year since the indoor space is heated during winter for comfort. However, the performance of the refrigeration system does vary as a function of the outdoor air wet bulb temperature. All yearly energy cost analysis estimates are based on typical weather data for Madison, Wis., determined with a simulation program written for the site.[3]

Problem 1: Plugged Suction Line Filter

Each of the two independent refrigeration system loops at the site is equipped with a refrigerant-side suction line filter located between the evaporator outlet and the suction line accumulator. The filter functions to protect the compressor from ingesting debris carried by refrigerant as it migrates through the system. Since the refrigeration system is a closed hermetic system (ideally), debris can enter the system only if it is opened for maintenance or if some unusual operating circumstance, such as a compressor problem, develops. When the system is operating in accordance with its design, the filters should remain clean.

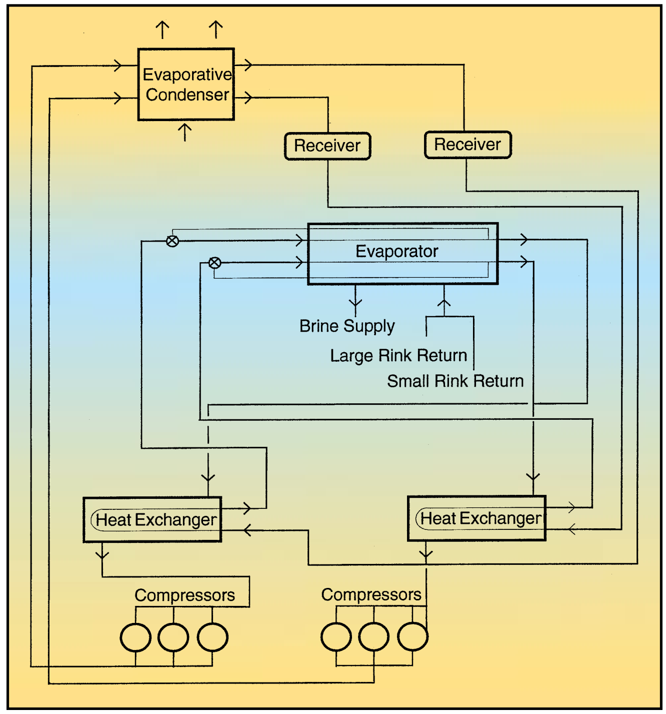

Figure 1: Schematic of the ice rink refrigeration system.

No direct method of measuring the pressure drop across the suction line filters was available on the system as-installed. However, when both the odd and even loop were simultaneously operating, it was apparent that the suction isolation valve at the inlet to the odd loop bank of compressors frosted up while

Recommended Additional Monitoring Equipment

A list of recommended monitoring equipment for any large refrigeration system is presented in the table below. For the site investigated, this monitoring equipment would be needed in both refrigeration loops. Pressure gages should be installed upstream and downstream of the suction and liquid line filters to detect any excessive pressure drop across the filter (indicating when it should be changed or flagging a larger problem in the system). Preferably, a single pressure gauge could be piped to a three-way valve, allowing the upstream and downstream side of the filter to be connected to the same gage. This arrangement eliminates gage-to-gage bias in pressure differential readings.

A pressure switch should be installed after the relief valve on the high-pressure receiver. This switch indicates whether the high-pressure receiver relief valve has opened (venting refrigerant from the high-pressure receiver) so the operators are aware of potential system problems and refrigerant loss. Receiver liquid level column, liquid level indicators and angle valves should to be installed to indicate the level of refrigerant in the receiver during normal system operation. If the level decreases below a pre-determined point, a refrigerant leak may be present in the system. The level indicators allow detection and repair of the leak before significant quantities of refrigerant have been lost to the atmosphere. ■

Component	Location	Number Required	Unit Cost	Extended Cost
Pressure Gage	Liquid Line Filter	2	$40	$80
Pressure Gage	Suction Line Filter	2	$40	$80
Pressure Switch	Receiver Relief Valves	1	$50	$50
Receiver Liquid Level Column	Receiver	1	$100	$100
Liquid Level Indicator	Liquid Level Column	3	$10	$30
Seal Cap Angle Valve	Liquid Level Column Shutoff	2	$40	$80

the valves on the even loop remained frost free. This frosting was the first indication that the odd loop compressors were operating with a lower suction pressure, and thus a lower suction temperature, resulting in increased frost formation on the valve. After this observation, a pressure gauge was installed to measure the pressure drop across the suction line filter. The measured pressure drop was 3 to 5 psi (20 to 35 kPa) *higher* than the expected 3 psi (20 kPa) drop for the filter at its rated refrigerant flow rate.

Figure 2 shows the contrasts of the dirty filter (bottom) with the clean replacement filter (top). As apparent by visual inspection, the filter was severely contaminated with debris. The additional pressure drop caused by the suction line filter (approximately 4 psi [28 kPa]) was included in the computer model developed for the system. The additional annual operational cost resulting from the dirty suction line filter was estimated to be $2,290 in the Madison climate, representing about 5% of the annual operating cost. This operating cost penalty could have been avoided by installing proper monitoring equipment on the system (see sidebar).

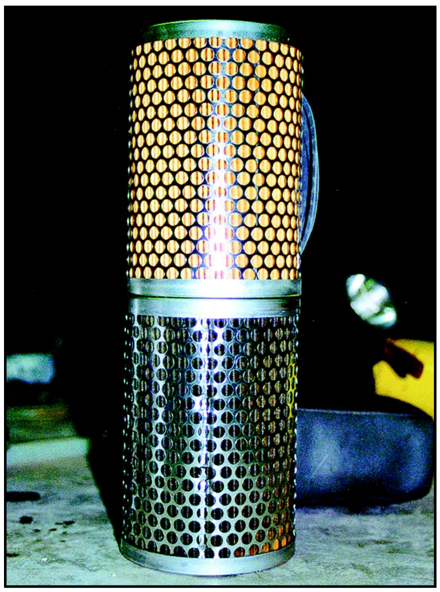

Figure 2: Clean (top) and dirty (bottom) refrigerant suction line filters.

Problem 2: Low Refrigerant Charge

As illustrated in *Figure 1*, the system has two separate refrigeration loops with separate charges of refrigerant that cannot mix. All refrigeration systems are designed and operated to deliver liquid refrigerant to the expansion device, in this case, a thermostatic expansion valve. A sufficient charge of refrigerant must be provided in the system to fill the volume of the piping and components between the condenser and a portion of the high-pressure receiver with liquid refrigerant.

Assuming the system is initially supplied with a proper charge of refrigerant, a low charge will occur, eventually, only if refrigerant leaks to the atmosphere from some point in the system. To ensure that the refrigerant is in a liquid state prior to entering the expansion valve, a sight glass is installed in the liquid line just upstream of the expansion valve. Vapor bubbles present in the sight glass are one indicator of low refrigerant charge. Vapor formation in the liquid line can also occur due to excessive pressure drop in the liquid line.

A visual inspection of the site glass revealed that "bubbles" were present in the odd loop. In addition, the evaporator pressure and compressor suction superheat temperature were monitored using the data acquisition equipment. Fluctuating pressure and/or high suction superheat indicates that the evaporator is being starved of refrigerant. *Figure 3* depicts actual data showing how the odd loop evaporator pressure fluctuates due to the presence of vapor in the liquid line. The evaporator pressure in the even loop, which is operating normally, is also shown. The expansion valve is sized to deliver liquid, which has a much higher density and therefore requires a smaller area for flow to admit a given amount of refrigerant.

In direct-expansion refrigeration systems, the refrigerant control device modulates the flow of refrigerant to the evaporator in response to the superheat generated at the evaporator outlet. In this system, the design superheat is 7°F to 8°F (3.9°C to 4.4°C). *Figure 4* shows that the superheat on the even loop is maintained at design operating conditions while the odd loop with vapor in the liquid line fluctuates between 10°F and 30°F (5.5°C to 16.7°C). The uncontrolled superheat is an indication that the expansion valve is out of control (i.e. hunting), resulting in the evaporator being starved of refrigerant on the odd loop.

In this case, when the valve is always wide open, the outlet superheat and thus the mass flow of refrigerant, is controlled only by the pressure drop across the valve. As can be seen by comparing *Figures 3* and *4*, the superheat increases as the pressure drop decreases, indicating the mass flow of refrigerant is decreasing with decreasing pressure drop. As the pressure drop increases, the superheat decreases, indicating the mass flow of refrigerant is increasing with increasing pressure drop.

To isolate the cause of vapor in the odd loop liquid line, the liquid line pressure just ahead of the expansion valve was measured using a pressure gage and compared to the transducer pressure measurement at the outlet of the condenser. From this information, a pressure drop was determined and compared to the calculated pressure drop between these two points due to fluid frictional effects. The calculated pressure drop and measured pressure drop were in close agreement so the problem was determined to be a low refrigerant charge and not excessive liquid line pressure drop.

To effectively resolve the problem of low refrigerant charge in the odd loop, a refrigerant leak check was performed on the entire system. A small leak in a diaphragm of the head (discharge) pressure controller was identified and repaired. Once the leak repair was made, a total of 135 lbs (61 kg) of HCFC-22 were added to the system.

The effect of low refrigerant charge on the system performance was calculated using actual operating data. The coefficient of performance (COP) for the system, when it is operating as designed with an evaporator pressure of approximately 40 psia (275 kPa) and a condenser discharge pressure of approximately 235 psia (1620 kPa), is about 1.7. The low refrigerant charge in one bank reduced the refrigeration system performance by approximately 33%. The reduction in system performance translates directly into increased annual system energy costs. Since only one of the two refrigerating

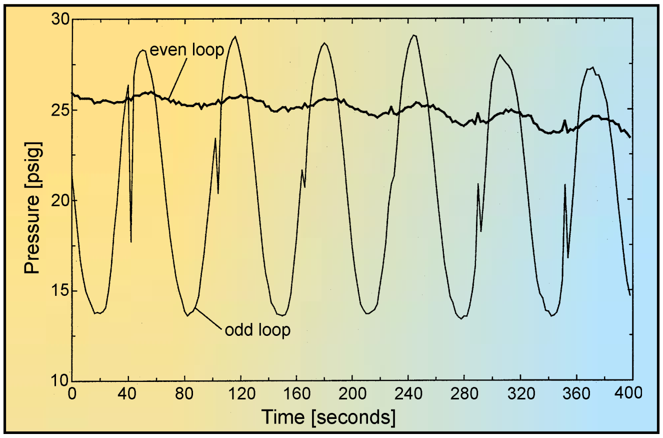

Figure 3: Evaporator pressure variation over time for even and odd loops.

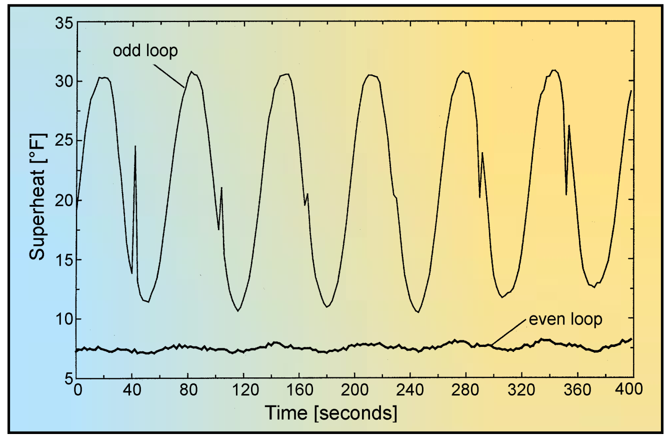

Figure 4: Refrigerant superheat at outlet of evaporator for the even and odd loops.

loops had this problem, the performance of the overall system was reduced by 17%, which equates to an annual operating cost penalty of $7,800.

This unnecessary operating cost could have been avoided by installing proper monitoring equipment on the system (see sidebar). For the site investigated, the material cost of the necessary monitoring components described in the sidebar is approximately $420 per bank. The pressure switch needs to be wired into the existing alarm system. Assuming an installation cost of $2,000 for these components, the net savings for the site (due to early detection and resolution of problems) could have been $7,439, assuming that the problems which were detected and corrected existed for one year.

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Refrigeration System Control

The compressor discharge (or head) pressure is a function of the heat transfer rate in the condenser, total heat rejected and the ambient wet bulb temperature (for evaporatively condensed systems). All else being the same, higher head pressures require additional compressor work and result in lower refrigeration system efficiency. However, a minimum pressure difference is required across the thermostatic expansion valve to ensure proper operation and stable control, so the head pressure is a controlled system parameter. The method used for head pressure control in this system is to cycle the evaporative condenser fan on and off as needed to maintain the head pressure (actually the saturated condensing pressure) between 220 and 250 psia (1517 to 1725 kPa) at all times during the year. During times when the fan operates, the heat rejection capacity of the evaporative condenser significantly increases and the condensing pressure (and discharge pressure) fall. When the fan cycles off, the heat rejection capacity of the condenser is diminished requiring the saturated condensing temperature (and pressure) to rise allowing heat to be rejected to the ambient environment.

Components on the refrigerant-side of the system are selected based on design refrigeration loads and design ambient conditions. For large capacity systems, the *1997 ASHRAE Handbook—Fundamentals* recommends two or more compressors with multistage thermostats to control the sequence of compressor operation. Compressors should be selected with sufficient refrigeration capacity to "pull-down" (rapidly decrease the temperature of) the rink during system start-up. The condenser selection and fan control should be based on:

- Maximum expected outdoor air wet-bulb temperature (evaporative condensers and cooling towers).
- Ability to operate over a wide range in capacities.
- Freeze protection for the water (for evaporative condensers and cooling towers).

The easiest way to account for the varying outdoor operating temperatures is to maintain a controlled high head pressure. The high head pressure also ensures a large pressure differential is available to deliver the proper flow of refrigerant through the expansion valve. Direct-expansion refrigeration systems for ice arenas and other applications are commonly designed in this manner[4]. One advantage of this control strategy is that it provides built-in freeze protection for the evaporative condenser by maintaining a high refrigerant saturated condensing temperature and thus a relatively high evaporative condenser return water temperature. Additionally, it ensures a high-pressure difference across the expansion valve and therefore favorable control characteristics.

Although a minimum head pressure is needed to maintain the required pressure drop across the expansion valve for refrigerant feed, it is easy to make the mistake of "over-designing" the system. Expansion valves are supplied with rated mass flows for given pressure drops. According to manufacturers' data and studies done by Vinnecombe and Ibrahim (1991), expansion valves have an approximately 25% reserve capacity margin above and beyond limits identified in data provided by manufacturers. To optimize system performance, it is necessary to determine the

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minimum allowable condenser pressure for a given evaporator pressure. This lower limit is determined by examining the minimum required expansion valve pressure drop as provided by the manufacturer. The reserve capacity of the expansion valve can be used as the system design margin of safety.

A minor change in the head pressure control strategy to further reduce or "float" the head pressure can have a significant effect on refrigeration system performance. With "floating head pressure control," the head pressure changes as a function of the ambient wet bulb temperature (for water cooled and evaporatively condensed systems). For the equipment installed in the ice arena, a minimum pressure difference across the expansion valve of 70 psi (483 kPa) was required to ensure proper thermostatic expansion valve control. In this system, the evaporator pressure is nearly constant at approximately 40 psia (275 kPa) year-round. A conservative minimum head pressure for this system needed to provide reliable thermostatic expansion valve control is 155 psia (1070 kPa), which corresponds to a saturated condensing temperature of 55°F (12.8°C) for HCFC-22.

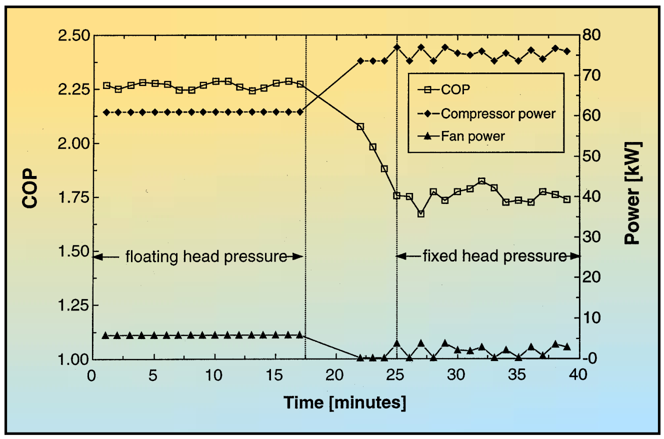

Figure 5: Measured COP, compressor power and fan power for floating and fixed head pressure control.

This minimum head pressure provides a pressure drop of approximately 115 psi (793 kPa) across the expansion valve which is well above the minimum required pressure drop of 70 psi (483 kPa). The original control strategy maintained the head pressure between 220 and 250 psia, (1517 kPa to 1725 kPa) providing far more pressure difference across the expansion valve than required.

After the problems noted earlier were corrected, the control system at the Madison Ice arena was modified to allow the head pressure to float and additional operating data were then taken. *Figure 5* depicts these results. The coefficient of performance is directly calculated from measurements on the refrigerant-side of the system. As evident in *Figure 5*, the system operates with a COP of approximately 2.25 when the head pressure is allowed to float. The COP decreases drastically to approximately 1.75 when the head pressure control is changed to its original setpoint of 220 psia (cut-in) and 250 psia (cut-out). The outside air wetbulb temperature was approximately 50°F during this experiment. According to simulation program[3], the COP using floating and fixed head pressure at 50°F (10°C) wetbulb temperature should be 2.15 and 1.65, respectively. This is a difference in operating efficiency of approximately 23%. When comparing the actual and predicted COP values, the difference was found to be approximately 5%, which is within the experimental error of the measurements.

One perceived disadvantage in using floating head pressure is that the evaporative condenser fan continuously operates rather than cycling on and off. This constant operation translates directly into increased energy consumption for the fan. However, as shown in *Figure 5*, the increased fan demand and energy consumption are more than compensated by the decrease in compressor demand and energy consumption. The data for the first 17 minutes in *Figure 5* show the compressor and fan power required during floating head pressure control. The compressor power substantially increases (from about 61 kW to 75 kW) and the fan power decreases (from approximately 5.7 kW to an average of about 2 kW) when the system is operated using fixed head pressure. Fan power was accounted for in our analysis of fixed and floating head pressure control strategies.

We recommend that all systems be evaluated for operation under a floating head pressure control strategy. System controls must be provided to allow the head pressure to float to predetermined minimum. The expansion valve installed in the system or the maximum heat rejection capacity of the condenser dictates the minimum head pressure. If the selected expansion valve is undersized, it will starve the evaporator of refrigerant during low outside air ambient conditions. If the expansion valve is oversized, it may cause liquid refrigerant to enter the compressor as well as extreme fluctuations in evaporator pressure.

The yearly operational cost of the site with its current "fixed head pressure" control strategy was calculated to be $45,618. If a "floating head pressure control strategy" is implemented on both loops the total annual operating cost would be $36,020, based on the predictions of the computer model and actual data taken after control changes where implemented. The floating head pressure control strategy results in an estimated yearly savings of $9,598—a 21% decrease in energy costs for a simple control change.

Conclusions

It is almost always cost effective to install proper monitoring equipment in large refrigeration systems to detect and correct problems early, before significant operating cost penalties or equipment damage occurs. Without monitoring equipment, the owner/operator of the equipment may, unknowingly, spend thousands of dollars on energy. Some of the recommended monitoring equipment are listed in the sidebar.

The major finding in this study is that the operating cost of large direct-expansion refrigeration systems can be significantly reduced by a simple change in the condenser control strategy. To improve system efficiency, the head pressure should be allowed to "float" with varying environmental conditions to a minimum value dependent upon the characteristics of the installed thermostatic expansion valves. The floating head control strategy would cause the fan to cycle on and off at conditions of extremely low outside air temperatures so that a minimum pressure difference across the thermostatic expansion valve is maintained. For the ice arena system investigated, the additional condenser fan and pump power required is greatly exceeded by the reduction in power needed to operate the compressors.

Although not specifically analyzed, floating head pressure control strategies will also prolong component life. Since floating head pressure reduces compressor operating pressure ratios, wear and tear on pistons, cranks, connecting rods and wrist pins are greatly reduced. In addition, condenser fan starts and stops are minimized prolonging fan belt and motor life.

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