USE OF FLUE GASES FROM OPEN CYCLE GAS TURBINES TO DRIVE MSF PLANTS, PILOT TEST WITH FLUIDIZED BED HEAT EXCHANGER

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Abstract

This paper presents the tests performed in 2015 and 2016, at Sabiyah Thermal Power and Desalination Plant with a fluidized bed self-cleaning heat exchanger test unit. This unit has been designed to test the non-fouling performance of the fluidized bed self-cleaning heat exchanger technology when heating up hot brine using flue gases at high temperatures of approximately 550 ºC. The aim of the test has been to assess the promising possibility of using hot flue gases (550 ºC) from an open cycle gas turbine as a heat source in a full-sized brine heater for an existing MSF unit at Sabiyah. During the test the locations of the inlet and outlet connections of the unit with the MSF were the same than those foreseen for a full-size unit. The flue gases in the test unit come from an integrated burner that simulates the gases coming from the open cycle gas turbines. The unit was first operated in February 2015 during approximately 50 hours at brine temperatures between 97 ºC and 98 ºC showing a constant heat transfer coefficient. In March 2016, the unit was operated at 110 ºC during 200 hours to confirm the non-fouling performance at higher temperatures. These tests have shown that the fluidized bed heat exchanger can be operated without fouling when heating brine of a MSF unit with the use of high temperature waste heat flue gases.

I. INTRODUCTION

1.2 Background

Sabiyah power and desalination plant located in Al Jahra, Kuwait, has eight 300MW boilers and steam turbines producing 2,400MW. Steam from the turbines is bled at low pressure to drive 8 MSF units with a capacity of 12.5 MGD of drinking water production each.

Additionally, six GE MS6001 and four GE MS7001 open cycle gas turbines provide 518 MW of peak load power capacity. These turbines are located near the brine heaters of four of the MSF units, at an average distance of 50 m from the MSF units. These turbines emit flue gases at temperatures of around 550 ^{o}C. The efficiency of them cannot be upgraded by means of a combined cycle due to space unavailability.

A way to increase the efficiency of the plant is to use the flue gases of the open gas cycle turbines to directly heat up brine of the MSF units by means of new brine heaters installed in series to the existing steam driven heaters.

Each combination of two GE MS7000 turbines or of three GE LM6000 turbines would be able to feed enough heat to one MSF unit. In the best-case scenario, the four MSF units located near the gas turbines could be driven by the heat input of flue gases. A MSF unit operating with flue gases would save on extraction steam currently taken from the steam turbines of the conventional units. Without extraction, steam would be further expanded generating up to 40 MWe of additional power. If all four MSF units would be linked to the open cycle gas turbines the power capacity of the plant would be increased by 160 MWe.

Use of flue gases of 550 ^{o}C to heat up brine implies high wall temperatures at the brine side when using regular gas to liquid shell and tube heat exchangers or boiler type arrangements. From experience, it is known that the Sabiyah desalination units suffer from significant scaling at temperatures above 105 ^{o}C. For this reason, top brine temperatures are always kept below this temperature. Using regular shell and tubes heat exchanger is not considered as an option since wall temperatures would be above 105 ^{o}C even when the top brine temperature would be kept below 100^{o}C.

As an alternative to regular heat exchangers the fluidized bed heat exchanger has been considered for this application. In a fluidized bed heat exchanger, solid particles are circulated with the fouling liquid. Solid particles have two effects in the liquid. First, they act as scouring agent avoiding any type of scaling layer to develop. Second, convective heat transfer coefficient is increased at the liquid side. As a consequence, wall temperatures in contact with brine are reduced.

1.2 History of fluidized bed heat exchangers and required design changes for gas to brine heating

Fluidized bed heat exchangers were developed parallel to the MSF/FBE technology in the Netherlands during the 70s. An MSF/FBE is a once through vertical MSF unit which uses a stationary fluidized bed of solid spherical particles of sizes in the range between 1.5 and 4 mm inside the tubes of its heating section. Particles scour the walls of the tubes avoiding the deposition of scale, thus keeping them clean, even at temperatures up to 115 ^{o}C and no antiscalant use. This technology and the results of its operational experiences were extensively described in the publications [1], [2] and [3].

Later, the use of this technology was generalized to other industries with the development of a self-cleaning fluidized bed heat exchanger. In this heat exchanger particles are circulated through the tubes of the heat exchanger together with the fouling liquid to avoid deposition. In this configuration, the shell of the heat exchanger remains a standard buffered shell. The full description of the standard fluidized bed heat exchanger technology is detailed in [4]. Given the successful references of the MSF/FBE technology it was concluded that the fluidized bed heat exchanger technology is a suitable technology to heat up brine without scaling problems when using high temperature exhaust gases.

The use of flue gases from open cycle gas turbines to heat up a liquid flow requires a number of configuration changes to the standard self-standing fluidized bed heat exchanger as described in [4]. A natural design choice could be to modify the concept into a flue gas duct with one pass hanging tube bundles as typically done for flue gas boilers. In this configuration, the fluidized bed would be developed inside the tubes where seawater would be circulated. However, pressure drop at the flue gas side must be kept to the minimum possible value because pressure drop is detrimental to the power output of the gas turbine. In order to keep the pressure drop low, it is required to select a large tube pitch in the bundle to limit flue gas velocities. However, a large tube pitch implies large particles and flow distribution areas. This means that velocities in the flow distribution channel below the tubes sheet are reduced to an extend where particles fluidization becomes impossible. At this condition, the bed would collapse at the inlet and particles could not be entered in the heat exchanger tubes.

In order to address this issue, it was decided to develop a novel configuration of the fluidized bed heat exchanger. In such configuration brine is circulated through the shell side of the heat exchanger in a vertical upwards direction. The vertical flow of brine allows the fluidization of a stationary bed of particles. A stable fluidization in this case is mainly related to a stable flow distribution of brine which is done by means of a distribution plate of proprietary design. Flue gases are circulated in horizontal direction through the tubes of the heat exchanger. A diagram of this configuration is shown in
Figure 1. The pressure-drop at the flue gas side can be optimized by lowering the velocity and increasing the number of tubes. Since the plot area of the heat exchanger is defined by a given required brine velocity for the stationary fluidization of particles, tubes can be added by adding tube rows in the vertical direction. An extensive description of the design criteria of this type of configuration is given in [5].

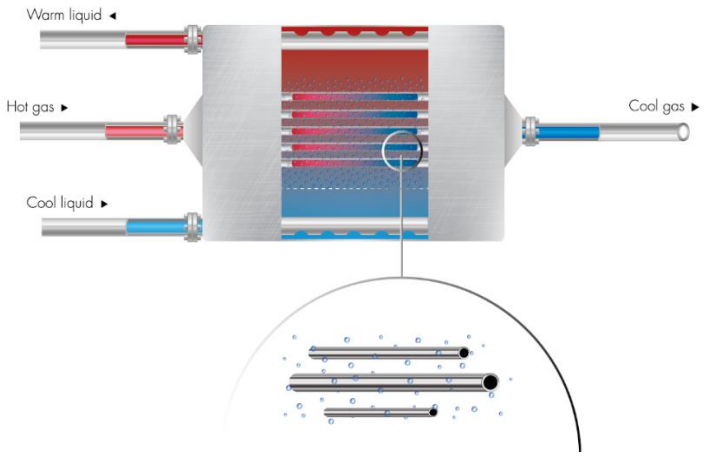

Figure 1. Diagram of a horizontal fluidized bed heat exchanger

Since this is a newly developed configuration it was decided to perform a test with a pilot installation of reduced sized to prove the concept.

II. EXPERIMENTAL SETUP

Figure 2 shows the process flow diagram of the hot pilot as originally built for the test carried out in February 2015. In this pilot combustion gases of a built-in burner fueled by propane circulate through four 0.5 m long horizontal tubes. Brine from the MSF conventional heater inlet and/or outlet is circulated in the heat exchanger shell from bottom to top in perpendicular direction to the tubes. Glass particles in the shell of the heat exchanger are fluidized by the brine flow giving a stationary bed. The glass particles act as scouring agents on the outer walls of the tubes avoiding any deposit build up, hence maintaining a stable heat transfer performance of the heat exchanger. With this so-called stationary bed a stable flow of brine is required for a correct particles fluidization. To this purpose a distribution plate with nozzles is required. Additionally, it is required to have a certain distance between the distribution plate and the start of the tubes bundle. This entrance height defines the volume called inlet channel.

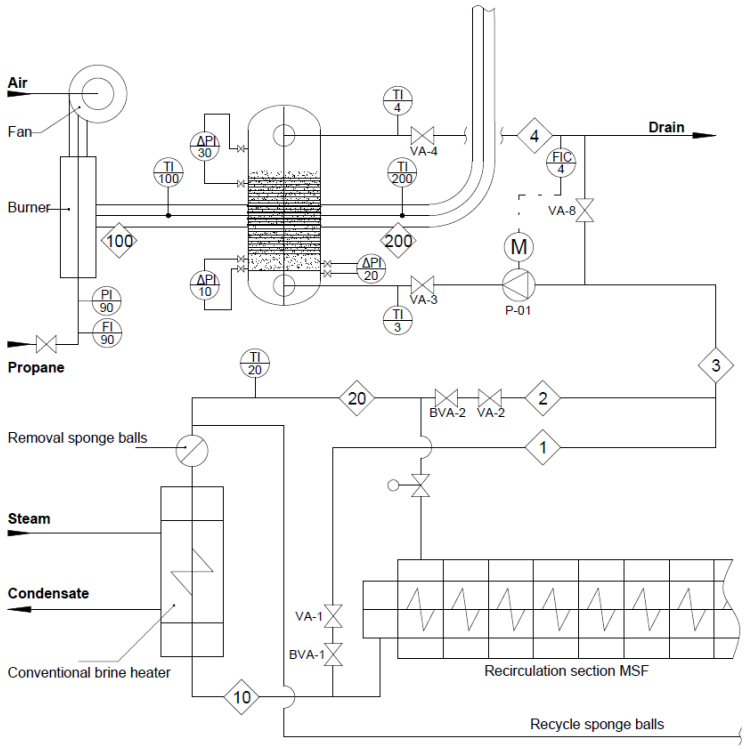

Figure 2. Hot Pilot Process Flow Diagram test February 2015

This hot pilot includes instrumentation and additional equipment to control and measure the main process parameters.

2.1 Brine flow and porosity control

The flow can be set to the desired value through the flow controller of a variable frequency pump (P-01) located in the inlet of the test unit. The flow is set such that the bed of particles is fluidized until it reaches above the tube bundle. The presence of the bed above the tube bundle is detected by a differential pressure measurement instrument. Similarly, a second differential pressure measurement measures the pressure difference over the height of the inlet channel. Based on this measurement pressure difference the

porosity which is the volume fraction of liquid in the bed defined according to (1) in the inlet channel can be calculated and controlled. The porosity is determined from the pressure drop according to (2).

$$\epsilon = \frac{V_l}{V_l + V_p} \quad (1)$$

$$dP = g\, L\, (1 - \epsilon)\, (\rho_l - \rho_p) \quad (2)$$

By changing connections, during operation also the pressure difference over the distribution plate can also be measured to verify that no clogging has occureds in the distribution place.

2.2 Temperature measurement

Temperature can be measured at the brine inlet and outlet through two screw in thermocouple sensor with temperature transmitter. Flue gas inlet and outlet are also measured with similar thermocouples with larger design temperature to monitor the flue gas temperature decrease. Accuracy of the temperature measurement is 0.1 K. The temperature measurements are required for the calculation of the heat transfer coefficient.

2.3 Burner

Flue gases to the heat exchanger come from the combustion of propane in a burner. At the propane inlet of the burner a flow measurement is located together with a pressure gauge. The flow of propane can be controlled by the position of a propane inlet valve. Furthermore, the air flow into the burner can be controlled by a variable speed fan. Tuning both air and propane flow allows to vary the flue gas inlet temperature and flow.

The value of the pressure and the volume flow of propane together with the flue gas temperature as measured after the burner allow to calculate the mass flow of the flue gas entering the heat exchanger by a heat and mass balance calculation. Knowing the amount of flue gas is required to calculate the overall heat transfer coefficient of the heat exchanger.

2.4 Data logging

A data logger allows to store all data mentioned above except for the propane flow and pressure. These parameters are stored manually on an hourly basis by an operator.

2.5 Brine temperature control

The setup as described above which is a once through configuration does not allow to control the brine temperature in the heat exchanger. The amount of heat transferred by the flue gas to the brine is in the range of 8 kW, which corresponds to a brine temperature increase of approximately 0.3 ^{o}C for the design flow of this once through configuration. As a result, brine inlet and outlet temperature are in the same range. Consequently, the maximum brine temperature inside the pilot is about equal to the temperature of the brine taken from the MSF at the brine heater outlet. During the test of February 2015, the brine taken from the MSF was always below 100^{o}C.

With the aim of being able to increase the brine temperature in the pilot independently of the plant's operating temperature a modification was installed in the hot pilot. The modification is based on recirculation of the outlet to the inlet of the heat exchanger and controlling the temperature with the discharge. The flow diagram of the new setup is shown in Figure 3. To control the discharge in the new configuration, a control valve (CVA-005) was installed at the discharge of the unit. The control valve is regulated by an added temperature controller TIC-3. With this configuration, it is possible to control the temperature by allowing recirculation and keeping VA-8 open. CVA-5 closes when TIC-3 is below the set temperature and opens when TIC-3 is above the set temperature. VA-5 is added as a by-pass valve in case a once through operation is desired.

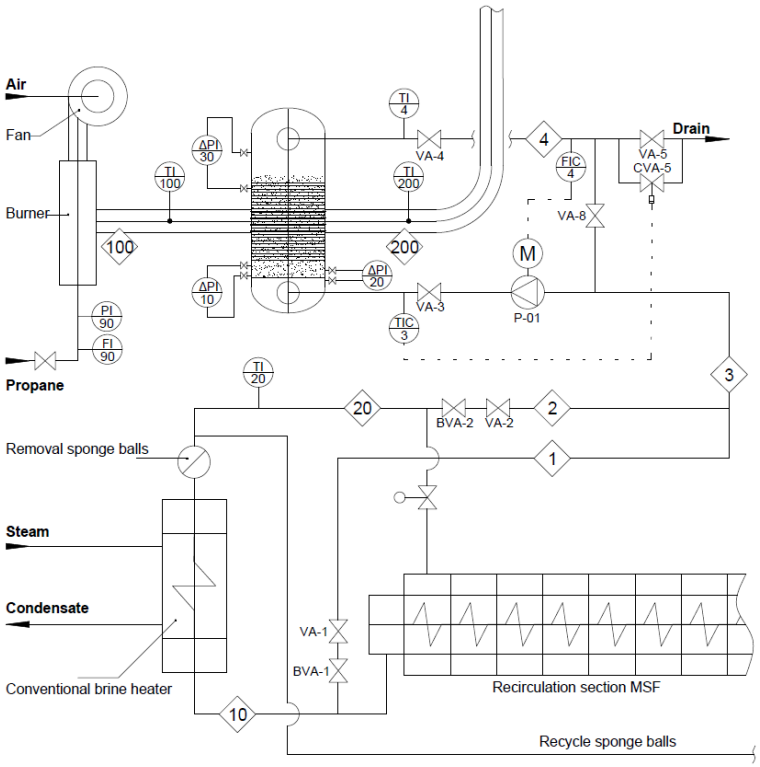

Figure 3. Hot Pilot Process Flow Diagram with temperature control

III. TESTS CONDITIONS

3.1 Process conditions

Two separate tests were carried out. A first test was carried out in February 2015. This test did not make use of recirculation and operated in once through at a brine temperature equal to the top brine temperature of the MSF plant. A second test was carried out making use of recirculation to reproduce a more challenging process temperature. The base process conditions at which both tests were carried out are summarized below.

Table 1. Base conditions during tests

Brine	**Test 1 – 02/2015**	**Test 2 – 03/2016**
Flow (m^3/h)	22.5	33.15
Inlet temperature (ºC)	97.0-98.0	100
Inlet pressure (barg)	0.65-1.00	0.65-1.00
Recirculation temperature (ºC)	-	110 - 118
Recirculation ratio	-	50-85
Flue gas		
Flow (m^3/h)	475 – 560	475 – 560
Tube inlet velocity (m/s)	75-110	75-110
Inlet temperature (ºC)	580-620	490 – 560
Inlet pressure (mbarg)	30-35	30-35
Particles		
Mass (kg)	80	52
Diameter (mm)	2	3
Material	Soda lime glass	Soda lime glass
Testing time		
Period 1	Run 1 (hours)	51	190

3.2 Brine composition

The feed brine composition has been determined using ICP and is disclosed in Table 2.

Table 2. Feed brine composition

	Unit	Concentration / Value
pH	pH	8.9
Bicarbonate	mg/l	190
Carbonate	mg/l	35
Sulphate	mg/l	4,600
Aluminium	µg/l	< 10
Arsenic	µg/l	< 5.0
Barium	µg/l	41
Calcium	mg/l	750
Cobalt	µg/l	< 1.0
Chrome	µg/l	< 5.0
Iron	µg/l	10
Potassium	mg/l	655
Magnesium	mg/l	2,225
Manganese	µg/l	0.63
Sodium	mg/l	17,825

Nickel	µg/l	9.9
Silicon	mg/l	6.0
Zinc	µg/l	< 5.0

3.3 Antiscalant

The antiscalant used in the MSF plant is a poly acrylic acid in aqueous solution. As specified by the supplier this product is effective at temperatures up to 105 ^{o}C.

IV. ANALYTICAL METHOD

The results of the test were analyzed on basis of the evolution of the measured overall heat transfer coefficient. A stable heat transfer coefficient indicates that the tubes of the heat exchanger are kept clean.

In clean conditions the overall heat transfer coefficient is dependent of the convective heat transfer coefficient at the flue gas side, the conductivity of the tubes and the convective heat transfer coefficient at the brine side. For fouled tubes the overall heat transfer coefficient is also dependent of the conductivity of the fouling layer. Because the brine flow during the test was constant, it can be assumed that the convective heat transfer coefficient on the brine side was constant. Also, the conductivity of the tube walls can be assumed constant. However, on the flue gas side, due to slight variations of flow and inlet temperature of the flue gas, the convective heat transfer coefficient suffers from small variations. In order to take into account this variation the experimental overall heat transfer coefficient was compared to the theoretical heat transfer coefficient according to the methodology shown below. A constant ratio of both coefficients indicates clean tubes. A decreasing ratio would indicate an additional heat resistance as it would happen with a insulating scale layer that develops on the tubes.

The total heat transferred from the flue gas to the brine can be calculated according to equation (3) which is calculated from the measured process values.

$$\mathbf{Q = \dot{m}_{fg}\, Cp_{fg}\, \Delta T_{fg}} \qquad (3)$$

The mass flow of the flue gas is calculated from the heat and mass balance over the burner according to (4) and (5).

$$\mathbf{\dot{m}_{pr} h_{pr} + \dot{m}_a h_a = \dot{m}_{fg} h_{fg}} \qquad (4)$$

$$\mathbf{\dot{m}_{pr} + \dot{m}_a = \dot{m}_{fg}} \qquad (5)$$

In the calculation of the flue gas mass flow the following assumptions are made:

- The propane feed temperature, T_{pr}, is 5 ^{o}C.
- The air feed temperature, T_a, is 20 ^{o}C.
- The air feed pressure, P_a, is steady at 1.03 bar.
- The flue gas discharge pressure, P_{fg}, is equal to the air pressure, 1.03 bar.
- The ideal gas model applies for the calculation of gases thermodynamic properties.
- Propane is pure.
- The average air composition in mol ratio is:
  - N2 0.7729
  - O2 0.2075

- H2O 0.0101
- AR 0.0092
- CO2 0.0003

In order to calculate the flue gas mass flow through equations (4) and (5), propane mass flow, propane enthalpy and air enthalpy, can be calculated out of the measured parameters and/or the above stated assumptions using explicit equations (6), (7) and (8). The enthalpy of flue gas can be calculated using equation (10). The resolution of equation (10) requires knowing the mass flow of both propane and air. Since the mass flow of air is not known there is an implicit system of three equations with (4), (5) and (10) having three unknowns, the flue gas and air mass flows and the flue gas enthalpy. The system is resolved iterating values of the air mass flow until all three equations are validated.

Propane mass flow and enthalpy:

The mass flow of propane is calculated from the measured volume flow and pressure of propane and the assumed feed temperature.

$$\dot{\mathbf{m}}_{\mathbf{pr}} = \dot{\mathbf{V}}_{\mathbf{pr}}\boldsymbol{\rho}_{\mathbf{pr}}\left(\mathbf{P}_{\mathbf{pr}}, \mathbf{T}_{\mathbf{pr}}\right) \quad (6)$$

The enthalpy of propane is obtained assuming ideal gas law as a function of the measured pressure, the assumed temperature and the molar mass of pure propane.

$$\mathbf{h}_{\mathbf{pr}} = \mathbf{f}\left(\mathbf{P}_{\mathbf{pr}}, \mathbf{T}_{\mathbf{pr}}, \mathbf{m}_{\mathbf{pr}}\right) \quad (7)$$

Air enthalpy:

The air enthalpy is obtained in the same way as the enthalpy of propane for the assumed pressure, temperature and composition.

$$\mathbf{h}_{\mathbf{a}} = \mathbf{f}(\mathbf{P}_{\mathbf{a}}, \mathbf{T}_{\mathbf{a}}, \mathbf{m}_{\mathbf{a}}) \quad (8)$$

Flue gas enthalpy

The enthalpy of flue gas can also be calculated as a function of the assumed pressure and measured temperature, $T_{fg\ in}$. However, in order to obtain the molar mass an additional calculation is required. The composition of the flue gas is calculated using the chemical equations of combustion (9).

$$\mathbf{C_3H_8 + 5O_2 \ --> \ 3CO_2 \ + \ 4H_2O} \quad (9)$$

$$\mathbf{h}_{\mathbf{fg}} = \mathbf{f}\left(\mathbf{P}_{\mathbf{fg}}, \mathbf{T}_{\mathbf{fg\ in}}, \dot{\mathbf{m}}_{\mathbf{p}}, \dot{\mathbf{m}}_{\mathbf{a}}\right) \quad (10)$$

The experimental overall heat transfer coefficient is calculated from the calculated heat duty as determined before, the log mean temperature difference as defined by the measured temperatures and the known tube geometry defining the heat transfer area using equation (11)

$$Q = K_{exp}\, A\, \Delta T_{mean\ log} \quad (11)$$

The theoretical heat transfer coefficient is derived from equation (12).

$$K_T = \frac{1}{\frac{1}{\alpha_{fg}} + \frac{d_{in}}{2\lambda_w} \ln \frac{d_{out}}{d_{in}} + \frac{d_{in}}{d_{out}\,\alpha_b}} \quad (12)$$

Here, the flue gas side film coefficient is calculated following the by Gnielinski modified Petukhov's Nusselt number equations (13) and (14). This relation is applicable for hydrodynamically developed and undeveloped flows in the turbulent range $10^4 < Re < 10^6$ and Pr 0.6 < Pr < 1000 [6] and [7]. The flue gas film coefficient is calculated from the Nu number using equation (15).

$$\mathbf{Nu} = \frac{\mathbf{Re\,Pr}\,\boldsymbol{\zeta/8}}{\mathbf{1+12.7}\sqrt{\boldsymbol{\zeta/8}}\,\mathbf{Pr^{2/3}-1}} \left[\mathbf{1} + \left(\frac{\mathbf{d}}{\mathbf{L}}\right)^{\mathbf{2/3}}\right] \quad (13)$$

$$\zeta = (0.78 \ln Re - 1.5)^{-2} \quad (14)$$

$$\alpha_{fg} = \frac{Nu\,\lambda_{fg}}{d_{in}} \quad (15)$$

In these relations the fluid properties are evaluated at the flue gas mean temperature across the tube length and diameter.

In the computation of the theoretical overall heat transfer coefficient the film coefficient on the brine side with the fluidized bed is evaluated using the Ruckenstein [8] correlation as given in equation $Nu = \frac{\alpha_b . d_{part}}{\lambda_b} = 000676 Pr^{0033} Re_{d_{part}}^{-00237} . Ar^{00522}$ (16). This results in a value of approximately 9000 W/m^{2}K for the parameters of the fluidized bed as given in table 1 independent of the type of particles used.

$$Nu = \frac{\alpha_b\, d_{part}}{\lambda_b} = 0.067\, Pr^{0.33} Re_{d_{part}}^{-0.237}\, Ar^{0.522} \quad (16)$$

In the evaluation of the conductive heat transfer, the thermal conductivity of the 25.6 x 1.65 mm tubes is taken as 15 W/mK.

As stated previously, the evolution of the heat transfer coefficient is evaluated as the quotient between the experimental heat transfer coefficient and the theoretical heat transfer coefficient according to (17).

$$\gamma = \frac{K_{exp}}{K_T} \quad (17)$$

The external wall temperature of the heat exchanger tubes on the brine side is an important parameter to evaluate the validity of the test. This is the wall temperature that the brine experiences close to the tubes which is the temperature that drives the fouling tendency at the brine side of the heat exchanger. The wall temperatures at both the inner and outer wall of the tubes are calculated through the heat balance along the diameter of the tube in cylindrical coordinates. The heat balance is written with equations (18) to (21) from which the wall temperatures can be computed.

$$\dot{\mathbf{Q}} = \boldsymbol{\pi}\, \mathbf{d_{in}}\, \mathbf{K_{exp}}\, (\mathbf{T_{fg}} - \mathbf{T_b}) \quad (18)$$

$$\dot{Q} = \pi\, d_{in}\, \alpha_{fg}\, (T_{fg} - T_{w\,fg}) \quad (19)$$

$$\dot{Q} = 2\pi\, \frac{\lambda_w}{\ln\left(\frac{d_{out}}{d_{in}}\right)} (T_{w\,fg} - T_{w\,b}) \quad (20)$$

$$\dot{Q} = \pi\, d_{out}\, \alpha_b\, (T_{w\,b} - T_b) \quad (21)$$

V. RESULTS

The process temperatures during this test held in February 2015 are shown in Figure 4 and the evolution of the heat transfer coefficient and wall temperatures is shown in Figure 5. The result of the first test show that the ratio between the experimental and theoretical heat transfer coefficient remains constant along the 50 hours of operation. The external wall temperatures of the tubes during the test were calculated based on the experimental heat transfer coefficient and were in the range between 104.5 ºC and 107 ºC. For around 40 hours they were above 105 ºC, thus above the typical operating temperature at Sabiyah. The expected wall temperature in case a regular shell and tube heat exchanger would have been used would have been even higher.

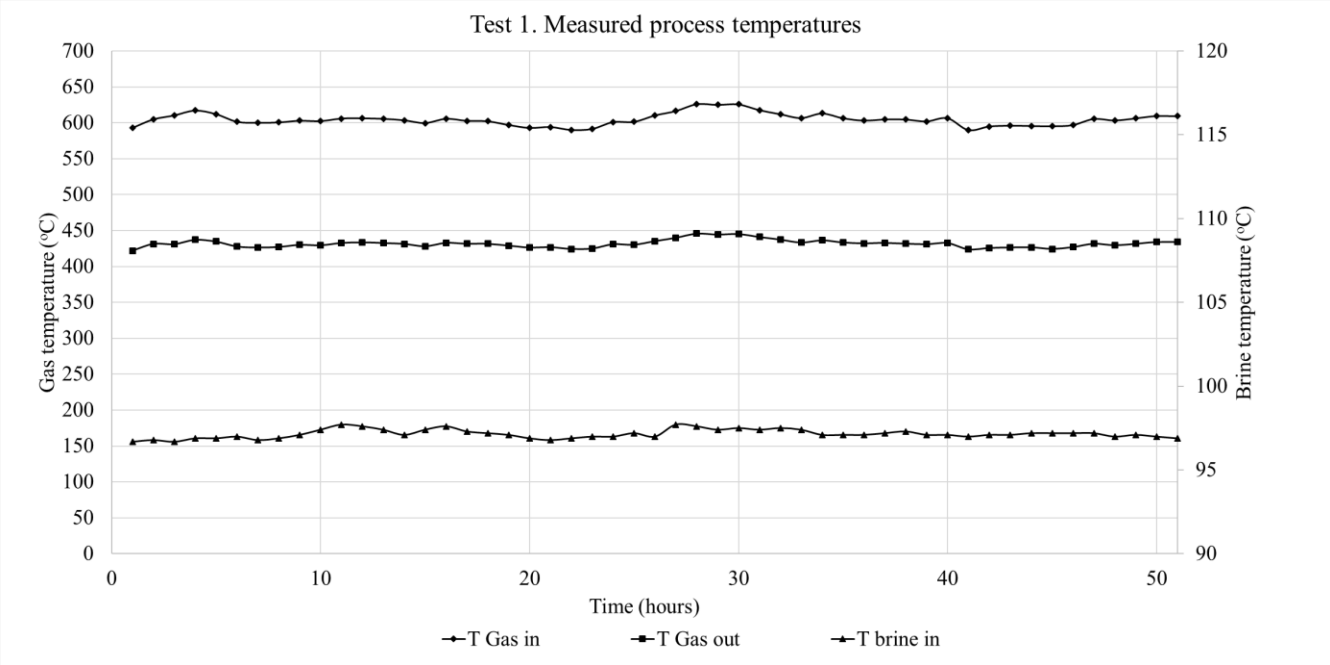

Figure 4. Evolution of measured process temperatures during test 1

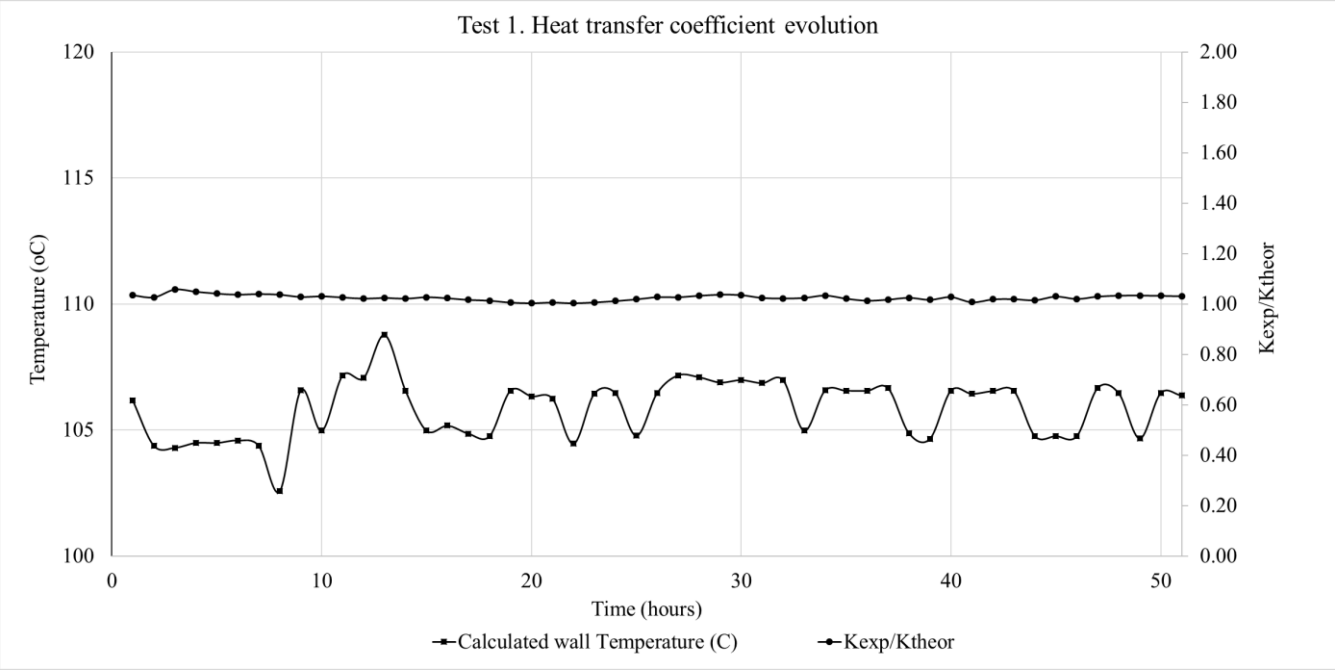

Figure 5. Evolution of the heat transfer coefficient and wall temperature during test 1[1]

After the test in 2015 the heat exchanger was physically inspected on the inside. It was observed that the heat exchanger tubes were shiny and clean. No scaling or dirt deposit was present in the heat exchanger.

Figure 6. Picture of heat exchanger tubes during inspection

The process temperatures and heat transfer results of the test held in March 2016 also showed satisfactory results as shown in Figure 7 and Figure 8. During 190 hours' operation the heat transfer coefficient stayed constant. In this case the brine bulk temperature, 110 ºC, was above the design temperature of the antiscalant. The wall temperatures were in the range between 116 ºC to 119 ºC.

[1] During Test 1 the propane flow was not hourly measured. The calculation of the flue gas mass flow is based on the average propane consumption and it assumed constant.

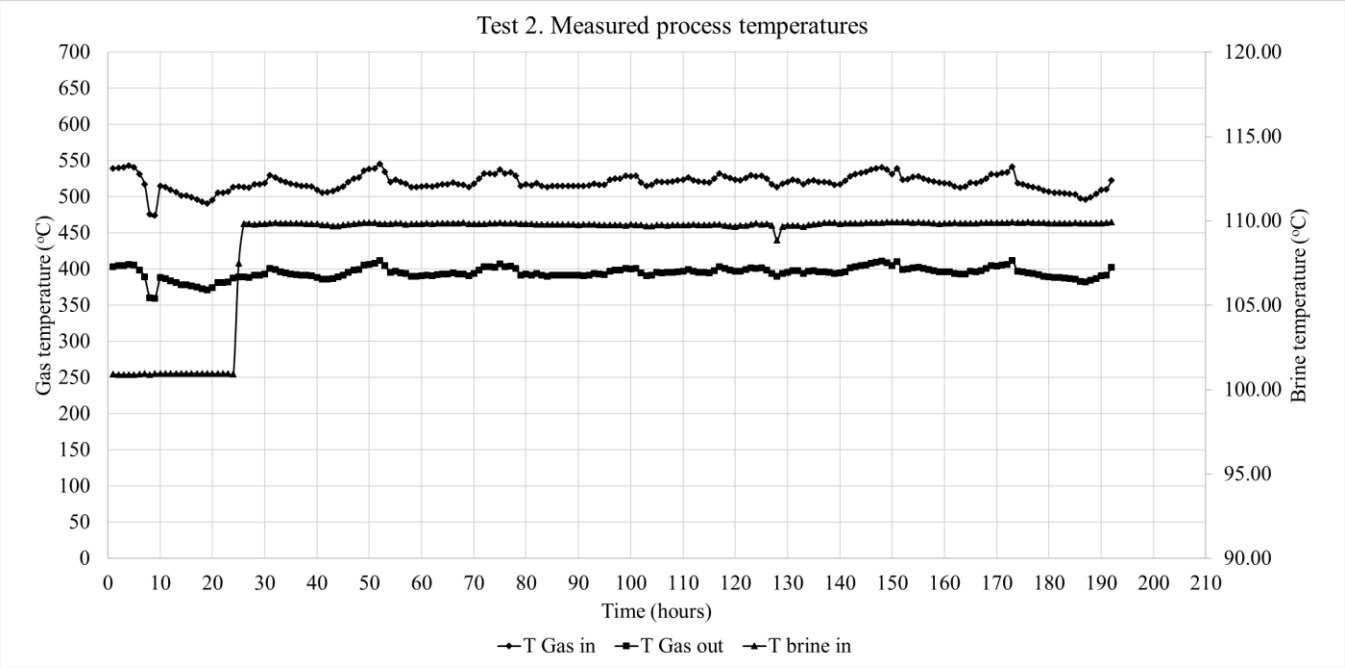

Figure 7. Evolution of the heat transfer coefficient and wall temperature during test 2

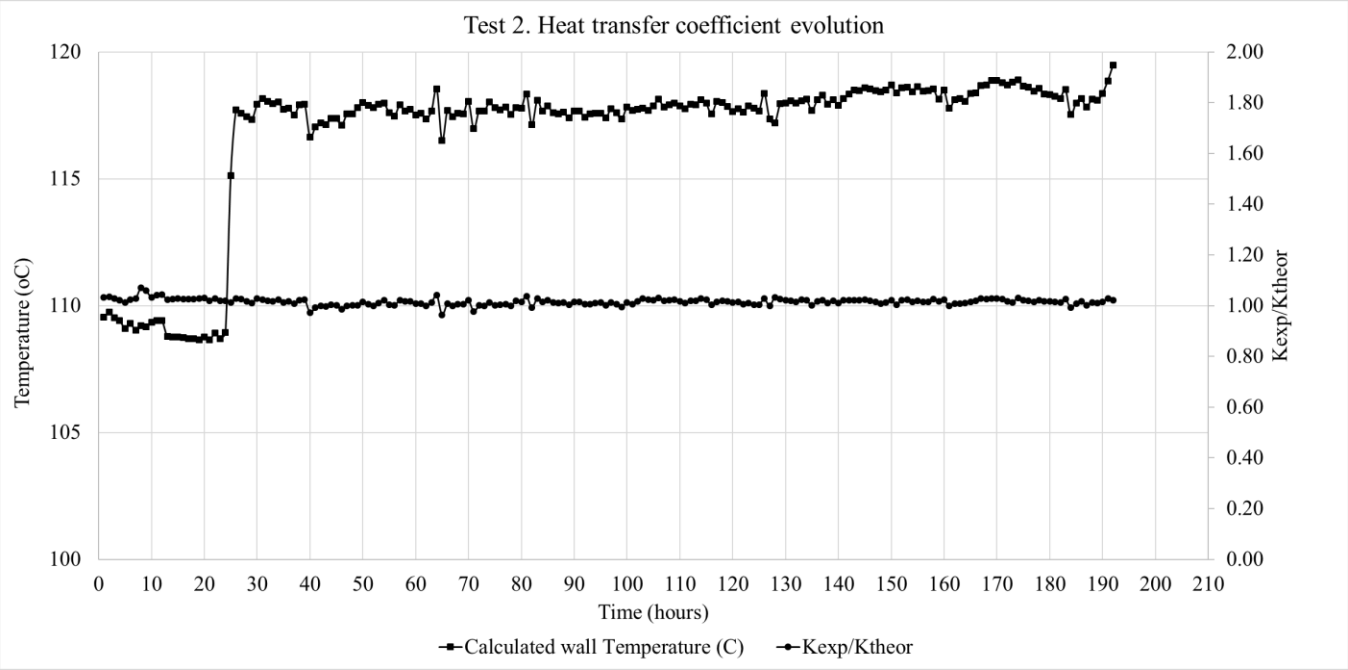

Figure 8. Evolution of the heat transfer coefficient and wall temperature during test 2

The physical inspection of the heat exchanger again showed clean tube walls. All areas where particles had been active were kept free from any deposit. However, the area above the tube bundle, where particles are by design not acting, showed a thick layer of fouling deposit in the range of 1 cm.

Fouling of this area does not hinder heat transfer, but its development is a risk for the correct operation of the fluidized bed. Further accumulation of deposits could cause that pieces of deposit break loose

which can lead to overall miss operation. The accumulation of these deposits is believed to be the results of to the high operating temperatures with brine bulk temperatures above the design temperature of the antiscalant. Such high bulk temperate caused the direct precipitation of super saturated species in the brine. In addition, with the applied recirculation in test 2 also the residence times were very significant thereby providing the time for the precipitates to grow. The deposition of precipitated solids affected the areas where particles were not active but the areas with particle motion were not affected.

To verify the reasoning behind why deposits were found in test 2 and not in test 1, a future test is planned in which the unit will be operating at similar conditions to test 1 but for a period as long as test 2.

VI. CONCLUSION

Through the described experiences, it has been demonstrated that the fluidized bed heat exchanger concept gives an opportunity to recover thermal energy from open cycle gas turbines for direct sea water heating.

Operation in test 1 at bulk temperatures of the sea water within the design temperature of the antiscalant chemicals used at the MSF plant was flawless even with wall temperatures above the antiscalant specification.

Operation in test 2 at elevated bulk temperatures of the sea water that are above the specification of the antiscalant chemicals showed that particles were efficient to keep the tubes of the heat exchanger clean despite the very high wall temperatures. However, significant deposition of dirt was observed in areas were cleaning particles were per design not active. This deposition could lead to miss operation when operating a full scale unit.

Overall it can be concluded that the fluidized bed heat exchanger concept has proven to be able to operate without suffering from scaling of heat transfer surfaces when heating brine of an MSF unit by hot flue gases. The surface even remained clean at wall temperatures 13 ^{o}C above the design temperature of the antiscalant chemical.

VIII. NOMENCLATURE

Symbol	Unit	Description
A	m^2	Heat transfer area
Cp	kJ/kg K	Heat capacity
d	m	Diameter
f	-	Function of
h	kJ/kg	Enthalpy
K	W/m^2K	Overall heat transfer coefficient
L	m	Pipe length
$\dot{m}$	kg/s	Mass flow
m	kg/kmol	Molar mass
Nu	-	Nusselt number
Q	W	Exchanged heat
$\dot{Q}$	W/m	Exchanged heat flux
Re	-	Reynolds number
P	bar	Pressure

Pr	-	Prandlt number
t	s	Time
T	K	Temperature
V	m^3	Volume
$\dot{V}$	m^3/s	Volumetric flow
α	W/m^2K	Fluid film coefficient
ΔP	bar	Pressure difference
ΔT	K	Temperature difference
ΔT_{log}	K	Logarithmic temperature difference
ρ	kg/m^3	Density
λ	W/mK	Thermal conductivity
ζ	-	Petukhov constant

Subscripts	**Description**
a	Air
b	Brine
exp	Experimental
fg	Flue gas
in	Internal
out	External
$part$	Particles
pr	Propane
w	Wall
T	Theoretical

VII. REFERENCES

1. Klaren, D.G., Development of a Vertical Evaporator, Ph.D. Thesis, Delft University of Technology, May, 1975
2. Klaren, D.G. and Windt, J., Design and Construction of a 500 m^3/d Multi-Stage Flash / Fluidized Bed Evaporator, Proceedings 6th International Symposium on Fresh Water from the Sea. Las Palmas. Vol. 2, 1978, p 15-30.
3. Spanhaak, G, A Comparison of the Operating Characteristics of Conventional Multi-Stage Flash Evaporation and Multi-Stage Flash / Fluidized Bed Evaporation, Desalination. Vol. 31, 1979, p. 511-519
4. Klaren, D.G. and de Boer, E.F., Self-Cleaning Fluidized Bed Heat Exchangers for Severely Fouling Liquids and their Impact on Process Design. Chapter 21 in 'Heat Exchangers- Basic Design Applications', edited by Jovan Mitrovic, 2012
5. van Beek, M.C. and Cancela Vallespin, M, Horizontal shell side fluidized bed heat exchanger, design considerations and experiences from a pilot unit, Proceedings of Heat Exchanger Fouling and Cleaning XII conference, vol 1, 2017, p TBP
6. Petukhov, B. S., Heat Transfer and Friction in Turbulent Pipe Flow with Variable Physical Properties. High Temperature Institrite. Academy of Science of the USSR. Moscow. USSR, 1970
7. VDI-Gesellschaft Verfahrenstechnik und Chemieingenieurwesen (GVC), VDI Heat Atlas Second Edition. Springer, Dusseldorf (2010). ISBN 978-3-540-77876-9

8. Ruckenstein, E., On the heat transfer between a liquid/fluidized bed and the container wall, *Rev. Roum. Phys.*, vol.10, 1959, pp. 235-246.