CFD Analysis of an Economizer for Heat Transfer Enhancement using Passive Technique

Himanshu Shekhar[1], Saddam Hussain[1], Abhinava C.R[1], Rajesh Babu C[2]
School of Mechanical Engineering, REVA University, Bengaluru

ABSTRACT - Economizers are used in conventional coal-fired thermal power plants to reduce the wastage of thermal energy through flue gases and increase the efficiency of the boiler. This helps the industry as there is an increase in profit. From the previous numerous researches, we have found that the efficiency of economizer can be increased by using different passive, active, and compound techniques. Several passive techniques can be used to enhance the heat transfer of an economizer. Some of the commonly used passive techniques are rough surfaces, treated surfaces, extended surfaces, etc. Our project mainly concentrates on the serrated fins extended surfaces. There has been work on serrated fins extended surfaces in case of heat exchangers but less amount of work has been done in case of economizers. Our work concentrates on the twisting of blades of serrated fins and the effect it produces on the efficiency of economizer using the CFD model. We will be comparing the CFD result with the industrial data.

1. INTRODUCTION

Economizer are one of the mechanical devices which are intended to reduce energy consumption or to perform some useful Functions such as preheating a fluid. In simple terms, the economizer is a type of heat exchanger. Till today many experiments andtheories have been performed to increase the efficiency of economizer by using different active and passive techniques andSome of them were quite successful in increasing efficiency. The economizer finds its use most required in boilers, power plants,Heaters, refrigerators, ventilators and air conditioners.

1.1 HEAT TRANSFER ENHANCEMENT

It is necessary to increase the heat transfer performance of working fluids in the heat transfer devices like heat exchanger andEconomizers. In the last few decades, a significant amount of effort had been made to develop the heat transfer enhancementTechniques to improve the overall performance of heat exchangers and economizers. Theefficiency of heat transfer equipmentWill be essential in energy conservation. And also, a more efficient heat exchanger can reduce the size of the heat exchanger, thusreducing the costs associated with both material and manufacturing of the heat exchanger.The rate of heat transfer can beIncreased passively by increasing the surface area, surface roughness, and also by changing the boundary conditions of theEconomizer.These Passive techniques, where these above-mentioned inserts are used in the flow passage to enhance the heatTransfer rate, are best suited when compared to active techniques.

1.2 EXTENDED SURFACES

The heat that’s conducted through solids, walls, or boundaries has got to be uninterruptedly dissipated to the surroundings orEnvironment to keep up the system in steady-state conduction. In several engineering applications, huge quantities of heat gotTobe dissipated from narrow areas. Heat transfer by convection between a surface and also the fluid encompassing it can beEnhanced by attaching to the surface lanky strips of metals referred to as fins. The fins increase the effective area of the surfaceThereby enhancing the heat transfer through convection. The fins are also attributed to “extended surfaces”.Extended surfaces (fins) are one in many of the heat exchange devices that are being used greatly to enhance heat transfer rates.The rate of heat transfer relies upon the surface area of the fin. It will increase the contact of the surface area.The heat whenTransferred through the fins poses the question of ‘determination of heat flow through a fin’ that requires the data ofTemperature distribution through it and This can be obtained by considering the fin as a metallic plate which is connected at itsBase to a heated wall and is transferring heat to a fluid through the means of convection and the heat flow through the fin is byConduction. Thus, the temperature distribution in a fin will depend upon the properties of both the fin material and theSurrounding fluid. In this, the heat transfer rate and potency for circular and elliptical annular fins wereAnalyzed for various environmental conditions.A finis a surface that extends from an object to enhance the rate of heat transferTo and from the surrounding environment by increasing convection. Different shape of the cavity is used to increase the surfaceArea of the fin with the fluid flowing aroundit.

2. METHODOLOGY

The CFD problem is solved as follows:

- Identification of flow domain.
- Geometry Modelling.
- Grid generation.
- Specification of boundary conditions and initial conditions.
- Selection of parameters and convergence criteria.
- Results and post-processing.

2.1 MODELING

Economizer is modelled through CATIA V5. It is assembly of all the parts. Mainly economizer pipe and serrated fins (without twisting).

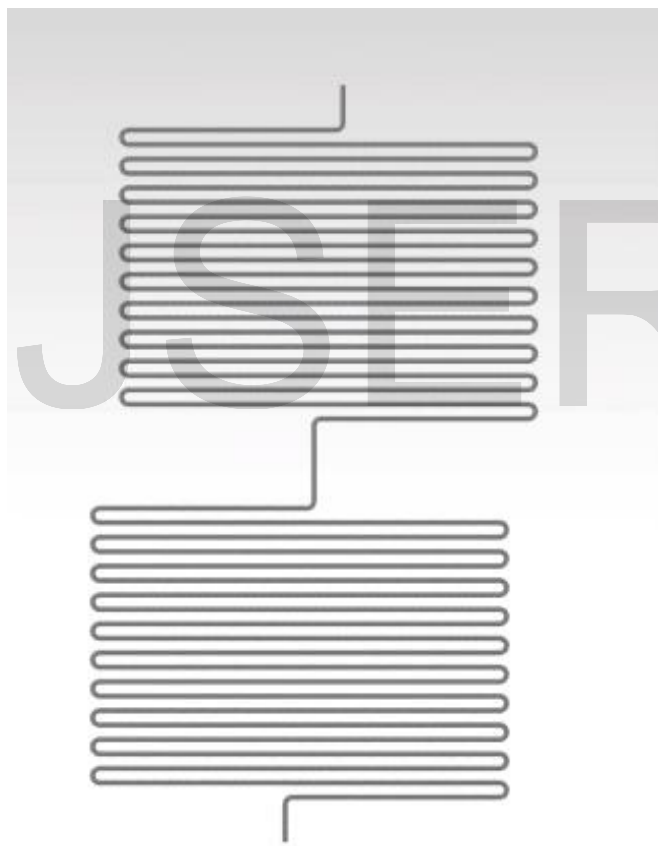

Fig 1. 3-D model of economizer

Table.1. Detail geometric parameters of the test samples:

Do (mm)	Di (mm)	PL (mm)	PT (mm)	NT	Nrow	Tube material
50.8	41.8	95	135	20	1	SA 210 GRA1

Table.2. Economizer specifications:[49]

Parameters	Values
Type of economizer	Plain tube, horizontal in-line
Type of flow	Multipass counter cross flow
Location of the economizer	Upstream of air preheater
The total heating surface of an economizer	1915 m2
Economizer duty	4.902 MMkcal/hr.
Total number of blocks	2
Economizer coil side pressure drop	0.2 kg/cm2
Economizer casing thickness	mm

2.2 GOVERNING FLUID FLOW EQUATIONS

To describe the fluid flow characteristics in the 3D computational domain, the following Continuity equations:

$$\frac{\partial u_i}{\partial x_i} + \frac{\partial \rho}{\partial t} = 0$$

Momentum equation:

$$\rho\left[\frac{\partial u_j}{\partial t} + u_i \frac{\partial u_j}{\partial x_i}\right] = -\frac{\partial p}{\partial x_j} - \frac{\partial \tau_{ij}}{\partial x_i} + \rho g_i \qquad (i,j = 1,2,3)$$

Energy equation

$$\rho\left[\frac{\partial h}{\partial t} + u_i \frac{\partial h}{\partial x_j}\right] = \frac{\partial}{\partial x_i}\left[\lambda \frac{\partial T}{\partial x_i}\right] - \tau_{ij}\frac{\partial u_j}{\partial x_i} + \frac{\partial p}{\partial t} + u_i \frac{\partial p}{\partial x_i}$$

Here ρ denotes the fluid density, t the time, ui the velocity component in i direction and xi the Cartesian coordinate, p is the pressure, gj is the gravitational acceleration, h is the specific enthalpy, T is the temperature and λ is the thermal conductivity coefficient. τij stands for the viscous stress tensor.

International Journal of Scientific & Engineering Research Volume 11, Issue 6, June-2020
ISSN 2229-5518

2.3 MESHING

A basic simulation model for a three-row finned tube with an inline arrangement is used. The boundary conditions are presented below. The conjugate heat transfer analysis is considered in the simulations performed. Therefore, the solid-fluid interface between the fins and the energy-carrying fluid are no-slip walls.

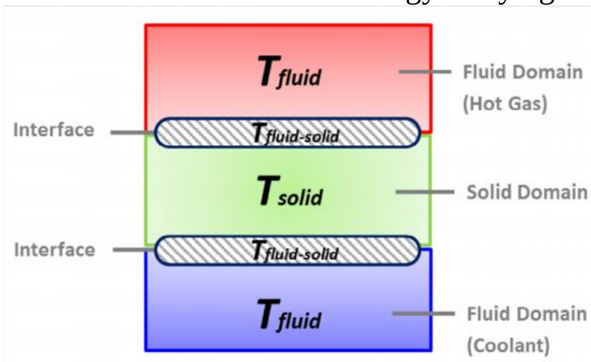

Fig 2. Boundary conditions

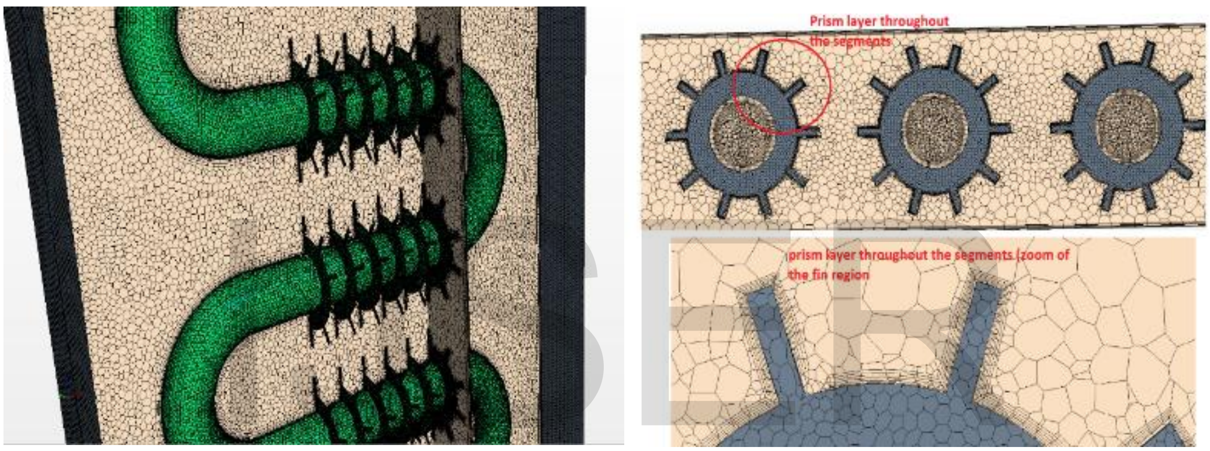

Fig 3. Grid used for the simulation model

The grid independence is checked for 10 Segments serrated fins with different unstructured grids like polyhedral cells and prism layers close to the no-slip walls. Grid independence for 10 segments as shown below in the graph.

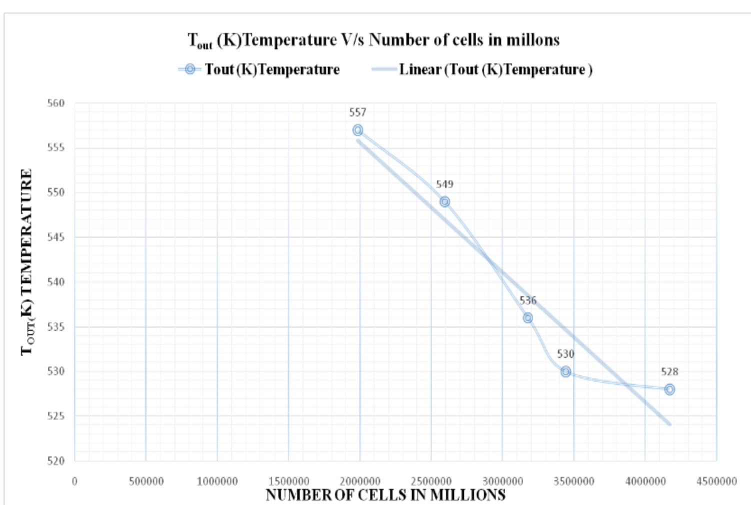

Fig 4. Grid Independence test (10 Segments)

TABLE3. GRID INDEPENDENCE WITH % INCREMENT IN OVERALL HEAT TRANSFER COEFFICIENT FOR 10 SEGMENTS

Mesh Size (millions)	Average Temperature Outlet T_{out} (K)	Surface area (m^2) T_{in}(K)	Overall Heat transfer coefficient (W/m^2 K) for 100% Indian coal		
			Existing plant with inline bare tube arrangements	Existing plant with Extended surfaces	% of increment
1983803	551				
2593137	549				
3177187	536				
3442248	530	1989.06651	43.87	44.98	2.53
4171024	527				

The grids checked have 1.9, 2.5, 3.1, 3.4, 4.1 million cells. The error between the finest grid having 3.4 million cells and grid having 1.9 million cells was less than 1.5% for heat transfer rate. By increasing the mesh size further, we can see that the 3442248 and 4171024 million cells simulation results in a value that is within my acceptable range. This indicates that we have reached a solution value that is independent of the mesh resolution, and for further analysis, we can use the 3442248 million cell case, as we will get a result within the user-defined tolerance.

Residual RMS Error-values have reduced to an acceptable value (typically 10-3)

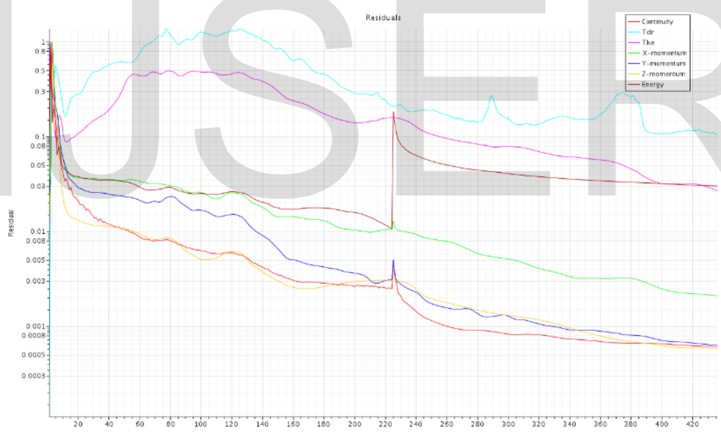

Fig 5. RMS residual values

Post-processing

Using STAR-CCM+, the temperature contour of the serrated fin is shown in fig 6 & 7.

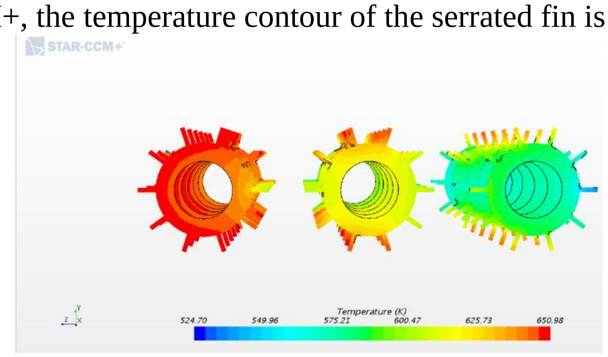

Fig 6. Temperature contours

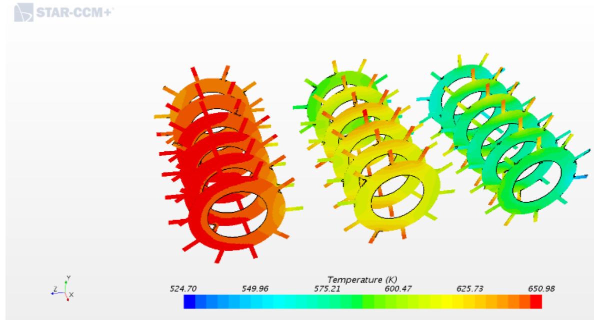

Fig 7. Temperature contours

From the above results, it is clear that existing plant with inline bare tube arrangement the overall heat transfer coefficient for 100% Indian coal is 43.87 (W/m2 K) provided heating surface area. By implementing passive heat transfer enhancement technique extended surface with 10 segments i.e with the fin area available for heat transfer is 1989.06 m2 the average temperature outlet is 530 K. As it is clear that for increasing fin surface area from 10 segments to 40 segments (Fin area from 1989.06 to 7956.25 m2) leads to the increase of the interruption of the boundary layer close to heat transfer surfaces, this results in a higher heat transfer rate. It is clear that the average temperature outlet is decreasing from 530 K to 506 K is a good sign that the capture of heat from the flue gases towards the feed water and with 24 K rise in feedwater temperature. And the percentage increment of overall heat transfer coefficient is from 2.53 % to 4.14%. It is clear that for the different heat transfer area, the case with serrated fins delivers better heat transfer performance than without fins.

REFERENCES

1. Næss, E., 2010. Experimental investigation of heat transfer and pressure drop in serrated-fin tube bundles with staggered tube layouts. Applied Thermal Engineering, 30(13), pp.1531-1537.
2. Mahmood, A.J. and Aldabbagh, L.B.Y., 2013, June. Double Pass Solar Air Heater with Transverse Fins and without Absorber Plate. In Proceedings of World Academy of Science, Engineering and Technology (No. 78, p. 1899). World Academy of Science, Engineering, and Technology (WASET).
3. Lemouedda, A., Schmid, A., Franz, E., Breuer, M. and Delgado, A., 2011. Numerical investigations for the optimization of serrated finned-tube heat exchangers. Applied Thermal Engineering, 31(8-9), pp.1393-1401.
4. Rahman, A. and Tafti, D., 2020. Characterization of heat transfer enhancement for an oscillating flat plate-fin. International Journal of Heat and Mass Transfer, 147, p.119001.
5. Hofmann, R., Frasz, F. and Ponweiser, K., 2007. Heat transfer and pressure drop performance comparison of finned-tube bundles in forced convection. WSEAS Trans. Heat Mass Transfer, 2(4), pp.72-88.
6. Yadav, S. and Kaushal, M., 2013. Nusselt number and friction factor correlations for solar air heater duct having protrusions as roughness elements on absorber plate. Experimental Thermal and Fluid Science, 44, pp.34-41.
7. Nuntaphan, Atipoang and Tanongkiat Kiatsiriroat. "Thermal behaviour of crimped spiral fin tube bank under dehumidifying process: A case study of inline arrangement." no. January (2004).
8. Yakar, G. and Karabacak, R., 2010. Effects of holes placed on perforated finned heat exchangers at different angles on the Nusselt and Reynolds numbers. Scientific Research and Essay, 5(2), pp.224-234.
9. Tian, E., He, Y.-L., and Tao, W.-Q., Numerical simulation of finned tube bank across a staggered circular-pin-finned tube bundle. Numerical Heat Transfer, Part A: Applications, 2015. **68**(7): p. 737-760.