

# DESIGN AND RATING OF AN EVAPORATOR FOR WASTE HEAT RECOVERY ORGANIC RANKINE CYCLE USING SES36

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## ABSTRACT

The paper presents a design and rating study of a 4MW evaporator having plain horizontal carbon steel tubes having diameters of 25,4 mm, 31,8 mm and 38 mm, to be used in waste heat recovery via Organic Rankine cycle (ORC). SES36 is chosen as working fluid due to its low boiling point, which makes it suitable for low-grade waste heat recovery with subcritical ORCs. Waste heat carrier industrial air arrives at the evaporator bundle at 280°C. Inlet temperature of the working fluid is 40°C and the evaporation occurs at 125°C and 1,09 MPa. Furthermore, a design sensitivity analysis is made by means of using 13 different in-tube flow boiling correlations. The resulting design and rating parameters yielded by each correlation are compared to each other. By those means, a design error margin of various thermo-hydraulic heat exchanger parameters is revealed, when different in-tube flow boiling heat transfer calculation methods are used. The change in the error margins are investigated with respect to changing tube outer diameter, tube wall thickness, fin density and tube layout (staggered and inline).

## 1. INTRODUCTION

Waste heat recovery as an alternative energy source is receiving more and more attention from the industry and scientific world, as the energy shortage and environmental concerns in the world are rising. Organic Rankine cycles (ORCs) are promising applications for waste heat recovery, due to their heat recovery efficiencies and environmental-friendly features (Quoilin *et al.*, 2013). ORCs are applicable with a wide range of waste heat sources rooting from industrial operations such as metallurgical industry, incinerators, combustion engines, annealing furnaces, drying, baking, cement production etc. ORCs are typically being applied on waste heat sources with the temperature range from 100°C up to 400°C, by being usually referred as low-temperature waste heat (100°C-250°C) and high-temperature waste heat (250°C-400°C). ORCs have a similar working principal with the conventional Rankine cycle which utilizes water or steam, however they utilize organic fluids as working fluid. Two among the most commonly used zero ozone depletion potential (ODP) working fluids for low-grade waste heat recovery are R245fa and Solkatherm® SES36. Their thermodynamic and environmental properties can be seen in the Table 1.

**Table 1:** Properties of proposed fluids

Working Fluid	GWP	ASHRAE Criteria	M. Mass (g.mol <sup>-1</sup> )	Boiling Point (K)	T <sub>crit</sub> (K)	P <sub>crit</sub> (MPa)
R245fa	950	B1	134,05	288,05	427,2	3,64
Solkatherm® SES36	3710	Non-Flammable	184,53	308,79	450,7	2,85

The case-specific working conditions are the main determining parameters for an ORC system design. The efficiency of the cycle strongly depends on the considered working fluid. A reasonable selection process can be done by taking the fluids' thermodynamic, stability, safety, legislative and environmental aspects into consideration for a particular case. Moreover, the critical temperature and

critical pressure values of the working fluid are the main criterion for distinguishing the cycle conditions (subcritical, transcritical and supercritical) of an ORC. R245fa is reported to be a suitable refrigerant for ORC applications by various researchers (Maalouf *et al.*, 2012; Saleh *et al.*, 2007; Shengjun *et al.*, 2011; Liu *et al.*, 2012). Although, there are empirical examples of promising ORC efficiency of SES36 (Mikielewicz *et al.*, 2012; Mikielewicz, 2010; Riva *et al.*, 2006; Siddiqi and Atakan, 2011; Galvez, 2009), the ORC research with SES36 is still at its infancy. With the light of its reported promising features in accordance to the aforementioned aspects, Solkatherm® SES36 is chosen as the working fluid for the present study related to the subcritical ORC evaporators.

Waste heat recovery can be performed efficiently through a direct evaporator (e.g. finned tube bundles) where the hot flue gas coming from a heat source (Ribatski and Thome, 2007) is in direct contact with the heat exchanger outer surface. However, the thermodynamic efficiency of an evaporator relies on heat transfer and pressure drops, the sizing of an evaporator needs to be performed accordingly (Quoilin *et al.*, 2013). In a design problem, the accuracy of the design method might have a significant impact on the aspects related to heat transfer and thermo-economic efficiency. A too small sized evaporator not capable to perform a complete evaporation might cause turbine or expander damage in some cases. On the other hand, a too large evaporator will cause excessive working fluid superheating, which may lead to a negative impact on system performance and a higher heat exchanger cost (Fischer, 2011). Design of evaporators is often being done by means of commercial software, where various generally applicable calculation methods are implemented, yet the methods are usually undisclosed. Moreover, the validity range of those methods usually do not overlap with present conditions. Thus, the accuracy of using different heat transfer correlations from the aspect of specific ORC evaporator design and rating parameters is not known yet. In that manner, the accuracy of in-tube flow boiling calculations might have an observable influence on the end design, even though the design of evaporator for waste heat recovery ORC applications is significantly dependent on flue gas heat carrier side. For having a concrete idea of the largest possible error margin of using various flow boiling heat transfer methods, a design sensitivity analysis is performed by means of using 13 different a priori flow boiling heat transfer correlations. The investigated correlations are listed in Table 2.

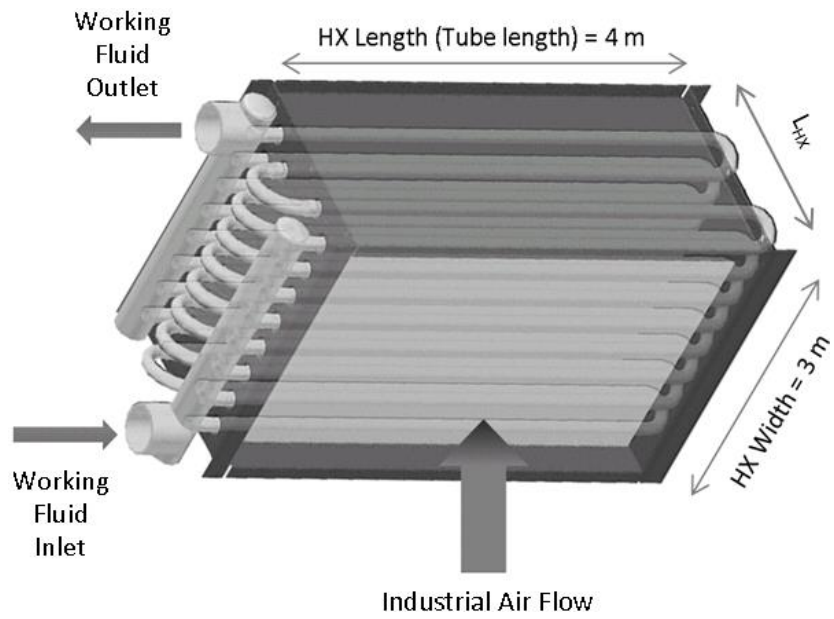
**Table 2:** Used flow boiling correlations

Author(s)	Year	Source
Kandlikar	1990	(Kandlikar, 1990)
Gungor & Winterton	1987	(Thome, 2004)
Gungor & Winterton	1986	(Gungor and Winterton, 1986)
Wattelet <i>et al.</i>	1994	(Dobson <i>et al.</i> , 1993)
Butterworth	1970	(Schlunder, 1986)
Chen	1966	(Chen, 1966)
Bennett <i>et al.</i>	1959	(Bennett and Chen, 1980)
Palen	1983	(Schlunder; 1986)
Shah	2009	(Shah, 1976)
Klimenko	1990	(Klimenko, 1988)
Liu & Winterton	1991	(Liu and Winterton, 1991)
Steiner & Taborek	1992	(Steiner and Taborek, 1992)
Chun & Seban	1971	(Chun and Seban, 1971)

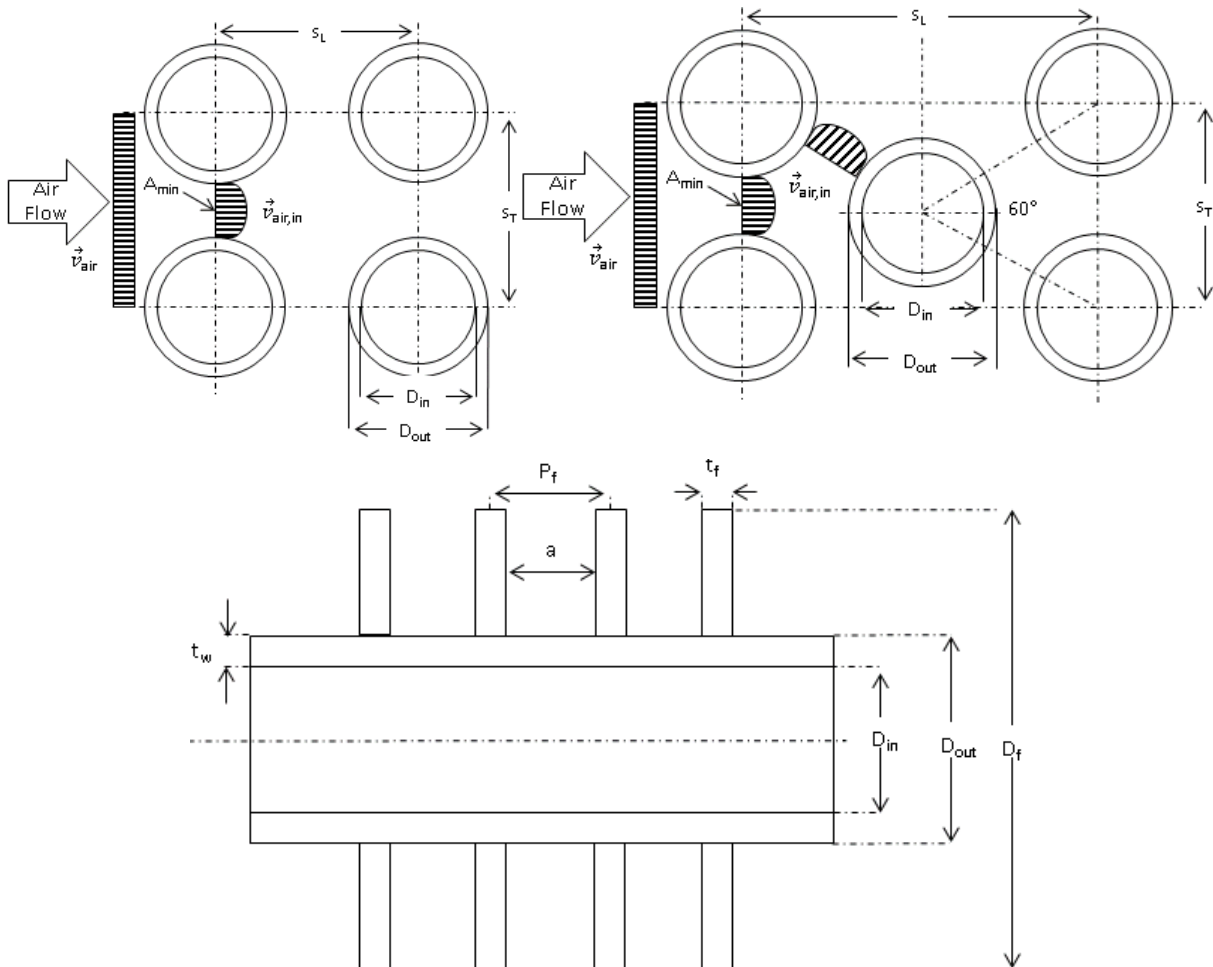
The error margin sensitivity analysis is made for a typical low-temperature ORC waste recovery case where the working fluid enters to the evaporator at 40°C and the evaporation occurs at 125°C and 1,09 MPa. The waste heat carrier fluid is industrial air and has a temperature of 280°C at the evaporator inlet. The required capacity is 4 MW. The fixed and variable geometrical parameters of the evaporator is given at the Table 3. It is important to mention that number of tubes and fin diameter changes directly with the outer tube diameter. Moreover, an illustration of the evaporator is shown in the Figure 1. Moreover, Figure 2 shows the geometric parameters of the inline and staggered layout, as well as the geometrical definitions related to the fins around tubes.

**Table 3:** Evaporator properties and parameters

$D_{out}$ (mm)	25,4 – 31,8 – 38
# of Tubes	66 – 59 – 54
$t_w$ (mm)	2,11 – 2,77
$L_{tube}$ (m)	3
$W$ (m)	4
Tube Material	Carbon Steel
$D_f$ (mm)	57 – 60,35 – 70
$t_f$ (mm)	0,4
$P_f$ (fins/m)	236 – 275 – 314 – 354 – 393 – 432
Fin Material	Aluminum
$D_{bend}$ (mm)	$3 \times D_{out}$
Tube Layout	Staggered (60°) - Inline

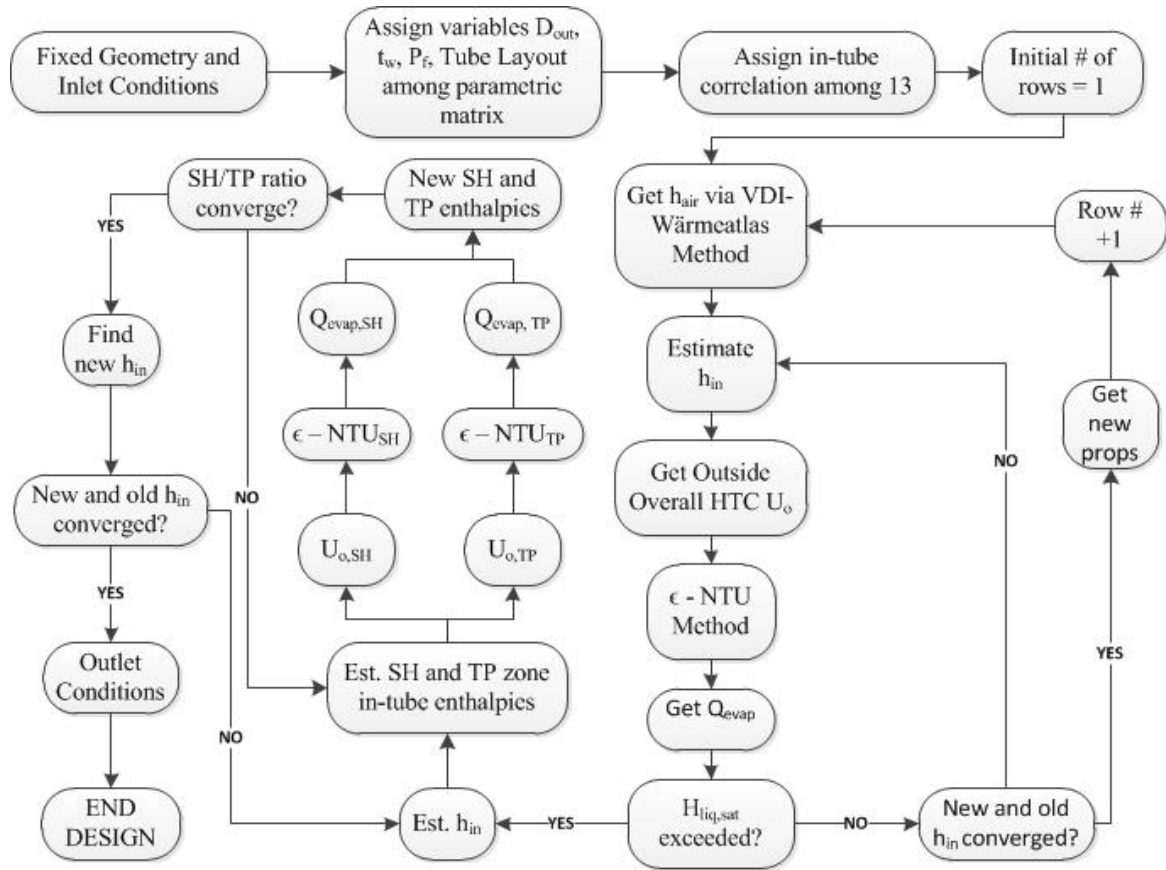


**Figure 1:** Evaporator drawing



**Figure 2:** Inline (upper left) and staggered (upper right) tube layouts; fin geometry (lower center)

## 2. DESIGN METHODOLOGY



**Figure 3:** Flow chart of iterative evaporator design methodology

A generic design methodology is implemented for finding the number of required rows in an iterative manner, and the flow of the method is illustrated in Figure 3. The method starts with single row of tubes in accordance with the diameter and heat exchanger width. The transferred heat (via  $\epsilon$ -NTU method, separate relations for superheated single-phase zone denoted as “SH” and two-phase zones denoted as “TP”) and subsequently the working fluid exit enthalpy are calculated for that particular row. The rows are increased by one incrementally until saturated (and also superheated) vapour is attained at the last tube. The vapour quality (if saturation conditions are reached) is calculated through linear ratio of latent heat and exit enthalpy at that particular tube. The air properties are calculated at each new row encountered, whereas the working fluid property change (i.e. temperature glide) is neglected due to very low pressure drops. At the end, a design solution for each combination of in-tube convective coefficient correlation, tube diameter, tube wall thickness, fin density and tube layout ( $13 \times 3 \times 2 \times 6 \times 2 = 936$  design combinations in total) is attained.

Air-side convective coefficient is found through VDI-Wärmeatlas method given for forced convection on finned tubes in cross flow (VDI-Wärmeatlas, 2010). The heat transfer equations for inline (Eqn. 1) and the staggered (Eqn. 2) layout are given as:

$$Nu_D = 0,22Re_D^{0,6} \left(\frac{A}{A_{to}}\right)^{-0,15} Pr^{1/3} \quad (1)$$

$$Nu_D = 0,38Re_D^{0,6} \left(\frac{A}{A_{to}}\right)^{-0,15} Pr^{1/3} \quad (2)$$

The in-tube convective coefficients at the subcooled zone and the superheated zone are calculated through Dittus-Boelter equation given as (for liquid and vapour phase):

$$h = 0,023Re^{0,8}Pr^{0,4} \left(\frac{k}{D_{in}}\right) \quad (3)$$

For evaluating the in-tube and outer pressure drops, Friedel correlation (Friedel, 1979) and Robinson & Briggs correlation (Thome, 2004) were used, respectively. Two-phase pressure drop at U-bends were calculated with Muller-Steinhagen and Heck correlation (Muller-Steinhagen and Heck, 1986). The fouling outside (industrial air) and inside the tubes (working fluid) are determined as 0,0004 m<sup>2</sup>K/W and 0,0002 m<sup>2</sup>K/W, respectively. The fan power is estimated as:

$$Fan\ Power = G_{air} A_{min} \frac{\Delta P_{air}}{\rho_{air}^{0.85}} \quad (4)$$

where the fan efficiency is assumed as 85%. The cost is estimated for comparative reasons and is calculated by taking European market values of carbon steel tubing and welding labor cost per U-bend into consideration, whereas the fin cost is excluded.

### 3. DESIGN SENSITIVITY ANALYSIS

The design solutions mentioned in the previous chapter are compared with each other by means of fixing all geometric parameters except tube diameter, tube wall thickness, fin density and tube layout, respectively. For each of those four variables, influence of the deviation of two-phase heat transfer coefficients  $h_{tp}$  on estimated heat exchanger cost, fan power, total transferred heat  $Q_{tot}$ , number of bends  $N_{bends}$ , total longitudinal heat exchanger length  $L_{hx}$ , total tube length  $L$ , overall heat transfer coefficient  $U$ , air-side pressure drop  $\Delta P_{air}$  and refrigerant-side pressure drop  $\Delta P_{ref}$  is assessed for each changing geometric variable. At a particular case, the error margin among the values yielded by 13 correlations were calculated for each parameter with the formula below:

$$Error\ Margin = \frac{Standard\ deviation\ of\ the\ 13\ values}{Average\ of\ 13\ values} \times 100 \quad (5)$$

The changes in the error margins are illustrated with 3D graphs and the exact values are provided at the corresponding tables underneath. At each table, maximum and minimum deviations are shown with yellow and green highlighting, respectively. The unchanging error margins and in-between values are not indicated with any color. It is important to note that the deviations of cost, total tube length, count of U-bends and longitudinal heat exchanger length are quite similar as they are directly related to each other. Table 4 and Figure 4 show the deviations of a design case having **staggered layout, SES36 as working fluid, 3 m of tube length, 4m of heat exchanger width, D<sub>out</sub>=1/2” and t<sub>w</sub>=2,11 mm** with respect to 6 values of  $P_f$  **changing between 236 - 432 fins/m**. Apparently the maximum deviation usually occurs (except  $Q_{tot}$  and  $L_{hx}$ ) when the fin density is 314 fins/m, which corresponds to a heat transfer coefficient deviation of 29,1%. The minimum deviation of convective coefficient occurs at the highest fin density (432 fins/m) as 24,6%. There is no linear but a Gaussian-like tendency in error margin change with changing fin density. Thus, it can be said that the error margins are relatively smaller at the lowest and then the highest fin density. Largest deviations occur at  $\Delta P_{ref}$  and cost (10,7% and 11,6%), where  $Q_{tot}$  is the least deviating parameter observed in all cases.

**Table 4:** Influence of  $h_{tp}$  deviation on all parameters for 6 fin density values

$P_f$	$h_{tp}$	$Q_{tot}$	Cost	U	Fan Power	L	$L_{hx}$	$N_{bends}$	$\Delta P_{air}$	$\Delta P_{ref}$
236	27,2%	3,1%	5,5%	4,0%	5,0%	4,6%	4,2%	4,3%	4,3%	10,2%
275	28,2%	2,9%	7,7%	5,6%	7,1%	6,4%	4,4%	5,9%	5,9%	9,8%
314	29,1%	3,8%	11,6%	8,3%	10,5%	9,5%	4,9%	8,7%	8,7%	10,7%
354	27,8%	4,3%	10,4%	7,3%	9,3%	8,4%	5,3%	7,7%	7,7%	10,7%
393	25,7%	4,3%	9,0%	6,3%	8,1%	7,3%	5,6%	6,7%	6,7%	9,3%
432	24,6%	3,6%	6,8%	4,7%	6,1%	5,5%	5,6%	5,0%	5,0%	7,1%

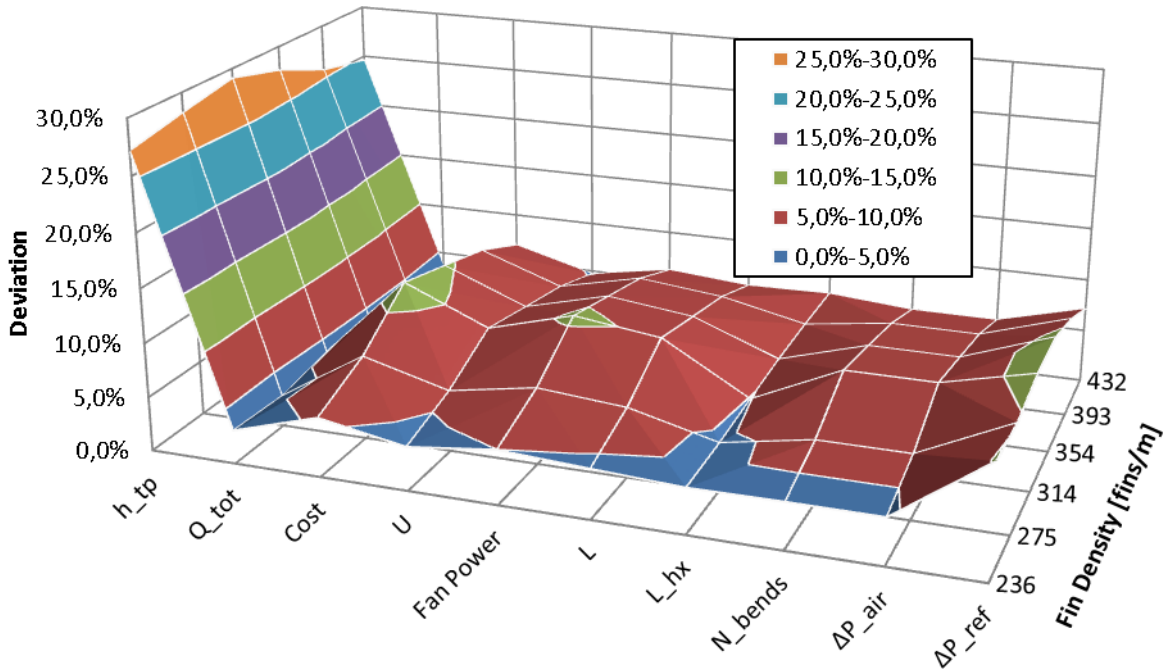


Figure 4: Influence of  $h_{tp}$  deviation on all parameters for 6 fin density values

Table 5 and Figure 5 show the deviations of a design case having **staggered layout, SES36 as working fluid, 3 m of tube length, 4m of heat exchanger width,  $D_{out}=1/2''$  and  $P_f=236$  fins/m** with respect to 2 values of  $t_w$  **changing between 2,11 – 2,77 mm**. As can be observed from the table, the change in error margin is considerably low as the tube wall thickness changes. The convective coefficients derive between 27% to 27,2%. As the wall thickness shrinks, the error margin gets somewhat larger, but at a negligible value (0,6%) at  $\Delta P_{ref}$ , where largest change in deviations occur.

Table 5: Influence of  $h_{tp}$  deviation on all parameters for 2 tube wall thickness values

$t_w$	$h_{tp}$	$Q_{tot}$	Cost	U	Fan Power	L	$L_{hx}$	$N_{bends}$	$\Delta P_{air}$	$\Delta P_{ref}$
2,11	27,2%	3,1%	5,5%	4,0%	5,0%	4,6%	4,2%	4,3%	4,3%	10,2%
2,77	27,0%	3,0%	5,5%	4,0%	5,0%	4,6%	4,0%	4,2%	4,2%	10,8%

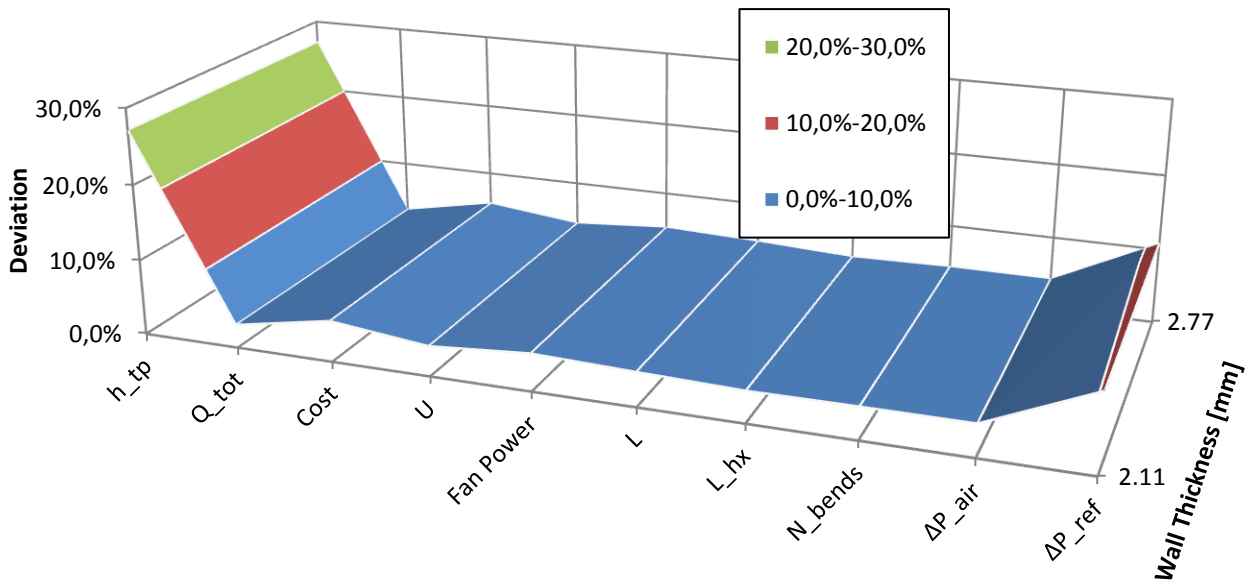
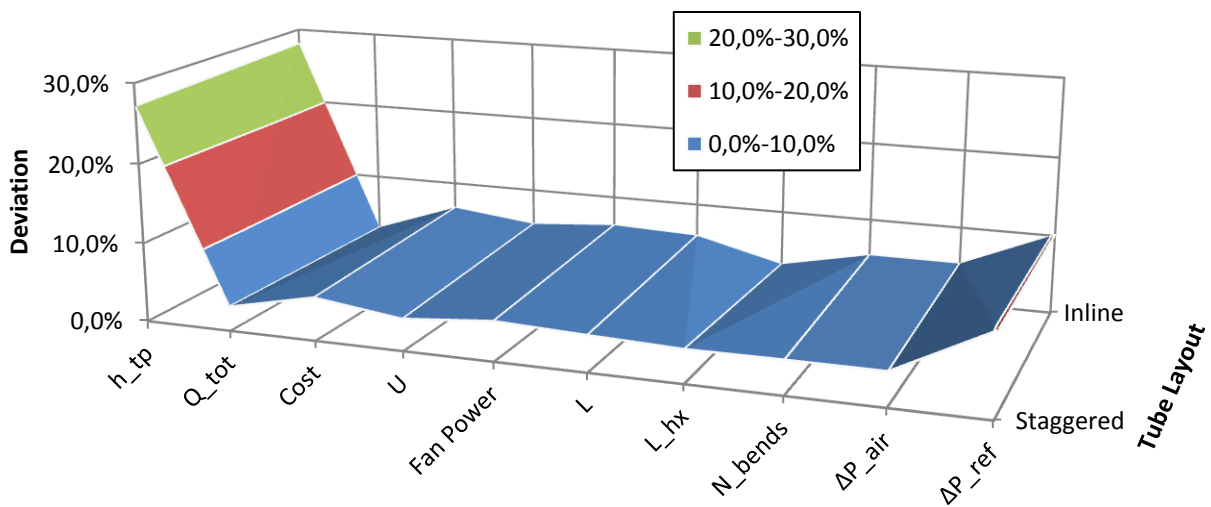


Figure 5: Influence of  $h_{tp}$  deviation on all parameters for 2 tube wall thickness values

Table 6 and Figure 6 show the deviations of a design case having **staggered layout, SES36 as working fluid, 3 m of tube length, 4m of heat exchanger width,  $D_{out}=1/2''$  and  $P_f=236$  fins/m** with respect to 2 values of  $t_w$  changing between 2,11 – 2,77 mm. Deviation of convective coefficients change between 27,2% and 28,1%. In most of the investigated parameters, the error margin increase as the tube layout is changed from staggered to inline. The largest deviation occur with  $\Delta P_{ref}$  as 10,2%. Smallest deviations are observed at  $Q_{tot}$  which changes between 2,7% and 3,1%.

**Table 6:** Influence of  $h_{tp}$  deviation on all parameters for 2 tube layouts

Layout	$h_{tp}$	$Q_{tot}$	Cost	U	Fan Power	L	$L_{hx}$	$N_{bends}$	$\Delta P_{air}$	$\Delta P_{ref}$
Staggered	27,2%	3,1%	5,5%	4,0%	5,0%	4,6%	4,2%	4,3%	4,3%	10,2%
Inline	28,1%	2,7%	6,6%	5,3%	6,2%	5,8%	3,0%	5,3%	5,3%	10,2%



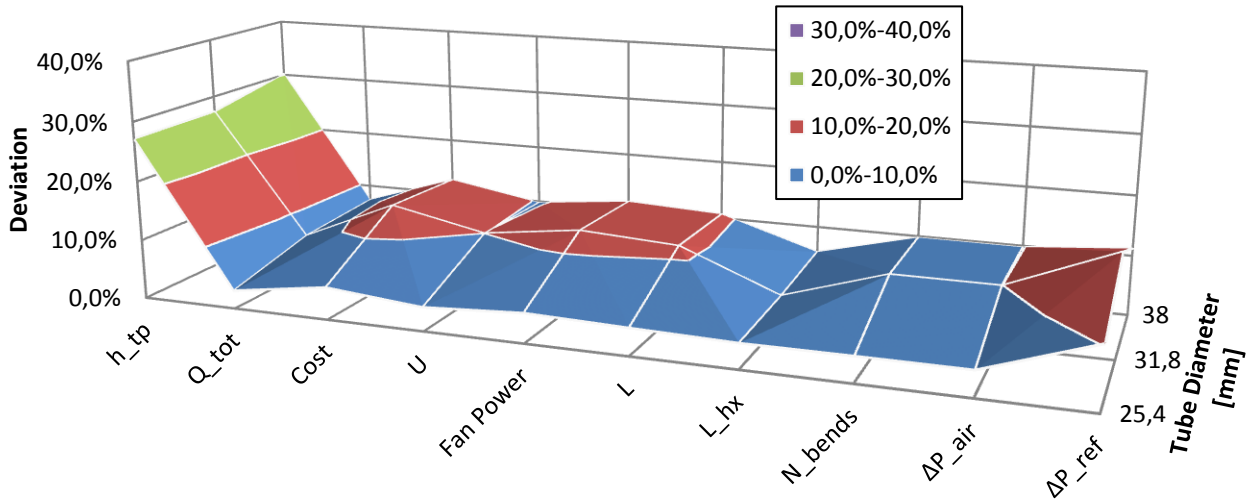
**Figure 6:** Influence of  $h_{tp}$  deviation on all parameters for 2 tube layouts

Table 7 and Figure 7 show the deviations of a design case having **staggered layout, SES36 as working fluid, 3 m of tube length, 4m of heat exchanger width,  $t_w=2,11$  mm and  $P_f=236$  fins/m** with respect to 3 values of  $D_{out}$  changing between 25,4 – 38 mm. The convective coefficients change between 27,2% and 30,2%. Among the three investigated outer diameter values, smallest diameter (25,4 mm) yields the smallest error margins at all parameters. In general, the least deviating parameter is observed to be  $L_{hx}$ , where the largest occurs at  $\Delta P_{ref}$ . There is again a non-linear but a Gaussian-like change in the error margins as the diameter increases. In other words, the largest deviations are observed at the middle-size diameter (31,8 mm).

**Table 7:** Influence of  $h_{tp}$  deviation on all parameters for 3 tube outer diameters

$D_{out}$	$h_{tp}$	$Q_{tot}$	Cost	U	Fan Power	L	$L_{hx}$	$N_{bends}$	$\Delta P_{air}$	$\Delta P_{ref}$
25,4	27,2%	3,1%	5,5%	4,0%	5,0%	4,6%	4,2%	4,3%	4,3%	10,2%
31,8	27,4%	6,4%	13,4%	10,0%	12,1%	11,2%	4,6%	9,9%	9,9%	17,0%
38	30,2%	7,3%	12,6%	9,8%	11,5%	10,7%	5,6%	9,8%	9,8%	11,2%





**Figure 7:** Influence of  $h_p$  deviation on all parameters for 3 tube outer diameters

It is important to note that Chun & Seban and Chen correlations usually underpredict the heat transfer coefficients in comparison to the other methods. Thus, they calculate larger tube lengths required for heat transfer. It can be said that those two correlations play a significant role in increasing the possible general error margin. Moreover, the deviation between all of the correlations occur due to their varying reported application ranges, conditions and own prediction errors. None of the investigated correlations were validated experimentally for Solkatherm® SES36 as well. As a result of all the mentioned reasons of error, the obtained percentage values represent the maximum error that the end design can have when a conventional flow boiling correlation is used for in-tube heat transfer calculations.

#### 4. CONCLUSIONS

For four changing geometric variables (fin density, tube wall thickness, tube layout and tube outer diameters) a design sensitivity analysis is done for investigating the influence of in-tube convective heat transfer coefficient calculated via 13 correlations on 10 thermo-hydraulic heat exchanger parameters. By including many flow boiling correlations, a general error margin in those parameters is deduced. The error margins can be regarded as the maximum error that exists in the end design of a low-temperature direct evaporator for waste heat recovery ORCs. The found conclusions include:

- Keeping the fin density as low or high as possible can help decreasing the error margin,
- Higher tube wall thickness can decrease the error margin,
- Using staggered tube layout instead of inline can decrease the error margin,
- Using smaller tube outer diameters can help decreasing the error margin,
- A high deviation of in-tube convective heat transfer coefficient usually causes a higher deviation in investigated parameters, although the changing geometric variables should be considered at the same time as well for having an idea of expected error margin,
- Using newer calculation methods can decrease the error margins in design
- In most of the cases and investigated parameters, the error margins remain under 10%, with a maximum observed error margin of approximately 30% at convective coefficients.
- More correlations can be included to the study for expanding the general applicability scope of sensitivity analysis.
- An experimental investigation is necessary for having a more accurate idea.



## NOMENCLATURE

A	area	(m <sup>2</sup> )
D	diameter	(mm)
G	mass flux	(kg/m <sup>2</sup> s)
h	convective coefficient	(W/m <sup>2</sup> K)
H	Enthalpy	(J/kg)
k	thermal conductivity	(W/mK)
L	Length	(m)
N	number of	(-)
P	pressure, pitch	(Pa, fins/m)
Pr	Prandtl number	(-)
Re	Reynold's number	(-)
Q	transferred heat	(W)
t	thickness	(mm)
T	temperature	(°C)
U	overall heat transfer coeff.	(W/m <sup>2</sup> K)
W	width	(m)

### Subscript

air	air-side
bends	U-bends
crit	critical
evap	evaporation
hx	heat exchanger
in	in-tube side
liq	liquid phase
min	minimum
out	outer-tube side
ref	refrigerant side
sat	saturation
tp	two-phase
tot	total

### Greek

$\rho$	density	(kg/m <sup>3</sup> )
$\Delta$	difference	(-)

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