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Ammonia Low Pressure Receivers

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Ammonia refrigerating systems should be like after dinner speeches – short, simple and clean. Unfortunately many traditional ammonia refrigerating systems have grown up over the years and are the exact opposite – widely distributed, complex and thoroughly contaminated with corrosion products and tarry fractions of lubricating oil.

It is the object of this paper to show that ammonia systems need not necessarily be so.

Unfortunately a myth is being perpetuated to the effect that central refrigerating systems give great advantages in efficiency and reliability over individual low charge systems. It is also an objective of this paper to show that the relatively minor advantages of common systems are often overshadowed by the benefits which can accrue from using short, simple, low charge, individual systems.

The low pressure receiver type of system is one method of achieving simplicity with high efficiency.

Description

The low pressure receiver can be used either with halocarbon refrigerants or with ammonia. It is more difficult to use the low pressure receiver system with ammonia because ammonia is practically immiscible with the more desirable types of lubricating oils and has a very high ratio of latent heat to specific heat. This means that the amount of overfeed which can be produced in an ammonia system is significantly lower than the overfeed which can be achieved in a halocarbon system.

A simplified version of a reversed cycle defrosting ammonia low pressure receiver system is indicated in **Figure 1**. (See Engineering Data pages 73,74). The system is so arranged that refrigerant leaves the evaporator in a wet condition and is evaporated to dryness by the heat exchanger of the low pressure receiver.

It can be seen that, if a liquid seal is maintained at the high pressure side and that if the expansion valve is capable of opening wide enough to pass any refrigerant mass flow which the compressor can provide, then the only place in the system where a variable amount of liquid refrigerant can be stored is the low pressure receiver. Insofar as liquid is present in the low pressure side of the low pressure receiver and insofar as this liquid can only be fed to the low pressure receiver through the evaporator, it follows that the evaporator must operate in a flooded condition.

The only conditions under which the evaporator will not flood are those in which the heat transfer to refrigerant in the evaporator is capable of evaporating more refrigerant than the expansion device can supply and the condition in which the system is grossly undercharged with refrigerant.

It should be noted that, under steady conditions, the mass flow of refrigerant through the evaporator is the same as the mass flow of refrigerant through the compressor. The “overfeeding” of the evaporator is caused by increasing the refrigerating effect per unit mass of refrigerant circulated rather than by increasing the mass flow, as could be done by a pump. As previously stated, the amount of overfeed which can be achieved when the refrigerant is ammonia is very limited. This makes good distribution within the circuits of the evaporator of great importance.

The operation of the low pressure receiver system can be followed with reference to the Mollier diagram shown in **Figure 2**. The enthalpy difference produced by subcooling in the liquid to liquid heat exchanger is precisely equal to the enthalpy difference between the condition at which the refrigerant leaves the evaporator and the dry saturated condition. It can be shown for a low pressure receiver system that, at the evaporator, the following terms can be derived:

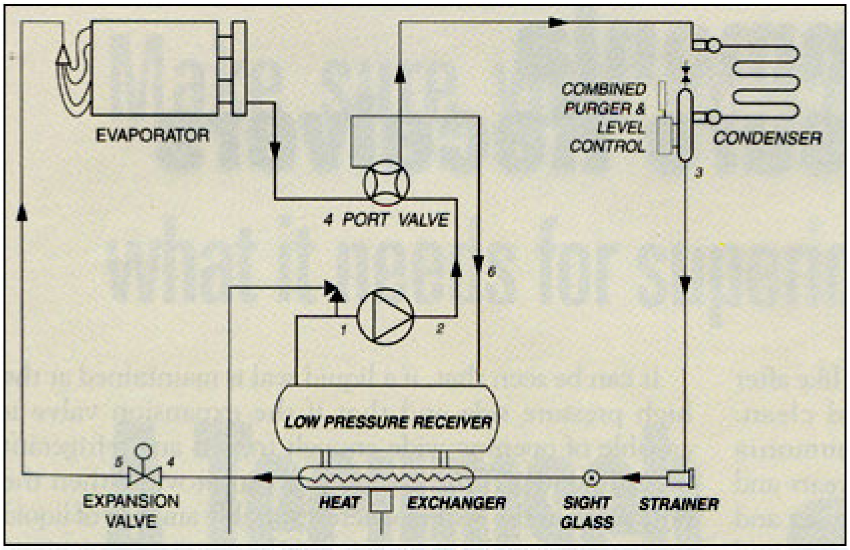

Figure 1 : Schematic diagram

$$\text{Inlet dryness} = \frac{C_p(T_c - T_e)e^{\frac{-hA}{mC_p}}}{L}$$

$$\text{Outlet dryness} = 1 - \frac{C_p(T_c - T_e)\left(1 - e^{\frac{-hA}{mC_p}}\right)}{L}$$

$$\text{Overfeed} = \frac{1}{1 - \frac{C_p(T_c - T_e)\left(1 - e^{\frac{-hA}{mC_p}}\right)}{L}}$$

Where

cp = specific heat of liquid refrigerant- kJ/kg° C

Tc = condensing temperature - °C

Te = evaporating temperature - °C

h = heat transfer coefficient in heat exchanger - kW/m^2°C

A = surface area of heat exchanger - m^2

m = refrigerant mass flow - kg/sec

L = latent heat of refrigerant at evaporation - kJ/kg

The operating conditions of a properly charged low pressure receiver refrigerating system are entirely predicated by the properties of the refrigerant, by the condensing and evaporating conditions and by the performance of the liquid to liquid heat exchanger.

The mathematics are rather more complicated for two stage systems or for screw compressor systems with economisers, however the principle is the same. Overfeed is greatest for refrigerants with low latent heat and for systems with high differences between condensing and evaporating temprerature. Increasing the heat transfer surface in the subcooler increases the amount of overfeed but subject to the law of diminishing returns as the subcooling approaches what is economically practicable.

The performance of evaporators producing low overfeeds or superheat at refrigerant exit is critically dependent on the inlet and outlet dryness conditions. The optimum design of low pressure receiver system therefore must consider not only the desirability of wetting the evaporator to its outlet but also the effect of reduced flash gas at evaporator inlet due to the high degree of subcooling which can be obtained in a low pressure receiver.

It is a significant advantage of the low pressure receiver system when used with ammonia that the refrigerant charge can be the absolute minimum possible for a system with a fully flooded evaporator. Direct expansion systems, which tend to have a high pressure receiver and a fully flooded liquid line would contain more refrigerant and pump circulated ammonia systems with accumulators and down legs would contain a charge greater by a factor of several times than the minimum charge obtainable with a low pressure receiver system.

The benefits of the low pressure receiver system and the reliability consequenty on having a limit charged, high pressure float controlled type of system are of course vitiated if low pressure receiver systems are connected in parallel because when controls fail, some parts of the system can be overcharged with refrigerant while others are starved.

Defrosting

Another advantage of the low pressure receiver system is that it lends itself to a very effective and reliable reversed cycle defrosting system using the four-port valve shown in When defrosting is required, the ball of the four-port valve shown in **Figure 1**.

When defrosting is required, the ball of the four-port valve is turned through 90° thus routing compressor discharge to the evaporator outlet and connecting the low pressure receiver inlet to the top of the condenser. The functions of evaporator and condenser are therefore exchanged. Refrigerant condenses in the evaporator, passes as liquid in the reverse direction through the liquid line, and enters the bottom of the condenser where it evaporates by heat exchange with ambient.

This type of defrost is very rapid and effective.

The defrosting is obviously much more efficient than electric defrosting because the system is acting as a heat pump and extracting the majority of the heat to defrost the evaporator from ambient. Surface temperatures on the evaporator are much less than with electric defrosting with the result that natural convection effects are reduced.

The reversed cycle defrosting system is also more effective than typical hot gas defrosting systems where a cooler is defrosted by heating it with discharge vapour at condensing pressure. In this case the heat source is the heat extraction of the other coolers in the cold store and it is, at first sight, surprising that the reversed cycle defrosting system should be more efficient. Part of the reason for this is that, during reversed cycle defrosting, there is no need to maintain an artificially high condensing temperature. Other reasons will be considered later in the paper when practical experiences are discussed.

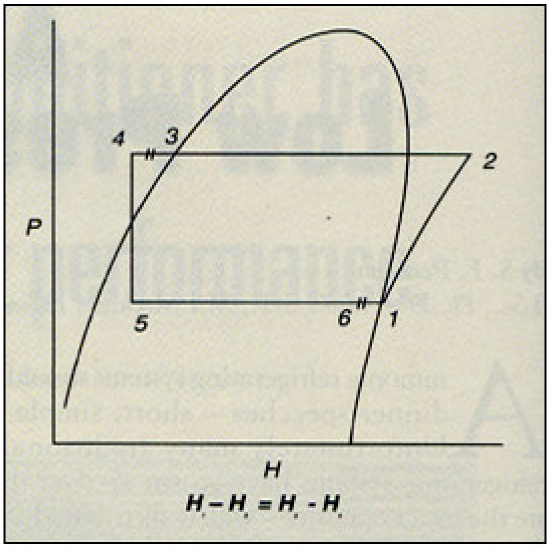

Figure 2 : Mollier diagram

Summary

The ammonia low pressure receiver performs the following functions:

1. It acts as a liquid receiver, storing excess refrigerant charge.
2. It acts as a suction trap, preventing liquid refrigerant flooding back to the compressor.
3. It allows excess refrigerant to be supplied to the evaporator, which can then operate in a flooded manner.
4. It allows evaporators to be operated efficiently at very low temperature differences because there is no need to produce a superheat signal in the evaporator. This is particularly important when the evaporating temperatures are below –10 °C.
5. It allows the ammonia expansion device to be of the high pressure float type, thus avoiding dependence on the notoriously unreliable ammonia thermostatic expansion valve.
6. It allows the collection of excess lubricating oil which can be returned to compressor suction.

The evaporator is overfed purely because the refrigerating effect has been increased by subcooling. The actual mass flow of refrigerant is not increased over what the compressor is pumping at the time.

Practical results

The effectiveness of the low pressure receiver system for air coolers using halocarbon refrigerants and for horizontal plate freezers and scraped surface evaporators has been demonstrated over many years, Ref/1/. The introduction of low pressure receiver systems using ammonia has indicated the suitability of such systems for plate type heat exchangers, scraped surface evaporators and low temperature air coolers.

Plate Type Heat Exchangers

It is a significant advantage of the ammonia low pressure receiver system when used with a plate type heat exchanger that the position of the low pressure receiver relative to the plate type heat exchanger is not at all critical. By contrast, the accumulator of a natural circulation system must be carefully positioned to achieve optimum performance. If the accumulator is too high, boiling will be suppressed in the lower parts of the plate heat exchanger. If the accumulator is too low, the refrigerant may not overfeed the evaporator, giving rise to oil recovery problems. In addition, the correct accumulator level applies at only one particular heat exchanger loading and for a particular chevron angle of the plates. Ammonia is relatively tolerant of accumulator position compared to R134a which it may be difficult to get to work at all. Even so, cases of mis-selection and use of high pressure drop plates in natural circulation systems are not unknown. The use of low pressure receiver systems overcomes all these selection dificulties and allows efficient operation at a wide variety of loads. In a recent case study, Ref/2/, the Energy Efficiency Office reported on a change from R12 shell and tube brine cooling chillers to ammonia operated plate type heat exchangers using screw compressors and a low pressure receiver system. The plate type heat exchangers were designed for the very low approach temperature of difference of 1.3K, giving a log mean temperature difference of 2.34K. It is doubtful whether a natural circulation system could have been induced to operate at such low temperature differences. As a result of the changes refrigeration energy use dropped by 20% giving a saving of £104,000 per year at 1994 prices. It is noteworthy that the five low pressure receiver packages were independent of each other and were not part of a central plant system. It is conventional wisdom to assume that a central plant system must be more efficient than a system made up of individual packages.

Such an assumption does not stand up in practice. There are two effects which give rise to the assumption that central plants are more effective. These are :

1. that under reduced load there is extra condensing surface available which will result in significantly more efficient operation of the refrigerating system, and
2. that also under reduced load additional evaporator capacity can be applied to a reduced number of compressors, thus giving more efficient operation.

Both assumptions are over simplistic.

At the Ind Coope installation where there were five low pressure receiver plate heat exchanger systems in parallel, the overall heat transfer coefficient from ammonia to the 25% calcium chloride brine was of the order of 600W/m2K. However, if the system were to be run at one-fifth capacity, cooling the brine through a smaller temperature range in all five coolers, the overall heat transfer coefficient would fall to about 200W/m2K. This means that, having reduced the log mean temperature difference from 2.35K to 1.43K, there would be practically no difference in evaporating temperature. The actual cost of running all five evaporators would be significantly greater than running a single evaporator because of the increased pump power required. Another method of using all five plate heat exchangers in parallel would be to reduce the brine flow rate by using a variable speed pump. By this means the log mean temperature difference could be kept at the same level but the internal brine velocity in the plate heat exchangers would be reduced to one-fifth of the design velocity. The nett result of this strategy would also be to reduce the overall coefficient of heat transfer to about the same order as previously, but in addition the specification of such low brine velocities would almost certainly result in maldistribution within the plate heat exchanger and very poor performance. It can be seen that operating on the appropriate number of plate heat exchangers to match the load is much more simple and also more economic.

The situation is rather different when one considers the condensing side of the systems. Boiling heat transfer coefficients in evaporators tend to increase with increasing heat flux. If the heat flux is reduced, the evaporator performance falls off drastically. There are also problems of refrigerant distribution and oil logging. In shell and tube condensers, however, the heat transfer coefficient tends to increase as the heat flux decreases. This is because the thickness of the condensate film is reduced at low heat fluxes. Unfortunately, in evaporative condensers of modern design the condensate film thickness depends to a large extent on the pressure drop in the refrigerant side of the condenser and on the slope of the tubes. Evaporative condenser manufacturers publish tables of heat rejection factors against condensing temperature for various ambinet wet bulbs. Condenser models are rated for heat rejection in proportion to their surface areas. This rating refers to a standard condition at which the heat rejection factor is 1. At lower condensing temperatures and high ambient wet bulbs, the heat rejection factors increase. **Figure 3** shows curves of evaporative condenser heat rejection factors against condensing temperature at ambient wet bulbs of 10°C and 20°C.

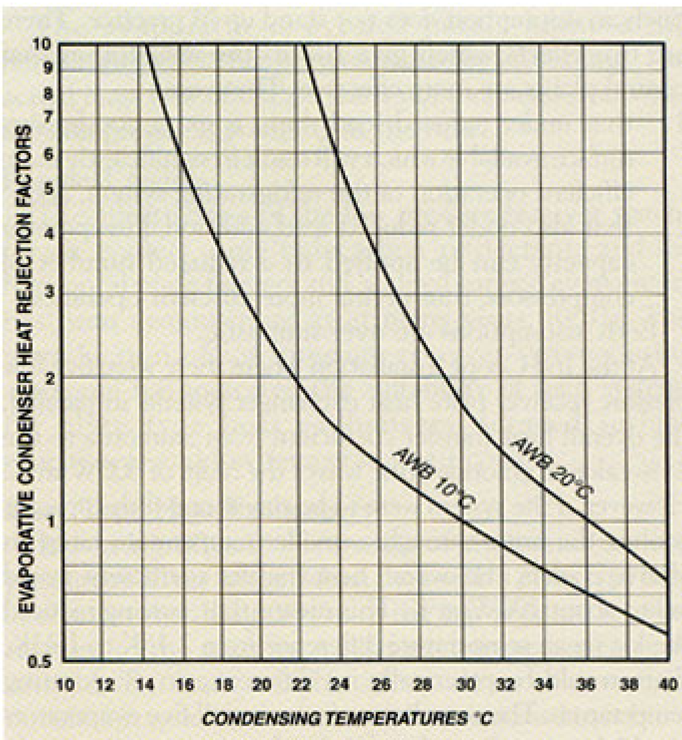

Figure 3 : Heat rejection factors for evaporative condensers

It can be seen that the heat rejection factors rise at a rate greater than exponential as the condensing temperature is reduced. In order to size an evaporative condenser the actual heat rejection must be multiplied by the rejection factor. It therefore follows that high rejection factors are a bad thing, resulting in large condenser surface requirements. Running evaporative condensers lightly loaded in paralled will produce high heat rejection factors which will tend to compensate for the extra condensing surface available. Despite the negative effect of increased heat rejection factors when condensing temperatures are decreased, there will still tend to be a reduction in compressor absorbed power. However this reduction is much less than would be calculated from simple theory and is unlikely to compensate for the increased complexity, the greatly increased refrigerant charge and the control difficulties associated with running the condensers in parallel at light load.

To summarise; there is no benefit in using a central plant on full load at design ambient.

There is little benefit in using a central plant on reduced load at reduced ambient because some measure of head pressure control is likely to be required for practical reasons.

The main advantage would be when operating at reduced load at or near the design ambient which is not a common occurrence under U.K. climatic conditions.

Scraped Surface Evaporators

Low pressure receiver systems have always worked particularly well in conjunction with scraped surface evaporators. The reasons for this are not entirely clear but it is a consistent feature of the application of low pressure receiver systems to scraped surface evaporators that the matching compressor selected as standard by the evaporator manufacturer always turns out to be too large by a factor of about two. This is presumably a reflection of the inefficiency of the conventional natural circulation arrangement for scraped surface evaporators. It appears that low pressure receiver systems keep the evaporator surface well wetted but that the small amount of overfeed allows correct operation of the evaporating pressure control valves in conjection with minimum pressure drop in the suction line. There is no danger of flooding the compressor if the system is correctly charged. Much the simplest form of scraped surface evaporator system using low pressure receiver is the single circuit with one low pressure receiver, one scraped surface evaporator and one condenser. However ways have been found to operate several scraped surface evaporators in parallel from a single low pressure receiver when this is deemed necessary for economic reasons. The expansion devices to the scraped surface evaporators are heated bulb expansion valves of the type normally used as level controls. If correctly sited in the outlet pipe from the scraped surface evaporator, the heated bulb can be arranged to sense “wetness” in the refrigerant vapour flow from the evaporator. The degree of wetness can be controlled by adjusting the voltage on the bulb heater by means of a variable transformer. This system has been successfully operated on several sites using both ammonia and R22.

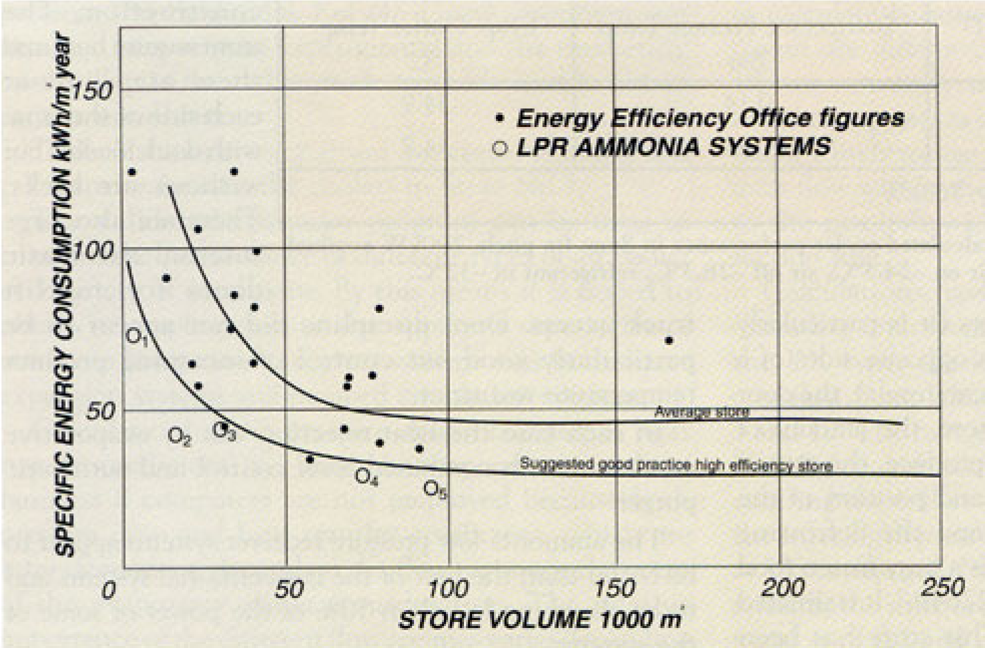

Figure 4 : Specific energy consumption of low temperature cold stores

Cold Stores

The most widespread and perhaps the most successful application of the ammonia low pressure receiver system has been to cold stores. Apart from the simplicity of the pumpless two pipe system which the low pressure receiver allows, it has been discovered that reversed cycle defrosting cold storage installations using low pressure receivers are generally more efficient than pump circulated overfeed systems. This is illustrated by **Figure 4** which is derived from the British Energy Efficiency Office "Best Practice Guide 37 to the Cold Storage Sector", Ref /3/. It shows specific energy consumption graphed against cold storage volume. The black circles in Figure 4 are the result of a postal questionnaire issued through the Cold Storage and Distribution Federation to its memebrs in 1992. Most of the stores were used totally or primarily for frozen food. The white circles, which have been added to the Energy Efficiency Office figures, represent cold stores using the ammonia low pressure receiver system. In the case of the low pressure receiver systems the power consumpton figures included the power of air circulating fans for air coolers and condenser fans, water pumps and heaters.

The Energy Efficiency Office figures show great variation in specific energy consumption between cold stores of similar sizes. Many of the stores appear to be operating inefficiently. The energy consumption of a cold store is affected by many factors including the effectiveness of the insulation, the position and sealing of the doors (it is particularly undesirable to have doors in use on opposite sides of a cold store because through draughts can result), the door discipline, the height of the cold store, the amount of heat to be removed from incoming produce, the rate of throughput of produce, the design and position of the air coolers within the cold store and the defrosting arrangements. The point marked 1 is a busy frozen food distribution centre of cubic capacity 2264m3. It is situated on the south coast of England. This store has been converted from a halocarbon refrigerant system to an ammonia low pressure receiver system. All the refrigerating equipment was replaced but the cold store structure, door arrangement and insulation were unchanged. The specific energy consumption of the store was reduced by about 50% after the conversion.

The point marked 2 is a well insulated frozen food cold store with air locks. The store is situated in north eastern Scotland. The coolers are situated on the rear wall as far as possible from the doors. The compressors are economised screws of 127mm rotor diameter and the economiser is a common vertical shell and coil vessel. The store has given consistently low specific power consumptions since it was installed in 1992.

The point marked 3 is an order picking cold store of about 30,000m3 capacity. The coolers are above the automatic doors. During the period of measurement the doors were not operating correctly and some of the strip curtains were also damaged. It is reasonable to assume that the specific energy consumption will decrease when the doors are mended.

The point marked 4 represents a production cold store associated with a frozen food factory. The air coolers are galvanised steel construction and are situated in the middle of the cold store. Door discipline is good but the product load is significant. During holiday weeks without significant production, power consumption fell by about 25%.

The point marked 5 is a very large frozen food distribution store in the south west of England. The air coolers again are in the centre of the store but, in this case, they are of stainless steel tube, aluminium fin construction. The store is quite busy and there are doors at each side of the store with dock loaders but without air locks. There are also large internal automatic doors for forklift truck access. Door discipline did not appear to be particularly good but control of incoming product temperature was strict.

In each case the heat rejection was by evaporative condenser with combined level control and automatic purger.

The ammonia low pressure receiver systems appear to be better than the best of the conventional systems and to be absorbing less than 50% of the power of some of the systems.

Practical and Theoretical Analysis

In order to provide a theoretical basis for the design of cold storage systems using ammonia low pressure receivers it was decided to measure the performance of the large coolers at store No 3. The low temperature cold storage installation at that site consisted of two identical ammonia low pressure receiver systems, each with three large coolers rated at 198kW and measuring 9000 x 2000 x 1000mm. Each cooler has six fans of 1000mm diameter, running at 580 rpm with air straighteners. It was noted that the power consumption of these large low speed fans used in conjunction with air straighteners was significantly less than higher speed fans which are more commonly used. One of the coolers was arranged so that temperatures and pressures could be measured at various points on the circuit. The system was then run with all three coolers in operation. The refrigerant pressure drops and temperature changes were measured under this condition, the refrigerant flow being estimated from the compressor performance. One of the coolers was then shut off and the measurement repeated, the refrigerant flow rate again being measured from the compressor performance. Two out of the three coolers were then shut off and the measurements taken with all the refrigerant going through the remaining cooler. Analysis of the results gave pressure drops and overall heat transfer coefficients at three different refrigerant flow rates and temperature differences.

The program for the theoretical calculation of refrigerant pressure drop and refrigerant internal heat transfer coefficient was developed, using the equations presented in Schlünder, Ref /4/. Good agreement was obtained between the experimental and the theoretical results despite the rather crude methods used to obtain experimental data.

Table 1 shows a comparison between measured and calculated values for the coolers in Store No 3.

Table 1 : Comparison of measured and calculated cooler performance in 8mm fin pitch, 200kW nominal duty air coolers in store 3. Air on -24.3°C, air off -28.3°C, refrigerant in -32°C.

kW/circuit		LMTD °C	Refrigerant Pressure Drop Bar	Evap. Outlet Temp. °C
Measured	3.9	7.2	0.14	−34.9
Calculated	3.9	7.0	0.144	−34.7

Number of circuits: 43 . . . actual duty: 168kW

The validated computer program can be used to compare the performance of different types of air cooler at a variety of conditions. By this means it is hoped to obtain an explanation of the apparent superiority of ammonia low pressure receiver systems over both direct expansion systems and pumped circulation systems.

Calculation of the pressure drop on internal heat transfer coefficients in a long tube evaporator is a tedious business if computers are not employed because both pressure drop and heat transfer coefficient, which are inter-dependent, depend on the flow regime at the point of the evaporator under consideration. The relative importance of the different flow regimes varies depending on the type of cooler being used.

Direct expansion coolers tend to start with a higher vapour fraction than is obtained at the inlet of a low pressure receiver type cooler but a significant part of the cooler surface is used to produce superheat under conditions of very poor internal heat transfer coefficient. As the signal for minimum stable superheat of the thermostatic expansion valve is of the order of 8–10k, it is normal to circuit the refrigerant in contraflow with the air to obtain maximum temperature difference at the coil exit.

In low pressure receiver coolers it is normal to run the refrigerant in parallel flow with the air to obtain maximum temperature difference and therefore increased boiling at the inlet end of the circuit.

Pump circulation coolers are normally arranged in crossflow which makes the performance more difficult to calculate because the refrigerant conditions in each circuit are different. Overfeed ratios of between 3 and 10 are normally used and the number of parallel refrigerant passes is increased to reduce pressure drop. A major disadvantage of high overfeed ratios is that the mass flow within the cooler is increased, thus giving rise to the possibility of serious pressure drop in the wet suction line.

Calculations have been carried out for four coolers operating under similar conditions. The coolers are of the DX type, the low pressure receiver type and the crossflow pump circulation type with overfeed ratios of 3 and 10. The refrigerant tubing is of 15mm bore and air is to be cooled from –22°C to –25°C with refrigerant entering the coil at –30°C. In the case of the DX cooler the entering dryness is taken as 0.35 and an exit superheat of 5°C is required. In the case of the low pressure receiver systems, the inlet conditions is taken as having a dryness fraction of 0.2 and exit dryness of 0.98. The calculated results are given in **Table 2**.

Table 2 : Theoretical comparison of similar sized coolers extracting 100kW from air

being cooled from -22°C to -25°C with refrigerant entering the 15mm i.d. evaporator circuits at -30°C.

	Cooler Capacity kW	Refrigerant Mass Flow kg/hour	Circuit Loading kW	Refrigerant Temperature at Cooler Exit °C	Required Length of Tubing m	Suction Line Loss °C/m in 65 mm tube
DX Cooler (Counter	100	396	3.3	–25.5	826	0.023
flow) LPR Cooler	100	334	4.0	–31.1	755	0.017
(Parallel flow) Pumped 3:1	100	815	3.6	–31.1	767	0.021
Pumped 10:1	100	2897	2.8	–31.9	715	0.176/0.025*

* in 80 mm bore tube

It can be seen that the pump circulation cooler with a 10 to 1 overfeed ratio gives a cooler which can be marginally smaller than the others but this advantage is completely overcome by excessive pressure drop in the wet suction line. In order to produce an acceptable pressure drop with this cooler design it is necessary to go up by at least one pipe size.

Taking all factors into consideration, it appears that the low pressure receiver system provides the smallest practical cooler design combined with the lowest suction line pressure drop. The higher suction line pressure drop obtained with the DX cooler was unexpected but is probably due to the higher total refrigerant mass flow required.

Although the low pressure receiver system appears to give marginally better performance than the direct expansion system or the pumped circulation system, the calculated improvements are not in themselves enough to explain the higher efficiencies which appear to be obtained from low pressure receiver systems in practice. Other reasons for the improved efficiency must be sought.

Effect of Defrost Method on System Efficiency

There are three types of defrost system in use for finned air coolers: the electric defrost system, the hot gas defrosting system and the reversed cycle defrosting system.

It is obvious that direct electrical defrosting of large air coolers is inefficient, undersirable and dangerous. The amount of electrical heat which is put into the air cooler to cause defrosting of the fins has to be paid for twice. Once when the heat is supplied and the second time when the heat is removed from the cold store air and the metal of the cooler. In addition, electric defrosting of coolers tends to produce understandably high surface temperatures at the heaters, which result in natural convection effects which circulate heat through the store. By contrast, the reversed cycle defrosting which can be associated with ammonia low pressure receiver systems puts heat into the cooler at the minimum practicable temperatures for defrosting. Defrosting can be terminated by internal pressure, thus eliminating unnecessary temperature rises within the cooler.

It is not so obvious why reverse cycle defrosting is more efficient that hot gas defrosting. During hot gas defrosting, discharge vapour from the compressors is fed into one or more coolers which are allowed to rise to condensing temperature. The fact that useful refrigerating effect is obtained when evaporating the refrigerant to produce the flow of hot vapour suggests that this system should be very efficient. In practice it does not seem to be so. There are two reasons; firstly, in order to get a rapid defrost it is normal to maintain an artificially high condensing pressure while defrosting is in progress. This increases running cost, especially during winter. Secondly, in order to operate the numerous valves which are required to seal the suction of the evaporator and to supply hot gas for defrosting purposes, it is normal to supply high pressure vapour to operate the valves. Such pilot operated valves can be of two types, those which are held open by a spring and closed by high pressure vapour and those which are held open by high pressure vapour and closed by a spring. The use of either type of valve is likely to lead to an escape of high pressure vapour from the high pressure side to low presure side. Such escapes are particularly serious in the type of system where the valve is normally held open by high pressure vapour because this is the mode of operation during normal refrigeration. On a large cold store there will be multiple valve stations, each one of which is a potential source of refrigerant leakage to the low pressure side. This is the most plausible reason for efficiency loss in hot gas defrosted cold stores.

In contrast, in the low pressure receiver system, the flow of refrigerant is controlled by a single four-ported ball valve. The ball valve has self-loading seats and is therefore is not prone to leakage. It is also possible for an experienced operative to confirm that the four-port valve is not leaking by studying the frost pattern on the valve.

Another possible source of inefficiency in pump circulated hot gas defrost coolers is oil contamination. The low pressure receiver type coolers were all operated with a polyalphaolefin oil whereas it is normal to use mineral oil in conventional refrigerated systems.

Conclusions

The low pressure receiver refrigerating system is a simple and effective system for use with scraped surface evaporators, plate type heat exchangers and finned air coolers for low temperature cold stores. It is important to optimise the design of plate type heat exchangers and finned air coolers for use with low pressure receivers. Programs have been developed to allow such design optimisation and are available for all common refrigerants and for all common secondary refrigerants. Site tests and continuing site experiences indicate that the programs are valid and produce useful results.

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