ECI Symposium Series, Volume RP5: Proceedings of 7th International Conference on Heat Exchanger Fouling and Cleaning - Challenges and Opportunities, Editors Hans Müller-Steinhagen, M. Reza Malayeri, and A. Paul Watkinson, Engineering Conferences International, Tomar, Portugal, July 1 - 6, 2007

ACHIEVEMENTS AND POTENTIAL OF SELF-CLEANING HEAT EXCHANGERS USING UNTREATED NATURAL SEAWATER AS A COOLANT

D.G. Klaren [1] and E.F. de Boer [1]

[1] KLAREN BV, Vincent van Goghsingel 40, NL - 2182 LP Hillegom, the Netherlands; info@klarenbv.com

ABSTRACT

The fluidized bed self-cleaning heat exchange technology has been developed in the 70s for the application in thermal desalination plants, like Multi-Stage Flash (MSF) evaporators. This paper summarizes the achievements of the Multi-Stage Flash / Fluidized Bed Evaporator (MSF/FBE) in the past, but also pays attention to its market potential today, whether or not in combination with the possible revamp of existing conventional MSF plants into a better performing configuration.

Further, this paper also pays attention to the application of the self-cleaning heat exchange technology for closed-loop coolers or so-called ‘mother-coolers’ for large chemical plants.

Last but not least, attention is paid to the application of the self-cleaning heat exchange technology on offshore platforms using untreated natural seawater as a coolant, where it might replace the conventionally used heavy, bulky, voluminous and energy-inefficient air coolers for the same service.

INTRODUCTION.

The fluidized bed self-cleaning heat exchange technology has been developed in the 70s for the application in thermal desalination plants, like Multi-Stage Flash (MSF) evaporators. As a matter of fact, a 500 ton / day MSF evaporator equipped with a fluidized bed heat exchanger, operated for several years on the Isle of Texel, a tourist resort located at the northern coast of the Netherlands. Besides the many advantages of this unique seawater distillation plant, its operation convincingly demonstrated that indeed chemically untreated natural seawater could be heated up to a temperature of 115 °C or even higher, without forming of a layer of hard scale deposits on the walls of the heat exchanger tubes.

In the late 70s and early 80s, the very successful introduction of the membrane processes in seawater desalination caused hardship for many thermal desalination processes, including the MSF evaporator equipped with a fluidized bed heat exchanger. However, today a revival for interest in this type of evaporator is observed, particularly for installations with a small production capacity at remote locations.

This paper pays attention to the operation and performance of the Multi-Stage Flash / Fluidized Bed Evaporator (MSF/FBE) in the past, but also discusses its market potential today, whether or not in combination with the possible revamp of existing conventional MSF plants into a better performing configuration.

Further, this paper pays attention to the application of the self-cleaning heat exchange technology for closed-loop coolers for large chemical plants and for the application of the self-cleaning heat exchange technology on offshore platforms.

PRINCIPLE MULTI-STAGE FLASH (MSF) EVAPORATOR.

The principle of a conventional Multi-Stage-Flash evaporator is presented in Fig. 1, while its corresponding temperature diagram is shown in Fig. 2. As follows from these figures, the evaporator can be best described as a counter-current heat exchanger. The raw feed or seawater is heated in the heat recovery section where condensers are mounted in the top of the flash chambers. After obtaining a final heat supply in the heat input section, the heated seawater flashes through all chambers or stages via the orifices in the intersection walls of the chambers and a gradual drop in saturation pressure and temperature occurs which results in a partial evaporation of the concentrated seawater or brine in each flash chamber. The produced flash

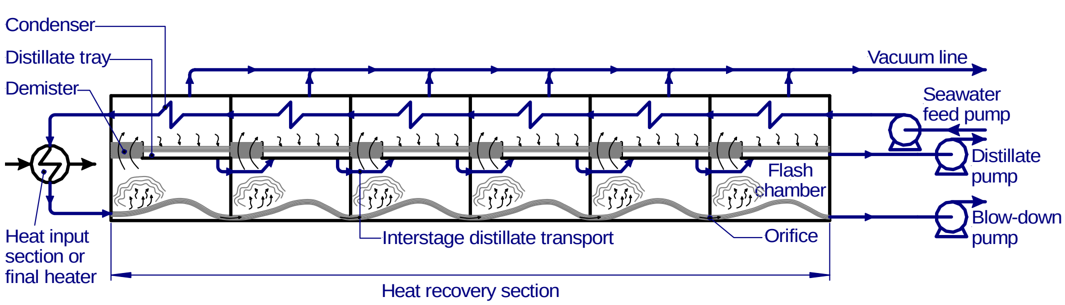

Fig. 1: Principle of conventional MSF evaporator.

vapour passes through the water-steam separators or demisters and finally condenses on the condenser surfaces, which are cooled by the incoming feed. The distillate is collected in the distillate trays and cascades down to the tray in the next chamber in the same way as the brine. The plant has to be completed with pumps for the removal of the brine and distillate out of the coldest flash chamber and for the feed supply. Dissolved gases are removed from the heated seawater in the hottest chamber by means of a vacuum line, which is also connected to all other chambers for the removal of in leaking non-condensables. After the heat input into the final heater, this conventional MSF evaporator can operate at a maximum temperature of approximately 95 °C, although it requires a chemical dosing to prevent the formation of hard scales. Tubes can be made of Aluminium Brass or alloys like CuNi 90/10 or CuNi 70/30, or currently most widely used, titanium. At the high-temperature end, the first two flash chambers should be made of stainless steel, while the remaining flash chambers can be made from carbon steel.

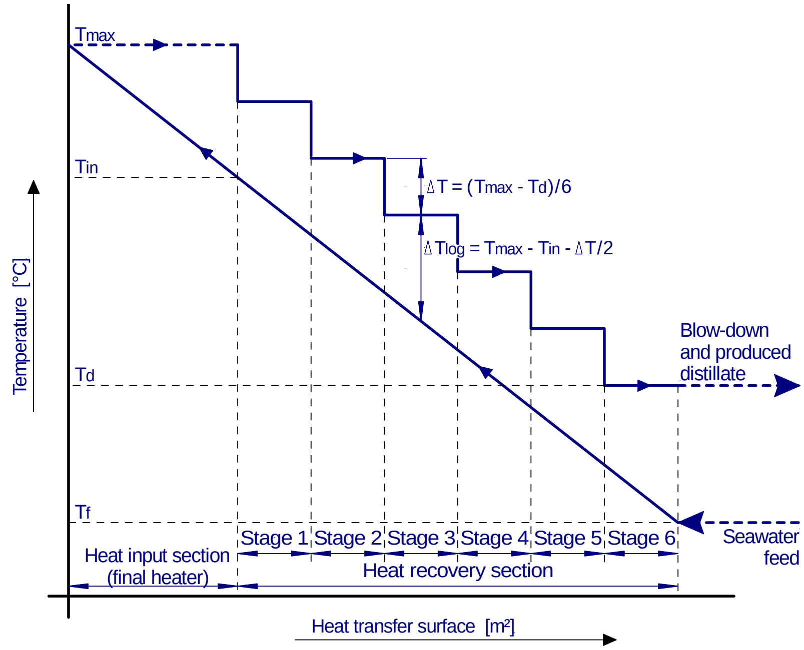

Fig. 2: Temperature diagram of a conventional MSF evaporator equipped with six stages.

SOME DEFINITIONS FOR RELEVANT PARAMETERS.

Specific heat consumption, q; i.e. the heat supplied into the final heater divided by the total amount of distillate produced, according to the equation:

$$q = h_{lg} \times \frac{T_{max} - T_{in}}{T_{max} - T_d} \tag{1}$$

For the meaning of the temperatures one is referred to Fig. 2, while h_{lg} is the average heat of evaporation in the temperature flash range (T_{max} - T_d).

Yield or gain-ratio, g_r; i.e. the total amount of distillate produced, divided by the quantity of saturated steam supplied into the final heater, according to the equation:

$$g_r = \frac{T_{max} - T_d}{T_{max} - T_{in}} \tag{2}$$

Again for the meaning of the temperatures, one is referred to Fig. 2.

Distillate production, Φ_d:

$$\Phi_d = \Phi_f \times c_l \times \frac{T_{max} - T_d}{h_{lg}} \tag{3}$$

Where Φ_f is the seawater feed flow and c_l is the average specific heat of the seawater in the temperature flash range (T_{max} - T_d).

Required specific heat transfer surface, F_s; i.e. the total heat transfer surface of the heat recovery section divided by the amount of distillate condensing per hour, according to the equation:

$$F_s = \frac{h_{lg}}{k \times (T_{max} - T_d) \times \left(\frac{q}{h_{lg}} - \frac{1}{2 \times n} \right)} \tag{4}$$

Where n is the number of stages.

CONSEQUENCES OF THE DESIGN FOR THE DIMENSIONS.

Conventional MSF evaporator.

The conventional MSF evaporator can be designed "cross-tube" or "long-tube". Both designs are shown in Fig. 3 and the differences are evident. For the "cross-tube" design the flow through the condensers is perpendicular to the flashing flow in the chambers. For a "long-tube" design both flows are parallel and counter-current. Although the "long-tube" design requires more space, its design is often preferred as it requires less water boxes for the condensers of the heat recovery section.

Efficient MSF evaporators may require 1 ton of steam for the production of 9 ton of distillate. In that case, we speak about an MSF evaporator with a yield or a gain-ratio of 9. Optimization shows that such an evaporator requires a number of stages which is often equal to approx. 3× gain-ratio, which corresponds for a gain-ratio of 9 with 27 stages. For a "long-tube" design, each stage length should be equal to the required tube length of its condenser.

In the text below, the design of a conventional MSF evaporator for comparison with an MSF evaporator equipped with fluidized bed condensers is given. For the design of the conventional MSF evaporator the tube length of each condenser has to be calculated. Therefore, the energy balance for the seawater flow through a tube is used which can be derived from the equation for the condenser tube length:

$$L_t = D_o \times \left(\frac{D_i}{D_o}\right)^2 \times \frac{\rho_l \times c_l \times V_l}{4 \times k} \times \frac{\Delta T}{\Delta T_{log}} \qquad (5)$$

Where:

- L_t = Tube length in meters to be calculated.
- D_o = Outer tube diameter, equal to 0.01905 m.
- D_i = Inner tube diameter, equal to 0.01723 m.
- ρ_l = Density of the seawater, approximated by 1 000 kg/m³.
- c_l = Specific heat of the seawater, approximated by 4 200 J/(kg·K).
- V_l = Velocity of the seawater in the tubes, equal to 1.8 m/s.
- k = Overall heat transfer coefficient, equal to 2 500 W/(m²·K).

The calculation of the values for ΔT and ΔT_{log} in the above equation are based on an installation with a gain-ratio of 9 and 27 stages, an inlet temperature of the seawater feed flow T_f = 25 °C and a maximum seawater temperature T_{max} = 95 °C. From the above parameters, it is possible to calculate the inlet temperature of the final heater T_{in} = 88 °C and the temperature of the produced distillate T_d = 32 °C.

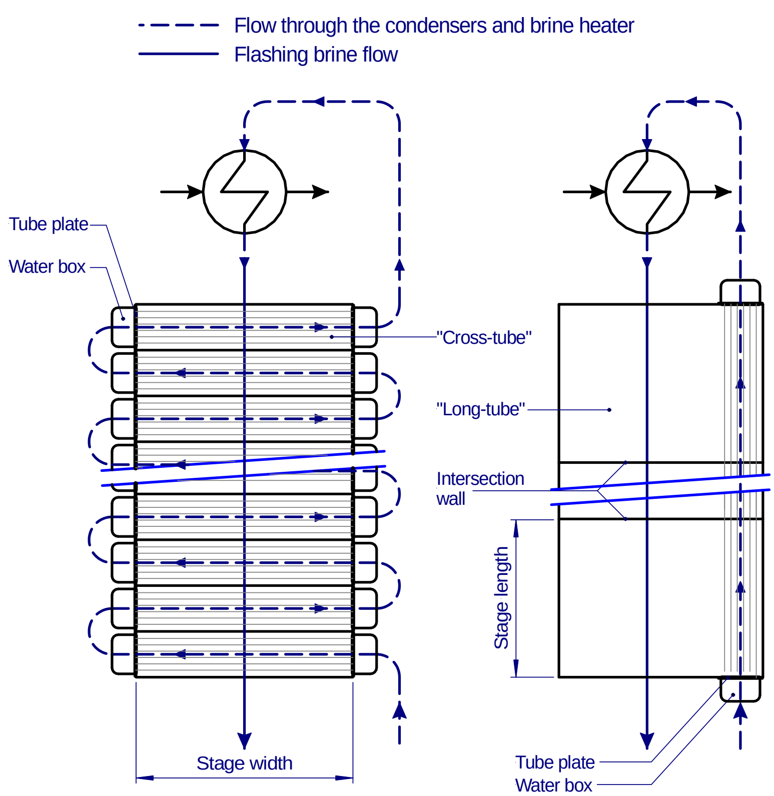

Fig. 3: Example of “cross-tube” and “long-tube” design.

Processing of all above temperatures in the temperature diagram of Fig. 2 for an evaporator with 27 stages, gives the following temperature differences:

- ΔT = Temperature increase of the seawater in each condenser is equal to 2.33 °C.
- ΔT_{log} = Mean logarithmic temperature difference for each condenser is equal to 5.84 °C.

Substitution of above relevant parameters in Eq. (5) yields for the tube length per condenser L_t = 4.70 m, and for the tube length for the condensers of all 27 stages 27 × 4.70 = 127 m. Of course, the installation of such a considerable condenser tube length in series, in combination with flash chambers and steam-water separators, requires several vessels where each vessel should contain a number of stages. For this design, 5 to 6 vessels in series would be required, each vessel with a length between 20 to 30 m.

MSF/FBE.

The principle of the MSF evaporator equipped with fluidized bed condensers and final heater is shown in Fig. 4. Its design is always “long-tube” and all condensers including the final heater require only one inlet channel and one outlet channel. Inlet channel and outlet channel have to be designed in such a way that the self-cleaning heat exchange technology using a stationary and stable fluidized bed of particles can be applied. The MSF/FBE under consideration applies the same primary design parameters as for the conventional MSF discussed above, viz. the seawater flow, the temperatures (and thus also the gain-ratio), the tube diameter and the physical properties of the seawater, with the exception of the seawater velocity in the tubes.

The differences between both installations are made by the application and the consequences for the design of the fluidized bed with the following specification:

- Diameter glass particles, equal to 2.0 mm.
- Density of the glass is 2 500 kg/m³.
- Porosity of the fluidized bed in the tubes equal to approx. 70%, which corresponds with a volume fraction of the glass balls in the tubes of approx. 30%.
- Superficial velocity of the seawater in the tubes equal to 0.125 m/s.
- Average k-value for all condensers calculated at 2 500 W/(m²·K).

Substitution of the values for the primary design parameters, the superficial velocity of the seawater in the tubes and the average k-value as used in the fluidized bed design into Eq. (5), gives for the tube length for each condenser L_t = 0.33 m and for the tube length for the condensers of all 27 stages 27 × 0.33 = 8.9 m.

This dramatic reduction in tube length by almost a factor 15 (!!) for the MSF/FBE in comparison with the conventional MSF *evaporator is caused by the typical characteristics of the fluidized bed heat exchangers which combine excellent heat transfer in their tubes with very low liquid velocities. This is due to the turbulence created by the fluidized particles in the tubes, which reduces the thickness of the laminar boundary layer in the tubes and, consequently, improves heat transfer.*

An example of an installation which applies this unique phenomenon in heat transfer is shown in Fig. 5. This figure shows the MSF/FBE demonstration plant, which operated on the Isle of Texel, in the Netherlands from 1978 till 1982. The specification of this plant is given in Table 1. As a matter of fact, the plant performed better than its specification.

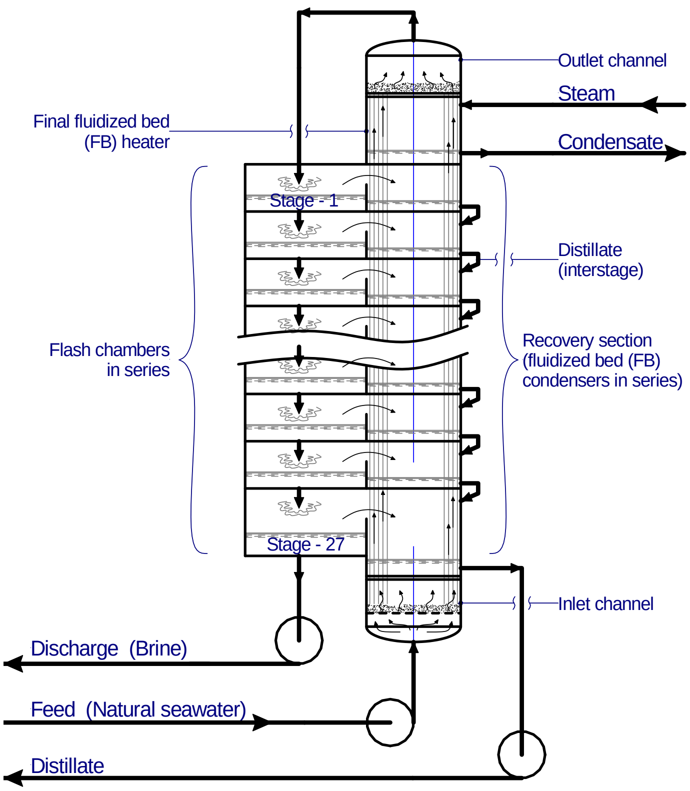

Fig. 4: Flow diagram of MSF/FBE.

More advantages of the MSF/FBE in comparison with the conventional MSF evaporator.

Apart from an enormous advantage of the MSF/FBE in comparison with the conventional MSF evaporator to install the complete MSF/FBE in only one vessel of modest height instead of five to six vessels of very large dimensions determined by the condenser tube length for the conventional MSF evaporator, there are many more advantages for the MSF/FBE:

- As a matter of fact the conventional MSF evaporator can only operate at a maximum temperature of approx. 95 °C by using chemicals to prevent the formation of calcium carbonate scale. The MSF/FBE shown in Fig. 5 has operated on chemically untreated natural seawater at maximum temperatures ranging from 93 to 115 °C without experiencing any deterioration of the k-value due to the formation of scale. Apparently, the glass particles do not only assure a very good heat transfer in spite of very low liquid velocities in the tubes, they also remove scale crystals at an early stage and keep the heat transfer surface impeccably clean.

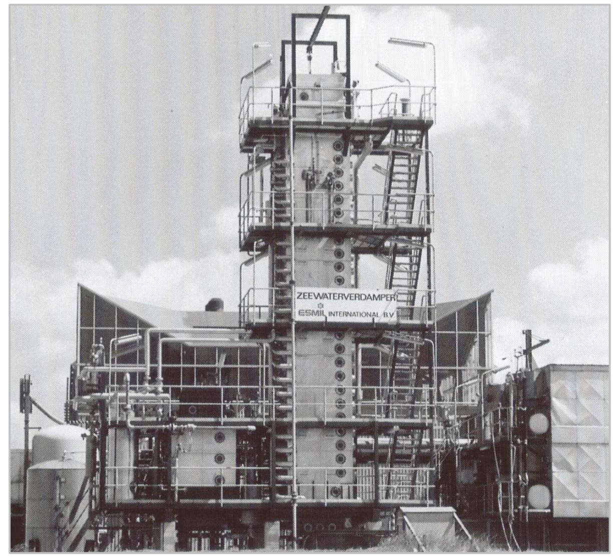

Fig. 5: Multi-stage flash/fluidized bed evaporator (MSF/FBE).

- It has been shown that an MSF/FBE with a gain-ratio equal to 9 or 10 can easily be installed in only one vessel with a height of approx. 10 m. Increasing the height of this vessel to e.g. 20 m makes it possible to increase the number of stages and, consequently, also the gain-ratio of an MSF/FBE increases to 15 or 20. These high gain-ratios have never been realized in conventional MSF plants and is a revolutionary improvement in efficiency for seawater desalination plants employing multi-stage-flash evaporation.
- Flash chamber volume of the MSF/FBE can be much less than for the conventional MSF. This is caused by the fact that the MSF/FBE reaches a complete flash-off of the brine in the flash chambers much faster and separation of water droplets from the flash vapour in the flash chambers can de realized by a very simple deflection plate instead of expensive and voluminous wire-mesh demisters.
- Distillate production of an MSF/FBE can be varied between 0 and 100% in a matter of 15 minutes. For a conventional MSF evaporator, the distillate production can be varied from 70 to 100% but this might take many hours. The flexibility in distillate production for the MSF/FBE is caused by the installation of an inter-stage valve system, which runs through all stages and creates the possibility of avoiding too high brine levels while changing the maximum seawater temperature which directly affects the distillate production.
- The total pumping power requirements are lower for an MSF/FBE than for a conventional MSF evaporator.

For more information about the development and the operation of the MSF/FBE, one is referred to the Ref. [1], [2], [3], [4], [5] and [6].

Table 1: Specification of MSF/FBE.

Distillate production	16 535	kg/h
Steam consumption	1 983	kg/h
Steam pressure	2.55	bara
Steam temperature	125.0	°C
Gain-ratio	8.34	-
Specific heat consumption	277	kJ/kg
Maximum brine temperature	93.3	°C
Feed temperature	25.0	°C
Total number of stages	26	-
Feed (untreated natural seawater)	164 000	kg/h
Total number of condenser tubes	1 557	-
Condenser tube diameter	19.05 × 0.91	mm
Average condenser tube length per stage	400	mm
Condenser tube material	AlBr	-
Diameter of fluidized particle	2.0	mm
Density of particle material	2 500	kg/m³
Total particle weight	~ 4 000	kg
Porosity	77	%

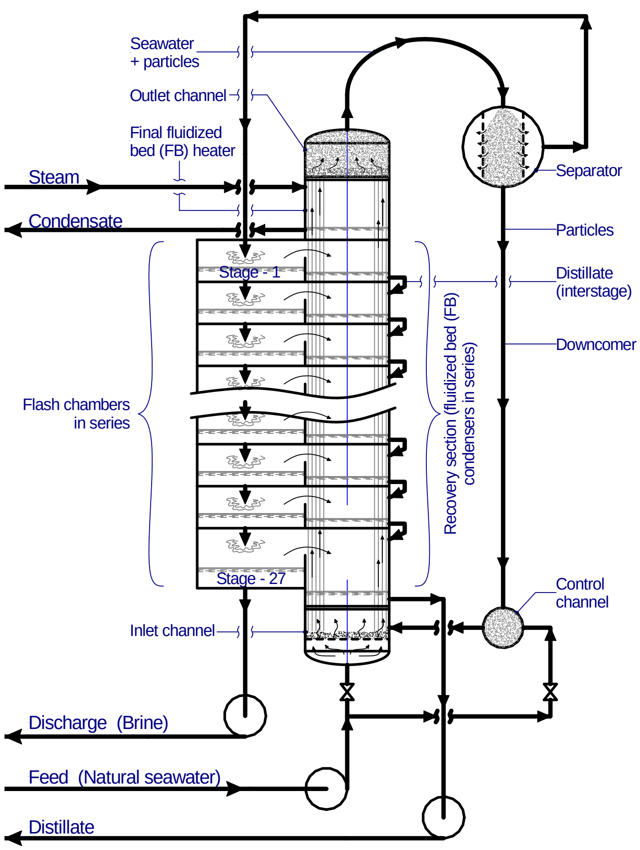

Fig. 6: Flow diagram of MSF/FBE employing external circulation of particles.

Current state-of-the-development of the MSF/FBE.

Of course, the MSF/FBE described above also suffered from some problems. There was always a chance that, in spite of the requirement to operate all parallel tubes as stationary fluidized beds with an evenly distributed upward seawater flow, severely uncontrolled internal circulation of seawater and particles could take place. In some tubes this created a down-flow of hot seawater and particles which caused temperature back-mixing in the inlet channel and immediately reduced the gain-ratio. This detrimental effect on the performance could be avoided by operating the installation with a very shallow layer of fluidized particles above the tube-plate in the outlet channel. However, such an operating condition required a very constant seawater feed flow and the attention of the operators. If such an uncontrolled internal circulation of seawater and particles occurred, it could be corrected by the operators in a matter of minutes.

The continued development and introduction of external circulation of particles for self-cleaning heat exchangers in the 90s improved the design of the MSF/FBE and solved the above problem. This improved design is shown in Fig. 6. It is characterized by the fact that a very small slip velocity of the particles in the tubes is allowed, which means that the fluidized particles are flowing upwards through all parallel tubes with a very low velocity of maybe only 1 to 2 cm/s. Each tube outlet is provided with a simple check (ball) valve to prevent any back flow or downward flow of particles and liquid through a tube. This upward flow of particles from the inlet channel into the outlet channel, requires the installation of a separator to remove the particles from the liquid, and a downcomer and control channel to return the particles into the inlet channel. The downcomer has to be designed in such a way that it only has a down-flow of particles and no down-flow of hot seawater, in order to prevent the unwanted phenomenon of temperature back-mixing.

POTENTIAL MARKETS FOR SELF-CLEANING HEAT EXCHANGE TECHNOLOGY IN SEAWATER DESALINATION.

It has already been mentioned that due to the fast introduction of the membrane processes in seawater desalination, the break-through of the successful MSF/FBE, which applies the self-cleaning heat exchange technology, stagnated. In the early 80s investors did not want to spend more money for the up-scaling of a very successful MSF evaporation technology, as the market for the MSF evaporators was indeed shrinking. And besides the MSF/FBE on the Isle of Texel described above and shown in Fig. 5, only two commercial plants were built for the production of boiler feed water from polluted harbour water for a waste incineration plant in Amsterdam. These MSF/FBE's each produced 100 ton / day with a gain-ratio of 2.5 and had 6 stages.

Today, some important changes are noticed. Also membrane processes have met their limits and surface waters have become polluted increasingly. This requires a sophisticated and costly pre-treatment of such waters, which is not always in favour of the membrane processes. As a consequence, we do observe a modest revival of interest for reliable, thermally efficient and easy to operate MSF/FBE's with distillate productions up to several thousands tons per day.

However, we also believe there are even more possibilities for the introduction of this technology in existing conventional MSF plants.

Many operators of existing conventional MSF evaporators are under pressure to increase their distillate production. The most obvious way to raise production is to increase the maximum temperature T_{max} of the seawater heated in the final heat exchanger, which then also increases the flash range (T_{max} - T_d) and, according to Eq. (3), the distillate production. However, all conventional MSF plants of the design shown in Fig. 1 are limited in their maximum seawater temperature to approx. 95 °C as the chemicals they use to prevent scaling deteriorate at higher temperatures and loose their effectiveness in preventing scale.

Combination of conventional MSF evaporator and MSF/FBE.

Now, it has been shown that the MSF/FBE can operate at much higher temperatures than 95 °C it is worthwhile to consider a design consisting of a combination of a conventional MSF evaporator for the lower temperatures and an MSF/FBE for the higher temperatures. Fig. 7 shows such a combination of both evaporators. While the maximum temperature of the seawater in the condensers of the conventional MSF evaporator is limited to 95 °C, in the high-temperature MSF/FBE the temperature range can be extended from 95 to 115 °C.

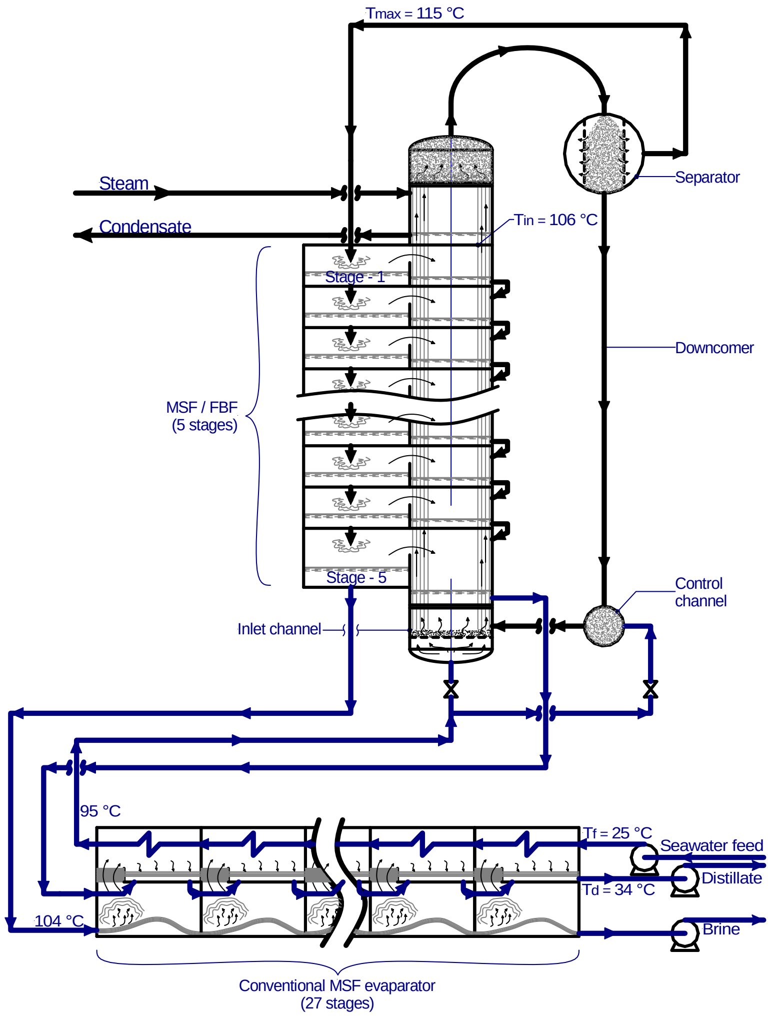

Fig. 7: Configuration of conventional MSF evaporator and MSF/FBE for the higher temperatures.

Also this combination of MSF evaporators operates with a gain-ratio of 9 and a seawater feed temperature T_f = 25 °C, which then for the other process temperatures, yields the following values: T_{in} = 106 °C, T_{max} = 115 °C and T_d = 34 °C.

In comparison with the conventional MSF evaporator operating with a maximum seawater temperature of 95 °C, a gain-ratio of 9 and a flash range (T_{max} – T_d) = (95 – 32) = 63 °C, this combination of MSF evaporators with a maximum temperature of 115 °C, has increased its distillate production by 28.5% in comparison with the conventional MSF evaporator. This was achieved by increasing its flash range from 63 to 81 °C (resulting from the temperature difference 115 – 34 = 81 °C) and keeping the seawater feed flow constant.

It should be realized that for the new situation a higher steam pressure for the heating of the seawater in the final heater is required. From experience, it is known that this higher steam pressure is usually available.

Conventional MSF evaporator with self-cleaning final heater.

Another even more simple method to increase the distillate production of an existing conventional MSF evaporator is to replace the conventional final heater by a self-cleaning final heat exchanger which allows heating of the seawater at a higher temperature than 95 °C, This design is shown in Fig. 8. Again maintaining a gain-ratio of 9 for the newly proposed installation, the temperature of the seawater in the final heater increases from T_{in} = 95 °C to T_{max} = 102.8 °C with a distillate temperature T_d = 32.8 °C. This then increases the flash range to (T_{max} – T_d) = (102.8 – 32.8) = 70 °C and the distillate production by a factor 70 / 63 = 1.11. This means that 11% more distillate is produced in comparison with the conventional MSF evaporator. It is worthwhile to investigate if it would be possible to revamp the existing conventional final heater into a vertical self-cleaning configuration. This could save on the investment cost for the self-cleaning final heater.

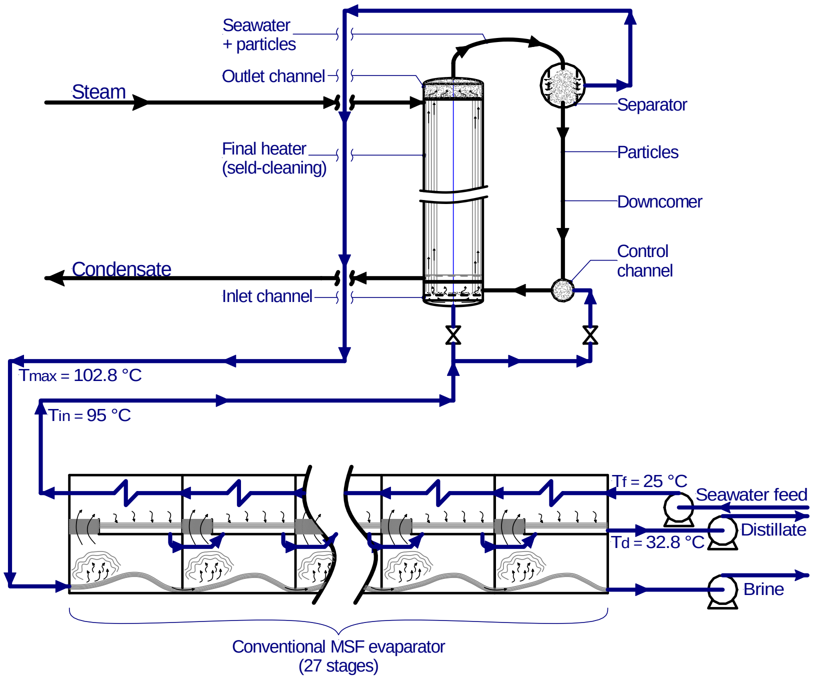

Fig. 8: Conventional MSF evaporator with high-temperature self-cleaning final heater.

risk of corrosion and makes it possible to consider the use of cheaper materials, e.g. duplex instead of titanium.

For a particular application, using very severely fouling seawater, the process conditions are:

- Heat load : 460 MW
- Closed-loop water flow : 40 000 m³/h
- Closed-loop water temperature in / out : 43.9 / 33.9 °C
- Seawater flow : 80 000 m³/h
- Seawater temperature in / out : 26.7 / 31.7 °C

For environmental reasons, the temperature increase of the seawater has been restricted to 5 °C, which explains the rather large seawater flow.

For the handling of this thermal duty, four parallel coolers were selected. The mechanical design parameters of which are specified in Table 2. For more information on this particular subject, one is referred to Ref. [7].

OTHER APPLICATIONS FOR SELF-CLEANING HEAT EXCHANGERS USING UNTREATED NATURAL SEAWATER AS A COOLANT.

Closed-loop coolers for large chemical plants.

Self-cleaning heat exchangers are now being considered for new plants that are to be built in coastal areas, such as the Caribbean and the Middle East where there is little or no fresh water available for cooling services. On these locations, seawater theoretically makes the best heat sink. However, seawater heat exchangers may suffer from a combination of very severe biological fouling, sedimentation fouling and, at higher temperatures, fouling due to scaling.

An excellent solution for this dilemma is to use the self-cleaning heat exchanger as closed-loop coolers or ‘mother-coolers’ for large chemical plants. The main purpose of the application of closed-loop coolers is to avoid the use of seawater in a large number of plant coolers and thereby concentrate the problems of corrosion and fouling in a few very large seawater-cooled closed-loop coolers which are responsible for the cooling of the closed-loop fluid used throughout the plant. Normally, this closed-loop fluid is conditioned water which does not foul or corrode the shell-side of either the closed-loop coolers or the exchangers in the plant. Neither does it foul or corrode the connecting piping, the pumps, the valves, etc. in the clean coolant circuit between the closed-loop coolers and the exchangers in the plant.

Self-cleaning closed-loop coolers have the advantage that they completely eliminate the fouling problem caused by the seawater due to the circulation of the cleaning particles through the tubes. Elimination of fouling lowers the

Table 2: Mechanical design parameters for closed-loop cooler.

Total number of tubes	11 500	-
Diameter of tubes	25.4 × 1.21	mm
Total length of tubes	7 500	mm
Heat transfer surface	6 879	m²
Overall height of cooler	14 000	mm
Width of cooler	3 000 / 4 000	mm
Length of cooler	8 000	mm
Material shell-side	Carbon steel	-
Material tube-side	Duplex	-
Material cleaning particles	3 mm Glass balls	-
Design pressure inlet channel	2	barg
Design pressure outlet channel	-1 / 1	barg
Design pressure shell	2	barg
Pressure drop tube-side	0.35	bar
Pressure drop shell-side	1.8	bar

Coolers on offshore platforms.

Many potential cooling applications which could make use of untreated natural seawater as a coolant are found on offshore platforms. Properties which influence the choice for heat exchangers performing a cooling service on an offshore platform, are the following:

- Reliability.
- Maintenance.
- Weight.
- Plot area.
- Power consumption.

It is evident that reliability and maintenance of the installed equipment are important as they directly influence production. Moreover and different from onshore installations, the weight and the plot area of heat exchangers on offshore platforms are very important, because they directly influence the cost of the platform.

In the case study presented below, we compare air coolers, often the preferred option for the cooling of natural gas on a platform, with self-cleaning exchangers using seawater as a coolant. The process conditions for the gas-side are given in Table 3 and the important characteristics for the actually installed air coolers for this particular application are the following:

- Total weight = 350 000 kg
- Total Plot area = 350 m²
- Required fan power = 750 kW

An impression of this platform, where the arrow pointing to the top-deck indicates the position of these air coolers, is shown in Fig. 9.

For the self-cleaning exchangers, we compare two different designs. The first design allows the high-pressure gas in the heat exchange tubes with untreated natural seawater at the shell-side. The self-cleaning performance is realized by the fluidization of glass particles by the upward seawater flow in the shell. It is quite clear that this design is directly derived from the MSF/FBE technology discussed before. It also answers the often asked question that, in principle, the self-cleaning heat exchange technology can be applied in the shell of tubular heat exchangers.

The second design applies the high-pressure gas in the shell, while the seawater is flowing through the tubes. For this application the self-cleaning performance is achieved by the fluidization and recirculation of metal particles through the tubes. Both self-cleaning designs, although quite different from each other, show large advantages in comparison with the use of air coolers for the same service.

Design nr.1: Self-cleaning cooler with untreated natural seawater in the shell and high-pressure gas in the tubes. The design principle of the proposed exchanger is shown in Fig. 10 and shows a vertical exchanger employing a fluidized bed in a shell with a rectangular cross section. The slip velocity of the fluidized bed requires recirculation of the cleaning particles between outlet channel and inlet channel through separator and downcomer.

For the operating conditions and the specification of the fluidized bed in the shell of the proposed self-cleaning cooler, we refer to Table 4, while the important design parameters of the proposed self-cleaning cooler are given in Table 5. Again we observe an excellent heat transfer film coefficient at very low superficial liquid velocities.

Fig. 11 shows an example of the flat-box inlet and outlet channels for the high-pressure gas-side for this application. All screwed plugs are in line with the tubes and removing the plugs makes it even possible to weld the tubes into the tube plates and to inspect the tubes. This flat-box channel design is often used in air coolers for the cooling of high

Table 3: Process conditions gas-side flow cooling service on offshore platform.

Heat load	46	MW
Gas flow	190	kg/s
Inlet temperature gas	120	°C
Outlet temperature gas	43	°C
Inlet pressure gas	150	bar
Density gas	118	kg/m³
Specific heat gas	3.2	kJ/(kg·K)
Thermal conductivity gas	0.05	W/(m·K)
Dynamic viscosity gas	0.016	mPa·s
Fouling factor gas side	0.00026	m²·K/W

Fig. 9: Offshore platform with air coolers on top deck indicated by arrow.

pressure fluids. The flat walls for the low-pressure rectangular shell of this self-cleaning cooler do require some reinforcement, but also have the advantage that removable flat covers can be applied for inspection of the proprietary re-fluidisation plates and the outside surface of the tubes.

Table 4: Operating condition and specification fluidized bed of self-cleaning cooler shown in Fig. 10

Heat load	46	MW
Seawater flow	1 600	m³/h
Inlet temperature seawater	26	°C
Outlet temperature seawater	50	°C
Material fluidized particles	Glass	-
Diameter particles	2.0	mm
Density particles	2 500	kg/m³
Porosity fluidized bed	78	%
Superficial seawater velocity	0.171	m/s
Slip velocity fluidized bed	0.021	m/s
Total glass particles transport	29.9	kg/s
Film coefficient	5 640	W/(m²·K)
Fouling factor	nil	m²·K/W

Table 5: Design and specification of self-cleaning cooler shown in Fig. 10.

Overall heat transfer coefficient (k-value)	863	W/(m²·K)
Total number of tubes	2 004	-
Diameter of tubes	19.05 × 3.25	mm
Tube length	11 870	mm
Total heat transfer surface	1 423	m²
Tube pattern	square	-
Pitch	40	mm
Tube bundle configuration	12 rows × 167 tubes/row	-
Total number of re-fluidization plated in the shell (also baffles for strength)	10	-
Inner dimensions rectangular flow area of the shell	475 × 6 675	mm
Pressure drop shell-side due to bed weight and re-fluidization plated	0.64	bar
Pressure drop tube-side	0.5	bar
Pumping power requirements for pressure drop shell-side	32	kW
Maximum total weight heat exchanger at operating conditions	107 000	kg
Plot area heat exchanger	45	m²

Design nr. 2: Self-cleaning cooler with untreated natural seawater in the tubes and high-pressure gas in the shell. The design for this self-cleaning cooler with high-pressure gas in a rather large diameter shell does not seem to be a logic choice. However, in this paragraph, it is shown that this configuration still offers an extremely interesting design.

Fig. 12 shows the design, which allows for the thermal expansion of shell with respect to the tube bundle by the application of a stuffing box for the throughput of the high-pressure inlet nozzle connected to the low pressure seawater inlet. This stuffing box does not seal high-pressure gas from the atmosphere but the liquid condensates from vapours present in the gas. By removing the low pressure flange from this high-pressure nozzle, the bundle can be pulled out the high-pressure outer shell, which offers the possibility to clean the actual heat transfer surface in contact with the gas after removing the half-cylindrical segments of the thin-walled inner shell. Another alternative to seal the high-pressure gas from the atmosphere, while still maintaining the possibility of thermal expansion of the shell versus the tube bundle, could be the application of a bellow on the high-pressure nozzle connected to the seawater inlet. However, with a bellow the removal of the bundle from the shell becomes more complicated.

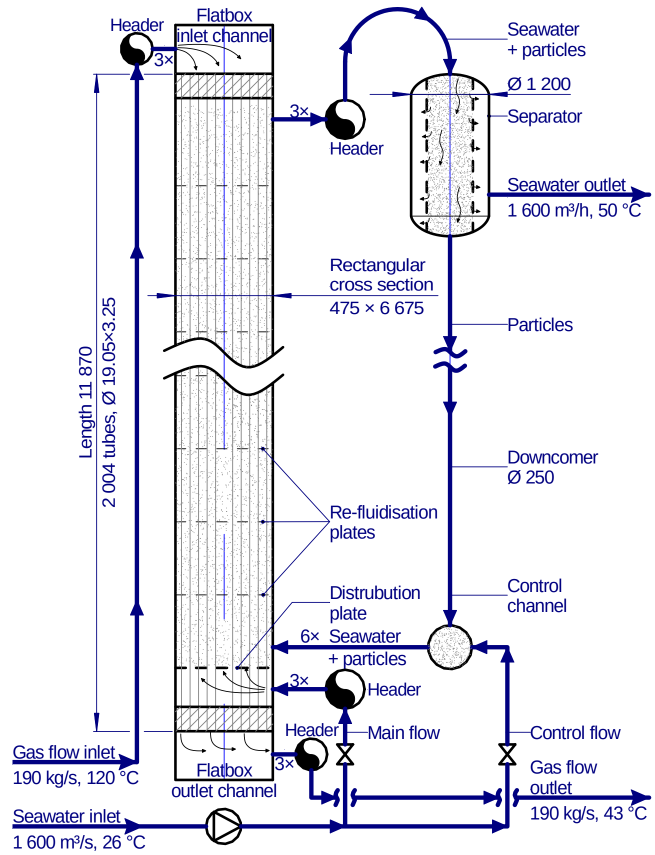

Fig. 10: Self-cleaning cooler with seawater in shell and high-pressure gas in tubes.

Also this self-cleaning cooler employs circulation of the particles. After being separated from the seawater the cleaning particles are returned through a downcomer and control channel into the inlet receiving channel and are carried from this receiving channel into the actual inlet channel where even distribution of the particles over all parallel tubes takes place.

The actual inlet channel is pulled against the lower tube plate by sufficient bolts to prevent this channel from leaking seawater into the shell at only modest seawater pressure and atmospheric pressure conditions in the shell. However, when the actual high pressure gas is supplied into the shell, the large force created by the gas pressure pushes the inlet channel to the tube plate and should provides the sealing to prevent leaking of gas from the shell into the inlet channel.

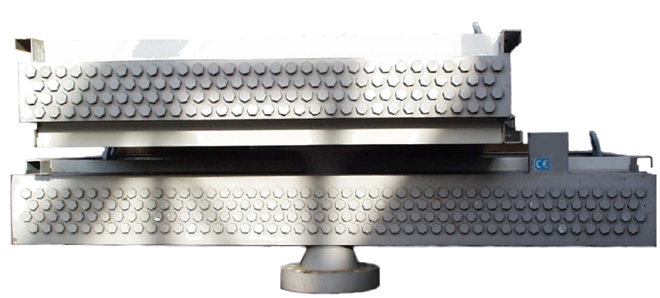

Fig. 11: Example of flat-box channels in high-pressure air coolers.

For the actual tube bundle in the thin-walled inner shell, we recommend the application of so-called Grid-baffles as referred to in Ref. [8], or EM baffles explained in Ref. [9] and shown in Fig. 13. The advantages of both types of baffles can be summarized as follows:

- Less flow induced vibrations of the tubes.
- Excellent shell-side heat transfer film coefficient.
- Low shell-side pressure drop.
- Much lower shell-side fouling.

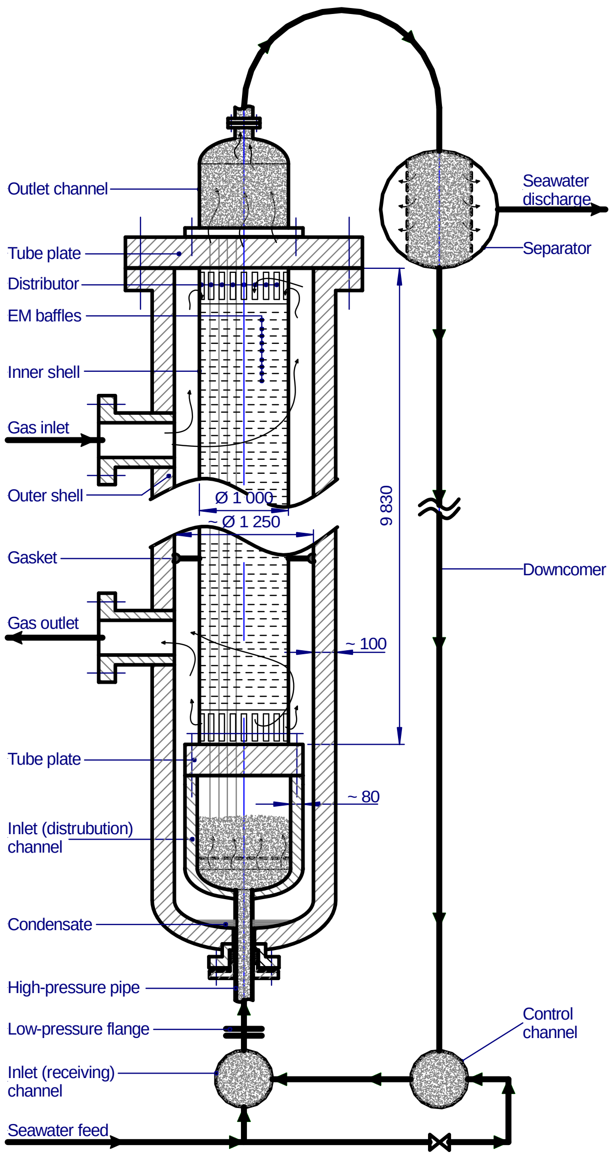

Fig. 12: Self-cleaning cooler with seawater in tubes and high-pressure gas in shell.

Table 6 specifies the conditions for the fluidized bed side of this self-cleaning cooler. In comparison with the self-cleaning cooler shown in Fig. 10, there are the following significant differences:

The seawater flow is reduced from 1 600 to 920 m³/h, which increases the seawater outlet temperature from 50 to 70 °C. Instead of glass particles, chopped metal wire particles with a diameter of only 1.2 mm are used. Velocities and film coefficient at the fluidized bed side are much higher than for the design shown in Fig. 10. In spite of the high outlet temperatures of the seawater, we know from experiences that the bed porosity of 98% (i.e. only 2% by volume of chopped metal wire particles) will be able to keep the tube wall clean.

Table 6: Operating condition and specification fluidized bed of self-cleaning cooler shown in Fig. 12.

Heat load	46	MW
Seawater flow	920	m³/h
Inlet temperature seawater	26	°C
Outlet temperature seawater	70	°C
Material fluidized particles	Stainless steel	-
Diameter particles	1.2	mm
Density particles material	7 800	kg/m³
Porosity fluidized bed	98	%
Superficial seawater velocity	1.13	m/s
Slip velocity fluidized bed	0.84	m/s
Total particles transport	30	kg/s
Film coefficient	11 336	W/(m²·K)
Fouling factor	nil	m²·K/W

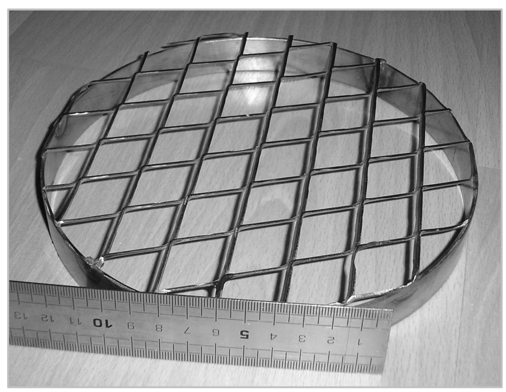

Fig. 13: Example of EM baffle.

Table 7 extends the specification with respect to the overall design of the self-cleaning cooler shown in Fig. 12 and we like to ask your attention for a number of interesting features of this design:

- As a result of the latest developments of the self-cleaning technology we have been able to install smaller tube diameters. These smaller tube diameters allow a very compact design as reflected by the plot area. These smaller tube diameters, in combination with the Grid baffles or EM baffles in the shell, also very much improves the overall heat transfer coefficient.
- Assuming that the pressure drop for the gas flow in the air coolers is approx. 1 bar, we have designed our self-

cleaning heat exchanger for the same pressure drop of the gas flow at the shell-side and still find a remarkable high overall heat transfer coefficient or k-value equal to 1 522 W/(m²·K).

- The maximum installation weight of the self-cleaning heat exchanger with high-pressure gas in the (large diameter) thick-walled shell is surprisingly low. This lower maximum installation weight is mainly caused by the very compact design of this self-cleaning heat exchanger, which reduces the weight of the maximum possible liquid contents in the heat exchanger to be taken into account in the maximum installation weight.

Table 7: Design and specification self-cleaning cooler shown in Fig. 12.

Overall heat transfer coefficient (k-value)	1 522	W/(m²·K)
Total number of tubes	2 107	-
Diameter of tubes	15.88 × 2.11	mm
Effective tube length	9 830	mm
Total heat transfer surface	1 033	m²
Tube pattern	Equilateral triangular	-
Pitch	19.88	mm
Diameter inner shell	1 000	mm
Inner diameter outer shell	1 250	mm
Pressure drop tube-side due to bed weight, tube wall friction and distribution system	0.8	bar
Pressure drop shell-side	1.0	bar
Pumping power requirements for pressure drop tube-side	25	kW
Maximum total weight heat exchanger at operating conditions	110 000	kg
Plot area heat exchanger	20	m²

Comparison of the various designs Table 8 compares the most important characteristics of the various types of self-cleaning seawater coolers, such as pumping power requirements, maximum installation weight and plot area, with conventional air coolers for the same service on offshore platforms. In this comparison we have not taken into account the conventional seawater cooler because its severe fouling at the seawater side makes a relevant comparison with coolers which do not foul at the seawater side irrelevant.

However, for a fair comparison of the air coolers versus the self-cleaning seawater coolers we also have to take into account the following facts:

The conventional air coolers as shown in Fig. 9 consist of multiple units operating in parallel. As a consequence, production is still guaranteed with one or two air coolers out of service for repair or maintenance. If we also like to achieve a certain level of guaranteed production for our self-cleaning coolers, it is necessary to divide the production over several parallel operating units. In case of an installation consisting of 2 × 50% production units, the maximum installation weight and plot area increases respectively by approximately 25% and 50% and the actual consequences in numbers are also shown in Table 8.

For the pumping power requirements of the self-cleaning coolers, we have not taken into account the pumping power which is necessary to lift the seawater from sea level to the floor level of the exchangers. We have assumed that we obtain our seawater from a multi-purpose ring-line on a floor level of the platform and discharge our heated seawater to a similar ring-line on the same floor level. When we should taken into account the power required to lift the seawater from sea level to the required floor level over a height of 15 m with a pump efficiency of 80% and we do not apply a recovery turbine for the seawater discharge, we have to increase the pumping power requirements for the self-cleaning coolers, which consequences are also shown in Table 8.

CONCLUSIONS.

Attention has been paid to a variety of applications where untreated natural seawater is used as a coolant in self-cleaning exchangers. This offers the possibility of heating the seawater to higher temperatures without the risk of scale formation.

First, we have paid attention to the influence of the self-cleaning heat exchange technology on the design of the MSF/FBE. The advantages in design and its simple operation explain the revival in interest for small production capacities and energy efficient MSF/FBE's at remote locations using chemically untreated natural seawater as a coolant and feed flow for the production of distillate. The production capacity of existing conventional MSF evaporators can be increased substantially by raising the maximum seawater temperature of the evaporator. However, to avoid problems due to the formation of scale at these higher temperatures, the conventional MSF evaporator should be combined with an MSF/FBE which then should operate in series with the conventional MSF evaporator at the high-temperature end of the installation. An even more simple approach to increase the distillate production of an existing conventional MSF evaporator may be the revamp of the existing final heater into a self-cleaning configuration. This would allow the self-cleaning final heater to operate at higher temperatures without the risk of scale formation.

Next, we have explained the advantages of the self-cleaning heat exchange technology for the application in closed-loop coolers of very large chemical plants which have to rely on severely fouling, and preferably untreated natural seawater, as a coolant.

Finally, we have shown the potential of the self-cleaning heat exchange technology for applications on offshore platforms using untreated natural seawater as a coolant. Here it could replace the conventionally used heavy, bulky, voluminous and energy-inefficient air coolers for the same service. This case study shows total weight reductions of the required heat exchangers by at least a factor 3, reduction in plot area for the heat exchangers by at least a factor 8 and a reduction of the external power input, i.e. fan power versus pumping power for the seawater, by more than a factor 20.

Table 8: Comparison of significant parameters air coolers versus self-cleaning seawater coolers.

	Unit	**Conventional air cooler**	**Self-cleaning cooler(s) with high pressure gas in the tubes**	**Self-cleaning cooler(s) with high pressure gas in the shell**
Fan power	kW	750	n.a.	n.a.
Pumping power	kW	n.a.	32	25
Extra pumping power to lift seawater from sea level to floor level	kW	n.a.	84	48
Maximum total weight coolers at operating conditions (self-cleaning coolers 1 × 100%)	kg	350 000	107 000	110 000
Maximum total weight self-cleaning coolers at operating conditions (2 × 50%)	kg	n.a.	134 000	137 000
Plot area coolers (self-cleaning coolers 1 × 100%)	m²	350	45	20
Plot area self-cleaning coolers (2 × 50%)	m²	n.a.	68	30

NOMENCLATURE

c	specific heat, kJ/(kg·K)
D	diameter, m
F_s	heat transfer surface, m²
g_r	grain-ratio, -
h_{lg}	heat of evaporation, kJ/kg
k	heat transfer coefficient, W/(m²·K)
L	length, m
n	number of stages, -
q	specific heat consumption, kJ/kg
T	temperature, °C
V	velocity, m/s
ρ	density, kg/m³
ΔT	temperature difference, °C
ΔT_{log}	mean logarithmic temperature difference, °C
Φ	flow, kg/s or m³/s

Subscript

d	distillate
f	feed
i	inner tube
in	inlet
l	liquid
max	maximum
o	outer tube
t	tube

REFERENCES

[1] Klaren, D.G. (1975); "Development of a Vertical Flash Evaporator", *Ph.D. Thesis*, Delft University of Technology, May.

[2] Klaren, D.G. (1976); "The MSF/FBE: An Improved Multi-Stage Flash Distillation Process", *Proceedings 5th International Symposium on Fresh Water from the Sea*, pp. 167 – 182, Alghero, Sardinia, May.

[3] Pieper, G.A. (1976); “A Comparative Cost Study Between an MSF/FBE and a Conventional Horizontal Multi-Stage Flash Evaporator”, *Proceedings 5th International Symposium on Fresh Water from the Sea*, pp. 183 – 192, Alghero, Sardinia, May.

[4] Veenman, A.W. (1976); “Construction and Initial Operation of the MSF/FBE”, *Proceedings 5th International Symposium on Fresh Water from the Sea*, pp. 193 – 211, Alghero, Sardinia, May.

[5] Klaren, D.G. and J. Windt (1978); “Design and Construction of a 500 m³/day Multi-Stage-Flash / Fluidized Bed Evaporator”, *Proceedings 6th International Symposium on Fresh Water from the Sea*, pp. 15 – 30, Las Palmas, September.

[6] Spanhaak, G. (1979); “A Comparison of the Operating Characteristics of Conventional Multi-Stage Flash Evaporation and Multi-Stage Flash / Fluidized Bed Evaporation”, *Desalination*, Vol. 31, pp. 511-519.

[7] Klaren D.G. and D.W. Sullivan (2000); “Self-Cleaning Heat Exchangers in Very large Closed-Loop coolers”, *Proceedings 34th National Heat Transfer Conference*, Pittsburgh, Pennsylvania, August.

[8] Klaren, D.G., de Boer, E.F. and D.W. Sullivan (2007); “Consider Low Fouling Technology for ‘Dirty’ Heat Transfer Services”, *Hydrocarbon Processing*, Vol. 86, No. 3, pp. 71-84, March.

[9] HP Innovations (2006); “Novel Heat Exchanger Design Offers Optimum Performance”, *Hydrocarbon Processing*, Vol. 87, No. 1, pp. 27 – 29, January.