Iranian Journal of Chemical Engineering
Vol. 14, No. 3 (Summer 2017), IAChE

Transient Thermal Study of Recuperative Tube in Tube Heat Exchanger Operating in Refrigeration System Using Experimental Test and Mathematical Simulation

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ARTICLE INFO

Article history:
Received: 2016-05-20
Accepted: 2016-10-19

Keywords:
Heat Exchanger
Joule-Thomson
Transient
Cryogenic
Refrigeration

ABSTRACT

Joule-Thomson cooling systems are used in refrigeration and liquefaction processes. There are extensive studies on Joule-Thomson cryogenic systems, but most of them cover steady-state conditions or lack experimental data. In the present work, transient thermal behavior of Joule-Thomson cooling system, including counter current helically coiled tube in tube heat exchanger, expansion valve, and collector, was studied by experimental tests and simulations. The experiments were carried out by small gas liquefier and nitrogen gas as working fluid. The recuperative heat exchanger was thermally analyzed by experimental data obtained from gas liquefier. In addition, the simulations were performed by an innovative method using experimental data as variable boundary conditions. A comparison was done between the presented and conventional methods. The effect of collector mass and convection heat transfer coefficient was also studied using temperature profiles along the heat exchanger. The higher convection heat transfer coefficient in low-pressure gas leads to the increase in exchanging energy between two streams and faster cooling of heat exchanger materials, but the higher convection heat transfer coefficient in high-pressure gas does not influence cool-down process.

1. Introduction

Joule-Thomson effect is used in refrigeration/liquefaction processes by adiabatic expansion of pressurized gases. Ever since its discovery, cooling to cryogenic temperatures with Joule-Thomson effect is one of the most widely known and experimented methods. Simple design, fixed parts, high reliability, less maintenance, and low cost are the main benefits of gas liquefiers operating with Joule-Thomson methods [1]. The most important element of such gas liquefiers is counter current recuperative heat exchanger operating in the final section of the system. Pacio and Dorao [2] reviewed the thermal hydraulic models of cryogenic heat exchangers. They introduced physical effects, such as changes in fluid

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properties, flow maldistribution, axial longitudinal heat conduction, and heat leakage, as the main challenges of cryogenic heat exchangers. Aminuddin and Zubair [3] studied various losses in a cryogenic counter flow heat exchanger numerically. They discussed the effect of longitudinal heat conduction loss as a parasitic heat loss by conducting heat exchanger cold end to the adjacent components, but they did not perform any experimental tests. Krishna et al. [4] studied the effect of longitudinal heat conduction in the separating walls on the performance of three-fluid cryogenic heat exchanger with three thermal communications. They reasoned that the thermal performance of heat exchangers operating at cryogenic temperature is strongly governed by various losses such as longitudinal heat conduction through the wall, heat in-leak from the surroundings, flow maldistribution, etc. Gupta et al. [5] investigated the second law analysis of counter flow cryogenic heat exchangers in the presence of ambient heat in-leak and longitudinal heat conduction through wall. They cited the importance of considering the effect of longitudinal heat conduction in the design of cryogenic heat exchangers. Nellis [6] presented a numerical model of heat exchanger in which the effects of axial conduction, property variations, and parasitic heat losses on the environment have been explicitly modeled. He concluded that small degradation exists in the performance of heat exchanger under the conditions in which the temperature of heat exchanger cold end is equal to temperature of the inlet cold fluid. Narayanan and Venkatarathnam [7] presented a relationship between the effectiveness of a heat exchanger losing heat at the cold end. They studied a Joule-Thomson cryo-cooler, and concluded that the hot fluid outlet temperature will be lower in the heat exchangers with heat in-leak at the cold end with respect to heat exchangers with insulated ends. Ranganayakulu et al. [8] studied the effect of longitudinal heat conduction in compact plate fin and tube fin heat exchanger using finite-element method. They indicated that the thermal performance deteriorations of cross flow plate-fin, cross flow tube-fin, and counter flow plate-fin heat exchangers due to longitudinal heat conduction may become significant, especially when the fluid capacity rate ratio is equal to one and longitudinal heat conduction parameter is large. Damle and Atrey [1] studied the effect of reservoir pressure and volume on the cool-down behavior of a miniature Joule-Thomson cryo-cooler considering the distributed Joule-Thomson effect in heat exchanger tube. Chou et al. [9] presented a preliminary experimental and numerical study of transient characteristics for a miniature Joule-Thomson cryo-cooler. Tzabar and Kaplansky [10] analyzed the cool-down process for Dewar-detector assemblies cooled with Joule-Thomson cryo-coolers by finite-element method. Hong et al. [11] studied the cool-down characteristics of a miniature Joule-Thomson refrigerator. They discussed the influence of the supply pressure and the temperature on the mass flow rate during the cool-down stage. Maytal [12] studied the cool-down periods of a fast Joule-Thomson cryo-cooler for nitrogen and argon as coolants. Chien et al. [13] performed an experimental and numerical study on transient characteristics of the self-regulating Joule-Thomson cryo-cooler. They developed modeling of the bellows control mechanism and the simulation of the cooler system. The performed studies commonly cover the Joule-

Thomson cryo-coolers with Hampson-type heat exchangers. Moreover, most of them lack experimental data and use constant pre-cooling temperature as boundary conditions. In the present work, an attempt was made to consider transient thermal behavior of Joule-Thomson cooling system operating in small gas liquefier by experimental tests and simulations. The tests were performed by a counter current helically coiled tube in tube heat exchanger and the obtained pre-cooling temperatures were used in simulations as variable boundary conditions. Installing temperature sensors inside the tube can change the flow regime and obtain unreal temperature profiles. So, mathematical simulations can solve the problem and eliminate the cost of experimental tests by using appropriate assumption and boundary conditions. Therefore, by applying the actual pre-cooling temperatures as inputs in mathematical model, the simulations results were used to study the temperature profiles along the heat exchanger with higher reliability. In addition, the effects of collector mass and convection heat transfer coefficient were analyzed by simulations not considered by other researchers until now.

2. Experimental test procedure

Experimental tests were performed by small gas liquefier and nitrogen gas as working fluid. Pre-cooling was performed by coiled tube immersed in liquid nitrogen bath. In order to avoid liquefying high-pressure nitrogen gas within the bath, liquid nitrogen level was adjusted at special point. The helically coiled tube in tube heat exchanger made of stainless steel 304 L and expansion valve was placed in two-layer stainless steel coldbox with 10^{-9} bar vacuum conditions. The vacuum conditions within the coldbox were established by a Woosung rotary vacuum pump (5 m^3 h^{-1}) and a DP-100 diffusion vacuum pump (250 l s^{-1}) in series. The temperatures were measured by Pt-100 sensors with accuracy of 0.1 K. The temperature sensors were installed on the tubes wall due to technical limitations; so, the temperatures measured by sensors have offset with respect to the actual temperature of gas flow. The details of experimental Joule-Thomson system are presented in Table 1.

Table 1
The details of experimental Joule-Thomson cooling system.

Parameters	Values
Working fluid	Nitrogen
Working pressure (bar)	20
Mass flow rate (kg h^{-1})	1
Inner tube internal diameter (mm)	1.671
Outer tube internal diameter (mm)	8.001
Tubes wall (mm)	0.762
Tube length (m)	4
Tube wall thermal conductivity (W m^{-1} K^{-1})	50

3. Modeling procedure

Finite-Element Method (FEM) was used to simulate the counter current helically coiled tube in tube heat exchanger. Forward, central, and backward forms of FEM were used to discretize the energy equations in warm fluid, tubes wall, and cold fluid, respectively. MATLAB m-file programming was employed to solve the FEM forms of energy equations by Gauss-Jordan method. The properties of gas at various temperatures were collected from references and added to a separate function m-file of MATLAB software. This function m-file was used in the main code. The assumptions applied to simulate the problem were as follows:

- The radial distribution of temperature was neglected in gas flows and tubes wall.

- Constant tube wall thermal conductivity was applied along the tube.
- Constant convection heat transfer coefficients were assumed for two fluids.
- Conduction and convection heat in-leak terms were neglected (high vacuum conditions).
- Pressure drops along the tubes were measured in gas liquefier and set to zero in simulation.
- The low-pressure gas temperature at collector outlet was set to collector temperature in equation (5).

The model was solved by direct use of heat capacities, radiation heat transfer into cold fluid flowing within the annular section, and longitudinal heat conduction through separating and external walls. The energy equations were established in five sections (warm fluid, cold fluid, separating wall, and external wall) as follows:

$$\rho_1 A_1 c_{p1}\frac{dT_1}{dt} = -m_1 c_{p1}\frac{dT_1}{dz} - h_1(2\pi r_1)(T_1 - T_2) \tag{1}$$

$$\rho_2 A_2 c_{p2}\frac{dT_2}{dt} = k_2 A_2\frac{d^2T_2}{dz^2} + h_1(2\pi r_1)(T_1 - T_2) - h_3(2\pi r_2)(T_2 - T_3) \tag{2}$$

$$\rho_3 A_3 c_{p3}\frac{dT_3}{dt} = m_3 c_{p3}\frac{dT_3}{dz} + h_3(2\pi r_2)(T_2 - T_3) + h_3(2\pi r_3)(T_4 - T_3) \tag{3}$$

$$\rho_4 A_4 c_{p4}\frac{dT_4}{dt} = k_4 A_4\frac{d^2T_4}{dz^2} - h_3(2\pi r_3)(T_4 - T_3) + Q_l(2\pi r_4) \tag{4}$$

$$m_c c_{pc}\frac{dT_c}{dt} = m_1 c_{p3} T_{J-T} - m_3 c_{p3} T_c \tag{5}$$

where subscript "c" indicates the collector properties. The value of T_{J-T} was calculated as follows:

$$T_{J-T} = f[T_1(z = l)] \tag{6}$$

where $f[T_1(z = l)]$ is a function of warm gas outlet temperature (upstream of expansion valve) to apply the Joule-Thomson effect in calculations. The term $f[T_1(z = l)]$ was coded by a separate function m-file and used in the main program. Q_l was defined as heat in-leak term by radiation heat transfer mechanism as follows:

$$Q_l = \epsilon\,\sigma(T_a^4 - T_4^4) \tag{7}$$

The boundary conditions are as follows:

$T_1(z = 0) = T_{Pre-cooling}$

$T_2(z = 0) = T_{Pre-cooling}$
and
$T_2(z = l) = f[T_1(z = l)]$

$T_3(z = l) = f[T_1(z = l)]$

$T_4(z = 0) = T_{Pre-cooling}$
and
$T_4(z = l) = f[T_1(z = l)]$

In order to estimate convection heat transfer coefficient within the helically coiled tube, the correlations proposed by Xin and Ebadian were used as follows [14]:

$$\mathrm{Nu_{ave}} = (2.153 + 0.318\mathrm{De}^{0.643})\mathrm{Pr}^{0.177} \tag{8}$$

$$20 < \mathrm{De} < 2000, \quad 0.7 < \mathrm{Pr} < 175, \quad 0.0267 < \frac{\mathrm{d}}{\mathrm{D_{coil}}} < 0.0884$$

$$\mathrm{Nu_{ave}} = 0.00619\mathrm{Re}^{0.92}\mathrm{Pr}^{0.4}\left(1 + \frac{3.455\mathrm{d}}{\mathrm{D_{coil}}}\right) \tag{9}$$

$$5 \times 10^3 < \mathrm{Re} < 10^5, \quad 0.7 < \mathrm{Pr} < 5, \quad 0.0267 < \frac{\mathrm{d}}{\mathrm{D_{coil}}} < 0.0884$$

Equation (8) and equation (9) were used in high-pressure gas and low-pressure gas respectively. Also, the program was able to select appropriate equation for the flows according to the flow regime during the computational run.

4. Results and discussion

4.1. Experimental based study

Usually, pre-cooling is the first step of gas liquefaction processes. Liquid nitrogen is commonly used for pre-cooling the gases due to low cost and safety aspects. The knowledge of evaluating the transient thermal behavior of gas in pre-cooling step can lead to predicting the gas temperature in the next step, accurately. Figure 2 shows the high-pressure gas temperature after pre-cooling by liquid nitrogen. As can be seen, the gas temperature decreases from room temperature to about -130 ℃ in two steps. In the first step, the temperature decreases with slope of -6 ℃ min^{-1} and in the second step, the slope of temperature decreases to -0.5 ℃ min^{-1}. The first step of temperature decrease is due to the direct cooling effect of liquid nitrogen on high-pressure nitrogen gas flowing within the helically coiled tube partially immersed in liquid nitrogen. The high-pressure gas after expansion in Joule-Thomson valve returns and cools the incoming gas before liquid nitrogen bath (see figure 1). This method is commonly used in gas liquefier to improve the efficiency. Therefore, the temperature of high-pressure gas incoming to liquid nitrogen bath decreases gradually, and accordingly, the high-pressure gas temperature leaving the liquid nitrogen bath decreases, as shown in figure 2 (the second step).

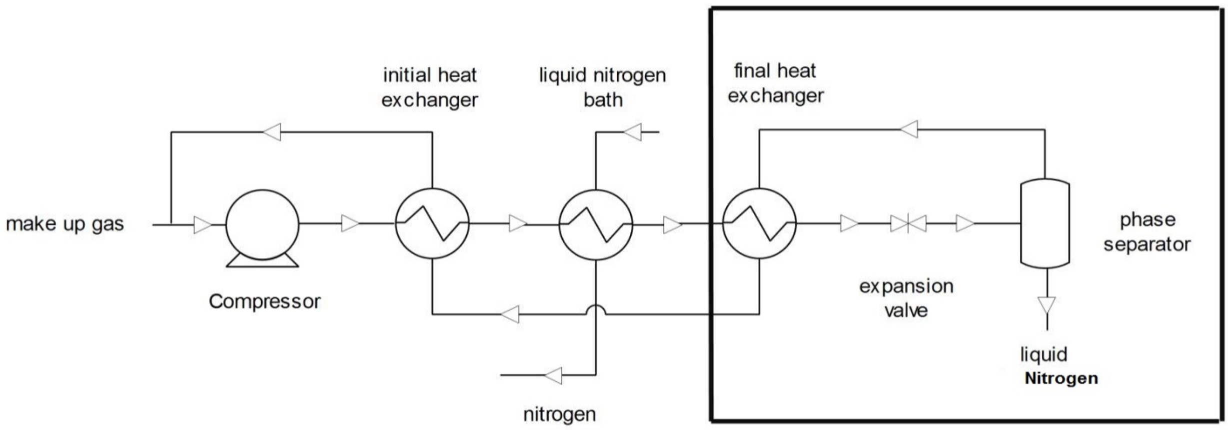

Figure 1. The PFD of small gas liquefier used in the present work. The Joule-Thomson cooling system is specified by bold line.

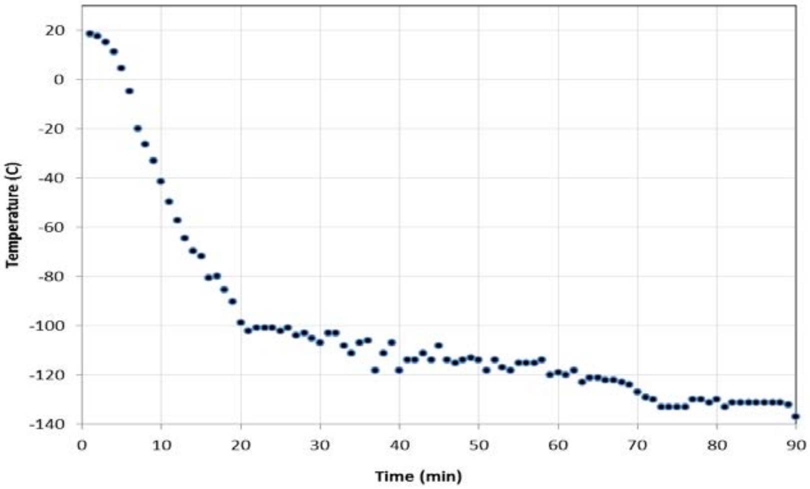

Figure 2. Temperature of nitrogen after pre-cooling by liquid nitrogen bath versus time.

These two steps usually take place in any liquefaction processes with pre-cooling section by the second refrigerant. So, this special form of the temperature decrease must be considered in any mathematical simulations.

Distance between pre-cooling section (liquid nitrogen bath) and final heat exchanger can influence the temperature of gas entering to the final heat exchanger. Figure 3 shows the high-pressure gas temperature at inlet and outlet of the final heat exchanger. As seen, the gas temperatures at the heat exchanger inlet are different from those for the pre-cooling section. The distance between the two sections led to about 60 °C temperature raise at the heat exchanger inlet. This temperature increase can be attributed to initial cooling of the final section of liquefier including heat exchanger, expansion valve, and collector. The high-pressure gas temperature at heat exchanger outlet is also shown in figure 3. As can be seen, the rate of temperature decrease is low for initial times, increases gradually, and equals to the rate of the temperature decrease at heat exchanger inlet, finally.

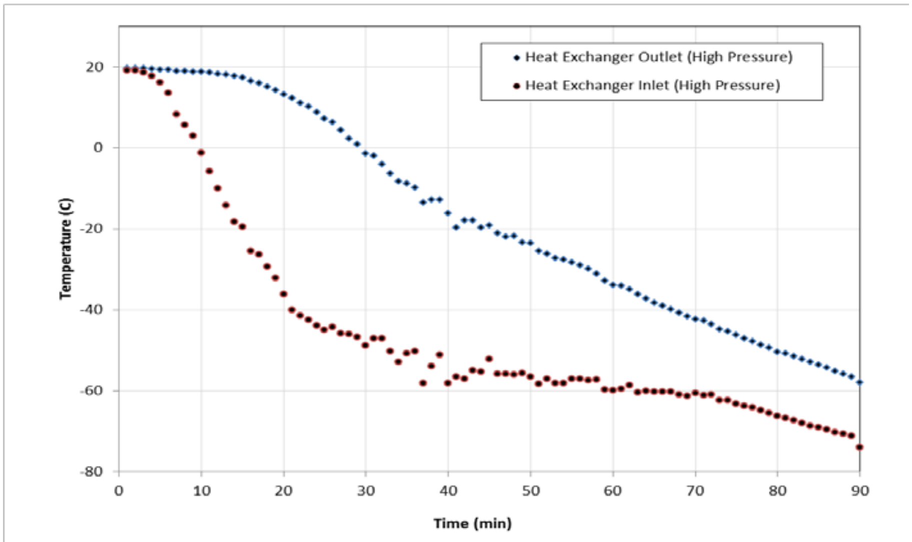

Figure 3. The high-pressure gas temperature at inlet and outlet of the final heat exchanger (obtained from experimental test).

Figure 4 shows the gas temperature at points in upstream and downstream of Joule-Thomson valve. In order to study the effect of collector heat capacity, upstream sensor is installed near the collector. As can be seen, downstream temperatures are higher than the upstream temperatures up to 70 min. In this time step, the cooling effect of Joule-Thomson valve does not appear due to initial temperature of collector and its heat capacity. Therefore, backward stream (expanded gas) cannot cool the incoming gas (high-pressure gas). At times longer than 70 min, the Joule-Thomson effect appears and the gas temperatures in backward stream become lower than the incoming gas temperatures.

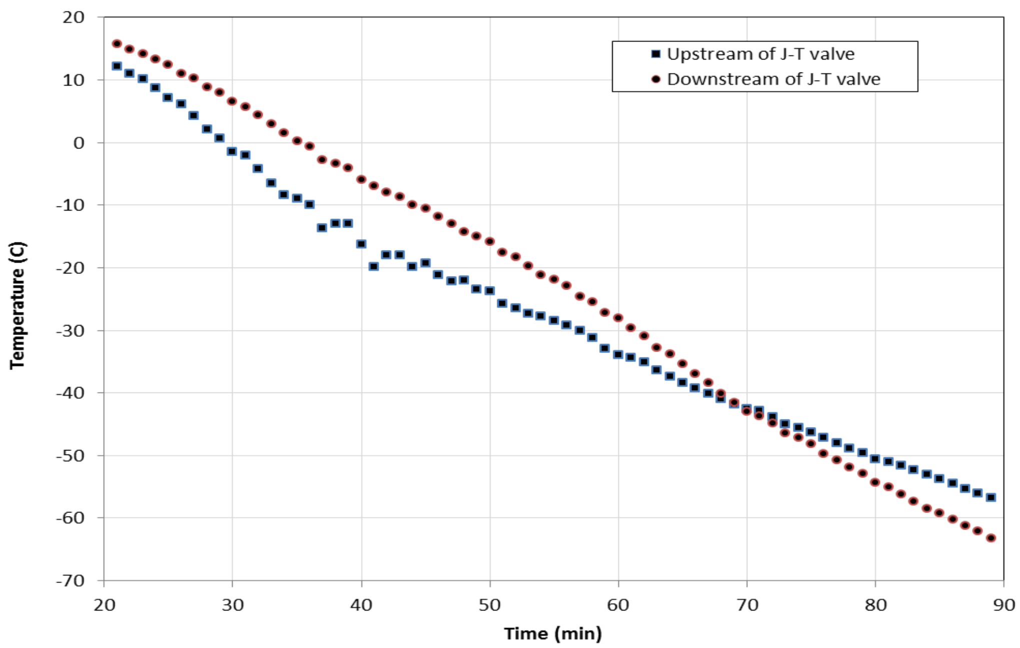

Figure 4. The gas temperature at points located in upstream and downstream of Joule-Thomson valve.

The final recuperative heat exchanger operating in gas liquefier is used for cooling the incoming high-pressure gas by exchanging heat to cooled low-pressure gas coming from backward stream. Therefore, the backward low-pressure stream must be cooler than the incoming high-pressure gas. This event normally occurs in steady-state conditions, but the conditions are different in unsteady state. Figure 5 shows the high-pressure gas temperature at heat exchanger inlet and low-pressure gas temperature at heat exchanger outlet versus time. As seen, the incoming high-pressure gas temperature is lower than the backward low-pressure gas temperature up to 70 min. This is due to initial cooling of heat exchanger, Joule-Thomson valve, and collector. In the initial cooling time step, the high-pressure gas with low temperature enters the heat exchanger and exchanges energy with the low-pressure gas leaving the heat exchanger. At times longer than 70 min, the high-pressure gas temperature at heat exchanger inlet becomes equal to the low-pressure gas at heat exchanger outlet. The complementary discussions will be presented in the next section using simulations.

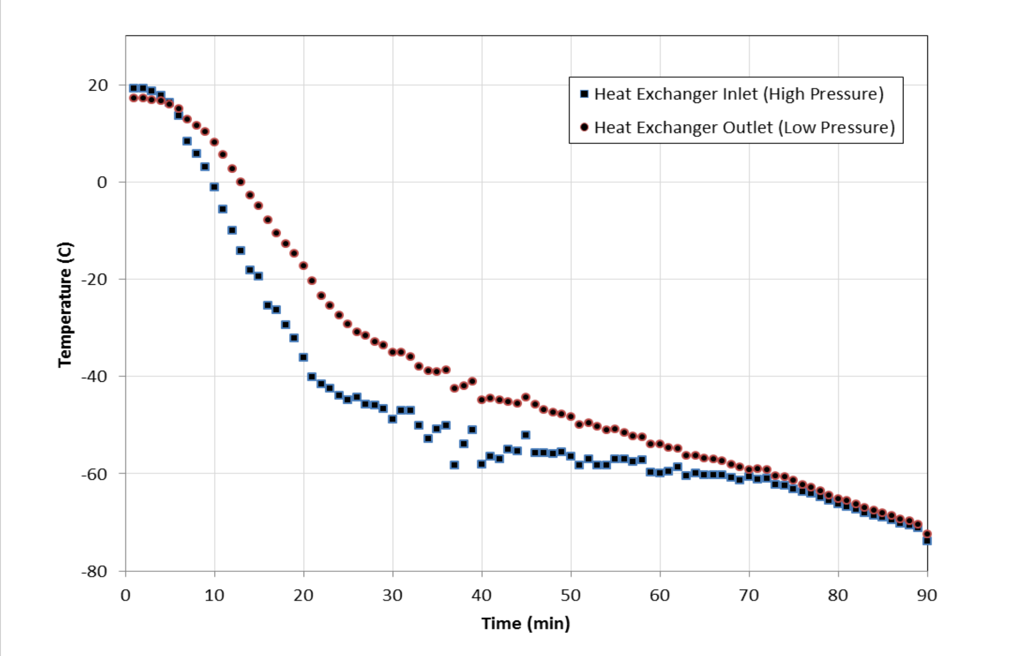

Figure 5. The high pressure gas temperature at heat exchanger inlet and the low-pressure gas temperature at heat exchanger outlet (backward stream) versus time.

4.2. Simulation based study

Figure 6 shows the high-pressure gas temperature at inlet and outlet of the final heat exchanger in which the gas temperatures in experimental tests were used as boundary condition at heat exchanger inlet, and the gas temperatures at heat exchanger outlet were obtained by simulation. As can be seen, simulation results have good agreement with respect to experimental data. This means that simulation using the current model and the applied assumptions can be used as useful prediction tools to evaluate the gas temperature profile along the heat exchanger.

To compare the current model with regular models, the simulations were also performed using constant pre-cooling temperature and without considering collector mass. Figure 7 shows the high-pressure gas temperature at heat exchanger inlet and outlet versus time with two different assumptions. As can be seen, using constant high-pressure gas temperature (equal to -80 °C) at heat exchanger inlet leads to a decrease in temperature level of high-pressure gas at heat exchanger outlet with respect to the experiment. Neglecting the collector mass in simulations leads to a further decrease in the temperature level. Table 2 presents the values of high-pressure gas temperatures at heat exchanger outlet for the special assumption considered in simulations. Relative errors for case 1 (simulation using actual temperatures as input and considering collector mass) are similar for both 60 and 90 min. These constant relative errors can be attributed to the position of temperature sensor and installing the sensor on tube walls. Similar discussion was reported by Saberimoghaddam and bahri [15]. Using constant high-pressure gas temperature at heat exchanger inlet (case 2) leads to higher relative errors for two time steps. For the time equal to 60 min, the relative error is considerable due to the high-temperature difference between the considered high-pressure gas temperature at heat exchanger inlet and the actual values obtained from experiment. At time 90 min, the relative error decreases due to the decrease in temperature difference between the considered high-pressure gas temperature at heat exchanger inlet and the actual values obtained from experiment. By neglecting the collector mass (case 3), the relative errors increase for both time steps. Although the collector mass does not influence cooling systems in steady-state conditions, neglecting it leads to about 60 and 25 % errors for times equal to 60 and 90 min, respectively. Therefore, the model presented here can be used for evaluating the thermal behavior of recuperative tube in tube heat exchanger operating in Joule-Thomson cooling systems efficiently.

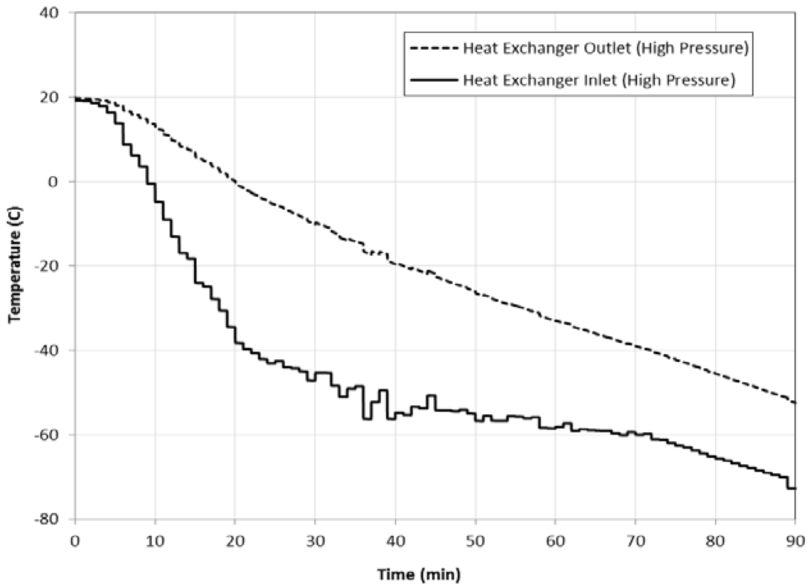

Figure 6. The high-pressure gas temperature at inlet and outlet of the final heat exchanger (obtained from simulation).

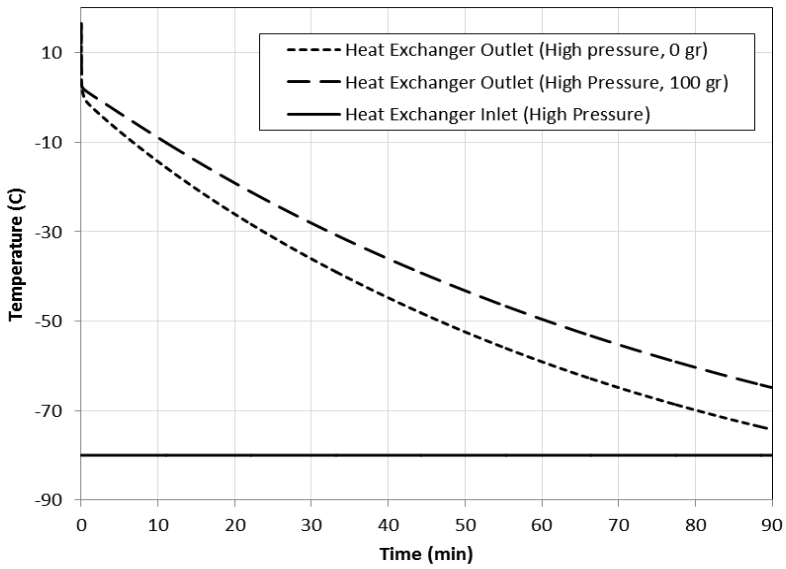

Figure 7. The high-pressure gas temperature at inlet and outlet of the final heat exchanger (obtained by simulation with assumptions: constant high-pressure gas temperature at heat exchanger inlet= -80 ℃, collector mass = 0 gr/ 100 gr).

Table 2
The comparison of simulations with different assumptions.

	Experimental test	Case 1: Simulation using actual temperatures as input and considering collector mass (current paper)	Case 2: Simulation using constant precooling temperature and considering collector mass	Case 3: Simulation using constant precooling temperature and without considering collector mass
High pressure gas temperature at 60 min (°C)	-36	-32	-50	-58
Relative error (%)	-	11.11	38.88	61.11
High pressure gas temperature at 90 min (°C)	-58	-52	-65	-72
Relative error (%)	-	10.34	12.06	24.13

If the convection heat transfer coefficients for two sides of heat exchanger are sufficiently high, exchanging energy between two streams will take place in the zone near the heat exchanger inlet. If the convection heat transfer coefficients are not sufficiently high, exchanging energy will not take place in the zone near the heat exchanger inlet, and the high-pressure gas reaches heat exchanger outlet with lower temperature. The lower temperature at heat exchanger outlet for the high-pressure gas results in higher Joule-Thomson effect and lower temperature at expansion valve outlet. Therefore, thermal behavior of a liquefier is different based on internal conditions of recuperative heat exchanger. The effect of various convection heat transfer coefficients will be considered in the following section. Modifying the recuperative heat exchanger after manufacturing and installing step is impossible due technical limitations.

Due to the limitation to installing temperature sensors within the tube, mathematical simulation was performed to determine temperature profile along the heat exchanger. For this reason, the high-pressure gas temperatures at heat exchanger inlet obtained from experimental test were used as boundary condition in the mathematical simulation. Figure 8 shows the high-pressure and low-pressure gas temperature profiles at time steps equal to 10 min, 30 min, 60 min, and 90 min (A to D). At the initial time step, the low-pressure gas temperatures are near the ambient temperature except for small zones at heat exchanger cold and warm end. As can be seen, a temperature drop in the low-pressure gas occurs at the heat exchanger warm end. At the heat exchanger warm end, the high-pressure gas expands and its temperature decreases due to Joule-Thomson effect. The Joule-Thomson effect on the expanded gas results in decreasing the collector temperature. Therefore, the low-pressure gas temperature rises and reaches to high-pressure gas inlet temperature at heat exchanger inlet (length = 4 m) due to energy exchange with collector. The low-pressure gas temperature rises further from length 4 m to 3 m due to initial temperature of heat exchanger materials and drops down gradually from length 3 m to 0.2 m. The high temperature drop in low-pressure gas takes place near the heat exchanger cold end due to proximity of this region to pre-cooling section. The tube wall near the heat exchanger cold end has lower temperature with respect to other sections of tube length. At time equal to 30 min, the level of temperature throughout the heat exchanger decreases for high-pressure and low-pressure gases within the tube due to heat exchanger cooled materials in the previous times. The cooling of heat exchanger materials starts from two sides (cold end and warm end), and the middle zone of the heat exchanger has higher temperature with respect to other zones. This event can be seen in the low-pressure gas temperature profiles. At time equal to 60 min, the collector materials cool sufficiently and the Joule-Thomson effect appears in the low-pressure gas temperature. Therefore, the low-pressure gas temperature becomes lower than high-pressure gas temperature at heat exchanger warm end. This phenomenon also takes place at time equal to 90 min with higher Joule-Thomson effect at lower temperature.

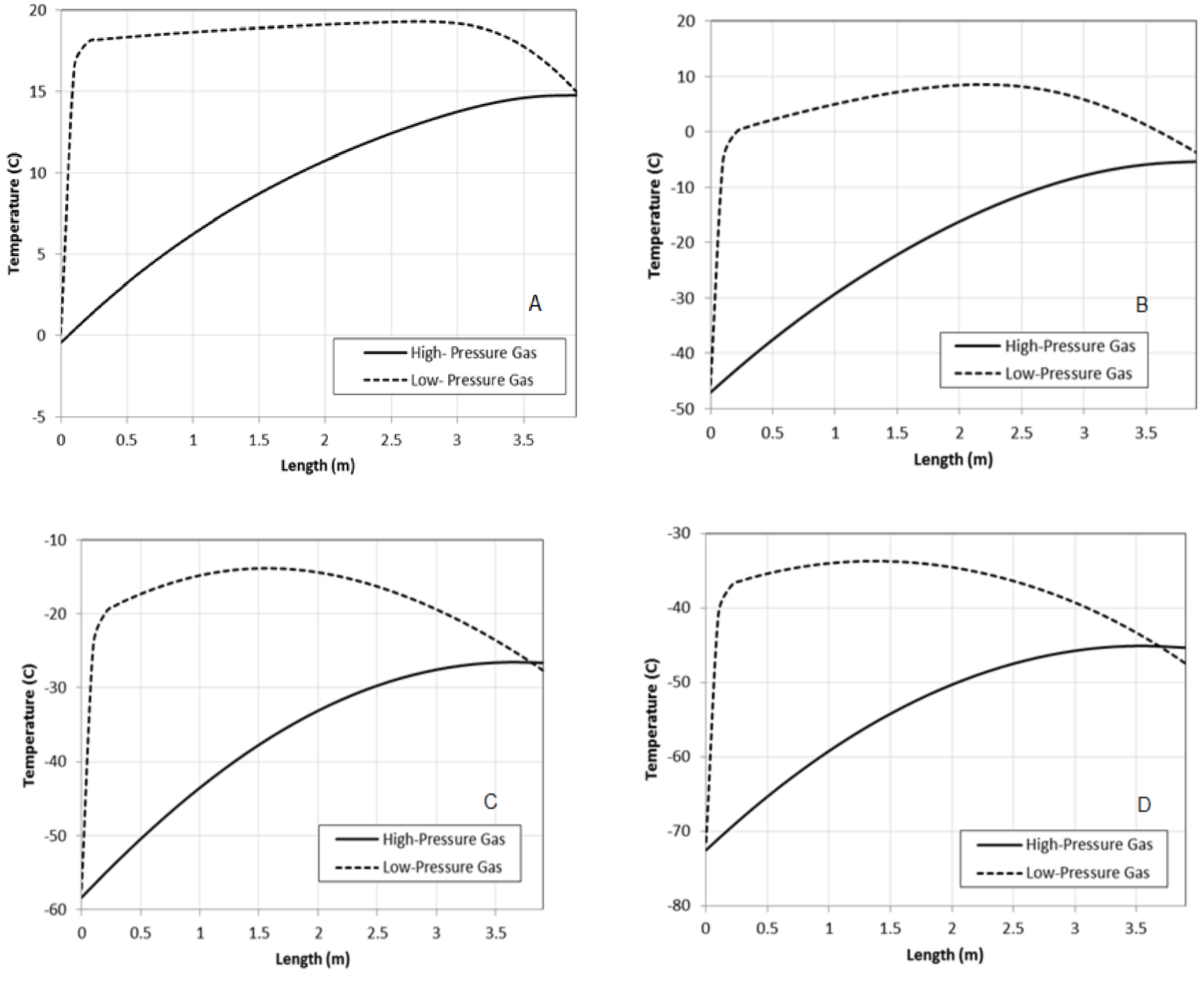

Figure 8. The high-pressure and low-pressure gas temperature profiles along the heat exchanger at various times A) 10 min. B) 30 min. C) 60 min. D) 90 min.

Mass heat capacity of collector is another important parameter in transient thermal behavior of a gas liquefier. An effective mass for any collector is engaged in the cooling process. The value of collector effective mass can influence transient time (the time needed for cooling the system from ambient temperature to steady state) and performance of system. Figure 9 shows the high-pressure and low-pressure gas temperatures for various collector masses at time equal to 90 min. As can be seen, the Joule-Thomson effect (temperature cross near the heat exchanger warm end) disappears in the cases in which the collector's effective mass is higher than 100 gr. Moreover, the overall temperature level increases for both the high-pressure and low-pressure gas profiles. This means that the material heat capacity associated with expansion valve and collector can greatly increase the cool-down process time. Neglecting the collector mass in mathematical simulations leads to considerable errors and unreal results in prediction of transient thermal behavior of cool-down process.

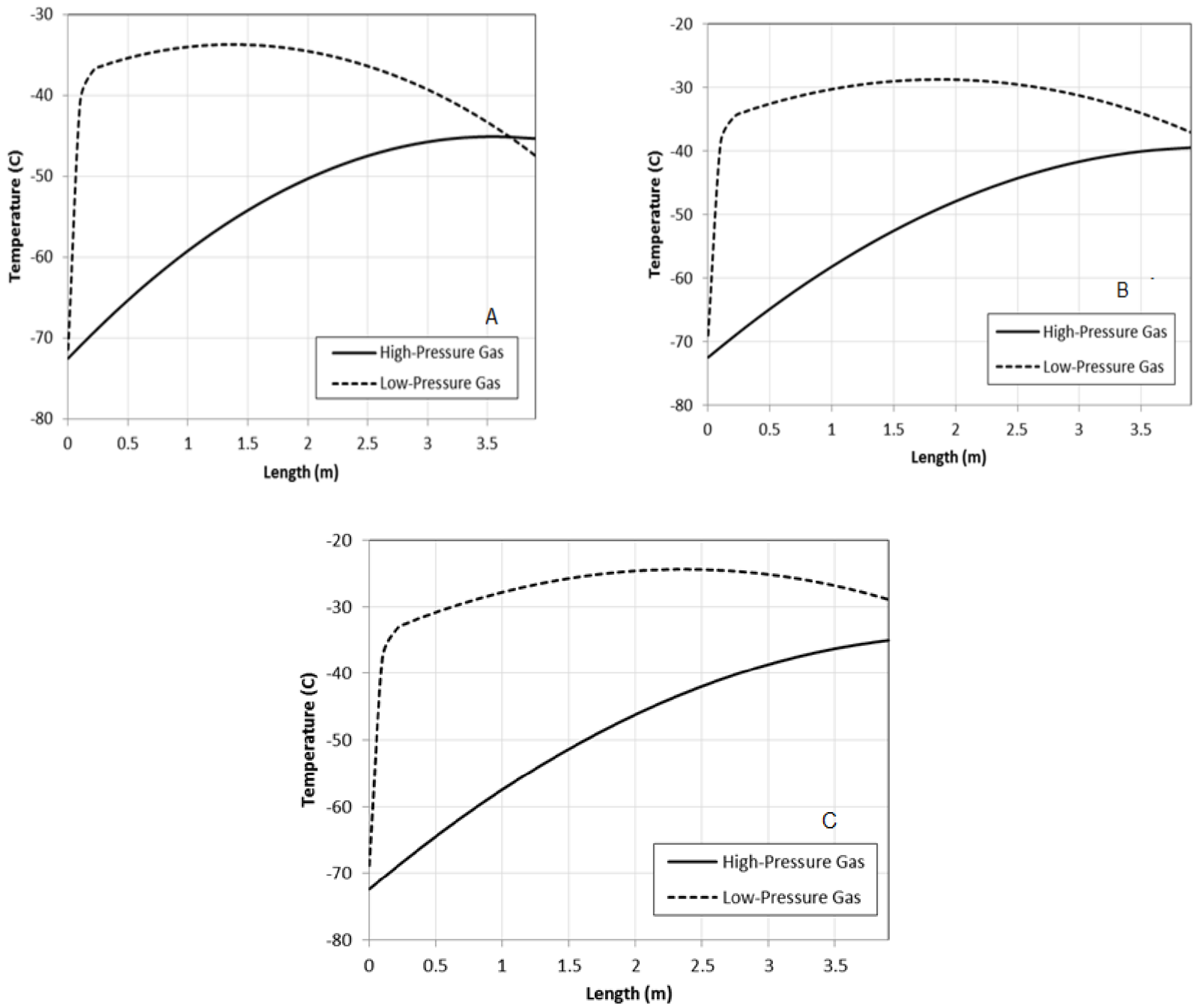

Figure 9. The high-pressure and low-pressure gas temperature profiles along the heat exchanger at time equal to 90 min for various effective masses of collector A) 100 gr B) 200 gr C) 300 gr.

Convection heat transfer coefficients in high-pressure and low-pressure gases are important parameters that can be considered for recuperative heat exchanger in the cool-down process. Although the gas mass flow rate determines the convection heat transfer coefficient, decreasing or increasing the convection heat transfer coefficient is possible by varying the cross section area, flow turbulence, etc. for any mass flow rate. In addition, using finned tube in heat exchanger can directly influence the flow regime. Therefore, the simulations were carried out with different convection heat transfer coefficients. The results showed that the convection heat transfer coefficient in high-pressure gas did not influence temperature profiles and cool-down process times. In the simulations, the convection heat transfer coefficient was increased up to 300 percent and no considerable changes were observed. Figure 10 shows the high-pressure and low-pressure gas temperature profiles along the heat exchanger at time equal to 90 min for various low-pressure convection heat transfer coefficients.

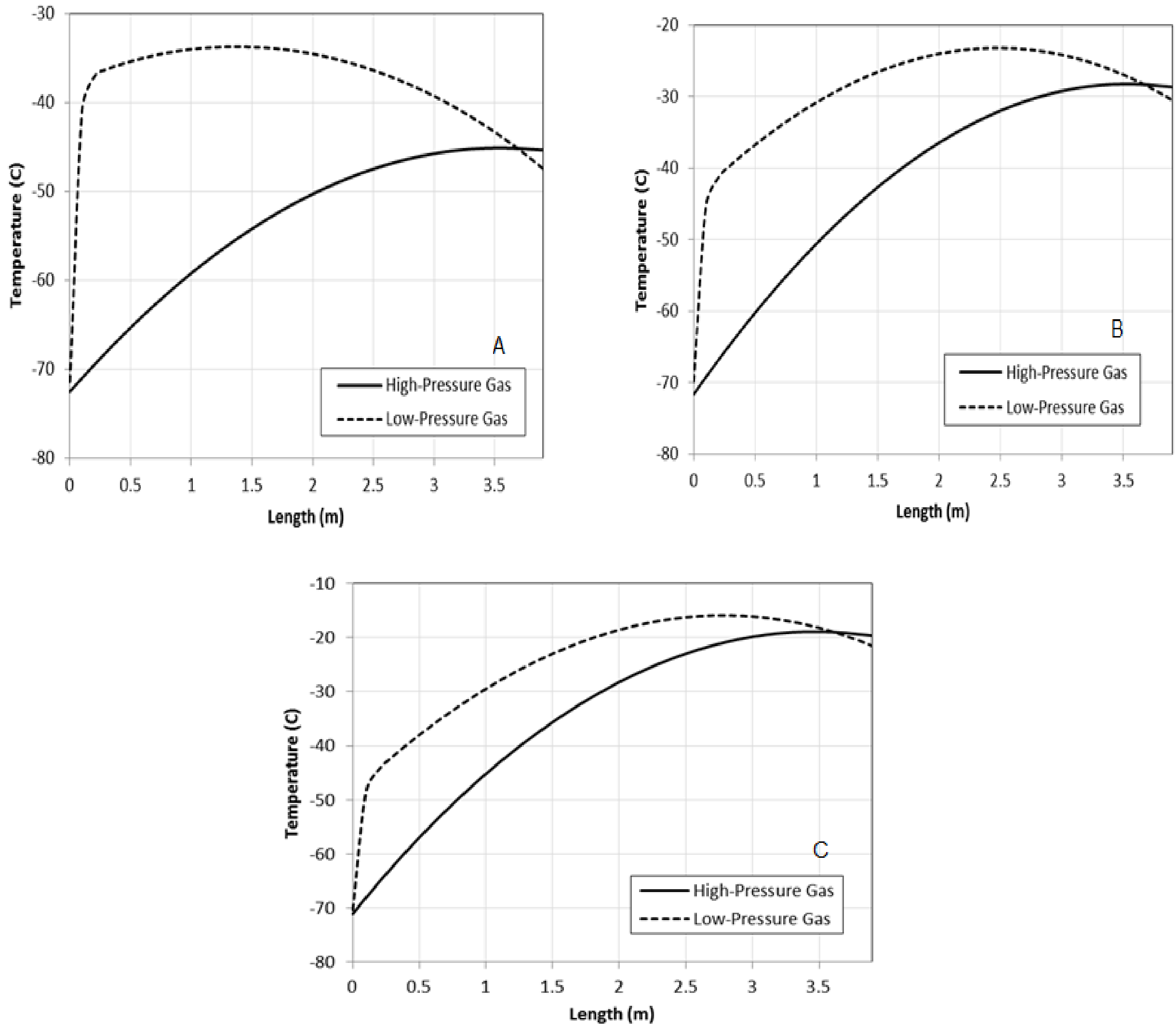

Figure 10. The high-pressure and low-pressure gas temperature profiles along the heat exchanger at time equal to 90 min for various low-pressure convection heat transfer coefficients: A) Normal conditions (10 J m^{-2} K^{-1}); B) 100 % increase; C) 200 % increase.

As can be seen, increasing the convection heat transfer coefficient in low-pressure stream leads to the decrease in low-pressure gas temperature in the zones near the cold end and the increase in low-pressure gas temperature in the zones near the warm end. Moreover, the high-pressure gas leaves the heat exchanger at warm end with higher temperatures. Therefore, increasing the convection heat transfer coefficient in low-pressure gas leads to the increase in exchanging energy between two streams and the faster cooling of the heat exchanger materials. In this situation, temperature difference between two streams and entropy generation decreases, and the initial cooling of heat exchanger materials is performed by the pre-cooled high-pressure gas instead of Joule-Thomson effect in expanded gas. As a result, initial cooling of heat exchanger materials can be performed by opening the expansion and lower operational pressures.

5. Conclusions

According to the experimental test and simulation results, the most important conclusions are as follows:

- Using liquid nitrogen as a pre-coolant cannot supply constant pre-cooling temperature at the next heat exchanger inlet.
- Initial cooling of Joule-Thomson system, including recuperative heat exchanger, expansion valve, and collector, was substantially performed by pre-cooled high-pressure gas with respect to Joule-Thomson effect. Therefore, the expansion valve can be fully opened in this period.
- The collector mass influences the cool-down process time greatly and must be considered in simulation to obtain real results.
- The higher convection heat transfer coefficient in low-pressure gas leads to the increase in exchanging energy between two streams and the faster cooling of heat exchanger materials, but the higher convection heat transfer coefficient in high-pressure gas does not influence cool-down process.

Acknowledgements

This research was supported by Malek Ashtar University of Technology, Department of Chemistry and Chemical Engineering. We are thankful to our colleagues in cryogenic laboratory such as Dr. Vahid Khebri, Dr. Ali Nozari, Eng. Asadollah Baghervand, and Mahdi Khatibi who provided expertise that greatly assisted the research. We are also immensely grateful to Dr. Mohammad Reza Rezaei and Dr. Hadi Rezaei Vahidian for their comments on an earlier version of the manuscript, although any errors are our own and should not tarnish the reputations of these esteemed professionals.

Nomenclature

A	cross-section area [m^2].
c_p	specific heat [J kg^{-1} K^{-1}].
D	coil diameter [m].
d	tube diameter [m].
De	Dean number.
h	convection heat transfer coefficient [W m^{-2} K].
k	thermal conductivity [W m^{-2} K].
l	length [m].
m	mass [kg].
Nu	Nusselt number.
Pr	Prandtl number.
Q_l	heat in-leak [W m^{-2}].
r_1	internal radius of inner tube [m].
r_2	external radius of inner tube [m].
r_3	internal radius of outer tube [m].
r_4	external radius of outer tube [m].
Re	Reynolds number.
T	temperature [K].
z	length [m].

Greek symbols

$\in$	emissivity.
σ	Stefan–Boltzmann constant [W m^{-2} K^{-4}].
ρ	density [kg m^{-3}].

Subscripts

1	warm fluid.
2	inner tube wall.
3	cold fluid.
4	outer tube wall.
a	ambient.
c	collector.
l	leakage.
coil	coil.
ave	average.

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