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THE USE OF SPIRALLY CORRUGATED TUBES WITH TWISTED TAPE INSERTS IN HORIZONTAL SHELL AND TUBE CONDENSER

Valentin Petkov*

Technical University of Gabrovo, 5300 Gabrovo, Bulgaria

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ABSTRACT

Performance evaluation criteria have been used to evaluate the merits of implementation of spirally corrugated tubes with twisted tape inserts in an real working condenser for different geometrical and operational constraints. A special attention has been taken into account to the calculation of the outside heat transfer coefficient and its variation according to the height of the tube bundle. Substantial decrease in the heat transfer area or increase in the heat transfer rate can be achieved when compound heat transfer enhancement is used.

NOMENCLATURE

D	tube diameter, m
L	tube length, m
$\dot{Q}$	heat transfer rate, W
P	pumping power, W
W	mass flow rate, kg/s
T	temperature, K
ΔT	temperature difference, K
A	heat transfer surface area, m^2
U	overall heat transfer coefficient, W/(m^{2}K)
H	360 deg twist pitch, m
h	specific entalpy, J/kg
g	gravity acceleration, m/s^2
$\dot{m}$	mass flow rate per tube, kg/s

Dimensionless groups

f	Fanning friction factor
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
x	steam quality
Λ	property index of fluid
E	heat transfer enhancement ratio
N	number of tubes in the bundle
N_S	augmentation entropy generation number $N_S = \dot{S}_{g,a} / \dot{S}_{g,s}$
A_*	ratio of heat transfer surfaces, $A_* = A_a / A_s$
D_*	ratio of tube diameters, $D_* = D_a / D_s$
L_*	ratio of tube lengths, $L_* = L_a / L_s$
N_*	ratio of number of tubes, $N_* = N_a / N_s$
P_*	ratio of pumping powers, $P_* = P_a / P_s$
Q_*	ratio of heat transfer rates, $Q_* = \dot{Q}_a / \dot{Q}_s$
W_*	ratio of mass flow rates, $W_* = W_a / W_s$
ΔT_m^*	ratio of mean temperature differences $\Delta T_m^* = \Delta T_{m,a} / \Delta T_{m,s}$

Greek symbols

α	heat transfer coefficient, W/(m^{2}K)
λ	thermal conductivity, W/(m K)
ρ	fluid density, kg/m^3
μ	dynamic viscosity, kg/(s m)
ν	kinematic viscosity, m^2/s

Subscripts

a	augmented
i	inside
o	outside
s	smooth
w	wall
l	liquid
g	steam
m	mean value

INTRODUCTION

The previous studies [1-3] revealed that the used of compound heat transfer enhancement technique as spirally corrugated tubes with twisted tape inserts can be very attractive for implementation in the condensers. Recent studies [4, 5] have presented the thermo-hydrodynamic characteristics of two spirally corrugated tubes (SCT) combined with four twisted tapes with different geometrical parameters. The benefits have been assessed using different

* Corresponding author. E-mail: petkov_v@tugab.bg

criteria for effectiveness when this augmentation technique is implemented for water heaters or condensers [6].

However, these characteristics and estimates are based on studies related to condensation of water steam on a horizontally placed single tube. The behavior of a particular tube in the tube bundle of the condenser (water heater) strongly depends on the parameters of the two-phase flow that reaches it, which can significantly affect the characteristics of the tubes located below it in the bundle [7].

The aim of this work is to evaluate the actual benefit of a real horizontal condenser in which smooth tubes are replaced by tubes with spirally corrugated tubes combined with twisted tape inserts, using some of the criteria developed by Webb [6]. Particular attention has been paid to determine the change of the heat transfer of the condensing steam along the height of the tube bundle.

EVALUATION METHODOLOGY

As evaluation methodology for calculation of the heat transfer coefficient of the condensing steam along the bundle height, the method described in [8] has been expanded and further developed.

The methodology refers to the thermo-hydrodynamic calculation of tubular horizontal condenser (water heater), for which the cooling fluid moves in the tubes, and the pure steam - across the tube bundle in a downward flow in one pass. It evaluates the local values of the steam pressure gradient in the shell and the heat transfer coefficient of condensation for different values of the steam quality, x.

The first calculations are combined with the pressure drop in the inlet manifold to obtain the total steam pressure drop. The next are combined with the heat transfer coefficient on the coolant side, the thermal resistance of the wall and the pollution layer on both sides to obtain the local value of the overall heat transfer coefficients, U.

The developed methodology can be used to solve the following four types of tasks [8]:

1. Verification calculation of a given structure of the condenser for the specified purpose - the required heat exchange area is calculated and compared with the available one;
2. Design calculation of a condenser based on a specific structure and through an iterative procedure to meet the specified limits.
3. Determining the output parameters of a condenser with certain dimensions at a given mass flow rates and inlet temperatures of the fluids.
4. Evaluation of the results of testing a working condenser and specifying the sum of the thermal resistances from pollution.

The algorithm developed in [8] gives particularly good results when the volumetric flow rate of the steam is much higher than that of the cooling fluid and even small values of the pressure drop in the intertube space are significant.

In the horizontal shell-and-tube condenser the steam velocity creates tangential stresses in the flowing condensate, reduces the thickness of its layer and increases the heat flow in the top row of bundle tubes. With each subsequent row down, this effect decreases due to the increasing flow of condensate on each subsequent row.

The Nusselt equation for the heat transfer coefficient, α_o, in the case of steam condensation on the outside of a horizontal tube [8] is

$$\alpha_o = 0.729\left(\frac{\lambda_l^3 h_{lg} g \rho_l}{D_o \nu_l \left(T_g - T_w\right)}\right)^{1/4} \tag{1}$$

After converting, Eq. (1) yields the type used by [8]

$$\alpha_o = 0.959\lambda_l \left(\frac{\rho_l^2 g L}{\mu_l \dot{m}_l}\right)^{1/3}. \tag{2}$$

Eq. (2) is valid for cases where the condensate flows under the action of gravity, the steam velocity is negligible and does not affect the condensate layer, and the wall surface temperature remains constant.

When the influence of gravitational force on the flowing condensate is insignificant compared to that of the steam velocity on the process, the value of the heat transfer coefficient can be correlated by $\alpha_o = \alpha_{sh}$ [8], where

$$\alpha_{sh} = 1.26\alpha_l \left(\frac{x_g}{1-x_g}\right)^{0.702} \Lambda^{-0,390}. \tag{3}$$

Here α_l is the heat transfer coefficient of the liquid phase of the two-phase flow and it is determined [8] by

$$\alpha_l = \frac{\lambda_l}{D_o} a\left(Re_{m,l}\right)^m \left(\frac{\mu_l c_{p,l}}{\lambda_l}\right)^{0.34} \tag{4}$$

where x_g is the steam quality and Λ is the property index, which is defined as the ratio of the pressure gradient with all liquid flow to that with all steam flow. If the flow is of liquid and steam phase only, it is given by [8],

$$\Lambda = \frac{\rho_g}{\rho_l}\left(\frac{\mu_l}{\mu_g}\right)^{0.2}. \tag{5}$$

The constants a and m in Eq. (4) depend on the Reynolds value of the condensate $Re_{m,l}$ and the tube-layout angle in the bundle ϑ, and can be found in [8].

When the steam velocity is negligible and the flow of condensate is entirely due to gravity, this is denoted by α_{gr}. For the top row of tubes there is no inundation, and α_{gr} can be calculated from Nusselt's theoretical equation, Eq. (2), which is transformed as

$$\alpha_{gr,1} = \Phi_l \Gamma_1^{-1/3} \tag{6}$$

where

$$\Phi_l = 0.959\lambda_l \left(\frac{\rho_l^2 g}{\mu_l}\right)^{1/3}. \tag{7}$$

$\Gamma_1 = \dot{m}_{l,1}/L$ is the mass flow of the condensate per unit length, for each tube of the first row of the bundle. The influence of the flowing condensate on the following down rows is done by correcting the Eq. (6) [9]

$$\alpha_{gr,k} = \Phi_l \Gamma_k^{-0.110}\left(\sum_1^k \Gamma_i\right)^{-0.223} \tag{8}$$

where Γ_i is the mass flow rate of condensate per unit length of the row i and Γ_k - that of the row k. It is known from experience that SCT have higher α_{gr} than smooth tube due to the better drainage of the flowing condensate. When the smooth tubes are replaced with SCTs, this effect can be modeled, as the values of α_{gr}, calculated by Eq. (8), are multiplied by the coefficient of external enhancement, E_o. For the top row of the bundle α_{gr} will be higher than that calculated by Eq. (6) due to the effect of steam velocity, such as $\alpha_{sh} > \alpha_{gr}$. For the lowest row of the bundle $\alpha_{gr} >> \alpha_{sh}$ and $\alpha_o = \alpha_{gr}$, which is lower than that one calculated by Eq. (6), due to the effect of the condensate flowing. The combined effect of these effects on the bundle height is determined [8] by

$$\alpha_{o,k} = \sqrt{\left(\alpha_{sh,k}^2 + \alpha_{gr,k}^2\right)} . \tag{9}$$

The heat transfer coefficient of the cooling fluid α_i in smooth tubes is determined by Gnielinski [11], namely

$$Nu_i = 0.012\left(Re_i^{0.87} - 280\right)Pr^{0.4}\left[1 + \left(\frac{D_i}{L}\right)^{\frac{2}{3}}\right] \tag{10}$$

where the values of Re_i and Pr are determined for each bundle pass. When using SCTs, the values of Nu_i, Eq. 10, are adjusted by the internal enhancement coefficient $E_i = Nu_{i,a} / Nu_{i,s} = f\left(Re_i\right)$. The overall heat transfer coefficient U_k, for each row of the bundle, is determined by

$$\frac{1}{U_k} = \frac{D_o}{D_i}\frac{1}{\alpha_{i,k}} + R_{f,i}\frac{D_o}{D_i} + 0.5\frac{D_o}{\lambda_w} ln\left(\frac{D_o}{D_i}\right) + R_{f,o} + \frac{1}{\alpha_{o,k}} \tag{11}$$

and it is related to the outer surface of the tubes. The thermal resistances $R_{f,i}$ and $R_{f,o}$, Eq. (11), are from the fouling on the inside and outside of the tubes.

The above methodology can be demonstrated to determine the characteristics of a horizontal shell-and-tube condenser (water heater), in which smooth tubes are replaced by heat transfer augmented tubes. Some of the cases for evaluation of the benefits using the criteria presented in [6] are considered.

TEST EXAMPLE

Horizontal shell-and-tube condenser (water heater) works under the following parameters:

1. Mass flow rate of heated water - $W_f = 90$ kg/s
2. Temperature of the incoming water in the device - $T_{f,i} = 20$ °C
3. Heating coolant - saturated steam with temperature - $T_s = 110$ °C
4. Mass flow rate of heating steam - $\dot{m}_s = 8$ kg/s
5. Brass tubes Ø16/14 mm.
6. The tubes are located in one pass.

When the water heater is made of smooth tubes, the tube bundle consists of $N = 395$ tubes with a length of $L = 3.35$ m. The required heat exchange area they provide is $A = 66.5$ m^2 for heat transfer rate $\dot{Q} = 17.844$ MW. The leaving water temperature is $T_{f,o} = 67.4$ °C, and the log-mean temperature difference at which the device operates is $\Delta T_m = 63.4$ K. The pumping power to overcome the hydraulic resistances in the tube bundle is $P = 0.559$ kW.

From all SCTs studied in [4, 5], the tube 340 has been selected having the best energy performance as follows: $E_i \equiv Nu_{i,a} / Nu_{i,s} = 2.85$; $E_o \equiv \alpha_{o,a} / \alpha_{o,s} = 1.05$; friction factor $f_a = 0.043\,Re^{-0.052}$. From the combinations of SCT with twisted tapes, the tube 344 has been selected, which has the following characteristics: $E_i = 8.142\,Re^{-0.016}$; $E_o = 0.99$; $f_a = 0.229\,Re^{-0.019}$.

Consider the effect of replacing smooth tubes with tubes with artificial turbulizers for the next cases:

Fixed geometry criteria (FG)

These cases involve a one-for-one replacement of smooth tubes by augmented tubes of equal length and may be regarded as "retrofit" applications. This group comprises the cases: FG-1a, FG-1b, FG-2a and FG-2b.

Case FG-1a

The objective of this case is to evaluate the increase in the flow rate $Q_* > 1$ with the constraints: $D_* = 1$, $L_* = 1$, $N_* = 1$, $W_* = 1$. The consequence is the increased pumping power, $P_* > 1$.

The result of the replacing the smooth tubes with SCTs (Tube 340) is that the heat duty of the heat exchanger has increased from 17,844 to 23,978 MW ($Q_* = 1.344$) or by 34.4%, Table 1; $T_{f,o} = 83.7$ °C; $\Delta T_{m,a} = 52.4$ K. The increased in the pumping power is $P_a = 2.956$ kW, which may require replacement of the existing pump.

When the tube 344 is used ($e/D_i = 0.0371$, $H/D_i = 5.98$), the heat transfer rate increases to $\dot{Q}_a = 25.651$ MW or $Q_* = 1.437$, i.e. there is an additional increase in the thermal power by 43.7%. The outlet water temperature is $T_{f,o} = 88.2$ °C and the log-mean temperature difference is reduced to $\Delta T_{m,a} = 48.1$ K. However, the pumping power increases significantly, $P_a = 7.737$ kW.

The summarized results are presented in Table 1, where the smooth tube is numbered as 200.

Table 1

Tube	Q	P	$T_{f,o}$	ΔT_m
№	MW	kW	°C	K
200	17.844	0.559	67.4	63.4
340	23.978	2.956	83.7	52.4
344	25.651	7.737	88.2	48.1

Case FG-1b

The objective of this case is the evaluation of the reduced driving temperature difference, $\Delta T_m^* < 1$, through the reduction of the inlet temperature difference between two fluids. The heat duty is fixed, $Q_* = 1$. The pumping power increases, $P_* > 1$. If the tube 340 is used, the results are: temperature of the inlet water $T_{f,i} = 45.0$ °C; outlet water temperature $T_{f,o} = 92.4$ °C, the temperature difference is $\Delta T_{m,a} = 36.3$ K, which is a reduction of nearly 43%. The pumping power for overcoming the hydraulic resistances is $P_a = 2.972$ kW.

When the tube 344 is implemented, the results are: water temperatures: $\Delta T_{f,i} = 49.3$ °C, $\Delta T_{f,o} = 96.7$ °C; the driving temperature difference $\Delta T_{m,a} = 31.2$ K. The reduction of the temperature difference is $\Delta T_m^* = 31.2/63.4 = 0.492$, Table 2, or nearly 51%. The pumping power is $P_a = 7.633$ kW. The summarized results are presented in Table 2

Table 2

Tube	P	$T_{f,o}$	ΔT_m
№	kW	°C	K
200	0.559	67.4	63.4
340	2.972	92.4	36.3
344	7.633	96.7	31.2

<u>Case FG-2a</u>

In this case, the requirement for constant pumping power $P_* = 1$ is added to the constraint for constant heat transfer area $A_* = 1$ ($N_* = 1$, $L_* = 1$). This can only be achieved by reducing the flow rate of the working fluid, $W_* < 1$. The objective is increased heat transfer rate $Q_* > 1$.

When the smooth tubes are replaced with tubes 340, in order to meet the constraint $P_* = 1$, the mass flow rate is reduced to $W_a = 51.0$ kg/s. In this case, the pumping power is $P_a = 0.556$ kW, the outlet water temperature is $T_{f,o} = 97.4$ °C, the heat rate is $\dot{Q}_a = 16.506$ MW and $Q_* = 16.506/17.844 = 0.925 < 1$, Table 3.

Obviously, it is not beneficial the condenser (heater) with SCTs to operate in this mode, since it does not achieve higher heat rate. This result is not surprising, because while maintaining the design of the exchanger there is an unnecessary heat transfer area at higher overall heat transfer coefficients.

Table 3

Tube	Q	P	$T_{f,o}$	W
№	MW	kW	°C	kg/s
200	17.844	0.559	67.4	90.0
340	16.506	0.556	97.4	51.0

<u>Case FG-2b</u>

The objective of this case is to evaluate the reduction of the driving temperature difference $\Delta T_m^* < 1$. The constraints are $N_* = 1$, $L_* = 1$, $P_* = 1$ and $Q_* = 1$. As a consequence, the mass flow rate of the working fluid must be diminished, $W_* < 1$.

If the tubes 340 are used instead of the smooth ones, the result is that it is not possible to increase the temperature of the water, because the temperature of the outlet water exceeds $T_{f,o} = 100$ °C. This can only be achieved if the inlet water temperature is below 20°C ($T_{f,i} < 20$ °C). For instance, $T_{f,i} = 12.0$ °C, $T_{f,o} = 95.7$ °C, $\Delta T_{m,a} = 43.5$ K and $T_m^* = 43.5/63.4 = 0.686$. Therefore, in this case, the replacement of the smooth tubes with SCTs is also inefficient.

Fixed flow area criterion

This criterion is based on the maintaining of a constant flow area. For a shell-and-tube exchanger having constant diameter tubes, this means that the number of tubes and shell diameter are held constant.

<u>Case FN-1</u>

The objective of this case is to evaluate the reduction of heat transfer surface area by reducing the length of the tubes (the bundle) $A_* < 1$ ($N_* = 1$, $L_* < 1$). The heat rate and pumping power are fixed, $Q_* = 1$ and $P_* = 1$. This can be only achieved by reducing the flow rate of heated water $W_* < 1$.

When tubes 340 are implemented, the result is that it is not possible to reduce the heat transfer area of the heat exchanger by reducing the length of the tubes, since it is not possible to meet the requirement for constant pumping power $P_* = 1$. The minimum pump power that can be achieved is $P_a = 0.771$ kW ($P_* = 0.771/0.559 = 1.38$) at $W_a = 57.0$ kg/s, and $W_* = 0.633$. The outlet water temperature is $T_{f,o} = 94.9$ °C.

Variable geometry criteria

The variable geometry (VG) cases have been developed by Webb [6] when the heat exchanger is “sized” for required heat flow with specified flow rate $W_* = 1$, and fixed pumping power, $P_* = 1$.

<u>Case VG-1</u>

In this case, it is necessary to increase the flow area to accommodate the higher fluid friction of the augmented surface. This can be achieved by increasing the number of tubes in single-pass heat exchanger, $N_* > 1$, or by reducing the number of passes in multi-pass design.

The objective is reduction of the heat exchange area of the apparatus $A_* < 1$ ($N_* > 1$, $L_* < 1$) under the constraints $W_* = 1$, $Q_* = 1$ and $P_* = 1$.

When the smooth tubes are replaced with SCTs 340, the heat exchange area of the condenser can be diminished as $A_a = 46.6$ m^2 or $A_* = 0.701$. The number of tubes in the tube bundle is $N_a = 630$; $N_* = 630/395 = 1.595$; the length of the tubes is $L_a = 1.52$ m, $L_* = 0.454$. Therefore, by changing the design of the heat exchanger, 30% reduction in the heat transfer area can be achieved by using SCTs. The outlet water temperature is $T_{f,o} = 67.4$ °C whereas the pumping power, $P_a = 0.554$ kW, remains unchanged.

When SCTs 344 are implemented, the required heat exchange area is $A_a = 39.5$ m^2, which is realized with $N_a = 840$ tubes, $N_* = 2.1$ and $L_a = 0.96$ m; the pumping power is almost the same $P_a = 0.548$ kW. The ratio $A_* = 39.5/66.5 = 0.594$ shows an additional reduction of the heat transfer area by about 11% compared to the device with SCT without tapes. The results are presented in Table 4.

Table 4

Tube	A	N	L	P
№	m^2	ps.	m	kW
200	66.5	395	3.35	0.559
340	46.6	630	1.52	0.554
344	39.5	840	0.96	0.548

<u>Case VG-2a</u>

The objective of this case is to evaluate the increase of the heat power, $Q_* > 1$. The constraints are $W_* = 1$, $P_* = 1$ and $A_* = 1$ ($N_* > 1, L_* < 1$).

When the smooth tubes are replaced with tubes 340 the results are: the heat power is $Q_a = 22.015$ MW or $Q_* = 1.234$ - the increase is 23.4%; the number of tubes is $N_a = 710$, $N_* = 1.797$; the length of the tubes is $L_a = 1.92$ m; the outlet water temperature is $T_{f,o} = 78.5$ °C; the driving temperature difference is $\Delta T_{m,a} = 55.7$ K, $\Delta T_m^* = 0.879$. The pumping power is $P_a = 0.545$ kW.

If tubes 344 are used in the same design, the heat power of the condenser is $\dot{Q}_a = 24.201$ MW, $Q_* = 1.356$ or an additional increase of 35.6% can be achieved. The number of tubes is $N_a = 1000$, $N_* = 2.532$, and the tube length is $L_a = 1.37$ m, $L_* = 0.41$. The outlet water temperature is $T_{f,o} = 84.3$ °C, $\Delta T_{m,a} = 51.3$ K, and the pumping power is $P_a = 0.558$ kW. The results are presented in Table 5.

Table 5

Tube	Q	N	L	P	$T_{f,o}$	ΔT_m
№	MW	ps.	m	kW	°C	K
200	17.844	395	3.35	0.559	67.4	63.4
340	22.015	710	1.92	0.545	78.5	55.7
344	24.201	1000	1.37	0.558	84.3	51.3

<u>Case VG-2b</u>

The objective of this case is to assess the reduction of the driving temperature difference $\Delta T_m^* < 1$, with constraints $Q_* = 1$, $P_* = 1$, $A_* = 1$ ($N_* > 1, L_* < 1$).

When the SCTs 340 are used the result is: inlet water temperature $T_{f,i} = 39.5$ °C; outlet water temperature $T_{f,o} = 86.9$ °C; the driving temperature difference between the two fluids is $\Delta T_{m,a} = 42.5$ K; $\Delta T_m^* = 0.670$, or a reduction of 33% can be achieved and respectively smaller distraction of exergy due to irreversibility of heat transfer at final temperature difference.

If the same SCTs are combined with twisted tape – SCT-344, the result is: pumping power $P_a = 0.55$ kW; inlet water temperature $T_{f,i} = 45.7$ °C; outlet water temperature $T_{f,o} = 93.1$ °C; $\Delta T_{m,a} = 35.5$ K, $\Delta T_m^* = 0.560$. There is an additional reduction of the temperature difference by 11% compared to the heat exchanger with tubes 340. The results are presented in Table 6.

Table 6

Tube	N	L	$T_{f,o}$	ΔT_m
№	ps.	m	°C	K
200	395	3.35	67.4	63.4
340	710	1.92	86.9	42.5
344	1000	1.37	93.1	35.5

CONCLUSION

It should be emphasized that the use of compound heat transfer enhancement technique, as that combination of SCTs (with appropriate geometry) with twisted tapes (with appropriate step), can lead to a significant reduction in capital investment for manufacture of heat exchangers $A_* < 1$ or a significant increase in the heat power $Q_* > 1$ of the condensers (water heaters) when new heat exchangers are bult up. For this purpose, the design of the existing apparatus must be changed. The easiest way is to reduce the number of passes (when the device is multi-pass), while maintaining the diameter of the casing. If the existing apparatus is single-pass, the change of construction is associated with increasing of the shell diameter and the reduction of the tube length.

The use of SCTs with twisted tapes to replace the smooth tubes in operating heat exchangers, without changing their design, can be done only if the available pumping power is sufficient to overcome the increased hydraulic resistance or can be replaced with more powerful. Reducing the flow rate of heated water is very often not desirable and such regimes should be avoided.

This study has revealed that the imposed constraint for fixed pumping power, $P_* = 1$ (cases FG-2a, FG-2b and FN-1) is a major obstacle to achieving greater benefit when different heat transfer enhancement techniques are used in the heat exchangers. In a recently publication paper, Zimparov et al. [12] have presented the idea that constraint $P_* = 1$ must be removed from the list of constraints. It has been proposed that the pumping power can be left to increase, $P_* > 1$, until the augmentation entropy generation number N_S is less than unity, $N_S < 1$. The maximum benefits can be achieved when $N_S = 1$ is achieved. The use of greater inside heat transfer enhancement (with the use of compound heat transfer enhancement techniques instead of simple ones), and with an enhanced outer tube surface of the augmented exchanger, $E_o > 1$, is an useful instrument to increase the benefits, if only $N_S < 1$.

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