

Performance Evaluation of Ejector Based CO_2 System for Simultaneous Heating and Cooling Application in an Indian Dairy Industry

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ABSTRACT

India is ranked first in milk production, and its goal is to increase annual production to 300 million tons by 2024. The dairy industry requires multi-temperature cooling and simultaneous heating. In the present study, the ejector based transcritical CO_2 system with two evaporators (LT at -10°C, MT at 0°C) is analyzed and proposed for fulfilling both cooling and heating demands in the dairy industry. Ejector based systems with and without internal heat exchanger are analyzed and compared. The effects of gas cooler outlet temperature, receiver pressure and gas cooler pressure on the performance of the system are investigated by energy and exergy analysis. The gas cooler pressure is analyzed for achieving the desired pasteurization temperature. The internal heat exchanger not only reduces the minimum gas cooler pressure required for the pasteurization temperature but also increases the system COP and exergy efficiency by 6.4% and 4.5% respectively. Second law analysis shows that the maximum exergy efficiency of the system is 38.4%.

Keywords: Trans-critical; Internal heat exchanger; Pasteurization; Exergy analysis; pressure optimization.

1. Introduction

Carbon dioxide was a widespread refrigerant in the early 18th century, which was used in the form of dry ice but was phased out with the invention of synthetic refrigerants. In 1985 Roland and Molina found that the use of halocarbon refrigerants causes the destruction of the ozone layer [1]. This, together with global warming, led to the search for alternate refrigerants. CO_2 being a natural refrigerant with zero ODP and GWP of 1, is conceived to be the best alternate refrigerant, especially for heat pump applications where simultaneous heating and cooling loads are in demand [2]. The operational pressure of CO_2 is higher than that of conventional refrigerants, making it a highly challenging refrigerant. However, 20% of energy saving is achieved for the same heating capacity and temperature requirement when compared with R12 [3].

At high ambient temperature CO_2 system works in the transcritical cycle. Therefore the maximum irreversibility occurs in throttling the gas. To minimize the loss during the expansion process and to utilize the available energy, researchers have proposed the use of ejectors. It is a simple device without any moving parts. A recent experimental study on a multi ejector pack reported a maximum work recovery efficiency of 36% [4]. Thermodynamic analysis of the CO_2 transcritical cycle [5] showed that the use of an ejector enhances the COP by 16%. Deng et al. [6] reported that the maximum cooling COP for the ejector expansion system is 8.2% and 11.5% higher than that of the internal heat exchanger cycle and conventional refrigeration cycle respectively. Fangtian and Yitai [7] reported that the ejector improves the COP by 30% and reduces the exergy loss by 25% when compared with the expansion valve.

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Nomenclature

AUX	auxiliary (-)
CIP	cleaning in process (-)
COP	coefficient of performance (-)
C_{pg}	specific heat at constant pressure of glycol(kJ/kgK)
GWP	global warming potential (-)
h	specific enthalpy (kJ/ kg)
HREX	heat recovery heat exchanger
I	exergy loss rate (kW)
IHX	internal heat exchanger (-)
LT	low temperature (-)
m	mass flow rate (kg/s)
MT	medium temperature (-)
ODP	ozone depletion potential (-)
P	pressure (bar)
Q	heat transfer rate (kW)
s	entropy (kJ/kgK)
T	temperature (°C)
W	work (kW)

Greek symbols

η	efficiency (%)
μ	entrainment ratio
ε	effectiveness

Subscripts

0	reference environment
4m	state point after heat recovery heat exchanger
4i, 16i	state point after the internal heat exchanger
c	cooling
com	compressor
dis	discharge
ej	ejector
ev	expansion valve
evap	evaporator
evapr	evaporator refrigerated medium
ex	exergy
gc	gas cooler
hr	heat recovery
i	i^{th} state, i = 1,2,3…….17
in	inlet
MN	motive nozzle
p	primary
s	suction
SN	suction nozzle
sys	system
tot	total
x, z	state point at outlet motive and suction nozzle in ejector efficiency

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The temperature glide available in the gas cooler when CO_2 system is operated in transcritical mode can be utilized in various industries like food processing, pulp and paper industry, chemical and petrochemical industry. The CO_2 system can be used for water heating applications, where the temperature required is up to 100°C [8]. Sawallah [9] presented the optimization of gas cooler pressure for heat recovery from CO_2 system for supermarket applications in Sweden, for ambient temperatures ranging from -5°C to 40°C.

CO_2 transcritical heat pump system can be utilized in food and beverage industries like dairy, sugar refineries, breweries, grain drying and canning units [8]. The use of a CO_2 heat pump system for milk processing showed the primary energy saving of 35% when compared with ammonia and CO_2 heat pump [10]. Also, the thermodynamic analysis of the CO_2 ejector system with an internal heat exchanger between two gas coolers showed an increased performance when compared to the system without an internal heat exchanger [11].

CO_2 systems are better for multi evaporator temperature in food retail applications like supermarkets. Supermarkets require refrigeration at different temperatures like MT for storage and LT for freezing. The performance comparison among the CO_2 parallel compression booster system consumed 15% and 16.6% less energy compared to the CO_2 booster system and R717/CO_2 cascade system respectively [12]. The analysis of a multi ejector CO_2 system for supermarkets with multi evaporators (MT and LT) and heat recovery at three temperature levels, using Modelica showed an increase in the COP by 20% [13]. The transcritical CO_2 booster system is more energy efficient and has 44% lower TEWI values when compared with the R134a parallel systems for MT and LT loads [17]. The CO_2 transcritical system with heat recovery integrated into a supermarket with LT and MT cabinets shows 3.6 to 6.5% energy savings when compared with the baseline system of R134a/CO_2 cascade refrigeration system with R410A heat pump system for hot water production and space heating [18].

The dairy industry requires simultaneous heating and cooling for milk processing, CIP, milk storage, etc. A thermodynamic study on cascaded ammonia and CO_2 system (for simultaneous heating and cooling) for the dairy industry reported that the cost of energy saving is approximately 33.8%, and the reduction in CO_2 emissions is 45.7% with a payback period of 40 months [14]. The utilization of solar energy for the preheating of water to a boiler in the dairy industry shows the payback period of 46 months with fuel oil savings of 41 kg/day [19]. However, the literature survey implies that less work has been carried out with CO_2 heat pump systems in dairy applications.

The production and consumption of dairy products are increasing, and so also the energy consumption in this sector. Besides, the present conventional method of heating used for pasteurization is unfavorable in terms of fossil fuel consumption and CO_2 emissions. Hence, this study focuses on the energy and exergy analysis of the CO_2 transcritical heat pump with an ejector and internal heat exchanger, exploring its suitability for a dairy application. Two evaporators for the cooling requirement at two different temperatures (4°C for milk chilling and -2.7°C for butter storage) and heat recovery at 72°C for pasteurization are considered. The CO_2 heat pump system is analyzed based on the gas cooler pressure required for fulfilling the heating load for achieving pasteurization temperature. The effect of parameters like gas cooler outlet temperature, gas cooler pressure and receiver pressure on the performance of the system is discussed

2. The milk processing system in a dairy plant

The plant Sholinganallur Dairy, located in Chennai, Tamil Nadu, India, is considered as an example for the present study for milk pasteurization and butter storage. The dairy requires cooling at two temperatures; approximately 4°C for milk chilling and -2.7°C for butter storage. It also requires heating above 72°C for pasteurization. Presently Ammonia and Freon based

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refrigeration systems are used to fulfill refrigeration load (Q_{tot}) of 280 TR at 0°C and 20 TR at -10°C respectively, while the boiler is used for 11 hours a day to meet 680 kW of heating demand (Q_h) for. Fig. 1 shows the conventional heating and chilling system used in the dairy industry, in which the ammonia refrigeration system is used for chilling water, and the boiler is used for hot water production.

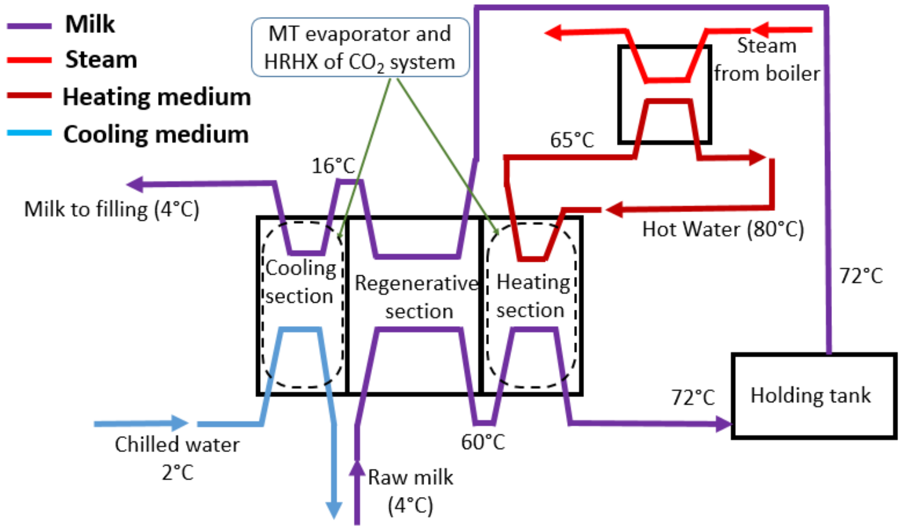

Fig. 1. Conventional heating and chilling system

3. Ejector based transcritical CO_2 system for a dairy industry

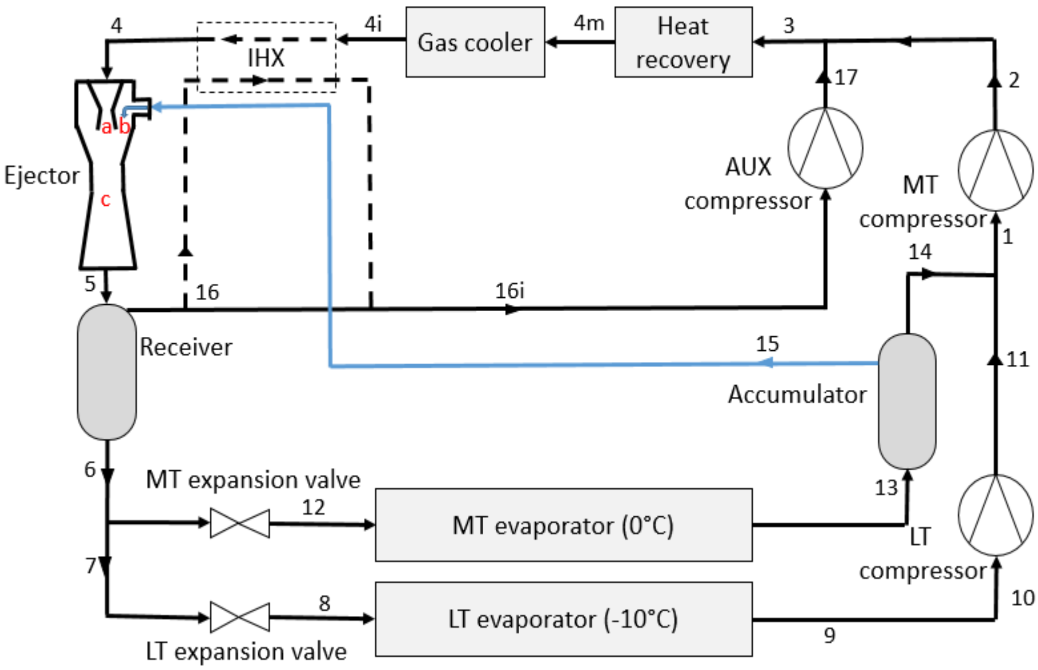

Fig. 2. Schematic of the transcritical CO_2 system

The schematic of the transcritical CO_2 system with and without IHX, and the corresponding cycle on the p-h diagram are shown in Figs. 2 and 3 respectively. The system consists of an ejector as an expansion device and two evaporators; medium temperature (MT) evaporator for milk chilling and low temperature (LT) evaporator for butter storage, and three compressors; LT compressor, MT compressor and an auxiliary compressor. LT compressor compresses CO_2 vapour from the LT evaporator pressure (10) to the accumulator pressure (11). The CO_2 vapour coming out from the MT evaporator (13) is collected in the accumulator. The total mass or fraction of the mass in the accumulator is used as the ejector suction (15) and the rest is supplied to the MT compressor (14). The vapour CO_2 from the accumulator and LT compressor (11) is compressed in the MT compressor. The high pressure and high temperature CO_2 gas at the discharge of MT (2) and auxiliary compressors (17) are fed to the heat recovery heat exchanger (3). The heat that is required for heating the milk is recovered in the heat recovery heat exchanger, and the excess heat is rejected to the ambient in the gas cooler. For the configuration with IHX, the CO_2 from the gas cooler outlet (4i) is allowed to cool lower than the gas cooler outlet temperature in the internal heat exchanger. This CO_2 from the gas cooler outlet is cooled with the help of vapour CO_2 stream from the receiver (16) and expanded in the ejector. For the configuration without IHX, the CO_2 after gas cooler (4) is expanded in the motive nozzle of the ejector to receiver pressure (5). At the same time, CO_2 from the MT evaporator (15) is expanded in the suction nozzle along with the motive flow. The two streams coming out from the motive nozzle (a) and suction nozzle (b) are mixed in the mixing section (c) at constant pressure. The mixed stream (c) regains the pressure in the diffuser to an intermediate pressure at the exit of the ejector (5). It is in a two-phase region and is separated in the phase separator. The vapour CO_2 (16) flows through the internal heat exchanger (16i). It is compressed in the AUX compressor (17). The liquid CO_2 (6) is expanded to MT and LT evaporators through expansion valves for cooling requirements of milk chilling (12) and butter storage (8) respectively.

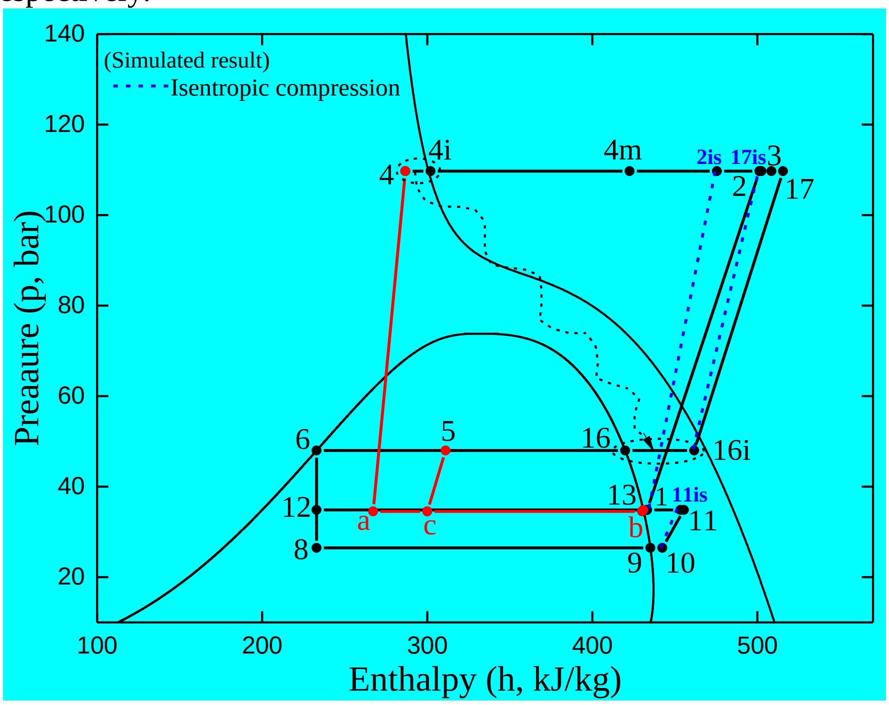

Fig. 3. *ph* plot of ejector based transcritical CO_2 system.

3.1. Experimental test setup for validation of a model

Fig. 4 shows the instrumented CO_2 test facility used for validation of the simulation results. The test facility is equipped with two shell and tube evaporators; MT (0°C, 10 kW) and LT (-10°C, 3 kW) evaporators, heat recovery shell and tube heat exchanger, gas cooler (copper tube and aluminum finned, air-cooled cross-flow heat exchanger) and three semi-hermetically

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sealed reciprocating compressors. The speed of the compressors is varied by three invertors according to the cooling loads in evaporators by feedback control.

Table 1. List of instruments used in the experimental setup and their specifications

S. No.	**Sensor**	**Company**	**Model**	**Type**	**Range**	**Accuracy** (% of measured value)
1.	Temperature	Danfoss	AKS 21M	B	-50 to 200°C	0.4
2.	Pressure	Danfoss	AKS 2050	----	up to 150 bar	0.5
3.	Energy meter	ISOIL	PT 500	IFX-M	3.0 $m^3 h^{-1}$	0.5
4.	Power meter	Danfoss	VFD	----	35-60 Hz	0.8

The glycol solution is used as a medium to load the evaporators and heat recovery heat, exchanger. The controlled heat transfer is allowed to take place from the latter to the former. The excess heat from the CO_2 after heat recovery is rejected in the gas cooler. The parameters viz., gas cooler outlet temperature, receiver pressure, MT and LT evaporator temperatures are adjusted in the control panel. Temperature and pressure readings at different locations are directly obtained from the control panel with a PT1000 temperature sensor and Danfoss pressure transmitter. The heating and cooling loads are obtained by measuring the flow rates and temperature of glycol across the heat recovery heat exchanger, MT and LT evaporators. The power input to the compressors is stored in the control panel. The instruments used for measuring parameters and their specifications are listed in Table 1.

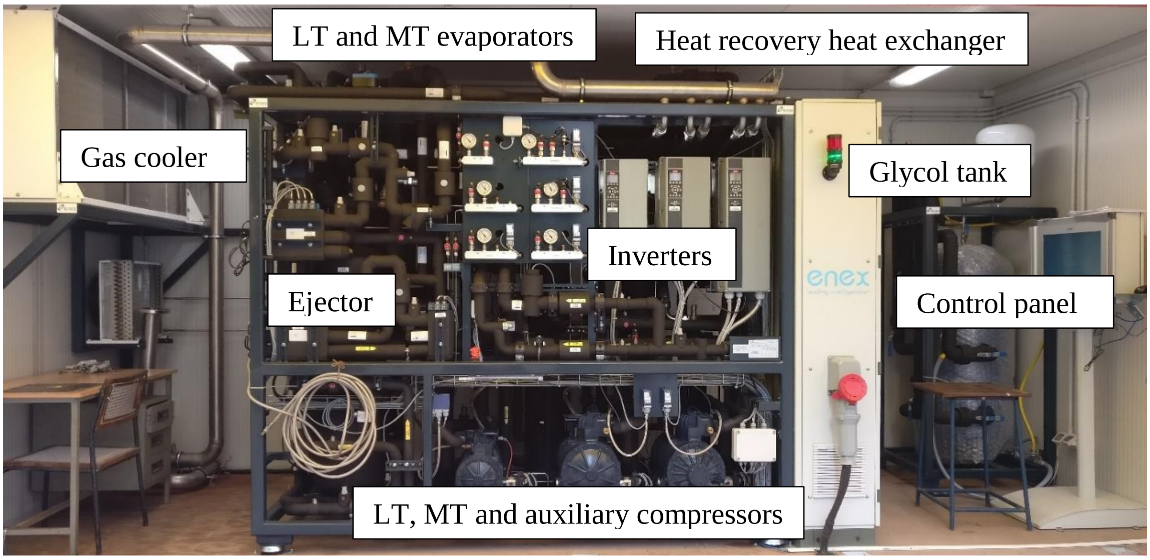

Fig. 4. Experimental test setup of a transcritical ejector CO_2 system.

The system is allowed to run for 1 hour to attain steady-state conditions. During this time, the readings for temperature and pressure at different locations and glycol flow rates through evaporators and heat recovery heat exchangers are stored in the control panel for every 10 seconds of operation. Steady-state readings for 30 minutes period are taken and averaged for the performance calculation of the system.

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The data stored in the control panel includes mass flow rates of glycol through MT and LT evaporators and heat recovery heat exchanger, compressors power consumption, temperatures of glycol at entry and exit of evaporators and heat recovery and pressure and temperature readings of CO_2 at various stages. The cooling and heating loads and COPs are calculated as follows

$$Q_{MT,evap} = m_{gly,MT,evap}\left(C_{pg,MT,in}T_{gly,MT,evap,in} - C_{pg,MT,out}T_{gly,MT,evap,out}\right) \quad (1)$$

$$Q_{LT,evap} = m_{gly,LT,evap}\left(C_{pg,LT,in}T_{gly,LT,evap,in} - C_{pg,LT,out}T_{gly,LT,evap,out}\right) \quad (2)$$

$$Q_{hr} = m_{gly,hr}\left(C_{pg,hr,out}T_{gly,hr,out} - C_{pg,hr,in}T_{gly,hr,in}\right) \quad (3)$$

The total cooling load, cooling and heating COPs and the total power consumed by the compressors of the system are

$$Q_{tot} = Q_{MT,evap} + Q_{LT,evap} \quad (4)$$
$$W_{tot} = W_{MT,com} + W_{AUX,com} + W_{LT,com} \quad (5)$$

Where $W_{MT,com}$, $W_{AUX,com}$ and $W_{LT,com}$ are the individual power consumptions of MT, AUX and LT compressors respectively. These are directly stored in the control panel.

$$COP_c = \frac{Q_{tot}}{W_{tot}} \quad (6)$$
$$COP_h = \frac{Q_{hr}}{W_{tot}} \quad (7)$$
$$COP_{sys} = \frac{Q_{tot}+Q_{hr}}{W_{tot}} \quad (8)$$

Where COP_c, COP_h and COP_{sys} are the cooling, heating and system COPs respectively, which are obtained experimentally and used for validation of simulation results with experimental results.

3.2. CO_2 system modeling and performance evaluation

Assumptions:

- One dimensional steady-state model.
- Pressure drop in gas cooler, heat recovery heat exchanger, receiver, accumulator and evaporators are negligibly small.
- Refrigerant (CO_2) exits MT and LT evaporators at a saturated vapour state.
- Refrigerant enters into HREX after mixing from the exits of MT and AUX compressors
- Variable parameters – gas cooler outlet temperature (T_4), receiver pressure (P_{rec}), gas cooler pressure (P_{gc}), IHX effectiveness
- Constant parameters – ejector efficiency, MT and LT evaporator capacities and temperatures.

Thermodynamic modeling of an ejector is considered by assuming its efficiency to be constant throughout the analysis. The ejector consists of the motive nozzle (primary nozzle), the suction nozzle, the mixing section and the diffuser. The refrigerant (CO_2) at high pressure ($P_{MN,in}$ or P_{gc}) from the gas cooler outlet (state 4 in Fig. 5) and the low pressure ($P_{SN,in}$ or $P_{MT,evap}$) refrigerant from the accumulator (state 15 in Fig. 5) are supplied to motive and suction nozzles respectively. These mix in the mixing section and get compressed to an intermediate

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pressure ($P_{DIF,out}$ or P_{rec}) in the diffuser. The ejector efficiency for the present study is defined as the ratio of work recovered in the ejector to the maximum work available in the motive nozzle [15]. The latter is the expansion work from the gas cooler pressure to the receiver pressure. Similarly, the former is the compression of suction fluid from the MT evaporator pressure to the receiver pressure. These expansion and compression processes are shown in Fig. 5.

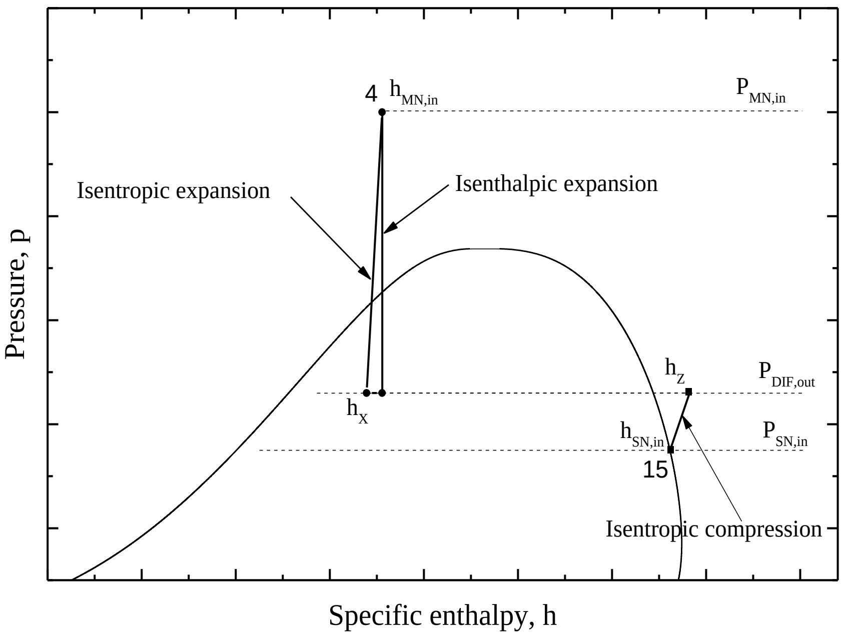

Fig. 5. Expansion of motive fluid in a two-phase ejector and compression of suction fluid.

Thus ejector efficiency is given by,

$$\eta_{ej} = \mu \, . \, \frac{h_Z - h_{SN,in}}{h_{MN,in} - h_X} \tag{9}$$

$$\mu = \frac{m_s}{m_p} \tag{10}$$

Where μ is the entrainment ratio of the ejector, defined as the ratio of the mass flow rate through the suction nozzle (m_s) to the mass flow rate through the motive nozzle (m_p). The ejector efficiency is considered as 0.25 [4]. The motive and suction mass flow rates (m_p and m_s) are unknowns in Eq. 9. The enthalpies h_X and h_Z are found by assuming the receiver pressure ($P_{DIF,out}$ or P_{rec}, variable parameter). h_X is the function of receiver pressure and entropy at state 4, and h_Z is the function of receiver pressure and entropy at state 15 (Fig. 5).

The mass flow rate through the evaporators required for the evaporator capacities are evaluated as

$$m_{MT,evap} = \frac{Q_{MT,evap}}{h_{13} - h_{12}} \tag{11}$$

$$m_{LT,evap} = \frac{Q_{LT,evap}}{h_9 - h_8} \tag{12}$$

Where $m_{MT,evap}$ and $m_{LT,evap}$ are the mass flow rates of refrigerant through MT and LT evaporators respectively. The enthalpies h_{12} and h_8 are equal to h_6 obtained by the initial assumption of the receiver pressure and saturated liquid state (state 6). $Q_{MT,evap}$ and $Q_{LT,evap}$ are the cooling capacities in MT and LT evaporators respectively.

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The mass flow rates through the ejector motive nozzle (m_p), suction nozzle (m_s) and auxiliary compressor ($m_{AUX,com}$) are found by taking the control volume across the ejector and the phase separator (receiver) by mass and energy balance equations along with ejector efficiency expression (Eq. 9).

$$m_p + m_s = m_{MT,evap} + m_{LT,evap} + m_{AUX,com} \tag{13}$$

$$m_p h_4 + m_s h_{15} = m_{AUX,com} h_{16} + (m_{MT,evap} + m_{LT,evap}) h_6 \tag{14}$$

have Three unknowns, namely, mass flow rates through the suction nozzle (m_s), motive nozzle (m_p) and auxiliary compressor ($m_{AUX,com}$) are found by solving Eqs. 9, 13 and 14 simultaneously. Then we get the expressions for m_s and m_p are as follows

$$m_s = \frac{(m_{MT,evap}+m_{LT,evap})(h_6-h_{16})}{\frac{A}{\eta_{ej}}(h_4-h_{16})+h_{15}-h_{16}} \tag{15}$$

$$m_p = \frac{(m_{MT,evap}+m_{LT,evap})(h_6-h_{16})}{\frac{A}{\eta_{ej}}(h_4-h_{16})+h_{15}-h_{16}} \cdot \frac{A}{\eta_{ej}} \tag{16}$$

Where $A = \frac{h_Z - h_{SN,in}}{h_{MN,in} - h_X}$ taken in Eq. 9.

$m_{AUX,com}$ is obtained by substituting m_s and m_p in Eq. 13 and the mass flow rate through the MT compressor is expressed as

$$m_{MT,com} = m_{MT,evap} + m_{LT,evap} - m_s \tag{17}$$

The temperatures of exit streams from internal heat exchanger are calculated by considering the effectiveness of IHX and energy balance equations expressed as follows

$$\mathcal{E}_{IHX} = \frac{h_{16i}-h_{16}}{h_{4i}-h_{16}} \tag{18}$$

$$m_p(h_{4i} - h_4) = m_{AUX,com}(h_{16i} - h_{16}) \tag{19}$$

Isentropic efficiency of the compressors is considered as a correlation defined as [16].

$$\eta_{com} = 1.003 - 0.121\left(\frac{P_{dis}}{P_s}\right) \tag{20}$$

The superheat at the inlet to the LT compressor is assumed to be 5 degrees. Based on the isentropic efficiency presented in Eq. 17 the exit enthalpies (h_2, h_{17} and h_{11}) of the compressors are estimated.

After determining the mass flow rates and exit enthalpies of compressors the work done by the compressors are expressed as

$$W_{LT,com} = m_{LT,com}(h_{11} - h_{10}) \tag{21}$$

$$W_{MT,com} = m_{MT,com}(h_2 - h_1) \tag{22}$$

$$W_{AUX,com} = m_{AUX,com}(h_{17} - h_{16s}) \tag{23}$$

$W_{LT,com}$, $W_{MT,com}$ and $W_{AUX,com}$ are works of LT, MT and auxiliary compressors. The total work of the system is calculated as

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$$W_{tot} = W_{LT,com} + W_{MT,com} + W_{AUX,com} \quad (24)$$

$$Q_{tot} = Q_{LT,evap} + Q_{MT,evap} \quad (25)$$

$$Q_{hr} = m_p(h_3 - h_{4m}) \quad (26)$$

Q_{tot} is the sum of cooling capacities of LT and MT evaporators and Q_{hr} is the heating capacity in the heat recovery.

The performance of the system is portrayed by cooling COP, heating COP and system COP as follows.

$$COP_c = \frac{Q_{tot}}{W_{tot}} \quad (27)$$

$$COP_{hr} = \frac{Q_{hr}}{W_{tot}} \quad (28)$$

$$COP_{sys} = COP_c + COP_{hr} \quad (29)$$

Where COP_c, COP_{hr}, and COP_{sys}, are cooling COP, heating COP and system COP respectively.

3.2. Exergy analysis

Exergy analysis helps in pointing out the direction of the system improvement and evaluating the exergy loss in each component of the system. The exergy loss for each component has been analyzed by the following equations.

For LT compressor

$$I_{LT,com} = m_{LT,com} T_0 (s_{11} - s_{10}) \quad (30)$$

For MT compressor

$$I_{MT,com} = m_{MT,com} T_0 (s_2 - s_1) \quad (31)$$

For auxiliary compressor

$$I_{AUX,com} = m_{AUX,com} T_0 (s_{17} - s_{16i}) \quad (32)$$

For heat recovery heat exchanger

$$I_{hr} = m_p T_0 (s_{4m} - s_3) + Q_{hr}\left(\frac{T_0}{T_{4m}}\right) \quad (33)$$

For gas cooler

$$I_{gc} = m_p[(h_{4m} - h_4) - T_0(s_{4m} - s_3)] \quad (34)$$

For internal heat exchanger

$$I_{IHX} = m_{AUX,com}[(h_{16} - h_{16i}) - T_0(s_{16} - s_{16i})] + m_p[(h_{4i} - h_4) - T_0(s_{4i} - s_4) \quad (35)$$

For ejector

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$$I_{ej} = m_p T_0 (s_5 - s_4) + m_s T_0 (s_5 - s_{15}) \quad (36)$$

For MT expansion valve

$$I_{MT,ev} = m_{MT,evap} T_0 (s_{12} - s_6) \quad (37)$$

For LT expansion valve

$$I_{LT,ev} = m_{LT,evap} T_0 (s_8 - s_6) \quad (38)$$

For MT evaporator

$$I_{MT,evap} = m_{MT,evap} T_0 \left[\frac{h_{12} - h_{13}}{T_{MT,evapr}} - (s_{12} - s_{13})\right] \quad (39)$$

For LT expansion valve

$$I_{LT,evap} = m_{LT,evap} T_0 \left[\frac{h_8 - h_9}{T_{LT,evapr}} - (s_8 - s_9)\right] \quad (40)$$

The total exergy loss of the system is

$$I_{tot} = I_{LT,com} + I_{MT,com} + I_{AUX,com} + I_{hr} + I_{gc} + I_{IHX} + I_{ej} + I_{MT,ev} + I_{LT,ev} + I_{MT,evap} + I_{LT,evap} \quad (41)$$

The exergy efficiency of the cycle is

$$\eta_{ex} = 1 - \frac{I_{tot}}{W_{tot}} \quad (42)$$

Where T_0 is ambient temperature assumed as 30°C and $T_{LT,evapr}$ and $T_{MT,evapr}$ are the temperatures of the refrigerated medium in LT and MT evaporators respectively. The variable parameters considered for analysis of the present model are gas cooler pressure (P_{gc} or P_4), gas cooler outlet temperature (T_{4i}), receiver pressure (P_{rec} or P_5) and IHX effectiveness (ε_{IHX}).

The complete simulation of the model is carried out in MATLAB by interfacing with REFPROP as a property calculator at various state points.

4. Results and discussion

The thermodynamic model of the CO_2 system suitable for the dairy industry application is validated comparing with experimental results and the model results are discussed.

4.1. Model validation with experimental results

The input parameters and the basic assumptions of the thermodynamic model are listed in Table 2. The parameters considered for the model are based on the possible operating conditions in the experimental test setup. The results presented in Figs. 6 to 9 are based on the parameters listed in Table 2 for the system without IHX. The analysis has done with these parameters are mainly for the validation of the model. After validation, the same model is extended for the performance evaluation of the CO_2 system for dairy industry application with the data listed in Table 3.

Table 2 Input parameters considered for validation with experimental results.

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Parameter	Value
MT evaporator capacity, $Q_{MT,evap}$ (kW)	10
LT evaporator capacity, $Q_{LT,evap}$ (kW)	3
HREX capacity, Q_{hr} (kW)	13
MT evaporator temperature, $T_{MT,evap}$ (°C)	0
LT evaporator temperature , $T_{LT,evap}$ (°C)	-10

The experimental test setup is designed to operate at optimum gas cooler pressure in a transcritical state, where the pressure and temperature are independent parameters. To compare the simulation results with those of experimental, the gas cooler pressure has to be optimized for the model too. Fig. 6 shows the optimized gas cooler pressure at its various outlet temperatures for the maximum system COP. Further, the optimized gas cooler pressure is used for calculating the cooling and system COP of the thermodynamic model, which are validated with the experimental results.

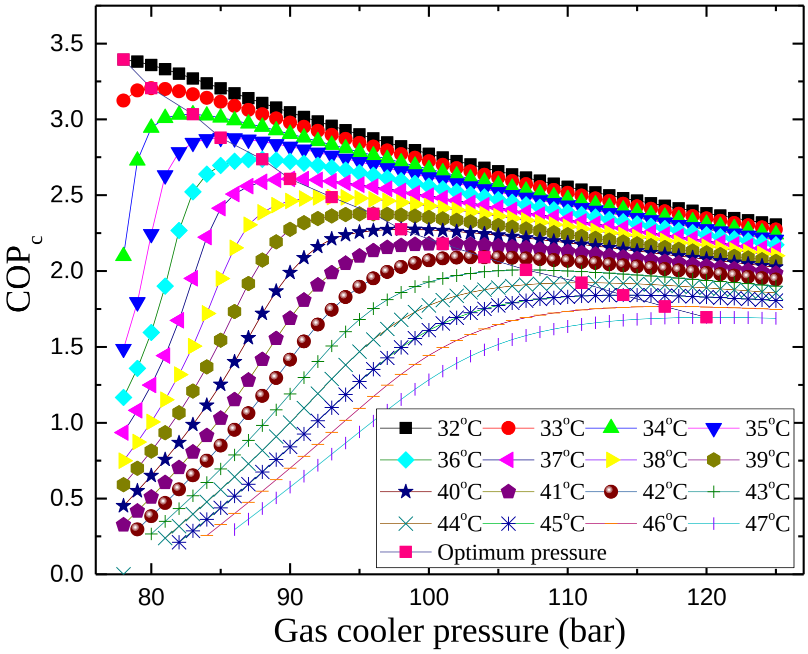

Fig. 6. Variation of COP with gas cooler pressure at various gas cooler outlet temperature without IHX.

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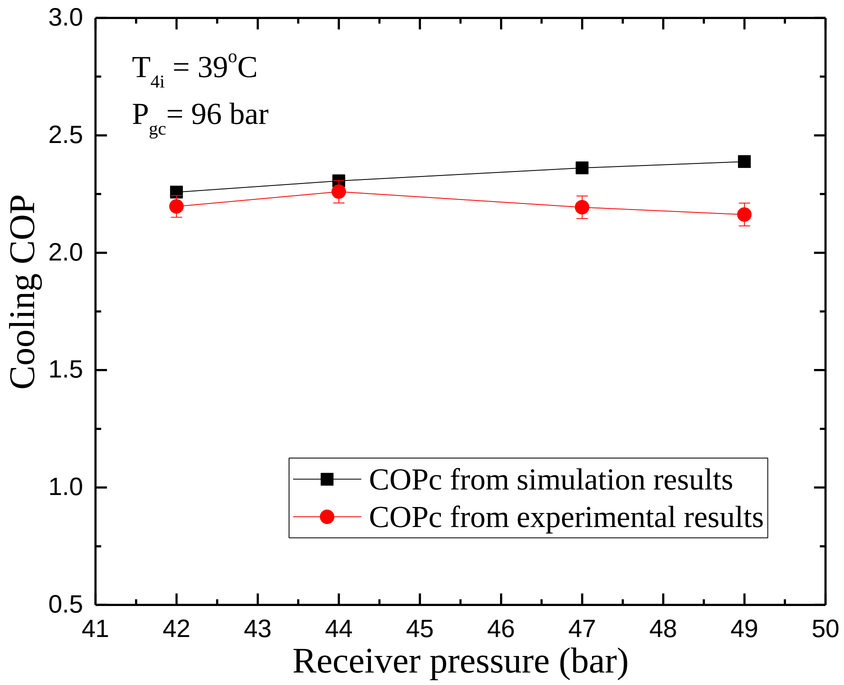

Fig. 7. Variation of cooling COP with receiver pressure.

The simulation results are validated against the present experimental results. Fig. 7 compares the variation of the two cooling COPs with receiver pressure at the gas cooler outlet temperature of 39°C. At this outlet temperature, the optimum pressure is 96 bar from Fig. 6, and the same is considered as the discharge pressure. The COP values match substantiating the model. The effect of receiver pressure on system COP is less. This is because the increase in receiver pressure reduces the pressure ratio of the auxiliary compressor and thereby its work. The increase in receiver pressure also increases the pressure lift of the ejector. This reduces the mass flow rate through the suction nozzle consequently pushing the remaining refrigerant to the MT compressor ($m_{MT,com}$) as shown in Fig. 8. Therefore, the decrease in auxiliary compressor work is compensated by the increase in MT compressor work which results in less variation of COP with receiver pressure. The simulation results are closely matching with the experimental results with a minimum and maximum COP error of 0.046 and 0.225 respectively. As the effect of receiver pressure on the performance of the system is less, therefore a value of 48 bar is assumed as receiver pressure for the performance evaluation of the thermodynamic model.

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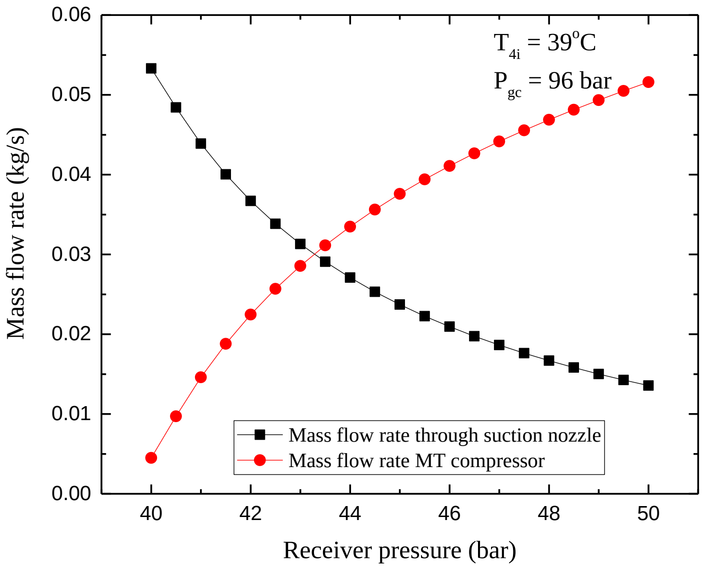

Fig. 8. Variation of ejector suction mass flow rate and mass flow rate through MT compressor with receiver pressure.

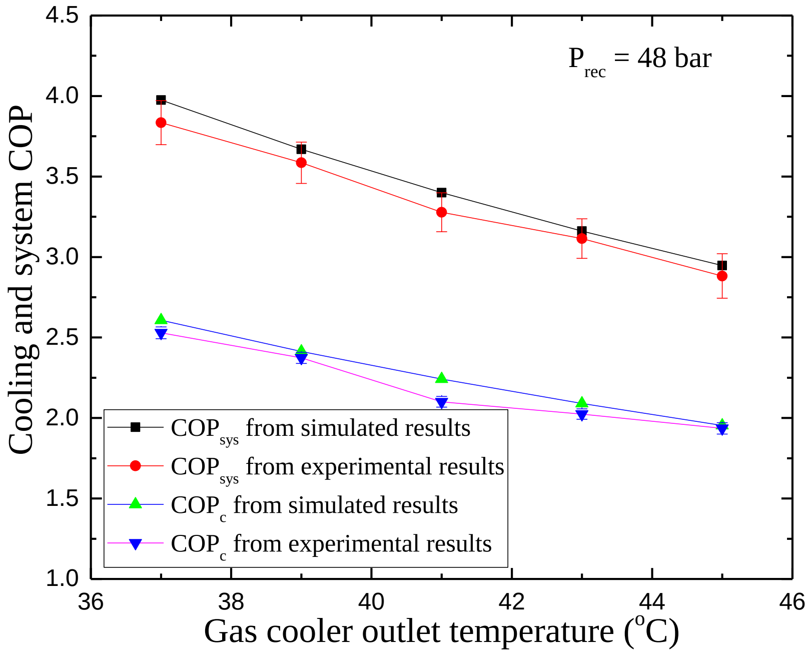

Fig. 9. Variation of cooling and system COPs with gas cooler outlet temperature.

Fig. 9 also validates the model by matching the variation of both cooling and system COPs of the two (model and experiment) with gas cooler outlet temperature. For every gas cooler outlet temperature (T_4), the optimum gas cooler (discharge) pressure is taken from Fig. 6. As the outlet temperature increases, optimum gas cooler pressure increases and both cooling and system COPs decrease. This is because the compressor power increases faster than the refrigeration capacity.

4.2. Performance evaluation of the proposed system

The input parameters considered for the performance evaluation listed are in Table 3. The parameters considered for the model are based on the requirement in the dairy industry application.

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Table 3 Input parameters considered for the performance evaluation.

Parameter	Value
MT evaporator load, $Q_{MT,evap}$ (kW)	980
LT evaporator load, $Q_{LT,evap}$ (kW)	70
HREX load, Q_{hr} (kW)	680
MT evaporator temperature, $T_{MT,evap}$ (°C)	0
LT evaporator temperature , $T_{LT,evap}$ (°C)	-10
HREX outlet temperature, T_{4m} (°C)	65
Chilled milk temperature in MT evaporator $T_{MT,evapr}$ (°C)	4
Butter storage temperature in LT evaporator, $T_{LT,evapr}$ (°C)	-2.7
Ambient temperature, T_0 (°C)	30

Fig. 10 shows the variation of the mass flow rate through the suction nozzle, and LT, MT and auxiliary compressors with gas cooler outlet temperature. As the gas cooler outlet temperature increases, the vapour quality (dryness fraction) at the ejector outlet increases. This reduces the liquid flow rate at the ejector outlet and therefore the cooling capacity. To fulfill the required cooling load in MT and LT evaporators, the mass flow rate at the ejector outlet should be higher. This necessitates more flow rate through the auxiliary compressor with an increase in gas cooler outlet temperature as shown in the figure.

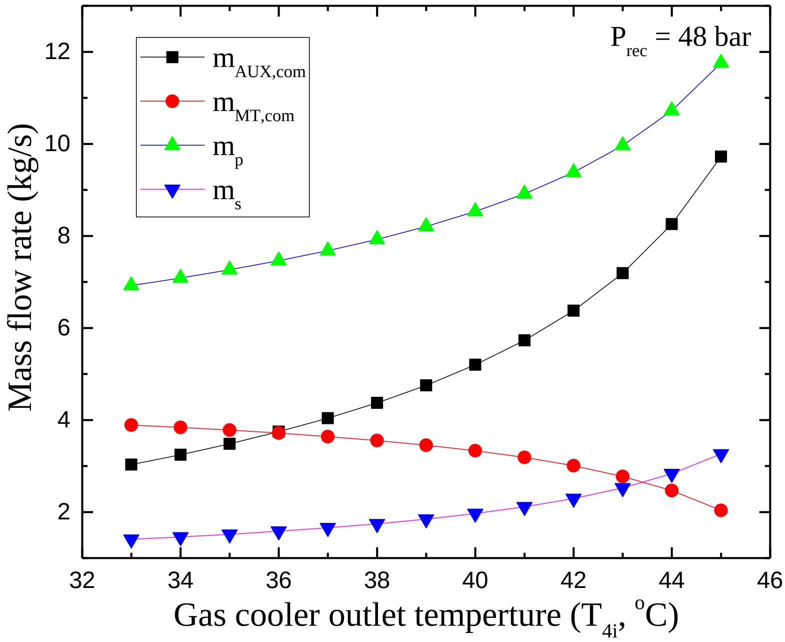

Fig. 10. Effect of gas cooler outlet temperature on mass flow rates through ejector suction nozzle, HREX and MT and AUX compressors.

With the increase in gas cooler outlet temperature, energy in the motive gas flow or the energy input to the ejector increases. This facilitates the ejector to draw more gas through the ejector suction nozzle and consequently the flow rate through MT compressor decreases. However, the total of these two increases leading to more flow rate through the heat recovery heat exchanger. Thus the heating capacity of the system increases with the outlet temperature of the gas cooler.

4.2.1. **Effect of IHX on gas cooler pressure and system COP**

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Milk is heated from 60 to 72°C in the heat recovery heat exchanger (Fig.1) of the CO_2 system for pasteurization. Considering 5°C approach, the minimum outlet temperature of CO_2 (T_{4m}) is 65°C. With limited gas flow in hand, thus the compressor discharge temperature should be high enough to meet the heating demand. Higher superheat or the compressor inlet temperature and higher pressure ratio or compressor discharge pressure are the two means to boost the discharge temperature. The former is accomplished by utilizing IHX.

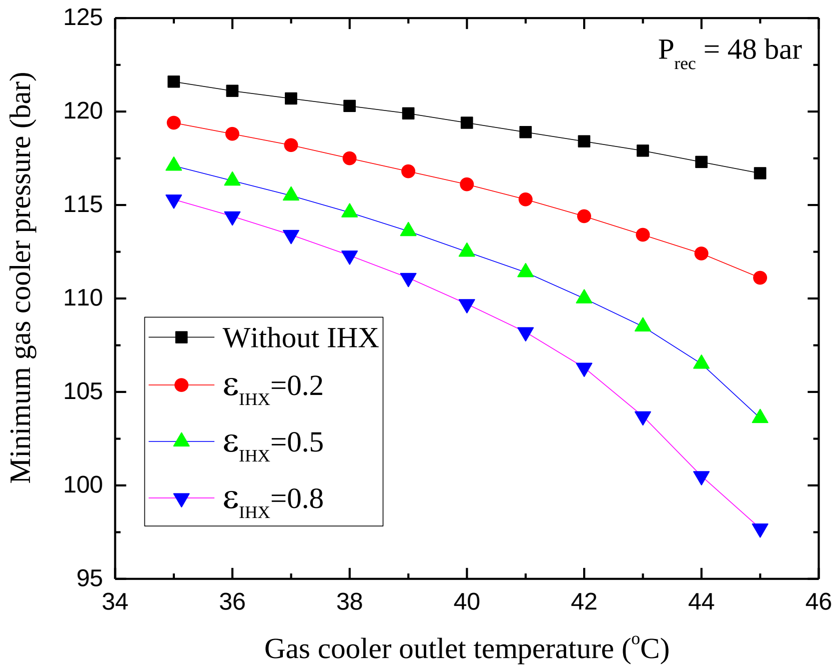

Fig. 11. Effect of gas cooler outlet temperature and IHX on minimum gas cooler pressure.

Fig. 11 shows the effect of gas cooler outlet temperature and effectiveness of IHX on the minimum gas cooler pressure, yet providing sufficient heat for milk pasteurization. This pressure decreases with an increase in the outlet temperature. As explained before in Fig.10, higher outlet temperature results in higher flow through the HREX for the same cooling and heating demands. The higher flow can deliver more heat, but to get only the required heat, gas cooler pressure is to be decreased as illustrated in the figure. The auxiliary compressor inlet temperature (T_{16i}) increases with an increase in the effectiveness of the internal heat exchanger. Therefore the required heating demand is achieved at lower pressures.

The influences of gas cooler outlet temperature and the effectiveness of IHX on the COP of the system are depicted in Fig. 12. IHX not only decreases the minimum gas cooler pressure but also decreases the gas cooler outlet temperature (subcooling), thereby increases the system COP. The maximum improvement of system COP is 6.4% with the addition of IHX (ε=0.8).

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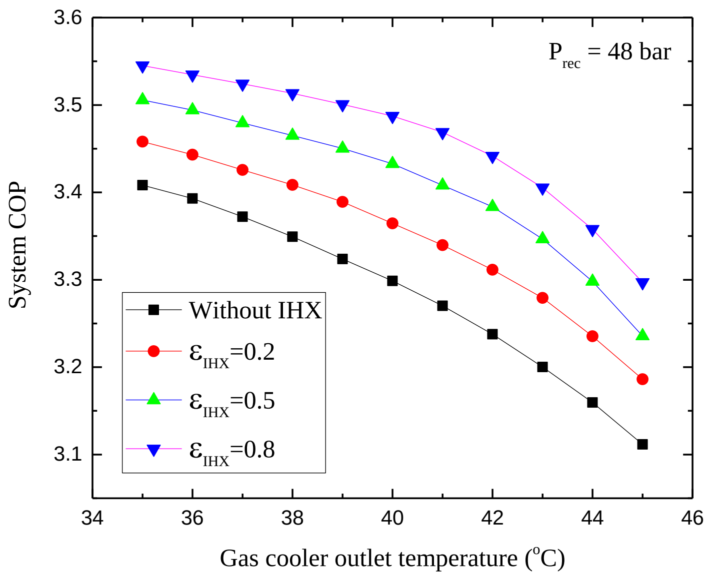

Fig. 12. Effect of gas cooler outlet temperature and IHX on COP of system.

4.3. Exergetic performance of the system

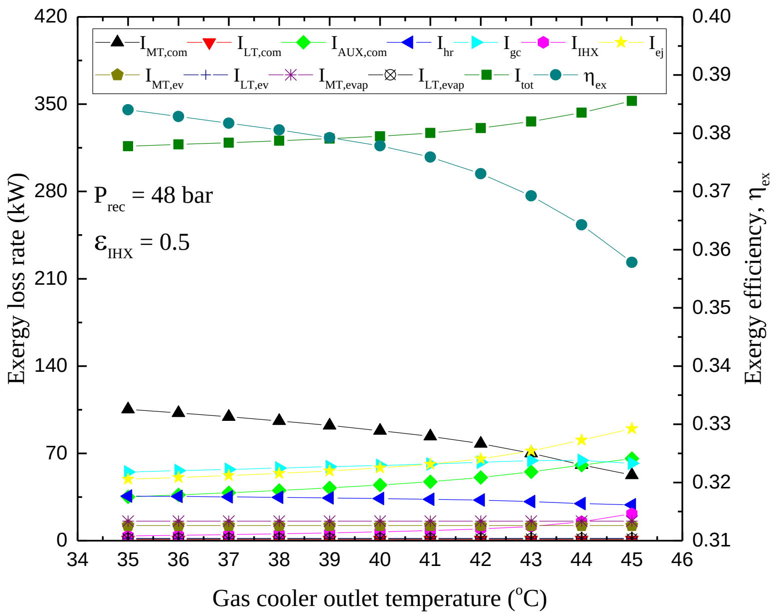

Fig. 13. Effect of gas cooler outlet temperature on exergetic performance of the system

Fig. 13 shows the effect of gas cooler outlet temperature on the exergetic performance of the system. The exergy losses in evaporators and expansion valves are minimum amongst all the components. This is because of a small temperature difference of only 4°C between the refrigerant and the milk to be cooled in MT evaporator, low cooling load (low refrigerant flow) in LT evaporator and small pressure drop (throttling loss) in expansion valves. Exergy loss in compressors depends only on the flow rate for the defined conditions of operation. Hence variations of exergy losses in MT and AUX compressors match that of respective flow rates in Fig. 10. The exergy loss occurs in the ejector mainly due to the irreversible mixing of motive and suction flow. In the model, this is represented by the inefficiency of the ejector. It is explained before that motive flow increases with gas cooler outlet temperature (Fig.10). For the specified inefficiency of the ejector, its exergy loss increases with gas cooler outlet

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temperature (Fig.10), which is in line with the variation in the motive flow rate. The exergy loss in the gas cooler remains nearly constant as an increase in its outlet temperature decreases the temperature drop (T_{4m}-T_{4i}) while correspondingly increasing the flow rate.

Fig. 14 illustrates that the exergy efficiency of the system decreases as the gas cooler outlet temperature increases. This is because the total exergy loss increases more than the power consumption. The figure also indicates that exergy efficiency increases by about 4.5% with the addition of IHX ($\mathcal{E}$=0.8). Its role in component-wise exergy loss is depicted in Fig. 15. The MT compressor, auxiliary compressor, ejector and gas cooler together contribute to about 88% of the exergy loss. The inclusion of IHX reduces the total exergy loss by 8.4%. Therefore IHX is recommended in the CO_2 system for simultaneous heating and cooling applications such as the dairy industry.

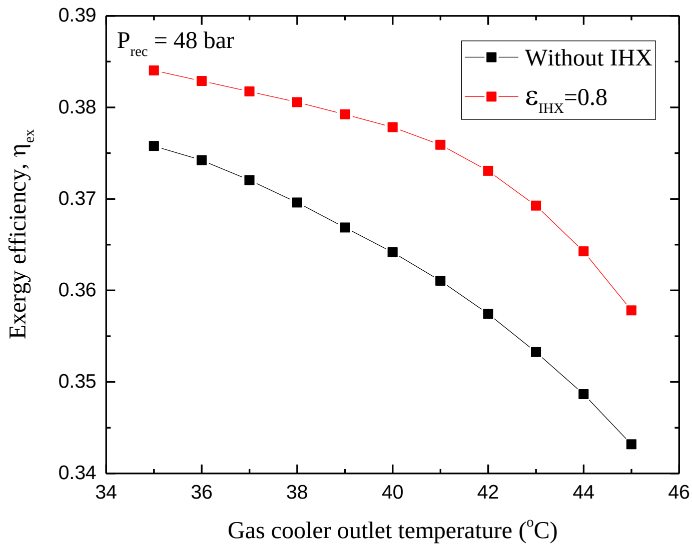

Fig. 14. Effect of internal heat exchanger and gas cooler outlet temperature on the exergy efficiency of the system.

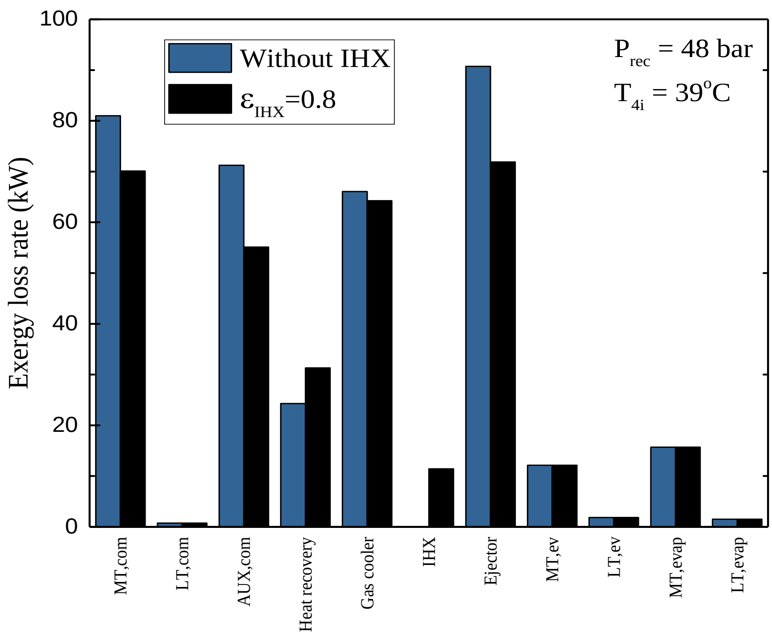

Fig. 15. Effect of internal exchanger on the exergy loss in each component of the system.

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5. Conclusions

The thermodynamic analysis of the ejector based CO_2 system with and without internal heat exchanger (IHX) for the application in the dairy industry is carried out. The internal heat exchanger is found to boost not only the discharge temperature but also the heating capacity of the system. Its effect on the optimum gas cooler pressure is analyzed. It decreases the minimum gas cooler pressure and increases the COP of the system. Internal heat exchanger with effectiveness 0.8 improves the COP of the system by 6.4% and the exergy efficiency by 4.5%. The maximum exergy efficiency of the system with IHX is observed as 38.4%. The ejector based CO_2 system with IHX is found appropriate for the Indian dairy industry to cater for its both cooling and heating demands simultaneously, thus enhancing the energy efficiency and also mitigating the global warming due to synthetic refrigerants.

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References

[1] Moulina, M. J., Rowland, F.S., 1974. Stratospheric sink for chlorofluoromethanes: chlorine atom-catalyzed destruction of ozone. Nature, 249, 810-812. https://www.nature.com/articles/249810a0

[2] Sarkar, J., Bhattacharyya, S., Ramgopal, M., 2004. Optimization of a transcritical CO_2 heat pump cycle for simultaneous cooling and heating application. Int. J. Refrigeration, 27, 830–838. https://www.sciencedirect.com/science/article/pii/S0140700704000520

[3] Lorentzen, G., 1994. Revival of carbon dioxide as a refrigerant. Int. J. Refrigeration, 17, 292–301. https://www.sciencedirect.com/science/article/pii/0140700794900590

[4] Palacz, M., Smolka, J., Andrzej, J., Nowak, K., Banasiak, K., Hafner, A., 2017. Shape optimisation of a two-phase ejector for CO_2 refrigeration systems. Int. J. Refrigeration, 74 212–223. https://www.sciencedirect.com/science/article/pii/S0140700716303425

[5] Li, D., Groll, E.A., 2005. Transcritical CO_2 refrigeration cycle with ejector expansion device. Int. J. Refrigeration, 28, 766–773. https://www.sciencedirect.com/science/article/pii/S0140700705000022

[6] Deng, J., Jiang, P., Lu, T., Lu, W., 2007. Particular characteristics of transcritical CO_2 refrigeration cycle with an ejector. App. Thermal Eng., 27, 381–388. https://www.sciencedirect.com/science/article/pii/S1359431106002705

[7] Fangtian, S., Yitai, M., 2011. Thermodynamic analysis of transcritical CO_2 refrigeration cycle with an ejector. App. Thermal Eng., 31, 1184–1189. https://www.sciencedirect.com/science/article/pii/S135943111000534X

This is the accepted version of an article published in Thermal Science and Engineering Progress
http://dx.doi.org/10.1016/j.tsep.2020.100626

[8] Sarkar, J., Bhattacharyya, S., Ramgopal, M., 2008. Transcritical carbon dioxide based heat pumps: Process heat applications. International Refrigeration and Air Conditioning conference. Purdue. https://docs.lib.purdue.edu/cgi/viewcontent.cgi?article=1690&conte=iracc

[9] Sawalha, S., 2008 Theoretical evaluation of trans-critical CO_2 systems in supermarket refrigeration. Part 1: Modeling, simulation and optimization of two system solutions. International Journal of Refrigeration, 31, 516-524. https://www.researchgate.net/publication/222683661_Theoretical_evaluation_of_trans-critical_CO2_systems_in_supermarket_refrigeration_Part_I_Modeling_simulation_and_optimization_of_two_system_solutions

[10] Liu, Y., Groll, E.A., Yazawa, K., Kurtulus, O., 2017. Energy saving performance and economics of CO_2 and NH_3 heat pumps with simultaneous cooling and heating applications in food processing: Case studies. Int. J. Refrigeration, 73, 111–124. https://www.sciencedirect.com/science/article/pii/S014070071630295X

[11] Ahammed, E., Battacharyya, S., Ramagopal, M., 2018. Analysis of CO_2 based refrigeration systems with and without ejector for simultaneous pasteurization and chilling of milk, Int. J. Refrigeration, 95, 61-72. https://www.sciencedirect.com/science/article/pii/S0140700718303013

[12] Konstantinos, M.T., Amarisb, C., Mylonaa, Z., Tassou, S., 2018. Analysis of typical booster configuration, parallel-compressor booster configuration and R717/R744 cascade refrigeration system for food retail applications. Part 2: energy performance in various climate conditions. 2nd International Conference on Sustainable Energy and Resource Use in Food Chains, ICSEF 2018, 17-19 October 2018, Paphos, Cyprus. https://www.sciencedirect.com/science/article/pii/S1876610219311713

[13] Hafner, A., Forsterling, S., Banasiak, K., 2014. Multi-ejector concept for R-744 supermarket refrigeration. Int. J. Refrigeration, 43, 1-13. https://www.sciencedirect.com/science/article/pii/S0140700714000668

[14] Singh, S., Dasgupta, M.S., 2017. CO_2 heat pump for waste heat recovery and utilization in dairy industry with ammonia based refrigeration. Int. J. Refrigeration, 78, 108-120. https://www.sciencedirect.com/science/article/pii/S0140700717301044

[15] Elbel, S., Hrnjak, P., 2008. Experimental validation of a prototype ejector designed to reduce throttling losses encountered in transcritical R744 system operation. Int. J. Refrigeration, 31, 411–422. https://www.sciencedirect.com/science/article/pii/S0140700707001508

[16] Liao, S.M., Zhao, T.S., Jakobsen, A., 2000. A correlation of optimal heat rejection pressures in transcritical carbon dioxide cycles. App. Thermal Eng., 20, 831–841. https://www.sciencedirect.com/science/article/pii/S1359431199000708

[17] Mylona, Z., Kolokotroni, M., Tsamos, K.M., Tassou, S.A., 2017. Comparative analysis on the energy use and environmental impact of different refrigeration systems for frozen food supermarket application. Energy Procedia, 123, 121-130. https://www.sciencedirect.com/science/article/pii/S1876610217328059

This is the accepted version of an article published in Thermal Science and Engineering Progress
http://dx.doi.org/10.1016/j.tsep.2020.100626

[18] Polzota, P., Agaroa, P.D., Cortellaa, G., 2017. Energy analysis of a transcritical CO2 supermarket refrigeration system with heat recovery. Energy Procedia, 111, 648 – 657. https://www.sciencedirect.com/science/article/pii/S1876610217302576

[19] Yadav, R.H., Jadhav, V.V., Chougule, G.A., 2016. Exploring the scope of renewable energy technologies in dairy sector. International Journal of Engineering Sciences & Research Technology, 5(7), 439-450. http://www.ijesrt.com/issues%20pdf%20file/Archive-2016/July-2016/50.pdf

[20] Megdouli, K., Ejemani, N., Nahdi, E., Mhimid, A., Kairouani, L., 2017. Thermodynamic analysis of a novel ejector expansion transcritical CO2/N2O cascade refrigeration (NEETCR) system for cooling applications at low temperatures. Energy, 128, 586-600. https://www.sciencedirect.com/science/article/pii/S0360544217306394

[21] Kohler, J., Richter, C., Tegethoff, W., Tischendorf, C., 2007, Experimental and theoretical study of a CO2 ejector refrigeration cycle. In: Presentation at the VDA Alternative Refrigerant Winter Meeting, Saalfelden, Austria. Available from: http://www.vda-wintermeeting.de/.

[22] Banasiak, K., Palacz, M., Hafner, A., Bulinski, Z., Smołka, J., Nowak A.J., Fic, A., 2014. A CFD-based investigation of the energyperformance of two-phase R744 ejectorsto recover the expansion work in refrigeration systems: An irreversibility analysis. International Journal of Refrigeration, 40, 328-337. https://www.sciencedirect.com/science/article/pii/S0140700713003812

[23] Lawrence, N., Elbel, S., 2018. Numerical investigation of the effect of microchannel evaporator design and operation on the improvement potential of ejector refrigeration cycles. Energy, 164, 21-34. https://www.sciencedirect.com/science/article/pii/S0360544218317158.

[24] Minetto, S., Brignoli, R., Banasiak, K., Hafner, A., Zilio, C., 2013. Performance assessment of an off-the-shelf R744 heat pump equippedwith an ejector. Applied Thermal Engineering, 59, 568-575 https://www.sciencedirect.com/science/article/pii/S1359431113004572

[25] Elbel, S., 2011. Historical and present developments of ejector refrigeration systems with emphasis on transcritical carbon dioxide air-conditioning applications. International Jornal of Refrigeration, 34, 1545-1561. https://www.sciencedirect.com/science/article/pii/S0140700710002720

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Performance Evaluation of Ejector Based CO_2 System for Simultaneous Heating and Cooling Application in an Indian Dairy Industry

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ABSTRACT

India is ranked first in milk production, and its goal is to increase annual production to 300 million tons by 2024. The dairy industry requires multi-temperature cooling and simultaneous heating. In the present study, the ejector based transcritical CO_2 system with two evaporators (LT at -10°C, MT at 0°C) is analyzed and proposed for fulfilling both cooling and heating demands in the dairy industry. Ejector based systems with and without internal heat exchanger are analyzed and compared. The effects of gas cooler outlet temperature, receiver pressure and gas cooler pressure on the performance of the system are investigated by energy and exergy analysis. The gas cooler pressure is analyzed for achieving the desired pasteurization temperature. The internal heat exchanger not only reduces the minimum gas cooler pressure required for the pasteurization temperature but also increases the system COP and exergy efficiency by 6.4% and 4.5% respectively. Second law analysis shows that the maximum exergy efficiency of the system is 38.4%.

Keywords: Trans-critical; Internal heat exchanger; Pasteurization; Exergy analysis; pressure optimization.

6. Introduction

Carbon dioxide was a widespread refrigerant in the early 18th century, which was used in the form of dry ice but was phased out with the invention of synthetic refrigerants. In 1985 Roland and Molina found that the use of halocarbon refrigerants causes the destruction of the ozone layer [1]. This, together with global warming, led to the search for alternate refrigerants. CO_2 being a natural refrigerant with zero ODP and GWP of 1, is conceived to be the best alternate refrigerant, especially for heat pump applications where simultaneous heating and cooling loads are in demand [2]. The operational pressure of CO_2 is higher than that of conventional refrigerants, making it a highly challenging refrigerant. However, 20% of energy saving is achieved for the same heating capacity and temperature requirement when compared with R12 [3].

At high ambient temperature CO_2 system works in the transcritical cycle. Therefore the maximum irreversibility occurs in throttling the gas. To minimize the loss during the expansion process and to utilize the available energy, researchers have proposed the use of ejectors. It is a simple device without any moving parts. A recent experimental study on a multi ejector pack reported a maximum work recovery efficiency of 36% [4]. Thermodynamic analysis of the CO_2 transcritical cycle [5] showed that the use of an ejector enhances the COP by 16%. Deng et al. [6] reported that the maximum cooling COP for the ejector expansion system is 8.2% and 11.5% higher than that of the internal heat exchanger cycle and conventional refrigeration cycle respectively. Fangtian and Yitai [7] reported that the ejector improves the COP by 30% and reduces the exergy loss by 25% when compared with the expansion valve.

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Nomenclature

AUX	auxiliary (-)
CIP	cleaning in process (-)
COP	coefficient of performance (-)
C_{pg}	specific heat at constant pressure of glycol(kJ/kgK)
GWP	global warming potential (-)
h	specific enthalpy (kJ/ kg)
HREX	heat recovery heat exchanger
I	exergy loss rate (kW)
IHX	internal heat exchanger (-)
LT	low temperature (-)
m	mass flow rate (kg/s)
MT	medium temperature (-)
ODP	ozone depletion potential (-)
P	pressure (bar)
Q	heat transfer rate (kW)
s	entropy (kJ/kgK)
T	temperature (°C)
W	work (kW)

Greek symbols

η	efficiency (%)
μ	entrainment ratio
ε	effectiveness

Subscripts

1	reference environment
4m	state point after heat recovery heat exchanger
4i, 16i	state point after the internal heat exchanger
c	cooling
com	compressor
dis	discharge
ej	ejector
ev	expansion valve
evap	evaporator
evapr	evaporator refrigerated medium
ex	exergy
gc	gas cooler
hr	heat recovery
i	i^{th} state, i = 1,2,3…….17
in	inlet
MN	motive nozzle
p	primary
s	suction
SN	suction nozzle
sys	system
tot	total
x, z	state point at outlet motive and suction nozzle in ejector efficiency

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The temperature glide available in the gas cooler when CO_2 system is operated in transcritical mode can be utilized in various industries like food processing, pulp and paper industry, chemical and petrochemical industry. The CO_2 system can be used for water heating applications, where the temperature required is up to 100°C [8]. Sawallah [9] presented the optimization of gas cooler pressure for heat recovery from CO_2 system for supermarket applications in Sweden, for ambient temperatures ranging from -5°C to 40°C.

CO_2 transcritical heat pump system can be utilized in food and beverage industries like dairy, sugar refineries, breweries, grain drying and canning units [8]. The use of a CO_2 heat pump system for milk processing showed the primary energy saving of 35% when compared with ammonia and CO_2 heat pump [10]. Also, the thermodynamic analysis of the CO_2 ejector system with an internal heat exchanger between two gas coolers showed an increased performance when compared to the system without an internal heat exchanger [11].

CO_2 systems are better for multi evaporator temperature in food retail applications like supermarkets. Supermarkets require refrigeration at different temperatures like MT for storage and LT for freezing. The performance comparison among the CO_2 parallel compression booster system consumed 15% and 16.6% less energy compared to the CO_2 booster system and R717/CO_2 cascade system respectively [12]. The analysis of a multi ejector CO_2 system for supermarkets with multi evaporators (MT and LT) and heat recovery at three temperature levels, using Modelica showed an increase in the COP by 20% [13]. The transcritical CO_2 booster system is more energy efficient and has 44% lower TEWI values when compared with the R134a parallel systems for MT and LT loads [17]. The CO_2 transcritical system with heat recovery integrated into a supermarket with LT and MT cabinets shows 3.6 to 6.5% energy savings when compared with the baseline system of R134a/CO_2 cascade refrigeration system with R410A heat pump system for hot water production and space heating [18].

The dairy industry requires simultaneous heating and cooling for milk processing, CIP, milk storage, etc. A thermodynamic study on cascaded ammonia and CO_2 system (for simultaneous heating and cooling) for the dairy industry reported that the cost of energy saving is approximately 33.8%, and the reduction in CO_2 emissions is 45.7% with a payback period of 40 months [14]. The utilization of solar energy for the preheating of water to a boiler in the dairy industry shows the payback period of 46 months with fuel oil savings of 41 kg/day [19]. However, the literature survey implies that less work has been carried out with CO_2 heat pump systems in dairy applications.

The production and consumption of dairy products are increasing, and so also the energy consumption in this sector. Besides, the present conventional method of heating used for pasteurization is unfavorable in terms of fossil fuel consumption and CO_2 emissions. Hence, this study focuses on the energy and exergy analysis of the CO_2 transcritical heat pump with an ejector and internal heat exchanger, exploring its suitability for a dairy application. Two evaporators for the cooling requirement at two different temperatures (4°C for milk chilling and -2.7°C for butter storage) and heat recovery at 72°C for pasteurization are considered. The CO_2 heat pump system is analyzed based on the gas cooler pressure required for fulfilling the heating load for achieving pasteurization temperature. The effect of parameters like gas cooler outlet temperature, gas cooler pressure and receiver pressure on the performance of the system is discussed

7. The milk processing system in a dairy plant

The plant Sholinganallur Dairy, located in Chennai, Tamil Nadu, India, is considered as an example for the present study for milk pasteurization and butter storage. The dairy requires cooling at two temperatures; approximately 4°C for milk chilling and -2.7°C for butter storage. It also requires heating above 72°C for pasteurization. Presently Ammonia and Freon based

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refrigeration systems are used to fulfill refrigeration load (Q_{tot}) of 280 TR at 0°C and 20 TR at -10°C respectively, while the boiler is used for 11 hours a day to meet 680 kW of heating demand (Q_h) for. Fig. 1 shows the conventional heating and chilling system used in the dairy industry, in which the ammonia refrigeration system is used for chilling water, and the boiler is used for hot water production.

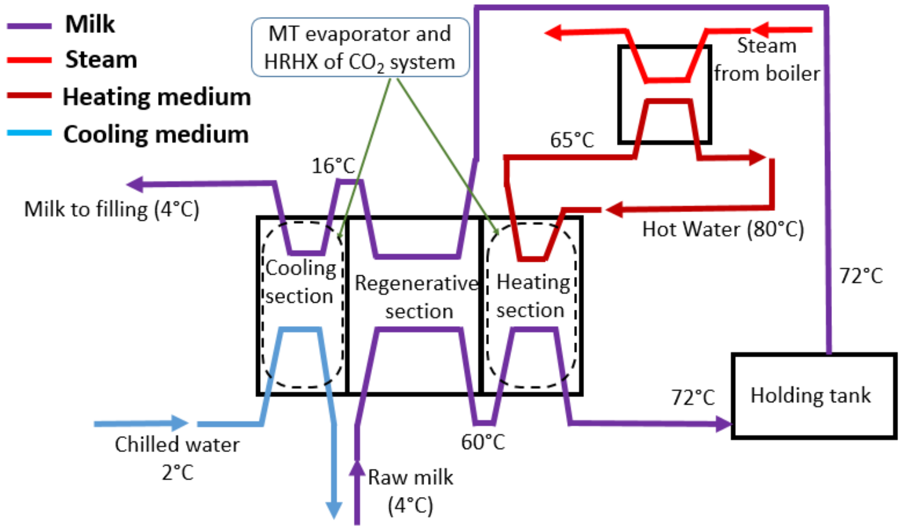

Fig. 1. Conventional heating and chilling system

8. Ejector based transcritical CO_2 system for a dairy industry

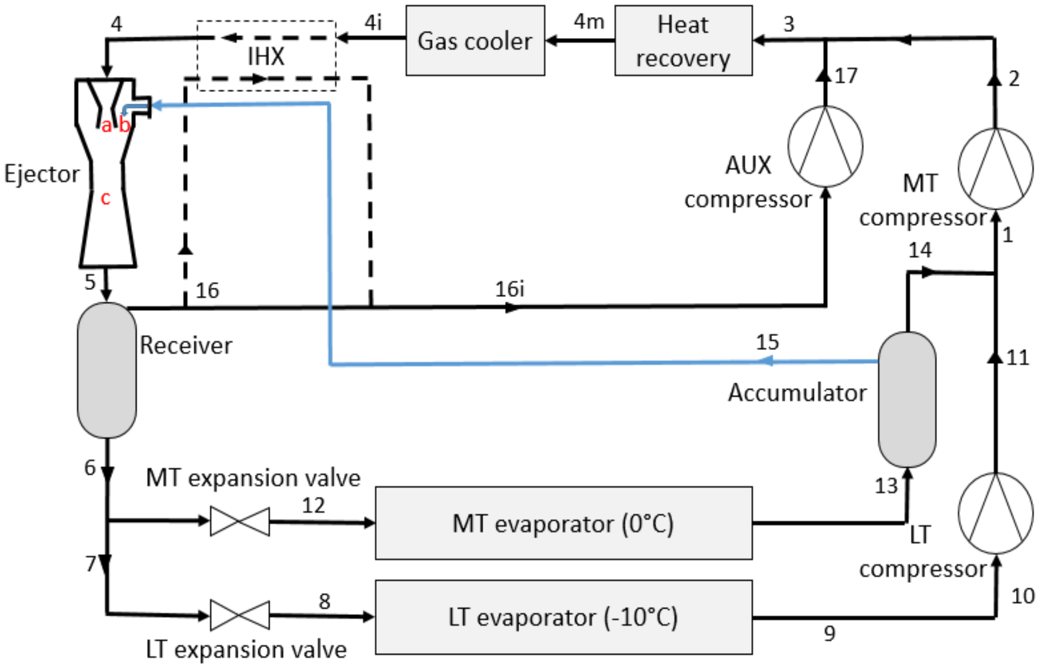

Fig. 2. Schematic of the transcritical CO_2 system

The schematic of the transcritical CO_2 system with and without IHX, and the corresponding cycle on the p-h diagram are shown in Figs. 2 and 3 respectively. The system consists of an ejector as an expansion device and two evaporators; medium temperature (MT) evaporator for milk chilling and low temperature (LT) evaporator for butter storage, and three

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compressors; LT compressor, MT compressor and an auxiliary compressor. LT compressor compresses CO_2 vapour from the LT evaporator pressure (10) to the accumulator pressure (11). The CO_2 vapour coming out from the MT evaporator (13) is collected in the accumulator. The total mass or fraction of the mass in the accumulator is used as the ejector suction (15) and the rest is supplied to the MT compressor (14). The vapour CO_2 from the accumulator and LT compressor (11) is compressed in the MT compressor. The high pressure and high temperature CO_2 gas at the discharge of MT (2) and auxiliary compressors (17) are fed to the heat recovery heat exchanger (3). The heat that is required for heating the milk is recovered in the heat recovery heat exchanger, and the excess heat is rejected to the ambient in the gas cooler. For the configuration with IHX, the CO_2 from the gas cooler outlet (4i) is allowed to cool lower than the gas cooler outlet temperature in the internal heat exchanger. This CO_2 from the gas cooler outlet is cooled with the help of vapour CO_2 stream from the receiver (16) and expanded in the ejector. For the configuration without IHX, the CO_2 after gas cooler (4) is expanded in the motive nozzle of the ejector to receiver pressure (5). At the same time, CO_2 from the MT evaporator (15) is expanded in the suction nozzle along with the motive flow. The two streams coming out from the motive nozzle (a) and suction nozzle (b) are mixed in the mixing section (c) at constant pressure. The mixed stream (c) regains the pressure in the diffuser to an intermediate pressure at the exit of the ejector (5). It is in a two-phase region and is separated in the phase separator. The vapour CO_2 (16) flows through the internal heat exchanger (16i). It is compressed in the AUX compressor (17). The liquid CO_2 (6) is expanded to MT and LT evaporators through expansion valves for cooling requirements of milk chilling (12) and butter storage (8) respectively.

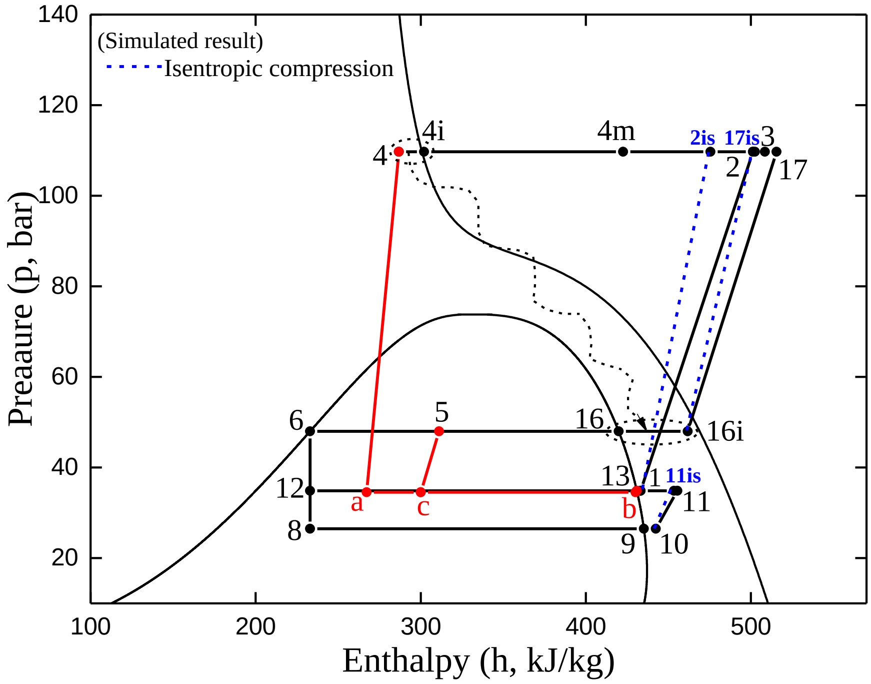

Fig. 3. *ph* plot of ejector based transcritical CO_2 system.

8.1. Experimental test setup for validation of a model

Fig. 4 shows the instrumented CO_2 test facility used for validation of the simulation results. The test facility is equipped with two shell and tube evaporators; MT (0°C, 10 kW) and LT (-10°C, 3 kW) evaporators, heat recovery shell and tube heat exchanger, gas cooler (copper tube and aluminum finned, air-cooled cross-flow heat exchanger) and three semi-hermetically

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sealed reciprocating compressors. The speed of the compressors is varied by three invertors according to the cooling loads in evaporators by feedback control.

Table 1. List of instruments used in the experimental setup and their specifications

S. No.	Sensor	Company	Model	Type	Range	Accuracy (% of measured value)
1.	Temperature	Danfoss	AKS 21M	B	-50 to 200°C	0.4
2.	Pressure	Danfoss	AKS 2050	----	up to 150 bar	0.5
3.	Energy meter	ISOIL	PT 500	IFX-M	3.0 $m^3\ h^{-1}$	0.5
4.	Power meter	Danfoss	VFD	----	35-60 Hz	0.8

The glycol solution is used as a medium to load the evaporators and heat recovery heat, exchanger. The controlled heat transfer is allowed to take place from the latter to the former. The excess heat from the CO_2 after heat recovery is rejected in the gas cooler. The parameters viz., gas cooler outlet temperature, receiver pressure, MT and LT evaporator temperatures are adjusted in the control panel. Temperature and pressure readings at different locations are directly obtained from the control panel with a PT1000 temperature sensor and Danfoss pressure transmitter. The heating and cooling loads are obtained by measuring the flow rates and temperature of glycol across the heat recovery heat exchanger, MT and LT evaporators. The power input to the compressors is stored in the control panel. The instruments used for measuring parameters and their specifications are listed in Table 1.

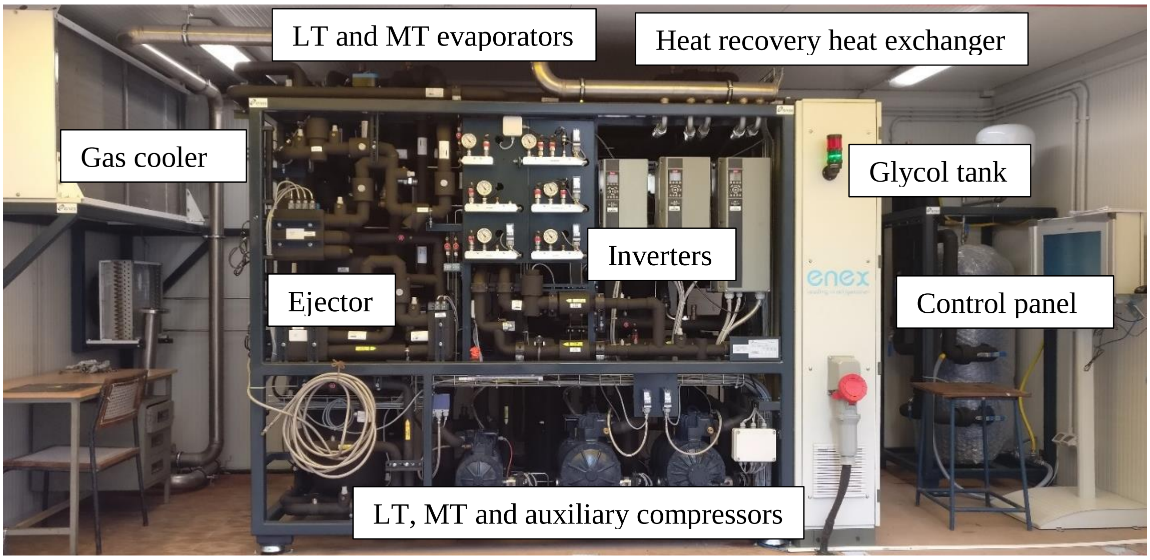

Fig. 4. Experimental test setup of a transcritical ejector CO_2 system.

The system is allowed to run for 1 hour to attain steady-state conditions. During this time, the readings for temperature and pressure at different locations and glycol flow rates through evaporators and heat recovery heat exchangers are stored in the control panel for every 10 seconds of operation. Steady-state readings for 30 minutes period are taken and averaged for the performance calculation of the system.

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The data stored in the control panel includes mass flow rates of glycol through MT and LT evaporators and heat recovery heat exchanger, compressors power consumption, temperatures of glycol at entry and exit of evaporators and heat recovery and pressure and temperature readings of CO_2 at various stages. The cooling and heating loads and COPs are calculated as follows

$$Q_{MT,evap} = m_{gly,MT,evap}\left(C_{pg,MT,in}T_{gly,MT,evap,in} - C_{pg,MT,out}T_{gly,MT,evap,out}\right) \tag{1}$$

$$Q_{LT,evap} = m_{gly,LT,evap}\left(C_{pg,LT,in}T_{gly,LT,evap,in} - C_{pg,LT,out}T_{gly,LT,evap,out}\right) \tag{2}$$

$$Q_{hr} = m_{gly,hr}\left(C_{pg,hr,out}T_{gly,hr,out} - C_{pg,hr,in}T_{gly,hr,in}\right) \tag{3}$$

The total cooling load, cooling and heating COPs and the total power consumed by the compressors of the system are

$$Q_{tot} = Q_{MT,evap} + Q_{LT,evap} \tag{4}$$
$$W_{tot} = W_{MT,com} + W_{AUX,com} + W_{LT,com} \tag{5}$$

Where $W_{MT,com}$, $W_{AUX,com}$ and $W_{LT,com}$ are the individual power consumptions of MT, AUX and LT compressors respectively. These are directly stored in the control panel.

$$COP_c = \frac{Q_{tot}}{W_{tot}} \tag{6}$$
$$COP_h = \frac{Q_{hr}}{W_{tot}} \tag{7}$$
$$COP_{sys} = \frac{Q_{tot}+Q_{hr}}{W_{tot}} \tag{8}$$

Where COP_c, COP_h and COP_{sys} are the cooling, heating and system COPs respectively, which are obtained experimentally and used for validation of simulation results with experimental results.

8.2. CO_2 system modeling and performance evaluation

Assumptions:

- One dimensional steady-state model.
- Pressure drop in gas cooler, heat recovery heat exchanger, receiver, accumulator and evaporators are negligibly small.
- Refrigerant (CO_2) exits MT and LT evaporators at a saturated vapour state.
- Refrigerant enters into HREX after mixing from the exits of MT and AUX compressors
- Variable parameters – gas cooler outlet temperature (T_4), receiver pressure (P_{rec}), gas cooler pressure (P_{gc}), IHX effectiveness
- Constant parameters – ejector efficiency, MT and LT evaporator capacities and temperatures.

Thermodynamic modeling of an ejector is considered by assuming its efficiency to be constant throughout the analysis. The ejector consists of the motive nozzle (primary nozzle), the suction nozzle, the mixing section and the diffuser. The refrigerant (CO_2) at high pressure ($P_{MN,in}$ or P_{gc}) from the gas cooler outlet (state 4 in Fig. 5) and the low pressure ($P_{SN,in}$ or $P_{MT,evap}$) refrigerant from the accumulator (state 15 in Fig. 5) are supplied to motive and suction nozzles respectively. These mix in the mixing section and get compressed to an intermediate

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pressure ($P_{DIF,out}$ or P_{rec}) in the diffuser. The ejector efficiency for the present study is defined as the ratio of work recovered in the ejector to the maximum work available in the motive nozzle [15]. The latter is the expansion work from the gas cooler pressure to the receiver pressure. Similarly, the former is the compression of suction fluid from the MT evaporator pressure to the receiver pressure. These expansion and compression processes are shown in Fig. 5.

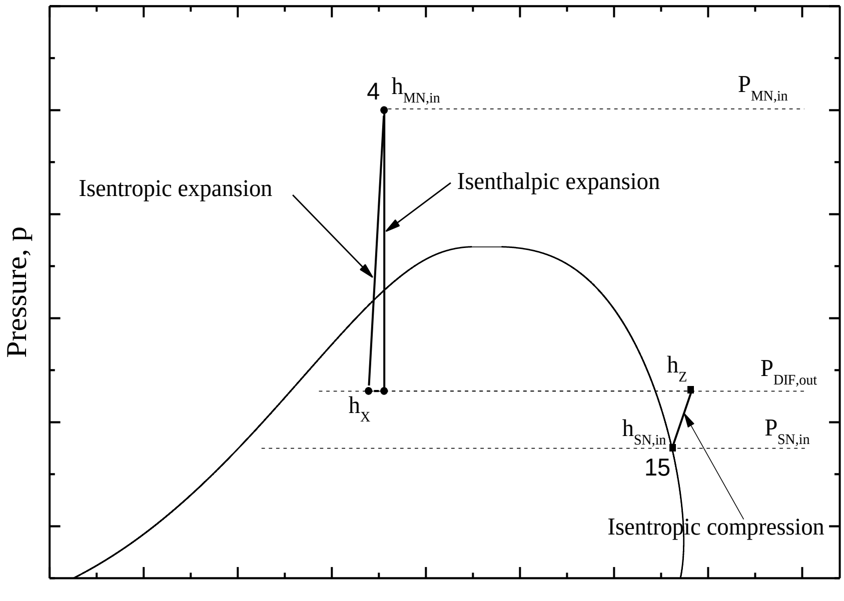

Fig. 5. Expansion of motive fluid in a two-phase ejector and compression of suction fluid.

Thus ejector efficiency is given by,

$$\eta_{ej} = \mu \, . \, \frac{h_Z - h_{SN,in}}{h_{MN,in} - h_X} \tag{9}$$

$$\mu = \frac{m_s}{m_p} \tag{10}$$

Where μ is the entrainment ratio of the ejector, defined as the ratio of the mass flow rate through the suction nozzle (m_s) to the mass flow rate through the motive nozzle (m_p). The ejector efficiency is considered as 0.25 [4]. The motive and suction mass flow rates (m_p and m_s) are unknowns in Eq. 9. The enthalpies h_X and h_Z are found by assuming the receiver pressure ($P_{DIF,out}$ or P_{rec}, variable parameter). h_X is the function of receiver pressure and entropy at state 4, and h_Z is the function of receiver pressure and entropy at state 15 (Fig. 5).

The mass flow rate through the evaporators required for the evaporator capacities are evaluated as

$$m_{MT,evap} = \frac{Q_{MT,evap}}{h_{13} - h_{12}} \tag{11}$$

$$m_{LT,evap} = \frac{Q_{LT,evap}}{h_9 - h_8} \tag{12}$$

Where $m_{MT,evap}$ and $m_{LT,evap}$ are the mass flow rates of refrigerant through MT and LT evaporators respectively. The enthalpies h_{12} and h_8 are equal to h_6 obtained by the initial assumption of the receiver pressure and saturated liquid state (state 6). $Q_{MT,evap}$ and $Q_{LT,evap}$ are the cooling capacities in MT and LT evaporators respectively.

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The mass flow rates through the ejector motive nozzle (m_p), suction nozzle (m_s) and auxiliary compressor ($m_{AUX,com}$) are found by taking the control volume across the ejector and the phase separator (receiver) by mass and energy balance equations along with ejector efficiency expression (Eq. 9).

$$m_p + m_s = m_{MT,evap} + m_{LT,evap} + m_{AUX,com} \tag{13}$$

$$m_p h_4 + m_s h_{15} = m_{AUX,com} h_{16} + (m_{MT,evap} + m_{LT,evap}) h_6 \tag{14}$$

have Three unknowns, namely, mass flow rates through the suction nozzle (m_s), motive nozzle (m_p) and auxiliary compressor ($m_{AUX,com}$) are found by solving Eqs. 9, 13 and 14 simultaneously. Then we get the expressions for m_s and m_p are as follows

$$m_s = \frac{(m_{MT,evap}+m_{LT,evap})(h_6-h_{16})}{\frac{A}{\eta_{ej}}(h_4-h_{16})+h_{15}-h_{16}} \tag{15}$$

$$m_p = \frac{(m_{MT,evap}+m_{LT,evap})(h_6-h_{16})}{\frac{A}{\eta_{ej}}(h_4-h_{16})+h_{15}-h_{16}} \cdot \frac{A}{\eta_{ej}} \tag{16}$$

Where $A = \frac{h_Z - h_{SN,in}}{h_{MN,in} - h_X}$ taken in Eq. 9.

$m_{AUX,com}$ is obtained by substituting m_s and m_p in Eq. 13 and the mass flow rate through the MT compressor is expressed as

$$m_{MT,com} = m_{MT,evap} + m_{LT,evap} - m_s \tag{17}$$

The temperatures of exit streams from internal heat exchanger are calculated by considering the effectiveness of IHX and energy balance equations expressed as follows

$$\mathcal{E}_{IHX} = \frac{h_{16i}-h_{16}}{h_{4i}-h_{16}} \tag{18}$$

$$m_p(h_{4i} - h_4) = m_{AUX,com}(h_{16i} - h_{16}) \tag{19}$$

Isentropic efficiency of the compressors is considered as a correlation defined as [16].

$$\eta_{com} = 1.003 - 0.121\left(\frac{P_{dis}}{P_s}\right) \tag{20}$$

The superheat at the inlet to the LT compressor is assumed to be 5 degrees. Based on the isentropic efficiency presented in Eq. 17 the exit enthalpies (h_2, h_{17} and h_{11}) of the compressors are estimated.

After determining the mass flow rates and exit enthalpies of compressors the work done by the compressors are expressed as

$$W_{LT,com} = m_{LT,com}(h_{11} - h_{10}) \tag{21}$$

$$W_{MT,com} = m_{MT,com}(h_2 - h_1) \tag{22}$$

$$W_{AUX,com} = m_{AUX,com}(h_{17} - h_{16s}) \tag{23}$$

$W_{LT,com}$, $W_{MT,com}$ and $W_{AUX,com}$ are works of LT, MT and auxiliary compressors. The total work of the system is calculated as

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$$W_{tot} = W_{LT,com} + W_{MT,com} + W_{AUX,com} \quad (24)$$

$$Q_{tot} = Q_{LT,evap} + Q_{MT,evap} \quad (25)$$

$$Q_{hr} = m_p(h_3 - h_{4m}) \quad (26)$$

Q_{tot} is the sum of cooling capacities of LT and MT evaporators and Q_{hr} is the heating capacity in the heat recovery.

The performance of the system is portrayed by cooling COP, heating COP and system COP as follows.

$$COP_c = \frac{Q_{tot}}{W_{tot}} \quad (27)$$

$$COP_{hr} = \frac{Q_{hr}}{W_{tot}} \quad (28)$$

$$COP_{sys} = COP_c + COP_{hr} \quad (29)$$

Where COP_c, COP_{hr}, and COP_{sys}, are cooling COP, heating COP and system COP respectively.

3.2. Exergy analysis

Exergy analysis helps in pointing out the direction of the system improvement and evaluating the exergy loss in each component of the system. The exergy loss for each component has been analyzed by the following equations.

For LT compressor

$$I_{LT,com} = m_{LT,com} T_0 (s_{11} - s_{10}) \quad (30)$$

For MT compressor

$$I_{MT,com} = m_{MT,com} T_0 (s_2 - s_1) \quad (31)$$

For auxiliary compressor

$$I_{AUX,com} = m_{AUX,com} T_0 (s_{17} - s_{16i}) \quad (32)$$

For heat recovery heat exchanger

$$I_{hr} = m_p T_0 (s_{4m} - s_3) + Q_{hr} \left(\frac{T_0}{T_{4m}}\right) \quad (33)$$

For gas cooler

$$I_{gc} = m_p[(h_{4m} - h_4) - T_0(s_{4m} - s_3)] \quad (34)$$

For internal heat exchanger

$$I_{IHX} = m_{AUX,com}[(h_{16} - h_{16i}) - T_0(s_{16} - s_{16i})] + m_p[(h_{4i} - h_4) - T_0(s_{4i} - s_4) \quad (35)$$

For ejector

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$$I_{ej} = m_p T_0 (s_5 - s_4) + m_s T_0 (s_5 - s_{15}) \quad (36)$$

For MT expansion valve

$$I_{MT,ev} = m_{MT,evap} T_0 (s_{12} - s_6) \quad (37)$$

For LT expansion valve

$$I_{LT,ev} = m_{LT,evap} T_0 (s_8 - s_6) \quad (38)$$

For MT evaporator

$$I_{MT,evap} = m_{MT,evap} T_0 \left[\frac{h_{12} - h_{13}}{T_{MT,evapr}} - (s_{12} - s_{13})\right] \quad (39)$$

For LT expansion valve

$$I_{LT,evap} = m_{LT,evap} T_0 \left[\frac{h_8 - h_9}{T_{LT,evapr}} - (s_8 - s_9)\right] \quad (40)$$

The total exergy loss of the system is

$$I_{tot} = I_{LT,com} + I_{MT,com} + I_{AUX,com} + I_{hr} + I_{gc} + I_{IHX} + I_{ej} + I_{MT,ev} + I_{LT,ev} + I_{MT,evap} + I_{LT,evap} \quad (41)$$

The exergy efficiency of the cycle is

$$\eta_{ex} = 1 - \frac{I_{tot}}{W_{tot}} \quad (42)$$

Where T_0 is ambient temperature assumed as 30°C and $T_{LT,evapr}$ and $T_{MT,evapr}$ are the temperatures of the refrigerated medium in LT and MT evaporators respectively. The variable parameters considered for analysis of the present model are gas cooler pressure (P_{gc} or P_4), gas cooler outlet temperature (T_{4i}), receiver pressure (P_{rec} or P_5) and IHX effectiveness (ε_{IHX}).

The complete simulation of the model is carried out in MATLAB by interfacing with REFPROP as a property calculator at various state points.

9. Results and discussion

The thermodynamic model of the CO_2 system suitable for the dairy industry application is validated comparing with experimental results and the model results are discussed.

9.1. Model validation with experimental results

The input parameters and the basic assumptions of the thermodynamic model are listed in Table 2. The parameters considered for the model are based on the possible operating conditions in the experimental test setup. The results presented in Figs. 6 to 9 are based on the parameters listed in Table 2 for the system without IHX. The analysis has done with these parameters are mainly for the validation of the model. After validation, the same model is extended for the performance evaluation of the CO_2 system for dairy industry application with the data listed in Table 3.

Table 2 Input parameters considered for validation with experimental results.

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Parameter	Value
MT evaporator capacity, $Q_{MT,evap}$ (kW)	10
LT evaporator capacity, $Q_{LT,evap}$ (kW)	3
HREX capacity, Q_{hr} (kW)	13
MT evaporator temperature, $T_{MT,evap}$ (°C)	0
LT evaporator temperature , $T_{LT,evap}$ (°C)	-10

The experimental test setup is designed to operate at optimum gas cooler pressure in a transcritical state, where the pressure and temperature are independent parameters. To compare the simulation results with those of experimental, the gas cooler pressure has to be optimized for the model too. Fig. 6 shows the optimized gas cooler pressure at its various outlet temperatures for the maximum system COP. Further, the optimized gas cooler pressure is used for calculating the cooling and system COP of the thermodynamic model, which are validated with the experimental results.

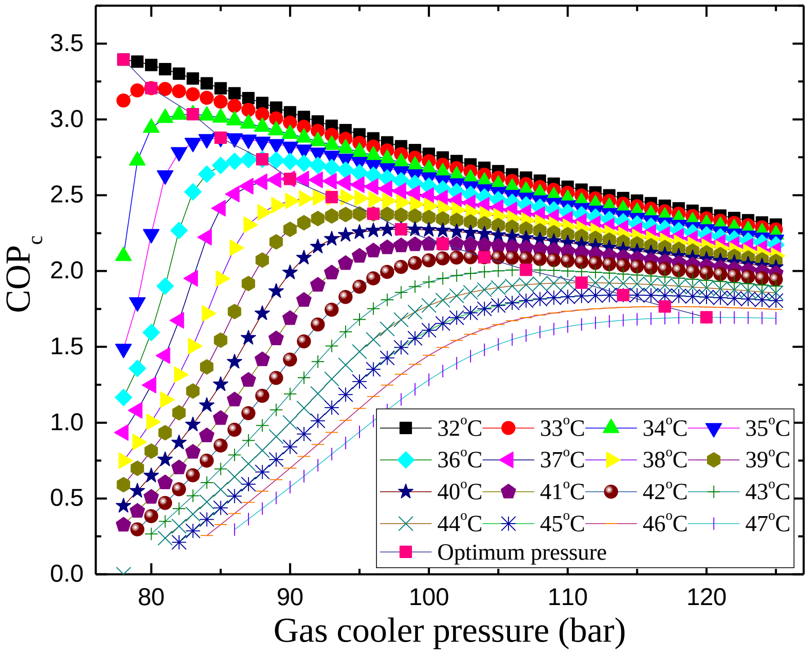

Fig. 6. Variation of COP with gas cooler pressure at various gas cooler outlet temperature without IHX.

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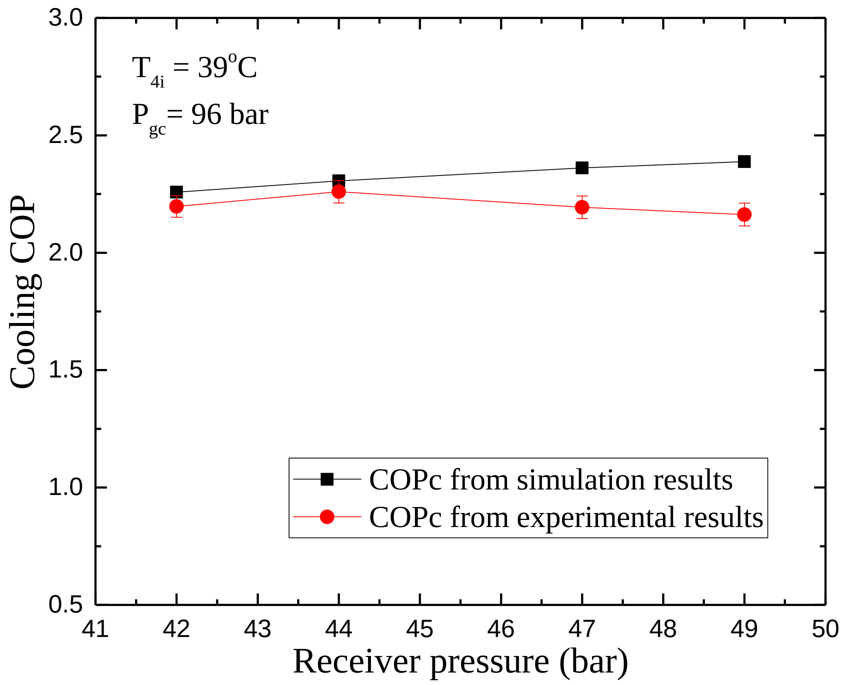

Fig. 7. Variation of cooling COP with receiver pressure.

The simulation results are validated against the present experimental results. Fig. 7 compares the variation of the two cooling COPs with receiver pressure at the gas cooler outlet temperature of 39°C. At this outlet temperature, the optimum pressure is 96 bar from Fig. 6, and the same is considered as the discharge pressure. The COP values match substantiating the model. The effect of receiver pressure on system COP is less. This is because the increase in receiver pressure reduces the pressure ratio of the auxiliary compressor and thereby its work. The increase in receiver pressure also increases the pressure lift of the ejector. This reduces the mass flow rate through the suction nozzle consequently pushing the remaining refrigerant to the MT compressor ($m_{MT,com}$) as shown in Fig. 8. Therefore, the decrease in auxiliary compressor work is compensated by the increase in MT compressor work which results in less variation of COP with receiver pressure. The simulation results are closely matching with the experimental results with a minimum and maximum COP error of 0.046 and 0.225 respectively. As the effect of receiver pressure on the performance of the system is less, therefore a value of 48 bar is assumed as receiver pressure for the performance evaluation of the thermodynamic model.

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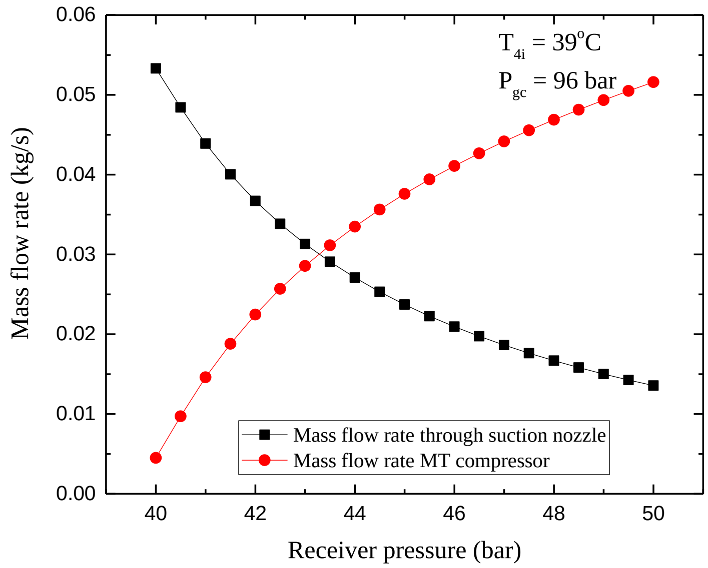

Fig. 8. Variation of ejector suction mass flow rate and mass flow rate through MT compressor with receiver pressure.

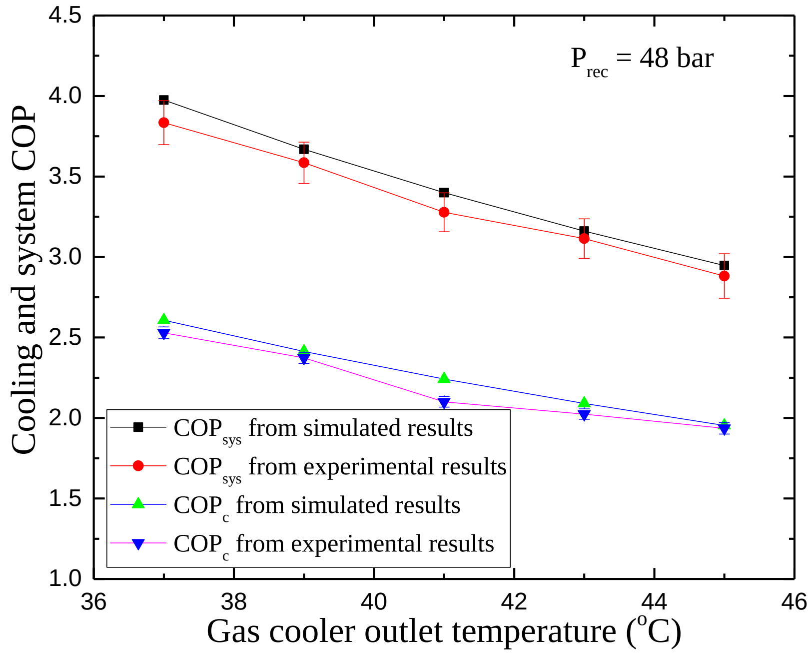

Fig. 9. Variation of cooling and system COPs with gas cooler outlet temperature.

Fig. 9 also validates the model by matching the variation of both cooling and system COPs of the two (model and experiment) with gas cooler outlet temperature. For every gas cooler outlet temperature (T_4), the optimum gas cooler (discharge) pressure is taken from Fig. 6. As the outlet temperature increases, optimum gas cooler pressure increases and both cooling and system COPs decrease. This is because the compressor power increases faster than the refrigeration capacity.

9.2. Performance evaluation of the proposed system

The input parameters considered for the performance evaluation listed are in Table 3. The parameters considered for the model are based on the requirement in the dairy industry application.

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Table 3 Input parameters considered for the performance evaluation.

Parameter	Value
MT evaporator load, $Q_{MT,evap}$ (kW)	980
LT evaporator load, $Q_{LT,evap}$ (kW)	70
HREX load, Q_{hr} (kW)	680
MT evaporator temperature, $T_{MT,evap}$ (°C)	0
LT evaporator temperature , $T_{LT,evap}$ (°C)	-10
HREX outlet temperature, T_{4m} (°C)	65
Chilled milk temperature in MT evaporator $T_{MT,evapr}$ (°C)	4
Butter storage temperature in LT evaporator, $T_{LT,evapr}$ (°C)	-2.7
Ambient temperature, T_0 (°C)	30

Fig. 10 shows the variation of the mass flow rate through the suction nozzle, and LT, MT and auxiliary compressors with gas cooler outlet temperature. As the gas cooler outlet temperature increases, the vapour quality (dryness fraction) at the ejector outlet increases. This reduces the liquid flow rate at the ejector outlet and therefore the cooling capacity. To fulfill the required cooling load in MT and LT evaporators, the mass flow rate at the ejector outlet should be higher. This necessitates more flow rate through the auxiliary compressor with an increase in gas cooler outlet temperature as shown in the figure.

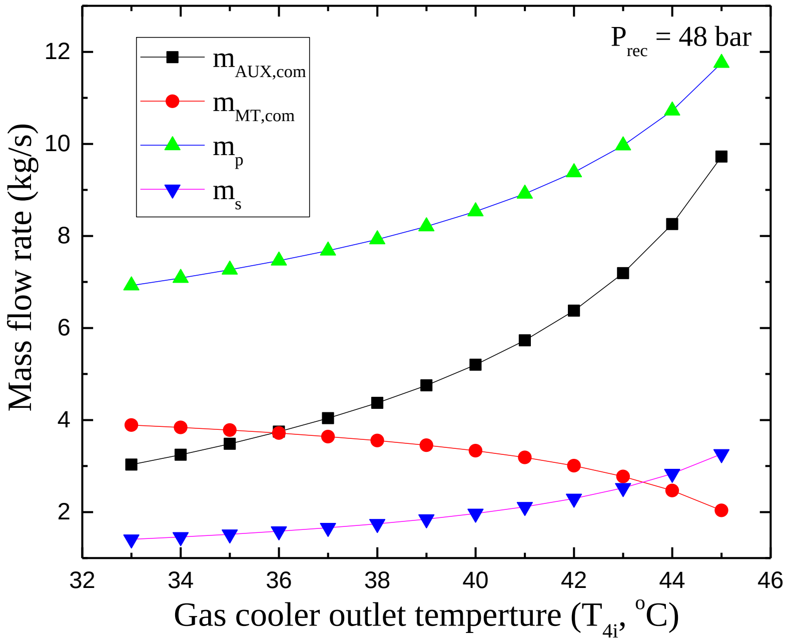

Fig. 10. Effect of gas cooler outlet temperature on mass flow rates through ejector suction nozzle, HREX and MT and AUX compressors.

With the increase in gas cooler outlet temperature, energy in the motive gas flow or the energy input to the ejector increases. This facilitates the ejector to draw more gas through the ejector suction nozzle and consequently the flow rate through MT compressor decreases. However, the total of these two increases leading to more flow rate through the heat recovery heat exchanger. Thus the heating capacity of the system increases with the outlet temperature of the gas cooler.

9.2.1. Effect of IHX on gas cooler pressure and system COP

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Milk is heated from 60 to 72°C in the heat recovery heat exchanger (Fig.1) of the CO_2 system for pasteurization. Considering 5°C approach, the minimum outlet temperature of CO_2 (T_{4m}) is 65°C. With limited gas flow in hand, thus the compressor discharge temperature should be high enough to meet the heating demand. Higher superheat or the compressor inlet temperature and higher pressure ratio or compressor discharge pressure are the two means to boost the discharge temperature. The former is accomplished by utilizing IHX.

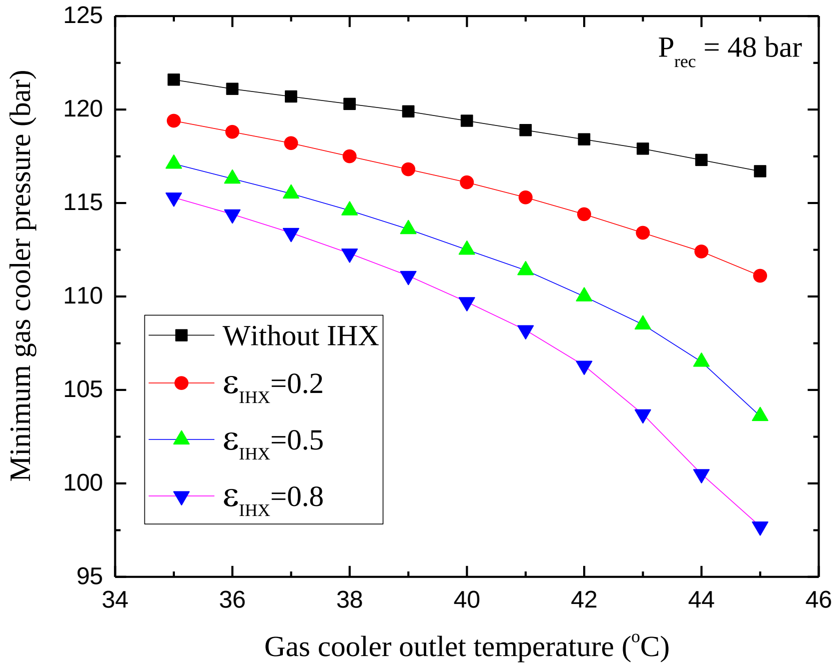

Fig. 11. Effect of gas cooler outlet temperature and IHX on minimum gas cooler pressure.

Fig. 11 shows the effect of gas cooler outlet temperature and effectiveness of IHX on the minimum gas cooler pressure, yet providing sufficient heat for milk pasteurization. This pressure decreases with an increase in the outlet temperature. As explained before in Fig.10, higher outlet temperature results in higher flow through the HREX for the same cooling and heating demands. The higher flow can deliver more heat, but to get only the required heat, gas cooler pressure is to be decreased as illustrated in the figure. The auxiliary compressor inlet temperature (T_{16i}) increases with an increase in the effectiveness of the internal heat exchanger. Therefore the required heating demand is achieved at lower pressures.

The influences of gas cooler outlet temperature and the effectiveness of IHX on the COP of the system are depicted in Fig. 12. IHX not only decreases the minimum gas cooler pressure but also decreases the gas cooler outlet temperature (subcooling), thereby increases the system COP. The maximum improvement of system COP is 6.4% with the addition of IHX (Ɛ=0.8).

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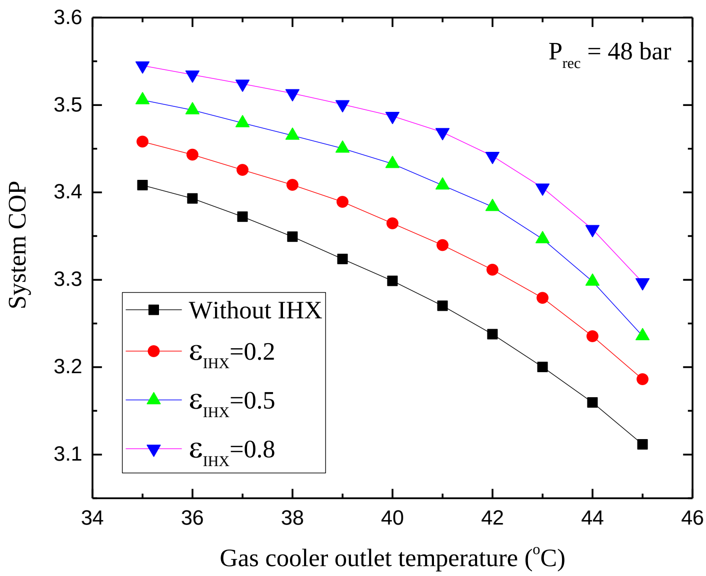

Fig. 12. Effect of gas cooler outlet temperature and IHX on COP of system.

9.3. Exergetic performance of the system

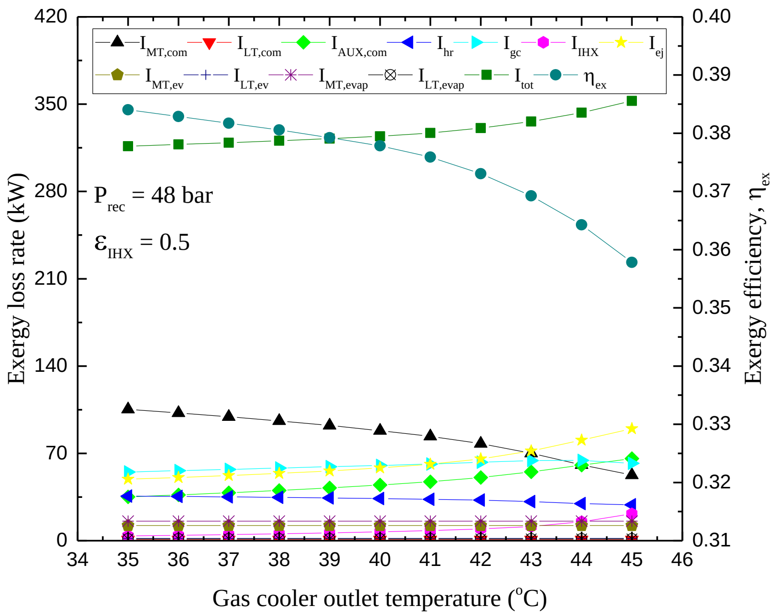

Fig. 13. Effect of gas cooler outlet temperature on exergetic performance of the system

Fig. 13 shows the effect of gas cooler outlet temperature on the exergetic performance of the system. The exergy losses in evaporators and expansion valves are minimum amongst all the components. This is because of a small temperature difference of only 4°C between the refrigerant and the milk to be cooled in MT evaporator, low cooling load (low refrigerant flow) in LT evaporator and small pressure drop (throttling loss) in expansion valves. Exergy loss in compressors depends only on the flow rate for the defined conditions of operation. Hence variations of exergy losses in MT and AUX compressors match that of respective flow rates in Fig. 10. The exergy loss occurs in the ejector mainly due to the irreversible mixing of motive and suction flow. In the model, this is represented by the inefficiency of the ejector. It is explained before that motive flow increases with gas cooler outlet temperature (Fig.10). For the specified inefficiency of the ejector, its exergy loss increases with gas cooler outlet

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temperature (Fig.10), which is in line with the variation in the motive flow rate. The exergy loss in the gas cooler remains nearly constant as an increase in its outlet temperature decreases the temperature drop (T_{4m}-T_{4i}) while correspondingly increasing the flow rate.

Fig. 14 illustrates that the exergy efficiency of the system decreases as the gas cooler outlet temperature increases. This is because the total exergy loss increases more than the power consumption. The figure also indicates that exergy efficiency increases by about 4.5% with the addition of IHX (Ɛ=0.8). Its role in component-wise exergy loss is depicted in Fig. 15. The MT compressor, auxiliary compressor, ejector and gas cooler together contribute to about 88% of the exergy loss. The inclusion of IHX reduces the total exergy loss by 8.4%. Therefore IHX is recommended in the CO_2 system for simultaneous heating and cooling applications such as the dairy industry.

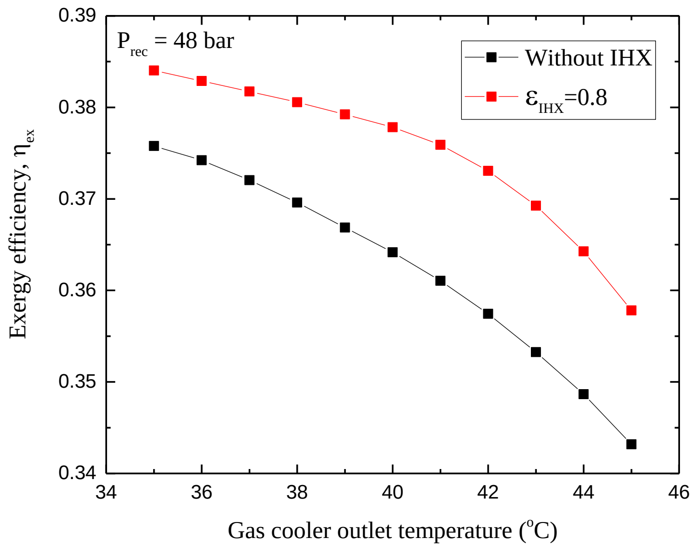

Fig. 14. Effect of internal heat exchanger and gas cooler outlet temperature on the exergy efficiency of the system.

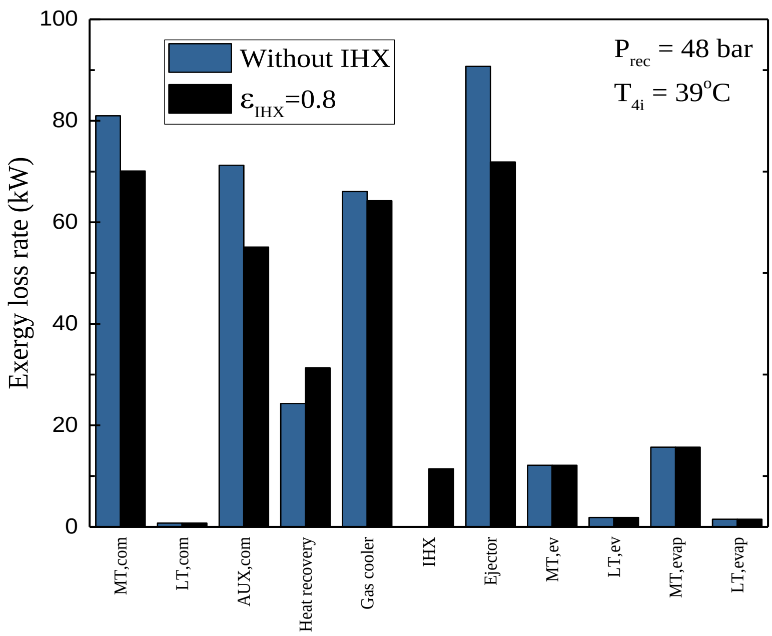

Fig. 15. Effect of internal exchanger on the exergy loss in each component of the system.

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10. Conclusions

The thermodynamic analysis of the ejector based CO_2 system with and without internal heat exchanger (IHX) for the application in the dairy industry is carried out. The internal heat exchanger is found to boost not only the discharge temperature but also the heating capacity of the system. Its effect on the optimum gas cooler pressure is analyzed. It decreases the minimum gas cooler pressure and increases the COP of the system. Internal heat exchanger with effectiveness 0.8 improves the COP of the system by 6.4% and the exergy efficiency by 4.5%. The maximum exergy efficiency of the system with IHX is observed as 38.4%. The ejector based CO_2 system with IHX is found appropriate for the Indian dairy industry to cater for its both cooling and heating demands simultaneously, thus enhancing the energy efficiency and also mitigating the global warming due to synthetic refrigerants.

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References

[26] Moulina, M. J., Rowland, F.S., 1974. Stratospheric sink for chlorofluoromethanes: chlorine atom-catalyzed destruction of ozone. Nature, 249, 810-812. https://www.nature.com/articles/249810a0

[27] Sarkar, J., Bhattacharyya, S., Ramgopal, M., 2004. Optimization of a transcritical CO_2 heat pump cycle for simultaneous cooling and heating application. Int. J. Refrigeration, 27, 830–838. https://www.sciencedirect.com/science/article/pii/S0140700704000520

[28] Lorentzen, G., 1994. Revival of carbon dioxide as a refrigerant. Int. J. Refrigeration, 17, 292–301. https://www.sciencedirect.com/science/article/pii/0140700794900590

[29] Palacz, M., Smolka, J., Andrzej, J., Nowak, K., Banasiak, K., Hafner, A., 2017. Shape optimisation of a two-phase ejector for CO_2 refrigeration systems. Int. J. Refrigeration, 74 212–223. https://www.sciencedirect.com/science/article/pii/S0140700716303425

[30] Li, D., Groll, E.A., 2005. Transcritical CO_2 refrigeration cycle with ejector expansion device. Int. J. Refrigeration, 28, 766–773. https://www.sciencedirect.com/science/article/pii/S0140700705000022

[31] Deng, J., Jiang, P., Lu, T., Lu, W., 2007. Particular characteristics of transcritical CO_2 refrigeration cycle with an ejector. App. Thermal Eng., 27, 381–388. https://www.sciencedirect.com/science/article/pii/S1359431106002705

[32] Fangtian, S., Yitai, M., 2011. Thermodynamic analysis of transcritical CO_2 refrigeration cycle with an ejector. App. Thermal Eng., 31, 1184–1189. https://www.sciencedirect.com/science/article/pii/S135943111000534X

This is the accepted version of an article published in Thermal Science and Engineering Progress
http://dx.doi.org/10.1016/j.tsep.2020.100626

[33] Sarkar, J., Bhattacharyya, S., Ramgopal, M., 2008. Transcritical carbon dioxide based heat pumps: Process heat applications. International Refrigeration and Air Conditioning conference. Purdue. https://docs.lib.purdue.edu/cgi/viewcontent.cgi?article=1690&conte=iracc

[34] Sawalha, S., 2008 Theoretical evaluation of trans-critical CO_2 systems in supermarket refrigeration. Part 1: Modeling, simulation and optimization of two system solutions. International Journal of Refrigeration, 31, 516-524. https://www.researchgate.net/publication/222683661_Theoretical_evaluation_of_trans-critical_CO2_systems_in_supermarket_refrigeration_Part_I_Modeling_simulation_and_optimization_of_two_system_solutions

[35] Liu, Y., Groll, E.A., Yazawa, K., Kurtulus, O., 2017. Energy saving performance and economics of CO_2 and NH_3 heat pumps with simultaneous cooling and heating applications in food processing: Case studies. Int. J. Refrigeration, 73, 111–124. https://www.sciencedirect.com/science/article/pii/S014070071630295X

[36] Ahammed, E., Battacharyya, S., Ramagopal, M., 2018. Analysis of CO_2 based refrigeration systems with and without ejector for simultaneous pasteurization and chilling of milk, Int. J. Refrigeration, 95, 61-72. https://www.sciencedirect.com/science/article/pii/S0140700718303013

[37] Konstantinos, M.T., Amarisb, C., Mylonaa, Z., Tassou, S., 2018. Analysis of typical booster configuration, parallel-compressor booster configuration and R717/R744 cascade refrigeration system for food retail applications. Part 2: energy performance in various climate conditions. 2nd International Conference on Sustainable Energy and Resource Use in Food Chains, ICSEF 2018, 17-19 October 2018, Paphos, Cyprus. https://www.sciencedirect.com/science/article/pii/S1876610219311713

[38] Hafner, A., Forsterling, S., Banasiak, K., 2014. Multi-ejector concept for R-744 supermarket refrigeration. Int. J. Refrigeration, 43, 1-13. https://www.sciencedirect.com/science/article/pii/S0140700714000668

[39] Singh, S., Dasgupta, M.S., 2017. CO_2 heat pump for waste heat recovery and utilization in dairy industry with ammonia based refrigeration. Int. J. Refrigeration, 78, 108-120. https://www.sciencedirect.com/science/article/pii/S0140700717301044

[40] Elbel, S., Hrnjak, P., 2008. Experimental validation of a prototype ejector designed to reduce throttling losses encountered in transcritical R744 system operation. Int. J. Refrigeration, 31, 411–422. https://www.sciencedirect.com/science/article/pii/S0140700707001508

[41] Liao, S.M., Zhao, T.S., Jakobsen, A., 2000. A correlation of optimal heat rejection pressures in transcritical carbon dioxide cycles. App. Thermal Eng., 20, 831–841. https://www.sciencedirect.com/science/article/pii/S1359431199000708

[42] Mylona, Z., Kolokotroni, M., Tsamos, K.M., Tassou, S.A., 2017. Comparative analysis on the energy use and environmental impact of different refrigeration systems for frozen food supermarket application. Energy Procedia, 123, 121-130. https://www.sciencedirect.com/science/article/pii/S1876610217328059

[43] Polzota, P., Agaroa, P.D., Cortellaa, G., 2017. Energy analysis of a transcritical CO2 supermarket refrigeration system with heat recovery. Energy Procedia, 111, 648 – 657. https://www.sciencedirect.com/science/article/pii/S1876610217302576

[44] Yadav, R.H., Jadhav, V.V., Chougule, G.A., 2016. Exploring the scope of renewable energy technologies in dairy sector. International Journal of Engineering Sciences & Research Technology, 5(7), 439-450. http://www.ijesrt.com/issues%20pdf%20file/Archive-2016/July-2016/50.pdf

[45] Megdouli, K., Ejemani, N., Nahdi, E., Mhimid, A., Kairouani, L., 2017. Thermodynamic analysis of a novel ejector expansion transcritical CO2/N2O cascade refrigeration (NEETCR) system for cooling applications at low temperatures. Energy, 128, 586-600. https://www.sciencedirect.com/science/article/pii/S0360544217306394

[46] Kohler, J., Richter, C., Tegethoff, W., Tischendorf, C., 2007, Experimental and theoretical study of a CO2 ejector refrigeration cycle. In: Presentation at the VDA Alternative Refrigerant Winter Meeting, Saalfelden, Austria. Available from: http://www.vda-wintermeeting.de/.

[47] Banasiak, K., Palacz, M., Hafner, A., Bulinski, Z., Smołka, J., Nowak A.J., Fic, A., 2014. A CFD-based investigation of the energyperformance of two-phase R744 ejectorsto recover the expansion work in refrigeration systems: An irreversibility analysis. International Journal of Refrigeration, 40, 328-337. https://www.sciencedirect.com/science/article/pii/S0140700713003812

[48] Lawrence, N., Elbel, S., 2018. Numerical investigation of the effect of microchannel evaporator design and operation on the improvement potential of ejector refrigeration cycles. Energy, 164, 21-34. https://www.sciencedirect.com/science/article/pii/S0360544218317158.

[49] Minetto, S., Brignoli, R., Banasiak, K., Hafner, A., Zilio, C., 2013. Performance assessment of an off-the-shelf R744 heat pump equippedwith an ejector. Applied Thermal Engineering, 59, 568-575 https://www.sciencedirect.com/science/article/pii/S1359431113004572

[50] Elbel, S., 2011. Historical and present developments of ejector refrigeration systems with emphasis on transcritical carbon dioxide air-conditioning applications. International Jornal of Refrigeration, 34, 1545-1561. https://www.sciencedirect.com/science/article/pii/S0140700710002720

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Declaration of interests

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Credit Author Statement

Performance Evaluation of Ejector Based CO_2 System for Simultaneous Heating and Cooling Application in an Indian Dairy Industry

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