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Research Paper

Transient model of a refrigerant-to-water helically coiled tube-in-tube heat exchanger with corrugated inner tube

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HIGHLIGHTS

- The homogeneous flow approach is suitable for refrigerant two-phase flow modeling.
- The superposition principle is appropriate where suitable correlations are missing.
- Transient models of refrigeration systems require correct OFF-cycle modeling.
- Helically coiled double tubes with inner fluted tube experience high pressure drop.

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ABSTRACT

A transient model of a refrigerant-to-water helically coiled double tube with corrugations in the inner tube is developed. The refrigerant flows in the annulus, and a secondary fluid (water, glycol/water solution, etc.) flows in the inner tube. The model addresses both the single-phase and the two-phase flows of the refrigerant and the single-phase flow of the secondary fluid. Both the ON/OFF operation of the compressor and of the secondary fluid pump (at compressor shut-down, the pump may be ON or OFF) are considered. To deal with the lack of suitable friction factor and heat transfer correlations, a superposition principle that allows to combine the effects of the curvature of the heat exchanger and of the flutes in the inner tube is adopted. The finite volume method is used to numerically solve the governing equations. The developed model is experimentally validated and achieves reasonable agreement with the experimental data in both steady-state and transient conditions, in both the heating and the cooling modes.

1. Introduction

Refrigerant-to-water helically coiled concentric tubes with corrugated inner tube are popular heat exchangers in the HVAC (Heating, Ventilating and Air Conditioning) industry, especially in water-source and ground-source heat pumps due to their compactness and high heat exchange efficiency. In the literature, the corrugated tubes have various other designations: “convoluted”, “fluted”, “grooved”, or simply “enhanced”.

The presence of a secondary flow due to the coiling [1–4] and of the corrugations in the inner tube [5–9] significantly enhances the heat transfer to, however, the detriment of the pressure drop which increases significantly when compared to straight smooth tubes. In this kind of refrigerant-to-water heat exchanger, typically the refrigerant flows in the annulus between the two tubes, and a secondary fluid (which may be water, water/glycol solution, etc.) flows in the internal tube, with either parallel flow or counter-flow depending on the mode of operation of the heat pump (cooling or heating). Generally, the internal tube is made from copper or cupronickel (an alloy of copper and nickel used when the secondary fluid is corrosive), and the external tube is made from steel.

In the literature, several investigators have addressed the modeling of either helically-coiled smooth tubes (e.g. [10–12]) or fluted tubes without coiling (e.g. [13–16]), but there is a shortage of modeling studies on tubes that combine both the coiling and the corrugations. In this paper, a transient model of a helically-coiled tube-in-tube heat exchanger with corrugated inner tube is being developed as part of an effort to create a detailed transient model of a direct-expansion ground-source heat pump, where this kind of heat exchanger is used as an indoor heat exchanger. Integrated into refrigeration systems, these heat exchangers are subjected to dynamic effects (variable refrigerant flow, variable secondary fluid flow, thermal capacitance of the tubes) that only a transient model can capture.

The model presented here accounts for the ON/OFF operations of the compressor and of the secondary fluid pump. The flow rate of the secondary fluid, when flowing, may be variable or not.

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Nomenclature

A	area (m^2)
Bo	boiling number
C	coefficient
Cp	specific heat capacity (J/kg.K)
d	condensation length (m)
D	diameter (m)
De	dean number
e	flute depth (m)
e^*	non-dimensional flute pitch
f	friction factor
F	friction force per unit volume (N/m^3)
g	gravitational acceleration (m/s^2)
h	specific enthalpy (J/kg)
h	heat transfer coefficient ($W/m^2.K$)
k	thermal conductivity (W/m.K)
L	length (m)
M	molecular mass (g/mol)
Nu	Nusselt number
P	pressure (Pa)
p	pitch (m)
p^*	non-dimensional flute pitch
Pr	Prandtl number
Q	heat flux per unit volume (W/m^3)
r	radius (m)
r^*	diameter ratio
Ra	Raleigh number
Re	Reynolds number
T	temperature (°)
t	thickness (m)
t	time (s)
u	velocity (m/s)
v	secondary fluid velocity (m/s)
x	vapor quality
X_{tt}	Martinelli parameter

Greek letters

α	flute helix angle (°)
α	void fraction
β	thermal expansion coefficient (K^{-1})
ϕ^2_{fo}	two-phase frictional multiplier
μ	viscosity (kg/m.s)
ρ	density (kg/m^3)
σ	surface tension (N/m)
θ^*	non-dimensional flute angle

Subscripts

1	internal side of inner tube
2	external side of inner tube
3	internal side of outer tube
A	area
b	bore
c	critical
c	helical coil
e	envelope
e	external
e	secondary fluid
f	friction
h	hydraulic
i	internal
l	liquid
m	modified
P	perimeter
r	refrigerant
v	vapor
w	tube wall

2. Heat exchanger geometry

Fig. 1 shows a schema of a section of the heat exchanger, and Fig. 2 shows the characteristic dimensions of the flutes. The flutes may be treated as triangles of height e and base s [17]. The "base" s is the part of the tube circumference occupied by the flute (Fig. 3).

D_{1A} is defined so as to represent the mean cross-sectional area, and D_{1P}, to represent the mean perimeter of the corrugated tube:

$$D_{1A} = \sqrt{D_e^2 - \frac{2\ N_n\ e\ s}{\pi}} \tag{1}$$

$$D_{1P} = D_e + \frac{N_n}{\pi}\left(\sqrt{4\ e^2 + s^2} - s\right) \tag{2}$$

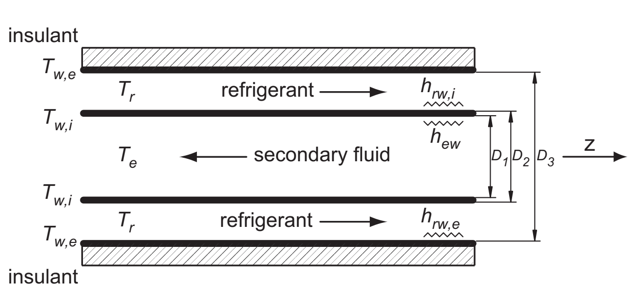

Fig. 1. Schematic representation of the double tube heat exchanger.

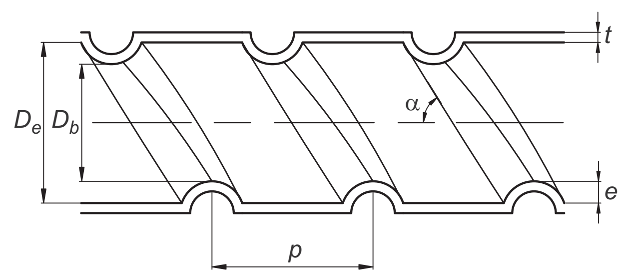

Fig. 2. Geometry of the corrugated inner tube.

The external equivalent diameter of the corrugated tube is then:

$$D_2 = D_1 + 2t \tag{3}$$

One can note that if the cross-sectional area A_{tube} or the perimeter P_{tube} of the corrugated tube are known with precision, then:

$$D_{1A} = \sqrt{\frac{4A_{tube}}{\pi}} \tag{4}$$

$$D_{1P} = \frac{P_{tube}}{\pi} \tag{5}$$

If the annular cross-sectional area A_{ann} is known, one can determine D_{2A} directly:

$$D_{2A} = \sqrt{D_3^2 - \frac{4A_{ann}}{\pi}} \tag{6}$$