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Determination of Korodense Tube Thermal Characteristics

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Abstract: Fouling in condenser tubing for the Multi-Stage Flash system reduces the system overall heat transfer coefficient and consequently system efficiency. In this study, a comparison between korodense and smooth tube thermal characteristics is investigated experimentally in a laboratory designed rig. A significance part of this manuscript is to design a research test-rig to simulate the heat exchange process in desalination plant, equipped in such a way to minimize mutual heat transfer between tubes, dismantle non-uniform distribution of steam, with uniform temperature distribution inside the steam condenser, maintained to achieve the asymptotic value of the overall heat transfer coefficient within a short time, operates smoothly without malfunctions for long operating hours; 160 hrs without creation of two-phase flow, or accumulation of gases inside the rig. The circumstances in a real desalination plant-form are simulated with regarding the constituents: coolant flow speed, tube diameter and the type of employed liquid coolant. Tested korodense and smooth tube at a time is to unify tested conditions on both. Other major parameters influenced the test circumstances are: inlet coolant temperature, inside temperature for the test-rig, existence of non-condensable gases, the used real brine water concentration (ppm). The tested tube characteristics are carefully identified, with a wise differentiation between properties of brine and fresh water. The study is performed for aluminum-brass tubes with inside diameters of 19.05, 23 and 29.5 mm. Three flow velocity of 0.1, 0.1645 and 0.2398 m/s are simultaneously examined with each time with these three tubes. Performance is detected for the two coolants fresh and real brine water. Distinguish results of fresh and brine water coolants are presented, indicates the overall heat transfer coefficient varies for both coolants, on various utilized velocities and diameters of both korodense and smooth tubes. Fresh water is examined first, for each tube diameter starting by; 19.05 mm and ending by 29.5 mm, with coolant speed increases from 0.1m/s to 0.2398 m/s. The higher the overall heat transfer coefficient is, the better the performance of the tube. However, the effect of change of these two parameters is more significant with korodense tube rather than the smooth. When brine water is utilized, a gain in the overall heat transfer coefficient by a margin factor over smooth tube has achieved.

Key words: Smooth tube · Corrugated tube · Condenser Tubing · MSF system · Desalination

INTRODUCTION

One of the most common phenomena for desalination of seawater is distillation. When saline solutions are boiled the vapour that comes off is pure water vapour containing no salt. The salt remains in the non-evaporated solution. The separation process is highly efficient and the vapour is easily separated from the solution. Multi Stage Flash Distillation (MSF) System is the most widely used system for desalting seawater in Gulf areas. Illustration of a typical MSF system is given in Figure 1. 95.7% of fresh water is claimed to be produced by the multi-stage flashing system plants [1]. MSF system has many important advantages, over other distillation processes, including: high-performance ratio which is defined as the ratio of mass of fresh water produced to energy input, simplicity of the structure, high output capacities of fresh water, low cost of produced water and the plant performance ratio is not rigidly tied to the number of stages used. However, MSF system experiences severe mechanical difficulties due to fouling. Seawater flows inside the condenser tubes and is being heated by the condensing steam on the outside of the tubes. Figure 2 shows the tubing of used type of condenser heat exchanger. Considerable amounts of fouling are formed on the inside surfaces of the tubes and

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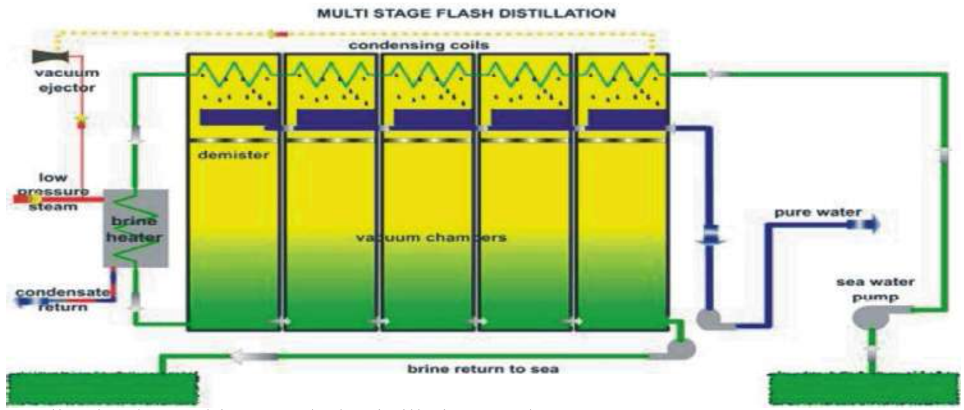

Fig. 1: How Desalination by Multi-stage Flash Distillation Works [3]

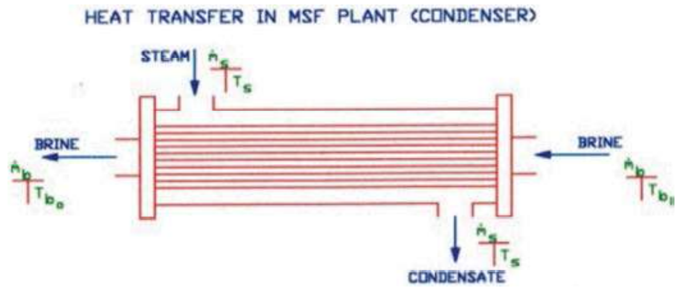

Fig. 2: Tubing of a Condenser Heat Exchanger [12]

act as heat insulation, decreases the efficiency and hydraulic performance of the system. Consequently, increases the cost of produced fresh water. To improve the heat- transfer performance of the MSF system enhanced heat transfer technique may be used. The idea of using enhanced tube for MSF system, if proved it will be a lifetime energy saver [2]. An enhanced heat transfer surface has a special surface geometry, provides a higher heat transfer coefficient and surface per unit space area than a traditional design. The enhanced transfer mechanisms have developed to stage that it has routinely used in heat exchanger of the refrigeration and automotive industries. However, each application must be tested to see if enhanced heat transfer makes sense. Unfortunately, all the current MSF systems are designed using smooth (plain) surface tubes. Heat transfer enhancement techniques in heat exchangers have been of interest for a number of years [3, 4]. Most fouling research has been connected with a single-phase flow in a smooth tube [2]. As a rule, fouling and enhancement have been investigated separately with little cross-referencing of the two. Watkinson [5] had attempted to bring the two areas together; however, Kim and Webb [6], Chamra and Webb [7] have obtained the major achievements. They concluded that the fouling data on enhanced tubes appears to show a higher fouling rate than plain tubes. However, the thermal performance of the enhanced tubes remained superior to that of the plain tubes. Tube enhancement can be achieved using two or three-dimensional roughness.

Two-dimensional rib roughness has been extensively investigated [3]. Many investigations were performed, to study the effect of enhanced tube geometrical factors on their performance characteristic. Gee and Webb [8] studied the effect of helix angle (α) on the Stanton number (*St*) and Fanning friction factor (f) characteristics of repeated rib roughness. Their results illustrated, for p/e = 15 and e/d_i = 0.01, that helical-rib roughness provides a better heat transfer coefficient than a smooth tube. Furthermore, they concluded that as α decreases the friction factor f drops faster than does Stanton number (St). It is clear from this literature review of two dimensional roughened tubes that enhancement techniques improve the heat transfer [9]. The above mention facts have influenced the research direction and throw the light on the details of the current work. These include, choosing tube roughness enhancement, studying the effect of tube diameters and Reynolds numbers on performance characteristics of the multi- stage flash system. Use is made of actual cooling water, avoiding of using artificial fouling which leads to misleading results. Furthermore, support the current idea of trying enhanced surface tubes with multi- stage flash system (MSF) distillation unit. If this idea succeeded it will be a lifetime energy saver. The novel approach of the research program can be itemized as follows: to design a research rig that simulates one stage, to examine and assess the feasibility of enhanced heat transfer mechanisms in MSF plant, to use actual brine water that typifies real process conditions, to study both the influence and effects of flow speeds on fouling resistance, to study the effect of tube diameter in fouling resistance and to examine the influence of the tube material on fouling resistance [9-13].

Overall Heat Transfer Coefficient Analysis: It's important to analyses the overall heat transfer coefficient. In the MSF condenser, a cooled brine water flow inside tubes while a saturated water vapour is outside these tubes as shown in Figure 2. The equation of heat transfer between two fluids is given by [12]:

$$Q = U\,(A\ \theta_m) \tag{1}$$

Q = Thermal load on the condenser in kW,
A = Nominal inside tube surface area in m^2
θ_m = Logarithmic mean temperature difference in °C,
U = Overall heat transfer coefficient, kW / m^2 °C

The logarithmic mean temperature difference □m, by using the vapor saturation temperature Ts can be calculated from [12]:

$$\theta_m = \frac{T_o - T_i}{Ln\left[\dfrac{T_s - T_i}{T_s - T_o}\right]}$$

T_s = Saturated vapor temperature in the condenser,
T_i = Coolant (fresh or brine) temperature at the entrance of the condenser tubes
T_o = Coolant (fresh or brine) temperature at the exit from the condenser tubes

The amount of heat- transfer Q between the saturated vapor and coolant water is equal to energy gain by coolant liquid, determines by [12]:

$$Q = \dot{m}\ Cp\ (T_0 - T_i) \tag{2}$$

$\dot{m}$ = Coolant mass flow rate, kg/s
C_p = Specific heat of coolant, kJ/kg °C
T_0 = Coolant temperature out, °C
T_i = Coolant temperature in, °C

Substituting equation (2) and by Om equation into equation (1) leads to overall heat transfer coefficient:

$$U = \frac{\dot{m}Cp\,(T_o - T_i)}{A\ \theta_m},\ kW/m^2\,°C \tag{3}$$

A is the nominal inside surface area:

A = $\pi D_i L$
D_i = Tube inside diameter, m
L = Tube Length, m

Experimental Set-Up: The apparatus consists mainly of condenser, boiler, circulating pumps, tanks, measuring instruments, data acquisition system and computer. The test rig condenser "Brine Heater Tank" schematic drawing is highlighted in Figure 3. The condenser consists of drum, cooling tubes and steam distribution tube. The drum is a stainless steel cylindrical box with an inside dimension of 78 cm diameter and 110 cm length. As shown in the figure, the drum has four sight glass windows two on each side to observe the condensation process on the outside surface of cooling tubes. The condenser is designed with eight cooling tubes, which evenly spaced in the drum as illustrated in the figure. The locations of the tested korodense and smooth tubes are indicated by number 20 and 24 in the schematic. The steam distribution tube is located at the center of the drum and contains holes in its entire surface to distribute the vapor evenly along the cooling tubes. The boiler is an electric resistance type; main supply is a three phase voltage 380V - 50 Hz. The maximum power is 18 kW and the maximum steam production is 20 kg/hr. The maximum operating pressure is 3-4 bars. The boiler is provided with centrifugal circulation pump, which provides the boiler with fresh water. Two pumps are assigned for circulating the coolant flow to the condenser. They are operated individually, sharing the experimental duration running time. They are single stage jet assisted centrifugal pumps with closed vane impellers, mechanical shaft seal and "O" ring casing seal. The pump is manufactured from quality corrosion resistant materials have a quick and easy installation. The capacity is 35 liter/min at a pressure of 140 kPa with maximum total head of 35 m. The flow rate of the coolant inside the condenser tubes was measured by using a formed glass flow tube type flow meter. Flow rates were determined by reading the scale graduation at the centre of the float. The maximum scale graduation available was 150 *mm*, which corresponds to 0.521 liter/ min. The thermocouples used are inserted in the important points where temperatures are to be taken. Those thermocouples are E-type and connected to the

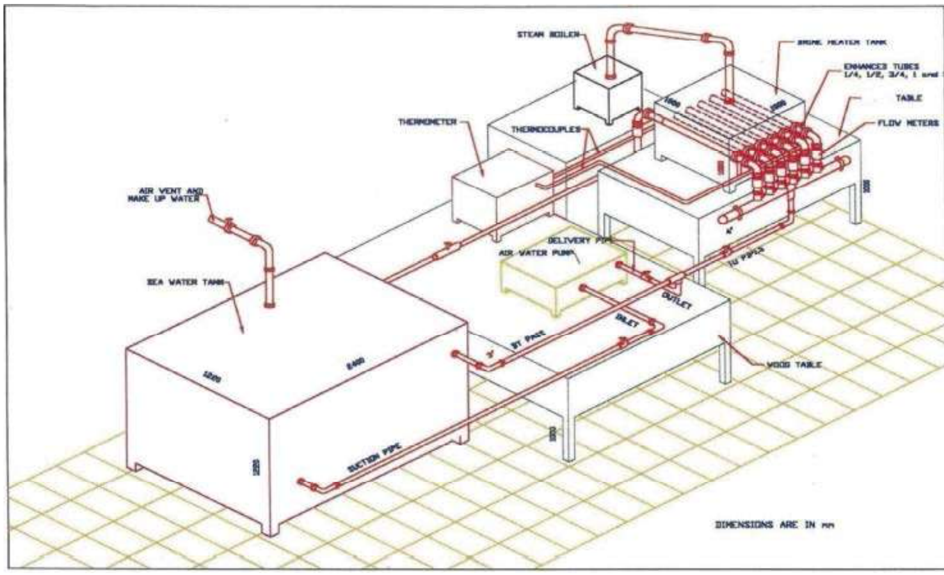

Fig. 3: Experimental Lay-out [12]

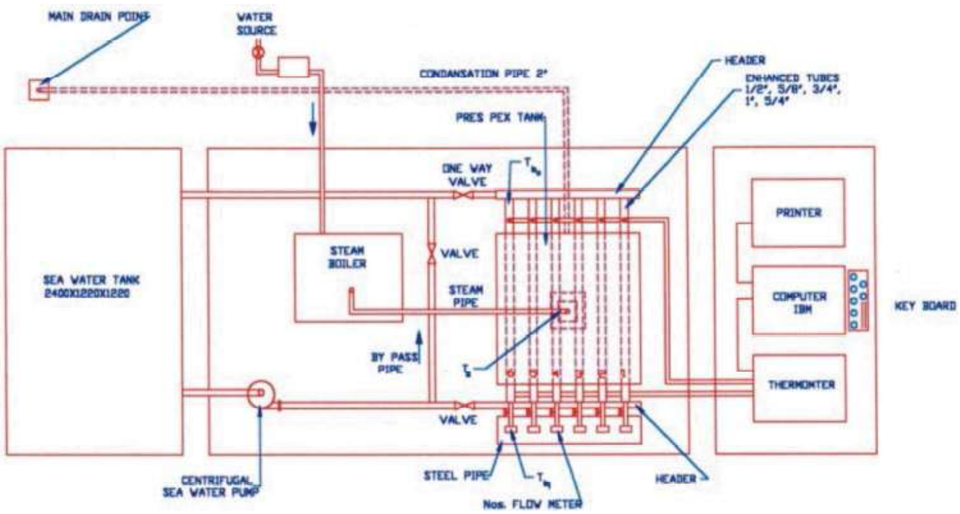

Fig. 4: A Horizontal plan of the experimental set-up [13]

Fig 5: Used Sort of korodense tubes [15]

measurement and control link units via a card containing nine channels. Each thermocouple is calibrated and connected to a channel in this card. Seven channels are used in our case, one for boiler outlet, one for the tubes inlet and one for each tubes outlet and three inside the condenser. Every channel is programmed as required, using software. Used thermocouples are Teflon insulated, with a wire diameter =0.005 inch, length= 240 inch and spade lugs termination. Denoted range for inlet and outlet water temperature, where the inlet temperature of usage water to the tested tubes, is ranged between 20 and 40°C. The outlet temperature of water from tested tubes expected to be less than 100°C. The test rig shown in Fig. 3 and 4 is instrumented in such a way that temperature at all important points in the apparatus (inlet and outlet of each tube and the boiler outlet) can be monitored, as well as the flow rate of all the fluid streams. Moreover, the measurements control system and is used to take the required temperature automatically as programmed. This control system consists of a controller, a measurement and control link units. The controller and measurement units work together via the control link units. Software, to form a flexible data measurement and control system for up to 100 input channels including a maximum of 32 digital channels. In addition, the software developed package provided allows the user to write his own algorithm to optimize the system to a particular application. The used type of tube is called Korodense. This is a specialized type of corrugated tubing. It was designed to improve heat transfer efficiency for certain applications, particularly in steam condensers, desalinated plant condensers and author similar plants applications [14]. In current experimental work, used tubes are developed by Wolverine Tube Inc. Figure 5 shows similar types.

Experimental Procedure adnd Test Results: Current experiment data is reordered every half-minute, for seven channels during the test. These data are averaged and printed out every half an hour for every channel. Used tubes are aluminum-brass with inside diameters 19.05, 23 and 29.5 mm. Investigation is done for three different coolant velocities; 0.1, 0.1645 and 0.2398 m/s. To gain experience with heat performance of enhanced korodense tube against smooth; fresh water is examined first as a coolant liquid. Smooth and korodense with the same diameter are mounted on the rig and is tested simultaneously with the same coolant speed. The control valves used on both tubes are adjusted sequentially to obtain the required and same coolant speed for both tubes. The results for this study are presented in Figures 6-10. Results for different tube inside diameters of 19.05 mm, 23 and 29.5 mm with flow speed 0.2398 m/s, when fresh water is applied are shown in Figures 6 and 7. Figure 6 highlights the smooth tube data results vs. time, with given three above mention diameters on the chosen critical speed 0.2398 m/s. The figure presents the variation of the overall heat transfer coefficient U with time, associated with a tube inner diameter 29.5 mm and the variation of U with time for tube diameter 23 mm, which has higher values than that of the tube diameter 19.05 mm. These results are acceptable by the fact that as tube diameter increases, $\dot{m}$ increases consequently and accordingly U increases. Figure 7 highlights the data results for korodense tube, when fresh water is applied for

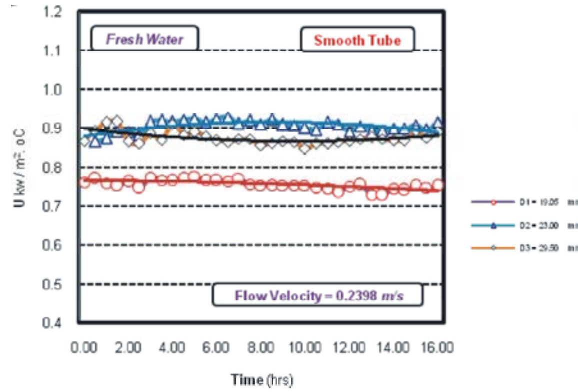

Fig. 6: The Overall Heat Transfer Coefficient vs. Time, utilizing smooth tube with fresh Water for the given three tube diameters, when flow speed=0.2398 m/s

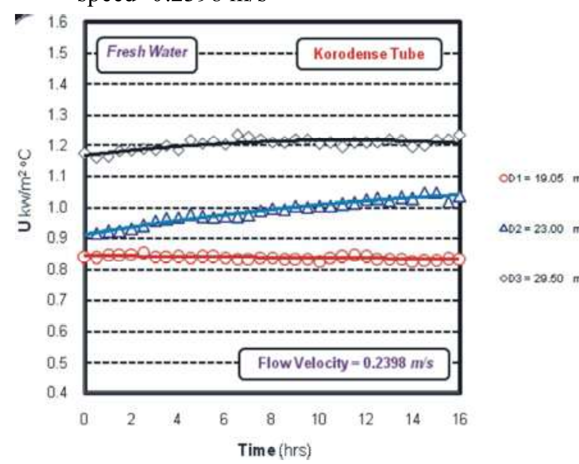

Fig. 7: The Overall Heat Transfer Coefficient vs. Time, utilizing Korodense tube with fresh Water for the given three tube diameters, when flow speed=0.2398 m/s

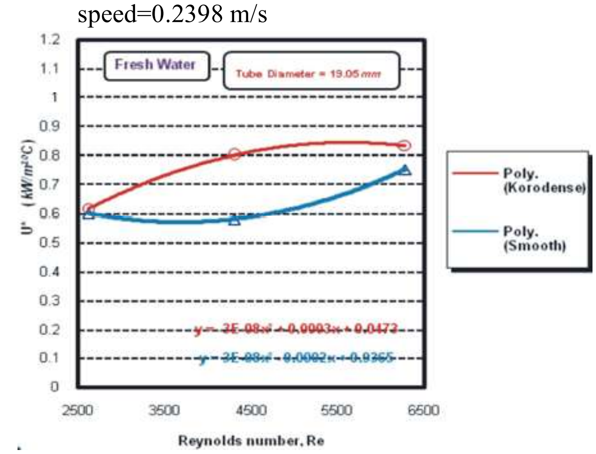

Fig. 8: Asymptotic Overall Heat Transfer Coefficient vs. Re, applying fresh water, with D=19.05 mm tube for both tubes, on the three studied flow velocities

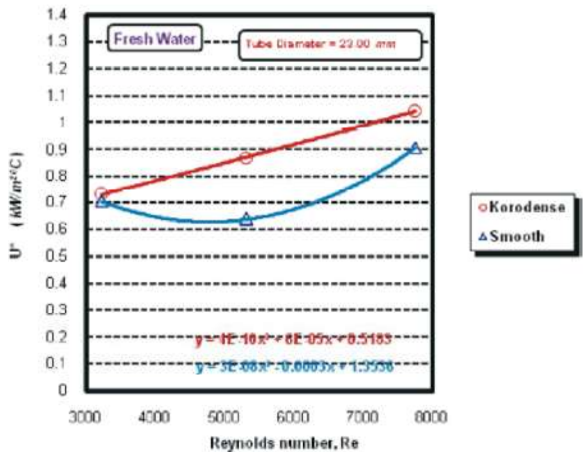

Fig. 9: Asymptotic Overall Heat Transfer Coefficient vs. Re, applying fresh water, with D=23.00 mm tube for both tubes, on the three studied flow velocities [13]

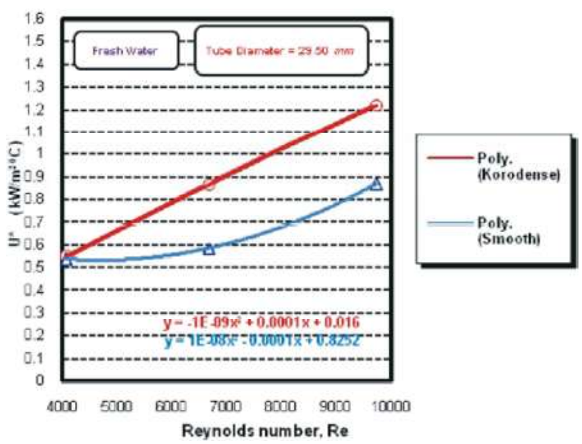

Fig. 10: Asymptotic Overall Heat Transfer Coefficient vs. Re, applying fresh water, with D=29.50 mm tube for both tubes, on the three studied flow velocities

the three above mention diameters, on the same previous chosen flow speed 0.2398 m/s. Here it gave clearer in difference values, agreed with the given theoretical formula. As seen from the figure, the highest overall heat transfer coefficient values are when D = 29.5 mm. The intermediate values of the overall heat transfer coefficient are obtained when D = 23 mm. The lowest values are achieved for the tube diameter equal 19.05 mm. Comparing Figure 6 and 7, we shows higher values of U for korodense tube over smooth by a quite margin. To compare results over different speeds on certain given diameter, for both smooth and korodense tubes when fresh water is applied, asymptotic values of the overall heat transfer coefficient U^* at different speeds with tube diameter 19.05 mm are plotted in Figure 8. As seen in this figure, the U^* for korodense tube is higher than smooth on the three tested speeds. However, the biggest difference between both tubes is accomplished on the intermediate speed of 0.1645 m/s. The difference between both tubes is getting narrower at lower and higher speeds. The data presented in Figure 9 shows repeated former test result, but with tube inside diameter of 23 mm on same studied coolant speeds. Same trend results are repeated as those data presented in Figure 8. Figure 10 shows again the same trend results displayed before on both figures 8 and 9, but with tube diameter 29.5 mm. Here, our data coincide with fluid flow laws. As a result with $\dot{m}$ increases, consequently the performance of a korodense tube improved dramatically as flow velocity increased.

Figure 11, presents the performance of the most effective smooth tube inside diameter, where D = 29.5 mm. The figure shows the overall heat transfer coefficient versus time, given for different studied flow speeds, with utilizing brine water. Figure 12 highlights the overall heat transfer coefficient versus time, given for different studied flow speeds, for korodense tube with inside diameter = 29.5 mm, again when utilizing brine water. The figure shows a better performance of the tube with higher values of the overall heat transfer coefficient versus time on different studied flow velocities. Figure 13 presents the performance of both smooth and korodense tubes in the form of the asymptotic values of the overall heat transfer coefficient U^*, when using brine water versus Reynold's number, utilizing the smallest inside diameter of the studied tubes; 19.05 mm, on the given coolant flow velocities 0.1, 0.1645 and 0.2398 m/s. The figure shows better performance of korodense tube over smooth, with narrower values at low speeds while wider gape values at higher speeds. Figure 14 is a repetition of Figure 13, but with a tube diameter 23 mm, on different studied flow velocities. The figure shows a better performance of the korodense tube on different studied coolant speeds. Figure 15 again is a repetition of the former two figures, but with a tube diameter 29.5 mm, where it shows a better performance of the korodense tube over smooth on different studied coolant flow velocities. The above results indicated that the korodense tube's heat performances are superior to smooth tube for most of the studied diameter sizes; 19.05, 23 and 29.5, on given coolant speeds investigated; 0.1, 0.1645 and 0.2398 m/s. A better overall heat transfer coefficient can be accomplished using enhanced korodense tube. A gain in the value of U can be achieved using enhanced tubes over smooth tubes. Increasing tube diameter and/or coolant speed, increases the overall heat

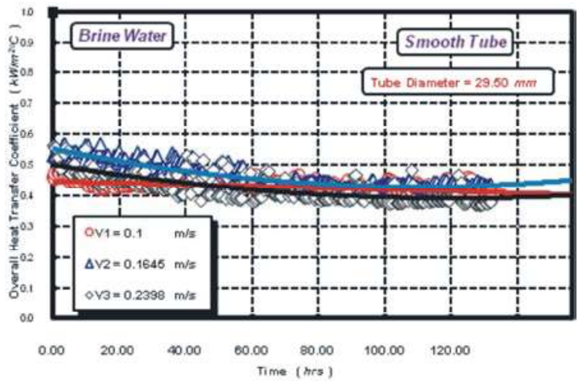

Fig. 11: The Overall Heat Transfer Coefficient vs. Time, utilizing smooth tube with Brine Water for the given three studied flow speeds, when the tube diameter= 29.5 mm

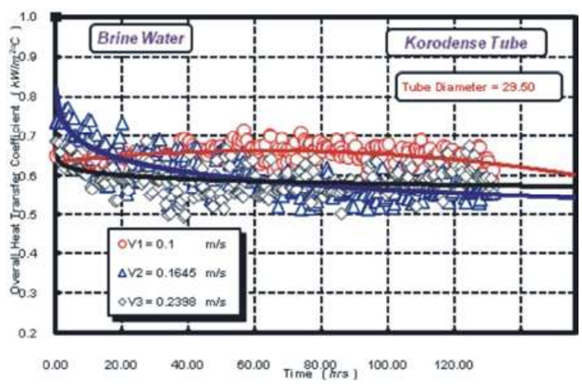

Fig. 12: The Overall Heat Transfer Coefficient vs. Time, utilizing Korodense tube with Brine Water for the given three studied flow speeds, when the tube diameter= 29.5 mm

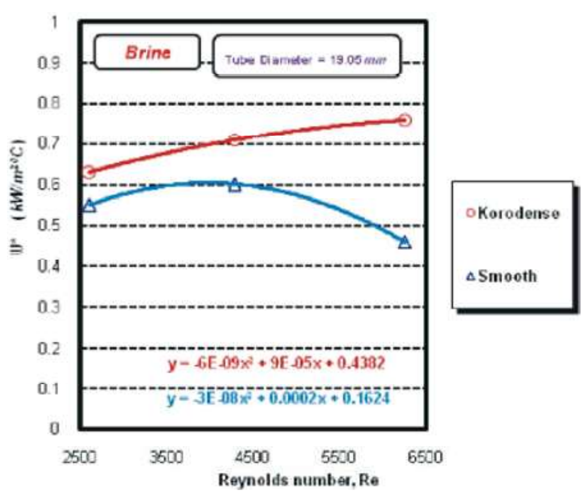

Fig. 13: Asymptotic Overall Heat Transfer Coefficient vs. Re, applying Brine water, with The tube diameter= 19.05 mm for both tubes, on the three studied flow speeds

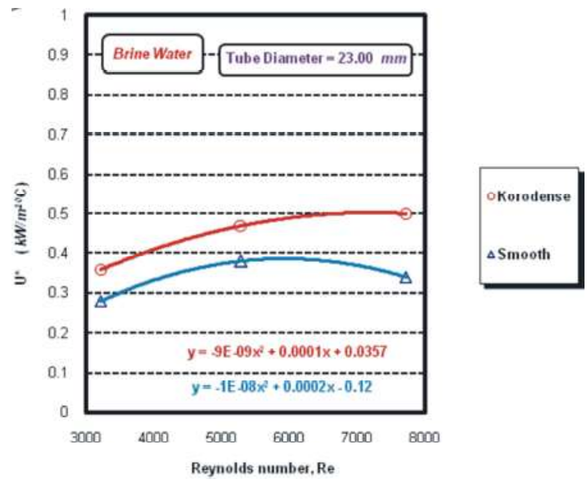

Fig. 14: Asymptotic Overall Heat Transfer Coefficient vs. Re, applying Brine water, with The tube diameter= 23.00 mm for both tubes, on the three studied flow speeds [13]

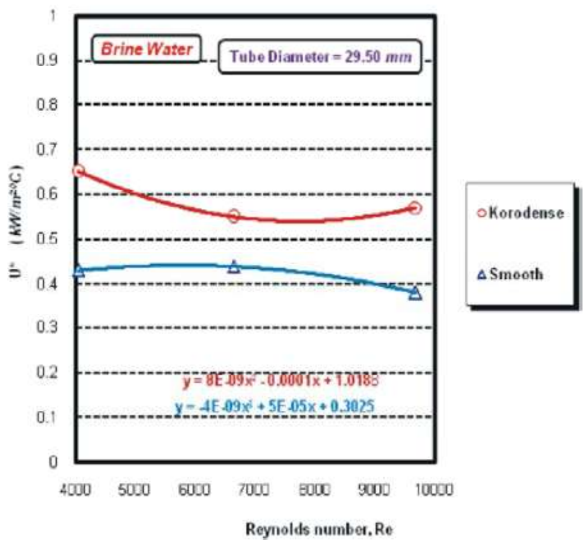

Fig. 15: Asymptotic Overall Heat Transfer Coefficient vs. Re, applying Brine water, with The tube diameter= 29.50 mm for both tubes, on the three studied flow speeds

transfer coefficient for both smooth and korodense tubes. However, the accomplished gains in U for korodense tube when associated with the variation of these two parameters are more significant. In other words, increasing the Reynolds number will improve the heat performance of both smooth and korodense tubes. At low Reynolds numbers the benefits of korodense tubes over smooth tubes are not significant and the results for both tubes are not quite different. However, at higher Reynolds numbers the korodense effect improves the local heat transfer coefficient. Consequently, leads to values of overall heat transfer coefficients for korodense tubes much advanced than that of smooth.

CONCLUSIONS

A heat performance study for smooth and korodense aluminum-brass tubes is done. Use is made of a simulated test rig for the condensation technique of multi-stage flash system. All running MSF systems are designed to use smooth tubes, this text support research efforts spent for koredense tubes replacement. The current study indicates that both tube diameter and coolant flow speed influence the mechanism's overall heat transfer coefficient. The higher the coolant speed, the higher the overall heat transfer coefficient. The larger inside diameter, the higher the overall heat transfer coefficient. However, the effect of these two parameters is more significant with korodense tube rather than smooth. Thus, a better overall heat transfer coefficient can be accomplished using korodense tube. In other words, increasing the Reynolds number improves the heat performance of both smooth and korodense tubes. At low Reynolds numbers the benefits of korodense tube over smooth tube are not significant. However, at higher Reynolds numbers the tube corrugations improve the local heat transfer coefficient. Consequently, leads to values of the overall heat transfer coefficients for korodense tube much higher than smooth tube. The gained research experiences through this work provide a base for a better performance of the MSF system, when using korodense tubes with real brine water as a coolant medium, while simulating actual real life environment.

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