

## BACK TO BASICS: CENTRIFUGAL REFRIGERANT PUMPS

### INTRODUCTION

In the last issue of the **GOLD FRONT**, we covered the basics of centrifugal pumps as applied to the movement of non-volatile secondary fluids (water, glycols, salts). In this issue, we will extend these concepts as we look at centrifugal-type **refrigerant** pumps. There are a couple of key differences between pumping a non-volatile fluid like water or glycol and a volatile fluid like a refrigerant: 1) a refrigerant is usually at (or at least near) saturation conditions (i.e. its boiling point); and, 2) from the pump's perspective the movement of a refrigerant within a system appears to be "open."

The challenge with the first difference is keeping the refrigerant from flashing to a vapor as it enters the pump. Refrigerant flashing is possible because the liquid refrigerant being pumped is usually at its boiling point (i.e. saturation temperature). Attempting to pump a flashing (i.e. vaporizing or boiling) liquid is not recommended and leads to a condition known as *cavitation*. During cavitation, a pump will sound like it is

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trying to circulate gravel. Under normal operation, centrifugal pumps should be dead quiet.

To some degree, the second difference noted above, open pumping circuits, is related to the operating characteristics that arise in the application of refrigerant pumping in industrial refrigeration systems. An open pumping loop is one in which the pump suction pressure has no direct coupling with the pump discharge. This certainly is the case when the fluid supplied to the pump suction is open to atmosphere at some point. In the case of a mechanically-pumped liquid overfed refrigerant loop, the liquid supplied to the pump is “open” to the compressor’s suction at the free surface of liquid refrigerant in the recirculating vessel.

In a closed loop, the pump only has to develop sufficient head (pressure) to overcome frictional and form losses in the closed circuit piping system. A good example of an open pumping loop found in most refrigeration systems is the condenser water loop.

A key difference in an open loop system is the absence of any “recovery” of the static head on the pump. In other words, a pump in an open system needs to develop sufficient head to overcome the static head due to the elevation difference that exists within the open portion of the pumping loop in addition to the frictional and form losses in the supply-side piping. In some cases within refrigerant circuits, the pump must also overcome the pressure exerted by regulators applied to evaporators or other sources of static pressure (e.g. transfer systems).

### KEY PUMP DEFINITION

Before we get into the details of centrifugal refrigerant pump operation, we need to revisit a term from the last newsletter. Remember **NPSH** stands for **Net Positive Suction Head**, and it represents the pressure (i.e. head) at the pump suction. The term “net positive” is intended to account for the *balance* of positive pressures (static, or height, and absolute pressure above the vapor pressure of the fluid) and negative pressure (losses) attributable to

## UPCOMING AMMONIA COURSES

*Ammonia Refrigeration System Safety*  
April 20-22, 2010 Madison, WI

*Design of NH<sub>3</sub> Refrigeration Systems for Peak Performance and Efficiency*  
September 20-24, 2010 Madison, WI

*Introduction to Ammonia Refrigeration*  
October 6-8, 2010 Madison, WI

*PHA for Industrial Ammonia Systems*  
September 15-17, 2010 Madison, WI

*Principles & Practices of Mechanical Integrity for Industrial Refrigeration Systems*  
November 3-5, 2010 Madison, WI

*Intermediate Ammonia Refrigeration*  
December 1-3, 2010 Madison, WI

*Process Safety Management Audits for Compliance and Continuous Safety Improvement*  
January 26-28, 2011 Madison, WI

*Energy Efficiency Improvement Strategies for Ammonia Refrigeration Systems*  
February 9-11, 2011 Madison, WI

*Introduction to Ammonia Refrigeration Systems*  
March 2-4, 2011 Madison, WI

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## NOTEWORTHY

- We have moved! Please make a note of our new address above.
- Mark your calendars and plan to join us for the **2010 IRC Research & Technology Forum** held at the Pyle Center in Madison on **May 5-6, 2010**.
- Send items of note for next newsletter to **TODD JEKEL**, [tbjekel@wisc.edu](mailto:tbjekel@wisc.edu).

fluid flow. A *NPSH* of zero (0) suggest the fluid being pumped will vaporize (boil) as it moves into the pump.

There are two (2) types of *NPSH*, the *required* (*r*) and the *available* (*a*). *NPSH<sub>r</sub>* is the minimum *required* net positive suction head in order to prevent the refrigerant from flashing; thereby, causing the pump to cavitate. The *NPSH<sub>r</sub>* characteristics of a pump will depend on its operating point and are given by the pump manufacturer. *NPSH<sub>a</sub>* is the *available* net head that exists at the pump suction considering all of the additive (static head) and negative (frictional losses, heat gains, and form losses) effects. In order to keep the pump from cavitating, the *NPSH* available to the pump must be greater than the minimum required by the pump.

$$NPSH_a > NPSH_r$$

The *NPSH<sub>a</sub>* is made up of the following pressures:

1. Pressure difference above the vapor pressure of the fluid (pressure above the fluid surface minus the fluid's vapor pressure or saturation pressure)
2. The static height of the fluid above the pump centerline
3. The pressure losses (frictional and form) due to fluid flowing through the suction piping, valves, and the pump's suction.
4. Heat gains in the piping to the pump suction.

A properly operating liquid refrigerant pump should be quiet, but keeping a refrigerant pump from cavitating is important for several reasons, and noise isn't one of them: 1) a cavitating pump does not deliver as much liquid flow, which in turn may starve some evaporators resulting in a loss of refrigeration capacity; 2) cavitation erodes the pump's impeller resulting in declining performance (see previous item) and ultimately increased frequency of impeller replacement; and 3) cavitation reduces the ability to cool the electric motor used in semi-hermetic pumps.

If your pump is cavitating, there are a couple of options to remedy the situation. First, obtain a pump curve for your pump and determine the impeller diameter (from tag). Next, read the discharge pressure on the pump to determine the pressure rise developed. With this information, you now can look at the manufacturer's pump curve and determine the flow rate that is being delivered and *NPSH<sub>r</sub>* where the pump is operating along the pump curve. If the pump is operating "out on the curve" (toward the right side of the pump curve), you may need to adjust (i.e. close) the metering valves on the liquid overfed evaporators being served by the pump to decrease the pump's overall flow rate. If the operating point is reasonable (meaning not requiring an excessive amount of *NPSH<sub>r</sub>*), and if you have a capacitance probe on the vessel, then you can raise operating liquid level in the vessel. Recognize however, when you do this the vessel will not have as much surge volume in the recirculator (before making this change, perform a "management of change" in accordance with your plant's PSM program). The surge volume is the volume between the fill level and the high-level compressor cutout and is required to accommodate unexpected return of liquid to the recirculator without shutting down the refrigeration system. Another remedy may be to re-insulate the pump suction piping. Visible frost on the suction piping is an indication of a loss of insulation integrity which translates into more heat gain to the refrigerant. Heat gain erodes the *NPSH<sub>a</sub>* by raising the temperature of liquid flowing to the pump.

## REFRIGERANT PUMP ANATOMY

A centrifugal refrigerant pump is, fundamentally, no different from the pump we showed in the last article; however, there are some design details that differ in order to decrease the pressure loss through the suction eye and into the pump. This means that the suction eyes and impeller diameters are usually larger than water pumps of the same capacity and the impeller's rotational speed is generally low - usually 1,800 rpm or less for open-drive configurations. While these changes are intended to make the pump less prone to cavitation and prolong the life of shaft seals for open drive pumps, they also may mean that the pump efficiency is comparatively lower. In contrast, semi-hermetic pumps are often designed to run at 3600 rpm. Keep in mind that the power associated with liquid refrigerant pumps in ammonia systems is low, so the energy penalty

associated with lower pump efficiency is a minor consideration.

### HOW MUCH LIQUID DO I NEED TO CIRCULATE?

In an overfeed system, the quantity of liquid refrigerant required by operating evaporators will depend on the prevailing refrigeration load as well as the additional quantity of liquid intended to keep the evaporator's interior surfaces wetted with liquid refrigerant. The total flow rate a pump needs to develop will be a function of the refrigerant conditions, the circulation number ( $N_R$ , overfeed rate + 1), and the load. The following equation can be used for determining how much liquid refrigerant needs to be pumped:

$$GPM = \frac{12,000 \cdot N_R \cdot TR}{8.021 \cdot \rho \cdot \Delta h}$$

where, 12,000 converts tons to Btu/hr and 8.021 converts gpm to ft<sup>3</sup>/hr,  $N_R$  is the circulation number,  $TR$  is the aggregate refrigeration load being served by the pump in tons,  $\rho$  is the density of the pumped refrigerant, and  $\Delta h$  is the enthalpy difference between the saturated vapor condition in the suction trap and saturated liquid in the feed vessel (i.e. the recirculator). Note that for a traditional mechanically pumped overfeed system the suction trap and feed vessel are the same recirculator vessel and the  $\Delta h$  simplifies to the heat of vaporization of ammonia at the evaporating temperature. Below is a table that shows the required recirculating gallons per minute for each ton of refrigeration load for a range of saturation temperatures and circulation numbers.

**Table 1 – Recirculated gallons per minute per ton of refrigeration (gpm/ton).**

		Recirculated Liquid Temperature [°F] / (°C)					
		-60 (-51)	-40 (-40)	-20 (-29)	0 (-18)	20 (-7)	40 (4)
Circulation Number	2:1	0.1117	0.1164	0.1216	0.1274	0.1340	0.1415
	3:1	0.1676	0.1746	0.1824	0.1911	0.2009	0.2122
	4:1	0.2235	0.2328	0.2431	0.2548	0.2679	0.2829
	5:1	0.2793	0.2910	0.3039	0.3185	0.3349	0.3537

Multiply by 0.0646 to convert table values to m<sup>3</sup>/hr per kW<sub>T</sub>

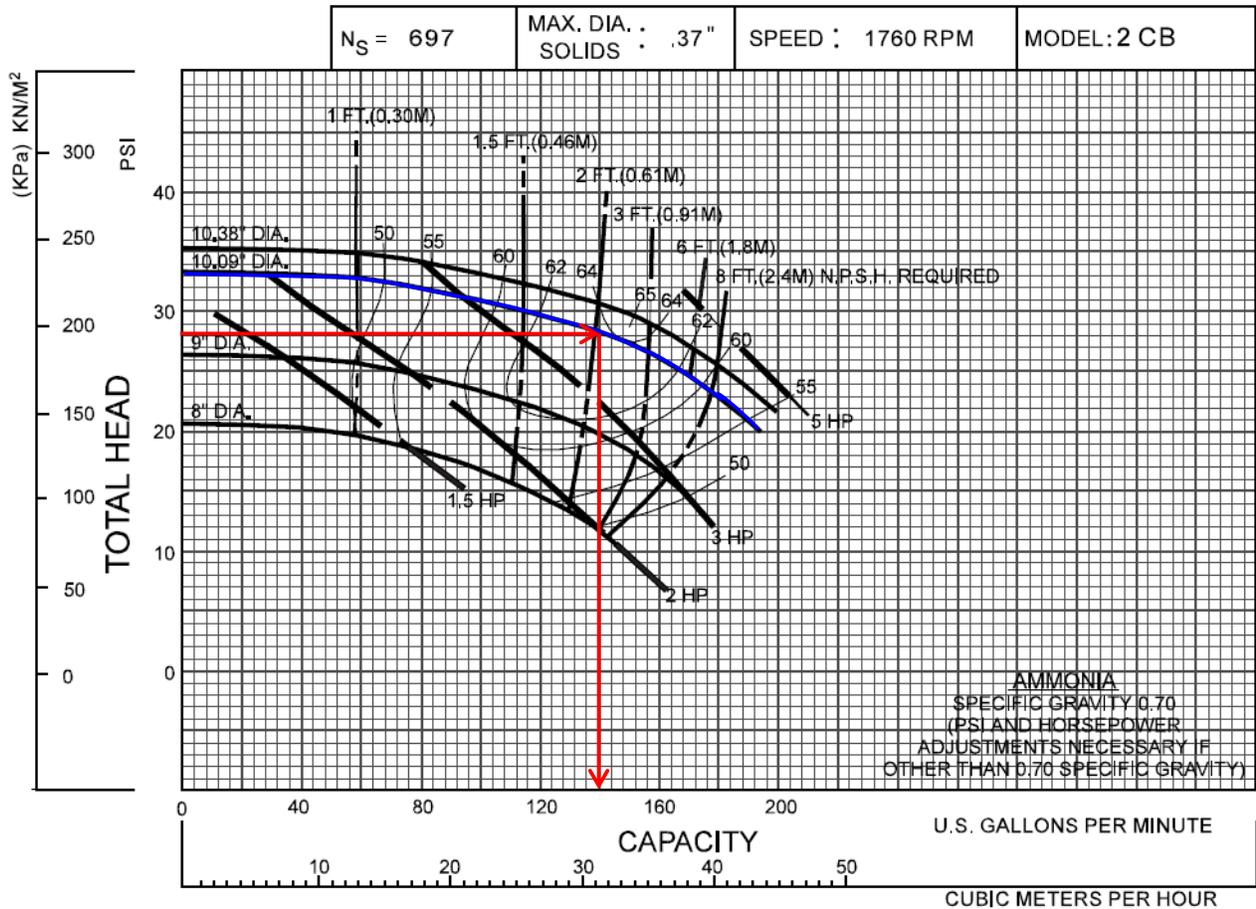
One caution with the above table is the potential misunderstanding of the difference between OVERFEED rate and RECIRCULATION rate (related to what we are calling the circulation number,  $N_R$ ). The two rates are interrelated, but they are NOT synonymous. The recirculation rate is important to designers selecting pumps and pump manufacturers because it specifies the TOTAL flow rate that the pump is required to deliver. However, evaporator manufacturers will often specify overfeed rates for their units. "Overfeed" is how much EXTRA liquid is delivered above and beyond the evaporation rate (that required to just meet the load). Hence the circulation rate is the overfeed rate plus the evaporation rate, or in equation form:

$$N_R = O_R + 1$$

A quick example: a 300 ton (1,055 kW<sub>T</sub>) load on a blast freezer operating at -40°F (-40°C) evaporating (i.e. recirculating) temperature with an overfeed rate of 3:1 (a 4:1 recirculation rate) would require the delivery of 69.8 gpm (15.85 m<sup>3</sup>/hr) to the evaporators.

### THE PUMP CURVE

Recall that a pump curve is a graphical representation of a pump's performance. So, let's review the basics of reading a pump curve for a liquid refrigerant pump. In this case, we will consider an open-drive Cornell Model 2 CB pump. The horizontal axis of the pump curve shows the pump's developed flow rate expressed in gal/min (or m<sup>3</sup>/hr). The vertical axis of the pump curve shows the generated pump head expressed in psi (or kPa, or ft or m).



The series, or “family”, of heavy lines that project horizontally from the vertical axis to the right sloping downward represent this model pump performance with varying but discrete individual impeller diameters (ranging from 8”-10.38”). The dashed lines that run on a diagonal from upper left to lower right represent the required pump power (1.5 HP-5 HP). The semicircular lines represent the pump efficiency (50%-65%). And the heavy dashed lines show the *NPSH<sub>r</sub>* (required) ranging from 1-8 ft.

As an example, consider a pump with a 10.09” (25.6 cm) impeller diameter operating with a discharge pressure of 23.7 psig (265 kPa) connected to an ammonia recirculator operating at a pressure of 8.8” Hg [-40°F (-40°C)]. Since the system is operating in a vacuum, we need to first determine the pressure developed by the pump. The saturation pressure in the recirculator vessel is 8.8 in Hg or *negative* 4.3 psig; therefore, the total head or pressure developed by the pump is 28 psi (294 kPa). The flow delivered at that head can be determined by drawing a horizontal line from the 28 psi hash mark to the 10.09” diameter impeller pump curve (blue line). This intersection point represents the pump’s operating point. Projecting a vertical line from the pump operating point down to the horizontal axis provides information on the flow delivered by the pump. In this case, the flow is 140 gpm (31.8 m<sup>3</sup>/hr). From the operating point, other operating characteristics of the pump can be obtained such as the pump efficiency (~63.5%) and operating power required. The brake horsepower tells you the required input power to drive the pump (~4 hp [~3 kW]). The last piece of information is the *NPSH<sub>r</sub>* which is 2.0 ft [0.61 m] in this case.

Alternatively, the pump horsepower requirement for the units given on the pump curve, the following formula can be used:

$$hp = \frac{\dot{V} \cdot \Delta p}{1,714}$$

where the flow,  $\dot{V}$ , is in gallons per minute and the head,  $\Delta p$ , is in psi. For the example above, the pump horsepower is 2.3 hp (1.7 kW), and the motor power is 3.6 hp (2.7 kW) which is obtained by dividing the pump horsepower by the 63.5% pump efficiency.

### WHEN DOES THE PUMP STOP MOVING LIQUID?

In a mechanically-pumped liquid overfed system, the liquid refrigerant pump is responsible for delivering liquid refrigerant into the evaporators, and that's it. Some people believe that the liquid refrigerant pump is responsible for delivering refrigerant through the evaporator and back into the recirculated liquid return (i.e. the suction header). That simply can't occur because once the saturated liquid refrigerant delivered to the evaporator begins to vaporize (i.e. by absorbing heat in the unit), the pump can no longer "push" because the liquid seal is broken. From the point within the evaporator where the liquid seal is broken, it becomes the compressor's job to return (pull) the vapor along with the overfed liquid to the suction header and, with the assistance of properly pitched piping and gravity, back to the recirculator package.

### OTHER CONSIDERATIONS

#### Open-drive vs. semi-hermetic pumps

Centrifugal liquid refrigerant pumps are available in two basic configurations: open-drive and semi-hermetic. An open-drive pump utilizes a wholly-external electric motor with a shaft open to atmosphere to drive the pump impeller. By its nature, an open drive centrifugal pump has a greater potential for refrigerant leakage due to the presence of a shaft-seal. The open-drive pump design does offer the advantage of being able to easily replace the electric motor should it fail. Generally, the impellers in open drive pumps can be trimmed once the required system head is determined.

Semi-hermetic centrifugal pump designs offer the advantage of reduced potential for refrigerant losses because the electric motor is sealed within a housing common to the pump. Disadvantages of a semi-hermetic refrigerant pump are that the electric motor is not easily serviceable and the impellers are not generally trimmable once the pump is installed.

#### Minimum flow protection

In addition to the potential for cavitating the pump as you "run out on the curve," liquid refrigerant pumps can cavitate at low flow conditions as well. Rather than cavitation being attributable to the dynamic pressure loss as the refrigerant enters the pump, this form of cavitation originates from heat caused by the inefficiency of the pump being added to the refrigerant stream. The added heat can cause the liquid refrigerant to boil leading to cavitation. To prevent this form of cavitation, a bypass line from the pump discharge back to the recirculator package supplying liquid is installed with an orifice (or metering valve) set at a manufacturer's required minimum flow. During normal operation, the heat gain from the pump power input is small, for the example we used earlier the heat gain would result in a temperature rise of 0.18°F (0.1°C).

#### Presence of EPRs on pump discharge pressure requirement

Pumping liquid to evaporators equipped with evaporator pressure regulators (EPRs) requires special attention. First, the pump discharge pressure must be higher than the EPR set pressure in order to get liquid into the evaporator. Second, it is important to realize that the pumped liquid refrigerant being supplied to an EPR-fitted evaporator is **subcooled** as it enters the evaporator. The refrigerant enters the evaporator essentially at the temperature in the recirculator (plus a little rise due to picking up heat in the pump and heat gain in the liquid piping). Since the liquid refrigerant supply temperature is below the saturation temperature corresponding to the EPR pressure setting, the liquid refrigerant entering the evaporator has to absorb heat until it reaches the saturation temperature and begins to evaporate. Evaporators operating with liquid refrigerant supply

temperatures more than 10°F (5.6°C) below its saturation temperature will likely result in poor evaporator performance.

### **Static head requirements in open loops (or why VFDs are not great for refrigerant pumps)**

Another issue with open loop pumps is that the static head requirement sets the minimum discharge pressure to get flow to the evaporators. What does that mean? It means that before there is any flow to the evaporators, the pressure required to lift the liquid to the roof must be overcome. If the pressure generated by the pump is not sufficient, the pump will operate at its minimum flow with a column of liquid standing in the riser to the roof. In other words, no flow is delivered to the evaporators. This is the main reason that variable frequency drives (VFDs) are not generally good applications for refrigerant pumping in liquid overfeed applications. Remember from the pump affinity laws in the previous newsletter, that the head (discharge pressure) generated by the pump falls with the square in the reduction in speed.

### **Hydrostatic lock-up**

Hydrostatic lock-up is the trapping of subcooled (or pressurized) liquid refrigerant with a subsequent increase in temperature which causes the trapped liquid to expand. The volume-constrained liquid results in a substantial increase in pressure with an increased likelihood of equipment failure due to the pressure rise. It is important to identify locations within a system using pumped liquid refrigerant that can trap liquid and provide suitable means of overpressure protection. [Addendum g](#) to ASHRAE Standard 15-2007 provides requirements for protection from hydrostatic lock-up. A pumped liquid line is a location where hydrostatic lock-up is a danger during maintenance procedures. In addition, the piping between the pump discharge check valve and the liquid feed solenoids must be protected from hydrostatic lock-up during a plant-wide power failure. Therefore, it is required to have a hydrostatic relief device downstream of the pump discharge service valve piped back to the recirculator vessel.

### **SUMMARY**

This article introduced concepts that are unique to refrigerant pumping with centrifugal pumps. We discussed cavitation in detail to help you troubleshoot this behavior. Hopefully this article will let you understand and troubleshoot potential problems that may exist in your refrigerant pumping systems.