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INVESTIGATION INTO THE PHASE CHANGE OF REFRIGERANT IN A WIRE-AND-TUBE CONDENSER OF A REFRIGERATOR

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Abstract. During the passage of the refrigerant through the condenser of a refrigerator, it generally undergoes the processes of desuperheating, condensation and sub-cooling. One of the important criteria in condenser design is to locate the points where superheated refrigerant vapour changes to saturated vapour and subsequently from wet vapour to saturated liquid. The present paper discusses the analysis of wire-and-tube condenser under normal operating conditions of free convection using FEM where Galerkin's weighted residual method is used to minimise the errors. Refrigerant R-134a is considered for the study and the condenser components are divided into several elements where the state of refrigerant is predicted. The effects of ambient temperature and mass flow rates of the refrigerant are also determined. This is used to find out the length of tube required for phase change for its initiation and completion. The methodology adopted is validated against the simulation data available in the literature for both sensible and latent heat transfer. The analysis also leads to the information about the adequacy of the number of tubes for complete condensation of the refrigerant vapour under given operating conditions. It is found that for a certain mass flow rate of refrigerant, sub-cooling does not occur at some higher ambient temperatures, thus showing the inefficiencies of the heat exchanger. The methodology can be used as a design tool for the design of wire-and-tube condenser of a small refrigerator. The derating of condenser under abnormal ambient conditions can also be predicted.

Keywords: Wire-and-tube condenser, refrigerator, free convection, finite element method, phase change

1.0 INTRODUCTION

The refrigeration cycle is a mean of removing heat from a place where it is not wanted and rejecting it to a place where it is not objectionable. The wire-and-tube condenser is the component in the refrigeration cycle, where heat of the refrigerant is removed and rejected. Condensers are thus heat exchangers designed to get rid of the heat absorbed by the refrigerant in the evaporator and the heat of compression added by the compressor. One of the commonly used condensers in domestic refrigerators is wire-and-tube condenser. Wire-and-tube condenser consists of a single

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steel tube, bent into serpentine parallel passes and solid steel wires are attached to the tube that serve as extended surfaces. The solid wires are brazed on to diametrically opposite sides of the tubes as shown in Figure 1(a). Figure 1(b) shows the wire-and-tube condenser parameters. Usually superheated refrigerant discharged from the compressor is cooled to the condensing temperature and thereafter it condenses to liquid refrigerant undergoing a phase change. Subject to availability of additional heat exchanger (i.e. condenser) surface the refrigerant may leave the condenser in a subcooled state. The heat transfer takes place from the outer surfaces of the wires and tubes to the external environment by free or forced convection.

Bansal and Chin [1] have developed a simulation model using the finite element and variable conductance approach, along with a combination of thermodynamic correlations. They optimised the condenser capacity per unit weight, using a variety of wire and tube pitches and diameters. Tanda and Tagliafico [2] gave a Nusselt number correlation as a function of the geometric and operating parameters to predict free convection heat transfer from a vertical wire-and-tube heat exchanger to ambient air based on their experimental results. Hoke *et al*. [3] carried out experiments to investigate the air side convective heat transfer for wire-on-tube heat exchangers used in most refrigerators. They were able to give a correlation valid for all the experimental data for seven wire-on-tube heat exchangers studied under forced convection regime. The importance of angle of attack for locating the wire-on-tube heat exchangers that are cooled by forced convection is also highlighted. A new type of coiled and spiral heat exchangers for cryogenic plant had been developed by Martynov [4]. Reeves *et al*. [5] discussed the U.S. Department of Energy (DOE) refrigerator-freezer model, and demonstrated its limitations for simulating refrigerator performance under off design conditions. They performed experiments in a nominal $0.51 m^3$ capacity, top-mount refrigerator-freezer. Their system was instrumented and operated over a wide range of room ambient and internal cabinet temperatures. Lee *et al*. [6] developed the correlation on the airside heat transfer coefficient of wire-and-tube type heat exchanger using single layer samples. Lozza and Merlo [7] evaluated the performance of air-cooled condensers and compared performance for various fin designs and for coil arrangements. Wang *et al*. [8] investigated the effect of circuit arrangement on the performance of air-cooled condensers. Finite element modeling for the analysis of general types of heat exchangers with and without phase change has been carried out by Ravikumar *et al*. [9]. Quadir *et al*. [10] analysed the wire-on-tube heat exchangers under normal operating conditions i.e. free convection environment using finite element method. Their analysis disseminated the information about the adequacy of the number of tubes for complete condensation of the refrigerant vapour for a range of operating conditions different from the design conditions. They also determined the effects of ambient temperatures and mass flow rates of the refrigerant on phase change location.

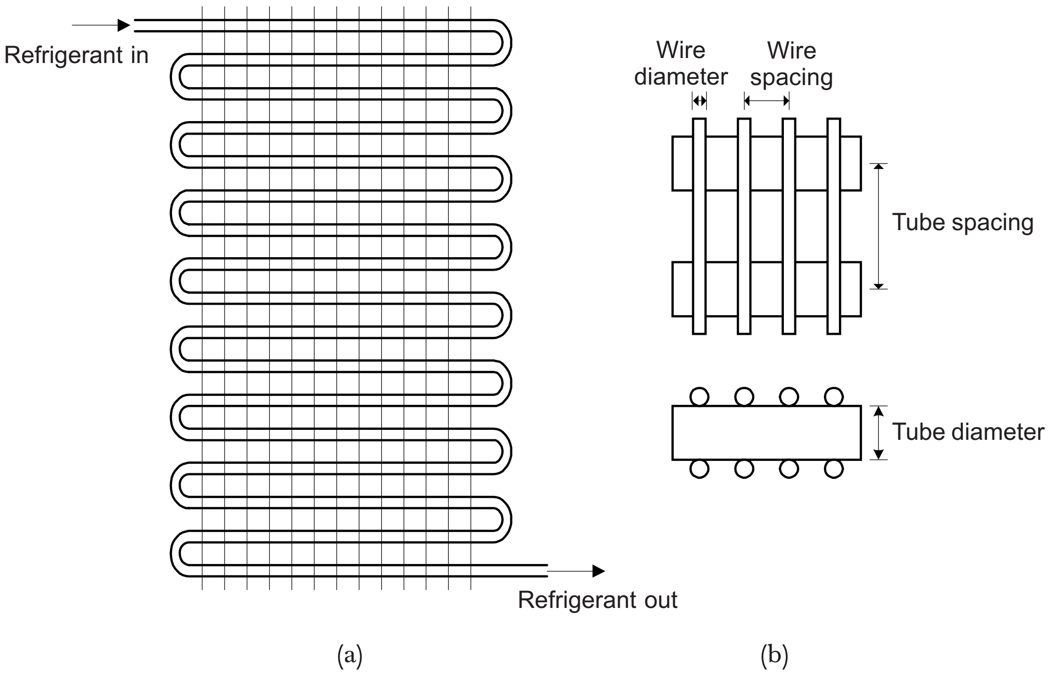

Figure 1 (a) Schematic of a wire-and-tube condenser and (b) wire-and-tube condenser parameters

Generally computer model is a very efficient tool for analysing the performance of any heat exchanger for different design parameters. This paper aims at presenting the development of such a model for designing the wire-and-tube condenser of a domestic refrigerator. As heat transfer takes place from outside of the condenser tubes of a refrigerator generally by free convection, the same was considered in the simulation, which was carried out by finite element method (FEM).

2.0 ANALYSIS

The numerical analysis is developed based on the finite element method. Figure 2(a) shows one tube length of a condenser consisting of bare portions of the tube as well as the wired portion. A typical one-element discretised model of the wire-and-tube condenser is shown in Figures 2(b) and 2(c) depending upon whether elements refer to the bare portion of the tube or wired portion of the tube, respectively. Every element has two nodes, one at entry and the other at exit point of tube side fluid. Refrigerant R-134a is considered as the working fluid entering the tube in a superheated vapour state. The fluid first follows desuperheating process bringing the state of the fluid to dry saturated condition. Then the fluid starts condensing in the tube. Depending upon the length of the tube available, the fluid may be sub-cooled or be in a two-phase mixture at the exit of the wire-and-tube condenser.

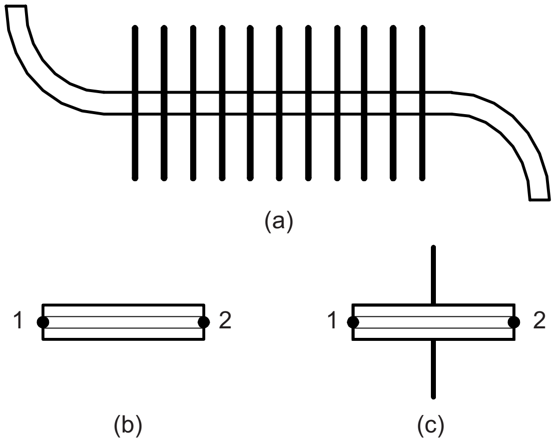

Figure 2 (a) One tube length of the wire-and-tube condenser, (b) bare portion of tube and (c) tube with wire attached

3.0 MODELING OF THE CONDENSER

A simulation model has been developed for the wire-and-tube condenser using the finite element method. The governing equations are solved by Galerkin's weighted residual method, which uses the integral form of governing equation. By substituting the approximate polynomial solution into the given partial differential equation, the result will not be zero, but will be a residual.

Since the refrigerant in a wire-and-tube condenser may follow three distinct regions as described above, it is important to describe the governing equations and their FEM formulations separately for each region.

3.1 Desuperheating Regime

The heat balance for a small element [11] can be written as,

$$\dot{m}_r . C_{pr} . \frac{dT}{dA} = -U\left(T - T_\alpha\right) \tag{1}$$

The differential equation governing heat transfer from superheated vapour to ambient air (free convection) over an elemental area dA in terms of enthalpy H is

$$\frac{dH}{dA} + \frac{U}{\dot{m}_r}\left[\frac{H}{C_{pr}} - T_\alpha\right] = 0 \tag{2}$$

where, $\dot{m}_r$ is the mass flow rate of refrigerant, C_{pr} is the specific heat of the refrigerant, U is the overall heat transfer coefficient for the element; T_α is the temperature of the ambient air. The enthalpy for the refrigerant in the element is assumed to vary linearly as,

$$H = N_1 H_1 + N_2 H_2 \tag{3}$$

where, the shape functions are given by:

$$N_1 = 1 - \frac{A}{\Delta A} \quad \text{and} \quad N_2 = \frac{A}{\Delta A} \tag{4}$$

and A is the area at any given location which varies from zero to ΔA, where ΔA is the area of an element. Applying the approximation to governing equation (2) and solving the above equations, the following element matrix is obtained

$$\begin{bmatrix} 2C - 0.5 & C + 0.5 \\ C - 0.5 & 2C + 0.5 \end{bmatrix} \begin{Bmatrix} H_1 \\ H_2 \end{Bmatrix} = \begin{Bmatrix} U.\Delta A.T_\alpha / (2\dot{m}_r) \\ U.\Delta A.T_\alpha / (2\dot{m}_r) \end{Bmatrix} \tag{5}$$

where, $C = \dfrac{U.\Delta A}{6\dot{m}_r.C_{pr}}$

The above formulation is valid till the enthalpy of the refrigerant equals to that corresponding to dry saturated condition.

3.2 Phase Change Regime

As the temperature of the refrigerant remains constant and equals to the saturation temperature in this region for a given pressure, the differential equation governing heat transfer is written as:

$$\frac{dH}{dA} + \frac{U}{\dot{m}_r}(T_{\text{sat}} - T_\alpha) = 0 \tag{6}$$

Assuming linear variation in enthalpy of the refrigerant in the element,

$$H = N_1 H_1 + N_2 H_2 \tag{7}$$

And following the same procedure as that in earlier region the following element matrix is found:

$$\begin{bmatrix} 1 & -1 \\ 1 & -1 \end{bmatrix} \begin{Bmatrix} H_1 \\ H_2 \end{Bmatrix} = \begin{Bmatrix} U.\Delta A(T_{\text{sat}} - T_\alpha)/\dot{m}_r \\ U.\Delta A(T_{\text{sat}} - T_\alpha)/\dot{m}_r \end{Bmatrix} \tag{8}$$

The above formulation will be valid till the enthalpy equals to that of saturated liquid at the operating condenser pressure.

3.3 Subcooling Region

The differential equation governing heat transfer from liquid refrigerant to ambient air in terms of temperature is written as:

$$\dot{m}_r . C_{pr} . \frac{dT}{dx} + U.P\left(T - T_\alpha\right) = 0 \tag{9}$$

where, P is the equivalent perimeter of the tube element. Assuming linear variation for the temperature of the refrigerant in the tube,

$$T = N_1 T_1 + N_2 T_2 \tag{10}$$

Applying the same procedure as before, the following element matrix is found:

$$\begin{bmatrix} 2C - 0.5 & C + 0.5 \\ C - 0.5 & 2C + 0.5 \end{bmatrix} \begin{Bmatrix} T_1 \\ T_2 \end{Bmatrix} = \begin{Bmatrix} 3.C.T_\alpha \\ 3.C.T_\alpha \end{Bmatrix} \tag{11}$$

where, $C = U.P.\Delta l/(6\dot{m}_r.C_{pr}) = U.\Delta A/(6\dot{m}_r.C_{pr})$ and Δl is the length of a small element.

A computer program (VISUAL FORTRAN 6.1) was developed to take into consideration all the three regions. The program is able to locate the position where the change of region takes place.

4.0 RESULTS AND DISCUSSIONS

Refrigerant R-134a enters into the condenser in a superheated vapour state at 1.2 MPa, and 60°C. The coil dimensions are tabulated in Table 1. The outside surface

Table 1 Design parameter and geometrical data of the current wire-and-tube condenser

Variable	**Dimension/Material**
D_{to} (mm)	4.9
D_{ti} (mm)	3.28
S_t (mm)	55.4
D_w (mm)	1.5
S_w (mm)	6.5
N_w	100
N_t	13
L_t (mm)	400
A_t (m^2)	0.097
A_w (m^2)	0.339
Tubing material	Stainless steel coated with copper

Table 2 Mass flow rate of refrigerant and ambient temperatures used in the parametric studies

No.	Number of runs	Mass flow rate of refrigerant, $\dot{m}_r$ (kg/s)	Ambient temperature, T_α (K)
1	7		
2	7		
3	7	0.0003 – 0.0007	283 – 313
4	7	with steps of 0.0001	with steps of 5
5	7		
Total	35		

area in the wire portion of the tube is different from that in the bare portion of the tube. Table 2 lists the number of parametric studies undertaken for varying mass flow rate of refrigerant and ambient temperatures.

In this study, each tube was divided into three distinct sections. The first and third sections are the bare portions of tube, while the second one is the wire portion where wires are welded. It is also to be noted that due to the gap between the wires in the second section, the outside surface area of elements are to be taken properly. For this section, the average element area method was considered for all calculations because the difference between the exact area method and average area method are very little (less than 2 %) [10].

Considering free convection, calculations were performed with 70 elements in the first, 348 elements in the second and 70 elements in the third section for the coil with a refrigerant mass flow rate of 0.0003 – 0.0007 kg/s and ambient temperature of 283 (10°C) – 313 K (40°C). The results are tabulated in Table 3. The overall heat transfer coefficient, U was considered as 10.82 W/m^{2}K obtained from [12].

The results in Table 3 shows the location of phase change, sub-cooling and sub-cooled temperature at the end of tubes, depending upon whether the flow condition is in the desuperheating region, phase change region or sub-cooling region. The variation of ambient temperature from 10°C to 40°C was considered in order to cater to different climatic conditions.

For example, at 10°C ambient temperature and 0.0007 kg/s mass flow rate of refrigerant the desuperheating of refrigerant vapour continues till 0.75 tube length (tube No.1) from the start. The phase change then follows and is completed at 9.7 tube length (tube No.10). Then the sub-cooling starts and the sub-cooled temperature of the refrigerant at the end of the condenser is 295.1 K (i.e. 22.1°C), which is 12.1°C higher than the ambient temperature. As the ambient temperature is increased, it is observed that the beginning of the phase change and sub-cooling are delayed.

Again at 20°C ambient temperature and same mass flow rate of refrigerant the desuperheating of the refrigerant vapour continues till 1.16 tube length (tube No.2) from the start. There is no sub-cooling taking place for the same mass flow rate of

Table 3 Location of phase change, sub-cooling and sub-cool temperature, varying with mass flow rate and ambient temperature. Refrigerant initial conditions: p =1.2 MPa, T_{sup} = 60°C

Mass flow rater $\dot{m}_r$ (kg/s)	Ambient temperature T_α (K)	Location of starting of phase change (tube length)	Location of starting of sub-cooling (tube length)	Sub-cooling temperature at the exit of last tube (K)
0.0003	283	0.38	4.20	283.03
	288	0.42	4.74	288.05
	293	0.46	5.67	293.08
	298	0.53	7.11	298.20
	303	0.62	9.22	303.83
	308	0.78	NSC	NSC
	313	1.31	NSC	NSC
0.0004	283	0.47	5.53	283.41
	288	0.52	6.44	288.62
	293	0.58	7.6	294.11
	298	0.67	9.4	300.60
	303	0.80	12.26	313.28
	308	1.22	NSC	NSC
	313	1.64	NSC	NSC
0.0005	283	0.56	6.83	285.0
	288	0.62	8.1	290.99
	293	0.70	9.5	298.20
	298	0.81	11.67	309.84
	303	1.2	NSC	NSC
	308	1.45	NSC	NSC
	313	2.2	NSC	NSC
0.0006	283	0.65	8.4	288.74
	288	0.72	9.6	296.43
	293	0.82	11.45	307.40
	298	1.2	NSC	NSC
	303	1.36	NSC	NSC
	308	1.7	NSC	NSC
	313	2.50	NSC	NSC
0.0007	283	0.75	9.7	295.10
	288	0.82	11.3	305.47
	293	1.16	NSC	NSC
	298	1.30	NSC	NSC
	303	1.53	NSC	NSC
	308	2.0	NSC	NSC
	313	2.83	NSC	NSC

NSC = No sub-cooling takes place

refrigerant and the refrigerant remains in two-phase condition. This suggests that additional tube length is required in order to complete condensation of refrigerant vapour for such cases.

Graphs for different refrigerant mass flow rates and ambient temperatures are shown in Figure 3. In this figure, PC-10, PC-15, etc. represent curves showing the location of phase change taking place in tubes at 10, 15°C etc. respectively. The other notations SC-10, SC-15, etc. refers to the starting of the sub-cooling similar to the phase change notations. Figure 3 does not show the variation of location for sub-cooling corresponding to 35°C and 40°C because there is no sub-cooling for 35°C and 40°C for all mass flow rates of the refrigerant. It is observed that phase change or sub-cooling is delayed with either the increase in the refrigerant mass flow rate or increase in ambient temperature. The variation in these figures, however, follows almost a straight-line behavior.

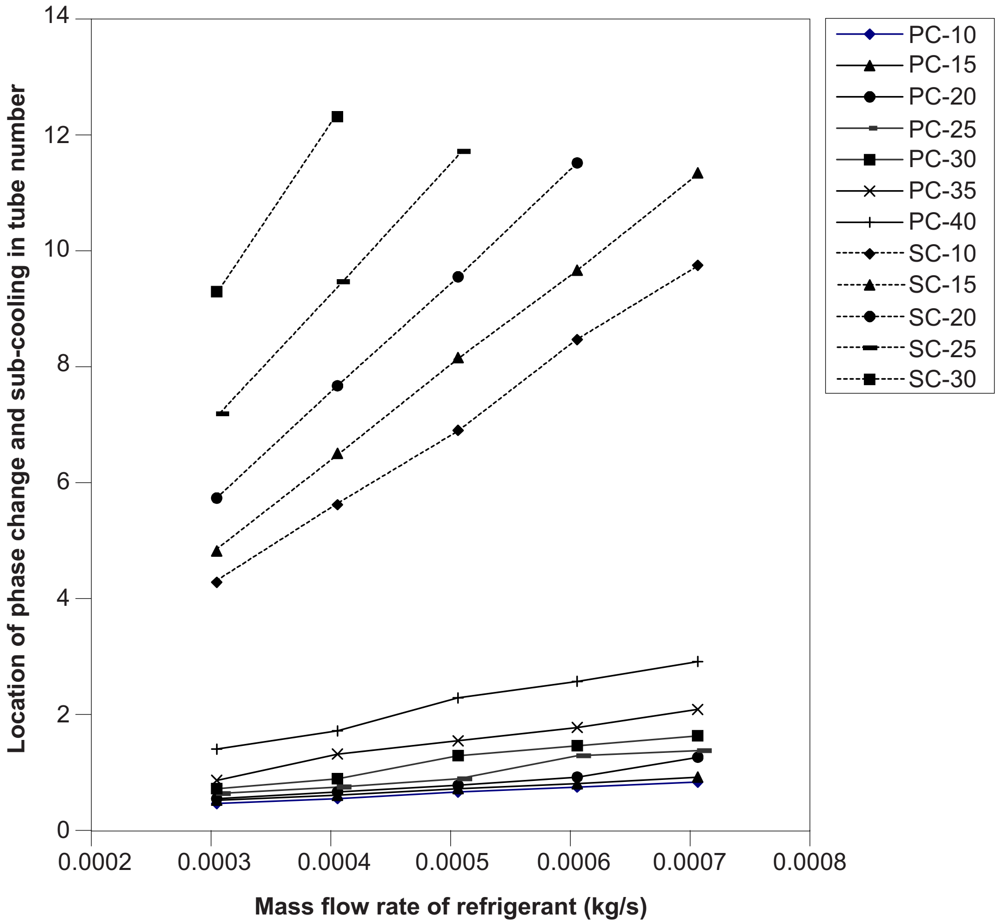

Figure 3 Location of starting of phase change and sub-cooling with variation in refrigerant mass flow rates and ambient temperature

The above results in respect of location of phase change and sub-cooling with variation in ambient temperatures for mass flow rate of refrigerant of 0.0006 kg/s are plotted in Figure 4.

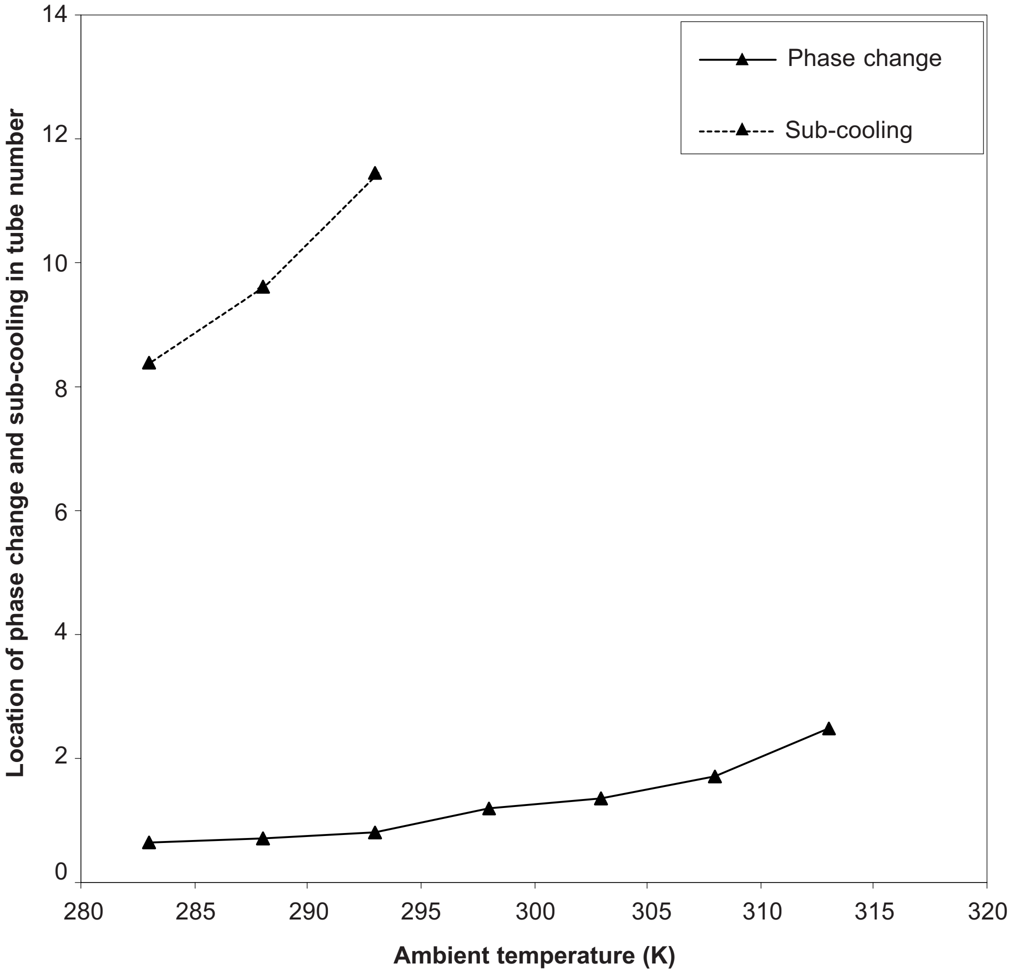

Figure 4 Location of starting of phase change and sub-cooling with variation in ambient temperature with refrigerant mass flow rate of 0.0006 kg/s

5.0 CONCLUSIONS

Galerkin's finite element method was used to analyze the performance of a wire-and-tube condenser under free convection conditions. The present method compares the sensible and latent heat transfer for refrigerant R-134a with the simulation data available in the literature, with a deviation of around ± 8%. The proposed method is used to study the effect of parameters like mass flow rates of refrigerant, ambient temperature, tube and wire geometry, etc. at the beginning and end of the phase change or sub-cooling taking place inside the tubes of the condenser. The predictions were carried out using a locally developed computer program (VISUAL FORTRAN 6.1). The present method can be used to check the number of tubes provided in an existing wire-and-tube condenser when the operating conditions are different from design conditions and also can be used as a design tool to design a new wire-and-tube condenser under abnormal ambient conditions.

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Nomenclature

A	area, m^2
C_p	specific heat capacity at constant pressure, kJ/kg K
D	diameter, m
FEM	finite element method
H	total enthalpy, kJ
h	specific enthalpy, kJ/kg
L	length, m

$\dot{m}$	mass flow rate, kg/s
N	number, shape factor
NSC	no sub-cooling takes place
P	perimeter, m
p	pressure, MPa
R-134a	Tetrafluoro ethane
S	pitch, m
T	temperature, K
U	overall heat transfer coefficient, W/m^{2}K

Subscripts

r	refrigerant
sat	saturated condition
sup	superheated condition
t	tube
t_i	inside tube
t_o	outside tube
w	wire
$\propto$	ambient temperature