Figure 1 : Horizontal Plate Freezer

Energy-Saving Ammonia Refrigeration Applications using Pump Circulation

By Ramesh Paranjpey
Technical Advisor & Consultant
Pune

Liquid overfeed systems are normally used for low temperature applications, where there are multiple evaporators operating at the same or at different temperatures. Of late, liquid overfeed pump circulation systems are becoming more and more popular. Basically, these are based on a flooded evaporator operation with higher-than-required liquid supply to the evaporator. There is lack of clarity on what should be the pump circulation rate and what is the exact meaning of overfeed rate. Nor is information available on whether to use hand expansion valves or flow control valves, what is the ideal pump discharge pressure, what should be the temperature difference between inlet and outlet of evaporator, how to adjust optimum flow etc.

At the time of designing and selecting the equipment for a particular project, the author had to struggle to get satisfactory design guidelines. This article has been prepared with a view to give some additional information and to serve as a guideline for persons who wish to design pump recirculation systems, based on the design of an actual system already commissioned and working satisfactorily.

A fish freezing plant using plate freezers of 100TR(351.7kW) refrigeration capacity was needed to be designed and installed for Gadre Marine Exports in Ratnagiri, Maharashtra. An ammonia refrigeration system using reciprocating compressors and pump circulation overfeed system was selected.

About the Author

Ramesh Paranjpey is a mechanical engineer with an M.Tech in refrigeration from IIT Bombay with over 35 years experience. He has worked in very senior positions starting with Kirloskar Pneumatic in Pune, Carrier Transicold in Bangalore and Singapore as well as Voltas-Air International Pune. Presently he works for himself as a technical advisor & consultant. He is an ASHRAE Fellow, past president ASHRAE W.I. chapter and past president ISHRAE Pune chapter. He can be contacted at *pramesh@vsnl.com*

Figure 2 : Five plate freezer bank operational in processing hall.

The plant consists of two systems each of 100 TR capacity and each system using five plate freezers, two 12-cylinder two-stage reciprocating compressors, evaporative condensers and a low temperature ammonia storage vessel with ammonia circulation pumps, using hot gas defrost arrangement.

While designing and selecting equipment for the refrigeration installation some of the major factors needed to be considered were:

1. Choice of refrigerant
2. Selection of system
3. Selection of components forming part of the system, like compressors, condensers, circulation pumps, low temperature/pressure vessels
4. Proper system piping design/adjusting/balancing of components

Why Ammonia Was Selected?

Ammonia is the most preferred refrigerant for cold storages above 50 ton (approx. 175kW) capacity for its obvious advantages such as being a natural refrigerant having zero ODP, GWP values, highest efficiency compared to other refrigerants, easy to detect leaks, excellent heat transfer characteristics, low cost and availability in all parts of the country.

Energy Saving – Besides the above main reason for selecting ammonia refrigerant system, is the smaller mass flow rate required to be handled with an ammonia system as compared with R-22 refrigerant systems. A 100 ton plant with –25°C evaporating temperature /+40°C condensing temperature requires about 1193kg/hr of saturated liquid ammonia to provide the desired refrigeration effect. If, for the same application R-22 is used then the liquid flow rate would be about 8317 kg/hr. This means that a pump having a circulation rate in the ratio 4:1 would require 10 kW pump motor for R-22, whereas for ammonia it would be only 3 kW. Since these pumps are running continuously, so long as plant is in operation, it contributes to a substantial additional operating cost.

Type of System – A two-stage refrigeration system was essential for this application, since the freezers are to operate at –32°C saturated evaporating temperatures and +38°C condensing temperature with cooling water available at 30°C temperature.

Normally, ammonia refrigeration systems use either gravity flooded design or a pump circulation system.

Why Overfeed Ammonia Pump Circulation System was Selected

As the number of evaporators increase and as the temperature requirement gets lower and lower, liquid recirculation/overfeed systems are preferred. Normally for more than 3 to 5 evaporators and located far away from the machine room, liquid recirculation is the best option.

In liquid overfeed systems the refrigerant liquid coming out of the receiver is expanded to the required pressure/temperature and this liquid is stored in a low pressure receiver. It is then pumped to the various operating evaporators, like product coolers, blast freezers or plate freezers. It thus forms an independent low side circuit. The compressor sucks vapours from this low pressure receiver and the cycle repeats.

Overfeed means, that much more liquid is fed to the evaporator than actually vaporizes. Excess liquid is called “overfeed”, which returns to the low pressure side accumulator or LP receiver. Thus the mass flow rate handled by the compressor is less than the mass flow rate circulated in the evaporator.

In a gravity flooded system, the refrigerant mass flow circulation rate in the evaporator and the compressor is the same.

The advantages of liquid overfeed or pump circulation systems are given below:

1. At low temperatures, achieving good heat transfer in the evaporator is crucial since the plant operates with high compression ratios, where quantities of flash gas are appreciable, affecting proper wetting of the surface. The pump circulation with overfeeding causes more wetting of the tubes associated with high velocity of circulating liquid refrigerant, resulting in higher heat transfer rate.
2. In a pump circulation system design, the advantage is, that it effectively decouples the refrigeration system from the evaporators allowing more efficient operation and a lot of flexibility in design and operation
3. Overfeeding also ensures that the vapours coming out of the evaporator are close to saturated condition, without any superheat, thus lowering compressor inlet gas temperature resulting in lower discharge gas temperature,

continued on page 106

continued from page 104

which is a critical factor for ammonia systems working at low temperatures. Higher discharge temperatures pose problems for the compressor, like deterioration of lubrication properties, leading to increased wear and premature failure.

4. The compressors are protected from liquid slugs resulting from load fluctuations and due to malfunctioning of controls, since suction gas first returns to the low pressure vessel and not directly to the compressor.

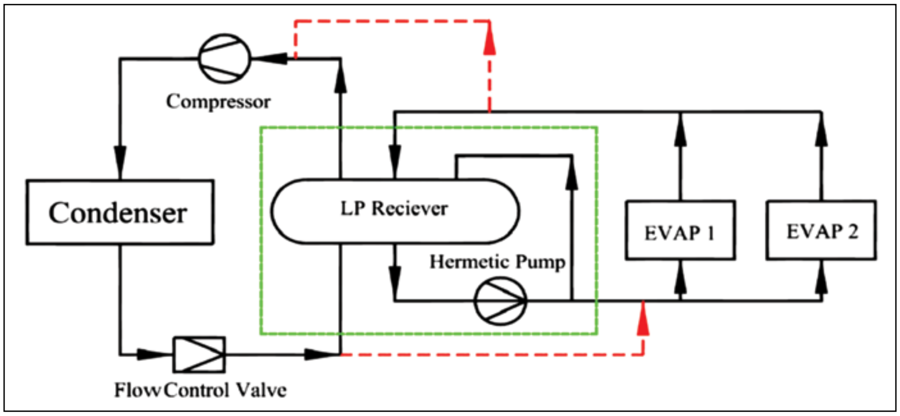

Figure 3 : Gravity system v/s pump circulation system.

5. Flash gas resulting from refrigerant throttling losses is removed at the low pressure receiver before entering the compressor. This gas is then directly drawn to the compressor and eliminated in the low side system design. It does not contribute to additional pressure drop in the evaporators or wet suction return line to the low pressure receiver.

6. There is uniform liquid distribution in the freezers and all perform equally. In gravity flooded systems, the evaporator closest to the compressor receives more liquid whereas the evaporator farthest may starve; also the pressures/temperatures are not equal in all evaporators.

7. Refrigerant controls, level indicators, alarms, refrigerant pumps and oil drains are located in the plant room under supervision of the operator and not far away.

8. With proper controls, automatic operation is convenient and evaporators can be defrosted with little disturbance to the system.

9. Refrigerant feed is unaffected by fluctuating ambient conditions and condensing temperatures. The flow control valves need not be adjusted after initial setting.

10.Oil does not accumulate in the evaporators and need not be drained from each evaporator. Oil draining is simple, at the low pressure receiver located in the plant room.

11.Fault finding and trouble shooting is also easier since, as long as enough liquid is available in the low pressure receiver at the required temperature to meet the demands of all the evaporators, then one can be sure that the refrigeration system design is OK. It is then easier to concentrate on performance analysis of the low/evaporator side independently, in case of malfunctioning of the system.

12.In case of sudden stoppage of plant, production does not come to a halt as some liquid at low temperature is available in the LP receiver acting as a reservoir for some duration.

The area indicated in the green rectangle in *Figure 3* shows additional components needed for a pump circulation system. If these are deleted and the red dotted lines are joined, then it becomes a standard gravity feed system layout

The distinguishing components in the overfeed system design, over and above the normal gravity flooded systems, are low pressure receiver, circulation pumps and refrigerant liquid and wet return pipe line. The required additional controls and installation/construction of this vessel, and pumps also need special attention.

Selection of Compressor

In refrigeration capacities less than about 350 kW, reciprocating compressors are slightly more efficient. Also at part load, a reciprocating compressor with cylinder unloading has a higher efficiency than the screw compressor. The first cost of a single reciprocating compressor is normally less than that of a screw compressor of the same pumping capacity.

Choice of both a reciprocating and a screw compressor was evaluated. Efficiency, leading to power saving, was the main consideration. As the load remains more or less constant, unlike in air conditioning plants, the peak load power performance of screw and reciprocating two-stage compressors was compared. A two-stage reciprocating compressor with 12 cylinders using 3 cylinders for high stage and 9 cylinders for low stage, using open flash inter-stage cooler was selected. Each system used two compressors thus giving flexibility of operating 50% freezers with one compressor only and thus by keeping the second compressor off, considerable power saving is achieved. For part load operation each compressor was provided with 5 steps of capacity control by a way of a cylinder unloading mechanism activated by a suction pressure controller. Capacity of each step of compressor was equal to the capacity of each freezer.

continued on page 108

continued from page 106

Figure 4 : Bank of four KC93 Kirloskar two-stage compressors.

For this particular project, the peak power consumed by the reciprocating compressor at –33°C and +38°C operating conditions was 1.654 kW/TR whereas equivalent screw models from various manufacturers had power consumption ranging from 1.98 kW/TR to 2.33 kW/TR

Fisheries plants are normally closed during the monsoon, as this is a breeding season and boats are not allowed to catch fish for a certain period and hence an operating time of 16 hours per day and for 8 months plant running time has been considered, for calculation of power savings.

For 100 ton refrigeration plant operating 16 hours per day and for 8 months duration, the power saving is:

(1.98 – 1.654) × 100 × 16 × 30 × 8 = 125,184 kW

This is a big saving in running cost besides the initial cost saving, as reciprocating compressors are available at lower initial costs.

Selection of Interstage Cooling Method

Out of the various methods used for inter-stage cooling of gas, open flash cooler gives the maximum efficiency, since the discharge gas from low stage is cooled to saturation condition in the open flash cooler by the liquid from high stage, which in turn gets cooled to the same saturation temperature, thereby increasing refrigeration output without any additional compressor power. In all the other methods of inter-stage cooling, like direct refrigerant liquid injection in the suction to high stage or shell and coil inter-stage cooling or a combination of any of these, results in higher power consumption per ton of refrigeration.

Figure 5 : Open flash cooler.

To highlight the power saving benefits, following figures are taken from a manufacturer's catalogue :

For a 12 cylinder two-stage ammonia reciprocating compressor, working at –40°C evaporating and +35°C condensing temperature.

System type	Method of cooling	Capacity –TR	Power Consumption - kW	kW/TR
System 'A'	Liquid injection inter-stage	45.07	97.9	2.172
System 'B'	Closed shell and coil cooler	51.55	102.2	1.9825
System 'C'	Open flash cooler	53.274	103.4	1.94

The yearly saving due to use of system 'C' instead of system 'A' are:

(2.172–1.94) × 100 × 16 × 30 × 8 = 89,088 kW.

Hence a system using two-stage reciprocating compressors with open flash cooler was selected.

Selection of Condenser

Normally for industrial ammonia refrigerant systems, the choice is between a shell-and-tube condenser/PHE with cooling tower or evaporative condensers. In case the distance between the compressor and the location of the heat rejection equipment is long, it is simpler for water be piped to the cooling tower, rather than running a long refrigerant line. However, if space is available nearby or on top of the plant room, the evaporative condenser is the best choice as it provides the lowest condensing temperature for a given wet bulb temperature. The lower condensing temperature leads to lower compressor power consumption and also results in lower compressor discharge temperatures.

The superior performance of an evaporative condenser is because of avoidance of the intermediate fluid i.e. the cooling tower water in the heat transfer process. The condensing temperature, in case of a shell-and-tube condenser is guided by the leaving water temperature from the condenser, whereas in case of an evaporative condenser the wet bulb temperature becomes the guiding factor. The major heat transfer mechanism in the evaporative condenser is the evaporation of water from the condenser tubes which carry high temperature refrigerant vapours.

continued on page 110

continued from page 108

Figure 6 : Evapco make evaporative condensers

The rate of heat transfer, therefore, depends not only on this temperature difference but also on the water vapour pressure difference of the water falling on the tubes and the vapour pressure surrounding the tubes. In case of a shell-and-tube condenser, the heat transfer is only sensible heat transfer process on the water side since no evaporation of water takes place in the heat exchanger.

For example, with air wet bulb temperature of 28°C to 29°C, one can easily achieve condensing temperature of 38°C with an evaporative condenser, where as in a water cooled condenser, with a water inlet temperature of 32°C and water leaving temperature of 36°C, the condensing temperature would be a minimum 40°C without making the heat exchanger unduly large and expensive.

Lowering the condensing temperature from 40°C to 38°C with the use of evaporative condensers for 100 TR plant capacity has resulted in compressor power saving of:

$$5.44 \times 16 \times 30 \times 8 = 20{,}889.6 \text{ kW}$$

Another factor in favour of the evaporative condenser, is the lower water pumping cost as the water circuit is short and there is less pressure drop in the piping compared to a S&T condenser with long water lines to the cooling tower.

The spray water flow rate in an evaporative condenser is nearly one third that of the flow rate required in a water cooled condenser/cooling tower combination.

Considering all the above aspects, two evaporative condensers were selected resulting in much lower energy consumption and costs.

One method of reducing power consumption with evaporative condensers is to use speed control for the fans, to vary fan speed and thus reduce power consumption when wet bulb temperatures, are lower than design wet bulb temperatures, which is the case most of the time. Shutting off of one condenser when the load is reduced is also another possibility of saving power one can use.

Low Pressure Receiver

The design and construction of a low pressure vessel with its proper piping and control arrangement is very important and in many plants which the author has seen, this aspect is not handled with due importance and hence a detailed write up on these specific items of pump circulation system is given below for the benefit of the designers of such systems.

The LP vessel performs two major roles in the system:

1. Liquid vapour separation to ensure only vapour is sucked by the compressor.

2. Liquid refrigerant storage at a required temperature, so that it is available as a utility to meet the requirement of either one or more operating evaporators

The important liquid refrigerant levels to be considered while designing the vessel are as under:

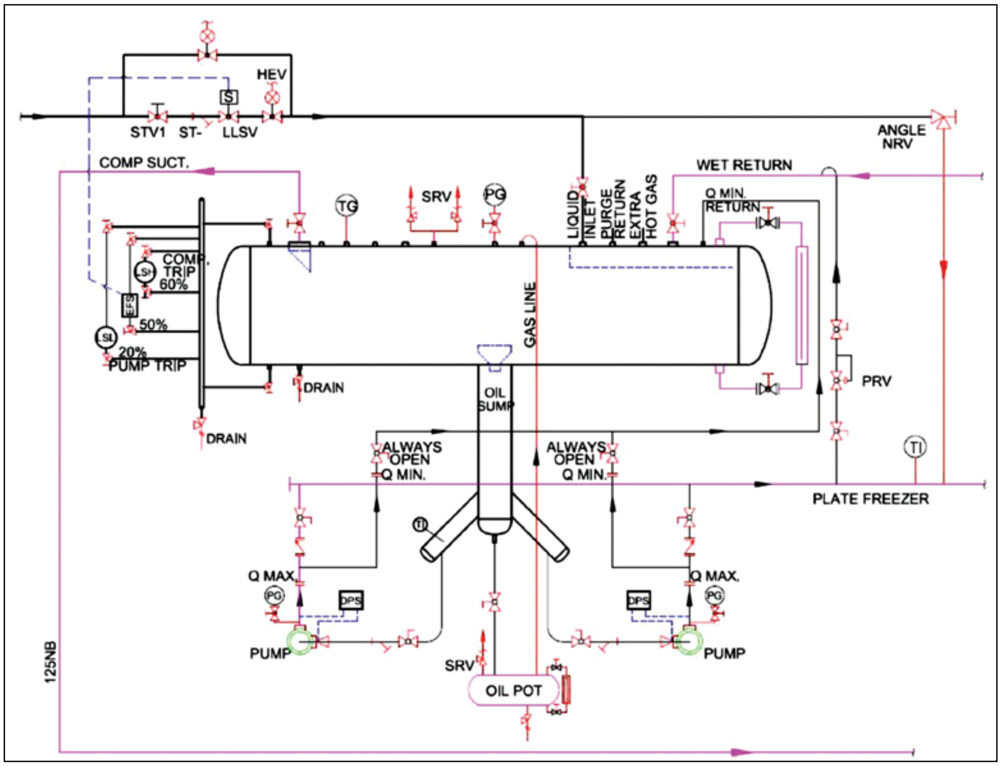

Figure 7 : Details of Low Pressure vessel -P&ID

continued on page 112

continued from page 110

Figure 8 : Low pressure vessel in operation

1. ***Working liquid level*** – A float switch regulates a solenoid valve in liquid line. When liquid level falls and solenoid valve opens, liquid refrigerant is admitted to the vessel and when the level reaches desired value, the solenoid valve closes. This operation is similar to standard gravity flooded systems. The major difference is, in many cases the high side may be a two-stage plant using open inter-stage cooler and in which case the pressure drop available across the hand operated expansion valve is very low and the valve should be sized accordingly. Similarly, the height at which the inter-stage liquid cooler is installed is important and should be equal or more than the liquid level in the low pressure receiver. If the inter-stage cooler is at a lower level than the LP vessel then due to elevation difference, the liquid line pressure drop takes place. If this pressure drop is more or equal to the pressure difference needed from inter-stage to LP vessel, then it becomes very difficult to admit liquid in the LP vessel or the rate of liquid filling becomes abnormally slow. Thus, location of various vessels and the sizing of expansion device are critical issues.

The quantity of liquid required to be stored in the LP vessel can be calculated based on the internal volume of all working evaporators and the associated pipe lines. The rate of circulation and the quantity required to be stored are two separate issues and should be treated independently.

2. ***Surge volume*** – the volume above working liquid level provided in the vessel is known as surge volume and serves the purpose of accommodating liquid that might be forced out of evaporators during defrosting of one or more evaporators. Another aspect that needs to be considered is the liquid in the wet return line from the evaporators to LP vessel, which may drain in the LP vessel if power shut down takes place and the liquid refrigerant pump becomes inoperative. In normal circumstances, the amount of liquid plus vapour returning is the same as the pump circulation feed to evaporators, but when the pump is not taking out liquid from vessel due to reasons mentioned, the extra liquid quantity gets added to the vessel from the wet suction line and the vessel design needs to take this point in to account in addition to defrost quantity. This level can be provided with alarm indication making the operator aware that liquid level is likely to reach dangerous levels and something is wrong with the plant, needing investigation.

3. ***Ballast Volume*** – The other important level on the lower side of the operating liquid level is the liquid required, either during start up after a pump down cycle or if additional evaporators are taken on line for operation. During this period, the liquid drawn from the vessel is at a higher rate than it is returning to LP vessel. The alarm indication for this minimum level should be provided making the operator aware about the falling liquid level. It does not mean that pump stops at this alarm indication and it continues to run. The ballast volume is generally calculated for 5 minutes period, meaning pump flow rate multiplied by 5.

4. ***Low Level Trip*** – A pump always needs liquid refrigerant at the inlet or on the suction side and hence further drop in liquid level should be set to trip the pump before the vessel empties. Getting vapour or bubbles at the entry of the pump due to any reason should be avoided for trouble free operation of the pump and the overall system.

5. ***High Level Trip*** – high level cutout set at a higher level than surge volume level will trip the compressor for its protection from liquid entry.

The high level and low level cutouts are actuated by independent float switches thus requiring in all, three float switches. The third one is for maintaining normal working liquid level whereas alarms can be actuated by 4-20 mA output signal from sensors.

6. ***Circulation Ratio/Overfeed Rate*** – In overfeed systems, deciding the pump circulation rate is an important factor and needs careful consideration. Firstly, it is essential to understand the difference in recirculation rate and overfeed rate. The circulation rate is the ratio of actual flow rate supplied to the evaporator by the pump to the flow rate at which refrigerant evaporates. The amount of liquid evaporating depends on the heat load and the circulation rate could be more by 3 to 4 times, depending on the type of evaporator design. A circulation rate of 4, means if the quantity of liquid entering the evaporator is 4 kg then out of it 1 kg liquid evaporates and 3 kg of liquid along with 1 kg of vapour returns to the LP receiver.

Overfeed rate is the ratio of liquid mass upon the vapour mass exiting from the evaporator. In the case mentioned above it would be 3. If all the liquid entering, is vapourized,

continued on page 114

continued from page 112

Circulation rate 4:1 =

Supply to freezer (4kg) / Evaporation (1 kg)

OR

Overfeed rate 3:1 =

Liquid return (3 kg) / Evaporation (1 kg)

Figure 9 : Circulation rate V/S overfeed rate

then overfeed rate would be zero and the recirculation rate would be 1, which happens in a case with normal flooded systems, where pumps are not used.

These excess or overfeed quantities of liquid may seem wasteful and unnecessary, but as mentioned earlier they perform the important function of totally wetting the inside of the evaporator coil surface with liquid refrigerant, from the beginning to the end of the coil, which gives a very high heat transfer coefficient and optimizes coil surface.

Construction and Nozzles for Low Pressure Receiver

Many designers think that low pressure vessel is just an ordinary tank like a high pressure receiver. However the following paragraphs have been specially added to impress upon designers that each and every aspect of the vessel construction also needs careful consideration.

Since a low pressure receiver has a large quantity of refrigerant stored at a low temperature and it may not be possible to pump down this liquid to the high pressure receiver every time when the plant is shut down, it is suggested that the vessel should be designed for maximum standing pressure corresponding to the maximum surrounding plant room temperature. Although standard literature mentions a design pressure as 150 psig, it is highly recommended that this vessel be designed for 300 psig pressure similar to a high pressure receiver. The second important point is, since it is storing low temperature liquid, whose temperature, in most cases would be lower than minus 20°C, the material of construction used should be low temperature steel like SA516/517 grade 60/70.

The sizing of a low pressure receiver, both for vertical and horizontal design is given in *ASHRAE Refrigeration Handbook 2006*, Chapter 1 and many manufacturers also give ready selections based on tonnage, refrigerant used and operating temperatures. The *Industrial Refrigeration Handbook* by W.F. Stoecker also gives design guidelines for sizing. A horizontal type vessel is preferred to provide sufficient surface for the settlement of the oil in the drum and to enable stable suction head conditions.

Besides selecting the diameter and length of the LP vessel, various nozzles fitted on the vessel need some special considerations. It is suggested that a channel/trough on one end of the vessel may be internally created, in which all the wet liquid return lines from evaporators, main liquid entry to vessel from HP or intermediate pressure vessel through expansion valve, as well as other pipes which are likely to carry liquid, enter. These pipes would also include defrost return, minimum flow return from refrigerant circulation pump/ and pump bypass. The liquid falling from all these nozzles first enters the trough and then flows into the vessel, thus eliminating chances of short circuiting directly to the compressor suction. The compressor suction connection is provided at the other end keeping a maximum distance between wet return and compressor suction. The suction pipe is also provided with a pipe extending in the vessel with 45° cut or 'U' bend on the opposite direction of liquid entry for the obvious reason of avoiding liquid droplets entering in the compressor suction line. The main liquid outlet connection is at the bottom of the vessel, from where the liquid refrigerant pump inlet connection is taken. This pipe is to be sized adequately in diameter and length so that the pump suction does not receive gas bubbles, as well as the liquid flow to the pump suction is laminar (3fps). The entry point has to be provided with a vortex breaker plate and the pipe should protrude in the vessel by about 2 inches to prevent oil accumulated in the vessel from entering the pumps. Normally two pumps are provided; one working and other as standby and hence two outlets from this vertical leg at 15° inclination, feeding to pump suction are provided. Independent connections to each pump from the vessel is the best option in case more than one pump is working. The oil drain at the bottom of the liquid leg is essential, from which oil can be drained in a pot or directly to the outside, depending on which refrigerant is used. In case the drop leg is protruding in the vessel then oil drain pot should be connected directly to the vessel drain. The drain pot should not be insulated and can have a 60W electric heater element as well. These are some of the special requirements of LP vessel construction.

What Should be the Optimum Circulation Rate?

ASHRAE Refrigeration Handbook 2006 recommends minimum circulating rates (Chapter/page1.4)

Ammonia Down feed	6 to 7
Ammonia up feed	2 to 4
R-22 up feed	3
R-134a	2

Many articles describe a rate of circulation as high as 20 incase of brine circulation where no phase transfer takes place and the increased velocity leads to increased heat transfer rate. Flow rates of 20 to 40 have been also tried with R-12 refrigerant in plate freezers.

continued on page 116

continued from page 114

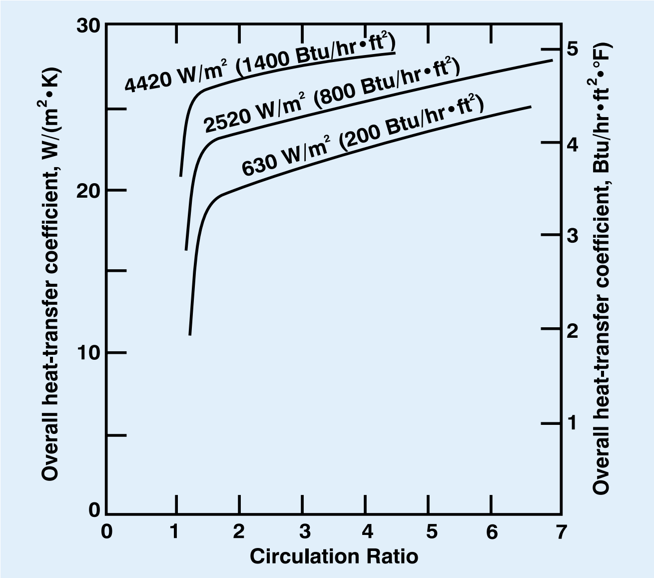

Figure 10 : Heat transfer rate

U.K. Institute of Refrigeration recommends 7-14 recirculation rates for plate freezers using ammonia refrigerant.

There are therefore no clear cut guidelines and explanations as to how to decide the circulation rate and the designer has to establish the same based on his experience and the system parameters.

If the circulation rate is more than 1, then it surely increases heat transfer coefficient, but on the other hand increases the cost of pumping by way of a larger pump and power cost. It also increases the pressure drop through the evaporator. It also means, if the inlet temperature to evaporator is to be controlled as per design requirements, then the outlet pressure would be lower than the inlet pressure equivalent to pressure drop and this means the compressor suction pressure would be lower leading to higher power consumption, reduced compressor capacity and higher discharge gas temperatures at the compressor outlet.

Many experiments have been tried to find optimum flow. The heat transfer coefficient increases as the circulation rate increases. The largest gain in heat transfer is when the entire evaporator inside area remains completely wet from inlet to outlet, as it is the latent heat transfer or phase change process which absorbs maximum heat from the fluid or product. This rate of circulation could therefore be any where from 1 to 2 when abrupt increase in 'U' value takes place and then further increase in circulation rate beyond 2 leads to a marginal increase in heat transfer coefficient as can be seen from the *Figure 10*.

Selection of Canned Motor Hermetic Pump

Considering all the above aspects for a capacity of 100 ton refrigeration plant operating at +38°C/–33°C conditions, initially a pump of 11.5 kW input power (10:1 ratio) was selected and during operation and after adjusting the flows it was noticed that most of the liquid is getting by-passed and we do not need such high circulation rates to be considered for selection of pumps.

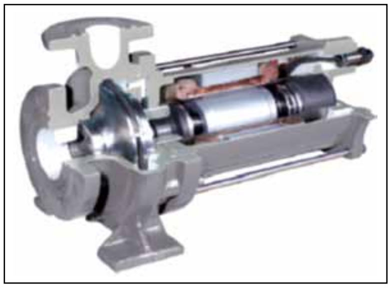

Figure 11 : Hermetic Pumpen – Germany ammonia pump

Figure 12 : Ammonia pumps in operation

In the subsequent installation of an identical plant a pump with 4.3 kW input power based on 4:1 circulation rate was installed and worked successfully with out affecting the plant performance.

Thus saving in power was to the tune of:

$$(11.5 - 4.3) \times 16 \times 30 \times 8 = 27{,}648 \text{ kW.}$$

It is therefore essential to select the right size pump and in addition to this it is recommended that a variable speed drive could be installed to further optimize the speed for required flows, so that the pump draws minimum power.

How to Adjust Optimum Flow Rate?

In the case of pump circulation systems using multiple evaporators, it may not be easy to adjust the hand expansion valve of each cooler accurately as adjustment of one cooler hand expansion valve would require readjustment of hand valves of other coolers. Flow regulating valves (FRV) installed at the inlet of each cooler are therefore recommended instead of hand expansion valves. The automatic flow regulating valves serve two functions: it maintains a constant, but adjustable liquid ammonia refrigerant flow rate to the evaporator and also acts as a check valve during hot gas defrost.

To adjust the flow, install temperature probes at the inlet and outlet of each evaporator. When plant operations stabilize, say after 30 to 45 minutes, then note the position of FRV and then gradually turn the valve more and more towards the close position, until the

continued on page 118

continued from page 116

gas outlet temperature gets a lot higher, all of a sudden, than the inlet liquid temperature. For example, if the inlet temperature is say –32°C, the outlet temperature could rise to –20°C. This indicates that at this point the amount of liquid supplied is equal to amount of liquid evaporated. This also means that all the liquid supplied has evaporated and at the gas outlet point, we are having approximately 10 to 12° superheated gas.

Measure how many rounds you are now, from having a totally closed position. This position is equal to a re-circulation rate of 1:1, same as we get in normal gravity flooded systems. Then open the FRV three times more from this position to get a recirculation rate of four. As mentioned earlier a higher circulation rate is of no benefit what so ever.

The temperature difference between the inlet and outlet of each freezer should be in the region of 1 to 1.5°C. Adjust the flow rates accordingly.

What Should be Inlet Pressure to Evaporator?

Most of the designers/users feel that higher the pressure, better is the pump flow to coolers and better is the performance. This thinking is incorrect. The inlet pressure to the evaporator should be just enough to overcome pressure drop inside the cooler and the wet refrigerant return line up to LP vessel. For example, if the plant has been designed for an evaporating temperature of –32°C, saturation pressure corresponding to –32°C is 1 bar absolute. If the pressure at the inlet is higher than this, say 2.0 bar, the corresponding saturated evaporating temperature is –18°C. This means, although the liquid supply temperature is –32°C, it will not be evaporated till the pressure inside the evaporator drops to 1.0 bar. It means, using part of the evaporator area for sensible cooling (overcoming 14°C sub-cooling) instead of evaporating and thus loosing valuable surface area from being effective as there would be no boiling, just a temperature increase.

In most of the installations it has been observed that it takes a much longer time to cool if the flow and inlet pressure is not properly adjusted. With higher flow rates and pressures, instead of evaporation, most of the area is used to overcome sub-cooling. In abnormal circumstances, evaporation may actually take place in the suction line or even in LP vessel, if the circulation rate is too high.

Piping Design and Valve Selection

Selection of pipe sizing based on pressure drop calculations, taking into account length of liquid and suction lines is recommended. The suction line is most critical and calculation of pipe sizing in pump circulation systems is complex. Any higher pressure drop would lead to the compressor operating at lower suction pressure, leading to higher power consumption. The slope of wet return line also is critical as it has two-phase flow and liquid of higher density may flow along the bottom of pipe. *ASHRAE Refrigeration Handbook 2006* (1.7) recommends, as a simple practice, one size higher than dry suction line selection for wet return line. The slope of lines should be towards the evaporators and receivers, away from the compressor.

Selections of valves – Globe/ball valves are normally used. Popularity of ball valves is increasing, due to full bore open characteristics leading to minimum pressure drop. Besides this, many such valves are quarter turn open/close valves making them quick opening. An angle valve causes nearly half the pressure drop compared to a globe valve and should be preferred if the physical arrangement permits. Valves should be installed so that they close against high pressure side. Globe valves should be mounted with stem horizontal.

One estimate calculated that a fully open globe valve in liquid/vapour line between evaporator and low pressure receiver, causing 7.5 kPa (1.1 psi.) pressure drop, adds 9400 US Dollars to the annual operating cost for a 600 ton plant, whereas a fully open ball valve would add only 43 US Dollars to the annual cost (*Industrial Refrigeration Hand Book*-W.F.Stoecker-page 394)

Hot Gas Defrost for Plate Freezers

The diagram above indicates

continued on page 120

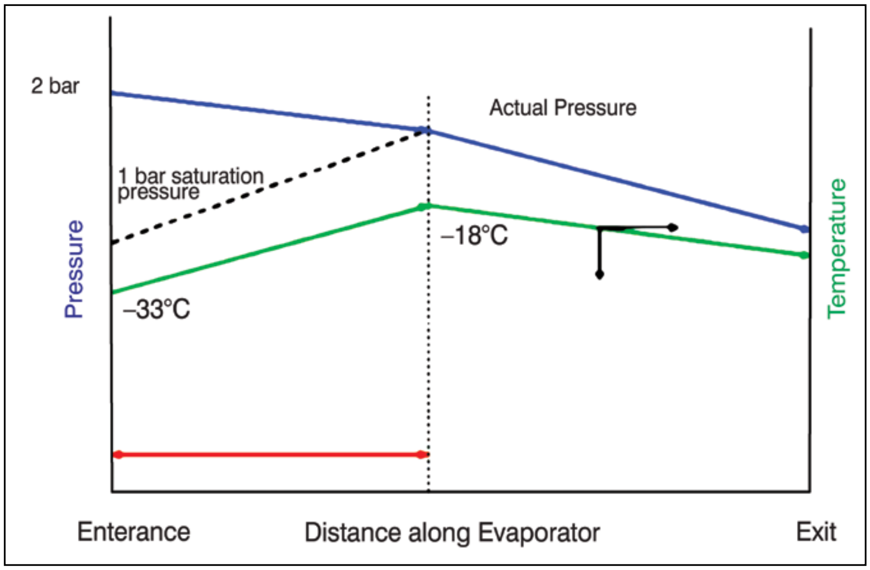

Figure 13 : Effect of inlet pressure on evaporator performance

continued from page 118

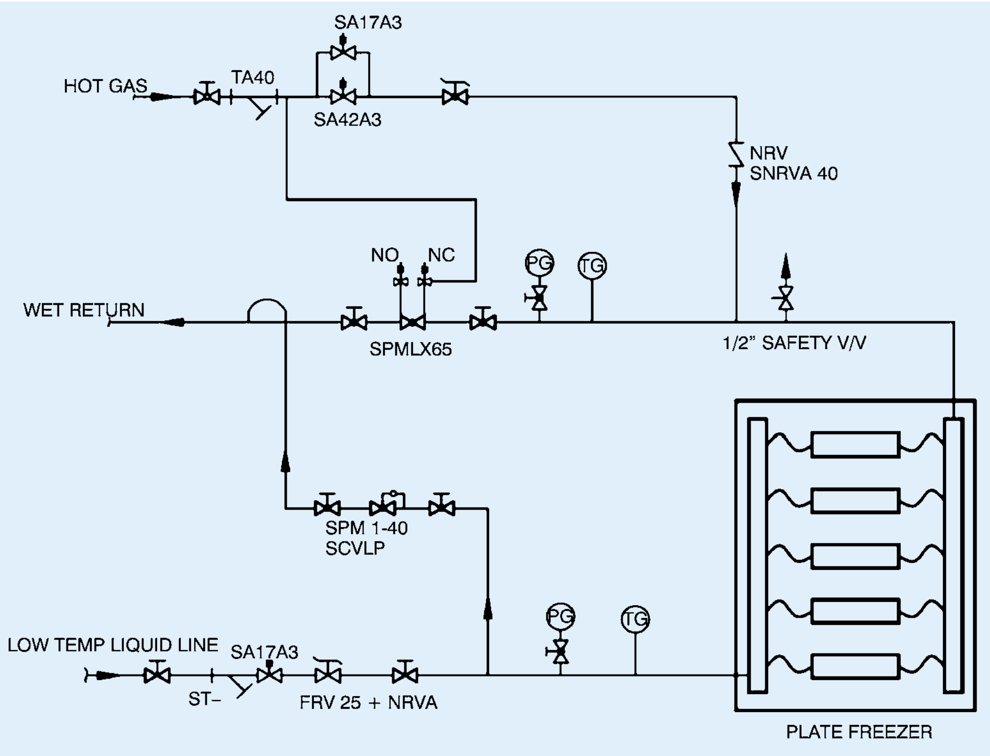

Figure 14 : Hot gas defrost arrangement

suggested method of providing piping and control arrangement. The success of hot gas defrost depends on many factors and some of the important ones are indicated below:

1. Hot gas defrost is the best alternative as the heat source acts from within, where as in a water/electrical defrost the heat source is from outside.

2. Although hot gas defrost is the most effective way of defrosting, it is equally complicated, troublesome and may be inefficient if not properly designed. Variations in system piping arrangements, management of pressures, temperatures and liquid refrigerant make implementation of a hot gas defrost arrangement complex.

3. Best results are obtained when upstream pressure hot gas is at 8 bar with reduction to 4/5 bar in the cooler/freezer and temperature of 5 to 13°C. Warmer temperature will not improve defrost efficiency. This is because most of the heat for melting frost comes from hot gas latent heat rather than sensible heat.

At 5°C and 5 bar, latent heat is 1467.38 kJ/hr for ammonia. At 22°C and 9.13 bar, latent heat is 1178.22 kJ/hr, which means 22°C defrost temperature would in fact require 5% more gas than 5°C to provide the same latent heat content.

4. Before hot gas is admitted, it is essential to empty the freezer by pumping out liquid. If this is not done most of the hot gas heat is used in evaporating residual liquid rather than heating metal.

5. The soft hot gas defrost arrangement is recommended for larger heat exchangers as shown above. This means double solenoid valves in hot gas as well as wet return lines to ensure slow pressure equalizing by opening smaller valve first and then followed by main valve to avoid vibrations, noise and undue stresses leading to rupture, damage to valves, piping and other components.

6. It is recommended that evaporator defrost should be returned to the intermediate pressure vessel and not to the low pressure vessel in case of a two-stage system. This has two advantages. Firstly, it does not disturb the LP vessel pressure/temperature conditions during defrost operation and thus other operating coolers work without any disturbance. Secondly, defrost liquid pressure and intermediate vessel pressure difference is much lower than defrost pressure and LP vessel pressure and thus saves considerable energy.

7. Example : A 70 kW coil defrosting for 12 minutes will condense up to 10.9 kg/min. of ammonia of total 130.67 kg, the enthalpy difference between returning low stage –40°C and intermediate vessel at –6°C is (172.34–19.17)=153.17 kJ/kg i.e. 20000kJ × 60 ÷ 12 = 100073.61kJ/hr ÷ 12666 = or 7.9 TR or nearly 27 kW removed from the –40°C booster compressor for 12 minutes during each defrost.

Purging

As the system working at –32°C or lower temperature operates below or at atmospheric pressures for most of the time, chances of air leaking in, exist. Automatic air purgers have been installed with connections from evaporative condenser outlets as well as high pressure storage receivers. The purgers are activated by sensing discharge pressure and when it exceeds preset limit the non-condensables are purged out.

Non-condensable gases present in the system lead to higher operating discharge pressures thereby consuming more power.

continued on page 122

continued from page 120

Water Contamination

Water contamination of the ammonia refrigerant is common in many refrigerating systems. The solubility of water in ammonia makes an aqueous solution to be formed. The water gradually accumulates in the system and the effects go unnoticed. This results in additional power consumption as the plant starts operating at lower suction pressures.

The refrigerant grade anhydrous ammonia having a purity of 99.95% or water content of less than 500 PPM is recommended. As operating temperatures are lowered, the ability to absorb water increases. Anhydrous ammonia has lower saturation pressure than aqueous solution. This means the saturation temperature increases.

So, to maintain desired temperatures of the product the system must operate at lower suction pressure. Any lowering of suction pressure results in extra power consumption and loss of capacity, deterioration of evaporator capacity leading to increase in kW/ton

There are many reasons, besides charging wet ammonia, by which water accumulates, such as system components are not properly dehydrated, air enters the system during pump out cycle or while operating the system below atmospheric pressures, inadequate evacuation, rupture of tubes in heat exchangers, loose suction pressure gauge connections, chemical reaction between aqueous ammonia and oxygen and mineral oils, and many more.

The purity of ammonia needs to be therefore periodically checked. Generally, the highest water content can be found in low pressure receivers feeding the ammonia pumps. Detection of water in the system may take years before the problem is recognized and it is prudent practice to periodically check quality of ammonia and take corrective measures. (For more details refer IIAR Bulletin No. 108)

Conclusion

This is a summary of energy savings achieved:

1	Reciprocating compressors instead of screw	1,25,184 kW
2	Open flash cooler instead of direct liquid injection in the inter-stage	89,088 kW
3	Evaporative condenser instead of shell and tube	20,889 kW
4	4:1 Ammonia pump circulation rate instead 10:1	27,648 kW
5	Total power savings in a year	2,62,809 kW
6	Total saving in Rs/year for two plants @5Rs/kW	Rs. 26,28,196/-

These savings are direct measurable values and additional savings due to liberal pipe sizing, selection of ball valves wherever possible, returning defrost condensate to intermediate vessel instead to LP vessel and selecting ammonia refrigerant instead of HCFC-22 refrigerant lead to considerable further savings which are additional savings, besides what has been mentioned in the table. Use of VFD'S for condenser fans and ammonia pumps would also lead to further power savings in case plants operate at lower loads and at favourable ambient conditions than designed.

The article concludes by mentioning that every project has some unique requirements and the design and selection of equipment cannot be generalized. None the less, the knowledge of basic fundamentals is essential to arrive at proper design and installation to achieve best performance, keeping in view the overall cost aspect. Design of good control system is also essential, since it is well known that a good control system can make a marginal installation operate acceptably, while a poor control system cannot make even the best installation operate satisfactorily. A micro level analysis of each and every aspect is therefore desired while designing systems. Selecting the components, piping and control system, proper installation and safety devices, as also good operating and preventive maintenance practices will certainly help in keeping the energy and operating costs to a bare minimum while giving optimum output.

References

1. Henry B. Bonar II Right, *Sized Recirculation Systems*, Jan16, 2006

Design & Safe Operation of Ammonia Refrigeration Systems-RETA-96-8

"Why you should avoid critical flow in ammonia refrigeration systems?"

2. Rudy Stegmann - "Liquid Recirculation" - *ASHRAE Journal-November 2000*

3. Ellis Norton - "Ammonia liquid recirculation" - *ASHRAE Journal, October 2001*

4. Rick VanSeters - "Centrifugal pumps in liquid recirculation refrigeration systems" - *ASHRAE Journal-August 2003*

5. George C Briley - "Liquid Recirculation Systems" - *ASHRAE Journal-September 2004 & October 2004*

6. *ASHRAE Refrigeration Volume 2006*

7. Wilbert F. Stoecker - *Industrial Refrigeration Handbook*

8. Institute of Refrigeration (IOR) UK - *Guidance Notes for Installation and Use of Hermetically Sealed Refrigeration Pumps*

9. Various IIAR bulletins - T163

10. Parker Hannifin Corp. - Bulletin Po-11a-Hot Gas Defrost Applications Manual for Ammonia Evaporators.

11.Case study and photos from Gadre Marine Exports plant in Ratnagiri, Maharashtra.