DESIGN SENSITIVITY ANALYSIS OF USING VARIOUS IN-TUBE CONDENSATION CORRELATIONS FOR AN AIR-COOLED CONDENSER FOR ORCS

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ABSTRACT

The study is related to the evaluation of using 19 condensation heat transfer correlations in an annular finned horizontal round tube V-shaped air-cooled condenser design problem for a representative low-temperature waste heat recovery Organic Rankine Cycle (ORC) case. The condensation is realized through cold air provided by the fan suction at a mass flow rate of 90,35 kg/s, whereas the working fluid mass flow rate is 7,8 kg/s. The considered condensation temperature is 40°C which corresponds to a saturation pressure of 1,17 bar. The ambient air is considered to be 15°C. The investigated working fluid is SES36. For a given set of geometrical constraints, an iterative condenser design model is implemented. All considered correlations are applied separately for the same boundary conditions. The design sensitivity on the overall heat transfer coefficient, total transferred heat, required fan power, air- and refrigerant-side pressure drops is assessed. By those means, the engineering error margin of using different calculation tools in designing air-cooled condensers for ORC is reported.

INTRODUCTION

The increasing energy demand and environmental concerns in the world encourage researchers and industrialists to consider utilizing sustainable energy sources. In that manner, waste heat recovery from industrial and domestic facilities is continuously receiving more attention as a solution alternative. Organic Rankine cycles (ORCs) have been reported to have promising heat recovery efficiencies and environmental-friendly features for waste heat recovery applications [1]. ORCs have a wide range of applications 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 divided as low-temperature waste heat (100°C-250°C) and high-temperature waste heat (250°C-400°C). Unlike the fact that conventional Rankine cycle utilizes water or steam, ORCs use an organic working fluids which have much lower boiling points. Lower boiling points of working fluids allow them to operate and recuperate waste heat at much lower temperatures.

The major trade-off of using organic fluids can be their high Global Warming Potentials (GWP) and Ozone Depletion Potentials (ODP). Many of the conventional refrigerants are being (or have been) phased out within the frame of Montreal and Kyoto Protocols due to their non-environmental features. Thus, usage of novel and more environment-friendly refrigerants for vapour compression cycles are being legislatively fostered.

Various research reporting the thermodynamic cycle efficiency capabilities of various working fluids for ORCs can be found in the literature [1-4]. Table 1 shows the critical properties of some of the promising working fluids for low- and high-temperature waste heat ORC applications at subcritical conditions.

NOMENCLATURE

A	[m²]	*Area*
d	[m]	Diameter
G	[kg/m²s]	Mass flux
h	[W/m²K]	Convective heat transfer coefficient
L	[m]	Length
P	[Pa]	Pressure
R_f	[m²K/W]	Fouling thermal Resistance
Q	[W]	Cartesian axis direction
T	[K]	Temperature
U	[W/m²K]	Overall heat transfer coefficient

Special characters

Δ	[-]	Difference
λ	[W/mK]	Thermal conductivity
ρ	[kg/m³]	Density

Subscripts

air	Air side
c	Critical point
cond	Condensing
fr	Frontal
h	Hot side
i	In-tube
o	Outside of tube
ref	Refrigerant side
tot	total
tp	Two phase
w	Tube wall

Table 1 Promising working fluids for low- and high-temperature ORCs

Working Fluid	T_c (K)	P_c (MPa)
R245fa	427,20	3,640
R265mfc	460	3,266
Solkatherm® SES36	450,70	2,849
R245ca	447,57	3,940
R1233zd	438,75	3,570
n-Butane	425,20	3,922
n-Pentane	469,65	3,370
Cyclopentane	511,70	4,510
MM	518,70	1,925
MDM	564,13	1,415
Toluene	591,80	4,109

Alongside the design considerations regarding expansion means (expander or turbine) and evaporator of an ORC, the condenser design has a very important effect on the overall system performance of ORC as the effective heat transfer at heat sink reflects on the overall energy and exergy efficiency [4]. A too small sized condenser will not be capable of condensing the refrigerant completely at the outlet, which might cause pump, compressor, turbine or expander damage in some cases. On the other hand, a rather large heat exchanger might cause excessive working fluid subcooling, which may lead to a negative impact on system components (freezing etc.), cycle performance and the cost of heat exchanger.

The thermo-hydraulic design of condensers are being done by using conventional heat transfer and pressure drop correlations. However, using a general correlation for a case-specific (specific geometrical and boundary conditions, working fluid etc.) design of an ORC condenser is prone to have an error margin. The magnitude of error on an end design is yet to be revealed for a large spectrum of boundary conditions.

Present study comprises a generally applicable iterative heat exchanger design methodology for performing a design sensitivity analysis of a representative ORC condenser boundary condition case, for which an air-cooled condenser is design is performed with each of the 19 different in-tube condensation heat transfer correlations. By that means, design sensitivity of the case-specific end designs using different conventional methods and the corresponding predicted error margins on important parameters such as total transferred heat, overall heat transfer coefficient, in-tube convective coefficient, air- and refrigerant-side pressure drop and required condenser fan power are reported.

BOUNDARY CONDITIONS AND GEOMETRY

The considered case-specific boundary conditions and heat transfer media are given in the Table 2. The values are representative for a low-temperature waste heat recovery ORC application. The thermodynamic conditions of the remaining components are not mentioned in this study, as the information is redundant for the focus of present paper.

Table 2 Boundary conditions

Heat Sink	Ambient air
$T_{air,i}$ (ambient)	288°C
$P_{air,i}$ (atm)	101,325 kPa
Q_{cond}	1 MW_{th}
Working fluid	Solkatherm® SES36
T_{cond}	40°C
P_{cond}	117 kPa
$R_{f,i}$ (Ambient air)	0,0002 m²K/W
$R_{f,o}$ (Refrigerant)	0,0004 m²K/W

A V-shaped air-cooled circular-finned plain tube condenser comprising two main batteries is considered. The working fluid is transported via one manifold and divided equally into the two batteries. All the tubes have only one pass. The amount of rows is calculated iteratively. The ambient air is suctioned through the two batteries by means of axial fans located on the top. The geometrical parameters of the considered V-shape condenser is given in the Table 3, whereas an illustration is provided in the Figure 1.

Table 3 Assumed specifications of the condenser

Tube Outer Diameter (mm)	25,4
Tube Wall Thickness(mm)	2,11
One Tube Length (m)	6
Heat Exchanger Width (m)	2,5
Tube Material	Carbon Steel
Fin Diameter (mm)	57
Fin Thickness (mm)	0,4
Fin Density (fin/m)	354
Fin Material	Aluminium
Tube Layout	Staggered (60°)
Transverse Pitch (mm)	60
Longitudinal Pitch (mm)	70,45

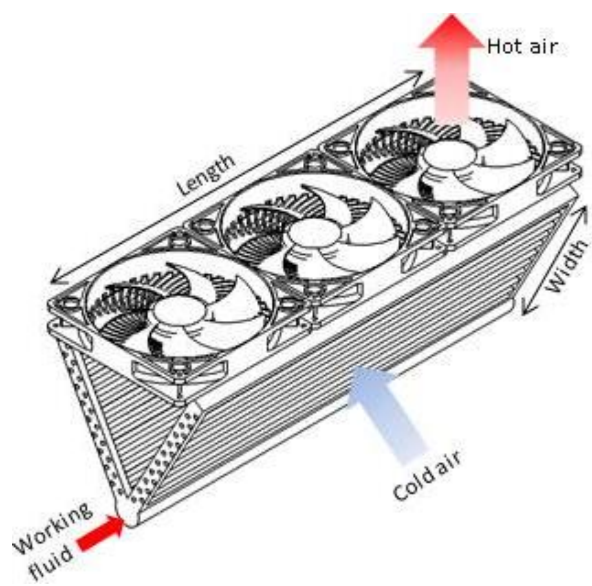

Figure 1 Condenser geometry

DESIGN METHODOLOGY

A generic design methodology is implemented for finding the number of required rows in an iterative manner. The method starts with one row of tubes, and calculates the total transferred heat Q_{tot} (via ϵ-NTU method) and vapour quality at the exit of tubes. If the exit quality is higher than zero, the number of rows is increased by one. Then the transferred heat in each row is calculated with the new flow conditions (i.e. decreased mass flow rate in tubes). The method terminates adding new rows when zero quality (and subcooling) is attained in most of the tubes. A portion of vapour flow (yet very small) might exist at the outlet of the some rows situated at the outwards end of the battery. However, the whole flow is mixed at the outlet manifold, where any remaining vapour is further condensed. For observing the sensitivity of the end designs, the same design method is applied by means of using 19 condensation heat transfer correlations for the in-tube calculations. Those correlations are listed in Table 4.

Table 4 Used condensation heat transfer correlations

Author(s)	Source
Shah (2009)	[5]
Akers & Rosson (1960)	[6]
Traviss et al. (1971)	[7]
Chato (1961)	[6]
Chen (1987)	[8]
Fujii (1995)	[6]
Tang (2000)	[5]
Cavallini & Zecchin (1974)	[8]
Koyama et al. (2003)	[9]
Bivens & Yokozeki (1994)	[10]
Dobson & Chato (1998)	[11]
Shah (1979)	[5]
Huang (2010)	[12]
Park (2011)	[12]
Moser (1993)	[12]
Cavallini et al. (2006)	[13]
Shah (2013)	[14]
Akers (1959)	[15]
Haraguchi (1994)	[12]

Moreover, the air-side convective coefficient is found through VDI-Wärmeatlas method [16]. The in-tube convective coefficients for the subcooled zone at the end of condenser tubes were calculated through Dittus-Boelter equation. For evaluating the in-tube condensation and outer pressure drops, Choi et al. correlation [17] and Robinson & Briggs correlation [18] were used, respectively. Single-phase pressure drops at subcooled sections are calculated separately though Fanning friction factors depending on the flow type. The required fan power is calculated via following equation:

$$Fan\ Power = G_{air} A_{fr} \frac{\Delta P_{air}}{0.85\ \rho_{air}} \qquad (1)$$

where the fan efficiency is assumed as 85%. The equation for overall heat transfer coefficient is given as:

$$\frac{1}{U_o A_o} = \frac{1}{A_i h_{tp}} + R_{f,i} + \frac{ln\left(\frac{d_o}{d_i}\right)}{2\pi\lambda_w L} + \frac{1}{A_o h_o} + R_{f,o} \qquad (2)$$

DESIGN SENSITIVITY ANALYSIS

The influence of the deviation of condensation convective coefficient on various parameters such as overall heat transfer coefficient, U_o, total transferred heat, Q_{tot}, fan power, air-side pressure drop, ΔP_{air} and refrigerant-side pressure drop, ΔP_{ref}. For each pair of bars, lower one represents the deviation of h_{tp}, whereas the upper one represents the deviation of the investigated parameter. All percentages are with respect to the values of Shah (2009) correlation, which is always represented with zero deviation. The smallest h_{tp} deviation happens between Akers (1959) and Moser (1993) correlations, whereas the largest is between Huang (2010) and Chen (1987) correlations. It is important to note that all correlations yielded the same amount of rows (i.e. same geometry). Figure 2 shows the deviations of U_o.

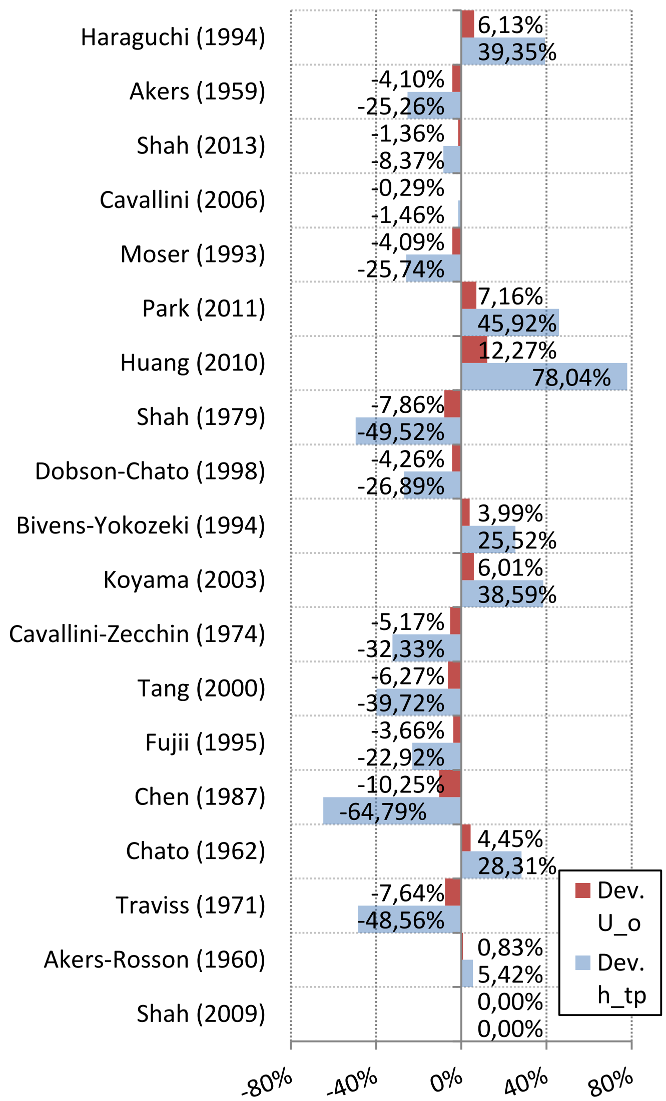

Figure 2 Deviation of U_o with respect to the deviation of h_{tp}

The largest deviation is 22,52%, which occurs between Huang (2010) and Chen (1987) correlations. Apparently, an error margin less than 60% in the condensation convective heat transfer coefficients yields less than 10% error margin in U_h. The smallest error margin occurs between Akers (1959) and Moser (1993) correlations.

Figure 3 represents the deviation of total transferred heat Q_{tot} from the condensing working fluid to the cooling air. The maximum deviation observed is 16,48% and occurs between Huang (2010) and Chen (1987) correlations, where the largest deviation of h_{tp} occur. Similarly to the previous results, the smallest difference happens between Akers (1959) and Moser (1993), which is 0,01%. As can be observed, behaviors of Q_{tot} are significantly similar to U_o due to the fact that they are correlated through ϵ-NTU method.

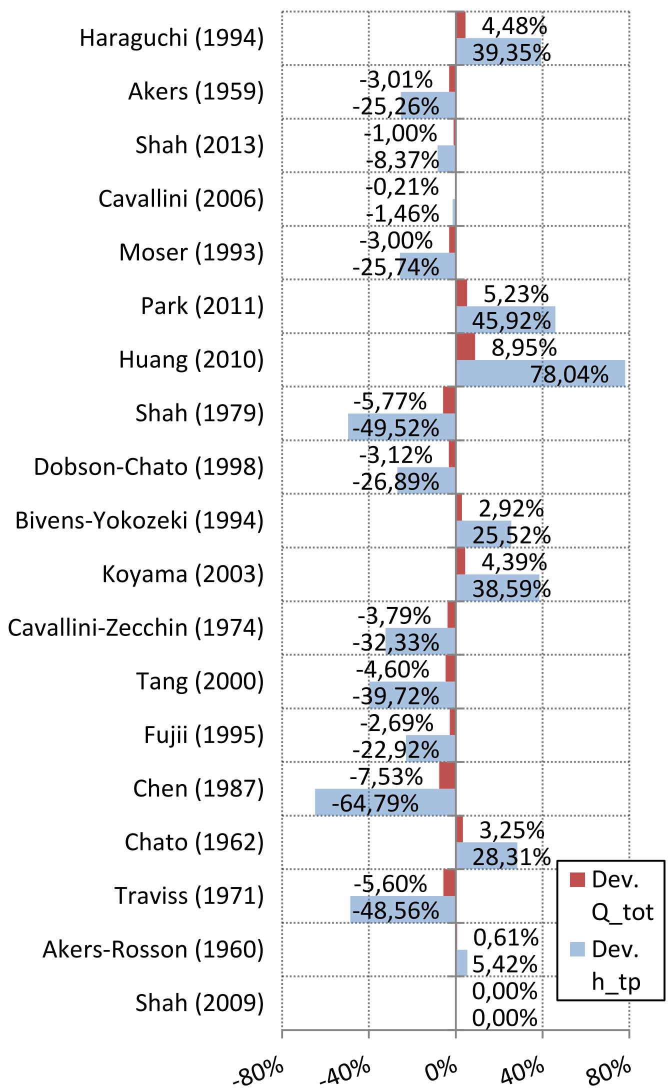

Figure 3 Deviation of Q_{tot} with respect to the deviation of h_{tp}

Figure 4 shows the deviations of required fan power with respect to changing h_{tp}. The maximum deviation occurs again between Huang (2010) and Chen (1987) correlations and is 0,73%. Some of the correlation pairs like Fujii (1995)-Moser (1993) and Dobson-Chato (1998)-Akers (1959) calculate the same value of fan power. Apparently, even a largely deviating convective coefficient has a negligible effect on the end calculation of required power. This is due to the indirect relationship of convective heat transfer coefficient and air-side pressure drop.

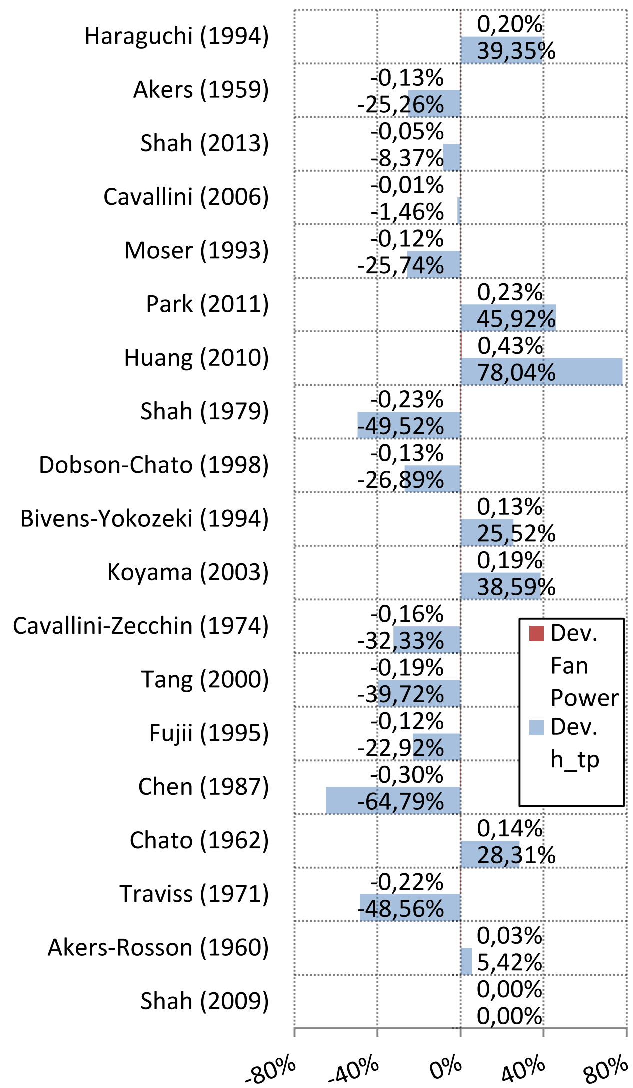

Figure 4 Deviation of fan power with respect to the deviation of h_{tp}

As expected, Figure 5 shows that the deviations of air-side pressure drop is quite similar to the fan power deviations. Akers (1959), Moser (1993), Dobson-Chato (1998) and Fujii (1995) correlations yield the same value, as well as Koyama et al. (2003) and Haraguchi (1995) correlations. The largest deviation happens again between Huang (2010) and Chen (1987) and is 0,4%.

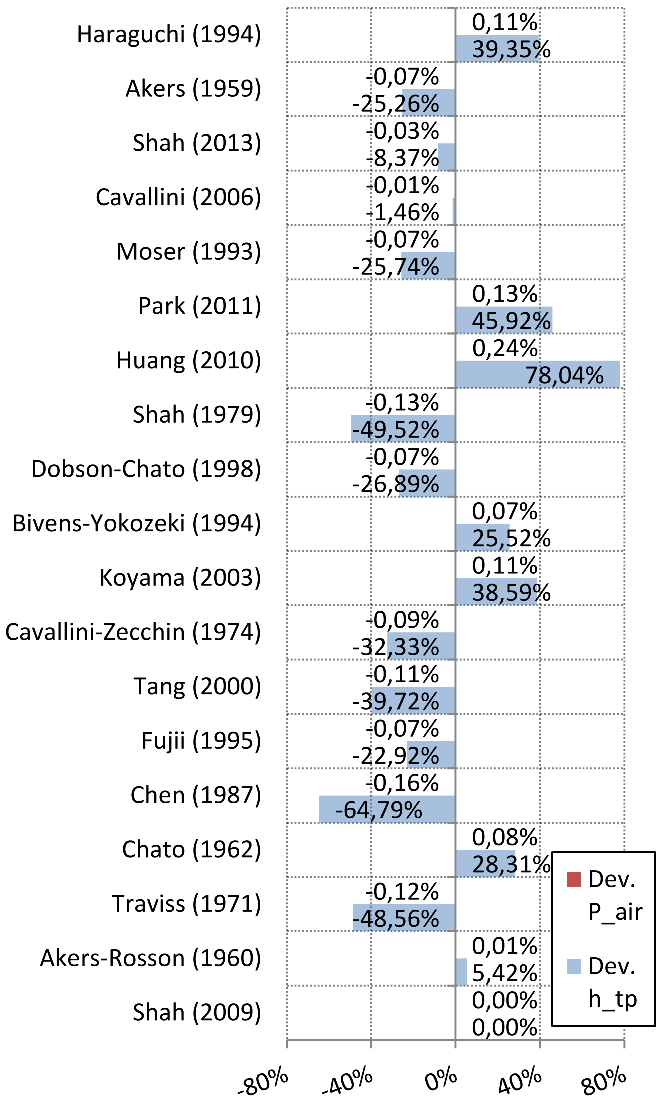

Figure 5 Deviation of ΔP_{air} with respect to the deviation of h_{tp}

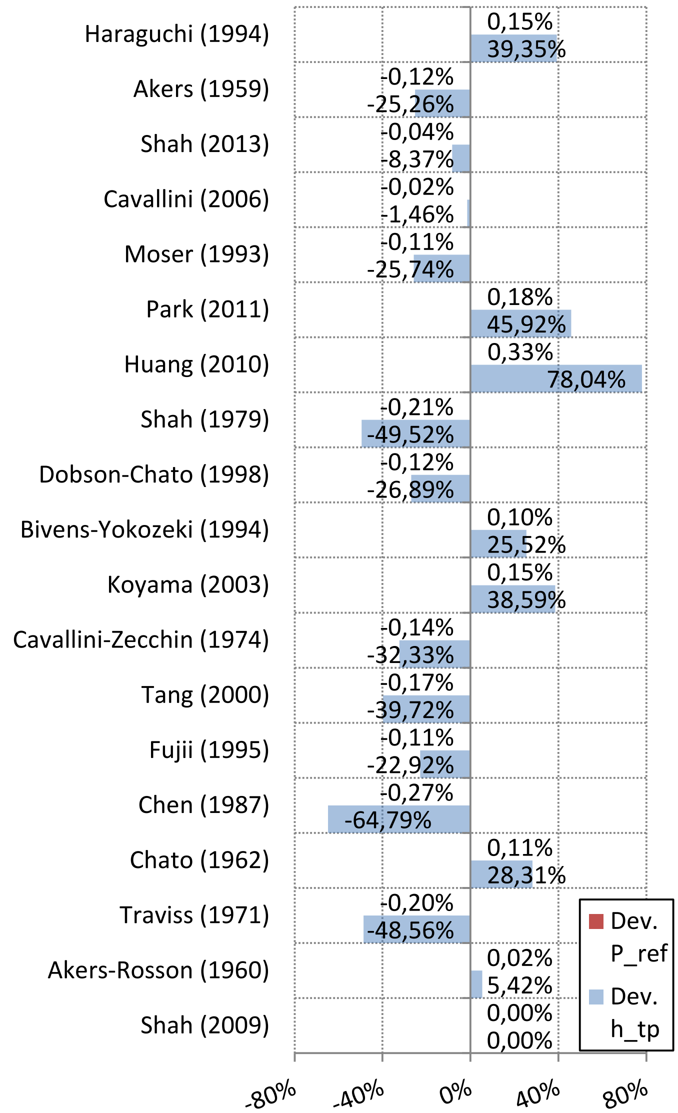

Figure 6 Deviation of ΔP_{ref} with respect to the deviation of h_{tp}

Figure 6 shows the deviation of refrigerant-side pressure drop with respect to convective condensation coefficient. Due to the fact that all designs yielded the same heat exchanger geometry, the deviations remain to be minor. The maximum deviation is 0,6% and occurs again between Huang (2010) and Chen (1987) correlations. On the other hand, Akers (1959)-Dobson-Chato (1998) and Moser (1993)-Fujii (1995) pairs yield the same values.

CONCLUSION

A design sensitivity analysis is performed through the evaluation of using 19 condensation heat transfer correlations in an air-cooled annular finned horizontal round tube V-shaped air-cooled condenser design problem for a representative low-temperature waste heat recovery Organic Rankine Cycle (ORC) case. The engineering error margins of using different calculation tools in overall heat transfer coefficient, total transferred heat, required fan power, air- and refrigerant-side pressure drops are reported. The findings are as follows:

- The error margin occurs in the calculation of overall heat transfer coefficient is between 0,01%-22,52%,
- The error margin occurs in the calculation of total transferred heat is between 0,01%-16,48%,
- The error margins that occur in the required fan power, air-side and refrigerant-side pressure drops are very low (<0,73%),

- When the accuracies of the investigated correlations are also considered, the methods can be used interchangeable for given conditions,
- For validation of the error margins, experimental investigation of same conditions is necessary,
- Method can be applied to different conditions as well.

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