Fixed and Floating Head Pressure Comparison for Madison Ice Arena

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A case study was performed on the refrigeration system serving the Madison Ice Arena which 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 semihermetic reciprocating compressors having a total capacity of 103 tons (355 kW). The shell-and-tube evaporator has two separate direct-expansion refrigerant circuits. A parallel rack of three compressors serves each refrigerant circuit. On the shell side of the evaporator, an ethylene glycol-water solution is cooled to the design leaving fluid 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 - 1725 kPa). A schematic of the system is shown in Figure 1.
Figure 1: Schematic of the ice rink refrigeration system
Refrigeration System Head Pressure Control

The compressor discharge (or head) pressure is a function of the heat transfer rate in the condenser 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, so the head pressure is a controlled system parameter. The original method employed 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 (1517-1724 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) falls. When the fan cycles off, the heat rejection capacity of the condenser is diminished requiring the saturated condensing temperature (and pressure) to rise for rejecting heat 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, ASHRAE (1993) recommends two or more compressors with multistage thermostats to control the compressor sequence of 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
- 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 in the condenser. The high head pressure ensures a large pressure differential is available to deliver the proper flow of refrigerant through the expansion valve. During the course of this study, it was found that ice arenas are commonly designed in this manner (Cox, D., Rink Tec International, Little Canada, MN, personal communication, 1998). 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 condenser return water temperature. Additionally, it ensures a high pressure difference across the expansion valve and therefore favorable control characteristics. Other large refrigeration systems, such as the systems for refrigerated warehouses, operate with a similar control strategy.

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 Vinnicombe and Ibrahim (1991), expansion valves have an approximately 25 percent reserve capacity margin compared to data provided in the manufacturers’ catalog. To optimize system performance, it is necessary to determine the 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 “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 control. In this system, the evaporator pressure is, essentially, 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. 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, (1517-1724 kPa) providing far more pressure difference across that expansion valve than required.

The control system at the Madison Ice arena was modified to allow for floating of the head pressure in accordance with the recommendations in Brownell’s study of the system (1998). Once these control changes were implemented, additional data was taken to determine the specific operating impact on the system. Figure 2 depicts these results. The coefficient of performance is directly calculated from data taken on the brine side of the system and the refrigerant side of the system. The refrigerant side COP is used in this analysis to allow for comparison with previous work (Brownell, 1998). As evident in Figure 2, 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 a set point of between 220 and 250 psia. This is a decrease in system operating efficiency of approximately 22 percent. The outside air wetbulb temperature was approximately 50°F during this experimentation. According to Brownell’s study, the COP using floating and fixed head pressure at 50°F wetbulb temperature were predicted to be 2.15 and 1.65, respectively. This is a difference in operating efficiency of approximately 23 percent. When comparing the actual and predicted COP values, the difference was found to be approximately five percent, which is within experimental error.

One perceived disadvantage in using floating head pressure is that the fan is constantly operating rather than cycling on and off. This constant operation translates directly into increased energy consumption for the fan. However, as shown in Figure 3, the increased energy costs are offset by the decrease in fan power.
Figure 2: System Coefficient of Performance

- \( \text{COP}_{\text{REFRIGERANT}} \)
- \( \text{COP}_{\text{DATA}} \) (GLYCOL)

TIME (Minutes)

COP

0.0 0.3 0.5 0.8 1.0 1.3 1.5 1.8 2.0 2.3 2.5 2.8

0 5 10 15 20 25 30 35 40
The first part of the graph depicts the compressor and fan power required during floating head pressure while the point where the compressor power substantially increases and the fan power decreases is where the system was being operated using fixed head pressure. As indicated in the figure, the compressor power increases from 61 kW to approximately 75 kW while the fan power decreases from approximately 5.7 kW to an average of 2 kW. The additional fan power was accounted for in this analysis.
Conclusions

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 control strategy for the condenser fan. Many systems are designed to maintain high condensing pressures. However, 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 only 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 fan power required is greatly exceeded by the reduction in power needed to operate the compressors.

The overall annual cost savings for this system using floating head pressure control is predicted to be $9600 (Brownell, 1998). The control change required to realize these savings cost the Madison Ice Arena approximately $1000. This amounts to a payback period of less than two months.
References

