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Transcritical CO_2 heat pump systems: exergy analysis including heat transfer and fluid flow effects

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Abstract

This paper presents the exergetic analysis and optimization of a transcritical carbon dioxide based heat pump cycle for simultaneous heating and cooling applications. A computer model has been developed first to simulate the system at steady state for different operating conditions and then to evaluate the system performance based on COP as well as exergetic efficiency, including component wise irreversibility. The chosen system includes the secondary fluids to supply the heating and cooling services, and the analyses also comprise heat transfer and fluid flow effects in detail. The optimal COP and the exergetic efficiency were found to be functions of compressor speed, ambient temperature and secondary fluid temperature at the inlets to the evaporator and gas cooler and the compressor discharge pressure. An optimization study for the best allocation of the fixed total heat exchanger inventory between the evaporator and the gas cooler based on heat transfer area has been conducted. The exergy flow diagram (Grassmann diagram) shows that all the components except the internal heat exchanger contribute significantly to the irreversibilities of the system. Unlike a conventional system, the expansion device contributes significantly to system irreversibility. Finally, suggestions for various improvement measures with resulting gains have been presented to attain superior system performance through reduced component irreversibilities. This study is expected to offer

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useful guidelines for system design and its optimisation and help toward energy conservation in heat pump systems based on transcritical CO_2 cycles.

Keywords: CO_2 heat pump; System simulation; Irreversibility; Exergetic optimization

1. Introduction

Discovery of the harmful effects of the synthetic refrigerants on the environment has created a renewed interest in eco-friendly natural refrigerants such as water, carbon dioxide, etc. Its ecologically benign nature, low price, easy availability, non-flammability, non-toxicity, compatibility with various common materials, equipment compactness due to high operating pressures and excellent transport properties are cited as some of the reasons behind the revival of carbon dioxide as a refrigerant. Carbon dioxide based heat pumps offer extensive possibilities in simultaneous heating and cooling applications due to the large temperature glide present in the gas cooler.

Exergy based second law analysis of various systems has become a very effective tool to measure system effectiveness and to design the system to maximize energy savings. Some researchers have conducted exergy analyses for conventional vapour compression refrigeration systems and its components. Bridges et al. [1] have reported second law analysis of a R410a based domestic refrigerator and a split air conditioner with fan coil optimization. Exergy analysis has been reported for an ammonia based vapour compression refrigeration system [2] and also for a conventional refrigerant based heat pump air conditioning system [3]. Apera et al. [4] conducted out exergy analyses of a compressor speed controlled vapour compression refrigeration system with refrigerant R22 and its substitutes R407c, R417a and R507 and chose R407C as the most suitable substitute. Exergy analysis has also been done for system components such as heat exchangers for optimized performance [5]. Reported exergetic analyses of transcritical CO_2 systems [6–9] have all been based only on the thermodynamic cycle using ideal assumptions. Such analyses for a complete CO_2 heat pump system including the heat exchangers and the secondary fluids and considering entropy generation due to both heat transfer and fluid flow effects have not been reported yet.

This paper presents a component level exergy analysis for the carbon dioxide based heat pump to provide heating and cooling services simultaneously. The irreversibilities of all components and the second law efficiency of the system for different values of the operating parameters such as water inlet temperature, ambient temperature, compressor speed and the area ratio of the heat exchangers have been estimated. System optimization based on these results has been presented as well. Finally, techniques to reduce the irreversibility for various components, which leads to improved system exergertic efficiency, have been suggested.

2. Mathematical modelling

A simplified sketch of a carbon dioxide based heating and cooling system showing the main components is shown in Fig. 1. Water is supplied as the heat exchanger fluid to both the gas cooler and evaporator through flow control valves. Both these heat exchangers are of the double pipe

Nomenclature

c_{pw}	water specific heat, $kJ\,kg^{-1}\,K^{-1}$
d	diameter of tube, m
E	exergy, W
f	friction factor
G_r	refrigerant mass velocity, $kg\,m^{-2}\,s^{-1}$
h	specific enthalpy, $kJ\,kg^{-1}$
I	irreversibility, W
L	length of tube, m
$\dot{m}_{ref}$	refrigerant mass flow rate, $kg\,s^{-1}$
$\dot{m}_{evw}$	water mass flow rate in evaporator, $kg\,s^{-1}$
$\dot{m}_{gcw}$	water mass flow rate in gas cooler, $kg\,s^{-1}$
Q	heat transfer rate, W
Re	Reynolds number
Re_l	Reynolds number for saturated liquid
s	specific entropy, $kJ\,kg^{-1}\,K^{-1}$
T	temperature, K
T_0	ambient temperature, K
T_{evw}	mean temperature of water in evaporator, K
T_{gcw}	mean temperature of water in gas cooler, K
T_{hexr}	mean temperature of refrigerant in internal heat exchanger, K
UA	overall heat transfer coefficient area of heat transfer product, $W\,K^{-1}$
x	refrigerant dryness fraction in evaporator
ΔP	pressure drop, bar

Greeks

ρ	density, $kg\,m^{-3}$
η_{ex}	second law efficiency

Subscripts

1–6	state point of refrigerant
ev	evaporator
evi	evaporator inlet
evo	evaporator outlet
evr	evaporator, refrigerant side
evw	evaporator, water side
gc	gas cooler
gci	gas cooler inlet
gco	gas cooler outlet
gcr	gas cooler, refrigerant side
gcw	gas cooler, water side

i	inner
o	outer
ref	refrigerant
wi	water inlet

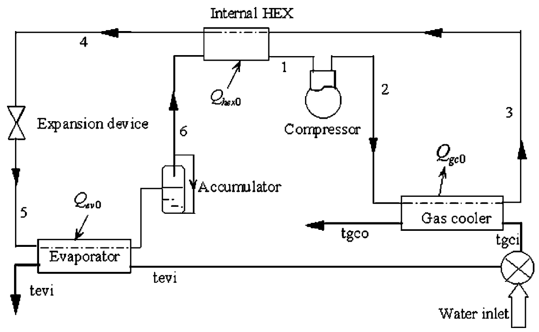

Fig. 1. Schematic layout of a transcritical carbon dioxide heat pump system.

counter flow type where the refrigerant flows through the inner tube and water flows through the outer annulus. The internal heat exchanger is also a double pipe counter flow type with the warmer refrigerant as the inner fluid, and the suction side refrigerant is the outer fluid in the annulus. Heat gain and heat loss with the environment are also indicated in the figure.

The entire system has been modelled based on energy balances of the individual components of the system, yielding conservation equations. To consider the lengthwise property variation, all three heat exchangers have been discretized, and momentum and energy conservation equations have been applied to each segment. The following simplifying assumptions have been made for the analysis:

1. Compression process is adiabatic but not isentropic.
2. Pressure drops in all the connecting pipes and heat transfer between the connecting pipes and the ambient are negligible.

Applying an exergy balance and energy conservation to each component of the system, the following modular relations can be developed to yield the system model.

2.1. Evaporator

The energy balance in the evaporator is given by

$$Q_{\mathrm{evw}} + Q_{\mathrm{evo}} = Q_{\mathrm{evr}} = \dot{m}_{\mathrm{ref}}(h_6 - h_5), \tag{1}$$

where the heat gain in the evaporator refrigerant Q_{evr} is given by

$$Q_{\mathrm{evr}} = \sum_{i=1}^{N} (\mathrm{UA})_{\mathrm{ev},i} (\mathrm{LMTD})_{\mathrm{ev},i}, \tag{2}$$

Q_{evo} is the heat gain from the ambient.

The cooling effect Q_{evw} is given by

$$Q_{\mathrm{evw}} = \dot{m}_{\mathrm{evw}} c_{\mathrm{pw}} (T_{\mathrm{evi}} - T_{\mathrm{evo}}). \tag{3}$$

The irreversibility or exergy loss in the evaporator is expressed as

$$I_{\mathrm{ev}} = T_0 \left(\dot{m}_{\mathrm{ref}} (s_6 - s_5) - \dot{m}_{\mathrm{evw}} c_{\mathrm{pw}} \ln \frac{T_{\mathrm{evi}}}{T_{\mathrm{evo}}} \right) + I_{\mathrm{evr}}^{\Delta P} + I_{\mathrm{evw}}^{\Delta P} + Q_{\mathrm{evo}} \left(\frac{T_0}{T_{\mathrm{evw}}} - 1 \right), \tag{4}$$

where the temperature related terms on the right hand side are due to the temperature difference and heat interaction with the ambient, respectively. The pressure related terms are due to the pressure drops in the refrigerant and water sides, respectively, and are given by

$$I_{\mathrm{evr}}^{\Delta P} + I_{\mathrm{evw}}^{\Delta P} = \frac{\dot{m}_{\mathrm{ref}}}{\rho_{\mathrm{evr}}} \Delta P_{\mathrm{evr}} + \frac{\dot{m}_{\mathrm{evw}}}{\rho_{\mathrm{evw}}} \Delta P_{\mathrm{evw}}, \tag{5}$$

ΔP_{evw} is the pressure drop on the water side. The refrigerant side pressure drop ΔP_{evr} (summation of pressure drops in all segments) is given by (using Lockhart and Martinelli equation)

$$\Delta P_{\mathrm{evr}} = \sum_{i=1}^{N} \left(4 \frac{L_{\mathrm{ev}}}{d_{\mathrm{evi}}} \frac{f_{\mathrm{evr}}}{2} (1-x)^2 \frac{G_{\mathrm{r}}^2}{\rho_1} \phi_1^2 \right)_i, \tag{6}$$

where the friction factor is expressed as $f_{\mathrm{evr}} = 0.0791 Re_1^{-0.25}$.

The two phase frictional pressure drop multiplier is evaluated from

$$\phi_1 = \left(1.376 + \frac{7.242}{X_{\mathrm{tt}}^{1.655}} \right)^{1/2}, \tag{7}$$

where X_{tt} is the Lockhart–Martinelli factor.

2.2. Compressor

The exergy input to the compressor is given by

$$E_{\mathrm{in}} = \dot{m}_{\mathrm{ref}} (h_2 - h_1). \tag{8}$$

The overall thermal efficiency for the semi-hermetic compressor has been calculated as [10]

$$\eta_{\mathrm{is,c}} = -0.26 + 0.7952 r_{\mathrm{p}} - 0.2803 r_{\mathrm{p}}^2 + 0.0414 r_{\mathrm{p}}^3 - 0.0022 r_{\mathrm{p}}^4, \tag{9}$$

with the compressor pressure ratio r_{p} ($=P_{\mathrm{dis}}/P_{\mathrm{suc}}$) varying between 1.5 and 6.5.

The irreversibility in the compressor is estimated from

$$I_{\mathrm{comp}} = T_0 \dot{m}_{\mathrm{ref}} (s_2 - s_1). \tag{10}$$

2.3. Gas cooler

The energy balance in the gas cooler yields

$$Q_{\mathrm{gcw}} + Q_{\mathrm{gco}} = Q_{\mathrm{gcr}} = \dot{m}_{\mathrm{ref}}(h_2 - h_3), \tag{11}$$

where Q_{gcr} is the heat transferred from the refrigerant in the gas cooler, given by

$$Q_{\mathrm{gcr}} = \sum_{i=1}^{N} (\mathrm{UA})_{\mathrm{gci}} (\mathrm{LMTD})_{\mathrm{gci}}, \tag{12}$$

Q_{gco} is the heat loss to the ambient.

The heating effect Q_{gcw} is given by

$$Q_{\mathrm{gcw}} = \dot{m}_{\mathrm{gcw}} c_{\mathrm{pw}} (T_{\mathrm{gco}} - T_{\mathrm{gci}}). \tag{13}$$

The resulting irreversibility is expressed as

$$I_{\mathrm{gc}} = T_0 \left(\dot{m}_{\mathrm{gcw}} c_{\mathrm{pw}} \ln \frac{T_{\mathrm{gco}}}{T_{\mathrm{gci}}} - \dot{m}_{\mathrm{ref}}(s_2 - s_3) \right) + I_{\mathrm{gcr}}^{\Delta P} + I_{\mathrm{gcw}}^{\Delta P} + Q_{\mathrm{gco}} \left(1 - \frac{T_0}{T_{\mathrm{gcw}}} \right). \tag{14}$$

The irreversibility due to pressure drop is given by

$$I_{\mathrm{gcr}}^{\Delta P} + I_{\mathrm{gcw}}^{\Delta P} = \frac{\dot{m}_{\mathrm{r}}}{\rho_{\mathrm{gcr}}} \Delta P_{\mathrm{gcr}} + \frac{\dot{m}_{\mathrm{gcw}}}{\rho_{\mathrm{gcw}}} \Delta P_{\mathrm{gcw}}, \tag{15}$$

where ΔP_{gcw} is the water side pressure drop in the gas cooler. The refrigerant side pressure drop ΔP_{gcr} is estimated by the correlation [11]

$$\Delta P_{\mathrm{gcr}} = \sum_{i=1}^{N} \left\{ \frac{G_{\mathrm{r}}^2}{\rho_{\mathrm{r}}} \left(f_{\mathrm{gcr}} \frac{L_{\mathrm{gc}}}{d_{\mathrm{gci}}} + 1.2 \right) \right\}_i \tag{16}$$

where the friction factor for the supercritical refrigerant flow is given by

$$f_{\mathrm{gcr}} = [0.79 \ln(Re) - 1.64]^{-2}. \tag{17}$$

2.4. Expansion device

The irreversibility during the expansion process is expressed as

$$I_{\mathrm{exp}} = T_0 \dot{m}_{\mathrm{ref}} (s_5 - s_4). \tag{18}$$

2.5. Internal heat exchanger

The heat balance in the internal heat exchanger yields

$$\dot{m}_{\mathrm{r}}(h_1 - h_6) = \dot{m}_{\mathrm{r}}(h_3 - h_4) + Q_{\mathrm{hex0}}, \tag{19}$$

where Q_{hex0} is the heat gain (+ sign) or loss (− sign) with the ambient.

The irreversibility in the internal heat exchanger is given by

$$I_{\text{hex}} = T_0\dot{m}_{\text{ref}}[(s_1 - s_6) - (s_3 - s_4)] + I_{\text{hex}}^{\Delta P} + Q_{\text{hex0}}\left(\frac{T_0}{T_{\text{hexr}}} - 1\right). \tag{20}$$

It may be noted that the last term on the right hand side of this equation is always positive whether $T_{\text{hexr}}>$ or $<T_0$, which indicates that the exergy is always degrading whether there is a heat loss or gain with the component. The heat transfers with the ambient for all the components have been estimated employing conventional natural convection equations assuming that no other heat transfer mode is existent. The UA values for all the heat exchanger segments were evaluated using conventional overall heat transfer coefficient relations. The heat transfer coefficient for water in both heat exchangers has been evaluated by the conventional Dittus–Boelter equation for annular flow. The refrigerant heat transfer coefficient for the gas cooler was calculated using the Pitla correlation [12] and that for the evaporator was estimated using the Wattelet–Carlo correlation [13].

The system performance measures are based on the system COP, which is the ratio of the total heating and cooling output to the work input, and on the exergetic efficiency, which is the percentage ratio of total exergy output to the exergy input. The output exergy can be found by subtracting the total system irreversibility (summation of irreversibilities of all the components in the system) from the exergy input to the system and is given by

$$\eta_{\text{ex}} = \frac{E_{\text{in}} - \sum I_{\text{Components}}}{E_{\text{in}}} \times 100\% = \left(1 - \frac{\sum I_{\text{Components}}}{E_{\text{in}}}\right) \times 100\%. \tag{21}$$

3. Solution procedure

A computer code has been developed for the exergetic analysis of the transcritical carbon dioxide based heat pump system envisaged for dairy application for various operating conditions. Using the new equation of state for CO_2 [14] and transport property correlations available in the literature [15], an exclusive code 'CO2PROP' employing the technique based on the derivatives of the Helmholtz free energy function using efficient iterative procedures has been developed to estimate the thermodynamic and transport properties of carbon dioxide where the transcritical zone has been included as well, the details of which have already been reported [9]. The property variation is very abrupt near the critical region, and the gas cooler encompasses this regime. To consider this variation, the entire length of the gas cooler has been divided into several equal discrete segments, and each segment has been treated as a counter flow heat exchanger. In each segment, the heat transfer coefficients for both refrigerant and water are calculated based on mean values. This way, the gas cooler is made equivalent to a number of counter flow heat exchangers arranged in series, and the combined heat transfer of all the segments is the total heat transfer of the gas cooler. Therefore, the fast changing properties of CO_2 have been modelled accurately in the gas cooler to yield good accuracy. For the evaporator and internal heat exchanger, similar discretization has been performed as well to obtain good accuracy. At first, the system was simulated based on the energy balances for each component by a Newton–Raphson iterative method, employing heat transfer correlations and the subroutine code CO2PROP to evaluate the state points with all the necessary thermodynamic and transport properties, to get maximum accuracy.

Then, the irreversibility of each of the components, exergetic output and efficiency have been evaluated using the exergy balance equations presented in the previous section.

4. Results and discussion

The System COP, component irreversibilities and system exergetic efficiency, based on combined heating and cooling, have been estimated for various inlet conditions for the heat exchanger fluid (water), compressor speed and heat exchanger dimensions of the heat pump envisaged to provide heating at 73 °C and cooling at 4 °C simultaneously as is typically required in a dairy application. Stainless steel tubes with outer diameters of 3/8 in. (10 mm) and 5/8 in. (16 mm) have been chosen as the inner and outer tubes, respectively, for both the evaporator and gas cooler and also for the internal heat exchanger. All these are concentric tube-in-tube type heat exchangers. The thickness of all the tubes has been taken as 0.815 mm. A Dorin compressor (TCS113) with a rated speed of 2900 rpm and a swept volume of 11.7 cm^3 has been chosen [16]. Results are presented for a total heat exchanger tube length of 25 m for both the evaporator and gas cooler. The internal heat exchanger length has been taken as 4 m, which yields an effectiveness of about 60%. Ambient temperature is taken to be 30 °C (average in the Indian sub-continent) for the analysis.

Figs. 2–7 show the effects of the different operating conditions for the same total heat exchanger length (25 m). The parameters varied are: compressor speed from 2000 to 3500 rpm, water inlet temperature from 20 to 40 °C and area ratio from 1.0 to 2.5. Unless otherwise specified, the mean values are compressor speed of 2900 rpm, water inlet temperature of 30 °C and area ratio of 1.8. In the plots presented, one of the parameters is varied within the specified range stated above, while the other parameters are kept constant at the mean values. For the entire study, the compressor discharge pressure is set at the optimum level for maximum COP, and the optimisation analyses have already been reported by the authors [9].

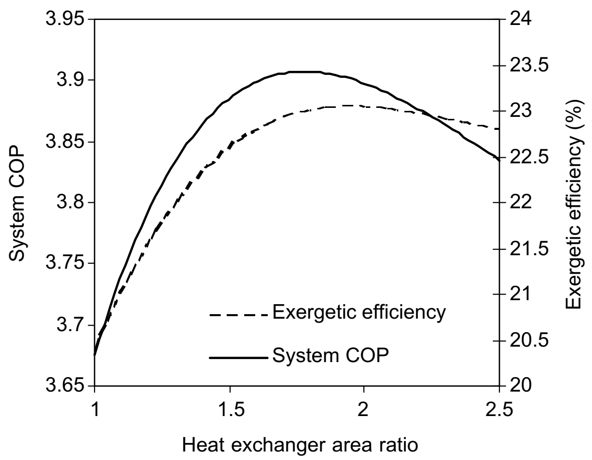

Fig. 2. System performance with varying heat exchanger area ratio.

Both system COP and exergetic efficiency increase initially with area ratio (ratio of gas cooler effective surface area to that of evaporator, here simply the length ratio as both have the same diameter) and, beyond a certain value, decrease as is evident from Fig. 2. Although the system COP attains a maximum at an area ratio of about 1.8, the exergetic efficiency reaches its maximum at an area ratio of about 1.85. The irreversibility of the evaporator increases and that of the gas cooler decreases with increase in area ratio (Fig. 3). This is attributed to the fact that the effective temperature difference in the evaporator rises while that in the gas cooler drops with an increase in area ratio. The irreversibility of the internal heat exchanger decreases for larger area ratio values since the pressure for both the fluids drops. However, as shown, the influence of the internal heat exchanger on system performance is marginal. Because of a decrease in the entropy

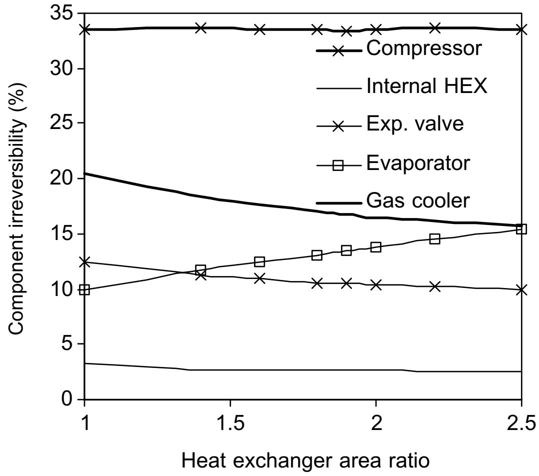

Fig. 3. Variation of component irreversibility with heat exchanger area ratio.

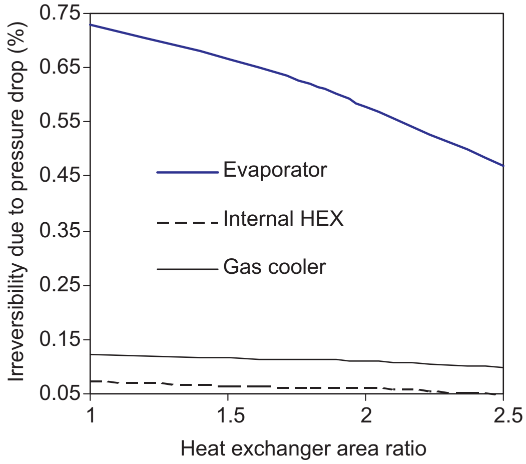

Fig. 4. Influence of heat exchanger area ratio on irreversibility due to pressure drop.

generation rate at the lower pressure in the expansion valve, the irreversibility decreases as the area ratio increases. Although the irreversibility due to pressure loss is negligible compared to the total exergy loss (Fig. 4), that for the evaporator is significantly more than that yielded by the internal heat exchanger and gas cooler. This could be attributed to the pressure drop in the evaporator being more than that in the gas cooler and internal heat exchanger due to both frictional and momentum effects.

The effect of water inlet temperature at a compressor speed of 2900 rpm, ambient temperature of 30 °C and area ratio of 1.8 is shown in Figs. 5 and 6. As the water inlet temperature increases, the system COP decreases due to an increase in compressor work and also due to the decrease in cooling output. The water inlet temperature has a negligible effect on the refrigerant mass flow rate. However, the optimum discharge pressure increases rapidly with increasing water inlet tem-

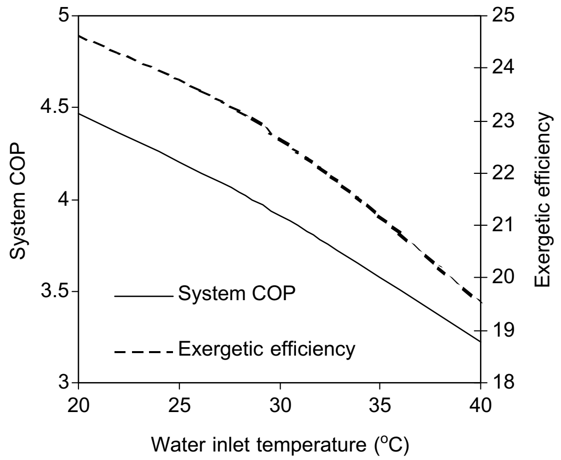

Fig. 5. System performance with varying water inlet temperature.

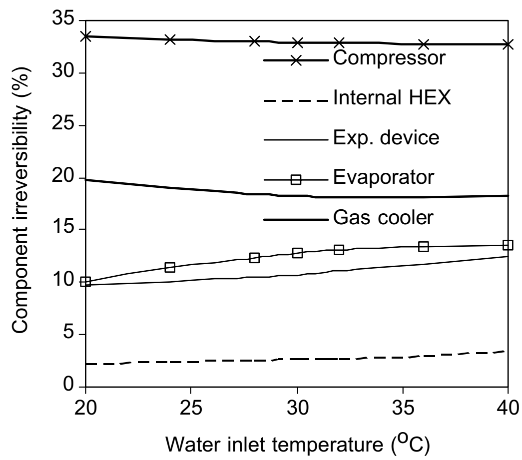

Fig. 6. Variation of component irreversibility with water inlet temperature.

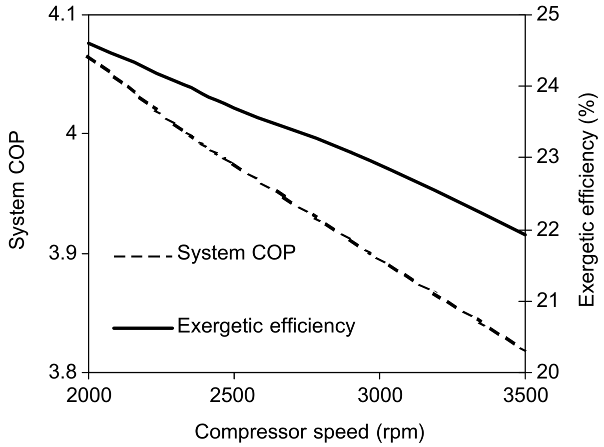

Fig. 7. Effect of compressor speed on system performance.

perature due to the rapid change of refrigerant outlet temperature in the gas cooler. Hence, the exergetic efficiency of the system deteriorates with a rise in water inlet temperature as the exergy losses in the evaporator and in the internal heat exchanger increase rapidly due to the increase in heat exchanger temperature differences. Although the irreversibility in the gas cooler remains fairly constant with changes in water inlet temperature, the exergy loss in the expansion valve increases. The effect of the compressor speed on the system performance at a heat exchanger area ratio of 1.8 and a water inlet temperature of 30 °C is presented in Fig. 7. It is observed that the system COP at optimum discharge pressure decreases as both compressor work and cooling output increase with compressor speed. The maximum increment of irreversibility occurred in the gas cooler. This may be attributed to an increase in the temperature difference in the gas cooler as well as the increase in pressure loss due to the rapid increase in flow velocity. So, an increase in compressor speed yields higher capacity due to the higher mass flow rate and higher frictional pressure loss due to the higher velocity that causes the increase in irreversibility as well.

The exergy loss in the different components (Grassmann diagram) is shown in Fig. 8. Similar to the behaviour reported for systems based on conventional refrigerants such as R22, R12, R502

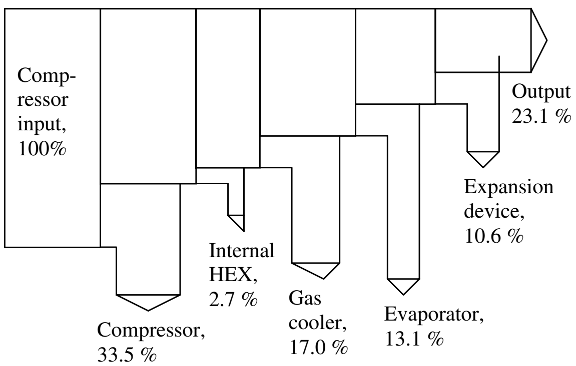

Fig. 8. Exergy flow (Grassmann) diagram at mean operating condition.

[4,17], the exergy loss is maximum in the compressor followed by that in the gas cooler, evaporator and expansion device, while the exergy loss in the internal heat exchanger is negligible. Because of the high pressure drop occurring in the system being studied, the expansion device contributes a much larger fraction of the irreversibility compared to conventional systems.

5. Measures to improve exergetic efficiency

The exergy loss is relatively high in the compressor, gas cooler, evaporator and expansion valve, while that in the internal heat exchanger is insignificant. Hence, the contribution of the internal heat exchanger towards exergy destruction and its influence on the system performance is not predominant, although by increasing the effective heat transfer area, we can modestly increase the effectiveness as well as the system COP and exergetic efficiency of the system. The primary challenge is to improve the system performance and exergetic efficiency by improving the performance (by controlling exergy loss) of the above stated four influential components. Some of the important improvement measures are presented with their associated betterment in system performance.

5.1. Compressor

Process irreversibility (due to mixing, throttling, internal convection, etc.), pressure loss due to friction in inlet and outlet valves and heat loss to the environment are the basic reasons for the exergy loss in the compressor, although the last two have been neglected since the heat loss is not so severe from having a modest temperature difference over the ambient. The system COP and the exergetic efficiency increase linearly with the isentropic efficiency as shown in Fig. 9. Isentropic efficiency primarily depends on the compressor design and the working pressure, so a superior compressor design will lead to higher isentropic efficiency, resulting in a reduction in the process irreversibility, within certain limits. With a 10% increase in isentropic efficiency, the exergetic efficiency improves by almost 3%.

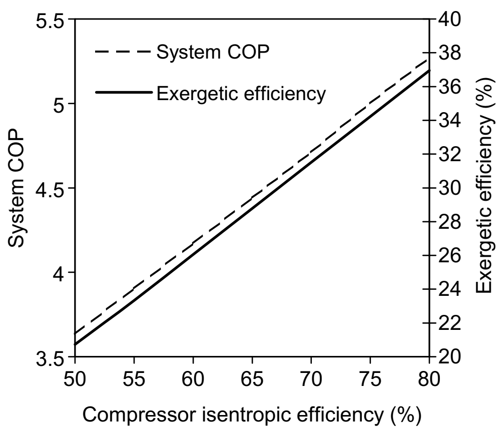

Fig. 9. System performance with varying compressor isentropic efficiency.

5.2. Evaporator and gas cooler

Irreversibilities in the evaporator and the gas cooler occur due to the temperature differences existing between the two heat exchanger fluids, pressure loss, flow imbalance and heat transfer with the ambient. The results show that almost 90% of the irreversibility occurs due to the fluid temperature difference and 10% due to the rest. At the mean conditions, the average fluid temperature differences in the evaporator and gas cooler are about 23 and 42 °C, respectively, whereas the pressure losses are 2.5 and 0.9 bar, respectively. The effective fluid temperature difference can be reduced by increasing the heat transfer area, either by increasing the heat exchanger length or incorporating fins, both of which will result in higher pressure drop. The system COP and exergetic efficiency of the system increase, first rapidly and then slowly, because of the increase in irreversibility due to pressure loss (Fig. 10). Heat transfer with the ambient is inconsequential for the system performance. If better insulation reduces the net outer wall conductivity from 20 to 1 $\mathrm{W\,m^{-1}\,K^{-1}}$, the exergetic efficiency will increase by a mere 0.2%. Irreversibility due to flow imbalance depends on the heat exchanger design. In this analysis, the heat exchanger was designed for nearly equal heat capacities of the two fluids, and hence, the flow imbalance is negligible for this system.

5.3. Expansion device

Replacement of the expansion valve by a turbine is the only option available to improve the performance of the system and reduce the irreversibility of the expansion process. For such a technique, it was reported [7] that an expansion work recovery turbine with isentropic efficiency of 60% would reduce the contribution of this process to total cycle irreversibility by 35% in the thermodynamic cycle. In this system, at the mean condition stated above, by using an expansion work recovery turbine of 85% isentropic efficiency, both the system COP and the exergetic efficiency will improve by about 22%. Hence, improvement in system performance through this technique is quite significant. However, such extensive hardware addition may not be economically feasible in many practical applications, especially for small capacities.

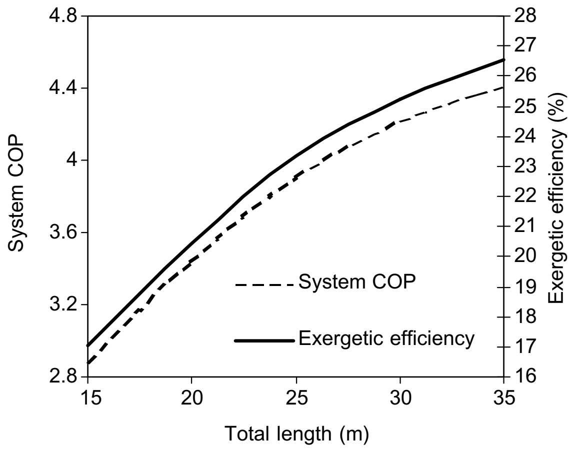

Fig. 10. System performance with varying total heat exchanger length.

6. Conclusions

Exergetic analyses of a transcritical carbon dioxide based heat pump to provide heating and cooling simultaneously have been presented here. Unlike previous studies reported in the literature, realistic heat transfer and fluid flow effects have been included. A computer model has been developed first to simulate the system at steady state for different operating conditions and then to evaluate the system performance based on COP as well as exergetic efficiency. Component level irreversibility analyses have been performed. The highly variable thermophysical and transport properties of the refrigerant near the critical point have been taken into account to obtain better accuracy. Spatial discretization of all the heat exchangers has been used as well to yield better precision. Results are obtained by varying the important operating and design parameters such as heat exchanger area ratio, compressor speed, water inlet temperature and ambient temperature over a given range. The results show that:

1. The optimum heat exchanger area ratio ranges between 1.8 and 1.9 for maximum system COP as well as maximum exergetic efficiency at optimum discharge pressure.
2. The favourable heat transfer properties of carbon dioxide in both the two phase and supercritical region and an efficient compression process contribute significantly toward high system COPs and exergetic efficiency values. The temperature differences in the heat exchangers contribute more that 90% of their irreversibilities, whereas the rest occurs due to pressure drop and the system imbalance in the heat exchanger.
3. It is more effective to maintain the secondary fluid inlet temperature as low as possible to get higher COP and exergetic efficiency within the given range.
4. The compressor, evaporator, gas cooler and expansion device contribute to system irreversibility to a larger extent, while the internal heat exchanger has a negligible effect. The expansion valve contributes a significant amount of exergy loss here, whereas it is negligible for a conventional system.
5. It is effective, in terms of improvement in COP and exergetic efficiency, to employ a large heat exchanger area by increasing the length or by using fins, which will also attract additional investment and higher pressure drop. Hence, there is an optimal trade-off between the two.
6. Replacement of the expansion valve with a turbine will increase the COP as well as the exergetic efficiency significantly, but it will also raise issues related to cost, design and dynamic balancing of the system. It is advisable to employ a turbine for large systems, such as a large dairy plant or other large system where simultaneous cooling and heating is useful.

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