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ScienceDirect

Energy Procedia 161 (2019) 300–308

Energy Procedia

www.elsevier.com/locate/procedia

2nd International Conference on Sustainable Energy and Resource Use in Food Chains, ICSEF 2018, 17-19 October 2019, Paphos, Cyprus

Experimental and CFD investigation of overall heat transfer coefficient of finned tube CO_2 gas coolers

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Abstract

The overall heat transfer coefficient of two CO_2 gas coolers was investigated through experiment and Computational Fluid Dynamics (CFD). The CFD modelling provided prediction accuracy for the overall heat transfer coefficient with a maximum error of 9% compared to the CFD predictions. Comparing the two gas cooler designs, and from the experimental and modelling results it has been shown that the performance of the gas cooler can be improved by up to 20% through optimization of the circuit design of the gas cooler. A horizontal slit between the 1st and 2nd row of tubes of the gas cooler can increase the overall heat transfer coefficient by 8% compared with the a fin without the slit.

Keywords: CO_2 refrigeration system; finned tube heat exchanger; gas cooler; overall heat transfer coefficient (U-LMTD); CFD.

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1876-6102
10.1016/j.egypro.2019.02.096

Nomenclature	
Air-off	air-side outlet heat exchanger (ºC)
Air-on	air-side inlet heat exchanger (ºC)
Ao	heat transfer surface area (m^2)
barg	pressure-gauge (bar)
h	enthalpy (kJ/kg)
LMTD	log mean temperature different (K)
$\dot{m}$	mass flow rate (m/s)
P	pressure (bar,Pa)
Q	heat rejection (W, kW)
R-744	CO_2 refrigerant
T	temperature (ºC)
U	overall heat transfer coefficient (W/m^2K)
u, v, w	velocity (m/s)
Greek symbols	
ρ	density (kg/m^3)
Δ	change in respective parameters
μ	dynamic viscosity (N•s/m²)
Subscripts	
ref	refrigerant (R744)
in	inlet
o	outlet
i	inner, inlet

1. Introduction

The gas cooler in a CO_2 refrigeration system has an important influence on the performance of the system in the transcritical region because of the high exergy loss in the gas cooler [1]. To reduce the thermodynamic losses the refrigerant exit temperature from the gas cooler should approach that of the coolant inlet temperature [2]. Generally, three major factors which affect the performance of supercritical CO_2 refrigeration systems are the design of the gas cooler, gas cooler pressure and gas cooler outlet temperature [2]. The most common type of gas cooler is the finned tube heat exchanger due to its good reliability, low resistance to air flow and manufacturing flexibility [3]. However, the heat exchanger needs to be further improved for better overall refrigeration system efficiency [4].

The design of finned-tube heat exchangers has a considerable impact on the overall heat transfer performance of the heat exchanger. The fin thickness, surface topology of the fins, fin and tube materials, the spacing and dimensions of the tubes and fins are essential parameters of the design [5]. Several design improvements were identified by previous researchers through experimental and modelling works. Tahsen et al. [6], Huang et al. [7], Chen and Lai [8] obtained the optimum spacing of tube-to-tube and fin-to-fin for maximum overall heat conductance (heat transfer rate). In addition, the staggered tube arrangement is better than the in-line tube arrangement under fixed air velocity and fin pitch conditions since higher heat transfer coefficients can be obtained. The average heat transfer coefficient on the fin increases with the air velocity and the difference between the ambient and tube temperature. It is therefore important to determine an optimum design for finned tube heat exchangers for specific applications [9]. For CO_2 gas cooler heat exchangers, a slit fin and a certain number of row and circuit combination was proposed for gas cooler improvement design, Zilio et al. [10]. Singh et al. [11] proposed a finned tube heat exchanger model improvement with a cut fin configuration and validated it with experimental results. It was found that gas cooler performance increased by 6%-12% with a slit fin design. Ge et al. [12] calculated the effect of row and circuit number combination on the performance of a fined tube gas cooler. It was reported that the number of rows and circuits has a significant effect on gas cooler performance.

U-LMTD is one parameter that can be used for heat exchanger performance evaluation alongside the Effectiveness –NTU method. The U-LMTD method also provides a more simple analysis and with CFD modelling enables the investigation of the heat transfer coefficient segment by segment [13,17].

This study involves experimental investigations and Computational Fluid Dynamics (CFD) modelling to evaluate the overall heat transfer coefficient of two gas cooler designs: gas cooler A (3 row-4 circuit) and gas cooler B (2 row-2 circuit) with and without a slit fin configuration. Furthermore, CFD allows data to be obtained for areas difficult to access in experimental investigations (such as within pipes or narrow sections) and avoid the physical disruptions caused by sensors.

2. Methodology

In this study, the overall heat transfer coefficient and Log Mean Temperature Difference (U value-LMTD) are investigated with both experimental and model methodologies. The experimental part is based on a CO_2 refrigeration system and employed a specific design of gas cooler test rig. The gas cooler type is a finned tube with a fan air cooling system. The U value-LMTD was investigated experimentally for the entire gas cooler. The second part of the research consisted of modelling using Computational Fluid Dynamics (CFD) to simulate the finned and tube of the gas coolers. The simulation considered individual segments of the heat exchanger geometry, where one segment consists of two fins, all tubes with refrigeration flow and air flow model in the fin gap. Two types of gas cooler design are investigated in this study; gas cooler A and gas cooler B which has specification of 3 row- 4 circuit and 2 row-2 circuit, respectively

The inlet and outlet temperatures of working fluid in gas coolers were specified by experimental procedures and CFD in the segment, so it is easy to determine overall heat transfer coefficient using the U-LMTD method and investigate the heat transfer coefficient profile at each segment in the gas cooler. The overall heat transfer coefficient (U-value) and LMTD are calculated using equations 1 - 4 as follows:

$$U = \frac{Q}{A_0 \Delta T_{LMTD}} \tag{1}$$

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \tag{2}$$

Where, Q is heat rejection in gas cooler and defined using equation as follow:

$$Q = \dot{m}_{air} \Delta h_{air} \tag{3}$$

And ΔT_1 and ΔT_2 are defined as follows:

$$\Delta T_1 = T_{ref,i} - T_{air,o} \text{ and } \Delta T_2 = T_{ref,o} - T_{air,i} \tag{4}$$

3. Experimental setup

3.1. Experimental test facilities

The measurements recorded during the tests included pressure, temperature and mass flow rate on the R-744-side and velocity, pressure drop and temperature on the air side. The schematic diagram of CO_2 refrigeration in which the gas cooler employed is shown in Fig.1.

The K-type thermocouples used had an uncertainty smaller than ± 0.5°C, the Danfoss MBS333® pressure transducers with measuring range of 0 -160 bar had an uncertainty of ± 0.3%, the Optimass-3000® mass flow meter had an uncertainty of ± 0.035%, and the TSI Velocicalc® Plus 8386A hot-wire air velocity meter measured within the range of 0 m/s to 50 m/s with uncertainty of ± 3 %. To enable the information to be read and recorded, the instrumentations were connected to a data logging system.

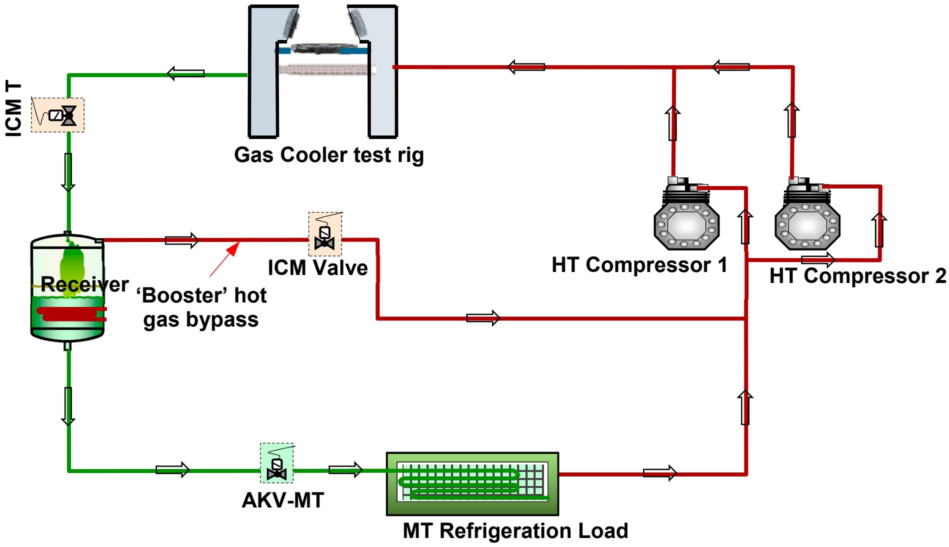


Fig.1. Schematic diagram of CO_2 refrigeration test rig.

3.2. Test results

The testing for gas cooler A was done with air-flow rates across the coil of 2000 l/s, 2400 l/s, 2800 l/s, or air velocity 1.7 m/s, 2.0 m/s, 2.4 m/s, respectively. In addition, tests for gas cooler B were done with 1600 l/s , 2000 l/s, 2400 l/s, 2800 l/s which correspond to 1.3 m/s, 1.7 m/s, 2.0 m/s and 2.4 m/s air velocity. The air temperature onto the coil was varied between 30°C and 35°C to get the supercritical mode. The experimental results for the gas cooler are shown in Fig.2 and Table 1. These results weree also subsequently used to establish the CFD inlet boundary conditions.

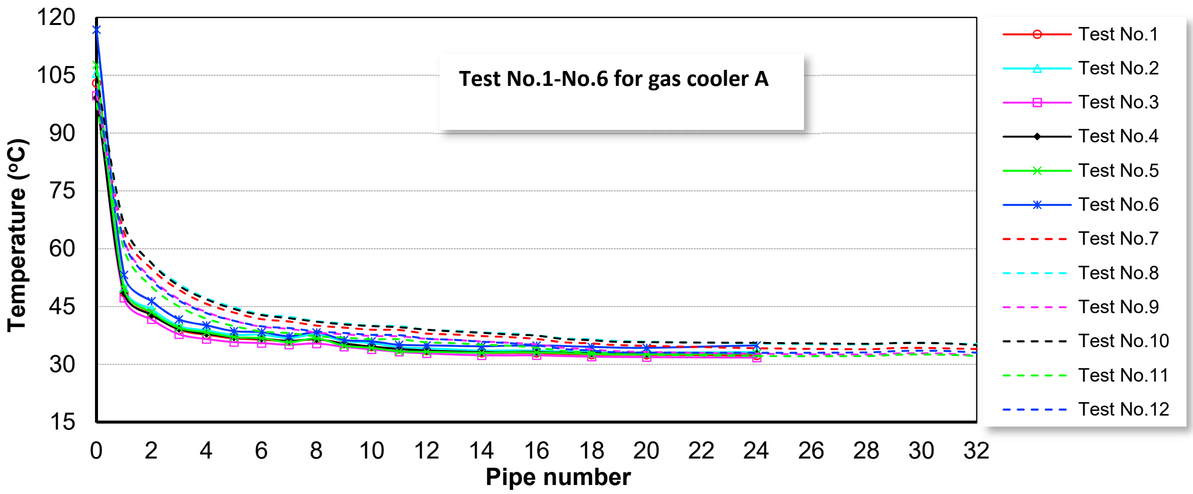


Fig. 2. Coil tube temperatures for gas cooler A and gas cooler B
(Note: conditions similar to No. 1 in Table-1)

Table 1. Representative gas cooler A and gas cooler B test results for different air-on temperatures and velocities at supercritical condition

Gas cooler A								
No.	*v* (m/s)	$T_{air\ on}$ (°C)	P_{ref_in} (barg)	T_{ref_in} (°C)	$\dot{m}_{ref}$ (kg/s)	T_{air_off} (°C)	T_{ref_out} (°C)	Q (kW)
Test No.1	1.7	32.2	82.8	102.9	0.041	36.2	32.3	9.4
Test No.2	1.7	32.8	85.1	105.5	0.042	36.9	33.2	9.6
Test No.3	2.0	31.8	81.4	99.7	0.039	35.0	31.8	8.9
Test No.4	2.0	32.8	84.2	99.2	0.040	35.9	32.8	8.9
Test No.5	2.4	32.4	85.4	107.7	0.038	35.1	32.8	9.0
Test No.6	2.4	34.3	86.6	116.8	0.041	37.2	34.9	9.6
Gas cooler B								
No.	v (m/s)	$T_{air\ on}$ (°C)	P_{ref_in} (barg)	T_{ref_in} (°C)	$\dot{m}_{ref}$ (kg/s)	T_{air_off} (°C)	T_{ref_out} (°C)	Q (kW)
Test No.7	1.7	33.7	84.9	100.3	0.042	37.6	34.0	8.9
Test No.8	1.7	35.1	86.3	100.8	0.038	38.7	35.3	8.2
Test No.9	2.0	32.6	82.5	100.2	0.039	35.7	32.3	8.6
Test No.10	2.0	35.2	86.5	104.6	0.043	38.5	35.0	9.2
Test No.11	2.4	32.0	81.5	97.6	0.042	34.9	32.2	9.2
Test No.12	2.4	33.0	83.9	101.3	0.042	35.9	33.0	9.3

4. Computational Fluid Dynamic (CFD) model

4.1. CFD governing equations

The equations governing the flow and associated heat transfer in a fluid are based on the conservation principles of mass, momentum and energy. These fundamental physical principles are expressed as the Navier-Stokes set of equations (Eq. 5-7), and because they are non-linear second-order equations, the solution procedure is complex [18]. CFD applies and solves the discretised form of these equations for a domain, through iterations, where the pressure (p), temperature (T), density (ρ) and velocity components (*u, v, w*) at each grid cell can be predicted with high accuracy. Convergence of a solution is obtained after the residuals between successive iterations are within the limits defined in the solver [19].

Continuity equation:

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_j}\left(\rho u_j\right)=S_M \tag{5}$$

Momentum equation:

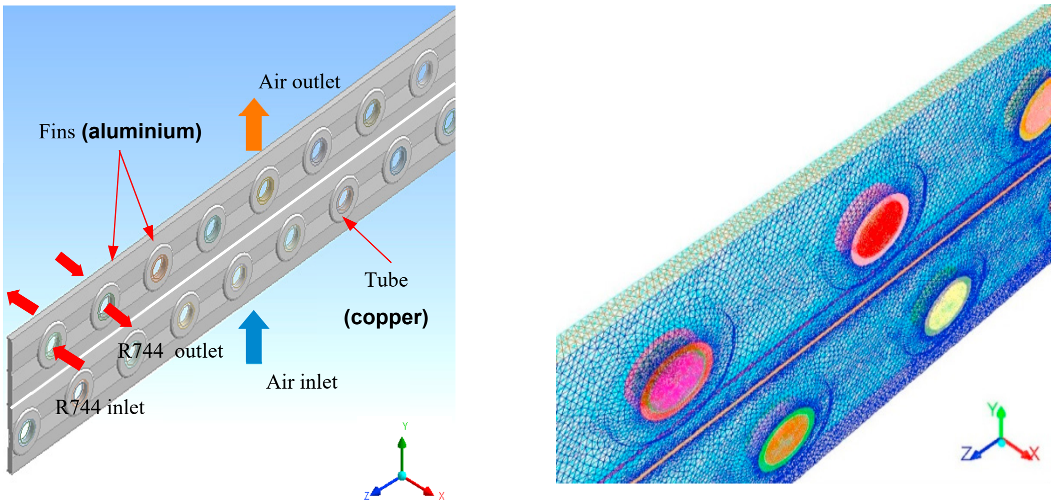
$$\frac{\partial}{\partial t}\left(\rho u_j\right)+\frac{\partial}{\partial x_j}\left(\rho u_i u_j\right)=-\frac{\partial}{\partial x_j}\left(P\right)+\frac{\partial}{\partial x_j}\left(\overline{\tau}\right)+\rho g_j+F_j \tag{6}$$

Energy Equation:

$$\frac{\partial}{\partial t}\left(\rho H\right)=-\frac{\partial}{\partial x_j}\left(\rho u_j c_p T\right)+\frac{\partial}{\partial x_j}\left[\lambda\frac{\partial T}{\partial x_j}\right]+S_E \tag{7}$$

4.2. CFD boundary conditions and meshing

The heat transfer coefficients are crucial parameters to assess the heat exchanger performance, and the model was designed to enable the investigation of overall/total heat transfer coefficients at each segment, for individual tubes. The boundary condition is shown in Fig.3 (a). The refrigerant inlet mass flow rate and temperature at inlet to each tube were input to the model. A linear variation of temperature was assumed for the tube segments as shown in Fig. 2. The air enters between two fins (y-direction), at a constant velocity and temperature obtained from the experiments (see Table 1). The fins and fin collar were modelled as thin-walls. The thermo-physical properties (density, viscosity, specific heat capacity, thermal conductivity) of air and refrigerant (R744) as a function of temperature and pressure were obtained using the Engineering Equation Solver (EES) software [20]. These were incorporated in FLUENT® using the piecewise-linear formulation. The thermo-physical properties of copper and aluminium were obtained from the FLUENT® database.



(a) **(b)**

Fig.3. CFD geometry and boundary approach: (a) segment positions consideration, (b) model geometry and boundary condition, and (c) meshing i.e, gas cooler B with slit fin.

The model was meshed using tetrahedral type elements and three different numbers of cells. The mesh sensitivity analysis was performed with respect to the residual convergence of the models. Using the coarse (1.2 million cells), medium (3.2 million cells) grids for gas cooler-A, and coarse (0.8 million), medium (2.1 million cells) for gas cooler-B, the residuals' convergence reached to a minimum of 10-4 for continuity, 10-7 for energy, 10-3 for x, y and z, 10-3 for k and 10-2 for ε, whilst the fine grid were found to have residuals in the order of 10-5, 10-8, 10-6, 10-4 and 10-4, respectively. Following the satisfactory residuals obtained from the fine grid, the latter was used for subsequent simulations. However, this more refined grid also involved a higher computing time. The final mesh (i.e. gas cooler B) is shown in Fig.3 (b), whereby high grid densities have been used in all areas where high temperature gradients were more likely to occur such as the fin collars and the close surroundings of the tube.

4.4. Model validation against experimental results

The *k-ε* turbulence models were found to have better performance for both the heat released with relative error (%) and air-outlet temperatures with absolute error (°C) as follows: Standard k-ε: (8.7%, 0.49°C errors; RNG k-ε: 7%, 0.2°C errors); Realizable k-ε: 5.9%, 0.15 errors) the k-ω models showed slightly worse performance (Standard k-ω: 9.3%, 0.6°C errors and SST k-ω: 9.5%,0.6°C errors) compared to the k-ε models; whilst the laminar model had errors of 38.3%, 2.6°C. Hence, as the Realizable k-ε model showed the best performance, it was adopted for subsequent simulations.

The models resulted in a maximum error of 10% for heat rejection rate, relative to the experimental heat rejection in the gas cooler, and a maximum absolute error of 1.5°C in the air-off temperature. However, the mean heat rejection rate error was found to be 4.7%, and the mean air-off temperature was 0.57°C. Hence for the purpose of this study, as the mean temperature error is similar to the uncertainty of the thermocouples and the relative mean error for the heat rejection rate is approximately 5%, the simulation results are considered valid.

4.5. CFD model post processing for U-LMTD calculation

Fig.4 shows the temperature contour of the CFD model in which the fin design is a horizontal slit mid-way between the top row and the middle row. It can be seen that gas cooler A comprised of 24 segments and gas cooler B of 32 segments. The performance was similar for each circuit, so one circuit could be used for the simulations.

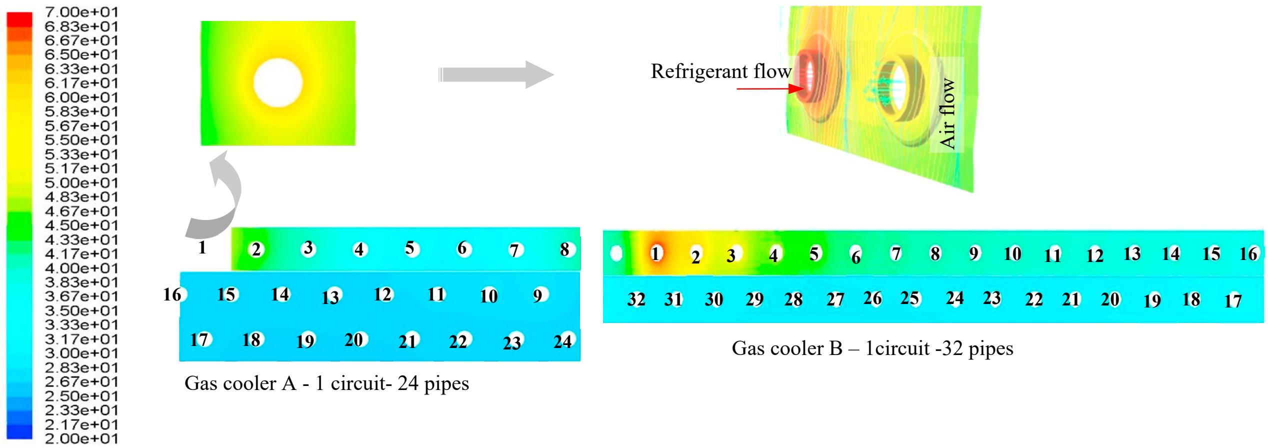


Fig.4. Temperature contour and segment based on the pipe number in one circuit

5. Results and discussions

Fig.5 presents the average U-value with respect to air velocity (m/s), obtained from the CFD modelling and experimental results. The overall heat transfer coefficient increases as air velocity increases, due to an increase in the Reynolds Number. An increase in Reynolds Number implies that more energy will be transferred from the refrigerant due to higher bulk movement (convection) of the air.

The CFD modelling has shown the overall heat-transfer coefficient of gas cooler A to vary between 638 W/m^2K – 665 W/m^2K with the air velocity varying from 1.7 m/s to 2.4 m/s. For gas cooler B, the heat transfer coefficient was found to vary between 438 W/m^2K – 558 W/m^2K with the air velocity varying from 1 m/s to 2.4 m/s. The gas cooler geometry with higher number of rows and circuit combination (2 rows -2 circuits to 3 rows - 4 circuits) led to increases in the overall heat transfer coefficient of the gas cooler by up to 20%. In addition, with horizontal slit fin, the performance of the gas cooler increased by 8%.

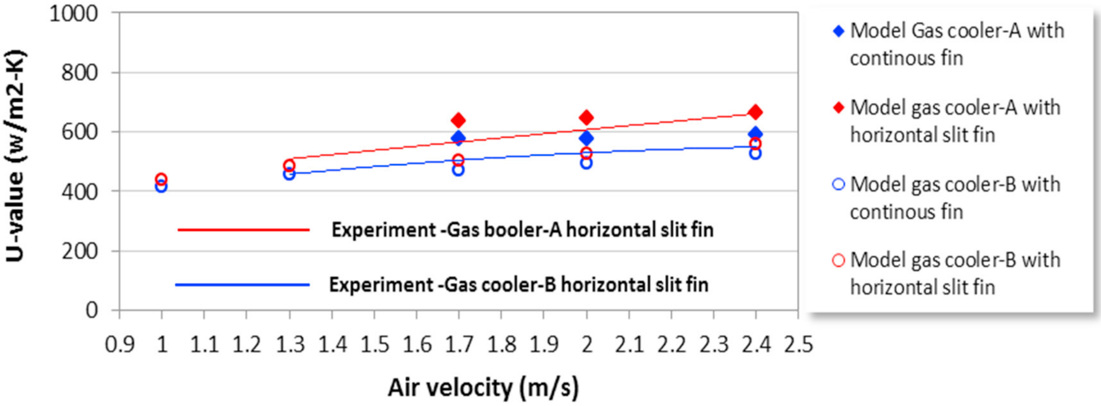


Fig.5. Variation of average U-value with air velocity of gas cooler-A and B

6. Conclusion

The experimental work and Computational Fluid Dynamics (CFD) model were developed and used to study the overall heat transfer coefficient (U-LMTD) in CO_2 gas coolers. The CFD model design considered a segment of gas cooler which comprised of two fins, all number of pipes, air flow and refrigerants flow, this was due to time and computer capacity effectiveness. The model was validated against experimental data with respect to heat rejection rate and air-off temperature. Prediction errors of less than 10% were obtained for the heat rejection rate, whilst the mean temperature errors were found to be within the uncertainties of the thermocouples employed in the experiment. The U-LMTD from experimental results for a circuit of the entire gas cooler is compared with the U-value and LMTD in segment obtained from the model. These investigations have shown that the overall heat transfer is significantly influenced by gas cooler design (i.e. configuration of the tube circuit and slit fin design). This investigation also has shown that up to 20% better efficiency can be obtained through circuit design optimisation. The slit fin arrangement can increase the coil performance by 8%.

Acknowledgements

This study was supported by the DRPM RISTEK DIKTI, Ministry of Research, Technology and Higher Education – Indonesian Government, Funding No: SP Dipa-042.06.1.401516/2018 and the RCUK Centre for Sustainable Energy use in Food chains (CSEF) of the Research Councils UK Energy programme, Grant No. EP/K011820/1. All data used in the analysis are presented in the paper. Any additional information and data can be obtained from the corresponding author of the paper.

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