Research Article

Design, Analysis and Optimization of Fin-and-Tube Type Heat Exchanger in Marine Diesel Engine

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Abstract

Despite diesel engine technology seems close to be ending with decrease of electrical vehicles, it still has potential in marine, heavy-duty, and industrial areas. Marine diesel engines are frequently encountered in yachts, towing boats, and sailing boats. The cooling fluid is sea water in marine engines. The engine coolant water is cooled by shell and tube heat exchangers. Charge air is cooled by fin-and-tube type heat exchangers. In this study, fin-and-tube type heat exchanger's design and flow analysis were investigated for the prototype six-cylinder inline engine. Heat exchanger's header geometry was optimized to increase heat transfer, and an acceptable header design was decided upon with the analysis results.

Keywords: *Marine Diesel Engine, Cooling System, Fin-and-Tube Heat Exchanger*

1. Introduction

Heat exchangers are a device used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particles and a fluid, at different temperatures and in thermal contact. Heat exchangers generally do not have external heat and work interactions. Typical applications include heating or cooling of fluid streams, evaporation, or condensation of single or multicomponent fluid streams. In other applications, the purpose may be to recover or reject heat or to sterilize, pasteurize, concentrate, crystallize, or control a process fluid. In most heat exchangers, heat transfer between fluids occurs through a separation wall or into and out of a wall [1].

The dominant thermal resistance is usually on the air side for air-cooled type heat exchangers. For this reason, to effectively reduce air-side resistance to save energy and resources, heat exchanger manufacturers seek ways to improve air-side performance. Fin-and-tube heat exchangers' performance data is very important for correct design [2].

Shell and tube type heat exchangers and fin-and-tube type heat exchangers are frequently used in marine diesel engines. Shell-tube heat exchangers consist of many tubes placed parallel to the shell axis. Fin-and-tube heat exchangers consist of many fins around the tubes.

In this study, fin-and-tube heat exchanger's design and analysis details were worked. Heat exchanger's header (which is coming charge air) geometry was optimized with analysis results. The header geometry was chosen with minimum pressure drop in working conditions. These results are going to check with test results in future work.

2. Materials and Methods

Exhaust, charge, and heat exchanger were included in the engine coolant for cooling system in the marinating process. In addition, it was aimed to have adaptable structure with in-line 4 and 6 cylinder, and different types of engine options planned to be developed in the future.

The calculation was investigated with the marine engine which is still working on push boat in shipyard. The calculation was checked with the test results in 6-cylinder marine diesel engine which is working actively. The new calculations were made with references these results. The marine diesel engine picture is given Figure 1.

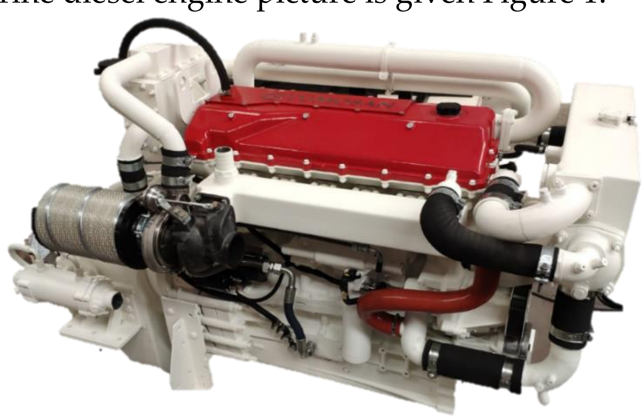

Figure 1: TUMOSAN 6 Cylinder Inline Marine Diesel Engine

In this project, engine power is 336 kW. All dimensions, design and analysis were referred with this engine power. The aim of the fin-and-tube heat exchanger side heat transfer was calculated 70 kW for this engine (with safety coefficient).

The fouling problem can be come with some reasons to fin-and-tube heat exchangers. Different types of fouling reasons are corrosion, biological deposits and settlement of sludge, rust or dust particles. The fin-and-tube heat exchanger fouling example is given Figure 2.

Figure 2: Fouling in Fin-and-Tube Heat Exchanger

Many efforts have been made to improve heat transfer in fin-and-tube heat exchangers with an affordable increase in pressure drop. Detailed parametric studies have been carried out on the effect of geometric parameters on the thermofluid properties of fin-and-tube heat exchangers. However, these results can be made more usable after optimization. Design correlations were also obtained in some of these studies. However, some analysis methods have been developed to examine the air side performance of fin-and-tube type heat exchangers [3]. Wang et al. a data reduction method is proposed to obtain the heat transfer coefficient and friction factor of fin-and-tube heat exchangers under dry conditions. They recommended selecting the appropriate epsilon-NTU relationship for data reduction [4]. Perrotin and Clodic investigated fin efficiency calculation by circular fin method. They determined that these methods have accurate estimates of fin efficiency for the straight blade, where P_l (longitudinal tube pitch) is nearly the same as P_t (transversal tube pitch) [5].

Thermodynamics performance of fin-and-tube heat exchangers depends on flow regimes and geometric parameters. Flow regimes such as laminar and turbulent can be determined by the Reynolds number is given equation 1.

$$Re = \frac{uL}{v} = \frac{\rho uL}{\mu} \tag{1}$$

where ρ is density. u and L are fluid velocity and characteristic length, respectively. μ and v are dynamic viscosity and kinematic viscosity, respectively.

Hydraulic diameter (D_h) is calculated with equation 2.

$$D_h = \frac{4A_cL_f}{A} \tag{2}$$

where A and A_c are total heat transfer surface area and minimum flow area, respectively. L_f is fin length in the flow direction.

Heat transfer amount in fin-and-tube heat exchangers are calculated with equation 3.

$$Q = \eta hA\Delta T \tag{3}$$

where Q is heat transfer amount, η is the fin surface efficieny, h is heat transfer coefficient, A is the total heat transfer surface area and ΔT is fluid's temperature differences.

2.1. Marinization System

Sea water enter the engine with sea water pump. Firstly, it passed through shell and tube heat exchangers. Sea water temperature differences from inlet to outlet is almost 9-11 °C. These heated sea water passed through fin-and-tube heat exchanger. In here sea water temperature difference is approximately 3-5 °C. Then sea water continues to be cooling for transmission oil cooler. After cooled oil cooler, sea water going out the engine system. The design for 6-cylinder inline marine diesel engine is given Figure 3.

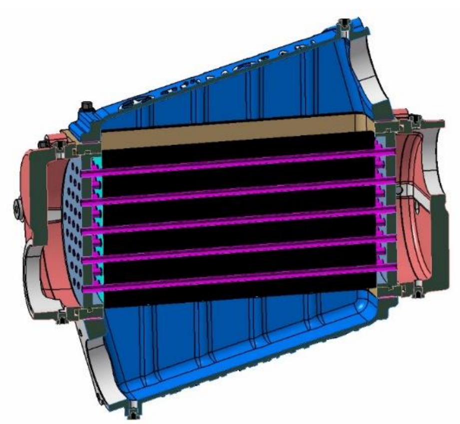

Figure 3: Fin-and-Tube Heat Exchanger Design

Charge air come inside the heat exchanger from top side and charge air go out the bottom side of heat exchanger. The seaside gets inside from right side in Figure 3. It passes through inside tubes and sea water go out from left side of heat exchanger.

Sea water and charge air flow were given in Table 1. The heat transfer quantity was calculated with these values.

Table 1: Charge Air and Sea Water Flow

Features	**Values**
Charge Air	0.5 kg/sec
Sea Water	4.8 kg/sec

Heat transfer value is increase with fins. The features were used in calculation was given in Table 2.

Table 2: Fin-and-tube Heat Exchanger Features

Features	**Values**
Tubes Diameter	8 mm
Tubes Thickness	1 mm
Tubes Material	CuNi10Fe1
Tubes Heat Transfer Coefficient	29 W/mK
Fin Diameter	170 mm
Fin Heat Transfer Coefficient	125 W/mK
Number of Fins	300
Number of Tubes	60
Fin Material	CuZn36
Fin Melting Temperatures	910°C - 950°C

2.2. Heat Exchanger Analysis

Fin-and-tube type heat exchanger has many fins, so distance between two fins are too narrow in the heat exchanger. For this reason, it is quite difficult to create a mesh in the regions between the fins and all fin surfaces. It is hard to get a solution from these meshes. It is necessary to simplify the geometry due to complex geometry. The dual cell heat exchanger model of Ansys Fluent was used for the solution of this problem [6]. The dual cell model uses the epsilon-NTU method for heat transfer calculation by modeling the heat exchanger core region with a porous structure. Thus, it allows analysis without including fins and tubes in the geometry. In addition, the geometry was halved from the symmetry surface to reduce computational cost.

Ansys Meshing program was used to create the mesh structure. The region in which the sweep mesh can be created was arranged by dividing the geometry. Thus, the mesh structure consisting of tetrahedral and hexahedral elements were created. The mesh involve about 8 million cells. These mesh view was given in Figure 4.

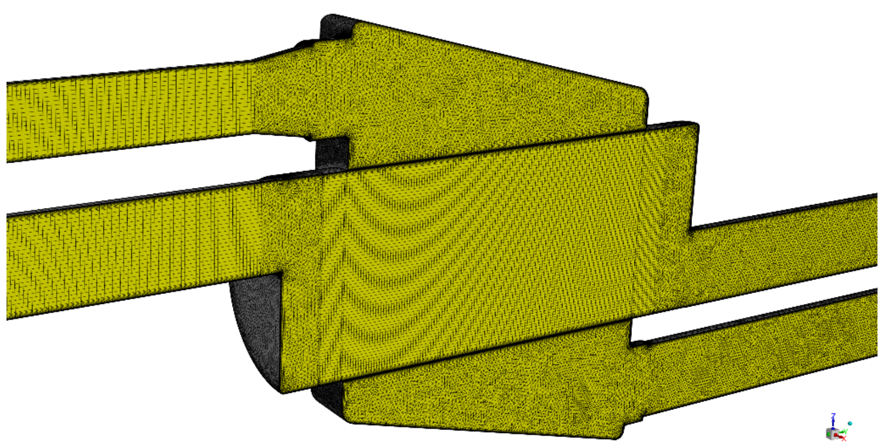

Figure 4: Symmetry Surface Geometry on Charge Air and Sea Water Side

3. Results

Flow and thermal analysis were performed with Ansys Fluent software. Velocity and temperature contours of the analysis results were given in Figure 5 for design 1.

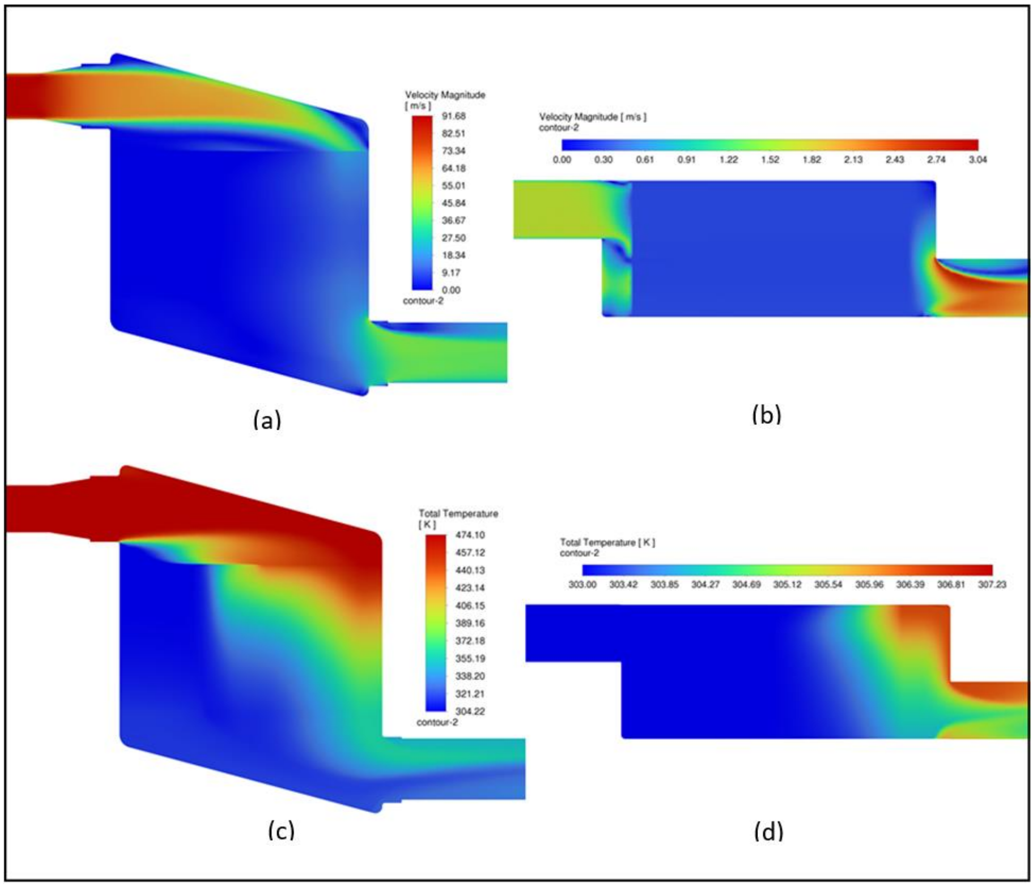

Figure 5: Left side: Turbocharge Air (a,c), Right Side: Sea Water (b,d)

The uniform distribution of the fluid between the fins or in the tube bundle is effective in properly performing heat transfer and increasing the efficiency of the heat exchanger. In basic heat exchanger design theory, it is assumed to fluid can distribute uniformly. However it is hard to catch uniform distribution on heat exchanger. Most of heat exchanger applications don't have uniform distribution [7].

When uniform distribution is provided, all heat transfer surfaces are used efficiently. On the other hand, maldistribution of flow which is occur in the header can cause existence local too hot and too cold regions, and directly affect the performance of total heat transfer [8]. Backflow occurred in about 36% of the region where the fins are located as seen in the analysis vectors were given in Figure 6. This shows that uniform flow cannot be achieved in the fin region.

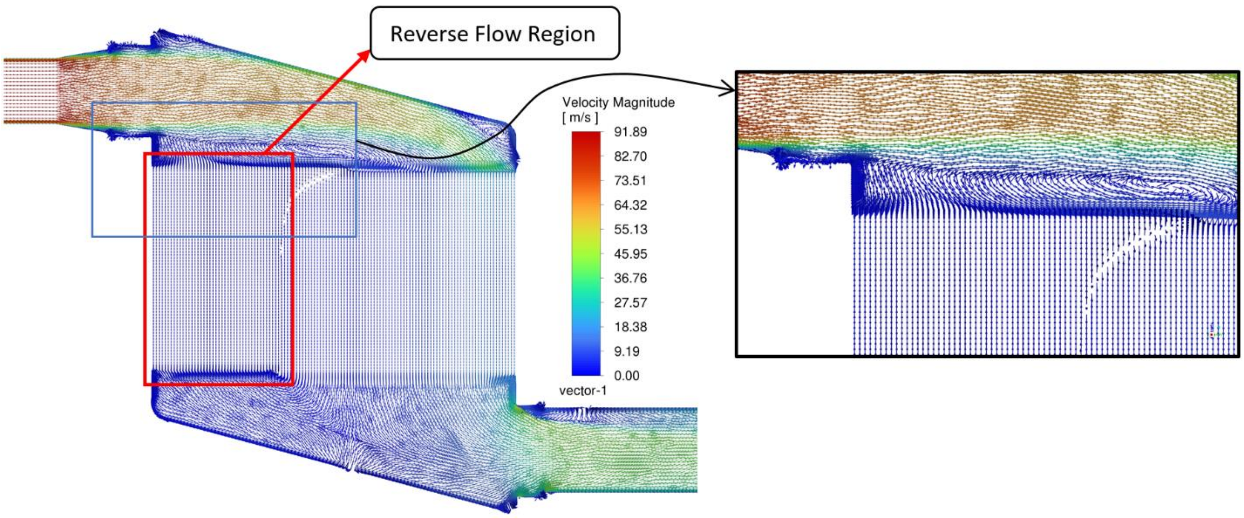

Figure 6: Intercooler Symmetry Surface Vector Plot and Scaled View (Design 1)

The direction of the inlet line is towards the inclined header surface was given in Figure 7(a) for design 1. Therefore, the turbocharge air was forced to pass through the area of the exchanger core area that is far from the inlet side with the dynamic effect. In order to direct the direction of the input line directly onto the heat exchanger core, design 2 was given in Figure 7(b) was applied by giving an angle to the input line.

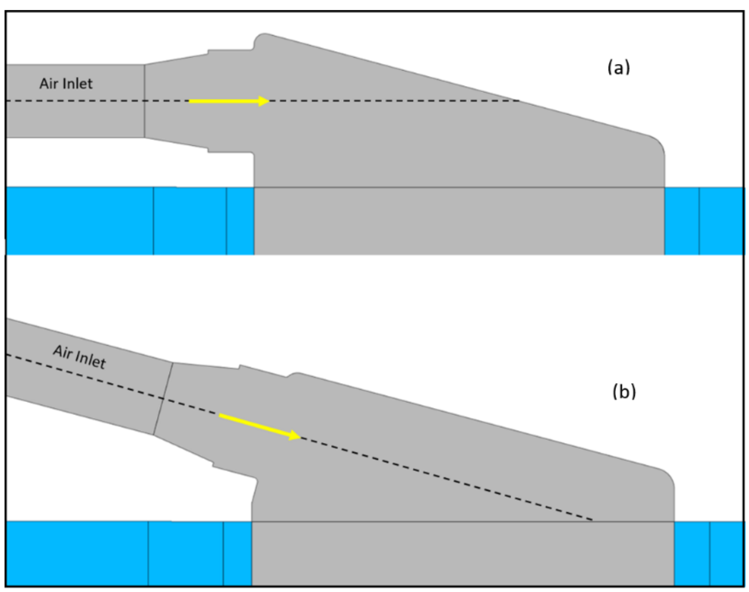

Figure 7: Design 1 (Base Design)(a) Design 2 (Angled Header)(b)

Velocity and temperature contours of the analysis results were given in Figure 8 for design 2.

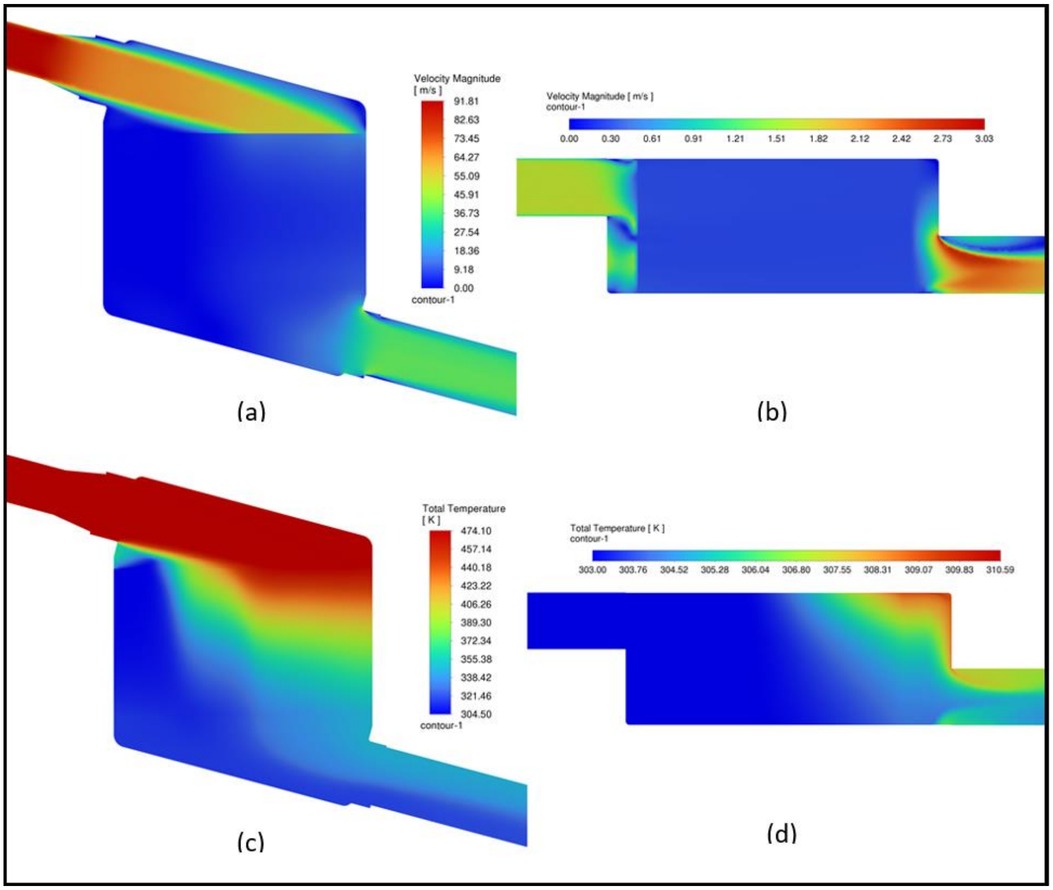

Figure 8: Left side: Charge Air (a,c), Right Side: Sea Water (b,d)

Backflow was observed at 28% in the design 2 heat exchanger core area was given in Figure 9. The heat transfer amount increase was obtained in with this design.

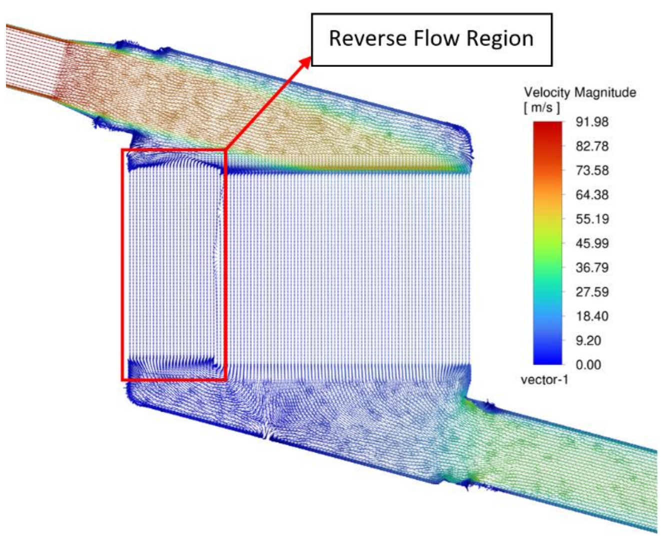

Figure 9: Intercooler Symmetry Surface Vector Plot View (Design 2)

Although a decrease in backflow was achieved, it was still continue in the region which is shown in Figure 9. The rapid charge air coming from the inlet line encounters a sudden expansion in the cross section. With the dynamic effect, a low-pressure zone is formed in the region shown in Figure 10. Thus the vortex is occurred. Due to the low-pressure zone, the flow that passes through the lower zone of the exchanger is directed upwards again through the fins.

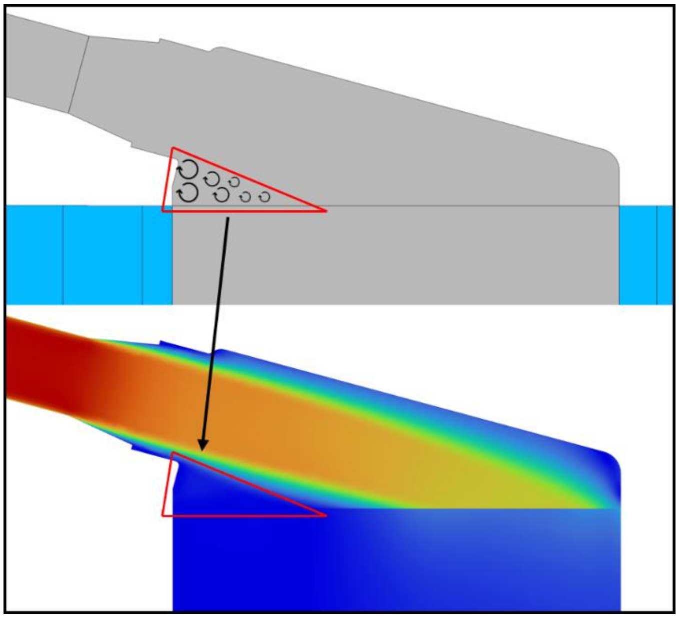

Figure 10: Vortex Area in Heat Exchanger

In order to reduce this low-pressure zone and to better distribute the charge air to the central region of the heat exchanger core, the inlet line has been moved down 15 mm along the inclined surface. This design was given in Figure 11(b).

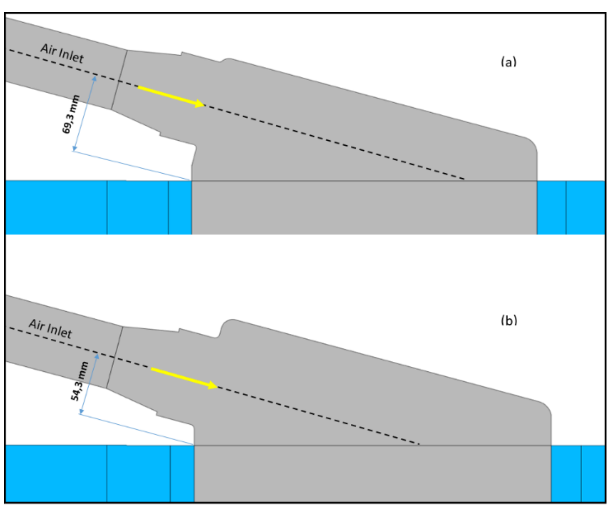

Figure 11: Design 2 (Angled Header)(a) Design 3 (Angled Header)(b)

The analysis result contour images made with this design was shown in Figure 11.

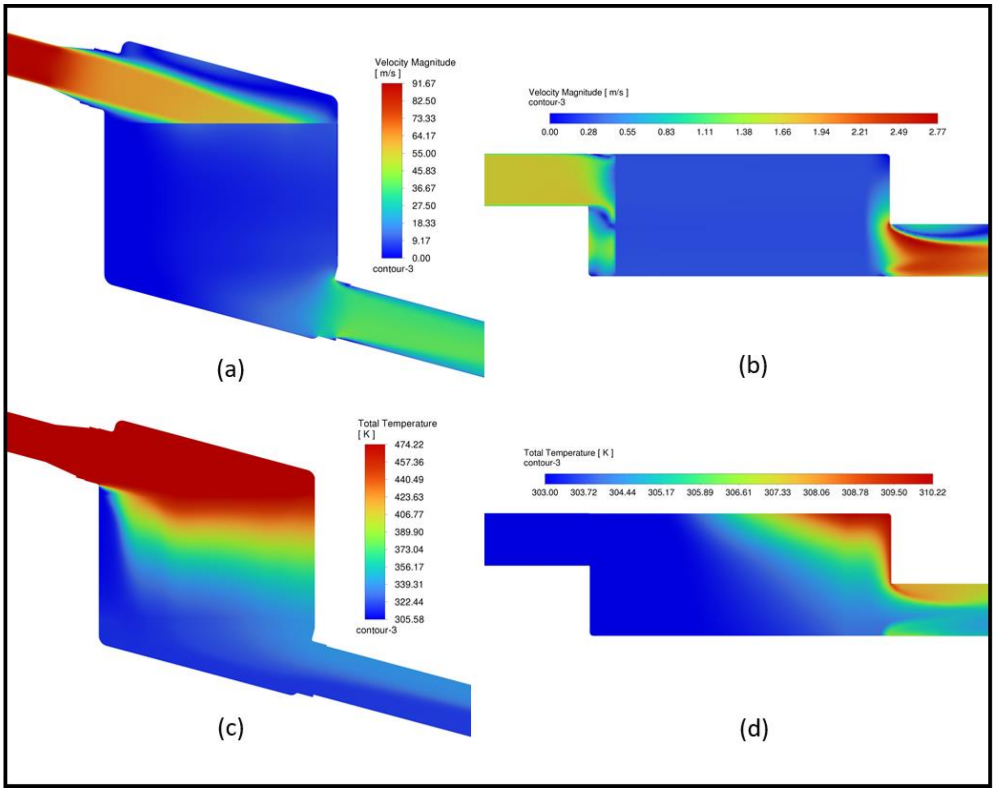

Figure 12: Left side: Charge Air (a,c), Right Side: Sea Water (b,d)

As a result, a 13% backflow was observed in the heat exchanger core region was given Figure 13, an increase in heat transfer and a decrease in pressure drop were obtained.

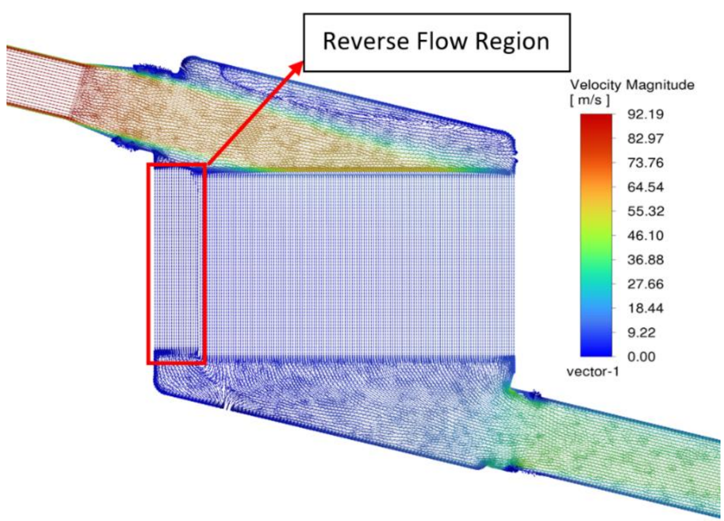

Figure 13: Intercooler Symmetry Surface Vector Plot View (Design 3)

4. Discussion and Conclusion

3 different header design were investigated with analysis for fin-and-tube type heat exchanger. The increase of about 8 kW has been achieved in the heat transfer from design 1 to design 3 with analysis results. This corresponds to a performance increase of about 13%. The difference between the three designs were indicated in the Table 3.

Table 3: Comparison of 3 Different Header Design

	Heat Transfer [W]	Charge Air Inlet Temperature [°C]	Charge Air Outlet Temperature [°C]	Sea Water Inlet Temperature [°C]	Sea Water Outlet Temperature [°C]	Charge Air Pressure Drop [Pa]
Design 1	63022	200	73.62	30	32.25	4898
Design 2	68512	200	66.77	30	32.36	5010
Design 3	71264	200	56.86	30	32.54	4871

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