SSRG International Journal of Industrial Engineering
ISSN: 2349 – 9362 / https://doi.org/10.14445/23499362/IJIE-V13I2P101

Volume 13 Issue 2, 1-9, May-Aug 2026

Original Article

Integrated Mechanical Design and Finite Element Evaluation of a Shell-and-Tube Heat Exchanger for Industrial Applications

M. Prabhahar[1*], K. Jayaraj[1], S. Prakash[1], S. Nallusamy[2]

[1]*Department of Mechanical Engineering, Aarupadai Veedu Institute of Technology, Vinayaka Mission's Research Foundation (Deemed to be University), Paiyanoor, Tamilnadu, India.*
[2]*School of Engineering Management and Continuing Education, Jadavpur University, Kolkata, India.*

[*]*Corresponding Author : mprabhahar@gmail.com*

Received: 02 June 2026 Revised: 04 July 2026 Accepted: 26 July 2026 Published: 17 August 2026

Abstract *- Shell and tube heat exchangers are widely used in industry owing to their dependability, ease of maintenance and ability to handle high-pressure and high-temperature fluids. They are mostly used in power plants, oil refineries and large-scale chemical operations. The purpose of this work is to build a shell and tube heat exchanger that meets ASME Section VIII Division 1 and TEMA standards. The primary purpose is to do mechanical design calculations to determine the permitted thickness of the shell, tube sheet, and other key components so that they meet the ASME allowable stress limitations. The design considers pressure, temperature, corrosion allowance, and material selection. The exchanger is then modeled in appropriate software and analyzed in ANSYS to confirm that the selected variables in the design process contribute significantly to the basic requirement of a safe design and are economically feasible.*

Keywords *- Heat Exchanger, Mechanical Design, ANSYS, TEMA, ASME.*

1. Introduction

Shell-and-Tube Heat Exchangers (STHEs) play a critical role in thermal management for process industries because of their reliability, mechanical strength, and versatility [1]. Their design requires precise engineering of components such as the shell, tube bundles, heads, tube sheets, baffles, tie rods, and nozzles [2-3]. Design and fabrication must comply with (TEMA) guidelines and the (ASME) Boiler and Pressure Vessel Code [4-5].

The shell is a large cylindrical enclosure that surrounds the tube bundle, and baffles direct shell-side fluid across the tubes to enable efficient heat exchange [6]. The tube sheet is a critical portion that holds the Heat Exchanger together [7]. A tube sheet is an important component of the heat exchangers [8]. It holds a number of long tubes that can be placed horizontally to transfer heat [9-10]. It also separates the shell-side and tube-side regions, which are exposed to varying pressures and temperatures [11]. Tube sheet and shell sheet joints are critical because they must bear significant mechanical and thermal stresses [12].

Head-end covers are devoted to the tube sheet on each side of the exchanger to allow tube-side fluid to enter and exit the exchanger [13-14]. These covers form a closed chamber that distributes fluid evenly through the tubes [15]. Their designs vary depending on the exchanger type, such as fixed-head, floating or U-tube configurations [16].

This paper outlines the design requirements for the shell/tube heat exchanger and its components, which are constructed as a tubular shell subjected to an external pressure of 2.5 bar in accordance with ASME Section VIII with, Division 1 TEMA [17-18]. The design is verified under critical load conditions using ANSYS [19]. The following sections present the mechanical design methodology and structural analysis [20]. The latest research methods with the research gap shown in table 1.

(STHE) is one of the most widely used thermal systems in various areas such as power generation, petrochemical dispensation, cooling and chemical industries because of its good strength, ease of maintenance and operation at high pressure and temperature. The mechanical integrity of these exchangers is extremely important because they can incur major economic and safety risks if the stresses are excessive, the temperature changes cause deformation, or the exchanger is improperly designed. Pressure-containing components like shells, tubesheets, channel covers and flanges are typically designed to meet the requirements of ASME Section VIII Division 1 and TEMA standards for reliability and safety.

Table 1. Literature gap analysis

Ref. No	Author(s) & Year	Objective	Methodology	Key Findings	Research Gap
[1]	Fernandes & Krishanmurthy (2022)	Evaluation of tube heat exchanger	Analytical design and simulation	Improved heat transfer performance through optimized design parameters	Limited structural and stress analysis considerations
[2]	SivaChandran et al. (2022)	Experimental investigation of tube heat exchanger performance	Experimental testing under varying operating conditions	Validated heat transfer effectiveness and pressure drop characteristics	Lack of numerical optimization and structural assessment
[3]	Bozorgan (2025)	Multi-objective optimization of a shell heat exchanger for helicopter applications	Design optimization considering weight, cost, and heat transfer coefficient.	Achieved an optimized balance between thermal performance and manufacturing cost	Does not address detailed structural integrity under pressure loading
[5]	Dhongade (2025)	CFD-based technique of shell-and-tube heat exchangers	Computational Fluid Dynamics and comparative analysis	CFD predictions showed better accuracy than traditional correlations	Structural stresses and fatigue behavior not considered
[6]	Rungla et al. (2022)	Design and analysis of a floating-head heat exchanger	Mechanical design calculations and analysis	Demonstrated advantages of the floating-head configuration in thermal expansion accommodation	Limited finite element validation of critical components
[7]	Koçak Soylu & Demirel (2025)	Thermal-hydraulic performance under fouling conditions	Aspen Plus simulation and thermal analysis	Fouling significantly reduced heat transfer efficiency and increased pressure drop.	Structural consequences of fouling-induced thermal stresses not evaluated
[9]	Dassa et al.	Development of a tubesheet geometry design tool	Computational geometry generation and validation	Improved efficiency and accuracy in the tubesheet design process	Requires integration with stress and fatigue assessment models
[13]	Xu et al. (2022)	Thermal deformation analysis of heat exchanger structures	Numerical thermal-structural coupling analysis	Identified thermal deformation effects on exchanger reliability	Focused on printed circuit heat exchangers rather than shell-and-tube systems
[18]	Li et al. (2025)	Investigation of tube fatigue failure in heat exchangers	Numerical analysis with experimental validation	Accurately predicted fatigue failure locations and mechanisms	Limited optimization strategies for fatigue mitigation
[20]	Saffiudeen et al. (2026)	Improvement of tubesheet integrity using GTAW cladding	New assessment and inspection methods	GTAW cladding improved tubesheet life and resistance to degradation	Long-term thermal-mechanical performance requires further study

The current research trends are mainly focused on the enhancement of thermal performance, CFD analysis of heat exchanger flow, fouling analysis, and cost optimization of the heat exchanger. Some researchers have studied the processes of heat transfer augmentation techniques, others have examined the thermal-hydraulic characteristics, and others have used finite element analysis (FEA) to assess the stresses in the structure due to the pressure loading. However, most of the previous investigations address separately the thermal and structural analysis of the heat exchanger separately, and only a few of them consider the coupled thermo-mechanical performance of a heat exchanger under real operating conditions. In addition, most of the existing literature focuses on the validation of traditional designs without considering more sophisticated structural optimization methods that can lead to lightweight and less material designs while still meeting the requirements for ASME code.

In order to overcome these drawbacks, the present research work suggests a complete design and analysis of a shell and tube heat exchanger according to ASME section with VIII division 1 and TEMA standard and then a detailed finite element analysis. The novelty of the proposed work is in the ability to combine code-based mechanical design with advanced thermo-structural evaluation and optimization methods. Special attention is given to studying the distribution of stress, deformation properties and structural integrity under combined pressure and thermal loading conditions. The results of this research will be used to aid the design of safer, lighter and less expensive heat exchangers for use in the modern industrial environment that meet the full requirements of international design codes.

2. Methodology

2.1. STHE Nomenclature

Shell-and-tube heat exchangers consist of several key components. Figure 1 illustrates the main parts of a fixed tube sheet shell-and-tube heat exchanger. TEMA standards specify various configurations for different applications. For this study, a 'B' type bonnet (integral cover) serves as the front stationary head, an 'M' type rear head with a fixed tube sheet is used as the rear stationary head, and an 'H' type double split-flow shell is selected. The primary objective is to calculate the required thicknesses for the shell, tube sheet, channel head, and nozzle neck, and to assess the need for stiffeners in accordance with ASME standards.

2.2. Terminology used in STHE

2.2.1. Design Pressure

Design pressure is a key factor in determining the minimum required thickness of heat exchanger components, as specified by ASME Section VIII with Division 1.

This should be a minimum of the maximum pressure that the pump will encounter during normal use, including vacuum, external loads and possible pressure surge. The shell is designed to be a vacuum (0.027 bar abs). The most important factor is the outside pressure on the shell, because the pressure inside is less than that of the air surrounding the shell. The shell thickness is calculated based on the external pressure design criterion from UG-28 of ASME Section VIII, Division 1. The design external pressure is conservatively taken as 2.5 bar to ensure safety and to take into account environmental variations.

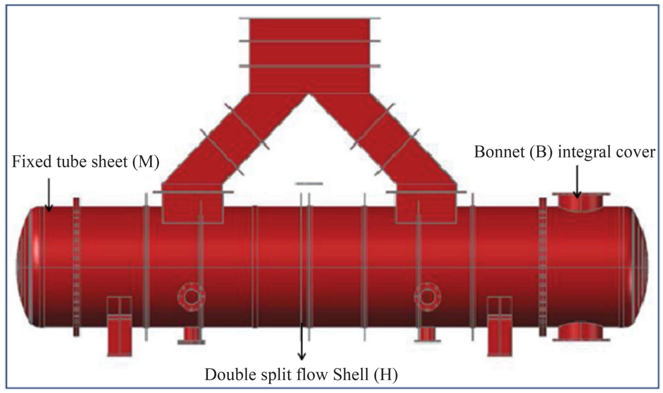

Fig. 1 Parts of Fixed tube Heat exchanger

2.2.2. Design Temperature

Design temperature is also an important factor in the calculation of minimum thickness and maximum stress in pressure-containing parts. As per ASME Section VIII with Division 1, it should be the maximum metal temperature during the operation, with consideration of temperature fluctuations [31]. Considering uncertainties, thermal gradients, as well as environmental variation, the design temperature is taken above the maximum operating temperature. Thus, the allowable stress values are chosen from the appropriate ASME Section II Part D material tables [32] for a design temperature of 40°C.

3. Materials Selection

The cooling medium on the tube side is seawater, and the water vapor condenses on the shell side of the heat exchanger. The shell is chosen to be made of austenitic stainless-steel SS 304 because of its corrosion and fouling resistance significant for seawater applications. ASME Section II, Part D confirms these properties, they are specified by section SA-240 type 304. Established key material parameters for design are a modulus of elasticity of 200 GPa, a yield stress of 205 MPa, a Poissons ratio of 0.30 and a mass density of 7800 kg/m³. These values support SS 304's suitability for resisting corrosion and fouling in seawater service.

Table 2. Shell and Tube Heat Exchanger Specifications

Type	**:**	**BHM (Horizontal)**
Material	:	Stainless Steel 304
Yield Stress	:	205 MPa
Modulus of elasticity	:	200 GPa
Shell Inside diameter	:	1.2 m
Tube diameter	:	0.01905 m
Thickness of tube	:	BWG 19 (1.067mm)
Tube Length	:	4.8 m
Baffle type	:	Segmental 25% cut
Tube sheet	:	Fixed tube sheet
No. of Pass	:	Two Pass
Tube pattern	:	Triangular pattern
Tube Pitch	:	1.5 Pitch (28.575)
No of tubes	:	1200 nos

3.1. Design of Components

3.1.1. Shell Thickness under External Pressure

A minimum thickness of the shell under external pressure was calculated using material properties and operating conditions following ASME Section VIII with std Division 1 under (UG 28). The shell was supposed to be a cylindrical vessel with a tori-spherical type of channel heads. For the calculation of initial shell thickness, a value of 6 mm was assumed, which is typical for the industry. The account for metal loss during service, corrosion allowance of 1 mm at the sides was added, giving an evaluated thickness of 8 mm to maintain the integrity for a long time. According to code requirements, the following parameters were decided: The length of the condenser (L_o):

$$L_o = 4.88 + \frac{1}{3}(1.093) + \frac{1}{3}(0.593) = 5.442 \text{ m} \quad (1)$$

Diameter-to-thickness ratio (D_o/t):

$$D_o/t = 152$$

Length-to-diameter ratio (L/D_o):

$$L/D_o = 4.4753$$

Using the calculated D_o/t and L/D_o ratios, the strain factor was obtained from Figure 2 with ASME Section II Part D. Then, the equivalent stress factor was determined from the HA-3 chart.

Strain Factor (Factor A):

$$A = 0.00015 \text{ (Figure 2, Section II, Part D)}$$

Stress Factor (Factor B):

$$B = 12{,}500 \text{ kPa (HA-3 Chart, Section II, Part D)}$$

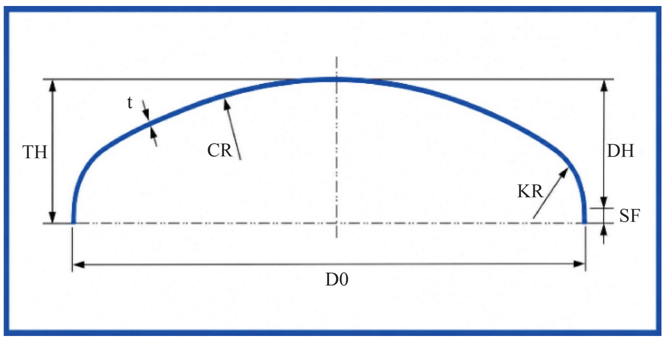

Fig. 2 Tori Spherical Head

The allowable external working pressure (P_a) is calculated as:

$$P_a = \frac{4B}{3(Do/t)}$$

$$P_a = 109.64 \text{ kPa}$$

The calculated permissible working pressure was compared with the design pressure.

Design Pressure > Allowable Working Pressure
250 kPa > 109.64 kPa

The design pressure exceeds the allowable working pressure, the provided shell thickness of 8 mm is insufficient per code requirements.

Increasing shell thickness would raise weight, material usage, and fabrication costs. To maintain structural safety while controlling these factors, stiffening rings are added to the shell design. Their geometry improves resistance to external pressure without significantly increasing weight or cost.

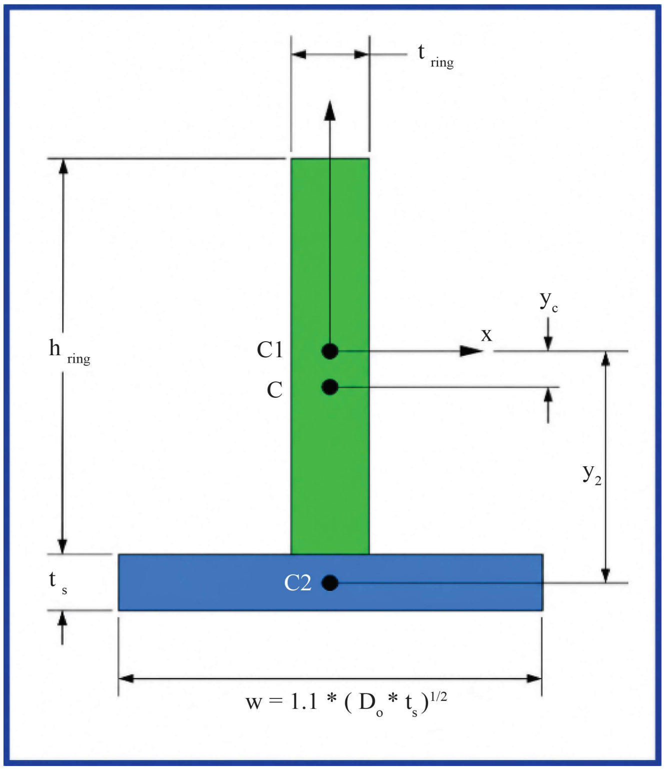

Fig. 3 Shell with Stiffener

3.1.2. Stiffeners Requirements

Stiffeners are provided to enhance structural stability and to prevent shell buckling under external pressure or vacuum conditions. Their use permits the adoption of thinner shell sections, resulting in a reduction in material thickness and overall cost. By increasing the local moment of inertia of the shell, stiffeners improve the vessel's resistance to deformation and structural failure, as shown in figure 3. As per UG 29, Sec VIII, Div 1, the contribution of the length of the condenser cross-section is restricted to the value of

(w)= 1.1*(Do*ts) 0.5 = 108.5 mm
Assume Thickness of Stiffener (tr):8mm
Thickness of ring (tr):8mm
Height of the ring (hr):130mm
Distance between the rings (Ls):685mm
Cross-sectional area of stiffener (As):1907mm^2
Area of Ring (Ar): 1040mm^2
Area of Base Plate (As):868mm^2

The value of stress factor B is calculated as per the following relation

Stress factor, (B): 21145 kPa

As per UG 29 Sec VIII, Div 1, the corresponding value is out of the chart HA3 of Section II, part D, and A is subsequently calculated using the formula and result shown in figure 4.

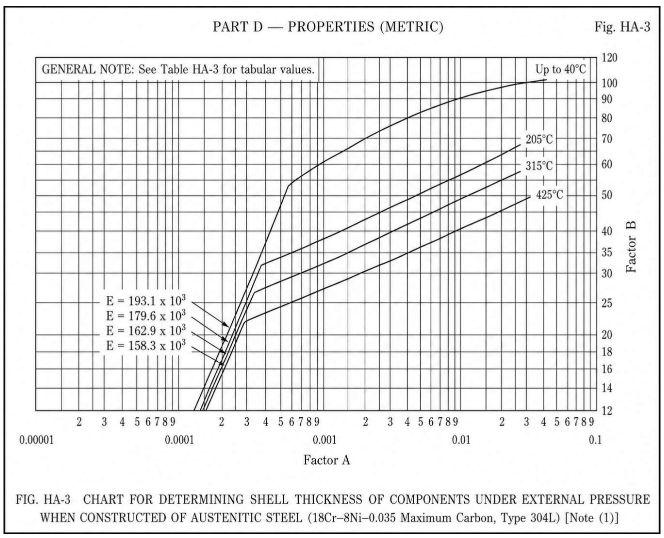

Fig. 4 HA3 Chart for stress factor

A = 2*B/E
Strain Factor A = 0.000219
Required Moment of Inertia for without shell combination of Stiffeners,
Is: Do2Ls (t + As/Ls) A/14 = 171124 mm4
Required Moment of Inertia with shell combination of for Stiffeners,
Is’:Do2Ls(t + As/Ls) A/10.9 =219793 mm4

The available Moment of Inertia for the ring shell cross section was obtained as geometry provided is calculated as follows:
Centroid of given section assembly along y axis (C): 41.61 m
Centroidal Distance for Ring (Rr): 31.38 mm
Centroidal Distance for Base Plate (Rs): 37.61 mm
I1 = tr*hr3/12 = 1464666 mm4
I2 = w*ts3/12 = 4629 mm4
Arrr2 = hr*tr*(65-41.61)2 = 568924 mm4
Asrs2 = w*ts*(41.61-4)2 = 1227794 mm4
Total Available Moment of Inertia = 3266014 mm4
Total Available Moment of Inertia (3266014) >Required Moment of Inertia (Is’=219793)

Since the available moment of inertia is greater than the required moment of inertia, the shell with its stiffeners is safe for the design condition.

3.1.3. Tori spherical Head Thickness

A tori spherical head is widely used as an end closure in pressure vessels and storage tanks in unity with ASME Section VIII Division 1 (UG-23). The spherical crown region is connected to the cylindrical shell through a knuckle section.

This configuration provides improved stress distribution compared to flat heads. Additionally, tori spherical heads are economical and relatively easy to fabricate.

In the present design, a tori spherical head is selected for both ends of the heat exchanger. The head thickness is assumed to be 10 mm, considering fabrication feasibility and handling requirements during assembly with channel heads.

Inner diameter of the shell (Di): 1200 mm
Thickness of the head (th): 10 mm
Design Pressure (Pd): 250 kPa
Young’s modulus (E): 193 x 106kPa
Crown (or) Tori spherical radius (Cr): 1220 mm
Knuckle radius (Kr): 0.1*1220 = 122 mm
Straight flange height (SFh): 3.5*th=3.5*10= 35 mm
Depth of dish end (Dh): 0.1935 Do – 0.455 th =231.52 mm
Total internal height of the head (THi): SFh+Dh=35+231.52 =266.52 mm
Ratio of (Ro/th):122

Find the strain factor (A) using below relations:

$$A = \frac{0.125}{(Ro/t)}$$

Strain factor (A) = 0.001025
Using the strain factor, find out the corresponding stress factor from HA3 chart
Stress factor (B) = 60000 kPa
Maximum Allowable Working Pressure (Pa):

$$Pa = \frac{B}{(Ro/t)}$$

Pa = 491.803 kPa

The design is validated by comparing the design pressure with the allowable pressure:

$$Pd < Pa \quad 250 < 491.803$$

Since the allowable working pressure exceeds the design pressure, the selected head thickness of 10 mm is adequate and satisfies ASME code requirements.

3.1.4. Inlet /out let Nozzle Neck Thickness

Nozzles are critical components of pressure vessels, allowing fluid entry, exit, and instrumentation. Their design must maintain structural integrity under pressure, external loads, and local stress concentrations. As per ASME Section VIII, Division 1, UG-45 specifies the minimum required thickness for nozzles, taking into account pressure and mechanical strength requirements. The nozzle thickness shall not be less than the thickness required for the shell or head to which it is attached.

Inside Diameter of Nozzle, (Di) : 474 mm
Outer Diameter of Nozzle (Do) : 490 mm
Effective Length of Nozzle (Le) : 200 mm
Design Pressure (Pd) : 250 kPa
Assumed Thickness (ta) : 6 mm
Corrosion Allowance : 1 mm per side
Young's Modulus (E) : 193 x 106 kPa
Diameter Thickness Ratio (Do/ta) : 61.25
Length Diameter Ratio (Le/Do) : 0.40816
Strain Factor, A (Fig. G, Sec II, part D): 0.008
Stress Factor, B (Curve HA3, Sec II, part D) : 85000 kPa
This value of A falls to the right of the material line in the curve HA3, that is used to determine the all-out allowable working pressure, Pa. (As per of UG45),
Allowable Pressure (Pa):1850 kPa

Since the project pressure is less than the allowable working pressure, the provided width is sufficient as per code. So the design is safe.

3.1.5. Tube Sheet Thickness as per TEMA standards

The tube sheet is a critical component of a tube heat exchanger. The shell-side and tube-side fluids and supports the tubes to ensure proper alignment and structural stability. In a fixed-tube-sheet design, the tubes are rigidly attached to the tube sheet, which is fused to the shell, resulting in a compact, cost-effective configuration.

The tube sheet design must account for pressure loading, mechanical loads. The present analysis is carried out in accordance with TEMA standards.

Inner diameter of the shell, (Di) : 1.2 m
Outer Diameter of the shell, (Do) : 1.216 m
Thickness of the shell (ts) : 0.008 m
Number of tubes (Nt) : 1200 nos
Outside dia of the tubes (do) : 0.01905 m
Tube wall thickness (tt) : 0.001067 m
Code allowable stress in tension (S): 115 MPa
Shells without expansion joints (J) : 1
Tube length between inner tubesheet faces (L): 4.784 m
Tube length between outer tubesheets faces (Lt): 4.888 m
Fixed tube sheet exchangers, G shall be id of shell: 1.2 m
Elastic mod of the shell material at mean metal temp (Es):193 GPa
Elastic mod of the tube material at mean metal temp (Et):109.6 GPa
For supported tube sheets, gasketed on both sides (F): 1
Coeff of thermal expansion of shell (αs): 0.000012
Coeff of thermal expansion of tube material (αt) : 0.000017
Tube sheet Thickness for Bending (TB):
Effective bolting pressure (Pbt): 408.31 kPa
Effective differential design pressure (P) : 449 kPa
Effective pressure for testing (Pe): ((Pbt+ P)*1.3)
: ((408.31+449)*1.3)=1114.50 kPa
Tube Pitch: 0.028575 m
Mean ligament efficiency (η):
Tube sheet Thickness for Shearing = 10.09 mm

The final tube sheet thickness was selected by comparing the bending and shearing thickness values; the maximum value, 51.04, was selected as the tube thickness.

4. Results and Discussion

4.1. FEA

The Heat Exchanger was modelled using PRO-E and ANSYS software. The geometry of the Dish end was created in Pro-E software and imported into the ANSYS software in IGES format. Except dish end all other components were created in ANSYS using its modelling commands. Glue and overlap commands were used to integrate the components. The Model of the Heat Exchanger was as shown in Figure 5.

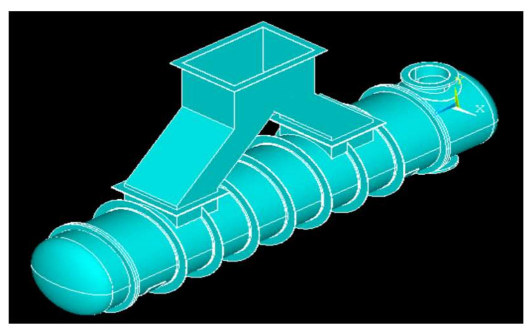

Fig. 5 Heat Exchanger Model

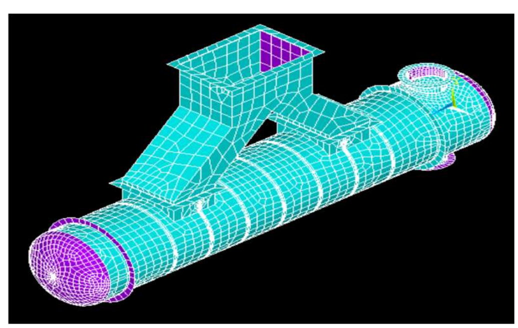

Fig. 6 Meshed Model

4.2. Meshing

The material of construction for the condenser was SS 304. The material properties for SS 304 are as in Table.2. After assigning the material properties and element properties, the model was meshed using the free mesh command. The meshed model was as in Figure 6.

Table. 3 Material Properties

Property	Value
Mod of Elasticity	200 Gpa
Poisson's ratio	0.3
Mass Density	7800 kg/m3

In ANSYS, all degrees of freedom were arrested on the areas where support legs were supposed to be welded in the model. The Pressure value of 99.0 kPa was applied at the outer surface of the condenser model. After assigning the loads and boundary conditions, the problem was solved in ANSYS.

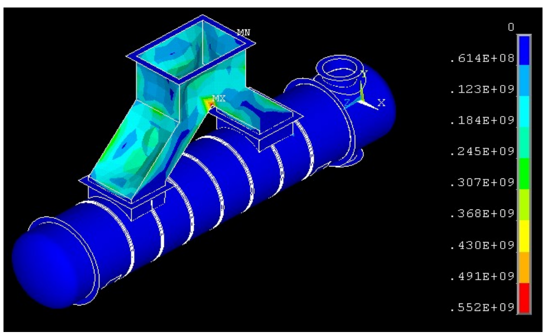

Fig. 7 without stiffeners Von Mises

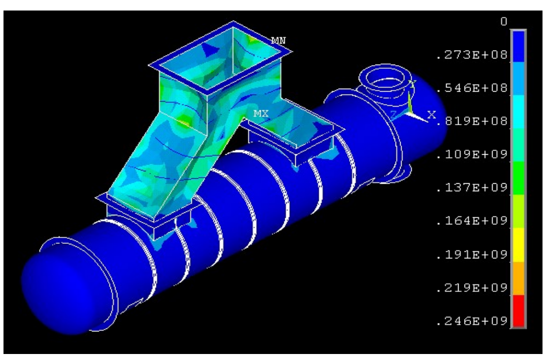

Fig. 8 with stiffeners Von Mises stress

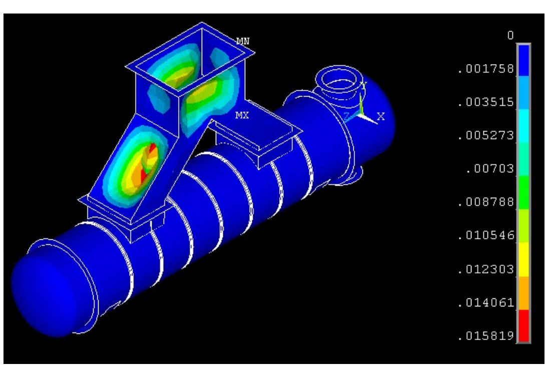

Fig. 9 without stiffeners Deformation

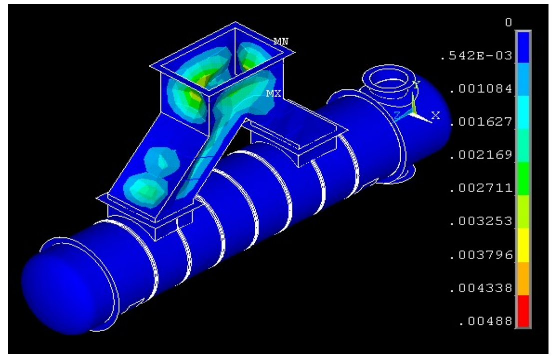

Fig. 10 with stiffeners Deformation

The model was analyzed in ANSYS with stiffeners at the duct portion and without stiffeners at the duct portion. The result was as in Table.3. The Von Mises stress values for the condenser model without stiffeners and with stiffeners were as shown in Figures 7 and 8. The deformation values for with stiffeners and without stiffeners were as shown in Figures 9 and 10. The results revealed that introducing stiffeners of size: 0.010 meter thicknesses x 0.150 meter width at the vapour duct portion reduces the stress value from 552 MPa to 246 MPa. Also, the results revealed that the introduction of stiffeners at the duct reduces the deformation from 0.015819 m to 0.00488 m.

Table. 4 Ansys Results

Analysis type Maximum	Deformation (m) Maximum	Von Mises Stress (Mpa)
Stiffeners at duct portion	0.00488	246
Without stiffeners at duct	0.015819	552

The mechanical design of the tube heat exchanger was carried out in accordance with ASME Section with VIII Division 1 and TEMA standards to ensure structural integrity under external pressure and operating conditions. The results obtained from the design calculations for the shell; tube sheet, nozzles, and stiffening elements are modelled using modelling software analyzed using Ansys software. The Yield stress value of SS 304 material is 205 MPa. The maximum allowable working stress for SS 304 is 136.67 MPa if the factor of safety considered is 1.5. In the above ANSYS results, it is revealed that the stress values and deformations are more in the vapour duct portion of the condenser. Also, it is revealed that the stiffeners are required at the duct portion. As shown in Figure.8, the maximum Von Mises stress values for the condenser with stiffener configuration is 109 MPa, which is well within the allowable working stress. Hence, it is proposed to provide stiffeners of 0.150 m width and 0.010-meter thickness at two equal intervals at the periphery of the vapour duct, as in the above Figure 8. The Yield stress value of SS 304 material is 205 MPa. The maximum allowable working stress for SS 304 is 136.67 MPa if the factor of safety considered is 1.5. In the above ANSYS results, it is revealed that the stress values and deformations are more in the vapour duct portion of the condenser. Also, it is revealed that the stiffeners are required at the duct portion. As shown in Figure. 5.5, the maximum Von Mises stress values for the condenser with stiffener configuration is 109 MPa, which is well within the allowable working stress. Hence, it is proposed to provide stiffeners of 0.150 m width and 0.010-meter thickness at two equal intervals at the periphery of the vapour duct.

5. Conclusion

The results show that all components meet code safety criteria. The shell and head thicknesses provide sufficient resistance to loads, while the tube sheet design meets structural and functional requirements under mechanical and thermal loads. Additionally, nozzle and reinforcement calculations align with code requirements, ensuring strength against localized stresses and discontinuities. The study demonstrates that standard design codes provide a reliable framework for mechanical heat exchanger design. The final design is safe, compliant and suitable for industrial application, meeting all objectives.

Future Work

Future work will focus on experimental validation, coupled CFD-FEA analysis, fatigue life assessment and optimization techniques to further improve heat exchanger performance.

Ethical Considerations

This research is a numerical and engineering design study that does not involve human participants or animals.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this manuscript.

Funding Statement

The authors received no specific funding.

Acknowledgments

The authors would like to thank Aarupadai Veedu Institute of Technology for providing the research facilities and support that made this work possible.

References

[1] Erica Jacqueline Fernandes, and Sachidananda Hassan Krishanmurthy, “Design and Analysis of Shell and Tube Heat Exchanger,” *International Journal for Simulation and Multidisciplinary Design Optimization*, vol. 13, 2022. [CrossRef] [Google Scholar] [Publisher Link]

[2] S. SivaChandran et al., “Experimental Study of Shell and Tube Heat Exchanger,” *AIP Conference Proceedings*, vol. 2473, no. 1, 2022. [CrossRef] [Google Scholar] [Publisher Link]

[3] Navid Bozorgan, “Design Optimization of a Shell and Tube Heat Exchanger for a Helicopter Considering Objective Functions of Heat Exchanger Weight, Overall Heat Transfer Coefficient, and Manufacturing Cost,” *Journal of Heat and Mass Transfer Research*, vol. 13, no. 2, pp. 147-157, 2026. [CrossRef] [Google Scholar] [Publisher Link]

[4] Ameya Mathkar, Shyam Gopalakrishnan, and Sujay S. Pathre, “Comparison of Stresses in Flexible Shell Element Expansion Joint of a Heat Exchanger for Code Application When Made of Multiple Flexible Shell Element,” *Pressure Vessels and Piping Conference*, 2022. [CrossRef] [Google Scholar] [Publisher Link]

[5] Prashant Rajesh Dhongade, “*Numerical Analysis and Optimization of Shell-and-Tube Heat Exchangers Using CFD: A Comparative Study with Traditional Correlation-Based Methods*,” RIT Autonomous, 2025. [Publisher Link]

[6] Shaad Rungla et al., “Design and Analysis of Floating Head Heat Exchanger,” *SSRN*, pp. 1-89, 2022. [CrossRef] [Google Scholar] [Publisher Link]

[7] Sezgi Koçak Soylu, and Yasar Demirel, “Thermal-Hydraulic Analysis of a Shell-and-Tube Heat Exchanger Due to Fouling in the Distillation Process Using ASPEN PLUS,” *Journal of Thermal Science and Engineering Applications*, vol. 17, no. 6, pp. 1-31, 2025. [CrossRef] [Google Scholar] [Publisher Link]

[8] Senthil Kumar Raman, “Stress Evaluation of a Three-Point Supported Heat Exchanger,” *Pressure Vessels and Piping Conference*, Las Vegas, Nevada, USA, 2022. [CrossRef] [Google Scholar] [Publisher Link]

[9] Isaak Dassa, Konstantinos Karamitsios, and Dimitrios Mertzis, “Development and Validation of a Tubesheet Geometry Generator Tool for Efficient Heat Exchanger Design,” *Proceedings of the 9th World Congress on Mechanical, Chemical, and Material Engineering*, London, United Kingdom, 2023. [CrossRef] [Google Scholar]

[10] Damylle Cristina Xavier Donati, “Experimental and Numerical Study of the Modeling of the Fluid-structure Phenomenon in Plate Heat Exchangers,” *UFSC Institutional Repository*, pp. 1-157, 2023. [Google Scholar] [Publisher Link]

[11] Robert Weyer, and Milad Haji Mohammad Karim, “Overview of Piping Stress Analysis using Shell Elements,” *Pressure Vessels and Piping Conference*, 2022. [CrossRef] [Google Scholar] [Publisher Link]

[12] Hossein Vossough, and Said Golabi, “Optimal Design Method for Elliptical Weld Neck-Flange and its Closure using Finite Element Analysis and NSGA-II to Comply with ASME Section VIII-Division 2, no Leakage Conditions and Bolt Integrity,” *Journal of Pressure Vessel Technology*, vol. 147, no. 5, 2025. [CrossRef] [Google Scholar] [Publisher Link]

[13] Zirui Xu et al., "Study on Thermal Deformation of Hybrid Printed Circuit Heat Exchanger for Advanced Nuclear Reactor," *Cleaner Energy Systems*, vol. 3, pp. 1-13, 2022. [CrossRef] [Google Scholar] [Publisher Link]

[14] Ishwar Gophane et al., "Theoretical and Finite Element Analysis of Pressure Vessel," *Indian Journal of Science and Technology*, vol. 17, no. 12, pp. 1148-1158, 2024. [CrossRef] [Google Scholar] [Publisher Link]

[15] Guodong Zhu et al., "Study on the Calculation Method of Fluid Distributor of Corner-Corner Type in Heat Exchangers," *Pressure Vessels and Piping Conference*, Atlanta, Georgia, USA, 2023. [CrossRef] [Google Scholar] [Publisher Link]

[16] J. Lum et al., "Comparative Creep–Fatigue Assessment of a Representative Intermediate Heat Exchanger in a High-Temperature Gas-Cooled Reactor System," *Nuclear Engineering and Technology*, vol. 58, no. 10, pp. 1-14, 2026. [CrossRef] [Google Scholar] [Publisher Link]

[17] Hyeong-Yeon Lee et al., "High-temperature Design of 700° C Heat Exchanger in a Large Scale High-Temperature Thermal Energy Storage Performance Test Facility," *Pressure Vessels and Piping Conference*, Nevada, USA, 2022. [CrossRef] [Google Scholar] [Publisher Link]

[18] Yeni Li et al., "Numerical Analysis with Experimental Validation of Tube Fatigue Failure in Feedwater Heaters," *Engineering Failure Analysis*, vol. 181, 2025. [CrossRef] [Google Scholar] [Publisher Link]

[19] Luke Liu et al., "Assessment of Inspection Resolution Acceptance Criteria for Local and General Metal Loss in API579-ASME/1 Code by Means of Finite Element Method," *Pressure Vessels and Piping Conference*, Quebec, Canada, vol. 89046, 2025. [CrossRef] [Google Scholar] [Publisher Link]

[20] Mohamed Fayas Saffiudeen et al., "Enhancement of Tubesheet Integrity and Longevity via GTAW Cladding: Experimental Assessment and Inspection Methods," *Materials Today Communications*, vol. 53, pp. 1-9, 2026. [CrossRef] [Google Scholar] [Publisher Link]