International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2002

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R18-3

COMPARATIVE PERFORMANCE OF TWO-STAGE CASCADE AND MIXED REFRIGERANT SYSTEMS IN A TEMPERATURE RANGE FROM –100 C TO –70 C

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ABSTRACT

This paper presents comparative refrigeration performance data and analysis of intrinsic and extrinsic losses for two-stage cascade and mixed refrigerant (MR) systems. Compressor characteristics, effectiveness of the counter-flow heat exchangers and mean temperature difference in evaporator and condenser were taken into account. Non-flammable commercially available HFC refrigerants were considered in both cascade and MR systems. The analysis has shown that at temperatures below –80 C, the MR system provides higher COP and requires smaller compressor displacement than the cascade system. The experimental performance data were obtained for a MR system operating with a reciprocating hermetic compressor.

BACKGROUND

Low-cost refrigeration systems using mixed refrigerants (MR) can be based on industrial oil-lubricated compressors. This allows long term, reliable operation. The high efficiency of MR technology was proven in developing coolers for different applications. Small-scale MR coolers provide refrigeration capacity up to 100 W in a wide temperature range below –100 C / 1 /. Industrial MR coolers have been built since 1975 / 2,3 /. They provide refrigeration capacity up to 3600 W at refrigeration temperatures T_R = - 153 C…-100 C.

A comparison of different cycles is of essential interest in systems development. This paper presents comparative data on refrigeration efficiency for systems operating with modern hydrofluorocarbon (HFC) refrigerants to provide constant temperature refrigeration. The calculated data were obtained with an identical thermodynamic model, which provides all thermodynamic parameters needed to analyze the exergy losses. Analysis of these losses allows optimization of system design.

A MR refrigeration system schematic is shown in Figure 1a. The cooler consists of a compressor unit (CM), condenser (CD) and cold box (CB) that includes a counter-flow heat exchanger (HX), a throttle device (THR) and an evaporator (EV) to remove heat Q_R from the object to be cooled at the temperature T_R.

A selected cascade system schematic is shown in Figure 1b. It operates with either individual or azeotropic refrigerants. A relatively warm circuit provides refrigeration at the precooling temperature (T_{PR}) in the condenser-evaporator (CE). The counter-flow heat exchanger HX can be installed to each circuit in order to increase capacity and reduce the exergy losses associated with the throttling process.

The thermodynamic efficiency of the cycles can be determined with the coefficient of performance (COP) at a specified T_R and ambient T_A: COP = Q_R / PW_C. Carnot efficiency coefficient (CEF) allows a comparison at different T_R and T_A: CEF = $Q_R*(T_A / T_R – 1) / PW_C$ .It can be presented in the form of: CEF = $1 – D_{SUM} / PW_C = 1 – (D_{INT} + D_{EXT}) / PW_C$, where PW_C is a power input, D_{SUM}, D_{INT}, D_{EXT} are summarized, intrinsic, and extrinsic exergy losses / 4 /. The performance of an actual refrigeration system depends on both D_{INT} and D_{EXT} constituents. An idealized cycle model can be formulated to analyze only selected factors that are important for a particular stage of the analysis.

The originally designed computer software based on the Soave equation of state was used for calculating the refrigeration capacity and efficiency at specified temperatures T_A, T_R and pressures: high (P_H) and low (P_L) in the cycle. In addition, a minimal temperature difference (ΔT_{MIN}) was specified to avoid a pinch point in the heat exchangers.

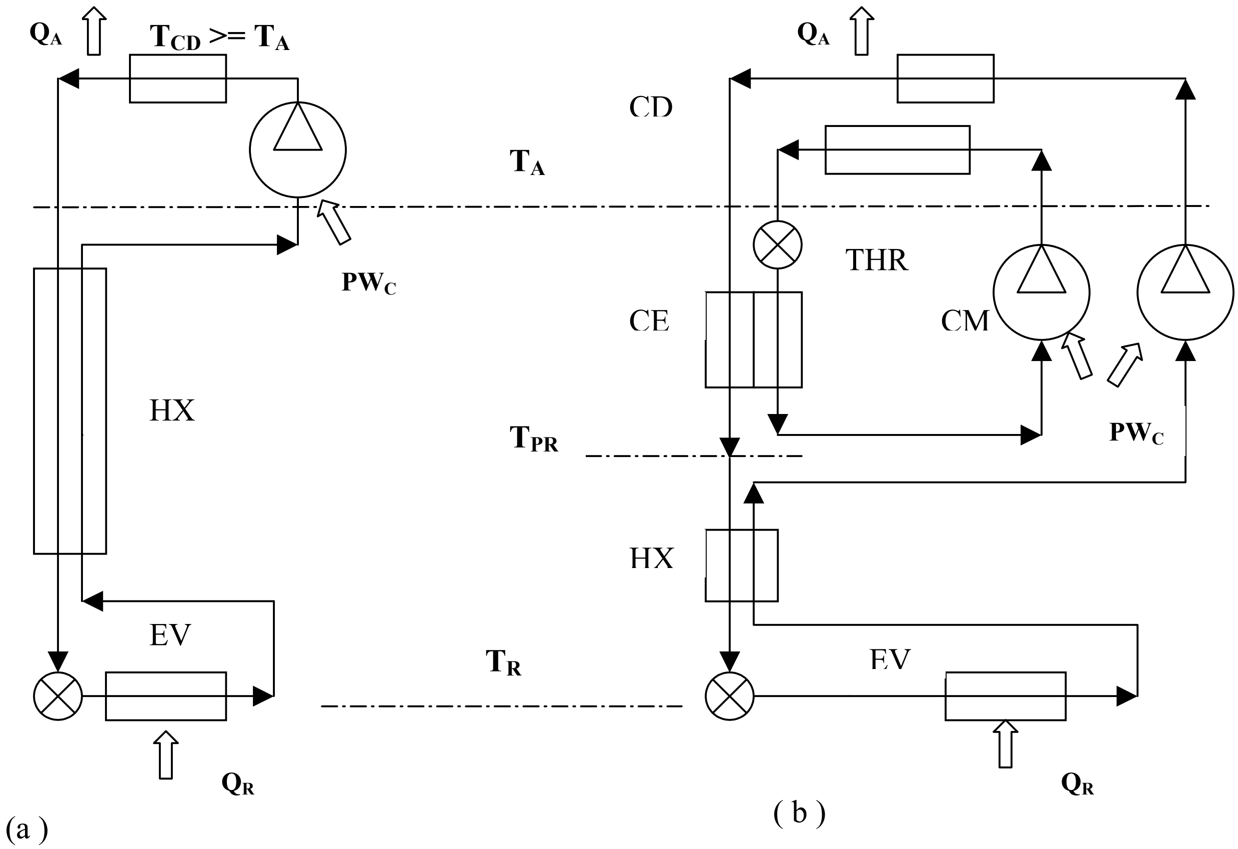

Figure 1. Schematics of the cascade and MR cycles.

IDEALIZED CYCLES ANALYSIS

A highly idealized cycle model takes into account only intrinsic exergy losses, those that inherently depend on a refrigerant's thermodynamic properties and irreversibility of the process. These losses cannot be eliminated even hypothetically. In the selected cycles summarized, intrinsic losses include the following constituents.

An adiabatic throttling conducted at h = const causes an entropy growth Δs and correspondent loss of exergy: $D_{THR} = T_A * \Delta s * G$, where G is a refrigerant flow rate.

The temperature difference in the HX depends on a ratio of specific heat capacities for both high and low pressure streams. An appropriate loss of exergy D_{HX} can be defined as: $D_{HX} = Q_{HX}$

* $\Delta\tau_e$, where Q_{HX} is a thermal load, $\tau_e = (T_A / T_{AV} - 1)$ is an exergetic temperature function at the average temperature T_{AV} of the appropriate stream in the HX.

The temperature glide in the evaporator results in the loss of exergy $D_{EV} > 0$ if the system is designed for the constant temperature refrigeration (T_R = const): $D_{EV} = Q_R * \Delta\tau_e$.

It is assumed that both cycles would operate with adiabatic compressors. A compressor with 1 cfm (0.47 dm^3/sec) displacement was selected for the MR cycle and the low stage of the cascade cycle. In general a compressor discharge temperature (T_{DCH}) is higher than T_A. This is because part of the compressor work is transformed into refrigerant thermal exergy. This exergy D_{ACM} is not used in the dedicated refrigeration cycle. The condenser rejects this heat to the environment. A penalty for using an adiabatic compressor can be calculated as follows: $D_{ACM} = G * (h_{DCH} - h_A) * \tau_e$, for $T_{AV} = (T_{DCH} + T_A)/2$.

Thus, a maximum achievable Carnot efficiency of the selected cycles can be calculated with a model that counts only for D_{INT}:

$CEF = Q_R*(T_A/T_R - 1) / PW_C = 1 - (D_{THR} + D_{HX} + D_{EV} + D_{ACM}) / PW_C$.

The performance parameters of such idealized cycles are presented in Table 1. Refrigerants R-32 and R-23 were selected for the cascade cycle to provide maximum thermodynamic efficiency. The condenser-evaporator mean temperature was optimized at T_{PR} = -40 C.

The mean temperature difference in the HX operating with MR is calculated as:

$\Delta T_{AV} = Q_{HX} / \Sigma (Q_I / \Delta T_I)$, where Q_I and ΔT_I are the parameters at the incremental segments of the HX. Results were obtained with an assumption that minimum temperature difference in the MR heat exchanger ΔT_{MIN} = 0.2 C.

The cascade cycle demonstrates high CEF = 0.70…0.72 values (Table 1). However, at T_R < -80 C the second-stage compressor suction pressure is lower than one atmosphere. This results in significant reduction of the specific refrigeration capacity q_v (W/cfm). Table 1 also presents data for the single-stage MR cycle. Composition of the four-component HFC-based mixed refrigerant and compression ratio P_H / P_L were optimized for each T_R. The optimized MR provides a relatively small temperature difference ΔT in the HX. However, CEF of the idealized MR cycle does not exceed CEF= 0.6. This is due to an increased value of D_{ACM} compared to the cascade cycle. In the MR cycle the compressor suction point temperature is close to T_A. This increases the compressor discharge temperature and D_{ACM} losses, which are about 70% of total D_{INT}. At the same time, the MR increases the efficiency of the processes in the cold box. The cold box CEF_{CB} is as high as: CEF_{CB} = 0.83…0.75.

Comparative performance of the highly idealized cycles is given in Table 2 at T_R = -70 C. Each constituent in loss of exergy is given in percentage of input power PW_C. These data clearly confirm the significant influence of compressor losses (D_{ACM}=32%) on the MR cycle efficiency. Meanwhile, the data also show the advantage of the MR cycle, which provides the higher refrigeration capacity q_v per unit of the compressor displacement. In addition, MR cycle allows operation at any T_R with suction pressure P_L above one atmosphere. This would facilitate the design parameters and improve the performance of the actual MR system.

EVALUATION OF ACTUAL SYSTEM PERFORMANCE

The analysis of actual systems involves the influence of extrinsic losses that depend on design features and equipment performance. The following factors cause these losses: hydraulic pressure drop in the supply and return lines, heat intakes through the insulation, increased temperature difference across the heat exchangers due to limited heat transfer coefficients, and

Table 1. Performance of idealized refrigeration cycles.

Parameter	Refrigeration Temperature, C		
	-70	-80	-90
Cascade Cycle : Upper Stage			
Discharge/Suction Pressure, bar	19.8 / 1.9	19.8 / 1.9	19.8 / 1.9
Power Input, W	294	197	105
Low Stage:			
Discharge/Suction Pressure, bar	7.7 / 1.6	7.7 / 1.1	7.7 / 0.6
Power Input, W	135	119	89
Refrigeration Capacity, W/cfm*	278	227	152
Total COP	1.37	1.25	1.09
Total Carnot Efficiency	0.70	0.72	0.72
Mixed Refrigerant Cycle			
Discharge/Suction Pressure, bar	18.7 / 5.6	17.5 / 4.9	17.5 / 4.1
Power Input, W	335	313	300
Refrigeration Capacity, W/cfm*	369	295	254
COP	1.10	0.94	0.85
Carnot Efficiency	0.56	0.54	0.56

Table 2. Intrinsic exergy losses in the components of the idealized cycle at T_R=-70 C.

Cycle Exergy Losses, % of PW_C	Cascade Cycle		MR Cycle
	Upper Stage	Low Stage	
Compressor	7	<1	32
Counter-Flow Heat Exchanger	n/a	<1	5
Throttle	16	2	<1
Evaporator	5	0	7
Total	30		44

losses related to compressor design.

The last constituent is a major factor in the reduction of refrigeration performance. The compressor performance can be measured by volumetric and isentropic efficiencies. The volumetric efficiency is the ratio of actual volumetric flow entering the compressor suction port to the geometric displacement of the compressor. The isentropic efficiency is the ratio of the work required for the isentropic compression to the work input to the compressor shaft. A series of tests was conducted to determine the efficiencies of a 4 cfm (1.9 dm^3/sec) reciprocating hermetic compressor on R22, nitrogen, non-flammable MIX1 and flammable MIX2 refrigerants. The efficiencies are shown as a function of compression ratio in Fig. 2 and Fig. 3. Both mixed refrigerants have demonstrated slightly higher isentropic and a little lower volumetric efficiencies than R22, at compression ratios 2...6. However, for further analysis we assumed the identical efficiencies for the compressors working with the conventional or mixed refrigerant.

The efficiencies were also evaluated for commercial refrigeration compressors with a displacement of 12 cfm (5.7 dm^3/sec) for reciprocating hermetic and 24 cfm (11.3 dm^3/sec) for scroll compressors. The volumetric and adiabatic efficiencies (Fig. 4) have been obtained by processing data from manufacturers' catalogs. It can be seen that scroll compressors demonstrate an advantage in performance over the reciprocating hermetic ones. The absence of valves, expansion volumes and the continuous flow process result in high volumetric efficiency over a wide range of operating pressures. It is very difficult to access the actual compressor data for the low stage compressor of the cascade system. Therefore, the identical performance has been used for all the compressors.

In order to achieve better and practically oriented results, refrigerants R-404a and R-508b have been selected for the cascade cycle. In addition, a counter-flow liquid line (LL) - suction line (SL) heat exchanger was considered in the upper stage (not shown in Fig.1).

The counter-flow HX is an essential part of the MR system, as the condenser-evaporator is for the cascade system. A mean temperature difference ΔT_{AV}=5 C in both heat exchangers was assumed. The regeneration ratio $\Delta T_{LL}/\Delta T_{SL}$ in the LL-SL heat exchangers was limited by a maximum allowable compressor discharge temperature in each stage of the cascade system. The regeneration ratio 50% was assumed for the upper HX and 90% for the low stage one. The losses due to the temperature difference in the condenser and evaporator were neglected.

The following common parameters were specified for the cascade and MR refrigeration

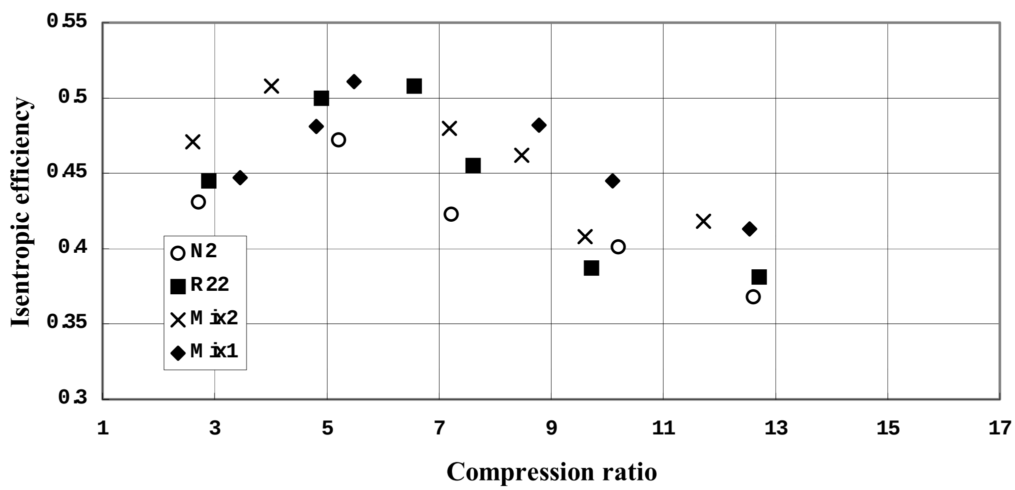

Figure 2. Test data on the isentropic efficiency for 4 cfm reciprocating hermetic compressor.

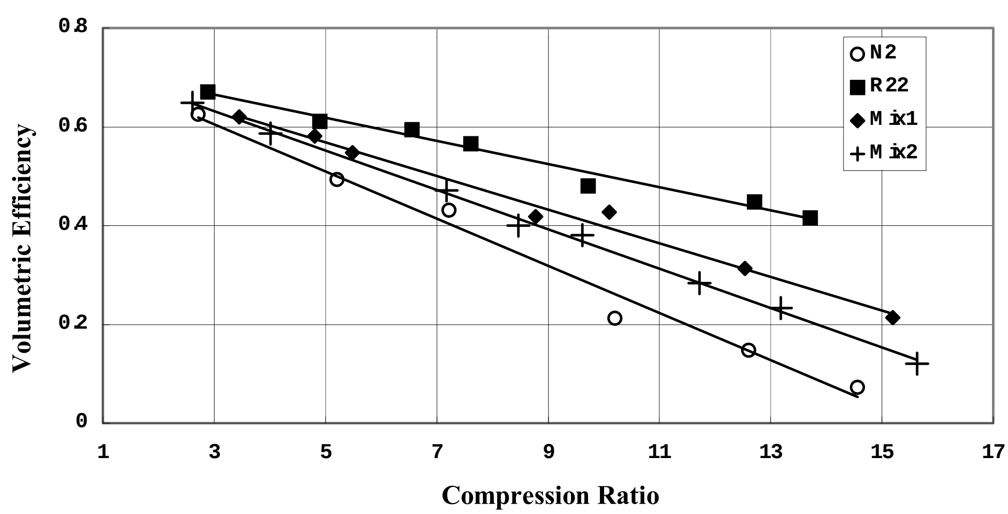

Figure 3. Test data on the volumetric efficiency for 4 cfm reciprocating hermetic compressor.

systems: condenser outlet temperature T_{CND}=25 C and evaporator outlet temperature T_R.

HFC refrigerant R-508B was chosen for the low stage of the cascade system in accordance to recommendations made in / 5 /, and R-404A was selected for the upper stage. The low stage refrigerant condensing temperature was set at T_{PR} = –40 C based on the results obtained for the idealized cycles.

Composition and suction pressure of the MR was also optimized to achieve maximum thermodynamic efficiency of the actual MR system. The refrigerant was composed of commercially available non-flammable, non-toxic, HFC components.

The comparative calculated COP data for the refrigeration capacity Q_R=1kW systems are shown in Tables 3-4. The actual cascade system with either compressor demonstrates a better efficiency above refrigeration temperatures T_R = –80 C. At lower temperatures the performance benefits of the MR system become quite obvious. The higher COP values are mainly due to low compression ratios that result in higher isentropic efficiencies of the compressors in the MR system (Fig. 4). For example, at T_R = –100 C the COP of the MR system with the reciprocating hermetic compressor is about 60% higher than that of the cascade system. The COP of MR system employing the scroll compressor is higher by 18% compared to a cascade system with two scroll units.

The configuration and size of the refrigeration system are important in many industrial applications. The compressor unit is a major component of the system, and its large physical size may become a limiting factor for system applicability. Lower volumetric efficiency of the compressor requires larger compressor displacement. In this respect the actual MR system offers a substantial improvement over the cascade technology, especially at T_R lower than –80 C. For example, at T_R = –90 C the required displacement of the reciprocating hermetic MR compressor is nearly one fifth the total size of compressors required for the cascade system. The required compressor displacement to provide Q_R=1kW as a function of T_R is shown in Fig. 5. Even the required size of the scroll compressor with MR is 50% smaller than that for the cascade system.

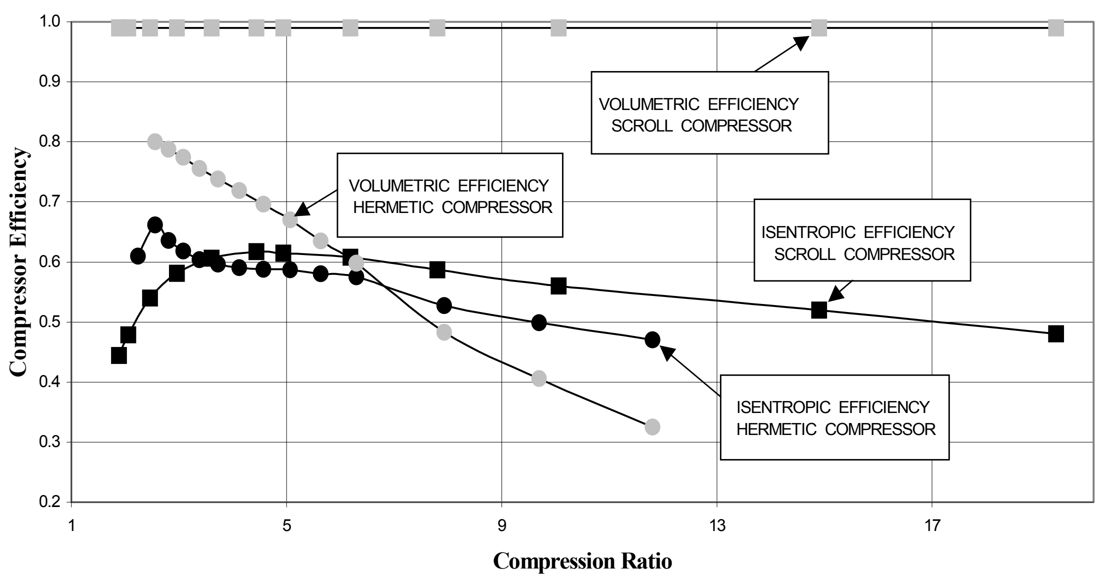

Figure 4. Efficiencies of large reciprocating hermetic and scroll compressors designed for R404a.

Table 3. Performance of actual systems with reciprocating hermetic compressors.

	MR SYSTEM			CASCADE SYSTEM							
				Low Stage			Upper Stage			TOTAL	
Evaporator Temp., C	Pressure Ratio	Power Input W	COP	Pressure Ratio	Power Input W	COP	Pressure Ratio	Power Input W	COP	Power Input W	COP
-70	2.38	1565	0.64	3.35	280	3.58	11.7	1221	1.05	1501	0.67
-80	2.85	1566	0.64	5.54	455	2.20	11.7	1388	1.05	1844	0.54
-90	4.44	1992	0.50	9.68	738	1.36	11.7	1658	1.05	2396	0.42
-100	6.25	2348	0.43	18.3	1423	0.70	11.7	2312	1.05	3735	0.27

Table 4. Performance of actual systems with scroll compressors.

	MR SYSTEM			CASCADE SYSTEM							
				Low Stage			Upper Stage			TOTAL	
Evaporator Temp., C	Pressure Ratio	Power Input W	COP	Pressure Ratio	Power Input W	COP	Pressure Ratio	Power Input W	COP	Power Input W	COP
-70	2.38	1730	0.58	3.35	293	3.41	11.7	1055	1.23	1348	0.74
-80	2.85	1725	0.58	5.54	433	2.31	11.7	1167	1.23	1599	0.63
-90	4.44	1895	0.53	9.68	659	1.52	11.7	1353	1.23	2011	0.50
-100	6.25	2233	0.45	18.3	1016	0.98	11.7	1644	1.23	2660	0.38

The actual MR cooler has been designed to provide a cooling capacity Q_R =500 W at T_R = –80 C for a semiconductor application. A minimum size and minimum power consumption were required. A 4.6cfm reciprocating hermetic compressor has been selected. MR composition and HX geometry were optimized via a customized computer model. Even at a very comfortable pressure ratio of 4:1 to 5:1, the compressor adiabatic efficiency didn't exceed 52% (Fig. 2). The COP of the cooler was influenced by a large mean temperature difference ΔT_{AV}=15 C in a reduced size heat exchanger, by pressure drops and heat intakes through the insulation. The experimental parameters of the MR cooler at various T_A are shown in Table 5. The maximum obtained COP is 0.36 which corresponds to Carnot efficiency CEF=0.21. This CEF value is comparable with that of conventional commercial freezers providing refrigeration at T_R = –40 C / 3 /.

SUMMARY

Mixed refrigerant systems based on a single industrial compressor operating at temperatures below -80 C provide refrigeration with a better coefficient of performance than a traditional cascade cycle operating with two compressors.

The flexibility of mixed refrigerant technology allows an optimal pressure ratio for existing compressors. This reduces compressor displacement when compared to a cascade system of the same capacity. In addition, the required displacement for a compressor used in a MR system is smaller than the total required displacement of a cascade system by a factor of 5 to 6.

The Carnot efficiency of mixed refrigerant systems is essentially dependant on the losses associated with an adiabatic compressor. Exergy losses due to mean temperature difference in the counter-flow heat exchanger also significantly influence the performance of the MR cycle.

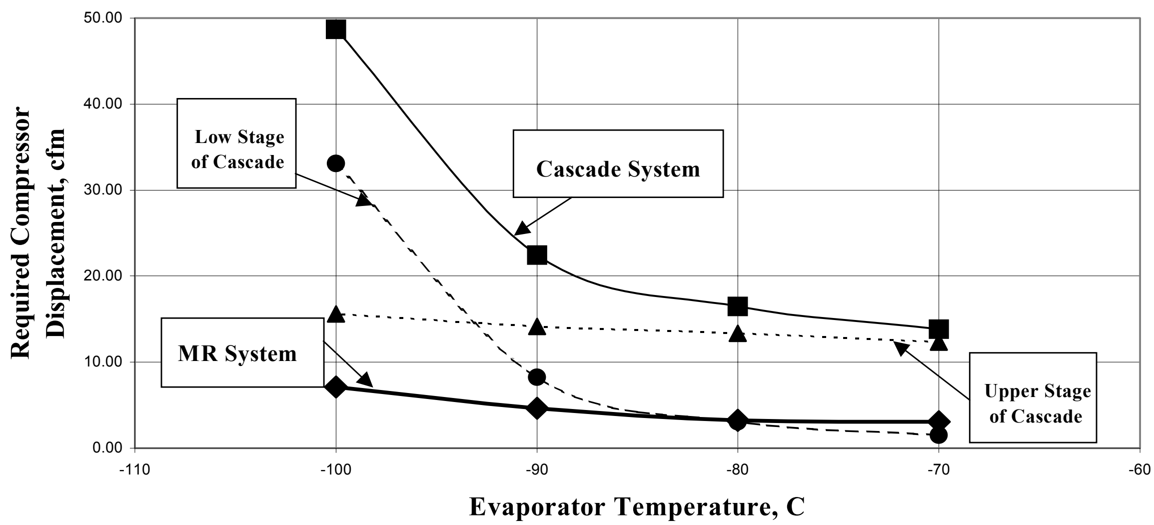

Figure 5. Actual Systems with reciprocating hermetic compressors.

Table 5. Experimental Parameters of MR System at T_R=-80 C.

Ambient Temperature C	Discharge Pressure bar	Suction Pressure bar	Refrigeration Capacity W	Power Consumption W	COP	Carnot Efficiency
20	18.7	4.0	530	1478	0.36	0.20
22	18.2	4.2	543	1516	0.36	0.21
26	18.9	5.1	500	1610	0.31	0.18
29	18.9	5.2	494	1636	0.30	0.18

ACKNOWLEDGEMENT

We thank Jennifer Hammond for her editing of this paper and Kevin Flynn for his valuable comments regarding this paper.

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