Purdue University
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International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

1998

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ABSTRACT

Centralized commercial refrigeration is becoming a sector where several technical options are competitive. These options are evaluated based on three major criterions: energy consumption, refrigerant emissions and initial cost. In supermarkets with machinery rooms, the competition is strong between previous centralized direct systems and new systems using heat transfer fluids. To achieve balanced comparisons between energy efficiency of these competitive techniques, measurements of direct expansion systems and of systems using monophase heat transfer fluids have been performed on site.

1. INTRODUCTION

Concern over the environmental impact of refrigeration, both from the point of view of refrigerant emissions and of energy consumption, has renewed interest in alternative refrigeration systems [1]. Two options are evaluated for centralized commercial refrigeration: previous direct systems and new indirect systems using heat transfer fluids [2, 3]. One of the main reason to promote this option is due to the dramatic limitation of the initial charge of refrigerant, consequently the quantities of fluid released to the atmosphere are lower. These two alternatives lead to different energy efficiencies compared by two methods. First, energy consumptions of both systems are calculated based on real efficiencies of pumps and compressors which are installed in different supermarkets. Second, measurements are performed in two cold rooms inside supermarket one with a usual direct expansion system, the other with a secondary loop system.

2. CONSUMPTION CALCULATIONS FOR THE TWO OPTIONS

A large French supermarket (sales area > 10,000 m^2) has been chosen as a reference for this study. Table 1 indicates power of compressor racks for the temperature range. Medium temperature cooling capacity represents around 90% of the whole cooling capacity. Usually, on the French market, it is closer to 75%.

Centralized direct expansion system

The cooling capacity is spread out over 12 compressor racks to limit consequences of major failure of the refrigerating system and to adapt the evaporating temperature to the required temperatures in display cases and cold rooms. The energy consumption is then improved.

All the racks of compressors share one single large air condenser.

Each rack is composed of 3 or 4 semi-hermetic reciprocating compressors. The refrigerant is R404A.

The calculations of the pressure drops and temperature variations take into account the real tube lengths and diameters. For the direct expansion system the pressure drops on the suction line vary from 1.5 to 2.5°C.

Table 1 - Compressor and Cooling Capacity Distribution.

Designation	Number of compressor racks	Cooling capacity (kW)	Compressor power (kW)
Medium Temperature -15°C < T < - 8°C			
Display cases	3	504,5	182,3
Cold rooms	5	307,7	86
Total medium temperature	8	812,2	268,3
Low Temperature < -38°C			
Display cases	2	71	57,4
Cold rooms	2	26,7	15,7
Total low temperature	4	97,8	73,2
Global cooling capacity	12	910	341,5

Secondary loop system

Data have been gathered from the contractors who installed indirect systems for medium and low temperature loop. Heat transfer fluids are respectively MPG (mono-propylene-glycol) for the medium temperature loop and Tyfoxit 1.2 for the low temperature loop. Both fluids represent a good compromise between cost and energy performances. The low temperature loop is a classical two-tubes circuit.

A new concept called "Single tube" is taken as a reference for the medium temperature secondary loop. The heat transfer fluid circulates in a single big tube made of monopropylene; each of the display cases or cold rooms is fed by a small pump which provides the coil with a given flow of cold fluid that is released at an increased temperature in the "single tube".

The circulation of the heat exchange fluid is realized by a primary pump on the main loop and small pumps for each coil.
Energy consumptions of pump and compressor are evaluated from real characteristics of available components. Secondary pumps available at acceptable costs show a very poor efficiency as it can be noticed in figure 1.1.

Pressure losses have been calculated for MPG and Tyfoxit in order to calculate the pumping power. Refrigerant for the primary refrigerating circuit is also R404A.

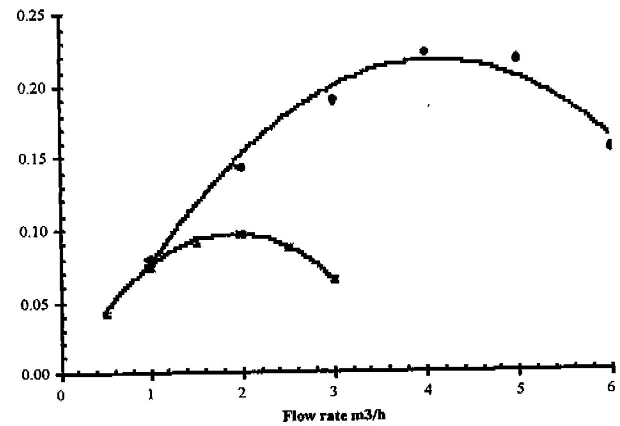

Figure 1.1: Efficiency of Secondary Pumps.

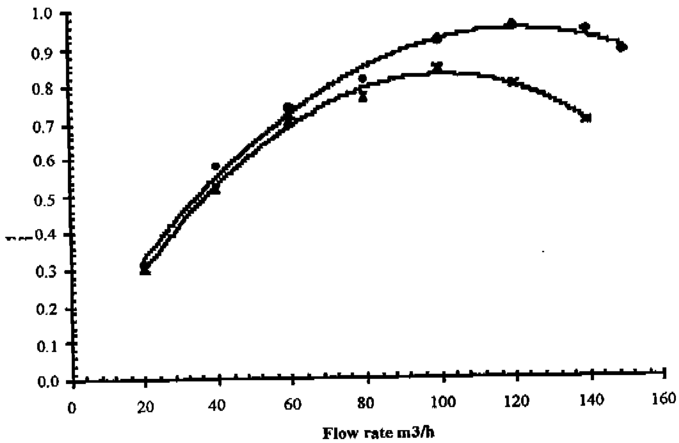

Figure 1.2: Efficiency of Primary Pump

Results of the comparison

The temperature of the air coil inside display cases are set in order to maintain the same temperature of the goods. It is claimed that the average temperature (calculated with the in/out temperatures of the coil) of the heat transfer fluid is 1°C above the usual evaporating temperature of the refrigerant.

Operating parameters and the results of comparison between the direct expansion system and the system using heat transfer fluid are presented table 2.

Table 2 - Comparison of the Two Different Systems.

Designation	Direct expansion system		Indirect system	
	Medium temperature	Low temperature	Medium temperature	Low temperature
Cooling capacity	812.1 kW	97.8 kW	1013.9 kW	114.1 kW
Tevaporating	-10°C (from 0 to -12°C)	-35°C	-16°C	-42°C
Tcondensing	+40°C	+40°C	+40°C	+5°C
Compressor capacity	268.3 kW	73.2 kW	376.6 kW	39.8 kW
Pump capacity			33.73 kW	7.29 kW
Operation time/day	18 h / 24 h	18 h / 24 h	18 h / 24 h	18 h / 24 h
Annual consumption	1766.7 MWh	480.9 MWh	2697 MWh	309.8 MWh
Charge of fluid	1735 kg		765 kg	

The efficiency of low temperature racks is much higher for the secondary loop system due to the fact that the refrigerant is condensed at -5°C using heat exchange with the medium temperature heat transfer fluid. But this heat which is removed by the medium temperature loop implies a higher power for the medium temperature racks.

The global consumption of this system compared to the previous direct system increases by more than 33%. Two main characteristics can explain this over consumption:

- the evaporating temperature for the heat transfer fluid alternative is lower by at least 4°C than the one of the direct expansion system;
- the capacity needed for the circulation of the heat transfer fluid is significant and represents around 12% of the added electric power absorbed. The efficiency of the small pumps used to feed each display case is particularly low, less than 20 % as shown in figure 1.1. However, even if the efficiency of these pumps was increased, the over consumption of the system working with heat transfer fluid would still represent 10 % of the compression power.
- The real gain of the indirect option is a reduction of the refrigerant charge by more than 55%.

3. MEASURES PERFORMED IN SUPERMARKETS

Temperature, pressure and energy consumption were registered over three weeks in two large supermarkets (S = 10,000 m^2) in December 1997. Both of the stores are located in North of France, so conditions of use of cold rooms and weather are identical.

Measures have been registered at various points of the cooling circuits: coils and refrigerating systems. One refrigerating system (IS) is indirect system working with heat transfer fluid, the other system (DE) is a direct expansion system.

It has been checked that comparison is based on appropriate conditions, in particular that ratio of heat exchange surfaces compared to the room volume do not exceed 5% (see Table 3). Heat exchange coefficients are higher for coils with heat transfer fluid but air speeds are 2/3 lower.

Table 3: Heat Exchange Coefficient and Heat Exchange Surface/ Cold Room Volume Ratio.

Cold room	Cold room volume (m^3)	Heat exchange surface per volume unit (m^2/m^3)	Heat exchange coefficient A ($W/m^2.K$)
Indirect system	1354	0.42	18.8
Direct system	888	0.446	11.2

Registered parameters are: outside temperature, blowing and intake temperatures of air coils, in/out temperatures of the heat transfer fluid or refrigerant, in/out temperatures of heat transfer fluid of the primary heat exchanger in the machinery room, intake pressure and temperature of compressors. Fluids are R22 and MPG as heat transfer fluid.

Figures 2.1 and 2.2 are synthetic presentations of temperatures permitting to compare operation between the indirect system and the direct expansion system. Outdoor air temperatures are identical. Major conclusions are indicated herebelow.

- Difference between blowing and intake temperatures is significant: 0.8 K for the direct expansion system and 4.6 K for the indirect system. Indeed, air flow shall be much lower so that air is cooled with a 4 K temperature glide at the heat transfer fluid side.
- In this study, saturating temperatures observed are - 10.9°C for the DE system and - 15,6°C for the ID system, that is a compression ratio increase of 20% for the same condensing temperature.
- At this time of the year, the condensing temperature is set at 30°C. Differences between saturating temperatures imply an increase of the energy consumption of 15% for the indirect system not taking into account power of pumps.
- The mean temperature of the cold room is higher than 3.7°C for the indirect system compare to the direct expansion system. In one case opposed to the other one, the operating temperature is kept.

It can be deduced that either the refrigerant capacity of the indirect system is not high enough or the temperature level is not appropriate. Two options exist: either increase the heat exchange surface or decrease the heat transfer fluid temperature. If temperature decreases while maintaining the same temperature differences, the saturated temperature at the compressor intake should be in the range of -19°C.

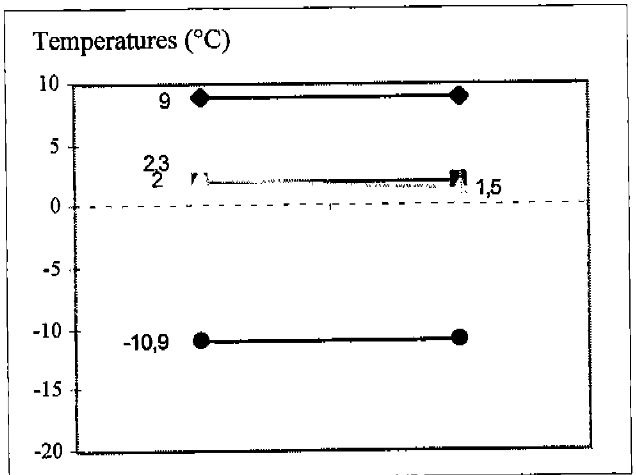

Figure 2.1: Key Temperatures of the Direct Expansion System

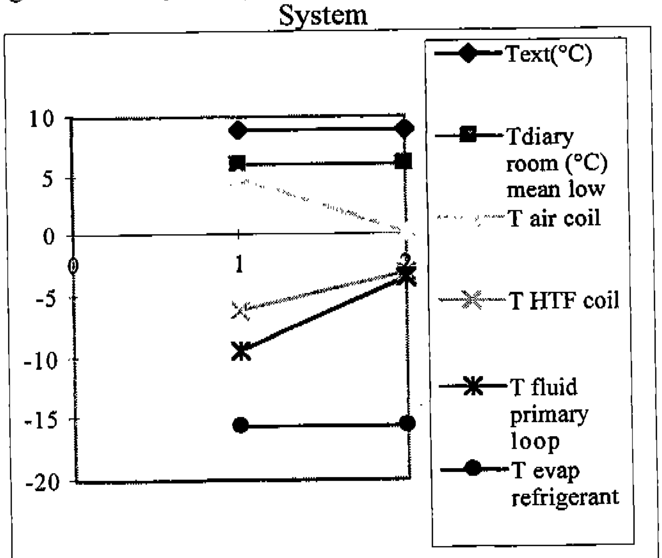

Figure 2.2: Key Temperatures of the Indirect System

• Defrosting water were recovered and weighted. On the referenced period, frost weight is higher in the cold room with direct expansion system (2.9 g/h.m^3) than with the indirect system (1.8 g/h.m^3). This is due mainly to the average room temperature: the higher the room temperature, the higher the moisture level. Defrosting lasts longer and temperature increases are more significant with the indirect system compared to the direct expansion system.

CONCLUSIONS

Based on data of existing systems, calculations show real increase of energy consumptions of indirect systems. This increase raises when products have to be kept at low temperature.

First on-site measurements confirm calculation predictions, that is saturated temperatures at the compressor intake significantly lower for indirect systems.

Competitive solutions are under evaluation in Europe so that balanced comparisons can be performed. Comparison criteria shall include maintenance, energy consumption, operation costs and impact of refrigerant emissions.

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