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The influence of secondary refrigerant air chiller U-bends on fluid temperature profile and downstream heat transfer for laminar flow conditions

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Abstract

This paper describes numerical investigations, using computational fluid dynamics, conducted to examine the heat transfer mechanisms by which air-chiller U-bends cause enhanced downstream internal convection, where single phase secondary refrigerants under laminar conditions are employed as the heat exchanger fluid. The numerical model, created using FLUENT, consists of a single heat exchanger tube pass incorporating an inlet pipe, a U-bend and an outlet pipe. The model was validated using experimental data from the literature. Numerical investigations indicate 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 20 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. A simple technique based on quantification of the degree of temperature inversion at the U-bend is proposed which provides a mechanism by which heat transfer enhancement can be estimated.

Keywords: CFD; Heat transfer enhancement; Laminar; Secondary refrigeration; Temperature profile; U-bend

1. Introduction

Indirect refrigeration systems (Fig. 1), often used in supermarket applications [1,2], present an alternative refrigeration design concept to direct expansion (DX) systems that can reduce, or even eliminate, the use of environmentally damaging CFC (chlorofluorocarbon), HFC (hydro-fluorocarbons) and HCFC (hydro-chlorofluorocarbon) refrigerant compounds. A major advantage of this system is that a smaller quantity of refrigerant is required in the primary loop than would be required if a direct expansion (DX) system alone were used. Horton and Groll [1] compared a DX system to an indirect system with an equivalent cooling capacity. The charge of primary refrigerant required for the indirect system was only 10% of the refrigerant charge required for the DX system.

Indirect refrigeration systems however, require an additional heat exchanger and a secondary refrigerant pump, typically resulting in increased energy requirements over equivalent DX systems [3]. In addition, a common feature of most antifreeze secondary refrigerants is that they operate in single-phase mode. Consequently, the high convection heat transfer coefficients associated with the evaporation of a fluid is unavailable. Recent studies into secondary refrigeration systems however, have determined that they can be surprisingly effective under the laminar flow regime, even outperforming direct expansion alternatives [4]. Laminar flow, typically associated with poor heat transfer, has been found by Haglund Stignor, [5] to provide the most energy efficient air-chiller performance in many

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Nomenclature

C_p	specific heat capacity (J kg^{-1} K^{-1})
D	pipe diameter (mm)
e	percentage error (%)
Gr	Grashoflf number $= g\beta(T_s - T_m)\delta^3 v^{-2}$
h	convection heat transfer coefficient (W m^{-2} K^{-1})
k	thermal conductivity (W m^{-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 (W m^{-2})
r	pipe radius (mm)
R	bend radius (mm)
Re	Reynolds number $= \rho VD/\mu$
T	temperature (K)
V	fluid velocity (m s^{-1})
x	axial distance along pipe (m)
x^*	dimensionless distance $= x/D\,Re\,Pr$

Greek symbols

β	coefficient of thermal expansion (K^{-1})
θ	sector angle (°)
μ	viscosity (kg m^{-1} s^{-1})
ν	kinematic viscosity (m^2 s^{-1})
ρ	density (kg m^{-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

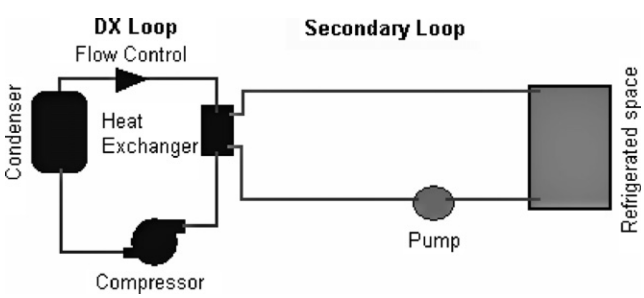

Fig. 1. Indirect refrigeration system.

situations, due to surprisingly good heat transfer performance and the reduced pumping power required for laminar flow. Hong and Hrnjak [6] proposed 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. Within the development region, which extends to a significant length for high Pr number secondary refrigerant fluids, high convective heat transfer can exist for laminar flow conditions. 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 research.

Other experimental investigations conducted to date [7–10] 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 $K = Re\sqrt{(r/R)}$, 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.

Abdelmessih and Bell [10] proposed a correlation (Eq. (1)) for the local Nusselt number following U-bends. This correlation attempts to incorporate the effects of forced convection, natural convection and secondary flow effects and is based on experimental data that lies within the ranges: $120 \leqslant Re \leqslant 2500$; $3.9 \leqslant Pr \leqslant 110$; $2500 \leqslant Gr \leqslant 1{,}130{,}1000$; $27 \leqslant x/D \leqslant 171$.

$$Nu = \left[4.36 + 0.327(Gr\,Pr)^{1/4} + 1.955 \times 10^{-6} Re^{1.6} K^{0.8} \mathrm{e}^{-0.0725(x/D)}\right] \times \left(\frac{\mu_m}{\mu_w}\right)^{0.14} \quad (1)$$

This correlation is applicable to the region downstream of a U-bend and exhibits the impact of the bend through incorporation of the Dean number, K. The influences of