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Enhanced film condensation of steam on a horizontal finned tube

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ABSTRACT

A heat transfer experimental setup was designed and fabricated to simulate the performance of steam condensers of the multistage flash process. The test section consists of a horizontal single tube shell and tube steam condenser. The experimental study was devised to evaluate the impact of using fin-tubes rather than plain tubes to augment the heat transfer rate during the condensation of steam. Three integral finned tubes of three different materials namely Cu/Ni 90/10, Cu/Ni 70/30, and Al/Br, were tested for heat transfer performance of steam. For each material, three tubes with fin densities of 11, 13, and 15 fins per inch (FPI) were tested. The heat transfer coefficient of the smooth plain tubes was used as a baseline for comparing the heat transfer performance of the high performance enhanced tubes. For the three tested materials, tubes of fin density 11 FPI consistently produced the highest heat transfer enhancement factor followed by 15 and 13 FPI tubes, respectively. For seawater velocity of 2.5 m/s, Al–Br finned tubes yielded the maximum enhancement level followed by Cu–Ni 70/30 and Cu–Ni 90/10, respectively.

Keywords: Steam; Condensation; Enhancement; Finned tube

1. Introduction

The multistage flash (MSF) distillation process produces about 26% of the total world desalinated water [1]. The popularity of MSF is due to its simplicity, robustness, and inherent reliability. Although the MSF process is the most reliable source for production of fresh water from seawater, it is considered an energy intensive process that requires both thermal and mechanical energy. Thermal energy, in the form of low pressure, bleed steam from a steam turbine at a pressure range of 1–3 bars is required for heating recycle brine, in the heat input section (brine heater). Medium pressure steam from higher pressure extraction point (in a steam turbine) is passed to ejectors to generate the required vacuum in the distiller. Mechanical energy is also required for driving various MSF pumps. High thermal energy input (typically 290 kJ/kg of product water) puts the MSF process in the highest energy consumption category in comparison to other commercially available desalination processes.

MSF units require condensing tubes of vast heat transfer surfaces. The heat transfer tubes are by far the major cost item in the evaporator amounting to 33% of the overall vessel cost [2]. The evaporator cost is normally 40% of the total plant cost [3]. Consequently, the cost of tubing represents around 13.5% of the total plant cost. Thus, if the heat transfer surfaces are made more effective, it will have a direct impact on the performance of the desalination units and consequently in the distiller capital costs.

Most condensers employed in MSF desalination plants are designed to operate under film-wise condensation mode. This process is characterized by the

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formation of a thin film of liquid that drains under the action of gravity or surface tension or both. The presence of a film creates a barrier between the vapor and the cooled surface and retards the condensation process. If condensation is to be enhanced, the film thickness must be reduced. This reduction can be achieved using a variety of condensation augmentation techniques such as drop-wise condensation (DWC), the use of external forces, and modification of surface geometry.

DWC has a significantly higher heat transfer coefficient (HTC) than film-wise condensation. A substantial amount of material and space could be saved if the presupposition of film-wise condensation, which is normally the basis for condenser design calculations, could be changed to DWC. The DWC mode has not been widely applied to practical industrial condensers because of the problem of maintaining DWC for sufficiently long periods to meet the requirements of the industrial applications. Therefore, the most important aspect for DWC heat transfer research at the present time is to explore techniques for preparing effective surfaces or surface films that prolong DWC for lifetimes appropriate for commercial applications [4].

Another approach employed to augment heat transfer in condensation is the use of external force to reduce film thickness of the condensate. Rotational, vibrational and electrostatic forces have been tested to decrease the condensate film thickness but have the disadvantage of requiring complicated equipment. Hamed and Sorial [5] reported forced convective heat transfer of condensation of steam taking place over a horizontal tube. They investigated the effect of steam impingement on a single vertical tube. The condensing vapor was admitted through a perforated tube. The vapor disintegrated and flowed in a jet-wise mode, impinging on the condensing surface. The high velocity vapor jets sweep the condensate layer as well as any condensing gas film off the condensing surface. This resulted in improvement in the HTCs.

Modification of condensing surface geometries is another method to reduce the thickness of the condensate layer. These enhancement surfaces may be finned tubes or profiled tubes such as roped, twisted, or fluted tubes. Modification of tube surface geometry will greatly enhance condensation heat transfer and lead to reduction in the size of heat exchangers.

A review on the condensation heat transfer of enhanced tubes was reported [6,7]. Ravi et al. [8] measured HTCs during condensation of steam and R 134a on five different horizontal circular integral-fin tubes (CIFTs) and on two-spine integral-fin tubes. Fin tip diameter of the copper integral fin tubes ranged from 23.82 to 24.98 mm and fin height from 0.6 to 1.11 mm. Longitudinal fin pitch varied from 0.53 to 1.07 mm for R 134a and was 2.57 mm for steam. It was found that the spine tubes give higher HTCs than CIFTs (for parameters investigated). Ravi et al. [9] tested also four specially tailored enhanced tubes with the same fluids—two with spines in the lower half and circular fins in the upper half and two with spines in the upper half and circular fins in the lower half. At constant temperature, difference in the first two tubes (spines on the lower half) gave higher HTCs than the other two, because of faster liquid drainage and/or less liquid retention.

Briggs [10] presented experimental data taken during condensation of steam at atmospheric pressure on four copper pin-fin tubes. The best performing tube showed a heat transfer enhancement ratio (ratio of heat fluxes for pin fin tube and plain tube at the same vapor side temperature difference of 20 K) of 3.95, comparable to the highest value found in the literature for a simple two-dimensional copper integral-fin tube.

Das et al. [11] studied steam film-condensation heat transfer on single horizontal 304 stainless steel integral-fin tubes to examine the effect of fin height. Experiments were conducted at both atmospheric and vacuum conditions on eight tubes with rectangular fins of nominal heights ranging from 0.16 to 1.5 mm. All the tubes had the same fin spacing (1.5 mm), fin thickness (1 mm), inside diameter (13.11 mm), and nominal root diameter (14.25 mm). The experimental results showed that condensation heat transfer was affected by surface tension and low thermal conductivity of tube material. The optimum fin height was 0.3 mm with a corresponding heat transfer enhancement for vacuum and atmospheric conditions of 1.4 and 1.6 respectively. For fin heights higher than 0.3 mm, the HTC decreased with increasing fin height.

Film condensation of stagnant water vapor on both vertical and horizontal spirally fluted tubes was studied theoretically [12]. The average HTC and Nusselt numbers were calculated and compared to that of smooth tubes. The analytical results showed that the enhancement due to fluting may reach five times in horizontal tubes, while for vertical tubes it is much lower. The theoretical results were compared to the available experimental results of film condensation on horizontal finned tubes and twisted vertical tubes, which were very similar to spirally fluted tubes and they showed good agreement.

A comparison between enhanced and plain tube performance characteristics is investigated experimentally [13]. The study was performed for aluminum-brass tubes with inside diameters of 19.0, 23.0, and 29.5 mm. Three different flow velocities of 0.10, 0.164, and 0.24 m/s were examined with these three tubes.

The study was carried outer for two different coolants, fresh and brine water. Results indicate that the overall heat transfer coefficient (OHTC) (U) varies with both coolant flow velocity and tube diameter. For both smooth and enhanced tubes, the higher the coolant speed, or/and pipe diameter, the larger the OHTC. However, the effect of these two parameters is more significant with enhanced tubes. An increase in OHTC by factor of 2.13, over smooth tubes, can be achieved. Results also show that for certain tube diameters there is an optimum flow speed for maximum value of U and for a certain flow speed, there is an optimum pipe diameter size. Results prove that HTCs of corrugated tubes are superior to plain tubes for most flow speeds and diameters.

Ali et al. investigated the possibility of utilizing roped tubes with special surface configurations in the steam condensers of multi-stage flash distillation plants (MSF modules) [14]. A mathematical model for condensation on fluted tube surfaces was developed and numerically solved. A single tube experiment, with four tube configurations was considered, namely plain, small pitch, medium pitch, and big pitch roped tubes. The convective HTCs inside the tubes as well as the OHTCs were correlated against both Reynolds number and steam loading. The range of Reynolds number covered in the experiments is up to 50,000. The study revealed that the roped tubes improved the convective HTC inside tubes by 70–160% compared to plain tubes.

Namasivayam and Briggs [15] reported the test results of the experimental work conducted to evaluate forced-convention condensation on a set of five single integral fin tubes of fin spacing ranging between 0.25 and 2.0 mm. The best performing finned tube was that with a fin spacing of 0.25 mm despite the fact that this tube was fully flooded with retained condensate between fins. This result, however, was in line with the simple theory of condensation of quiescent vapor.

An experimental study was performed to investigate the heat transfer and pressure drop characteristics of serrated finned tube banks with staggered layouts [16]. The influences of varied fin densities, transversal tube spacing, and longitudinal tube spacing were reported. For a constant fin height, an increase in the fin density resulted in an increase in the Euler number, and a gradual decrease in the Nusselt number was observed as the Reynolds number increased. An increase in the transversal tube spacing corresponded to a significant reduction in the Euler number whereas the Nusselt number essentially remained unchanged. The longitudinal tube spacing had an insignificant effect on the Nusselt and Euler numbers and the optimum ratio of the transversal tube spacing to longitudinal tube spacing increased with an increase in the transversal tube spacing. Scaling of the tube spacing had little effect on the Nusselt number but had a significant influence on the Euler number.

Seara et al. [17] evaluated the condensation of R-134a on horizontal smooth and integral-fin (32 fins per inch [FPI]) titanium tubes of 19.05.mm outer diameter. Experiments were carried out at saturation temperatures of 30, 40, and 50°C. The results showed that the condensation HTCs on the smooth tubes were well predicted by the Nusselt theory with an average error of +2.38% and within a deviation between +0.13 and +5.42%. The enhancement factors provided by the integral-fin tubes on the overall condensation HTCs ranged between 3.09–3.94, 3.27–4 and 3.54–4.1 for condensation temperatures of 30, 40, and 50°C, respectively. The enhancement factors increased by increasing the wall subcooling and with the rise of the condensing temperature. The condensate flooded fraction of the integral-fin tubes perimeter varied from 25 to 20% at saturation temperatures of 30 and 50°C, respectively.

An experimental investigation has been carried out to find the condensing side HTC, during condensation of steam over a plain tube, a CIFT, and a spine integral-fin tube (SIFT) [18]. The CIFT and SIFT have enhanced the outside HTC by a factor of 2.5 and 3.2, respectively.

Claire et al. [19] reported experimental results on the effect of vapor velocity on condensate retention between fins during condensation of low-finned tubes. Condensation was simulated using three liquids (water, ethylene glycol, and R113) supplied to the tube via small holes between the fins along the top generator of the tubes. Eight tubes with different fin dimensions were used. Results indicated that when the retention angle (which is a measure of the extent of condensate retention between fins) was less than about $\pi/2$ at zero air velocity, it increased with increasing air velocity. On the other hand, for cases where the retention angle was greater than about $\pi/2$ at zero air velocity it decreased with increasing air velocity. In all cases (all fluids and all tubes tested) the retention angle approached a value near $\pi/2$ with increasing air velocity.

Murase et al. [20] investigated the effect of inundation during condensation of steam in tubes banks. Most of the data related to wire-wrapped enhanced tubes but measurements were also reported for low-finned and smooth tubes. The technique of artificial inundation was used where liquid was supplied above a single horizontal test condenser tube to simulate condensate draining from higher tubes. Inundation rates were used to simulate a column of up to almost 30 tubes. The temperature and flow rate of the simulated inundation were carefully controlled. All tests were carried out at atmospheric pressure with constant vapor down flow approach velocity and constant coolant flow rate. For the given coolant and vapor flow rates and temperatures (same for all tests), and in the absence of inundation, the vapor-side heat-transfer coefficient for the finned tube was around four times that of the smooth tube while the heat-transfer coefficient for the wire-wrapped tubes was independent of winding pitch and around 30% higher than that of the smooth tube. The heat-transfer coefficient for the finned tube was virtually unaffected by inundation up to a depth of 20 finned tubes in a bank. At this depth level, the heat-transfer coefficient for the finned tube was around six times that of smooth tubes. For the wire-wrapped tubes, deterioration in performance with increasing inundation was least for the smallest winding pitch used for which the heat-transfer coefficient fell by 9% at an equivalent depth, in a bank of 25 tubes. At this depth level the heat-transfer coefficient for the wire-wrapped tube was almost twice that of the smooth tube.

An approximate analytical method has been suggested for solving the governing equation for horizontal pin fins subject to condensation while saturated steam was flowing under laminar forced convection [21]. Adomian decomposition method was used for the determination of the temperature distribution, performance, and optimum dimensions of pin fins with temperature dependent thermal conductivity under condensation of steam on the fin surface. From the results, a significant effect on the temperature distribution in the fin and its performances are noticed with the variation in fin-geometric parameters and thermo-physical properties of saturated vapor. Next, a generalized scheme for optimization has been demonstrated in such a way that either heat-transfer duty or fin volume can be taken as a constraint. Finally, the curves for the optimum design have been generated for the variation of different thermo-physical and geometric parameters, which may be helpful to a designer for selecting an appropriate design condition.

Literature review revealed that investigations on the use of finned tubes to enhance heat transfer were to a large extent focused on condensation of refrigerants. This is mainly attributed to very low HTCs associated with organic fluids. In spite that the HTCs of condensing steam are relatively high, the use of finned tubes will further increase the OHTC and the outside surface area which in turn will reduce the heat transfer equipment size and cost.

The aim of present study is to design and fabricate a multipurpose test rig incorporating a single tube condensing test section simulating actual operating conditions of MSF condensers. The experimental setup was developed to examine the impact of the use of integral finned tubes of different fin densities on the performance of heat transfer tubes of a variety of materials of construction which are conventionally employed in MSF condensers.

2. Experimental

2.1. Experimental set-up

Schematic diagram and photograph of the experimental setup are shown in Figs. 1 and 2, respectively. The test section incorporated a single tube steam condenser which was maintained in a horizontal orientation. The tube was cooled internally by seawater. Pure steam condensing outside the heat transfer tube was obtained from an existing boiler through a pressure control system and a steam separator where entrained water droplets separate from steam flow and collect at the bottom. The condensate flow rate was measured by a rotameter as well as a condensate-measuring jar and a condensate-collecting tank. The condensate-collecting tank is attached to a suction pump, to remove the condensate. A glass window, in the test section, allows observation of the test condenser tube. Suitable instruments are provided to measure different parameters like pressure, flow, and temperature. Inlet and outlet temperatures of cooling seawater as well as the steam shell temperature were measured with resistance temperature detectors, with accuracy

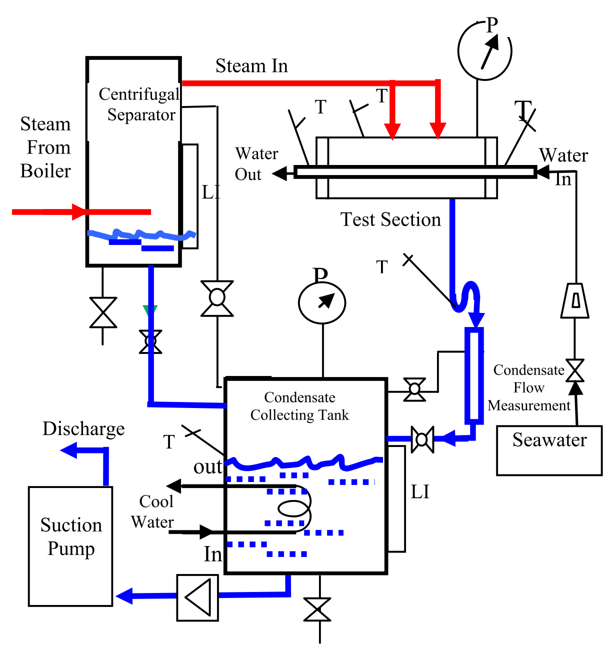


Fig. 1. Schematic diagram of the experimental setup.

Fig. 2. Photograph of the bench top experimental unit.

better than ±0.5%. The flow rate of the cooling sea water is measured by an electromagnetic flow transmitter with accuracy which was better than ±1% as well as with a calibrated flow meter for comparison.

2.2. Materials

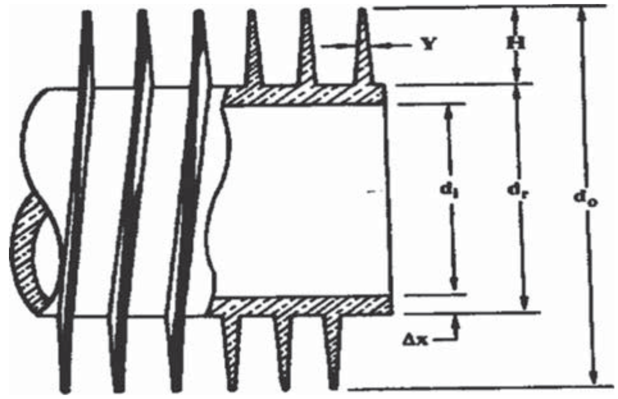
Three integral fin tubes of three different materials namely Cu–Ni 90/10, Cu–Ni 70/30, and Al–Br, which are specially manufactured for the research study, were tested. A smooth (plain) tube of each of the three selected materials was also tested as a baseline for comparison with the heat transfer performance of finned tubes. For each material, three finned tubes were tested with fin densities of 11, 13, and 15 FPI. This lies within the range of fin densities normally employed for steam condensation [22,23]. The geometric parameters of the finned tube are shown in Fig. 3. Fig. 4 shows a photograph of one of the finned-tubes. The tests were performed at a pressure slightly above atmospheric pressure and covered cooling seawater velocities between 1.0 and 3.0 m/s.

3. Enhancement heat transfer factor analysis

The enhancement heat transfer factor (EF) is the figure of merit used in this study to quantitatively evaluate performance of the finned tube and is defined as the ratio of OHTC of the finned tube ($U_{\text{finned tube}}$) to that of a plain tube of same material ($U_{\text{plain tube}}$) at the same coolant (sea water) velocity.

$$\mathrm{EF} = U_{\text{finned tube}}/U_{\text{plain tube}} \tag{1}$$

In the heat transfer test section sea water flow inside tubes while saturated water vapor is condensing outside these tubes. The equation of heat transfer between the two fluids is given by.



D_o = Diameter over fins = 39.58 mm
D_r = Root diameter of finned section = 35.38 mm
D_i = Inside diameter of finned section = 32 mm
Δ_x = Wall thickness of finned section = 1.7 mm
Y = Mean fin thickness = 1.59 mm
H = Fin height = 2.1 mm

Fig. 3. Geometric parameters of the finned tube.

Fig. 4. Photograph of a finned tube.

$$Q = UA\theta_m \tag{2}$$

Using vapor saturation temperature, the logarithmic mean temperature difference, θ_m may be calculated by:

$$\theta_m = \frac{T_o - T_i}{In\left[\frac{T_s - T_i}{T_s - T_o}\right]} \tag{3}$$

The amount of heat transfer Q between the saturated vapor and coolant (seawater) is equal to energy gain by the coolant which is determined by:

$$Q = mC_p(T_o - T_i), kW \tag{4}$$

The specific heat of cooling sea water is calculated from the corresponding values of sea water salinity (total dissolved solids of around 43,000 ppm) and the mean average temperature of the sea water inlet and outlet temperatures.

Substituting Eq. (4) into Eq. (1) leads to OHTC of:

$$U = \frac{mC_p(T_o - T_i)}{A\theta_m} \tag{5}$$

A is the nominal inside surface area and may be determined by:

$$A = \pi D_1 L$$

Eq. (5) is used to determine the OHTC (U) of the finned and plain tubes, experimentally. The coolant mass flow rate (m), coolant inlet and outlet temperature (T_i and T_o). and saturated vapor temperature (T_s), are measured. The logarithmic mean temperature difference (θ_m) is calculated from Eq. (3). Substituting m, T_i, T_o and θ_m into Eq. (5) the OHTC U is determined.

4. Results and discussion

Extensive heat transfer tests were carried out to evaluate thermal performance of single horizontal integral fin tubes during condensation of pure steam outside the tube while seawater (coolant) was flowing inside the tube.

Three materials [Al–Br, Cu–Ni (90/10), and Cu–Ni (70/30)] which are conventionally employed in MSF desalination plants, were tested. For each material, the performance of three finned tubes of fin densities 11, 13, and 15 FPI as well as the plain smooth tubes, was examined. The condenser shell pressure was maintained slightly above atmospheric pressure.

Fig. 5 shows the base-line heat transfer test results in which the three plain condenser tubes made of Cu–Ni 90/10, Cu–Ni 70/30, and Al–Br, were used. For each material, increase of coolant seawater velocity results in the increase of the OHTC. The increase of the OHTC results from a reduction of the seawater coolant resistance side. Increase of the coolant velocity induces turbulence which will in turn increase the sea water film HTC. At the same coolant velocity, Al–Br material of thermal conductivity 100 W/mK exhibits consistently the highest OHTC, followed by Cu–Ni 90/10, and Cu–Ni 70/30 of thermal conductivities of 50 and 20 W/mK, respectively. At a sea water velocity of 2 m/s, which is conventionally employed in the condenser tubes of MSF desalination plants and at a condensing temperature of 100℃, the OHTC of aluminum brass, Cu–Ni 90/10, and Cu–Ni 70/30 are 4.8, 4.5 and 3.9 kW/m^{2}K, respectively.

Fig. 6 shows the impact of variation of water velocity on the enhancement factor for Al–Br finned-tubes. Tubes of fin density 11 FPI are consistently producing the highest enhancement factor followed by tubes of finned densities of 15 and 13 FPI, respectively. Keeping the fin density constant, increasing the coolant velocity results in the increase of the enhancement factor.

Figs. 7 and 8 show the impact of variation of water velocity on the enhancement factor for Cu–Ni 90/10, and Cu–Ni 70/30, respectively. For the same coolant velocity, tubes of low fin density (11 FPI) produce the highest enhancement factor followed by tubes of fin densities 15 and 13 FPI, respectively. For the two materials, the largest enhancement factors are obtained at low water velocity.

Fig. 9 shows a plot of enhancement factor against fin density for the three tested tube materials of construction at a constant seawater velocity of 2.5 m/s. It can clearly be seen that integral-fin tubes of fin den-

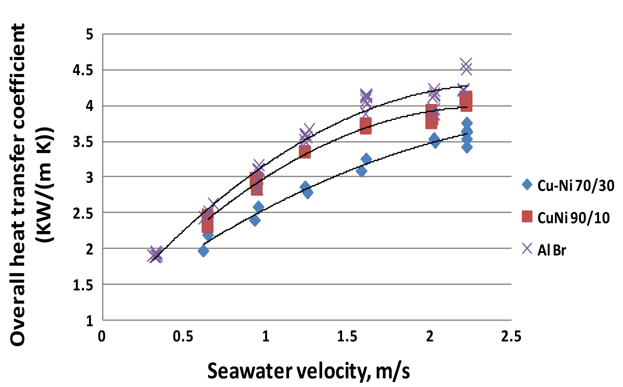


Fig. 5. Baseline tests—OHTC vs. sea water velocity for plain tubes.

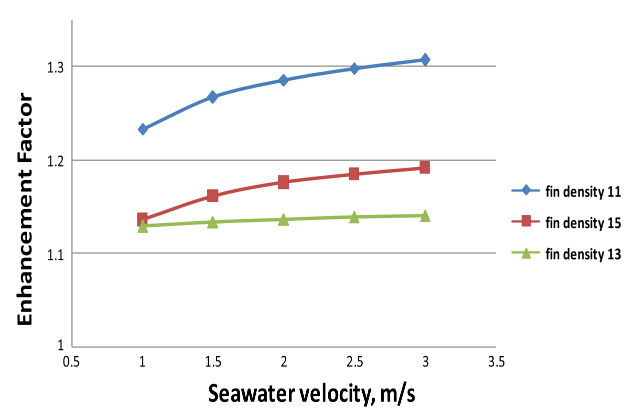


Fig. 6. Variation of enhancement factor with sea water velocity for Al Br finned tube.

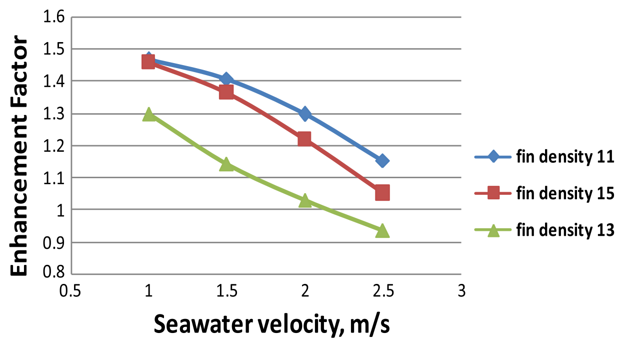


Fig. 7. Variation of enhancement factor with sea water velocity for Cu–Ni 90/10 finned tube.

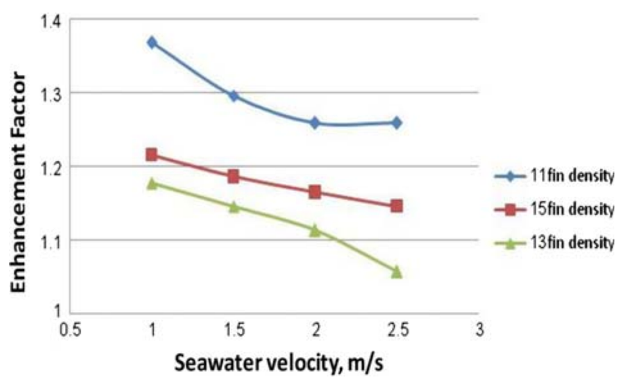


Fig 8. Variation of enhancement factor with sea water velocity for Cu–Ni 70/30 finned tube.

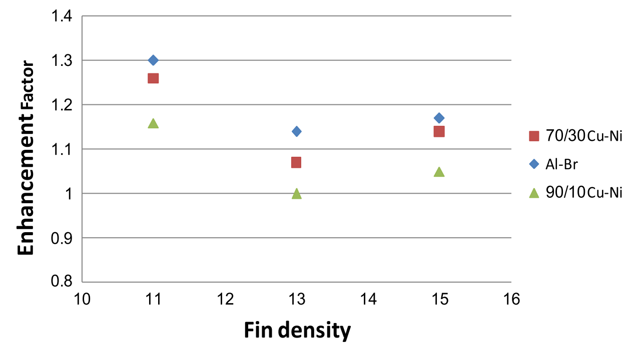


Fig. 9. Variation of enhancement factor with fin density for the three tested materials at sea velocity of 2.5 m/s.

sity of 11 FPI are the best performing tubes and are consistently yielding the highest enhancement factor for the three tested materials. Conversely, tubes of fin density 13 FPI are yielding the lowest enhancement factor. Meanwhile tubes of fin density of 15 FPI are exhibiting enhancement factors in between the 11 and 13 FPI tubes.

There are two opposing factors which influence the impact of fin density on the enhancement factor. As the fin density is increased, more fin tip area becomes available and consequently heat transfer enhancement level is increased. Conversely, as fin density increases the amount of condensate retained by capillary forces between fins is increased and the area available for heat transfer is considerably reduced [15]. In this study, tubes of the lowest fin density (11 FPI), although are relatively generating the lowest fin tip area, exhibit the best performance as a result of a less amount of condensate retained between fins. Increasing the fin density to 13 FPI, causes finned tubes become more flooded and consequently impedes heat transfer enhancement rate. As the fin density is increased further to 15 FPI, increase of the fin tip area due to decrease of fin spacing supersedes the negative impact of condensate flooding between fins. Results of this study are in agreement with the test results presented by Jaber and Webb [23] who report that the economic optimum fin density is expected to be <13 fin/in. Fig. 9 also shows that at constant fin density, Al–Br finned tubes with the highest tube thermal conductivity are consistently exhibiting the best enhancement level followed by Cu–Ni 70/30 and Cu–Ni 90/10 finned tubes.

The present experimental study is focused on the performance of a single finned heat transfer tube. In designing finned tube heat exchangers, which can be installed as steam condensers or brine heater in a large sized MSF desalination plant, it is necessary to perform pilot plant studies on a heat exchanger consisting of a shell and bundle of finned tubes. Optimum design conditions such as fin geometry, shell diameter, number of tubes, tube size, pitch arrangement, and condensate inundation and the effect of fins on fouling will then be identified.

5. Conclusions

(1) The present experimental study identifies the best performing integral finned tube geometries of three different materials of construction namely Cu–Ni 90/10, Cu–Ni 70/30, and Al–Br, for steam condensation. For each material, three tubes with fin densities of 11, 13, and 15 FPI were evaluated. The heat transfer performance of smooth plain tubes made of the same material of construction, were used as a baseline for comparison.

(2) For the three tested materials, tubes of low fin density 11 FPI are the best performing tubes and are consistently producing the highest heat transfer enhancement factor followed by tubes of finned densities 15 FPI and 13 FPI, respectively.

(3) For seawater velocity of 2.5 m/s, Al–Br finned tubes yielded the maximum enhancement level followed by Cu–Ni 70/30 and Cu–Ni 90/10, respectively.

Symbols

A	—	nominal inside tube surface area (m^2)
C_p	—	specific heat of coolant (kJ/kg ℃)
D	—	tube inside diameter (m)
EF	—	enhancement factor
L	—	tube length (m)
M	—	coolant mass flow rate (kg/s)
Q	—	thermal load on the condenser (kW)
T_s	—	saturated vapor temperature in the condenser (℃)
T_i	—	coolant (seawater) temperature at the entrance of the condenser tube (℃)
T_o	—	coolant (seawater) temperature at the exit from the condenser tube (℃)
U	—	overall heat transfer coefficient (kW/m^2℃)
$U_{\text{finned tube}}$	—	overall heat transfer coefficient of the finned tube (kW/m^2℃)
$U_{\text{plain tube}}$	—	overall heat transfer coefficient of the plain tube (kW/m^2℃)
θ_m	—	the logarithmic mean temperature difference (℃)

References

[1] IDA Desalination, Water Desalination Report, Yearbook 2011–2012.

[2] C. Sommariva, H. Hogg, K. Callister, Cost reduction and design lifetime increase in thermal desalination plants: Thermodynamic and corrosion resistance combined analysis for heat exchange tubes material selection, Desalination 158 (2003) 17–21.

[3] C. Sommariva, H. Hogg, K. Callister, Maximim economic design life for desalination plant: The role of auxiliary equipment materials selection and specification in plant reliability, Desalination 153 (2002) 199–205.

[4] X. Ma, J. Chen, D. Xu, J. Lin, C. Ren, Z. Long, Influence of processing conditions of polymer film on dropwise condensation, Int. J. Heat Mass Transf. 45 (2002) 3405–3411.

[5] O.A. Hamed, G.A. Sorail, Forced convective heat transfer in the condensation of pure vapor jets on a vertical tube, Arab Gulf J. Sci. Res. 2 (1984) 731–742.

[6] A. Cavallini, G. Gensi, D. Del Col, L. Dorett, G.A. Longo, L. Rossetto, C. Zilio, Condensation inside and outside smooth and enhanced tubes—a review of recent research, Int. J. Refrig. 26 (2003) 373–392.

[7] M.W. Browne, P.K. Banal, An overview of condensation heat transfer on horizontal tube bundles, Appl. Therm. Eng. 19 (1999) 565–594.

[8] K. Ravi, H.K. Varma, B. Mohanty, K.N. Agrawal, Prediction of heat transfer coefficient during condensation of water and R-134a on single horizontal integral-fin tubes, Int. J. Refrig. 25 (2002) 111–126.

[9] K. Ravi, H.K. Varma, B. Mohanty, K.N. Agrawal, Augmentation of heat transfer during filmwise condensation of steam and R-134a over single horizontal finned tubes, Int. J. Heat Mass Transf. 45 (2002) 201–11.

[10] A. Briggs, Condensation of steam on pin-fin tubes, in: Proceedings of 1st International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, South Africa, Kruger National Park, 2002.

[11] A.K. Das, G.A. Incheck, P.J. Marto, The effect of fin height during steam condensation on a horizontal stainless steel integral-fin tube, J. Enhanc. Heat Transf. 6 (1999) 237–250.

[12] N.H. Aly, S.D. Bedrose, Enhanced film condensation of steam on spirally fluted tubes, Desalination 101 (1995) 295–301.

[13] A.Y. Kalendar, A.J. Griffiths, Performance study of enhanced and smooth surface tubes in a system condenser of a multistage flash desalination unit, Desalination 134 (2001) 269–293.

[14] S.A. Ali, K.A. Fathalan, A. Radhwan, Impact of Enhanced Condensation on MSF Desalination Plant Performance, King Abdulaziz City for Science and Technology, Report APR-4–7, 1989.

[15] S. Namasivayam, A. Briggs, Effect of vapor velocity on condensation of atmospheric pressure steam on integral-fin tubes, J. Appl. Therm. Eng. 24 (2004) 1353–1364.

[16] Y. Ma, Y. Yuan, Y. Liu, X. Hu, V. Huang, Experimental investigation of heat transfer and pressure drop in serrated finned tube banks with staggered layouts, Appl. Therm. Eng. 37 (2012) 314–323.

[17] J.F. Seara, F.J. Uhia, R. Oiz, A. Oopaz, Condensation of R-134a on horizontal integral-fin titanium tubes, Appl. Therm. Eng. 30 (2010) 295–301.

[18] R. Kumar, H.K. Varma, B. Mohanty, K.N. Agrawal, Augmentation of outside tube heat transfer coefficient during condensation of steam over horizontal copper tubes, Intl. Comm. Heat Mass Transfer 25(I) (1998) 81–91.

[19] C.L. Fitzgerald, A. Briggs, J.W. Rose, H. S. Wang, Effect of vapour velocity on condensate retention between fins during condensation on low-finned tubes, Int. J. Heat Mass Transf. 55 (2012) 1412–1418.

[20] T. Murase, H.S. Wang, L.W. Rose, Effect of inundation for condensation of steam on smooth and enhanced condenser tubes, Int. J. Heat Mass Transf. 49 (2006) 3180–318921.

[21] B. Kundu, G.K. Ghosh, An approximate analytical prediction about thermal performance and optimum design of pin fins subject to condensation of saturated steam flowing under forced convection, Int. J. Refrig. 32 (2009) 809–825.

[22] R. Kumar, H.K. Varma, B. Mohanty, K.N. Agrawal, Augumentation of heat transfer during filmwise condensation of steam and R-134a over single horizontal finned tubes, Int. J. Heat Mass Transf. 45 (2002) 201–211.

[23] M.H. Jaber, R.L. Webb, Steam condensation on horizontal integral-fin tubes of low thermal conductivity, Journal of enhanced heat transfer 3 (1996) 55–71.