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Pumped Recirculators vs CPR Feed

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Abstract

The controversy over CPRs vs. pumped recirculators has gone on for many years. Despite numerous articles and papers, each claiming energy savings for its approach, the problem has not been fully analyzed from a fundamental standpoint. Using the simple concept of conservation of mass, and employing thermodynamic properties of refrigerants, a mass and energy balance can be modeled for each type of system. This approach yields equations which can then be used to predict the mass flows required for each type of system. This allows the systems to be compared to each other in a scientific manner. The paper will illustrate how feeding cold liquid to the evaporators lowers pressure drops and increases overall system efficiencies.

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Introduction

The IIAR archives contain several papers presented over the last 30 years arguing the benefits and drawbacks of Control Pressure Receivers (CPRs) versus pumped recirculators. Often these papers have focused on the energy consumption of pumps vs. transfer drums as a method of moving ammonia liquid. The arguments have gone back and forth about warming factors, liquid/vapor heat transfer, part load operation, and the effect of vapor quality at the evaporator inlet. Proponents of CPRs use calculations which show transfer drums to be just as efficient as pumps, sometimes better than pumps at low circulation ratios. Opponents of CPRs argue that transfer drum energy losses are understated and that the operational costs for transfer drums are far greater than equivalent pumped systems.

The curiosity of this focus on pumping costs is that refrigerant pumps represent approximately 1% of the total power consumption of a typical pumped ammonia refrigeration system. On the other hand, refrigeration compressors represent approximately 66%, and evaporators and their associated line losses represent 25% of the total power consumption of a typical system. So it makes little sense to focus energy analysis on refrigerant pumps—especially for a refrigerant like ammonia which has very high latent heat. Small improvements in compressor or evaporator efficiency will far outweigh even dramatic improvements in pumping energy consumption.

This paper analyzes the evaporators and compressors for each type of system from a mass and energy standpoint. Using conservation of mass as a foundation, this paper will derive equations used to predict mass flow rates for each type of system. Using the equations developed in this paper each type of system will be compared for energy consumption.

Definitions

A complete list of the terms and abbreviations used in this paper are included in Table 1 in the appendix. This section introduces several of the concepts used throughout this paper.

Circulation Ratio

Our industry uses many different terms to describe the same thing; overfeed rate, overfeed ratio, recirculation ratio, circulating number, circulating rate, etc. However, despite the lack of agreement on a single word to use, the definition of the concept remains the same in the textbooks and handbooks.

Will Stoeckers "*Industrial Refrigeration Handbook*," p. 302, defines circulation ratio as

$$\text{Circulation ratio, n} = \frac{\text{refrigerant flowrate supplied to evaporator}}{\text{flowrate of refrigerant vaporized}}$$

The *ASHRAE Refrigeration Handbook*, 1990, page 2.3, states: "In a liquid overfeed system; the mass ratio of liquid pumped to the amount of vaporized liquid is the circulating number or rate."

The circulation ratio defines the quality of the refrigerant at the outlet of the evaporator. If the vapor quality is 50%, then it is a 2:1 circulation ratio. If the vapor quality is 33%, it is a 3:1 circulation ratio. Note that these definitions do not distinguish whether the work done by the evaporating refrigerant is useful or non-useful work. It is a simple ratio between the amount of vapor generated and the amount of liquid fed to the evaporator.

Enthalpy

The word *enthalpy* comes from the Greek word *enthalpos* meaning *to put heat into*. Its original definition was the heat change which occurs when 1 mol of a substance reacts completely with oxygen to form products at 298 K and 1 atm. Refrigeration calculations use changes in enthalpy to predict mass flows of refrigerant based on the amount of heat transferred. It is interesting to note that different thermodynamic tables may have different values of enthalpy for ammonia liquid and vapor. That is because the selection of the datum (enthalpy = 0) is arbitrary. Enthalpy, like entropy, cannot be measured directly. Tabular enthalpies are only meaningful when added to or subtracted from other enthalpies, to come up with a difference. There should be good agreement between thermodynamic tables for such things as latent heat or refrigerating effect, both of which are differences.

This paper uses thermodynamic data from The National Institute of Standards and Technology (NIST). For a datum, the ASHRAE convention is used, which designates –40°(F/C) liquid as being 0 enthalpy.

h_{fg}, The Latent Heat of Evaporation

The symbol h_f is used to represent the enthalpy of a saturated liquid, and h_g for saturated vapor. The symbol h_{fg} represents the *latent heat of evaporation*, or the difference in enthalpy between saturated liquid and saturated vapor at constant temperature. This is often found listed in thermodynamic tables.

Breaking Down Latent Heat of Evaporation

Evaporation always occurs at saturation. However, not all processes in a refrigeration system occur solely with saturated liquid and saturated vapor. For example, if warm liquid is fed to a cold evaporator, the liquid must be cooled to saturation before it can evaporate—this is called liquid cooling (LC). Flash Gas (FG) is generated when cooling the liquid to saturation. The now saturated liquid evaporates, becoming a saturated vapor, removing heat equivalent to the latent heat of evaporation (h_{fg}). The net thermodynamic result of cooling the liquid to saturation and then evaporating it is called the refrigerating effect (RE). One way to look at the latent heat of evaporation is to consider it to be the sum of the refrigerating effect (RE) and the liquid cooling (LC)

$$h_{fg} = h_f - h_g = RE + LC$$

The Refrigerating Effect (RE)

The **refrigerating effect** is the difference in enthalpy between the entering liquid and the saturated vapor in the process under consideration.

The equation for calculating refrigerating effect is

$$RE = (h_{fin} - h_{gout}) \tag{1}$$

It is common to use the absolute value of this term so that it is always a positive number.

Liquid Cooling (LC)

Work, in the form of mechanical cooling, must occur when there is a reduction in the temperature of a liquid refrigerant within a closed system. For the example of warm liquid entering a cold evaporator, the warm liquid must be cooled to its saturation temperature. This cooling load is the difference between the entering liquid enthalpy and the leaving liquid enthalpy. This difference in enthalpy will be called liquid cooling (LC).

$$LC = (h_{fin} - h_{fout}) \tag{2}$$

Flash Gas (FG)

Flash gas is the vapor generated when cooling the incoming liquid to the saturated temperature in the process being examined. Calculating the flash gas requires the use of the terms described above for refrigerating effect and liquid cooling. The flash gas load is the product of the mass flowrate of liquid (L), and the liquid cooling enthalpy difference (LC). The flash gas is simply that load, divided by the refrigerating effect (RE).

$$FG = \frac{L * LC}{RE} \tag{3}$$

Comparing CPRs and Pumped Recirculators — Evaporators

Evaporator performance is critical to refrigerating systems operating properly and efficiently. Evaporators and their respective line losses represent approximately 25% of the energy consumption of a typical refrigeration system. Mass flow differences between the two approaches are significant, and have an effect on piping pressure drops.

Calculating Evaporator Mass Flows—Pumped Systems

One of the differences between CPR and pumped systems is the temperature of the liquid fed to the evaporators. In the case of the pumped system, the liquid is at its saturation temperature in the recirculator, where the pump pressurizes it (in effect making it a subcooled liquid) and pushes the saturated liquid out to the evaporators. (Figure 1)

The vapor generated by the evaporator would be:

$$VE = \frac{W}{RE} \tag{4}$$

Where W is the refrigeration load (BTU/min) (kJ/min). (To convert Tons Refrigeration, TR, to BTU/min multiply by 200) (To convert kW to kJ/min multiply by 60).

The liquid supplied to the evaporator would be

$$L = n * VE \tag{5}$$

Calculating Evaporator Mass Flows—CPR Feed

A CPR mixes saturated liquid makeup and return overfed liquid to create a subcooled blend. This blend is pushed out to the evaporators by the difference in pressure between the CPR and the evaporator. The liquid is at a temperature above the saturated temperature of the evaporator. (Figure 1)

Since the supply liquid is warmer than the liquid coming out of the evaporator, the overfed liquid coming out of the evaporator has to be cooled along with the evaporated liquid while it is in the evaporator. This is non-useful work which the evaporator must perform in a CPR fed system.

Given the following:

$$L = n * VE \tag{6}$$

Liquid is n times vapor

$$RL = (n - 1) * VE \tag{7}$$

RL is returned (overfed) liquid

The total amount of vapor generated is the sum of the evaporator load plus the return liquid cooling load. The return liquid cooling load is essentially the flash gas generated by the liquid cooling, thus it is calculated by using Equation (3).

VE = Evaporator Load + Return Liquid Load

$$VE = \frac{W}{RE} + \frac{RL * LC}{RE}$$

Substituting for RL

$$VE = \frac{W}{RE} + \frac{(n - 1) * VE * LC}{RE}$$

Rearranging the equation to solve for VE

$$VE = \frac{W}{RE - (n - 1) * LC} \tag{8}$$

Comparing Equation (8) to Equation (4), the vapor generated by the evaporators, VE, will always be greater for a CPR system than for a pumped system, partially due to the lower refrigerating effect, and the rest due to the return liquid cooling. The CPR fed evaporator vapor (VE) is a function of circulation ratio, whereas in the pumped system, it is not. The conclusion is that the higher the circulation ratio, the greater the difference in mass flow between the two systems. It is interesting to note that the result of this equation is to lower the net refrigerating effect by the amount of the overfeed rate minus one times the difference in liquid enthalpies.

Example Calculation

Determine the difference between the mass flowrates of a pumped system and a CPR system for the following conditions:

Given:

20 TR (70.336 kW) Evaporator Load

0°F (–17.78°C) Evaporator Temperature

3:1 Overfeed

20°F (–6.67°C) CPR Feed

0°F (–17.78°C) Recirculated feed

Using:

$h_{\text{f in (Liquid at 20°F)}}$ = 64.579 BTU/# = 150.11 kJ/kg

$h_{\text{f out (Liquid at 0°F)}}$ = 42.779 BTU/# = 99.438 kJ/kg

$h_{\text{g (Vapor at 0°F)}}$ = 611.560 BTU/# = 1421.5 kJ/kg

Solving for the Pumped System

Calculate the Refrigerating Effect using equation (1)

$RE = (h_{fin} - h_{gout})$

RE = (611.56-42.779) BTU/# = 568.78 BTU/#

RE = (1421.5-99.438)kJ/kg = 1,322.06 kJ/kg

Determine vapor mass flow from the evaporator, VE using Equation (4)

$$VE = \frac{W}{RE}$$

$$VE = \frac{20 * 200 \text{ (BTU/min)}}{568.78 \text{ (BTU/\#)}} \qquad VE = \frac{70.336 * 60 \text{ (kJ/min)}}{1{,}322.06 \text{ (kJ/kg)}}$$

VE = 7.03 #/min VE = 3.192 kg/min

Determine liquid mass flow, L using equation (5)

$$L = n * VE$$

L = 3*(7.033#/min) = **21.10 #/min** L = 3*(3.192 kg/min) = **9.576 kg/min**

Solving for the CPR System

Calculate RE, the Refrigerating Effect, using equation (1)

$$RE = (h_{fin} - h_{gout})$$

RE = (611.56-64.579) BTU/# = 546.98 BTU/#

RE = (1421.5-150.11)kJ/kg = 1,271.39 kJ/kg

Calculate LC, the Liquid Cooling, using equation (2)

$$LC = (h_{fin} - h_{fout})$$

LC = (64.579-42.779) BTU/# = 21.8 BTU/#

LC = (150.11-99.438) kJ/kg = 50.672 kJ/kg

Determine vapor mass flow from the evaporator, VE using Equation (8)

$$VE = \frac{W}{RE - (n - 1) * LC}$$

$$VE = \frac{(20\ \text{TR}) * 200(\text{BTU/min})}{(546.97) - (3-1) * (21.8)(\text{BTU/\#})} \quad VE = \frac{(70.336) * 60(\text{kJ/min})}{(1271.39) - (3-1) * (50.672)(\text{kJ/kg})}$$

VE = 7.95 #/min   VE = 3.607 kg/min

Determine liquid mass flow, L using equation (5)

$$L = n * VE$$

L = 3*(7.95#/min) = **23.85 #/min**   L = 3*(3.607kg/min) = **10.82 kg/min**

For this example, the mass flowrate required for the CPR system is 13% more than for the pumped system.

Pressure Drop in Wet Suction Lines

Both the liquid and vapor mass flows from evaporators in CPR systems are greater than the corresponding flows for an equivalent pumped system. One of the reasons for this is that much of the flash gas for the CPR system is generated at the evaporator. Some of this flash gas cools the liquid which is about to be evaporated, and the rest cools the overfed return liquid. In contrast, the pumped system generates all of its flash gas at the recirculator, where it is taken to the compressor in the dry suction line. This difference in where the flash gas is generated causes an increase in the wet suction line mass flows which results in higher pressure drops in the wet suction lines for CPR systems.

Wet suction lines reach from the recirculator or accumulator to all of the overfed evaporators in the system. These lines are typically some of the longest and largest diameter pipes in the system, thus they are the most costly to install and insulate. Pressure losses in these lines are unrecoverable and represent inefficiency in the overall refrigeration system. Pressure drop in wet suction lines directly affects the performance of evaporators (with the exception of those whose suction pressure is controlled by a back pressure regulator). The only way to make up for these losses would be to operate the compressors at a lower suction pressure, which increases the energy required to operate the compression system.

Comparing Two Phase Pressure Drops for CPR and Pumped Systems

Pressure drop in a wet suction line presents a calculation challenge, as it has both liquid (incompressible) and vapor (compressible) flow in the same line at the same time. Back in the day, it was traditional to simply calculate the desired pressure drop in the line for the vapor flow, and increase the selected pipe by one size to accommodate the returned liquid. However, this method did not differentiate between different circulation ratios. The higher the circulation ratio, the higher the pressure drop in the pipe, due to the increased liquid flow.

The Beattie method of estimating two phase pressure drop (available in the IIAR Pipe Sizing Spreadsheet) provides a consistent and reasonable estimate of two phase pressure drop, and this method is used to determine piping losses. See Table 2a and b in the appendix for calculation of piping pressure drops. The tonnages, suction temperatures, circulation ratios, and CPR temperatures for these examples all came from previous IIAR papers. The pressure drop in wet suction lines is considerably higher (17%–62%) for CPR systems than for their equivalent pumped system. Circulation ratios of 3:1 result in approximately 30% higher pressure drops than 2:1 ratios.

Which Circulation Ratio is Best?

The wet suction line pressure drop calculations show that type of feed—or more importantly—temperature of feed, has a significant effect on piping pressure drops. Calculations in the previous section show that circulation ratio has a significant effect on piping pressure drops. All else being equal, the lower the circulation ratio the better. But the bigger question is whether all else is actually equal. And—does the refrigerant quality at the inlet to the evaporator have an effect on its performance? In other words, is some flash gas beneficial to evaporator performance?

Research Studies

A study by Gustav Lorentzen published in 1965 shows the overall coefficients of heat transfer for the same air cooler coil increasing for increasing circulation ratio. His article recommends circulation ratios of 3 to 5 for pumped systems.

More recently, a paper published by Bruce Nelson (ASHRAE, 1990) concluded that, "for the typical design TD of 10°F, the minimum downfeed/overfeed ratio be set at n = 4 for ammonia…The minimum recommended upfeed ratio is n = 3 for ammonia." Note that downfeed refers to a top fed coil, and upfeed refers to a bottom fed coil.

The same Nelson article highlights a study done by Chen which shows the convective heat transfer coefficient as a function of overfeed ratio and tube loading for ammonia at 20°F. One thing that is interesting to note is that the heat transfer coefficient is higher for increasing vapor quality, up to 90%, after which the heat transfer drops off dramatically. This lends some credence to the argument that some flash gas at the inlet (increased vapor quality) may help coil performance.

Manufacturer Ratings

Taking this one step further, an informal survey of evaporator manufacturers was conducted to find out the following:
- Is there an improvement in performance for coils at higher circulation ratios?
- Is there an improvement in performance for coils that are fed with liquid above the saturation temperature (CPR fed units)?

Most of the manufacturers contacted responded with coil ratings and were helpful in trying to answer the question. The results were surprising, as they did not necessarily reflect what the articles cited above indicated they would be.

First, for the pumped feed air units, the manufacturer's ratings did not show the expected performance increase for the higher circulation ratios. In fact, some of them showed a slight increase in performance for decreasing circulation ratios. Others showed no difference at all. There was at most a 1–2% difference in ratings between coils. Note that the ARI standard is 5%, so all of these coil ratings are well within the "margin of error." Despite the theoretical advantage of higher circulation ratios, coil manufacturer ratings do not give any advantage to circulation ratios higher than 2:1.

Results from the CPR fed evaporator study were similar. Some of the manufacturers showed virtually identical performance for CPR vs. pumped air units, while one showed approximately a 5% improvement in coil performance for a CPR-fed unit. Even with the one rating improvement, the ratings of the coils were all within 5% of each other (within the ARI *margin of error*). Similarly, the manufacturers showed no significant difference in performance between a CPR-fed coil and a pump-fed coil.

Evaporator Conclusions

So why is it that theory and industry practice are seemingly in opposition to each other? The answer is complicated. Some manufacturers are secretive about their ratings protocol. There has been very little research done recently on tube and fin evaporators at an academic level—all of the recent interest has been in microchannel. So, despite the fact that the questions posed were simple—the answers to those questions get more complex the harder you look at them. A possible theory as to why CPR-fed evaporators don't outperform pump-fed evaporators is that even though they have a vapor quality advantage, the mass loading through each circuit is higher, resulting in higher pressure drops. This cancels out the heat transfer improvements. A possible theory as to why 4:1 does not outperform 2:1 with a pumped system is that the tube surface is assumed to be wet no matter the circulation ratio.

Perhaps what can be taken away from this is that we, as an industry, need to have a better fundamental understanding of how circulation ratio affects coil performance. If there is no advantage to 3:1 or 4:1 over 2:1, then we would all be better off, from a wet suction line pressure drop standpoint, at a 2:1 circulation ratio.

The equations and calculations both show that mass flows and pressure drops are higher for CPR-fed units. The calculations in Figure 2a and b show that, even for low circulation ratios, the wet suction line pressure drops are 20–60% higher than for the equivalent pumped system. Consider for a moment, a system with a wet suction line loss of 5°F (2.7°C) on a pumped system. The equivalent CPR system will have 1–3°F (0.55–1.65°C) more losses than the pumped system. For a 10°F TD (5.55°C) coil, that is a 10–30% loss in capacity. These losses increase for higher circulation ratios.

Temperature Rise in Refrigerant Pumps

It is assumed in the above calculations that the pumped liquid fed to the evaporators is saturated. Pumps are not perfectly efficient at pressurizing liquid. Some of the energy input to the pump is lost due to friction and hydraulic losses. These inefficiencies show up as heat in the pumped liquid, which leads to a temperature rise in the pumped fluid.

The temperature rise in a centrifugal pump can be calculated as

$$dt = P_s \frac{(1 - \mu)}{c_p Q \rho} \tag{9}$$

Note that specific heat (c_p) for ammonia varies about 5% over the range of temperatures typically used in refrigeration. (Table 3. Source: NIST)

Table 4 shows the results from a variety of pumps. These calculations show that the temperature rise in an ammonia pump is on the order of 0.1°F (0.05°C). High head pumps may have as much as 0.4°F (0.22°C) temperature rise. Such a small temperature rise in the liquid does not significantly affect the thermodynamic calculations.

Comparing CPRs and Pumped Recirculators – Mass Flowrates

To show the mass flow differences between the systems, equations are needed to solve the mass balance for recirculators and CPRs. The methodology for developing these equations is simple—do a mass balance around the vessel, then substitute known relationships into the mass balance and solve the equations for L_{in}, liquid feed to the vessel, and VC, vapor to the compressors.

Calculating Recirculator Mass Flows

The mass and energy balance for a recirculator is pretty straightforward
(See Figure 2).

The vapor generated in the evaporators (VE), and the liquid makeup to the vessel (L_{in}) are independent of the overfeed rate. The only place overfeed rate enters into the calculations is for pumping rate (PL) and amount of liquid returned in the wet suction header (RL). The vapor to the compressor (VC) is simply the sum of the vapor generated in the evaporators (VE) plus any flash loads.

Thermodynamically there are three enthalpies to be concerned with:

h_{fin}	entering liquid enthalpy
h_f	saturated liquid enthalpy at vessels pressure
h_g	saturated vapor enthalpy at vessels pressure

Using equations (1) and (2):

$$RE = RefrigeratingEffect = (h_{fin} - h_g)$$

$$LC = LiquidCooling = (h_{fin} - h_f)$$

The first recirculator equation is a mass balance. The masses entering the vessel are the Liquid in (L_{in}), the Vapor generated by the evaporators (VE), and the returned liquid (RL). The masses leaving the vessel are the pumped liquid (PL), the liquid out to the next vessel, (L_{out}), and the vapor to the compressors (VC).

Recirculator Mass Balance

$$L_{in} + VE + RL = PL + L_{out} + VC \tag{10}$$

The pumped liquid, in terms of mass flow, equals the vapor from the evaporators plus the returned liquid

$$PL = VE + RL \tag{11}$$

Substituting (11) into (10)

$$L_{in} = VC + L_{out} \tag{12}$$

The load to the compressors is the sum of the vapor from the evaporators (VE) plus the flash gas.

$$VC = VE + FlashGas \tag{13}$$

From equation (3) the flash gas is

$$FG = \frac{L_{in} * LC}{RE}$$

Equation (13) can be rewritten

$$VC = VE + \frac{L_{in} * LC}{RE} \tag{14}$$

Substituting equation (14) into equation (12) yields (see appendix 1 for details)

$$L_{in} = (VE + L_{out})\left(\frac{RE}{RE - LC}\right) \tag{15}$$

Using these equations the steady state mass flows for a recirculator can be predicted based on evaporator load.

Calculating CPR Mass Flows

The mass balance for a CPR follows along the same lines as a recirculator, except that there is a lot more going on in the CPR (Figure 5). Assume that the CPR feeds and receives two different temperature levels of liquid, and that it feeds and receives liquid from other vessels.

Thermodynamically there are six enthalpies to be concerned with:

h_{fin}	entering liquid enthalpy
h_{fcpr}	saturated liquid enthalpy at vessel pressure
h_g	saturated vapor enthalpy at vessel pressure
h_{f1}	returning liquid enthalpy from Accumulator 1
h_{f2}	returning liquid enthalpy from Accumulator 2
h_{favg}	liquid enthalpy at CPR "Blend" temperature

The other terms are

n_1	Circulation ratio from accumulator 1
RL_1	Return liquid from accumulator 1
SL_1	Supply liquid to accumulator 1 evaporators
n_2	Circulation ratio from accumulator 2
RL_2	Return liquid from accumulator 2
SL_2	Supply liquid to accumulator 2 evaporators

As before, use Equations (1) and (2) to get RE and LC.

$RE = Refrigerating\ Effect = (h_{fin} - h_g)$

$LC = Liquid\ Cooling = (h_{fin} - h_{fcpr})$

Then do a mass balance around the CPR.

$$L_{in} + RL_1 + RL_2 = VC + L_{out} + SL_1 + SL_2 \qquad (16)$$

Because the circulation ratio is defined

$$SL_1 = \frac{n_1}{(n_1 - 1)} * RL_1 \quad \text{and} \quad SL_2 = \frac{n_2}{(n_2 - 1)} * RL_2 \qquad (17a,b)$$

The only load to the compressors is the flash gas, Equation (3)

$$VC = FG = \frac{L_{in} * LC}{RE} \qquad (18)$$

Substituting for VC, SL_1 and SL_2 and solving for L_{in} (see Appendix 1 for details)

$$L_{in} = \left(L_{out} + \frac{RL_1}{(n_1 - 1)} + \frac{RL_2}{(n_2 - 1)} \right) \left(\frac{\mathrm{RE}}{RE - LC} \right) \qquad (19)$$

The only remaining challenge is to determine the *lossless* mix temperature. The reason for the *lossless* term is the assumption of no thermodynamic losses. To average the mix temperature, use a weighted average of the mass flowrates times the enthalpies divided by the sum of the mass flowrates.

$$h_{favg} = \frac{L_{in} * h_{fin} + RL_1 * h_{f1} + RL_2 * h_{f2}}{L_{in} + RL_1 + RL_2} \qquad (20)$$

These equations (18 and 19) predict the mass flowrates around a CPR and (20) calculates its lossless mix temperature. These can be used to perform a mass and energy balance on a CPR system.

Thermodynamic Losses in a CPR

One of the things that a CPR is trying to do is maintain a cold pool of liquid below a warm area of vapor. It is fighting the laws of thermodynamics, because these two would, if left alone for a period of time, both reach equilibrium at some saturated temperature and pressure.

Some of the factors working in favor of the CPR are that cold liquid is slightly denser than warmer liquid, thus the liquid in the CPR will stratify leaving the coldest liquid at the bottom, away from the warm vapor. The other factor working in favor of the CPR is that the internal design of these vessels attempts to limit contact between the cold liquid and the warm vapor.

However, these factors only partially mitigate the inevitable result that the cold liquid will condense the warm vapor above. There are two primary areas of heat transfer: 1) the surface of the cold liquid interacting with the warm vapor, and 2) the vessel walls which conduct heat and act as a conduit for heat transfer between the two. The driving force for this thermodynamic reaction is the temperature difference between the two. If there is a large temperature difference, there will be a greater amount of heat transfer; in fact the heat transfer will be proportional to the temperature difference.

The thermodynamic losses in a CPR are a good news/bad news kind of thing. The good news is that the vapor that is condensed by the liquid does not need to be compressed by the compressors drawing off the CPR, lowering their required mass flow. The bad news is that this warms the liquid fed to the evaporators, increasing the amount of flash gas generated at the evaporator and raising the mass flow for lower temperature compressors.

Flash Gas Removal

Depending on the system being analyzed, the greatest energy difference between CPR systems and an equivalent pumped recirculated system is the ability of a pumped system to remove flash gas from each temperature level before feeding that saturated liquid to the next lower temperature level (Figures 4 and 5). In CPR systems, most of the flash gas is generated at the evaporators, and the remainder is generated at the CPRs. In a recirculated system, all of the flash gas is generated at the recirculator vessels. This allows for advantageous removal of flash gas at each temperature/pressure level in the recirculated system, which lowers its total required compressor energy.

When examining compression systems for pumped and CPR systems, it is expected that the pumped system would have higher mass flow rates to the higher temperature compressors, and lower mass flow rates to the lower temperature compressors. This is due to the pumped system removing flash gas at the highest possible suction pressure. The energy savings comes from the fact that higher temperature compressors require less energy to do their work of compression than lower temperature compressors.

Comparing the Two Systems

Now that there is a basis for calculating mass flows in the two different systems, the two approaches can be compared on an example system, to demonstrate the differences in compressor energy consumption. Compressor energy requirements are estimated for this example using the compressor performance data shown in tables 5 and 6. Two stage performance was used for the low temperature loads, and high stage performance was used for the high temperature loads. Pumping energy was not considered as part of the analysis for either system.

Recirculated System

Condensing Temp: 90°F		Condensing Temp: 32.22°C	
+30°F Load	300 TR	–1.11°C Load	1055 kW
+20°F Load	1200 TR	–6.67°C Load	4220 kW
–20°F Load	600 TR	–28.9°C Load	2110 kW
–30°F Load	200 TR	34.4°C Load	703.36 kW

The recirculated system will use a flash tank at 95 psig, 61°F (655 kPa, 16.11°C) which feeds the +30°F (–1.11°C) recirculator, and each recirculator in turn feeds the next lower temperature recirculator. The assumption for both systems is that the gas from the flash tank or CPR 1 is used for floor warming or defrosts and does not require mechanical compression.

CPR System

The CPR system uses two CPRs, which feed the following loads:

Condensing Temp: 90°F		Condensing Temp: 32.22°C	
CPR1	95 psig (sat 61°F)	655 kPa (sat 16.11°C)	
+30°F Load	300 TR	–1.11°C Load	1055 kW
+20°F Load	1200 TR	–6.67°C Load	4220 kW
CPR2	45 psig (sat at 30°F)	310 kPa (sat –1.11°C)	
–20°F Load	600 TR	–28.9°C Load	2110 kW
–30°F Load	200 TR	–34.4°C Load	703.36 kW

The source of liquid makeup for the CPR 2 is CPR 1.

The CPR system is analyzed for a *lossless* CPR, where it is assumed there are no thermodynamic losses in the CPR, and for 4°F, 8°F, and 12°F (2.22, 4.44, 6.67°C) temperature rises in the CPR due to thermodynamic losses. As discussed above, the losses show up as condensed gas in the CPR, sufficient to warm the CPR liquid the specified amount.

Results of the Analysis

This system was analyzed at both 2:1 and 3:1 circulation ratios. The summary results of the analysis, shown in Table 7 a and b, show that for every case, the pumped system has lower compressor energy consumption than the CPR system—even if the CPR system is *lossless*. Higher circulation rates than 3:1 would result in higher compressor energy consumption for the CPR system, while the pumped system energy consumption remains constant. Even when the model was run on fractional (less than 2:1) circulation rates, the pumped system always had lower compressor energy consumption. This is due to the fact that the pumped system removes flash gas in a more efficient manner than the CPR system.

Tables 8 and 9 show the detailed analyses for a 2:1 system with a lossless CPR, and a 3:1 system with 8°F losses in the CPR, respectively.

Other Considerations

Since 1989 at least four papers have been presented at the IIAR warning the members of the dangers of hydraulic shock, vapor-propelled liquid, and condensate-induced hydraulic shock. Bulletin 116 was issued in 1992 warning the members of these dangers. The lesson from each of these papers and the bulletin is much the same—that hot, high-pressure gas and cold liquid are a potentially dangerous combination. The consequences of these two getting together in the same place range from banging and shaking of pipes to major releases, injuries, and conflagrations.

Much of the effort of the ammonia refrigeration system designer should be directed towards keeping hot gas and cold liquid apart. Soft hot gas initiation, two step solenoids, long pre-defrost pump outs, and suction valves that will not open against a high differential are a few of the steps that are taken in the design of a system to avoid having cold liquid and hot gas in the same place at the same time.

A gas powered transfer drum is the only place in an ammonia refrigeration system where cold liquid and hot gas are put together on purpose. Consider for a minute the frequency of transfer drum cycles. A transfer drum, operating at the frequency of four dump cycles per hour, cycles 96 times per day. That calculates to 35,000 cycles per year, or 700,000 cycles over a 20 year lifetime. With this many repetitions, even if there is a low frequency of failure, there is a relatively high probability of something going wrong. Even if there were energy improvements with transfer drums, which there are not, the safety risks of their use alone should dissuade any designer from advocating them.

Conclusions

CPR fed systems are not as energy efficient as pumped systems. To even stay close to the pumped systems' efficiency they must operate in a narrow band of low circulation ratios and hope for low thermodynamic losses in the CPR. If the circulation ratio falls too low, there is insufficient subcooling in the CPR, and the liquid will flash in the mains running out to the air units. This can also occur in CPR systems on startups or during periods of high loads. If the circulation ratio gets too high, the vapor generated at the evaporators and CPR losses make the system very inefficient. The first costs of each system can certainly be argued, and it is likely fair to say that for some systems, especially smaller systems, a CPR approach may be less costly. However, the operating costs will always be lower for a pumped system.

Any potential advantage of CPR feed in terms of evaporator performance is more than wiped out by the disadvantage in wet suction line pressure drops. In terms of compressor energy, regardless of the circulation ratio, whether 5:1 or 1.2:1, a pumped recirculator system consumes less energy because of the advantageous removal of flash gas. The advantage of a pumped system is really quite simple—pumping cold liquid to an evaporator is the most energy efficient way to supply overfed evaporators. It results in lower mass flows and pressure drops through the evaporators and piping system, and lower overall compression energy.

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Appendix 1

Pumped Recirculator Algebraic Rearrangement

$$L_{in} = VC + L_{out} \tag{12}$$

$$VC = VE + \frac{L_{in} * LC}{RE} \tag{14}$$

Substituting (14) into (12)

$$L_{in} = VE + \frac{L_{in} * LC}{RE} + L_{out}$$

$$L_{in} - \frac{L_{in} * LC}{RE} = VE + L_{out}$$

$$L_{in} \left(1 - \frac{LC}{RE} \right) = VE + L_{out}$$

$$L_{in} \left(\frac{RE}{RE} - \frac{LC}{RE} \right) = VE + L_{out}$$

$$L_{in} \left(\frac{RE - LC}{RE} \right) = VE + L_{out}$$

$$L_{in} = (VE + L_{out}) \left(\frac{RE}{RE - LC} \right) \tag{15}$$

CPR Algebraic Rearrangement

$$L_{in} + RL_1 + RL_2 = VC + L_{out} + SL_1 + SL_2 \tag{16}$$

$$SL_1 = \frac{n_1}{(n_1 - 1)} * RL_1 \quad \text{and} \quad SL_2 = \frac{n_2}{(n_2 - 1)} * RL_2 \tag{17a,b}$$

$$VC = FG = \frac{L_{in} * LC}{RE} \tag{18}$$

Substituting (17) and (18) into (16)

$$L_{in} + RL_1 + RL_2 = \frac{L_{in} * LC}{RE} + L_{out} + \frac{n_1}{(n_1 - 1)} * RL_1 + \frac{n_2}{(n_2 - 1)} * RL_2$$

Rearranging

$$L_{in} - \frac{L_{in} * LC}{RE} = L_{out} + \frac{n_1}{(n_1 - 1)} * RL_1 - RL_1 + \frac{n_2}{(n_2 - 1)} * RL_2 - RL_2$$

$$L_{in} * \left(\frac{RE}{RE} - \frac{LC}{RE} \right) = L_{out} + RL_1 \left(\frac{n_1}{(n_1 - 1)} - 1 \right) + RL_2 \left(\frac{n_2}{(n_2 - 1)} - 1 \right)$$

$$L_{in} * \left(\frac{RE - LC}{RE} \right) = L_{out} + RL_1 \left(\frac{n_1}{(n_1 - 1)} - \frac{(n_1 - 1)}{(n_1 - 1)} \right) + RL_2 \left(\frac{n_2}{(n_2 - 1)} - \frac{(n_2 - 1)}{(n_2 - 1)} \right)$$

Simplifying

$$L_{in} * \left(\frac{RE - LC}{RE} \right) = L_{out} + \frac{RL_1}{(n_1 - 1)} + \frac{RL_2}{(n_2 - 1)}$$

$$L_{in} = \left(L_{out} + \frac{RL_1}{(n_1 - 1)} + \frac{RL_2}{(n_2 - 1)} \right) \left(\frac{RE}{RE - LC} \right) \qquad (19)$$

Table 1. Terms Used in the Paper

	IP	S.I.
n is the circulation ratio		
W is the evaporator load (TR*200 = BTU/min) (kW*60 = kJ/min)	BTU/min	kJ/min
L is the liquid mass flow to the evaporator	#/min	kg/min
RL is the returned, or overfed liquid	#/min	kg/min
PL is the Pumped Liquid (Pumped)	#/min	kg/min
SL is the Supplied Liquid (CPR)	#/min	kg/min
VE is the vapor generated by the evaporator	#/min	kg/min
VC is the vapor to the compressor	#/min	kg/min
FG is the Flash Gas	#/min	kg/min
RE is the Refrigerating Effect	BTU/#	kJ/kg
LC is the Liquid Cooling	BTU/#	kJ/kg
h_f is the enthalpy of a saturated liquid	BTU/#	kJ/kg
h_g is the enthalpy of a saturated vapor	BTU/#	kJ/kg
dt is the temperature rise in the pump	°F	°C
Q is the volumetric flow through the pump	ft^3/min	m^3/s
μ is the pump efficiency	%	%
ρ = fluid density	lb/ ft^3	kg/m^3
c_p = specific heat	BTU/lb °F	kJ/kg – °K
P_s = Brake Power	BTU/min	kW

Table 2a. Two Phase Piping Pressure Drop – IP Units

Examples								
CPR Liquid Temperature	0	20	46	46	12	12	12	°F
Suction Temperature	–40	0	20	20	–32	–22	–22	°F
Recirculation Rate	3	2	2	3	2	2	3	:1
Evaporator Tons	120	75	1353	1353	162	555	555	TR
Enthalpy								
CPR Liquid, h_f	42.78	64.58	93.35	93.35	55.83	55.83	55.83	BTU/#
Saturated Liquid, h_f	0.00	42.78	64.58	64.58	8.47	19.12	19.12	BTU/#
Vapor Out, h_g	597.39	611.56	617.59	617.59	600.42	604.08	604.08	BTU/#
CPR System								
(1) RE, Refrigerating Effect	554.61	546.98	524.24	524.24	544.59	548.25	548.25	BTU/#
(2) LC, Liquid Cooling	42.78	21.80	28.78	28.78	47.36	36.71	36.71	BTU/#
(8) VE, Vapor Generated	51.17	28.56	546.16	579.83	65.16	216.99	233.77	#/min
(5) L, Liquid Flow	153.50	57.12	1092.32	1739.50	130.32	433.98	701.30	#/min
Equivalent TR	152.83	81.23	1510.16	1603.27	192.86	634.66	683.73	TR
Pumped System								
(1) RE, Refrigerating Effect	597.39	568.78	553.01	553.01	591.95	584.97	584.97	BTU/#
(4) VE, Vapor Generated	40.17	26.37	489.32	489.32	54.73	189.75	189.75	#/min
(5) L, Liquid Flow	120.52	52.74	978.64	1467.96	109.47	379.51	569.26	#/min
Equivalent TR	120.00	75.00	1353.00	1353.00	162.00	555.00	555.00	TR
Comparison								
Increase in Mass Flow for CPR	27.36%	8.30%	11.62%	18.50%	19.05%	14.35%	23.19%	
Wet Suction Pipe Size	8"	4"	10"	10"	6"	10"	10"	
CPR System Pressure Drop	0.34	0.41	0.53	0.89	1.06	0.5	0.86	°F/100'
Pumped System Pressure Drop	0.21	0.35	0.42	0.64	0.76	0.38	0.57	°F/100'
Increase in Pressure Drop for CPR	61.90%	17.14%	26.19%	39.06%	39.47%	31.58%	50.88%	
Note: Numbers in parenthesis refer to equation numbers.								

Table 2b. Two Phase Piping Pressure Drop - SI Units

Examples								
CPR Liquid Temperature	–17.8	-6.7	7.8	7.8	–11.1	–11.1	–11.1	°C
Suction Temperature	–40.0	–17.8	–6.7	–6.7	–35.6	–30.0	–30.0	°C
Recirculation Rate	3	2	2	3	2	2	3	:1
Evaporator Load	422.0	263.8	4,758.2	4,758.2	569.7	1,951.8	1,951.8	KW
Enthalpy								
CPR Liquid, h_f	99.44	150.11	217.00	217.00	129.76	129.76	129.76	kJ/kg
Saturated Liquid, h_f	0.00	99.44	150.11	150.11	19.68	44.43	44.43	kJ/kg
Vapor Out, h_g	1388.60	1421.50	1435.60	1435.60	1395.60	1404.10	1404.10	kJ/kg
CPR System								
(1) RE, Refrigerating Effect	1289.16	1271.39	1218.60	1218.60	1265.84	1274.34	1274.34	kJ/kg
(2) LC, Liquid Cooling	99.44	50.67	66.89	66.89	110.08	85.33	85.33	kJ/kg
(8) VE, Vapor Generated	23.22	12.96	247.89	263.17	29.58	98.49	106.11	kg/min
(5) L, Liquid Flow	69.67	25.93	495.77	789.51	59.15	196.99	318.32	kg/min
Equivalent kW	537.48	285.66	5310.94	5638.41	678.24	2231.96	2404.52	kW
Pumped System								
(1) RE, Refrigerating Effect	1388.60	1322.06	1285.49	1285.49	1375.92	1359.67	1359.67	kJ/kg
(4) VE, Vapor Generated	18.23	11.97	222.09	222.09	24.84	86.13	86.13	kg/min
(5) L, Liquid Flow	54.70	23.94	444.18	666.27	49.69	172.26	258.39	kg/min
Equivalent kW	422.02	263.76	4758.23	4758.23	569.72	1951.82	1951.82	kW
Comparison								
Increase in Mass Flow for CPR	27.36%	8.30%	11.62%	18.50%	19.05%	14.35%	23.19%	
Wet Suction Pipe Size	8"	4"	10"	10"	6"	10"	10"	
CPR System Pressure Drop	0.189	0.228	0.294	0.494	0.589	0.278	0.478	°C/100'
Pumped System Pressure Drop	0.117	0.194	0.233	0.356	0.422	0.211	0.317	°C/100'
Increase in Pressure Drop for CPR	61.90%	17.14%	26.19%	39.06%	39.47%	31.58%	50.88%	
Note: Numbers in parenthesis refer to equation numbers.								

Table 3. Specific Heat of Ammonia Liquid

Temp (°F)	Temp (°C)	cp (BTU/lb °F)	cp (kJ/kg °K)
–40	–40.00	1.0549	4.413
–20	–28.89	1.0684	4.470
0	–17.78	1.0814	4.525
20	–6.67	1.0948	4.581
40	4.44	1.1094	4.642
Source: NIST			

Table 4. Temperature Rise in Liquid Pumps

The temperature rise in a centrifugal pump can be calculated as

$$dt = P_s \frac{(1 - \mu)}{c_p Q \rho}$$

Where

dt = Temperature rise in the pump, °F, °C

P_s = Power input to pump, BTU/min (kW)

Q = volumetric flow through the pump, ft^3/min (m^3/s)

μ = pump efficiency, %

ρ = fluid density, lb/ft^3 (kg/m^3)

c_p = specific heat, BTU/lb °F (kJ/kg °K)

Input Variables, IP Units					
Fluid Temperature	–20	20	0	–40	°F
Pump efficiency	61.00%	40.00%	55.00%	32.00%	
Flow through the pump	250	64	80	15	GPM
Total Head	34	61	36	16	psid
Power input to pump	8	6	3	0.4	BHP
Calculated Values					
Fluid Density	42.229	40.429	41.344	43.085	lb/ft^3
C_p	1.068	1.095	1.081	1.055	BTU/#°F
Flow through the pump	33.422	8.556	10.695	2.005	ft^3/min
Power input to pump	339.270	254.453	127.226	16.964	BTU/min
Result					
Temp Rise	0.088	0.403	0.120	0.127	°F
Temp Rise	0.049	0.224	0.067	0.070	°C

Table 5a. Compressor Performance, IP Units

Single Stage +90°F Condensing

Suction Temp, °F	Capacity #/min	BHP	HS Perf BHP/#/min
–40	26.3	199.1	7.570
–35	30.4	205.7	6.766
–30	35	213.7	6.106
–25	40.2	222.9	5.545
–20	45.9	232.1	5.057
–15	52.2	244.8	4.690
–10	59.2	251.5	4.248
–5	66.9	261.2	3.904
0	75.4	270.2	3.584
5	84.7	278.7	3.290
10	94.8	286	3.017
15	105.9	292	2.757
20	117.9	296.4	2.514
25	130.9	299.2	2.286
30	145.1	300.4	2.070

Table 5b. Compressor Performance, SI Units

Single Stage +32.2°C Condensing

Suction Temp,°C	Capacity kg/min	kW	HS Perf kW/kg/min
–40.00	11.93	148.45	12.444
–37.22	13.79	153.37	11.122
–34.44	15.88	159.33	10.036
–31.67	18.23	166.19	9.114
–28.89	20.82	173.05	8.312
–26.11	23.68	182.52	7.709
–23.33	26.85	187.52	6.983
–20.56	30.35	194.75	6.418
–17.78	34.20	201.46	5.891
–15.00	38.42	207.80	5.409
–12.22	43.00	213.24	4.959
–9.44	48.04	217.72	4.532
–6.67	53.48	221.00	4.132
–3.89	59.38	223.08	3.757
–1.11	65.82	223.98	3.403

Table 6a. Two Stage Performance, IP Units

Booster with +30°F / +90°F High Stage

Suction Temp, °F	Capacity #/min	BHP	Booster BHP/#/min	HS Load #/min	HS Load Mult	HS Perf BHP/#/min	HS BHP	Total BHP/#/min
–40	27.6	104.5	3.786	36.1	1.308	2.070	2.708	6.494
–35	31.9	107.9	3.382	41.7	1.307	2.070	2.706	6.088
–30	36.7	110.5	3.011	47.5	1.294	2.070	2.679	5.690
–25	42.1	112.9	2.682	54.2	1.287	2.070	2.665	5.347
–20	48.0	114.4	2.383	61.6	1.283	2.070	2.657	5.040

Table 6b. Two Stage Performance, SI Units

Booster with –1.11°C /32.2°C High Stage

Suction Temp, °C	Capacity kg/min	kW	Booster kW/kg/min	HS Load kg/min	HS Load Mult	HS Perf kW/kg/min	HS kW	Total kW/kg/min
–40.00	12.52	77.92	6.224	16.37	1.308	3.403	4.451	10.675
–37.22	14.47	80.45	5.560	18.91	1.307	3.403	4.448	10.008
–34.44	16.65	82.39	4.949	21.55	1.294	3.403	4.404	9.354
–31.67	19.10	84.18	4.408	24.58	1.287	3.403	4.381	8.789
–28.89	21.77	85.30	3.918	27.94	1.283	3.403	4.367	8.285

Table 7a. Recirculator vs CPR

Compressor Mass Flow and Energy Consumption, IP Units						
Temperature	61	30	20	–20	–30	°F
Load	0	200	1200	600	200	TR
2:1 Recirculation Ratio						
Mass Flow, #/min						
Recirculated	64.92	133.23	449.17	229.59	69.00	
CPR-Lossless	65.58	93.12	484.40	225.64	77.29	
CPR- 4°F Loss	58.41	92.19	493.31	229.40	78.61	
CPR- 8°F Loss	50.95	91.24	502.59	233.29	79.98	
CPR- 12°F Loss	43.20	90.25	512.24	237.32	81.41	
Energy Consumption, BHP						Total
Recirculated		275.8	1129.2	1157.1	392.6	**2954.8**
CPR-Lossless		192.8	1217.8	1137.3	439.8	**2987.6**
CPR- 4°F Loss		190.9	1240.2	1156.2	447.3	**3034.5**
CPR- 8°F Loss		188.9	1263.5	1175.8	455.1	**3083.3**
CPR- 12°F Loss		186.8	1287.8	1196.1	463.2	**3133.9**
3:1 Recirculation Ratio						
Mass Flow, #/min						
Recirculated	64.92	133.23	449.17	229.59	69.00	
CPR-Lossless	66.32	92.32	489.71	224.22	78.40	
CPR- 4°F Loss	59.69	92.18	503.43	229.79	80.44	
CPR- 8°F Loss	52.67	92.04	517.98	235.66	82.60	
CPR- 12°F Loss	45.22	91.90	533.45	241.86	84.89	
Energy Consumption, BHP						Total
Recirculated		275.8	1129.2	1157.1	392.6	**2954.8**
CPR-Lossless		191.1	1231.1	1130.1	446.1	**2998.4**
CPR- 4°F Loss		190.8	1265.6	1158.2	457.7	**3072.3**
CPR- 8°F Loss		190.5	1302.2	1187.7	470.0	**3150.5**
CPR- 12°F Loss		190.3	1341.1	1219.0	483.0	**3233.3**

Table 7b. Recirculator vs CPR

Compressor Mass Flow and Energy Consumption, SI Units						
Temperature	16.1	–1.1	–6.7	–28.9	–34.4	°C
Load	0.0	703.4	4220.2	2110.1	703.4	kW
2:1 Recirculation Ratio						
Mass Flow, kg/min						
Recirculated	29.45	60.43	203.74	104.14	31.30	
CPR-Lossless	29.74	42.24	219.72	102.35	35.06	
CPR- 4°F Loss	26.49	41.82	223.76	104.05	35.66	
CPR- 8°F Loss	23.11	41.38	227.97	105.82	36.28	
CPR- 12°F Loss	19.60	40.94	232.35	107.65	36.93	
Energy Consumption, kW						Total
Recirculated		205.7	842.1	862.9	292.8	**2203.4**
CPR-Lossless		143.8	908.1	848.0	328.0	**2227.8**
CPR- 4°F Loss		142.3	924.8	862.1	333.6	**2262.8**
CPR- 8°F Loss		140.9	942.2	876.8	339.4	**2299.2**
CPR- 12°F Loss		139.3	960.3	891.9	345.4	**2337.0**
3:1 Recirculation Ratio						
Mass Flow, kg/min						
Recirculated	29.45	60.43	203.74	104.14	31.30	
CPR-Lossless	30.08	41.88	222.13	101.71	35.56	
CPR- 4°F Loss	27.07	41.81	228.35	104.23	36.49	
CPR- 8°F Loss	23.89	41.75	234.95	106.90	37.47	
CPR- 12°F Loss	20.51	41.68	241.97	109.70	38.50	
Energy Consumption, kW						Total
Recirculated		205.7	842.1	862.9	292.8	**2203.4**
CPR-Lossless		142.5	918.1	842.7	332.6	**2235.9**
CPR- 4°F Loss		142.3	943.8	863.6	341.3	**2291.0**
CPR- 8°F Loss		142.1	971.1	885.7	350.5	**2349.3**
CPR- 12°F Loss		141.9	1000.1	909.0	360.2	**2411.1**

Table 8a. 2:1 Circulation, Lossless CPR, IP Units

CPR #1 Evaporators			
Suction temp	30	20	°F
Recirc Rate	2	2	:1
Evap Load, TR	200	1200	TR
Liquid Temp	46	46	°F
Enthalpy			
CPR Liquid Enthalpy	93.35	93.35	BTU/#
Saturated Liquid Enthalpy	75.59	64.58	BTU/#
Vapor Enthalpy	620.31	617.59	BTU/#
Calculations			
(1) RE, Refrig Effect	526.96	524.24	BTU/#
(2) LC, Liquid Cooling	17.77	28.78	BTU/#
(8) VE, Vapor from Evaporator	78.56	484.40	#/min
(7) RL, Return Liquid	78.56	484.40	#/min
CPR #1 Calculations			
Saturated Temperature	61	°F	
Liquid Feed Temperature (in)	90	°F	
Liquid Out to CPR #2	317.15	#/min	
Pressure	94.92	psig	
Enthalpy			
h_f, Entering Liquid	143.42	#/min	
Liquid Enthalpy (sat)	110.21	#/min	
Vapor Enthalpy (sat)	627.29	#/min	
Calculations			
(1) RE, Refrigerating Effect	483.87	#/min	
(2) LC, Liquid Cooling	33.21	#/min	
(19) L_{in}, Liquid In	944.96	#/min	
(18) VC, Vapor to Compressor	64.86	#/min	
(20) Lossless Blend Enthalpy	93.75	BTU/#	
Lossless Blend Temp	46	°F	

Table 8a. 2:1 Circulation, Lossless CPR, IP Units (continued)

CPR #1 Liquid Warming Calculations			
Enthalpy			
(20) Lossless Blend Enthalpy	93.75	BTU/#	
Evap Blend Enthalpy	93.35	BTU/#	
Calculations			
Difference	–0.39	BTU/#	
Liquid In	944.96	#/min	
Heat Transferred	–372.14	BTU/min	
Latent Heat at Saturation	517.08	BTU/#	
Mass Flow	–0.72	#/min	
VC, revised	65.58	#/min	

Table 8b. 2:1 Circulation, Lossless CPR, IP Units

CPR #2 Evaporators			
Suction temp	–20	–30	°F
Recirc Rate	2	2	:1
Evap Load, TR	600	200	TR
Liquid Temp	4	4	°F
Enthalpy			
CPR Liquid Enthalpy	47.12	47.12	BTU/#
Saturated Liquid Enthalpy	21.25	10.59	BTU/#
Vapor Enthalpy	604.79	601.16	BTU/#
Calculations			
(1) RE, Refrig Effect	557.67	554.04	BTU/#
(2) LC, Liquid Cooling	25.86	36.53	BTU/#
(8) VE, Vapor from Evaporator	225.64	77.29	#/min
(7) RL, Return Liquid	225.64	77.29	#/min
CPR #2 Calculations			
Saturated Temperature	30	°F	
Liquid Feed Temperature (in)	51	°F	

Liquid Out	0.00	#/min	
Pressure	45.03	psig	
Enthalpy			
h_f, Entering Liquid	98.95	#/min	
Liquid Enthalpy (sat)	75.59	#/min	
Vapor Enthalpy (sat)	620.31	#/min	
Calculations			
(1) RE, Refrigerating Effect	521.36	#/min	
(2) LC, Liquid Cooling	23.36	#/min	
(19) L_{in}, Liquid In	317.15	#/min	
(18) VC, Vapor to Compressor	14.21	#/min	
(20) Lossless Blend Enthalpy	47.71	BTU/#	
Lossless Blend Temp	4	°F	
CPR #2 Liquid Warming Calculations			
Enthalpy			
(20) Lossless Blend Enthalpy	47.71	BTU/#	
Evap Blend Enthalpy	47.12	BTU/#	
Calculations			
Difference	–0.60	BTU/#	
Liquid In	317.15	#/min	
Heat Transferred	–188.95	BTU/min	
Latent Heat at Saturation	544.73	BTU/#	
Mass Flow	–0.35	#/min	
VC, revised	14.56	#/min	

Table 8c. 2:1 Circulation, Lossless CPR, IP Units

Pumped Evaporators						
Suction temp	61	30	20	–20	–30	°F
Recirc Rate	0	2	2	2	2	:1
Evap Load, TR	0	200	1200	600	200	TR
Enthalpy						
Liquid Enthalpy	110.21	75.59	64.58	21.25	10.59	BTU/#
Vapor Enthalpy	627.29	620.31	617.59	604.79	601.16	BTU/#
Calculations						
(1) RE, Refrigerating Effect	517.08	544.73	553.01	583.54	590.57	BTU/#
(4) VE, Vapor from Evaporators	0.00	73.43	433.99	205.64	67.73	#/min
Pumped Recirculator Calculations						
Inlet Liquid Temperature	90.00	61.00	30.00	20.00	-20.00	°F
Enthalpy						
h_{fin}, Entering Liquid	143.42	110.21	75.59	64.58	21.25	BTU/#
h_f, Saturated Liquid	110.21	75.59	64.58	21.25	10.59	BTU/#
h_g, Vapor Enthalpy	627.29	620.31	617.59	604.79	601.16	BTU/#
Calculations						
(1) RE, Refrigerating Effect	483.87	510.10	542.01	540.21	579.91	BTU/#
(2) LC, Liquid Cooling	33.21	34.63	11.01	43.33	10.66	BTU/#
L_{out}, Liquid Out	880.99	747.76	298.59	69.00	0.00	#/min
(15) L_{in}, Liquid In	945.92	880.99	747.76	298.59	69.00	#/min
(3) FG, Flash Gas	64.92	59.80	15.18	23.95	1.27	#/min
(14) VC, Vapor to Compressors	64.92	133.23	449.17	229.59	69.00	#/min

Table 8d. 2:1 Circulation, Lossless CPR, IP Units

Energy Consumption		(CPR1)	(CPR2)			
Temperature	61	30	20	–20	–30	°F
Recirculation Ratio		2	2	2	2	:1
Load	0	200	1200	600	200	TR
Recirculator						
VE, Vapor from Evaporators	0.00	73.43	433.99	205.64	67.73	#/min
Flash Gas	64.92	59.80	15.18	23.95	1.27	#/min
Vapor to Compressors	64.92	133.23	449.17	229.59	69.00	#/min
CPR						
Lossless Blend Temperature	46	4				deg F
Actual Blend Temperature	46	4				deg F
VE, Vapor from Evaporators		78.56	484.40	225.64	77.29	#/min
CPR Flash Gas	65.58	14.56				#/min
VC Vapor to Compressors	65.58	93.12	484.40	225.64	77.29	#/min
Comparison						
Increase in Mass flow for CPR	1.0%	–30.1%	7.8%	–1.7%	12.0%	
Compressor Performance		2.07	2.51	5.04	5.69	BHP/#/min
Recirculator BHP		275.83	1129.22	1157.10	392.61	2954.75
CPR BHP		192.78	1217.77	1137.21	439.80	2987.56
Difference		83.05	–88.56	19.88	–47.18	
Net Increase in BHP for CPR		32.80				
Percentage Increase		1.11%				

Table 9a. 3:1 Circulation, 8°F CPR loss, IP Units

CPR #1 Evaporators			
Suction temp	30	20	°F
Recirc Rate	3	3	:1
Evap Load, TR	200	1200	TR
Liquid Temp	47	47	°F
Enthalpy			
CPR Liquid Enthalpy	94.47	94.47	BTU/#
Saturated Liquid Enthalpy	75.59	64.58	BTU/#
Vapor Enthalpy	620.31	617.59	BTU/#
Calculations			
(1) RE, Refrig Effect	525.84	523.12	BTU/#
(2) LC, Liquid Cooling	18.89	29.89	BTU/#
(8) VE, Vapor from Evaporator	81.96	517.98	#/min
(7) RL, Return Liquid	163.91	1035.97	#/min
CPR #1 Calculations			
Saturated Temperature	61	°F	
Liquid Feed Temperature (in)	90	°F	
Liquid Out to CPR #2	333.20	#/min	
Pressure	94.92	psig	
Enthalpy			
h_f, Entering Liquid	143.42	#/min	
Liquid Enthalpy (sat)	110.21	#/min	
Vapor Enthalpy (sat)	627.29	#/min	
Calculations			
(1) RE, Refrigerating Effect	483.87	#/min	
(2) LC, Liquid Cooling	33.21	#/min	
(19) L_{in}, Liquid In	1001.90	#/min	
(18) VC, Vapor to Compressor	68.76	#/min	
(20) Lossless Blend Enthalpy	86.16	BTU/#	
Lossless Blend Temp	39	°F	
CPR #1 Liquid Warming Calculations			
Enthalpy			
(20) Lossless Blend Enthalpy	86.16	BTU/#	
Evap Blend Enthalpy	94.47	BTU/#	

Calculations			
Difference	8.31	BTU/#	
Liquid In	1001.90	#/min	
Heat Transferred	8324.46	BTU/min	
Latent Heat at Saturation	517.08	BTU/#	
Mass Flow	16.10	#/min	
VC, revised	52.67	#/min	

Table 9b. 3:1 Circulation, 8°F CPR loss, IP Units

CPR #2 Evaporators			
Suction temp	–20	–30	°F
Recirc Rate	3	3	:1
Evap Load, TR	600	200	TR
Liquid Temp	3	3	°F
Enthalpy			
CPR Liquid Enthalpy	46.03	46.03	BTU/#
Saturated Liquid Enthalpy	21.25	10.59	BTU/#
Vapor Enthalpy	604.79	601.16	BTU/#
Calculations			
(1) RE, Refrig Effect	558.76	555.13	BTU/#
(2) LC, Liquid Cooling	24.78	35.44	BTU/#
(8) VE, Vapor from Evaporator	235.66	82.60	#/min
(7) RL, Return Liquid	471.33	165.20	#/min
CPR #2 Calculations			
Saturated Temperature	30	°F	
Liquid Feed Temperature (in)	51	°F	
Liquid Out	0.00	#/min	
Pressure	45.03	psig	
Enthalpy			
h_f, Entering Liquid	98.95	#/min	
Liquid Enthalpy (sat)	75.59	#/min	
Vapor Enthalpy (sat)	620.31	#/min	
Calculations			
(1) RE, Refrigerating Effect	521.36	#/min	

Table 9b. 3:1 Circulation, 8°F CPR loss, IP Units (continued)

(2) LC, Liquid Cooling	23.36	#/min	
(19) L_{in}, Liquid In	333.20	#/min	
(18) VC, Vapor to Compressor	14.93	#/min	
(20) Lossless Blend Enthalpy	38.11	BTU/#	
Lossless Blend Temp	–5	°F	
CPR #2 Liquid Warming Calculations			
Enthalpy			
(20) Lossless Blend Enthalpy	38.11	BTU/#	
Evap Blend Enthalpy	46.03	BTU/#	
Calculations			
Difference	7.93	BTU/#	
Liquid In	333.20	#/min	
Heat Transferred	2641.20	BTU/min	
Latent Heat at Saturation	544.73	BTU/#	
Mass Flow	4.85	#/min	
VC, revised	10.08	#/min	

Table 9c. 3:1 Circulation, 8°F CPR loss, IP Units

Pumped Evaporators						
Suction temp	61	30	20	–20	–30	°F
Recirc Rate	0	3	3	3	3	:1
Evap Load, TR	0	200	1200	600	200	TR
Enthalpy						
Liquid Enthalpy	110.21	75.59	64.58	21.25	10.59	BTU/#
Vapor Enthalpy	627.29	620.31	617.59	604.79	601.16	BTU/#
Calculations						
(1) RE, Refrigerating Effect	517.08	544.73	553.01	583.54	590.57	BTU/#
(4) VE, Vapor from Evaporators	0.00	73.43	433.99	205.64	67.73	#/min
Pumped Recirculator Calculations						
Inlet Liquid Temperature	90.00	61.00	30.00	20.00	-20.00	°F
Enthalpy						
h_{fin}, Entering Liquid	143.42	110.21	75.59	64.58	21.25	BTU/#

h_f, Saturated Liquid	110.21	75.59	64.58	21.25	10.59	BTU/#
h_g, Vapor Enthalpy	627.29	620.31	617.59	604.79	601.16	BTU/#
Calculations						
(1) RE, Refrigerating Effect	483.87	510.10	542.01	540.21	579.91	BTU/#
(2) LC, Liquid Cooling	33.21	34.63	11.01	43.33	10.66	BTU/#
L_{out}, Liquid Out	880.99	747.76	298.59	69.00	0.00	#/min
(15) L_{in}, Liquid In	945.92	880.99	747.76	298.59	69.00	#/min
(3) FG, Flash Gas	64.92	59.80	15.18	23.95	1.27	#/min
(14) VC, Vapor to Compressors	64.92	133.23	449.17	229.59	69.00	#/min

Table 9d. 3:1 Circulation, 8°F CPR loss, IP Units

Energy Consumption		(CPR1)	(CPR2)			
Temperature	61	30	20	–20	–30	°F
Recirculation Ratio		3	3	3	3	:1
Load	0	200	1200	600	200	TR
Recirculator						
VE, Vapor from Evaporators	0.00	73.43	433.99	205.64	67.73	#/min
Flash Gas	64.92	59.80	15.18	23.95	1.27	#/min
Vapor to Compressors	64.92	133.23	449.17	229.59	69.00	#/min
CPR						
Lossless Blend Temperature	39	–5				deg F
Actual Blend Temperature	47	3				deg F
VE, Vapor from Evaporators		81.96	517.98	235.66	82.60	#/min
CPR Flash Gas	52.67	10.08				#/min
VC Vapor to Compressors	52.67	92.04	517.98	235.66	82.60	#/min
Comparison						
Increase in Mass flow for CPR	–18.9%	–30.9%	15.3%	2.6%	19.7%	
Compressor Performance		2.07	2.51	5.04	5.69	BHP/#/min
Recirculator BHP		275.83	1129.22	1157.10	392.61	2954.75
CPR BHP		190.55	1302.21	1187.71	470.01	3150.47
Difference		85.28	–172.99	–30.61	–77.40	
Net Increase in BHP for CPR		195.72				
Percentage Increase		6.62%				

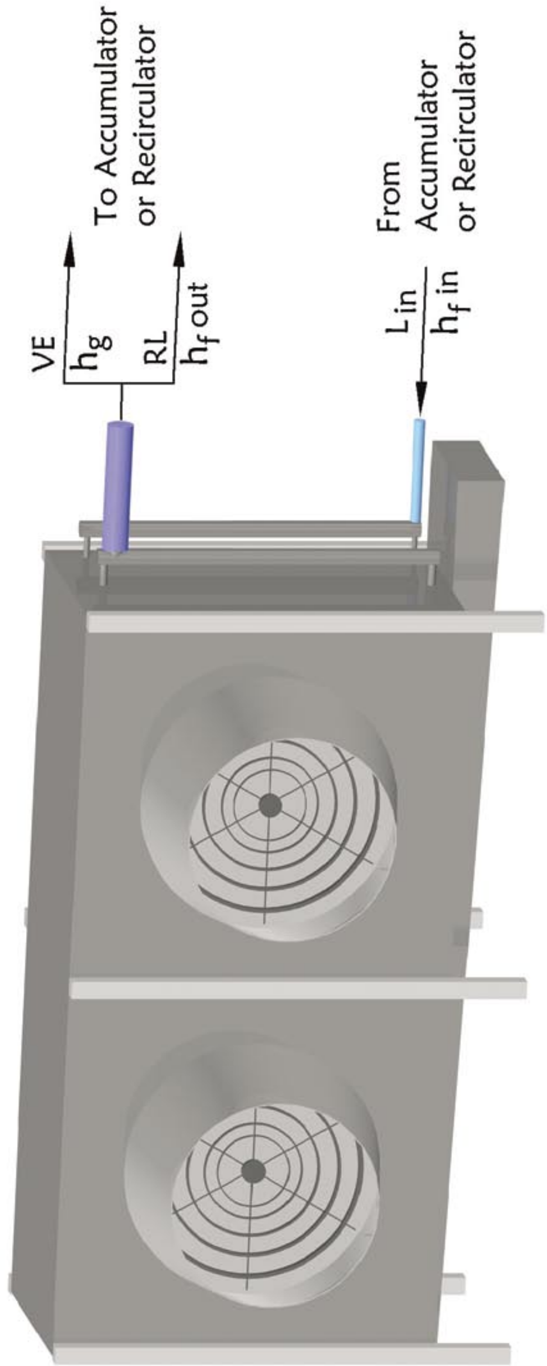


Figure 1: Evaporator Mass Flow Diagram

Figure 2: Recirculator

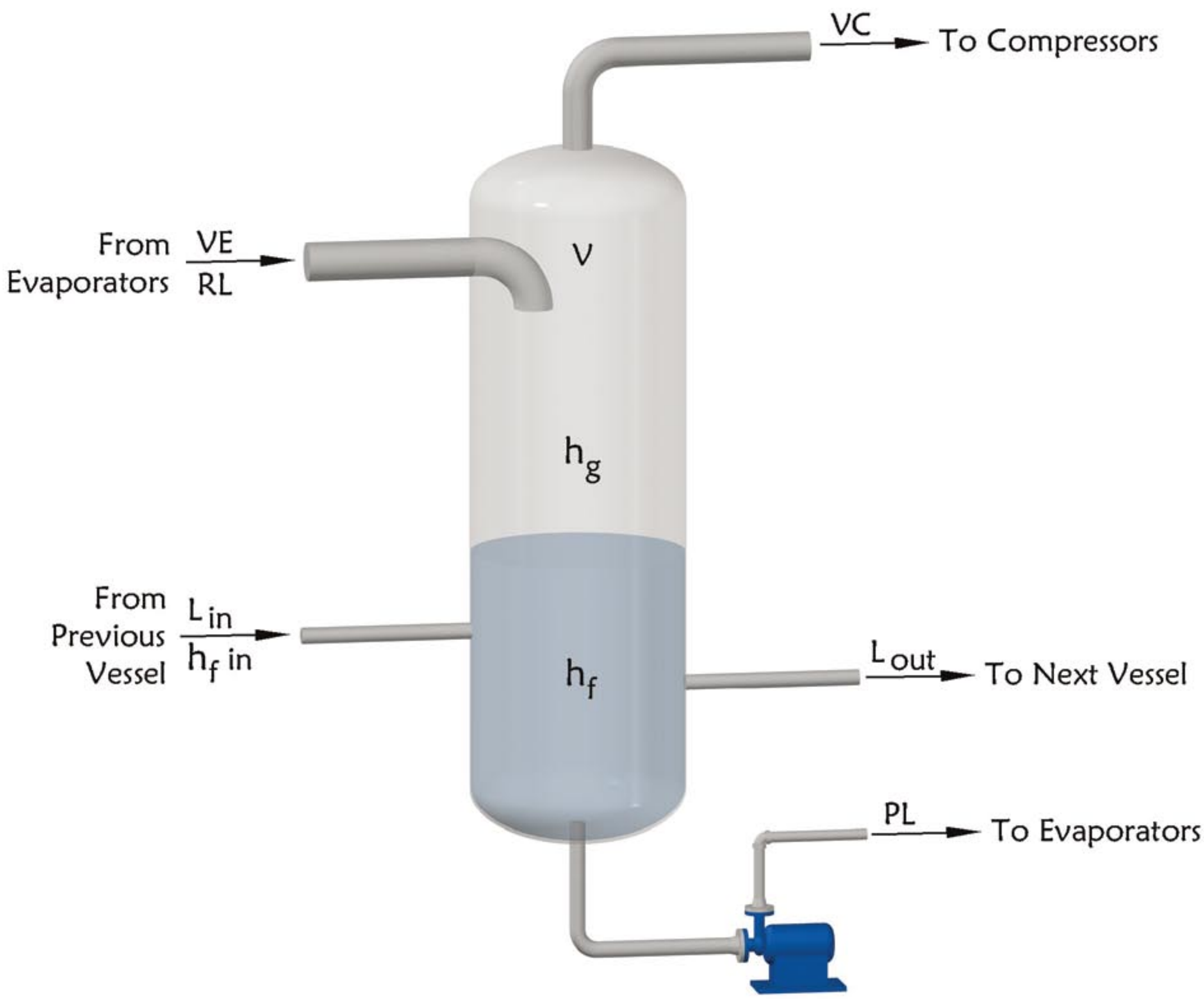
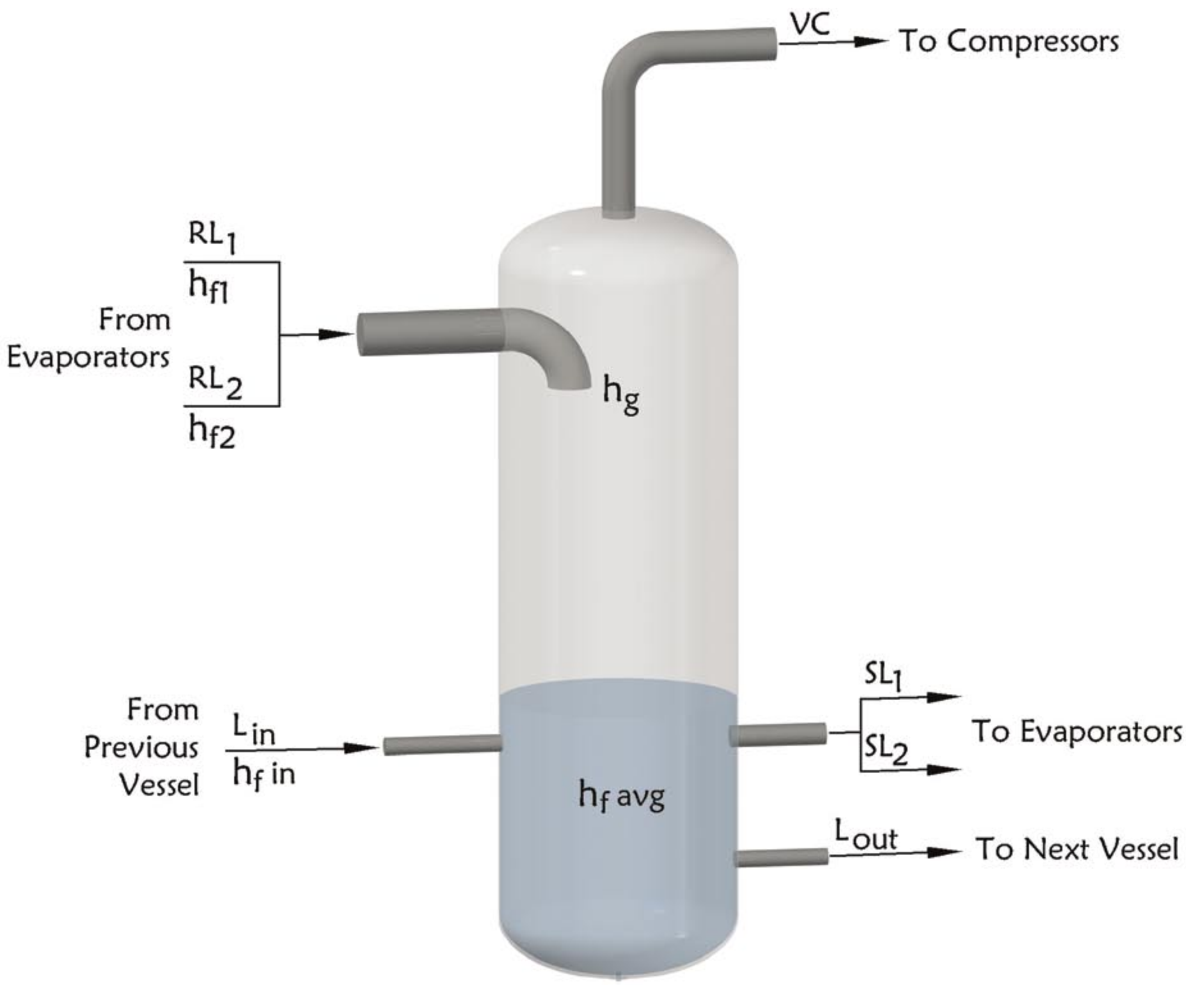


Figure 3: Control Pressure Receiver



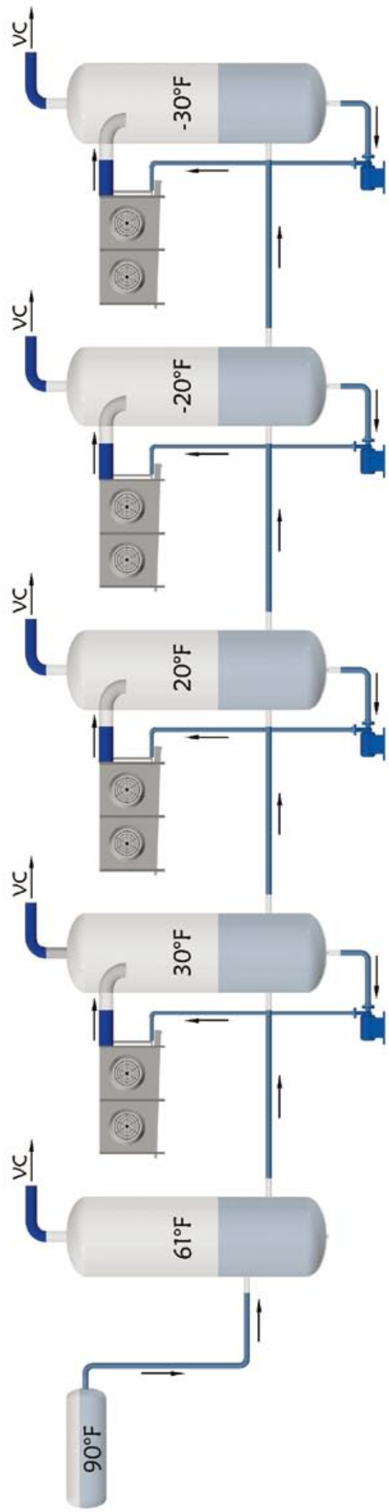


Figure 4: Pumped System Flow diagram

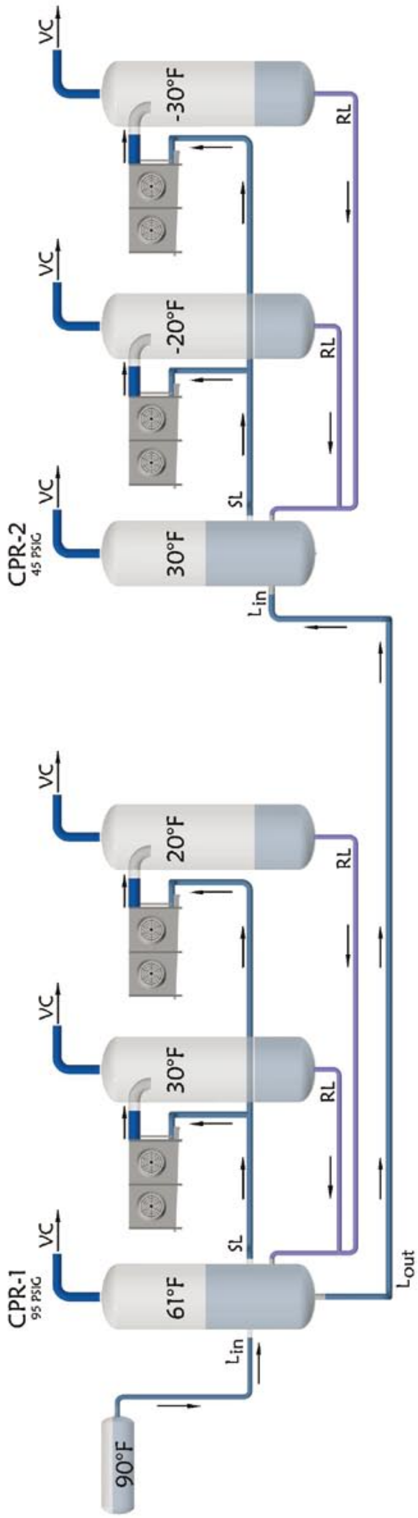


Figure 5: Control Pressure Receiver System Flow diagram

Notes:

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