World Applied Sciences Journal 18 (9): 1296-1309, 2012
ISSN 1818-4952

DOI: 10.5829/idosi.wasj.2012.18.09.273434

Design Development for Enhanced Desalinate Tubing Test Apparatus

A. Calendar, T. Galal, A. Al-Saftawi, M. Zedan, S.S. Karar and R. El-Shiaty

Department of Power and Ref., College of Technological Studies, Kuwait

Abstract: The aims of current research programme are itemised as to design a research rig simulates the desalination heat exchange process, assess the feasibility of enhanced heat transfer mechanisms in MSF plant, examining actual brine water that typifies real process conditions, study the influence of flow speeds on fouling resistance and the effect of tube diameter on its behaviour. The first apparatus designed is test-Rig (A), consists mainly of a cubic box simulating the steam condenser. Six horizontally tube are mounted, through which the coolant solution is flowing and the vapour from a boiler source is condensed outside the tubes, but inside the box. These tubes are aluminium-brass made. The test rig is instrumented in such a way that temperature at all important points in the apparatus can be monitored, as well as the flow rate of all the fluid streams. The thermocouples are inserted in the important points. Experimental results are discussed for both smooth and corrugated tube, when applying fresh and real brine water on different flow speeds. This apparatus has to be redesigned, since buckling, as well as cracks developed in the plixcy-glass box's wall due to thermal stresses, existence of mutual heat transfer between tested tubes, existence of non-uniform distribution of steam through-out the box, venting valves need to be added to water circulation loop, modification of pumping system through automation and redesign a water reservoir to keep the inlet temperature to the rig constant. The problems revealed above with using test-rig A, are given a careful consideration in assessment and assembly of test-rig (B). The modified set-up facilities included a new configured design for the steam condenser, equipped in such a way to minimize mentioned obstacles, dismantling faced problems associated with; circulated pumps, boiler, flow meters and its place of fitting, careful identifying the tested tube characteristics, a wise differentiation between properties of brine and fresh water, a creation of uniform steam temperature distribution inside the rig, preventing of two phase flow creation and simulating the actual circumstances in real desalination platform. The data collection system previously described is used in conjunction with the test rig (B). Use is made of two horizontally mounted tubes through which the coolant solution is flowing. Corrugated and smooth tubes are examined at a time to unify the tested conditions on both. A three different coolant flow velocity of 0.1, 0.1645 and 0.2398 m/s are examined. The study is carried-out for two different coolants, fresh and brine water. The effect of fouling of actual Brine water with the avoidance of using artificial fouling, as well as the effect of used corrugated tube with its chosen comprehended roughen enhancement, effect of changing tube diameters and Reynolds numbers on experimental data results are provided in the form of: overall heat transfer coefficient vs. time for both tubes, the rate of condensed water mass out of corrugated tube with respect to the rate of condensed water mass out of smooth tube. The fouling resistance vs. time, the overall enhancement ratio vs. time, the cleanliness factor vs. time and the overall fouling resistance ratio vs. time. Chosen applications are included in this paper.

Key words: Enhanced tube · Condenser Tubing · MSF system · Desalination

INTRODUCTION

The majority of research on fouling has been conducted in smooth tubes. Brine water flows inside the tubes of the condenser, is being heated by the condensing steam on the outside of the tubes [1]. Considerable amounts of fouling are formed on the inside surface of the tubes, acting as thermal insulation, decreasing the thermal efficiency of the system. In this study, an experimental investigation is carried out, displaying the essence of fouling influences on the performance of the smooth tube condenser. To improve tubing heat transfer performance in MSF system, enhanced heat transfer technique is been

Corresponding Author: T. Galal, Department of Power and Ref., College of Technological Studies, Kuwait.

suggested for use. The idea of using enhanced tube if proved; it would be a life-time energy saver. Enhanced heat transfer surface has a special geometry, provides a higher heat transfer coefficient or a wider surface per unit space area's than a traditional design. Enhanced heat transfer mechanisms have developed to be used in heat exchanger of refrigeration and automotive industries. Such technique can be used with any heat exchanger. However, each must be tested to see if this particular application makes sense. Unfortunately all current MSF systems are designed using smooth surface tubes, mainly because of the complexity effects of fouled brine water on available heat transfer mechanisms. Enhanced technique and fouling versus heat exchanger have been of interest for a number of years and for a number of researchers [2-16]. The above mention facts have influenced our research direction and throw lights on the current work. This includes: a) choosing our tube roughened enhancement, b) study the effect of flow speed on performance characteristic of the multi-stage flashing system and c) utilizing actual coolant brine, avoiding the use of artificial fouling will not leads to misleading results. Achieving good results will support current idea of trying enhanced tube surface on a distillation unit with multi-stage flashing system. In other words, the approach of current research programme is itemised to use a research rig that simulates one stage of an MSF desalination heat exchange process. Assess feasibility of enhanced heat transfer mechanism, using actual brine water that typifies real process conditions. Study the influence and effect of flow speeds on tube performance and fouling resistance. Future study is to examine the effect of both tube material and diameter on fouling resistance. Formulation of fouling is by no means an easy task due to the number/ types of parameters involved. Enhanced tubes have been discussed in many publications. The analytical treatment of these two subject areas is complicated. However, an experimental study; fouling vs. smooth pipes and fouling vs. enhanced pipes, provides the basic foundation for comprehension, with the prospective formulation of an empirical engineering model to describe its behaviour. Rudy T. M. [17] in his condensate retention test used an apparatus where condensation occurs inside a horizontal bell-jar that is flanged to a flat brass back-plate. This provides a test cell that can be pressurized. The test sections of integral-fin tube are approximately 51mm long. Insulated 3.2 mm copper tubing carries the cooling water through the back-plate to and from the test tube. A pan, positioned under the test tube, collects the condensate to allow calculation of the liquid loading and heat flux. Vapour is generated by a cartridge–heater in the lower portion of the test cell. Thermocouples are inserted to measure the cooling water temperatures in and out. Chamra L. M. [18] in his experimental work used an apparatus which is capable of testing four 19 mm O.D. or three 22 mm O.D. tubes simultaneously. The instrumentation includes an on-line data acquisition system driven by a personal computer, which is used to measure thermostat and pressure transducer outputs. The apparatus is designed to operate and record data 24 hour/day. Heat is transferred to the tube long test sections by condensing the coolant on the annulus side of the test section. Condensed coolant is returned to electric heated boilers. Wei Li [19] used an apparatus to study the effect of fouling in enhanced tubes in cooling tower systems. A fouling heat exchanger is connected in parallel with the 250 ton chillers. The fouling test tubes are installed in the heat exchanger. Eight special I.D. enhanced tube geometries are made by Wolverine for the fouling tests. The aims of the approach for current research can be itemised as follows: design a research rig that simulates the desalination heat exchange process to study and assess the feasibility of enhanced heat transfer mechanisms in MSF plant, with: examining actual brine water that typifies real process conditions, influence and effect of flow speeds on fouling resistance [20-25]. Discussion is concerned with the design development of our test-rig (A) to test-rig (B). A comparative analysis for the overall heat transfer coefficient, obtained through testing portable water in smooth tube, with equivalent brine water in a similar tube are discussed on both designs. Reported measurements and results provide in global the methodology as well as parametrical interpretation, where many new proposition, inquiry and problems had been identified. Also, discussion embraced some of these new problems. Revealed suggestions and recommendations by the acquired test-results (A) are given careful consideration in carrying-out the design, calibration and measurements for test-rig (B) is highlighted.

Experimental Set-Up of Test–Rig (A): Figure1 shows a condenser heat exchanger Tubing. A three dimensional diagram for the lay-out test-rig facility is shown in figure 2, the test rig with an over-view of the main facility's components are shown in figure 3. The apparatus consists mainly of a boiler and six horizontally mounted tube through which the coolant solution is flowing and the vapour condenses outside these tubes, but inside a cubic box (1*m* x1*m* x 1*m*) simulating the steam condenser (brine heater). These tubes are 1m long, with a nominal (commercial) inside diameter of ¾ inch. A closed graph for the tested tubes is presented in figure 4.

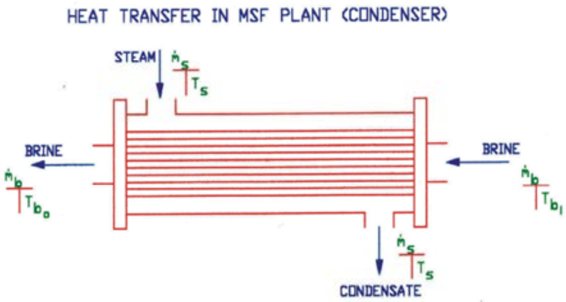


Fig. 1: Tubing of a Condenser Heat Exchanger

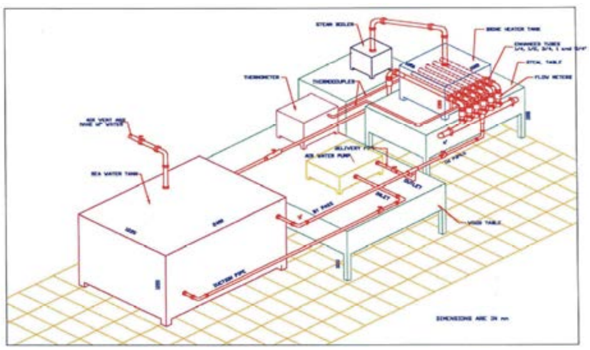


Fig. 2: Experimental layout.

1-Circulating pump
2-Steam Boiler
3-Corrugated tube
4-Flow meters

Fig. 3: Overall view of test rig-(A).

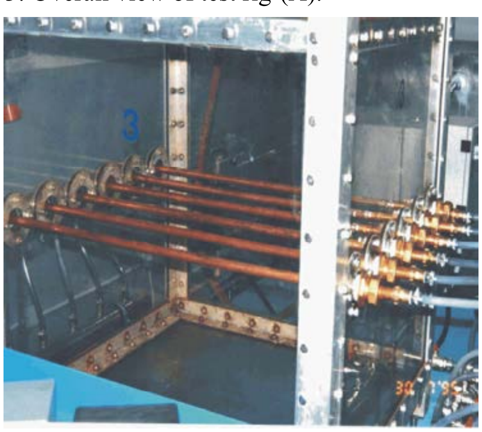


Fig. 4: Close graph of tested tubes location

Overall Heat Transfer Coefficient: Reported equations by references [1] display expression for overall heat transfer coefficient, function of system properties as follow:

$$U = \frac{\dot{m}\, C_P}{A} \ln\left[\frac{T_s - T_i}{T_s - T_o}\right], \qquad kW/m^2\,{}^oC$$

With, A is a nominal inside surface area, with $A = \pi\, D_i, L$, D_i is tube inside diameter and L is a tube Length. $\dot{m}$ is coolant mass flow rate and Cp = 4.2 *kJ/kg. °C* for fresh water coolant, while Cp for brine water is been calculated function of the temperature as given by ref. [20]. T_S is saturated vapour temperature. T_i and T_o are coolant inlet and outlet temperature respectively. Above system properties; $\dot{m}$, T_s , T_i and T_o are determined by measurement. Consequently, overall heat transfer coefficient U is determined from equation above. Current experimental data is recorded every half-minute for seven channels during test and these data are averaged and printed out every half-hour for each channel.

Involved Parameters: The pre-measured parameters recorded during the experiment, are listed in Table 1.1. Temperatures are recorded and stored each half-second. A first average is taken for each sixty readings, i.e. in each half-minute. Consequently, a second average is processed for each sixty readings of those recorded every half minute, i.e. in each half hour.

Table 1.1: Pre-measured parameters.

Used coolant	Fresh/Brine water
Recorded rates of mass flow " *mb* " &	0.522 L / min
Flow Speed "V"	0.035 *m / sec*
Fresh & Brine water inlet temperature " T_{bi}	20°C

Where:

- Flow speed: with the inner diameter of the tested tube is; D_i = 0.0177 m, with the recognition of the scale graduation on the flow meter tube = 150 mm, for this case it is equivalent to 0.521.6 L/min. Table 1.2 highlights the tested tube data.
- Flow Rates are determined by reading the scale graduation at the center of the float. The maximum scale graduation given is 150 mm, which corresponds to 521.6 mL /min.

Table 1.2: Test tube information.

Materials	Copper-alloy
Sort	Smooth
Dimensions	Inner diameter (D_i) = 17.70 *mm*,, Outer diameter (D_o) = 22.2 *mm*
Tube length (L) = 1.0 *m*	
Number of processed tubes	6 tube
Number of tubes for data correlations	4 tube (*tube no.2,3,4 and 5, excluding no.1 and 6)*

Calibration:

- Flow meters: water is pumped, passed through the flowmeter and collected in a container for a given duration time. The weight of water is measured with a Dial- O – Gram balance and converted onto flow rate in *mL/min*. The determined accuracy is ± 0.03.
- Thermocouples: are calibrated at two points, i.e. 0 *°C* and 100°C. Where, the accuracy of basic measurements is encountered as follows:
- Inlet temperature of water (*°C*) " T_{bi} " ± 0.1 to 0.2
- Outlet temperature of water (*°C*) " T_{bo} " ± 0.3
- Steam outlet temperature (*°C*) " T_s " ± 0.5

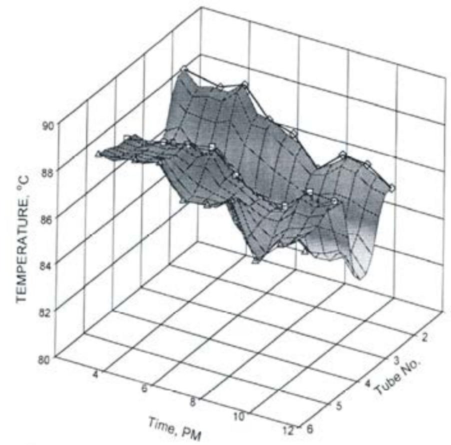
Test Procedures:

- Check the main tank for water supply, to make sure it is opened.
- Open the header valves.
- Starting the circulating pumps on.
- Put on the data acquisition system and run the computer program.
- Make sure that the boiler steam outlet valve and the water drain valve is closed. When the pressure inside the boiler $P_b = P_{max} > P_d$ the demanded outlet pressure, start control of the steam outlet Valve by slit opening, until you possess an outlet pressure = P_d is recorded.
- Also, indicate the temperature of the outlet superheated steam and make-sure it is steady.

Plotted Data & Results [26]: Following are collected data plotted on three dimensions, with z axis represents the outlet temperature from the tube or the Overall Heat Transfer Coefficient for each, y axis represents the corresponding tested elapsed time, while x axis feature the number of tubes under test. Experiment is done to show the temperature variation through each tube, the overall heat transfer variation through the tube, compare the values of overall heat transfer coefficient as well as the difference exist between Fresh water collected data and Brine water collected data, and finally to show the effect of fouling on reducing the value of the overall heat transfer coefficient. Presentations are shown on the figures 5-8.

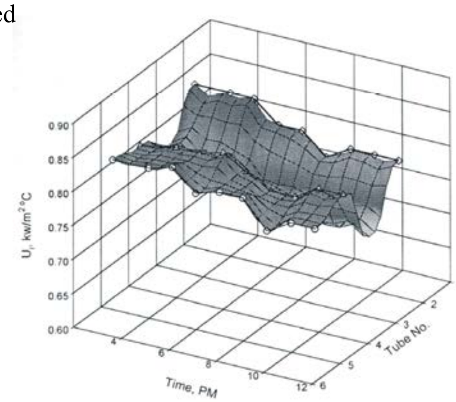
Fresh Water Brine Water

Data Comparison: When potable water is used, figure 9 gives a two dimensional presentation for some chosen/studied tubes. The variation of Ui as a function of time for each tube, i.e the parameter on z axis is the overall heat transfer coefficient versus the other on y axis



Type of used tubes: Smooth
No. Of tested tubes: 5
Inlet Dia. =17.7mm
Outlet Dia. =22.2mm
Inlet fluid temp. =20°C
Mass flow rate=0.522 kg/min

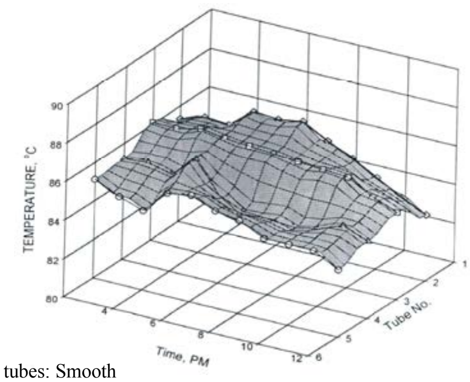
Fig. 5: Outlet Temperature distribution for the tested five tubes Vs. Tube no. Vs. Time When Fresh water is used



Type of used tubes: Smooth
No. Of tested tubes: 5
Inlet Dia. =17.7mm
Outlet Dia. =22.2mm
Inlet fluid temp. =20°C
Mass flow rate=0.522 kg/min

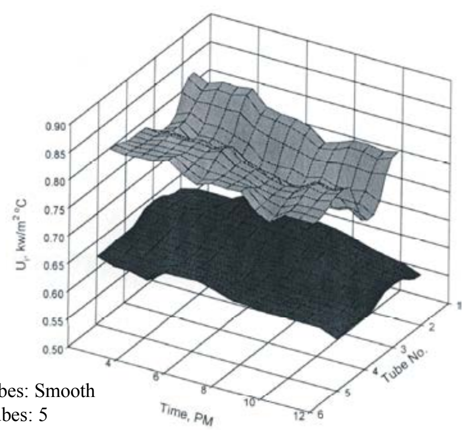
Fig. 6: Overall Heat Transfer Coefficient Vs. Tube no. Vs. Time

which represents the time, these data is collected from the three dimensional presentation of figure 6. While, when Brine water is used, figure 10 shows the variation of Ui as a function of time for some chosen tubes. The decline of the overall heat transfer coefficient for all tubes, with time goes-by until approaching the steady state conditions. While, for potable water in figure 9, system is approaching the quasi-equilibrium state as the time goes-by, but at a higher value of overall heat transfer coefficient.



Type of used tubes: Smooth
No. Of tested tubes: 5
Inlet Dia. =17.7mm
Outlet Dia. =22.2mm
Inlet fluid temp. =20°C
Mass flow rate=0.522 kg/min

Fig. 7: Outlet Temperature distribution for the tested five tubes Vs. Tube no. Vs. Time When Brine water is used



Type of used tubes: Smooth
No. Of tested tubes: 5
Inlet Dia. =17.7mm
Outlet Dia. =22.2mm
Inlet fluid temp. =20°C
Mass flow rate=0.522 kg/min

Fig. 8: Comparison between Overall Heat Transfer Coefficients Fresh and Brine Vs. Tube no. Vs. Time

Sources of Errors: Errors are classified in this work as:

- Uncertainty; available for Test-Rig (B) only.
- Certainty, with sources is given as follows:

***Flow-Meters*:**

- Reading a flow scale graduation is a source of error. To obtain correct flow readings, locate the centre of the float, then read the scale.

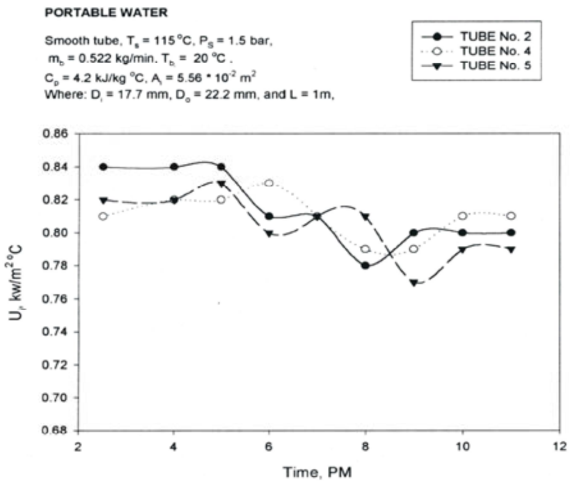


Fig. 9: Overall Heat Transfer Coefficient for Potable water for some chosen tubes Vs. Time

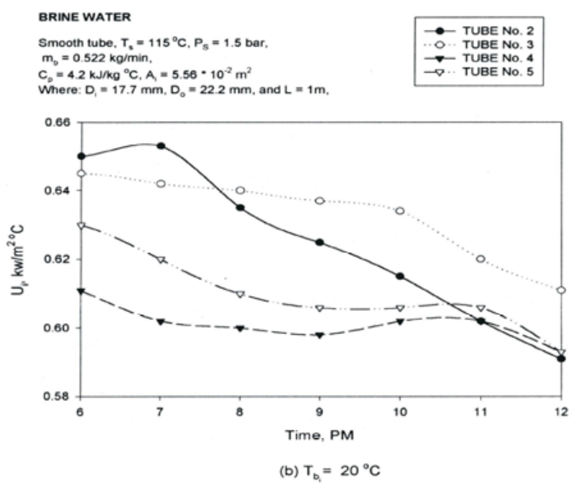


Fig. 10: Overall Heat Transfer Coefficient for Brine water for some chosen tubes Vs. Time

- The clearance between the internal surface of the flow-meter tube and the sphere (float) surface are too tiny, where any formation or deposition of a film could cause sticking.
- Flow meters are fitted up-stream of the tested tubes. This could cause unfilled tube with coolant, producing an error in measurements and a vibrated locator line on scale graduation for the considered flow-meter.

***Flow Temperature, Range and Thermocouples*:** Are given by ref [20]. Disregarded variation for any of the precedent given parameters will accumulate to the deficiency of the measured coolant outlet flow temperature.

- In the comparative analysis; fresh water in a smooth tube vs. brine water in a like tube, where same temperature range is preferable in application, to avoid deficient results.
- Thermocouple's direction of insertion would be a source of error. Example: Insertion in a perpendicular position to the direction of the fluid flow will develop vortices around the tip and consequently producing a bubbled flow.

***Inlet Temperature Variation*:** diversity of coolant inlet temperature is thoroughly due to:

- Wide variation of the seasonable ambient temperature.
- Employing a closed cycle of cooling water.

***Back Pressure to the Flow*:** the vertical assigned level of the down- stream header played role with the existing accuracy of data results. The downstream header is raised up vertically, in relative to the up-stream one, until certainty that the tubes are filled fully with coolant.

***Steam Distribution*:** Steam is introduced into test-rig through a single port on top of the upper-side of the condenser. The design provides unsymmetrical distribution of steam through-out the chamber.

***Ventilation*:** Coolant outlet temperature is significantly higher with test-rig used. Bubbles existence and appearance are also due to the entrainment of air into the circulating cooling system. The bubbles accumulated inside the tested tubes. Bubbled flow altered the measurement characteristics of the main stream, since it gives:

- Misleading temperature
- Misleading flow rate

***Creation of Two Phase Flow*:** The problem of creating such flow is a result of using temperature application, leads the fluid to boiling point. In this experiment avoidance of such problem is taken into account.

***Steady-State Duration Time Period and How to Be Minimized*:** For certain period of time, beginning by system start-up, thermal equilibrium has to be attained, required to define the boundary of real measured parameters.

Fig. 11: Test-rig (A) after a period of operation with buckling, as well as cracks

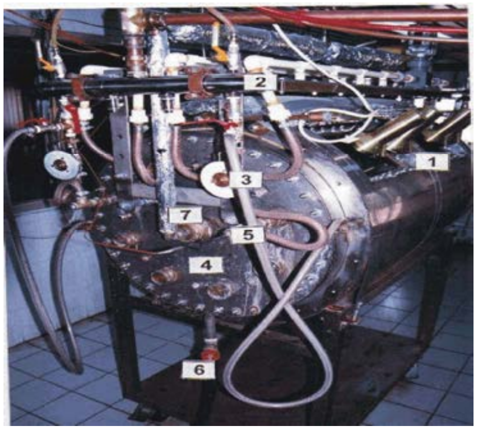
***Mutual Heat Transfer Interaction*:** The horizontal measured distance between two adjoining tube axes is 145 *mm*, see figure 4. The result of such close distance between tubes is a creation of mutual interaction heat transfer effect. Consequently, deficient results are expected.

***Boiler and Used Circulating Pump*:**

- Used type of boiler is the source of errors in measurement, where many problems are countered during its service.
- Used type of pump, if not suitable for fouled water. Accordingly, clogging is happened frequently when brine water took-place in circulation.

The cubic box of test rig (A), simulating the steam condenser has to be redesigned. Since, buckling as well as cracks developed in the plixcy-glass (12 *mm* thickness) box's wall due to the thermal stresses. Figure 11 shows the walls condition, after a period of operation.

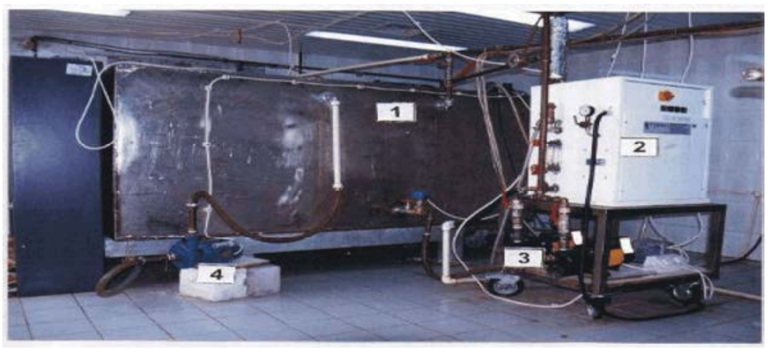
Experimental Set-up of Test-Rig (B): The apparatus consists mainly of a condenser, boiler, circulating pumps with piping loop, tanks, measuring instruments, data acquisition system and a computer. An overall front view of test rig is shown in figure 12, with the type of used tube is given by figure 12.1 and while main components of test rig facilities is presented in figure 13. The condenser consists of a drum, cooling tubes evenly spaced in the drum and steam distribution pipe as illustrated on horizontal plan of the experimental set-up given in figure 14. Location of simultaneous tested tubes; corrugated and smooth, are indicated on a given schematic drawing of figure15, where use is made of two horizontally mounted tubes through which the coolant solution are flowing.



1	*A lighted glass window*
2	*An outlet header*
3	*An outlet control valve*
4	*A backside tube opening flange*
5	*A thermocouple*
6	*A condensate drain valve*
7	*A backside steam inlet*

Fig. 12: An Overall Back View of Test Rig (B).

Fig. 12.1- Enhanced Tube used



1	**Brine water storage tank.**
2	Steam boiler.
3	Brine water circulating pumps.
4	Cooling brine water feed pump.

Fig. 13: Auxiliary components of test rig facilities.

The design is capable for testing eight cooling tubes at a time to unify the tested conditions. They distributed on an equal circular pitch angle. This symmetrical distribution provides the primary means for a uniform coolant temperature distribution, with the avoidance of mutual heat transfer. Where, actual Brine water is used with the avoidance of use the artificial fouling, which leads to misleading results. The geometry of the rig is been devised in a cylindrical shape rather than a cubic one, where better circumstances for similarity and symmetry are obtained. The rig frame is designed using stainless steel 316 in two shells; outer and inner, thickness "2 *mm*", with providing a thermal insulation in between for the purpose of minimizing heat dissipation. Four windows built in the upper half of the considered shelled frame, with a heat resistance glass (up to 700 *°C*) of a thickness "6 *mm"*; to help in watching the development inside the condenser during the experimental duration running time. A steam distribution pipe is located at the centre of the drum, contains holes in its entire surface to distribute the vapour evenly along cooling tubes. Coolant flow rate is measured by using a formed glass tube-type flow meter. Flow rates are determined by reading the scale graduation at centre of the float. Temperatures are to be taken by inserted thermo-couples onto important points, where each thermo-couple is calibrated and connected to a channel in the computer data card. Eight channels are used in our case; four inside the condenser, one for boiler outlet, one for coolant tube inlet, one for each coolant tube outlet, with outlet temperature is expected to be less than 100°C. The test rig with its auxiliary components shown on figures 12 and 13 is simulating actual environmental real life of one stage heat exchange process in MSF desalination. The control system consists of a controller, a measurement and control link units. The controller and measurement units work together via control link units. Software is prepared to form a flexible data measurement and control system for up to 100 input channels. A provided software developed package allows user to write his algorithm to optimise system to a particular application. The test rig (B) is a cylindrical box, used is made of two horizontally mounted tubes through which the coolant solution are flowing and vapour condenses on their outside surface, but inside the cylindrical box of dimension: $d_{internal}$=78 *cm*, $L_{int.}$ =110 *cm*. Steam is introduced to the condenser in-through two central entrances. The steam distributor tubing inside the condenser consists mainly of two overlapped tubes. The inner steam tube is designed to scatter the steam whereas the outer one is built-in for the purpose of avoiding a creation of steam jet, with distributing the steam uniformly though out the condenser. For measuring the steam temperature inside the cylindrical condenser, four thermo-couples are built-in the rig; two on the upper half and two on the lower half. Through this arrangement, it has been proved that the steam temperature is distributed uniformly equally throughout the rig "T_{srig}". The temperature is taken almost constant throughout the experiments also. Where, T_{srig} being adapted by controlling the steam supply valve (in the given undergone experiments, this temperature is checked, then adapted every about eight hours). Used thermocouples are chosen to be capable of standing for corrosive Medias, with covering the required range of measured temperatures. Different tube diameters are considered for trial; 19.05, 23 or 29.5 mm. The data collection system previously described is used in conjunction with the test rig (B). Flow-meters are fitted downstream of tested tubes rather than upstream to avoid previous cited problems; Bleeding valves are added throughout the water circulation loop. A three different coolant flow velocity of 0.1, 0.1645 and 0.2398 m/s are examined. The study is carried-out for two different coolants, fresh and brine water. The fouling experimental data results would be provided in the form of: overall heat transfer coefficient vs. time, the rate of condensed water mass out of corrugated tube with respect to the rate of condensed water mass out of smooth tube, the fouling resistance vs. time, the overall enhancement ratio vs. time, the cleanliness factor vs. time and the overall fouling resistance ratio vs. time. The experimental results are expected to be fitted to a theoretical model to show if a matching exists. These comprehended chosen tubes roughen enhancement, effect of tube diameters and Reynolds numbers influenced the current research direction and throw its conclusions on the performance characteristic of the multi-stage flash system.

Justifications for the Design of a Novel Model (B) of Test-Rig:

- The cylindrical box simulated the steam condenser of test-rig, which is designed with following modifications: a new configuration of steam distribution across the condenser is introduced, owning to the symmetrical distribution and the type of used diffuser. Resist buckling, as well as cracks due to the thermal stresses. Regards is given enough to eliminate mutual heat transfer effect between tubes.
- The design is to help-in reaching the quasi-equilibrium state.
- A creation of two-phase flow, with this design is much more avoided.
- The applied flow speeds could be increased owing to simulating the actual circumstances in real desalination platform. Where the condenser tube long is 17.7 *m*, with the associated flow speed is 2 m/sec. In this rig, the pipe long is taken 1.10 *m* with associated flow speeds are chosen as 0.1, 0.1645 and 0.2398 *m* /*sec* respectively.
- Used corrugated tube is of Wolverine Tube Inc. [27].
- Two sorts of flow meters are used: a classical glass flow-tube and a vortex type to fit the necessity of increased flow speed accuracy.
- A higher discharge capability pump is installed, to overcome the new set of selected range of flow velocities, in addition to reduced the problem of over-heating, by introduced an additional pump on stand-by position, while minimized the running time of each. Developed an automatic switching "on & off "electric circuit to operate both pumps, with the capability of diminishing elapsed interval time.
- A more sophisticated boiler than the former used one is considered for use.
- Bleeding valves is being arranged in several consequence locations up-stream of the tested tubes.
- The flow meters are fitted on down-stream of the tested tubes, rather than up-stream to avoid previous cited problems.

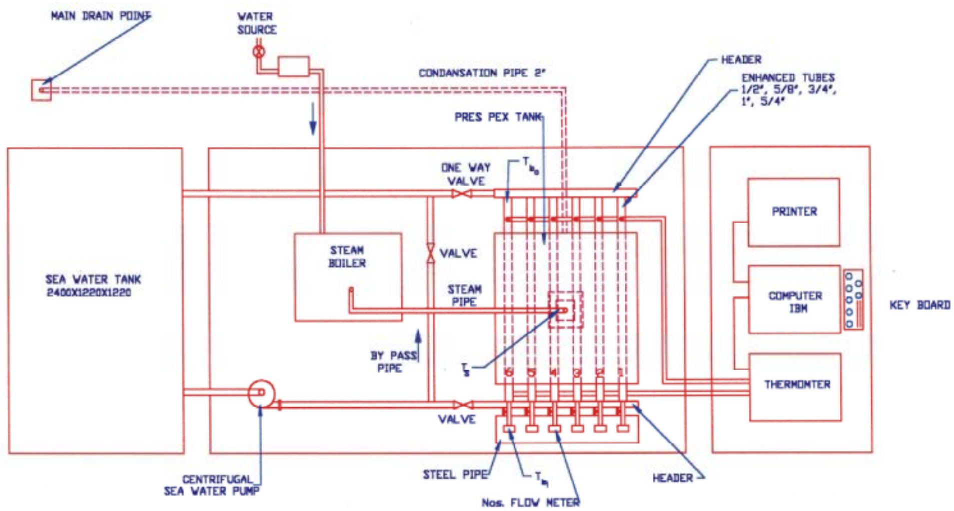
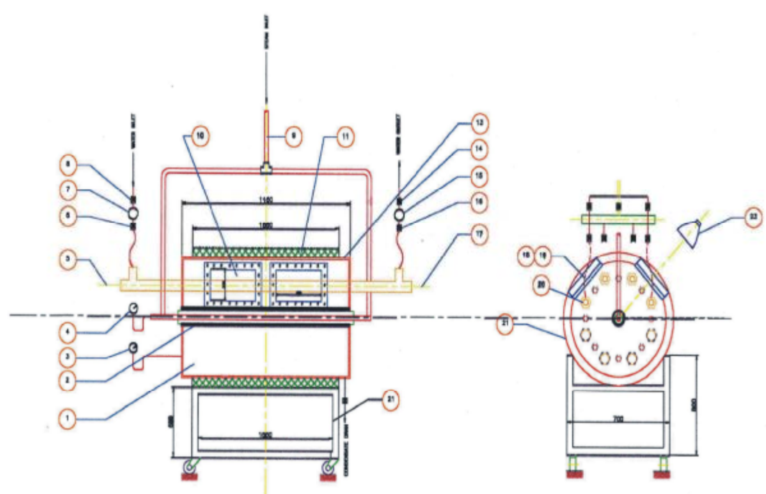


Fig. 14: A Horizontal plan of the experimental set-up.



Position	Item
1	Test Drum
2	Steam Distribution Tube
3	Drum Pressure Gauge (Test drum)
4	Steam Drum Inlet Pressure gauge
5	Water Inlet Thermocouple
6	Inlet tube Control Valve (2)
7	Inlet Header
8	Main Inlet Header Valve
9	Steam Inlet Valve
10	Sight Glass Windows (4)
11	Insulation
12	Lower Test Drum Thermocouple
13	Inner Shell
14	Main outlet Header Valve
15	Outlet Header
16	Steam Inlet Thermocouple
17	Water Outlet Thermocouple
18	Upper Test Drum Thermocouple
19	Opening for Testing Tubes (8)
20	Testing Tubes (Corrugated)
21	Outer Shell
22	Stainless Steel Frame
23	Lamps (4)
24	Testing Tubes (Smooth)
25	Condensate Drain Valve
26	Inlet Venting Valves (3)
27	Outlet Venting Valves (3)

Fig. 15: Schematic Drawing of Test Rig B

Procedures and Test Results: In present work, tube material is aluminium-brass; with inside diameter 23 mm. Both smooth and corrugated with same diameter are mounted onto the rig, tested simultaneously with same coolant speed. To obtain required flow speed on each, control valve mounted on the inlet-side of each tube are adjusted sequentially. Flow speed is chosen to simulate the actual flow process. Upon which, our model investigates three different flow speeds: 0.1, 0.1645 and 0.2398 m/s, in relevant to a tube length of 1100 mm. In actual process (MSF system, Doha east [28] flow speed in the order of 2 m/s, in relevant to a tube length within 18 m.

Overall Heat Transfer Coefficient: Use is made of Fresh water first to gain experience with the heat performance of enhanced versus smooth tube, with achieving the asymptotic value after 16 running hour. On speed 0.1 m/s, figure 16 highlighted the data for enhanced and smooth tubes. The figure shows a slim difference in variation for the Overall Heat Transfer Coefficient *U* between both tubes versus time. Same test is done, but with coolant speeds 0.1645 m/s and 0.2398 m/s respectively. Data presented on Figs.17 and 18, shows variation of corrugated tube's Overall Heat Transfer Coefficient, with significant higher values over smooth tube. *U* values for each tube are almost constant, with exceptional for

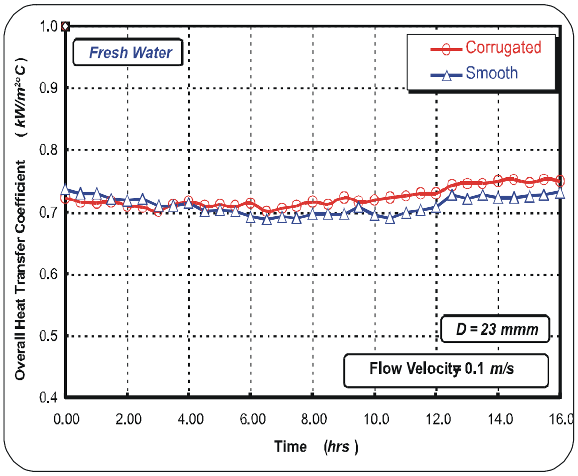


Fig. 16: U versus time for both tubes, with fresh water v = 0.1 m/s & D = 23 mm

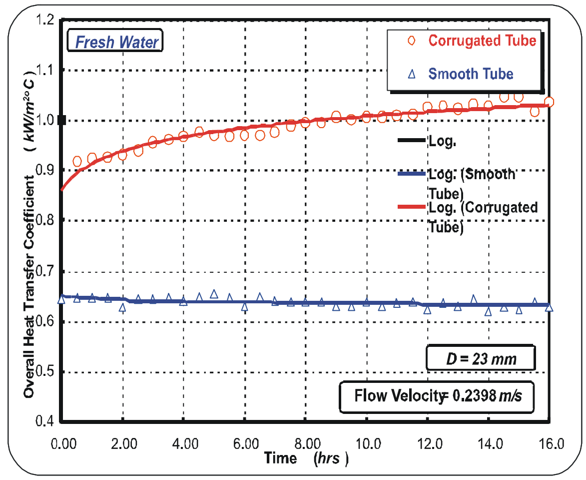


Fig. 18: U versus time for both tubes, with fresh water v = 0.2398 m/s & D = 23 mm

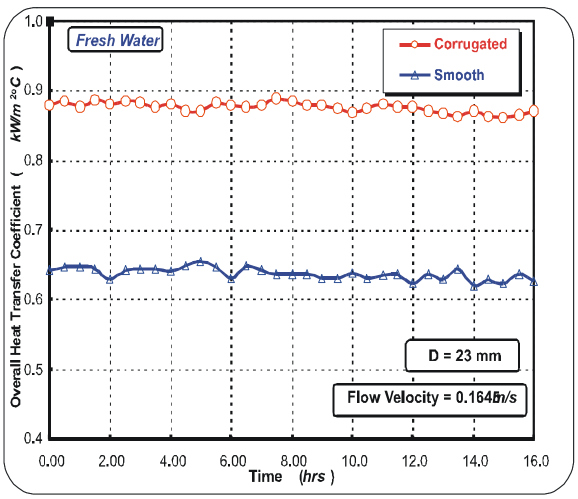


Fig. 17: U versus time for both tubes, with fresh water v = 0.1645 m/s & D = 23 mm

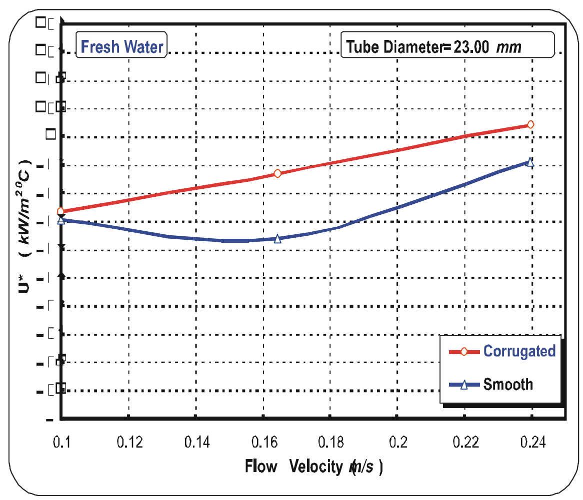


Fig. 19: Asymptotic U* versus flow speed for both tubes, Fresh water, D = 23.00 mm

corrugated tube on speed 0.2398 m/s, since it elapsed through a transition period at the beginning of the test, while reaches the asymptotic value then after. This would be due to low temperature of the tubes at the starting, with relatively high coolant speed. Consequently, asymptotic value of the Overall Heat Transfer Coefficient U^* at different speeds is plotted on figure 19, while figure 20 shows asymptotic value of the Overall Heat Transfer Coefficient U^* is plotted versus Reynolds Number. High values of Reynolds number is due to dramatic drop in water viscosity as temperature increases, due increasing coolant flow speed from 0.1 m/s to 0.1645 m/s and 0.2398 m/s respectively and due slight reduction in density with increased temperature. Figure 19 shows the values of U^* for enhanced tube are significant higher than that of the smooth. Highest difference occurs at speed 0.1645 m/s, with U^* for the enhanced is greater in magnitude than that of smooth by factor of 1.36. Work is repeated, but with applying genuine brine water onto the system. The steady state condition is developed lately, after about 140 running hours. Fig.21 shows variation of U versus time for both tubes on speed 0.1m/s, with tube diameter D=23mm,. Comparing graphs, it is realized the effect of fouling on lowering the values of U versus time by much. Figures 22 and 23 are done, but with speeds 0.1645 m/s and 0.2398 m/s respectively. Through the above given graphs the difference between the values of U for smooth and enhanced on specified speeds 0.1 m/s and 0.1645 m/s are not significant. However, on flow speed 0.2398 m/s, the values of U are been raised dramatically over given values

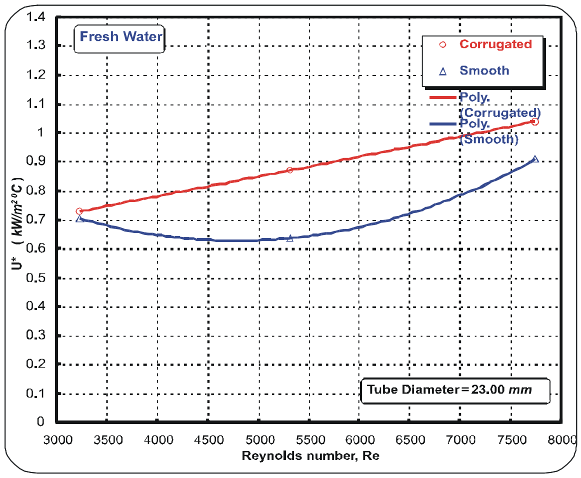


Fig. 20: Asymptotic U* versus Re for Both tubes, Fresh water, D = 23.00 mm

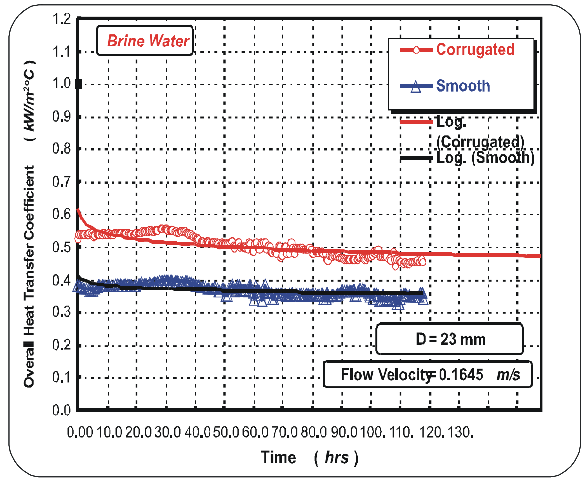


Fig. 22: (U) versus time for both tubes, with brine waterv = 0.1645 m/s & D = 23 mm.

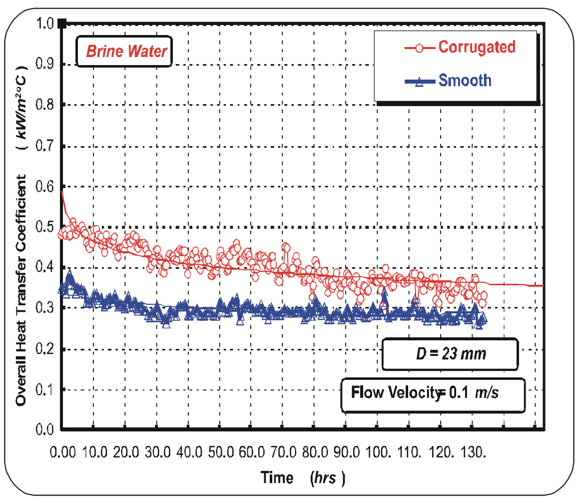


Fig. 21: (U) versus time for both tubes, with brine water, v = 0.1 m/s & D = 23 mm

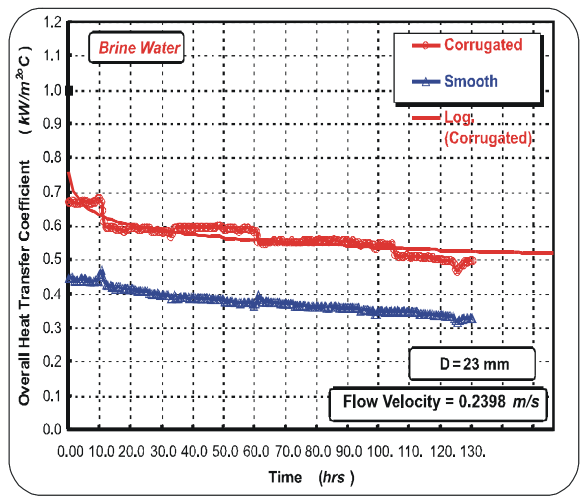


Fig. 23: (U) versus time for both tubes, with brine water, v = 0.2398 m/s & D = 23 mm.

for flow speeds 0.1 m/s and 0.1645 m/s. Also, the effect of fouling are reduced versus time as flow speed increased. Asymptotic values of U^* at different speeds are plotted on figure 24, while figure 25 is plotted versus Reynolds number. On figure 24, as coolant flow speed increases in corrugated tube, the asymptotic values of the overall heat transfer coefficient U^* increases. As a result the values of U^* for enhanced are much higher than that of the smooth tube. The highest difference occurs at velocity 0.2398 m/s, with the value U^* for enhanced tube is higher than that of smooth tube by a factor of about 1.5.

Condensed Water Mass Ratio [24]: With m_{wc} is the condensed water mass by corrugated tube (*c*), while m_{ws} is the condensed water mass by smooth tube (*s*), where [24]:

$$\frac{m_{w,c}}{m_{w,s}} = \frac{m_b C_p \left(\Delta T_{b,c}\right)}{m_b C_p \left(\Delta T_{b,s}\right)}$$

Fig. 26 shows the undergoing change of condensate mass ratio $m_{w,c}/m_{w,s}$ versus time, illustrating the rate of condensate water out of corrugated tube is the more significant. Therefore, the rate of condensate water collected out of the specified enhanced tube with respect to the rate of condensate water collected out of the smooth tube varies as shown on the figure between 1.2 up to 1.44.

Fouling Resistance Versus Time [1, 24]: Fouling resistance is typically performed by measuring the total thermal resistance *(1/UA$_i$)* for clean and fouled cases, Webb [1], using the following formula:

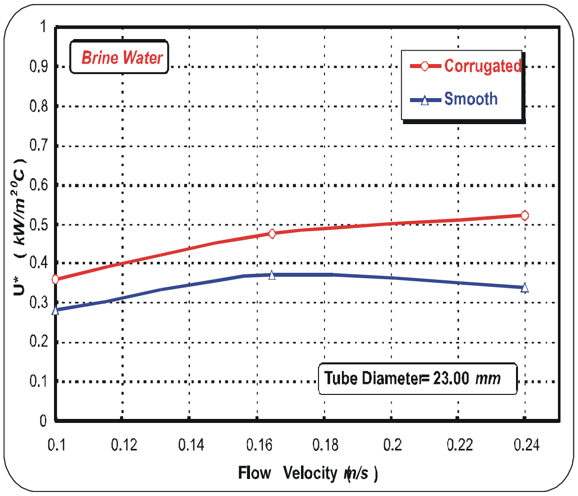


Fig. 24: (U*) versus flow velocity, for both tubes, Brine water, D = 23 mm

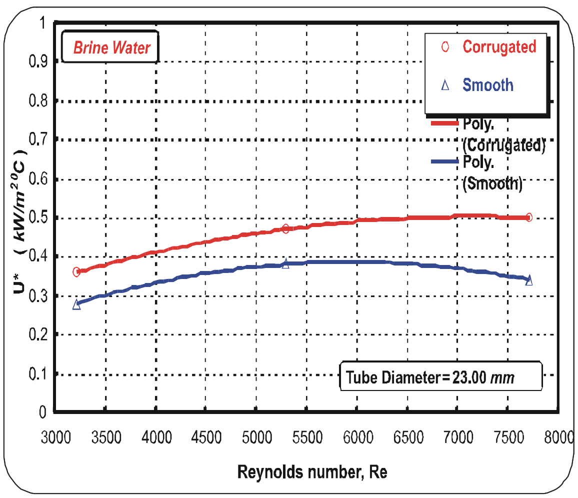


Fig. 25: (U*) versus Reynolds number for both tubes, Brine water, 23 mm

$$R_f = \frac{1}{(U)_f} - \frac{1}{(U)_{cl}} \quad {}^oC/kW$$

Where, subscript f and cl refer to fouled and clean conditions. Results for fouling resistance (R_f) for both corrugated and smooth tubes are illustrated on figure 27, as shown fouling thermal resistance for both tubes increases with time, then attain about a constant asymptotic value after 144hr. The figure also shows the performance for corrugated tube is superior to that of smooth tube.

CONCLUSIONS

The originality significance of this manuscript is to design a research test-rig to experimentally simulates the

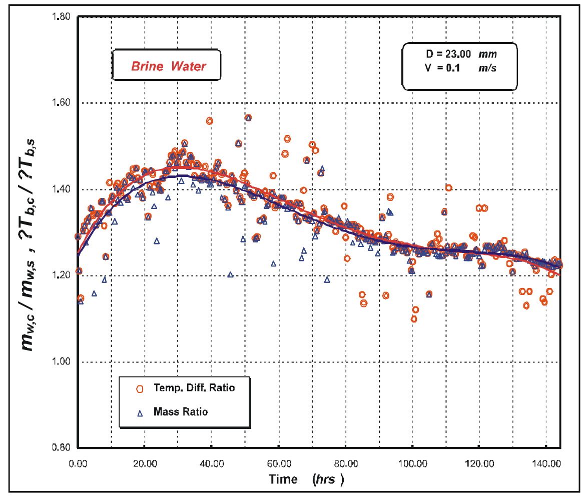


Fig. 26: The condensate mass ratio $(m_{w,c}/m_{w,s})$ versus Time.

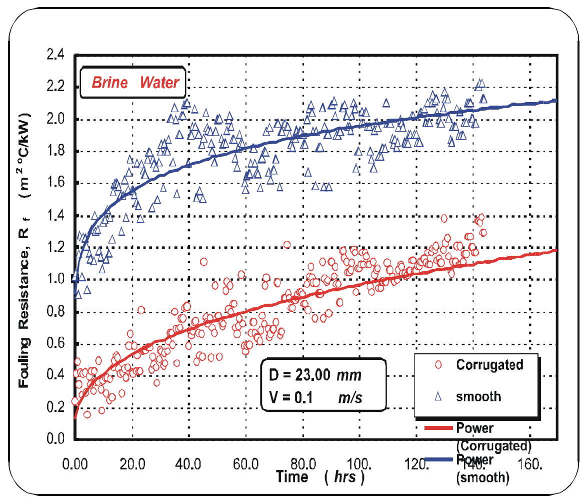


Fig. 27: Fouling resistance R_f for both tubes versus time

heat exchange process in actual 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 simulated 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, without creation of two phase flow, or accumulation of gases inside the rig. The continuous experimental running hours used for this designed test-rig is 160 hrs. The actual 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 corrugated 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, Brine water concentration (ppm). When used test rig (A), preliminary data results are obtained. The description of design methodology, disadvantage with malfunction of this rig, brought the upgrade design development to test rig (B), with its given advantages. This presented work provides researchers with background knowledge and methodology to build a laboratory test rig, has the power to operate continuously for long hours, under sever operating conditions, with highly reliable anticipated results. By using the experimental set-up for this research, comparable results are obtained for asymptotic values of overall heat transfer coefficient for Fresh and Brine water, at different flow speeds, between smooth and corrugated tubes. Values of overall heat transfer coefficient for enhanced tube are found significant higher than that of the smooth. The condensate water mass out of enhanced tube is more significant than condensate water mass out of smooth tube versus time. Fouling resistance versus time is determined for smooth and corrugated tube, with the performance of corrugated tube is found superior to that of smooth tube.

REFERENCES

1. Webb, R.L., 1994. Principles of Enhanced Heat Transfer, John Wiley & Sons Inc. New York,
2. Han, J.C., L.R. Glicksman and W.M. Rohsenow, 1978. "An Investigation of Heat Transfer and Friction for Rib-Roughened Surfaces," InternationalJournal of Heat Mass Transfer, 21: 1143-1156.
3. Webb, R.L., 1979. "Toward a Common Understanding of the Performance and Selection of Roughness for Forced Convection," Studies in Heat Transfer: A Festschrift for E.R.G. Eckert, J.P. Hartnett et al. Eds. Hemisphere Publishing Corp. Washington, D.C. pp: 257-272.
4. Fenner, G.W. and E. Ragi, 1979. "Enhanced Tube Inner Surface Heat Transfer Device and Method." U.S. Patent. 4, 154, 291 may15,
5. Gee, D.L. and R.L. Webb, 1980. "Forced Convection Heat Transfer in Helically Rib-Roughened Tubes," International Journal of Heat Mass Transfer, 23: 1127-1136.
6. Nakayama, W., K. Takahashi and T. Daikoku, 1983. "Spiral Ribbing to Enhance Single-Phase Heat Transfer Inside Tubes," Proceeding of the ASME-JSME Thermal Engineering Joint Conference, Honolulu, HI, Vol. 1, ASME, New York, pp: 503-510.
7. Sumitomo 1983. "Technical Data of Tred-Fin," Sumitomo Light Metal Industries, Aichi, Japan,
8. Bergles, A.E., 1985. "Techniques to Augment Heat Transfer- second- Generation Heat Transfer Technology", Journal of Heat Transfer, 110: 1082-1096.
9. Sethumadhavan, R. and M. Raja Rao, 1986. "Turbulent Flow Friction and Heat Transfer Characteristics of Single-and Multi-Start Spirally Enhanced T ubes," Journal of Heat Transfer, 108: 55-61.
10. Raja Rao, M., 1988. "Heat Transfer and Friction Correlations for Turbulent Flow of Water and Viscous Non-Newtonian Fluids in Single –Start Spirally Corrugated Tubes," Proceedings of the 1988 National Heat Transfer Conference, HTD-96, Vol. 1, ASME, New York, pp: 677-683.
11. Takahashi, K., W. Nakayama and H. Kuwahara, 1988. "Enhancement of Forced Convection Heat Transfer in Tubes Having Three-Dimensional Ribs," HeatTransfer Japanese Research, 17(4): 12-28.
12. Watkinson, A.P., 1990. "Fouling of Augmented Heat Transfer Tubes." Heat Transfer Engineering, 11(3): 57.
13. Cuangya, L., G. Chuanyun, W. Chaosu, H. Jinshu and J. Cun, 1991. " Experimental Investigation of Transitional Flow Heat Transfer of Three-Dimensional Internally Finned Tubes," in Advances in Heat Transfer Augmentation and Mixed Convection, M.A. Ebadain, D.W. Pepper and T. Diller, Eds. ASME Symposium, Vol. HTD-Vol. 169, ASME, New York, pp: 45-48, 1991.
14. Kim, N.H. and R.L. Webb, 1991. "Particulate Fouling of Water in Tubes Having Two Dimensional Roughness Geometry." Int. J. Heat MassTransfer, 34(11): 2727.
15. Chamra, L.M. and R.L. Webb, 1994. "Modelling Liquid-Side Particulate Fouling in Enhanced Tubes." Int. J. Heat Mass Transfer, 37(4): 571.
16. Bergles, A.E. and E.F.C. Somerscales, 1995. "The Effect of Fouling on Enhanced Heat Transfer Equipment ", Journal of Enhanced Heat Transfer, 2(1-2): 157-166.

17. Rudy, T.M., 1982. “A Theoretical and Experimental Study of Condensation on Single, Integral-fin Tubes,” PhD Dissertation, Pennsylvania State University,
18. Chamra, L.M., 1992. “A Theoretical and experimental study of particulate fouling in enhanced tubes”, Ph.D. thesis, The Penn State University,
19. Wei Li, A., Theoretical and experimental study of fouling in enhanced tubes in cooling tower systems, Ph.D. thesis, The Penn State University,
20. Ahmad Y. Kalendar, 2000. “An Investigation of Scale Depositing on Enhanced Distillation Heat Transfer Tube Surfaces”, Ph.D. Thesis, Cardiff University 2000.
21. Kalendar, A.Y. and A.J. Griffiths, 2001. “Performance Study of Enhanced and Smooth Surface Tubes in a System Condenser of a Multi-Stage Flash Desalination Unit”, Desalination Journal, ELSEVIER, 134: 269-283.
22. Galal, T., A. Kalendar, A. Al-Saftawi and M. Zedan, 2007. “Condensate Water Quantity Function of Condenser Tubing Type for Innovation MSF System”, ADST Proceedings, International Conference Desalination Technologies and Water Reuse, pp: 7-8.
23. Kalendar, A. Al-Saftawi and T. Galal, “Comparative Thermal Performance Analysis of Enhanced Condenser Tubing Type for Innovation MSF System”, ADST Proceedings, International Conference Desalination Technologies and Water Reuse, pp: 7-8.
24. Galal, T., A. Kalendar, A. Al-Saftawi and M. Zedan, 2010. ”Heat transfer performance of condenser tubes in an MSF desalination system”, Journal of Mechanical Science and Technology, 24(11): 2347-2355.
25. Kalendar, A., T. Galal, A. Al-Saftawi and M. Zedan, 2011. “Enhanced Tubing Thermal Performance for Innovative MSF System”, Journal of Mechanical Science and Technology, Published Recently,
26. Ahmad Kalendar, 1998. “Enhanced Condenser Tubing versus Energy Saving”, Proceeding of the International Conference on Energy Research and Development, Vol. II (Kuwait), pp: 603-619.
27. Private communication, Wolverine Tube Inc. Huntsville AL35801, USA.
28. Private communications, Training department, Doha East power & water production station, ministry of electricity & water, Kuwait.