Parasitic Effect of Tube Wall Longitudinal Heat Conduction on Cryogenic Gas Temperature

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ARTICLE INFO	ABSTRACT
	Longitudinal heat conduction is an important parameter in the cryogenic science, especially in cryogenic heat exchangers. In the present work, the parasitic effect of tube wall longitudinal heat conduction on temperature measurement has been studied in cryogenic laminar hydrogen flow. The effects of various parameters such as wall cold end temperature, wall thermal conductivity, gas volumetric flow, and tube wall thickness have been investigated by finite element method. The model was also validated versus the data obtained from experiments. The simulations showed that temperature drop occurs in gas flow at the end section of tube length. This section is independent of tube cold end temperature and leads to large temperature measurement error in laminar flows. Results showed that a few millimeters change in temperature sensor position results in measurement errors up to 80 %. The higher tube wall thermal conductivity and tube wall thickness result in higher parasitic effects of longitudinal heat conduction.

1. Introduction

Heat transfer and its related fields are the main subjects of cryogenic science that have attracted several researchers to focus on identifying and solving the obstacles involved. All the cryogenic processes operate at very low temperatures and many traditional principles are different in such conditions. In MUT cryogenic laboratory, an attempt was made to obtain a correlation for convection heat transfer coefficient in hydrogen laminar flow at cryogenic temperatures. Therefore, a helical coiled tube was immersed in liquid nitrogen to establish constant tube wall temperature. Hydrogen flow controlled by a mass flow controller was entered in immersed tube. It was expected that the gas flow temperature at tube inlet was near ambient temperature, but the PT-100 sensor showed different temperatures. In addition, the sensor showed different values when the distance

between sensor and liquid nitrogen surface varied a few millimeters. Despite all efforts performed to measure the correct temperature and insulation of external tube wall, the gas temperature at tube inlet (immersed coil) was in the range of cryogenic temperatures. This phenomenon disturbed the achievement of correct results. So, several tests were done in order to identify the problem. Finally, it was concluded that the problem was due to wall longitudinal heat conduction. Unfortunately, in spite of extensive study performed on the effects of wall longitudinal heat conduction at cryogenic temperatures, there is no attention to its parasitic effects in temperature measurement at experimental work scales. In the following, some research works performed by researchers have been presented to illustrate the importance of wall longitudinal heat conduction.

Pacio and Dorao [1] published a paper and reviewed the thermal hydraulic models of cryogenic heat exchangers. They also introduced physical effects such as changes in fluid properties, flow maldistribution, axial longitudinal heat conduction, and heat leakage as the main challenges of cryogenic heat exchangers. Aminuddin and Zubair [2] studied the 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 conduction from heat exchanger cold end to the adjacent components, but they did not perform any experimental tests. Krishna et al. [3] presented a paper about the effect of longitudinal heat conduction in the separating walls on the performance of three-fluid cryogenic heat exchanger with three thermal
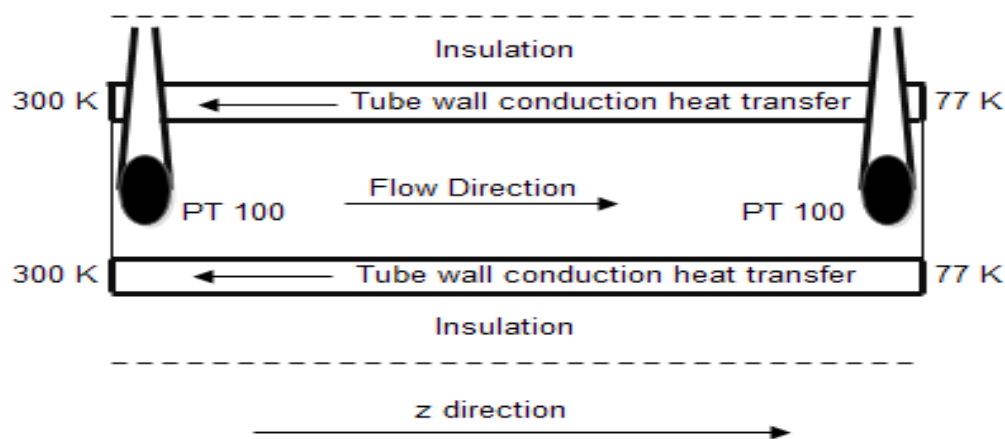
communications. They concluded that the thermal performance of heat exchangers used in cryogenic temperature, is strongly governed by various losses such as longitudinal heat conduction through the wall, heat-in-leak from the surroundings, flow maldistribution, etc. Gupta et al. [4] investigated the Second law analysis of counter flow cryogenic heat exchangers in 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 [5] presented a numerical model of heat exchanger in which the effect of axial conduction, property variations, and parasitic heat losses to the environment have been explicitly modeled. He concluded the small degradation effect in the performance of heat exchanger at conditions in which the temperature of heat exchanger cold end is equal to temperature of the input cold fluid. Narayanan and Venkatarathnam [6] 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 heat exchangers with heat leak at the cold end with respect to heat exchangers with insulated ends. Ranganayakulu et al. [7] 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 when the longitudinal heat conduction parameter is large. It is clear that longitudinal heat conduction has great importance in the cryogenic engineering, but almost all the research has been focused on the cryogenic heat exchangers. In the present paper, an attempt was made to study the longitudinal heat conduction from the new approach. Sometimes, as mentioned previously, evaluating the exact temperature of fluid is important. For example, in order to estimate the effectiveness of an experimental heat exchanger, the exact temperature of input and output must be specified. In the case of cryogenic heat exchanger in which temperature approaches are small, the errors associated with temperature measurement are not acceptable. These errors can be due to incorrect positions of temperature sensors. In the cryogenic operations, the temperature of fluid between two distinct points of tube with a distance of one or two centimeters can vary several degrees while no heat transfer occurs with secondary fluid or ambient temperature. This temperature difference is due to tube wall longitudinal heat conduction. This temperature difference is high when one end of the tube is at the cryogenic temperature and the other end of the tube is at the ambient temperature. Usually, the gas enters the coil with temperature lower than that measured by the sensor, because the sensor is not placed in correct and accurate position. This temperature difference in the laminar flow regime is not negligible. In the present study, the influence of longitudinal heat conduction through the wall on the temperature of laminar flow has been discussed. A finite element method was used to simulate the longitudinal heat conduction through the tube wall and the model was validated versus the data obtained from special experimental setup. Then, the model was used to evaluate the effects of different parameters.

2. Mathematical modeling

In order to study the problem, it was assumed that one end of a tube was at cryogenic temperature (77 K, normal boiling point of liquid nitrogen) and the other end was at different temperatures (higher than the other end of the tube). Fig. 1 shows the geometric model of tube in detail.

Figure 1. Scheme of model used for simulating the tube wall longitudinal heat conduction.

First, the governing energy equations were established in two sections of tube wall and gas flow within the tube as follows:

$$A_1 \frac{d^2 T_s}{dz^2} = Q - h_i(T_i - T_s), \quad A_1 = \frac{A_s K}{\pi D} \quad (1)$$

$$A_2 \frac{dT_i}{dz} = h_i(T_i - T_s), \quad A_2 = \frac{m_i C_i}{\pi D} \quad (2)$$

Where T_s and T_i are tube wall and gas flow temperatures respectively. h_i is convective heat transfer coefficient, K is tube wall thermal conductivity, D is tube diameter, mi is mass flow of gas, and As is tube wall cross section area. The term Q indicates the energy coming through insulation. This term was added to energy balance in order to fit model with experimental results. The boundary conditions are as follows:

$$T_s(z = 0) = 300 \text{ K} \quad \text{and} \quad T_s(z = l) = 77 \text{ K}$$

$$T_i(z = 0) = 300 \text{ K}$$

Finite Element Method (FEM) was used to solve the model and simulate the effect of tube wall longitudinal heat conduction on the temperature of gas within the tube. Central and forward difference forms of FEM were used to discretize the energy equations in tube wall and gas flow respectively. Matlab m-file programming was used to solve the FEM forms of energy equations by Gauss–Seidel iterative method. Entire length of tube was divided into 100 equal sections and energy balance equations were established on each section. The properties of gas (hydrogen) at various temperatures were collected from a paper published by McCary et al. [8]. These

properties were added to a separate function m-file of Matlab software and this function file was used in the main m-file. The radial distribution of temperature within the gas flow was neglected. Convective heat transfer coefficient between tube wall and gas flow was evaluated by the relation presented for laminar flow as follows [9]:
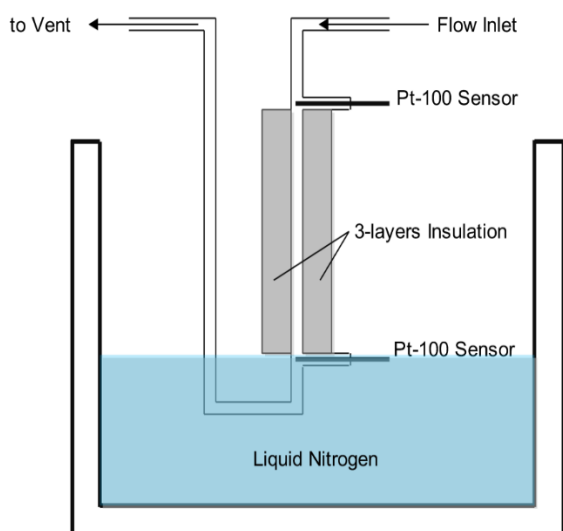
$$h_i = 3.66 \frac{K}{D}, \quad Re < 2300 \quad (3)$$

The model was validated versus data obtained from experiments at various mass flows. Experimental procedure of doing tests will be introduced in the next section.

3. Experimental procedure

Experimental tests were performed at various mass flows considering laminar regime within the tube in order to validate the mathematical model. Experimental setup consisted of a tube with three-layer insulation. The first, second, and third layers of insulation were made from styrofoam, polyethylene foam, and glass wool respectively. This three-layer insulation was used to minimize the energy transfer from the tube (hydrogen gas flow) to ambient. In the case of our study, external surface area of tube wall was in liquid nitrogen vapor, so in spite of three-layer insulation, it was expected that the gas flow cools while flowing through the tube. Accordingly, the term of Q was considered in the modeling of the tube. This temperature was determined by measuring the temperature of liquid nitrogen vapor at various distances from the surface of liquid nitrogen in the bath. Two PT-100 temperature sensors were installed at two tube ends. Gas flow was

controlled by ALICAT Mass Flow Controller (MFC). One end of tube was placed in liquid nitrogen and the tube inlet and outlet temperatures were recorded at various mass flows. Scheme of experimental setup has been shown in Fig. 2.

Figure 1. Scheme of experimental setup used for validating the mathematical simulation.

Hydrogen gas was used as working fluid within the tube. The MFC had a maximum capacity of one normal l/min and so experimental tests were performed in the range of 100 cc/min to 1000 cc/min as expected. A 3/8" stainless steel (304 L) tube was used in the experimental setup. The experimental data were used to validate the results obtained from modeling at conditions presented in Table 1 and due to technical limitations further tests were not performed at different conditions. In addition, only two temperature sensors (tube inlet and outlet) were used to avoid disturbance in the laminar flow regime, because mounting more PT-100 sensors along the tube results in gas flow turbulence.
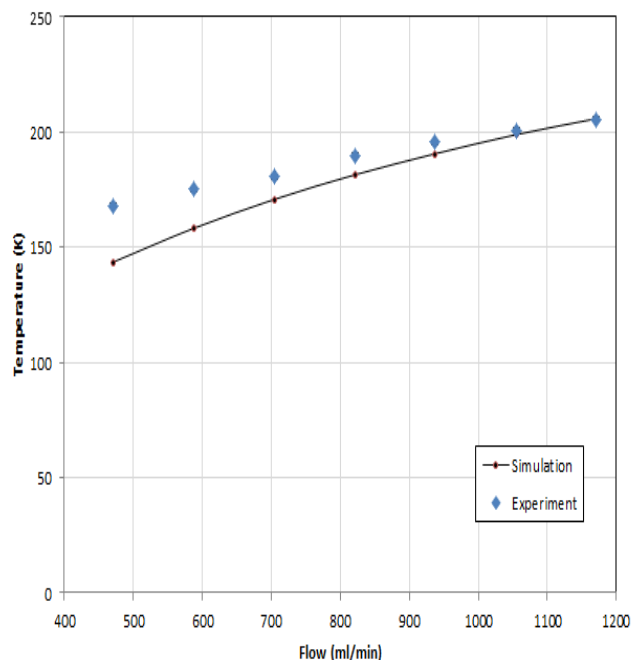
4. Results and discussion

Validating the mathematical model was the first step of our work and in the second step, the model was studied at various conditions with different parameters. Fig. 3 shows the results obtained from experimental tests and modeling.

Table 1
The parameters used for simulating the experimental setup.

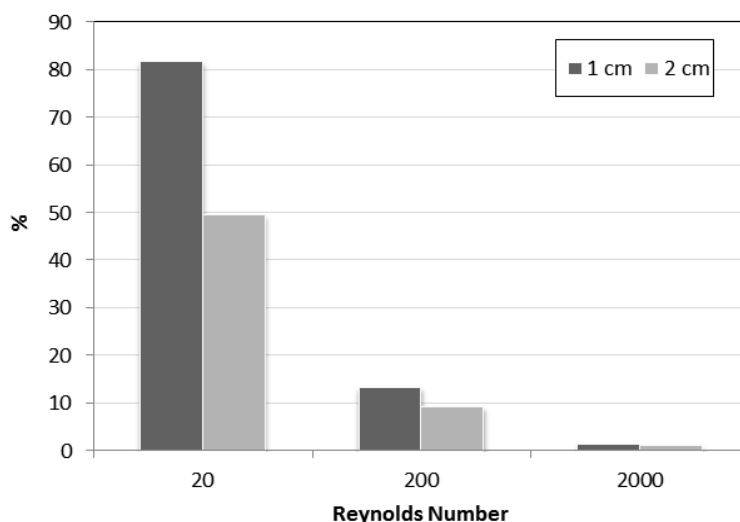| Parameters	Values
Tube length (m)	0.3
Tube diameter (m)	0.008001
Space interval (m)	0.003
Wall thermal conductivity (W/(m.K))	10
Tube wall diameter (m)	0.000762
Wall cold end temperature (K)	77
Wall warm end temperature (K)	300
Working fluid	Hydrogen

Figure 2. Tube outlet flow temperature under the effect of wall longitudinal heat conduction obtained from simulation and experimental tests.

As can be seen, the model presents a good estimation for tube outlet flow temperatures at relatively high volumetric flows. However, the model predicts the tube outlet flow temperatures with higher difference with respect to experimental results at lower volumetric flows. This difference can be attributed to the position of the sensor in the tube outlet. Fig. 4 shows the temperature difference of gas flow with respect to gas flow temperature at tube cold end versus various distances from tube cold end. These results were obtained from simulation. As seen, a distance of one centimeter from tube cold end causes 80 % temperature difference in gas flow with Reynolds number of 20. By increasing the Reynolds number, this temperature difference decreases and the effect of longitudinal heat conduction through the wall tube diminishes. Therefore, the differences shown in Fig. 3 are probably due to inaccurate position of temperature sensor that has been revealed at low Reynolds numbers. Consequently, at distances smaller than one centimeter, a few millimeters change

in position of temperature sensor causes large measurement errors. However, the trend of results obtained from simulations and experiments is similar and it can be concluded that the simulation results can be used to study different conditions.

Fig. 5 shows the effect of Reynolds number variations in the hydrogen gas temperature leaving the tube cold end. Fig. 5 shows the temperature profiles of the last 15 cm of tube end with warm and cold end wall temperature of 300 K and 77 K respectively. The term Q in the equation (1) was set zero for simulation. As seen, in the laminar flow regime, hydrogen gas temperature leaving the tube decreases with decrease in Reynolds number. It means that the gas temperature within the tube with a small Reynolds number is more influenced by wall temperature with respect to hydrogen flow with high Reynolds number. So, accurate position of the sensor in the gas flow with high Reynolds number is not a very important parameter.

Figure 4. Percentage of temperature difference with respect to the gas temperature at tube cold end (based on the results obtained from simulations).
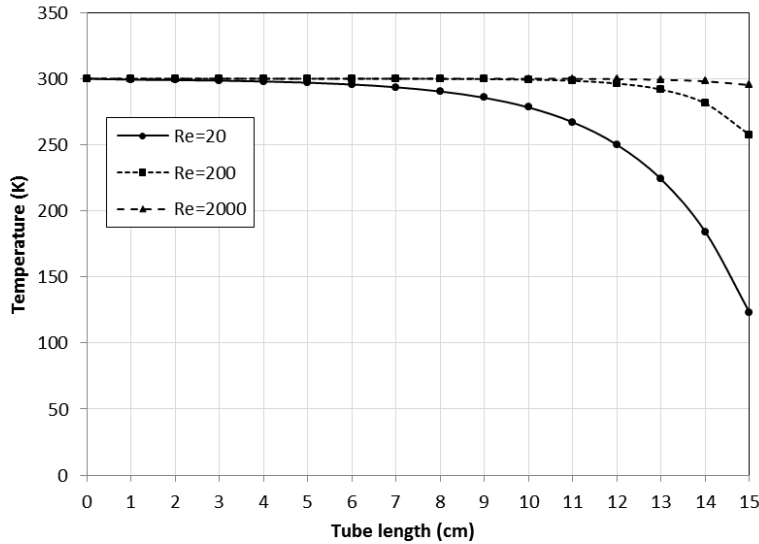
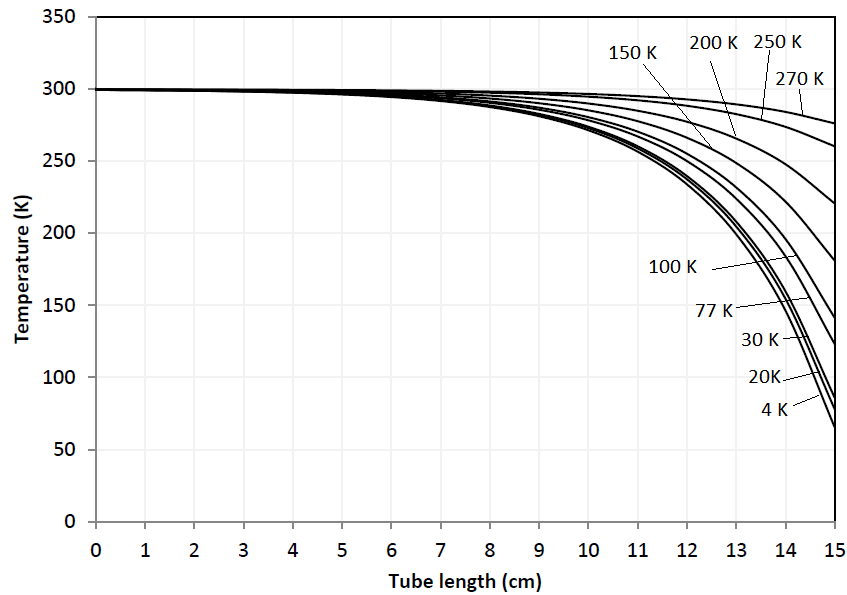**Figure 5.** Temperature of hydrogen gas leaving the tube with various Reynolds number.

In the laminar flow, Nusselt number is approximately constant within the tube with zero heat flux from tube wall (equation (3))[1]. Therefore, the variations of convective heat transfer coefficient are negligible while the gas thermal conductivity is constant. Consequently, gas resident time within the tube will be decreased by increasing the Reynolds number (mass flow) and gas cannot transfer energy with tube wall. For this reason, the temperature of gas remains constant along the tube. On the other hand, in some applications such as experiments for estimation of gas convective heat transfer coefficient by Wilson plot method, accurate temperatures of tube inlet and outlet are essential requirements to determine the actual correlations. Therefore, neglecting the wall longitudinal heat conduction in cryogenic tests and using liquid nitrogen as a constant temperature bath cause large measurement errors and result in the unreal correlations.

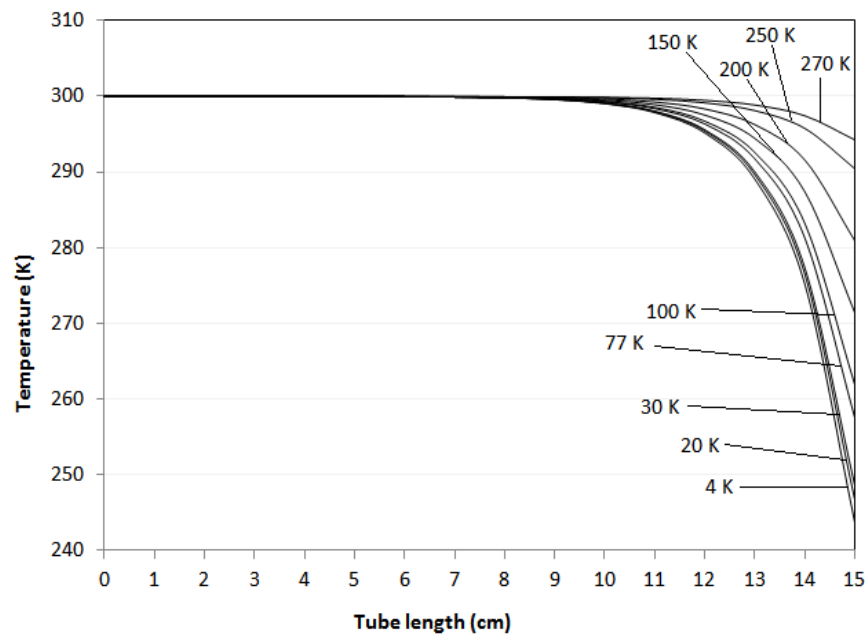Fig. 6 shows the temperature profiles of gas flow within the tube with various wall cold end temperatures. As seen, gas flow at the last 10 cm of tube end has been affected by wall longitudinal heat conduction and this phenomenon is independent of wall cold end temperature. In other words, at the constant Reynolds number and constant wall thermal conductivity, the main cooling effect of wall longitudinal heat conduction occurs in the small region of the tube end section. In contrast, as can be seen in Fig. 7, by increasing the Reynolds number up to 200, the effective length of the tube decreases because of decreased gas resident time. This means that the wall longitudinal heat conduction effect can be neglected at high Reynolds number and errors due to incorrect position of temperature sensors are minimized at these conditions. Fig. 8 shows the effect of wall thermal conductivity on the temperature profile of gas within the tube. The temperature profiles were plotted at Reynolds number of 200 and wall cold end temperature of 77 K. As seen, by increasing the metal thermal conductivity, longitudinal heat conduction

effect diffuses toward the tube inlet and gas has the benefit of longer time of transferring energy with tube wall. Consequently, the gas temperature within the tube is greatly affected by tube wall thermal conductivity and a gas with Reynolds number of 200 can be cooled from 300 K down to 170 K through the copper tube. Although the range of Reynolds number studied here is not used at industrial scales, it is widely used at experimental scales in order to investigate the heat transfer correlations.
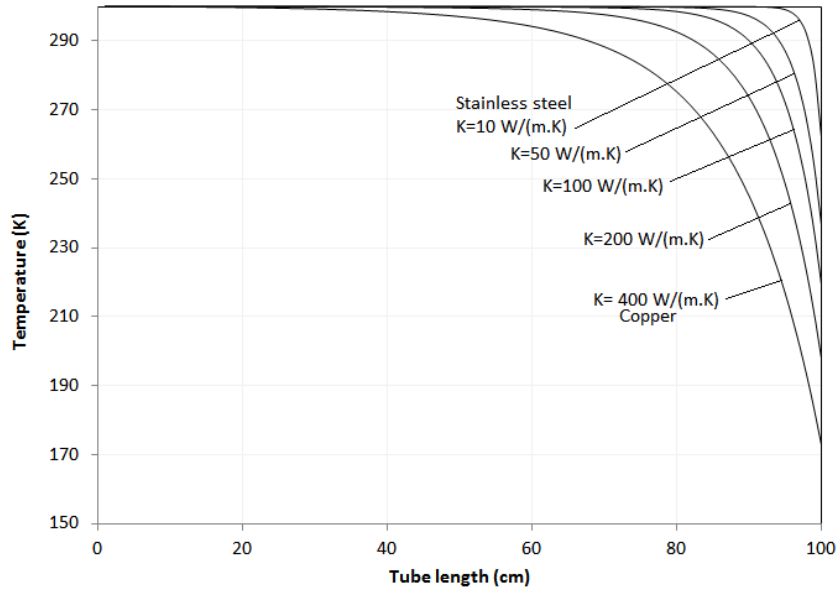
Figure 6. The effect of tube wall longitudinal heat conduction on temperature profiles of gas at various wall cold end temperatures (Re=20 and k=10 W/(m.K)).

Figure 7. The effect of wall longitudinal heat conduction on temperature profiles of gas at various wall cold end temperatures (Re=200 and K=10 W/(m.K)).

Figure 8. The effect of wall longitudinal heat conduction on temperature profiles of gas at various wall thermal conductivities (Re=200 and wall cold end temperature= 77 K).

Longitudinal heat conduction occurs when two ends of a metal tube have temperatures with large difference. Fig. 9 shows the temperature profiles along the tube within the gas flow and the tube wall for different wall diameter. As seen, the effect of longitudinal heat conduction has been increased by increasing the wall diameter. As the wall diameter increases, the heat capacity (mC$_p$) of tube wall increases while the mass flow of gas is constant. The mean heat capacity of gas flow with respect to the heat capacity of the tube wall is lower in the case of high tube wall diameter. Therefore, the gas flow temperature is influenced more by tube wall temperature and cold end temperature of gas flow becomes lower than those with a smaller tube wall diameter. Consequently, the tubes with small wall diameter are an appropriate choice for cryogenic application in order to eliminate the effect of longitudinal heat conduction through the tube wall. It must be noted that the small tube wall diameter causes the mechanical

failure in the case of high pressure applications, so the selection of appropriate tube wall diameter is a trade-off between the effect of longitudinal heat conduction and mechanical aspects.

5. Conclusions

As mentioned in the previous sections, wall longitudinal heat conduction has an important role in cryogenic field. In the present work, the parasitic effect of tube wall longitudinal heat conduction on temperature measurement in laminar flow was studied. According to simulation, the following conclusions were obtained:

• Simulation fitted the experimental results at high volumetric flows (high Reynolds number). Although the data obtained from simulation could not satisfy the experimental results at low volumetric flows, the overall trend of temperature decrease in both procedures (simulation and experiment) was similar and simulation results can be used for

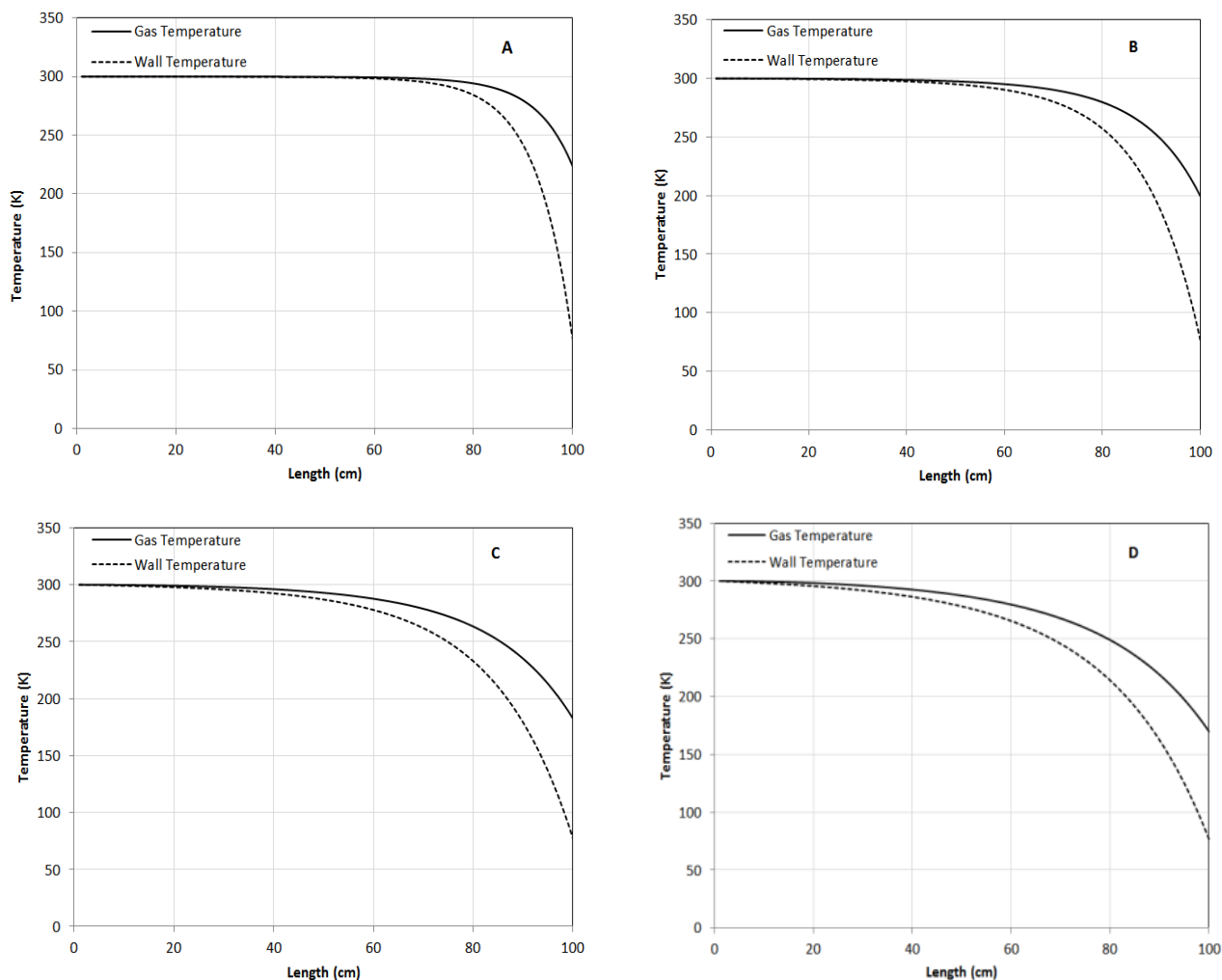different conditions and study of various parameters.

• Although tube wall longitudinal heat conduction exists in all ranges of Reynolds number, it can be neglected at high Reynolds numbers (e.g. higher than 2000 in the present study).

• Gas flow temperature has been influenced by wall longitudinal heat conduction at the end section of tube and this effective length is independent of wall cold end temperature. The effective length depends on tube wall thermal conductivity.

• The higher tube wall thermal conductivity results in longer effective length and consequently lower gas temperature at the tube cold end.

• Regardless of mechanical aspect, the tubes with small wall diameter are an appropriate selection for cryogenic application in order to eliminate the effect of longitudinal heat conduction through the tube wall.

Figure 9. Temperature profiles in gas flow and tube wall influenced by wall longitudinal heat conduction (Re=400, k=10 W/(m.K), (A). wall thickness=0.5 mm, (B). wall thickness=1 mm, (C). wall thickness=1.5 mm, (D). wall thickness=2 mm.

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