Thermodynamic analysis of transcritical R170 refrigeration cycle with internal heat exchanger

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***Abstract*— Energy as well as exergy analysis have been performed in this article. The natural refrigerant ethane (R170) has been used in the basic transcritical vapor compression refrigeration cycle because it's ODP (Ozone layer depletion potential) is zero and GWP (Global Warming potential) is very low. Thermodynamic modelling and simulation have been used to calculate the maximum COP, Optimum discharge pressure and second law efficiency of the basic transcritical vapor compression cycle. Furthermore, the internal heat exchanger is also used in the basic cycle and its effect is also studied. It is observed that the maximum COP, second law efficiency increases on using the internal heat exchanger in the basic cycle, whereas its use decreases the optimum discharge pressure significantly. Therefore, the present study reveals that the use of an internal heat exchanger in the basic transcritical vapor compression cycle with refrigerant ethane is advisable for the improvement of maximum COP and second law efficiency under-considered operating conditions.**

Keywords— Transcritical refrigeration cycle, maximum COP, second law efficiency, internal heat exchanger, Optimum discharge pressure.

INTRODUCTION

As the presence of natural resources in nature is limited, energy saving has always been an important topic of concern. The idea of refrigeration performance enhancement is used to reduce energy consumption. The Vapor compression refrigeration system is mainly used in refrigeration. The basic transcritical vapor compression refrigeration system incorporates the compressor, gas cooler, throttle valve or expansion device and evaporator.

Although, we use synthetic refrigerants in the refrigeration systems but they cause ozone layer depletion and global warming, so to mitigate the problem of ozone layer depletion and global warming, the use of natural refrigerant has increased interest confirming its eminent heat transfer characteristics and nontoxic property [1]. Ethane (R170) is a natural non-toxic refrigerant and has zero ODP (Ozone Depletion Potential) and a very low GWP (Global Warming Potential). Ethane is a hydrocarbon, good natural refrigerant, which can be used in refrigeration systems [2]. Transcritical cycle's COP depends on the high side pressure so it requires an optimum value of the heat rejection pressure. Its optimum value is generally associated with the gas-cooler exit and evaporation temperatures. Nemati et al. [3] carried out an extensive comparison between CO_2 and ethane as a refrigerant in a two-stage ejector-expansion transcritical refrigeration cycle integrated with an organic Rankine cycle (ORC) and concluded that the COP and second law efficiency for the ethane refrigerant are about 9.37 and 9.43% greater than that of the CO_2. Manoj et al. [4] studied the substitute refrigerant for Freon 22 in Vapour Compression Refrigeration System and found that hydrocarbon refrigerant have good performance characteristics compared to R22 and concluded that Hydrocarbon can be used to replace the R22 in vapour compression refrigeration system in future. Kasi [5] used refrigerant ethane in a cascade refrigeration system to achieve the very low temperature of -70˚C to -50˚C. Nasruddin et al. [6] have shown that ethane can help to reach a low-temperature of around -80˚C. Gong et al. [7] have shown that the COP of R508B is 10% lesser than the R170 + R116 binary mixture. Moreover, to explore the alternatives of conventional refrigerants, a three-stage cascade refrigeration system (TCRS) have been studied by sun et al. [8]. The results show that R14 can be replaced by R1150 in the low-temperature cycle. In the medium-temperature cycle, R23 can be replaced by R41 and R170. Miran et al. [9] proposed a transcritical refrigeration cycle integrated with mechanical sub-cooler using three different refrigerants ethane, CO_2, and N_2O. It was observed that the application of sub-cooler improved the COP of ethane by 36.1%, COP of N_2O and CO_2 by 26.48% and 30.74% respectively.

The above literature shows that ethane can be used in vapor compression refrigeration cycles. Hence, in the present work, energy and exergy analysis of basic transcritical vapor compression cycle and transcritical vapor compression cycle with internal heat exchanger have been carried out using refrigerant ethane.

THERMODYNAMIC MODELLING AND SIMULATION

In Fig 1, the cycle layout of the basic transcritical R170 vapor compression refrigeration cycle is shown.

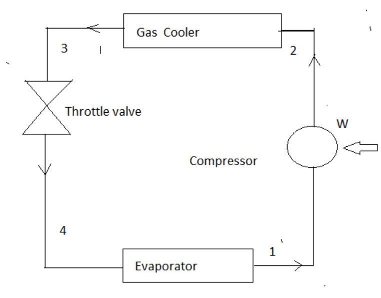

Fig 1. Cycle layout of a transcritical R170 refrigeration cycle

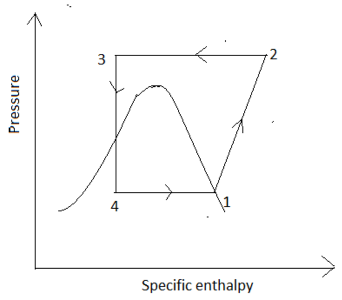

Fig 2 Diagram of pressure and enthalpy of a transcritical R170 cycle

The saturated ethane vapor from state 1 is compressed to the superheated state 2 in the compressor and then cooled from (2-3) in a gas cooler by dissipating heat to the atmosphere. From state 3, ethane is expanded in the throttle valve (3-4) and then passes through the evaporator (4–1) to provide a useful cooling effect.

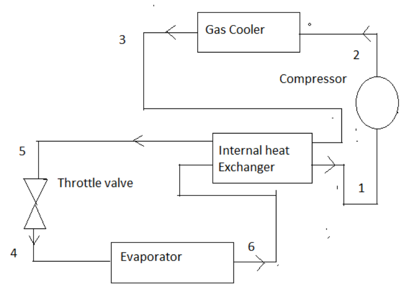

Fig 3. Cycle layout of transcritical R-170 refrigeration cycle using an internal heat exchanger

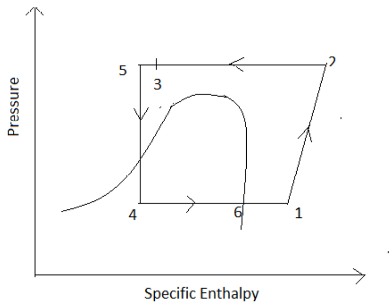

Fig 4. P-h diagram of a transcritical R170 refrigeration cycle using an internal heat exchanger

The cycle layout and correlating P-h diagram of a basic transcritical R170 cycle using an internal heat exchanger are shown in Figures 3 & 4 respectively. As shown, the ethane saturated vapor from state 6 get superheated to state 1 in the internal heat exchanger and from then state 1, it is compressed in the compressor to state 2 and changes to superheated gas. Then the superheated gas from state 2 get cooled in the gas cooler to state 3 by dissipating heat to the atmosphere. The high-pressure gas get sub-cooled from 3 to 5 in the internal heat exchanger and then from there it is expanded in the expansion device to state 4, which is the inlet to the evaporator.

The thermodynamic modelling of the transcritical vapor compression cycle using ethane has been presented. Some of the assumptions used in the thermodynamic modelling are as follows:

1. Only Gas cooler transfers the heat with the environment.

2. There is irreversible adiabatic compression in the compressor.

3. All the processes inside the evaporator, internal heat exchanger and, gas cooler are isobaric.

4. At evaporator exit, the refrigerant is dry saturated.

5. The expansion inside the expansion valve is isenthalpic.

For the basic expansion cycle (Fig. 1), the specific work input to the compressor is given by:

$$w_c = (h_2 - h_1) \quad (1)$$

From the evaporator, the specific refrigerating effect obtained is given as:

$$q = (h_1 - h_4) \quad (2)$$

Hence, the cooling COP is expressed by

$$cop = \left(\frac{q}{w_c}\right) \quad (3)$$

Volumetric cooling capacity:

$$V_c = q\rho \quad (4)$$

For exergetic performance, applying the second law of thermodynamics to each and every component, the following relations can be established:

(i) Compressor irreversibility:

$$i_c = T_o(s_2 - s_1) \quad (5)$$

(ii) Throttling device irreversibility:

$$i_{ed} = T_o(s_4 - s_3) \quad (6)$$

(iii) Gas cooler irreversibility

$$i_{gc} = (h_2 - h_3) - T_o(s_2 - s_3) \quad (7)$$

(iv) Evaporator irreversibility

$$i_{ev} = T_o(s_1 - s_4) - q\frac{T_o}{T_{ev}} \tag{8}$$

Second law (exergy) efficiency for the system can be expressed as the ratio of net exergy output to the exergy input to the compressor, i.e.:

$$\eta_{ii} = \frac{w_c - (i_c + i_{ed} + i_{ev} + i_{gc})}{w_c} \tag{9}$$

For the R170 transcritical cycle with an internal heat exchanger, the specific work input to the compressor is expressed by:

$$w_c = (h_2 - h_1) \tag{10}$$

From the evaporator, the specific refrigerating effect obtained is expressed as:

$$q = (h_6 - h_5) \tag{11}$$

Hence, the cooling COP:

$$cop = \frac{q}{w_c} \tag{12}$$

(i)Throttling device irreversibility:

$$i_{ed} = T_o(s_4 - s_5) \tag{13}$$

(ii) Evaporator irreversibility:

$$i_{ev} = T_o(s_6 - s_4) - q\frac{T_o}{T_{ev}} \tag{14}$$

(iii) Heat exchanger irreversibility can be given as :

$$i_{he} = T_o\left((s_1 - s_6) - (s_3 - s_5)\right) \tag{15}$$

For the internal heat exchanger, the effectiveness and energy balance can be given by equation (16) & (17) respectively.

$$\varepsilon_{ihe} = \frac{T_1 - T_6}{T_3 - T_6} \tag{16}$$

$$h_1 - h_6 = h_3 - h_5 \tag{17}$$

RESULTS AND DISCUSSION

A computer program in Engineering Equation Solver [10] is developed for the energy and exergy analysis of the transcritical ethane refrigeration cycle at the different operating conditions, compressor efficiency is presumed to be 75% in the present study. The secondary fluid temperature in the evaporator is assumed to be 5°C above the evaporator temperature and ambient temperature is considered to be 25˚C.

VAPOUR COMPRESSION REFRIGERATION SYSTEM (BASIC CYCLE)

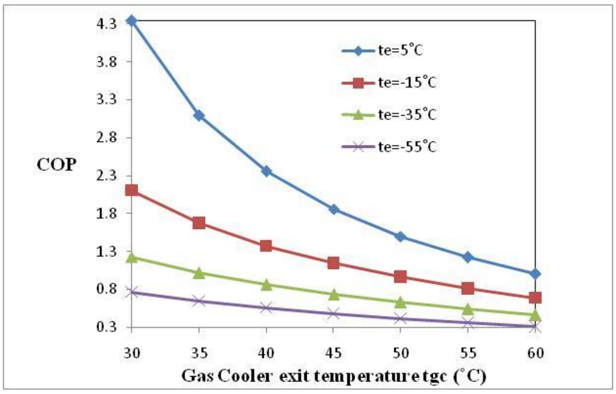

Fig.5 Deviation of COP with Gas cooler exit temperature

In Fig.5 the deviation of COP with the gas cooler exit temperature is shown. The range of the gas cooler exit temperature is taken from 30˚C to 60˚C. The result shows that when the evaporator temperature increases and gas cooler exit temperature decreases, the COP increases. The COP is maximum corresponding to the gas cooler exit temperature of 30˚C and evaporator temperature of 5˚C, whereas it is minimum corresponding to gas cooler exit temperature of 60˚C and evaporator temperature of -55˚C

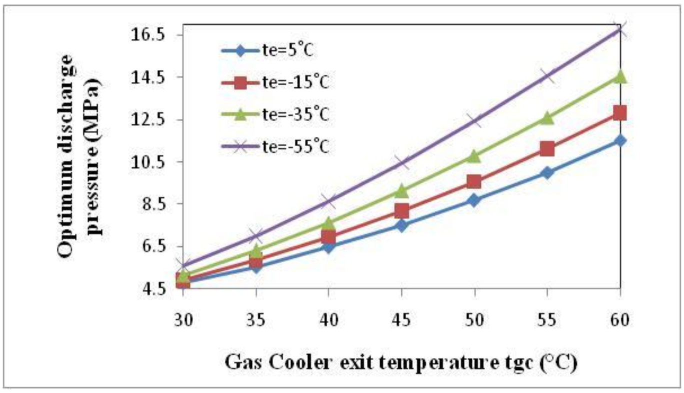

Fig. 6 Deviation of Optimum discharge pressure with gas cooler exit temperature

Fig. 6 shows the pressure deviation along the ordinates and gas cooler exit temperature along abscissa, the value of the optimum discharge pressure is obtained at different operating conditions of the gas cooler exit temperature ranging from 30°C to 60°C, it is clear from the graph that there is increase in the optimum discharge pressure on decreasing the evaporator temperature i.e. 5°C, -15°C, -35°C and -55°C respectively.

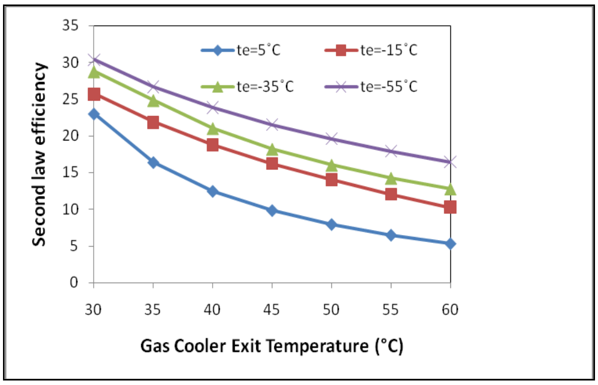

Fig. 7 Deviation of second law efficiency with gas cooler exit temperature

The deviation between the second law efficiency and the gas cooler exit temperature is shown at different evaporator temperature values in Fig.7.When the evaporator and gas cooler exit temperatures decreases the second law efficiency increases. The second law efficiency is maximum corresponding to the gas cooler exit temperature of 30˚C and evaporator temperature of -55˚C, whereas it is minimum corresponding to the gas cooler exit temperature of 60˚C and evaporator temperature of 5˚C.

COMPARISON BETWEEN THE BASIC TRANSCRITICAL ETHANE CYCLE AND BASIC TRANSCRITICAL ETHANE CYCLE WITH AN INTERNAL HEAT EXCHANGER:

The transcritical R170 cycle performance is improved with an internal heat exchanger, which can be easily observed from Fig.8. Deviation of COP with the gas cooler exit temperatures of transcritical R170 cycle with and without heat exchanger is compared at different evaporator values. The increment elevates from 4.8% at 30°C exit temperature to about 35.65% at exit temperature value of 60°C with evaporator temperature set at 5°C, whereas at fixed -35°C, the COP percentage increment goes from 12.62% at 30°C exit temperature to 47.02% at 60°C exit temperature.

——— Solid line shows basic cycle with IHE

---------- Dotted line shows basic cycle without IHE

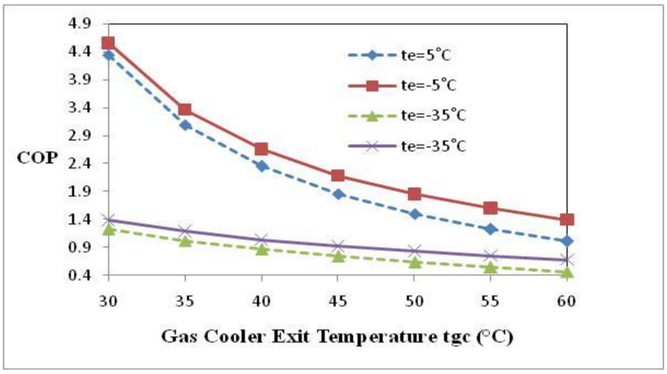

Fig.8 Deviation of COP with Gas cooler exit temperature

——— Solid line shows basic cycle with IHE

--------- Dotted line shows basic cycle without IHE

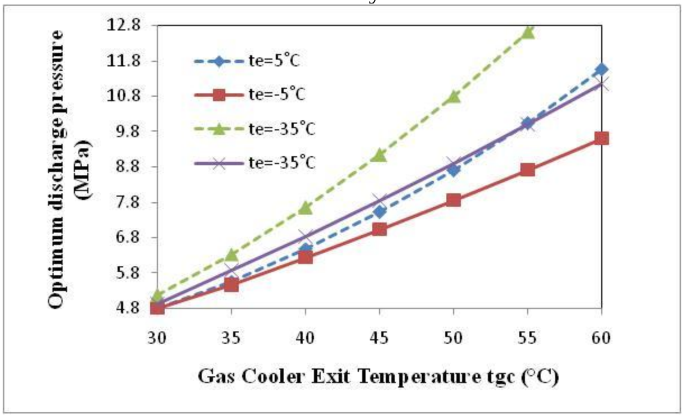

Fig. 9 Deviation of optimum discharge pressure with gas cooler exit temperature

It is clear from Fig. 9 that optimum discharge pressure is reduced with the employment of internal heat exchanger in the transcritical ethane cycle. Optimum discharge pressure variation with gas cooler exit temperature of the transcritical ethane cycle is compared with and without heat exchanger at fixed evaporator values. The optimum discharge pressure is comparable at gas cooler exit temperature of about 30˚C but with higher gas cooler exit temperature the difference seems to be notable with optimum discharge pressure reduced about 16.8% at 5°C and 23.6% at -35°C evaporator temperature.

——— Solid line shows basic cycle with IHE

---------- Dotted line shows basic cycle

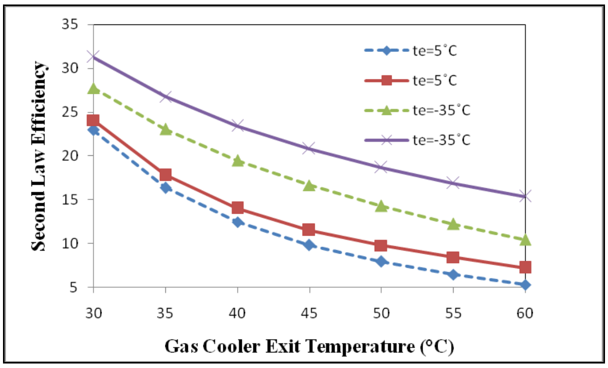

Fig. 10. Deviatioin of second law efficiency with gas cooler exit temperature

The Second law efficiency of the transcritical ethane cycle is also enhanced with an internal heat exchanger which can be observed from Fig.10. Deviation of Second law efficiency with gas cooler exit temperatures of transcritical ethane cycle with and without heat exchanger is compared at different evaporator values. The percentage increase in Second law efficiency values increases on increasing the gas cooler exit temperature values. The increment elevates from 4.8% at 30°C exit temperature to about 35.62% at an exit temperature value of 60°C with evaporator temperature being fixed at 5°C. Percentage increment goes from 12.62% at 30°C exit temperature to 47.02% at 60°C exit temperature with -35°C evaporator temperature value.

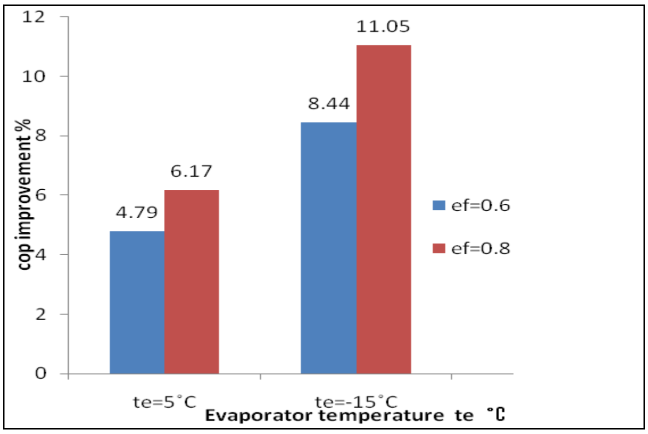

Fig 11. Percentage COP improvement at different evaporator temperature

Fig.11 shows the COP improvement of the transcritical vapor compression system after using an internal heat exchanger at two different effectiveness of the internal heat exchanger. At the evaporator temperature of 5˚C, the increase in COP of the system at effectiveness of 0.6 is 4.79% and at effectiveness of 0.8, it is 6.17%, whereas at the evaporator temperature of -15˚C the increase in COP at effectiveness of 0.6 is 8.44% and at effectiveness of 0.8, it is 11.05%. Hence COP increases on increasing the internal heat exchanger effectiveness.

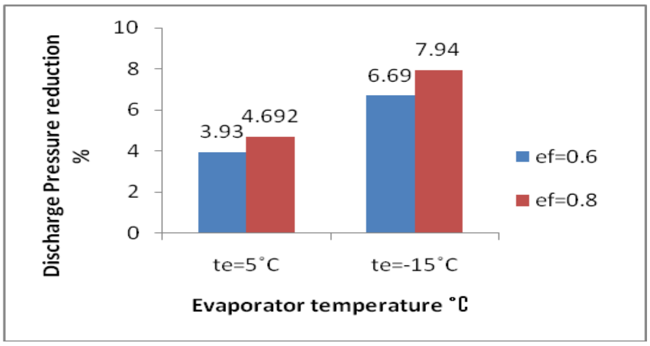

Fig 12 . Discharge pressure reduction percentage at different evaporator temperature

Fig. 12 shows the percentage reduction in the discharge pressure after using the internal heat exchanger at two different effectiveness of the heat exchanger. At the evaporator temperature of 5˚C and, when the effectiveness is 0.6 the discharge pressure reduction is 3.93% and effectiveness of 0.8 the reduction is 4.692 %. Similarly, at the evaporator temperature of -15˚C the discharge pressure reduction is 6.69% at the effectiveness of 0.6 and at 0.8, it is 7.94%. Hence, the use of an internal heat exchanger reduces the discharge pressure and percentage reduction increases on increasing the internal heat exchanger effectiveness and decreasing the evaporator temperature.

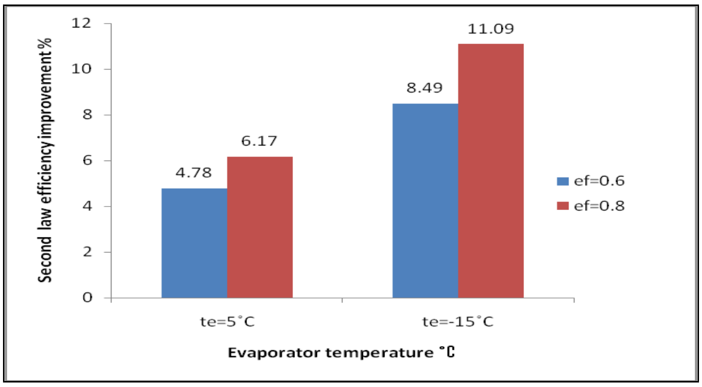

Fig.13 Percentage improvement in the second law efficiency at the different evaporator temperature

Fig. 13 shows the percentage improvement in the second law efficiency after employing the internal heat exchanger in the transcritical vapor compression system using ethane. There is increase in the second law efficiency of the system at evaporator temperature of 5˚C when the effectiveness is 0.6, the percentage increase is 4.78 % and at effectiveness of 0.8, it is 6.17%, whereas at the evaporator temperature of -15˚ C and effectiveness of 0.6 the increase is 8.49% and at effectiveness of 0.8 the percentage increase in the second law efficiency is 11.09%. Hence the second law efficiency increases at a moderate rate after using the internal heat exchanger in the transcritical vapor compression system.

Table 1 shows the comparison between the basic system and system with the internal heat exchanger based on energy and exergy analysis.

TABLE 1

S. no.	**Energetic and exergetic comparison at te=5°C and tgc=40 ˚C**		
		Without an internal heat exchanger	***With an internal heat exchanger***
1	Optimum discharge pressure (MPa)	6.479	6.224
2	Volumetric cooling capacity (kJ/m^3)	7587	8221
3	Compressor discharge temperature (°C)	67.97	86.66
4	Cooling COP	2.354	2.651
5	Compressor irreversibility (%)	22.34	21.24
6	Expansion device irreversibility %	34.39	20.11
7	Gas cooler irreversibility (%)	22.11	29.15
8	Evaporator irreversibility (%)	4.53	5.10
9	Internal Heat exchanger irreversibility (%)	------	5.69
10	Second law efficiency (%)	16.63	18.71

CONCLUSION

Based on the detailed thermodynamic analysis of both the cycles using ethane, the following conclusions can be drawn.

(1) Ethane is a good natural refrigerant, which can be employed in the basic transcritical vapor compression system and transcritical vapor compression system with an internal heat exchanger.

(2) For the basic transcritical cycle as well as for the transcritical cycle with an internal heat exchanger, evaporation temperature should be high and gas cooler exit temperatures should be low to obtain the maximum COP and minimum optimum discharge pressure.

(3) For both the cycles, the second law efficiency rises on reducing the evaporator and gas cooler exit temperatures.

(4) On increasing the internal heat exchanger effectiveness, the COP of the cycle increases, whereas its optimum discharge pressure decreases.

(5) The second law efficiency also increases on increasing the value of internal heat exchanger effectiveness.

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