3
Heat exchangers model

As previously stated, the present study encloses the use of either gaseous or liquid waste heat fluids. For this reason, the heat exchanger models will be different, depending on weather the waste heat fluid is a gaseous or liquid source.

For both cases, though, compactness and high heat exchange coefficients are essential criteria of selection.

The heat transfer at each of the control volumes carries out a different purpose. In the economizer the working fluid is heated from liquid phase at ambient temperature to the saturation point, in the evaporator the working fluid is boiled and in the superheater, in vapor phase, it is heated up to a temperature below the thermal stability of the fluid, so as to avoid decomposition.

The different control volumes are represented as three separate heat exchangers with different denominations according to the nature of the hot and cold fluids.

3.1.
Model for liquid heat source exchangers

The researches conducted by Nguyen et al. [27] and Yamada et al. [13] have proven the possibility of using brazed plate heat exchangers as boilers and condensers when working with organic fluids in a Rankine cycle. The brazed plate heat exchanger (BPHE), as shown in Fig.15, has high effectiveness with a compact size. In the BPHE the working fluid and the heat transfer fluid are separated into alternate passages between the plates, creating counter flow conditions. The corrugation of the plates, or chevron plate pattern, with angles of 20°, 35° and 45° is presented in the figure.

The heat transfer coefficients of both the working fluid and the waste heat fluid will vary in the different control volumes, not only because of the different temperatures, but, in the case of the working fluid, because of the phase changes.

The heat transfer, for every zone, is respectively: economizer (liquid to liquid), evaporator (liquid to phase change), and superheater (liquid to gas).

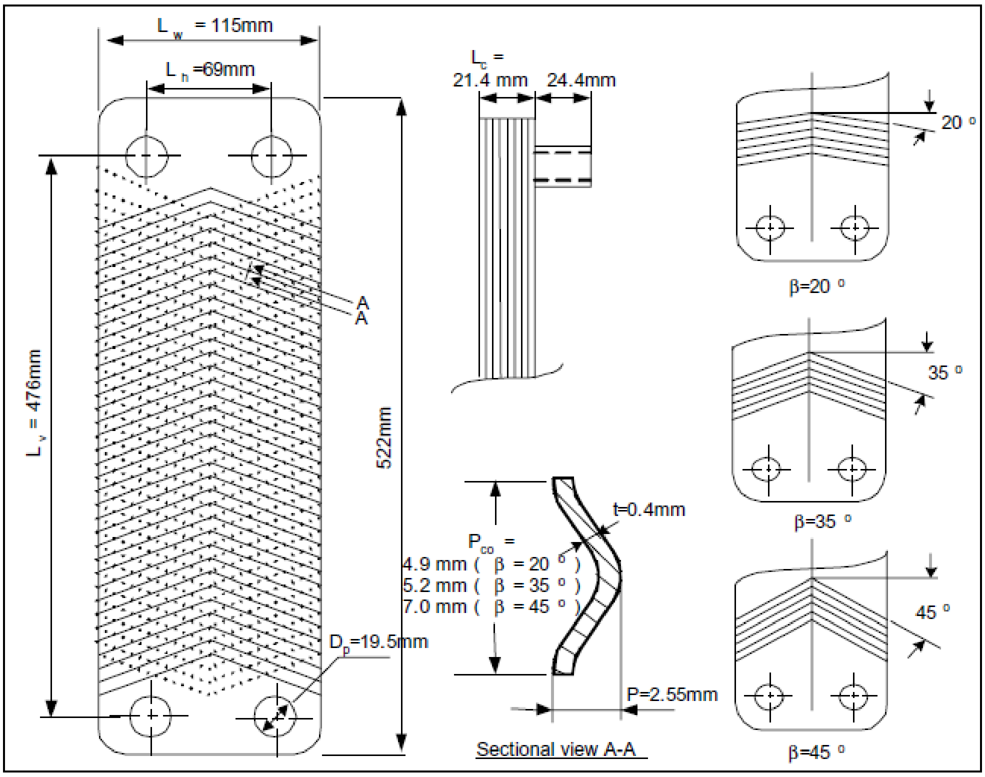

Figure 15. Schematic diagram of a brazed plate heat exchanger [43].

3.1.1.
Economizer heat transfer model

The heat transfer coefficients for both the working fluid and the waste heat fluid are single phase, both of them in liquid phase.

The correlation of the single phase heat transfer coefficient as a function of the Reynolds number, Re, the Prandtl number, Pr, and the chevron angle, β, was obtained from Han et al.[43], and is as follows:

$$\alpha_f = 0.295 \left(\frac{k_f}{D_h}\right) Re^{0.64} Pr^{0.32} \left(\frac{\pi}{2} - \beta\right)^{0.09} \quad (3.1)$$

where k_f is the thermal conductivity of the fluid and D_h the hydraulic diameter of the channel, defined as:

$$D_h = \frac{2b}{\varphi} \quad (3.2)$$

where b is the mean channel spacing and φ the ratio of the developed length to the projected length (=1.17).

The fluid Reynolds number is defined as:

$$Re = G_f \frac{D_h}{\mu_f} \tag{3.3}$$

where G_f is the mass flux of the fluid and μ_f the fluid viscosity.

The Prandtl number is defined as:

$$Pr = \frac{\mu_f c_{p,f}}{k_f} \tag{3.4}$$

The economizer overall heat transfer coefficient is obtained from the heat transfer coefficients of the working fluid and waste heat fluid, as follows:

$$U_{ec} = \frac{1}{\frac{1}{\alpha_{wf}}+\frac{1}{\alpha_{hf}}} \tag{3.5}$$

The single-phase heat exchange rate in the economizer is defined by the following equation:

$$\dot{Q}_{ec} = (UA)_{ec}\, F \Delta T_{\mathrm{lm,ec}} \tag{3.6}$$

where $(UA)_{ec}$ is the conductance of the economizer, $\Delta T_{\mathrm{lm,ec}}$ the logarithmic mean temperature difference, and F the correction factor.

The equation used for F is provided by Roetzel and Nicole [40], as follows:

$$F = \sum_{i=1}^{m} \quad \sum_{k=1}^{n} \quad [a_{ik}(1-r_m)^k \sin(2i \tan^{-1} R_m)] \tag{3.7}$$

where,

$$R_m = \frac{T_{1i}-T_{1o}}{T_{2o}-T_{2i}} \tag{3.8}$$

and

$$r_m = \frac{\Delta T_{lm}}{T_{1i}-T_{2i}} \tag{3.9}$$

Values for coefficient aik, for different heat exchanger configurations, are provided by [40].

From Eq.(3.6), the conductance of the economizer is known, and so the economizer area can be derived from:

$$A_{ec} = \frac{UA_{ec}}{U_{ec}} \tag{3.10}$$

3.1.2.
Evaporator heat transfer model

In the evaporator the waste heat fluid remains liquid, as it does throughout the entire cycle and so the heat transfer coefficient is calculated with Eq.(3.1). On the other hand, the working fluid is changing phase and requires an equation to describe this process.

The two-phase heat transfer coefficient for a brazed plate heat exchanger, obtained from Han et al. [43] is:

$$\alpha_{ev} = Nu\,\frac{k_l}{D_h} \tag{3.11}$$

where k_l is the thermal conductivity of the working fluid in liquid phase and Nu is the two-phase Nusselt number, calculated as:

$$Nu = Ge_1 Re_{Eq}^{Ge_2} Bo_{Eq}^{0.3} Pr^{0.4} \tag{3.12}$$

where Ge_1 and Ge_2 are non dimensional geometric parameters, Re_{Eq} is the equivalent Reynolds number, and Bo_{Eq} the equivalent boiling number.

$$Ge_1 = 2.81\left(\frac{p_{co}}{D_h}\right)^{-0.041}\left(\frac{\pi}{2}-\beta\right)^{-2.83} \tag{3.13}$$

$$Ge_2 = 0.746\left(\frac{p_{co}}{D_h}\right)^{-0.082}\left(\frac{\pi}{2}-\beta\right)^{0.61} \tag{3.14}$$

where p_{co} is the corrugation pitch of the heat exchanger (Figure 14).

$$Re_{Eq} = \frac{G_{Eq} D_h}{\mu_f} \tag{3.15}$$

$$Bo_{Eq} = \frac{q^{"}}{G_{Eq}\, h_{fg}} \tag{3.16}$$

where G_{Eq} is the equivalent mass flux, $q^{"}$ the heat flux in the evaporator, and, h_{lv} ,the enthalpy difference between the liquid phase and the vapor phase.

The evaporator overall heat transfer coefficient is obtained from the two-phase heat transfer coefficient of the working fluid and the single phase heat transfer coefficient of the heat transfer fluid.

$$U_{ev} = \frac{1}{\frac{1}{\alpha_{ev}} + \frac{1}{\alpha_{hf}}} \tag{3.17}$$

The evaporator conductance is:

$$\dot{Q}_{ev} = (UA)_{ev} F \Delta T_{lm,ev} \tag{3.18}$$

Then, the evaporator area is defined as:

$$A_{ev} = \frac{UA_{ev}}{U_{ev}} \tag{3.19}$$

3.1.3. Superheater heat transfer model

The superheater presents single phase heat transfer coefficients for both the waste heat fluid (liquid) and working fluid (vapor). Therefore, the same correlating equations as for the economizer, Eqs. (3.1) to (3.6), can be utilized.

The superheater area is obtained from the following equation:

$$A_{sh} = \frac{UA_{sh}}{U_{sh}} \tag{3.20}$$

3.1.4. Plate heat exchanger model prescribed values

For the sizing of the plate heat exchangers, several non dimensional parameters need to be defined. Table 3 presents the values considered in the model and were obtained from Han et al. [43].

The hydraulic diameter of the channel is defined as:

$$D_h = \frac{2b}{\varphi} \tag{3.21}$$

where b is the mean channel spacing defined as the plate pitch minus the plate thickness. And φ is the ratio of the developed length to the projected length, generally given by the manufacturer.

Table 3. Value of BPHE parameters, from Han et al. [43].

Chevron angle, β [radian]	Ratio of the developed length to the projected length, φ	Mean channel spacing, b [m]	Corrugation pitch, p_{co} [m]
$\frac{\pi}{4}$	1.17	$(2.55 - 0.4) \times 0.001$	5.2×0.001

3.2. Model for gaseous heat source exchangers

In a gas-to-liquid heat exchanger, the heat transfer coefficient on the gas side is 1/10 to 1/100 of that on the liquid side. Therefore, for a ‘‘thermally balanced’’ design (i.e., having a value for αA of the same order of magnitude on each fluid side of the exchanger), fins are employed to increase the gas-side surface area. Thus, the common heat exchanger constructions used for a liquid-to-gas heat exchanger are the extended surface and tubular Shah. [26]

The tube-fin heat exchanger is largely used in gas-to-liquid exchangers where the heat transfer coefficient on the liquid side is generally one order of magnitude higher than on the gas side. Since an extended surface is needed on the gas side of the heat exchanger, a tube-fin exchanger is selected with circular fins on circular tubes, as shown on Fig.16.

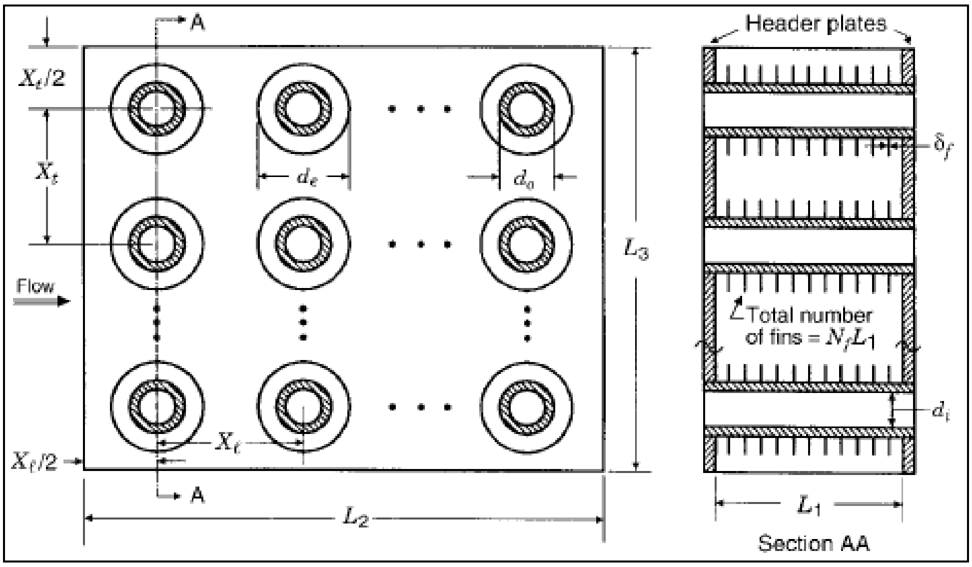

Figure 16. Circular-finned tubular exchanger, from Shah [26].

The layout of the tubes can have different arrangements, as shown on Fig.17; however, all the staggered tube arrangements were avoided to prevent deposition of gas stream particulates in the fins surfaces, especially if the exhaust comes from heavy fuels. The inline 90° or square arrangement is preferred in order to have a larger free-flow area. [26]

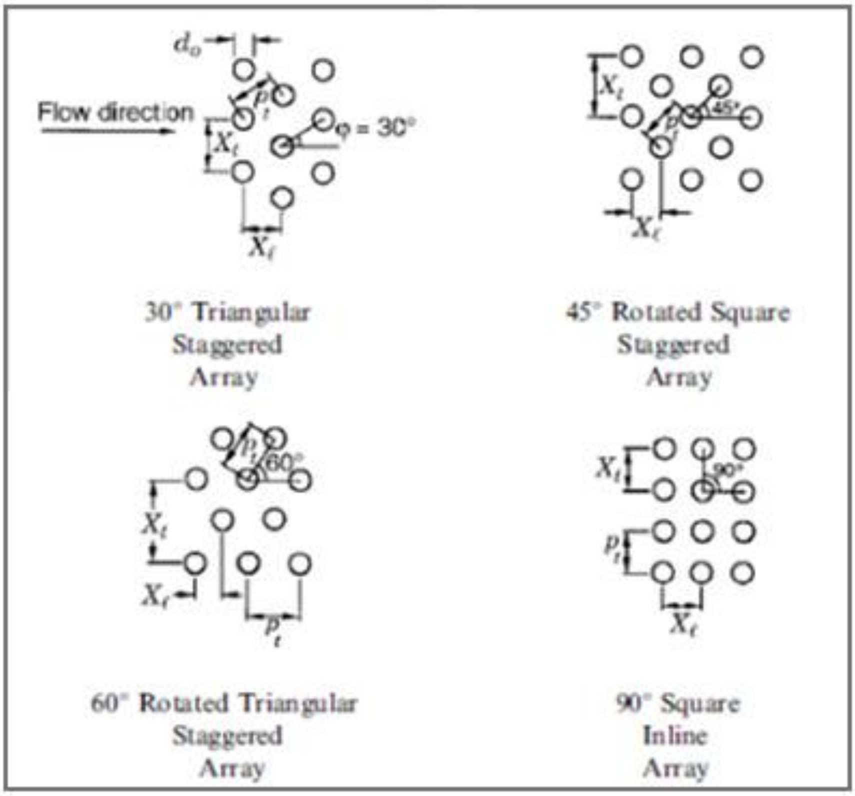

Figure 17. Various geometrical tube layout arrangements, from Shah [26].

From the thermodynamic model it was possible to determine the overall thermal conductance, UA, of each of the three zones (or heat exchanger). In order to specify the area of the heat exchangers, the overall heat transfer coefficient must be determined as follows:

$$U = \frac{1}{\frac{1}{\alpha_o} + \frac{d_o \ln(d_o/d_i)}{2 k_w} + \frac{d_o}{\alpha_i d_i}} \tag{3.22}$$

where α_o is the convective heat transfer coefficient outside the exchanger (gas side), d_o and d_i are the outside and inside diameters of the tube, respectively, k_w is the thermal conductivity of the tube wall, and α_i is the convective heat transfer coefficient inside the exchanger tubes.

3.2.1.
Calculation of gas-side heat transfer coefficient

The convective heat transfer coefficient outside of the exchangers is modeled considering heat transfer flow across tube bundles. The Zukauskas correlation is used to determine the Nusselt number outside of the exchangers, with the expression [26]:

$$\mathrm{Nu} = \mathrm{Pr}^{0.36}\,\mathrm{fn}(\mathrm{Re}_D) \tag{3.23}$$

where Pr is the Prandtl number, as defined in Eq. (3.4), and the Reynolds number, Re_D, defined in Eq. (3.3).

The function $\mathrm{fn}(\mathrm{Re}_D)$ takes the following form for the various flow circumstances in an inline tube bundle arrangement:

$$100 \leq \mathrm{Re}_D \leq 10^3: \ \mathrm{fn}(\mathrm{Re}_D) = 0.52\mathrm{Re}_D^{0.5}$$

$$10^3 \leq \mathrm{Re}_D \leq 2\times 10^5: \ \mathrm{fn}(\mathrm{Re}_D) = 0.27\mathrm{Re}_D^{0.63}$$

$$\mathrm{Re}_D > 2\times 10^5: \ \mathrm{fn}(\mathrm{Re}_D) = 0.033\mathrm{Re}_D^{0.8}$$

Finally the outside convective heat transfer coefficient can be obtained from:

$$\alpha_o = \frac{k\mathrm{Nu}}{D} \tag{3.24}$$

where D is the outside diameter of the tubes.

3.2.2.
Calculation of the internal heat transfer coefficient

The process of calculating the convective heat transfer coefficient inside the tubes will not be the same for all the heat exchangers. In the economizer the working fluid inside the tubes is single phase liquid, in the evaporator a phase change takes place due to boiling and in the superheater a single phase vapor is flowing through the tubes.

3.2.2.1. Internal heat transfer coefficient for the economizer and superheater

Since in the economizer and superheater the working fluid is in single phase the same equations are used and are determined by the Dittus-Boelter correlation, as follows:

For the economizer: $$\alpha_l = 0.023\mathrm{Re}_l^{0.8}\mathrm{Pr}_l^{0.4}\frac{k_l}{D} \quad (3.25)$$

For the superheater: $$\alpha_v = 0.023\mathrm{Re}_v^{0.8}\mathrm{Pr}_v^{0.4}\frac{k_v}{D} \quad (3.26)$$

3.2.2.2. Internal heat transfer coefficient for the evaporator

In the evaporator a phase change of the working fluid is taking place inside the tubes and so an appropriate correlation must be used to determine the convective heat transfer coefficient.

The Thome model, Kattan et al. [45], for intube flow boiling in horizontal tubes, considers the effects of the two-phase flow patterns.

Figure 18 shows the different flow patterns of evaporation in horizontal tubes. The single phase liquid entering the tube reaches its saturation temperature and the boiling process takes place along various flow regimes.

Figure 19 shows the transition from annular flow to stratified flow. This happens when dryout occurs at the top of the tube where the liquid film thickness is thinnest.

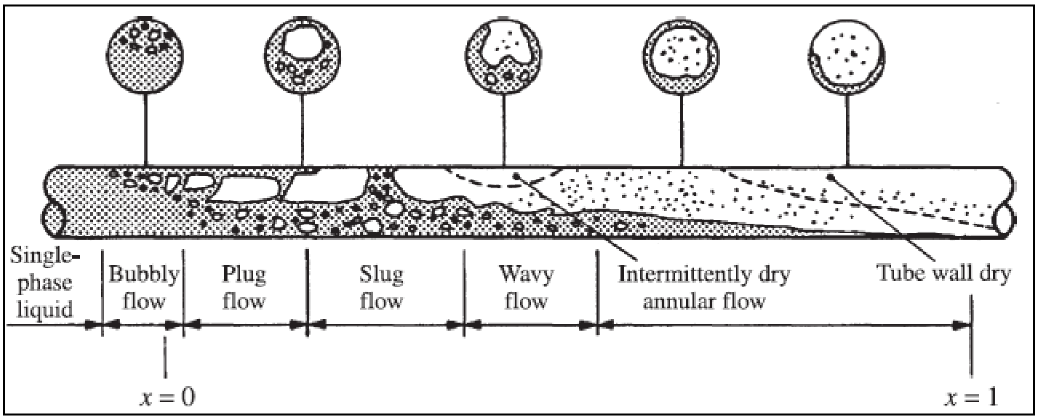

Figure 18. Two-phase flow patterns during evaporation in a horizontal tube, Collier and Thome [45].

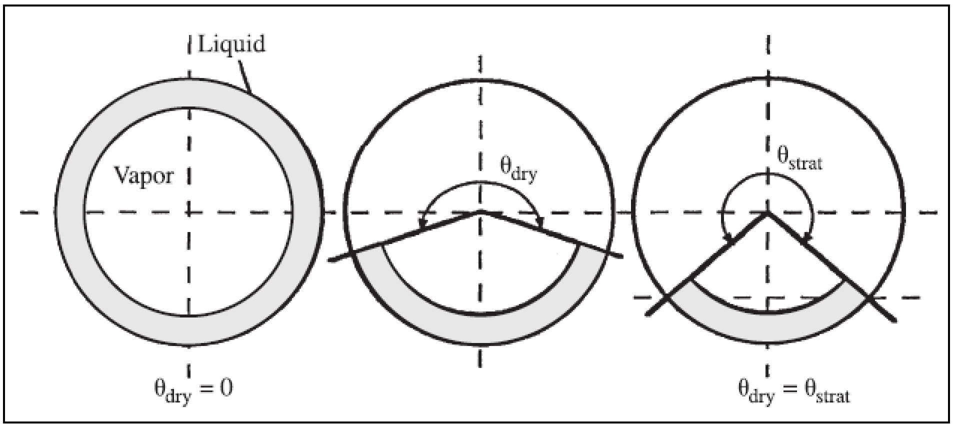

Figure 19. Transition from annular flow to stratified flow, Kattan et al. [45].

The general equation is defined as a two-phase heat transfer coefficient α_{tp} , which combines the liquid and vapor heat transfer coefficients for the wet and dry perimeter segments, as follows:

$$\alpha_{tp} = \frac{\theta_{dry}\alpha_v + (2\pi - \theta_{dry})\alpha_{wet}}{2\pi} \tag{3.27}$$

where θ_{dry} is the dry angle in radians, and is defined as follows:

$$\theta_{dry} = \theta_{strat} \frac{(G_{high} - G)}{(G_{high} - G_{low})} \tag{3.28}$$

where G_{high} is the mass velocity at the transition to annular flow, G_{low} the mass velocity at the transition stratified-wavy flow, and G the mass velocity of the liquid inlet.

The stratified angle θ_{strat} , in radians, can be obtained from a simplified expression:

$$\theta_{strat} \approx 2[\pi - \cos^{-1}(2\gamma - 1)] \tag{3.29}$$

where γ is the void fraction of the cross sectional area and is calculated using the Rouhani-Axelsson [46] correlation, that is given by:

$$\gamma = \frac{x}{\rho_v}\left[\left(1 + 0.12(1-x)\left(\frac{x}{\rho_v} + \frac{1-x}{\rho_L}\right)\right) + \frac{1.18(1-x)[g\sigma_L(\rho_L - \rho_v)]^{0.25}}{G\rho_L^{0.5}}\right]^{-1} \tag{3.30}$$

where x is the vapor quality, ρ_v and ρ_L are the densities of the vapor and liquid, respectively, g is the acceleration of gravity and σ_L the liquid surface tension.

The vapor phase heat transfer coefficient α_v is determined through the Dittus-Boelter correlation, as in the case of the superheater:

$$\alpha_v = 0.023\mathrm{Re}_v^{0.8}\mathrm{Pr}_v^{0.4}\frac{k_v}{D} \tag{3.31}$$

However, the Reynolds number Re_v for this case is defined as:

$$\mathrm{Re}_v = \frac{GxD}{\gamma\mu_v} \tag{3.32}$$

The heat transfer coefficient on the wetted portion of the tube is comprised of the nucleate boiling and the convective boiling contributions linked through the Steiner and Taborek [45] model in the following equation:

$$\alpha_{\mathrm{wet}} = (\alpha_{nb}^3 + \alpha_{cb}^3)^{1/3} \tag{3.33}$$

where α_{nb} is the nucleate boiling heat transfer coefficient defined by the Cooper [45] dimensional correlation:

$$\alpha_{nb} = 55P_r^{0.12}(-\log_{10} P_r)^{-0.55}M^{-0.5}q^{0.67} \tag{3.34}$$

where P_r is the reduced pressure, M is the fluid molecular weight, and q is the heat flux.

The convective liquid flow boiling heat transfer coefficient α_{cb} is determined by the following equation:

$$\alpha_{cb} = 0.0133\mathrm{Re}_L^{0.69}\mathrm{Pr}_L^{0.4}\frac{k_L}{\delta} \tag{3.35}$$

The Reynolds number of the liquid phase Re_L is defined as:

$$\mathrm{Re}_L = \frac{4G(1-x)\delta}{(1-\gamma)\mu_L} \tag{3.36}$$

and the thin liquid film thickness δ is calculated as follows:

$$\delta = \frac{\pi D(1-\gamma)}{2(2\pi-\theta_{\mathrm{dry}})} \tag{3.37}$$

Once the convective heat transfer coefficients are known for each of the heat exchangers the values of their corresponding overall heat transfer coefficients can be determined. This value is then divided by each conductance, obtained in the thermodynamic model, as in Eqs. (3.10), (3.19) and (3.20), to obtain the exchangers' area.

3.2.3. Tube-fin heat exchanger model prescribed values

The design of the fin and tube heat exchangers comprises many parameters that need to be set in order to obtain a solution.

Table 4 indicates the restrictions imposed in order to calculate the areas of the evaporator, economizer and superheater.

Table 4. Prescribed values of heat exchangers design.

Tube inside and outside diameters		Pressure Drop inside tubes	Pressure Drop across tube bundle
d_i (mm)	d_o (mm)	kPa	kPa
10	16	10	<50

3.2.4. Validation of the heat exchangers model

The available literature on ORC heat exchanger design is, to the author's knowledge, limited. Many experimental studies detail the data results obtained and explain the optimization of the process; however the description of the equipment used is scarce. Being a recently developed technology, especially for low grade waste heat, industry suppliers seem very cautious by not mentioning any information on the type of heat exchangers used as boilers in their particular ORC generating sets.

For that reason, finding reliable experimental data for the purposes of mode validation was not an easy task. Nevertheless, the heat exchanger simulation model was validated against data obtained from the parametric optimization study of an ORC by Zhang et al. [47]

The heat exchanger results are compared to the data of the parametric optimization study, in Table 5.

Table 5. Validation of heat exchangers model with parametric study data from Zhang et al. [47].

Parameter	Parametric ORC study data	Simulation input data	
Heat source mass flow rate (kg/s)	1	1	
Heat source temperature (°C)	90	90	
Cooling water source temperature (°C)	20	20	
Turbine inlet pressure (bar)	11	11	
Turbine efficiency	0.8	0.8	
Pump Efficiency	0.75	0.75	
Turbine outlet pressure (bar)	4.39	4.39	
Gross generator power (kW)	250	250	
Pump power (kW)	40	40	
Working Fluids	**Heat Transfer Area (m²)**	**Simulation result**	**% Relative error**
R134a	9.6	9.18	4.33
R123	12.2	11.33	7.13
R245fa	9.2	8.44	8.26

3.3. Heat exchanger simulation results

For the liquid heat source case the working fluids chosen to simulate the cycle were R134a, R1234yf and R1234ze. This selection was carried out according to the temperature range of application of the substances, matched to the temperatures commonly available in waste heat recovery from liquid sources.

Figures 20 to 22 show the calculated total area of the boiler for R134a, R1234yf and R1234ze under different inlet temperatures and mass flow rates of the liquid heat source. It can be observed that, as expected, the required area increases with the mass flow rate of the waste heat fluid and with the operating temperature of the boiler.

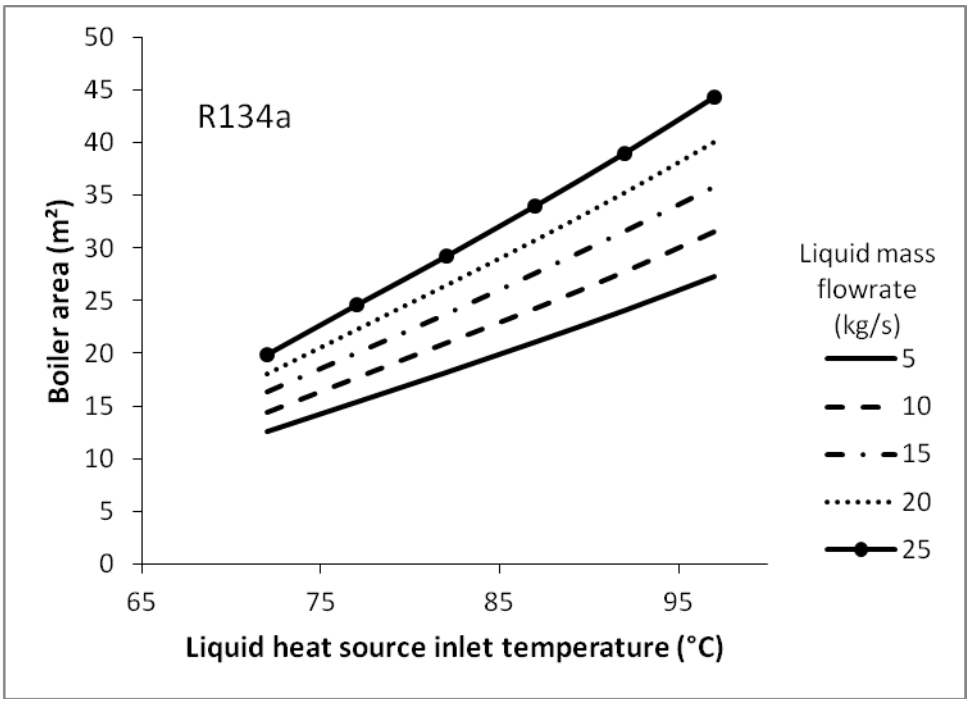

Figure 20. PHE boiler area using R134a at a high pressure of 15.52 bar.

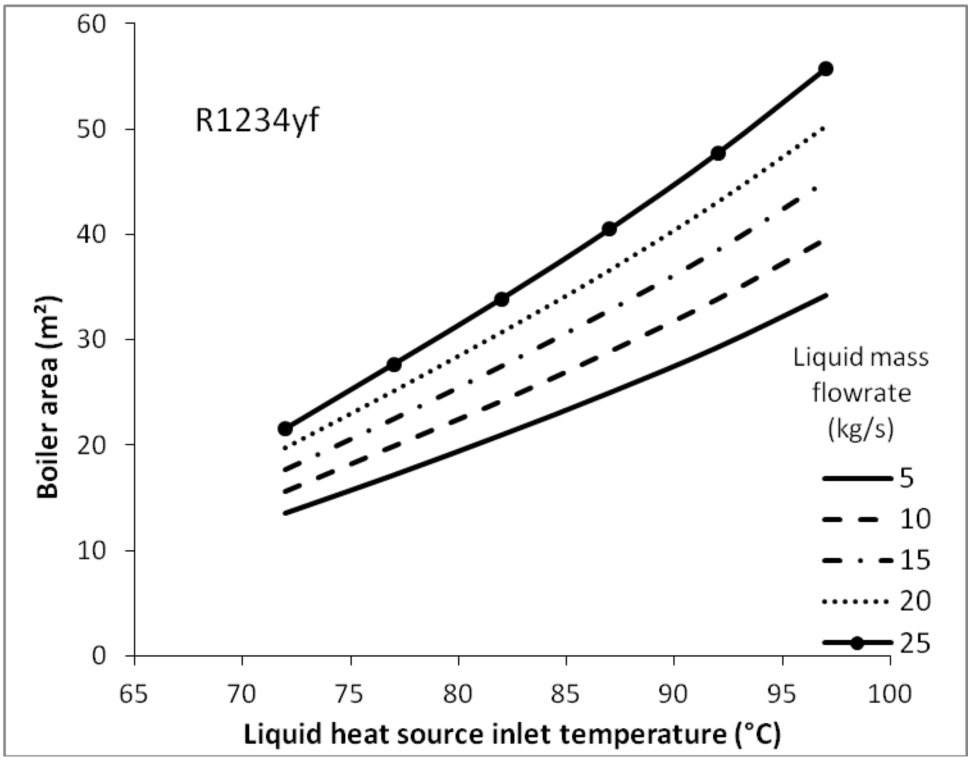

Figure 21. PHE boiler area using R1234yf at a high pressure of 15.56 bar.

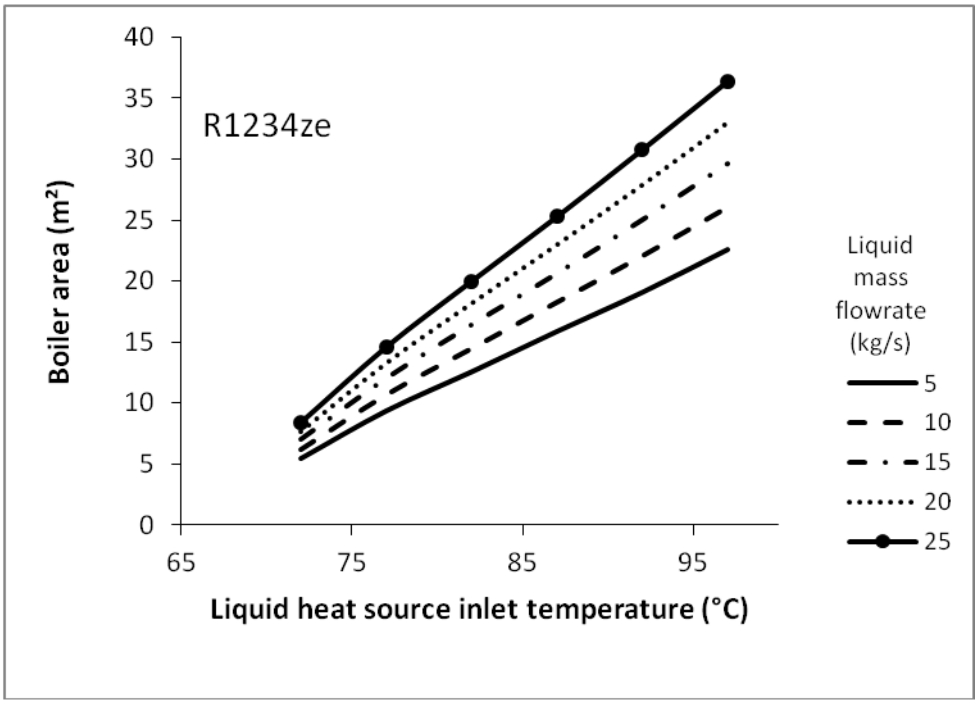

Figure 22. PHE boiler area using R1234ze at a high pressure of 15.34 bar.

Since R1234yf and R1234ze are considered to be new organic substances, compared to R134a, it is interesting to match up the calculated areas of these new substances with the area obtained for R134a. A dimensionless variable $\boldsymbol{a}$ is defined as the area of the new organic substances divided by the area of R134a.

$$a = \frac{A_{R1234yf}}{A_{R134a}} = \frac{A_{R1234ze}}{R_{134a}} \tag{3.38}$$

Figure 23 shows the variation of $\boldsymbol{a}$ for the different inlet temperatures and mass flow rates of hot water.

Comparatively, over the entire heat source temperature range, the required area for R1234yf was greater than that for R134a (a > 1).

Inversely, R1234ze required lower areas, by an amount ranging from 60% to 20% with increasing heat source temperature.

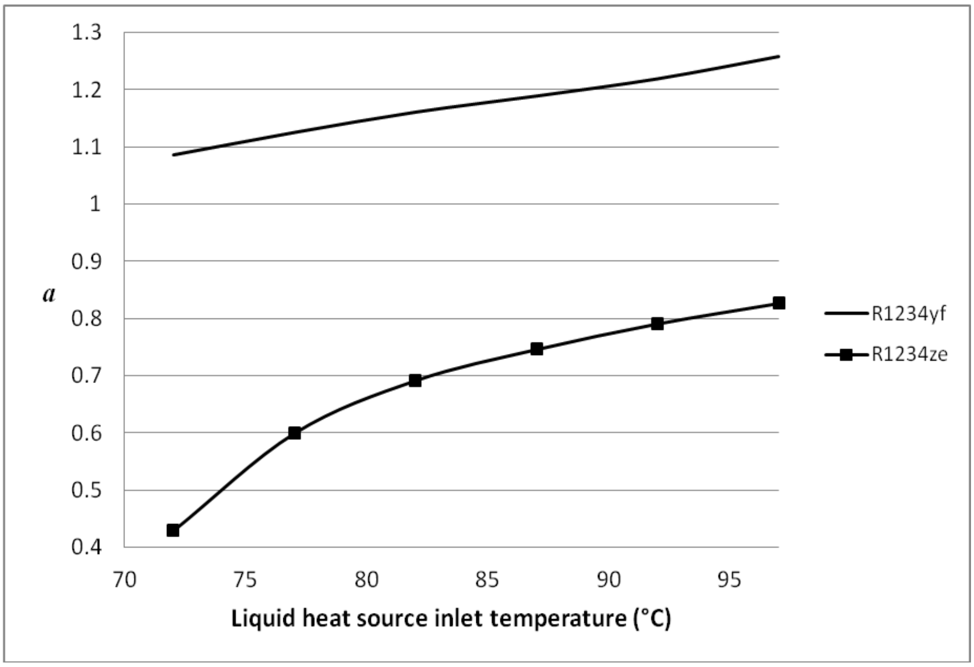

Figure 23. Variation of a at different temperatures.

For the gaseous heat source, the results for the tube-fin heat exchanger boiler are presented on Figs. 23 and 24 for R245fa and R123. The temperatures and mass flow rates were considered according to the available literature on gaseous waste heat recovery.

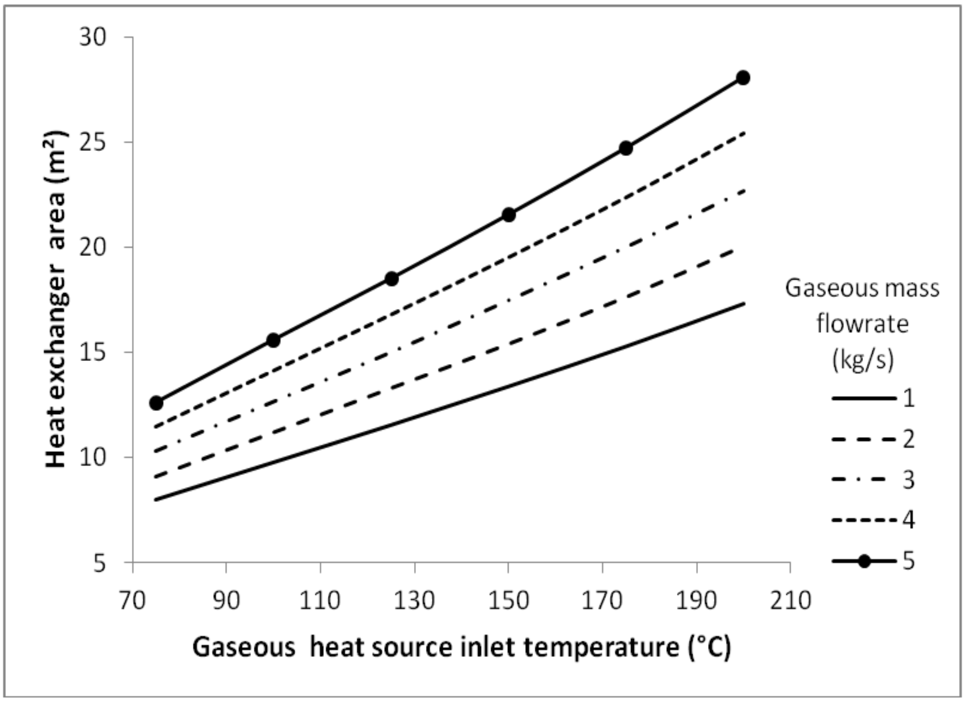

Figure 24. Tube-fin heat exchanger using R245fa at a high pressure of 16 bar.

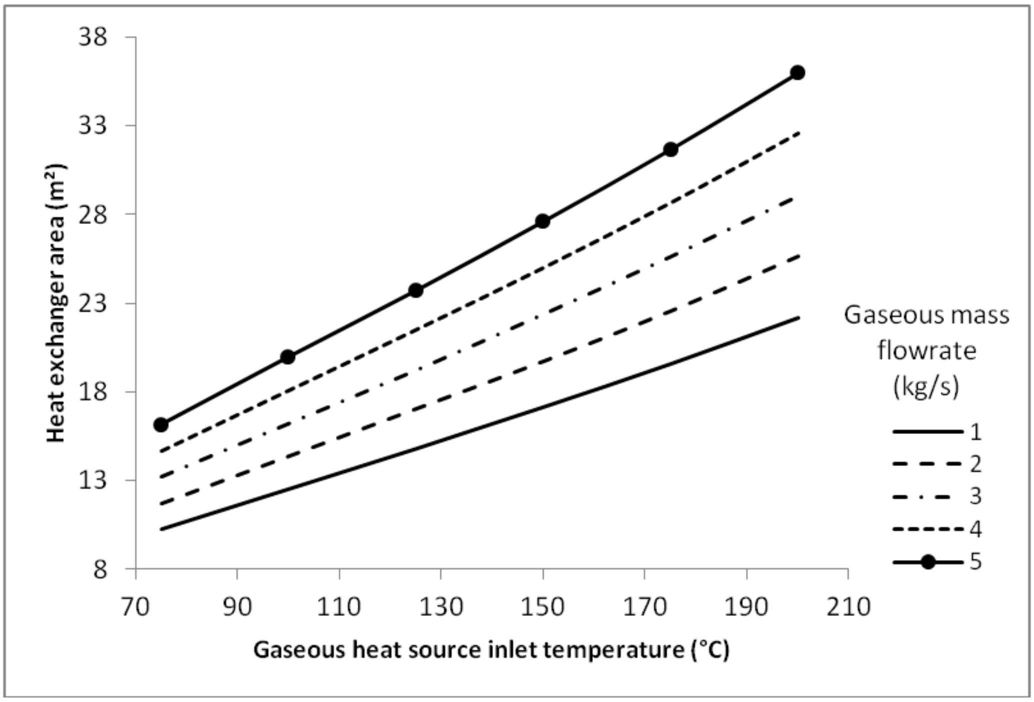

Figure 25. Tube-fin heat exchanger using R123 at a high pressure of 16.3 bar.