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ORC Condenser Heat Exchanger Design and Modelling

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Abstract- **The paper presents work done on the development of a condenser heat exchanger model suitable for incorporation into a low temperature solar thermal power cycle based on the organic Rankine cycle (ORC); it presents the mathematical and computer models of a flow of vapour over a bundle of horizontal tubes. Although the process of condensation finds application in a lot of industrial systems, the concept itself is still not yet well defined as a scientific notion and is still a subject of investigation; a number of empirical correlations that have been proposed are reviewed in this research and are to be evaluated against the results of experimental investigations. The working fluid is modelled from five organic fluids being R123, R134a, R245fa, n-butane and isobutene; the cooling liquid is placed on the tube side and consists of ethylene glycol at 50 % concentration. The condenser model is implemented on the engineering equation solver platform. The current study presents the preliminary results of the mathematical and computer analyses. The results are in the form of condensation heat transfer coefficients and rate of heat transfer for each tube in a vertical tire tube bundle.**

Keywords: Condensation heat transfer coefficients, bundle of horizontal tubes, engineering equation solver

Nomenclature:

Roman Symbols

A	surface area at nominal diameter (m^2)
A_f	fin surface area (m^2)
A_r	tube surface area at the base of the fins (m^2)
D	tube diameter (m)
D_r	diameter at fin root (m)
F	a constant accounting for the effect of physical properties
h_{fg}	latent heat of condensation (J/kg)
g	gravitational acceleration (m/s^2)
h	heat transfer coefficient ($W/m^2.K$)
k	thermal conductivity of condensate ($W/m^2{}^\circ C$)
Lc	Lc is the characteristic length (m)
N	number of tubes
Nu	Nusselts' number
T	Temperature (^oC)

Greek Symbols

η	fin efficiency
μ	dynamic viscosity (kg/m.s)
ρ	Density (kg/m^3)

Subscripts

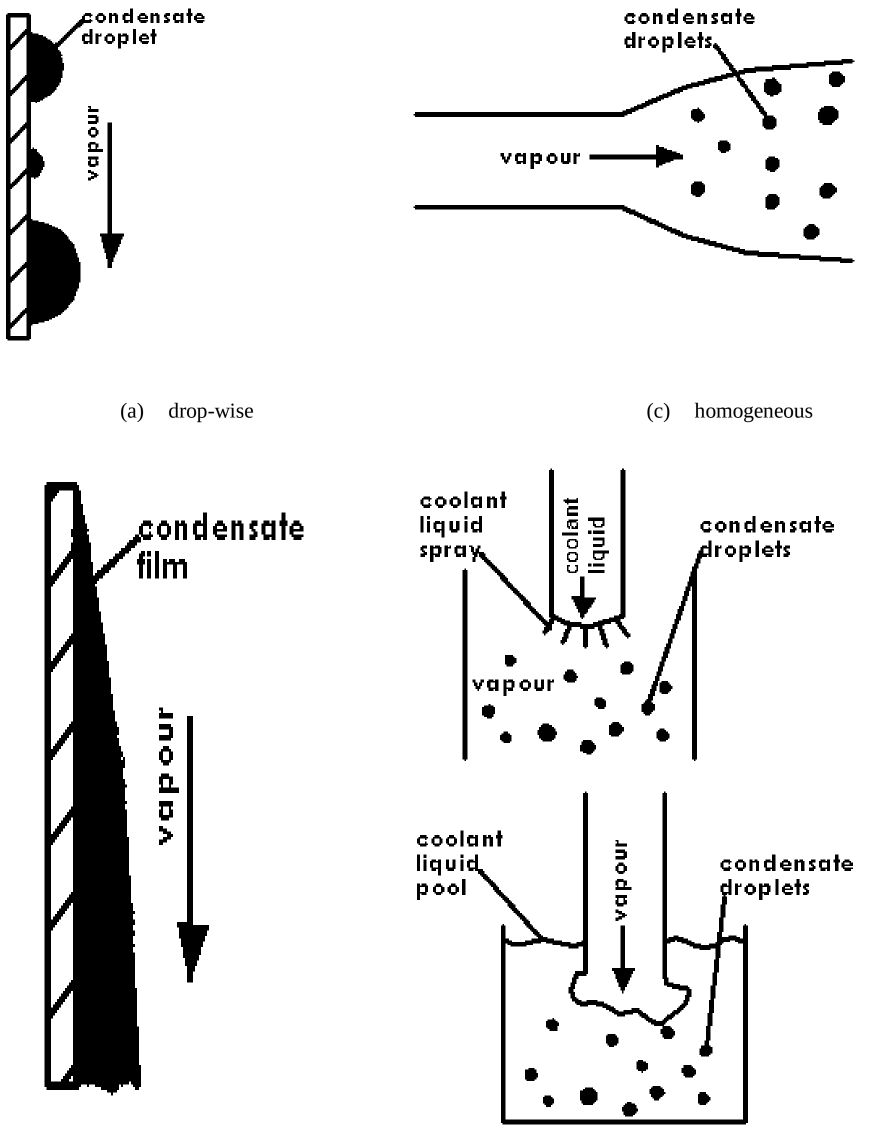
G	vapour phase
i	inside
L	liquid phase
o	outside
sat	saturation state
w	tube wall

I. INTRODUCTION

Condensation is the heat transfer process by which a saturated vapour is converted into a saturated liquid by means of removing the latent heat of condensation; it occurs when the enthalpy of the vapour is reduced to the state of saturated liquid.

Condensation occurs through one or a combination of four basic mechanisms of drop-wise, film-wise, direct contact, and homogeneous [1]. Drop-wise refers to a situation whereby the vapour condenses as liquid drops at particular nucleation sites on the cooling surface and remains as drops until carried away by gravity or vapour shear; in film-wise condensation, the liquid drops coalesce into a continuous thin liquid film; and in direct contact condensation, the vapour condenses directly onto the coolant liquid that has been sprayed into the vapour; whilst in homogenous condensation the super-saturated vapour condenses in space away from any macroscopic surfaces similar to the formation of fog possibly on dust or other particles acting as nucleation sites [2]; refer to figure 1.

The majority of industrial condensation processes are considered to be based on the film-wise mode of condensation. In this research work the study of the condenser is particularly directed towards the development of a low temperature solar thermal power plant as depicted in figure 2, below:



(a) drop-wise
(c) homogeneous
(b) film-wise
(d) direct-contact

Figure 1: four condensation mechanisms

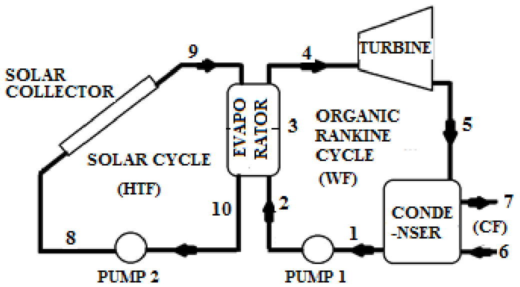


Figure 2: solar thermal power plant concept [3]
(HTF: heat transfer fluid; WF: working fluid; CF: cooling fluid)

II. THE THEORY OF CONDENSATION

The theory of film condensation based on the Nusselt (1916) integral model for film condensation on a vertical plate and later extended to cover condensation on a horizontal bundle of tubes has been well documented by Thome J.R. [1] and this theory forms the basis for the condenser model presented in this chapter; derivations of the formulae are considered outside the scope of this work and thus only the key equations are shown here.

Comparison of condensation heat transfer correlations for flow of refrigerants in horizontal smooth tube bundles has been presented by Sánta Róbert [4].

A detailed comparison of condensation heat transfer correlations for different flow configurations including flow inside and outside of horizontal, vertical and inclined tubes has also been presented by Fang Xiande et al [5].

Two correlations can be identified for flow outside a single horizontal tube and these are the Nusselt (1916) and the Dhir and Lienhard (1971) correlations; these are similar in every respect and only differ in the prefix multiplier constant (number):

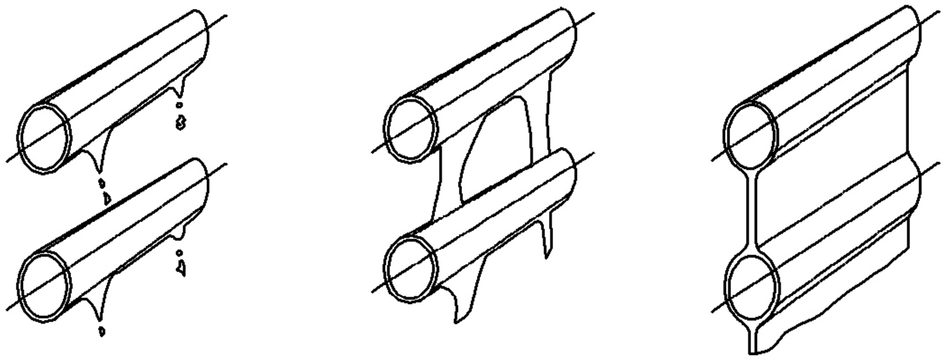
The correlation due to Nusselt (1916) is given as:

$$h=0.725\left[\frac{\rho_L(\rho_L-\rho_G)gh_{fg}k_L^3}{\mu_L D_o(T_{sat}-T_w)}\right]^{\frac{1}{4}} \tag{1}$$

The correlation given by Dhir and Lienhard (1971) proposed the following average heat transfer coefficient:

$$h=0.729\left[\frac{\rho_L(\rho_L-\rho_G)gh_{fg}k_L^3}{\mu_L D_o(T_{sat}-T_w)}\right]^{\frac{1}{4}} \tag{2}$$

The flow over a bundle is affected by condensate inundation; condensate inundation refers to the effect of condensate falling onto a tube from upper tubes in the tube bank. Condensate inundation modes include droplet, column and sheet; at higher inundation rates the inundation may progress into a spray mode which may involve side drainage, splashing and ripples on the lower tubes; see figure 3, below, [6].



(a) droplet (b) column (c) sheet

Figure 3: condensate inundation modes

Three correlations can be identified for flow outside a bundle of horizontal smooth tubes; these are Nusselt's (1949), Kern's (1958) and Eissenberg's (1972).

Nusselt's correlation gives the mean heat transfer coefficient as well as a coefficient for each of the tubes based on the coefficient for the first tube, as expressed by (1); the two equations for this correlation are as follows in (3) and (4):

$$\frac{\bar{h}}{h(N=1)}=N^{-\frac{1}{4}} \tag{3}$$

$$\frac{h(N)}{h(N=1)}=N^{\frac{3}{4}}-(N-1)^{\frac{3}{4}} \tag{4}$$

Incropera and DeWitt (2002) also proposed a similar average heat transfer coefficient for vertically aligned horizontal tube bundles; their correlation, however, suggests using the Dhir and Lienhard (1971) correlation for the reference heat transfer coefficient, [7].

The correlation due to Kern (1958) gives:

$$\frac{\bar{h}}{h(N=1)}=N^{-\frac{1}{6}} \tag{5}$$

$$\frac{h(N)}{h(N=1)}=N^{\frac{5}{6}}-(N-1)^{\frac{5}{6}} \tag{6}$$

The Butterworth and Eissenberg's (1972) correlations are only for the mean heat transfer coefficient and for a staggered arrangement of vertical tubes; the correlations are expressed as, [8]; [9]:

$$\frac{Nu}{Re_G^{1/2}}=0.416\left[1+\left(1+9.47\frac{gD\mu_L h_{fg}}{u_G^2 k_L(T_{sat}-T_w)}\right)^{0.5}\right]^{0.5} \tag{7}$$

$$\frac{\bar{h}}{h(N=1))}=0.6+0.42N^{-\frac{1}{4}} \tag{8}$$

Beatty and Katz (1948) correlation provides for condensation on low-finned tubes and their correlation together with supporting equations is given as: [5].

$$h=0.689F^{0.25}\left[\frac{A_r}{A}\frac{1}{D_r^{0.25}}+1.3\eta\frac{A_f}{A}\frac{1}{L_c^{0.25}}\right] \tag{9}$$

$$F=\left(\frac{\rho_L^2 gk_L^3 h_{fg}}{\mu_L(T_{sat}-T_w)}\right) \tag{10}$$

$$L_c=\frac{\pi(D_o^2-D_r^2)}{4D_o} \tag{11}$$

$$A = A_r + A_f(4d) \tag{12}$$

where η is the fin efficiency, A is the tube surface area at nominal diameter (m^2), A_r is the tube surface area at the base of the fins (m^2), A_f is the fin surface area (m^2), D_r is the diameter at fin root, F is a constant accounting for the effect of physical properties, and L_c is the characteristic length.

III. MATHEMATICAL MODELLING

The governing equations used in this model are as follows: The overall heat transfer coefficient for each tube is given by:

$$U[i]A_o = \frac{1}{\frac{1}{h_{CF}A_i} + \frac{R``_{fi}}{A_i} + \frac{\ln\left[\frac{D_o}{D_i}\right]}{2\pi k_W L_{tube}} + \frac{1}{h_{WF}[i]A_o} + \frac{R``_{fo}}{A_o}} \quad \text{for } i=1 \text{ to } N \tag{13}$$

where $h_{WF}[i]$ is determined from the correlations of section 3 and h_{CF} is determined from the Gnielinsk correlation given by:

$$\frac{h_{CF}D_i}{k_{CF}} = \frac{\frac{f_{CF}}{8}(ReD_{CF}-1000)Pr_{CF}}{1+12.7\left[\frac{f_{CF}}{8}\right]^{1/2}\left(Pr_{CF}^{2/3}-1\right)} \tag{14}$$

These are used to determine the heat transfer for each tube; total heat exchange is then given as:

$$Q_{COLUMN} = \sum_{i}^{N} Q[i] \tag{15}$$

$$Q_{TOTAL} = M * Q_{COLUMN}$$

Where N is the number of tubes in each column and M is the number of columns. The logarithimic mean temperature difference (LMTD) method is used together with the number of transfer units (NTU) method to determine the heat transfer for each tube.

The input conditions are taken as:

- The working fluid is any of: R123, R134a, R245fa, n-butane and isobutene;
- The cooling fluid is ethylene glycol at 50% concentration;
- T_{WF} = 50ºC as the condensate temperature; $T_{CF,i}$ = 25ºC as the cooling fluid inlet temperature;
- Copper tubing is used for the tube bundles: inside diameter 16.5 mm, outside diameter 19 mm and length 2.85 m

Equations that are also useful in formulating the mathematical models are those obtained from Nusselt's Integral method and may be expressed as:

$$\Gamma = 1.924\left[\frac{r^3 k_L{}^3 (T_{sat}-T_w)^3(\rho_L-\rho_G)g}{h_{fg}{}^3\frac{\mu_L}{\rho_L}}\right]^{1/4} \tag{16}$$

where Γ is the condensate mass flow rate on one side of the tube, per unit length of the tube, (kg/m.s); from this equation the overall heat transfer for a single tube can be expressed as:

$$\dot{Q} = 2\Gamma L_{tube} h_{fg} \tag{17}$$

where h_{fg} is the latent heat of condensation, (J/kg)

IV. COMPUTER SIMULATIONS

Four sets of simulations have been performed as follows:

i) The first simulations are based on Nusselt's correlations for the heat transfer coefficient for the first tube, for the mean heat transfer coefficient for a column of N tubes, and the heat transfer coefficients for tubes 2 up to N.
ii) The second set of results use Nusselt's correlations for the first tube; and then uses Kern's method for the mean heat transfer coefficient as well as for the tubes 2 to N.
iii) The third set of simulations uses the Dhir and Lienhard correlation for the first tube and then uses the Nusselt's correlations for the mean and for tubes 2 to N.
iv) The fourth set uses the Dhir and Lienhard correlation for the first tube and then uses the Kern's method for the mean and for tubes 2 to N.

Working fluids used are R245fa, R134a, R123, n-butane and isobutene. The results are compiled for each working fluid and compared on the basis of the correlations.

The simulations were done for only the 10 kWe concept cycle plant; sizing is based on the thermal load of 84.2 kW_{th} determined from cycle efficiencies in the range 10-15% obtained from the initial system model thus giving an average cycle efficiency of 12.5% [3].

V. RESULTS

- Condensate Heat Transfer Coefficients

The results for the condensate heat transfer coefficients based on different correlations are shown in the charts of figures 4 to 7.

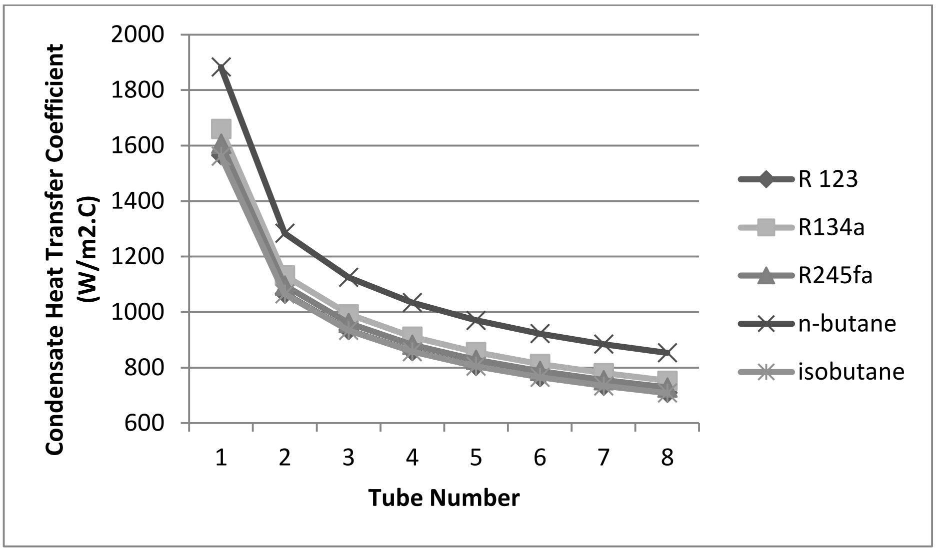


Figure 4: tube heat transfer coefficients based on Nusselts Correlations

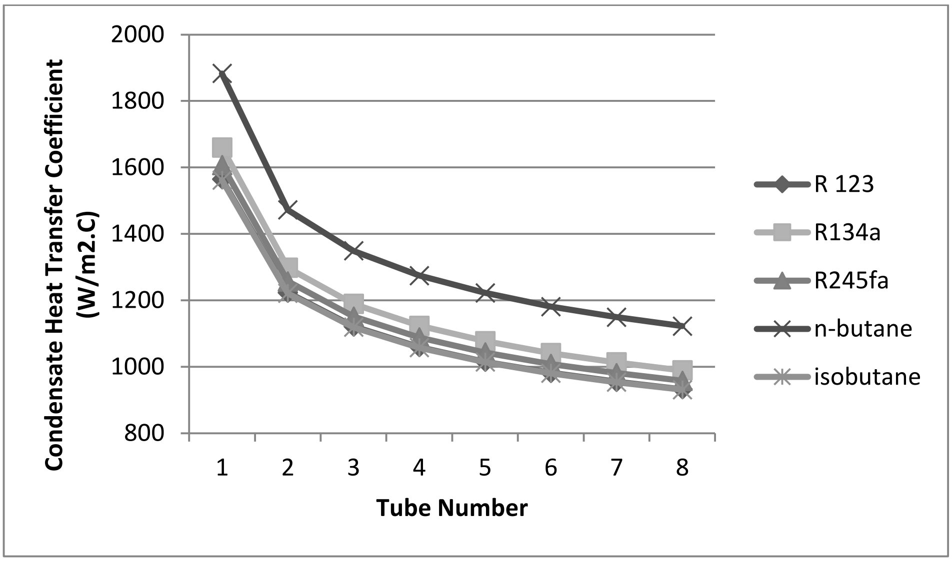


Figure 5: tube heat transfer coefficients based Nusselts and Kerns Correlations

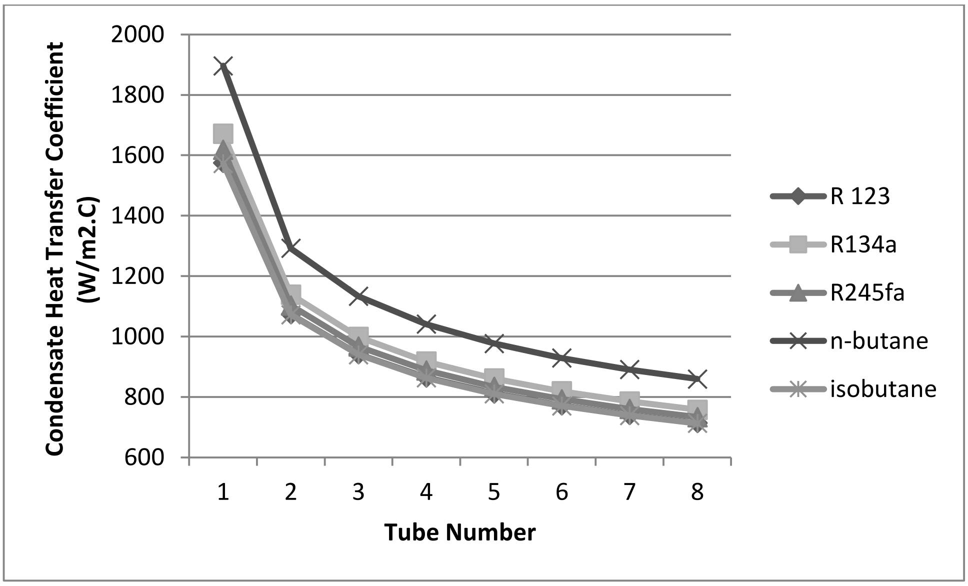


Figure 6: tube heat transfer coefficients based on Dhir and Lienhard, and Nusselts Correlations

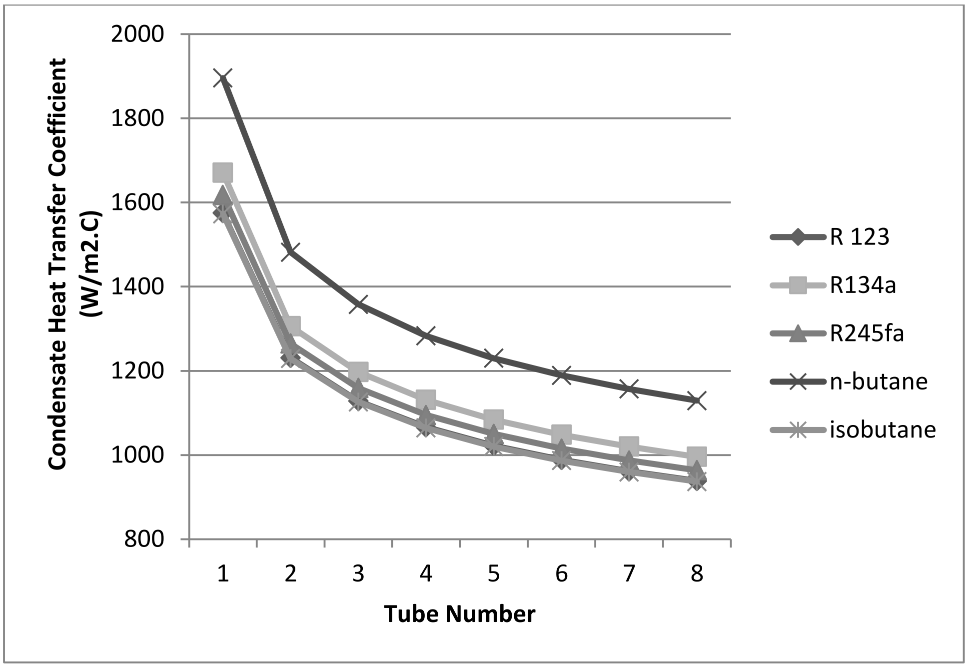


Figure 7: tube heat transfer coefficients based on Dhir and Lienhard, and Kerns Correlations

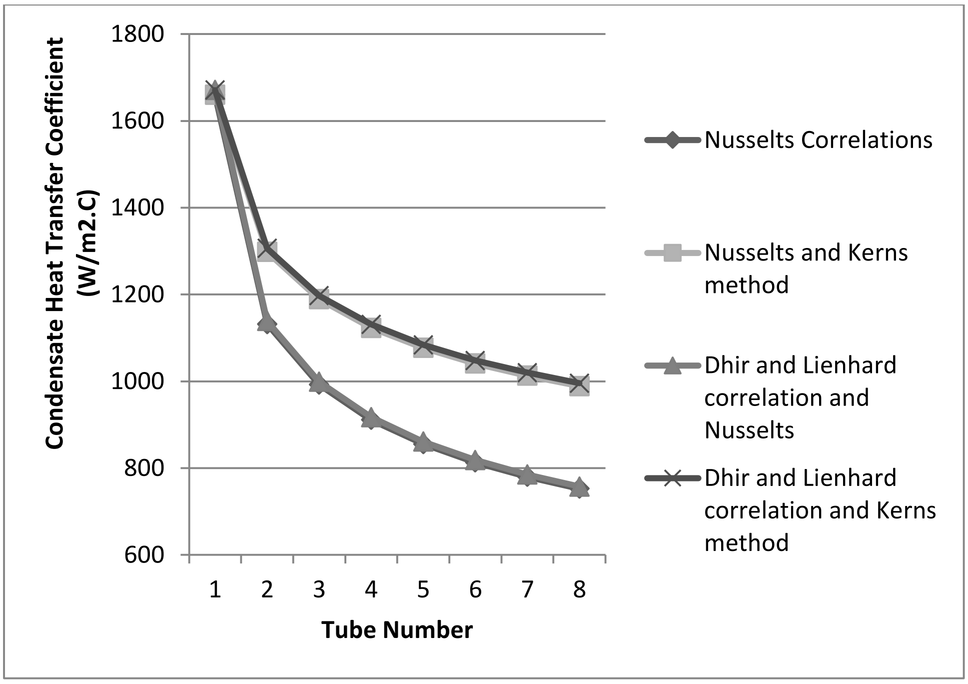


Figure 8: Condensate Heat Transfer Coefficient for R134a based on different combinations of Correlations

- Heat Transfer Rates

The total thermal loads per column data is captured for each working fluid and for each combination of correlations and the results are presented in the following table 1.

The overall heat exchanger thermal loads for each working fluid are shown in figure 9.

- Cooling Fluid Outlet Temperature

The average outlet temperatures of the ethylene glycol cooling fluid for each of the working fluids are shown in figure 10; the inlet temperature is assumed as 25ºC whilst the condensation temperature is taken as 50ºC.

TABLE 1: THERMAL LOADS PER COLUMN FOR EACH WORKING FLUID AND FOR EACH CORRELATIONS COMBINATION

WORKING FLUIDS	R 123	R134a	R245fa	n-butane	isobutane
CORRELATIONS	(W)	(W)	(W)	(W)	(W)
Nusselt's	14199	14585	14374	15402	14189
Nusselt's & Kerns	15470	15851	15643	16647	15461
Dhir and Lienhard & Nusselt's	14242	14628	14418	15444	14231
Dhir and Lienhard & Kern's	15512	15892	15685	16687	15502

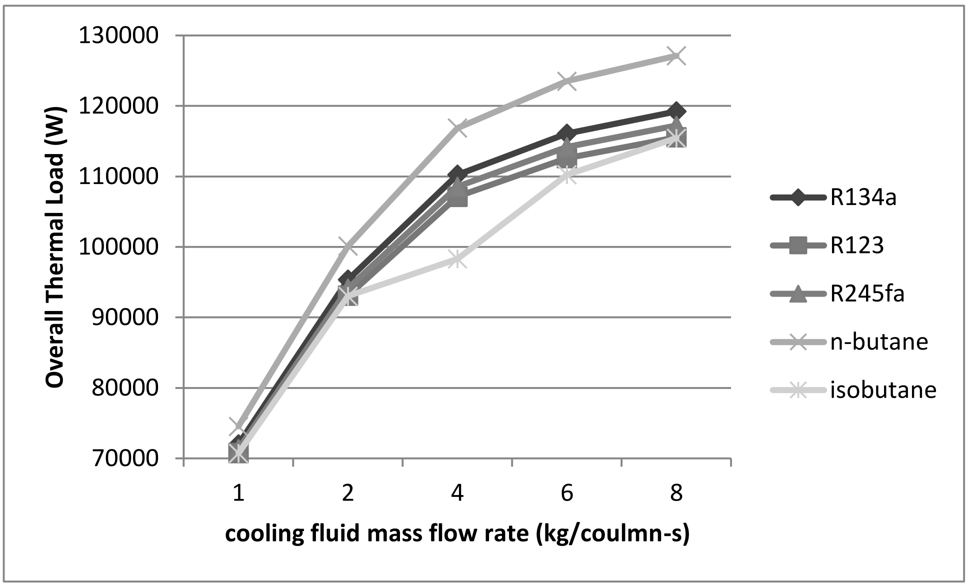


Figure 9: Overall Thermal Load versus cooling fluid column mass flow rate
(based on Dhir and Lienhard, and Kerns Correlations)

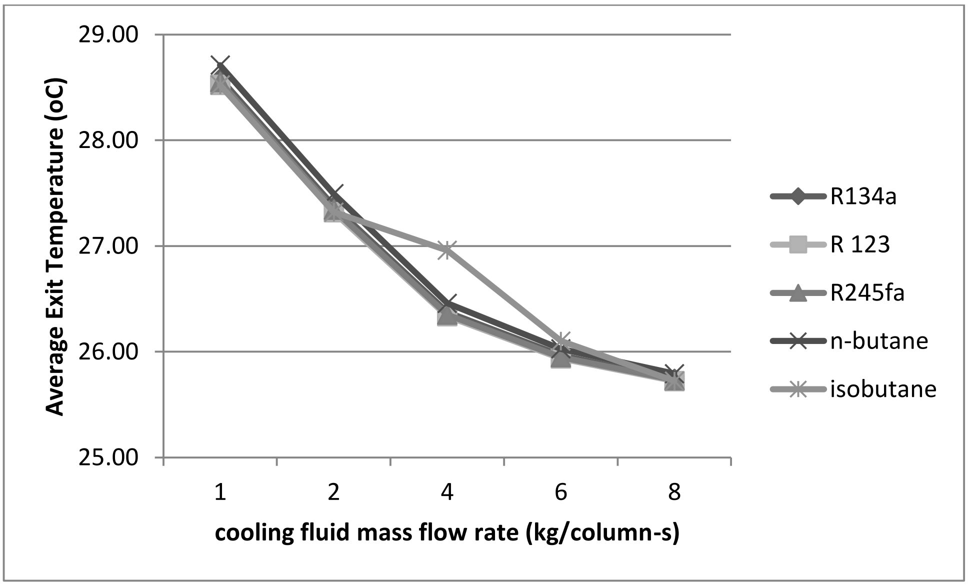


Figure 10: average cooling fluid exit temperature versus cooling fluid column mass flow rate
(based on Dhir and Lienhard, and Kerns Correlations)

VI. DISCUSSIONS

Figure 4 to 7 show that for all combinations of correlations two observations can be made: condensate heat transfer coefficient varies from a high on the first tube to a minimum on the last tube; and condensate heat transfer coefficients are highest with n-butane, followed by R134a, R245fa, and are lowest for R123 and isobutene; plots of values for the latter two appear superimposed.

Figure 8 shows that Kerns method gives higher values of the condensate heat transfer coefficient, varying from about 900 to 1900 $W/m^2.^oC$, as compared to the Nusselts correlation, where corresponding values vary from about 700 to 1900 $W/m^2.^oC$. The heat transfer coefficient for the first tube is almost the same regardless of whether the Nusselts or the Dhir and Lienhard correlation is used.

In terms of thermal loads, table 1 and figure 9 show that n-butane gives the highest value followed by R134a, R245fa, and R123, with the lowest being obtained with isobutene.

The average outlet temperatures of the ethylene glycol cooling fluid for each of the working fluids, figure 10, show a continuous decline with increases in cooling fluid mass flow rate; also n-butane gives the highest outlet temperatures; all the other fluids show superimposed plots of the outlet temperatures; outlet temperatures vary from a maximum of about 28.5^{o}C to a minimum of about 25.8^{o}C.

VII. CONCLUSIONS

The paper has presented a condenser heat exchanger model suitable for incorporation into a low temperature solar thermal power cycle. The model consists of a flow of vapour over a bundle of horizontal tubes. The simulations have shown the effect of condensate inundation in reducing the heat transfer capacity of tubes low down in a column of horizontal tubes. The simulations have also shown that Nusselts correlations give more conservative values when compared to the Kerns method. All the working fluids depict similar device thermal exchange characteristics; n-butane performs better than the other fluids whilst R123 and isobutene give the worst performance. Sizing of the heat exchanger is determined by the number of tube rows and columns; in this case, the simulations showed that 8 rows by 6 columns would be adequate for the given heat exchanger configuration and thermal load. These results are to be compared with the results of the on-going experiments.

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