A UNIVERSAL FLOATING HEAD SYSTEM

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ABSTRACT

The power consumed by a direct expansion refrigeration system in providing a given refrigeration effect is reduced if the condensing temperature is lowered, if the liquid is sub-cooled, if the evaporator temperature is raised, if the amount of refrigerant circulating is reduced, if the circulating velocities are reduced to the minimum required for oil return and if vapour formation in the liquid lines is suppressed.
This paper describes an engineering solution to the problem of designing a large direct expansion system suitable for use in supermarkets which incorporates all the above desirable features and which consequently is predicted to save up to 30% of the energy used by existing systems to meet a given refrigeration load, depending on climatic region. The refrigerant inventory is reduced and consequently so are losses.
Furthermore this new system is readily adapted to use environmentally acceptable refrigerants such as carbon dioxide, hydrocarbons and ammonia.

INTRODUCTION

The Montreal Protocol calls for the phase out of chlorinated hydrocarbon refrigerants by 2020, and some countries have already taken steps to prohibit the use of ozone damaging refrigerants. The Kyoto Protocol calls for the phasing out of flourinated refrigerants on the grounds of their greenhouse effect and so efforts are being made to design refrigeration systems which will use environmentally friendly refrigerants, which will operate with lower refrigerant inventories and which will use less power and so reduce carbon dioxide emissions by utility companies.
Supermarket refrigeration systems represent a large part of the refrigeration sector. It is estimated that there are in excess of 100,000 supermarkets worldwide, with 30,000 in the USA and 44,000 in Europe, [Billiard, 2002]. In the UK the supermarket refrigeration sector consumes 5% of the national power output.
An international project aimed at identifying the potential for improvements in supermarket refrigeration systems (reduction in energy consumption, reduction in refrigerant losses) is currently in progress. It is being run by the International Energy Agency from the Oak Ridge National Laboratory [Annex 26,2002].
At a fundamental level the scope for improving the efficiency of a refrigeration cycle is confined to lowering the condensing temperature, raising the evaporator temperature, increasing sub-cooling of the liquid refrigerant and controlled suction superheat**.** Practical methods of improving the cycle efficiency in these ways are well known, [Benstead, 2003].
In reality, international efforts to improve the energy efficiency and environmental impact of supermarket refrigeration systems amount to limited application of liquid pressurisation pumps and liquid line sub-coolers, and design changes to display cabinets (power savings, some CO2 reduction) and the use of secondary loop systems (reduced refrigerant loss).
The issues surrounding refrigerant losses were recently summarised [Editorial, 2002].
This paper describes a refrigeration system which minimises the lift between the evaporator and the condenser by means of floating the head pressure down to the lowest level available, which is limited by the ambient temperature, by taking full advantage of the opportunity to reject heat from hot compressor discharge to the environment. The system philosophy lends itself to adaptation so that the design can be used for a cascade system using only environmentally acceptable refrigerants (eg carbon dioxide and ammonia). A two-stage system variant with a flash intercooler using a high efficiency refrigerant such as R410A represents the limit of achievable energy efficiency. A thermal energy storage variant in which ice is made off-peak to be used during periods of peak electricity demand to provide load shifting offers the lowest running cost option. The suggested systems do not require any new equipment, but rather use existing components assembled in such a way as to provide an engineering solution to the problem of reducing power consumption in many large scale refrigeration systems.

1 THE BASIC SYSTEM

When an idealised refrigeration system is designed to reject the maximum possible amount of heat from the compressor exhaust to the environment then it will also consume the least amount of power. Figure 1 shows the layout of such a system, which is the subject of an International Patent Application.

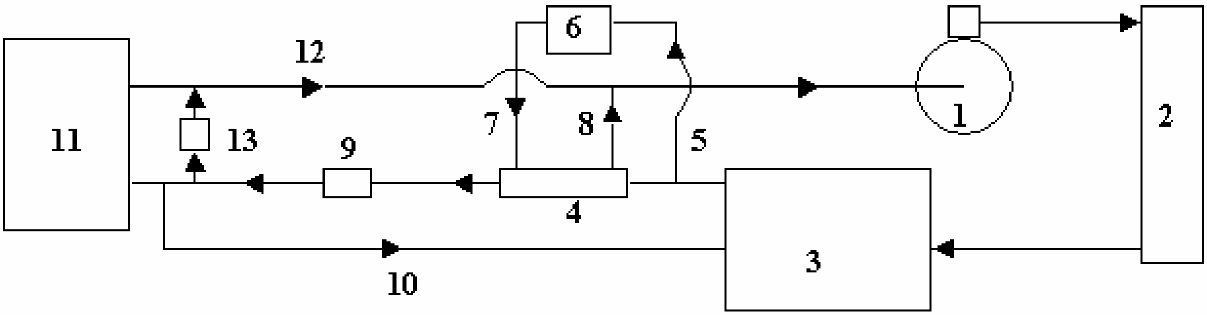

Figure 1. Single stage vapour compression floating head system

Compressor 1 produces hot vapour which passes to the air-cooled de-superheater/condenser 2, where de-superheating, condensation and possibly sub-cooling are allowed to occur depending on the ambient temperature. Thus the liquid entering the receiver 3 will be about 10C warmer than the ambient temperature and the effectiveness of 2 will not be limited by a fan speed controller. Liquid from 3 passing to expansion device and evaporator 11 passes through a heat exchanger 4 where it is further sub-cooled to any desired operating temperature below that existing in 3 in order to prevent flash back. This cooling is achieved by means of a bleed flow 5 passing through an expansion valve 6 producing cold vapour in line 7 which chills the main liquid flow in 4 before returning to the suction line via 8. The sub-cooled liquid leaving 4 (which may be below 0C when using carbon dioxide as a refrigerant) enters the recirculation loop (4,9,10,3) where flow is maintained by low pressure pump 9. Any cooling demand by evaporator 11 is met by drawing from the circulating flow of cold liquid. Spent cold vapour from 11 returns to the compressor via suction line 12. The liquid delivery line, the liquid re-circulation line and the suction line are housed in common insulation to minimise the risk of boiling in the liquid lines. In effect these lines form an elongated suction gas heat exchanger. Optional de-gassing valve 13 passes any vapour formed in the delivery line back to the suction line before the expansion valve in the main evaporator. An optional liquid pressure reducing valve (sometimes referred to as automatic expansion valve) fitted in the delivery line after 4 can be used to reduce line pressures to acceptable levels when using high pressure refrigerants and low levels of sub-cooling.
To exemplify the operation and performance of this system consider a low temperature pack designed to deliver 100kW of refrigeration at –30C using the high efficiency refrigerant R410A and operating in the UK. The number of hours that the ambient temperature Ta is at various selected levels in the central part of the British Isles is shown in Table 1:

Table 1: Temporal distribution of ambient UK temperature for design purposes

Ta(C)	0	5	10	15	20	25	33
Hours	789	2106	2863	2152	692	145	13

The condensing temperature Tc was taken to be 10C above these ambient levels. The floating head system is compared with a conventional system running on R404A with the setting for the fan speed controller on the air-cooled condenser assumed to be 30C, a normal scenario in most UK supermarket systems.
Standard cycle calculations were performed using a software package available as a free download from the Internet [Coolpack, 2003]. Similar operating conditions were specified for each system compared so that differences in performance were attributable entirely to system design and refrigerant. The compressors were assumed to have isentropic efficiencies of 70%, natural liquid sub-cooling was assumed to be 5C, evaporator superheat was taken as 6C, the pressure drop in the suction line was the equivalent of 2K, and in the liquid line was the equivalent of 0.8K.

When analysing the floating head system the amount of heat rejected by the heat exchanger 2 in Figure 1 was assumed to be that associated with de-superheating the compressor exit vapour, condensation of this vapour and any subsequent sub-cooling available, assumed to be 5C. The amount of liquid refrigerant required to bleed to the heat exchanger 4 was calculated using a heat balance between the cooling created by the evaporation of the bleed flow and loss of enthalpy of the remaining liquid flow passing through exchanger 4, using physical properties of the refrigerant evaluated at appropriate temperatures. Latent heat was evaluated at the liquid inlet temperature and specific heat evaluated at the mean temperature of the liquid passing through 4. Referring to the data in Table 2 below, when the condensing temperature T_C is 10C then no further sub-cooling is required and the heat exchanger 2 will deliver M_T=0.480 kg/s of liquid at 10C directly to the expansion devices in 11 using W=17.3 kW of compressor power. When, for example the condensing temperature in 2 is 25C then ΔT_S=15 degrees of sub-cooling are needed to bring the liquid to its design condition of 10C. This will require the evaporation of M_X=0.065 kg/s of liquid in unit 4 to remove Q_T=13 kW of heat from the 0.480 kg/s destined for the main evaporator. The total mass flow now being handled by the compressor is M_T= 0.545 kg/s and the compressor power has increased to W=30.2 kW, which is still less than the power needed if the condensing temperature was set at 30C as in a conventional system.

Table 2: Analysis of sub-cooler effect for an HT system as shown in Figure 1, providing 100kW of cooling at –15C

T_C C	ΔT_S K	T_M C	C_{PL} kJ/kg.K	LH kJ/kg	M_X kg/s	M_T kg/s	W kW	H hrs	Q_T kW
10	0	10	1.64	212	0	0.480	17.3	789	0
15	5	12.5	1.67	206	0.019	0.499	21.3	2106	4
20	10	15	1.70	199	0.041	0.521	25.6	2863	8
25	15	17.5	1.73	191	0.065	0.545	30.2	2152	13
30	20	20	1.76	182	0.093	0.573	35.2	692	19
35	25	22.5	1.80	173	0.125	0.605	40.8	145	26
43	33	26.5	1.87	156	0.190	0.670	51.5	13	39

In terms of the overall cycle, exchanger 4 is ideally energy-neutral since the extra compressor work required to produce the cooling bleed flow is recovered by the enhanced efficiency of the evaporation process created by the extra sub-cooling. The sub-cooler 4 can be used to fix the supply temperature of the liquid at the main evaporator to a very constant level thus reducing the required dynamic range of the expansion device and eliminating the need for expensive computer controlled motorised expansion valves. The sub-cooler also ensures that no flash back of vapour to the liquid receiver occurs should there be a fall in temperature in this unit caused by a drop in ambient temperature. In the example here it was assumed that operating conditions required liquid to be fed to the main expansion valves at a steady rate (0.48 kg/s) and a steady temperature of 10C, ensuring enough pressure to operate existing valves in the event of an assumed retrofit. When using carbon dioxide as a refrigerant the sub-cooler can be set to operate at 0C or below, particularly if the evaporator is close-coupled and the liquid supply lines are short as clearly shown in Figure 2 below. The benefits of this system increase as the ambient temperature falls and more heat can be rejected via exchanger 2. For a system designed to produce liquid at 10C then the sub-cooler 4 can be disabled when the ambient temperature falls below 0C.

Taken over a whole year of operation in the UK and using the data in Table 1 the floating head system with Te=-30C is predicted to use 23% less power than the conventional system. For a high temperature pack delivering 100kW of cooling at –15C a floating head system running on R410A (Table 2) would use 26% less energy than a conventional system running on R404A and with the condensing temperature again assumed to be limited to 30C.

2 TWO STAGE SYSTEM

As the ambient temperature rises the compressor must work harder to meet the cooling demand and it becomes more economic to use a two-stage system. The design shown in Figure 1 can be modified to meet this requirement as shown in Figure 3 below.

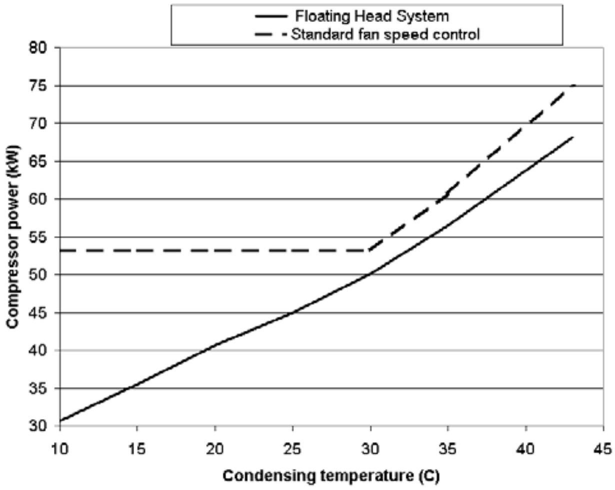

Figure 2. Comparison of the predicted power consumption of a 100kW system with an evaporating temperature of –30C. The full line is the performance of the floating head system described by Figure 1 running on R410A. The dashed line shows the performance of a conventional system running on R404A with the fan speed controller set to fix the lowest condensing temperature at 30C. Note that the pressures generated in a conventional system running on R410A would be unacceptably high.

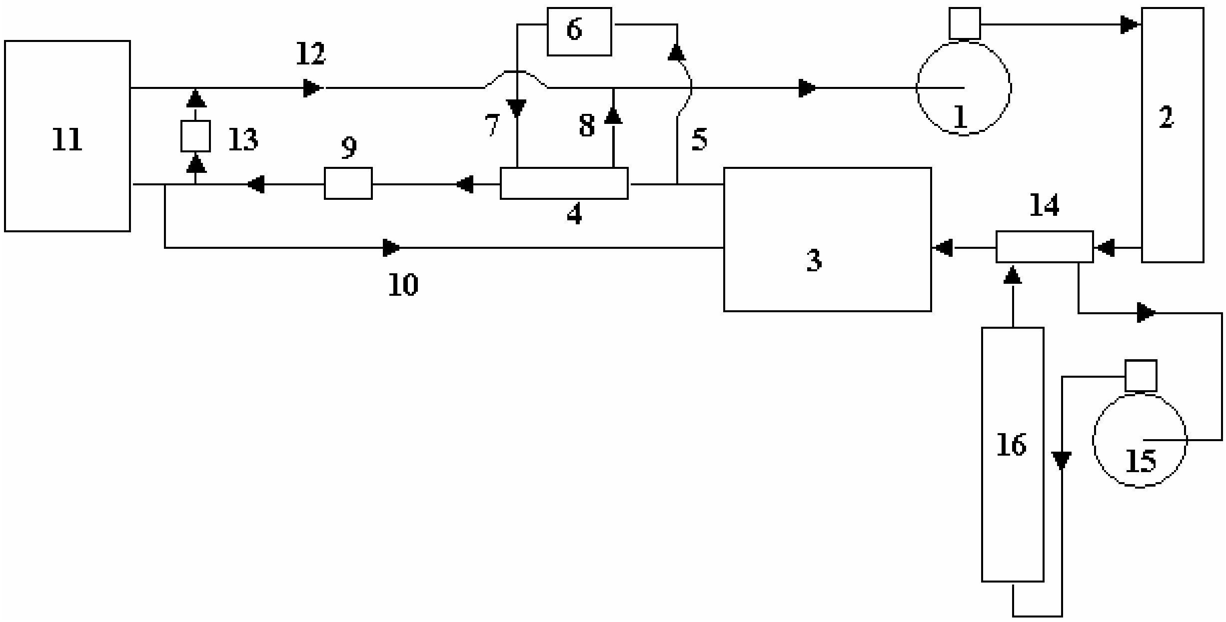

Figure 3. A universal floating head system comprising two stages with the second stage being used only when ambient temperatures are high enough to justify a seamless change from single to two stage operation..

An indirect heat exchanger 14 is used to link the first stage with a second stage compressor 15 and condenser 16. As the ambient temperature rises then at some point determined by various factors it will become economic to switch on the second stage and transfer some of the cooling load to this stage using the evaporator which forms part of heat exchanger 14. Heat exchanger 14 now becomes the principle means of sub-cooling liquid in the low stage circuit and sub-cooler 4 is set to prevent flashback. To exemplify this point consider the predicted variation in compressor power with ambient temperature shown in Figure 4 for the basic single stage system shown in Figure 1 and for the two stage system shown in Figure 3, both systems providing 100kW of cooling at –30C.

Table 3: Analysis of the two stage system shown in Figure 3 providing 100 kW of cooling at Te = -30C

T_C (C)	W_1 (kW)	ΔT_S (K)	Q_S (kW)	W_2 (kW)	W_T (kW)
15	34.2	5	3.7	0.3	34.5
20	37.4	10	7.5	0.8	38.2
25	40.6	15	11.5	1.6	42.2
30	43.6	20	15.4	2.7	46.3
35	46.7	25	19.6	4.1	50.8
40	49.7	30	23.8	6.0	55.7
43	51.4	33	26.2	7.4	58.8

As an example of how the performance of the floating head system was predicted consider the condition of a condensing temperature of 30C in Table 3 above. The first stage, operating between T_E = -30C and T_C = 30C requires compressor power W_1 = 43.6 kW. To lower the liquid temperature from 30C to the design value of 10C requires that the second stage removes Q_S=15.4 kW of heat in exchanger 14 by sub-cooling the liquid line in the low stage by ΔT_S=25K. This second stage operates between an evaporator temperature of 5C in 14 and a condensing temperature of 30C in heat exchanger 16. This requires compressor work of W_2 = 4.1 kW, giving the total work for the complete system to be W_T = 50.8 kW.

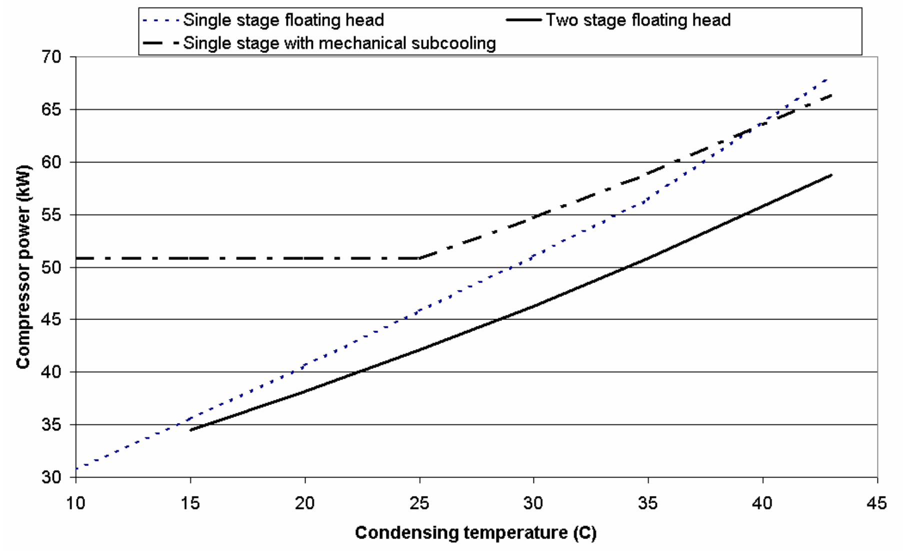

Figure 4. Comparison of power consumptions of three systems each providing 100kW of cooling at –30C, indicating the temperature at which a single stage floating head system should begin to operate as a two stage system. The best conventional system (a single stage unit with optional mechanical sub-cooling) is assumed to be set to produce liquid in the delivery line at 25C.

At an ambient temperature below about 5C (T_C~15C) it is probably not economic to use a second stage (see Figure 4), but above this temperature it becomes increasingly attractive to utilise a second stage. The best conventional system for use at high ambient temperatures, a single stage unit with a mechanical sub-cooler (a 5C temperature difference assumed in the connecting heat exchanger 14) running on R410A in both stages and with the control on condensing temperature set to 25C, is clearly outperformed by the floating head systems. The difference between this conventional system and the one shown in Figure 3 is simply that of maximum heat rejection in heat exchangers 2 and 16 which minimises the load on the second stage, coupled with the capability to continue heat rejection to the lowest of ambient temperatures. The annual energy consumption of the system shown in Figure 3 would be 30% less than a conventional system using the annual UK ambient temperature data in Table 1. The simple single stage system shown in Figure 1 would save 20% of the energy used by the conventional system. Operation in cold climates or during the generally colder night period would bring clear benefits not available to conventional systems. With this in mind a system which would be cheaper to run than any other system, but not the most economical in energy consumption, incorporates an ice generator in the second stage. In this design the second stage produces slurry ice at off-peak rates during the night and this ice is then used in the heat exchanger 14 in Figure 3 to chill the first stage refrigerant down to the lowest desirable temperature consistent with the design of the main expansion valves.

3 NEW SYSTEM USED WITH CARBON DIOXIDE AND AMMONIA

There is renewed interest in the use of natural refrigerants to replace existing HCFC and HFC compounds and also in the use of cascade systems (Taylor, 2003). The system shown in Figure 3 can be used to with carbon dioxide in the first or low stage and ammonia in the second or high stage. A major drawback of carbon dioxide as a refrigerant is the high pressure of the refrigerant even at normal ambient temperatures which precludes its use in supermarkets on safety grounds. However using the system described in this paper the pressure in the low stage can be kept at levels similar to those associated with refrigerants such as R22 by operating the heat exchanger 14 in Figure 3 and/or the sub-cooler 4 in Figures 1 and 3 to cool the liquid in the distribution line down to 0C or below. Figure 5 shows the predicted performance of a cascade system with CO_2 and NH_3 in the low and high stages of the circuit shown in Figure 3, with the heat exchanger 14 operating to keep the liquid CO_2 at about 0C. The system is designed to provide 100kW of cooling at –35C. Figure 5 also shows the predicted performance of a two stage system using R22 in conjunction with a liquid pressure amplification pump installed in the liquid delivery line of the low stage. This may be considered to be one of the most efficient systems currently available. Using the following temporal temperature distributions, [ASHRAE,2001] for three representative climatic zones in the USA (Table 4) the predicted annual energy savings available from the CO_2/NH_3 system compared with the R22 system are Cleveland 28%, St Louis 24% and Dallas 16%.

Table 4: Temperature distributions in hours per year for three US cities.

Ambient temperature (F)	0	10	20	30	40	50	60	70	80	90	100
Hours per year, Cleveland	76	219	504	1235	1559	1360	1490	1447	708	162	0
Hours per year, St. Louis	0	156	305	1011	1555	1156	1233	1673	1092	541	38
Hours per year, Dallas	0	0	29	346	887	1230	1587	1652	1904	888	237

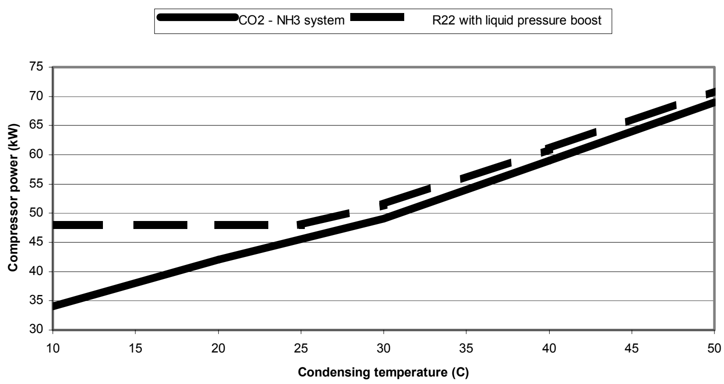

Figure 5. Power consumption of a CO_2/NH_3 cascade system delivering 100kW of cooling at –35C as a function of condensing temperature compared with an R22 two stage system with pressurisation of the liquid delivery line in the low stage and with head pressure control maintaining condensing temperature at 25C and mechanical sub-cooling via a second stage compressor being used above condensing temperatures of 25C..

4 DISCUSSION

The refrigeration systems described in this paper have the capability to float head pressure down as the ambient temperature falls and to maintain the liquid supply to the expansion device at a constant flow rate and constant temperature without vapour locking. Consequently the cycle efficiency is improved in almost direct proportion to the reduction in head pressure. For a system running on R22 with an evaporation temperature of –30C and set to maintain a head pressure at a condensing temperature of 40C the effect of floating down the head pressure is typified by Figure 6.

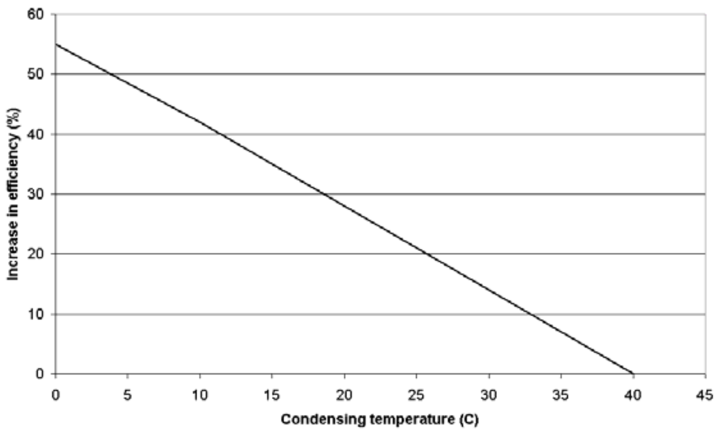

Figure 6. Effect of condensing temperature on cycle efficiency for a system with T_E=-30C running on R22

Low operating pressures also mean that high pressure high efficiency refrigerants such as R410A or carbon dioxide can be used, at least in the low stage. These further increase cycle efficiency by virtue of their improved heat transfer characteristics and smaller heat exchangers can be used [Hellmann et al, 1996]. For example it is possible to use higher evaporator temperatures with carbon dioxide. Since the low stage will be running at constant (low) head pressure there will be fewer start/stop events and lower flow pressure losses. There will be no need for motorised valves and computer controllers since the dynamic range of liquid flows will be minimal. Maintenance costs will be reduced because of reduced stress on components. Suction lines can be optimised for oil return and generally pipe sizes can be reduced.

When the second or high stage is indirectly connected to the first stage an optimal refrigerant can be chosen for each stage. Since the second stage pipework will be outside the store area then ammonia can be used.

There will be fewer compressors needed in the low stage circuit and refrigerant inventory will be lower than in a conventional system, thereby reducing losses and the seriousness of total losses.

5 CONCLUSIONS

A modified vapour compression refrigeration system design has been described and some performance characteristics have been predicted using cycle analysis software. An engineering solution to the problem of providing floating head capability has been suggested whereby maximum heat rejection to the environment from the low stage is allowed for all ambient temperatures and constant cold liquid feed to the expansion devices is provided at any selected low temperature. Flashback to the condenser is prevented by means of a sub-cooler driven by the low stage compressor at near-zero energy cost. Vapour formation in the liquid delivery line is inhibited by maintaining a circulating flow in a cold environment provided by the suction line. In the event of vapour formation in the low temperature liquid line an optional de-gassing valve vents the vapour to the suction line thus lowering the temperature in the liquid line. At low ambient temperatures the system operates as a single stage device, but as ambient temperature rises a point is reached where it becomes desirable to perform liquid sub-cooling by a second stage for maximum overall energy efficiency. For a system in which the two stages are connected indirectly by a heat exchanger the energy savings are predicted to be between 16% and 30% compared with the best existing systems in a range of climatic zones. The colder the climate the better the savings.

The system is ideally suited to take advantage of high pressure refrigerants, since the low stage pressure can easily be kept within acceptable limits by means of the sub-cooler, augmented if necessary by a liquid pressure reducing valve. Predictions for the performance of a low temperature carbon dioxide-ammonia system show that such a system will reduce the energy consumption when compared with one of the best available conventional systems.
If the two stages are directly linked by an open intercooler (flash chamber) the savings are significantly increased (by about 5%) since the inefficiency of the indirect heat exchanger is eliminated. Such a system precludes the use of different refrigerants in each stage. If the two stages are linked via an intermediate ice storage system which can be charged at off-peak rates then annual cost savings can be higher than 30% depending on local tariffs for electrical power [Davies and Lowes, 2002].

6 REFERENCES

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RESUME: La puissance consommee par un systeme de refrigeration par expension afin de produire un effet de refroidissement est reduit si la temperature de condesation est diminuee a l'extreme, si la temperature de l'evaporationest elevee, si la quantite de refrigerant qui circule est diminuee, si les vitesses de circulation sont diminuees au minimum necessaire pour le retour d'huile et si la formation de vapeur est supprimee.
Ce document decrit une solution technique au probleme de conception systeme d'expension directe adaptee aux besoins des super marches qui integrerait toutes les caracteristiques decrites precedemment et qui par consequent devrait permettre d'economiser jusqu'a 30% de l'energie utilisee dans les systemes existants pour une charge de refroidissement donne selon les regions climatiques. De plus ce nouveau systeme est pret a utiliser des refrigerants respectant l'environnement tels que le dioxide de carbonne, hydrocarbure et ammoniac.