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MODELLING OF AN INTEGRATED SUPERMARKET REFRIGERATION AND HEATING SYSTEM USING NATURAL REFRIGERANTS

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ABSTRACT

This paper describes the development of a computer simulation model of an integrated supermarket refrigeration and heating system using natural refrigerants (namely ammonia and hydrocarbons). Initial results of the model's performance are presented, showing the influence of some of the operating parameters on system thermal efficiency. This will be validated using data collected from monitoring of a similar plant. The model and monitoring of a novel refrigeration plant are part of the UK's contribution to IEA Annex 22 work program - Compression Systems with Natural Working Fluids (NWFs).

INTRODUCTION

The growing acceptance of TEWI (total equivalent warming impact) as a measure of a systems contribution to global warming, puts greater emphasis on the energy efficiency associated with the system, rather than the direct contribution from the refrigerant. Natural refrigerants, such as ammonia, hydrocarbons and carbon dioxide, are widely seen as offering environmental benefits for retail refrigeration. The inherently high efficiency, and negligible direct global warming effect of natural refrigerants, in comparison to the widely used HFCs, will have a positive impact on the TEWI rating of a refrigeration system. Also, many see tighter controls on refrigerant emissions and potential restrictions in supply of HFCs as an impetus to move towards natural refrigerants. However, many have problems associated with their use - ammonia for example is toxic, whilst hydrocarbons are highly flammable.

Conventional supermarket refrigeration systems use direct cooling, with refrigerant circulated around the building to individual display case air coolers. The refrigerant leakage rate for this type of system is significantly greater than indirect systems, adversely effecting the TEWI rating for refrigerants with a global warming effect. In comparison to direct systems, the charge of primary refrigerant for an indirect system can be reduced by as much as 80-90%. Furthermore, the risk of leakage is greatly reduced for an indirect system, since the primary refrigerant is contained in a factory assembled and leak tested unit. Ammonia and hydrocarbons are only likely to be acceptable as refrigerants for indirect systems, where a limited charge can be contained safely.

Recent changes to the British standards has widened the scope for 'natural' refrigerant usage; these are favoured more positively in Europe than in the USA. The safe and effective use of ammonia, coupled with a secondary system, represents a very sound solution with the potential for excellent energy efficiency in supermarkets. Until recently ammonia has not been used widely in supermarket refrigeration plant. However, recent developments, and considerable investment, have seen a number of full scale field trials by the majority of the main retailers, and the demand is steadily increasing in the UK.

SUPERMARKET REFRIGERATION MODEL

A common approach to HVAC systems has been adapted to meet the chilling, freezing, store heating and cooling demand of a typical supermarket. A single phase secondary fluid is pumped around the shop floor ring mains acting as a heat sink and heat source for freezer, chiller and heat pump compressor packs. See figure 1. Any surplus heat from the secondary fluid is rejected to the

outdoor environment with a primary ammonia chiller. In the interest of maintaining a system restricted to 'natural' refrigerants the refrigerated cabinet and heat pump compressor packs use hydrocarbon refrigerants. The number of display cases per compressor pack will be limited by the low charge restriction imposed by the new BS4434:1995 standard (this may be as low as 1.5 kg for a sealed direct expansion system for supermarket use in some cases).

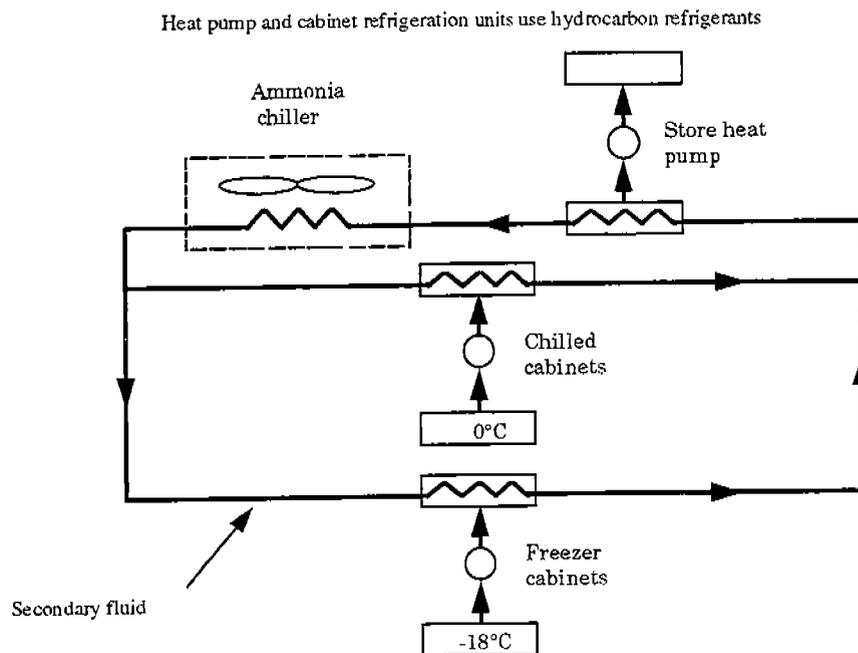Heat pump and cabinet refrigeration units use hydrocarbon refrigerants

Figure 1 Schematic of integrated refrigeration and store heating system for a supermarket

The benefit of this system is that a heat pump can be used to upgrade the heat rejected into the secondary fluid to provide heating for the store, reducing the refrigeration load of the ammonia chillers. Also the pumping requirements of the coolant fluid is reduced by circulating the coolant fluid at temperatures greater than the freezer units. The coolant fluid has to have sufficiently low viscosity at the required operating conditions (viscosity increases dramatically at the operating temperatures of freezers) not to need a huge input of pump power to drive it around the system.

The choice of secondary fluid depends on the minimum temperature expected at the outlet of the ammonia chiller evaporators. Below 5°C propylene glycol is used with the minimum concentration selected to avoid the adverse effects on physical properties viscosity and specific heat. Above 5°C water is chosen.

The overall energy efficiency of the system includes the pumping requirement of the secondary fluid. An approximate figure for the model pumping power is derived by multiplying the monitored store design figure, by scalar values of the variables which influence the pressure drop of the secondary fluid.

Refrigeration units and heat pumps

The performance of the ammonia chiller, refrigeration units and heat pumps are predicted using manufacturer's data. Performance data is not available for the high efficiency scroll compressors using hydrocarbons. This was generated from isentropic and volumetric efficiencies obtained from R22 data. For comparison purposes, the range of refrigerants that can be selected for use in the scroll compressors are R22, R404a (the refrigerant used in the monitored refrigeration plant), propane and CARE 50 (a HC blend).

The model can cope with up to three temperature levels in the display cases. A different refrigerant can be used for the refrigeration units if one refrigerant is more energy efficient than another at a particular temperature lift. There is also the option to cool the high temperature cases directly with the secondary fluid. If the temperature lift between the evaporator and condenser of the high temperature (HT) refrigeration unit is too small to perform well, then the secondary fluid should provide the cooling directly, saving energy from not using HT compressor sets.

Condenser and evaporator coolant heat exchangers

In most cases there will be a point of minimum temperature difference (pinch), between the secondary fluid and the refrigerant temperature profiles in the evaporators and condensers. The pinch temperature of the ammonia evaporator, ΔT_{PTE}, occurs between the secondary fluid outlet temperature, and the ammonia inlet temperature. The display case refrigeration condenser pinch temperature, ΔT_{PTC}, occurs between the secondary fluid outlet temperature, and the refrigerant condensing temperature. To simulate the heat transfer process correctly the refrigerant and coolant temperature profiles must never cross: that is ΔT_{PTC} and ΔT_{PTE} are always positive. The method uses ΔT_{PTC}, ΔT_{PTE}, and the temperature rise setting across the HC refrigeration condenser to define the evaporating and condensing temperatures from the secondary fluid supply temperature.

The full range of input data is as follows:

1. Refrigerant performance data for the chillers and refrigeration units, and the secondary fluid thermophysical data
2. Secondary fluid outlet temperature from the ammonia chillers, velocity of fluid, and temperature rise across the Refrigeration condenser.
3. specifications for pinch point values, subcooling in the condensers, and superheating in the evaporators.
4. Refrigerant choice, duty, and evaporation temperatures of the refrigeration units
5. Condensing temperature of ammonia chiller

PRELIMINARY RESULTS

Examples of the effect on system efficiency of changes in secondary fluid operating temperature are shown in figures 2 and 3. These both show the system at its simplest, with one refrigeration temperature and no heat pump. Figure 2 shows a refrigeration temperature of -30°C (typical of frozen produce) whilst figure 3 shows -10°C (typical of chilled produce). In general the power drawn by the refrigeration units installed in the display cabinets (hydrocarbon scroll compressors) increases with increasing secondary fluid temperature (due to the increasing condensing temperature), whilst the power drawn by the ammonia chiller decreases with secondary fluid temperature (due to the increased evaporating temperature). Thus, there tends to be one distinct secondary fluid temperature which optimises system efficiency for a given set of chilled and frozen cabinet control temperatures, duties, and ambient temperature.

In figure 3, there is a discontinuity in the total power consumption at -10°C secondary fluid temperature, representing the secondary fluid chilling the HT cabinets directly. At a chiller condensing temperature of 45°C the overall power consumption is slightly in favour of using the secondary fluid to provide the cooling directly, whereas at 35°C it is the only solution, to avoid a significant energy penalty. Although not shown, one would also expect a discontinuity for the low temperature graph at -30°C secondary fluid temperature, with the chiller evaporating at approximately -35°C. Ignoring the pumping power of the secondary fluid at -30°C, the power consumption of the chiller (164kW @ -35°C evaporation, 35°C condensation) alone, is greater than the minimum overall power consumption using refrigeration units for the low temperature duty.

Note that there is a minimum temperature lift for satisfactory operation of both the chiller and refrigeration units. As a result, above approximately 15°C secondary fluid temperature, the

evaporating temperature of the chiller remains fixed at 10°C, whilst the chiller duty and power increase. Also, at less than -10°C and -5°C coolant temperatures, the condensing temperature of the LT and HT refrigeration units remains fixed at 0°C and 5°C respectively.

Most supermarkets will operate both low temperature (LT) and high temperature (HT) cabinets. The effect on system efficiency with change in secondary fluid temperature and split in duties between LT and HT refrigeration units operating at temperature levels -30°C and -10°C is shown in figure 4 and 5. Figure 4 shows the power consumption for a typical supermarket duty ratio of 1 LT : 2 HT. Using the secondary fluid directly for the HT refrigeration is slightly more efficient than the best case using HT scroll compressor units, when the chiller condenses at 35°C. However, condensing at 45°C, the total power consumption is less using HT scroll compressor units.

Figure 5 shows the power of the secondary fluid pump, chiller and refrigeration units for the most efficient secondary fluid temperature, with the chiller condensing at 35°C, for different LT:HT duty ratios. For both the 1:5 and 1:2 LT:HT duty ratios, the best performance is obtained with direct cooling of the HT cabinets using the secondary fluid - hence the relatively low power consumption from the hydrocarbon refrigeration units (low temperature only). At the 1:1 ratio, however, the optimum performance is achieved using the hydrocarbon refrigeration units for both the LT and HT cabinets - hence the larger power input to these units.

Instead of using the ammonia chiller to reject heat to ambient, it is possible to raise the condensing temperature of the hydrocarbon refrigeration units and reject heat directly or through a separate water circuit. Using a water circuit with a dry, forced air cooled heat exchanger (of a similar size to the ammonia condenser), the condensing temperature of the refrigeration units is approximately 42°C for the same air temperature profile (compared to the 35°C for the ammonia plant). In all cases, (even ignoring the extra pumping power of the water heat rejection loop) the raised condensing temperature means that the power consumption when just using the hydrocarbon refrigeration units is always worse than the best case when using the ammonia plant as well.

CONCLUSION

A computer simulation model of an integrated supermarket refrigeration and heating system has been designed to assess the performance of 'natural' refrigerants, ammonia and hydrocarbons against a conventional HFC system. The results indicate that the use of HT refrigeration units instead of secondary coolant directly chilling is advantageous at a high ambient temperature and high LT:HT duty ratio only. The advantage is likely to increase at a lower ambient temperature once the store heating requirements and the use of heat pumps are included. The model is still under development at present, and is in the process of being validated with data collected from the monitored plant. It is hoped that this approach will result in an environmentally sound solution to the problem of supermarket refrigeration at minimum energy cost.
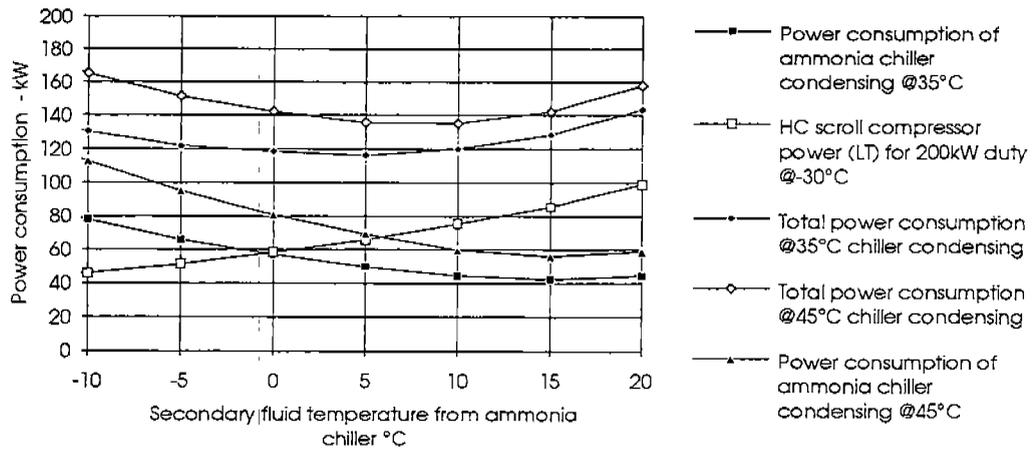
Figure 2 Influence on power consumption with change in secondary fluid temperature and ammonia chiller temperature for freezer cabinets
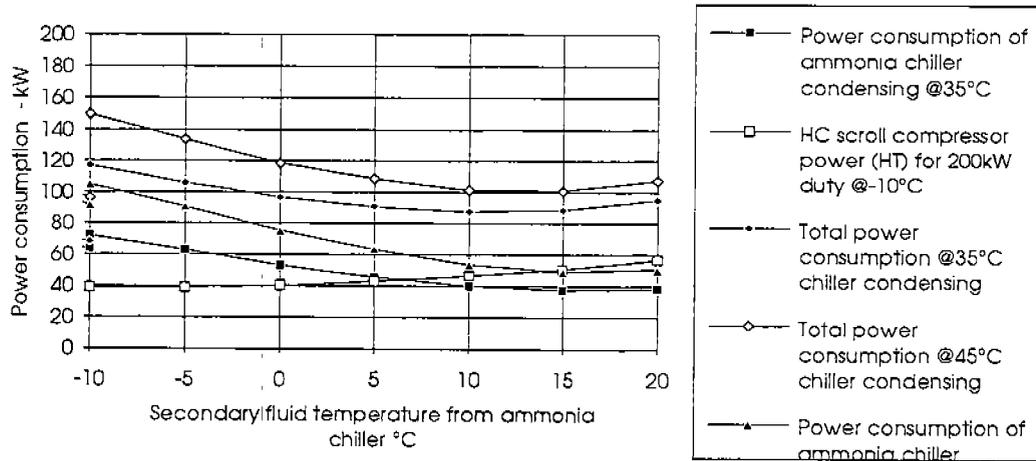

Figure 3 Influence on power consumption with change in secondary fluid temperature and ammonia chiller temperature for chiller cabinets
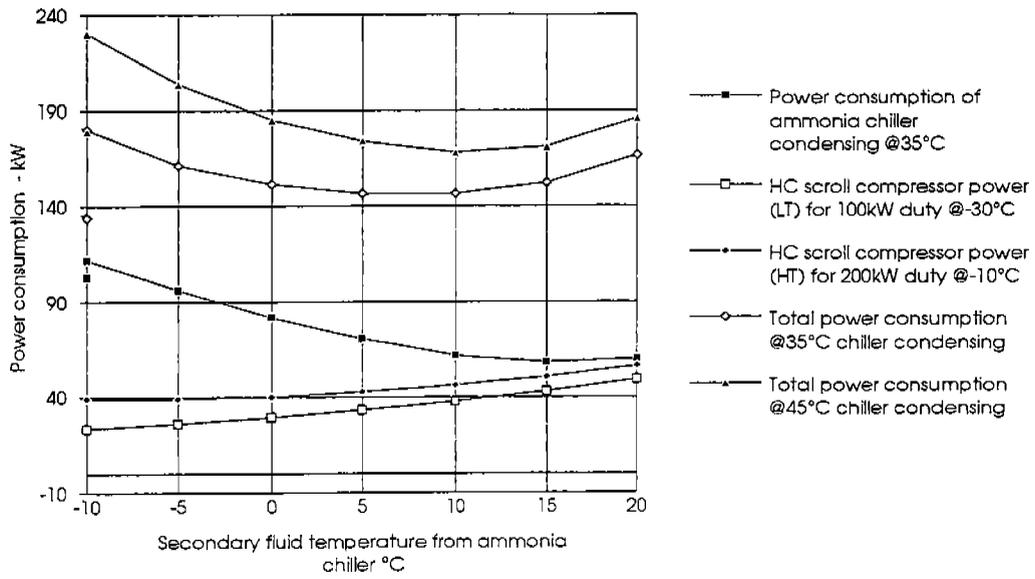
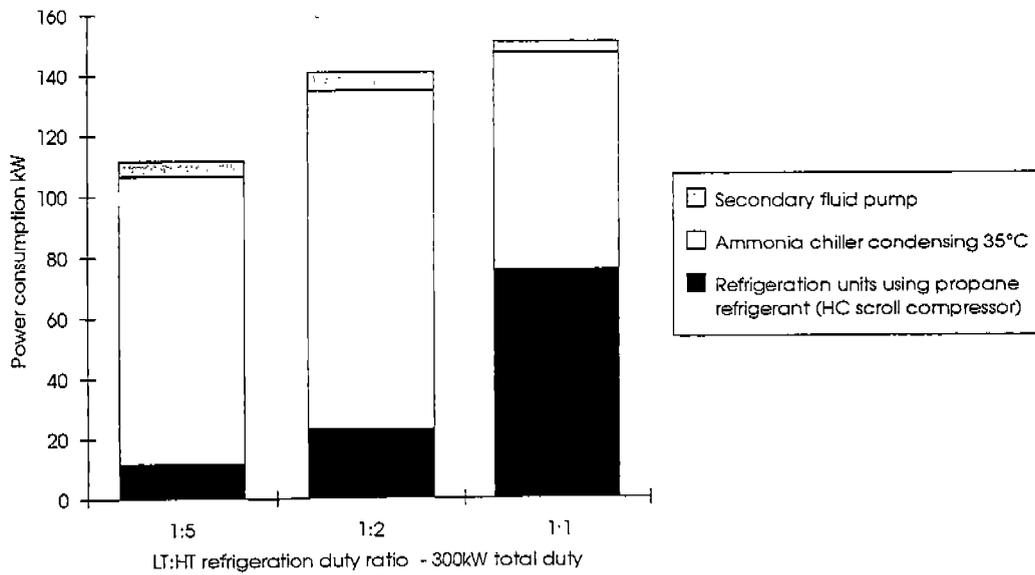Figure 4 System performance with change in secondary fluid temperature for LT:HT duty ratio = 1:2

Figure 5 Power consumption of secondary pump, chiller and refrigeration units for the most efficient operating condition for different LT:HT duty ratios