Effect of pressure drop and longitudinal conduction on exergy destruction in a concentric-tube micro-fin tube heat exchanger

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Abstract: A numerical study is carried out to predict the heat transfer characteristics and various exergy losses in a micro-fin concentric-tube double pipe heat exchanger. The results are compared with the experimental data from the literature and are found to be in a good agreement. The study is extended to include the effect of pressure drop and longitudinal conduction on the exergy losses. Furthermore, Ethylene Glycol-Water as a hot fluid is considered to investigate the effect of higher viscosity on the frictional irreversible losses. It is found that, at lower mass flow rates, fluid frictional irreversibility can be ignored for a low viscous fluid; however, there is a substantial exergy loss for a highly viscous fluid. At higher mass flow rates, fluid frictional irreversibility dominates over thermal irreversibility for both the fluids investigated in this study.

Keywords: micro-fin tube heat exchanger; exergy analysis; longitudinal conduction.


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Syed M. Zubair is a Distinguished Professor in Mechanical Engineering Department at King Fahd University of Petroleum & Minerals (KFUPM). He earned his PhD from Georgia Institute of Technology, Atlanta, Georgia, USA, in 1985. He has participated in several research projects at KFUPM and has published over 200 research papers in internationally refereed journals.
Due to his various activities in teaching and research, he was awarded Distinguished Teacher and Researcher awards by the university. In addition, he received best Applied Research award on Electrical and Physical Properties of Soils in Saudi Arabia from GCC-CIGRE group in 1993.

1 Introduction

Energy saving is an important aspect of the design, construction, and operation of heat exchangers. For this reason, development of heat transfer technologies has been the centre of research interest and many improvements have been applied to, in the past decades. A large number of attempts have been carried out to reduce the size and cost of heat exchangers. The main design philosophy is to enhance the heat transfer capability and reduce the pressure drop across the unit. It is expected that an improved heat transfer coefficient reduces the temperature gradient and hence reduces the irreversibility.

It is important to note that first law deals with the amount of heat transferred, while the minimisation of losses can be studied by the second-law analysis. The second-law mainly deals with the degradation of energy during a given process; that is, entropy generation or the lost work potential. The entropy generation in heat exchangers is primarily due to the temperature gradient and fluid friction.

The second-law-based performance evaluation of heat exchangers has been extensively studied by Yilmaz et al. (2001). They presented both the entropy generation and exergy losses as evaluation parameters; in addition, they also discussed the thermoeconomic analysis of heat exchangers. Akpinar (2005, 2006) experimentally investigated heat transfer, friction factor and exergy losses in a double pipe heat exchanger equipped with helical wires and swirl generators. An increase of up to 16% in exergy loss was observed for the heat exchanger equipped with the helical wires, while 25% increase in the exergy loss was observed for that equipped with the swirl generator. Second law analysis of a cross-flow heat exchanger has also been studied by Fan and Luo (2009). Naphon (2006) conducted an experimental study as well as numerical investigation on the temperature profile, entropy generation and exergy losses in a concentric-tube double-pipe heat exchanger. He further extended his investigation with the same setup with the introduction of micro-fins on the hot fluid side (Naphon, 2011). In both of these studies, he ignored the contribution of pressure drop across the pipe on the total irreversible losses in the heat exchanger.

The effect of pressure drop on exergy transfer effectiveness of heat exchangers for both gaseous and incompressible fluids has been investigated by Wu et al. (2007). Various configurations such as parallel-, counter-, and cross-flow have been compared and presented. Gupta and Das (2007) performed second-law analysis of a cross-flow heat exchanger in the presence of non-uniformity of flow. Performance evaluation in terms of first- and second-law analysis of a cross-flow serpentine heat exchanger has been extensively studied and presented by San et al. (2009) and San and Pai (2009).

Eslamain et al. (2009) studied the entropy generation in heat exchangers that are used in indirect evaporative coolers. They calculated an optimum tube length corresponding to a specific tube diameter, heat transfer area and flow conditions of a tube-in-tube type heat exchanger that gives the lowest entropy generation. Similarly, for plate type heat exchangers, they found an optimum primary flow length that gives the highest
thermodynamic efficiency. A detailed analysis of a micro-channel heat exchanger based on the second law of thermodynamics was presented by Askar et al. (2012). They found that the entropy generation number increases with the Reynolds number. It increases more rapidly at higher Reynolds number as the exergy loss due to pressure drop rises. Kuddusi (2016) derived analytical expressions for entropy generation calculation in a microchannel and found that the heat transfer is the main source of irreversibility in a heat exchanger. Fakheri (2010) reiterated that the entropy generation and exergy losses in a heat exchanger should not be criteria for the performance evaluation of a heat exchanger, neither effectiveness has any correlation with the irreversibility. He proposed a new performance evaluation parameter, entropy flux, which represents some desirable attributes of a heat exchanger. The loss of thermal and mechanical exergy for an ideal and incompressible flow in heat exchangers with waste heat recovery has been studied by San (2010).

The above literature survey suggests that although the second-law analysis has been widely studied for various types of heat exchangers, a thorough examination including the effect of frictional losses and longitudinal conduction for a double-pipe heat exchanger is yet to be investigated. In this work, a mathematical model has been developed for the analysis of exergy losses in a micro-fin tube heat exchanger. The model is verified against the experimental results obtained by Naphon (2011). The study is extended to include the effect of frictional losses and losses due to longitudinal conduction as well as the contribution of each component to the overall losses. Furthermore, the influence of fluid viscosity on total exergetic losses and the contribution of frictional loss is presented and discussed.

2 Mathematical model

In the present mathematical model, the energy conservation equations for the elemental volume on both the hot- and cold-fluid sides, and the tube wall are solved simultaneously, to obtain temperature profiles along the length of the tube. The governing differential equations were discretised using the finite-difference method and the resulting algebraic equations were solved simultaneously using engineering equation solver (EES) software.

2.1 Energy conservation

Heat transfer coefficient for the convective turbulent heat transfer in the tube was found by using Gnielinski (San, 2010) correlation:

\[
\text{Nu}_h = \frac{(f/8)(\text{Re}_D-1000)\text{Pr}}{1+12.7(f/8)^{1/2}(\text{Pr}^{2/3}-1)},
\]

where \(f\) is the Darcy friction factor given as:

\[
f = \left(0.79 \ln \left(\text{Re}_D \right) - 1.64\right)^{-2}.
\]

For the elemental section (refer to Figure 1), energy conservation model for the hot fluid can be expressed as,
\[
\dot{C}_\text{h} T_h - \dot{C}_\text{h} \left( T_h + \frac{dT_h}{dx} \delta x \right) - h_h \left( \frac{S_h}{L} \right) \left( T_h - T_w \right) = 0
\]  
(3)

or,
\[
\dot{C}_\text{h} \frac{dT_h}{dx} + h_h \left( \frac{S_h}{L} \right) \left( T_h - T_w \right) = 0
\]  
(4)

where \( \dot{C}_\text{h} \) is the hot fluid capacitance rate, \( T_h \) is hot-side fluid temperature, \( T_w \) is wall temperature, \( h_h \) is hot-side heat transfer coefficient, \( S_h \) is heat transfer surface area at the hot side and \( L \) is the total length of the tube.

**Figure 1** Schematic diagram of the elemental concentric tube

Energy balance for the tube wall for elemental control volume (refer to Figure 1) can be written as:
\[
\left( -kA \frac{dT_h}{dx} + kA \frac{dT_h}{dx} \left( T_h + \frac{dT_h}{dx} \delta x \right) + h_h \left( \frac{S_h}{L} \right) \left( T_h - T_w \right) \right) - h_h \left( \frac{S_h}{L} \right) \left( T_h - T_c \right) = 0
\]  
(5)

or,
\[
kA \frac{d^2 T_h}{dx^2} + h_h \left( \frac{S_h}{L} \right) \left( T_h - T_w \right) - h_h \left( \frac{S_h}{L} \right) \left( T_h - T_c \right) = 0,
\]  
(6)

where \( k \) is the tube-wall thermal conductivity, \( A \) is tube-wall cross-sectional area, \( T_c \) is cold-side fluid temperature and \( h_h \) is heat transfer coefficient on the cold side.

Similarly, energy conservation equation for the cold side control volume can be written as:
\[
\dot{C}_\text{c} T_c - \dot{C}_\text{c} \left( T_c + \frac{dT_c}{dx} \delta x \right) - h_c \left( \frac{S_c}{L} \right) \left( T_c - T_w \right) = 0
\]  
(7)

or,
\[
\dot{C}_\text{c} \frac{dT_c}{dx} \delta x + h_c \left( \frac{S_c}{L} \right) \left( T_c - T_w \right) = 0,
\]  
(8)
where $\dot{C}_c$ is the heat capacitance rate of the cold fluid and $S_c$ is heat transfer area on the cold side.

The fluid temperatures and wall temperature can be written in the dimensionless form as:

$$\theta = \frac{T - T_{c,in}}{T_{h,in} - T_{c,in}}.$$  \hspace{1cm} (9)

### 2.2 Second law analysis

For an internal flow, mass and energy transfer in an elemental control volume is represented in Figure 2. The entropy balance equation for the control volume can be expressed as,

$$\frac{\dot{q}'}{T + \Delta T} + d\dot{S}_{gen} = \dot{m} ds,$$  \hspace{1cm} (10)

where, $\dot{q}'$ is the heat transfer rate per unit length of the tube, $T$ is fluid temperature, $\dot{S}_{gen}$ is entropy generation rate and $\dot{m}$ is the mass flow rate of the fluid.

**Figure 2** Schematic diagram of elemental control volume

For any pure substance, we can write as,

$$\frac{dh}{dx} = T \frac{ds}{dx} + \frac{1}{\rho} \frac{dp}{dx},$$  \hspace{1cm} (11)

where, $h$ is the specific enthalpy, $s$ is specific entropy, $\rho$ is fluid density and $p$ is a pressure drop.

Substituting equation (11) in equation (10) and after some algebraic manipulation, gives (Bejan, 1995):

$$\dot{S}'_{gen} = \frac{\dot{q}' \Delta T}{T^2} + \dot{m} \left( \frac{dp}{dx} \right)^{\rho T}.$$  \hspace{1cm} (12)
where, $S_{\text{gen}}'$ is the entropy generation rate per unit length of the tube. The first term on the right side of the above equation represents the irreversibility due to temperature gradient while the second term represents the irreversibility due to the pressure gradient.

The entropy balance in a simple system, represented by Figure 1 can be expressed as,

$$S_{\text{system}} = \frac{\dot{q}}{T_s} - \frac{\dot{q}}{T_c} + S_{\text{gen}},$$  \hspace{1cm} \text{(13)}$$

where $\dot{q}$ is the rate of heat conduction. It is important to mention that there will be entropy generation in the system because of the existence of the temperature gradient along the tube wall; for example, say $T_s = T$ and $T_c = T - \Delta T$. At steady state, $S_{\text{system}} = 0$; thus, entropy generation due to the temperature gradient in the tube wall can be expressed as:

$$S_{\text{gen}} = \dot{q} \left[ \frac{\Delta T}{T(T - \Delta T)} \right],$$ \hspace{1cm} \text{(14)}$$

where $S_{\text{gen}}$ is the entropy generation due to the longitudinal conduction. Since the temperature difference along the infinitesimal length is small, the entropy generation per unit length of the tube wall can be expressed as:

$$S'_{\text{gen}} = \frac{1}{T^2} \frac{dT}{dx}$$ \hspace{1cm} \text{(15)}$$

Substituting $\dot{q} = kA dT / dx$ in the above equation gives,

$$S'_{\text{gen}} = kA \frac{1}{T^2} \left( \frac{dT}{dx} \right)^2.$$ \hspace{1cm} \text{(16)}$$

$S'_{\text{gen}}$ is the entropy generation due to the longitudinal conduction per unit length of the tube wall, $k$ is the thermal conductivity of the tube material and $A$ is the inner tube cross-sectional area.

2.3 Problem description

A double pipe heat exchanger with concentric-tube having a circular cross-section pipe of 2 m length is taken into consideration for the present analysis. The dimensions are taken from the experimental setup that was used by Naphon (2011). The experiments were conducted by Naphon using water both as hot and cold fluids in a closed loop having storage tanks of 0.5 m$^3$ each. An electric heater is used for heating the hot fluid and a chiller using R22 as a refrigerant is used to cool the cold fluid. Water flow rates were controlled by using flow meter having an accuracy of ±0.2%. T-type thermocouples, with ±0.1% accuracy, were used to measure the temperature of the tube at nine different sections. A data acquisition system with 40 channel connections was used to collect the data. The details of the geometry are given in Table 1. The hot fluid enters the inner tube at a temperature of 45°C and the cold fluid enters the annular section at a temperature of 15°C. The mass flow rates of both the fluids are varied to study different operating conditions.
Table 1  Details of the geometry used for the analysis

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner and outer diameter of micro-fin tube (mm)</td>
<td>8.92/9.52</td>
</tr>
<tr>
<td>Length of micro-fin tube (mm)</td>
<td>2000</td>
</tr>
<tr>
<td>Inner surface area of micro-fin tube (mm²/m)</td>
<td>$430 \times 10^2$</td>
</tr>
<tr>
<td>Thickness of micro-fin tube (mm)</td>
<td>0.30</td>
</tr>
<tr>
<td>Number and height of fin (mm)</td>
<td>60/0.2</td>
</tr>
</tbody>
</table>

Source: Naphon (2011)

3 Results and discussion

The above mathematical model is solved for various inlet conditions of the double-pipe heat exchanger and the results are compared with the experimental data available from the literature (Naphon, 2011). Figure 3 presents the variation of cold water outlet temperature with the hot water mass flow rate, for two different hot water inlet conditions. The mass flow rate of cold water was set to 0.1 kg/s and cold water inlet temperature is set to 15°C. The numerical results show an excellent agreement with the experimental values found for cold water outlet temperatures.

Figure 3  Variation of outlet cold water temperature with hot water mass flow rate for different inlet hot water temperature (see online version for colours)

The average heat transfer rate in the heat exchanger varied with the hot water mass flow rate is shown in Figure 4. Two cases of cold water mass flow rates are considered, 0.03 kg/s and 0.10 kg/s, at the cold and hot water inlet temperatures of 15°C and 45°C,
respectively. The predicted results for lower cold water mass flow rate deviates more than that at higher mass flow rate with the maximum deviation of 30% from the experimental data. At higher cold water mass flow rate, the deviation from the experimental data does not exceed more than 10%.

Figure 4 Variation of average heat transfer rate with hot water mass flow rate for different cold fluid mass flow rates (see online version for colours)

Entropy generation due to heat transfer in the double pipe heat exchanger is presented in Figure 5. The results ignore the irreversibility due to pressure drop along the pipe as was reported in the earlier experimental studies (Naphon, 2011). Two cases of cold water mass flow rates have been considered namely, 0.03 kg/s and 0.10 kg/s, with the inlet cold and hot water temperatures of 15°C and 45°C, respectively. The results show good agreement for the entropy generation at a higher mass flow rate of cold water; however, at lower mass flow rate the deviation is around 30%.

Another parameter, which is described as entropy generation number (EGN), is defined as the ratio of entropy generation per unit minimum fluid capacitance rate \(\frac{S_{gm}}{C_{min}}\), has been investigated. Figure 6 presents the variation of EGN as a function of hot water mass flow rates at the same operating conditions. It can be seen from the figure that EGN increases with the increase in hot water mass flow rate as well as at a lower cold water mass flow rate. However, at a higher cold water mass flow rate, the entropy generation decreases with the increase in hot water mass flow rate. This trend is found to be in line with the experimental data reported earlier in Naphon (2011).
3.1 Effect of pressure drop on entropy generation

As discussed above, the irreversibilities due to pressure drop and longitudinal conduction have been ignored by Naphon (2011). To analyse the contribution of irreversibilities due to fluid friction and longitudinal conduction, the mathematical model discussed in the
previous section considers both the entropy generation equations due to the pressure drop across the tube as well as the temperature gradient in the tube wall, in addition to the irreversible losses due to the temperature difference. Figure 7 presents the profiles of the non-dimensional temperature of the cold fluid, hot fluid and tube wall along the tube length, for the balanced flow case. These temperature profiles are typical of a counter-flow double-pipe heat exchanger, wherein there is a clear indication of the temperature gradient in the tube wall along the length of the heat exchanger.

**Figure 7** Dimensionless temperature profiles of hot fluid, cold fluid, and tube wall across the tube length (see online version for colours)

The EGN as a function of a number of transfer units (NTU) is plotted in Figure 8. Hot water inlet temperature is set at 45°C and cold water inlet temperature and mass flow rate at 15°C and 0.03 kg/s, respectively. The hot side Reynolds number is varied from 3600 to 12,000 while the cold side Reynolds number is fixed around 1800. It is important to mention that thermal loss represents the irreversibility due to the temperature gradient across the fluids while overall loss also takes into account the losses due to the longitudinal conduction and pressure drop across the heat exchanger. With increasing NTU, both thermal and overall EGN increases; however, thermal EGN is almost linear, while the overall EGN increases exponentially with the NTU. Initially, there is a drop in the percentage increase of EGN due to the fact that increasing EGN until the value of 0.275 does not affect the frictional irreversibility significantly, and the thermal irreversibility contributes to almost all the irreversibility increase. However, at higher flow rates, the frictional irreversibility rises exponentially with NTU, causing a significant rise in the percentage increase curve. We can see that at NTU = 0.262, the percentage increase in EGN by including both the losses is 38%. However, there is a significant increase in EGN with almost 100% increase in the value at NTU = 0.296. This clearly demonstrates that it is not appropriate to neglect irreversible losses due to the pressure drop and longitudinal conduction.
Effect of pressure drop and longitudinal conduction on exergy destruction

Figure 8  Entropy generation number as function of NTU at a lower cold fluid mass flow rate \((\dot{m}_c = 0.03 \text{ kg/s})\)  (see online version for colours)

![Figure 8](image)

For further insight, the irreversibility contribution by each component is presented in Figure 9. It is apparent that the thermal irreversibility is dominant in the selected range, contributing to more than 50% of the total value, followed by the pressure drop. Irreversible losses due longitudinal conduction do not constitute more than 10% of the total irreversible losses. The fractional irreversibility due to the longitudinal conduction as well as a thermal gradient across the fluids decreases with increasing NTU, while the frictional irreversibility increases. This suggests that at higher flow rates, the frictional irreversibility becomes significant, and cannot be ignored in any design calculations.

Figure 9  Irreversibility due to longitudinal conduction, pressure drop and temperature gradient across the fluids at lower cold fluid mass flow rate \((\dot{m}_c = 0.03 \text{ kg/s})\)  (see online version for colours)

![Figure 9](image)
To further enlighten the influence of flow rates on total irreversibility, cold water mass flow rate was increased to 0.1 kg/s, corresponding to Reynolds number of around 6100. The results are presented as a function of NTU in Figures 10 and 11. With increasing NTU, both thermal EGN and overall EGN increases almost linearly; however, the slope of later is higher, suggesting pressure drop irreversibility getting more dominant. The percentage increase in EGN due to the consideration of pressure and longitudinal conduction irreversibility increases from around 120% at NTU value of 0.34 to more than 200% at the value of 0.78.

Figure 10   Entropy generation number as function of NTU at higher cold fluid mass flow rate ($\dot{m}_c = 0.1$ kg/s) (see online version for colours)

Figure 11   Irreversibility due to longitudinal conduction, pressure drop and temperature gradient across the fluids at higher cold fluid mass flow rate ($\dot{m}_c = 0.1$ kg/s) (see online version for colours)
The fractional contribution of each component is presented in Figure 11. It clearly shows that the pressure drop irreversibility, unlike the previous case, has the major contribution constituting more than 50% of the total value. The irreversibility due to the longitudinal conduction remains lower than 5% for the selected range; however, the fractional contribution increases with increasing NTU. Irreversibility fraction due to thermal gradient across the fluids decreases with increasing NTU because of the fact that the pressure drop becomes more dominant at higher Reynolds numbers. It is, therefore, evident that the irreversibility due to the frictional losses is as significant as that due to thermal gradient for the micro-fin double-pipe heat exchanger.

### 3.2 Effect of fluid viscosity

As has been shown in the above results, the contribution of frictional irreversibility plays a major role in the exergy loss for the double pipe heat exchanger under consideration, especially at higher flow rates. The above results are based on Water as both the cold and hot fluids. In this section the hot fluid is replaced by 50–50% dilution of ethylene glycol (EG) and Water, having viscosity around three times higher than that of water.

Figure 12 shows the comparison of irreversibility fractions due to the temperature gradient, longitudinal conduction and pressure drop for

- water
- EG-Water as a hot fluid at a lower cold water mass flow rate \( \dot{m}_c = 0.03 \text{ kg/s} \).

It is evident from Figure 12 that the change in viscosity of the fluid has a significant effect on the irreversibility due to the pressure drop. There is around 20% rise in the irreversibility fraction due to a pressure drop at lower NTU values, while the gap increases to around 40% at higher NTU values when the hot fluid is changed from Water to EG-Water solution. Subsequently, the irreversibility fraction due to the thermal gradient can be seen to drop in the same ratio. Irreversibility due to thermal gradient remains the major contributor for the Water-Water case in the selected range of NTU; however, the frictional irreversibility can be seen to be dominant when the hot fluid is replaced with an EG-Water solution. For the base case, irreversibility fraction due to longitudinal conduction varies from around 5% to 10% within the selected NTU range; nevertheless, with EG-Water as hot fluid, the contribution of longitudinal conduction irreversibility reduces to less than 5% of the overall irreversibility.

At a higher cold fluid mass flow rate \( \dot{m}_c = 0.1 \text{ kg/s} \), which corresponds to cold-side Reynolds number of around 6000, the frictional irreversibility constitutes more than 50% of the total irreversibilities for the base case, which can be seen to further increase with higher viscous fluid as a hot fluid (refer to Figure 13). Thermal irreversibility reduces to around 30% when using EG-Water as a hot fluid when compared to more than 40% for the Water-Water case, within the selected NTU range. The irreversibility fraction due to the longitudinal conduction remains less than 3% of the total for both the cases. Hence, it is clearly demonstrated that ignoring frictional irreversibility in a micro-fin double pipe heat exchanger, especially dealing with highly viscous fluid, can give misleading results in the context of the second law of thermodynamics.
Figure 12 Irreversibility fraction with: (i) water and (ii) Ethylene Glycol-water as hot fluid as a function of NTU at lower cold fluid mass flow rate \((m_c = 0.03 \text{ kg/s})\) (see online version for colours)

Figure 13 Irreversibility fraction with: (i) water and (ii) ethylene glycol-water as hot fluid as a function of NTU at higher cold fluid mass flow rate \((m_c = 0.1 \text{ kg/s})\) (see online version for colours)
4 Concluding remarks

A numerical model is used to simulate thermal-hydraulic characteristics of a micro-fin concentric-tube double-pipe heat exchanger. The numerical results obtained from the model are compared with the experimental data from the literature. It is found that the results show a good agreement with the experimental values reported in the literature. The study was extended to analyse the effect of frictional and longitudinal exergy losses on the total losses in the heat exchanger. It was found that even at lower mass flow rates, the frictional irreversibility constitutes a significant portion of the total value; however, the thermal irreversibility is the dominant one in most of the cases investigated in this paper. However, at higher flow rates, frictional irreversibility constitutes the major share of the overall irreversible losses. While the exergy loss due to the longitudinal conduction is, around 10% of the total value at lower flow rates, whereas at higher flow rates it reduces to less than 5%.

Furthermore, the effect of viscosity on the exergy loss is analysed by considering Water with Ethylene Glycol-Water as the hot fluid. It was found that the contribution of frictional exergy loss is even more prominent when the working fluid has higher viscosity. For Ethylene Glycol-Water case, the fraction of frictional irreversibility is around 50% or more and hence, it can be said that ignoring the exergy loss due to pressure drop gives an unrealistic insight of the total exergy loss for higher viscosity fluids. The irreversibility fraction due to the longitudinal conduction is almost insignificant with a share of less than 5% of the total value when using the highly viscous fluid.

Acknowledgements

The authors acknowledge the support provided by King Fahd University of Petroleum and Minerals through the project IN151001.

References


**Nomenclature**

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<tr>
<th>Symbol</th>
<th>Description</th>
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<tr>
<td>$A$</td>
<td>Tube wall cross-sectional area, m²</td>
</tr>
<tr>
<td>$C_C$</td>
<td>Cold side heat capacitance, WK⁻¹</td>
</tr>
<tr>
<td>$C_H$</td>
<td>Hot side heat capacitance, WK⁻¹</td>
</tr>
<tr>
<td>$C_{\min}$</td>
<td>Minimum of the two heat capacitance, WK⁻¹</td>
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<tr>
<td>$f$</td>
<td>Darcy friction factor,</td>
</tr>
<tr>
<td>$h$</td>
<td>Specific enthalpy, J kg⁻¹</td>
</tr>
<tr>
<td>$h_C$</td>
<td>Cold side heat transfer coefficient, Wm⁻²K⁻¹</td>
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<tr>
<td>$h_H$</td>
<td>Hot side heat transfer coefficient, Wm⁻²K⁻¹</td>
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<tr>
<td>$k$</td>
<td>Thermal conductivity, Wm⁻¹K⁻¹</td>
</tr>
<tr>
<td>$L$</td>
<td>Total length of the tube, m</td>
</tr>
<tr>
<td>$m$</td>
<td>Mass flow rate of the fluid, kg s⁻¹</td>
</tr>
<tr>
<td>$N_{\text{Nu},D}$</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure, Pa</td>
</tr>
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<td>$Pr$</td>
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<tr>
<td>$q'$</td>
<td>Heat transfer rate per unit length, Wm⁻¹</td>
</tr>
<tr>
<td>$Re_D$</td>
<td>Reynolds number based on diameter</td>
</tr>
<tr>
<td>$S_C$</td>
<td>Heat transfer surface area on the cold side, m²</td>
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<thead>
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<th>Symbol</th>
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<tr>
<td>$S_h$</td>
<td>Heat transfer surface area on the hot side, m²</td>
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<td>$\dot{S}_{gen}$</td>
<td>Entropy generation rate, W K⁻¹</td>
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<td>$s$</td>
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<td>$T_h$</td>
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<td>$T_w$</td>
<td>Tube wall temperature, °C</td>
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**Greek symbols**

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<tr>
<td>$\theta$</td>
<td>Dimensionless temperature</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>Dimensionless length</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Fluid density, kg m⁻³</td>
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**Subscripts**

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<thead>
<tr>
<th>Subscript</th>
<th>Description</th>
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<tr>
<td>$D$</td>
<td>Hydraulic diameter</td>
</tr>
<tr>
<td>$h$</td>
<td>Hotter side</td>
</tr>
<tr>
<td>$in$</td>
<td>Inlet</td>
</tr>
<tr>
<td>$out$</td>
<td>Outlet</td>
</tr>
<tr>
<td>$w$</td>
<td>Tube wall</td>
</tr>
</tbody>
</table>