1998

Enhancement of Supermarket Freezers to Reduce Energy Consumption and Increase Refrigeration Capacity

M. Boiarski
InterCool Energy Corporation

O. Podchernyaev
CANMET

B. Yudin
InterCool Energy Corporation

D. Giguere
CANMET

ENHANCEMENT OF SUPERMARKET FREEZERS TO REDUCE ENERGY CONSUMPTION AND INCREASE REFRIGERATION CAPACITY

Mikhail Boiarski*, Oleg Podchernyaev**, Boris Yudin*, Daniel Giguère**,

(*) InterCool Energy Corporation, *450 Old Niskayuna Rd., Latham, NY 12110, USA*

(**) CANMET, Energy Diversification Research Lab.,
1615 Lionel-Boulet Blvd. Varennes, Quebec J3X 1S6, Canada

ABSTRACT

An auxiliary mixed refrigerant cycle for subcooling of a liquid refrigerant in the main cycle can provide up to 40% increase in the coefficient of performance of a conventional vapor compression cycle. This is twice as efficient compared to the conventional mechanical subcooling. In addition, an increase in the cooling capacity of up to 100% can translate into a reduction in the size of the whole system by up to 30%. This auxiliary precooling system can be built using mass produced components. The relatively small size of the unit makes it possible to use a packaged design compatible with mixed refrigerants. Experimental test data prove the proposed new system concept.

INTRODUCTION

Energy efficiency is a major characteristic that influences the market value of any refrigeration system. It also impacts the global environment which has become a focus of the international community. Refrigeration technology is one of the most highly developed industries with more than 150 years of its history. This makes it a real challenge to find new ways to further enhance the energy efficiency of modern refrigeration systems. For many applications even 10% increase in the coefficient of performance (COP) is considered significant. A preferred development would be to modify a system using generally available parts without major redesign of components.

One of the possible markets for new systems with improved energy efficiency is supermarket refrigeration systems. The 30,000 supermarkets in the United States consume 5% of all the electricity produced in the country /1/. The electricity bill for these stores equals 1% of their total sales - the same percentage as their net profit margin which makes them especially sensitive to refrigeration system energy efficiency. In this field one of the common methods to improve energy efficiency of a low-temperature refrigeration system is to subcool the liquid refrigerant leaving the condenser with a mechanical subcooler - an auxiliary refrigeration cycle /2/. This method is known to give power savings of 10% to 15 % /3,4/.

Our experience in building hermetic low-temperature systems working with mixed refrigerants (MR) /5/ and in designing CFC substitutes /6/ led to the development of a new mechanical-subcooling technology. It is based on a combination of both auxiliary cycle modifications and a new MR design /7/. We call that system a precooler to differentiate from mechanical subcooling using a pure refrigerant.

The goal of this paper is to compare the new precooler concept and conventional mechanical subcooling.

THERMODYNAMIC BACKGROUND

Schematics of the refrigeration system with the precooling cycle and the temperature (T) - enthalpy (h) diagram are presented in fig.1. A main cycle (MC) of this system operates in a temperature range from ambient T_A to T_R - refrigeration temperature. The main vapor-liquid cycle without precooling includes the following processes : 1 - 3' - throttling, 3' - 4 - evaporation, 4 - 5 - superheat in the suction line, 5 - 6 - compression and 6 - 1 - condensation. The system needs power P_{MC} in the compressor to provide refrigeration capacity Q_{RMC} in the evaporator.

It is well known that the greater the temperature ratio T_A / T_R the lower both the Q_{RMC} and the energy efficiency. To have a thermodynamically sound baseline for comparison, the refrigeration performance can be evaluated in terms of exergy efficiency which is equivalent to the Carnot efficiency (CEF). Using CEF makes it possible to compare systems performance at different T_R and T_A.

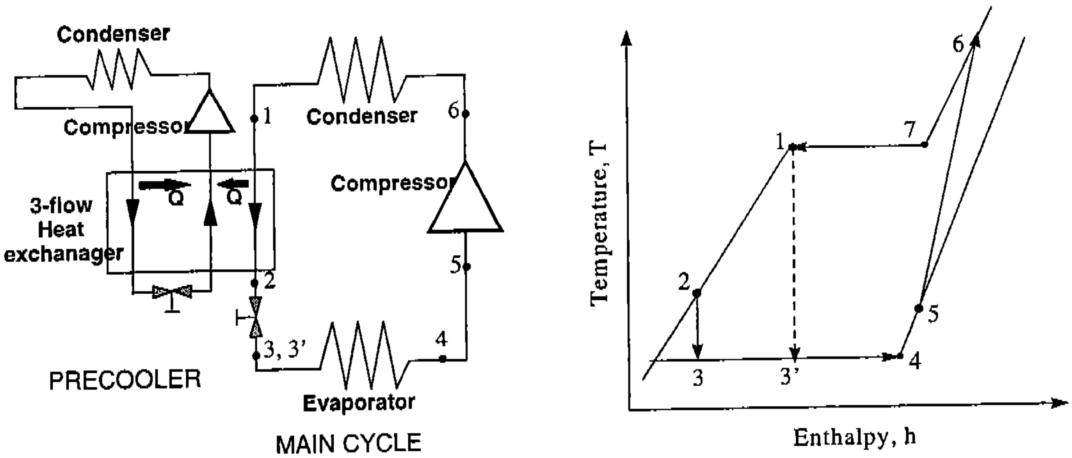

Fig.1. Refrigeration system with a precooling cycle

Lets consider first a thermodynamically idealized cycle with intrinsic exergy losses only. It means that a minimal temperature difference in the condenser and evaporator $DT_{min} = 0$; there is no parasitic heat loss and no pressure drop. The process in the compressor is assumed to be isoentropic. The results of the calculations of the intrinsic Carnot efficiency of the idealized cycle are presented in the fig. 2 together with the exergy losses D_{TXV} and D_{CND} in the TXV and the condenser. The following equations were used for calculation:

$CEF = COP * (T_A - T_R) / T_R$, where $COP = Q_{RMC} / P_{MC}$.
$CEF = 1 - (D_{TXV} + D_{CND}) / P_{MC}$,
$D_{TXV} = T_A * (S_3 - S_2)$ and $D_{CND} = (H_6 - H_7)*(T_A - 0.5 *(T_6 + T_7)) / T_R$.

The CEF of the system noticeably decreases at lower T_R primarily because of increasing the D_{TXV}. One way to reduce the D_{TXV} is to lower the TXV inlet temperature T_2. This can be done by means of an auxiliary refrigeration cycle.

A traditional mechanical subcooler with a pure refrigerant will only partially improve the overall system performance. In this case additional exergy losses occur in the evaporator /8/ of the subcooler where the liquid refrigerant of the main cycle is in the heat transfer with the auxiliary cycle refrigerant boiling at constant temperature. The results of the calculations for this case are presented in Fig.3 with dashed lines.
The graph shows the CEF increase as a function of the temperature difference ($DT_{12} = T_1 - T_2$). These data are plotted for various values of the subcooler cycle efficiency (CEF). If $DT_{12} = DT_{MAX} = T_A - T_R$, the pure refrigerant used in the subcooler does not give any improvement in the overall system efficiency. Calculations predict a maximal gain in CEF at intermediate values of $DT_{12} = 25...35$ °C (77...95 °F).

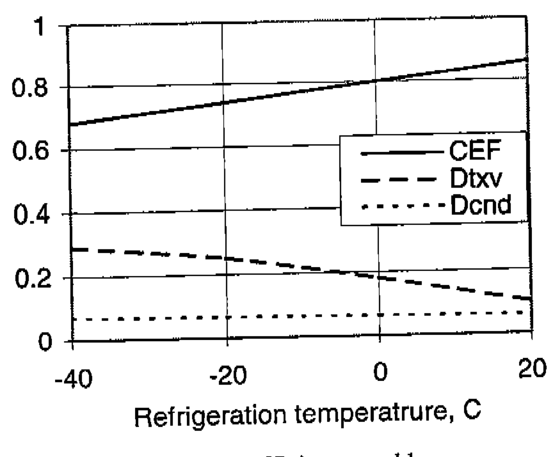

Fig. 2. Idealized cycle efficiency and losses

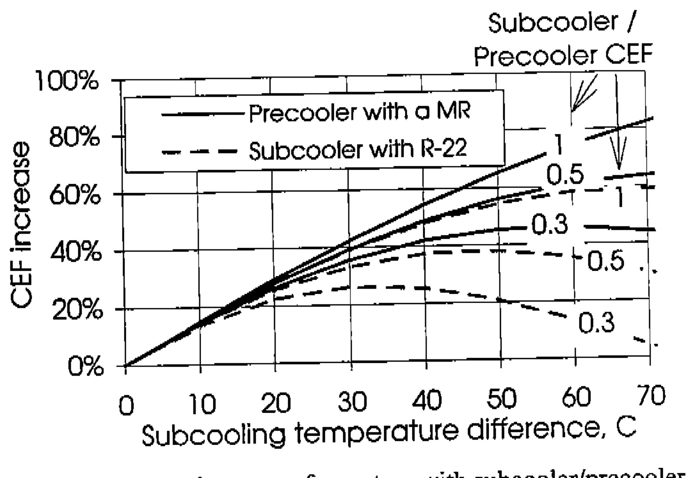

Fig. 3. Efficiency increase of a system with subcooler/precooler

Further improvements can be achieved by modification of both the auxiliary cycle configuration and its refrigerant /7/. Precooling of the liquid refrigerant between the condenser outlet to the TXV inlet is a temperature distributed process. It is known /9/ that zeotropic mixed refrigerants can be designed in a way to match any temperature load profile in the evaporator which reduces the heat transfer exergy losses. At the same time the TXV inlet temperature in the preccooler itself should also be as low as possible in order to reduce the D_{TXV}. One way to provide this is to use a three-flow heat exchanger (fig.1). The CEF increase for this case is presented by solid lines on the fig.2. For the expected values of the precooler CEF in the range of 0.25 to 0.30, the overall efficiency increase can reach 40%. This is twice as it is for conventional mechanical subcoolers.

CHARACTERISTICS OF A FREEZER WITH A PRECOOLER

A calculation model was developed to simulate a refrigeration system with a precooler. The thermodynamic properties of pure and mixed refrigerants were calculated using an original equation of state. It predicts phase equilibria and multicomponent mixture properties and is sufficiently accurate even when experimental information is limited. The calculation comparison was conducted for a 43.3 °C (110 °F) condensing temperature, and -30.0 °C (-22 °F) evaporating temperature. Having those temperatures fixed allows to use conventional values of COP instead of CEF.

The summarized simulation results for freezers using R-502 is presented in the fig.4. The total COP increase with a precooler is up to 40 % when the precooling temperature difference is close to its maximum value. If R-507 or R-404A are used in place of R-502, the gain in COP can be even higher reaching 60% for high ambient conditions.

In addition the cooling capacity of the system increases. Fig.5 shows the increase in the cooling capacity that can be obtained in a freezer with R-502 and R-507 refrigerants.

The possible use of the extra capacity depends on the particular system application. For a retrofit situation when a precooler is added to the existing system, extra capacity will enable to either increase the number of display cases or to lower storage temperature. The capacity gain can be used especially effectively for new installations. In this case the size of the compressor and the components of the main system can be almost halved. Even though the precooler adds another compressor to the system, the total size and cost of the installation may be lower. Fig.6 shows a total size of the compressors in a low temperature system with a precooler. More than 30% decrease in overall size can be realized when a precooler is used, bringing the cost of the installation down.

Efficient operation of a precooler depends largely on a proper design of its mixture refrigerant. As a starting point, the MR glide in the precooler return flow should match the precooling temperature difference of the main

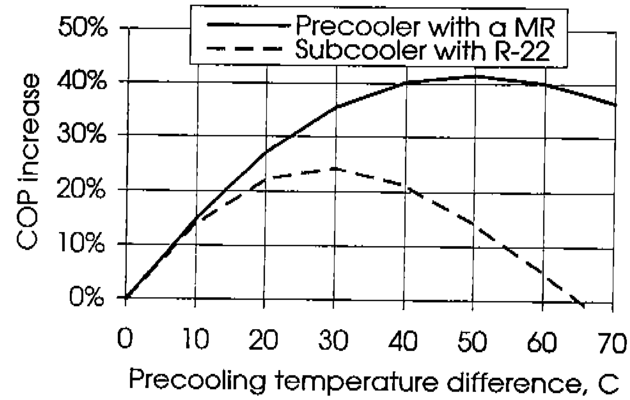

Fig.4. R-502 freezer COP increase with a precooler

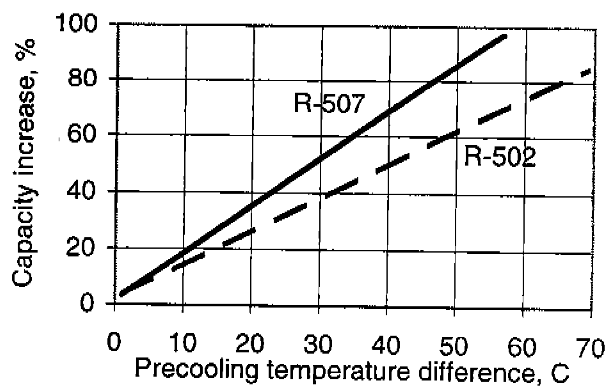

Fig.5. Cooling capacity increase with a precooler

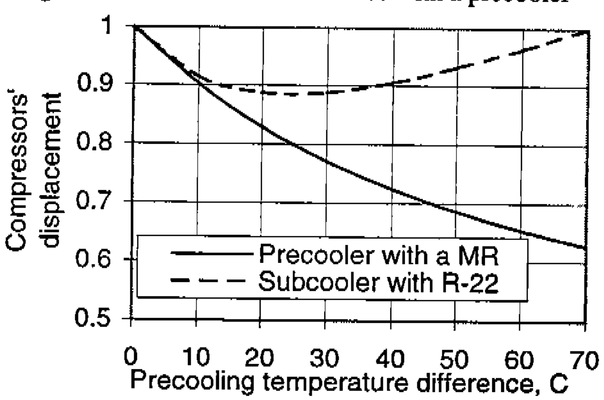

Fig.6. Total compressor displacement needed.

refrigerant. Other parameters should also be taken into account such as compressor discharge temperature, pressure ratio, etc. Environmental properties of the MR should also be acceptable.

The first suitable blend identified was tested in a lab as described below. Since then a new more efficient blend was developed. The comparison of R-22 as a refrigerant for a precooler with the two blends is presented in the table 1.

Table 1. Refrigerants comparison for a precooler

		R-22	Blend 1	Blend 2
Pressure Ratio		5.8	4.8	2.3
Compressor Displacement needed	cfm	14	17	6
Discharge Temperature	°C (°F)	61 (142)	82 (180)	65 (149)
Precooler Power consumption (compressor adiabatic eff.=0.5)	kW	5.8	4.8	3.8
Efficiency relative to R-22		1	1.21	1.53
R-507 freezer capacity increase	%	90	90	90
R-507 freezer COP increase	%	**17**	**26**	**37**

Comparison conditions: Condensation temperature 33 °C (91 °F)
*Precooling **50 °C** (90 °F), Precooler refrigeration capacity 10 kW.*

Experimental tests of the Blend 2 are now in progress to confirm the predicted improvement in performance.

LAB TESTS

Test Facility and Instrumentation.

A test facility has been built and a series of tests has been conducted to demonstrate the benefits from using a precooler with a MR. Figure 7 shows the installation to test a freezer with a precooler . It consists of the four loops. The main component of the basic refrigeration circuit is a semi-hermetic, two-cylinder compressor with a 2 hp motor. An accumulator is located in the compressor suction line. Refrigerant flow is controlled manually by using a thermal

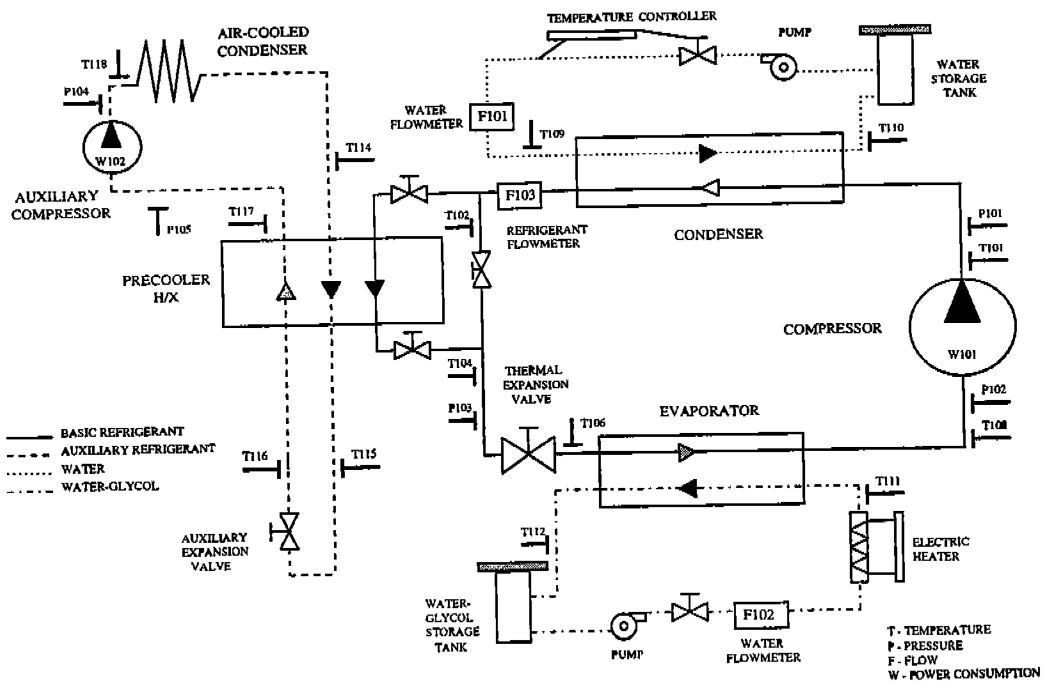

Fig.7. Schematics of the experimental setup.

expansion valve (TEV). The water-cooled counterflow condenser has a spiral coaxial configuration. A brazed-plate counter-flow evaporator is used to provide the heat exchange between the water-glycol and the refrigerant flows.

The heat source for the evaporator is provided by an aqueous glycol solution circuit that contains a storage tank, a circulating pump, a flowmeter, an electric heater and an interchanger. Heat is transferred from the condenser into a water circulating loop which contains a storage tank, a circulating pump and an automatic control system. The water flow rate is modulated manually by using globe-valves. The water temperature in the storage tank is controlled automatically by injecting a definite amount of the city water. Liquid line precooling is provided by an auxiliary refrigeration circuit which is composed of an air-cooled condensing unit, expansion valve and three-flow heat exchanger. The condensing temperature is controlled by an "on/off" switch on two of the four cooling fans. A custom design expansion valve regulates the refrigerant flow. The three-flow heat exchanger is designed to provide the minimal temperature difference between counter flows. The new design is able to respond to a possible temperature glide from 30 °C (54 °F) to 70 °C (126 °F) across the heat exchanger.

The temperature was measured using platinum 4-wire RTDs having an accuracy of 0.01 °C. The basic unit suction line transducer has an accuracy of $\pm$0.2 psi (1.5 kPa) and the liquid line transducer $\pm$0.8 psi (6 kPa). Other pressure transducers are calibrated to $\pm$1.0 psi (7 kPa). The main refrigerant flow is measured with a Coriolis effect mass flowmeter with an accuracy of $\pm$0.2%. Water and water-glycol flows are measured with paddlewheel type flow sensors providing an accuracy of $\pm$3.0 ml/sec ($\pm$2.0%). The power consumption of the main and the auxiliary compressors are measured with an accuracy of $\pm$0.3%. All data inputs are supplied to a data acquisition system and processed by a computer.

Test Results.

The tests were conducted at three different evaporator temperatures -33 °C (-28 °F), -28.9 °C (-20 °F) and -23.3 °C (-10 °F) by adjusting the temperature and flow rate of the water-glycol solution. The suction line superheat, measured from the dew-point temperature in the evaporator, was in the range of 3 °C to 5 °C. It was maintained by adjusting the refrigerant flow with the expansion valve.

The baseline test was run with R-502 in the main cycle for precooling temperature difference DT_{12} = 30 °C.

The format of this paper does not allow to present much of the obtained results. An example of the data calculated from the experimental measurements at the condensing temperature of 43.3°C is presented in table 2. Performance data for a R-502 freezer with a precooler using a new MR and the R-22 refrigerant are given for 30 °C precooling. The freezer's cooling capacity increased by about 60 %. The overall system COP increased by 30% to 35% with a precooler using a MR, which is in a good agreement with the calculations presented earlier.

Table 2. Experimental data for precooler with a MR at 30°C (54 °F) subcooling.

Refrigerator	Evaporating Temperature °C (°F)	Cooling Capacity, W (Btu/h)	Total Power Consumption, W	COP	Cooling Capacity Increase, %	COP Increase %
Without precooler	-28.9 (-20.0)	2130 (7270)	1960	1.09	x	x
	-33.0 (-27.4)	1460 (4980)	1680	0.87	x	x
Precooler With R-22	-28.9 (-20.0)	3260 (11130)	2430	1.34	53	23
	-33.0 (-27.4)	2400 (8190)	2130	1.13	64	30
Precooler With MR	-28.9 (-20.0)	3350 (11430)	2350	1.42	57	30
	-33.0 (-27.4)	2380 (8120)	2040	1.17	63	35

Main Refrigerator: Condensing Temp. +43°C (109 °F),
Precooler: Condensing Temp. +40°C (104 °F).

Precooler compressor operating parameters such as the pressure ratio and the discharge temperature were closely monitored. With the MR tested, they were within acceptable limits in all regimes, whereas with R-22 compressor discharge temperature was much higher exceeding 110 °C (230 °F) especially at higher subcooling.

CONCLUSIONS

Precooling of a liquid refrigerant in a freezer is identified to be an attractive application for mixture refrigerants. Well designed precooler system and mixture refrigerant can provide up to 40% increase in the overall efficiency of a conventional vapor compression cycle. This results in twice as much improvement compared to the conventional mechanical subcooling. An increase in the cooling capacity of up to 100% is also realized which can translate into a reduction in the overall size of the compressors by up to 30% for new installations. A precooler can be built using standard mass produced components. The relatively small size of the unit makes it possible to use a packaged design which makes it easier to handle zeotropic refrigerants.

REFERENCES

1. Thomas A. Mahoney, Food industry is huge market for refrigeration manufactures, The Air Conditioning, Heating and Refrigeration News, Nov. 14, 1994, *p.38-39*
2. Brown T., Refrigeration system with subcooler, US Patent No.3,852,974, 12.10.74.
3. Couvillion R.J., Larson M.W., Somerville M.H., Analysis of a vapor-compression refrigeration system with mechanical subcooling. *ASHRAE Trans., 1988, Vol.94(2), p.641-659.*
4. Thoronton J.W., Klein S.A., Mitchell J.W., Dedicated mechanical subcooling design strategies for supermarket applications. *Int.J.Refr., 1994, Vol.17, No8, p.508-515.*
5. Boiarski M., Brodianski V., Longsworth R., Retrospective of mixed-refrigeration technology and modern status of cryocoolers based on one-stage, oil-lubricated compressors, Proceedings of the Cryogenics International Conference, 1997, Portland.
6. Boiarski M., Munday C., Yudin B., Thermodynamic properties and efficiency of FRIGC FR-12 as a R-12 substitute", Proceedings of the International Conference On Ozone Protection Technologies, 1996, Washington.
7. Boiarski M., Podcherniaev O., Precooled vapor-liquid refrigeration cycle, US Patent No.5,739,993, 3.24.98.
8. Brodianski V., Boiarski M., Lunin A., The exergy analysis of the throttle refrigerating systems on pure and mixed refrigerants, Proceedings of the Energy Systems Design Analysis Conference, Vol.64-3, ASME, 1994.
9. Mulroy W., Domanski P.A., Didion D., Glide matching with binary and ternary zeotropic refrigerant mixtures, Proceedings of the IIR meeting "Energy efficiency in refrigeration and global warming impact", May 1993, Paris.