HORIZONTAL SHELL SIDE FLUIDIZED BED HEAT EXCHANGER, DESIGN CONSIDERATIONS AND EXPERIENCES FROM A PILOT UNIT

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ABSTRACT

Fluidized bed heat exchangers are used in applications where heat is transferred to or from fouling liquids. In this type of equipment, the scouring action of a fluidized bed of particles proves to keep the heat transfer walls clean at the most severe fouling conditions.

The main requirement for this technology so far has been that the fouling liquid must circulate in the tube side and must flow upwards in order to fluidize the cleaning particles. However, in applications where high pressurized gases are cooled with sea-water, mechanical reasons dictate to have the gas inside the tubes and the cooling sea-water in the shell.

For these applications a zero-fouling heat transfer solution with a horizontal configuration of the fluidized bed technology has been developed. In this configuration, the fouling liquid is circulated upwards through the shell of the heat exchanger developing a stationary fluidized bed. The non-fouling liquid or gas is circulated through horizontally orientated tubes.

By comparing it to an air cooler, it is shown that the horizontal configuration that cools the hot gas by direct sea water cooling uses only 3% of the plot size making it a very compact alternative while maintaining the zero-fouling operation.

The new horizontal configuration has proven its performance in a pilot unit as tested at the Sabiyah power and desalination plant located in Kuwait. Test results show that the heat exchanger remained clean during the test run.

The design approach of a horizontal shell side fluidized bed heat exchanger is different than for the standard arrangement with the fluidized bed in the tubes and it starts by selecting the bed porosity. The influence of the porosity on the intensity of the impacts is shown by flow tests in a 1:1 replica of the hot pilot. Based on the porosity the design approach is further described.

INTRODUCTION

Use of fluidized bed technology for self-cleaning heat exchangers has been applied since 1970's starting with the application of a stationary bed in the condenser section of a vertical MSF for seawater desalination (Klaren, 1978). The main concept behind the use of a fluidized bed inside the heat exchanger tubes is that the particles give a scouring effect on the walls of the heat exchanger tubes, thus removing any fouling layer that develops. After the use of the fluidized bed for MSF units the technology has been successfully applied in many other industries. The full description of the standard fluidized bed heat exchanger technology is detailed in Klaren, 2012.

Next to the scouring effect, the particles also improve heat transfer because the movement of the particles breaks up the boundary layer making it thinner. The improved heat transfer lowers the wall temperature for an application with heating which reduces the driving force for crystallization.

Figure 1. Pilot unit installed at one of the MSF distillers at the Sabiyah power plant.

For a correct and uniform fluidization, it is required that the fluid flows upward meaning that for the application in a regular shell and tube heat exchanger it needs to be oriented vertically. So far, in all applications applying a fluidized bed self-cleaning heat exchanger, the fluidized bed was inside the vertical oriented tubes and the cooling or heating medium was on the shell side. However, there can be conditions where it would be beneficial to have the fluidized bed on the shell side.

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One of such conditions can be the pressure of the medium applied. This is for example the case when hot pressurized gas needs to be cooled down before it can be transported from an off-shore platform to land. In this situation based on mechanical considerations one would prefer to have the gas inside the small diameter tubes.

This paper first describes the horizontal configuration, the benefits of it and the intended applications. Next, the pilot unit as installed at a thermal desalination plant in Kuwait with the new configuration is described and the operational experiences gathered with this unit for the heating of brine (concentrated sea water) using flue gases are discussed. Lastly, in this paper the design criteria to take into account for the horizontal configuration are elaborated.

HORIZONTAL CONFIGURATION AND INTENDED APPLICATIONS

In figure 2 a schematic representation is given of the horizontal configuration with the fluidized bed on the shell side.

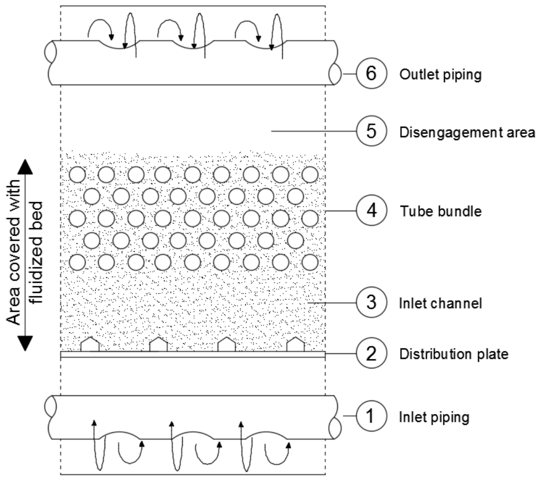

Figure 2. Schematic representation shell side fluidized bed heat exchanger with horizontal tubes.

In the horizontal arrangement the flow is entered below the distribution plate through inlet piping configured such that the flow is equally distributed over the length of the heat exchanger. The flow then passes through a distribution plate with nozzles to ensure a uniform flow over the cross section of the heat exchanger. Above the distribution plate in the inlet channel the particles are fluidized. The flow then passes in cross-flow across the tube bundle, where a second medium is circulated. For the sake of discussion in this paper, pressurized gas is taken as reference for the second medium. Above the tube bundle a certain area is maintained which functions as a disengagement area. The required length of the disengagement area is discussed in the design considerations.

In the design of a heat exchanger with a fluidized bed on the shell side the volume fraction of the particles is extremely important. As will be discussed later the flow velocity of the fluid determines the porosity in the shell and around the tubes and this porosity is a measure of the impact frequency and intensity of the particles that give the scouring action on the tubes. The porosity is defined as:

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

meaning that with a porosity of 1 there are no particles present in the flow. The flow velocity (assuming no particles present) of the fluidized bed is for a stationary bed equal to the falling velocity of a swarm of particles and is therefore given by:

$$U_{fb} = U_\infty \, \varepsilon^{2.39} \qquad (2)$$

This relation is valid for a Reynolds number above 500 with the Reynolds number based on the particle diameter. In more detailed models the exponent of 2.39 as used in equation 2 is given as a function of the Reynolds number. In equation 2 the falling velocity of a single particle in an infinite volume, U_∞, can be found from the very known equations derived from a balance between the gravitation force, the buoyancy force and the drag force acting on a particle, Richardson 1997.

Figure 3. Sample of a fouling layer as found on the tube inlet of the brine heater of a Multi Stage Flash (MSF) Distiller at the Sabiyah plant.

A very promising application of the concept with the fluidized bed on the shell side is the cooling of hot pressurized gas as required before transport from off-shore platforms to land. For the cooling of hot pressurized gas a very common solution is to use air cooled tube banks. A disadvantage of the use of air coolers is the large required plot size which is very relevant for off-shore platforms where plot area is very expensive. A more compact solution could be using direct cooling using sea water. A problem when using sea water for direct cooling, is bio-fouling in combination with particulate fouling and sometimes crystallization of salts. The zero-fouling operation of a fluidized bed heat exchanger could mitigate this.

For the application of cooling pressurized gas, the standard configuration with the fluidized bed on the tube side and the pressurized gas on the shell side would result in a wall thickness of 80 mm when assuming a shell diameter of 1 m. A shell with such a thickness would become very expensive and out of the ordinary. The new configuration with the fluidized bed on the shell side overcomes this limitation with tube thicknesses of only several millimeters at the given pressure ratings. In table 1 a comparison is given between the

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plot size required for a shell-side fluidized bed heat exchanger compared to that of an air cooler using finned tubes. The plot area of a cooler using sea water is only 3% of that of an air cooler.

Table 1 Comparison between plot size required for air cooled gas cooling and by shell-side fluidized bed cooler.

		Air Cooler (finned tubes)	Shell-side Fluidized Bed Heat Exchanger
Heat Duty	MW	10	
Gas temperature	^{o}C	in; 120	out; 40
Gas pressure	bar	150	
Sea/air temperature in	^{o}C	25	25
Sea/air temperature out	^{o}C	40	40
K	W/m^{2}K	40	1050
Heat transfer area	m^2	4700	251
Total tube length	m	4800	5037
Plot size		4 x 17 m	3 x 0.6 m
Plot area	m^2	68	1.8

PILOT UNIT

In figure 1 a picture is shown of the hot pilot as was built and installed in 2015 at the Sabiyah power and desalination plant located in Kuwait. The design and detailed findings from this unit are described and discussed in Cancela, 2017. The steam heated brine heater experiences significant fouling as can be seen in figure 3.

A schematic diagram of the hot pilot is given in figure 4. In this pilot unit flue gases from a burner connected to the unit and fueled by propane circulates through four 0.5 m long horizontal tubes. These four tubes are part of a tube bundle having in total 25 tubes (ø25.40 x 1.65 mm) in a staggered arrangement. The tubes transfer their heat to the brine in the unit which is taken from the outlet of the final brine heater of the distiller and circulates in the heat exchanger shell from bottom to top. The cleaning particles as applied for the fluidized bed are made from soda-lime glass and have a size of 2 mm. Glass particles were chosen since they in the past have proven to prevent fouling for sea water concentration and glass is attractive from the cost perspective.

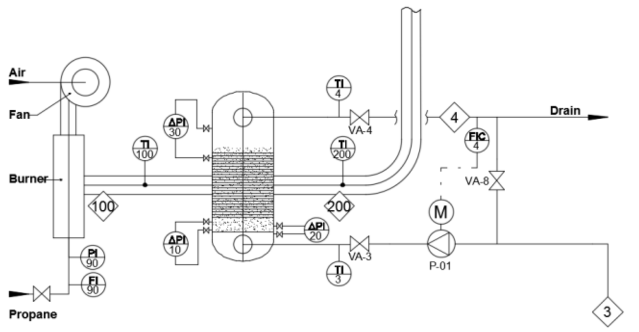

Figure 4. Schematic diagram hot pilot unit.

In the pilot unit the inlet and outlet temperatures are measured for both the flue gas and the brine flow. Also, the brine flow through the unit is measured. The mass flow of the flue gas is determined from the measured propane consumption and the temperature of the flue gas at the exit of the burner.

The experimental overall heat transfer coefficient can be calculated from the measured values for temperatures and flow and the known area of heat transfer using equation 3.

$$\mathrm{Q} = \mathrm{K_{exp}}\ \mathrm{A_{HT}}\ \mathrm{LMTD} \qquad (3)$$

In the equation above the heat transfer area is equal to the surface area of the 4 heated tubes and the heat load is derived from the flue gas.

From the experimental data also the wall temperature is derived using the following equation

$$\mathrm{Q} = \pi\ \mathrm{d_{t,o}}\ L_t\ \alpha_{\mathrm{fb}}\ (\mathrm{T_{w,br}} - \mathrm{T_{b,br}}) \qquad (4)$$

The external wall temperature of the heat exchanger tubes, 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 film coefficient of the fluidized bed can be calculated using the correlation as by Ruckenstein, 1959.

$$\mathrm{Nu} = \frac{\alpha_{fb}\, d_p}{\lambda_{br}} = 0.067\, Pr^{0.33} Re_{d_p}^{-0.237}\, Ar^{0.522} \qquad (5)$$

where the Archimedes number is defined as:

$$\mathrm{Ar} = \frac{g\, d_p^3}{v_l^2}\, \frac{(\rho_p - \rho_l)}{\rho_l} \qquad (6)$$

which for the conditions as applied in the test unit results in a film coefficient of 9000 W/m^{2}K with the fluid properties evaluated at the flue gas mean temperature across the tube length and diameter.

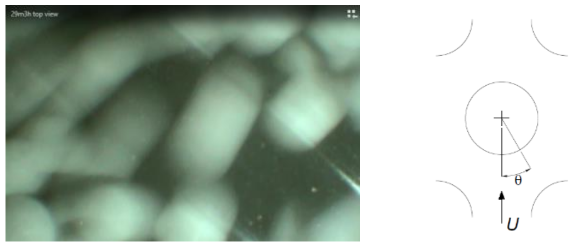

Figure 5. Picture from particle movement during experimental verification of impact intensity. On the right the definition of the angle of approach is given.

Before the hot pilot was manufactured a 1:1 replica of the heat exchanger shell with tube bundle was built and tested using cold water to optimize the flow and consequently the porosity as was previously explained. To study the impact intensity of the particles, the tubes were made of Plexiglas which made it possible to examine the particle flow from the inside by using an endoscope. An example of a picture as recorded is shown in figure 5. The impact intensity was studied in a qualitive way by evaluation of the recordings.

PILOT UNIT RESULTS

In the cold tests as done in the replica, the flow was varied between 12 and 21.5 m^3/h for the case with 2 mm glass particles. Also, one run was done with 3 mm particles where the flow was selected such that the porosity matched the maximum porosity, 0.65, as applied for the 2 mm particles. As the flow increases, the fluid velocity increases and the porosity will increase (lower particle volume) following equation 2. The porosity as given in the table below for the inlet channel has a strong effect on the impact frequency and velocity. As it can be seen in the table below that the most significant effect is on the downstream side (top) of the tube where the angle of approach as defined in figure 5 is between 135° and 225°. At the lowest flow rates, the particles were stagnant on the tube surface. At higher flow rates they start moving and above a porosity of 0.65 the intensity is regarded as acceptable.

Table 2 Results from cold test done on a 1:1 scale model to study particle impact intensity.

Flow (m3/h)	Particle size (mm)	Inlet channel porosity (-)	Impact frequency (*)		Impact velocity (*)	
			upstream 135° to 225°	downstream 225° to 135°	upstream 135° to 225°	downstream 225° to 135°
12	2	0.51	5	0 - (N1)	5	1
14	2	0.55	5	2 - (N1)	5	2
15.5	2	0.59	4	2 - (N2)	5	2
20	2	0.63	3	3 - (N2)	5	3
21.5	2	0.65	3	3	5	4
30	3	0.65	3	3	5	4

Notes (N1) Constant contact with low velocity
(N2) Constant contact with medium velocity
(*) Very high 5, High 4, Medium 3, Low 2, Very low 1 and None 0.

Based on the findings of the cold test, the test conditions as given in table 3 were selected of the first test with the hot pilot.

Table 3 Test conditions hot pilot test February 2015.

Test conditions

Period		February 2015
Run length	hrs	51
Brine flow	m^3/h	22.5
Brine Inlet temperature	°C	97.0-98.0
Flue gas flow	m^3/h	475 – 560
Flue gas tube velocity	m/s	75-110
Flue gas tube inlet temp.	°C	580-620
Particle diameter	mm	2
Particle material		Soda lime glass

The main result from the test was that the experimental heat transfer coefficient as shown in figure 6 remained constant over the duration of the test. Based on the measured heat load and a film coefficient of 9000 W/m^2K the external wall temperature of the tube had been calculated. The wall temperature was in the range between 102.6 °C and 108.7 °C. Such relatively low temperatures as compared to the 600 °C of the flue gases are the result of the several degrees of magnitude higher heat transfer coefficient on the shell side than on the tube side. Also, the expected wall temperature for a regular shell and tube heat exchangers would have been even higher given its lower shell-side heat transfer coefficient.

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 tube bundle of the heat exchanger.

During inspection of the unit no evidence of erosion of the tubes or shell was seen. It has to be said that to study erosion longer runs would be required but, given the low velocities in the heat exchanger and past experience, erosion on these parts is not expected.

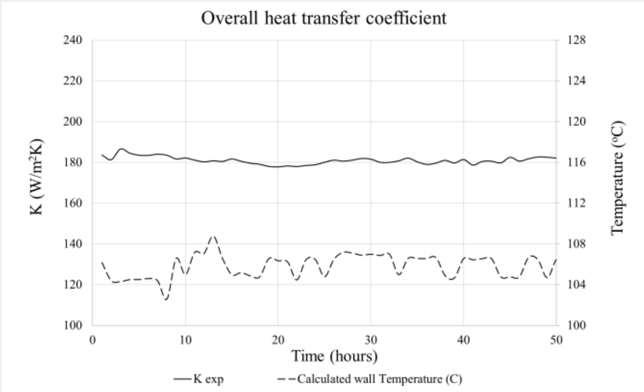

Figure 6. Experimentally determined heat transfer coefficient for hot pilot unit in the first test.

DESIGN CONSIDERATIONS

For the design of the horizontal fluidized bed heat exchanger different criteria need to be taken into account of which some are comparable to the standard heat exchanger and some are different.

Heat load; Starting point of the design is the heat load or cooling capacity of the heat exchanger which is defined by the flow rate of the to be cooled gas and the temperature drop to be accomplished. This heat load can be converted to a required flow rate of the sea water given the maximum sea water temperature over a year at the location of the platform and the maximum allowable temperature at which the seawater is allowed to be disposed.

Shell side flow velocity; with the known flow rate of the seawater the fluidized bed can be designed. Starting point of this is the design porosity. The latter is chosen on the basis of the required cleaning intensity and an analysis like this was done for the hot pilot unit discussed in this paper. For now, we assume a minimum porosity of 0.65 which by using equation 2 and a given falling velocity of a single particle diameter results in the flow velocity in the inlet channel. With the flow velocity and the flow rate known in the rectangular inlet channel its cross sectional area is defined.

Dimensions inlet channel; one of the dimensions of the cross section of the inlet channel comes from the thermal design of the cooling of the hot gases. After selecting the appropriate gas velocity in the tubes, the number of tubes is known and the length can be calculated as a function of the heat transfer coefficient and the required heat load. This length will set one of the sides of the rectangular area. If the length becomes relatively too long, the width may get too small. In that case, one could choose to lower the velocity in the tubes or even decide to use a multi pass arrangement.

Tube bundle arrangement; with the number of tubes known and the cross sectional area of the determined size of the cross

section, the height of the tube bundle can be calculated when assuming a tube pitch and pitch angle.
Amount of particles and length of disengagement area; starting point is the porosity as designed for and as discussed before. This porosity is a function of the design flow and independent from the mass of particles as put into the heat exchanger. If too little amount of particles is introduced, the expansion of the bed when a flow is applied will be limited. If too many particles are entered, it can cause an overflow of particles. Therefore, the right amount of particles is chosen such that the expansion of the particles is enough to cover the full tube bundle in a way that all tubes are protected against fouling. Above the bundle a disengagement area is included in the design. In design flow conditions, bed expansion is such that it does not only covers the tube bundle, but also covers a portion of the disengagement area. This portion is defined such that if the flow decreases and consequently the porosity decreases giving a smaller bed expansion then the tube bundle always stays covered by the fluidized bed. In the same way, the disengagement area is long enough to cope with a flow increase that results in an increased porosity and an increased bed expansion, without risk of particle overflow. It can be said that the length of the disengagement area is selected as a function of the required flow variability.

DISCUSSION

The test results as presented in this paper are regarded valid to evaluate the zero fouling operation of a fluidized bed heat exchanger using flue gases to heat up brine in a MSF unit. Still in 2016 the tests were continued to show the performance over a longer period and to assess the performance going to higher brine temperatures. For this the test unit was modified to run with brine recirculation. The results and findings of these tests are reported in Cancela, 2017.
The intensity of the cleaning action has been shown to be very dependent on the applied porosity. The porosity of the bed in relation to the particle movement upstream and downstream of the tubes has been assessed in a qualitive way and for one arrangement of tube pitch and tube angle. For a further development of this configuration further study needs to be carried out in which the intensity should be evaluated more quantitively.

CONCLUSIONS

The main conclusions from this work:

1. A configuration of a heat exchanger applying a fluidized bed on the shell side in cross flow with horizontal tubes containing high pressure gases provides in terms of plot size an very promising alternative compared to a standard applied air cooler.
2. The new configuration with the shell-side fluidized bed requires careful tuning of the porosity as to have an active bed on all sides of the tube. With the standard fluidized bed on the tubes side where the flow is parallel there is more freedom in the selection of the porosity.
3. The horizontal shell-side fluidized bed configuration was tested at an MSF plant in Kuwait and showed its zero fouling performance in a 50 hour test run. New tests have been carried out to show the performance over a longer period and at elevated temperatures. The results of these tests are reported in Cancela, 2017.

NOMENCLATURE

A	Heat transfer area, m^2
Ar	Archimedes, -
c_p	Specific heat, J/kgK
d	diameter, m
g	gravitation constant, m/s^2
K	Overall heat transfer coefficient, W/m^2K
L	Length, m
LMTD	Log Mean Temperature Difference, K
Pr	Prandtl number ($=\eta_l\, c_{p,,l} / \lambda_l$), -
Q	Heat load, W/m^2K
Re_{dp}	Particle Reynolds number ($=d_p\, U_{fb} / \nu_l$), -
T	Temperature, K
U	Velocity, -
V	volume, m^3
α	Film coefficient, W/m^2K
ε	Porosity, -
λ	heat conductivity, W/mK
ν	kinematic viscosity, m^2/s

Subscript

b	bulk
br	brine
exp	experimental
fb	fluidized bed
HT	heat transfer
b	bulk
l	liquid
o	outside surface
p	particle
t	tube
w	wall

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