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Performance Investigation of Two Two-Stage Trans-Critical Carbon Dioxide Refrigeration Cycles Ejector and Internal Heat Exchanger

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ABSTRACT

In the present work, the performances of improved two-stage multi inter-cooler trans- critical carbon dioxide (CO_2) refrigeration cycles with ejector and internal heat exchanger have been examined. In the new improved cycles, an internal heat exchanger is append to the cycles. Also, second inter-cooler in improved cycles, cooled with the refrigeration of the cycle, so that in first cycle it is a branch of saturated vapour flow from separator and in second cycle it is a branch of supersaturated steam from internal heat exchanger as well. Results are validated against those available in the literature. Comparisons of the results indicate that there is an excellent agreement between them. The influences of important operational parameters in the cycle performance such as pressure of gas-cooler, temperature of evaporator and temperature of gas-cooler on the performance of cycle have been analysed. The obtained results present that if the cooling flow for second inter-cooler supply from saturated vapour from separator, maximum coefficient of performance can be improved 25% in comparison with the conventional cycle at the considered specific states for operation.

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1. Introduction

Refrigeration cycle is a thermodynamic process whereby a refrigerant accepts and rejects heat in a repetitive sequence.

Owing to universal environmental concerns, the usage of natural working fluid as refrigerant is becoming more interesting to be considered. Trans-critical CO_2 cycle is currently regard as one of the most effective refrigerant for its characteristics such as non-flammability and non-toxicity regardless of the drawback of high working pressure. For years, authors investigated the application of trans-critical CO_2 cycle for refrigeration. Peihuna *et al.* [1] reviewed the flow condensation of CO_2 as a refrigerant. In their study empiric studies of CO_2 condensation have been investigated. Refrigeration expansion in the throttling process, causes that much frictionless heat is dissipated to the refrigerant as a result to the increment of the flow's velocity. So, large kinetic energy enhances as the pressure of refrigerant decreases. In a trans-critical vapour compression refrigeration cycle, the suffocation loss is higher than with conventional refrigerants because of the higher pressure change at the moment of the expansion. Different equivalents and techniques have been handled in order to reduce this loss. In refrigeration cycles one of the most effective part is ejector which is a kind of pump that utilizes the venture effect of a converging-diverging nozzle to modify the energy pressure of an incentive flow to kinetic energy. This action makes a low pressure zoon that causes fluid injection. After transition through the diffuser, the mixed fluid extends and the flow velocity is decreased which conduces to recompress the mixed fluids by converting velocity energy back into pressure energy.

Based on the Denso Corporation [2] investigation, ejector device can enhanced coefficient of performance (COP) of a CO_2 trans-critical cycle with 25% compared with the cooling COP of a convention vapour compression refrigeration cycle. Kornhauser [3] examined the

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performance of the ejector expansion R-12 refrigeration cycle thermodynamically. Results indicate that at ideal conditions and constant mixing pressure in the ejector maximum COP could be improved about 21% over the standard cycle. Domanski [4] obtained that ejector efficiency significantly influence on the performance of the ejector-expansion refrigeration cycle.

Exergy analyses of a trans-critical CO_2 refrigeration cycle with ejector expansion has been done by Tao *et al.* [5]. They separated the exergy destruction rate into two parts consist of endogenous/exogenous and unavoidable/avoidable, and provided valuable data about the interactions between the components of system and the improvement potential of components. Disawas and Wongwises [6] inspected the performance of the ejector-expansion refrigeration cycle without the valve of expansion. Their result showed that at low heat sink temperature, COP improved relative to the convention R-134a refrigeration cycle. Deng *et al.* [7] investigated the effect of the ejector as the principal expansion device. At the operating situation in their investigation, the performance coefficient is improved about up to 18.6% in comparison with the internal heat exchanger cycle and 22% in comparison with the conventional refrigeration cycle. Yari [8] studied a new two-stage ejector-expansion trans-critical CO_2 refrigeration cycle. He applied an internal heat exchanger and intercooler to augment the performance of the cycle. So that, the COP and second law efficiency values are on average 8.6% and 8.15% higher than those of the conventional ejector-vapour compression refrigeration cycle with R-12 as refrigerant. Eskandari and Yavari [9] examined a two-stage multi inter-cooling trans-critical CO_2 refrigeration cycle. At their study, the first intercooler cooled with external coolant and the second one with refrigerant flow. Their results indicate that the performance of the new cycle in the surveyed high-side pressure interval ameliorated about 19.6% compared to original cycle with ejector. Huai *et al.* [10] experimentally surveyed the performance of double-throttling device trans- critical CO_2 ejector- refrigeration system. The first valve of expansion is for control the high-side pressure, and the second valve of expansion was recouped by a two-phase flow ejector. It is for recuperate the system expansion work. Their obtained results indicate that the COP of the improved cycle is improved compared with the traditional system and the maximum increase is up to 32.4%. Abdellaui and Kairouani [11] inspected on a dual evaporator CO_2 trans-critical refrigeration cycle. Their simulation outcomes reveals that the performance of the new cycle improved about 46% compare to the single evaporator system. Bai *et al.* [12] investigated an improved dual-evaporator CO_2 trans-critical refrigeration cycle with two-stage ejector. Obtained results manifest that this system reveals more effective performance betterment than the single ejector in CO_2 dual-temperature refrigeration cycle, and both the maximum COP and exergy efficiency of system enhanced about 37.61% and 31.9% over those of the conventional dual-evaporator cycles.

In the present study, two new improved two-stage trans-critical CO_2 refrigeration cycles with an ejector, internal heat exchanger and multi-intercooler (MIHEC1 and MIHEC2) are introduced. In order to enhance the coefficient of performance, a new idea has been applied. At these new cycles, second intercooler is cooled using an external coolant, which is a portion of saturated vapour coming out of vapour-liquid separator for first system and supersaturated vapour coming out of internal heat exchanger for second system. Moreover, some of the important parameters of refrigeration cycles are obtained and compared with original cycle.

2. Cycle's definitions

Two improved cycles and corresponding pressure-enthalpy diagrams of them are given in figs. 1 and 2. Both cycles have multi-inter-cooling system, an ejector, valve of expansion, internal heat exchanger, separator, gas-cooler and two compressors. The difference between two improved cycles is that in one of them, Second intercooler cools down with saturated vapour flow from separator exit while in other system, second inter-cooler cools with super-heated vapour from internal heat exchanger exit. Original cycle and its related pressure-enthalpy diagram are presented in Fig. 3. Two improved multi-intercooler refrigeration cycles will be introduced as MIEHC1 and MIEHC2 in the following paragraphs and related figures also the original cycle is identified with METSC.

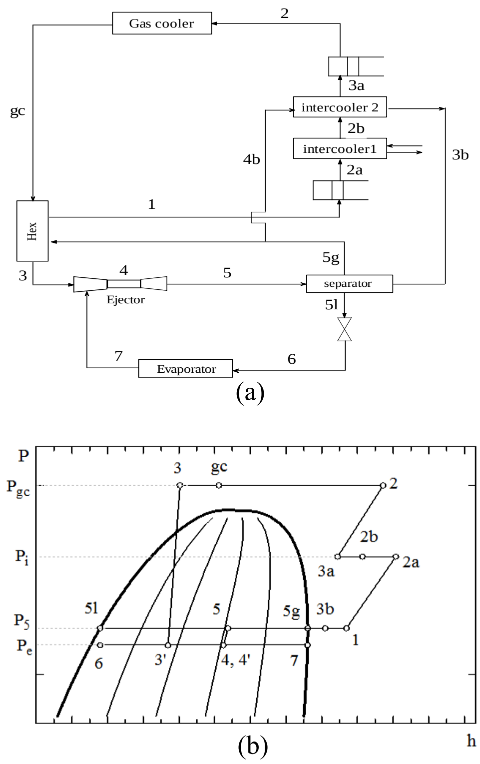


Figure 1. Cycle (a) and pressure-enthalpy (b) diagram of the improved two-stage multi inter-cooling refrigeration cycle, Type1 (MIEHC1).

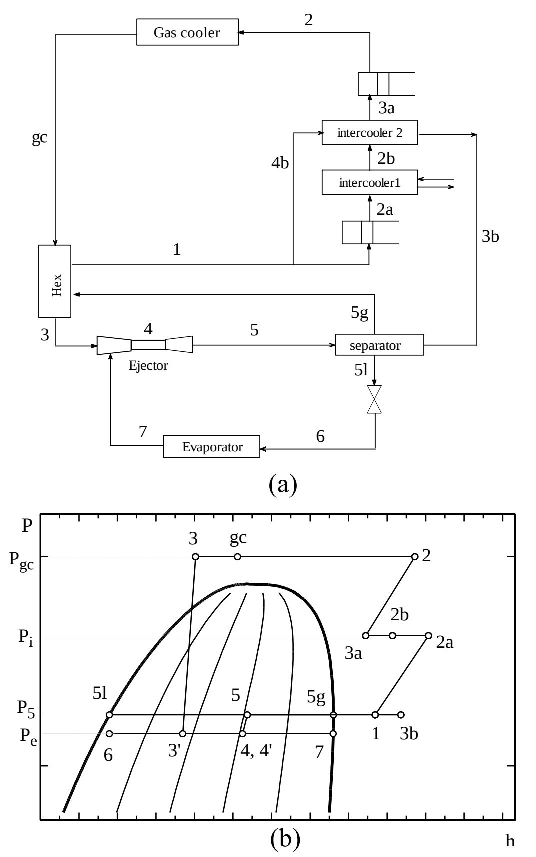


Figure 2. Cycle (a) P and pressure-enthalpy (b) diagram of the improved two-stage multi inter-cooling refrigeration cycle, Type2 (MIEHC2).

As can be seen in Fig. 1, the inlet saturated vapour flow to the first compressor compressed to the intermediate pressure; the compressed fluid moves into the first intercooler and is cooled at constant pressure by the cycle refrigerant flow. The cooled fluid moves into the second intercooler at constant pressure conditions.

Output vapour from second intercooler inters to the second compressor at intermediate pressure PI and is condensed to the outlet pressure P_2, with an isentropic efficiency η_{c2}; the supercritical flow enters to the gas-cooler in a constant pressure and is cooled (T_{gc}). Outlet flow from gas-cooler moves into the internal heat exchanger and cause more sub cooling to the temperature of T_3. Subsequently, the flow enters the nozzle part of the ejector with an efficiency of η_n and expands lower pressure. Second saturated vapour stream moves into the ejector mixing part at pressure of P_e in state 4. The mixture flow then inters through the diffuser, and exit flow recovers to the pressure P_5. Outlet flow from the ejector inters into the separator device and mixed with returned flow from inter-cooler at state (3b) in constant pressure P5. The outlet mixture is divided into two branches saturated liquid flow (state 5l) and saturated vapour (state 5g). The saturated liquid streams enter the valve of expansion and at constant enthalpy expands to pressure P_e, and exit flow from the valve of expansion enters to the evaporator. In MIEHC1 cycle, the saturated vapour is divided in two streams; one of them flows into the internal heat exchanger and the other one moves into the inter-cooler with mass flow rate (ξ). While in MIEHC2 cycle the cooling flow for second inter-cooler is supplied from internal flow to the first compressor. The cooling vapour returned to the inter-cooler, returns to separator at state (3b).

3. Thermodynamic analysis

Based on defined states on fig. 1 for MIEHC1 and fig. 2 for MIEHC2 thermodynamic formulation for improved cycles would be computed.

3. 1 Energy analysis

Steady state condition is assumed in the simulation. The pressure losses in all lines are negligible, adiabatic conditions assumed for compressor, ejector, valve of expansion and separator, the velocity of the refrigerant flow at the inlet and outlet of the ejector are negligible. The pressure of mixture at mixing section of the ejector is constant and equal to the pressure of evaporator. It is contemplated that exit vapour at evaporator has saturated vapour conditions. Total refrigerant flow rate in the cycle is considered 1 kg/s and other flow rates are evaluated according to it. Inlet flow rate to the second inter-cooler is identified by ξ symbol. In this simulation for both new improved cycles, ξ is fixed as $0.4\,Kg/s$. Some the important equations based on first and second law of thermodynamic that applied in the simulation are considered for each component in the related paragraphs.

Compressors:

The specific work of the first compressor is given as,

$$w_{c1} = (h_{2a} - h_1)(\dot{m}_{5g} - \xi) \tag{1}$$

Isentropic efficiency of the compressor is obtained experimental relation as [13]:

$$\eta_{c1} = \frac{1.003P_1 - 0.121\,(P_I)}{P_1} \tag{2}$$

$$\eta_{c1} = (h_{2as} - h_1)/(h_{2a} - h_1) \tag{3}$$

The specific work and isentropic efficiency in the second compressor is similarly [13],

$$w_{c2} = (h_2 - h_{3a})(\dot{m}5g - \xi) \tag{4}$$

$$\eta_{c2} = \frac{1.003P_I - 0.121(P_2)}{P_I} \tag{5}$$

$$\eta_{c2} = (h_{2s} - h_{3a})/(h_2 - h_{3a}) \tag{6}$$

Intercooler:

The optimized intercooler pressure that causes achieving maximum coefficient of performance for cycle is obtained by the geometric mean of inlet pressure to the first compressor and outlet pressure at the second compressor:

$$P_{2a} = P_{3a} = \sqrt{P_1 P_2} \tag{7}$$

First inter-cooler cools with ambient air stream. Inter-cooler effectiveness depends on refrigerant temperatures at the outlet and inlet of it and ambient temperature by Eq. (8).

$$\varepsilon_{IC} = (T_{2a} - T_{2b})/(T_{2a} - T_{amb}) \tag{8}$$

Saturated vapour flow is superheated by 5℃ inside the intercooler [14]. As a result,

$$T_{3b} = T_5 + 5 \tag{9}$$

According to the thermodynamics first law in the second intercooler Eq. (10) is obtained.

For MIEHC1:

$$(\dot{m}_{5g} - \xi)(h_{2b} - h_{3a}) = \xi(h_{3b} - h_{5g})$$

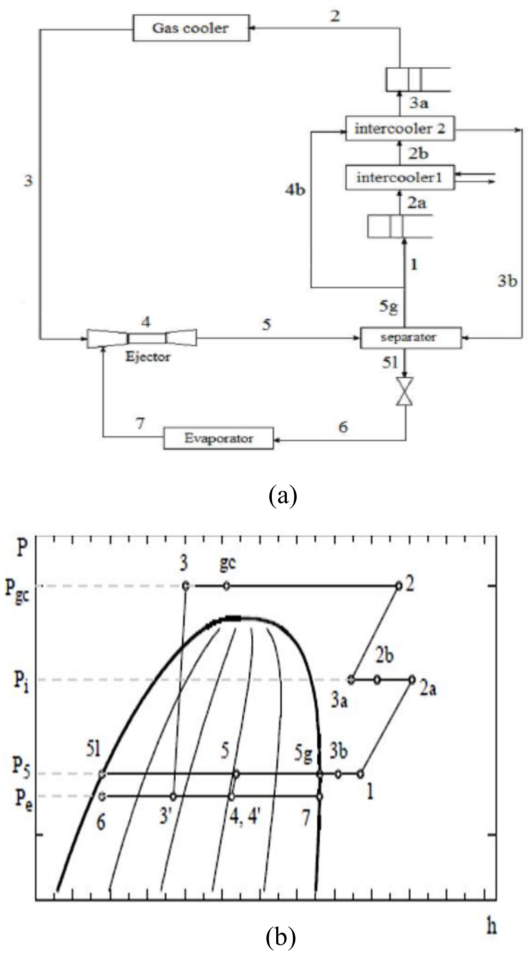
(10)

For MIEHC2:

$$(\dot{m}_{5g} - \xi)(h_{2b} - h_{3a}) = \xi(h_{3b} - h_1)$$

Internal heat exchanger:

The energy equilibrium equation for the internal heat exchanger for two cycles are as Eqs (11) and (12).



(a)

(b)

Figure 3. Cycle (a) and pressure-enthalpy (b) diagram of the two-stage multi inter-cooling refrigeration cycle with ejector, Original cycle (MEST).

For MIEHC1:

$$h_{gc} - h_3 = h_1 - h_{5g} \tag{11}$$

For MIEHC2:

$$(\dot{m}_{5g} - \xi)(h_{gc} - h_3) = \xi(h_1 - h_{5g})$$

The effectiveness of the internal heat exchanger is achieved from Eq. [14].

$$\varepsilon_{Hex} = (h_1 - h_{5g})/(h_{gc} - h_{5g}) \tag{12}$$

Ejector:

The driver stream moves into the ejector and expands with a nozzle which its efficiency is determined by Eq. (13),

$$\eta_n = (h_3 - h_{3'})/(h_3 - h_{3',s}) \tag{13}$$

The energy equilibrium in the ejector that two flows are joined each other is,

$$h_3 - h_{3'} = u_{3'}^2/2 \tag{14}$$

The momentum conservation in mixing section of the ejector is obtained as (more detail is given in Appendix),

$$(1 + \dot{m}_{5l}/(\dot{m}_{5g} - \xi))u_4 = u_{3'} \tag{15}$$

The efficiency in the mixing part of the ejector is defined as Eq.(16), [15]:

$$\eta_m = u_{4'}^2/u_4^2 \tag{16}$$

The equation of energy balance between states (4) and (5) is written as Eq. (17). In this equation, $u_{4'}$ is the corrected form of u_4, due to the account for losses of mixing section,

$$h_5 - h_4 = u_{4'}^2/2 \tag{17}$$

Using the energy balance for the ejector, Eq. (18) is given as follows,

$$(1 - \xi)h_5 = (\dot{m}_{5g} - \xi)h_3 + (\dot{m}_{5l})h_7 \tag{18}$$

The efficiency of the diffuser is described as the ratio of the enthalpy change that situate between the entrances to exit stagnation pressure to the kinetic energy.

$$\eta_d = (h_{4s} - h_4)/(h_5 - h_4) \tag{19}$$

Separator:

The energy balance in the separator is expressed as Eq. (20):

$$\xi h_{3b} + (1 - \xi)h_5 = (\dot{m}_{5g})h_{5g} + (\dot{m}_{5l})h_{5l} \tag{20}$$

Valve of expansion:

Refrigerant flow pressure reduced to the evaporator pressure by valve of expansion. The changes of the kinetic energy through valve of expansion can be neglected to obtain Eq. (21),

$$h_{5l} = h_6 \tag{21}$$

Evaporator:

The cooling capacity of the evaporator is,

$$q_e = (h_7 - h_6)(\dot{m}_{5l}) \tag{22}$$

The performance of system is appraised by coefficient of performance (COP). It is the ratio of the cooling capacity of the cycle to the total work of the compressors.

$$\mathrm{COP} = q_e/(w_{c1} + w_{c2}) = (h_7 - h_6)(\dot{m}_{5l})/(h_{2a} + h_2 - h_1 - h_{3a})(\dot{m}_{5g} - \xi) \tag{23}$$

3.2 Exergy analysis

Entropy generation rate for a fixed control volume is given as [16]:

$$\dot{s}_{gen} = \sum \dot{m}_o s_o - \sum \dot{m}_i s_i - \sum \frac{\dot{Q}_{C.V}}{T_K} \tag{24}$$

The exergy destruction rate can be acquired from Eq. (25) [17]:

$$I = T_0 \dot{s}_{gen} \tag{25}$$

For all states, the equation of the exergy destruction rate for each part of the cycle can be obtained as follow.
For compressors,

$$I_{c1} = T_o(s_{2a} - s_1)(\dot{m}_{5g} - \xi) \tag{26}$$

$$I_{c2} = T_o(s_2 - s_{3a})(\dot{m}_{5g} - \xi) \tag{27}$$

For inter-coolers,

$$I_{IC1} = \left(h_{2a} - h_{3a} - T_0(s_{2a} - s_{2b})\right)(\dot{m}_{5g} - \xi) \tag{28}$$

For MIEHC1:

$$I_{IC2} = \left(T_0(s_{3a} - s_{2b})\right)(\dot{m}_{5g} - \xi) + \left(T_0(s_{3b} - s_{5g})\right)\xi$$

For MIEHC2:

$$I_{IC2} = \left(T_0(s_{3a} - s_{2b})\right)(\dot{m}_{5g} - \xi) + \left(T_0(s_{3b} - s_1)\right)\xi \tag{29}$$

For gas-cooler,

$$I_{gc} = (h_2 - h_{gc} + T_0(s_{gc} - s_2))(\dot{m}_{5g} - \xi) \tag{30}$$

For internal heat exchanger,

$$I_{Hex} = T_0(s_3 + s_1 - (s_{gc} + s_{5g}))(\dot{m}_{5g} - \xi) \tag{31}$$

For ejector,

$$I_{ej} = T_0((\dot{m}_{5g} + \dot{m}_{5l} - \xi)s_5 + (\xi - \dot{m}_{5g})s_3 - (\dot{m}_{5l})s_7) \tag{32}$$

For evaporator,

$$I_e = T_0(\dot{m}_{5l})(s_7 - s_6 + (h_6 - h_7)/T_r)(\dot{m}_{5l}) \tag{33}$$

For valve of expansion,

$$I_{EV} = T_0(\dot{m}_{5l})(s_6 - s_{5l}) \tag{34}$$

Sum of the exergy destruction rate in each ingredient equal to the total exergy destruction rate. So,

$$I_t = I_{c1} + I_{c2} + I_{Ic1} + I_{Ic2} + I_{gc} + I_{Hex} + I_e + I_{EV} + I_{ej} \tag{35}$$

According to the analytical analysis, a thermodynamic simulation program for new two-stage multi-inter-cooling trans-critical CO_2 refrigeration cycle with ejector and internal heat exchanger was expanded. Table 1 outlines the initial presumption and main parameters of the simulation and analysis of system. We consider the thermodynamic conditions of the present cycle match to the cycle which was studied by Yari [7]. Notice that the mass flow rate of outlet saturated vapour from separator change with changing the parameters of the cycle.

4. Results and discussions

In order to validate the results, in a particular operational condition of the base cycle at gas cooler temperature of 36℃, COP versus pressure of gas-cooler and the entrainment ratio in the ejector for three distinct

Table 1. The parameters utilized in the present work

Parameters	Value
T_{amb}	27 °C
T_e	-25 to 0 °C
T_{gc}	35-45 °C
T_0	300K
T_r	$(T_e + 5)$K
P_{gc}	80-120 bar
ε_{Hex}	80%
η_d	80%
η_m	95%
η_n	80%

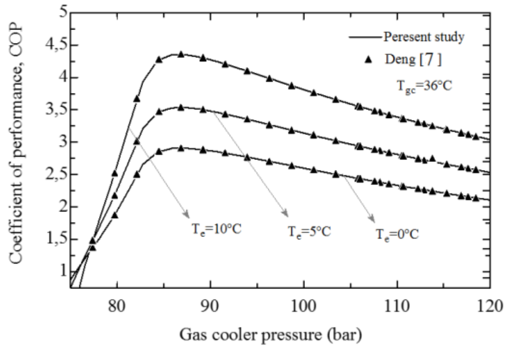
temperatures of evaporator is calculated as well. As indicated in figs. 4a and 4b, there is perfect agreement between two set of results. It should be exhibited that entrainment ratio in the ejector modifieswith variation of gas cooler pressure.

Figure 5 compares the COP of three cycles under the different pressures of gas-cooler. It can be observed that with increasing of the pressure of gas-cooler, COP at first increases and then decreases, so it has an optimum value according to maximum COP. Moreover, from the above comparison, it can be found that internal heat exchanger can enhance COP of the two-stage multi-inter-cooling ejector-expansion system. Results show that if second intercooler cooled with statured vapour from separator's exit, cycle has better performance than when it cooled with supersaturated steam from internal heat exchanger's exit. The reason for it, associated to variations of specific cooling capacity and specific work versus the pressure of gas-cooler at both the determined gas-cooler and evaporator temperatures. Also this figure shows that MIEHC2 cycle has higher COP compared to METSC cycle at low gas-cooler pressure. After some particular gas-cooler pressure, COP of the first improved cycle advances towards the COP of the original cycle. Figure 6 compares the coefficient of performance of the three cycles compared with the temperature of gas-cooler at different evaporator temperatures when pressure of gas-cooler is equal to 100 bar. It obviously indicates that, coefficient of performance of the MIEHC1 cycle is consistently larger than that of the original cycle and MIEHC2 at the determined pressure of gas-cooler. This difference enhanced as temperature of gas-cooler increases. Also, the figure emphasis that COP has high dependency on temperatures of both the gas-cooler and evaporator, such that as temperature of gas-cooler increase or temperature of evaporator reduced causes the COPs of the cycles decrease.

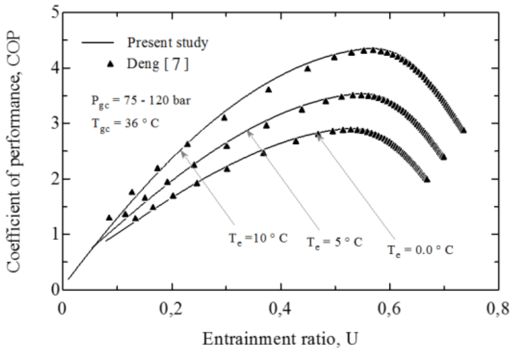
Figure 7 illustrates the comparison of the specific work vs. temperature of gas-cooler for different temperatures of evaporator at gas-cooler pressure of 100 bar. As shown in this figure, as temperature of gas-cooler or temperature of evaporator increases, the specific work of the cycle decreases. Furthermore, it can be found that the sensitivity of MIEHC1 and MIEHC2 cycles to gas-cooler temperature changing are less than original cycle.

Figure 8 exhibits the comparison of the specific cooling capacity vs. temperature of gas-cooler for various temperatures of evaporator at gas-cooler pressure of 100 bar. In addition, in all ranges under investigation for gas-cooler temperature MIEHC1 cycle has the most specific cooling capacity than other two cycles. Moreover, as temperature of evaporator increases, specific cooling capacity of the cycles decreases. Meantime as temperature of gas-cooler increases causes specific cooling capacity of the cycles decreases. As the previous results in fig. 6 indicated, in all three cycles coefficient of performance decrease with increasing gas-cooler temperature. Its reason can be found from comparison fig. 7 and also fig. 8. It is clear that with enhancing gas-cooler temperature the rate of the reduction of specific cooling capacity is greater than its rate of specific work and the coefficient of performance is the ratio between these two parameters.

The COP variation of the two improved cycles versus gas-cooler pressure at constant gas-cooler temperature in one part and also for a fixed evaporator temperature in another part are given if figs. 9 and 10. From this results, it can be obtained that pressure of gas-cooler corresponding to maximum COP for the constant gas-cooler temperature of 40℃ is about 92 bar. Eke for the constant evaporator temperature of -25℃, as temperature of gas-cooler increases, the pressure of gas-cooler related to maximum COP increases too.



(a)



(b)

Figure 4. Analogy between present and Deng's results [7] (a) COP vs. pressure of gas-cooler, (b) COP vs. entrainment ratio.

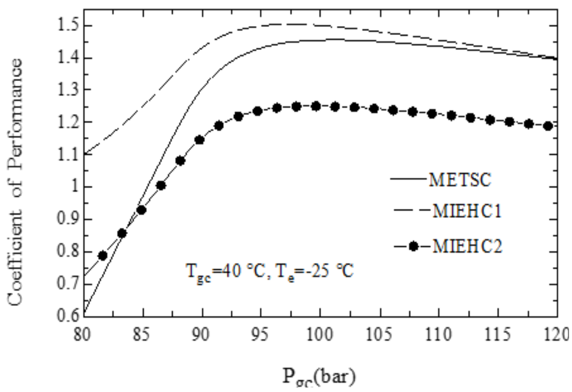


Figure 5. COP comparison of the improved and original cycles vs. pressure of gas-cooler at constant gas-cooler and evaporator temperature.

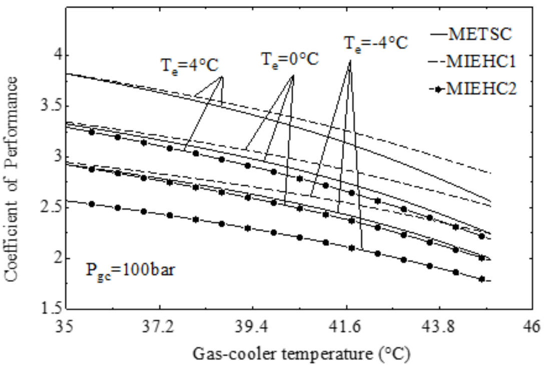


Figure 6. COP comparison of the improved and original cycles vs. temperature of gas-cooler at constant pressure of gas-cooler for various evaporator temperatures.

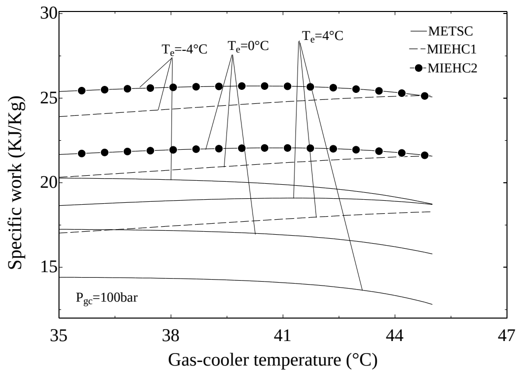


Figure 7. Specific work comparison of the improved and original cycles vs. temperature of gas-cooler at constant pressure of gas-cooler for various evaporator temperatures.

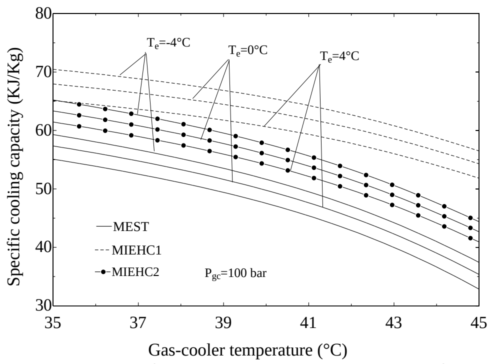


Figure 8. Specific cooling capacity comparison of the improved and the original cycles vs. temperature of gas-cooler at constant pressure of gas-cooler for different evaporator temperatures.

Figure 11 illustrated the alteration of the exergy destruction rates of the cycles against the pressure of gas-cooler at constant operations of both gas-cooler and evaporator. The figure clearly reveals that the total exergy destruction rate of the improved cycles is consistently higher than that of the original cycle. Also at pressure of gas-cooler less than 98 bar, MIEHC1 has more exergy destruction rate than MIEHC2 but as pressure grows from about 98 bar its rate at MIEHC2 cycle is greater.

The temperature of refrigerant at the first compressor enhanced as a result of adding an internal heat exchanger to the original cycle. It imposes high heat load to the compression process and therefor enhances exergy destruction rate in the ingredients of compression process. Internal heat exchanger induces augmentation exergy destruction as well. However, the internal heat exchanger because of reduction of the refrigerant temperature at the inlet of expansion system decreases the exergy destruction rate within the ingredients of cycle. A quantitative item should be introduced for comparison purposes. In this study, a parameter is introduced as ϵ_{ov} . It is the ratio of the COP improvement to the percent of increasing in the total exergy destruction. It is called as "overall correction" and is given by the Eq. (36).

$$\epsilon_{ov} = \frac{(COP_{new} - COP_{original})/COP_{original}}{(I_{t,new} - I_{t,original})/I_{t,original}} \tag{36}$$

Pursuant to Eq. (36), if overall correction is higher than 1.0, it means that the internal heat exchanger causes improvement in the performance of cycle. Giving consideration to previous figures, overall betterment for MIEHC1 is averagely 1.23 and when the pressure of gas cooler is equal to 90 bar, which stipulates that the improved refrigerant cycle has better performance at low pressure of gas cooler. In MIECH2, ϵ_{ov} at low pressure of gas cooler (about 80 bar) is averagely 0.6.

Both the coefficient of performance and total exergy destruction rate at temperature of gas-cooler of 40℃ and evaporator temperature of -25℃ for three different gas cooler pressures that usually accrued in refrigeration industrial are listed in table 2.

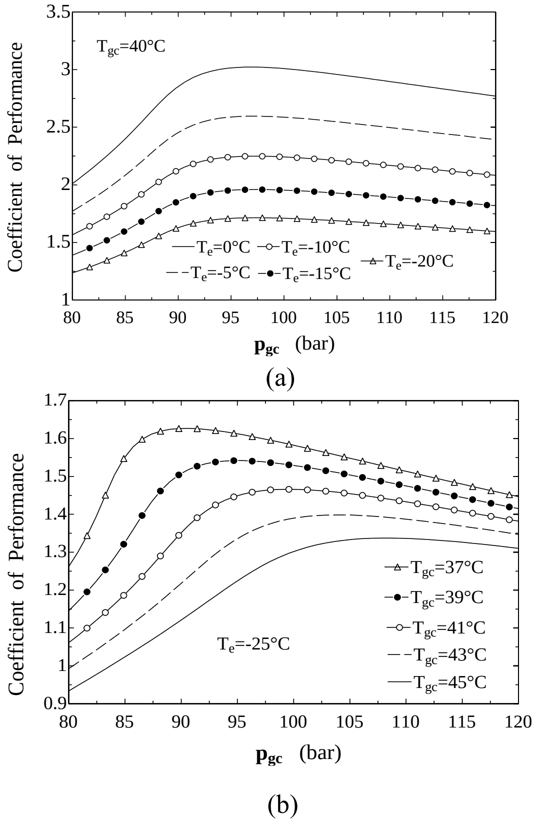
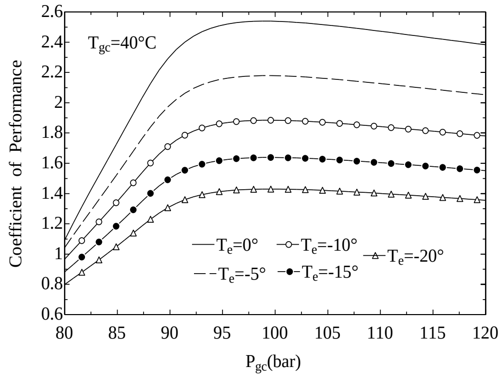
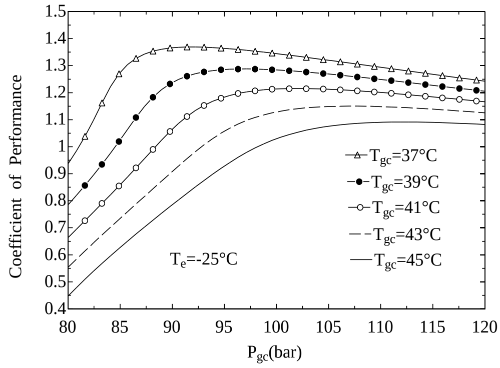


Figure 9. COP changes vs. pressure of gas-cooler (a) for various temperatures of evaporator at constant gas-cooler temperature (b) for diverse temperatures of gas-cooler at constant temperature of evaporator, in MIEHC1 cycle.



(a)



(b)

Figure 10. COP changes vs. pressure of gas-cooler (a) for various evaporator temperature at constant gas-cooler temperature (b) for various gas-cooler temperatures at constant temperature of evaporator, in MIEHC2 cycle.

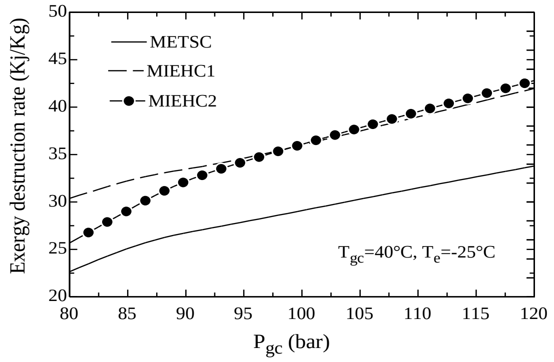


Figure 11. Exergy destruction rate modification of the improved and original cycles vs. pressure of gas-cooler at constant gas-cooler and evaporator temperatures.

Table 2. The coefficient of performance and total exergy destruction rate at Tgc= 40℃ and Te= -25℃

Cycle	**Coefficient Of Performance (COP)**			**Exergy destruction rate (Kj/Kg)**		
	P_{gc}= 80bar	P_{gc}= 100bar	P_{gc}= 115bar	P_{gc}= 80bar	P_{gc}= 100bar	P_{gc}= 115bar
METSC	0.6	1.44	1.4	22.5	27.9	31.8
MIEHC1	1.1	1.5	1.42	30.8	35.8	38.9
MIEHC2	0.73	1.23	1.18	25.9	35.7	40

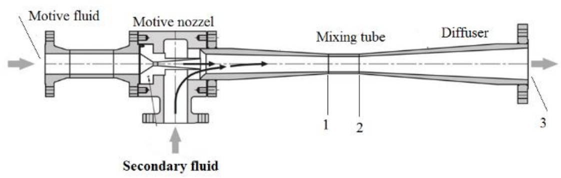


Figure 12. Ejector

5. Conclusion

Comparative study on the coefficient of performance and specific cooling capacity of the improved cycles indicates improvement, if an internal heat exchanger be added to the cycle and the saturated vapour from separator is used as cooling flow at second inter-cooler. At constant pressure of 100 bar in gas-cooler that is a conventional pressure in refrigeration cycles and in all evaporator temperatures that in this study are considered, as temperature of gas-cooler increases, the performance of the first improved cycle (MEEHC1) further improve to the original cycle. Also, from the results of this study, it is obtained that using of output supersaturated flow from internal heat exchanger at second inter-cooler isn't proper for cycle performance improvement. Affixing an internal heat exchanger, although can improve COP of the first improved cycle, but it enhances total exergy destruction rate at same conditions in both improved cycles. In this paper, the effect of some important parameters in two improved cycles are studied too and the pressure of gas-cooler related to optimum COP value is obtained. Its value for MIEHC1 at base predestined conditions (T_{gc}=40℃, T_e=-40℃) is about 92 bar whereas for MIEHC2 cycles is about 94 bar.

Appendix

The control volume between sections 3 and 5 in fig. 2a is separated into two regions, 1–2 and 2–3 that are shown in fig. 12.

Nozzle:

The outlet velocity of the nozzle is given by:

$$u_1 = \sqrt{2\eta_n (h_e - h_1)}$$

where h1 is the flow enthalpy at the motive nozzle outlet, considering the isentropic process.

$$h_1 = h(s_e, P_1)$$

The ejector mass flow rate is:

$$\dot{m}_{gc} = \rho_1 u_1 A_1$$

Flow in the mixing chamber:

The mass flow rate in the mixing part in the ejector is computed as:

$$\dot{m}_{gc} + \dot{m}_7 = \rho_2 u_2 A_2$$

The relation between pressure and velocity in the ejector can be obtained from the momentum equilibrium.

$$(P_2 - P_1)A_2 = \dot{m}_{gc}\, u_1 - (\dot{m}_{gc} - \dot{m}_7)u_2$$

The pressure variations in the mixing section is obtained from the following equation:

$$\frac{P_2 - P_1}{\frac{1}{2}\rho_1 u_1^2} = 2\frac{A_1}{A_2} - 2(U+1)^2 \frac{\rho_1}{\rho_2}(\frac{A_1}{A_2})^2$$

$$U = \dot{m}_7 \Big/ \dot{m}_{gc}$$

$\rho_1 \big/ \rho_2$ Can be approximate as [18];

$$\frac{\rho_2}{\rho_1} = \frac{U}{1+U}\frac{\rho_\upsilon}{\rho_1} + \frac{1}{1+U}$$

where ρ_v is the density of the refrigerant flow at the evaporator outlet. Refrigerant velocity at the mixing part of the ejector is determined as follows:

$$u_2 = \frac{1}{1+U} u_1$$

where U is the ejector entrainment ratio which is the ratio of the ejector suction mass flow rate to the stimulus mass flow rate [8, 15]. In this simulation, total mass flow rate is equal to 1 Kg/s.

$$m_{5g} + m_{5l} + \xi = 1$$

$$m_7 = m_{5l} = suction\, mass\ flow\, rate$$

$$m_{5g} - \xi = motive\, mass\ flow\, rate$$

Furthermore, in this simulation, ξ is assumed to be constant and is equal to 0.4Kg/s, and from energy balance in Eq. 18, we obtain:

$$m_{5g} + m_{5l} = 0.6$$

$$(m_{5g} - 0.4)h_3 + m_{5l} h_7 = 0.6 h_5$$

Where m_{5g} and m_{5l} are obtained by solution of above two equations.

Nomenclature

amb	Ambient
COP	Coefficient of Performance
H	Specific enthalpy [KJ/Kg]
Hex	Internal heat exchanger
I	Exergy destruction rate [KJ/Kg]
M	Mass flow rate [Kg/s]
P	Pressure [bar]
q	Specific cooling capacity [KJ/Kg]
S	Specific entropy [KJ/Kg.K]
T	Temperature [°C]
U	Velocity [m/s]
W	Specific work [KJ/Kg]

Subscripts

0	Reference environment
c	Compressor
d	Diffuser
e	Evaporator
ej	Ejector
ev	Valve of expansion
g	Saturated vapour
gc	Gas-cooler
gen	Generation
I	Intermediate
IC	Inter-cooler
L	Saturated liquid
mix	Mixing
n	Nozzle

Greek letters

ξ	Cooling mass flow rate in second Inter-ooler [Kg/s]
η	Efficiency [%]
ε	Inter-cooler effectiveness [%]

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