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Irreversibility minimization of heat exchangers for transcritical CO_2 systems

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Abstract

Irreversibility analyses of both evaporator and gas cooler of a CO_2 based transcritical heat pump for combined cooling and heating, employing water as the secondary fluid, have been reported. The analysis includes both operational and material associated irreversibilities. Optimization of heat exchanger tube diameter and length and effect of design parameters on overall system performance is also presented. Results clearly show that higher heat transfer coefficient can be achieved by reducing the diameter only to a limited extent due to rapid increase in pressure drop. The minimum possible diameter depends on mass flow rate (capacity) and division of flow path. The right combination of optimum diameter and length depends on the number of passes, capacity and operating parameters. It is noteworthy that due to higher pressure drop occurring in the evaporator compared to the gas cooler, zero temperature approach is attained before the optimum length is reached in case of the evaporator. Presented results are expected to help choose effective heat exchanger size in terms of diameter, length and number of passes.

Keywords: Transcritical CO_2 system; Heat exchanger; Life cycle irreversibility; Irreversibility minimization; Optimization

1. Introduction

Within the last decade, carbon dioxide has drawn ample interest as a natural refrigerant in refrigeration, air-conditioning and heat pump applications due to its various advantages including zero ODP and negligible GWP. Although lately manufacturers have laid more emphasis on mobile air-conditioning and water heating, it is possible to extend its use to various industrial applications especially in food industries for both heating and cooling. Recent studies indicate that CO_2 based heat pump systems perform better in simultaneous cooling and heating applications [1]. In the present study, design optimization of heat exchangers for a water heating and cooling system having certain capacity for similar applications (e.g., dairy units) is presented. For effective sizing of heat exchangers, detailed knowledge of irreversibility is essential. For example, for the same capacity (constant mass flow rate), heat exchanger area can be reduced by reducing diameter, so that irreversibility due to material reduces; however, the irreversibility due to pressure drop will increase rapidly. To accommodate this, a multi-pass arrangement can be introduced using very small tubes (for example, in a microchannel heat exchanger); but this will give rise to higher manufacturing irreversibility. So it is not an easy task to choose an effective set of diameter, length and number of passes for heat exchangers and hence the necessity of an optimization exercise arises.

It is well-known that irreversibility occurs in fluid flow systems through three mechanisms of entropy generation: *molecular thermal dissipation*, *viscous dissipation* and *chemical dissipation*. Several approaches such as entropy generation minimization, life cycle analysis, and exergoeconomic or thermoeconomic analysis have been employed for optimization of heat exchangers. Nummedal et al. [2] introduced equipartition forces ($\Delta\{1/T\}$) for entropy generation minimization. Johannessen et al. [3] stated that minimum entropy generation of a heat exchanger occurs corresponding to equipartition of entropy production rather than equipartition of thermal driving force. Cor-

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Nomenclature

C	irreversibility per unit mass or length	$\mathrm{MJ\,kg^{-1}}$, $\mathrm{MJ\,m^{-1}}$
c_p	specific heat	$\mathrm{kJ\,kg^{-1}\,K^{-1}}$
d	inner tube diameter	mm, m
D	outer tube diameter	mm, m
f	friction coefficient	
g	acceleration due to gravity	$\mathrm{m\,s^{-2}}$
G	mass velocity	$\mathrm{kg\,m^{-2}\,s^{-1}}$
h	heat transfer coefficient	$\mathrm{W\,m^{-2}\,K^{-1}}$
h_{fg}	latent heat of vaporization	$\mathrm{J\,kg^{-1}}$
I	irreversibility	W
L	heat exchanger length	m
M	mass of material	kg
$\dot{m}$	mass flow rate	$\mathrm{kg\,s^{-1}}$
n	number of passes	
q_w	heat flux	$\mathrm{W\,m^{-2}}$
Nu	Nusselt number	
Pr	Prandtl number	
Δp	pressure loss	bar
Re	Reynolds number	
r_p	compressor pressure ratio	
s	specific entropy	$\mathrm{kJ\,kg^{-1}\,K^{-1}}$
T	temperature	K
T_0	reference temperature	K
x	quality of refrigerant	
X_{tt}	Lockhart–Martinelli parameter	

Greek

μ	viscosity	$\mathrm{kg\,m^{-1}s^{-1}}$
η	efficiency	
ρ	density	$\mathrm{kg\,m^{-3}}$

Subscripts

II	second law
b	bulk
c	compressor
cr	critical
evi	evaporator inlet
evo	evaporator outlet
evr	evaporator, refrigerant side
evw	evaporator, water side
fab	fabrication
gci	gas cooler inlet
gco	gas cooler outlet
gcr	gas cooler, refrigerant side
gcw	gas cooler, water side
i	inner
ins	insulation
is	isentropic
man	manufacturing
o	outer
oper	operational
p	tube material
ref	refrigerant
tot	total
TP	two-phase
v	volumetric
w	wall

nelissen et al. [4,5] performed combined exergy analysis and life cycle analysis of a balanced counterflow water-to-water heat exchanger; life cycle analysis yields optimum dimensions, where life cycle irreversibility has the minimum value. A few irreversibility studies on balanced heat exchangers for constant fluid properties have also been reported [6]. Although, some studies on exergoeconomic optimization of refrigeration or heat pump systems and its components employing synthetic refrigerants have become available recently [7–9], exergoeconomic or irreversibility analysis of a CO_2 based transcritical system components has not been reported yet. High pressure, distinct dry out phenomenon and near critical operation in evaporator, and supercritical operation and abrupt variation of thermophysical and transport properties of CO_2 in gas cooler make this analysis unique and very interesting as well compared to sub-critical heat pump systems.

In the present study, detailed irreversibility analyses of both the evaporator and the gas cooler have been carried out for a CO_2 based transcritical heat pump system with water as a secondary fluid for both the heat exchangers. This analysis includes both operational and material irreversibilities. Optimization of heat exchanger tube diameter and length has been presented as well. Effect of the three basic design parameters namely, tube diameter, length and number of passes have been studied in detail.

2. Total irreversibility analysis

The methodology of total irreversibility analysis includes the effects of all phases of production, use and recycling on the environment. So the total irreversibility of a heat exchanger includes:

1. Operational (thermal dissipation, viscous dissipation, and cumulative losses in power and heat generation) irreversibility.
2. Irreversibility associated with use of material.

Irreversibility associated with the operation of a heat exchanger can be written as:

$$I_{\mathrm{oper}} = I^{\Delta T} + I^{\Delta p} \tag{1}$$

where, $I^{\Delta T}$ and $I^{\Delta p}$ indicate the irreversibilities due to thermal and viscous dissipation, respectively. The irreversibility associated with use of material considers total life cycle of material and its effect on environment. However, neglecting recycling and the environmental effect, the irreversibility associated with use of material can be simply written as:

$$I_{\text{man}} = \frac{1}{t}\sum_{n}\big[(M_{i,p}+M_{o,p})C_p + (M_{i,\text{man}}+M_{o,\text{man}})C_{\text{man}} + LC_{\text{fab}} + M_{\text{ins}}C_{\text{ins}}\big] \tag{2}$$

where, the first term on the right hand side denotes the irreversibility due to raw material for inner and outer tubes, the second term denotes the same for the inner and outer tube manufacturing processes, the third and fourth terms represent that for heat exchanger fabrication and insulation, respectively, and t indicates the total life cycle time. Hence the total irreversibility is given by,

$$I_{\text{tot}} = I_{\text{oper}} + I_{\text{man}} \tag{3}$$

For certain applications and working fluids, if we try to reduce $I^{\Delta T}$, I_{man} and $I^{\Delta p}$ will increase. Likewise, with the same diameters, we can reduce $I^{\Delta T}$ by increasing the length, which will also cause I_{man} and $I^{\Delta p}$ to increase. This opposing trend will lead to the existence of an optimum length where the total irreversibility will be a minimum. Analytical solution to obtain optimum length based on minimum total irreversibility is impossible to implement in case of a real heat exchanger due to variation in fluid properties. Hence rigorous numerical analysis with reliable property, heat transfer and pressure drop correlations is essential.

The following performance evaluation parameters based on second law evaluation criteria have been used to show the relative influence of thermal and viscous dissipation on heat exchanger irreversibility [10]:

Irreversibility distribution ratio

$$\phi = \frac{I^{\Delta T}}{I^{\Delta P}} \tag{4}$$

Rational (second law) effectiveness

$$= \frac{\text{Exergy gained by the cold stream}}{\text{Exergy donated by the warm stream}} \tag{5}$$

3. Mathematical modeling and numerical simulation

To obtain the total irreversibility, first the steady state simulation of the transcritical CO_2 heat pump cycle with internal heat exchanger (Fig. 1) has been implemented. Both the heat exchangers are considered to be multi-pass double-pipe counterflow type, where the refrigerant flows through the inner tube and water flows through the outer annular space. The following simplifying assumptions have been made in the analysis:

1. Mass flow rates in all the passes are equal.
2. Single-phase heat transfer has been considered for water.
3. Pressure drop in all the connecting pipes and heat transfer between the connecting pipes and ambient, and between compressor and the ambient have been neglected.
4. The thermal insulation of the heat exchanger has been assumed to be perfect.

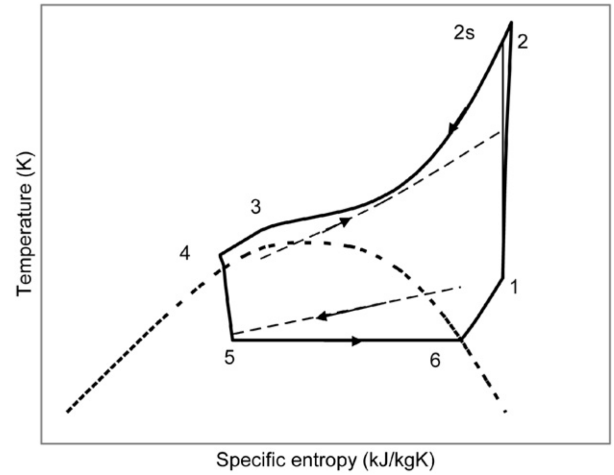

Fig. 1. Transcritical CO_2 heat pump cycle on T–s plane.

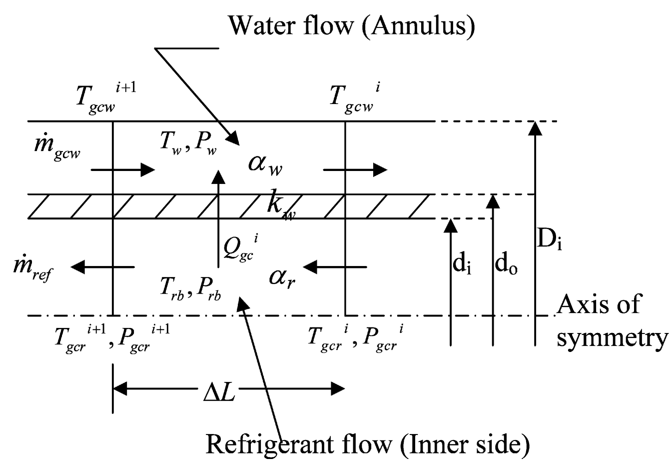

Fig. 2. A computational segment of the gas cooler.

To capture the longitudinal property variation for CO_2 and water, both the heat exchangers have been discretized appropriately. Subsequently momentum, energy and exergy conservation equations have been applied to each segment.

One of the computational segments of gas cooler (a symmetric half) of length ΔL is shown in Fig. 2. Employing LMTD (Logarithmic Mean Temperature Difference) expression and energy balance, the heat transfer in the ith segment of the gas cooler is given by,

$$(UA)_{\text{gc}}^{i}\frac{(T_{\text{gcr}}^{i}-T_{\text{gcw}}^{i})-(T_{\text{gcr}}^{i+1}-T_{\text{gcw}}^{i+1})}{\ln\Big(\frac{T_{\text{gcr}}^{i}-T_{\text{gcw}}^{i}}{T_{\text{gcr}}^{i+1}-T_{\text{gcw}}^{i+1}}\Big)} = \dot{m}_{\text{ref}}\big(h_{\text{gcr}}^{i}-h_{\text{gcr}}^{i+1}\big) = \dot{m}_{\text{gcw}}c_{\text{pw}}\big(T_{\text{gcw}}^{i}-T_{\text{gcw}}^{i+1}\big) \tag{6}$$

Similarly, for the evaporator, the energy balance equation for each computational segment can be written as:

$$(UA)_{\text{ev}}^{i}\frac{(T_{\text{evw}}^{i+1}-T_{\text{evr}}^{i+1})-(T_{\text{evw}}^{i}-T_{\text{evr}}^{i})}{\ln\Big(\frac{T_{\text{evw}}^{i+1}-T_{\text{evr}}^{i+1}}{T_{\text{evw}}^{i}-T_{\text{evr}}^{i}}\Big)} = \dot{m}_{\text{ref}}\big(h_{\text{evr}}^{i+1}-h_{\text{evr}}^{i}\big) = \dot{m}_{\text{evw}}c_{\text{pw}}\big(T_{\text{evw}}^{i+1}-T_{\text{evw}}^{i}\big) \tag{7}$$

The overall heat transfer coefficients for the segment for both gas cooler and evaporator have been estimated from the fundamentals. For the simulation, the internal heat exchanger effectiveness is assumed to be 0.6. For thermophysical and transport properties of CO_2, an exclusive code, CO2PROP, based on recent correlations [11–13] was developed and employed [14]. The simulation code solves the system equations by Newton–Raphson iterative method integrated with the subroutine code CO2PROP and the heat transfer calculations. The total irreversibility of evaporator and gas cooler has been evaluated by summing up the irreversibility in each segment. Specifications of a CO_2 compressor (Dorin Model TCS113) have been used for simulation and the following correlations have been used for volumetric and isentropic efficiency respectively, based on regression of experimental data [15]:

$$\eta_v = 1.1636 - 0.2188 r_p + 0.0163 r_p^2$$
$$\eta_{\mathrm{is},c} = 0.61 + 0.0356 r_p - 0.0257 r_p^2 + 0.0022 r_p^3 \tag{8}$$

The irreversibility due to thermal dissipation in the evaporator is expressed as:

$$I_{\mathrm{evr}}^{\Delta T} = \sum_{\mathrm{pass}} \sum_{\mathrm{segment}} T_0 \left(\dot{m}_{\mathrm{ref}}(s_6 - s_5) - \dot{m}_{\mathrm{evw}} c_{\mathrm{pw}} \ln \frac{T_{\mathrm{evi}}}{T_{\mathrm{evo}}} \right) \tag{9}$$

and the irreversibility due to pressure drop is given by:

$$I_{\mathrm{evr}}^{\Delta P} = \sum_{\mathrm{pass}} \left\{ \dot{m}_{\mathrm{ref}} \sum_{\mathrm{segment}} \left(\frac{T_o \Delta P_{\mathrm{evr}}}{T_{\mathrm{evr}} \rho_{\mathrm{evr}}} \right) + \dot{m}_{\mathrm{evw}} \sum_{\mathrm{segment}} \left(\frac{T_o \Delta P_{\mathrm{evw}}}{T_{\mathrm{evw}} \rho_{\mathrm{evw}}} \right) \right\} \tag{10}$$

Similarly, the irreversibility due to thermal dissipation in the gas cooler is expressed as:

$$I_{\mathrm{gc}}^{\Delta T} = \sum_{\mathrm{pass}} \sum_{\mathrm{segment}} T_0 \left(\dot{m}_{\mathrm{gcw}} c_{\mathrm{pw}} \ln \frac{T_{\mathrm{gco}}}{T_{\mathrm{gci}}} - \dot{m}_{\mathrm{ref}}(s_2 - s_3) \right) \tag{11}$$

and the irreversibility due to pressure drop in the gas cooler is given by:

$$I_{\mathrm{gc}}^{\Delta P} = \sum_{\mathrm{pass}} \left\{ \dot{m}_{\mathrm{ref}} \sum_{\mathrm{segment}} \left(\frac{T_o \Delta P_{\mathrm{gcr}}}{T_{\mathrm{gcr}} \rho_{\mathrm{gcr}}} \right) + \dot{m}_{\mathrm{gcw}} \sum_{\mathrm{segment}} \left(\frac{T_o \Delta P_{\mathrm{gcw}}}{T_{\mathrm{gcw}} \rho_{\mathrm{gcw}}} \right) \right\} \tag{12}$$

The second law efficiency of the system has been calculated by,

$$\eta_{\mathrm{II}} = \frac{1}{\dot{W}_c} \left\{ \dot{Q}_{\mathrm{ev}} \left[\frac{T_o \ln(T_{\mathrm{evi}}/T_{\mathrm{evo}})}{(T_{\mathrm{evi}} - T_{\mathrm{evo}})} - 1 \right] + \dot{Q}_{\mathrm{gc}} \left[1 - \frac{T_o \ln(T_{\mathrm{gco}}/T_{\mathrm{gci}})}{(T_{\mathrm{gco}} - T_{\mathrm{gci}})} \right] \right\} \tag{13}$$

where, $\dot{Q}_{\mathrm{ev}}$, $\dot{Q}_{\mathrm{gc}}$ and $\dot{W}_c$ are cooling capacity, heating capacity and compressor power input, respectively.

4. Heat transfer and pressure drop correlations

4.1. Gas cooler

The heat transfer in gas cooler tubes occur at supercritical pressures where the thermo-physical properties of carbon dioxide change drastically. The large variation in the thermophysical properties causes the heat transfer coefficient to be greatly dependent on both the local temperature and the heat flux. The variation includes two aspects: variation along the direction of flow and variation perpendicular to the direction of fluid flow. Longitudinal discretization accommodates the former effect. To capture the variation in the perpendicular direction, Pitla et al. [16] presented a new correlation employing the *mean Nusselt number* concept based on a numerical and experimental study, and is given by:

$$Nu = \left(\frac{Nu_w + Nu_b}{2} \right) \frac{k_w}{k_b}, \qquad h = \frac{Nu}{D} k_b \tag{14}$$

where, k_w and k_b are thermal conductivities of CO_2 at wall and bulk temperatures, respectively. Nu_w and Nu_b are Nusselt numbers based on the thermophysical properties at the wall and bulk temperature respectively, evaluated from the Popov–Kirilov or Gnielinski correlations respectively, as shown below:

$$Nu = \frac{f/8 Re\, Pr}{12.7\sqrt{f/8}(Pr^{2/3} - 1) + 1.07} \tag{15}$$

$$Nu = \frac{(f/8)(Re - 1000)Pr}{1.07 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \tag{16}$$

Friction coefficient for constant thermophysical property, f, is given by Filonenko's equation:

$$f = \left[0.79 \ln(Re) - 1.64\right]^{-2} \tag{17}$$

The pressure drop has been estimated by the following correlations [17]:

$$\Delta p = \frac{G^2}{2\rho} \left(f \frac{L}{D} + 1.2 \right) \tag{18}$$

$$f = \left(1.82 \ln(Re) - 1.64\right)^{-2} \frac{\rho_w}{\rho_b} \left(\frac{\mu_w}{\mu_b} \right)^s; \quad \text{where}$$
$$s = 0.023 \left| \frac{q_w}{G} \right|^{0.42} \tag{19}$$

4.2. Evaporator

High pressure, very low viscosity and surface tension, and near critical operation make the flow boiling heat transfer and pressure drop phenomenon of carbon dioxide distinct from conventional refrigerants. Distinct film breakdown and dry-out phenomena make most of the general correlations unusable. In this analysis, the recently developed Yoon et al. [18] correlation has been employed to estimate the boiling heat transfer coefficient. The following correlation is proposed to predict critical quality [18]:

$$x_{\mathrm{cr}} = 38.27 Re_l^{2.12} (1000 Bo)^{1.64} Bd^{-4.7} \tag{20}$$

Where, Bond number, $Bd = g(\rho_l - \rho_g)d_i^2/\sigma$, and boiling number, $Bo = q/Gh_{fg}$.

Yoon et al. [18] proposed the following correlation for the heat transfer coefficient of CO_2:

For region $x < x_{cr}$

$$h_{tp} = \left[(Sh_{nb})^2 + (Eh_l)^2\right]^{1/2} \tag{21}$$

Nucleate boiling heat transfer coefficient and parameters S and E are given as:

$$h_{nb} = 55Pr^{0.12}(-\log_{10}Pr)^{-0.55}M^{-0.5}q^{0.67}$$

where M is the molecular weight.

$$S = \left[1 + 1.62 \times 10^{-6}E^{0.69}Re_l^{1.11}\right]^{-1}$$

$$E = \left[1 + 9.36 \times 10^3 \times Pr_l\left(\frac{\rho_l}{\rho_g} - 1\right)\right]^{0.11} \tag{22}$$

For region $x \geqslant x_{cr}$

$$h_{tp} = \frac{\theta_{dry}h_g + (2\pi - \theta_{dry})h_{wet}}{2\pi} \tag{23}$$

where h_{wet} is the heat transfer coefficient on the whetted portion of the tube given by,

$$h_{wet} = Eh_l$$

$$E = 1 + 3000Bo^{0.86} + 1.12\left(\frac{x}{1-x}\right)^{0.75}\left(\frac{\rho_l}{\rho_g}\right)^{0.41} \tag{24}$$

where θ_{dry}, the angle of dry portion, is closely related to flow pattern and is suggested as:

$$\frac{\theta_{dry}}{2\pi} = 36.23Re^{3.47}Bo^{4.84}Bd^{-0.27}\left(\frac{1}{X_{tt}}\right)^{2.6} \tag{25}$$

h_l and h_g are the heat transfer coefficients corresponding to saturated liquid and vapor, given by Dittus–Boelter equation.

Pressure drop has been calculated from modified Martinelli et al. correlation [19]

$$\Delta p_{TP} = \frac{2f_{fo}G^2L}{d\rho_l}\left[\frac{1}{\Delta x}\int_0^x \phi_{TP}^2\,dx\right]$$

$$f_{fo} = 0.046Re^{-0.2} \tag{26}$$

where, the two-phase multiplier is given by [19]:

$$\phi_{TP} = (1-x)^2 + 2.87x^2(P/P_{cr})^{-1} + 1.68x^{0.8}(1-x)^{0.25}(P/P_{cr})^{-1.64} \tag{27}$$

5. Results and discussion

The numerical model has been validated with authors' own test data [20] obtained from a system employing a single pass evaporator and gas cooler, without an internal heat exchanger. The double pipe evaporator has an inner tube with a diameter of 9.5 mm, wall thickness of 1 mm, an outer tube with a diameter of 16 mm and a wall thickness of 1 mm and a total heat exchanger length of 7.2 m. The gas cooler, a double pipe configuration as well, has an inner tube diameter of 6.35 mm, wall thickness of 1 mm, outer tube diameter of 12 mm with a wall thickness of 1 mm and a total length of 14.0 m. Validation of model prediction with test results for water mass flow rates of 1.5 and 1 kg/min, and water inlet temperatures of 30 and 30.5 °C in evaporator and gas cooler, respectively, at an evaporator pressure of 40 bar, shows a modest agreement with a maximum deviation of 14% for system COP and 15.5% for system second law efficiency. The trends for compressor discharge pressure are fairly similar between experimental and theoretically predicted results [20].

For a given capacity and operating conditions, the total irreversibility of a double pipe counterflow multi-pass heat exchanger (both evaporator and gas cooler are of this type) depends on the outer and inner tube diameters, length and number of passes, i.e.

$$I_{tot} = f(d, D, L, n) \tag{28}$$

To simplify the analysis, the area ratio between annulus and inner cross section are assumed to be constant ($= 2.0$). This is a more reasonable assumption compared to the case of constant diameter ratio because relative mass velocity and Reynolds number will retain approximately proportional as individual cross-sectional areas are varied. This will subsequently result in approximately equal pressure drop ratio (between refrigerant and secondary fluid) and heat transfer rate for certain operating condition. Hence, following these arguments, results are presented here for the three independent geometric parameters: d, L and n. Thickness of all the tubes has been taken as 1/10th of the outer diameter. Tube material for both evaporator and gas cooler is considered to be stainless steel.

Information on exergy losses for material use is limited. The exergy loss in steel production depends on the manufacturing processes and is reported to vary from 4 to 18 MJ/kg [5,21]. Here, an average value of 11 MJ/kg has been taken for exergy losses for primary steel. The exergy losses due to tube manufacturing, fabrication and insulation (density: 30 kg/m^3 and thickness = 5 mm) have been taken as 5.7 MJ/kg, 0.26 MJ per meter length and 71 MJ/kg, respectively [5].

The following operating parameters have been assumed in the analysis for the evaporator and the gas cooler. In case of the evaporator, inlet and outlet temperature of water are chosen to be 30 and 4 °C, respectively. For the gas cooler, inlet temperature and mass flow rate of water are taken as 30 °C and 2 kg/min, respectively, leading to an outlet water temperature of about 70–80 °C. The heating capacity and the corresponding outlet temperature may vary. The ambient temperature has been taken as 30 °C. Calculations are based on 1.0 TR (3.5167 kW) of cooling capacity; compressor power input (exergy input) and heating capacity may vary based on the design conditions.

5.1. *Irreversibility minimization of gas cooler*

The effect of tube diameter and length of a 2-pass gas cooler on the total irreversibility is shown in Fig. 3. The minimum irreversibility lies in the range of 6 mm diameter (inner tube) and 26 m length. Reducing the diameter below 4 mm yields a

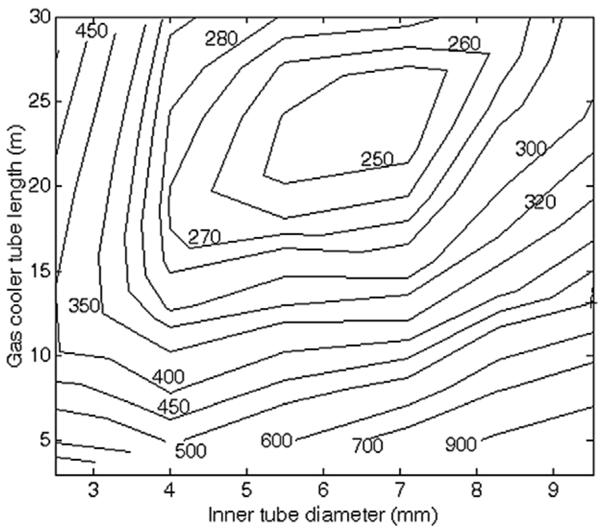

Fig. 3. Total irreversibility (in W) of a 2-pass gas cooler.

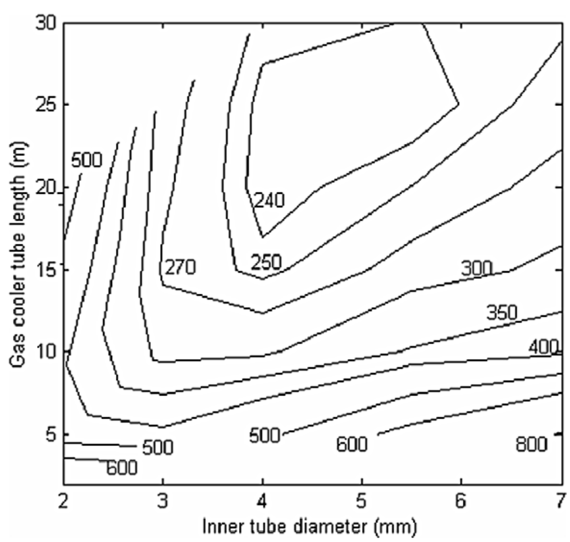

Fig. 4. Total irreversibility (in W) of a 5-pass gas cooler.

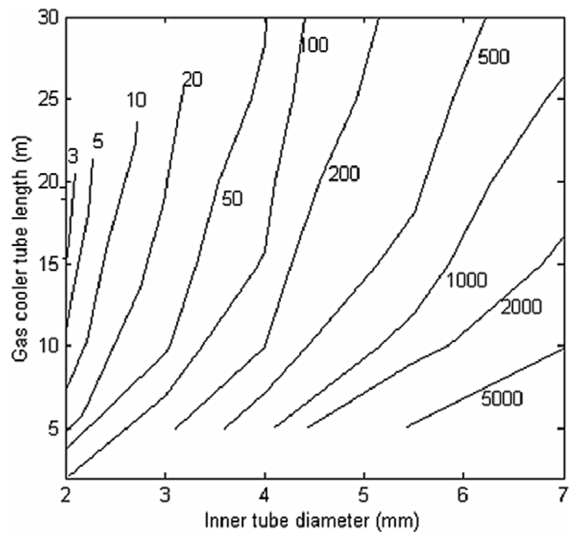

Fig. 5. Irreversibility ratio in a 5-pass gas cooler.

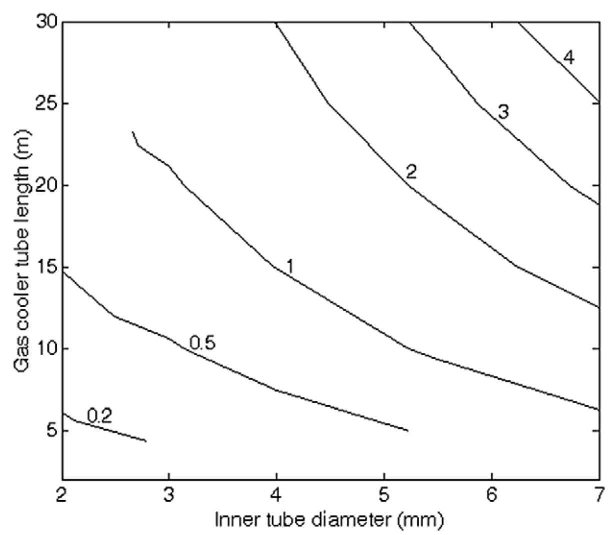

Fig. 6. Material related irreversibility I_{man} (in W) contours for a 5-pass gas cooler.

rapid increase in irreversibility due to a rapid increase in pressure drop. This can be avoided by increasing the number of passes. For a 5-pass unit, the optimum diameter and length are 4.5 and 23 m, respectively, as can be seen in Fig. 4. For a larger diameter and smaller length, the irreversibility due to thermal dissipation is high because of lower heat transfer coefficients; increasing the length can reduce this irreversibility, but that will also give rise to higher I_{man} and $I^{\Delta P}$. However, the effect of $I^{\Delta P}$ ($< 0.1\%$) and I_{man} ($< 2\%$) at larger diameter is fairly small as shown in Figs. 5 and 6. On the other hand, for a very small diameter, the fluid velocity rises (25 m/s for $d = 2$ mm and $n = 5$) leading to high Re (1.6×10^5 approx.) and a larger pressure drop as shown in Fig. 5, although the heat transfer coefficient will also increase in this case. For smaller diameter and longer heat exchangers, two problems will arise: (i) Due to increase in heat transfer coefficient, the temperature approach becomes zero, which is practically not feasible and hence there is no further decrease in thermal dissipative irreversibility, (ii) A rapid increase in pressure drop related irreversibility. It may be recalled from the pressure drop equation that for a specified size and operating condition, $G \propto n^{-1}$ and $Re \propto n^{-1}$, and thus the pressure drop is given by: $\Delta P \propto n^{-x}$. For water, $x = 1.75$ and for CO_2, $x > 1.5$. So, the irreversibility due to pressure drop $I^{\Delta P} \propto n^{1-x}$ (according to Eq. (12)). In the present case, for a 4 mm diameter and 30 m long heat exchanger, $I^{\Delta P} = 20$ W with $n = 2$ and $I^{\Delta P} = 9.5$ W with $n = 5$, and hence the value of x will be approximately 1.8. This small deviation is due to variation in other properties. I_{man} will increase linearly with n. It is very clear from the above discussion that with increase in n, $I^{\Delta P}$ will decrease and I_{man} will increase, so the optimum diameter and length will both decrease, but it is not a very easy task to find the relationship. With increase in number of passes from 2 to 5, the optimum diameter decreases as a function of $n^{-0.4}$ approximately; however, value of the exponent ($= 0.4$) will reduce marginally with increase in n due to increase in I_{man}, which provides a rough guideline on the minimum number of passes required to be used in a

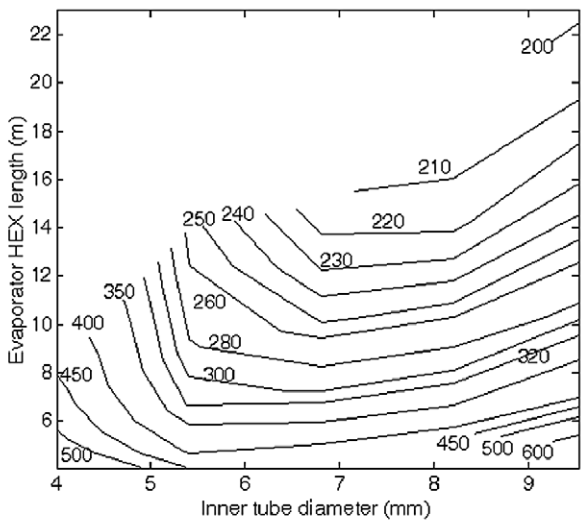

Fig. 7. Total irreversibility (in W) of a 2-pass evaporator.

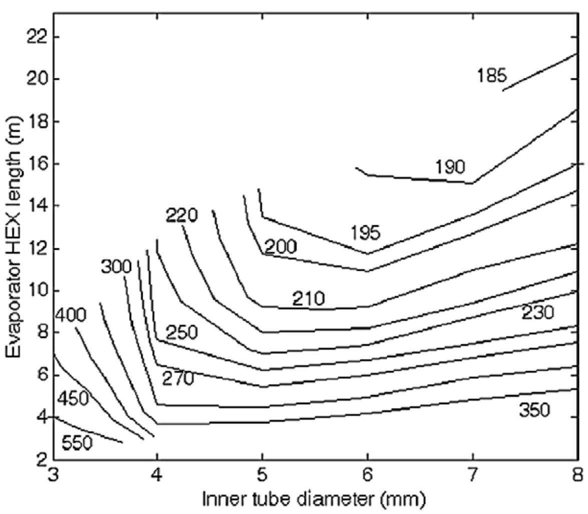

Fig. 8. Total irreversibility (in W) of a 5-pass evaporator.

heat exchanger of given tube diameter to attain minimum irreversibility.

5.2. *Irreversibility minimization of evaporator*

Effects of diameter and length of the evaporator with 2 and 5 passes on total irreversibility are shown in Figs. 7 and 8, respectively. Unlike the gas cooler, an optimum length does not exist for the evaporator, although an optimum diameter exists having values of 7.5 and 6 mm for 2 and 5 passes, respectively. Reducing the diameters (below 4 mm for 2 passes and 3 mm for 5 passes approximately) leads to very rapid increase in irreversibility due to pressure drop. This can be avoided by increasing the number of passes. As shown in the figures, the portion of the contour at lower diameter and higher length is incomplete. This can be attributed to the fact that the simulation is not feasible due to temperature approach tending to zero because of higher pressure drop. Beyond certain length, the temperature approach becomes negative which is an unacceptably trivial situation as the heat transfer direction reverses. Thus the

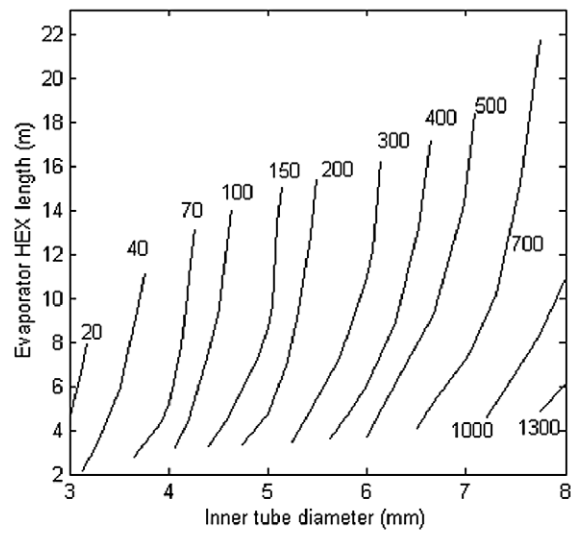

Fig. 9. Irreversibility ratio in a 5-pass evaporator.

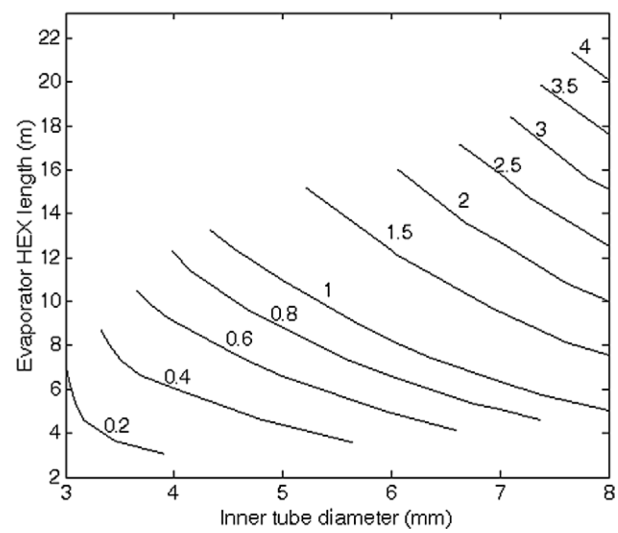

Fig. 10. Material related irreversibility I_{man} (in W) contours for a 5-pass evaporator.

minimum temperature difference approaches zero before the optimum length is attained. The effects of $I^{\Delta P}$ ($< 0.1\%$) and I_{man} ($< 2\%$) at higher diameter is very small as shown in Figs. 9 and 10, respectively, but the effect of $I^{\Delta P}$ ($> 5\%$) is high at smaller diameters. However, the contribution of pressure drop to the total irreversibility of evaporator is more (about 4 times) than that in case of the gas cooler; this may be attributed to the well-known dual effect of frictional and momentum pressure drop.

For both the evaporator and the gas cooler, the rational effectiveness (defined in Eq. (5)) varies between 75 to 95%. The effect of design parameters on the overall system performance is also very similar to that for individual heat exchangers. For minimum gas cooler irreversibility, the irreversibility of compressor and expansion valve are lower due to lower optimum gas cooler pressure and variation in evaporator irreversibility with gas cooler dimensions are not so significant. Hence the maximum overall second law efficiency of the system occurs at

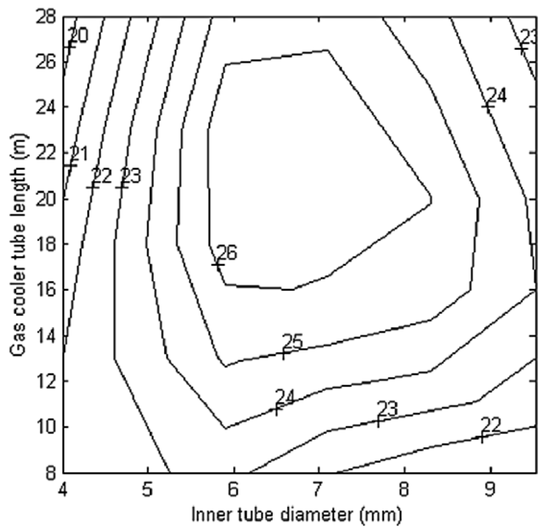

Fig. 11. Variation of second law efficiency with gas cooler dimensions.

nearly same ranges of diameter and length of 2-pass gas cooler as shown in Fig. 11. The heat transfer correlations, used in the present study, are based on normal tube diameters and hence may not be applicable for micro-channel configurations. It may be noted that although the presented optimum combination of dimensional parameters are for specific operating condition, the present simulation code can be used to find the optimal design combination for other operating conditions as well.

6. Conclusions

Irreversibility analyses of both the evaporator and gas cooler of a CO_2 based heat pump system have been carried out in this study. Water has been employed as the secondary fluid in both the heat exchangers. Typical operating conditions, as required in dairy plants, have been chosen for the analysis. Results clearly show that higher heat transfer coefficient can be achieved by reducing the diameter to a limited extent only due to rapid increase in pressure drop with reduction in diameter. With increase in length, thermal irreversibility decreases but irreversibilities due to both pressure drop and material become larger. Thus for a specified operating condition and capacity of gas cooler, a set of optimum diameter and length is possible for each set of passes and the optimum diameter and length will decrease with increase in number of passes. For the evaporator, although an optimum diameter has been obtained, optimum length could not be obtained as the temperature approach becomes zero before the optimal length could be attained. For the input data considered here, the effect of material use on irreversibility is found to be negligible. Although the effect of pressure drop on the irreversibility can be neglected for higher diameter, it is considerable for smaller diameter tubes. The effect of pressure drop depends on the number of passes and mass velocity of the fluid. Irreversibility due to pressure drop is higher for the evaporator compared to that in the gas cooler. The results presented here are expected to provide useful guidelines for the design of heat exchangers in terms of optimum geometry (diameter, length and number of passes) for similar systems; however, it must be noted that the optimum set depends on the capacity and the operating parameters.

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