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ENVIRONMENT-FRIENDLY REFRIGERATION PACKS FOR INDIAN SUPERMARKETS: EXPERIMENTAL INVESTIGATION OF ENERGY PERFORMANCE OF A MULTIEJECTOR-DRIVEN R744 INTEGRATED COMPRESSOR RACK

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ABSTRACT

An integrated CO_2 refrigeration facility is designed for hot climate conditions to improve the basic knowledge of the CO_2 refrigeration in India, especially in the supermarket sector. The operational modes can be adapted to simulate different applications as well as the gas cooler outlet temperature can be regulated to simulate various ambient conditions. For Southern India, a parallel compression configuration with ejectors for expansion work recovery is the most efficient mode (mode 4). It can deliver freezing, refrigeration and air conditioning loads, and has a heat reclaim option. The different operation configurations help to understand the behavior of CO_2 refrigeration for a wide range of applications as well as reveal the respective influences between components. A first test phase running mode 4 for an ambient temperature of 46°C and various receiver pressures in a range of 44 bar - 50bar, proves the feasibility of R744 transcritical systems as a non-HFC based alternative to HCFC-22 in retail applications in countries with high ambient temperature. The CO_2 refrigeration test facility is a preparation and demonstration site for a full-scale replacement of existing commercial refrigeration installations in India.

Keywords: Refrigeration System, R744, Carbon Dioxide, Transcritical Cycle, Multi-Ejector, Experiment

1. INTRODUCTION

Worldwide it is committed to eliminate the usage of ozone depleting HCFC and HFCs due to their high global warming potential (GWP). Therefore, the development of energy efficient and integrated refrigeration and Air-Conditioning (A/C) systems based on natural working fluids, adapted to hot climatic conditions in countries like India, is necessary, Sharma (2014), Gullo (2016). Therefore, a demo test facility using CO_2 as the only working fluid is designed to improve the basic knowledge of CO_2 refrigeration and its applications, especially for the supermarket sector. For high ambient temperature regions like Southern India, a parallel compression configuration with ejectors, Hafner et al. (2014), or expanders, Singh et al. (2016), for expansion work recovery is the most efficient mode, therefore the applied layout included all the options of energy efficiency boosting, i.e.:

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parallel compression and expansion work recovery with the use of multi-ejectors, Banasiak et al. (2015). The newly designed integrated system can deliver freezing, refrigeration and air conditioning loads, with a heat reclaim option. Higher ambient temperature conditions can be simulated by reducing the rotational speed of the gas cooler fans. For each ambient temperature the receiver pressure can be manually controlled maintaining the A/C evaporating temperature. By changing the glycol inlet temperature of the evaporators, the superheat, and therefore the evaporators efficiency can be adjusted. All those different configurations resulting in the possibility to understand the behavior of CO_2 refrigeration for a wide range of applications as well as the respective influences between components. A first test phase running mode 4 for an ambient temperature of 46°C and various receiver pressures in a range of 44-50bar, proves the feasibility of R744 transcritical systems as a non-HFC based alternative to HCFC-22 in retail applications in countries with high ambient temperature. The newly commissioned CO_2 refrigeration test facility at IIT Madras, Chennai, India, is a preparation and demonstration site for a full-scale replacement of existing commercial refrigeration installations in India.

2. INDEE CO_2 TEST FACILITY

The INDEE multifunctional test facility, as shown in Figure 1, uses CO_2 as working fluid consists of three evaporators for freezing (16), cooling (17) and Air Conditioning (18). A parallel compression system configuration with ejectors for expansion work recovery as well as a heat reclaim option (5) was applied. Low (LT) and medium temperature (MT) compressors (1, 3) suck the saturated and precooled (14, 15), for better efficiency, gas of the LT and MT evaporators (16, 17) respectively and compress it to a high-pressure level. Before entering the evaporators, the pre-heated working fluid gets expanded to the required pressure level by passing a throttling valve (h-j). LT and MT compressors are series-connected with an intermediate gas cooler, de-superheater (2), reducing the MT suction temperature (temp). Typically, for CO_2, a decreasing evaporating temperature with 1 K decrease the compressor power demand by about 0,5% and reduce the cooling capacity by 3%. The small amount of oil in the subcritical gas gets removed by the oil separator (20) before entering the gas cooling process. The evaporated gas of the A/C evaporator (18) gets compressed by ejectors utilizing this high-pressure gas coming from the gas cooling process. There is a possibility to operate with only one ejector, the low (low ER EJ, 9) and high-pressure ratio (high ER EJ, 11) ejector block, or with both ejector blocks in series or parallel-connected. CO_2 EJ can achieve very high pressure lift due to large work recovery. Another feature of this rack is the usage of liquid ejector (liq EJ, 10) allowing the evaporators to stay flooded all year. The gas gets entrained by the auxiliary compressors (AUX, 4) or/and MT, passing the oil receiver (21) for cooling down the oil stored there, and suction liquid accumulator (ACC, 19). A Middle Pressure Receiver (MPR, 13) is installed at the lowest temp and pressure point to keep the costs low. It adjusts the high-pressure level by removing gas when the ambient temp increases and is a liquid storage in case of leakage. Parallel compression of MT and AUX compressors resulting in direct compression of flash gas from MPR instead of flashing vapor to the main compressors suction group. The high-pressure gas gets then cooled down by the pre-cooler (5), gas cooler (6) and sub-cooler. While cooling the gas cooler follows the constant pressure line at gliding temps. The specific heat capacity of the refrigerant is not constant during this process but changes significantly with temp. Based on this property the gas coolers (GC) are designed. Therefore, three GC for different applications such as preheating and heating of hot tap water as well as space heating can be integrated. The temp approach (T_a) is limited by the pinch point and is typically located on the outlet of the gas cooler, to get the minimum of T_a = 2-4K, means on low pressure side. Therefore, GC inlet pressure should be high resulting in high cooling capacity, low throttling losses and thus, high COP. The higher the GC outlet temp the higher the pressure, at T_{GC} =46°C the optimum p~105bar for the highest COP. Sub-cooling is an application for preheating the tap water as well as improving the COP. Before entering the ejector, another improvement is the installation of a heat exchanger. The passing glycol (G) takes the heat and transfers it to the individual evaporator. Two G loop circuits are arranged, one for MT, A/C load and another one for LT load with a different G concentration (42% and 56% respectively). The temperature of the G inlet temperature of each heat exchanger, regardless of the gas cooler, can be maintained by changing the opening degree of the 3-way valves (m-r). G is stored in the G tank (22). By changing the opening degree of the

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gate valves (a, b, d, e, k, l) manually or the 3-way-valves (c, g) electronically the CO2 mass flow of each heat exchanger can be maintained, for e.g. testing the efficiency of those heat exchangers and their different applications. A simplified flow diagram of the multifunctional test facility is shown in Figure 1 and a list of the components are listed in Table 1.

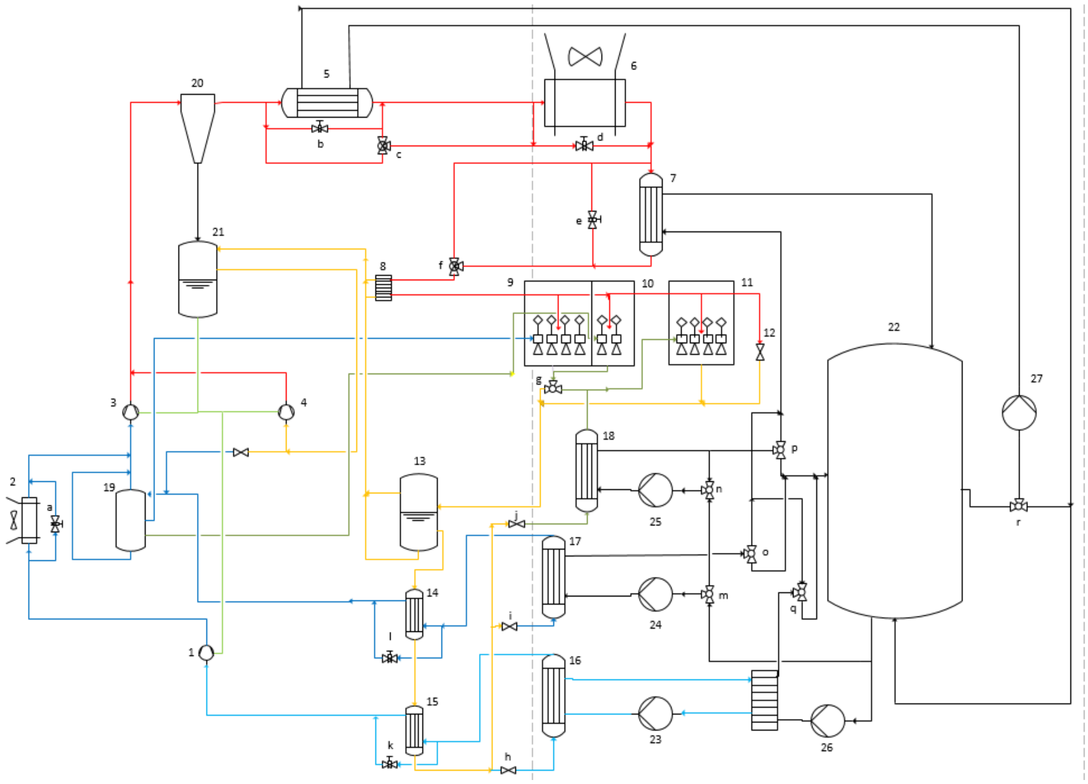

Figure 1 Flow diagram of INDEE container (simplified).

Table 1 Key of INDEE Main Components.

1	Low Temperature Compressor	15	Heat Exchanger Low Temperature
2	De-superheater, Fan	16	Low Temperature Evaporator
3	Medium Temperature Compressor	17	Medium Temperature Evaporator
4	Auxiliary Compressor (AUX)	18	Air Conditioning Evaporator
5	Pre-cooler	19	Liquid Suction Accumulator (ACC)
6	Gas Cooler, Fan	20	Oil Separator
7	Sub-cooler	21	Oil Receiver
8	Heat Exchanger	22	Glycol Tank
9	Low Pressure Ratio Ejector (low ER EJ)	23	LT Pump
10	Liquid Ejector (liq EJ)	24	MT Pump
11	High Pressure Ratio Ejector (high ER EJ)	25	A/C Pump
12	High-pressure Expansion Valve	26	Secondary Pump
13	Middle Pressure Receiver (MPR)	27	Heat Reclaim Pump
14	Heat Exchanger Medium Temperature	a, b, d, e, k, l	Gate Valves, Manually Operated
c, g, m-r	3-Way Valves, Electronically Operated	h-j	Throttling Valves, Electronically Operated

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3. EXPERIMENT

3.1 GOAL

The goal is to demonstrate the feasibility of R744 transcritical systems as a non-HFC based alternative to HCFC-22 in retail applications in countries with high ambient temp. Comparing the calculated exergy efficiencies (EEF) and power input ratios (PIR) of experimental work on the supermarket test facility running mode 4 for an ambient temp of 46°C with the theoretical data of a combined R22 A/C stand-alone, and R404A MT and LT stand-alone system.

3.2 SOUTH INDIAN CONDITION

South India has a tropical climate, means high humidity in the western part all year around and dry periods in the eastern part. In monsoon season, the minimum temp of $T_{amb} = 25^{\circ}C$ can be reached. In dry season the maximum temp of $T_{amb} = 50^{\circ}C$ can occur. The design temp therefore is $T_{amb} = 46^{\circ}C$. Based on this high ambient condition, the pressure of the CO2 in the REC can increase up to p=80bar limited by a controller.

3.3 LIMITATIONS

The glycol side is not representable since air for fans and water for heat exchanger (HX) will be used. Calculations are made based on assumptions, e.g. exergy is calculated by the utility temps of each HX outlet temp.
$T_{amb} = 46^{\circ}C$ must be forced and set by maintaining the GC outlet temp. LT evaporator is under-dimensioned influencing the efficiency and adjustment of the LT load.

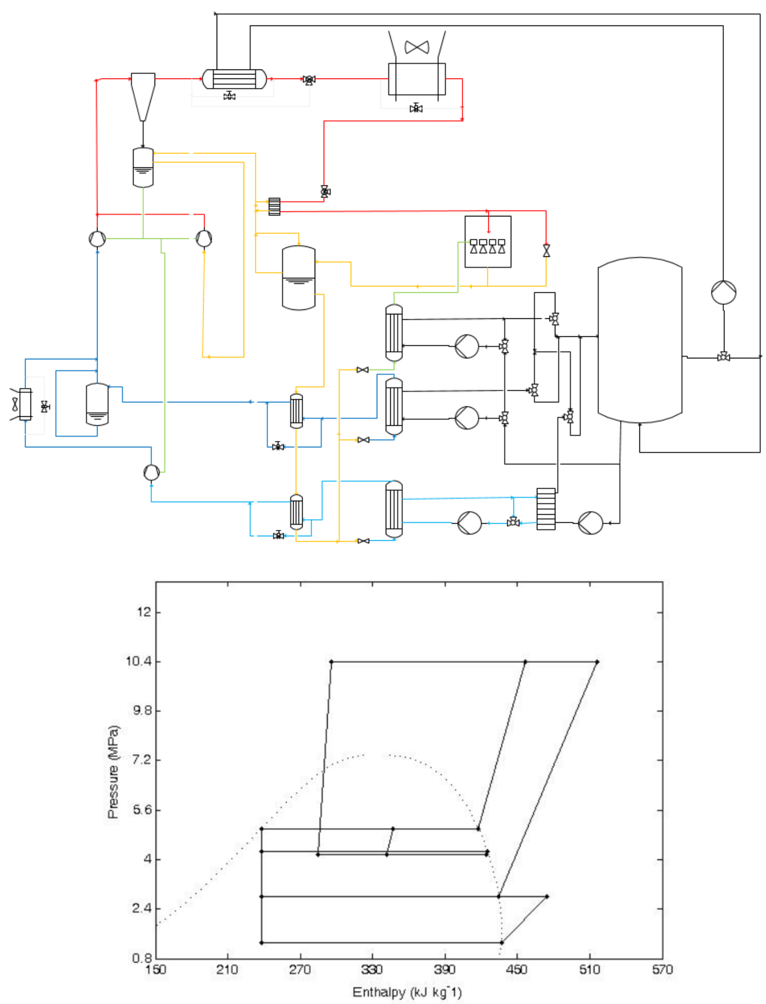

Figure 2 Simplified flow diagram of mode 4 (upper) and its p-h diagram (bottom).

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3.4 MODE 4

Based on R744 practical properties, it is a particularly suited working fluid for high ambient temp countries. For Southern India, a parallel compression configuration with ejectors for expansion work recovery is the most efficient mode. It can deliver freezing, refrigeration and air conditioning loads, and has a heat reclaim option.

In this mode, the sub-cooler is switched off, so the cooled high-pressure gas leaving the GC gets sucked and expanded by the high ER EJ. The 2-phase fluid then gets separated and maintained in MPR. The preheated and by throttling valves expanded CO2 liquid evaporates in LT, MT and A/C evaporator respectively. The saturated gas coming from LT and MT evaporator get sub-cooled to avoid superheat before entering MT and LT compressor. The low-pressure gas gets sucked by the MT compressor passing a fan for further pre-cooling. The saturated and pre-cooled gas of MT evaporator gets collected in the ACC and the saturated vapor of A/C evaporator gets sucked by the high ER EJ and utilized to expand the high-pressure gas. The vapor stored in the MPR is utilized to cool down the oil in the oil receiver and then gets sucked by the AUX compressor. In Figure 2, there is a simplified flow diagram of mode 4 as well as a p-h diagram.

3.4.1 SET-POINT

The ambient temp is set by the gas cooler outlet temp $T_{GC} = 46°C$ and kept constant. The MPR pressure is set within a range of $p_{REC} = 44 - 52$ bar by changing the suction temp of the AC evaporator. For each MPR pressure the AC evaporation temp is adjusted, $T_{ev} = 8 - 12°C$ by maintaining the glycol inlet temp. The evaporation temp of the MT and LT evaporators are kept constant at -6°C and -29°C respectively. These values can be maintained by the G inlet temp of each evaporator. The glycol outlet temps are kept constant. In Table 2, the input parameters are shown.

The test durations are 30 minutes starting after steady state condition is reached. Steady state condition means the fluctuation of evaporation temps at their set-points is minimal, the opening degree (OD) of the 3-way valve in front of the AC evaporator is about 95-99% and the superheat (SH) is beneath 5°C. Furthermore, the MPR pressure and GC outlet temp show the same value as initially set.

At the end of one test period all the necessary data get extracted and evaluated. Following parameters, shown in Table 3, are needed to calculate the exergy efficiency (EFF) and Power Input Ratio (PIR) (refer to Section 4).

Table 2 Input Parameters.

$T_{amb} = 46°C$ $p_{MPR} = 44 - 52$ bar $T_{e,AC} = 8 - 12°C$

Heat Exchanger:	Evaporation Temps	Glycol Inlet Temp [°C]:	Glycol Outlet Temp [°C]
MT	[°C]:	5-11	1
LT	-4, -6, -8	2-6	0
AC	-27, -29, -31	40-29	28
HR	8 - 12	40	70

Table 3 Extracted parameters of the supermarket test facility running mode 4.

G side:	GC outlet temp (= ambient temp)	Evaporator Volumes:	Compressor Power:	A/C evaporation pressure
G inlet temp MT	MT Evaporation pressure			
G outlet temp MT		AC	AUX	
G inlet temp LT		MT	MT	
G outlet temp LT		LT	LT	

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G inlet temp AC
G outlet temp AC
G inlet temp HR
G outlet temp HR

HR (Heat Reclaim)

4 CALCULATION

To compare the CO_2 test facility and R22 refrigeration plants in India supermarkets, EFF need to be calculated. EEF, or 2nd Law, is a ratio of the exergy output divided by the exergy input. It takes in account the quality of energy and is therefore more accurate than the “First Law of Thermodynamics” equation. It is shown in Eq. (1).

$$\text{EFF}=\frac{B_{MT}+B_{LT}+B_{AC}+B_{HR}}{P_{MT,}+P_{LT,}+P_{AUX}} \tag{1}$$

The exchanged exergy for Heat pumps such as HR is

$$B_i = \dot{Q}_i * (1-\frac{T_{amb}}{T_{ut,i}}) \tag{2}$$

And for Refrigeration such as MT, LT and AC

$$B_i = \dot{Q}_i * (\frac{T_{amb}}{T_{ut,i}}-1) \tag{3}$$

Where $T_{ut,i}$ is the temp of the heat source in K, and T_{amb} is the temp of the surrounding in K. $\dot{Q}_i$ is the cooling capacity:

$$\dot{Q}_i = \rho_i * V_i * \Delta h_i \tag{4}$$

The densities ρ_j, in kg/m^3, are calculated by using Coolpack and the glycol outlet temp of the individual evaporators. The enthalpy differences Δh_i, in kJ/kg, are based on average specific heat capacity of the water-glycol solution for a given temp range. The values for each volume V_j, in m^3, are measured by energy meters, i.e. ultrasound volumetric flow meters. Then the average of $\dot{Q}_j$is taken using Eq. (5) to calculate the average exergy B_i by using Eq. (3) of each heat exchanger.

The sum of each exergy exchange divided by the sum of all extracted compressor power consumption averages using Eq. (1) resulting then in EEF.

Introducing the reciprocal of exergy efficiency, Power Input Ratio *(PIR):*

$$PIR = \frac{P_{MT}+P_{LT}+P_{AC}}{P_{MT,c}+P_{LT,c}+P_{AC,c}} \tag{5}$$

$$P_{i,c} = B_i \tag{6}$$

The “Carnot” power consumption is calculated by the average cooling capacity $\dot{Q}_j$, the constant utilization temperature T_{ut} of the evaporators and the average ambient temp T_{amb}.

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5 RESULTS

Adjusting a MPR pressure to a specific set-point the GC outlet temp as well as the A/C evaporation temp change accordingly. Increasing the MPR pressure to p=52 bar the GC outlet temp increases to $T_{GC} = 51.4^{o}C$ and the A/C evaporation temp stays at $T_{e,AC} = 12^{o}C$, in total independency of the glycol inlet temp of the A/C evaporator. In Table 3, the influence of the MPR pressure to the ambient and AC evaporation temp is shown. Adjusting the A/C load by maintaining the glycol inlet temp only effects the SH and therefore the OD. The higher the temp difference, the bigger the SH, thus the OD. For $T_{amb} = 46^{o}C$, the load of MT and LT evaporator needs to be maintained by the glycol inlet temps once, independently of the MPR pressure. The set-points are shown in Table 3.

The PIR decreases from 3.71 to 2.59, and the EFF increases from 0.27 to 0.39 with increasing MPR pressure of p10T6 and p16T12 respectively. Signatures from p10 to p16 denote the MPR pressure of 46bar to 52bar expressed as temp in degree Celsius. Regarding to these results, for an ambient temp of 51°C, a MRP pressure of 52 bar and A/C evaporation temperature of 12°C are the most efficient set-points.

In Table 3 for each setting the compressor power demands and refrigeration capacities are listed. It is shown that the LT and MT power demand stay nearly constant as the AUX power demand decreases significantly. The AC / HR cooling capacities decrease significantly, LT decreases slowly, and MT cooling capacity stay nearly constant.

Table 3 Results of the first test campaign running mode 4 for various pressures

	R744 - p10T6	R744 - p12T8	R744 - p14T10	R744 - p16T12
$T_{amb,act}$	43.5	46.7	49.1	51.4
p_{MPR} [bar]	46	48	50	52
$T_{ev,AC}$ [°C]	6	8	10	12
$T_{ev,MT}$ [°C]	-6	-6	-6	-6
$T_{ev,LT}$ [°C]	-29	-29	-29	-29
P_{AC} [W]	17000	14700	13500	12200
P_{LT} [W]	1100	1100	1100	1100
P_{MT} [W]	6000	6100	6200	6100
$\dot{Q}_{AC}$ [W]	52800	53000	51100	46500
$\dot{Q}_{LT}$ [W]	4300	4300	4100	4000
$\dot{Q}_{MT}$ [W]	10400	10500	10500	10300
$\dot{Q}_{HR}$ [W]	76400	77200	75200	68100
η_{ex} (EFF)	**0.270**	**0.327**	**0.360**	**0.387**
PIR	**3.709**	**3.058**	**2.585**	**2.585**

6 CONCLUSION AND FUTURE WORK

Regarding the first test results the PIR of 2.59 – 3.71 decrease with increasing MPR pressure and therefore, increasing ambient temp. Table 3 shows a good EFF / PIR for high ambient temps resulting in the feasibility of integrated CO_2 refrigeration facilities running a parallel compression configuration with ejectors for expansion work recovery in high ambient temp regions.

In this paper only the calculated EEF and PIR of mode 4 for four different MPR pressures, 46-52bar, are shown and compared.

The next step is running test for several high ambient temps $T_{amb} = 36 - 46^{o}C$, following the same procedure as explained above, to achieve the most comparable results. Followed by the comparison with nearby supermarket refrigeration and air conditioning installation as, so far, only theoretical results exist. The CO_2 refrigeration test

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facility is a preparation and demonstration site for a full-scale replacement of existing commercial refrigeration installations in India (Phase II).

NOMENCLATURE

HCFC	Hydrochlorofluorocarbons	T_{amb}	Ambient Temperature
HFCs	Hydrofluorocarbons	T_a	Temperature Approach
CFCs	Chlorofluorocarbons	T_{ut}	Utilization Temperature
CO_2	Carbon dioxide	T_e	Evaporation Temperature
ODP	Ozone Depletion Potential	p_{REC}	Receiver pressure
GWP	Global Warming Potential	p_{GC}	Gas Cooler Pressure
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers	p_e	Evaporation Pressure
INDEE	Energy-efficient and environmentally friendly refrigeration and air conditioning for supermarkets in India	P_j	Power Consumption
Temp	Temperature	$P_{j,c}$	Carnot Power Consumption
MT	Medium Temperature Compressor	$\dot{Q}_j$	Cooling Capacity
LT	Low Temperature Compressor	B_i	Exergy Exchange
A/C	Air Conditioning	ρ_j	Density
		V_j	Volume
COP	Coefficient of Performance	Δh_j	Enthalpy Difference
PIR	Power Input Ratio	τ	Time
MPR	Middle Pressure Receiver		

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