The final publication is available at

https://doi.org/10.4271/2021-01-0388

Additional Information

On the design of heat exchangers for altitude simulators

Alberto Broatch, Pablo Olmeda, Jorge Garcia-Tíscar, and Ferran Roig

CMT-Motores Térmicos. Universitat Politècnica de València. Camino de Vera s/n. 46022 València (Spain)

Abstract

Altitude simulators for internal combustion engines are broadly used in order to simulate different atmospheric pressure and temperatures on a test bench. One of the main problems of these devices is their outlet temperature and in order to control it, at least one heat exchanger is needed. A methodology to define, select and analyses the best heat exchanger that fulfill the requirements is presented. The methodology combines CFD and 0D models with experimental test. The combination of these tools allows to adjust the 0D model that later it is used to perform parametric analysis that will help to select the best geometrical combinations taking into account heat transfer and pressure losses.

Introduction

In the new homologation process for automotive engines [1], real driving tests must be performed up to 1300 m over sea level. To perform the test at those altitude conditions two main solutions have been employed: the use of hypobaric chambers and the use of altitude simulators [2-4]. In these devices, altitude is only generated inside the engine intake and exhausts lines while engine is kept at room pressure. No-significant differences were obtained in the analysis of main engine parameters performed in [5] when using the two altitude simulating technologies.

In the altitude simulators (AS) a cooler is placed upstream the mechanical compressor. The main function of this cooler is to ensure the safe operation of the compressor, but it also reduces the compressor power consumption [6] since the lower the temperature at compressor inlet the best. When the requirements of altitude and engine inlet mass flows are very high, the demand behavior of the cooler will be higher.

One of the main problems of these coolers is the fact that to avoid high pressure losses on gas side, this fluid passes through the tubes. In the coolant media usually the pressure loss requirements are lower since the supply of the coolant is usually external to the AS equipment. The arrangement of the fluids in the heat exchanger causes a decrease in the thermal behavior of the heat exchanger

There are in the open literature a huge work about HEX models for rating and design [7-11], but from authors' knowledge there is neither a methodology that combines the use of three tools: 0D models, CFD models and experiments on these types of exchangers neither parametric studies reflecting the effect of the main geometric characteristics on HEX performance, not only from thermal point of view but also from pressure losses analysis.

The main objective of this work is to develop a 0D tube and shell HEX model adjusted with experimental data who can predict accurately the performance of this HEX. The model will be used to perform parametric studies in order to get different design combinations that fit with the design requirements. These studies will serve as a first step for the manufacturing of new HEX.

Materials and Methods

The methodology followed in this work is summarized in Figure 1, and it is based on both experimental measurements and the use of calculation models. The methodology can be summarized as follows:

- The starting point was a completely known geometry of the heat exchanger, that is, the planes of the complete heat exchanger from the internal geometry of the tubes, their spatial arrangement within the shell, the number of baffles on the shell side, the exact geometry of these baffles, etc. This exchanger is called the base heat exchanger
- Two different calculation models are built using the geometric characteristics of the base heat exchanger. On the one hand a CFD model with this complete geometry and, in parallel, a 0D model following the method developed by Bell-Delaware.
- Besides, experimental measurements were performed with the base heat exchanger that will be installed in a specific test room. They will be used for validation of the 0D model. The adjustment consisted on fit some model parameters until the results obtained with the model were similar to those obtained experimentally.
- The CFD model, due to its greater complexity and therefore to its greater requirements of computational times will be validated by comparing the obtained results with the results obtained with the validated 0D model. The CFD model was validated in this way since the operating conditions modeled were far away from those experimentally measured. So, once the 0D model was validated with the experimental measurements it will serve as an "experimental test bench" to check the results of the CFD model. Secondly, the CFD model was used to analyze in detail the internal flow through the exchanger to propose possible improvements to the HEX design.
- Finally, the 0D model (once validated with the experimental measurements) will be used to perform parametric studies to determine both the performance of the base heat exchanger under different boundary conditions and to define the basic geometry of the heat exchanger necessary to achieve certain operating conditions. In other words, on the one hand, the first studies will be carried out to determine both the thermal behavior (fluid outlet temperatures) and to estimate the pressure drop on both sides of the base heat exchanger. On the other hand, the basic geometries (diameter and length of tubes and casing diameter) will be determined so that thermal conditions can be achieved at the outlet knowing the boundary conditions at the exchanger inlet.

Experimental campaign

The experiment designed for performing the experimental campaign which it will be used for validating the numerical models consists of a shell and tube HEX which main characteristics are presented in Table 1.

The base HEX was installed on a test bench (as Figure 2 shows). On the top side of Figure 2, the location of the main measurements' devices are indicated. These are: gas inlet and outlet temperatures and pressures, coolant inlet and outlet temperatures and pressures, gas and coolant mass flows. As observed, the pressures were measured in piezo-metric way while gas temperatures were acquired by 4 different thermocouples in order to minimize the errors [12-13]. On the bottom side of Figure 2 the final configuration, with the HEX externally insulated to diminish the external heat losses, is presented.

Table 1. HEX main characteristics.

Tub length	826 mm
Shell diameter	264 mm
Tube inside diameter	10 mm
Tube outside diameter	12 mm
Tube number	237
Shell side fluid	Coolant
Tube side fluid	Gas
Tube material	Stainless Steel s
Baffles	Single-segmental baffles

The experimental measurements were performed using a constant supply on the coolant side: inlet temperature of 24ºC and inlet coolant mass flow of 15 m3/h since it is the most representative condition of the HEX in real conditions. In the tube side two different inlet gas temperatures were tested. First different tests varying gas mass flow at maximum allowable test bench temperature and, second, tests at the temperature reached by the system. Due to those limitations the tested conditions were:

- Seven experimental test with gas inlet temperature (GIT) of 190ºC with gas mass flows ranging from 100 kg/h to 700 kg/h with a step of 100 kg/h.
- Eleven test with GIT of 120ºC being, in this case, the measured gas mass flows were varied from 100 kg/h to 1100 kg/h

The main objective with the variation on gas mass flows is to measure the behavior of the HEX in a wide range of conditions: starting from laminar flow, passing through transitional flow and reaching turbulent flow in tube side.
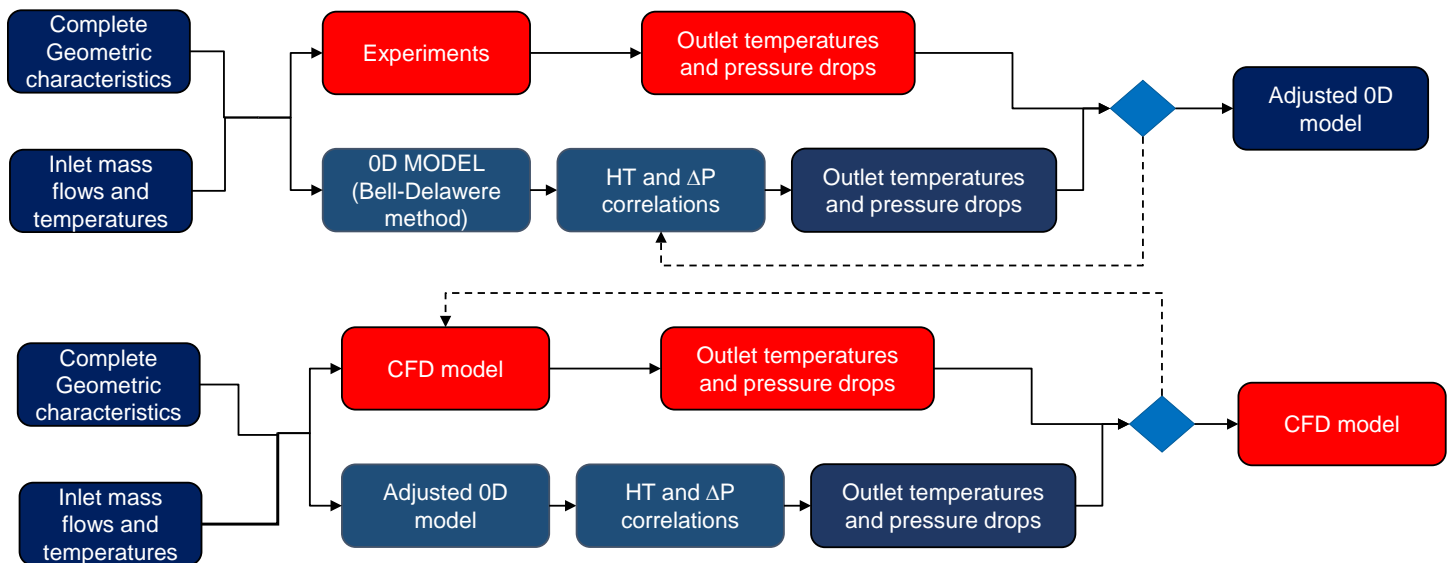

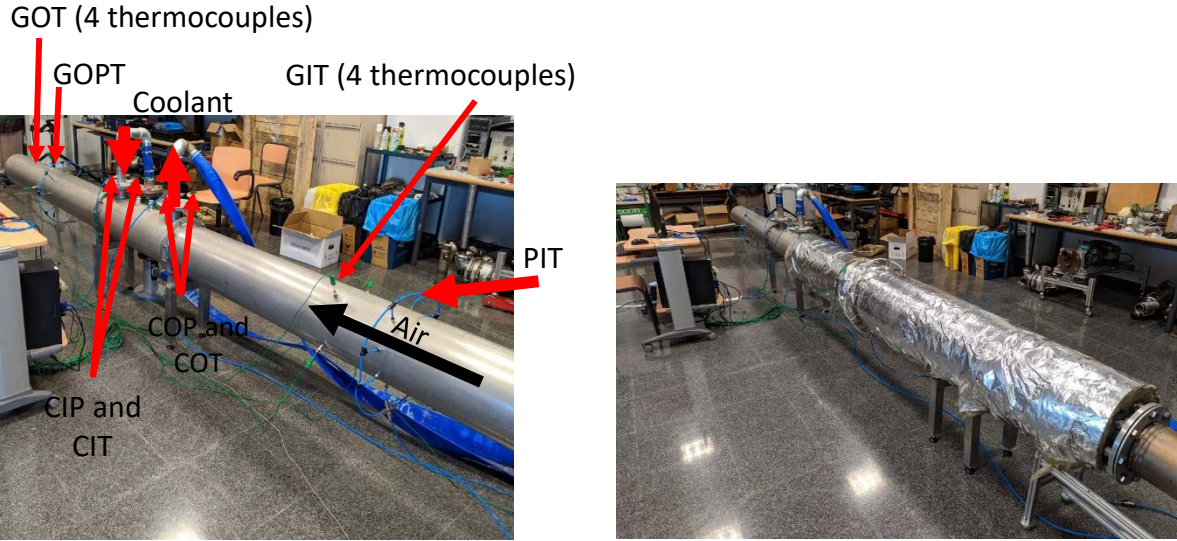Figure 1. Methodology followed on the work

Figure 2. HEX installed on the test bed

Numerical Models

In this section a brief description of the two models used for evaluating the performance of the HEX are presented. On one hand the 0D model which will allow to perform fast and reliable parametric studies, on the other hand, a CFD model who will allow to understand the main flow patterns inside the HEX.

The main important part for rating or design a heat exchanger rely on both the heat transfer and pressure drop calculations in both sides of the HEX. The heat transfer rate equation for the heat exchanger is shown in Eq. 1

$$Q = U_o A_0 LMTD \qquad (1)$$

Where U_o , A_o and $LMTD$ refers to the heat exchanger thermal coefficient referred to the tube outside diameter, the tube outside area and the log-mean temperature difference respectively. The heat exchanger thermal coefficient can be calculated as Eq. 2 shows:

$$\frac{1}{U_o} = \frac{1}{h_o} + R_{f,o} + \frac{d_0 ln(d_0/d_i)}{2k_w} + R_{f,i} + \frac{1}{h_i}\frac{d_o}{d_i} \qquad (2)$$

Where h_o and h_i are the convective heat transfer coefficients on tube outside (i.e. shell side) and tub inside; $R_{f,o}$ and $R_{f,i}$ refer to fouling outside and inside the tubes; the term in the middle of the right-hand of Eq. 2 refers to heat conduction through the tube wall and d_o and d_i are the outside and inside tube diameters. Since the HEX used in this work is completely new and clean, both fouling factors has been considered as null values.

0D Model

In this work, the 0D model for evaluating the performance of the shell and tube heat exchanger is based on the Bell-Delaware method [14-15] where:

Tube-side calculations are as follows*: The heat transfer coefficient (h_i) is computed using available correlations for internal forced convection depending on Reynolds number [16-17]. Table 2 shows the used correlations.

Table 2. Correlations for internal forced convection.

$Re \leq 2100$	$Nu = 3.66 + \dfrac{0.688 Re \dfrac{Pr \cdot d_i}{L}}{2\rho_i A_{o,t}^2}$
$2100 \leq Re \leq 4000$	$Nu = 0.023 Re^{0.8} Pr^{1/3} \left(\dfrac{\mu}{\mu_w}\right)^{0.14}$
$4000 < Re < 10^7$	$Nu = \dfrac{\left(\dfrac{f}{2}\right) Re Pr}{C + \left(\dfrac{f}{2}\right)^{1/2} (Pr^{2/3} - 1)} \left(\dfrac{\mu}{\mu_w}\right)^{0.25}$

For pressure drop calculations, Eq. 3 is used where friction (represented by the term $\frac{4fL}{d_i}\rho_i\left(\frac{1}{\rho_m}\right)$), entrance $(1-\sigma^2+K_c)$, momentum $\left(2\left(\frac{\rho_i}{\rho_o}-1\right)\right)$ and exit $\left((1-\sigma^2+K_e)\frac{\rho_i}{\rho_o}\right)$ effects are taken into account.

$$\Delta p = \frac{\dot{m}^2}{2\rho_i A_{o,t}^2}\begin{pmatrix}\frac{4fL}{d_i}\rho_i\left(\frac{1}{\rho_m}\right)+(1-\sigma^2+K_c)+2\left(\frac{\rho_i}{\rho_o}-1\right)-\\-(1-\sigma^2+K_e)\frac{\rho_i}{\rho_o}\end{pmatrix} \quad (3)$$

Where the friction factor, f, is calculated using Bhatti-Sha [18] correlation as Eq. (4) shows:

$$f = A + BRe^{-1/m} \quad (4)$$

The constants A, B and m depends on Reynolds number as Table shows:

Table 3. Values of the constants of Eq. (4).

	A	B	m
Re<2100	0	16	1
2100≤Re≤4000	0.0054	$2.3 \cdot 10^{-8}$	-2/3
4000<Re<$\cdot 10^7$	0.00128	$0.1143 \cdot 10^{-8}$	3.2154

The shell-side is divided in different flow streams. In each of these streams, the flow fraction is calculated bearing in mind both the flow areas and the flow resistances. The shell side heat transfer coefficient is calculated by correcting an ideal heat transfer coefficient for a pure crossflow stream (h_{id}) by using correction factors that consider the presence of different streams on the shell side. The shell-side heat transfer coefficient h_o is finally given by

$$h_0 = J_c \cdot J_l \cdot J_b \cdot J_r \cdot J_s \cdot h_{id} \quad (5)$$

where the correction factors are as follows:

- The correction factor for baffle cut and spacing:

$$J_c = 0.55 + 0.72 \cdot F_c \quad (6)$$

Where the fraction of tubes in pure cross flow between the baffle cut tips (F_c) is calculated as $F_c = 1 - 2 \cdot F_w$, where the fraction of tubes in baffle window is $F_w = \frac{N_{tw}}{N_t}$, being N_t the total number of tubes and N_{tw} the number of tubes in the window

- The correction factor for baffle leakage area is:

$$J_l = 0.44(1 - r_s) + [1 - 0.44(1 - r_s)]e^{-2.2r_{lm}} \quad (7)$$

Where:

$$r_s = \frac{S_{sb}}{S_{sb}+S_{tb}} \quad (8)$$

$$r_{lm} = \frac{S_{sb}+S_{tb}}{S_m} \quad (9)$$

Being the shell—to-baffle area $S_{sb} = \pi D_s \frac{L_{sb}}{2}\left(\frac{2\pi-\theta_{ds}}{2\pi}\right)$ and the tube-to-baffle area: $S_{tb} = \frac{\pi}{4}[(2+L_{tb})^2 - d^2]N_t(1-F_w))$

- The correction factor for bundle by-pass flow is:

$$J_b = exp\{-C_{bh}F_{sbp}[1-(2r_s)^{1/3}]\} \quad (10)$$

Where $F_{sbp} = \frac{S_b}{S_m}$ is the ratio of the bypass area to the overall cross flow area and $r_{ss} = \frac{N_{ss}}{N_{scc}}$ is the ratio of the number of sealing strips to the number of rows crossed between baffle section. The number of tube rows crossed between baffle tips in one baffle section is calculated as $N_{tcc} = \frac{D_s}{L_{pp}}\left(1-\frac{2B_c}{100}\right)$, the bundle-to-shell bypass area is $S_b = L_{bc}(D_s - D_{otl} + L_{pi})$

- The correction factor for adverse temperature gradient in laminar flow is:

$$J_r = \frac{1.51}{N_c^{0.18}}, Re_s \leq 20 \qquad (11)$$

$$J_r = \frac{1.51}{N_c^{0.18}} + \left(\frac{20-Re_s}{80}\right)\left(\frac{1.51}{N_c^{0.18}} - 1\right), Re_s > 20 \qquad (12)$$

Being the total number of tube rows crossed in the entire exchanger $N_c = (N_{tcc} + N_{tcw})(N_b + 1)$, where the effective number of tube rows crossed in the baffle window is calculated as:

$$N_{tcw} = \frac{0.8}{L_{pp}}\left(\frac{D_s B_c}{100} - \frac{D_s - D_{ctl}}{100}\right) \qquad (13)$$

-   The correction factor for unequal baffle spacing at inlet and/or outlet:

$$J_s = \frac{(N_b-1)+(L_i^*)^{1-n}+(L_o^*)^{1-n}}{(N_b-1)+(L_i^*) \ +(L_o^*)} \qquad (14)$$

Where $L_i^* = \frac{L_{bi}}{L_{bc}}, L_o^* = \frac{L_{bo}}{L_{bc}}, n = 0.6$ for turbulent flow

The heat transfer coefficient for a pure crossflow stream (h_{id}) is calculated by using Zukauskas correlation [15-16]

The calculation of the pressure drop on the shell side is a bit more complicated due to the presence of the different streams but in anyway the methodology is similar to the one for the heat transfer coefficient, i.e. the pressure drop is evaluated first for an ideal crossflow section and an ideal window section and then correction factors for the different streams are applied as Eq. (3) shows:

$$\Delta p_s = \Delta p_{cr} + \Delta p_w + \Delta p_{i-o} + K_{adj} \cdot \dot{m}^2 \qquad (15)$$

Where the subscripts mean: cr refers to central section, w for the window area and $i - o$ for the inlet and outlet section; K_{adj} and $\dot{m}^2$ refer to the friction effect and the mass flow

CFD Model

The employed CFD model in this study is based on a Conjugate Heat Transfer (CHT) methodology and it will be used for understanding the flow patterns in both sides: the shell and the tube in order to search for a possible improvements on heat exchanger design in two ways: better heat flow between the two fluids and a lower pressure drop on both sides.

Figure 3 shows a scheme of the geometry of the HEX studied in this work while its main characteristics are shown in Table 1. In Figure 3 only a couple of tubes, the 3 baffles are shown in order to clarify the used HEX. The solver used was StarCCM+ 15.02.007 Steady S-RANS, and the k-Omega turbulence model was employed. The used grid considered a 3-D unstructured polyhedral mesh scheme. The cell size was ranged between 65 μm (near the wall of the tubes to capture the laminar viscous sub-layer) and 3 mm (far from tube wall). The total number of cells was approximately 48 million
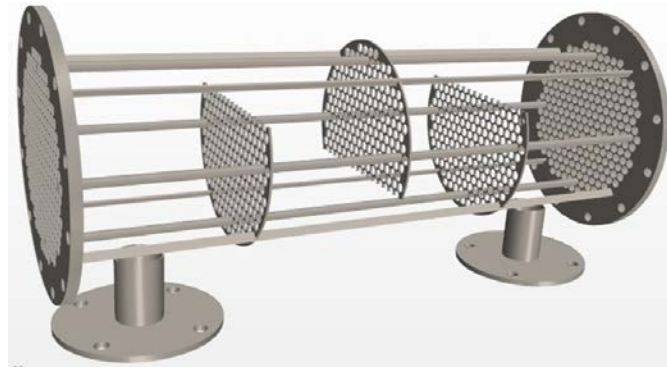

Figure 3. HEX internal geometry

Models Validation

The 0D developed model for this study has been fitted by using the available experimental results. The applied boundary conditions in all the cases were: coolant and gas inlet mass flows and temperatures while the most important obtained results were outlet temperatures and pressure drop of both fluids. The comparison between the experimental results and model results is presented in Figure 4. Gas outlet temperature estimated by the model ($T_{g,out\ model}$) fits quite precisely with those obtained experimentally ($T_{g,out\ measured}$) for the two different experimental gas inlet temperatures (120 and 190ºC). In the case of the coolant water, results are also satisfactory bearing in mind that the temperature drop in the coolant side is quite small. The results obtained with the model for both the temperature drop and the pressure drop on the tube side (gas side) is well captured as Figure 5 shows.
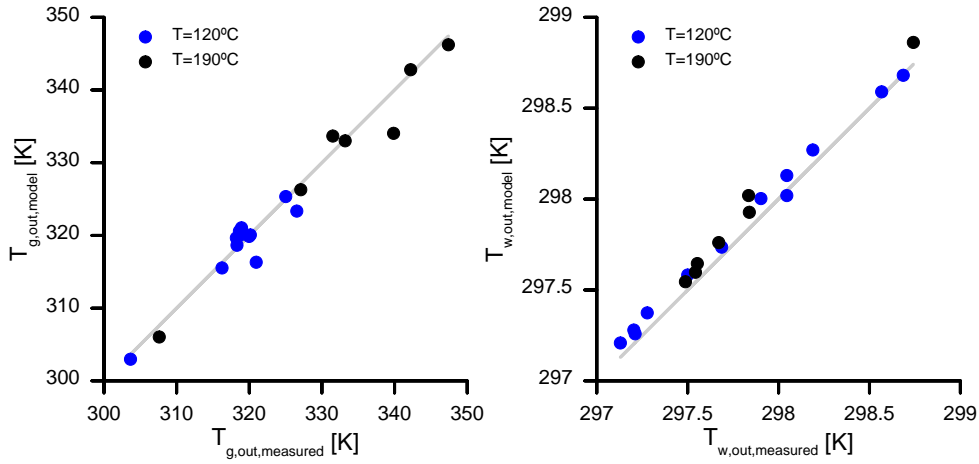
Figure 4. Comparison between 0D model and experimental results for gas and coolant outlet temperatures.

Once the 0D model has been fitted, it can be used, on one hand, to validate the obtained global results with the CFD model and, on the other hand, to perform parametric studies. The fitted CFD will be used to evaluate the internal behavior of the HEX, while the parametric study will allow the selection of the HEX for this application

The modelled conditions for validated de CFD model against the obtained results with the validated 0D model are presented on Table 4. Those conditions have been selected bearing in mind the most demanding running conditions that the HEX will meet in real application, i.e the maximum expected gas mass flow and its maximum temperature at HEX inlet. Starting from this demanding condition other studies have been performed in order to take into account the effect of the shell side fluid (coolant water) on the performance of the HEX and the effect of gas mass flow.

Table 4. Tested conditions for CFD Validation.

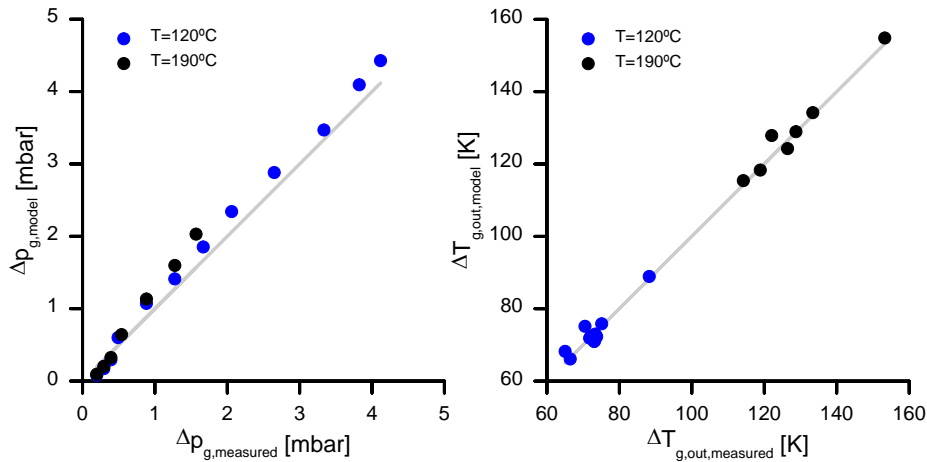| Gas mass flows [kg/h]	500 -1050 - 1550
Coolant mass flows [m3/h]	2.5 – 5.0 -7.5 -10.0-12.5 – 15.0
Inlet gas temperature [ºC]	450
Inlet coolant temperature [ºC]	10

Figure 5. Comparison between 0D model and experimental results for gas temperature and pressure drop

Figure 6 shows the heat estimated by the two models under the 18 simulated operating conditions. As observed in Figure 6 the 0D model predicts slightly higher heat exchange at high gas mass flows but the differences are almost negligible. Likewise, the effect of the coolant mass flows on the behavior of HEX is lower the effect of the gas mass flows at least in the studied ranges, since the variation of the predicted heat exchanged at the six different coolant mass flows keeping constant the gas mass flow is lower than the effect on this heat when changing the gas mass flow

Figure 7 shows the estimated outlet temperature of both gas and coolant and the pressure drops of both fluids as they pass through the HEX for the 18 simulations carried out to validate the CFD model. Figure 7 also shows that in the case of the coolant the two models give almost identical results both for the outlet temperature and for the pressure drop, with small differences on the gas side,
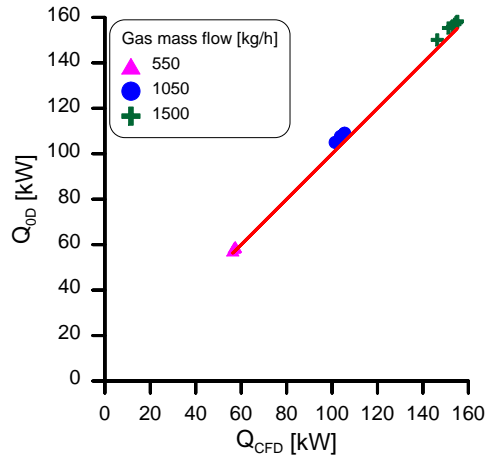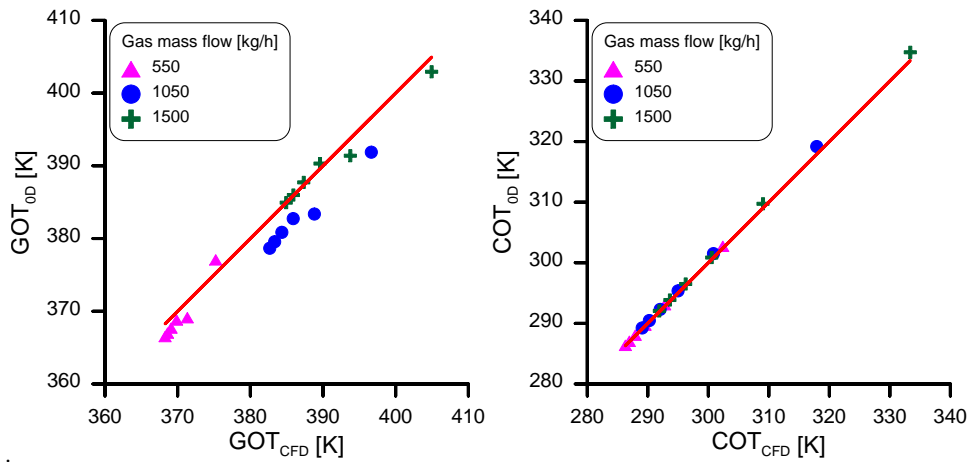
Figure 6. Comparison between 0D model and CFD model results for Heat exchanged on the base HEX

It is also noted that the change in gas mass flows does not affect the pressure drop on the refrigerant side, however, the change in refrigerant mass flow slightly affects the pressure drops observed on the gas side. This is due to the change in gas temperature with the change in coolant flow rates, which causes a change in gas viscosity and thus a change in pressure drop

The results obtained with the two models indicate that the overall behavior of the base exchanger has been captured satisfactorily and therefore the 0D model can be used to carry out parametric studies. Likewise, since the global results of the CFD model are almost identical to those obtained with the 0D model, it can be interpreted that the internal behavior predicted by the CFD model is well captured

The mentioned small influence on the gas outlet temperature of the change in the coolant mass flow rate can be explained by analyzing the figure 8. Figure 8 shows, on the one hand, the average gas, coolant and tube wall temperatures estimated by the 0D model for the 18 simulated cases. On the other hand, it shows the ratio of the 3 thermal resistances (inside the tube, outside the tube and wall) against the total resistance. In the first place, it is observed that the average temperature of the gas, in all cases, is around 550 K while the temperature of the wall and the reflector are around 300 K, being slightly higher than that of the wall. Secondly, it can be seen that the resistance on the gas side is approximately the total resistance. Due to this fact, the predominant resistance is that of the gas side (inside the tube) and, therefore, the change in any parameter that does not modify this resistance should not have an effect on the thermal behavior of the HEX.
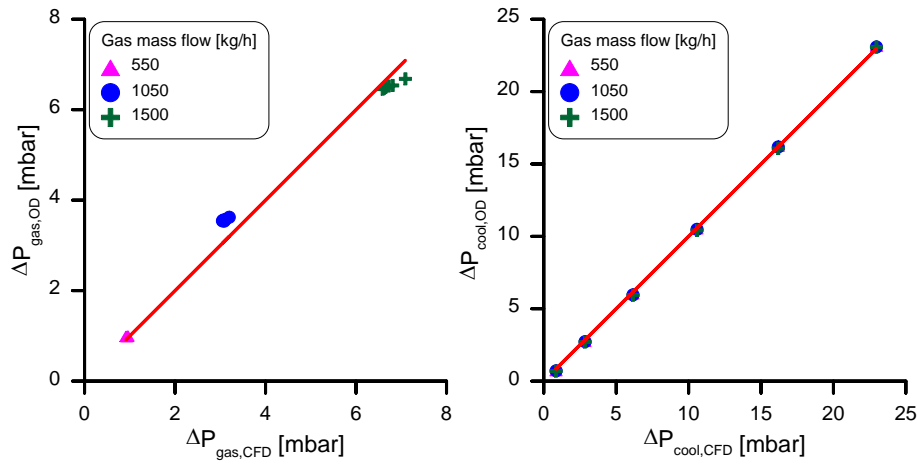
Figure 7. Comparison between 0D model and CFD model results for Gas Outlet Temperature (GOT), Coolant outlet temperature (COT), gas pressure drop and coolant pressure drop for the 18 simulated conditions
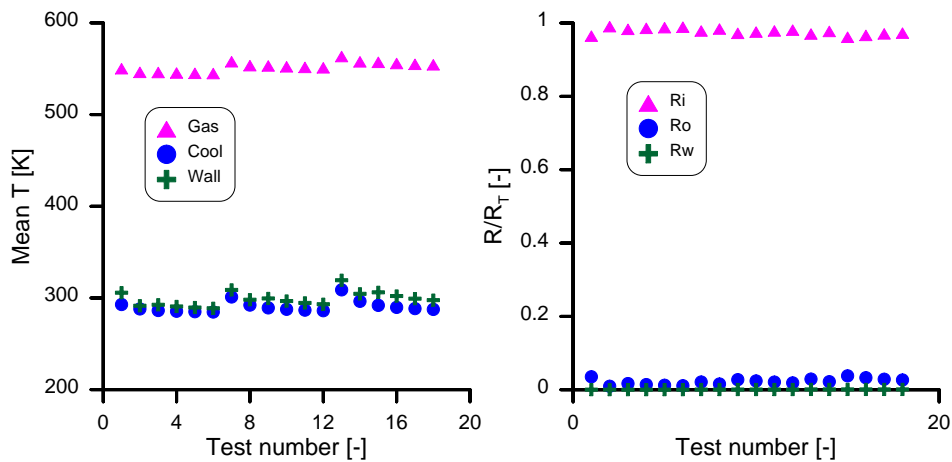

Figure 8. Mean temperatures predicted by the 0D model and heat resistances for the 18 simulated cases

Results

CFD Model

For one of the running conditions (similar results were obtained for the other conditions) a more detailed study on internal flows in shell side was performed. Figure 9 shows the flow path on the side of the casing (coolant) as it passes through the HEX from two perspectives: top cut and front cut. Firstly, the higher speeds on the right side of figure 9 are due to the fact that this is the inlet section of the coolant into the HEX. On the other hand, there are areas of stagnation (zero speeds) on the back of the 3 existing baffles, which are much more evident in the last one, depending on the direction of flow of the fluid (from right to left in the picture). Besides some recirculation bubbles appear behind top of baffles.
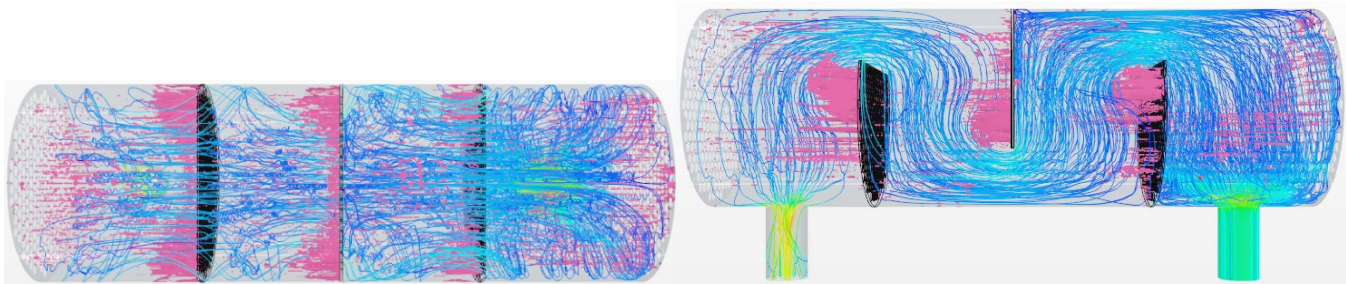

Figure 9. Comparison between 0D model and experimental results for gas temperature and pressure drop

Figure 10 shows the temperature evolution along the length of the shell of the gas, the tube and the coolant. As can be observed the gas temperature decay following an exponential function while the temperature change on tube and coolant is almost negligible. This result is similar to that obtained previously with the 0D model. This in fact tell us that the changes on shell side (for this particular configuration) will have no effect on the global performance of the HEX.
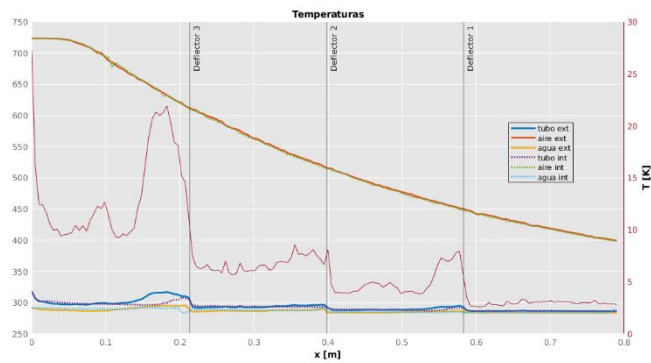
Figure 10. Comparison between 0D model and experimental results for gas temperature and pressure drop

0D Model

As mentioned, with the 0D model different parametric studies can be performed in order to study the effect of changing any of the geometric characteristics on the performance of the HEX. Before explaining the parametric studies performed the main hypothesis applied must be listed since in all the parametric studies carried out, certain geometric parameters were kept constant. These were:

- The thickness of the tubes, so if in the parametric study the inner diameter of the tubes is modified, it implies that the outer diameter of the tubes is also modified
- The clearance between the holes in the deflectors for the passage of the tubes and the tubes themselves. Therefore, as in the previous case, if the internal diameter of the tubes was modified in the study, it would also affect the diameter of the holes made in the baffles.
- Both the thickness between the baffle holes and their geometric distribution, that is, the triangular array is maintained.
- The distance between baffles, so when the length of the tubes is changed, the number of baffles can also change
- The free area/total are of the baffle, that means that ratio between the area for positioning the tube holes in the baffle and the total area is also kept constant. This means that when shell diameter is changed, the baffle area is also changed and the number of tubes can be also changed.

As an example, the change of tube diameter will imply at least the following modifications on HEX geometric characteristics: change of outlet tube diameter since tube thickness was kept constant, change in the deflector holes as the clearance between the tube and the deflector was kept constant, a change in both pitch and pitch to diameter ratio. Other implications on the geometrical characteristics due to a change on one of the changed dimensions in the parametric study can be easily estimated bearing in mind the parameters that were kept constant.

With these assumptions three different studies have been performed with the model. In all the studies the boundary conditions used were: gas and coolant inlet temperatures of 450ºC and 10ºC respectively, gas and coolant flows of 1550 kg/h and 15 m3/h respectively. These are the expected values in a real running condition.

- Analyze the effect of tube length and tube diameter on the performance of the base HEX keeping constant the rest of geometric characteristics and the boundary conditions.
- Study the effect of shell length and tube diameter on the performance of the base HEX keeping constant the rest of geometric characteristics and the boundary conditions
- Find the combinations of tube internal diameter, tube length and shell diameters which gave a predefined outlet gas temperature

Effect of tube length and diameter

The only geometric characteristics that affect the tube of the HEX, taking into account previous assumptions, are tube length and diameter, so the first study is related to the change of these two parameters. Unfortunately, the change of these two geometrical parameters also implies a change in the side of the housing since, on the one hand, the change in the length of the tube will cause a change in the total length of the housing and, as already mentioned, from certain increases or decreases in the length of the tube it could lead to an increase or decrease in the number of baffles

Figure 11 shows, firstly, the effect of the change in the internal diameter of the tube on the number of tubes in the HEX taking into account the above-mentioned geometrical limitations and restrictions. As can be seen, the larger the internal diameter of the tube, the fewer tubes that can be arranged in the HEX. Besides, the lower the internal tube diameter, the higher the number of tubes. From a mechanical and economic point of view, there must be a limit on the maximum number of tubes As an example, the number of tubes corresponding to the basic configuration of the HEX has been marked with a dot in Figure 11. The effect of changes in these two geometric parameters on the heat dissipated in the exchanger, on the outlet temperature and on the pressure drop on the gas side is shown below. These 3 parameters have been chosen because they are the most affected, as seen in the section on model validation. In Figure 11, it also can be seen that the longer the pipe, the more heat is exchanged between the two fluids (and therefore the lower the gas outlet temperature). However, increasing the diameter of the pipe implies less heat exchanged. This is due to the fact that, despite increasing the gas/internal wall contact area of the tube, the decrease in the number of tubes and the reduction of the heat transmission coefficient by gas/wall convection as a consequence of the speed of the gas passing through the interior of the tubes cancels out the possible positive effects expected from the increase in area.

In the case of the pressure drop the lower the internal diameter, the higher the pressure drop as expected. On the other hand the higher the tube length, the higher the pressure drop but the effect of tube diameter is more important than the tube length as the almost parallel iso-gas pressure drop lines show in Figure 11.
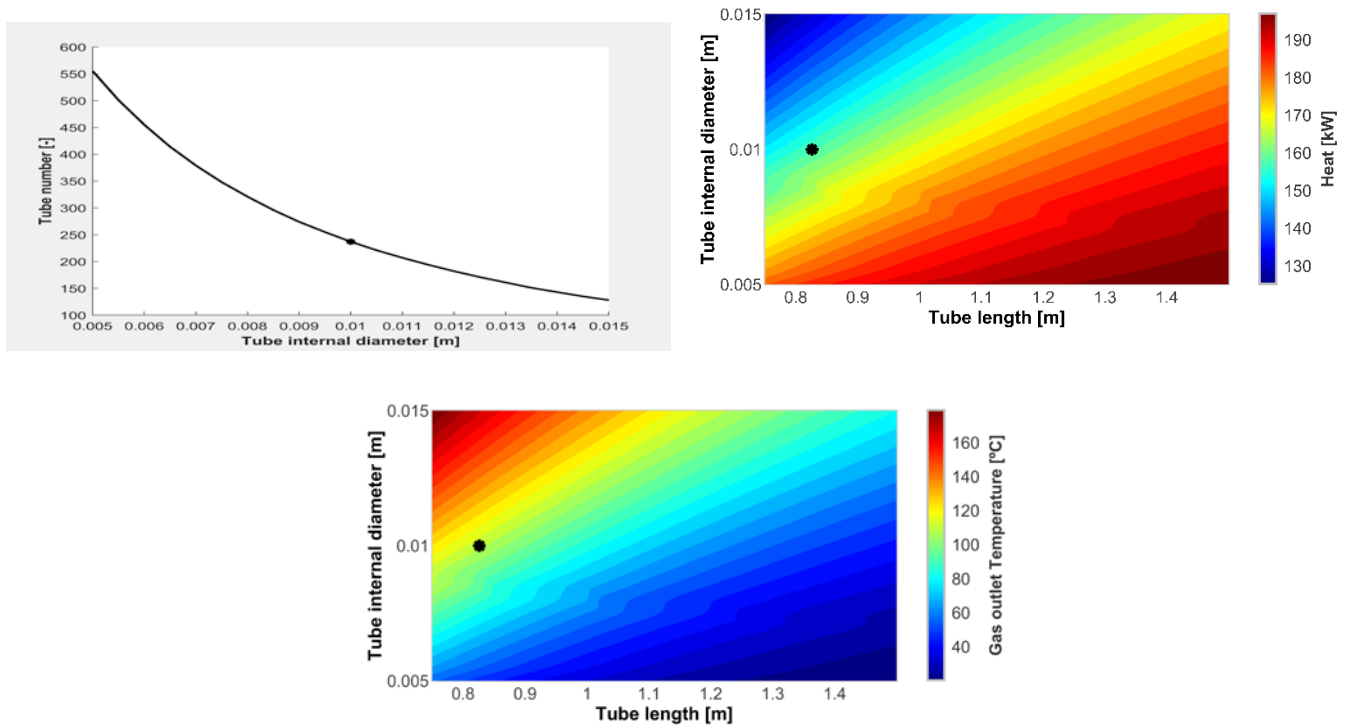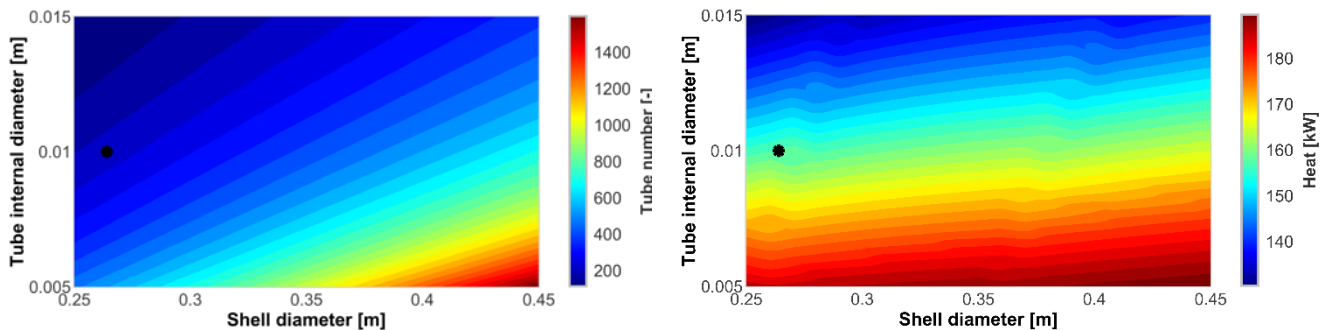

Figure 11. Effect of tube length and tube diameter on tube numbers, heat exchanged, gas outlet temperature and gas pressure drop.

Effect of tube internal diameter and shell diameter

The effect on tube number, heat exchanged, gas outlet temperature and gas pressure drop of the changes on tube internal diameter and shell diameter are shown in Figure 12. As expected both geometries (tube length and shell diameter) affect on tube number. The former, as explained in the previous parametric study, ie the larger the internal diameter, the lower the tube number while the higher the shell diameter, the higher the number of tubes as expected. The effect of tube internal diameter on heat transfer is more important that shell diameter. The main reason is due to the higher the shell diameter, the higher the number of tubes. This fact could lead to the idea that due to the increase in the contact area, the heat transferred should also increase. However, due to this increase in the number of tubes, the speed of the gas inside the tubes decreases and, as a consequence, the heat transfer coefficient is also reduced. So the effect of shell diameter on heat transfer is very small compared to the effect of tube inlet diameter. Similar analysis explains the observed effects on gas outlet temperatures.

Finally, both the increase of the casing diameter and the decrease of the tube diameter cause an increase in the pressure drop on the gas side. The increase in the casing diameter, as mentioned, causes an increase in the number of tubes whose effect on the pressure drop is greater than the reducing effect of the lower flow speed through the tubes
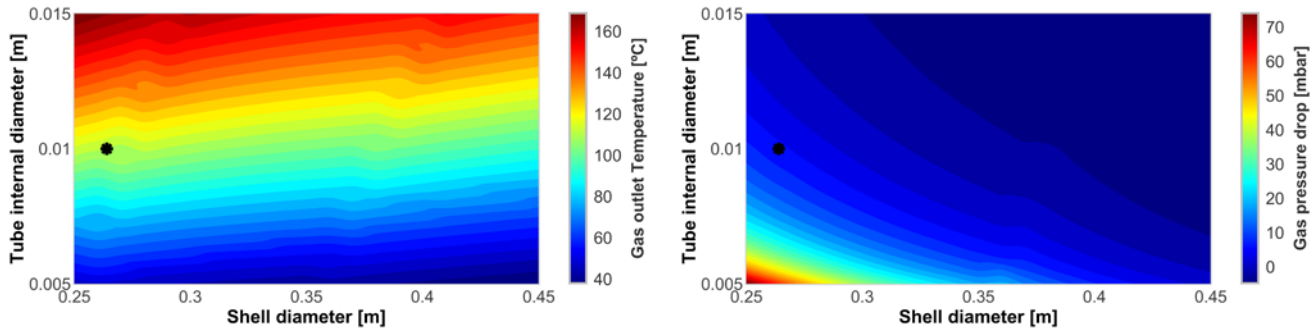
Figure 12. Effect of shell diameter and tube internal diameter on tube numbers, heat exchanged, gas outlet temperature and gas pressure drop.

Geometrical parameters that complain with a predefined outlet conditions

The final study performed in this work consisted of finding the combinations of tube internal diameter, tube length and shell diameters which gave an outlet gas temperature of 50ºC. These temperature is the maximum desired temperature in order to avoid the damage of the whole system. This study will allow the selection of the HEX.
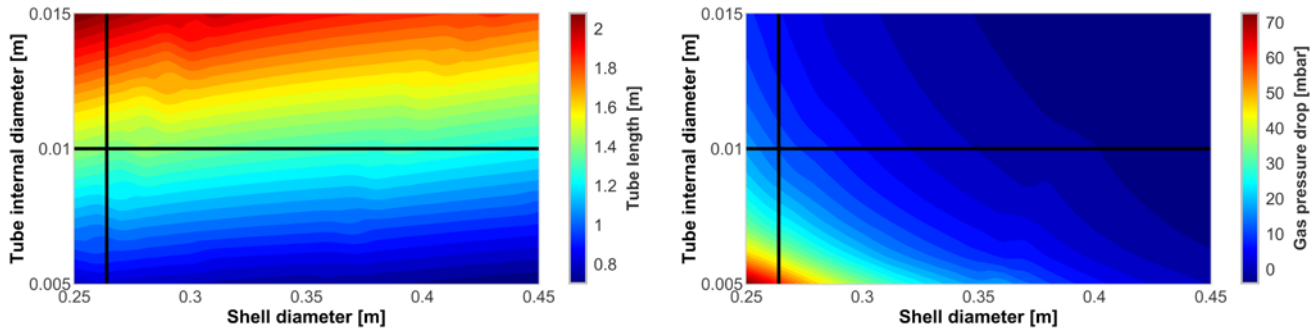

Figure 13. Geometric combinations to obtain the predefined gas outlet temperature.

In this case, the number of tubes in the HEX is not shown since, for a combination of tube diameter and housing diameter, this number is the same as shown in figure 12. In figure 13, as a reference, the inside diameter of the tube and the casing diameter corresponding to the base exchanger have been marked with black lines. The upper part of figure 13 shows the required length of each of the tubes for different combinations of tube and casing diameters that would make the gas outlet temperature 50ºC. As can be seen, the larger the tube diameter, the longer the tube is required. The reason has already been explained above and is that the positive effect on the heat flow of the increased contact area by increasing the tube diameter is counteracted by two negative effects. On the one hand, the lower number of tubes which reduces the total area and, on the other hand, the lower speed of gas circulation through the tubes. From the upper part of figure 13, the effect of increase shell diameter has a very low effect on the needed tube length to reach the desired GOT but, as shown, the higher the shell diameter, the lower the tube length. In this case the increase of contact area due to the increase of tube number has a great effect on heat transfer than the reduction of heat transfer coefficient (due to the reduction of gas velocity) has.

The lower part of figure 13 shows the pressure drop for the different geometric configurations that achieve a gas outlet temperature of 50ºC. As can be seen, the increase in the pipe diameter causes a reduction in the pressure drop. This same effect is caused by increasing the diameter of the casing.

Using figure 13 it can be concluded that a decrease in the tube diameter implies a lower number of tubes but causes an increase in the pressure drop. The effect in both cases is fundamentally due to the increase of the gas speed when passing through the tubes

Summary/Conclusions

The main conclusions of the work are:

- A 0D model for shell and tube heat exchanger (HE) have been programmed. The model have been adjusted with experimental data for a base HEX, where very good agreement has been obtained for gas outlet temperatures, coolant outlet temperature, gas pressure drop and coolant pressure drop in the two tested inlet gas temperatures.
- The CFD model has been constructed and validated against the results of the 0D model in 18 simulated cases
- The results of the 0D and CFD model indicate that the most important resistance is on the gas side (tube) and the improvements of the HEX should be focused on modifying this side of the HEX rather than trying to change coolant side (shell).

- The results from the three performed parametric studies with the 0D model indicate, on one hand, that the lower internal diameter and the higher tube length implies higher heat transfer rates and, on other hand, the effect of shell diameter seems to be related with the increase in tube number rather than the shell diameter itself.
- The results have been used for manufacturing new HEX for altitude simulators.

References

1. European Commission, 2007, "Regulation (EC) No 715/2007 of the European Parliament and of the Council," Off. J. Eur. Union, L171, pp. 1–16.

2. Testa, D., 2011, "Apparatus and Method for Altimetry Conditioning of Internal-Combustion Engines," European Patent No. 09425346.5–EP2295955.

3. Desantes, J. M., Galindo, J., Payri, F., Serrano, J. R., and Piqueras, P., 2015, "Device for Conditioning the Atmosphere in Alternative Internal Combustion Engine Tests, Procedure and Use of Said Device," Patent No. ES2544516B1.

4. Payri, F., Desantes, J. M., Galindo, J., and Serrano, J. R., 2011, "Unit for Simulating the Pressure and Temperature Conditions of the Air Drawn in by a Reciprocating Internal Combustion Engine," U.S. Patent No. 9038578B2.

5. A. Broatch, V. Bermúdez, J.R. Serrano, R. Tabet, J. Gómez and S. Bender. Analysis of Passenger Car Turbocharged Diesel Engines Performance When Tested at Altitude and of the Altitude Simulator Device Used. Journal of Engineering for Gas Turbines and Power 141, 081017-1 (2019).

6. V. Bermúdez, J.R. Serrano, P. Piqueras, J. Gómez and S. Bender. Analysis of the role of altitude on diesel engine performance and emissions using an atmosphere simulator. International J of Engine Research 2017, Vol. 18(1-2) 105–117, doi: 10.1177/1468087416679569

7. Ivanka Milcheva, Florian Heberle, Dieter Brüggemann. Modeling and simulation of a shell-and-tube heat exchanger for Organic Rankine Cycle systems with double-segmental baffles by adapting the Bell-Delaware method. Applied Thermal Engineering 126 (2017) 507–517, doi: 10.1016/j.applthermaleng.2017.07.020

8. Reza Aminia, Mohsen Aminib, Alireza Jafariniac, Mehdi Kashfia. Numerical investigation on effects of using segmented and helical tube fins on thermal performance and efficiency of a shell and tube heat exchanger. Applied Thermal Engineering 138 (2018) 750–760, doi: 10.1016/j.applthermaleng.2018.03.004.

9. Wagner Henrique Saldanha · Felipe Raul Ponce Arrieta · Gustavo Luís Soares. State-of-the-Art of Research on Optimization of Shell and Tube Heat Exchangers by Methods of Evolutionary Computation. Archives of Computational Methods in Engineering 2020, doi: 10.1007/s11831-020-09476-4

10. Julia C. Lemos, André L. H. Costa and Miguel J. Bagajewicz. Set Trimming Procedure for the Design Optimization of Shell and Tube Heat Exchangers. Ind. Eng. Chem. Res. 2020, 59, 14048−14054, doi: 10.1021/acs.iecr.0c02129

11. Caroline de O. Gonçalves1 | André L. H. Costa | Miguel J. Bagajewicz. Linear method for the design of shell and tube heat exchangers using the Bell–Delaware method. AIChE Journal. 2019;65:e16602. DOI: 10.1002/aic.16602

12. SAE International. Turbocharger gas stand test code. SAE International. Turbocharger gas stand test code, SAE J1826. Technical Report, Society of Automotive Engineers Inc, Warrendale, PA, 1995

13. SAE International. Supercharger testing standard, SAE J1723. Technical Report, Society of Automotive Engineers Inc, Warrendale, PA, 1995

14. Bell, K. J., 1988, Delaware method for shell-side design, in Heat Transfer Equipment Design, R. K. Shah, E. C. Subbarao, and R. A. Mashelkar, eds., Hemisphere Publishing, Washington, DC, pp. 145–166.

15. Bell, K. J., 1998, Approximate sizing of shell-and-tube heat exchangers, in Heat Exchanger Design Handbook, G. F. Hewitt, exec. ed., Begell House, New York, Vol. 3, Sec. 3.1.4.

16. Bejan, A. and Kraus, A. D. 2003. Heat transfer Handbook. John Wiley and sons, New Jersey.

17. Incropera, F. P., and DeWitt, D. P., 1996, Fundamentals of Heat and Mass Transfer, 4th ed., Wiley, New York

18. Sha, R. K. and Sekulic, D. P. 2003. Fundamentals of Heat Exchanger Design. John Wiley and sons, New Jersey.