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# EMBaffle® Heat Transfer Technology Step-Up in CO<sub>2</sub> Reduction

*Marco Rottoli, Daniele Agazzi, Marcello Garavaglia  
and Fabio Grisoni*

## Abstract

EMBaffle® is a proprietary shell and tube heat exchanger technology, designed to improve performance by pressure drops control, with suppressed tube vibrations. Developed to minimize fouling accumulation in crude oil units, the technology has proved to be effective in Gas treatment and Petrochemical, supporting the increase in gas-gas and cooling water flow rate per-unit demand, and in Renewable CSP, where Molten Salt units get a primary role in thermal storage and power efficiency. Diamond shape and baffle-grids number are the instruments for the design engineer to exploit exchanger efficiency against pressure drops, aimed to the desired performance with the optimized power consumption. Further to introduce the base of the technology, this work will also address the design of higher compact units by combining the grids performance with the improved exchanger tube surface. Experimental data to support the grid life under critical working conditions and actual performances with fluids density and viscosity are reported.

**Keywords:** shell&tube heat exchanger, increased thermo-hydraulic performance, reduced maintenance costs, improved plant reliability, energy efficiency & CO<sub>2</sub> reduction

## 1. Introduction

Shell & tube (S&T) heat exchangers are among the main process equipment involved in oil refineries, power industry and chemical plants. They are made of a pressure vessel in which is inserted a bundle of tubes. One fluid pass inside the tubes, while the other passes outside them. Tubes are generally supported by segmental baffles.

Although conventional segmental-baffles units prove well in the wide majority of the services, in several cases performances are not outstanding with negative impacts on maintenance costs and exchanger life, especially in case where large flow rates have to be processed, which can induce tube vibration issues.

If a dirty fluid is processed, fouling can be accumulated in the stagnation zones that are inherently created by segmental baffle geometry.

A possible remedy is the application of a longitudinal flow at the shell side. EMBaffle® technology promotes longitudinal flow at shell side, supporting the tubes with expanded metal grids.

The following paragraphs describes this technology, from general features to its inherent advantages, pressure drop and heat transfer characteristics.

Finally some technical advancements are reported, together with some design cases related to services where EMbaffle® technology proves its advantages over conventional segmental exchangers.

## **2. Longitudinal type heat exchanger technology**

Conventional S&T heat exchangers, widely used in Oil&Gas, Petrochemical, Chemical and Power Plants, are of the “segmental baffle” type, where the baffles support the tubes and govern the cross/longitudinal ratio of the shell flow direction through the bundle. The turbulent motion originated by the cross direction, normal to the exchanger tubes, determines the shell side heat transfer coefficients (HTC), that in many configurations controls the global value, and the consumed pressure drops. Decades of operational experience led to a widespread know-how in design and manufacture of safe, high performing and long-life segmental baffles heat exchangers, driving, at the same time, to claim for alternate design concepts in order to overcome the few critical limits of the technology.

Two matters in particular have been deeply addressed, depending on process flow rates and fluid nature (clean or dirty) [1, 2]:

- potential vibrations, induced by the cross component of high flow rates, may affect the exchanger tubes reliability;
- the flow recirculation, at the dead areas formed by the baffle outer diameter with the shell inner diameter, may induce, with dirty fluids, to progressive fouling accumulation, thus reducing the heat transfer surface and leading to potential tube local overheating and corrosion issues.

Further, the lowest design temperature approach, (typical for example of the very few degrees in Power generation pre-heaters), is limited because of the cross-flow component.

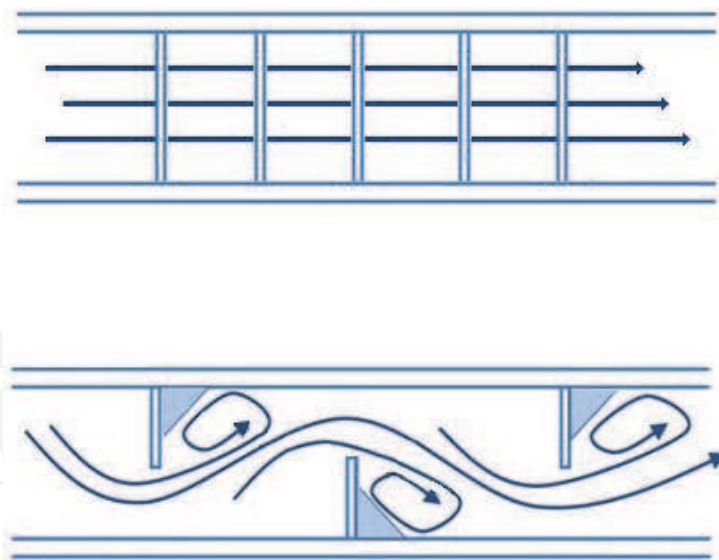
Helical and pure longitudinal flow design concepts have been exploited and exchangers have been developed to overcome some of the above criticalities.

In longitudinal flow solutions, design aims to reduced, up to the total suppression, the dead areas and the cross-flow component of a non-cross type Shell&Tube exchanger (**Figure 1**). The associated reduced drift and hydraulic resistance lead to significantly reduced pressure drops and the reduced span of tubes support elements grants a bundle compact assembly preventing potential vibration phenomena.

Among the longitudinal flow type heat exchangers, the Rod-Baffle is the pioneer. Conceived to suppress vibration issues by reducing the baffles span in shell side high flow rate and pure cross flow applications, in this technology, the tubes are supported by a repeated series of four regularly spaced-apart rod lines welded to a peripheral ring. The rod lines are disposed in alternate horizontal and vertical directions (one set of parallel rod lines at any single baffle), to form a square support, so that four baffles are required to fully confine the tube. The large free flow area left by the baffle determines a pure-like longitudinal flow in non-cross flow type HE.

It is the intent of this technology to govern a low value of pressure drops to suppress vibrations, accepting the resulting total Heat Transfer Coefficient as a consequent outcome.

Rod baffle basic concept was initially replicated by making use of strips in place of rods; only more recently solutions aimed to further reinforce the tubes



**Figure 1.**  
*Longitudinal vs. conventional S&T heat exchanger.*

confinement but taking care at the same time of the heat transfer performance have been developed.

### 3. EMbaffle® development

Aiming to grant all tubes confinement at any individual baffle, the EMbaffle® longitudinal flow type heat exchanger was then conceived and developed. Rods and strips are replaced by a patented Expanded Metal grid, (that's where EMbaffle name comes from). By making best use of the available shell side pressure drops, the unsupported tubes span can be easily managed to design the most stiff cage solution today available in the S&T heat exchangers market (**Figure 2**).

Initially thought to eliminate the dead areas in fouling applications, the grid geometries and baffles span impacts on fluid flow paths have been progressively explored, by both CFD analysis and experimental measurements, to establish design criteria aimed to maximize the shell side HTC making best use of the available pressure drops.

Two significant improvements in the longitudinal baffle technology are so achieved:

- Tubes full support at any baffle makes the technology ready to replace the Rod Baffle and “No Tube in the window” TEMA designs in most applications where



**Figure 2.**  
*EMbaffle® design – One baffle fully supporting each individual tube.*

vibration issues govern, with performance increasing under increased process flow rates within the same shell diameter constraints.

- Further, by making use of the reduced fouling accumulation and so the better use of the available pressure drops to improve global Heat Transfer Coefficient, the technology can replace in several processes different standard TEMA segmental baffle exchangers granting same/improved performances with reduced capex/opex costs.

Plant data and case studies will be proposed to offer a general view of EMbaffle design performance advantages when replacing traditional TEMA solutions.

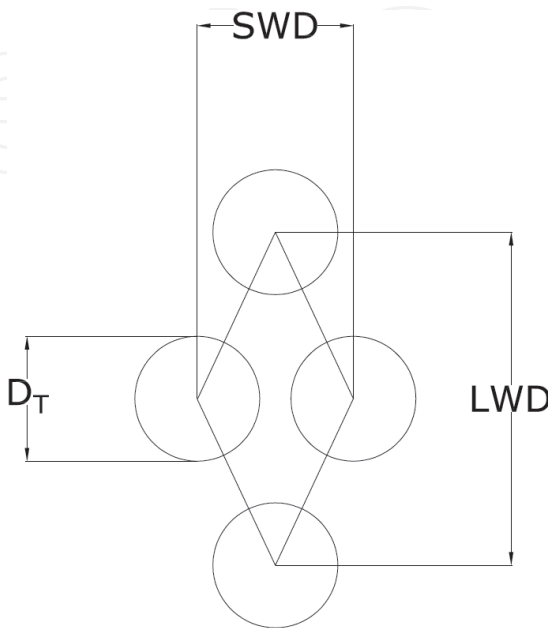
#### 4. Principles of EMbaffle® technology

To overcome the lower performance in heat transfer, intrinsic of the longitudinal flow design when compared to cross flow, EMbaffle® technology makes use of the rhombus-like shape of the expanded metal baffle mesh to promote turbulence. The two tube pitches defined by the layout are named “long way of the diamond” (LWD) and “short way of the diamond” (SWD) (**Figures 3 and 4**).

Essentially, the baffle grids generate a local turbulence whose longitudinal extension and amplitude, other than by the fluid properties, are determined by the peculiar geometry of the grid mesh (**Figure 5**).

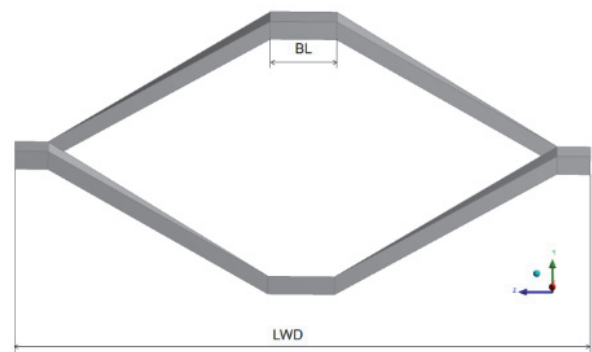
In **Figures 6 and 7**, the turbulence kinetic energy, as a measurement of the turbulence grade, is shown for different type of grid mesh shape with a specified grid span. The turbulence amplitude and extension are quite different for the different grids type.

Imposing a higher order of magnitude to the tube side heat transfer coefficient, the effect of the grids on the global heat transferred is studied by CFD analysis. As reported in **Figure 8** the heat transfer coefficient development substantially replicates the local turbulence peak at the grid, but the decay slope is significantly lower granting the maintenance of a quite homogeneous value from grid to grid.

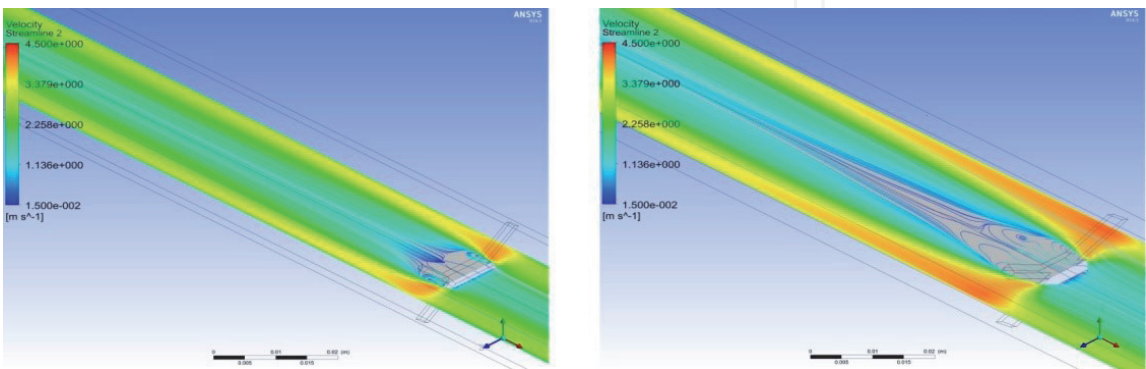


**Figure 3.**  
*EMbaffle® –typical tube layout.*

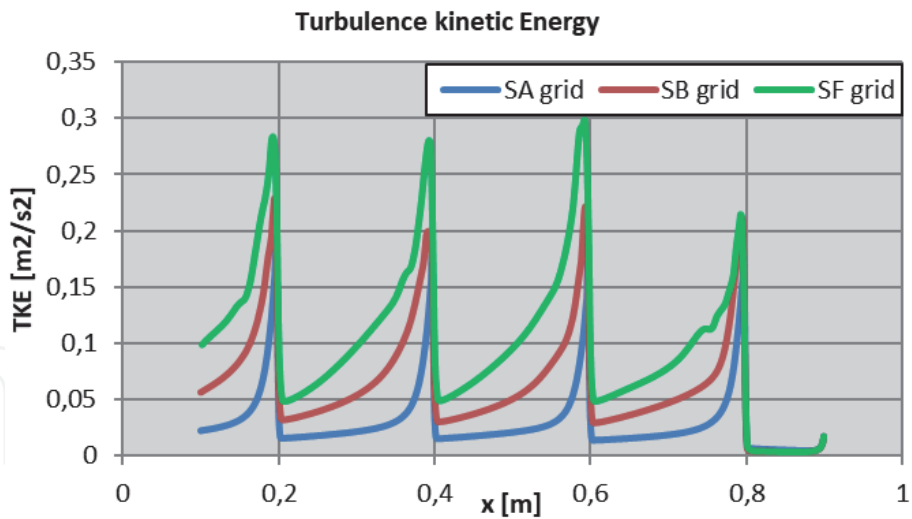




**Figure 4.**  
EMbaffle® – Typical rhombus-like shape of grid mesh [3].



**Figure 5.**  
EMbaffle® – Local flow velocity profile governed by the grid mesh shape induced turbulence.

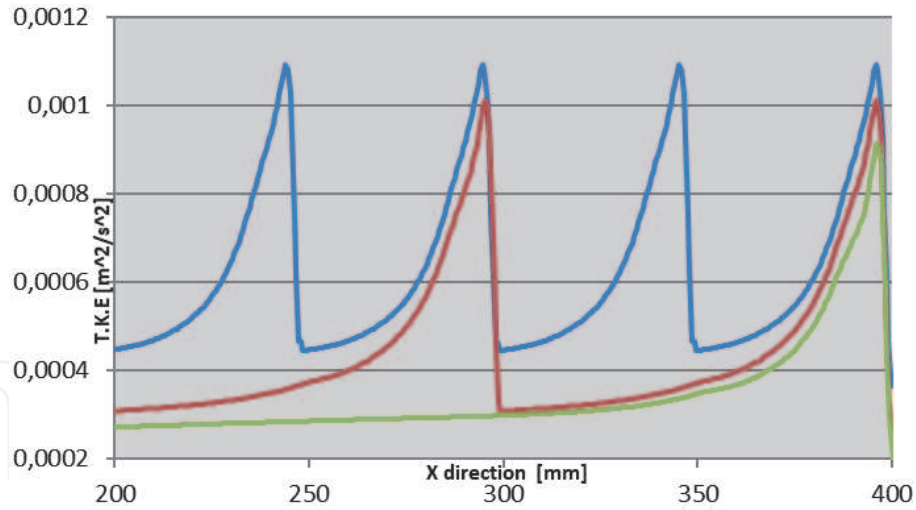


**Figure 6.**  
CFD analysis of turbulent kinetic energy generated by different grid types, gas case [3].

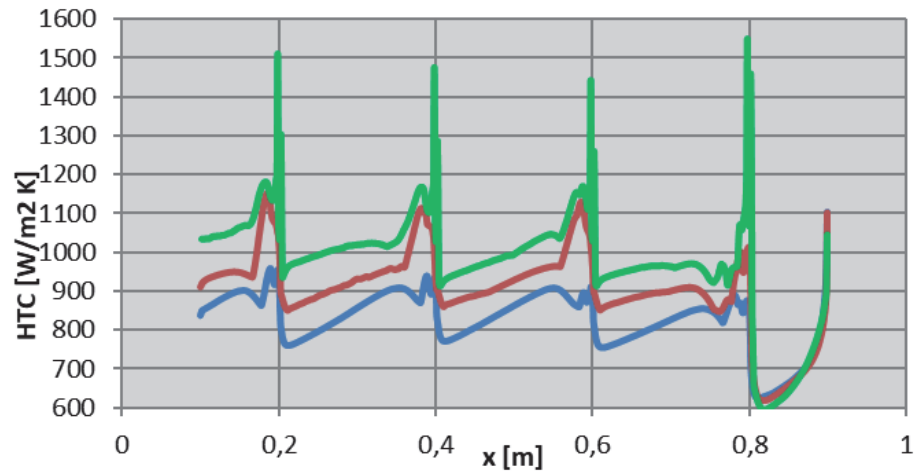
Grid mesh shape also allows for different tube count to be allocated within the same shell diameter, determining the total available heat exchange surface and the mean average flow velocity that governs the longitudinal contribution to the HTC.

Finally, increase or reduction in baffles span contributes, further to stronger or lighter tubes confinement, to the overall shell side HTC, with reversed impact on pressure drops.

The selection of grid type and grids span shall therefore be guided by the relevant boundary conditions as higher turbulence means higher pressure drops and overall HTC, while lower turbulence means lower pressure drops and lower total heat transferred.



**Figure 7.**  
CFD analysis of turbulent kinetic energy generated by different grid spans, liquid case [3].



**Figure 8.**  
From turbulence to heat transfer – Plot of HTC generated by different grid types [3].

## 5. Heat transfer in EMbaffle® technology

An important feature to design a S&T heat exchanger is the average temperature driving force  $\Delta T_m$  that can be calculated from the general global heat transfer equation:

$$Q = UA\Delta T_m \quad (1)$$

Where  $Q$  is the duty or heat transferred per unit time,  $U$  the overall heat transfer coefficient and  $A$  the heat transfer surface.

In general,  $\Delta T_m$  is determined by the approach temperatures, fluid properties and fluid arrangement. It can be calculated from the logarithmic mean temperature difference applying a correction factor:

$$\Delta T_m = \Delta T_{lm} F_t \quad (2)$$

$F_t$  is the correction factor and it depends on the S&T exchanger geometry (number of shell/tube passes and flow orientation), and distortion of the shell and tube fluid temperatures profile (thermal leakage through the longitudinal baffle, close approaches, temperature cross, bypass streams).

The correction factor  $F_t$  ranges from 0 to 1. Typically, values smaller than 0.8 indicate close temperature approaches and therefore an inadequate design for the given process conditions; the design may be easily improved by increasing the correction factor  $F_t$  switching to a counter-current type exchanger.

EMbaffle® allows to achieve a 100% counter-current configuration thanks to its pure longitudinal flow, maximizing the correction factor  $F_t$  to 1 and making the exchanger extremely performing where very tight temperature approaches are specified.

### 5.1 Heat transfer correlations

In EMbaffle® technology, the shell-side HTC is calculated using the following correlations for the Nusselt number in case of laminar and turbulent flow respectively:

$$Nu = C_L Re_h^{0.6} Pr^{0.4} \left( \frac{\mu_b}{\mu_w} \right)^{0.14} \tag{3}$$

$$Nu = C_T Re_h^{0.8} Pr^{0.4} \left( \frac{\mu_b}{\mu_w} \right)^{0.14} \tag{4}$$

The geometry coefficient functions,  $C_L$  and  $C_T$ , account for the enhancement due to the cross flow at the shell entrance and exit conditions. The Reynolds number is calculated as follows:

$$Re_h = \frac{\rho V_S D_h}{\mu_b} \tag{5}$$

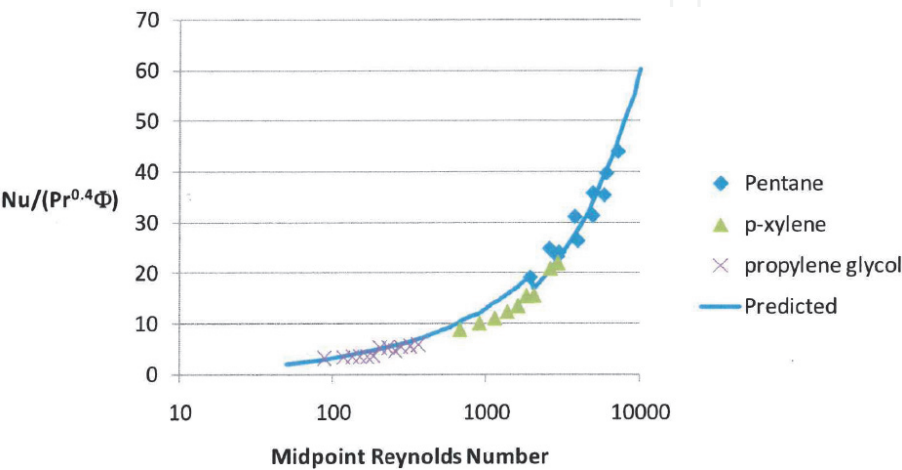
where  $V_S$  is the shell-side velocity and  $D_h$  is the characteristic diameter.

The shell-side velocity is calculated with the continuity equation, using the following expression for the shell-side flow area:

$$A_s = \frac{\pi}{4} (D_s^2 - N_T D_o^2) \tag{6}$$

The characteristic diameter is four times the nominal flow area divided by the wetted perimeter:

$$D_h = \frac{4 \left( \frac{1}{2} (LWD \times SWD) - \frac{\pi}{4} D_o^2 \right)}{\pi D_o} \tag{7}$$



**Figure 9.**  
Measured Nusselt number as a function of Reynolds number.



The above factors offer a large range of parameter options to provide the best solution in the light of the design constraints requested by the specific application.

Experimental tests had been conducted by different Institutions in order to validate the general heat transfer correlations and the coefficient  $C_L$  and  $C_T$  for different grid types. In **Figure 9** the measured Nusselt number  $A$  as a function of Reynolds Number is represented. The shift in prediction curve follows the change of Reynolds exponential dependence.

## 6. Pressure drops in EMbaffle® technology

Given the peculiar shape of the grids and the longitudinal flow patterns, EMbaffle® is characterized by reduced hydraulic resistance compared to conventional technologies. Due to this feature, in all cases where limited pressure drops are available EMbaffle® can still achieve low pressure drops for widely used TEMA types like E and F, while conventional segmental designs are forced to switch to “Low pressure drop” TEMA-types (G-, H-, J- or X). This results in a definitely more compact and thermo-hydraulically optimized design.

In EMbaffle® technology, shell-side pressure drop is the sum of the longitudinal flow component and the baffle flow component:

$$\Delta P = \Delta P_L + \Delta P_B \quad (8)$$

The expression for the longitudinal component is:

$$\Delta P_L = \frac{2\rho f_F L_T V_S^2}{D_p} \quad (9)$$

where  $D_p$  is the characteristic diameter,  $f_F$  the Fanning friction factor and  $L_T$  the length of the tubes. The characteristic diameter is calculated as follows:

$$D_p = \frac{4 \left[ \frac{\pi}{4} (D_s^2 - N_T D_o^2) \right]}{\pi D_o} \quad (10)$$

The friction factor is calculated with the following expression:

$$f_F = \begin{cases} \frac{16}{Re_p}, & Re_p < 1189 \\ \frac{0.079}{Re_p^{0.25}}, & Re_p \geq 1189 \end{cases} \quad (11)$$

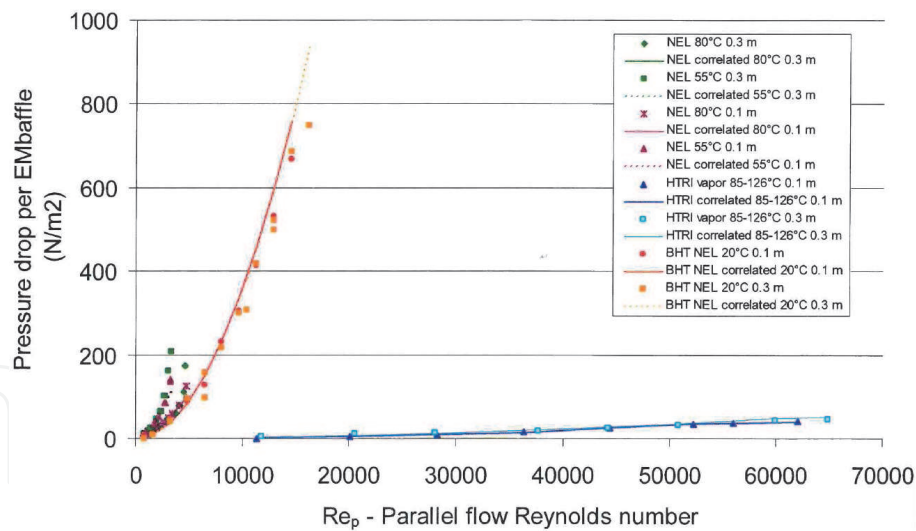
The baffle pressure drop is calculated using the baffle velocity  $V_B$  and a baffle loss coefficient  $K_B$ :

$$\Delta P_B = K_B N_B \frac{\rho V_B^2}{2} \quad (12)$$

$N_B$  is the number of the baffles. The baffle velocity is determined using the continuity equation with the following definition of the baffle flow area:

$$A_B = A_S - A_R - A_{EM} \quad (13)$$

$A_R$  is the ring area, while  $A_{EM}$  is the projected area of the EMbaffle grid.



**Figure 10.**  
*Measured pressure drops as a function of Reynolds number.*

$K_B$  is the correlation factor accounting for the effect of entrance and exit cross flow, depending on the ratio  $A_B/A_S$  and the shell length and diameter ratio.

Experimental measurements have been conducted by different Institutions and heat exchangers Manufacturers to validate the above correlations.

The global measured pressure drops are strongly influenced by the entrance and exit cross flow, especially with short experimental heat exchangers, requiring the cross check of different experimental data.

In general, the correlations do not fit properly for very high viscous fluids and for extremely high Reynolds number, while fits with proper margin for low viscosity liquid and gases in Reynolds ordinary range of design (**Figure 10**).

In a straight comparison between a conventional S&T heat exchanger and the equivalent EMbaffle® heat exchanger under the same duty, EMbaffle® design often results in significant shell-side lower pressure drops, allowing in several experienced cases to sensibly increase the flow rate without asking for increased pump or compressor consumption.

## 7. Vibrations in EMbaffle® technology

Flow-induced vibrations are determined by the interaction of a cross flow with a physical body; this produces the shedding of alternating vortices, that transfers mechanical energy to the body. If one of the natural frequencies of the body is matched, such a configuration starts to vibrate. Vibration can be mechanical vibration of the tubes or acoustic resonance of the exchanger shell.

In all gas services and high flow-rate cooling services, prevent vibration is a relevant issue for equipment design. While demand of higher and higher flow-rates to be processed is growing, No-Tubes-In-Window (NTIW) design (i.e. the cut portion of the baffles do not accommodate exchanger tubes) with intermediate supports is often the conventional design solution adopted. The same solution approach can also be adopted when low pressure drops are available at the shell side.

However, removing tubes from the windows ends up in a larger shell diameter with impact on the capital cost; furthermore, NTIW heat exchangers are usually prone to acoustic vibrations, frequently imposing the adoption of a not desired

detuning longitudinal plate to suppress the phenomenon (this is typical for shell side Gas service heat exchangers).

Thanks to the strong bundle consistency and the full confinement of all tubes at any grid, EMbaffle® makes use of the full tube layout ensuring the filling of the complete shell section with consequent reduction of the equipment diameter and/or improved heat exchanger performance, while suppressing the risk of acoustic vibrations due to his longitudinal flow design (**Figure 11**).

The unsupported tubes span of the conventional TEMA heat exchanger is governed by the balance between longitudinal and cross flows, limiting the minimum value that can be reached.

The natural frequency of the tubes depends on the tube diameter and thickness, tube material and unsupported tube span, according to the following formula [4]:

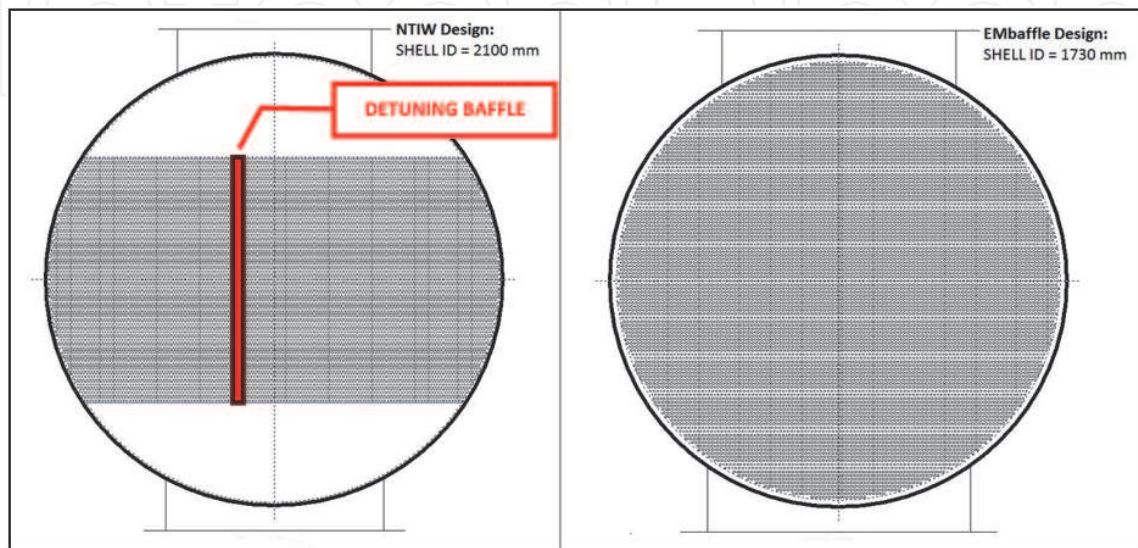
$$f_N = 0.04944C \left[ \frac{EIg_c}{W_e L^4} \right]^{0.5} \quad (14)$$

In an EMbaffle® exchanger, each tube is fully supported at every grid with a typical span ranging between 200 and 300 mm. This very close tube span significantly increases the natural frequency of the tubes, suppressing the risk of a frequency match and consequent vibration.

EMbaffle® is prone to good performances in condensing and boiling services too, e.g. cross-flow condensers, kettle-type reboilers, etc. where heat transfer coefficient is not substantially depending by the flow rate. Allowing the unrestricted shell-side flow thanks to the open structure, potential vibrations phenomena induced by phase transition are prevented, again allowing for a possible increase of shell side flow rate within the same exchanger constrains.

Concerns may apply to the shell-side fluid entrance region: here the flow suddenly changes from radial to longitudinal direction (vice versa at fluid exit), potentially stressing the tubes, specifically at bundle periphery as no annular space is left. Reducing the grids span in correspondence of the inlet/outlet nozzles, stronger tubes confinement can be configured as required to guarantee no vibrations.

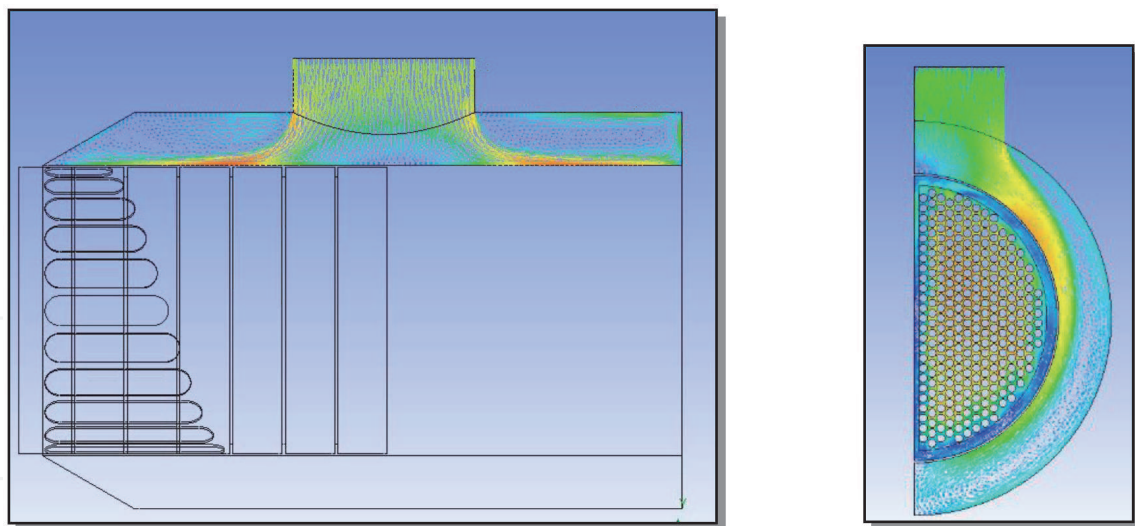
The use of an annular chamber to distribute the flow entrance in homogeneous way through the full bundle circumference, further to provide an impingement protection to the directly exposed tubes, ensures at the same time the development of the longitudinal flow through the complete shell section since the first baffle pass.



**Figure 11.**

*EMbaffle® exploiting of the full shell area in comparison to NTIW in gas applications.*





**Figure 12.**  
 Flow velocity distribution at EMbaffle® annular distributor – Top to bottom increased slot size case [5].

Dedicated CFD analysis has been performed to study different annular distributor configurations aimed to optimize the fluid-dynamics through the distributor and reduce the relevant correlated pressure drops (**Figure 12**).

Several geometries were modeled in order to analyze the flow distribution and the performances of each case. The flow velocity distribution at the inlet nozzle is showing a large area of the annular distributor to be interested by flow recirculation, addressing the flow to concentrate on lateral and bottom sides, trend accentuated by clearance reduction.

Decreased Top to Bottom exchanger slots size, contrary to what it could be expected, seems to address to a better uniform flow speed trend, but the dispersion of the flow rates at the entrance cannot be avoided. The average pressure drops are not significantly impacted by the shape of the cut and this supports the simplest and cheapest construction solution of the annular inner shell.

Thanks to all above provisions, no relative motion between tube and grid is permitted and, therefore, no wearing nor fretting is observed and reported after years of continuous operations in potential vibration services.

## 8. Fouling in EMbaffle®

EMbaffle technology was originally conceived to enhance the shell side heat transfer by reducing fouling in heat exchange specific applications in refineries and petrochemical plants. By creating a uniform flow in the bundle, dead zones are omitted. By supporting the tubes using expanded metal grid the boundary layer is continuously interrupted thanks to the local increased velocity. By this approach the balance between fouling disposition and removal results at a lower fouling layer than in conventional heat exchangers.

EMbaffle® technology has then been applied to a variety of processes, where complexity of fouling mechanisms does not allow a predictable behavior. Further to the preliminary experimental results coming from authoritative Bodies, the actual performances in fouling reduction are finding systematic confirmation by the outcomes from a number of units installed and operating for several years.

Detailed monitoring of fouling development and study of growing rate had been originally concentrated on crude oil application, where fouling is strongly impacting the thermal and hydraulic performances of the exchangers. The overall heat transfer coefficient over time of a segmental baffle type heat exchanger and the same

exchanger with EMbaffle replacement bundle, have been monitored, adjusting shell side velocity and pressure drops in order to reproduce close process parameters for the two measurement campaigns.

**Figures 13 and 14** report the plotted measurements and the related fitted distributions showing, over the constantly higher value, the quicker decay of the OHTC as a clear indication of the higher fouling grow rate of the segmental baffle exchanger.

From the measurements, the overall fouling factors can be extrapolated by using the following model:

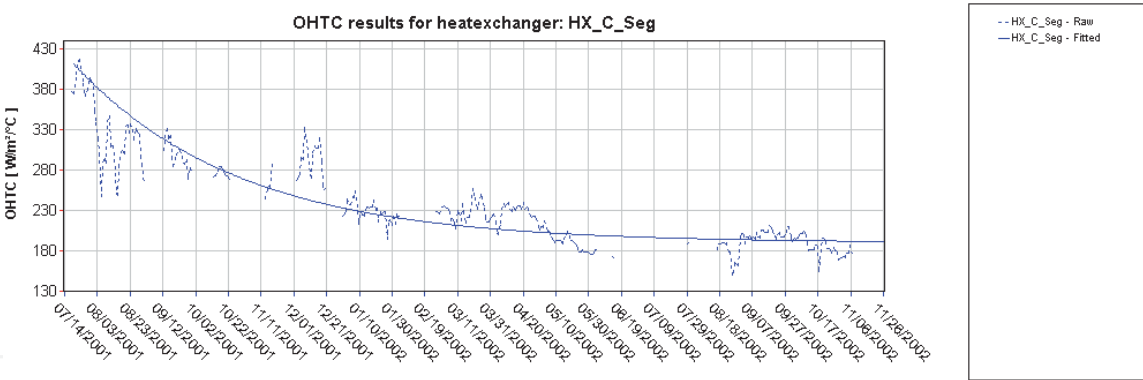
$$U(t) = U_{\infty} + (U_0 - U_{\infty})e^{-\left[\frac{t-t_0}{t_f}\right]} \tag{15}$$

The fouling rate is derived from:

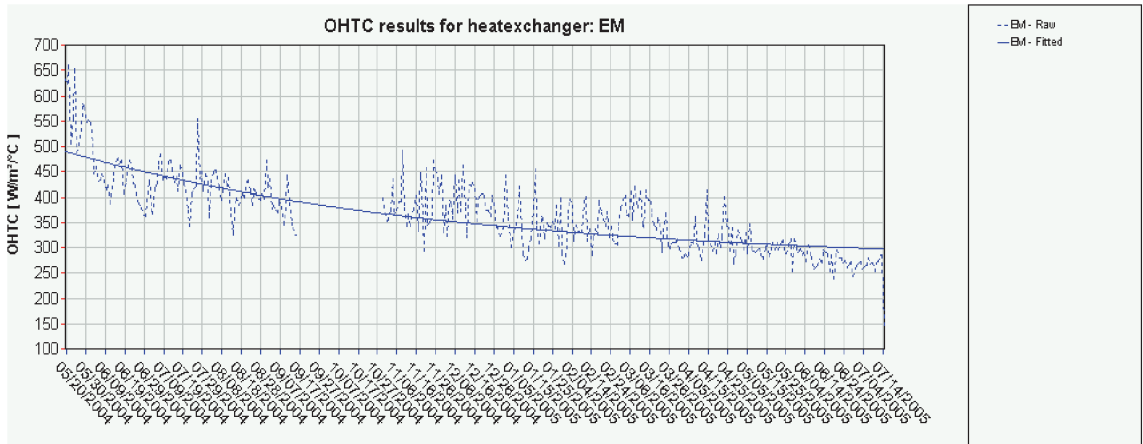
$$R(t) = \frac{1}{U(t)} - \frac{1}{U_0} = \frac{1}{U(t)} - \frac{1}{U_{\infty}} \ln \left( \frac{U_0}{U(t)} \right) - \frac{1}{U_{\infty}} \tag{16}$$

The large variation in the early phase of both the segmental as of the EMbaffle run is reflected in the first part of the fouling plot. On the longer run the fouling of the EMbaffle is increasing relatively slow (**Figure 15**).

In order to assign the right value to exchanger performances, the method of optimum clean out time is used, where the optimum run time of the heat exchanger is based on cost evaluation, i.e. cost of decreased performance versus the cost of a clean-out (**Figure 16**).

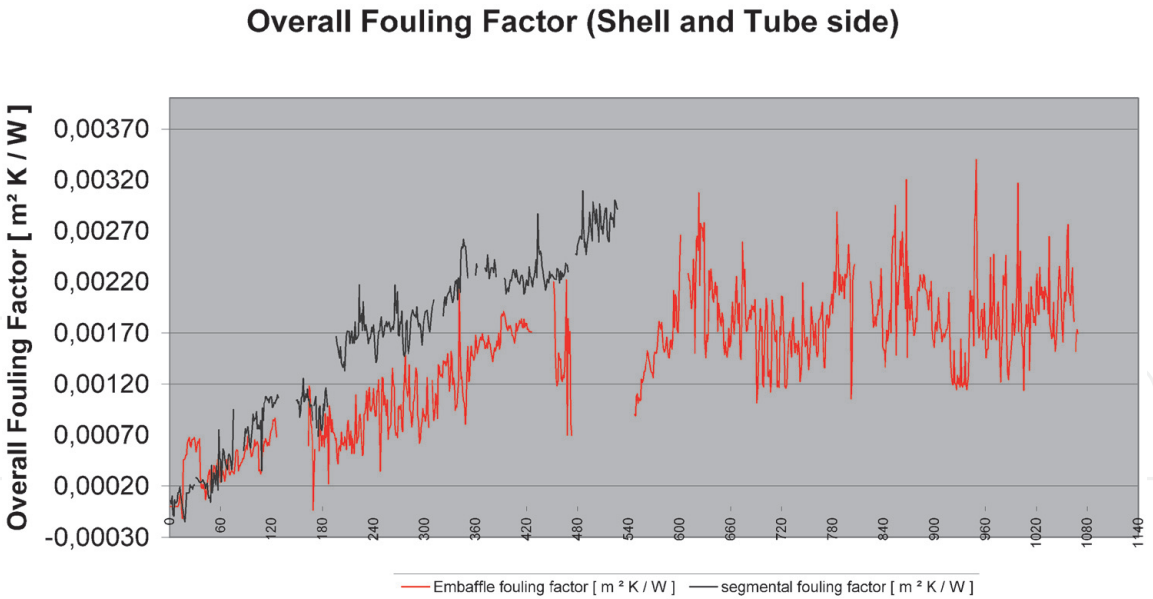


**Figure 13.**  
OHTC plot for the segmental baffle heat exchanger bundle [6].



**Figure 14.**  
OHTC plot for the EMbaffle® heat exchanger bundle [6].





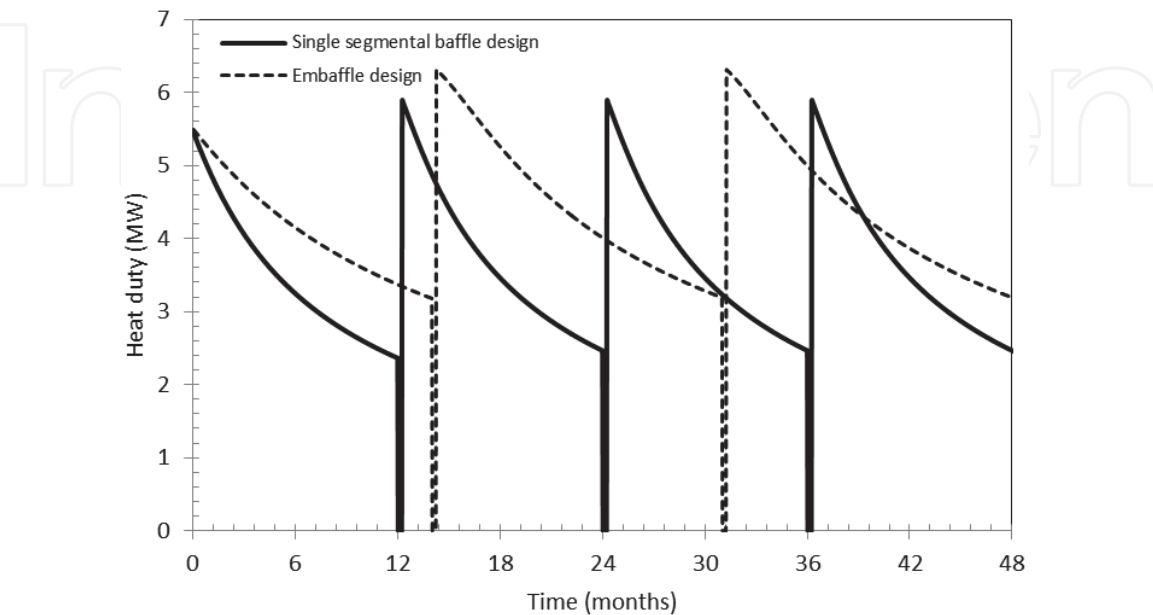
**Figure 15.**  
Overall fouling factors plot for the EMbaffle® and segmental baffle heat exchanger bundles [6].

The following equation is used to calculate the maximum economic benefit connected with the ratio between the run time and number of cleaning steps

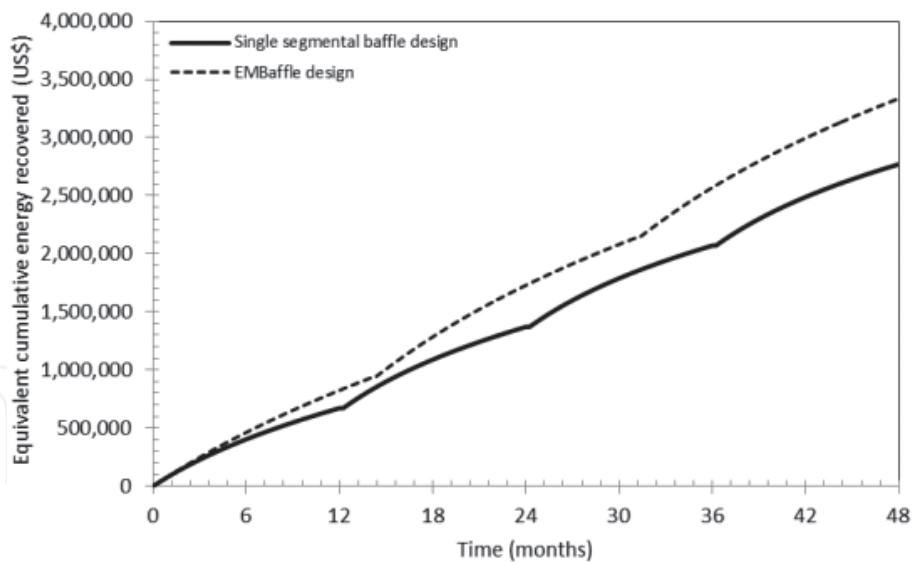
$$obj = C_E \int_0^{t_F} Q_E dt - N_c C_{cl} \tag{17}$$

Integral is calculated for the selected operating time.

In **Figure 17**, the economic benefit for a real case evaluated by comparing EMbaffle® performance with a parallel conventional unit on a base of 48 months operation is represented. Similar figures are of help in developing the best shut-down time at the light of the global plant configuration and performance.



**Figure 16.**  
Comparison of overall heat duty performance of the EMbaffle® (dashed line) and the segmental baffle (solid line) heat exchanger bundles [7].



**Figure 17.**  
Comparison of energy recovered (US\$) using EMBaffle® vs. segmental S&T in a crude preheating unit [7].

## 9. Advancement in EMBaffle® design

Finned tubes are widely used when equipment size and weight reduction play an important role. EMBaffle® developed a dedicated low fin “enhanced tube” helical profile (profile and finning process under patenting), conceived to fit longitudinal flow design aimed to increase the heat transfer based on two mechanisms: increase of active external tube surface and promotion of turbulence.

Two interesting cases of fin application have been addressed and will be presented in following paragraphs: gas cooling and oil to molten salts heat transfer in CSP applications.

### 9.1 Gas cooling

Several experimental measurements have been taken to check the EMBaffle® correlations precision in predicting the global heat transfer coefficient for gas cooling with plain tubes.

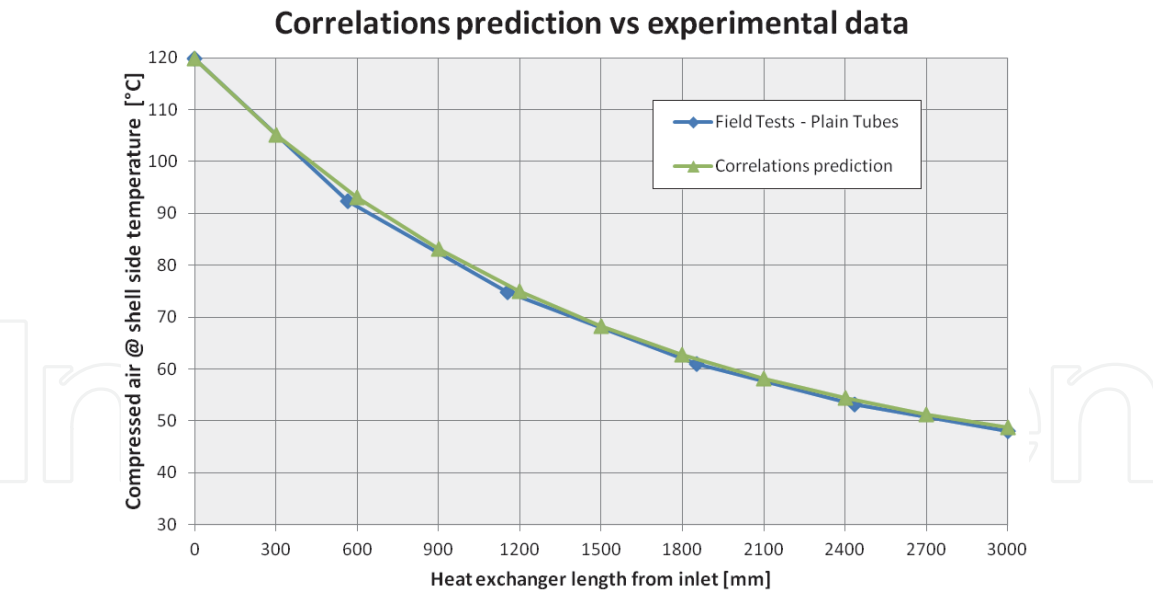
Water flow rate at the tube side has been sized to grant a ten times higher tube side coefficient with respect to the predictable shell side coefficient, so that changes in exchanger performance can be attributed to shell side heat transfer only.

In **Figure 18**, the correspondence between the correlations predictions and the experimental measures is reported: the theoretical curve fits perfectly with the measured temperature values, with predicted outlet temperatures differing less than 1%.

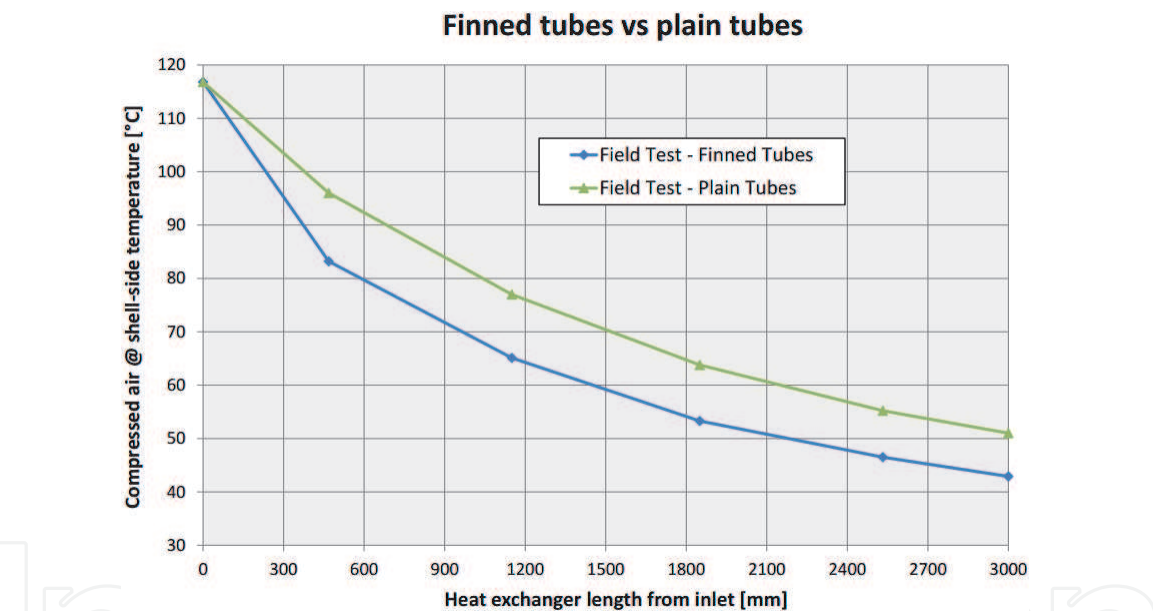
In **Figure 19**, experimental data to compare finned against plain tubes heat transfer performances are reported. In test case, the outlet air temperature reduces from 51 °C of the plain tubes case to 43 °C for the finned case, showing a significant improvement in heat transfer and global duty.

In the finned tubes test case, the air outlet temperature recorded for the plain tubes case has been reached at approx. 70% of the total tube length, showing a potential 30% tube length reduction.

These data allow to perform a design of the exchanger making use of the finned tube standard correlations to predict the temperature distribution profile and validate the global heat transferred.



**Figure 18.**  
*Experimental data for gas cooling application: Comparison between experimental data versus correlations prediction for plain tubes.*



**Figure 19.**  
*Experimental data for gas cooling application: Comparison between EMbaffle® proprietary low fin tubes design vs. plain tubes.*

Achieving a further significant reduction in the overall required tubes number and therefore of the equipment dimensions, the EMbaffle® finned tubes exchanger design is expected to prove successfully especially in offshore applications where compactness and lightness are of the essence.

More in general, the technology has a relevant impact on equipment costs containment for almost gas–gas and gas cooling processes and further tests shall be conducted to grant the continuous improvement of the performances in all gas applications.

Where tube-side can be the limiting factor, the use of enhanced features (inserts, inner surface micro-fins, etc.), to be applied in combination with shell side EMbaffle® grids, further to enhance the heat transfer, may also contribute to mitigate the fouling deposition on tubes side. The benefit of this combined

approach is therefore not only the increased heat recovery but also prolonged exchanger operating time through the reduction of fouling progress on both shell and tube sides.

For applications such as LNG vaporization, the combination between EMbaffle® and tube inserts is expected to be quite effective. On shell side, the EMbaffle® open structure will prevent the formation of dead zones guaranteeing, with the selection of proper grids span, the required tubes support, while inserts can mitigate the vaporizing issues at tubes inner surface by increasing the radial mixing.

## **9.2 Oil to molten salts in CSP**

A test campaign was carried out on the molten salt/thermal oil case to compare the performance between the use of bare tubes with finned tubes.

A heat exchanger based on EMbaffle technology with finned tubes was installed at the Concentrating Solar Platform centre in Almeria (Spain), the largest development and test center in Europe for molten salt application in CSP.

A test campaign was carried out using molten salt and thermal oil as media. A comparison has been made between the field test and an equivalent plain tube case calculated with the correlations.

Results shows an average 8% reduction in the overall heat transfer resistance. The consequent increase in performances is significant, even if not so high in absolute value: application of low fin tubes for this process shall be carefully evaluated.

## **9.3 Mechanical performance test**

The use of baffles made with “metal grid” instead of the more common “metal plate” suggests the need to verify their mechanical strength characteristics, especially in cases finned tubes are used and in the presence of processes with repeated thermal transients.

The different temperature distribution between the bundle support cage and the tubes during thermal transient brings to sliding of the tube inside the grid mesh, which could result mechanically harmful especially in the case of finned tubes.

In addition to the FEA for checking the static and dynamic stresses due to the accelerations induced on the tubes and on the grid, an experimental test was carried out to verify the consequences onto the grid subjected to the periodic longitudinal displacement of finned tubes in the most stringent conditions.

Two vertical baffles were positioned inside a horizontal cylindrical chamber and a finned tube was passed through them; weight and dimensions of tube were representative of the real exchanger conditions. A servomotor and a screw-nut type transmission were used to move the tube by operating a mechanical arm designed to transfer only a horizontal movement, minimizing any vertical thrust. The cylindrical chamber was filled with molten salts kept liquid with a system of heating resistances to maintain a temperature constantly above 380 °C.

Horizontal oscillatory movements (5 mm) equivalent to a 10-years working period of a exchanger with two daily transients were simulated to evaluate the effects of the relative wearing between the exchanger tube fin diameter and contact support points of the baffle grid diamond.

At the end of the test period the measurements of the outer diameter of the finned surface did show variation in height of the fins within 5% while no evidence of surface defect was registered on the contact profile of the grid mesh. Such a result is of extreme importance.

Other than confirming the good corrosion resistance of grid material in critical ambient conditions registered with standard corrosion tests formerly performed, it gives solid confirmation to the mechanical strength of the grid excluding at the same time any potential erosion defect on exchanger tubes surface in all EMbaffle technology application. This is of course further supported by the several years of service of the EMbaffle exchangers in different process services without reporting grids and/or tubes defect.

## 10. EMbaffle® design cases

Few design cases are presented in this paragraph as examples of how the application of EMbaffle® technology brings evident benefits.

### 10.1 DesignCase-1: overhead gas cooler

Two identical units (each one with two exchangers in parallel) have been installed in a platform. Using Sea water, the Overhead Gas Coolers were designed to cool high pressure acid natural gas from 110 °C down to 33 °C.

For this process the temperature approach between the fluids dictated a pure countercurrent arrangement, and the high water flow rate on the shell side did not allow the use of an F-shell TEMA type. Consequently a conventional segmental design in this case would have resulted in a much bigger and not-optimized geometry. A single pass for both tube and shell side exchanger would have been applied, with straight tubes and two tube sheets per exchanger.

The very limited shell-side available pressure drop in combination with the ability to accommodate large flow rates made this application very suitable for EMbaffle®, making possible the use of a F-shell TEMA type (**Figure 20**). The result was an optimized design, able to achieve a pure counter current arrangement with the application of U-tubes, which granted a single tube sheet per exchanger, reducing the weight. During the design stage a higher OHTC has been also exploited, with a consequent reduction in required heat transfer surface. Given the off-shore application, the reduction in size and weight obtained for the exchangers was particularly beneficial.



**Figure 20.**  
*EMbaffle® overhead gas cooler.*



Overhead Gas Cooler	Conventional design	EMbaffle® design	Units
TEMA type	BEM [8]	BFU [8]	—
Number of equipments	2 in parallel	2 in parallel	—
Shell ID	1780	1610	mm
Tube length	10000	7315	mm
Baffle arrangement	NTIW	EMbaffle	—
Installed area	3530	2609	m <sup>2</sup>
SS pressure drop	0.3	0.3	bar
Duty	50800	50800	kW
Duty / Installed area	14.4	19.5	kW/m <sup>2</sup>
Weight	118.2	73.3	tons

**Table 1.**  
Design comparison between EMbaffle® and conventional S&T for an overhead gas cooler.

**Table 1** reports a comparison between a conventional S&T exchanger and the EMbaffle® type exchanger for this case.

The improvement described above are clearly depicted: EMbaffle® design is able to exploit the same duty of the conventional case with a 25% reduction of the installed surface area, providing the same shell-side pressure drop.

10.2 DesignCase-2: cycle gas cooler

**Figure 21** depicts a Cycle Gas Cooler, installed in a large chemical plant in North America. The function of the exchanger is to use Cycle water to cool the hot gas (placed at the tube side) from 100 °C to 40 °C.

Water flow rate was huge (more than 4000 tons per hour) and simply could not be accommodated in a single conventional baffle equipped heat exchanger. Two conventional units operating in parallel would have been necessary in order to guarantee a vibration-free design.

From the pressure drops point of view also, the single conventional unit would not have been an option resulting in pressure drops far above the allowable ones. In **Table 2** the straight comparison between the two designs is reported.



**Figure 21.**  
EMbaffle® cycle gas cooler.

Cycle Gas Cooler	Conventional design	EMbaffle® design	Units
TEMA type	BEM [8]	BEM [8]	—
Number of equipments	2 in parallel	1	—
Shell ID	1740	1800	mm
Tube length	9760	11200	mm
Baffle arrangement	NTIW	EMbaffle	—
Installed area	3173	2335	m <sup>2</sup>
SS pressure drop	0.7	0.7	bar
Duty	69400	69400	kW
Duty / Installed area	21.9	29.7	kW/m <sup>2</sup>
Weight	126.6	79.2	tons

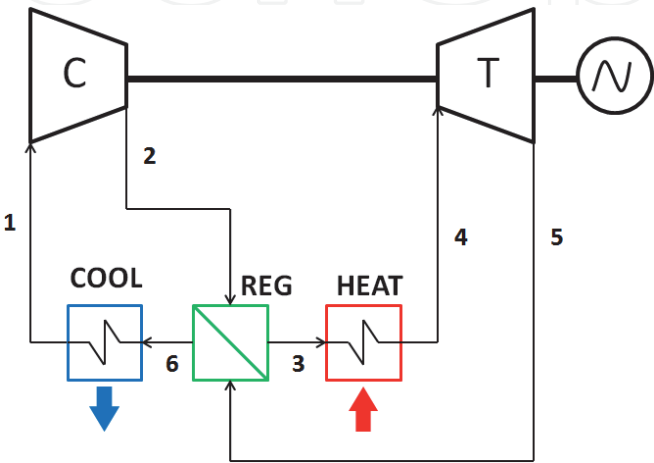
**Table 2.**  
Design comparison between EMbaffle® and a conventional S&T for a cycle gas cooler.

10.3 CO<sub>2</sub>-based power generation plants

Challenge to avoid/reduce emission of carbon dioxide in power generation industry has been addressed in many ways, being its use as working fluid in power production plants one of the most promising.

**Figure 22** illustrates the supercritical Brayton-cycle and the relevant heat transfer units. High pressures involved and typically large gas flow rates may suggest adoption of S&T heat exchanger, being EMbaffle® one of the promising layout in consideration of the benefits envisaged in Gas Treatment and Purification chapter. Technology usually proves either as Gas Regenerator (path 2–3, 5–6), even in consideration of typical low temperature approach and pure countercurrent layout, and Gas Cooler (path 6–1); whereas large compression factors have to be achieved multistage Gas Intercoolers (not represented in the figure, along the path 1–2) are adopted. Depending on the application, Gas Heater design may rely on S&T layout or onto other piece of equipment (WHRU as example) depending onto the heat source medium.

**Table 3** reports a comparison between technologies for Gas Regenerator.



**Figure 22.**  
Basic regenerative Brayton cycle for CO<sub>2</sub>-based power production plant.

CO <sub>2</sub> Regenerator	Conventional design	EMbaffle® design	Units
TEMA type	BEM [8]	BEM [8]	—
Number of equipments	1	1	—
Shell ID	2040	1600	mm
Tube length	23500	23500	mm
Baffle arrangement	NTIW	EMbaffle®	—
Installed area	5279	4324	m <sup>2</sup>
SS pressure drop	1.5	1.5	bar
Duty	48190	48190	kW
Duty / Installed area	9,1	11,1	kW/m <sup>2</sup>
Weight	207,1	124,4	tons

Table 3.  
Design comparison between EMbaffle® and a conventional S&T for a CO<sub>2</sub> regenerator.

Nomenclature

Symbol	Description	Units
$A$	Heat transfer area	[m <sup>2</sup> ]
$A_B$	Baffle flow area	[m <sup>2</sup> ]
$A_{EM}$	EMbaffle grid projected area	[m <sup>2</sup> ]
$A_R$	Ring area	[m <sup>2</sup> ]
$A_s$	Shell flow area	[m <sup>2</sup> ]
$C$	Tube span constant	[-]
$C_{cl}$	Cost of cleaning	[US\$/unit]
$C_E$	Cost of energy	[US\$/J]
$C_L$	Laminar heat transfer geometry function	[-]
$C_T$	Turbulent heat transfer geometry function	[-]
$D_h$	Characteristic diameter for Nu and Re <sub>h</sub>	[m]
$D_i$	Tube internal diameter	[m]
$D_P$	Characteristic diameter for Re <sub>P</sub>	[m]
$D_S$	Shell inner diameter	[m]
$D_o$	Tube outer diameter	[m]
$E$	Modulus of elasticity of tube material	[Pa]
$f_F$	Fanning friction factor	[-]
$F_t$	Correction factor depending on exchanger arrangement and approaches temperatures	[-]
$f_N$	Natural frequency	[Hz]
$g_c$	Conversion constant	[-]
$h$	Film transfer coefficient	[W/m <sup>2</sup> K]
$h_i$	Inner heat transfer coefficient	[W/m <sup>2</sup> K]
$h_o$	Outer heat transfer coefficient	[W/m <sup>2</sup> K]
$I$	Moment of inertia of tube	[m <sup>4</sup> ]
$K_b$	Hydraulic loss coefficient of baffle	[-]
$L$	Unsupported tube span	[m]
$LWD$	Long way of diamond	[m]
$N_B$	Number of baffles	[-]
$N_C$	Number of cleaning events	[-]
$N_T$	Number of tubes	[-]

$Nu$	Nusselt number	[-]
$obj$	Objective function value	[US\$]
$Pr$	Prandtl number	[-]
$Q$	Heat transferred	[W]
$Q_E$	Total accumulate heat exchanged	[MW]
$R(t)$	Actual fouling resistance at a specified time	[m <sup>2</sup> K/W]
$Re_h$	Heat transfer Reynolds number	[-]
$Re_p$	Longitudinal flow Reynolds number	[-]
$SWD$	Short way of diamond	[m]
$t$	Time	[s]
$t_0$	Time the heat exchanger has just been cleaned out or installed	[s]
$t_f$	Operating campaign time	[s]
$V_B$	Baffle velocity calculated from $A_b$	[m/s]
$V_s$	Shell-side velocity calculated form $A_s$	[m/s]
$U$	Overall heat transfer coefficient	[W/m <sup>2</sup> K]
$U_\infty$	Overall heat transfer coefficient at equilibrium	[W/m <sup>2</sup> K]
$U_0$	Overall heat transfer coefficient, initial	[W/m <sup>2</sup> K]
$W_e$	Effective mass per unit length	[kg/m]
$\Delta T_{lm}$	Logarithmic mean temperature difference	[K]
$\Delta T_m$	Average temperature driving force	[K]
$\Delta P$	Pressure Drop	[Pa]
$\Delta P_B$	Baffle flow pressure drop	[Pa]
$\Delta P_L$	Longitudinal flow pressure drop	[Pa]
$\lambda$	Wall thermal conductivity	[W/m K]
$\mu_b$	Bulk viscosity	[Pas]
$\mu_w$	Wall viscosity	[Pas]
$\rho$	Mass density	[kg/m <sup>3</sup> ]

### Author details

Marco Rottoli\*, Daniele Agazzi, Marcello Garavaglia and Fabio Grisoni  
EMbaffle® Technology @ Brembana&Rolle SpA, Valbrembo, Italy

\*Address all correspondence to: [mrottoli@brembanarolle.com](mailto:mrottoli@brembanarolle.com)

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