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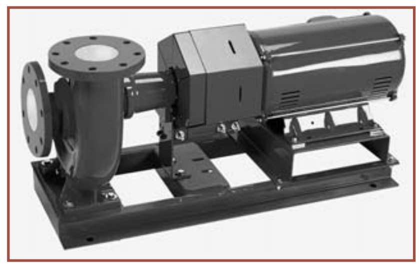
Pump Selection for Secondary Refrigerant Systems

Think of secondary refrigerant systems simply as hydronic systems that operate at unusually low temperatures compared to more common cooling systems. Given the variety of secondary refrigerants available and the increasing popularity of using secondary refrigerants instead of direct expansion coils, it is wise to consider some of the problems that may result if some basic pump selection principles are ignored. Here, we will explain how the usual practices in hydronic system pump selection must be modified when secondary refrigerants are being pumped through large systems.

In cold storage facilities and warehouse freezers in food processing plants and supermarket freezers, dairy products, meats, etc., must be stored at temperatures less than 41°F (5°C). Water at 35 to 40°F (2 to 4°C) certainly could be used to keep the food cold. It wouldn't freeze at that temperature, but the log mean temperature difference (LMTD) – a measure of the average overall temperature difference between the water and the food products – would be small, so very large case coils would be required. Also, there would be little room for error in operating the refrigerant plant at temperatures so close to the freezing point. Secondary refrigerants are liquids that can operate at much lower temperatures, increasing the LMTD, resulting in the use of smaller coils.

Secondary refrigerant systems also reduce the possibility of primary refrigerant loss. Some primary refrigerants are hazardous or environmentally un-friendly, and many are expensive to replace.



Direct expansion coils require the use of long primary refrigerant piping systems with many fittings. Each fitting is a potential point of refrigerant loss. In contrast, the secondary refrigerant in the coil piping poses little risk or cost if it should leak. In secondary refrigerant systems, it is a lot easier to control and contain the smaller volume of primary refrigerant in the chiller equipment room (figure 1).

The required pump head can be calculated by applying Bernoulli's equation (figure 2). Bernoulli's relationship applies to any pumping system, but the typical hydronic system is a special case. Points a and b could be defined at the same point because the liquid re-circulates, picking up heat at one heat exchanger, and dropping it off at the other. In closed systems, the first three terms of the Bernoulli equation go to zero, and pump head is determined solely by the friction loss of the system.

The pump head required in a closed-loop system is always exactly equal to the friction loss of the system.

System Friction Loss

The system friction loss includes the losses in components such as the evapo-

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rator heat exchanger, system air separator, and cooling coils. Equipment manufacturers provide these component head losses measured in feet of head loss or pounds per square inch difference (psid) at some specific flow rate. In large systems, pipes and fittings can represent a significant amount of friction loss. The total friction loss of all these components, calculated at the design flow rate, determines the pump head required.

The Darcy-Weisbach equation is a commonly used empirical expression for friction head loss in the piping:

$$h_{friction} = f\left[\frac{L}{D}\right]\frac{V^2}{2g}$$

where:

$h_{friction}$ is the friction head loss measured in feet

f is a friction factor

L is the length of the pipe in feet

D is the pipe diameter in feet

V is the average flow velocity in ft/sec

g is the gravitational constant in feet per second/second

Some obvious conclusions from the Darcy-Weisbach relationship include:

- Long, slim pipes have greater friction loss than short, wide pipes, all else being equal.
- Small changes in fluid velocity result in large changes in friction head loss because the velocity term is squared.

Interpreting the Darcy-Weisbach relationship a little bit more broadly, we will use it to represent the friction loss of all the components in the system, which results in the following system curve relationship.

$$\frac{h_1}{h_2} = \left[\frac{Q_1}{Q_2}\right]^2$$

where:

Figure 1. Determining the Flow Rate Required

The liquid flow rate required to transport a given heat load is determined by the density of the liquid, the specific heat of the liquid, and the expected temperature change (DT) as the liquid transfers heat from the process. Flow rate is:

$$\text{Liquid flow rate}\left[\frac{\text{gal}}{\text{min}}\right] = \frac{\text{Heat Load}\left[\frac{\text{BTU}}{\text{hr}}\right]}{\frac{8.34\ \text{lb}}{\text{gal water}}\ x\ \frac{60\,\text{min}}{hr}\ x\text{Liquid Specific Gravity}x\text{Liquid Specific Heat}\left[\frac{\text{BTU}}{\text{lb °F}}\right]x\Delta T[°F]}$$

Figure 1. Heat Load is the rate of heat transfer from the case coils required to maintain acceptable product temperatures.

h is head loss in feet

Q is flow in gallons/minute

Pump Selection

Pump performance is displayed on a pump curve based on standard tests using water. When a system curve is superimposed on a pump curve, their intersection must occur at the flow rate that will make the pump head and the system head loss equal.

For example, assume that one of these pumps will be used for a system that requires a design heat load of 167 tons refrigeration at 8°F ΔT. Water flow rate required, assuming specific heat equals 1, is 500 gal/min. Total system friction head loss, calculated at design water flow, is 50′.

Figure 3 is the pump curve and system curve generated by selection software for a 1,750 rpm pump. It is a closed system with design point at Q_1 of 500 gal/min and h_1 of 50′ of friction head loss. Note that the computer has been programmed to select a non-overloading motor, in this case, 10 hp.

So far, all calculations and the selection of the pump have been based on the use of water as the circulating medium. We'll see that the properties of typical secondary fluids can have a marked effect on the pump and motor selection.

Figure 2. Determining the Pump Head Required

Bernoulli's equation:

$$E_{pump} = \left[\frac{P_b - P_a}{W}\right] + \left[Z_b - Z_a\right] + \left[\frac{V_b^2 - V_a^2}{2g}\right] + h_{friction}$$

where:

E_{pump} is the pump head required to move water from some point "a" in the system to another point "b", measured in feet of head

P is the pressure at points a or b, in lb/ft^2

W is the density of the liquid in lb/ft^3 (In this expression, we're assuming it doesn't change in passing from a to b, or the changes are small enough to ignore.)

Z is the elevation of the liquid at a or b, in feet, measured from any constant reference level

V is the velocity at a or b, in ft/sec

g is the gravitational constant, 32.2 ft/sec/sec

h is the friction loss in the pipes, heat exchangers, coils and fittings in ft-lb of work per pound of liquid required to overcome friction.

Secondary Refrigerants

There are many fluids that can be used as secondary refrigerants. These liquids have a wide range of properties; many are quite different as compared to water. For example, at 32°F:

- Specific heat ranges from 0.21 to 0.84 BTU/lb-°F
- Specific gravity ranges from 0.77 to 1.77
- Viscosity ranges from 0.37 to 66 cP

The effect of these properties on pump selection are seen in the following example: Assume heat load is 167 tons refrigeration effect. Instead of water with specific heat equaling 1, a fluid with spe-

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cific heat of 0.85 will be used. The decrease in specific heat increases the required flow from 500 to 588 gal/min, assuming that heat load and ΔT are held constant. Let's look at how this increase will affect the pump and motor selection.

Using water at a design point of 500 gal/min and 50′ of head, figure 3 shows that a 7.75" impeller would have an efficiency of about 81 percent, operating just to the left of the pump's best efficiency flow, a strategy often used in closed loop pump selections. Brake horsepower (BHP) required to pump standard density water at this point would be:

$$BHP = \frac{Flow(gpm)xHead(feet)xSG}{3960x\eta_{pump}}$$

$$BHP = \frac{500x50x1}{3960x0.81}$$

$$BHP = 7.8hp$$

where:

BHP is the brake horsepower required by the pump at the specified flow, head and efficiency.

h_{pump} is the pump efficiency at the specified conditions.

The 10 hp motor could easily provide the 7.8 hp required by the pump at this point. Notice that the 7.75" impeller cannot intersect the system curve at 588 gal/min. A new value for system head loss must be calculated based on the increased flow rate. The system curve didn't change, it simply has more head loss at the increased flow.

$$\frac{h_1}{h_2} = \left[\frac{Q_1}{Q_2}\right]^2$$

$$\frac{50}{h_2} = \left[\frac{500}{588}\right]^2$$

$$h_2 = 69\,feet$$

To provide that increased head, a larger, 8.875" impeller would be required. Suppose that the secondary refrigerant had a specific gravity equaling 1.1, as well as having the lower specific heat. If this were to happen, pump horsepower requirements would rise to:

$$BHP = \frac{588x69x1.1}{3960x0.825}$$

$$BHP = 13.7hp$$

Even if the 10 hp motor had a service factor of 15 percent, it would still be overloaded under these conditions; and remember, not all motors have a 15 percent service factor. While it is not necessarily true that decreases in specific heat are always accompanied by increases in specific gravity in a secondary refrigerant, this example illustrates that the pump can require a larger impeller and more horsepower when these properties change.

In many secondary refrigerant systems, the increased flow required by the decreased specific heat is offset by an increase in design ΔT. In this example, increasing the design ΔT from 8 to 9.4°F would reduce the required flow rate back to 500 gal/min.

In selecting coils for secondary refrigerant systems, consider the combined effects of liquid supply temperature and design ΔT on coil performance. In general, a larger design ΔT, supply temperature held constant, means that the coil must be provided with a higher proportion of design flow in order to achieve some given percentage of design heat transfer. In other words, system flow balance becomes more important. The selection of coils, piping circuits, pipe sizes, control valves, and balancing devices will determine the proportion of flow going to each coil.

Changes Due to Increased Viscosity

Another important change at the pump occurs because the secondary refrigerant may be more viscous than water. Viscosity and density are not the same thing. For example, oil is clearly more viscous than water, but it also floats on water, demonstrating that it is less dense.

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Figure 3. Pump Curve and System Curve

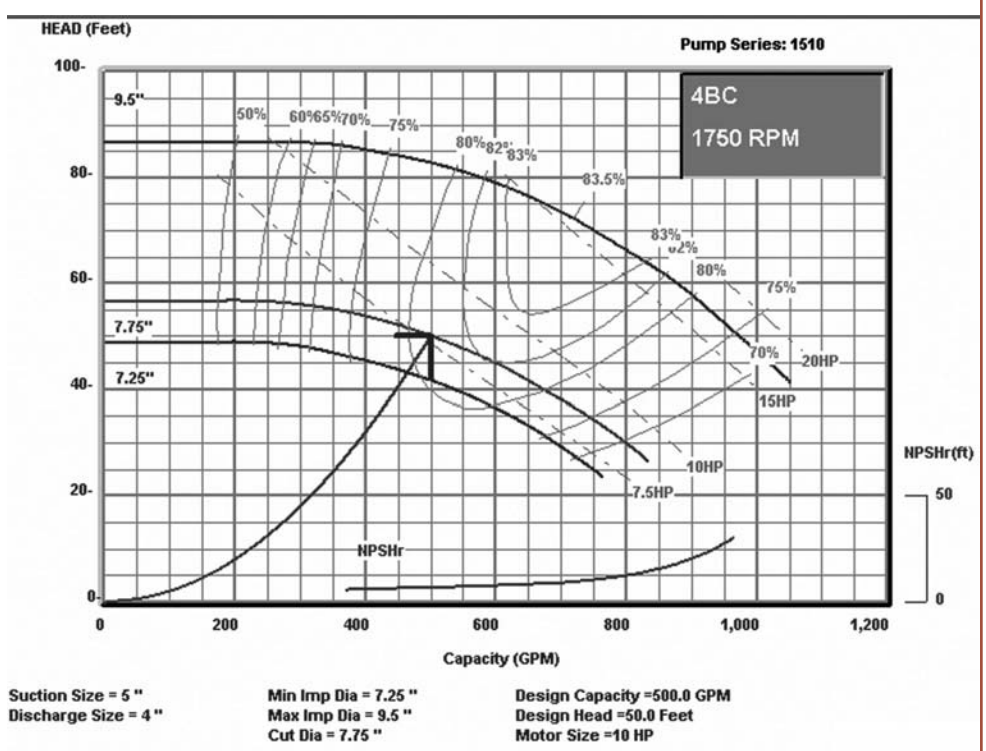

Figure 3. A pump curve and a system curve is generated by software that shows how different criteria affect a pump's efficiency. In this instance, a 7.75" impeller has an efficiency of about 81 percent.

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Viscosity is usually described as "a resistance to shear." Imagine a pipe full of liquid. Further, imagine that the liquid flows in an infinity of concentric cylinders from the largest at the pipe wall, to the tiniest at the center of the pipe. If the cylinders near the center of the pipe are moving faster than those close to the pipe wall, then there must be a slippage between cylinders, a shearing action as each cylinder slides past its neighbors. In fact, that's about what happens when viscous liquids flow. The velocity as measured across the pipe section is not a constant. In "turbulent" flow situations velocities are more nearly constant, especially near the center of the pipe, and therefore the "sliding cylinders" example is not very accurate. In "laminar" flow situations, there's a marked decrease in velocity from the center toward the pipe wall, and the sliding cylinders analogy holds up quite well.

For a given piece of pipe with a given flow, the difference between laminar and turbulent flow is a result of higher viscosity, which tends to make the flow more nearly laminar, everything else being equal. This becomes important at the pump because the pump must apply more work to the liquid to overcome the increased shearing action in viscous liquids.

The viscosity of a liquid can be measured in many different ways. The centipoise (cP) is one common way of doing it. Standard water at 60°F (16°C) has a viscosity of about 1.13 cP. Some secondary refrigerants are a lot more viscous, ranging as high as 66 cP. An increase in liquid viscosity has the effect of:

- Changing the pump curve, at least a little bit.
- Reducing the pump efficiency, at least a little bit.
- Increasing friction loss in the piping system, which means that the pump must apply more head to overcome that friction.

The pump curve in figure 4 was plotted from tests carried out under standard conditions using water at about room temperature. Because the pump curve ordinate is stated in feet of head, the same performance curve can be used with liquids of different density without modification as long as they have similar viscosity. One foot of head represents the application of 1 ft-lb of work to each pound of liquid passing through the pump. Because a pound of one liquid weighs the same as a pound of every other liquid, the pump curve is a general energy statement. The extra shearing action required to pump viscous liquids can deform the curve. The greater the difference in viscosity between water and the pumped liquid, the greater will be the effect on the pump curve. Compared to water, the viscous liquid will get less work applied per pound (less head) at the same flow rate (figure 4).

Changes in Head and Efficiency

The efficiency of a pump, like everything else on the pump curve, is determined by standard tests. As usual, efficiency is a fraction; "what you get out of the machine" divided by "what you have to put into the machine." We get "water horsepower" as a result of operating the pump. Water horsepower (WHP) is the time rate of work applied to each pound of liquid passing through the pump. We pay for the electrical energy converted by the motor into the "brake horsepower" (BHP) applied at the pump shaft. Incidentally, that's also the derivation of the brake horsepower relationship we used earlier.

The pump must apply more work to overcome the shearing action of the more viscous liquid, so the pump becomes less efficient when compared to the water-tested pump curve. If head, flow, and specific gravity didn't change at all, a loss of efficiency would mean greater BHP, a larger motor, and higher operating costs.

The effect of increased viscosity on system friction loss is the final, and possibly the most important part of this discussion. Increasing the viscosity of the liquid can increase the friction factor in the Darcy-Weisbach equation, and that can make the system curve steeper than the one based on water (figure 5).

Figure 4. Viscosity Effect on Pump Performance

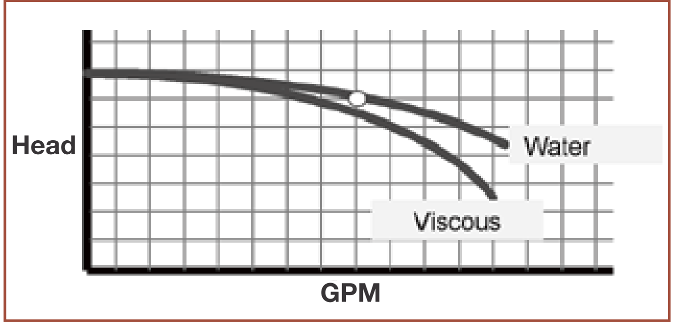

Figure 4. The greater the difference in viscosity between water and the pumped liquid, the greater the effect will be on the pump curve.

Figure 5. Viscosity Effect on the System Curve

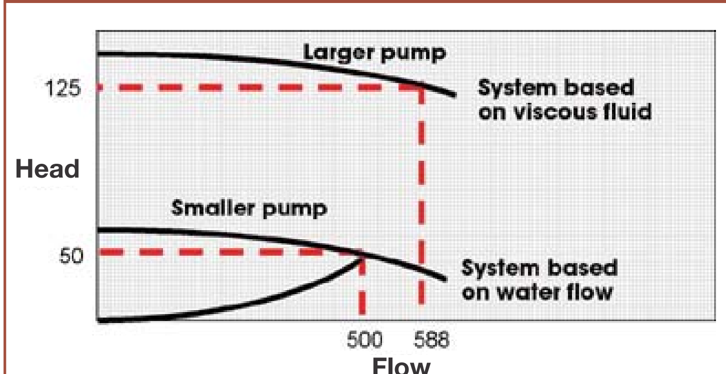

Figure 5. Increasing the viscosity of the liquid can increase the friction factor in the Darcy-Weisbach equation, and that can make the system curve steeper than the one based on water.

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