FUNDAMENTAL BASIS AND IMPLEMENTATION OF SHELL AND TUBE HEAT EXCHANGER PROJECT DESIGN: EVAPORATOR STUDY

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ABSTRACT
A shell and tube heat exchanger is used as an evaporator in this theoretical study. Parametric performance analyses for various actual refrigerants were performed using well-known correlations in open sources. Condensation and evaporation were occurred in the shell side while the water was flowing in the tube side of heat exchanger. Heat transfer rate from tube side was kept constant for evaporator design. Evaporating temperatures were ranging from -15 °C to 10 °C for the refrigerants of R12, R22, R134a, R32, R507A, R404A, R502, R407C, R152A, R410A and R1234ZE. Variation of convective heat transfer coefficients of refrigerants, total heat transfer coefficients with Reynolds numbers and saturation temperatures were given as validation process considering not only fouling resistance and omission of it but also staggered (triangular) and line (square) arrangements. The minimum tube lengths and necessary pumping powers were calculated and given as case studies for the investigated refrigerants and their determination with regard to cost and economy.

KEYWORDS- Heat exchangers, Shell and tube, Evaporator, Alternative refrigerants.

I. INTRODUCTION
The importance of especially shell and tube heat exchangers has ascended in terms of energy recovery, conservation, conversion, and effective utilization of novel energy potentials in today's world. In addition to this, it has also become crucial with regard to the point of modern environmental problems for instance, the pollution of air, water, and waste removal. Heat exchangers are widely employed in the, cooling, heating, air conditioning, recovery of waste heat, cooling towers, thermal and nuclear plants and chemical processes [1-4]. Also, they are base elements of various industrial goods which are found in the marketplace. There are many studies over the enhancement methods of different heat exchangers in the literature because of their significance in industry as well as public life. Besides, the present paper has come from the thought of limited numbers of studies have taken place in the literature that present comprehensive analysis regarding the issue of different refrigerants in shell and tube heat exchangers and their determination with regard to cost and economy.

Salimpour [5] conducted an experimental research to examine the heat transfer behavior of engine-oil inside shell and tube heat exchangers. Three heat exchangers that have different coil pitches have been chosen as the experimental setup. In the experimental implementation, as engine-oil flows inside the inner coiled pipe, coolant water passes through the shell.

Benefitting from the data acquired through the study, a correlation has been derived to calculate the heat transfer coefficients of the shell and coiled tube heat exchangers.

Garcia et al. [6] suggested a simplified model to use for the examination of shell and tube heat exchangers. The model they suggested purposes to be in accordance with the shell and tube heat exchangers while they're being used as condensers and evaporators. Though the model seems simple it is substantially useful in terms of
designation and appropriate selection of shell and tube heat exchangers. Additionally, the model has been validated in the modeling of a refrigeration cycle and the results were compared with the experimental setup.

Lee et al. [7] carried out an experimental test of a mixed refrigerant R407C in a chiller with shell and tube heat exchangers that are in fact designed for R22. A drastic decrease in the performance has been observed when R22 was replaced with R407. The cause of the decrease in the performance was thoroughly examined and the main factor induced the decrease performance was evaluated as the deterioration of the heat transfer in utilizing the mixed refrigerant, R407C. Furthermore, evaporator has been found as the most deteriorated element in terms of heat transfer and the element most worsens the performance of the chiller.

Assaf et al. [8] found a new method to design the dry-expansion shell and U-tube evaporators. They used Modelica language which provides a model with a general flow arrangement. The authors validated the model using a commonly seen shell and tube evaporator employed with R134a. Pure R134a, R407C and a special chosen glide matching refrigerant were flowed. As result, it was seen that the influence of temperature profile of refrigerant mixtures can be considerable on the relative performance of a specific heat exchanger configuration.

Abed et al. [9] reviewed the improvement techniques as well as falling film flow, particularly the impact of nanoparticles that took place in refrigerants. The review study also includes effect of geometry of surfaces, low fins, developed geometrical tubes and problems on refrigerant-based nanofluids. It has been concluded that the interaction between heat and mass transfer on falling film flow and disagreements of thermal properties of nanofluids should be regarded as well. As result, the study shed light on the factors that influence effectiveness, compactness and cost of spray evaporators as well as available improvement methods.

Hosseini et al. [10] experimentally investigated the heat transfer coefficient and pressure drop on the shell side of a shell and tube heat exchanger for different types of copper pipes. Novel correlations for both pressure drop and Nusselt number were explored valid for different tube types (smooth, corrugated, and with microfin). The tube groups that have the identical geometric and structural arrangements except different external tube surfaces inside the shell were utilized for the experimental work. In conclusion, corrugated and micro-fin tubes have revealed deterioration of efficiency at a Reynolds number field (Re < 400). Also, at a greater Reynolds number for micro-finned tubes the effectiveness of the heat exchanger has shown considerable enhancement.

Although there has been quite number of studies on shell and tube heat exchangers that perform various refrigerants in the literature, the current study presents the behavior of various refrigerants with different parameters, when they are used in both condensers and evaporators. The impact of the utilization of different refrigerants on convective heat transfer coefficient on shell-side, the variation of this coefficient with Reynolds number, the effect of evaporation on total heat transfer coefficient under the triangular or square pitch arrangements in tube-side and under the conditions of whether the fouling resistance is encompassed in the calculations, have been investigated. Besides, the influence of the same performed refrigerants on tube length under the evaporation condition when different evaporating temperatures and triangular and square pitch alignments were applied and according to whether the fouling resistance was considered, was examined. Also, the required pumping power for the abovementioned conditions was determined at varying Reynolds numbers for evaporation.

II. CALCULATIONS OF THE DESIGN PROCEDURE FOR THE INVESTIGATED SHELL AND TUBE EVAPORATOR
In the case studies, different refrigerants are evaporated at varying temperature intervals within the range of -15°C to 10 °C through the heat transfer by a cooling water received from a mini cooling tower at 25°C. A shell and tube heat exchanger, shown in Figs 1 and 2, with 4 pass designs (Np) on the water side is employed in the simulations. Copper tubes with the inside and outside diameters of 0.012573/0.01588 m (di/do) respectively have been selected owing to the accordance with the refrigerants. The design stage of a shell and tube heat exchanger for a 91.92 kW evaporating capacity and either including the fouling resistance or not including and triangular or square pitch arrangements has been presented in the following sections.

**A.1 Tube-side calculations**

In the conditions of working as an evaporator the heat amount that has been transferred through the tube-side is the same. Here, it should be stated that in all case studies the fluid used in tube-side is water.

The bulk temperature of water is found through Eq. (1) to calculate the thermo-physical properties:

\[ T_{\text{bulk}} = \frac{T_{c1} + T_{c2}}{2} \]  

The flow area of the cold-side is obtained from Eq. (2):

\[ A_{cs} = \pi d_i \]  

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The flow velocity of water (U) is assumed as constant (1.5 m s\(^{-1}\)); and thus, the mass flux is acquired by Eq. (3):

\[ G = \rho_w U = \frac{U_w}{U} \]  

The flow rate of cooling water in each tube is determined as follows:

\[ m_w = GA_{cs} \]  

The coolant flow rate (m\(_c\)) is taken from the heat rejection duty (Q\(_c\)) as follows:

\[ Q_c = m_c h_c \]  

The number of tubes (N\(_t\)) are found by Eq. (6). Besides, it should be stated that the rounding process can be employed to the number of tubes according to its calculated value.

\[ m_c = m_t N_t / N_p \]  

The Reynolds number of water flowing through the tubes has been determined in Eq. (7). Its values were calculated higher than 10,000 for all the case studies, hence the behavior of the flow was determined as turbulent. Eq. (8) has been suggested for Nusselt number by Gnielinski [49] for turbulent flow behavior.

\[ \text{Re}_t = \frac{Gd_i}{\mu_w} \]  

\[ \text{Nu} = \frac{\left( \frac{f}{2} \right) (\text{Re}_t - 1000) \text{Pr}}{1 + 12.7 \left( \frac{f}{2} \right)^{1/2} (\text{Pr}^{1/3} - 1)} \]  

as the fanning friction factor is shown as follows:

\[ f = \left[ 1.58 \ln(\text{Re}_t) - 3.28 \right]^{2} \]  

The convective heat transfer coefficient of water (h\(_c\)) is taken from the Nusselt number expression:

\[ \text{Nu} = \frac{h_c d_i}{k_i} \]  

**A.2 Shell-side calculations**

**A.2.1 Evaporator study**

From energy balance and as result of definite values on both-shell side and tube-side, evaporation mass flow in shell-side is found in Eqs. (11-12).
Heat flux is equal to the ratio of net energy to the total pipe surface area and shown in Eq. (17).

\[
Q_c = \frac{q}{A_{\text{tube surface}}}
\]  

(11)

\[
m_c \cdot \Delta h_c = m_R \Delta h_R
\]  

(12)

For determining the heat flux on shell-side, pipe surface area is calculated through Eq. (13)

\[
A_{\text{tube surface}} = N \pi D_o l
\]  

(13)

Due to the square alignment of pipes, hydraulic diameter is calculated in Eq. (14).

\[
D_{\text{square}} = \sqrt{\frac{4 \left( Pr^2 - \frac{\pi}{4} D_o ^2 \right)}{N \pi D_o ^2}}
\]  

(14)

Also, due to the triangle alignment of pipes, hydraulic diameter is calculated in Eq. (15).

\[
D_{\text{triangular}} = \sqrt{\frac{4 \left( Pr^2 - \frac{\pi}{4} D_o ^2 \right)}{2 N \pi D_o ^2}}
\]  

(15)

For the mass flow calculation on shell-side, cross-area is calculated by Eq. (16).

\[
A_{\text{crossing}} = \frac{\pi}{4} (Pr^2 - N \pi D_o ^2)
\]  

(16)

Heat flux is equal to the ratio of net energy to the total pipe surface area and shown in Eq. (17).

\[
q = \frac{Q}{A_{\text{crossing}}}
\]  

(17)

Mass flux is the proportion of the mass flow of vapor to the cross area and shown in Eq. (18).

\[
G = \frac{m_v}{A_{\text{crossing}}}
\]  

(18)

Boiling number is equal to the ratio of the heat flux to the product of latent heat coefficient and mass flux as expressed in Eq. (19).

\[
Bo = \frac{q}{G \cdot \Delta h_L G}
\]  

(19)

Also, Reynolds number is found from the proportion of the product of mass flux and hydraulic diameter to viscosity by Eq. (20):

\[
Re = \frac{G \cdot D_e}{\mu}
\]  

(20)

Prandtl number is determined by the proportion of the product of viscosity and specific heat to conductive heat transfer coefficient and shown in Eq. (21)

\[
Pr = \frac{\mu c_p}{\kappa}
\]  

(21)

Nusselt number is obtained as follows, in Eq. (22):

\[
Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4}
\]  

(22)

From Eq. (23) convective heat transfer coefficient is calculated:

\[
h_r = Nu \cdot \frac{k}{De}
\]  

(23)

By incorporating boiling, the convective heat transfer coefficient on the shell-side is determined from Eq. (24):

\[
h_r = h_{r,s} \cdot [4.3 + 0.4(Bo + 10^4)^{1.3}]
\]  

(24)

For clean surfaces the overall heat transfer coefficient is obtained according to Eq. (25) as follows:

\[
\frac{1}{U_{O,-\text{clean}}} = \frac{1}{h_r} + \frac{D_o}{D_t} \cdot h_l + \frac{r_o \cdot \ln(D_o/D_t)}{k}
\]  

(25)

By inserting the dirt factor into the equation, the overall heat transfer coefficient for dirty surfaces is found by Eq. (26):

\[
\frac{1}{U_{O,-\text{dirty}}} = \frac{1}{U_{\text{clean}}} + R_{Rs}
\]  

(26)

The fluid outlet temperature of the fluid on shell side is found as follows

\[
T_{c2} = \frac{(m \cdot c_p)_{h} \cdot (T_{h2} - T_{h1})}{(m \cdot c_p)_{c}} + T_{c1}
\]  

(27)

Average logarithmic temperature is determined by Eq. (28):

\[
\text{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})} = \frac{(T_{h} - T_{c1}) - (T_{h} - T_{c2})}{\ln(\frac{T_{h} - T_{c1}}{T_{h} - T_{c2}})}
\]  

(28)

Shell-side clean and dirty surface areas are found through the Eq. (29) as follows:

\[
A_o = \frac{Q}{U_{O \cdot \text{LMTD}}}
\]  

(29)

Similarly, dirty and clean pipe lengths are calculated as follows:
The calculations have been made by taking square and triangular alignments into account. The 4-pass system has had 96 pipes. For both evaporation and condensation, this has been valid, as shown in Figs. (1-2), due to the fact that tube-side has been accepted as the same in both two systems.

A.3 In tube pressure drop

The pressure drop at the water-side for the whole pipes is found by means of Eq. (31) as follows:

\[ \Delta P_t = \frac{4L N_t G^2 \nu_t}{2d_t} \]  

as the friction factor is obtained through Eq. (9) for the a turbulent flow phenomena in the tubes.

Besides, an additional loss develops in the fluid owing to the rapid expansions and contractions during a return [52].

\[ \Delta P_r = \frac{4N_r \rho u_m^2}{2} \]  

The total pressure drop is calculated as the sum of the pressure losses in the tube (Eq. 31) and due to return losses (Eq. 32).

\[ \Delta P_t = \Delta P_t + \Delta P_r \]  

The pumping power the fluid is proportional to the pressure drop in the fluid across a heat exchanger and given in Eq. (34) as follows:

\[ W_p = \frac{m \Delta P}{\rho \eta_p} \]  

while \( \eta_p \) is the pump efficiency and accepted as 0.85.

In addition, it should be stated that the details of this calculations are found in Kakac [11].

III. RESULTS AND DISCUSSION

The costs of evaporators, condensers, and other heat exchangers compose of nearly 30% of the overall cost of a thermal power plant. These costs are important in applications in which heat exchangers must have very large surface areas because of low logarithmic mean temperature differences. Many techniques have been examined in recent years to enhance heat transfer to diminish the sizes and costs of the heat exchangers as a solution for the cost problem.

Given that the above mentioned advances, in the present paper, a simple analysis is given in terms of the use of alternative refrigerants with the variation of many parameters including the fouling resistance and square or triangular pitch arrangements. Fouling is a broad term which covers any type of deposit materials that result in as a resistance on the heat transfer surfaces of a heat exchanger. By its nature, a resistance on heat transfer surfaces leads to deterioration of the capability of the heat exchanger. Therefore, in the designing stage of a heat exchanger, the designer must consider the effect of fouling on performance and to see the influence of fouling on performance, case studies that take fouling into account and not taking into account were conducted in this study. It is clear through the literature that square pitch arrangements of the pipes in heat exchangers permit easy cleaning, however the heat transfer performance is not as good as triangular pitch arrangements, due to the fact that much less tubes can be laid. From this point of view, to see the effect of square and triangular pitch arrangements in different working conditions, these alignments were applied to both evaporation and condensation case studies.

R12, R22, R134a, R32, R507a, R404a, R502, R407c, R152a, R410a and R1234ze were used as shell-side condensing and evaporating refrigerants. The evaporation temperature of the refrigerants was taken within the range of (-15 °C to 10 °C) for evaporation, and the condensing temperatures were varied from 35 °C to 60 °C, and the water-side conditions were the same for all investigated conditions. Also, it should be noted that all calculations were realized for 4 pipe pass arrangement for a constant cooling load of 91.92 kW.
Fig. 3 Alteration of shell side heat transfer coefficients of investigated refrigerants with arrangement types according to evaporation situation in the shell side

Fig. 3 gives the convective heat transfer coefficients at shell sides of condensation and evaporation, respectively. According to the figures, R32 usage at the square pitch alignment of 35 °C for condensation temperature and -15 °C evaporation temperature results in the highest level of convective heat transfer coefficient in condensation and evaporation, respectively. Also, it is seen through these figures that R507a at the square pitch alignment and at 60 °C temperature for condensation and R12 at the triangular pitch alignment and at 10 °C temperature for evaporation lead to least convective heat transfer coefficients.

Figs. 4a and b illustrate the change of shell-side convective heat transfer coefficients with Reynolds numbers at the shell-side of exchangers for various refrigerants that flow in evaporators and triangular and square arrangements, respectively. The points on these figures were calculated through varying the evaporation temperature of the shell-side within the range of (-15 to 10 °C). In line with the figures, the convective heat transfer coefficient decreases with increasing Reynolds numbers for both pitch arrangements. Furthermore, it should be stated that all the case studies have had the same thermal capacity and the fluid flowing in the tubes is water at the constant velocity (uc) value of 1.5 m/s.

In Figs. 5a and b, the total heat transfer coefficients for the omission of fouling resistance and at various evaporation temperatures (T_{evap}= (-15 °C)-(+10 °C) of different refrigerants were shown under the conditions of triangular and square pipe arrangements, respectively. Although the no significant difference was observed among the total heat transfer coefficients, highest level was obtained at R410a for both square and triangular arrangements, while the lowest level was observed for R152a. According to the results, taking the fouling resistance into account in evaporation becomes more influential on total heat
transfer coefficient by around 85%.

In Fig. 6, the required tube lengths at the same load amounts for evaporator at different evaporation temperatures (10 °C and -15 °C) and for square and triangular tube alignments have been thoroughly demonstrated. It is clear that in evaporation, square or triangular alignment does not effect on tube lengths remarkably. Also, in evaporation mode fouling resistance has increased the tube lengths by 70%. Additionally, for evaporation, the longest and shortest tube lengths are found as at the square and dirty arrangements (T_{evap}= 10 °C) for R152a (L=0.59 m) and at the triangular and clean arrangements (Tc= -15 °C) for R507a (L= 0.09 m), respectively.

In Fig. 7, the variation of pumping power for pipes including fouling resistance and for different refrigerants at varying evaporation temperatures ranging from -15 °C to 10 °C and triangular and square pipe arrangements has been demonstrated. It is observed that as shell-side Reynolds number increases, pumping power also increases and the most dependent on and most independent of shell-side Reynolds numbers with respect to pumping power are R507a and R152a, respectively.

IV. CONCLUSION

The analyses help researchers on the selection of parameters of the evaporators such as their basic geometries, flow rates, heat transfer coefficients, pressure drops, and pumping powers to increase heat transfer rate, reduce energy consumption and heat exchangers’ sizes according to alternative refrigerants for a constant heat duty. Selection of refrigerant type is found to be a significant criteria regarding the determination of these parameters. Some of new refrigerants should be considered as a promising one in comparison to the old ones according to the results. In spite of their good heat transfer and low energy consumption results, all investigated refrigerants’ chemical compositions will have changed in the near future due to the necessity of zero global warming potentials and zero ozone depleting potentials in the world. Their negative characteristics, such as toxicity, flammability, and oil miscibility, will still keep their importance. Consequently, works on this subject will continue in the near future because of the
problems of global warming and ozone depletion. Particularly, the usage of hydro fluorocarbon (HCF) blends involve larger care caused by the alteration in their physical specifications in the event of explosion as a result of leakage.

ACKNOWLEDGEMENTS

The authors are indebted to the Department of Mechanical Engineering, King Mongkut’s University of Technology Thonburi (KMUTT), the Thailand Research Fund and the National Research University Project for supporting this study. Especially, the first author wishes to thank KMUTT for providing him with a Post-doctoral fellowship.

Nomenclature

A inside surface area, m²
CFC chlorouorocarbon
d diameter, m
f friction factor
F correction factor
G mass flux, kg m⁻²s⁻¹
GWP global warming potential
h convective heat transfer coefficient, W m⁻²K⁻¹
HC hydrocarbon
HCFC hydrochlorouorocarbons
L tube length, m
LMTD logarithmic mean temperature difference
m flow rate, kg s⁻¹
N number of tubes
ODP ozone depleting potential
r tube radius, m
P pitch, m
R resistance, m²K W⁻¹
Q heat rejection duty, kW
U overall heat transfer coefficient, W m⁻²K⁻¹
U∞ water flow velocity, m s⁻¹
Δh enthalpy of phase change, kJ kg⁻¹
ΔP pressure drop, Pa
ΔTm mean temperature difference

Greek symbols

ρ density, kg m⁻³
u specific volume, m³ kg⁻¹
ν kinematic viscosity, m² s⁻¹
ηp pump efficiency

Subscripts

c cold
C clean
cs cold surface
D dirty
f fouling
g gas
h hot
i inside
l liquid
o outside
m mean
p pass
r return
s shell
sat saturation
T total
t tube
w water

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