FLOW OVER FINNED TUBE HEAT EXCHANGER

&

HEAT TRANSFER IN A SHELL AND TUBE HEAT EXCHANGER

GUIDED BY: PROF .DR. MEHMET SARIMURAT

-JAYAPRAKASH CHANDRAN
522242685

PROBLEM DESCRIPTION FOR FINNED TUBE HEAT EXCHANGER:

Problem Description

- The geometry consists of a generic fin and tube heat exchanger.
- Tubes arranged in staggered layout with spanwise and streamwise pitch of 30 mm and 34 mm respectively.
- Thickness of the fins is 0.5 mm, and the fin pitch is set to 3 mm
- Flow of air at 290 K is induced at a mass flow rate of 9.1×10^{-5} kg/s.
- Reynolds number based on the fin pitch is ≈ 305 and therefore, the flow is laminar.
- Since the main goal of this simulation is to analyze and compare the heat transfer from the extended surfaces, the flow inside the tube is not simulated. Instead the walls of the tube are set at a constant temperature of 350 K.
- The periodicity in the configuration helps to reduce the computational domain by only simulating the section that repeats in both the spanwise and streamwise directions as shown.

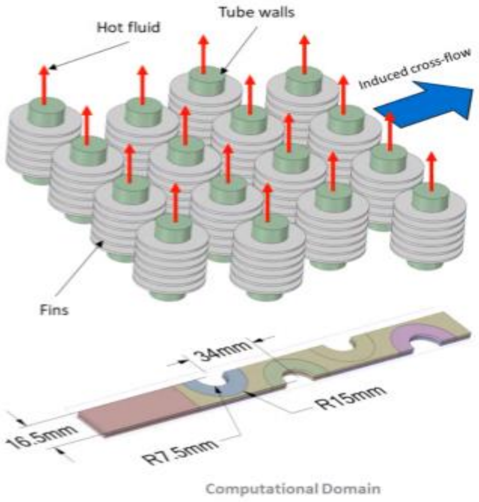


FIG A1 – Problem Description

Boundary Conditions

- Air at standard properties and temperature of 290 K is induced into the domain through the pressure inlet by pumping the air out of the domain through the mass flow outlet.
- Two fin materials are compared: aluminum and steel.
- The tube walls are maintained at the constant temperature of 350 K.
- Fin thickness is resolved by the computation mesh and conduction of heat through the fin is included.
- Boundary conditions are:
  - Pressure inlet at Inlet at 0 Pa gauge total pressure and 290 K total temperature.
  - Mass flow outlet at outlet with mass flow rate 9.1×10^{-5} kg/s.
  - Symmetry at the top, bottom and the two sides of the domain.
  - No Slip walls for tube walls at constant temperature of 350 K.
  - No Slip walls for fin walls with coupled thermal condition.
- A 3D steady-state simulation is performed using the pressure-based solver in Ansys Fluent.

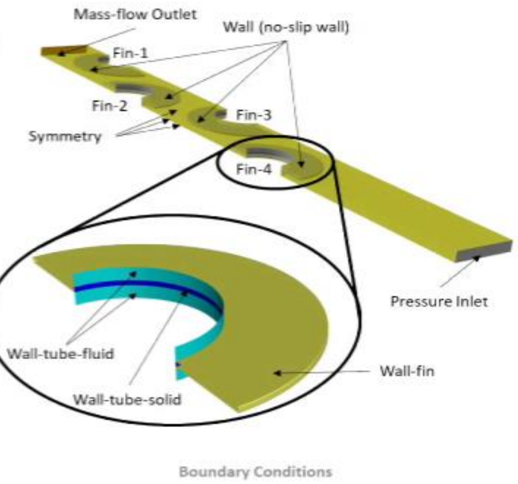


FIG A2 – Problem Description - Boundary Conditions

MESH:

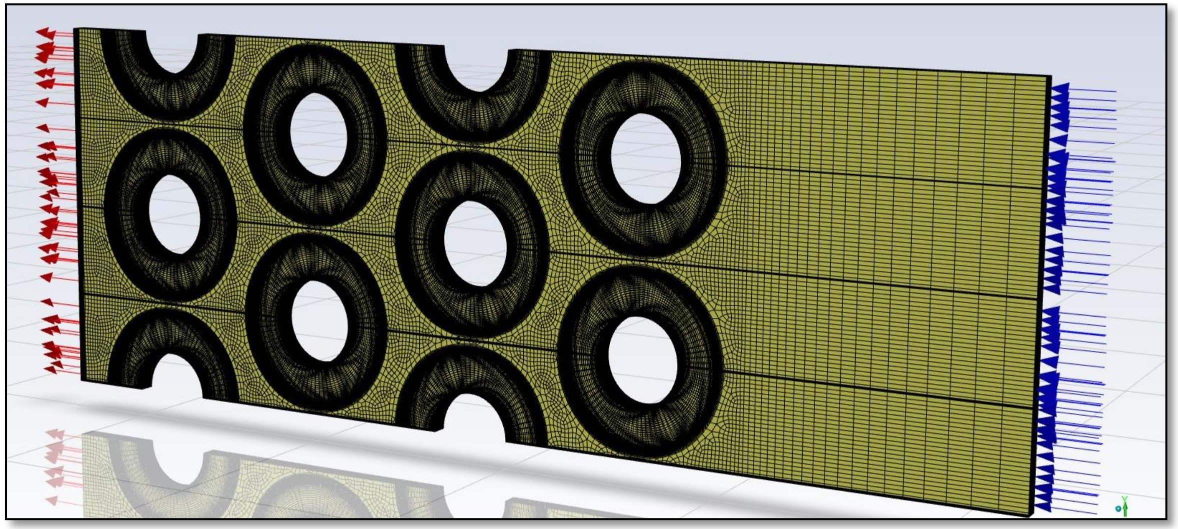


FIG. B1 – MESH

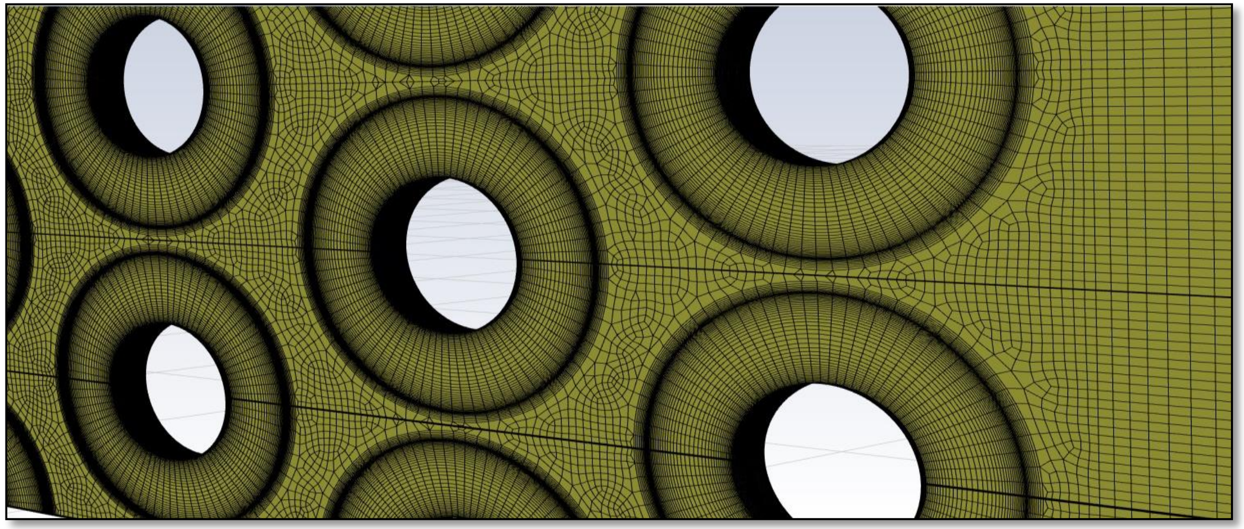
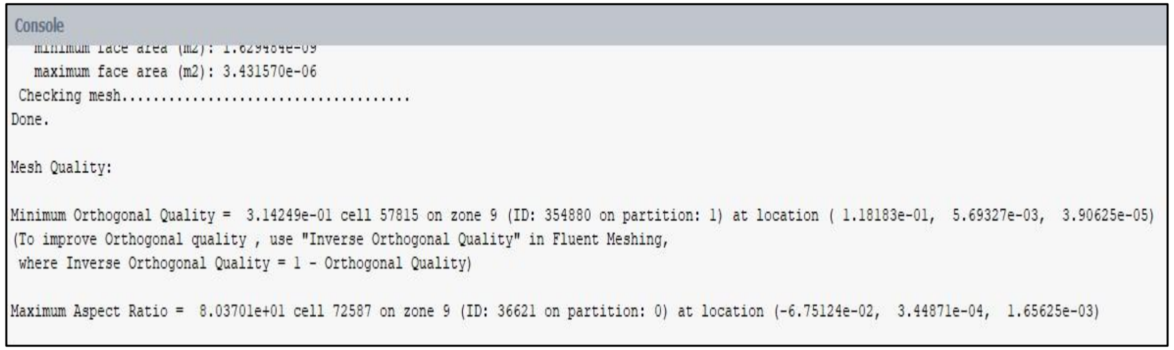


FIG. B2 – DETAILED MESH



```
Console
  maximum face area (m2): 3.431570e-06
 Checking mesh.....................................
Done.

Mesh Quality:

Minimum Orthogonal Quality =  3.14249e-01 cell 57815 on zone 9 (ID: 354880 on partition: 1) at location ( 1.18183e-01,  5.69327e-03,  3.90625e-05)
(To improve Orthogonal quality , use "Inverse Orthogonal Quality" in Fluent Meshing,
 where Inverse Orthogonal Quality = 1 - Orthogonal Quality)

Maximum Aspect Ratio =  8.03701e+01 cell 72587 on zone 9 (ID: 36621 on partition: 0) at location (-6.75124e-02,  3.44871e-04,  1.65625e-03)
```

FIG. B3 – MESH QUALITY CHECK

With a mesh value exceeding 1.0, the mesh quality is optimal for analysis purposes.

GEOMETRY CONDITIONS:

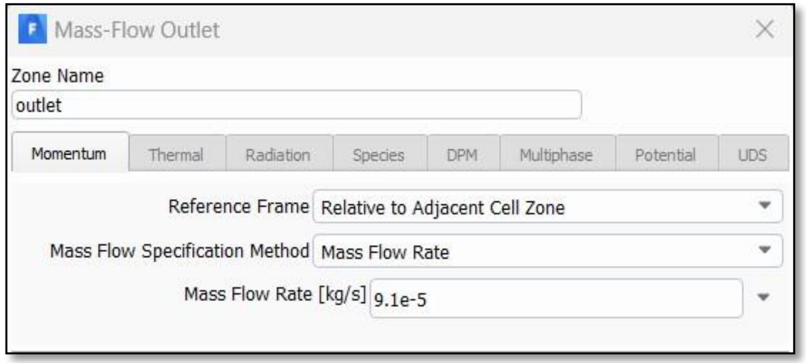


Fig C1: Mass flow outlet

Fig C2: Pressure Inlet

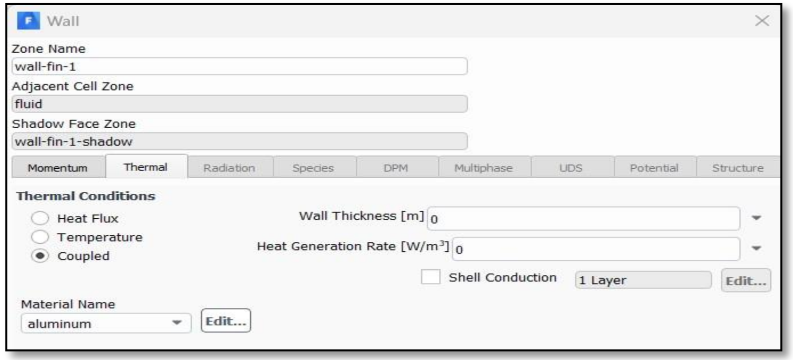


Fig C3: wall conditions

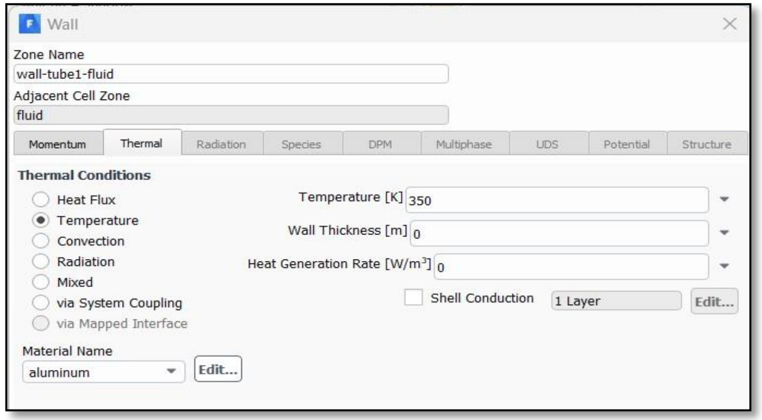


Fig C4: wall conditions - fluid

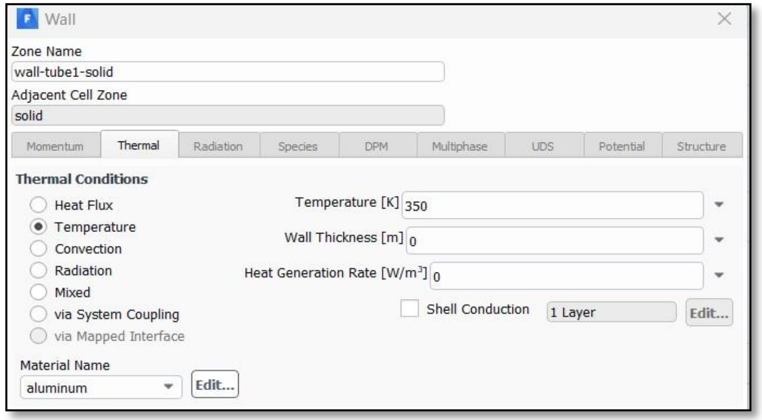


Fig C5: wall conditions - Solid

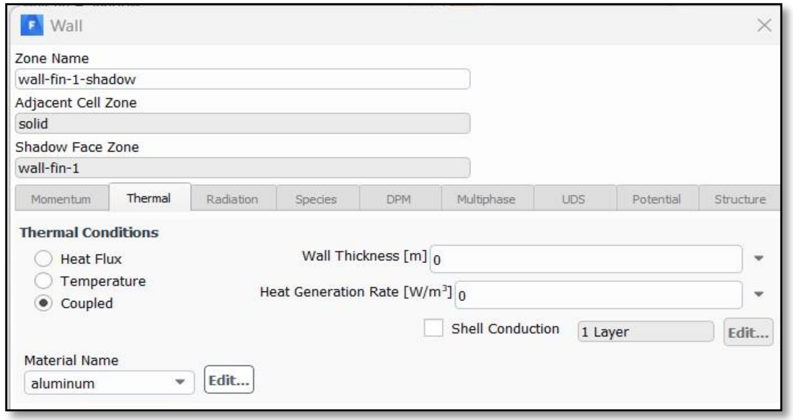


Fig C6: Wall fin shadow

GEOMETRY CONDITIONS CONTD:

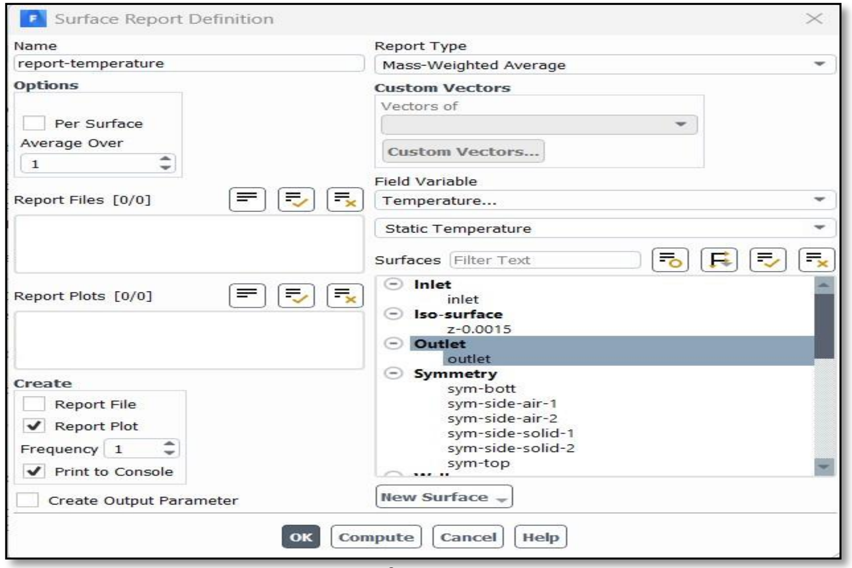


Fig D1: Surface report

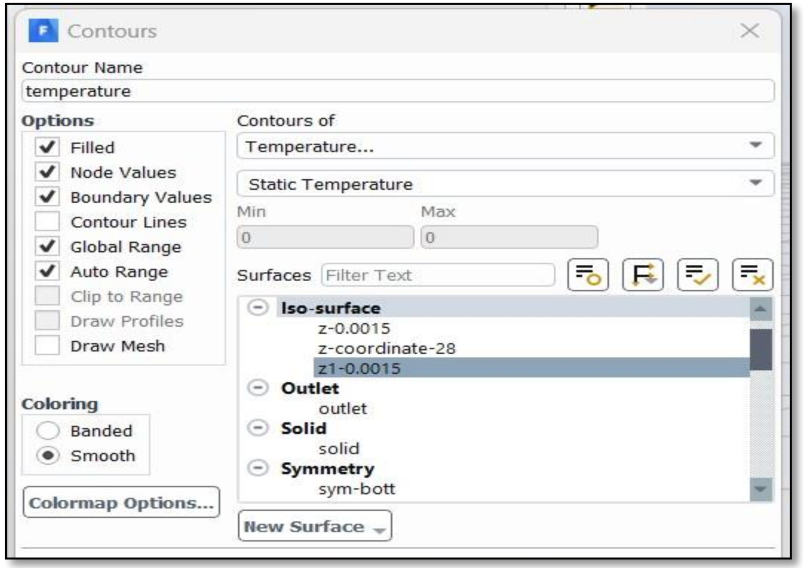


Fig D2: Contours

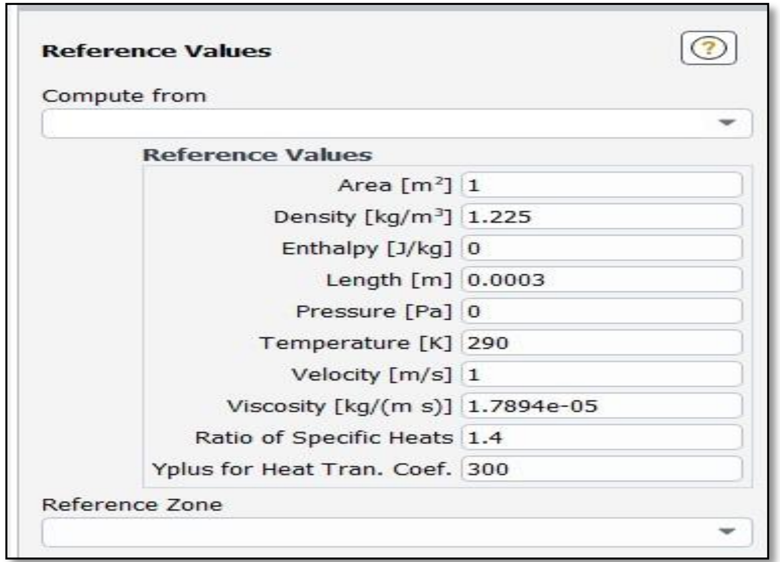


Fig D3: Reference Values

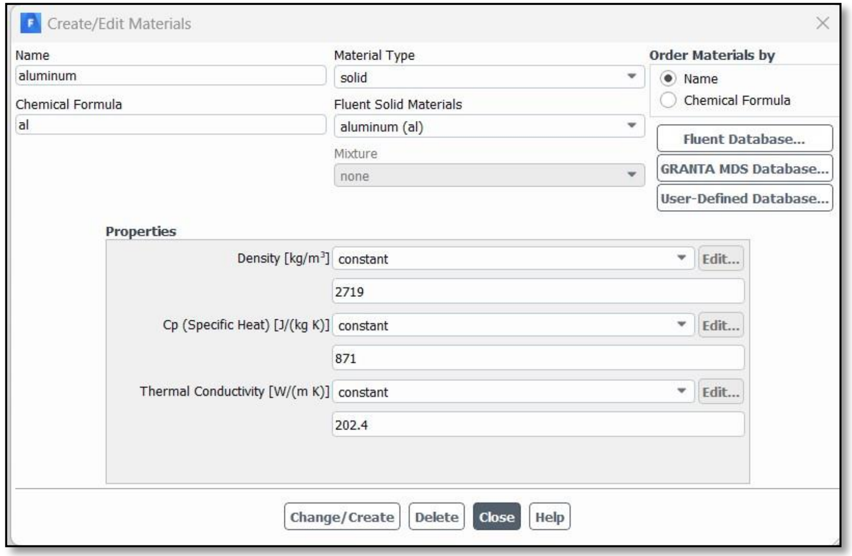


Fig D4: Materials Report

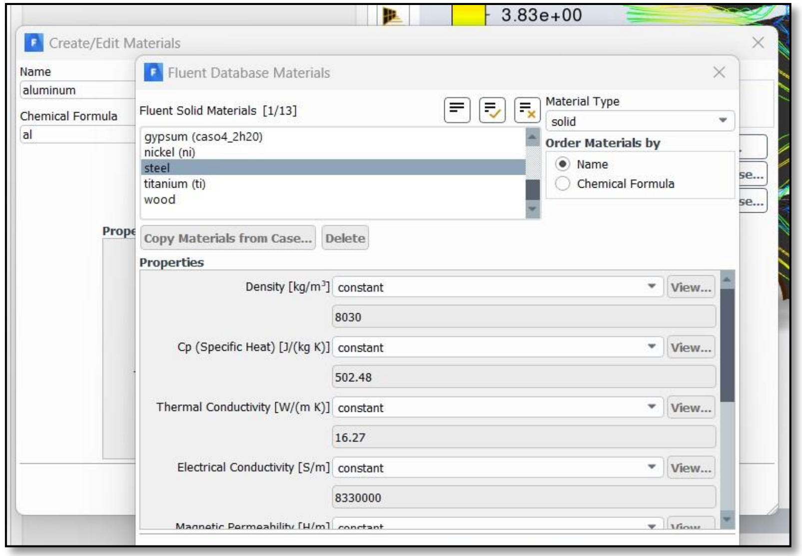


Fig D4.1: Materials Report

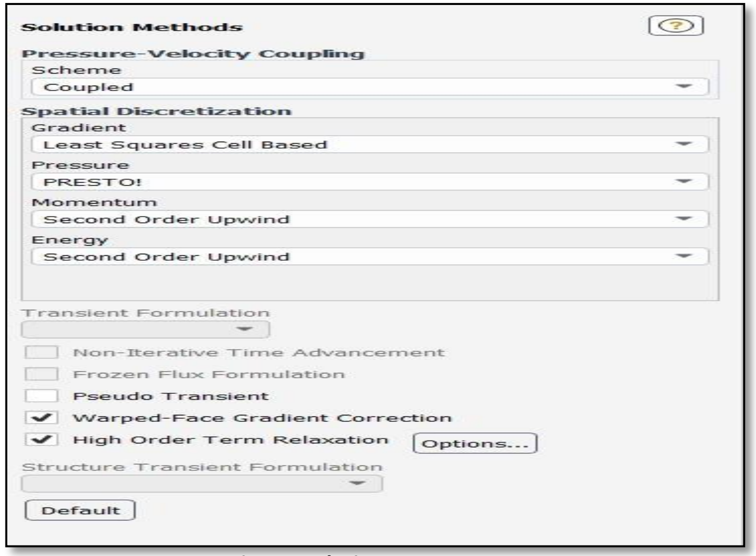


Fig D5: Solutions reports

MESH WITHOUT TUBULATOR:

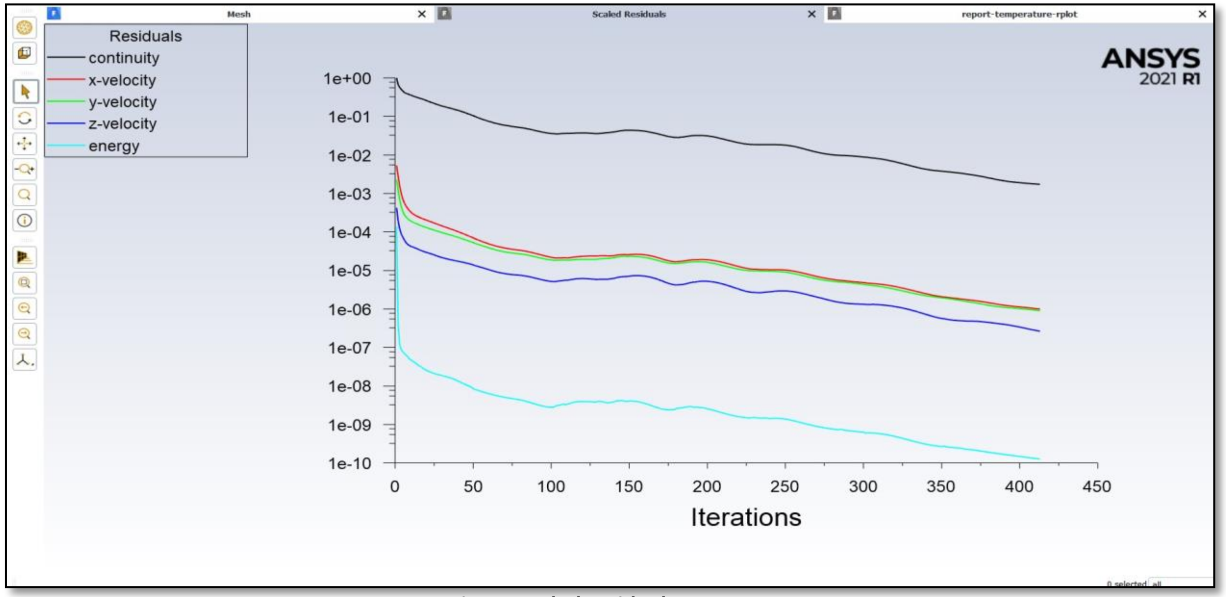


Fig E1: scaled residuals

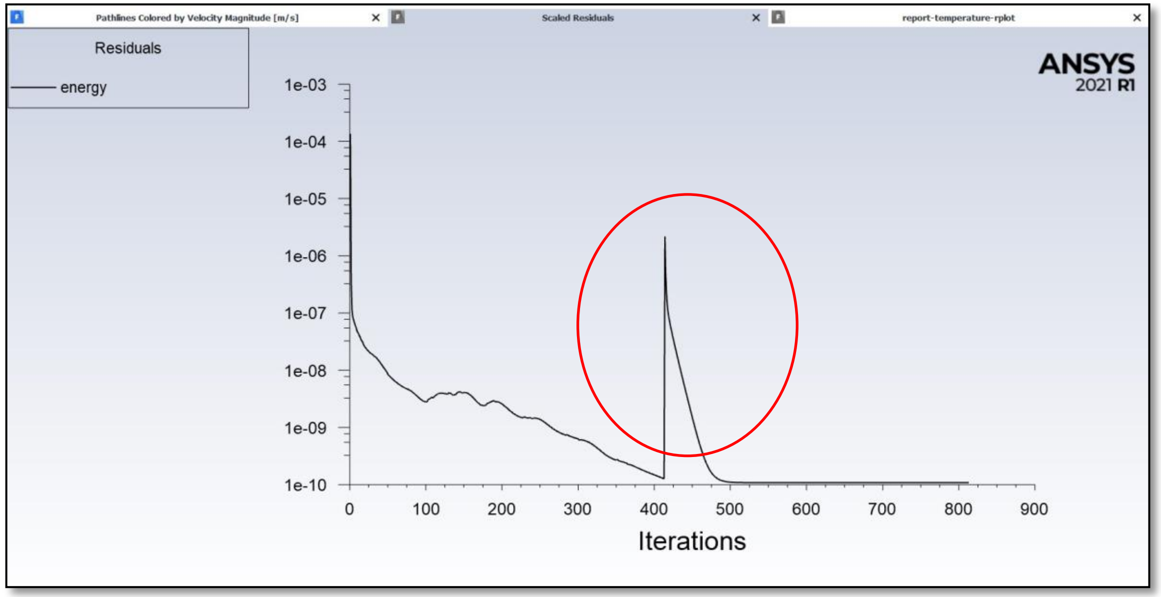


Fig E2: Energy plot

The observed variations arise from elevating the pseudo-transient factor to assess its impact on flow levels.

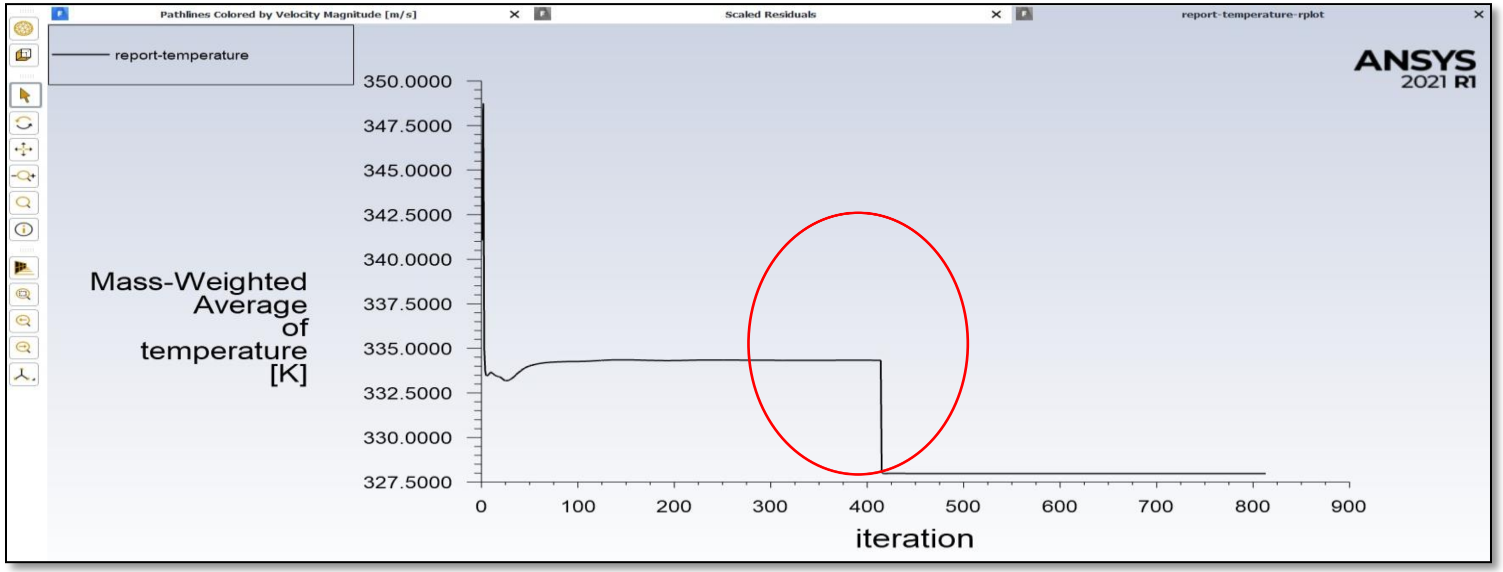


Fig E3: Temperature plot

MESH WITHOUT TUBULATOR:

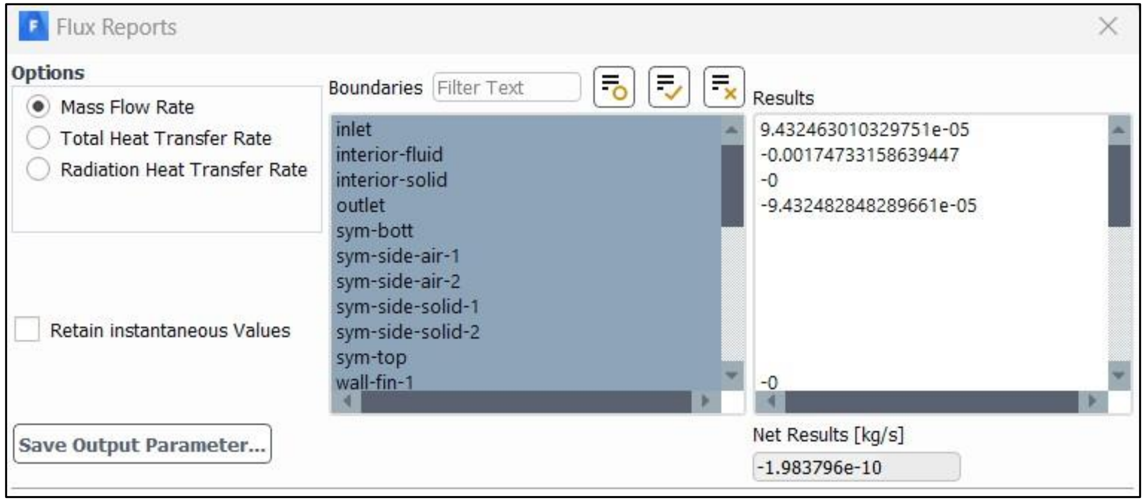


Fig F1: Mass flow Net results

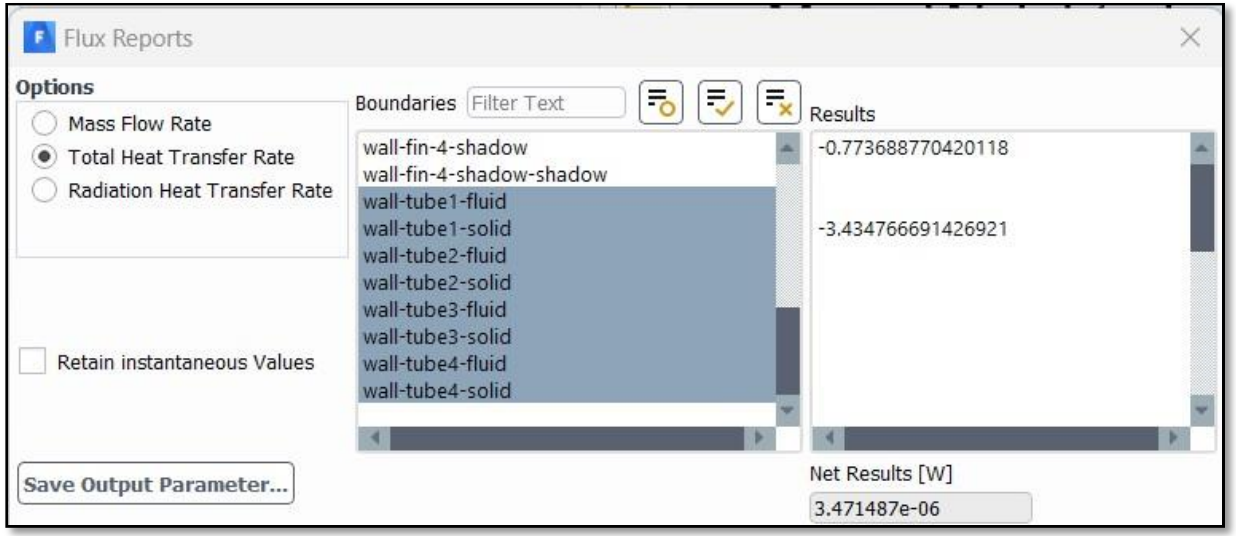


Fig F2: THT - Net results

RESULTS – THERMAL FIELD:

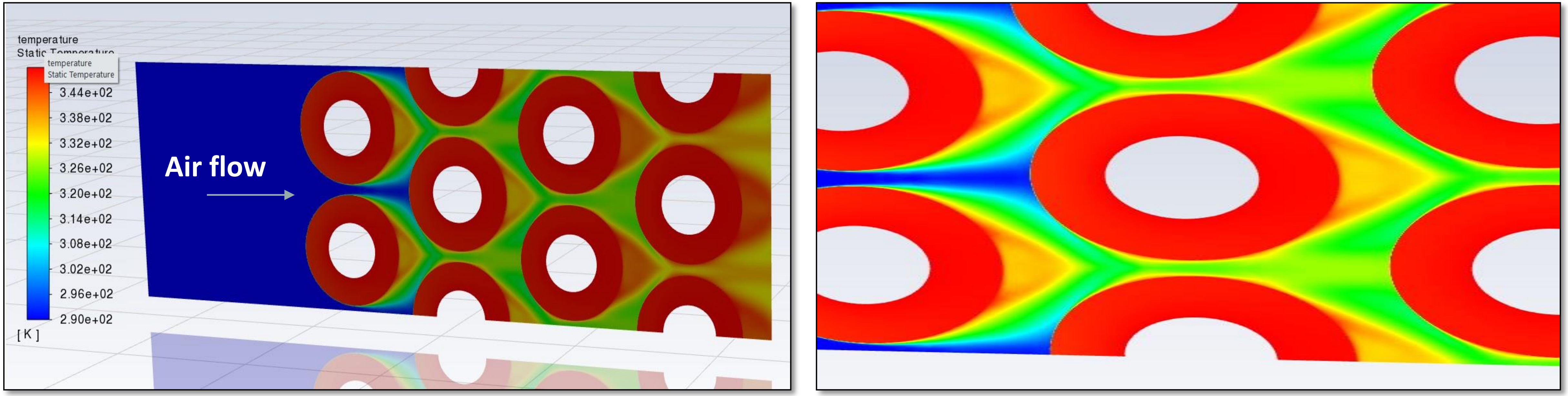


Fig G: Thermal Field results

- The heat transfer across the tube walls and extended surfaces is facilitated by cross-flow convection.
- This phenomenon is evident in the temperature contour plots displayed on the right-hand side.
- Due to its excellent thermal conductivity, aluminum offers minimal resistance to heat conduction. As a result, the aluminum fins exhibit higher temperatures in comparison to the steel fins.
- Consequently, the aluminum fins enable greater heat extraction from the tubes.
- An additional noteworthy observation is that the forward tubes and fins experience a more significant cooling effect than those situated at the rear. This is since the front row is exposed to the incoming cold air stream, while the subsequent rows encounter air that has already been warmed by the preceding rows.

RESULTS – FLOW FIELD :

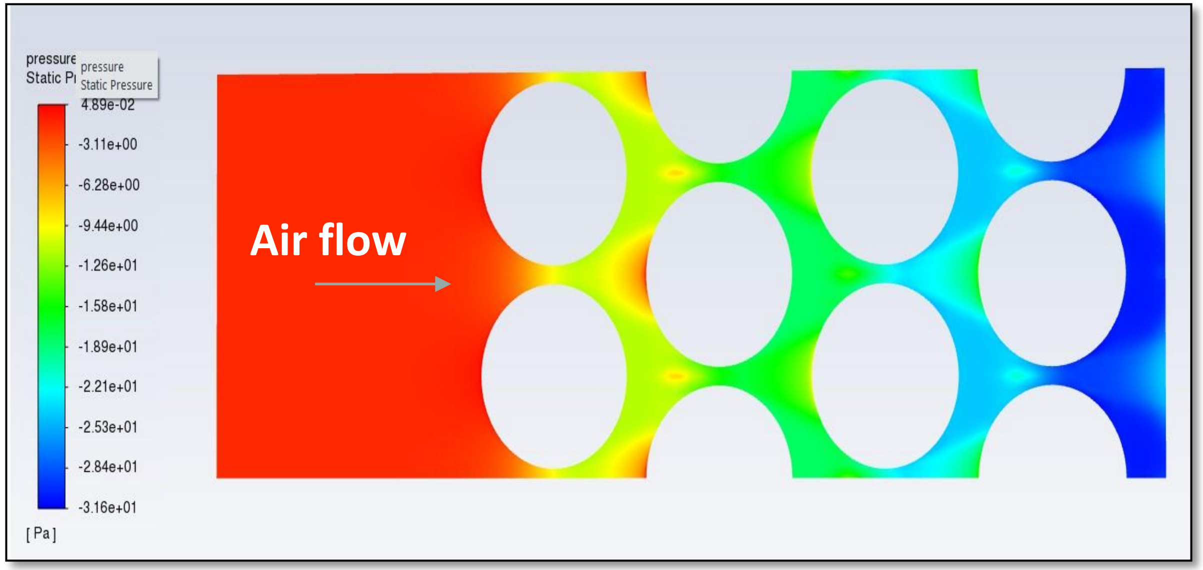


Fig H: Flow Field results

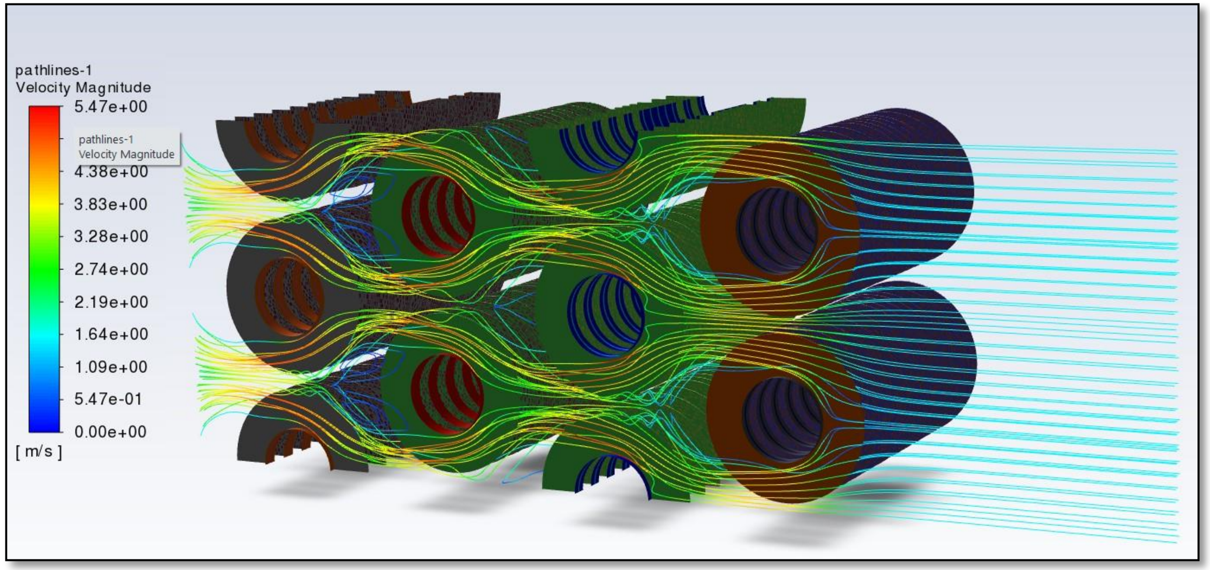


Fig H1: Path lines

- Airflow experiences stagnation zones at the leading edges of the fins and tube walls, halting momentarily before redirecting and enveloping the tube.
- Progressing downstream through the inter-fins gaps, the flow encounters resistance from the viscous friction against the fin and tube surfaces, leading to a gradual reduction in velocity.
- The strategically staggered placement of the tubes results in substantial sections of downstream tubes consistently intercepting the flow. This configuration promotes enhanced heat transfer in contrast to an aligned arrangement.
- Evident in the pressure contour plot below is the pressure drops across the array of tubes.

RESULTS – HEAT TRANSFER ANALYSIS:

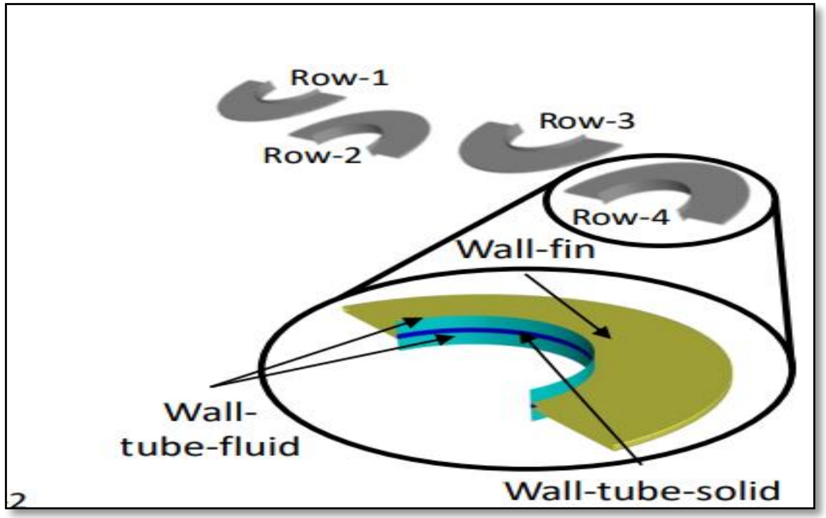


Fig I1: Rows

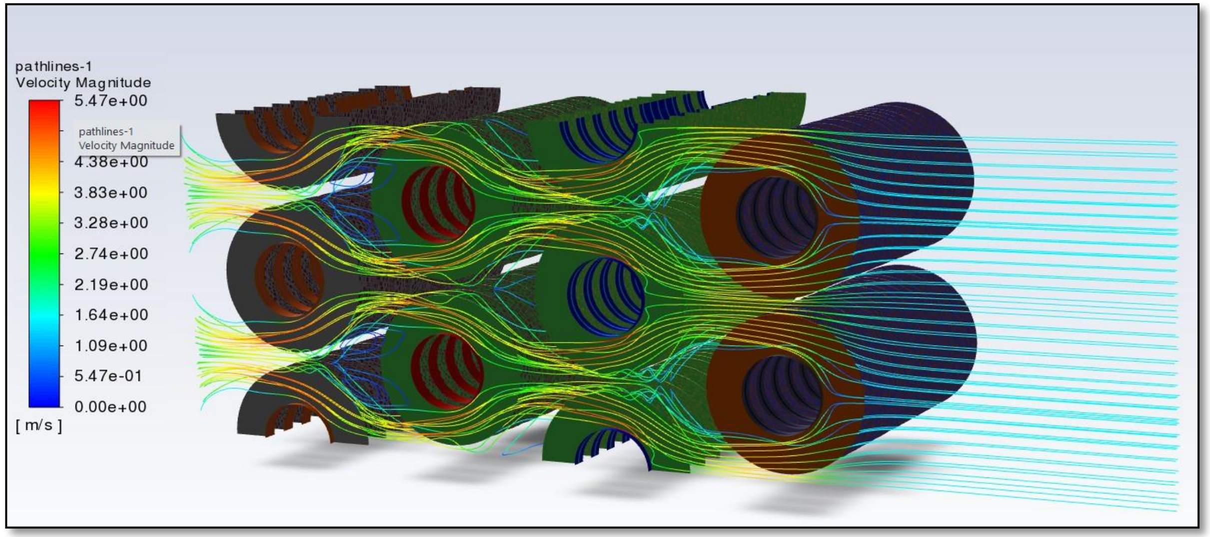


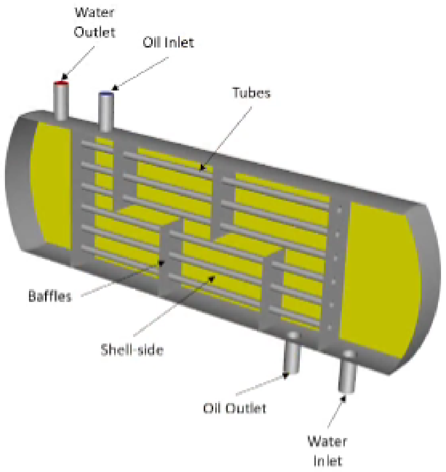
Fig I2: Path lines

- Clearly, the average temperature of the aluminum fins outpaces that of their steel counterparts, attributed to aluminum's heightened conductivity.
- As a direct outcome, the total heat transfer rate experiences a significant upswing with aluminum fins, presenting a contrast to the performance of their steel counterparts.
- Rows 3 and 4 demonstrate a marked reduction in heat transfer when juxtaposed with rows 1 and 2. This differentiation emanates from the influence of a stream of chilly ambient air on rows 1 and 2, whereas rows 3 and 4 grapple with the warmer wake generated by the preceding rows.
- The heat transfer rate from the fins surpasses that from the tube walls due to the expansive surface area conducive to convective heat exchange. Notably, an intriguing observation is the higher heat flux from the tube walls of row 2 compared to those of row 1.
- As the flow negotiates bends and turns around the row 1 tubes, it gains acceleration. This effect is evident in the swifter incident flow encountering the tube walls of row 2, as depicted by the crimson-hued path lines.

PROBLEM DESCRIPTION HEAT TRANSFER IN A SHELL AND TUBE HEAT EXCHANGER :

Problem Description

- In this example, we will be simulating flow and heat transfer in a shell-and-tube heat exchanger that is being used for cooling the transformer oil.
- The model used in this example is a section of a shell-and-tube heat exchanger*.
- The fluid moving through the tube is water:
  - $\rho = 998.2\ kg/m^3$
  - $c_p = 4182\ J/(kg \cdot K)$
  - $\mu = 0.001003\ kg/(m \cdot s)$
- The fluid on the shell-side is oil:
  - $\rho = 850\ kg/m^3$
  - $c_p = 2000\ J/(kg \cdot K)$
  - μ is modeled as temperature dependent using a piecewise linear function.
- The walls of the tank and the inlet and outlet tubes are modeled as adiabatic.



*This model represents a section of a shell and tube heat exchanger and is intended for instructional purposes only. It does not reflect any real-world geometry.

FIG A1 – Problem Description

Boundary Conditions

- Boundary conditions:
  - inlet-shell: Mass-flow Inlet with $0.05\ kg/s$ mass flow rate and constant total temperature of $383\ K$
  - inlet-tube: Mass-flow Inlet with $0.8\ kg/s$ mass flow rate and constant total temperature of $290\ K$
  - outlet-shell: Pressure outlet at $0\ Pa$ gauge pressure and $280\ K$ backflow temperature
  - outlet-tube: Pressure outlet at $0\ Pa$ gauge pressure and $300\ K$ backflow temperature
  - wall-baffles: No-slip wall with coupled thermal boundary condition.
  - wall-in: No-slip wall with zero heat flux.
  - wall-out: No-slip wall with zero heat flux.
  - wall-partition: No-slip wall with coupled thermal boundary condition.
  - wall-tank: No-slip wall with zero heat flux.
  - wall-tubes: No-slip wall with coupled thermal boundary condition.
- A steady-state solution is obtained using the pressure-based solver in Ansys Fluent

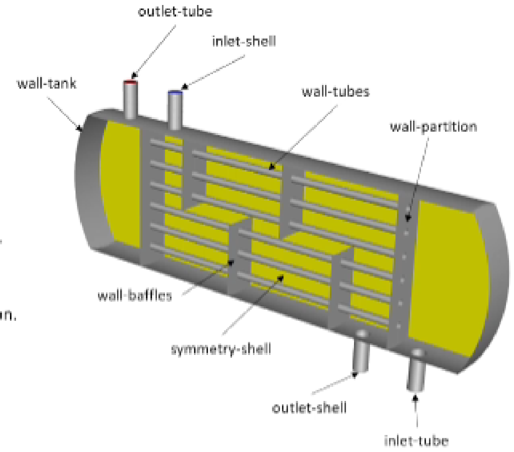


FIG A2 – Problem Description - Boundary Conditions

MESH:

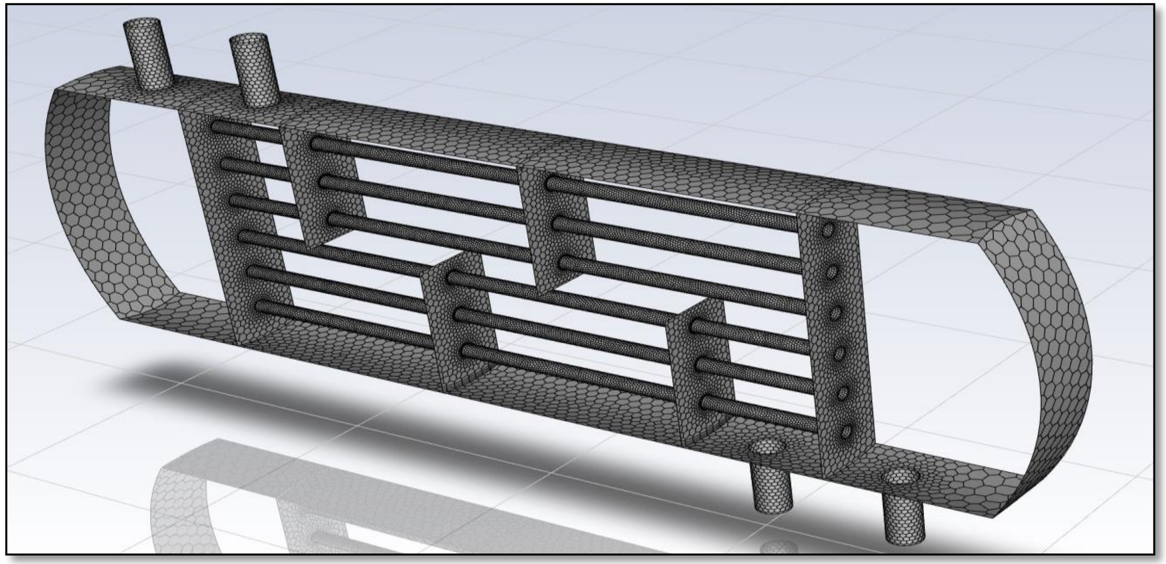


FIG. B1 – MESH

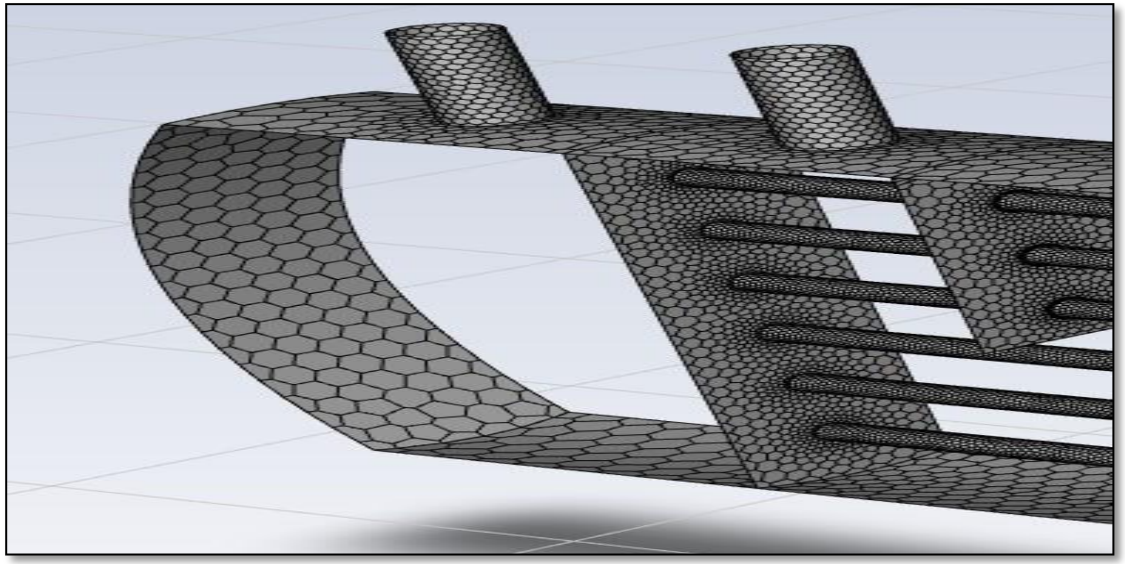


FIG. B2 – DETAILED MESH

```
Mesh Quality:

Minimum Orthogonal Quality =  2.02423e-01 cell 26933 on zone 171 (ID: 58713 on partition: 0) at location ( 3.01440e-01,  1.12285e-01,  7.35561e-03)
(To improve Orthogonal quality , use "Inverse Orthogonal Quality" in Fluent Meshing,
 where Inverse Orthogonal Quality = 1 - Orthogonal Quality)

Maximum Aspect Ratio =  2.15259e+01 cell 20634 on zone 171 (ID: 74800 on partition: 3) at location (-3.71808e-01,  1.52245e-01,  8.53333e-03)
```

FIG. B3 – MESH QUALITY CHECK

With a mesh value exceeding 1.0, the mesh quality is optimal for analysis purposes.

GEOMETRY CONDITIONS:

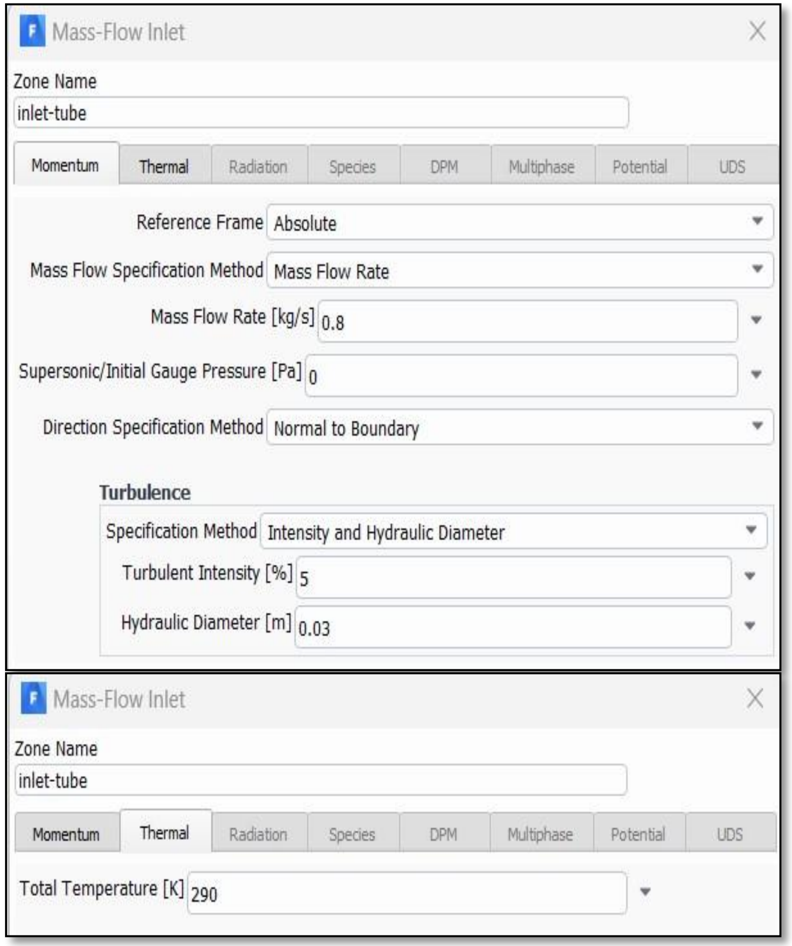


Fig C1: Mass flow Inlet - Tube

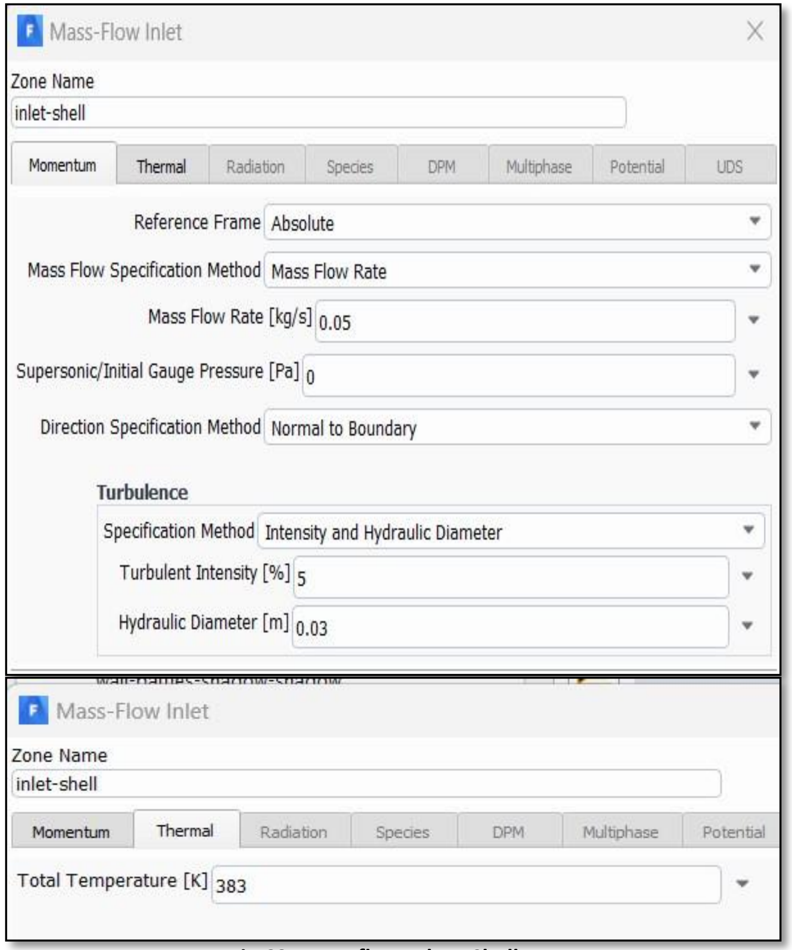


Fig C2: Mass flow Inlet - Shell

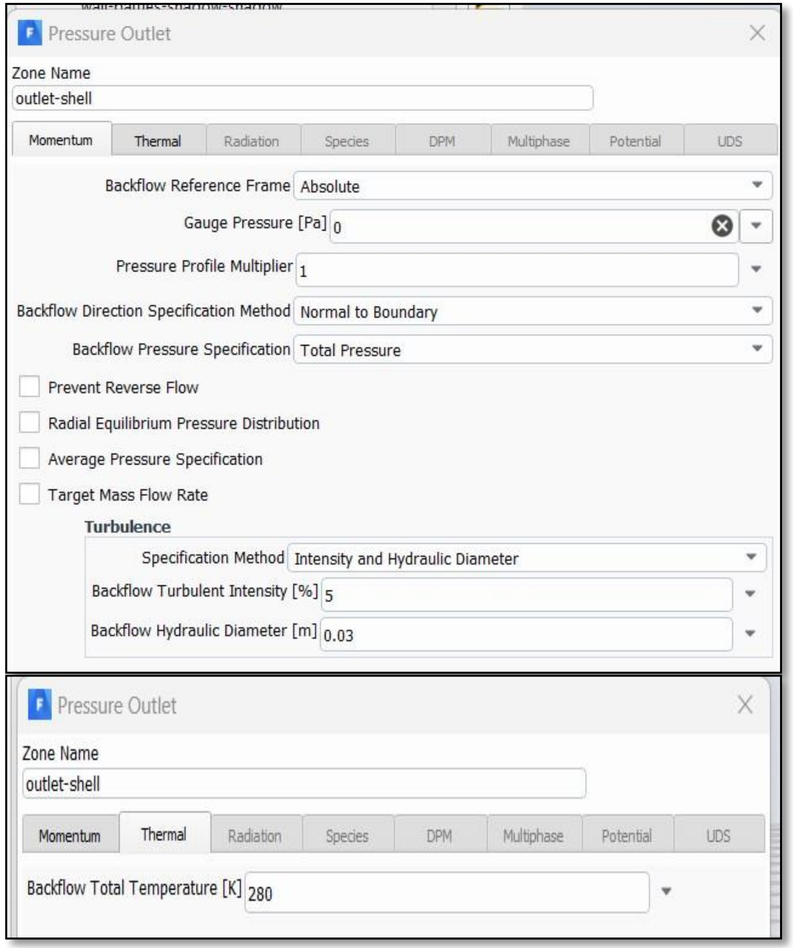


Fig C3: Mass flow Outlet - Shell

GEOMETRY CONDITIONS contd:

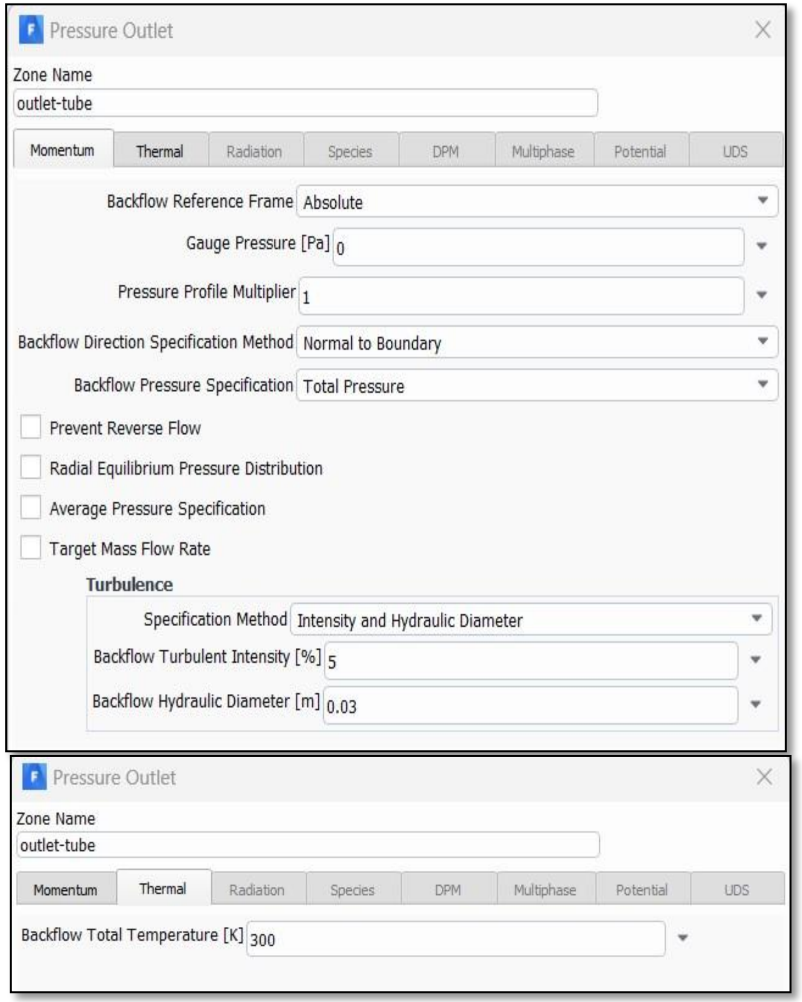


Fig C1: Mass flow outlet - Tube

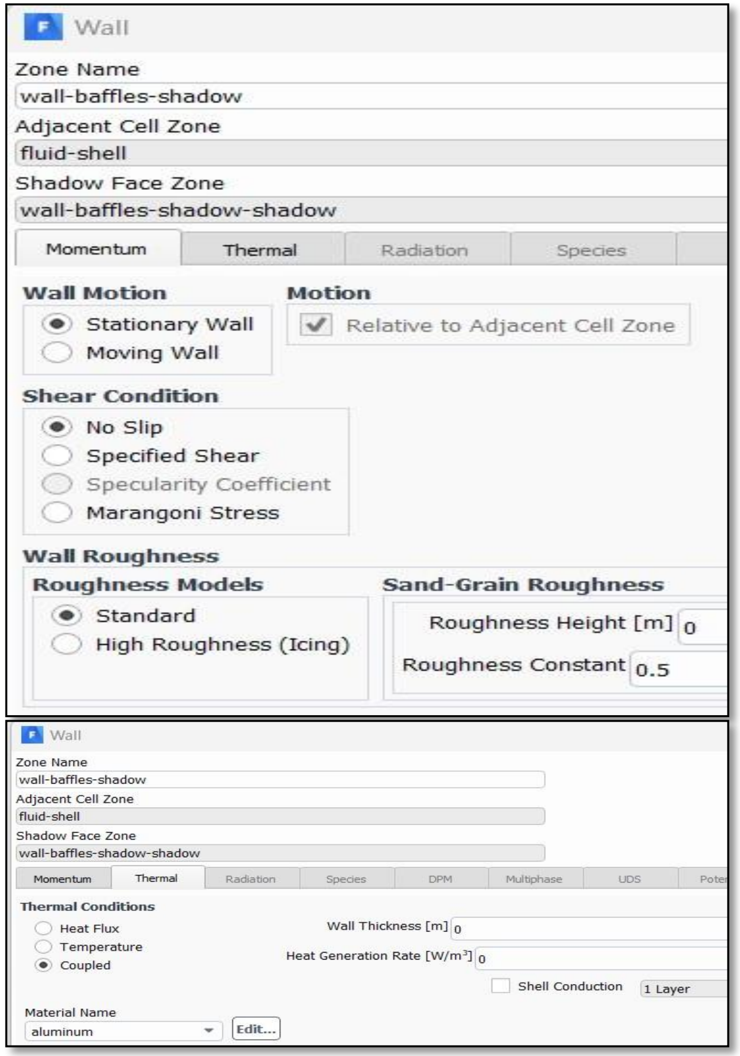


Fig C2: Walls baffles

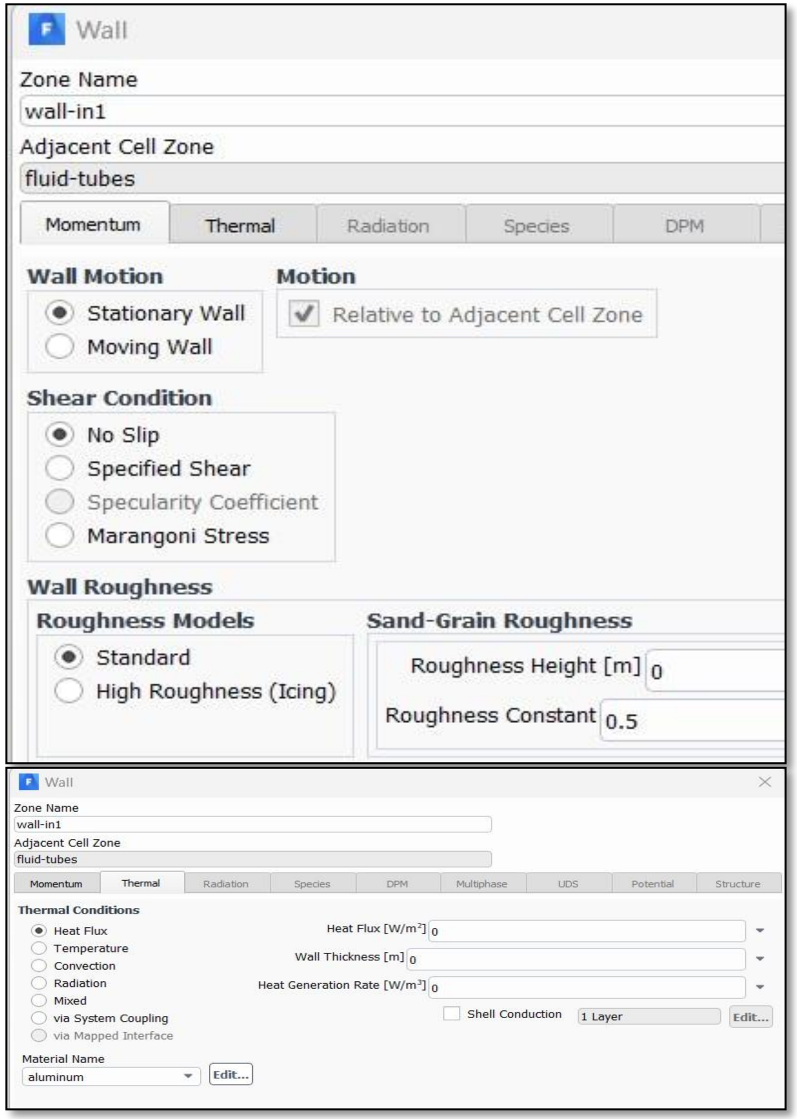


Fig C3: Wall in

SCALED RESIDUALS:

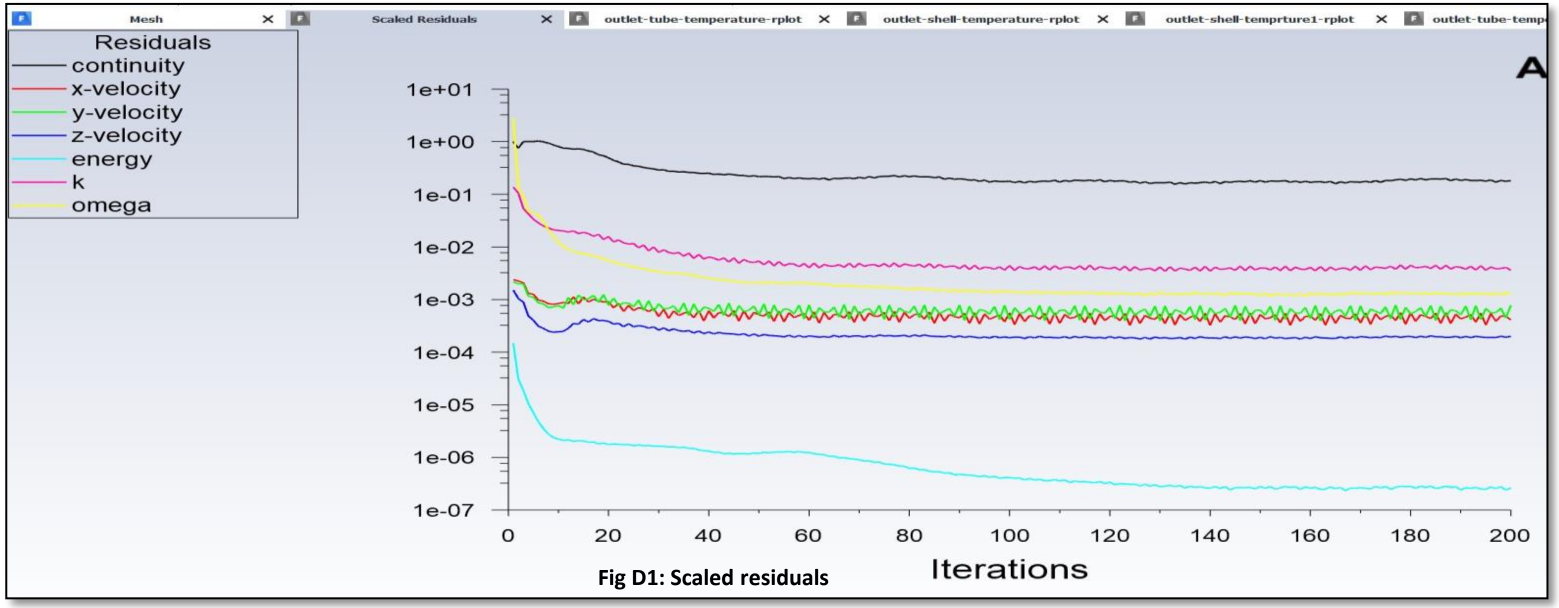


Fig D1: Scaled residuals

- In ANSYS, the term "pseudo-transient" generally refers to a technique used to simulate transient behavior in a steady-state analysis. This technique is particularly useful when you want to simulate dynamic or time-dependent phenomena using steady-state solvers.

- The primary purpose of using the pseudo-transient approach in ANSYS is to replicate transient effects within a steady-state simulation, providing an approximation of how the system would behave over time without explicitly solving the time-dependent equations. This can be beneficial for several reasons:

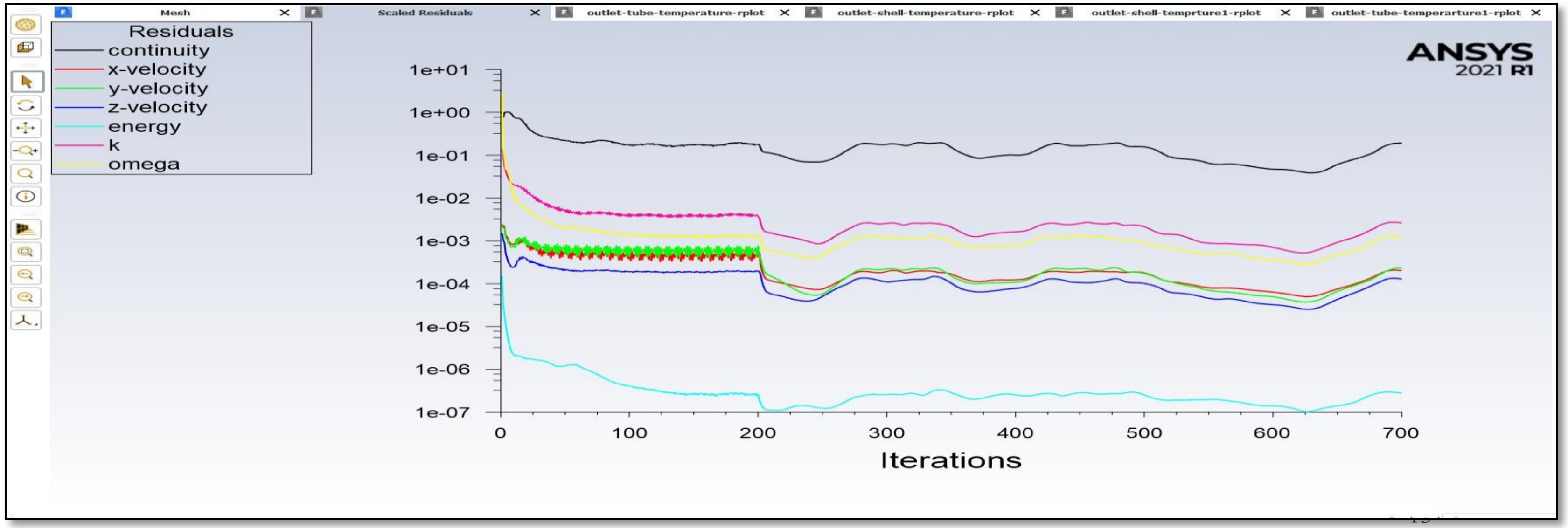


Fig D2: Scaled residuals – Pseudo transient

OUTLET TEMPERATURE PLOT:

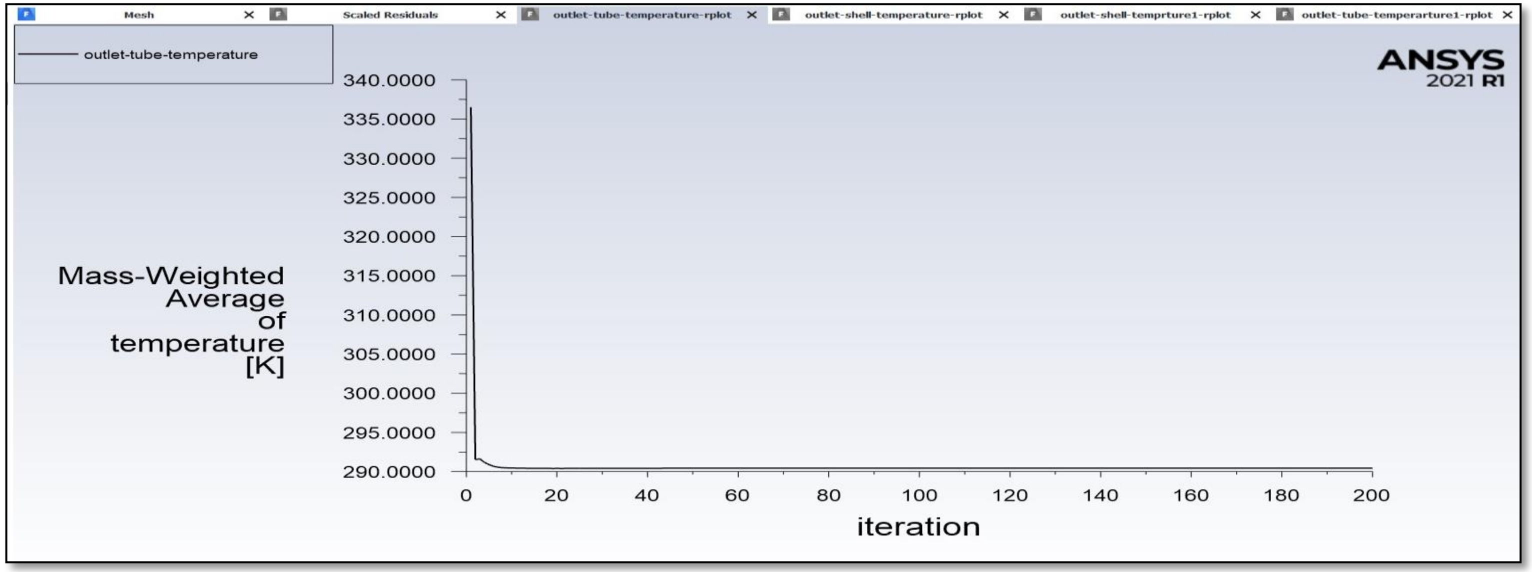


Fig F1: Temperature plot - WATER

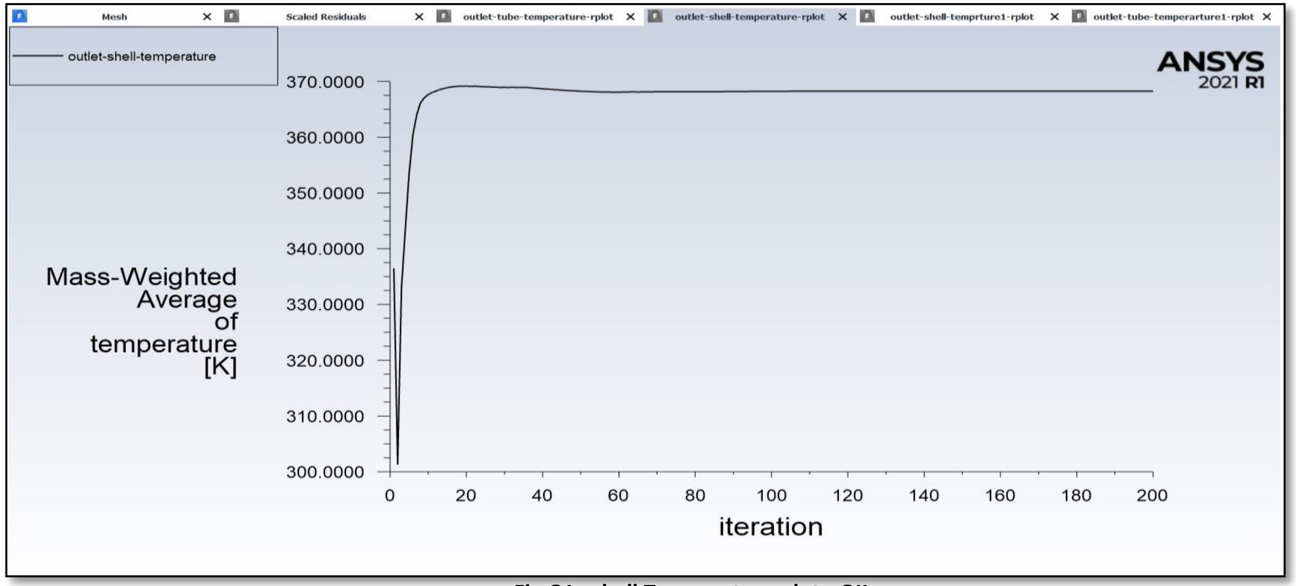


Fig G1: shell Temperature plot - OIL

OUTLET TEMPERATURE PLOT:

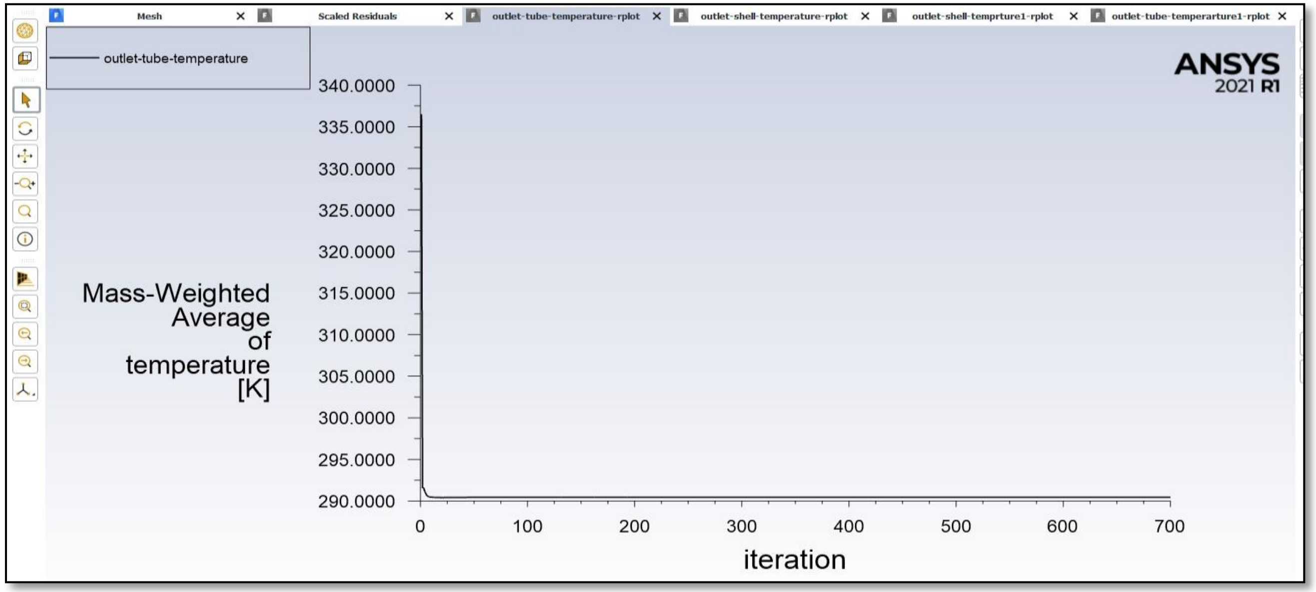


Fig F2: Temperature plot – Pseudo transient - WATER

Convergence

- Nevertheless, the even distribution of outlet temperatures and the successful equilibrium of fluxes attained meet the objectives of this example.

- The primary aim here is to illustrate the fundamental principles of a shell and tube heat exchanger and approximate the outlet temperatures of both oil and water.

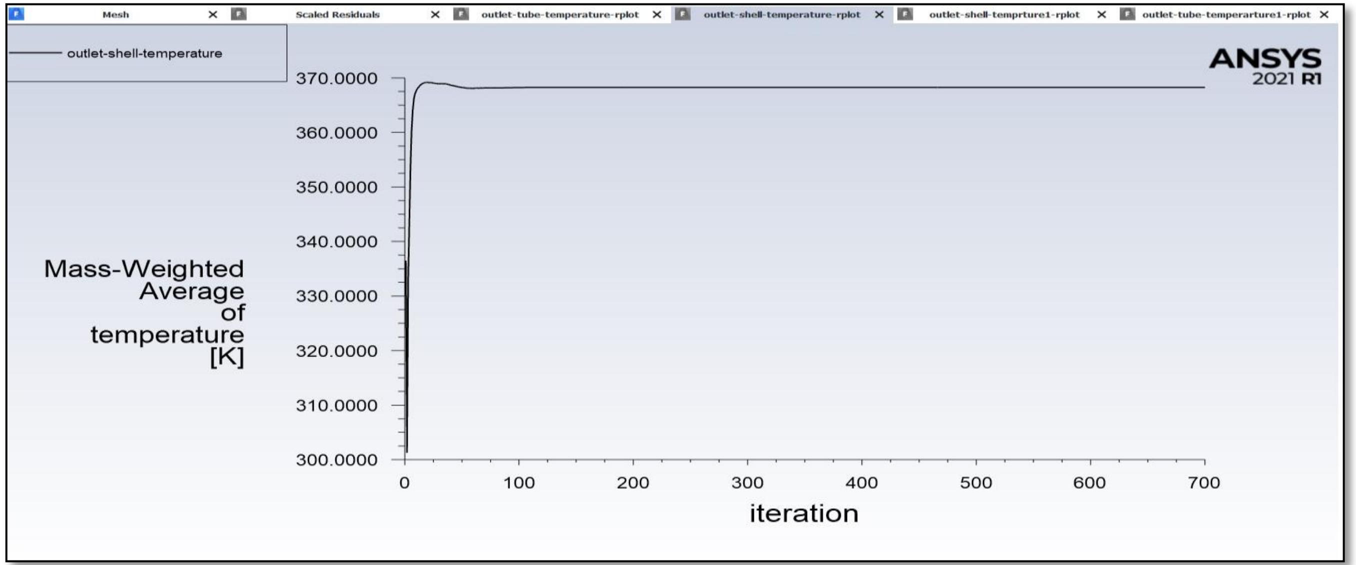


Fig G2: shell Temperature plot – Pseudo transient OIL

FLUX REPORTS:

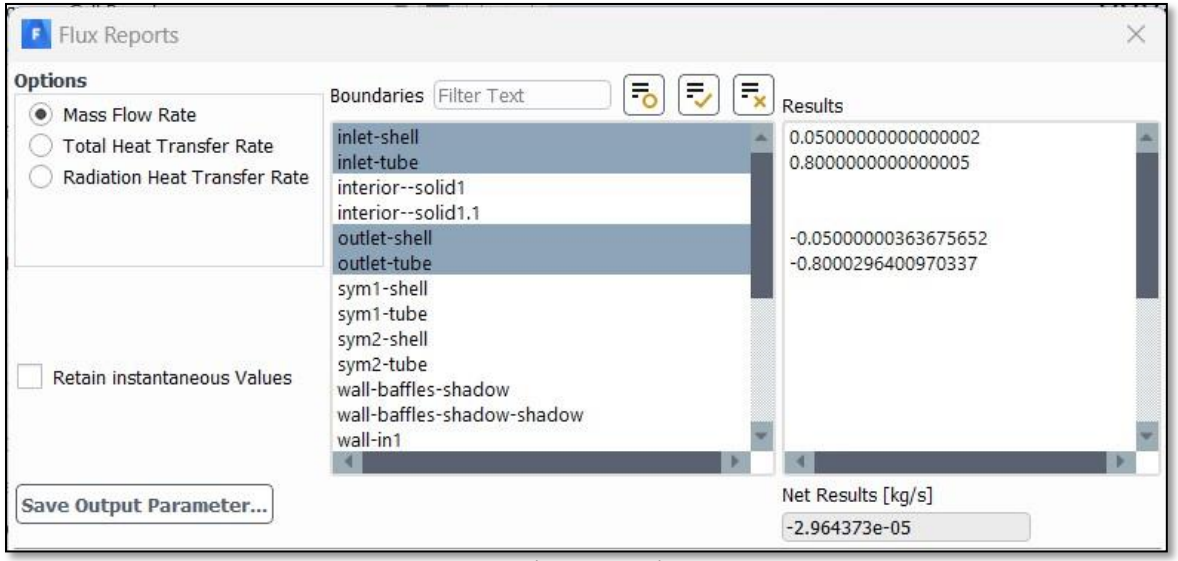


Fig H1: Mass Flow Rate - Flux Report

OIL - Δqo = $|8485 - 7011|$ = 1474W

WATER - Δqw = $|-27266 + 25795\ |$= 1471W

$$\Delta qavg = \Delta qo + \Delta qw \ / \ 2 = 1473$$

- The net heat transfer of 1473 W constitutes a significantly substantial portion of the heat loss experienced by the oil as it flows from the inlet to the outlet. In comparison, this value is comparatively minor when considered in the context of water's heat transfer percentages.

- Consequently, it's reasonable to anticipate a more pronounced temperature reduction in the oil compared to the corresponding temperature rise in the water.

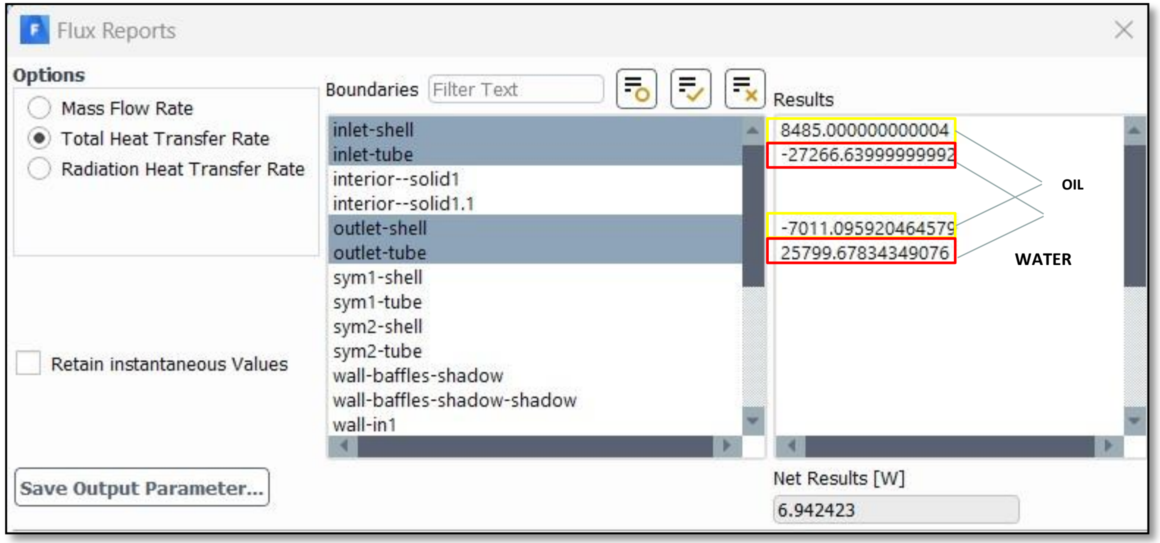


Fig H2: THT – FLUX REPORT

CONTOURS - VELOCITY & TEMPERATURE:

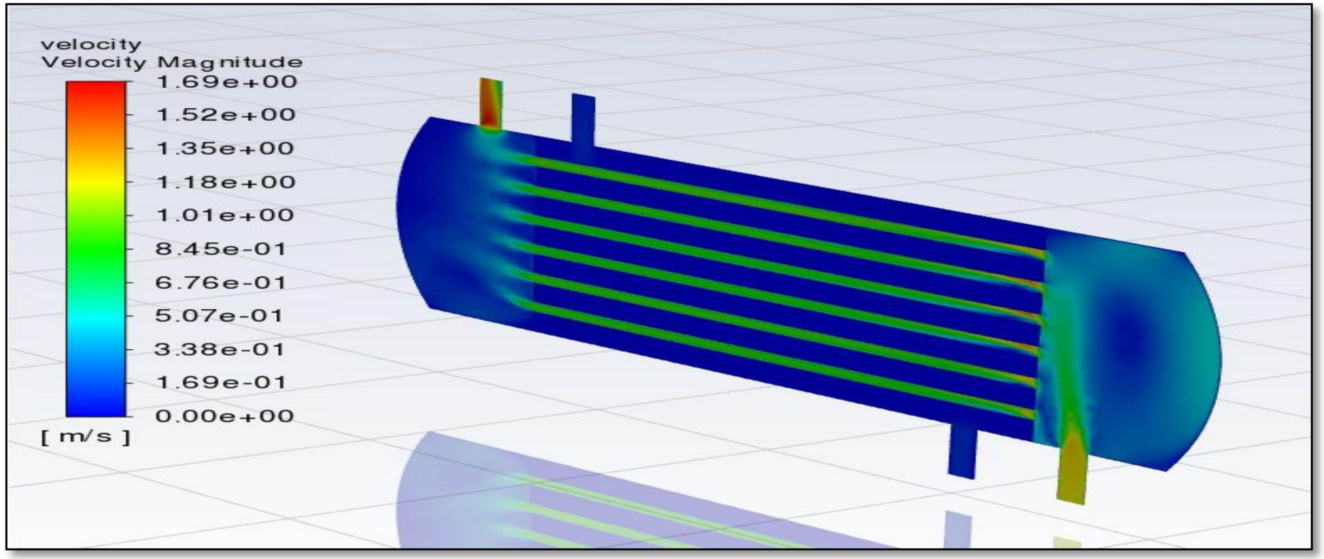


Fig I1 – Velocity Contour

- The image displayed below illustrates static temperature distributions across both the tube and shell regions. The distinct color variations serve to visually represent the temperature fluctuations within the two fluids. Upon entry into the shell, the oil possesses an initial temperature of 383 K. As it progresses, it sheds heat to the tube-side fluid (cooler water) and ultimately exits the shell at approximately 368 K.

- Conversely, the water undergoes a warming process due to heat transfer from the oil while traversing the tubes. Given the water's higher mass-flow rate relative to the oil, the resulting temperature increase is relatively modest compared to the oil. Consequently, the water exits the system nearly at the same temperature it entered.

- The tube walls, as depicted in the contour plots, showcase higher temperatures for water and lower temperatures for oil compared to their respective outlet temperatures.

- Enhancements in heat transfer can be achieved through strategies such as employing more densely packed tubes or incorporating additional baffles.

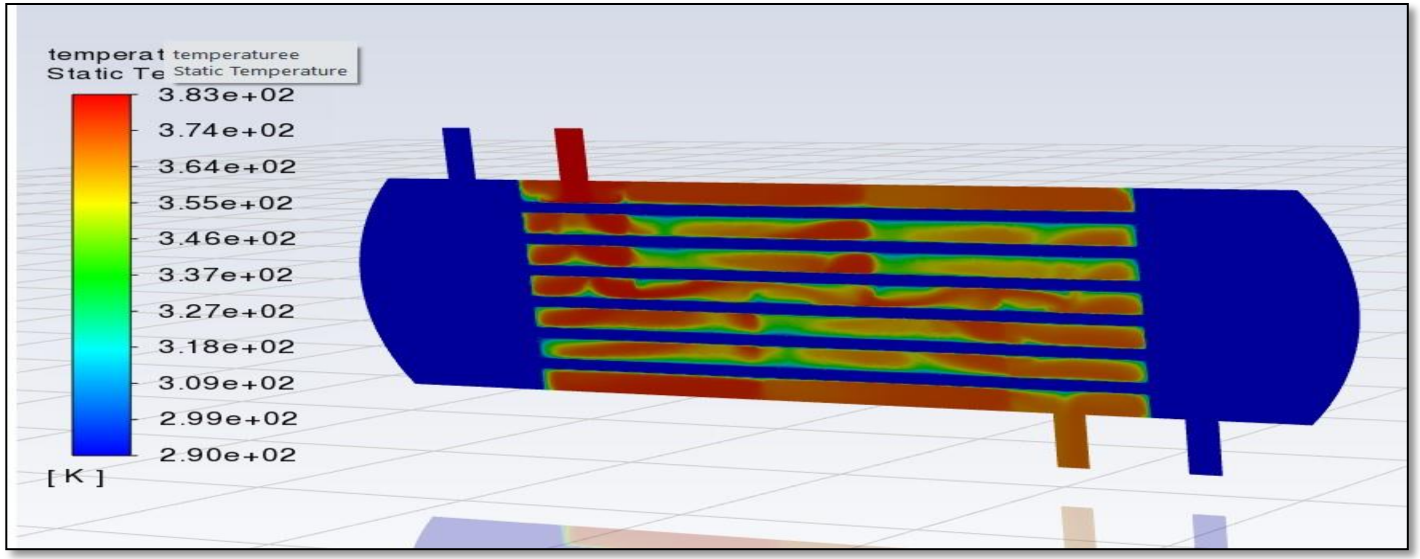


Fig I2 –Temperature Contour

PATHLINES:

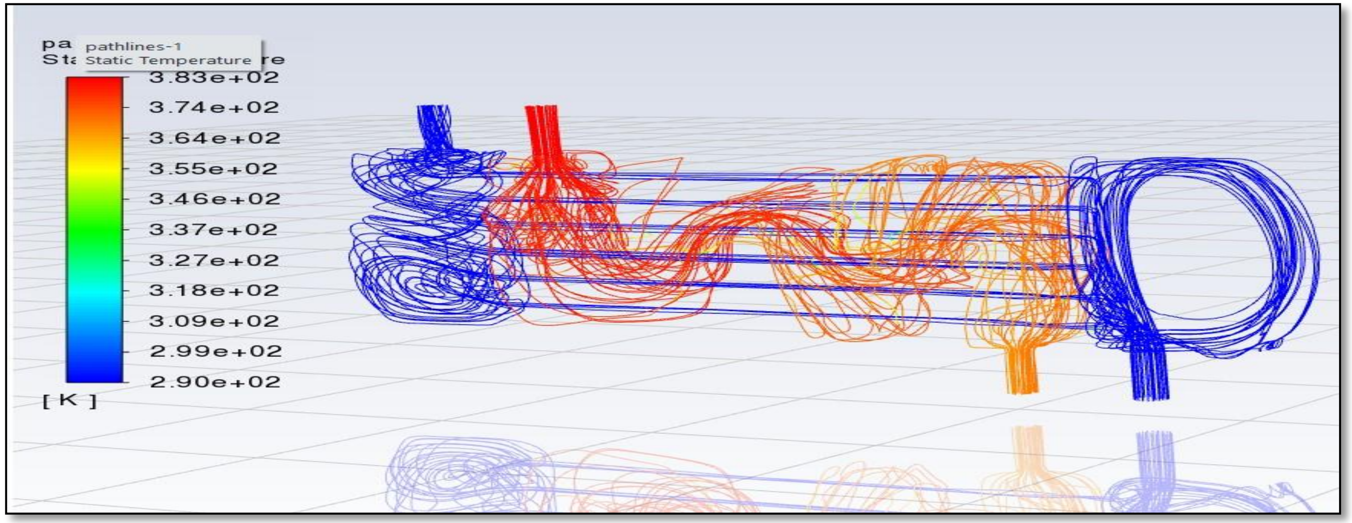


Fig J1 – PATHLINES

- The illustration depicts the turbulent movement of oil within the shell, as it navigates its way around the tubes and baffles. In contrast, the water exhibits a swirling pattern both before entering and after exiting the tubes.
- The highly unsteady motion of the oil and water is what makes this example hard to converge in a steady-state fashion.

- Notably, the water's velocity surpasses that of the oil due to the greater mass flow rate introduced at the system's inlet.
- This dynamic swirling motion of water is a result of its higher mass flow rate at the inlet.

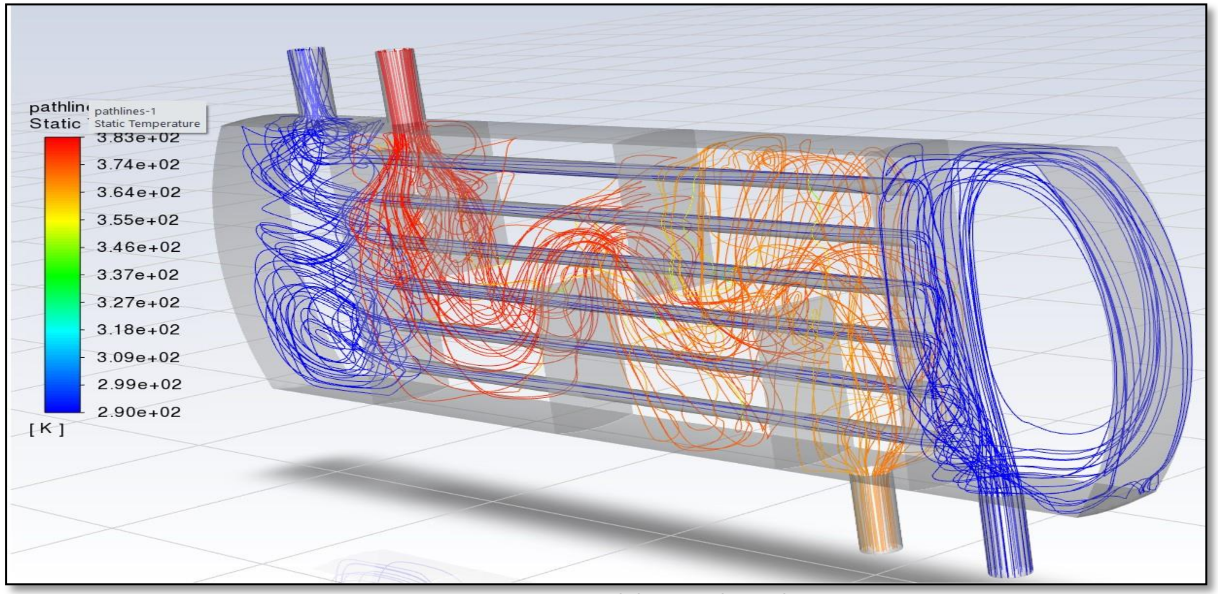


Fig J2 – Path lines with Mesh

THANK YOU!