Energy, Environment and Storage (2021) 01-01-1-6

Energy, Environment and Storage

Journal Homepage: www.enenstrg.com

Numerical Investigation of The Effect of Finned Obstacle on Heat Transfer Characteristics in a Rectangular Channel

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ABSTRACT: Nowadays, with the development of technology and science, heat transfer holds an important place in engineering applications. In industrial areas, heat increases give rise to overheating, causing system errors. Passive techniques are frequently used to prevent from such disruptions. In this work, fins which are passive techniques which provide heat transfer development with high efficiency and low cost were investigated. To improve heat transfer, finned structures should be well optimized. On the other hand, the designer can prevent mixing the incoming air with the heated air with a bad design, which may cause a negative effect rather than improve heat transfer.In this work, in contrast to previous works that is smooth tube and 0 degree rectangular finned tube, four fin structures were designed and flow and heat-transfer characteristics numerically analyzed. These are crescent finned tube, 20 degree symmetrical imperforated rectangular finned tube, 20 degree asymmetrical imperforated rectangular finned tube and 20 degree symmetrical perforated rectangular finned tube banks with six rows. In this investigation, the geometric parameters were not changed and their effects on flow and heat transfer properties in different Reynolds numbers on these models were examined.The results indicate that the symmetrical structure has better heat transferability and higher friction loss compared to the asymmetrical structure and the perforated fin is higher than imperforated fin but the overall performance is not always superior. Therefore, both symmetrical and perforated finned tube is designed and analyzed with the highest heat transfer potential, it is seen that in terms of heat transferability this model is better than other designs.

Keywords:Finned-tube heat exchangers, Nusselt number, CFD-Fluent, Performance evaluation criteria, Model analysis, Numerical simulation.

Article History: Received:10.10.2020; Revised:03.03.2021; Accepted:04.04.2021 Availableonline: 28.04.2021
Doi: https://doi.org/10.52924/DSMP5780

1. INTRODUCTION

Many overheating can cause some problems for industrial applications. Active and passive methods to overcome this heating problem has been described [1]. One of them is active methods [2]. In these methods, mixing the liquid, vibrating surface; Some external power output is needed, such as Magnetic field use and jets shot is needed as some external power output. Another method is passive methods [3]. Additional geometric or surface modification techniques often implemented in devices such as existing material or expanded surface [4].

Expanded surfaces, one of the passive methods, are widely used in engineering disciplines that require heat movement related to energy transitions. Addition the component named fin to the heating surface improves the convective heat transfer coefficient or increases the heat transfer area of the surface, which leads to an increase in heat dissipation performance, maintaining the reliability and durability of devices. Since they are proven to be easy to manufacture, inexpensive and efficient, fin arrays are used in heat exchangers, cooling of gas turbine blades, cooling of electronic devices and other application areas that require high heat flux removal rates [5-8].

With each new design, the thermal losses of power electronic devices increase and the dimensions decrease, so various fin geometries such as rectangular, cylindrical, ring, pin wings and square are used to strengthen the heat transfer area of the surface[8-14].

2. MODEL DEFINITION AND NUMERICAL SOLUTION METHOD

2.1 Physical Model

Figure 2.1 and figure 2.2 demonstrate the schematic diagram of the six-row circular pipe set and the geometric definition of the finned pipe.

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Figure 2.3 - 2.7 represent the geometric shapes 0 degree rectangular finned tube, crescent finned tube, 20 degree symmetric imperforated rectangular finned tube, 20 degree asymmetric imperforated rectangular finned tube and 20 degree symmetric perforated rectangular finned tube respectively. Air flows over the model in the xdirection. Figure 2.2. shows the geometric properties and structure of the finned piece. The abbreviations Fw, Fh, Fp, Di, Do, S_1 and S_2 are fin width, fin height, fin pitch, inner diameter, outer diameter, transverse and longitudinal pipe spacing, respectively. The fins are spread evenly around the pipe and are advanced in the Z direction. The scheme of the pipe row arrangement and the geometric details of the finned tube are as follows;

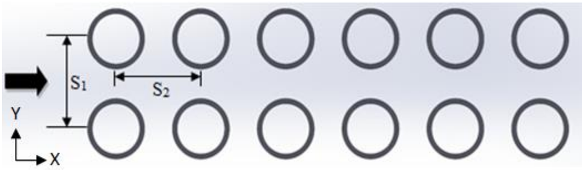

Figure 2. 1. Pipe set sequence.

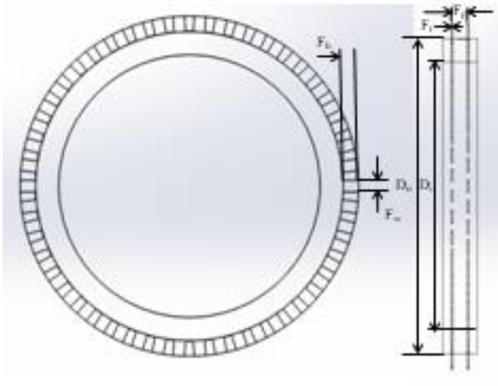

Figure 2. 2. Geometric definition of finned tube.

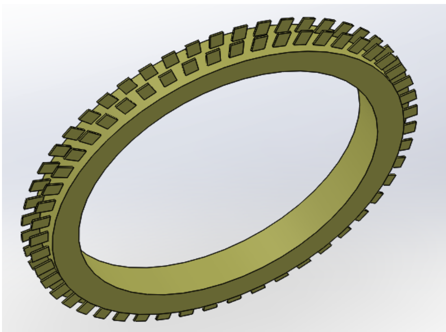

Figure 2.3. 3D view of 0 degree rectangular finned tube, Model 2

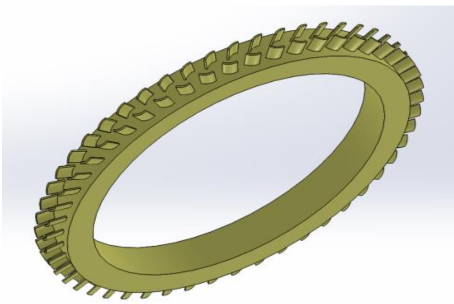

Figure 2. 4. 3D view of crescent finned tube, Model 3

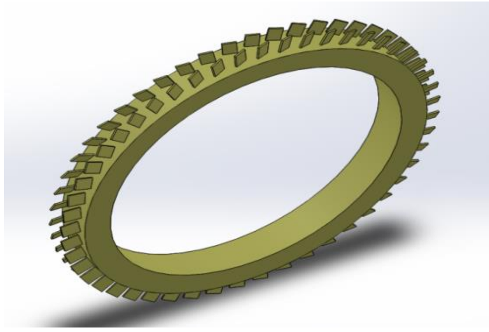

Figure 2. 5. 3D view of a 20 degree symmetric imperforated rectangular finned tube, Model 4

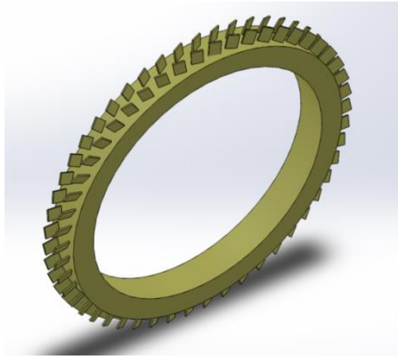

Figure 2.6. 3D view of a 20 degree asymmetric imperforated rectangular finned tube, Model 5

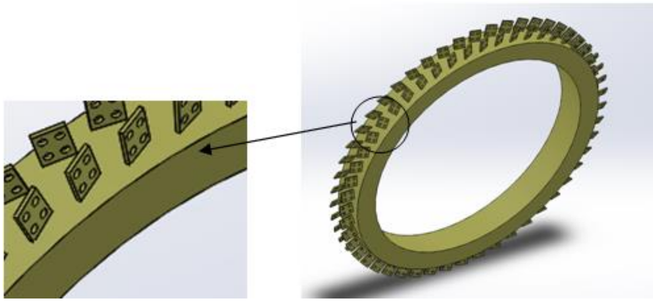

Figure 2. 7. 3D view of a 20 degree symmetrical perforated rectangular finned tube, Model 6

Table 2.1. Geometric parameters of finned tube clusters (six rows).

D_0 (mm)	D_i (mm)	$\frac{S_1}{D_0}$	$\frac{S_2}{D_0}$	$\frac{F_h}{D_0}$	$\frac{F_w}{D_0}$	$\frac{F_p}{D_0}$	Ft (mm)
40	34	1,5	1,5	0,05	0,04	0,05	0,2

Table 2.2. Smooth tube and finned tube models

Model 1	Smooth (without fins around) tube
Model 2	0 degree rectangular finned tube
Model 3	Crescent finned tube
Model 4	20 degree symmetric imperforated rectangular finned tube
Model 5	20 degree asymmetric imperforated rectangular finned tube
Model 6	20 degree symmetrical perforated rectangular finned tube

2.2 Boundary Conditions

The calculation area consists of eight boundaries as shown in Figure 2.8: Inlet velocity, outlet pressure, two symmetrical boundaries, two periodic boundaries and two wall boundaries. The boundary conditions of the computational area and finned tube wall are as follows;

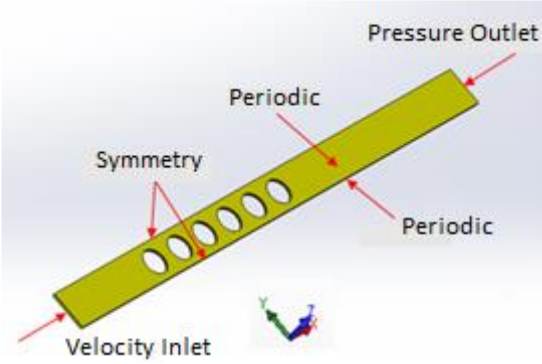

Figure 2. 8. Boundary conditions of the computational area.

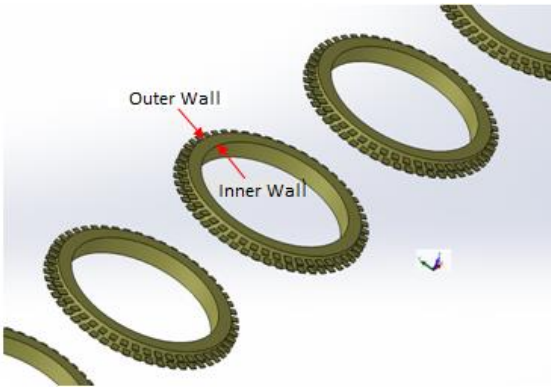

Figure 2. 9. Boundary conditions of the finned tube wall.

The working area is extended to three times the pipe diameter in upstream and seven times the pipe diameter in the downstream to stabilize the velocity at the inlet and avoid creating a non-recirculating flow at the outlet. At the entrance, the air enters the calculation area along the x direction at a constant temperature (423 K).

2.3 Numerical Solution Method

Finite volume method based on computational fluid dynamics software ANSYS FLUENT was used to solve a series of smooth finned tube.To guarantee accuracy and improve convergence, the Coupled - Pseudo Transient algorithm was chosen to solve the pressure and velocity domain. Standard discretization programs for pressure terms and quadratic spatial extraction arrangements for convection and discretization terms are used. Convergent criteria were defined for all simulations. Therefore, their residual values are less than 10^{-6} for energy equation and 10^{-4} for other equations.

2.4 Mesh Production

The computational area created using GAMBIT program is shown in Figure 2.10-14. This entire computing area consists of three parts: the entrance zone, middle zone, and exit zone. Different mesh shapes were applied to different regions. We applied a structured hexagonal mesh for both the entrance zone and exit zone. Because the fluid and interface of the finned tube are irregular and complex, an unstructured tetrahedral mesh has been applied to the central region. Moreover, a finer tetrahedral mesh was applied to the pipe wall region to ensure computational accuracy. Computational mesh configuration is as follows;

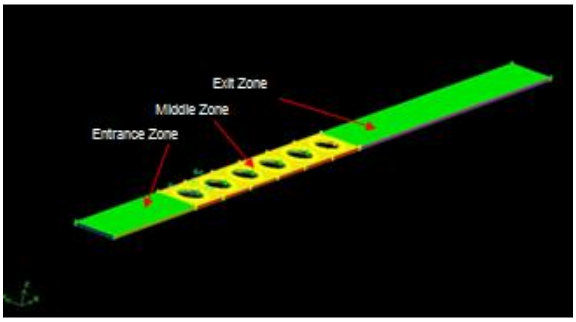

Figure 2. 10. General mesh area.

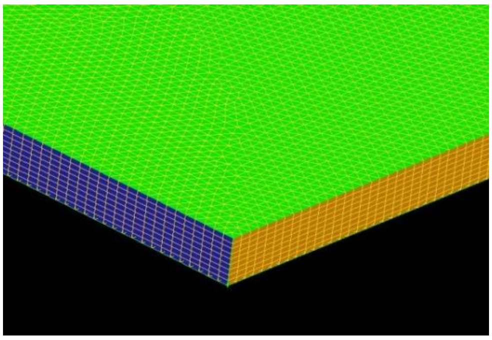

Figure 2. 11. Part of the entrance zone mesh

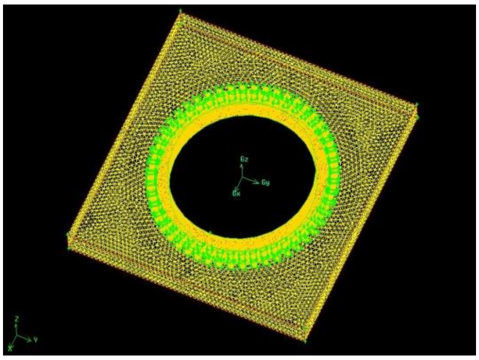

Figure 2. 12. Part of the middle zone mesh.

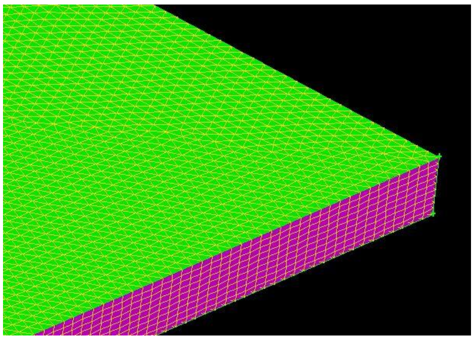

Figure 2. 13. Part of the exit zone mesh.

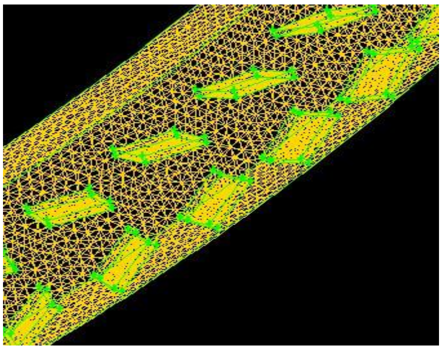

Figure 2. 14. Part of the finned tube mesh structure.

2.5 Mesh Independence

Mesh independence testing is necessary to ensure the accuracy of the numerical results obtained as a result of the analysis. Two separate works were made for approximately 2 million and 6 million meshes. When Re = 6000, Nu values were obtained as 48.08 and 50.55, respectively.Here, number of mesh was chosen as 6 million cells. Therefore, the mesh structure was adjusted to be around 6000000 in each geometry.

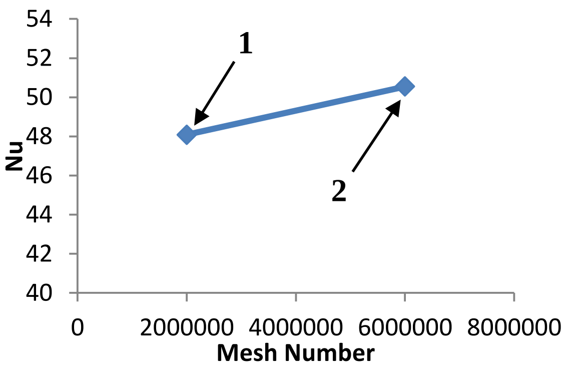

Figure 2. 15. Mesh structure vs Nu

It was inevitable to demonstrate the accuracy of the model and the solution method to ensure the accuracy of the simulation results. Because there was no experimental data for the finned tube in this study. Therefore, a numerical analysis was performed on a smooth pipe for verification. The smooth pipe used has the same geometry parameters as the set of smooth pipes tested by analysis.

The inlet velocity is 2-10 m / s in the range of values corresponding to Re number between 5000 and 10,000. Figure 2.16 shows the comparison of numerical resultsand experimental correlations [15,16]. When the results obtained in the verified Model 1 are compared with the correlations of Nu-Lu and Nu-Zhukauskas, validation has been achieved since we reached close values.

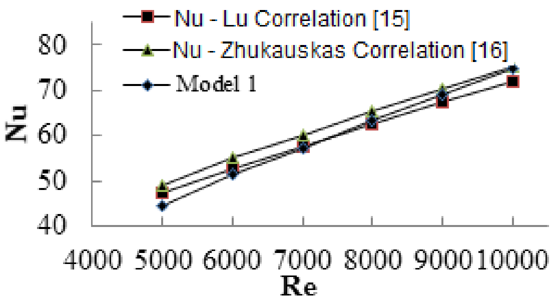

Figure 2. 16. Comparison of numerical results and experimental correlations

Consistent results between both numerical computation and experimental correlation results in Model 1 show that the applied numerical solution method can accurately predict the heat transfer and flow properties of finned tube arrays.

After verification, the results were obtained by analyzing Model 3, Model 4, Model 5 and Model 6.

2.6 Model Validation

Table 2.3. Pressure, velocity and temperature data of models at Re = 4000

	Re=4000					
	Model 1	Model 2	Model 3	Model 4	Model 5	Model 6
Pressure (Pin-Pout)	32-6	315.7-12.5	43.9-6.3	49.3-6.4	49.3-6.4	48-6.4
Velocity (Maximum)	5	14.2	5.43	5.98	5.9	5.9
Temperature (Finned circumference)	330-396	313-390	318-417	319-417	318-418	319-418

Table 2.4. Pressure, velocity and temperature data of models at Re = 6000

	Re=6000					
	Model 1	Model 2	Model 3	Model 4	Model 5	Model 6
Pressure (Pin-Pout)	60-9.8	1918-65.1	84.7-10.4	94-10.7	91.8-10.7	91-10.7
Velocity (Maximum)	7.18	37.9	7.78	8.39	8.29	8.28
Temperature (Finned circumference)	316-396	318-417	318-418	319-418	319-418	319-418

Tablo 2.5. Pressure, velocity and temperature data of models at Re = 8000

	Re=8000					
	Model 1	Model 2	Model 3	Model 4	Model 5	Model 6
Pressure (Pin-Pout)	95.4-14.6	4641.9-149.2	138.5-15.6	152.4-15.9	148.2-15.9	146.6-15.9
Velocity (Maximum)	9.39	62.8	10.2	10.8	10.7	10.6
Temperature (Finned circumference)	316-409	318-412	319-417	318-418	319-418	319-418

3. CONCLUSIONS AND RECOMMENDATIONS

Basically, model validationwas done toconfirmtheaccuracy of thisstudy [17]. As a result of thiscomparison, compliance has beenachievedaccordingtoreference [17]. Itwasobservedthatbetterresultswereobtained in terms of heat transfer in Model 2, which has thesamevalues as theliteraturebetween Model 1 and Model 2. Andthen Model 3, Model 4, Model 5 and Model 6 weredesignedandanalyzedandcomparedwiththespecified Reynoldsrange (2000-10000) using CFD-FluentCode [18].

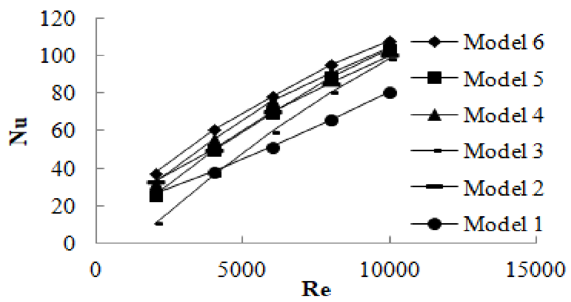

Figure 3. 1. Comparison of performanceparameters (Re-Nu) in finnedtubes.

As shown in Figure 4.1 Nusselt number is one of the most important parameters used in the heat transfer comparison. According to the increasing Reynolds number, designed model shows the variation of the Nusselt number. Nusselt numbers of the models increase with the increase of Reynolds numbers. As can be seen, in the specified Reynolds range, the Nusselt number, in other words, the heat transference, is the highest geometric design in the 20degree symmetrical perforated rectangular finned tube (Model 6), while the lowest geometric structure is seen in the smooth pipe (Model 1). Thelowest Nusselt number was obtained from crescent-finned tube (Model 3) among the designed models.

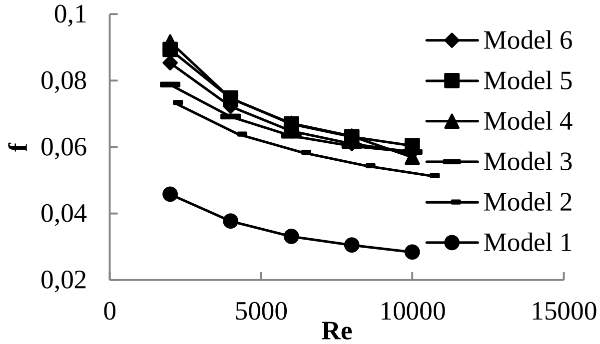

Figure 3. 2. Comparison of performance parameters (Re-f) in finned tubes.

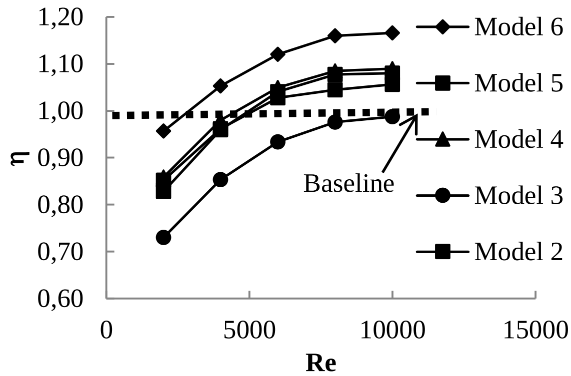

Figure 3. 3. Comparison of performance parameters (Re-η) in finned tubes.

Nusselt number is the most significant indicator of the heat transfer improvement method. In addition to this, the friction coefficient is another important parameter. The thermal performance factor obtained from the Nusselt and friction factor is an indicator of how much the heat transfer amount has improved compared to the previous situation. Analyzes were performed using CFD-Fluent Code [18] (Standard Model was selected as k-ε model and Enhanced Wall Treatment was applied near the wall) according to 6 different models and 5 different Reynoldsnumbers. While performing the analyzes, the flow and heat transfer properties of these models designed at different Reynolds numbers were examined by keeping the geometric parameters constant.

As a result of the analysis, it has shown that the performance evaluation criteria are not always superior to Model 1 taken as the baseline.

It has shown that the symmetrical structure has a higher heat transfer and friction loss compared to the asymmetric structure and perforated finned geometry has a higher heat transfer and friction loss compared to imperforated finned geometry. Based on this, when both symmetrical and perforated finned pipes (Model 6), which have the highest heat transfer potential, are designed and analyzed. It has been seen as a result of the analysis that they can transfer heat better than other finned pipes.

Furthermore, in this work, Nusselt numbers (Nu), friction losses (f) and performance evaluation criteria (η) were calculated against Reynold numbers and the results were analyzed using the CFD-Fluent Code [18].

- The model is consistent with the literature.
- The highest Nusselt number was obtained in Model 6, andtheleast in Model 1.
- The friction factor decreases as the Reynolds number increases and occurs at least Model 1.
- As the Reynolds number increases, so does the performance evaluation criteria. Performance evaluation criteria are mostly seen in Model 6. The structure with the least evaluation criteria is Model 3.
- Each model designed was found to have a higher Nusselt number than a straighttube (Model 1), but the crescent finned tube (Model 3) was worse in its design than the straight tube as opposed to improving the heat transfer, but resulting in more friction loss due to the finned structure of the design.
- The 0 degree rectangular finned tube (Model 2) has an advantage over the smooth tube (Model 1) in terms of heat transferability. Furthermore, more friction loss was determined as expected due to its fin structure.
- As a result of the analyzes made by changing the fin profile, it has been calculated that the heat transfer is at most 20 degrees symmetrical perforated rectangular finned tubes (Model 6).
- Numerical data for Nu and f correlate well; this can aid engineering applications of the heat exchanger for clean gas situations.
- In subsequent studies, analysis can be made by changing the finned angles or increasing or decreasing the number of holes on the finned structure.
- It will be appropriate to make designs according to the appropriate Reynolds number when it is desired to increase the heat capacity in machines such as economizers, recuperators, heat exchangers.

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