Development of a Design Tool for Heat Exchangers in ORC Plants

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Abstract. The present paper deals with the development of a three-step design chain for the simple and efficient design of tube bundle heat exchangers or evaporators respectively with a focus on ORC applications. It contains a design calculation tool, a CAD geometry model and a CFD analysis. The programming of the calculation tool, the construction of the CAD model and the analysis with a commercial CFD program are supposed to be interlinked with each other in order to create a fully integrated design chain.

INTRODUCTION

Tube bundle heat exchangers are some of the most essential and often used apparatuses in power plant engineering and are appreciated worldwide for their efficiency and durability. It comes to no surprise that the technology of transferring heat from one fluid to another by means of a tube bundle is also commonly used in the ORC application. The development of these heat exchangers of course has its difficulties, for example the selection of the tube material, the calculation of the heat transfer coefficients, the calculation of the tube bundle length, the construction of the model and finally the verification of the calculation and design of the heat exchanger. The present paper deals with all of these design steps in an attempt to integrate the solutions to these problems into a single design chain for ORC tube bundle heat exchangers. This approach will significantly decrease the time that is needed to calculate and design new heat exchangers which is a great competitive advantage in a free market scenario.

CALCULATING THE MEASUREMENTS OF A TUBE BUNDLE HEAT EXCHANGER

The first of the three elements of the design chain is a calculation tool programmed in Microsoft Excel. The goal is to calculate the length of the tube bundle if essential geometric data like the diameter, thickness and number of tubes involved in the tube bundle are given. In addition to that, the types of fluids involved, the temperatures, the pressures and the mass flows need to be specified by the input data. One of the biggest challenges when designing a tube bundle heat exchanger is the calculation of the heat transfer coefficients, especially in a complex geometric structure like a tube bundle being streamed through by an outside fluid stream. The essential formula is the connection between the transferred heat flow $\dot{Q}$, the k-Factor, the logarithmic temperature difference $\Delta\vartheta_m$ and the transfer area A. There are sources in which the product kA is referred to as the “transmission capacity” of the heat exchanger [1] [2]:

$$\dot{Q}=kA\Delta\vartheta_m \tag{1}$$

A very important mathematical relation can be developed from this formula - that is the determination of the minimum heat transfer area A.

$$A=\frac{\dot{Q}}{k\Delta\vartheta_m} \tag{2}$$

The variables $\Delta\vartheta_m$ and $\dot{Q}$ are still pretty simple to determine. The logarithmic temperature difference $\Delta\vartheta_m$ can be calculated from the temperature data only. While the index 1 is used for the heating fluid, the absorbing fluid temperature is marked with the number 2. The high commas signal whether the value is referring to the inlet or the outlet of the heat exchanger. As an example ϑ_2'' stands for the temperature of the fully evaporated working fluid at the outlet of the heat exchanger [1]:

$$\Delta\vartheta_m=\frac{\left(\vartheta_1'-\vartheta_2''\right)-\left(\vartheta_1''-\vartheta_2'\right)}{ln\left(\frac{\vartheta_1'-\vartheta_2''}{\vartheta_1''-\vartheta_2'}\right)} \tag{3}$$

Calculating the transferred heat flow is also rather simple, as the Excel plug-in RefProp [3] is used, that allows the user to determine fluid properties by specifying two other fluid values. For example the enthalpy can be determined by delivering the temperature and the pressure of the fluid in question. As the mass flow must be specified as part of the input data, Formula 4 can be solved as follows [2]:

$$\dot{Q}=\dot{m}_1\left(h_1'-h_1''\right)=-\dot{m}_2\left(h_2'-h_2''\right) \tag{4}$$

Looking at Formula 2, it is obvious, that the main problem with calculating the needed transfer area for a tube bundle heat exchanger is the determination of the heat transition number k. This coefficient is depending on three components which are the heat transfer coefficients on the inside and the outside of the tube and the thermal resistance of the tube material itself. The following Formula 5 shows, how the heat transition number k can be calculated [4]:

$$k=\left(\frac{d_o}{d_i}\frac{1}{\alpha_{fi}}+\frac{d_o}{2\lambda}ln\left(\frac{d_o}{d_i}\right)+\frac{1}{\alpha_{fo}}\right)^{-1} \tag{5}$$

The two coefficients α_{fi} and α_{fo} are representing the heat transfer coefficients on both sides of the tube. The middle term can be understood as the thermal resistance of the tube material itself, as mentioned previously [4]:

$$\alpha_{Wall}=\frac{2\lambda}{d_o ln\left(\frac{d_o}{d_i}\right)} \tag{6}$$

The variable λ in the case of Formula 5 and 6 stands for the thermal conductivity of the tube material, for which there is an easily modifiable dropdown menu implemented in the calculation tool. That being said, the main focus is on the determination of the heat transfer coefficients α_{fi} and α_{fo}. The heat transfer coefficient from the gas to the tube material – meaning on the inside of the tube – referred to as α_{fi} is significantly easier to calculate as there is no complex geometry involved in this heat transfer process. This cannot be said about the coefficient α_{fo} referring to the heat transfer from the hot tube material to the surrounding ORC-fluid. The transfer of heat is far more complicated inside a structure like a tube bundle, than it is just from a hot fluid to the tube it is flowing through. For the heat transfer from the gas to the tube material, it needs to be recognized whether the gas stream is turbulent or laminar, as it affects the formula, which will come to use in the calculation of the Nußelt number. For the laminar case, the *Formula by Martin for all tube lengths* can be utilized to determine the Nußelt number [1]:

$$Nu_{m,lam}=\left[49{,}37+\left\{1{,}615\left(RePr\frac{d}{l}\right)^{0{,}33}-0{,}7\right\}^{3}+\left\{\left(\frac{2}{1+22Pr}\right)^{0{,}167}\left(RePr\frac{d}{l}\right)^{0{,}5}\right\}^{3}\right]^{0{,}33} \tag{7}$$

When the gas flow is turbulent, the universal formula for streams including the transition stage $\left(2300 < Re < 10^{6}\right)$ is used. The components f_1 and f_2 are correction factors calculated from specific geometric and thermal data. The factor ξ stands for the pressure loss coefficient, which depends on the Reynolds number [1]:

$$Nu_{m,turb}=\frac{\frac{\xi}{8}(Re-1000)Pr}{1+12{,}7\sqrt{\frac{\xi}{8}}\left(Pr^{0{,}66}-1\right)}f_1 f_2 \tag{8}$$

The calculation of the Nußelt number on the outside and the resulting heat transfer coefficient α_{fo} is far more complicated than that, as there are a lot more correction factors involved than for the calculation of the heat transfer at the inner wall. In addition to that, these correction factors depend on more specific data than the ones in Formula 8. Therefore not all of the calculation can be shown to a full extend, as that would exceed the format of this paper. Nevertheless it is important to understand the general structure of the calculation process of the Nußelt number regarding an outside stream through a tube bundle. The basic Nußelt number $Nu_{l,0}$ is calculated from the turbulent and the laminar Nußelt number according to Formula 9 [5]:

$$Nu_{l,0}=0{,}3+\sqrt{Nu_{l,lam}^{2}+Nu_{l,turb}^{2}} \tag{9}$$

The two Nußelt numbers only depend on the Reynolds and the Prandtl number. The first correction factor is the so called arrangement factor f_A, which varies dependent on whether there is an aligned or shifted arrangement of the tube rows in the tube bundle [5]:

$$Nu_{0,Bundle}=f_A\,Nu_{l,0} \tag{10}$$

This Nußelt number $Nu_{0,Bundle}$ needs to be further modified with two factors that are called f_N and the K-Factor. The first factor f_N can be set to 1, if there is a high number of baffles in the heat exchanger. The determination of K is more complicated, but the factor is generally dependent on the relation of the Prandtl number in the free stream and the Prandtl number in the near wall region. With these two factors being known, Nu_{Bundle} can be calculated, which can be understood as the ideal Nußelt number for the tube bundle [5]:

$$Nu_{Bundle}=f_N\,K\,Nu_{0,Bundle} \tag{11}$$

The last step for the determination of the actual Nußelt number $Nu_{0,AW}$ is rather laborious, as it mainly deals with the calculation of correction factors from a broad variety of data. The central adjustment factor is the f_W-factor, which consist of the three components f_G, f_L and f_B representing the geometric factor, the leakage flow factor and the bypass factor [5]:

$$f_W=f_G f_L f_B \tag{12}$$

$$Nu_{0,AW}=f_W\,Nu_{Bundle} \tag{13}$$

At this point both the Nußelt number of the gas and the ORC-fluid are known, therefore the actual heat transfer coefficients that are needed for Formula 5 can be determined. The general formula looks like the following [1]:

$$\alpha = \frac{Nu\lambda_{Fluid}}{l} \tag{14}$$

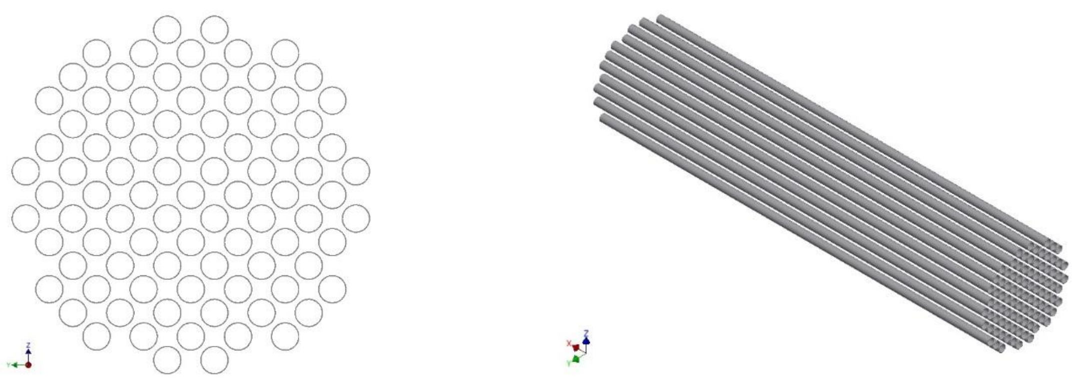
Now all the components to solve Formula 2 are known. The needed area A in order to transfer a specific amount of heat can be determined and from that area – knowing the number of tubes n and their outside diameter – the length finally can be calculated.

$$l = \frac{A}{nd_a\pi} \tag{15}$$

This length then can be transferred to the CAD file, where the length of the tube bundles and the shells are automatically adapted. Of course all of the connecting dimensions regarding the headpieces and flanges also need to be adapted automatically to the new length of the tube bundles.

CONSTRUCTION OF AN ORC HEAT EXCHANGER

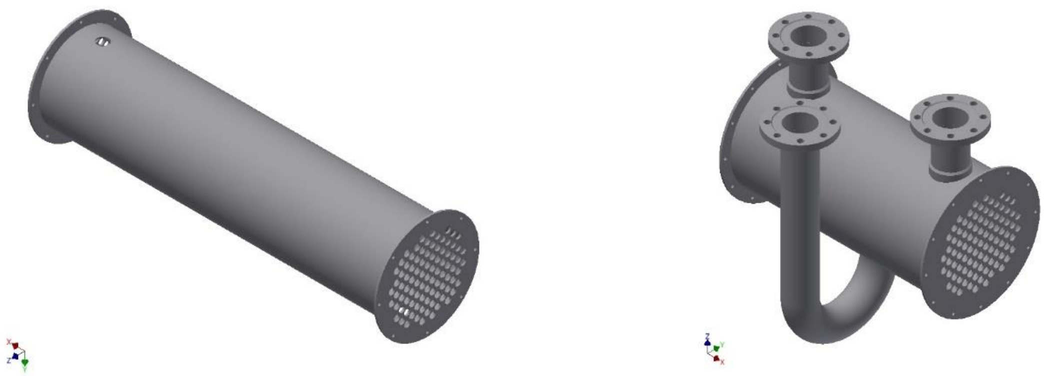
As previously mentioned the most important data transferred from the calculation tool in Excel to the CAD model is the length of the tube bundle which is determining all of the other dimensions of the heat exchanger e.g. the shell length for example. Therefore it can be stated, that the construction and parameterization of the tube bundle model is a central task. The construction of the tube itself is rather simple. The design of the tube bundle however is more complicated. In the following case the form of the bundle will be made of multiple rows of tubes arranged in a shifted way. There are numerous reasons, why this arrangement of the tubes in the bundle is superior to alternative structures. The main reason is, that all of the tubes in the tube bundle have the same distance from their surrounding tubes. This leads to a homogenous thermic stress field in the front and back side of the heat exchanger's shell, where the tube bundle is held. Apart from that, a bore pattern that is arranged in that way is relatively simple to manufacture. The following Fig. 1 shows the tube bundle in the front view (a) and in isometric view (b).



(a) (b)

Figure 1. Front view and isometric view of a tube bundle

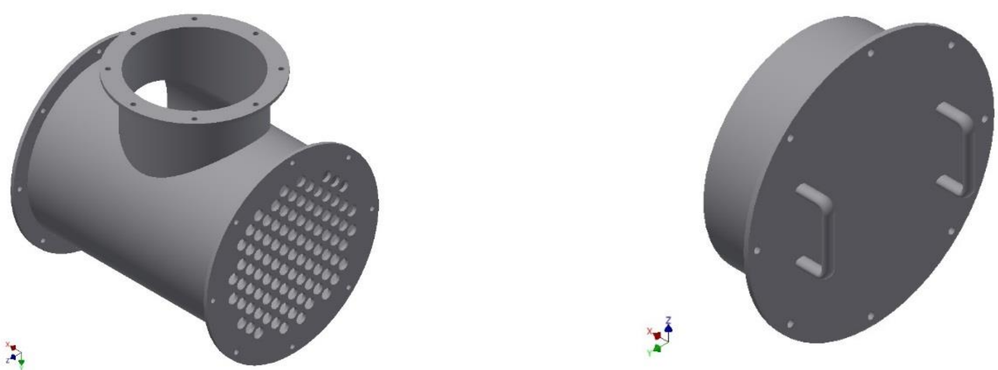
Another important component of the heat exchanger is the shell, of which two different versions are needed, one for the preheater and one for the evaporator. The shell contains the ORC-fluid in the later operation of the heat exchanger. It needs to have an inner diameter bigger than the outer diameter of the tube bundle and flanges on both sides, where the headpieces of the module can be attached. The main difference between the shell of the preheater and the shell of the evaporator is not only that the evaporator shell needs to be bigger, because the two-phase flow has a larger volume, but it also needs a downpipe and two rise pipes for the vapor. That is where the steam drum is connected to the evaporator shell. Apart from that, it is important to be cautious about the pressure development in the evaporator module and to dimension the connecting flanges in a sufficient way for the working pressures. In the following Fig. 2 both of the relevant shells are displayed. Figure 2 (a) shows an example of the preheater's shell in isometric view, while the evaporator shell with the connecting pipes and flanges can be seen in Fig. 2 (b).



(a) (b)

Figure 2. Isometric view of both the preheater shell (a) and the evaporator shell (b)

As described in the previous characterization of the shell part, a headpiece has to be attached to the shell on both sides. The main reason for that is, that in combination with a cap piece, this design guarantees much easier access to the front side of the tubes. That is especially important, considering that the tubes are constantly in contact with exhaust gas during operation. Therefore it can be assumed that cleaning of the heat exchanger will have to be performed in relatively short periods of time. For many mechanical cleaning processes, an access to the tube bundle is required. An alternative version of the headpiece, which would only consist of one piece of course also has some advantages e.g. a reduced number of parts, a reduction of potential assembling mistakes and less production expense in general. But for reasons that have already been mentioned, the additional effort is accepted as the better solution with regard to cleaning purposes. In Fig. 3 (a) the headpiece is shown together with its corresponding cap piece (b).



(a) (b)

Figure 3. Isometric view of the headpiece (a) and the cap piece (b)

The basic modules of the heat exchanger can be constructed by using all of the three core elements that have been described in this chapter. The tube bundle is held by the shell part, while the headpiece and the cap piece are attached to the ends of the shell and thereby creating the preheater or evaporator module. One main component for the proper functioning of the heat exchanger is still missing and that is the steam drum. The steam drum is a part that is mounted on top of the evaporator module and is connected to that module by three pipes: two riser pipes and one downpipe. The wet vapor rises from the evaporator shell to the steam drum, where the remaining liquid is separated from the hot vapor. The hot vapor then is drawn out of the steam drum on the top end of it.

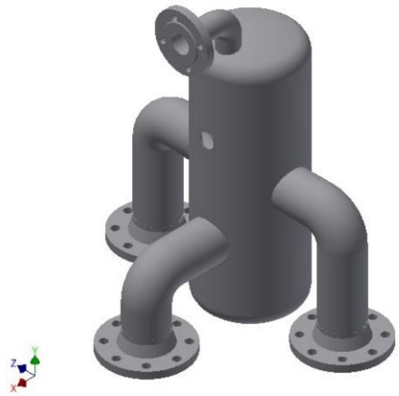


Figure 4. Isometric view of the steam drum

There are more parts needed for the construction of the heat exchanger than just the four essential ones already mentioned, which are the tube bundle, the shell, the headpiece and the steam drum. These additional parts are for example the baffles inside the preheater, the stand for the modules, the type plates and the transport hooks. As all of these parts are rather small, they will not be explained in detail. A full three-module example of the heat exchanger can be seen in Fig. 5.

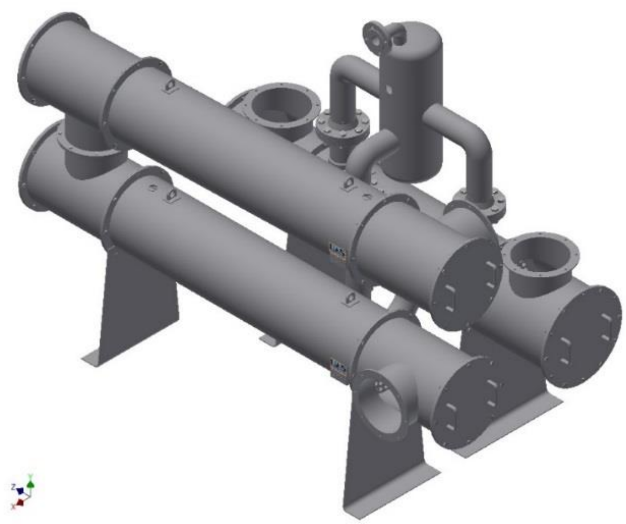
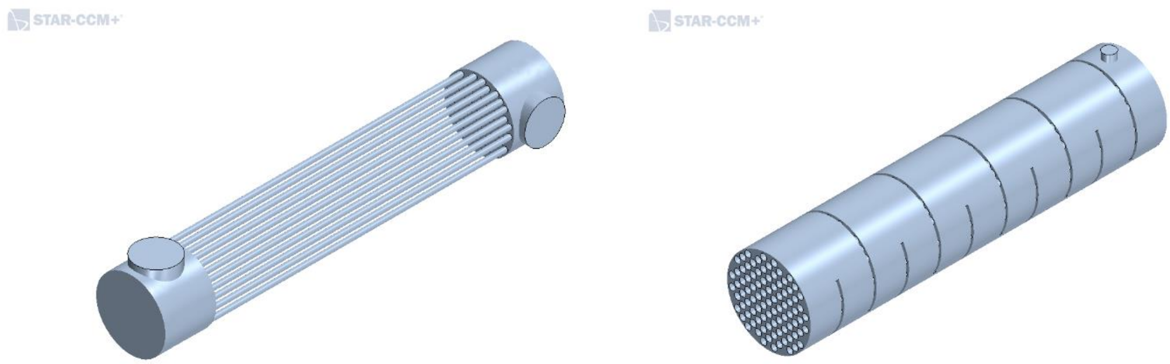


Figure 5. Isometric view of the heat exchanger

SIMULATION OF A TUBE BUNDLE HEAT EXCHANGER

The simulation of the flow and the heat transfer with a commercial CFD software like STAR-CCM+ [6] can substantially contribute to a deeper understanding of the heat transfer process. Apart from that it opens up possibilities for all kinds of optimization and can verify what has been previously calculated in the dimensioning tool. Therefore, the last element of the three-step design chain is a CFD analysis of the tube bundle heat exchanger. The first task with regard to the CFD analysis is the building of the model. It is easily understood, that the actual fluid volumes, which are needed for the CFD analysis, are not explicitly modelled in the CAD, as they only exist as negatives. The gas volume consists of the inner volume of the tube bundle and the inner volumes of the two headpieces on both sides of the tube bundle. The working fluid volume consists of the inner volume of the heat exchanger shell minus the volume of the tube bundle and the baffles. Both of these volumes are displayed in the following Fig. 6.

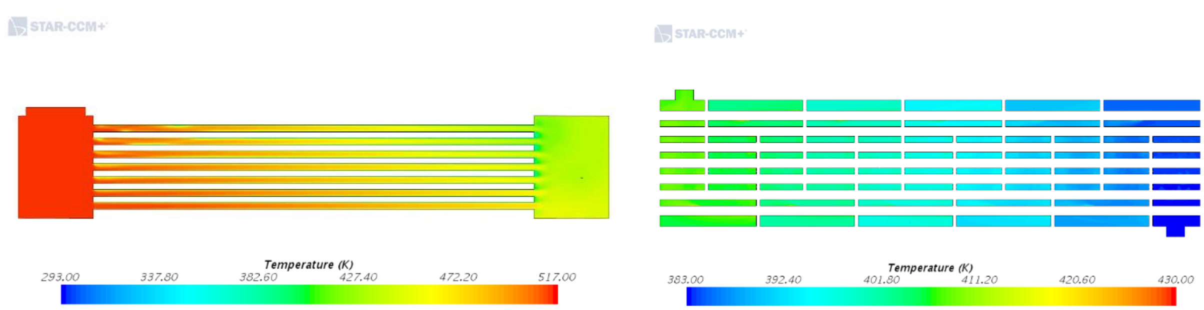


(a) (b)

Figure 6. Isometric view of the CFD models for the gas (a) and the working fluid (b)

Another extremely important step in the simulation process is the meshing of the models. A combination of prism layer meshing for the near wall regions and polyhedral meshing for the core volume is used. The prism layer mesher creates multiple layers of prismatic cells around the walls, so that these regions can be resolved in a fine way. The polyhedral mesher is the standard volume mesher in STAR-CCM+. It creates a high quality volume mesh by using polyhedral cells as a base element. This meshing method results in about 10 million cells for each of the parts. The selection of a turbulence model is the next step that must be performed in order to achieve a running simulation. The Menter-k-ω-SST-model is a hybrid model which combines the K-ω- and the K-ε-model and therefore has good attributes for near wall regions and free flow regions. It can be described as the current basic model for flow simulations in STAR-CCM+, which is the reason why it is chosen for the simulation of the heat exchanger. The main simplification that remains in the simulations as it is currently set up, is that a co-simulation is used. That means, that the transferred heat flow from the gas to the ORC-fluid is implemented as a boundary condition.

A reasonable first step in the evaluation of the simulation is to check the behavior of the temperature, as the exit temperature is easily comparable to the value calculated in the dimensioning tool. For the following results, which are displayed in Fig. 7, the difference between the calculated and the simulated exit temperature in the gas is 441 K to 437 K and for the ORC-fluid it’s 407 K to 403 K. In the case of the gas part it can clearly be recognize that the distribution of the temperature is not homogenous in the tube bundle, as the lower tubes have longer high temperature zones in flow direction. The assumption can be made, that there are vortices in the headpiece of the gas inlet, which lead to that temperature distribution. For the ORC-fluid part it can be said, that the temperature increment between the single sections, which are separated by the baffles, is stable and overall more homogenous than in the gas.



(a) (b)

Figure 7. Temperature development in the gas (a) and the working fluid (b)

From the evaluation of the temperature, the investigation of the overall fluid behavior and the velocity follows. The assumption already was made, that there are vortices in the inlet headpiece of the gas, which lead to a suboptimal temperature distribution in the tube bundle. In Fig. 8 (a), it can be seen, that this assumption has been correct and that there are vortices in the inlet area. In order to further confirm this point, the inlet velocity right in the vertical plane of the tube bundle inlet is displayed in Fig. 8 (b), where the velocity distribution between the single tubes can be recognized. The analysis of this velocity distribution explains the temperature distribution in the tube bundle, as a high fluid velocity results in a longer high temperature zone in the corresponding tube. Furthermore this behavior of the gas is problematic, as it leads to a significant pressure loss even before the gas enters the inside of the tube bundle where the actual heat transfer process starts. In an optimal case scenario the gas would enter the headpiece through the inlet and from there move through the tubes in a much smoother way without a significant pressure loss. Additional optimization work that can follow from here is an investigation on the shape of the headpiece, or slowing down the inlet velocity in order to reduce the pressure loss following from the vortex development in the inlet headpiece. An alternative to this might be to take a closer look at the transition area from the headpiece to the tube bundle and to consider a re-design of the preheater shell's front side, that doesn't contain sharp edges.

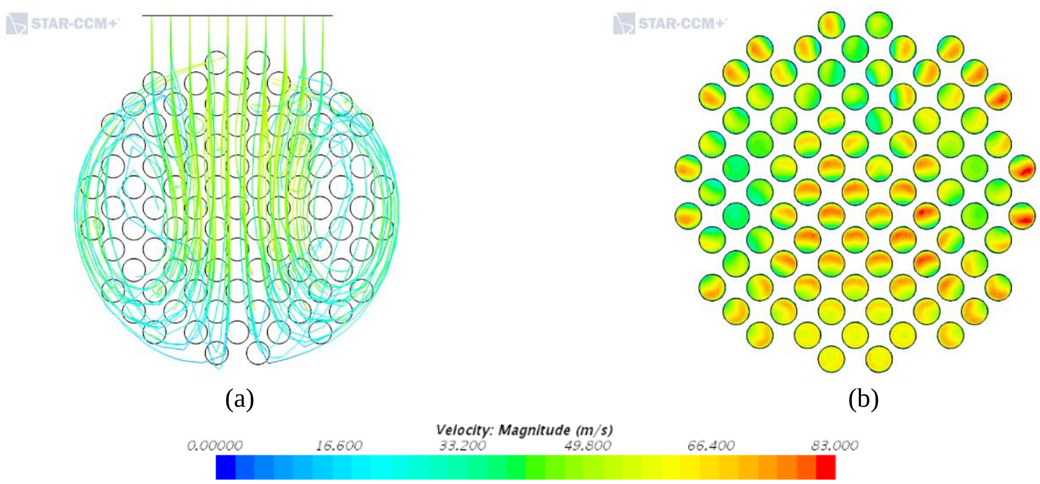


Figure 8. Velocity of the gas in the headpiece (a) and the front plane of the tube bundle (b)

In the last figure, the working fluid velocity is shown with the same streamline technique, that the vortices in the gas are shown. The overall fluid velocity is plausible with regard to the results from the dimensioning tool, also there are local velocity maxima right at the inlet and the outlet, as expected. Generally it can be said, that the high velocity zones are right around the top of the baffles, where the fluid is redirected. There are low velocity regions in the near shell areas of the baffles. To sum up, it can be stated, that the behavior of the working fluid is far less critical than the behavior of the gas, as there are no unexpected vortices in the surrounding ORC-fluid.

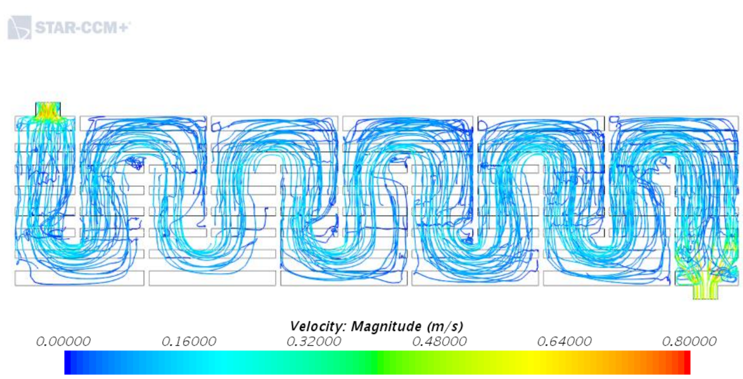


Figure 9. Velocity in the working fluid

CONCLUSION

With the three-step design chain for tube bundle heat exchangers in ORC plants, the operator has a great tool at his hands for designing basic heat exchangers as they are used in standard applications. The design chain can only be partially applied for more specific designs that have a completely different base architecture than the combination of separate preheaters and evaporators or heat exchangers that are not based on a natural circulation of the fluids involved. Additional optimization might be directed at the shape of the heat exchanger modules in the future, especially at the headpieces. Eventually it can be said, that the three-step design chain is very well suited for a quick and efficient first approach to the dimensioning task of tube bundle heat exchangers. However it cannot replace personal expertise, general knowledge of heat transfer processes and constructional know-how of the operator.

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