NUMERICAL INVESTIGATION OF THE INFLUENCE OF HEAT EXCHANGER U-BENDS ON TEMPERATURE PROFILE AND HEAT TRANSFER OF SECONDARY WORKING FLUIDS

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Abstract
In this paper, numerical investigations, conducted using computational fluid dynamics, on the enhancement of internal convection heat transfer following a heat exchanger U-bend under laminar flow conditions in secondary working fluids are described. Under laminar flow conditions enhanced mixing within the heat exchanger U-bends is known to occur due to the development of secondary flows, known as Dean vortices. Numerical investigations indicated that within the U-bend, secondary flows partially invert temperature profiles resulting in a significant localised decrease in average fluid temperature at the pipe surface. As a result, downstream heat transfer enhancement is observed, the magnitude of which can exceed that typical of a pipe combined entry condition in some circumstances by greater than 20% for up to twenty pipe diameters downstream. Heat transfer enhancement was found to increase with increasing U-bend radius, but to decrease with increasing heat exchanger pipe radius and internal Reynolds number.

Nomenclature

C_p	Specific heat capacity [$Jkg^{-1}K^{-1}$]
D	Pipe diameter [mm]
e	Percentage error [%]
Gr	Grashoff number = $g\beta(T_s-T_m)D^3\nu^{-2}$
h	Convection coefficient [$Wm^{-2}K^{-1}$]
k	Thermal conductivity [$Wm^{-1}K^{-1}$]
K	Dean number
L_i	Inlet pipe length [m]
L_o	Outlet pipe length [m]
Nu	Nusselt number = hDk^{-1}
Pr	Prandtl number = $\mu C_p k^{-1}$
Q	Surface heat flux [Wm^{-2}]
r	Pipe radius [mm]
R	Bend radius [mm]
Re	Reynolds number = $\rho VD\mu^{-1}$
T	Temperature [K]
V	Fluid velocity [ms^{-1}]
x	Axial pipe distance [m]
x*	Dimensionless distance = x/(DRePr)

Greek Symbols

β	Thermal Expansion Coefficient [K^{-1}]
θ	Sector Angle [°]
μ	Viscosity [$kgm^{-1}s^{-1}$]
ν	Kinematic Viscosity [m^2s^{-1}]
ρ	Density [kgm^{-3}]

Subscripts

exp	from experimental data
in	at inlet
i	at circumferential location i
m	mean
max	maximum
min	minimum
out	at outlet
sim	from simulation
tot	total
w	at pipe wall
x	at axial location, x

1 Introduction

Environmental concerns regarding the use of synthetic refrigerants have resulted in increased interest in the potential for indirect or secondary refrigeration systems. Indirect refrigeration systems, often used in supermarket applications [1,2], present an alternative refrigeration design concept to direct expansion (DX) systems. One advantage of these systems is that a smaller quantity of refrigerant is required in the primary loop than would be required if a direct expansion system alone were used. Indirect refrigeration systems however, require an additional heat exchanger, a secondary working fluid and an associated secondary pump, typically resulting in increased energy requirements over equivalent DX systems [3]. A further disadvantage associated with many secondary working fluids is that they operate in single-phase mode. Consequently, the high convection heat transfer coefficients associated with the boiling heat transfer is unavailable. Recent studies into secondary systems however, have determined that they can be surprisingly effective under the laminar flow regime, even outperforming direct expansion alternatives [4,5]. Hong & Hrnjak [6] observed that secondary flows developed within air chiller pipe bends cause significant mixing of the flow. This effect, it is suggested, eliminates the hydrodynamic and thermal development that occurs prior to the bend, resulting in a new development length immediately downstream of the U-bend. Specific investigation of the precise transport mechanisms that cause this heat transfer enhancement however, remain to be conducted and this forms the basis for the current paper.

Other experimental investigations conducted to date [7,8,9,10] have found that heat transfer may be enhanced immediately downstream of a U-bend. Unlike the current study, these investigations concentrated upon the magnitude of the enhancement effect and not upon the transport mechanisms that cause it. In general, the heat transfer enhancement is attributed to the mixing effect of centrifugally induced secondary flows known as Dean vortices that develop within the bend. These secondary flows, first described by Dean [11,12], are a result of centrifugal forces and a transverse pressure gradient that develop within the pipe as a fluid traverses a bend. Secondary flows have been characterised by a dimensionless number $K = Re(r/R)^{0.5}$, the Dean number [13]. The heat transfer enhancement effect of the secondary flow downstream of a bend is most pronounced for laminar flow [7,8], under which conditions heat transfer can also be influenced by natural convection [8,9]. Moshfeghian [8] noted that the surface temperatures following the bend vary circumferentially and suggested that it is the redistribution of temperature that occurs within the bend that leads to the downstream heat transfer enhancement. While the work conducted to date has suggested mechanisms for the heat transfer enhancement that occurs downstream of a U-bend, these mechanisms have not been confirmed or investigated in detail. Hong & Hrnjak [6] suggested that secondary flows potentially cause flow mixing, thereby creating a new development length downstream of the bend somewhat similar to a combined entry situation. Moshfeghian [8] however, notes that the surface temperatures vary circumferentially both within and downstream of the bend. This suggests that while the fluid is redistributed within the bend, it is not mixed sufficiently to result in a homogenous temperature at the bend exit, as is the case for a combined entry.

The current work therefore has been motivated by the requirement for a greater understanding of the transport mechanisms that cause this heat transfer enhancement effect. By examining the developing temperature profile both within and downstream of the U-bend, a more complete understanding of the enhancement mechanisms may be developed. This work aims to advance the fundamental understanding of the transport mechanisms by which U-bends distort temperature profiles under laminar flow conditions within a heat exchanger and consequently enhance downstream heat transfer.

2 Methodology and Validation

The investigation was conducted using the FLUENT CFD software package. The method employed facilitated the observation of velocity vectors and temperature contours upstream of, within and downstream of the U-bend. The model geometry utilised in this study consisted of a U-bend preceded by a straight circular inlet pipe and followed by an identical straight circular outlet pipe (Figure 1). For the validation models, dimensions were obtained directly from the literature [6,9]. Based on the experimental work of Haglund-Stignor [5] and Hong and Hrnjak [6], where heat fluxes between 0 and $20kWm^{-2}$ were utilised, a constant heat flux of $20kWm^{-2}$ was specified on the walls. This provided for a temperature rise over a 1m pipe section of 6-13K for Re=1500-500 respectively. The outlet boundary condition was set as an outflow condition which applies a zero diffusion flux for all flow variables and an overall mass balance correction.

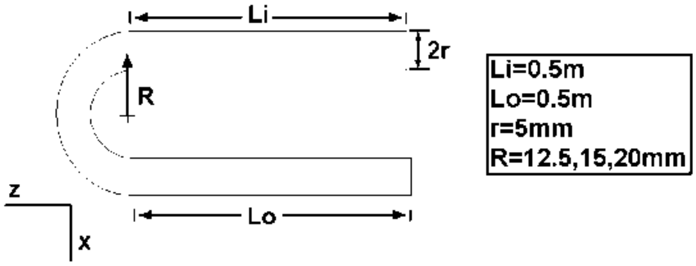

Figure 1 Key Model Dimensions

The model and solving procedure were validated by comparison with experimental data from two sources in the literature [6,9]. Data from Hong & Hrnjak [6] was particularly suitable for validation as, similar to the current investigation, the study dealt with the heat transfer from secondary working fluids downstream of an air-chiller U-bend. Local Nu number values, as predicted by Fluent for the outlet pipe, compared to similar data from Hong & Hrnjak are illustrated in Figure 2. A curve fit for the experimental data used by Hong & Hrnjak is also included. The average difference, over the range of values of x* for which experimental data was available (x*=0.00075 to 0.0032), was found to be 2.75%. In addition, surface temperature data along the length of the outlet pipe at eight evenly spaced circumferential locations was available from the work of Mehta and validation was also performed using this data [9]. Figure 3 shows surface temperature data for one such circumferential location, along the top surface of the outlet pipe (Line 1) for both a Fluent generated simulation and the experimental work of Mehta. Both data sets compared well with each other for all 8 circumferential locations, with a maximum difference for all circumferential and axial locations of 1K and an average difference of <0.5K. It was concluded from these validation studies that the simulation procedure and model mesh was capable of modelling the heat transfer enhancement of laminar flow fluids downstream of a U-bend sufficiently accurately to justify their use for further investigations into the transport mechanisms that cause this enhancement.

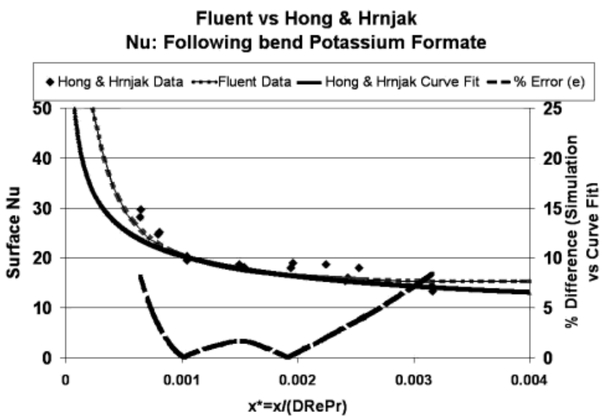

Figure 2 Validation against Hong & Hrnjak

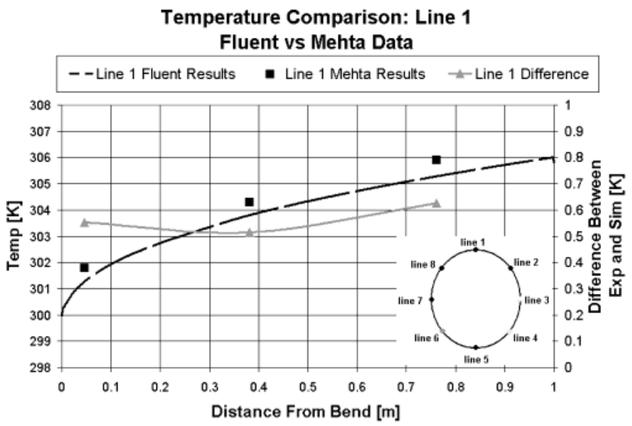

Figure 3 Validation against Mehta Data

3 Results

For high Pr number secondary refrigerant fluids, such as potassium formate, the temperature profile develops at a slower rate than the velocity profile. Consequently the development of the temperature profile exhibits a greater influence upon the heat transfer to these fluids under developing flow conditions than does the hydrodynamic development. In the following sections, the development of the fluid temperature profile upstream of, within and downstream of a 15mm radius U-bend is examined. For the model examined, the inlet temperature was specified as 250K and an inlet mass flow rate specified as 0.06kg/s provided Re≈1000 for the pipe diameter, D=10mm. The average fluid temperature at the pipe exit was found to be approximately 256K.

3.1 Upstream of the bend

Due to the laminar nature of the flow, transverse mixing is not significant upstream of the bend, although the cold fluid at the core tends to descend towards the bottom of the pipe while the fluid with the highest temperature is found towards the top of the pipe as a result of natural convection effects. As the temperature at the inside surface of the pipe increases along its length, the difference between the average fluid temperature and the inside surface temperature increases. This results in a steady decrease in Nu number along the inlet pipe in the direction of flow. This is illustrated in Figure 4 in which the Nu number preceding the bend as predicted by the FLUENT simulation is illustrated alongside the Nu number as predicted by a correlation developed by Churchill & Ozoe [14]. The correlation is applicable for laminar forced convection following a combined entry subject to a constant surface heat flux. For both the simulation and correlation data, the Nu number is observed to decrease steadily along the length of the pipe as the flow develops. The simulation results compare favourably to the correlation values. The Churchill & Ozoe correlation however, does not incorporate the effects of natural convection and as a result an increasing divergence between the two sets of data develops downstream of the pipe inlet as natural convection effects become more significant. The average difference between the data sets is within 4.5% of the correlation values.

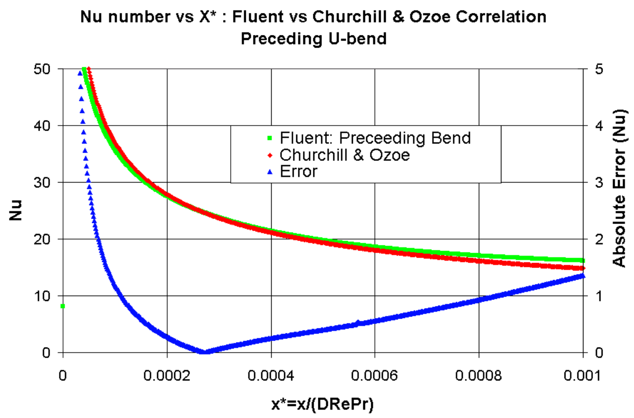

Figure 4 Nu number preceding the U-bend

3.2 Within the bend

Fluid enters the U-bend with the temperature profile approaching a parabolic shape developed within the straight heated inlet section. Within the bend this situation rapidly changes as a result of the Dean vortices that develop within the bend. A pressure gradient develops as fluid enters the bend, before impinging on the pipe walls at the outside of the bend and final redirecting around the bend. As illustrated in Figure 5a, higher pressure is found towards the outside wall of the bend, on which the fluid impinges, while lower pressures are experienced towards the inside wall of the bend. Centrifugal forces, induced due to the rotational motion of the fluid about the bend axis, tend to drive the fast moving fluid located at the core of the flow towards the outside of the bend. Towards the pipe walls however, the fluid velocity, and thus the centrifugal forces, are significantly lower than at the centre of the pipe. As a result, towards the walls the transverse pressure gradient illustrated in Figure 5b is sufficient to overcome the centrifugal force and to drive fluid from the bend outside around the pipe wall towards the bend inside. In addition, transverse velocity components that reveal the Dean vortices at 90° through the U-bend as predicted by the FLUENT model are also illustrated. The centres of the Dean vortices are found to be located away from the vertical axis of the pipe, towards the inside of the bend.

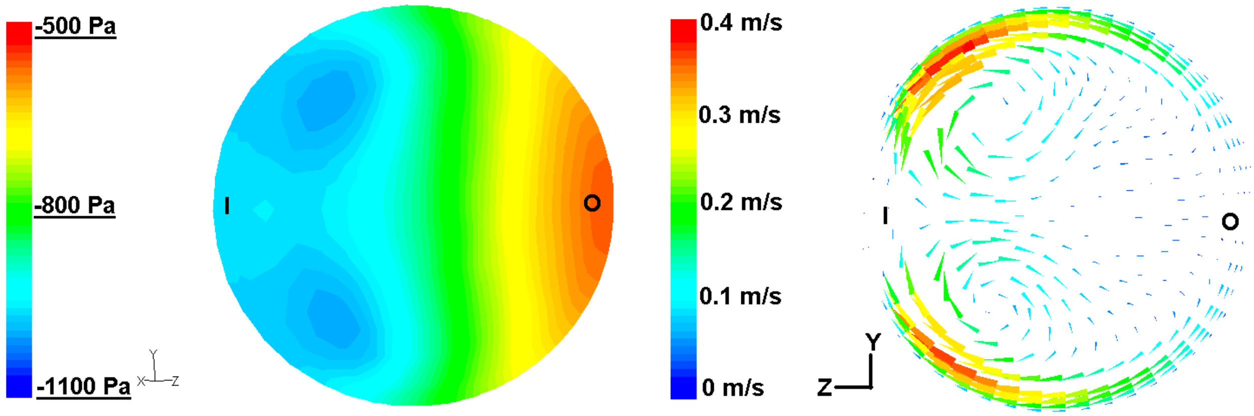

Figure 5 (a) Absolute pressure at 60° and (b) transverse velocity components at 90°

The effect of the secondary flow upon the temperature profile development, as predicted by the current model is illustrated in Figure 6. Temperature contours are illustrated at four locations (30°, 60°, 120°, 150°) through the U bend. The secondary flows are observed to drive cold fluid from the core of the pipe towards the outside pipe walls, with further circulation along the outside walls from the outside of the bend to the inside of the bend. This cold fluid replaces warm fluid that had developed at the pipe walls as the fluid moved through the inlet pipe. The warm fluid is simultaneously driven by the secondary flow, around the pipe walls towards the bend inside wall and onwards towards the pipe core. Consequently the thermal boundary layer is eliminated and a sudden drop in fluid temperature at the surface of the pipe occurs within the bend. It is this temperature drop and elimination of the thermal boundary layer that causes the subsequent downstream heat transfer enhancement. Notably however, the inversion process is not complete. A region of fluid with elevated temperature remains at the pipe wall located in a region towards the bend inside, I. The relative size of this region gives a measure of the completeness of the fluid inversion process. A more complete inversion would minimise the size of this region, reducing the average fluid surface temperature and maximising the downstream heat transfer enhancement.

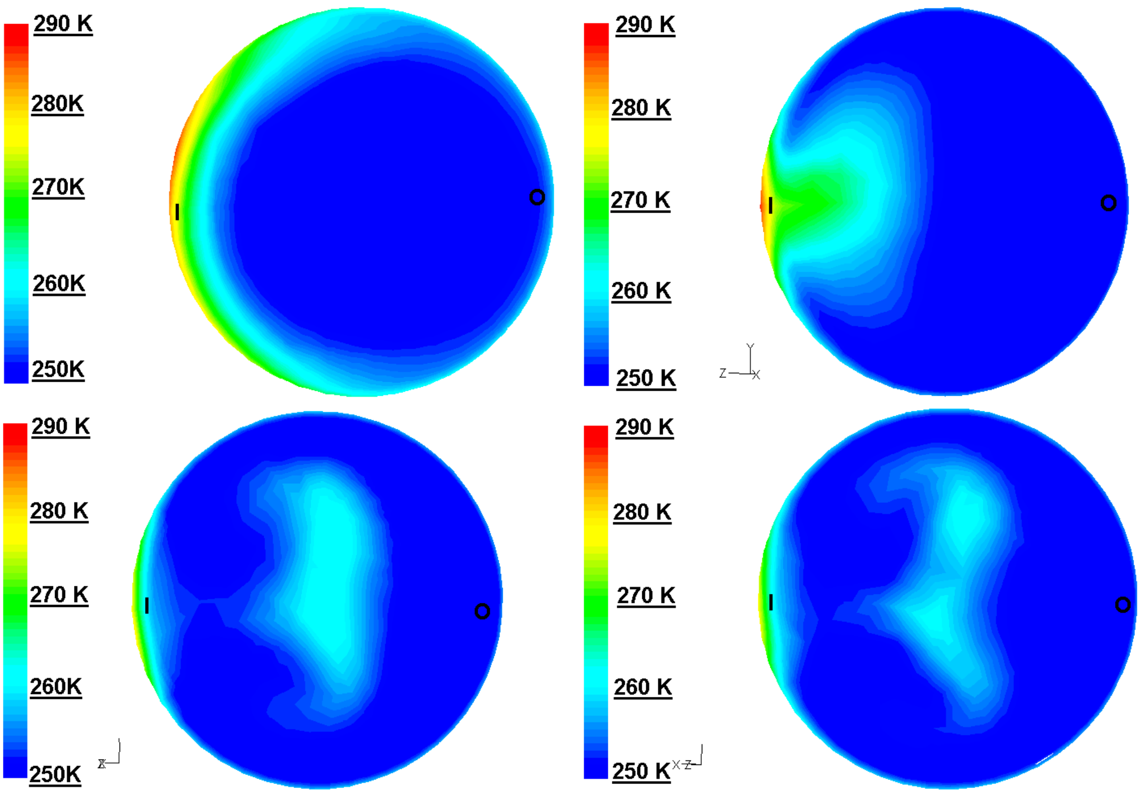

Figure 6 Temperatures contours at the 30°, 60°, 120°, 150° locations in the U bend.

3.3 Downstream of the bend

Figure 7 shows the transverse components of velocity at 90° through the U-bend and at one diameter downstream of the bend exit. Vectors have been scaled up by a factor of 2 for the in-bend vectors but were scaled up by a factor of 5 for the downstream vectors to ensure that they were large enough to view clearly. Upon exiting the bend, the centrifugal forces and pressure gradient that caused the secondary flow to develop within the bend are observed to have disappeared. As a result the transverse velocity components diminish swiftly downstream of the bend. For the current model, at 90° through the bend, the maximum transverse velocity was approximately 0.4m/s. At one diameter downstream of the bend however, the maximum transverse velocity reduced to <0.1m/s. By five diameters downstream of the bend the maximum transverse velocity reduced to approximately 0.04m/s, 10% of the value achieved within the bend. Consequently the secondary flow had a minimal impact upon the temperature profile development beyond the bend exit.

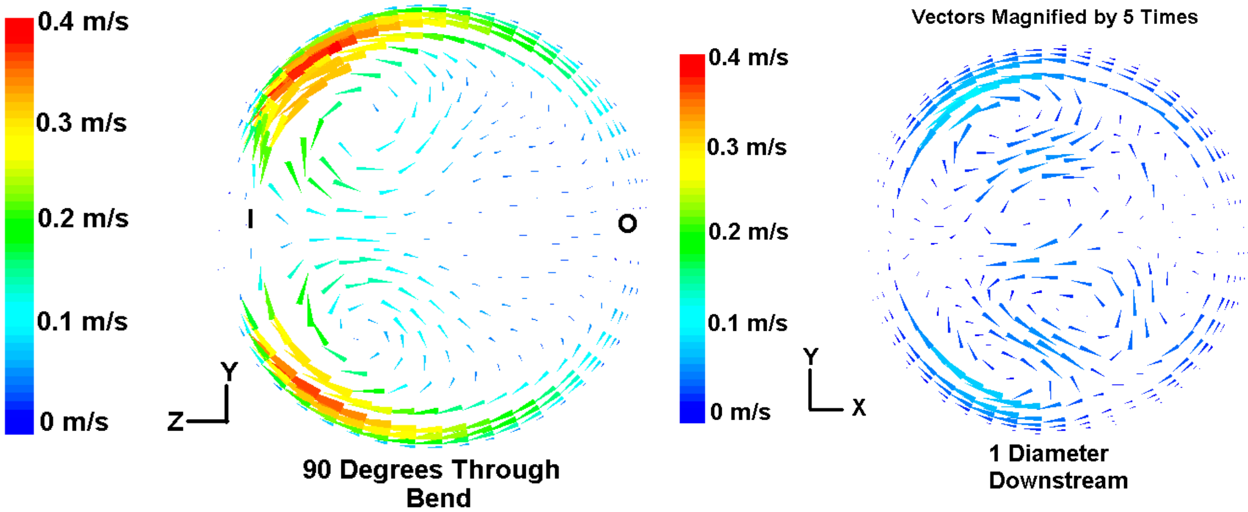

Figure 7 Velocity vectors at 90° through the bend and 1D downstream of the U-bend

Figure 8 illustrates the temperature contours at the bend exit and at 1, 5 and 10 diameters downstream of the U-bend. The effect of the secondary flow was observed to diminish significantly. The fluid from the core is no longer driven from the centre of the pipe towards the pipe walls. The temperature profile instead develops in a manner consistent with that of laminar pipe flow not preceded by a bend, i.e., as the fluid progresses through the outlet pipe, the fluid close to the walls swiftly increases in temperature while fluid at a greater depth heats at a much lesser rate. Mixing by the secondary flow decreases substantially and the fluid starts to approach a parabolic temperature profile once again. Some warm fluid at the core of the pipe remains in place for up to 20 diameters downstream. This warm fluid is observed to distort the developing parabolic profile somewhat, before eventually dissipating. Hong & Hrnjak [6] suggested that the heat transfer situation downstream of a U-bend was equivalent to that of a combined entry situation. Further to this, in the investigations of Haglund-Stignor [5], it was assumed that the region following a U-bend could be treated as a combined entry. However, from the current investigations, it was observed that the temperature contours at the bend exit differ significantly from the uniform temperature profile associated with a combined entry situation. As a result, it is apparent that the assumption that the heat transfer situation downstream of a U-bend is identical to that of a combined entry may not be entirely accurate.

Figure 9 illustrates the Nu number values predicted by the model for the region downstream of the U-bend. These values are compared to the simulation values determined for the combined entry region upstream of the U-bend. It was found that the Nu number values downstream of the U-bend are not identical to those for a combined entry. The downstream Nu values are observed to exceed the equivalent combined entry values, remaining greater than 20% higher than the combined entry value for up to x*=0.0004 (corresponding in this case to approximately 20 diameters downstream).

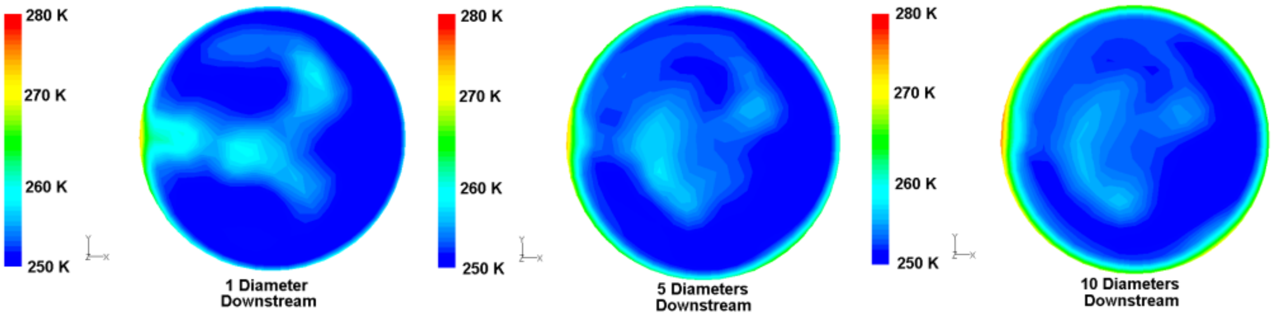

Figure 8 Temperature contours at 1D, 5D and 10D downstream of the U-bend

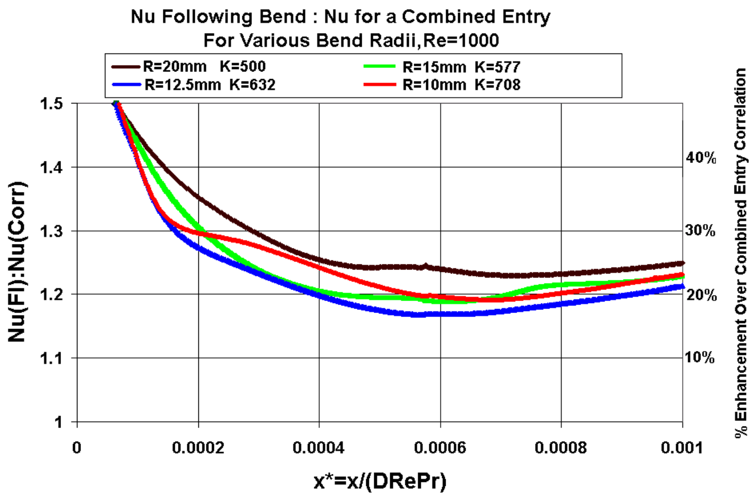

Figure 9 Nu number following the bend: Nusselt number for a combined entry, for a variety of Radius Ratio, r/R, values

This phenomenon is a consequence of the temperature profile inversion process. As the fluid progresses down the pipe, both for the combined entry situation and for the region following the bend, the surface temperature increases. As a result, the difference between surface temperature and average fluid temperature increases and the convective heat transfer coefficient decreases. For the combined entry situation, cold fluid located at the core of the flow results in the swift development of an increasing difference between surface temperature and average fluid temperature as the fluid progresses down the pipe. Following the bend however, the situation is altered insofar as cold fluid re-locates to the pipe surface, while warm fluid re-locates towards the core of the pipe. This warm fluid ensures that the difference between average fluid temperature and surface temperature increases at a lesser rate for the region following a bend than it does for a combined entry situation. Consequently, the surface heat transfer coefficient and Nu number values downstream of a U-bend exceed those achieved for a combined entry situation.

4. Conclusions

This work was motivated by the requirement for a greater understanding of the mechanism by which U-bends enhance downstream heat transfer from laminar flow secondary refrigerants in air-chillers. This work is of particular interest with regard to heat transfer from secondary refrigerants in finned tube air-chillers and other applications where high Pr number, single phase fluids are used for heat transfer applications in tubes that incorporate return U-bends. A model of a heat exchanger U-bend was developed using the FLUENT CFD software package to investigate the development of temperature profiles upstream of, within and downstream of a U-bend for a laminar flow secondary refrigerant. Validation of the model heat transfer was carried out by comparison with data from the literature. The mechanisms by which U-bends enhance refrigerant side heat transfer downstream of a secondary refrigerant air-chiller U-bend were examined. It was shown that centrifugally induced secondary flows, known as Dean Vortices, partially invert the temperature profile. Cold fluid is driven from the core of the pipe towards the pipe walls while warm fluid is driven around the circumference of the pipe towards the inside of the bend and onwards towards the centre of the pipe. Consequently, at the bend exit, cold fluid existed at the pipe walls and warm fluid was located at the core of the flow. Thus, the surface temperature of the pipe dropped significantly within the bend, resulting in an improved heat transfer situation downstream. It had been suggested previously [6] that heat transfer following a U-bend was very similar to that associated with a combined entry. In this work however, in some circumstances, Nu values downstream of a U-bend were found to exceed Nu values for a combined entry situation by greater than 20% for up to 20 pipe diameters downstream.

Acknowledgements

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