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Hopefully we will get a quick response, thank you for your consideration for this manuscript.

Best Regards,

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Optimized Design of Tube Heat Exchanger for Food Industry– A Numerical Study

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ABSTRACT: This work reports the design optimization of a simple configured tube heat exchanger, which is widely employed and integrated in the stove system in small scale food processing industry in Indonesia. Optimization parameters were the tube diameter and the flow rate of working fluids, which later corelated to the maximum performance of tube heat exchanger reflected by its effectiveness (ε). The study was based on the numerical investigation using Computational Fluid Dynamics (CFD) modeling with viscous model of k-ε RNG. The tube diameter was varied by 0.5, 1, and 1.5 in, and the flow rate was varied by using initial inlet velocity of 1.6, 3.6, and 5.6 m·s^{-1}. The results show that the best ε of tube heat exchanger as high as of 1.36 can be achieved using the tube diameter of 0.5 in with the slowest inlet velocity (1.6 m·s^{-1}). This result is manifested by the higher temperature difference between the outflow and inflow in the tube heat exchanger, and the considerably low pressure drop amongst all variation. Therefore, this study recommends food processing industry to redesign the existing tube heat exchanger in order to increase the production efficiency.

INTRODUCTION

Heat exchangers render a very extensive technology that can be easily found in the food and beverage industry as well as other processing and manufacturing industry. In principle, heat exchanger is an apparatus that exchanges heat from a flowing fluid to another flowing fluid through either direct or indirect contacts [1]. In addition, heat exchanger also enables transferring heat between two or more fluids which are separated by solid surfaces or between solid particulates and a fluid, at different temperatures and in thermal contact [2]. In daily life, heat exchanger can be found in car radiator, condenser of an air conditioner, heat exchanger in refrigerator, and others. A typical heat transfer mode involves an indirect contact type heat exchanger which the heat is transferred by convection way of one fluid to a separating wall, then transferred in conduction way through the wall and transferred to other fluid in convection way [3]. Moreover, the tube type heat exchanger can also be

used as a preheater system to get advantage from the heat generated by the burner in the food processing industry.

In the context of its application in food processing industry especially in Indonesia, a simple tube heat exchanger has been developed and integrated into a traditional firewood stove that is widely used in the home industry of tempe, *i.e.* Indonesian traditional soy product made from fermented soybeans [4,5]. Fig. 1a depicts the conventional firewood stoves for boiling the soybeans while the tube heat exchanger is integrated into a modified firewood stove in Fig. 1b. As shown, the tube heat exchanger was used to harvest the heat irradiated from the burning firewood and to thermally drive a natural convection of water inside the tube that is accumulated into hot water reservoir. This hot water reservoir was employed to boil another batch of soybeans and hence, improves the process efficiency as well as cuts the production costs. Improvement of the tube heat exchanger integrated firewood stove was studied by Hantoro and co-workers using computational fluid dynamic (CFD) approach [4,5] which includes the variation of tube number, tube materials, and the stove wall. In this study, it was found that heat exchanger with 6 iron-based tubes is considered technically and economically viable for substitution of 4 copper-based tubes that demand high cost of raw materials and processing.

Figure 1. (a) A firewood stove for a traditional soybean processing in Surabaya, East Java. (b) An improved firewood stove with two heating/burning outlets and additional tube heat exchanger installed in a firing chamber. Images modified from the study reported in literature [5].

As already mentioned above, enhancement of heat transfer rate in the tube heat exchanger can be readily achieved by geometrical design optimization, *e.g.*, diameter, length, and number of tube [6-8]. However, the geometry of the tube and the operating parameter was not optimized. Syyedvalilu and Ranjbar [6] reported that heat transfer rate is affected by the changes in tube diameter, slope ratio, and pitch coil reduction. Coelho and co-workers have also emphasized that operating parameter also plays role to tube the performance of the shell and tube heat exchangers besides the geometrical design [9]. This is corroborated by the experimental and numerical study by Jouhara and co-workers that heat transfer rate in a cross-flow air-to-water pipe-based heat exchangers is dependent to the mass flow rate [10]. Further experimental evidence was brought by Napon whose study focused on the thermal performance and pressure drop of a shell and helically coiled heat exchanger with and without helical crimped fins [11,12]. The study indicated that the inlet mass flow rates of both hot and cold water and the temperature of inlet hot water control the heat exchanger effectiveness. The finding highlighted that configuration of coil and tube determines the effectiveness and increasing of mass flow rate within the tubing system decreases of effectiveness of the heat exchanger. Taherian and Allen [13] investigated the effect of tube diameter, coil diameter, coil surface and shell diameter to the effectiveness of a shell and coiled tube heat exchangers using aqueous solution of propylene glycol. In the investigated system, the working fluid was pumped from a tank into the coils through an electric heater and a manifold distributor and recirculated after passing through rotameters. In agreement with the results reported by Naphon [11,12], the effectiveness is affected by the tube geometry and coil configuration, and the effectiveness also decreases with increasing mass flow rate ratio.

Considering the above-mentioned strategy to improve the effectiveness of tube heat exchanger, this study is aiming at optimizing the thermal performance of copper-based tube heat exchanger whose design has existed in tempe processing industry in Indonesia. The benchmark geometry and the operating are set as

follows: the number of tubes is 4 while the inlet velocity is fixed at $3.6 \text{ m} \cdot \text{s}^{-1}$. Optimization considered the maximum effectiveness (ε) upon varying the tube diameters and the inlet velocity (or mass flow rate). The thermophysical parameters, e.g. temperature profile as well as temperature difference, and pressure drops, were extracted from computational fluid dynamics (CFD) calculation at steady state. Particular attention was emphasized to the temperature distribution within the tubing system in heat exchanger.

METHOD

This study was carried out by means of numerical simulation based on computational fluid dynamics (CFD). CFD enables the investigation of tube heat exchanger performance upon modification without performing experimental work. Particularly, we investigated the effect of tube diameter and inlet velocity to the NTU-effectiveness of tube heat exchanger, whose detail technical specification is summarized in Fig. 2a – Fig. 2b. The numerical calculation using CFD involved three main stages including pre-processing, solving, and post-processing.

Preprocessing: Prior to perform numerical analysis using CFD, pre-processing step was undertaken by building a model of the geometry of tube heat exchanger using ANSYS 16.0 software. Subsequently, discretization (meshing) was carried out using different interval sizes. Detail of the tube heat exchanger geometry and the result of the model of tube heat exchangers after discretization step is depicted in Fig. 2.
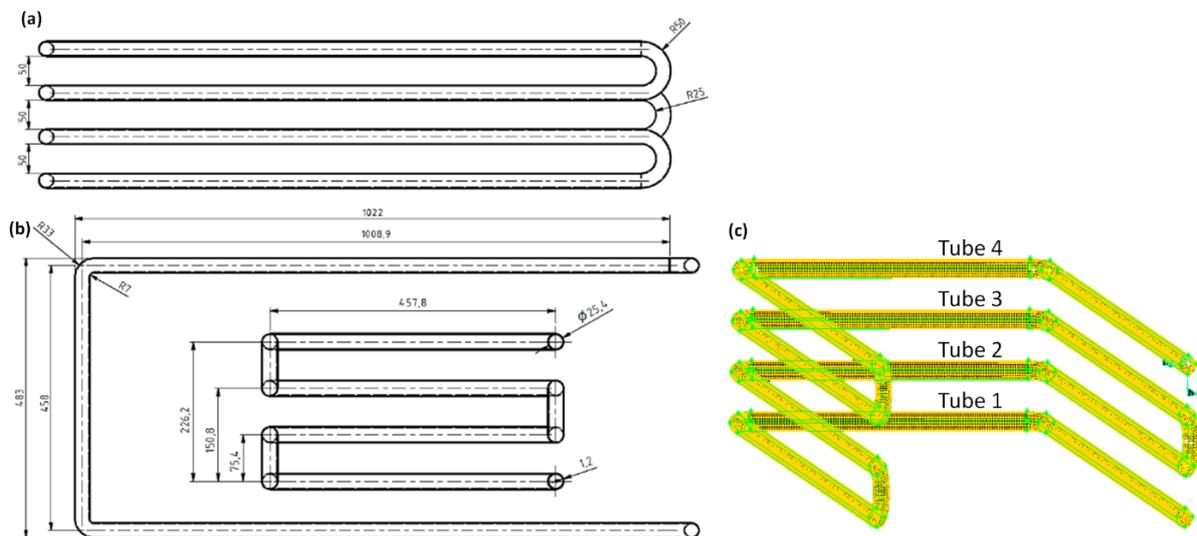

Figure 2. (a,b) Detail geometry of tube heat exchanger and (c) the model geometry after hex/wedge cooper meshing.

The geometry of the tube heat exchanger limits the simulation, where the number of tubes was fixed at 4 while the tube diameter was varied by 0.5, 1.0, and 1.5 in. Meshing geometry, as displayed in Figure 2c, was done using mesh type of hex/wedge cooper, *i.e.*, a mixture of hexahedral and wedge mesh, and cooper denotes that the number of volumes (cells) built is divided based on

the selected surface [20]. Final stage in pre-processing step was defining the boundary conditions which is summarized in Table 1 as follows:

Table 1. Boundary Conditions of Tube Heat Exchanger

Part of Tube Heat Exchanger	Conditions
Inlet	Velocity Inlet
Outlet	Outflow
Right tube	Wall
Middle tube	Wall
Left tube	Wall

Solving: This stage solves the governing equations in fluid dynamics as well as heat transfer within the discretized volumes. Warzi [17] considered zero-equation modeling in shear layer, one equation modeling and two-equation modeling as the solving methods that can be used in turbulence modeling. Nonetheless, two-equation modeling which is based on Eddy viscosity and depends on two equations of continuity, including standard k-ε, RNG k-ε, realizable k-ε, standard k-ω, and SST k-ω. The k-ε model is a turbulence model with kinetic energy and dissipation rate (ε). The transport phenomena can be solved by the Navier-Stokes equation [18], i.e., the change in momentum of fluid continuum depends only on the internal viscous force and the viscous force of the external pressure occurred on the fluid. Hence, the Navier-Stokes equation explains the equilibrium of the forces occurred on the fluid. In brief, Navier-Stokes equation can be stated in the equation of continuity, momentum, and energy as follows:

- Equation of Continuity:

$$\frac{\partial\,(\rho.u)}{\partial x} + \frac{\partial\,(\rho.v)}{\partial y} + \frac{\partial\,(\rho.w)}{\partial z} = 0 \tag{1}$$

- Equation of Momentum:

  Momentum towards x axis:

$$\frac{\partial\,(\rho.u^2)}{\partial x} + \frac{\partial\,(\rho.u.v)}{\partial y} + \frac{\partial\,(\rho.u.w)}{\partial z} = -\frac{\partial p}{\partial x} + \frac{1}{Re}\left(\frac{\partial\tau xx}{\partial x} + \frac{\partial\tau xy}{\partial y} + \frac{\partial\tau xz}{\partial z}\right) \tag{2}$$

  Momentum towards y axis:

$$\frac{\partial\,(\rho.u.v)}{\partial x} + \frac{\partial\,(\rho.v^2)}{\partial y} + \frac{\partial\,(\rho.v.w)}{\partial z} = -\frac{\partial p}{\partial y} + \frac{1}{Re}\left(\frac{\partial\tau xy}{\partial x} + \frac{\partial\tau yy}{\partial y} + \frac{\partial\tau yz}{\partial z}\right) \tag{3}$$

  Momentum towards z axis:

$$\frac{\partial\,(\rho.u.w)}{\partial x} + \frac{\partial\,(\rho.v.w)}{\partial y} + \frac{\partial\,(\rho.w^2)}{\partial z} = -\frac{\partial p}{\partial z} + \frac{1}{Re}\left(\frac{\partial\tau xz}{\partial x} + \frac{\partial\tau yz}{\partial y} + \frac{\partial\tau zz}{\partial z}\right) \tag{4}$$

- Equation of Energy:

$$\frac{\partial\,(u.Er)}{\partial x} + \frac{\partial\,(v.Er)}{\partial y} + \frac{\partial\,(w.Er)}{\partial z} = -\frac{\partial(\rho.u)}{\partial x} - \frac{\partial(\rho.v)}{\partial y} - \frac{\partial(\rho.w)}{\partial z} - \frac{1}{Re.Pr}\left(\frac{\partial qx}{\partial x} + \frac{\partial qy}{\partial y} + \frac{\partial qz}{\partial z}\right) + \frac{1}{Re}\left(\frac{\partial}{\partial x}\left(u.\tau_{xx} + v.\tau_{xy} + w.\tau_{xz}\right)\right.$$

$$\left. + \frac{\partial}{\partial y}\left(u.\tau_{xy} + v.\tau_{yy} + w.\tau_{yz}\right) + \frac{\partial}{\partial z}\left(u.\tau_{xz} + v.\tau_{yz} + w.\tau_{zz}\right) \tag{5}$$

The turbulence kinetic energy equation model k-ε is stated as follows:

$$\frac{\partial\rho k}{\partial t} + \frac{\partial}{\partial x_j}\left(\rho u_j\frac{\partial k}{\partial x_j} - \left(\mu + \frac{\mu_\tau}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right) = \tau_{tij}S_{ij} - \rho\varepsilon + \varnothing k \tag{6}$$

and the energy dissipation rate equation model k-ε is expressed as follows:

$$\frac{\partial \rho \varepsilon}{\partial t} + \frac{\partial}{\partial x_j}\left(\rho u_j \varepsilon - \left(\mu + \frac{\mu_\tau}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right) = c_{\varepsilon 1}\frac{\varepsilon}{k}\tau_{tij}S_{ij} - c_{\varepsilon 2}\frac{\varepsilon^2}{k}\rho f_2 + \emptyset\varepsilon \tag{7}$$

On the modeling of the standard k-ε, the turbulence effect is represented in average value. Here, k-ε using the method of renormalization group (RNG) was considered [19]. The physical parameters used in this simulation was a heat exchanger made of copper while the inlet fluid temperature, inlet velocity, and the temperature of the tubes are summarized in Table 2. The inlet velocity of working fluid (water) was varied into 1.6, 3.6, and 5.6 m·s⁻¹.

Table 2. Physical Parameters of Tube Heat Exchanger

Physical Parameter	Value
Material	Copper
Inlet temperature (°C)	26
Outlet temperature (°C)	60
Inlet velocity (m·s⁻¹)	1.6; 3.6; 5.6
Right tube temperature (°C)	84
Left tube temperature (°C)	73
Middle tube temperature (°C)	78

Post-processing: This stage after the numerical calculation was converged was done to display and visualize the results, *i.e.* the temperature distribution along the tube heat exchanger. The performance calculation (effectiveness, ε) was undertaken every variation of tube diameter and inlet velocity (or mass flow rate) using NTU method.

RESULTS AND DISCUSSION

Grid independence study constitutes an important aspect in modeling and simulation for validation. This is particularly crucial in thermal systems due to simplifications and idealizations that are usually employed, lack of accurate material property data, and various complexities in the process. Unless the models are satisfactorily validated using experimental study and the accuracy of the results obtained established, the models cannot be used as a basis for design and optimization. It should be noted that the physical behavior of the system, elimination of the effect of some parameters, e.g. grid size and time step, comparisons with available analytical or numerical results, and comparisons with the experimental data can be used for model validation [21]. In this regard, verification and validation process is based on the grid independence study.
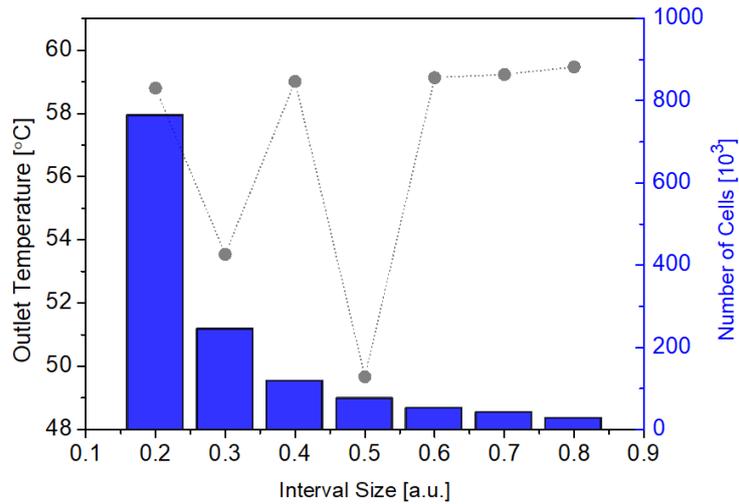
Figure 3. (a) Hexahedral/wedge discretized geometry of shell and tube heat exchanger, and (b) Nusselt number of heat exchanger using different distances upon meshing with different number of discrete cells.

The grid independence study shows a steady state temperature upon meshing with the interval size of 0.2, 0.4, 0.5, 0.6, 0.7, and 0.8 as displayed Fig. 3. As shown, the significant reduction of number of cells after changing the interval size from 0.2 to 0.3 whilst the number of cells is comparable among the geometry with interval size of 0.5-0.8. The steady state temperature of 59 ºC is achieved upon meshing with 0.2 interval size, and this value can be maintained by using interval size of at least 0.6. Therefore, pondering on the stability of data generated on mesh variation and the computational efficiency, the selected volume mesh size is set to the interval of 0.6 for all numerical calculation which will be discussed later (*vide infra*).
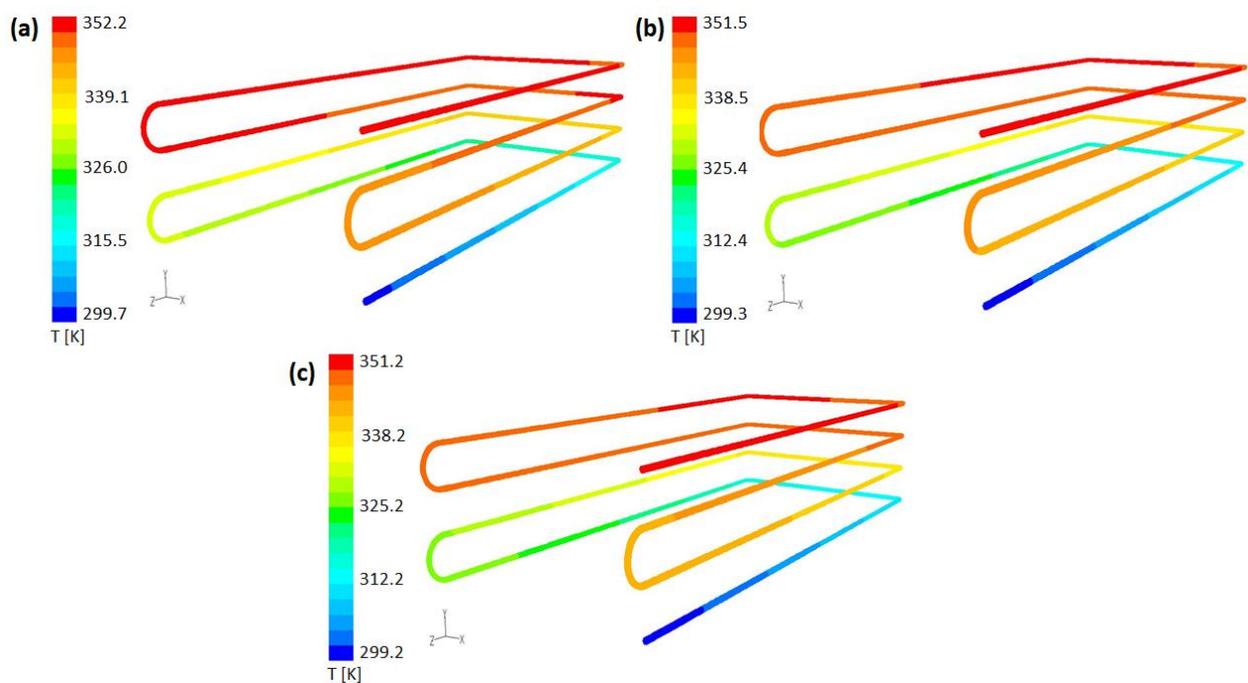

Figure 4. The steady state static temperature (in K) distribution in tube heat exchanger with 0.5 in tube diameter operated at inlet velocity of (a) 1.6, (b) 3.6, and (c) 5.6 m·s^{-1}.

Figure 5. The steady state static temperature (in K) distribution in tube heat exchanger with 1.0 in tube diameter operated at inlet velocity of (a) 1.6, (b) 3.6, and (c) 5.6 m·s^{-1}.
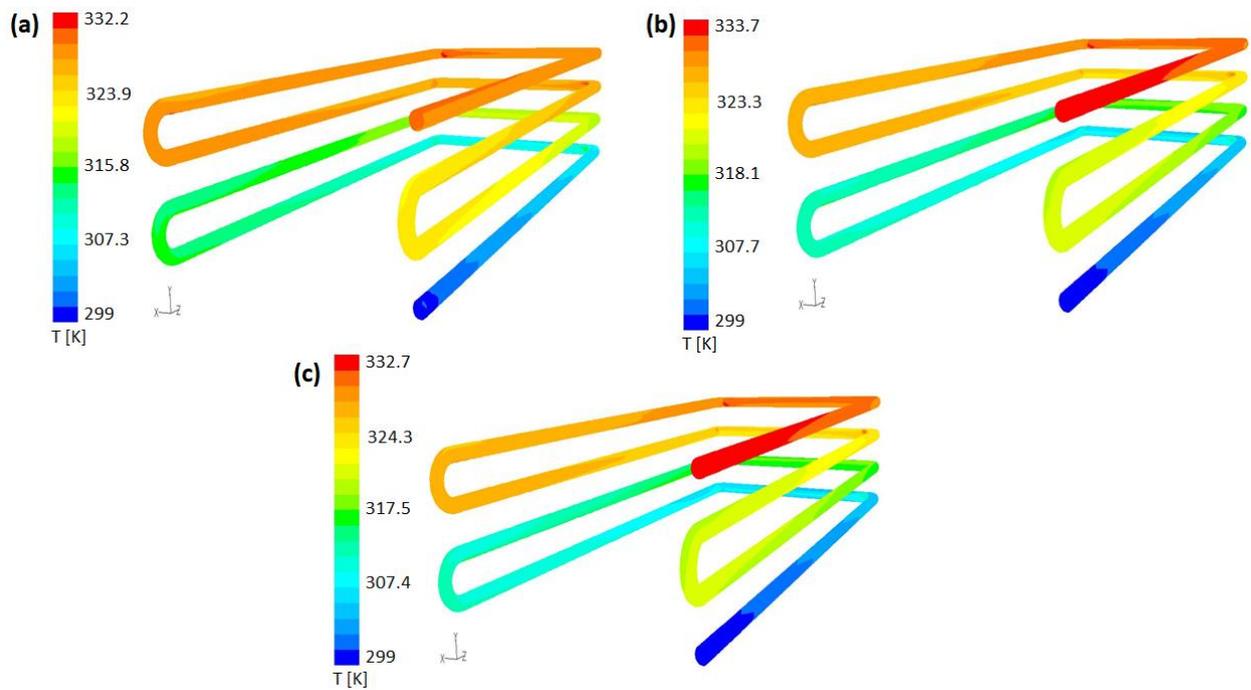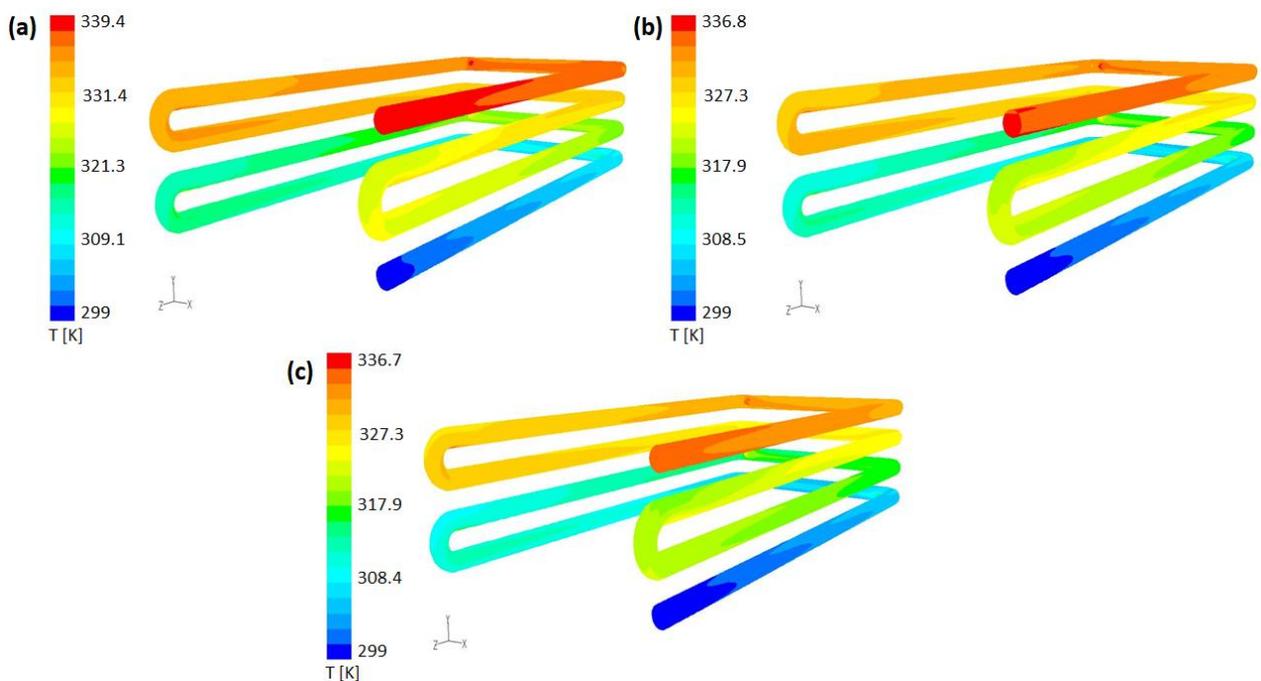

Figure 6. The steady state static temperature (in K) distribution in tube heat exchanger with 1.5 in tube diameter operated at inlet velocity of (a) 1.6, (b) 3.6, and (c) 5.6 m·s^{-1}.

The results of numerical calculation are depicted by the steady state temperature distribution upon varying tube diameter and inlet velocity (mass flow rate) in Fig. 4 – Fig. 6. The displayed temperature distribution, which is color coded, also indirectly indicates the profile of fluid flow inside the tubing system. Irrespective of the tube diameter and the inlet velocity, the

temperature is increasing from tube 1 (the lowermost tube) to tube 4 (the uppermost tube) as the total heat accumulated following the flow imposed to the heat source is increasing. Fig. 4 shows that increasing the inlet velocity in heat exchanger with tube diameter of 0.5 in tends to yield lower outlet temperature. With the lowest inlet velocity of 1.6 m·s^{-1} it is seen that temperature distribution is almost completely homogeneous at 352.2 K within tube 4. Such uniform distribution of the highest temperature is not observed for heat exchanger with tube diameter of 1.0 and 1.5 in. For heat exchanger with tube diameter of 1.0 in, the outlet temperature increases from 332.3 to 333.7 K when accelerating the inlet velocity up to 3.6 m·s^{-1}. Meanwhile, similar to the tube diameter of 0.5 in, heat exchanger with tube diameter of 1.5 in (Fig. 6) shows decreasing outlet temperature with increasing inlet velocity.

Comparing the heat exchanger based on the tube diameter, the resulting contours of temperature show that tube diameter of 0.5 gives a higher outlet water temperature than other tube diameters. With a constant heat source and the same inlet temperature of water, larger tube diameter implies larger area and hence, the total heat transferred into water within the tubing system is smaller as it is normalized to area. In addition, the operating inlet velocity also give a notable impact to the resulting temperature contour patterns, in which a low inlet velocity leads to a higher temperature of water output than the tube with a high inlet velocity.

The heat exchanger performance was evaluated from the effectiveness. Prior to discuss the quantitative performance of tube heat exchange, it is also necessary to assess the thermophysical properties to understand the underlying factor affecting the overall performance. In this regard, the effect of tube diameter and the mass flow rate to the thermophysical parameters, i.e. temperature difference (ΔT) and pressure drop (ΔP), is shown in Fig. 7. The temperature difference (ΔT), that is the difference between the inlet water temperature and outlet water temperature, reflects the amount of heat transferred into the water flowing in the tubing system. The numerical investigation yields that ΔT tends to decrease with increasing mass flow rate for tube diameter of 0.5 and 1.5 in. This decreasing ΔT is likely due to the decreasing travel time of the fluid volume from the inlet to the outlet, which limits the contact between the water with the tube wall exposed to the heat source, and hence the steady state heat transfer within the investigated control volume, *i.e.* tube heat exchanger, becomes minimum. The highest ΔT of 49 ºC and 39 ºC is obtained for tube heat exchange with diameter of 0.5 (ṁ = 0.2 kg·s^{-1}) and 1.5 in (ṁ = 1.8 kg·s^{-1}), respectively.
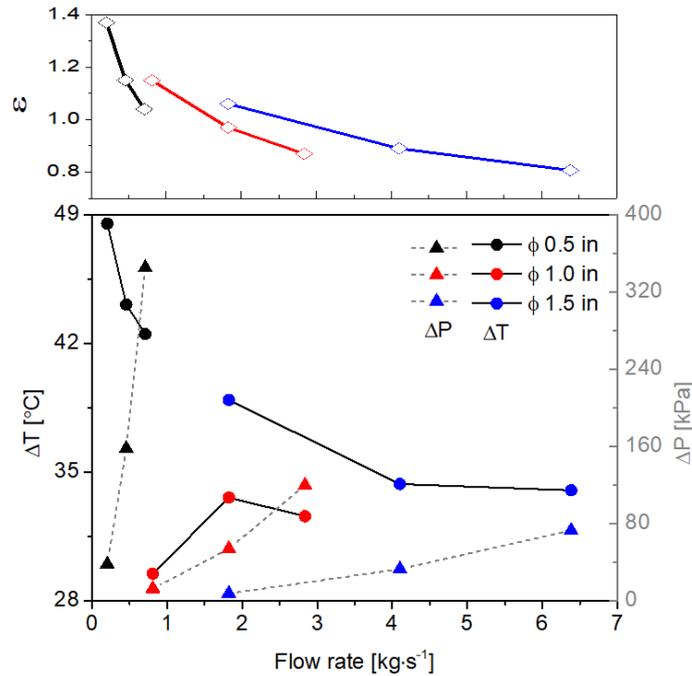
Figure 7. (*bottom*) Thermophysical parameters deduced from numerical calculation, including temperature difference (ΔT) and pressure drop (ΔP), and (*top*) the effectiveness of tube heat exchanger with variation of tube diameter and mass flow rate.

As already indicated from temperature distribution, it is interesting that a curving trend is observed for the heat exchanger with tube diameter of 1 in, which shows a maximum ΔT of 33 ºC. A flow rate higher than 1.8 kg·s^{-1} results in decreasing ΔT as similar to the trend observed in heat exchanger with tube diameter of 0.5 and 1.5 in. The maximum ΔT at this flow rate can be due to the interplay of highest convective combined with more efficient conductive heat transfer along the tubing system. In addition, it should be noted that the trend of ΔT for tube diameter of 1 in somehow implies that the maximum ΔT value is not reached with the lowest mass flow rate in heat exchanger with tube diameter of 0.5 and 1.5 in. Therefore, a lower mass flow rate will be necessary for further investigation.

Another essential parameter affecting the performance of heat exchanger is the pressure drop (ΔP). It is considered important in heat exchanger since it affects the pump power which contribute to the operational cost of tube heat exchanger. As shown in Fig. 7, irrespective of the tube diameter the higher pressure drops for tube heat exchanger is observed with increasing the mass flow rate. The lowest ΔP for heat exchanger with tube diameter of 0.5, 1, and 1.5 in is 38.0, 29.4, and 7.8 kPa, respectively, which is obtained at the respective lowest flow rate. This ΔP can be understood as a contribution of major and minor loss within the tubing system which accounts for the mechanical (viscous) wall friction between the fluid flow and the tube wall, and for sudden contraction, expansion, fitting, or bent, respectively. As the minor loss (h_{minor}) is independent to the Reynold number and the number of U-bent is similar for all tube heat exchanger variation, the minor loss contribution to the ΔP is solely affected by the velocity of the fluid. In this context, higher mass flow rate implies higher minor loss and so does the ΔP. Furthermore, according to the Darcy's law, where the major loss (h_{major}) = $f \times (L/D) \times (u^2/2g)$, it is obvious that the loss is inversely proportional to

the tube diameter and quadratic proportional with the fluid velocity. Thus, the higher inlet velocity with smaller tube diameter yields the highest possible ΔP, i.e. for tube heat exchanger with tube diameter of 0.5 in and an inlet velocity of 5.6 m·s^{-1} ($\dot{m}$ = 0.2 kg·s^{-1}).

Finally, the performance of heat exchanger is reflected by the NTU (number of transfer unit)-effectiveness (ε). The results as depicted in Fig. 7 (top) reveal that the heat exchanger with tube diameter of 0.5 in operated at m·s^{-1} ($\dot{m}$ = 0.2 kg·s^{-1}) exhibits the highest ε of 1.36. In general, this study finds that enlarging the tube diameter and increasing the inlet velocity potentially deteriorate the heat exchanger effectiveness. This finding is in line with the experimental study reported in literature [11-13]. As discussed earlier, the highest effectiveness is manifested by the higher ΔT and lower ΔP. In addition, it is known that ε-NTU is determined by the heat capacity of the working fluid, which is temperature dependent, and the total heat transfer coefficient. Since the heat capacity from the working fluid presumably doesn't differ significantly, the highest ε-NTU reflects that maximum heat transfer coefficient rooted from both convection and conduction is achieved.

CONCLUSION

We have demonstrated a numerical study to optimize the geometrical as well as the operational design of tube heat exchanger for food processing industry. With a tube diameter being varied by 0.5, 1.0, and 1.5 in and the inlet velocity being varied by 1.6, 3.6, and 5.6 m·s^{-1}, we have shown the dependency of two important parameters, *i.e.,* temperature difference and pressure drop, toward the variation of tube heat exchanger. The results indicate that smaller tube diameter with the lowest inlet velocity yields the highest effectiveness. Compared to the referenced existing tube heat exchanger (tube diameter of 1.0 in, and inlet velocity of 3.6 m·s^{-1}), the optimum design in this work increases the temperature difference from 33 to 49 °C while maintain the pressure drop as low as possible to reduce the cost of pumping power. As this tube heat exchanger is designed for food processing industry, it is recommended that the current benchmark system should be redesigned.

NOMENCLATURE
x Coordinate of x axis
y Coordinate of y axis
z Coordinate of z axis
u Velocity component of u
v Velocity component of v
w Velocity component of w
$\partial\rho$ Differential of fluid density
∂p Differential of force
∂q Differential of heat flux.
Re Reynold number
Pr Prandtl number
Er Total energy
μ_τ Turbulent eddy viscousity

μ Molecular viscousity

k Turbulent kinetic energy

x_j Space coordinate component

τ_{tij} Turbulent Reynolds stress tensor

S_{ij} Mean strain rate tensor

q   Heat transfer (Watt)

A   Heat transfer area (m^2)

T   Temperature (ºC)

Cp   Specific heat (J/kg.K)

Δp Pressure drop (N/m^2)

L   Tube Length (m)

D   Tube diameter (m)

Q   Volume rate (m^3/s)

f Friction factor

ρ Density (kg/m^3)

$\dot{m}$ Mass flow rate (kg/s)

U_m Velocity (m/s)

C_{min} Minimum heat rate capacity

μ_τ Turbulent eddy viscosity

x_j Space coordinate component

u_j Velocity at $_j$ coordinate

$\sigma_k \, \sigma_\varepsilon$ Turbulent Schmidt number

τ_{tij} Turbulent Reynolds stress tensor

S_{ij} Mean strain rate sensor

ε Turbulent dissipation rate

$\emptyset$ Volume fraction

c_ε Empirical constant

REFERENCES

[1]   I. M. Arsana, K. Budhikardjono, Susianto, A. Altway, "Modelling of The Single Staggered Wire and Tube Heat Exchanger," *International Journal Applied Engineering Research*, vol. 11, no. 8, pp. 5591-5599, 2016.

[2]   P. Kalpesh D, M. Chopra, "Performance Analysis of Cross Counter Flow Shell and Tube Heat Exchanger By Experimental Investigation & Mathematical Modelling," *International Journal of Engineering Research & Technology*, vol. 2, no. 7, pp. 271-276, 2013.

[3]   K. N. Seetharamu, G. A. Quadir, Z. A. Zainal, G. M. Krishnan, "FEM Analysis of Multifluid Heat Exchanger," *International Journal of Numerical Methods for Heat & Fluid Flow*, vol. 14, no. 2, pp. 242-255, 2004.

[4]   I. P. A. Kristyawan, R. Hantoro, G. Nugroho, "Effects of Indifferent Number and Tube Material on Temperature Distribution Tube Heat Exchanger in Stove," *Jurnal Teknik POMITS* Vol. 1, No. 1, pp. 1-5, 2013.

[5]   S. D. Kurniawati, R. Hantoro, D. Sawitri, "Effect of material type and thickness to distribution of temperatre on the wall furnaces using CFD Study – Study Case in The Industrial Tempe at District Tenggilis Mejoyo Surabaya," *Jurnal Sains dan Seni POMITS*, vol. 1, no. 1, pp. 1-6, 2013.

[6]   M. H. Seyyedvalilu, S. F. Ranjbar, "The Effect of Geometrical Parameters on Heat Transfer and Hydro Dynamical Characteristics of Helical Exchanger". *International Journal of Recent advances in Mechanical Engineering,* vol.4, no.1, pp. 35-46, 2014.

[7]   D. K. Yang, K. S. Lee, S. Song, "Fin spacing optimization of fin-tube heat exchanger under frosting condition," *International Journal of Heat and Mass Transfer,* vol. 49, pp. 2619-2625, 2006.

[8]   I. M. Arsana, K. Budhikarjono, Susianto, A. Altway, "Optimization of The Single Staggered Wire and Tube Heat Exchanger," *MATEC Web of Conference*. 2016, vol. 58, pp. 01017.

[9]   E. H. V. Segundo, A. L. Amoroso, V. C. Mariani, L. S. Coelho, "Economic optimization design for shell-and-tube heat exchangers by a Tsallis differential evolution," Applied Thermal Engineering 2016, vol. 111, pp. 143-151.

[10]   J. Ramos, A. Chong, H. Jouhara, 2016. "Experimental And Numerical Investigation Of A Cross Flow Air-to-Water Heat Pipe-Based Heat Exchanger Used In Waste Heat Recovery". Elsevier Science Inc. *International Journal of Heat and Mass Transfer* 102 (2016) 1267–1281

[11]   P. Naphon, "Thermal Performance And Pressure Drop Of The Helical–Coil Heat Exchangers With And Without Helically Crimped Fins," *International Communications in Heat and Mass Transfer,* vol. 34, pp. 321-330, 2007.

[12]   S. Wongwises, P. Naphon, "Heat transfer characteristics of a spirally coiled, finned-tube heat exchanger under dry-surface conditions," *Heat Transf. Eng.,* vol.. 27 , pp. 25–34, 2006.

[13]   Taherian, H. Peter, L. Allen. Experimental study of natural convection shell-and coil heat exchanger, in: *Proceedings of the 7th AIAA/ASME, the American Society of Mechanical Engineers*, HTD-vol. 357-2, 1998.

[14]   G. Y. Zhou, L. Y. Zhu, H. Zhu, S. T. Tu, J. L. Lei, "Prediction of temperature distribution in shell-and-tube heat exchangers," *Energy Procedia,* vol. 61, pp. 799 – 802, 2014.

[15]   B. Sunden, "Compact Heat Exchanger Performance," *International Journal of Numerical Methods for Heat & Fluid Flow.*, vol. 20, no. 5, pp. 551-569, 2010.

[16]   R. A. Wahyuono, R. Hantoro, G. Nugroho. "Study on Dry Heat Loss of a Very Low Birth Weight (VLBW) Newborn Nursed in an Infant Incubator with Overhead Screen", *Proceedings of Seminar on Intelligent Technology and Its Application* (SITIA), 2012.

[17]   Z. U. A. Warzi, "Fluid Dynamics, Theoretical and Computational Approaches Second Edition," CRC Press, 1998.

[18]   H. K. Versteeg, W. Malalasekera, "An Introduction to Computational Fluid Dynamics," Pearson Education Limited, 2007.

[19]   J. E. Bardina, P. G. Huang, T. J. Coakley, "Turbulence Modeling Validation, Testing, and Development," NASA Technical Memorandum 110446: California, 1997.

[20]   X. He, H. Zhao, X. Chen, Z. Luo, Y. Miao, "Hydrodynamic Performance Analysis of the Ducted Propeller Based on the Combination of Multi-Block Hybrid Mesh and Reynolds Stress Model," *Journal of flow control, Measurement, and Visualization*, vol. 3, no. 2, 67-74, 2015.

[21]   Y. Jaluria, "*Design and Optimization of Thermal Systems.* Second Edition," McGraw-Hill, New York, 2007.

[22]   D. Imen, L. Ammar, "Distribution of the temperature in the coaxial tube heat exchanger with spherical end," *Case Studies in Thermal Engineering*, vol. 10, pp. 621-627, 2017.

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I have sent my manuscript about 2 months ago.
How is the progress now? When will it be published? Thank you for your attention.

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Head of Heat Transfer Laboratory, Department of Mechanical Engineering, Universitas Negeri Surabaya.
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With kind regards

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