ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

Pressure Drop in Shell Side of Shell and Tube Heat Exchanger Using CFD

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Abstract:

Heat exchanger is a device which is utilized to exchange the thermal energy of one liquid to the next liquid. The shell and tube heat exchanger is one of the classification of heat exchangers which consists of a bundle of tubes encased inside a barrel shaped shell. Shell-and-tube heat exchangers are ordinarily utilized as a part of an extensive variety of utilizations like power plant, refrigeration, oil refinery, waste heat recuperation and substance handling and so forth. The thermal performance (pressure drop) in shell side of the shell and tube heat exchanger mainly depends on different baffles. For getting minimum pressure drop in shell side of shell and heat exchanger, staggered baffles are replaced by helical baffle with five different pitch values. The main advantage of helical baffle is it can reduce stagnation points in between baffles and shell portion and also increases the flow of fluid from inlet to outlet. In the present work it is proposed to conduct fluid flow analysis of shell and tube heat exchanger by using CATIA and FLUENT software. Andrew Ozden explained about shell side pressure drop in shell and tube heat exchanger with staggered baffle and his experimental values were validated in this research with numerical results by using CFD analysis. The percentage of error in between Andrew Ozden's research and this research is about 1%. For further analysis helical baffles with five different pitches (100mm, 125mm, 150mm, 175mm and 200mm) were considered in a turbulent region within a range of 5000-20000 for getting minimum pressure drop in shell side of the heat exchanger. Fluid flow analysis was conducted on shell and tube heat exchanger with water as a working fluid for shell and tube side. Copper and Stainless Steel are considered as a structural material for Tube and Shell respectively. Here we acquired k-ε as turbulent model for all analyses. Shell side pressure drop of shell and tube heat exchanger are calculated at different pitches of helical baffles and tabulated. Among all pitches of helical baffles, 150mm pitch helical baffle gives minimum pressure drop in shell side.

Keywords:

Shell and tube heat exchanger, generalized regression neural network, heat transfer, pressure drop, elliptical tubes.

1. Introduction:

D.Kral et.al. [1] Demonstrated on execution of heat transfers with helical baffles, or helixchangers, and utilizing the consequences of tests led on units with various baffle geometries. They also proved that an ideal helix edge is distinguished at which the change proficiency for changing over pressure drop to heat transfer on the shell side of helix changers is augmented. They also said that plans for standard industry applications are improved utilizing the examination of test outcomes. Qiuwang Wang et.al [2] stated that helical confuses are utilized progressively in shell-and-tube heat transfers (helix changers) for their noteworthy points of interest in lessening pressure drop, vibration, and fouling while keeping up a higher heat transfer execution. With a specific end goal to make great utilization of helical confuses, serial upgrades have been made by numerous scientists. In this paper, a general survey is given of advancements and changes on helix changers, which incorporates the broken helical baffles, nonstop or consolidated helical

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baffles, and the joined various shell-pass helix changers. They also proved that Broad outcomes from analyses and numerical recreations demonstrate that these helix changers have better stream and heat transfer execution than the routine segmental baffled heat transfers. At last they concluded by saying in view of these new changes, the ordinary heat transfers with segmental baffles may be supplanted by helix changers in modern applications to spare vitality, decrease cost, and delay the administration life and operation time. Stehlik et.al [3] described that heat transfer and pressure drop revision variables in view of the Bell-Delaware strategy have been looked at for an advanced segmental puzzle heat transfer and a helical confound heat transfer. When all is said in done, the outcomes demonstrated that appropriately composed helical baffles offer a noteworthy change in heat transfer while giving a decreased transfer pressure drop. They also quoted that the improvement in heat transfer for helical puzzles was reflected by the supposed turbulence upgrade redress consider, which represented the expansion in heat transfer saw at a basic astound slant point of 25°.

As the puzzle slant point was expanded past this basic edge, the turbulence upgrade figure kept on expanding and in the long run delivered a most extreme mend transfer improvement of 1.39 circumstances that for perfect cross-stream conditions. The decrease in pressure drop because of the helical baffles was found to fluctuate from 0.26 to 0.60 contingent upon the helical slant point. Qincheng Bi et.al [4] In this review, their investigations were done to concentrate the impacts of puzzle cover extent on the shell-side stream resistance and heat transfer execution of the shell-and-tube heat transfers with helical baffle (STHXsHB). Three STHXsHB with a cover extent of 10% and helix edges of 20°, 30°, and 40° were tried. Correlations were made of the exploratory information of the STHXsHB with the same helix angles however half cover extent. They also investigated the hypothesis of entrance dissemination was utilized to assess the irreversible misfortune in STHXsHB with various helix points and cover extents. Finally they came to know that the outcomes showed that both the baffle cover extent and the helix edge greatly affect the shell-side stream resistance and heat transfer. For a given helix edge, the thorough execution of STHXsHB with little cover extent is constantly superior to anything that with vast cover extent at a similar mass stream rate or Reynolds number on the shell side. In any case, for a similar heat transfer region, working conditions, and helix point, the STHXsHB with substantial baffle cover extent has less irreversibility in the heat trade prepare, as indicated by the hypothesis of entrance scattering. Moreover, exploratory outcomes showed that the arrangement of the moderately expansive helix edge and baffle cover extent is the favoured option in STHXsHB. Peng Q et.al [5] demonstrated that Two shell-and-tube heat transfers (STHXs) utilizing ceaseless helical baffles rather than segmental astounds utilized as a part of routine STHXs were proposed, planned, and tried in this review. The two proposed STHXs have a similar tube package however unique shell designs.

The stream design in the shell side of the heat transfer with ceaseless helical baffles was compelled to be rotational and helical because of the geometry of the nonstop helical baffles, which brings about a critical increment in heat transfer per unit pressure drop in the heat transfer. Appropriately outlined persistent helical puzzles can decrease fouling in the shell side and keep the stream instigated vibration also. They also investigated on the execution of the proposed STHXs and it was considered tentatively in this work. The heat transfer coefficient and pressure drop in the new STHXs were contrasted and those in the STHX with segmental confounds. They confirmed that the outcomes demonstrate that the utilization of ceaseless helical baffles brings about almost 10% expansion in heat transfer coefficient contrasted and that of routine segmental puzzles for a similar shell-side pressure drop. In view of the trial information, the no dimensional relationships for heat transfer coefficient and pressure drop were produced for the proposed

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nonstop helical astound heat transfers with various shell setups, which may be helpful for modern applications and further investigation of ceaseless helical baffle heat transfers. This paper additionally introduces a straightforward and achievable technique to manufacture consistent helical astounds utilized for STHXs. F. Zhang et.al [6] stated that a strategy for plan and rating of shell-and-tube heat transfer with helical baffles (STHXHB) has been created in present review in light of people in general written works and the broadly utilized Bell–Delaware technique for shell-and-tube heat transfer with segmental baffles (STHXSB). Various bend sort considers the writing have all been swapped by numerical expressions for the comfort of building plan. They also said that the itemized computation methodology of the technique is given. The precision of present technique is approved with some exploratory information. Finally they concluded that four plan instances of supplanting unique STHXsSB by STHXsHB are provided, and the examination comes about demonstrate that the greater part of the STHXsHB have preferable execution over the first heat transfers with segmental baffles.

Jian-Fei Zhang et.al [7] quoted that presented in this paper are exploratory test and correlation for a few shell-and-tube heat transfers, one with segmental confuses and four with helical baffles at helix points of 20◦, 30◦, 40◦ and 50◦, individually. They also demonstrated that the outcomes demonstrate that, in view of a similar shell-side stream rate, the heat transfer coefficient of the heat transfer with helical baffles is lower than that of the heat transfer with segmental baffles while the shell-side forced drop of the previous is even much lower than that of the later. Promote upgrade systems ought to be consolidated with a specific end goal to improve shell-side heat transfer in light of a similar stream rate. They came to know that The examination of heat transfer coefficient per unit pressure drop (and pumping power) versus shell-side volume stream rate demonstrates that the heat transfer with helical baffles have critical execution advantage over the heat transfer with segmental confuses; for a similar shell inward measurement, the execution of heat transfer with helical baffles with 30° helix edge is superior to anything that of 20°, and the execution of 40° helix point is superior to anything that of 50° helix edge. Finally, they concluded that the heat transfer with helical baffles of 40° point demonstrates the best execution among the five heat transfers tried. Jian-Feng Yang et.al [8] demonstrated that a joined serial two shell-pass shell-and-tube heat transfer (CSTSP-STHX) with constant helical confounds has been proposed to enhance heat transfer execution. This CSTSP-STHX isolates the shell side into two individual shell passes. They stated that the internal shell pass is ordinary segmental puzzled, and the external shell pass is ceaseless helical confused. The working liquid courses through the external and inward shell goes in grouping. The thermo pressure driven exhibitions of CSTSP-STHX are tentatively contrasted and the twofold shell-pass shell-and-tube heat transfer with segmental puzzles (SG-STHX).

They also said that the outcomes demonstrate that the CSTSP-STHX gets more prominent shell-side heat transfer coefficient and pressure drop, besides it additionally has better heat transfer coefficient under a similar pressure drop than those of the SG-STHX. At last it ought to be accentuated that the spillage on annulus separator must be as conceivable as lessened. Finally they concluded by saying the present reviews are advantageous for the outline and viable operation of CSTSP-STHX. Bin Gao et.al. [9] described that stream resistance and heat transfer of a few shell-and-tube heat transfers with broken helical baffles are tentatively examined and analysed at the five helix points of 8°, 12°, 20°, 30° and 40°. They also demonstrated that the second-law based thermodynamic examination is utilized to investigate the impacts of astound helix edge on the irreversible loss of heat transfers. The outcomes demonstrate that the shell-side pressure drop and heat transfer coefficient of the heat transfer with littler helix edge are higher than those with bigger helix point at a given

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International Journal & Magazine of Engineering, Technology, Management and Research
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shell-side volume stream rate. They also stated that in any case, in the state of a similar shell-side Reynolds number, the stream resistance with bigger helix edge is lower and the heat transfer execution is better. Usman Salahuddin et.al [10] validated that this paper gives a survey about the significant work done on helical astounds to enhance the execution of shell and tube heat transfers. A portion of the main considerations influencing the execution of shell and tube heat transfer are examined. A correlation between segmental puzzles and helical astounds is additionally displayed to demonstrate that helical confounds are more favourable than segmental baffles. They also stated that Much of the time, broken, collapsed, sextant helical confuses, 40° puzzle slant point and in addition low baffles dividing will give the best outcomes when coordinated in some mix, while ceaseless helical baffles dispense with dead districts. Finally, they said that in addition, fixing strips will probably enhance the execution of shell and tube heat transfers with consistent helical baffles.

Reza Tasouji Azar et.al [11] stated that in this review, cost capacities were dissected in light of exergy rate and aggregate life cycle for a shell and tube heat transfer as a remain solitary unit with segmental confounds and helical puzzles with helix edges from 5° to 45° in a retrofit extend. At first, the progressions of shell-side exergy obliteration rate were examined. After that the aggregate life cycle costs, including capital, establishment, working and upkeep expenses, were independently ascertained for all cases. They also quoted that in order to decide the impact of puzzle sorts, the net exergy sparing cost capacity was viewed as in light of the contrast between the exergy benefit cost and the net capital cost of the baffles' change. The consequences of exergy investigation demonstrated that the shell-side exergy annihilation rate diminished with the expansion in helix point. At last they concluded by saying besides, the net exergy sparing cost work examination exhibited that helical astounds from 25° to 45° enhanced thermodynamic execution in a financially savvy route in patches up and overhauls.

2 MODELING:

In the process of the Catia modelling of Shell and Tube Heat Transfer we have to design four Parts. They are,

2.1 TUBE SHEET:

Dimensions:

Diameter	= 100mm
Pitch	= 30mm
Hole bundle geometry	= Triangular
No. of Holes	= 7
Hole diameter	= 20mm

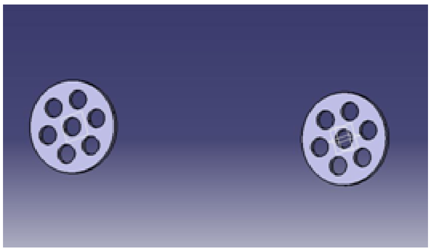


Figure 1 Designed Catia model of Tube Sheet

2.2 TUBES:

Dimensions:

Tube outer Diameter	= 20mm
Thickness	= 1mm
Tube Length	= 600mm

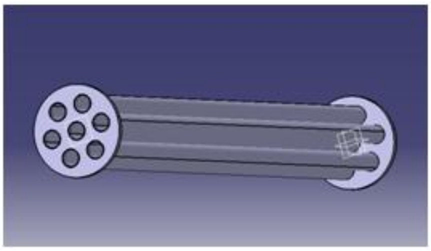
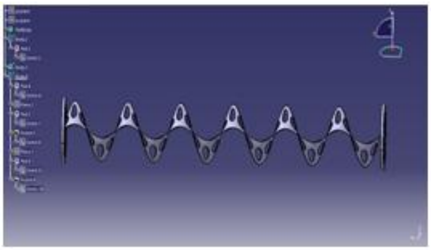


Figure 2 Designed Catia model of Tubes

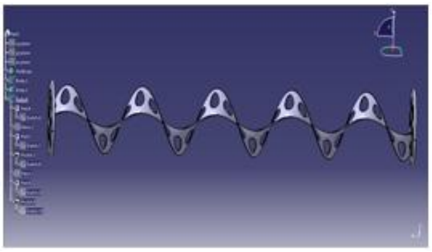
Helical Baffle:

Dimensions:

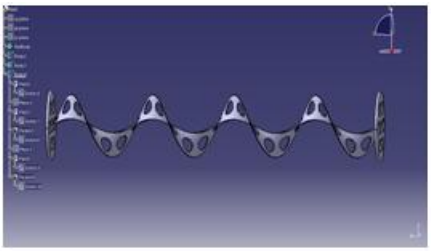
Helix Diameter	= 90mm
Helix Length	= 600mm
Helical Pitch	=100mm,125mm, 150mm, 175mm, 200mm
Baffle Thickness	= 2mm



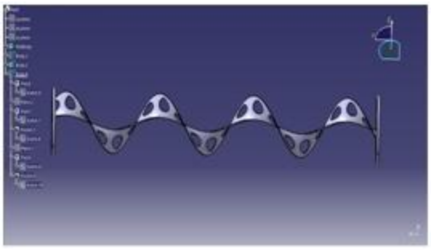
100-pitch baffle



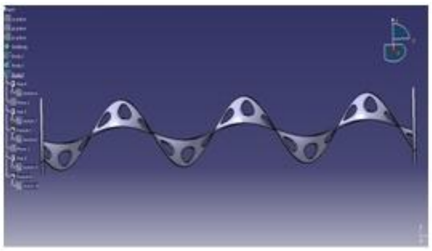
125-pitch baffle



150-pitch baffle



175-pitch baffle



200-pitch baffle

Figure 3 Baffles with various pitches

2.3 SHELL:

Dimensions:

Shell inner Dia. = 90mm
Shell Thickness = 5mm
Shell Length = 600mm

Used Catia Tools:

Project 3D Elements, Pad, Plane and Pocket.

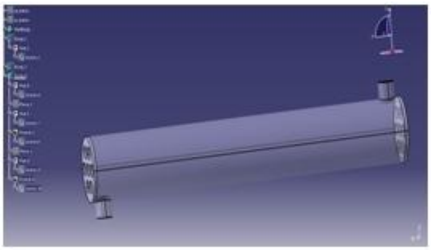


Figure 4 Designed Catia model of Shell

2.4 GEOMETRY:

Heat exchanger is built in the ANSYS workbench design module. It is a counter-flow heat exchanger. First, the fluid flow (fluent) module from the workbench is selected. The design modeller opens as a new window as the geometry is double clicked.

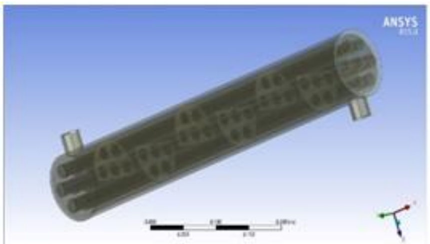


Figure. 5 Imported model in geometry

Table 1 Fluid And Solid States Are Assigned

PART NUMBER	PART OF THE MODEL	STATE TYPE
1.	INNER FLUID	FLUID
2.	OUTER FLUID	FLUID
3.	BAFFLE	SOLID
4.	SHELL(1)	SOLID
5.	TUBES(7)	SOLID
6.	TUBE SHEET	SOLID

2.5 MESHING:

At first a generally coarser work is produced. This work contains blended cells (Tetra and Hexahedral

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International Journal & Magazine of Engineering, Technology, Management and Research
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cells) having both triangular and quadrilateral countenances at the boundaries or limits. Care is taken to utilize organized hexahedral cells however much as could be expected. It is intended to diminish numerical dissemination however much as could be expected by organizing the work in a well way, especially close to the divider locale. Later on, a fine work is produced. For this fine work, the edges and areas of high temperature and pressure slopes are finely coincided.

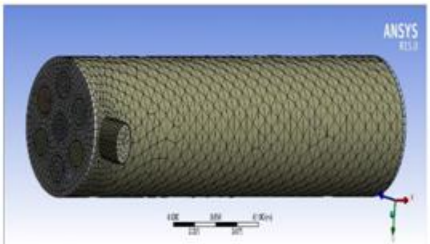
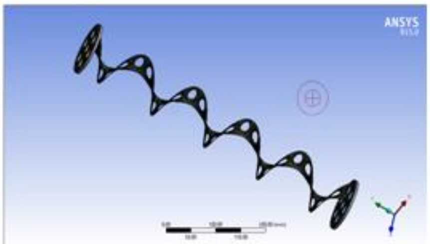
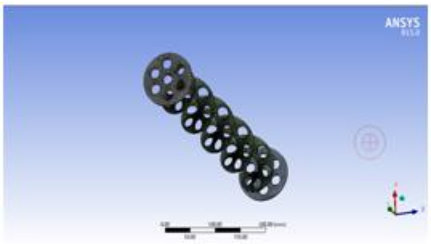
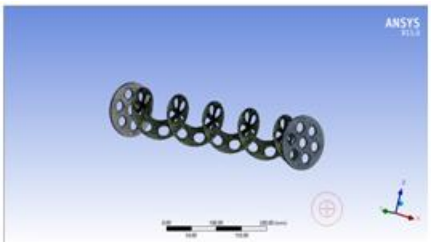


Figure 6 Shell and tube model after Meshing







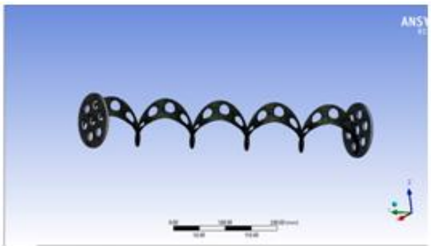


Figure 7 Baffle Model after Meshing

The different surfaces of the solid are named as per required inlets and outlets for inner and outer fluids.

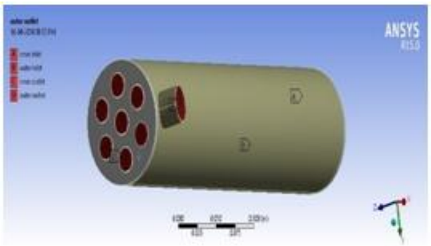


Figure 8 Named selections

2.6 SETUP:

The mesh is checked and quality is obtained.

2.7 MATERIALS:

The create/edit option is clicked to add water-liquid, steel and copper to the list of fluid and solid respectively from the fluent database.

2.8 CELL ZONE CONDITIONS:

In cell zone conditions, we have to assign the conditions of the liquid and solid.

Table 2 Cell Zone Conditions

S.No.	Part/Body	Material
1.	Inner Fluid	Water-Liquid
2.	Outer Fluid	Water-Liquid
3.	Tube Sheet	Steel
4.	Tubes	Copper
5.	Baffles	Copper
6.	Shell	Steel

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2.9 BOUNDARY CONDITIONS:

Boundary conditions are utilized by the need of the model. The bay and outlet conditions are characterized as speed delta and pressure outlet. As this is a counter-stream with two tubes so there are two gulfs and two outlets. The dividers are independently indicated with particular limit conditions. No slip condition is considered for every divider. But the tube dividers every divider is set to zero heatth flux condition. The insights about all limit conditions can be found in the table as given beneath.

Table 3 Boundary Conditions

	Boundary Condition Type	Mass Flow Rate(Kg/S)	Temperature (K)
Inner Inlet	Mass flow inlet	1	450
Inner Outlet	Pressure outlet	-	-
Outer Inlet	Mass flow inlet	1	300
Outer Outlet	Pressure outlet	-	-

Validation:

Ender Ozden in his research studied about pressure drop in shell side of shell and tube heat exchanger with segmented baffles and CFD analysis was carried out and finally he concluded that at 36% baffle cut gives minimum pressure drop and maximum thermal performance.According to Ender Ozden's research the boundary conditions were the shell inlet temperature is set to be 300 K, no slip condition is assigned to the surfaces, zero heat flux boundary is assigned to outer shell wall and constant wall temperature of 450 K is assigned to tube walls. And if we compare the value obtained in his research with this research the difference in the output is less than 2%, and the Ender Ozden's output values were provided below,

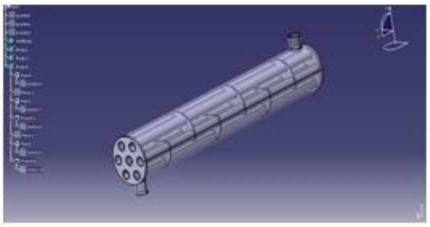

Figure 9 Ozden's Shell and tube heat exchanger with segmented baffles

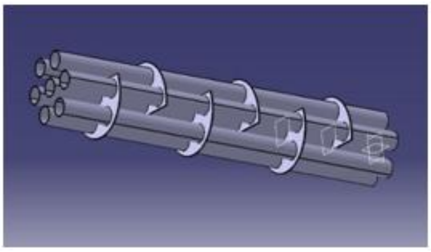

Figure 10 Ozden's Shell and tube heat exchanger with segmented baffles and tubes

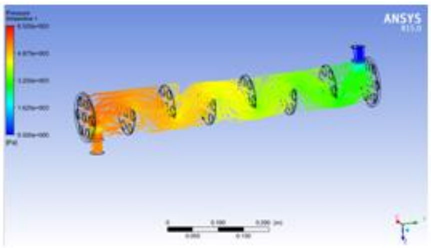

Figure 11 Ender Ozden's CFD analysis

Viscous Model	Mesh	Mass Flow Rate (Kg/S)	Shell Side Pressure Drop (Pa)
K-Ɛ Standard	Coarse Mesh	1	6648
K-Ɛ Standard	Coarse Mesh	1	6346

3.0 The Effect of Baffle Arrangement over Pressure Drop in Shell and Tube Heat Exchanger

In this paper we are calculate the shell side pressure drop of Shell and Tube Heat Exchanger by varying different types of helical baffle with varies pitch values. The flow pattern in the shell side of the heat exchanger with continuous helical baffles was forced to be rotational and helical due to the geometry of the continuous helical baffles, which results in an effective pressure drop in the heat exchanger. So helical baffle is preferred in pressure drop conditions. The helical baffle with different pitches are explained below.

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3.1 Case – 1 Shell and tube heat exchanger with 100 pitch helical baffle:

The shell and tube heat exchanger with helical baffle of 100 pitch considered for this analysis at different mass flow rates in turbulent region. In this analysis four mass flow rates are considered and analysed the shell side pressure drop in between the Reynolds number range 5000 to 20000.

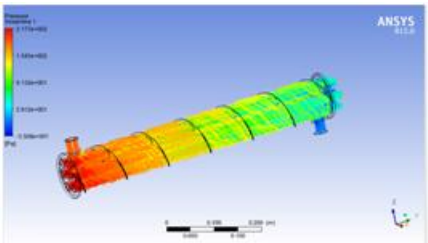


Figure 12 Flow line of pressure drop in shell and tube side at Re = 5555

Here mass flow rate of the fluid flowing in the shell side of the heat exchanger is taken as 0.184kg/s. depending on the mass flow rate the Reynolds number and pressure also changes. The flow of the fluid in the shell side of the heat exchanger is mentioned in the form of flow lines

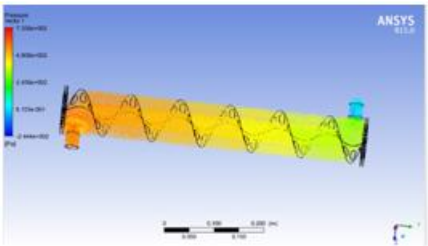


Figure 13 Velocity vector of pressure drop in shell side at Re = 10100

Greater pressures may damage the heat exchanger and also decreases the performance of the heat exchanger. The mass flow rate considered in this case is about 0.335 kg/s in tube side and 0.333 kg/s in shell side

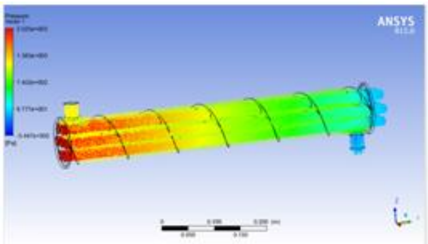


Figure 14 Velocity vector of pressure drop in shell and tube side at Re = 15155

The mass flow rate considered here in this case is 0.503 kg/s in tube side and 0.500 kg/s in shell side of the heat exchanger which drastically changes the behaviour of fluid which may leads to change in pressure depends upon mass flow rate

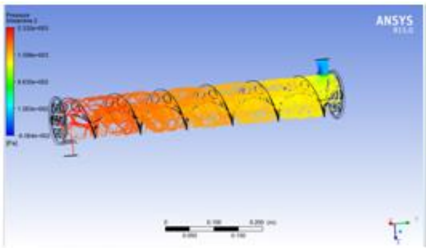
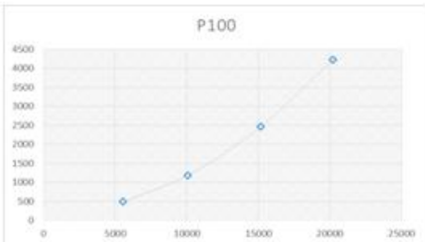


Figure 15 Streamline of pressure drop in shell side at Re = 20200

Table 4 Pressure drop values at different Reynolds numbers at 100 mm pitch

Reynolds number	5555	10100	15155	20200
Pressure drop (Pascal)	496.3882	1189.592	2462.091	4232.691



Graph – 1 Pressure drop at various Re Numbers

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International Journal & Magazine of Engineering, Technology, Management and Research
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Maximum pressure obtained for mass flow rate of 0.6663 kg/s in shell side of Pitch-100 baffle heat exchanger is about 4232 Pascal at 20200 Reynolds number and minimum pressure drop of 496 Pascal at 5555 Reynolds number.

3.2 Case – 2 Shell and tube heat exchanger with 125 pitch helical baffle:

The shell and tube heat exchanger with helical baffle of 125 pitch considered for this analysis at different mass flow rates in turbulent region of Reynolds number 5000 to 20,000.

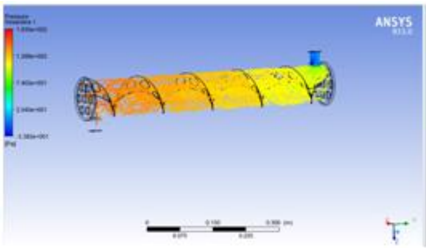


Figure 16Stream line of pressure drop in shell and tube side at Re = 5555

Here mass flow rate of the liquid flowing in the shell side of the heat exchanger is considered as 0. 184kg/s. The flow of the liquid in the covering side of the heatexchanger is specified as flow lines

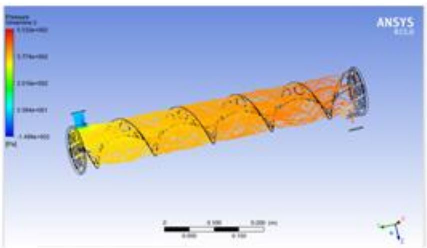


Figure 17Stream line of pressure drop in shell side at Re = 10100

The mass circulation rate considered in this case is about 0.335 kg/s in pipe side and 0. 333 kg/s in shell part

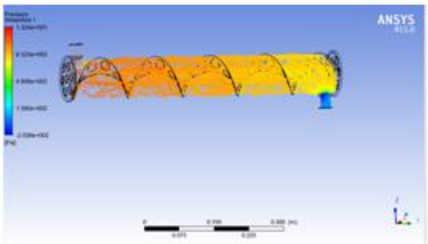


Figure 18Stream line of pressure drop in shell side at Re =15155

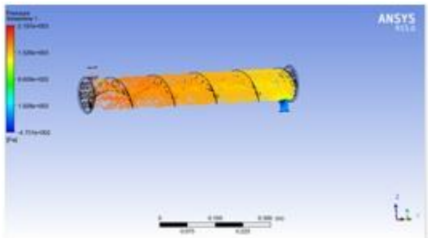
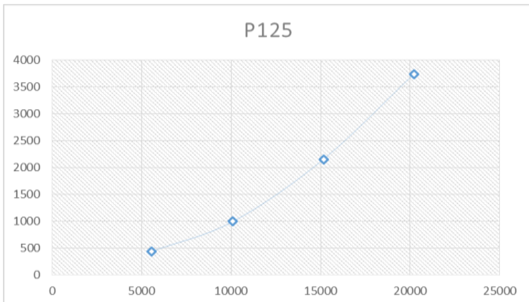


Figure 19Stream line of pressure drop in shell side at Re = 20200

Table 5 Pressure drop values at different Reynolds numbers at 125 mm pitch

Reynolds number	5555	10100	15155	20200
Pressure drop (Pascal)	433.6109	991.9478	2153.247	3742.742



Graph – 2 Pressure drop at various Re Numbers

Maximum pressure obtained for mass flow rate of 0. 666 kg/s in tube part of P-125 type high temperature exchanger is about 3742.742 Pascal at 20200 Reynolds number and minimum pressure drop of 433.6109 Pascal at 5555 Reynolds number.

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3.3 Case – 3 Shell and tube heat exchanger with 150 pitch helical baffle:

The shell and tube heat exchanger with helical baffle of 150 pitch considered for this analysis at different mass flow rates in turbulent region.

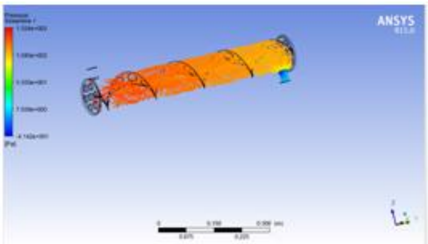


Figure 20 Velocity vector of pressure drop in shell side at Re = 5555

Here mass flow rate of the fluid flowing in the shell side of the heat exchanger is taken as 0.184kg/s. depending on the mass flow rate the Reynolds number and pressure also changes.

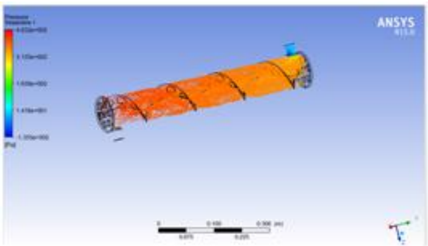


Figure 21Stream line of pressure drop in shell side at Re = 10100

Greater pressures may damage the heat exchanger and also decreases the performance of the heat exchanger. The mass flow rate considered in this case is about 0.335 kg/s in tube side and 0.333 kg/s in shell side

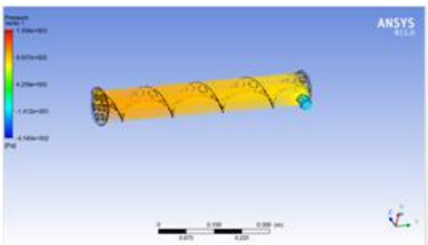


Figure 22 Velocity vector of pressure drop in shell and tube side at Re = 15155

The mass flow rate considered here in this case is 0.503 kg/s in tube side and 0.500 kg/s in shell side of the heat exchanger which drastically changes the behaviour of fluid which may leads to change in pressure depends upon mass flow rate.

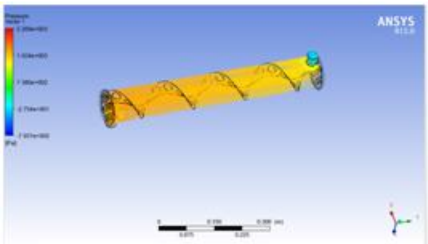


Figure 23 Velocity vector of pressure drop in shell and tube side at Re = 20200

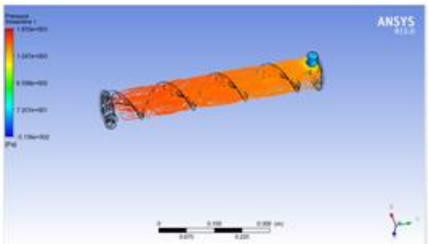
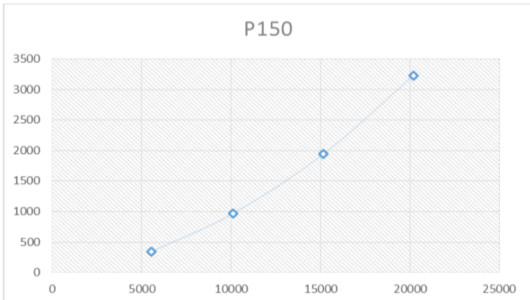


Figure 24Stream line of pressure drop in shell and tube side at Re = 20200

Table 6 Pressure drop values at different Reynolds numbers at 150 mm pitch

Reynolds number	5555	10100	15155	20200
Pressure drop (Pascal)	344.5621	966.0106	1941.968	3226.858



Graph – 3 Pressure drop at various Re Numbers

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Maximum pressure obtained for mass flow rate of 0.666 kg/s in shell side of Pitch-150 baffle heat exchanger is about 3226.8 Pascal at 20200 Reynolds number and minimum pressure drop of 344 Pascal at 5555 Reynolds number.

3.4 Case – 4 Shell and tube heat exchanger with 175 pitch helical baffle:

The shell and tube heat exchanger with helical baffle of 175 pitch considered for this analysis at different mass flow rates in turbulent region. In this analysis four mass flow rates are considered and analysed the shell side pressure drop in between the Reynolds number range 5000 to 20000

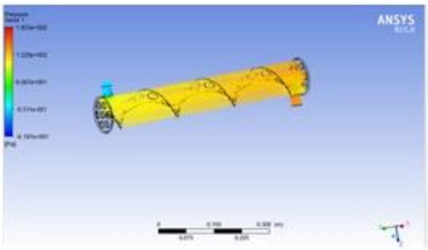

Figure 25 Velocity vector in both shell and tube side at Re= 5555

Here mass flow rate of the liquid flowing in the shell side of the heat exchanger is taken as 0.184kg/s. contingent upon the mass flow rate the Reynolds number and pressure likewise changes. The flow of the liquid in the shell side of the heat exchanger is specified as flow lines

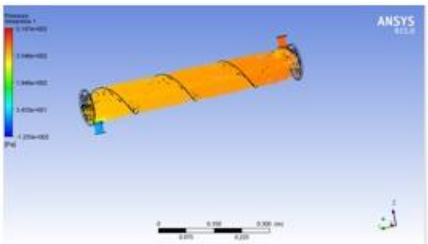

Figure 26 Flow Lines at both shell and tube side at Re= 10100

The mass flow rate considered for this situation is around 0.335 kg/s in tube side and 0.333 kg/s in shell side

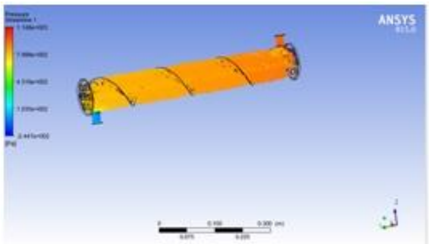

Figure: 27Streamlines at both shell side of Pressure Drop

For a given exchanger, the more prominent the flow rate, the higher the pressure drop. The mass flow rate considered here for this situation is 0.503 kg/s in tube side and 0.500 kg/s in shell side of the wheat exchanger which radically changes the conduct of liquid which may prompts to change in pressure relies on mass flow rate.

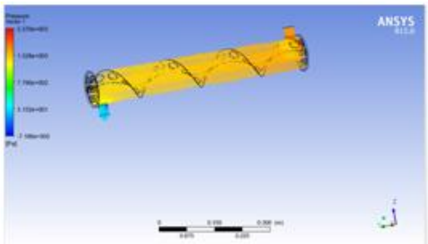

Figure 28 Velocity vectors at both shell and tube at Re = 20200

Stream Lines in the above figure clarifies the development of liquid in the shell side of the heat exchanger with mass flow rate of 0.671 kg/s in tube side and 0.666 kg/s in shell side of the heat exchanger

Table 7 Pressure drop values at different Reynolds numbers at 175 mm pitch

Reynolds number	5555	10100	15155	20200
Pressure drop (Pascal)	466.1469	1097.426	2260.187	3831.829

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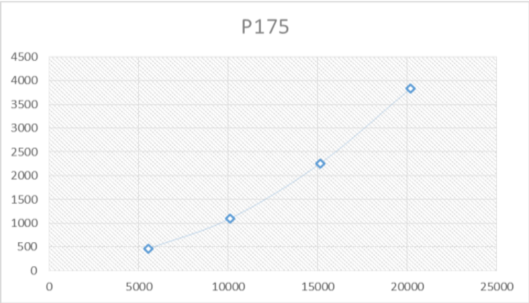


Graph – 4 Pressure drop at various Re Numbers

Most extreme pressure acquired for mass flow rate of 0.666 kg/s in tube side of P-175 sort heat exchanger is around 3831.8 at 20200 Reynolds number and least pressure drop of 0.466 Kilo Pascal at 5555 Reynolds number

3.5 Case – 5 Shell and tube heat exchanger with 200 pitch helical baffle:

The shell and tube heat exchanger with helical baffle of 200 pitch considered for this analysis at different mass flow rates in turbulent region. In this analysis four mass flow rates are considered and analysed the shell side pressure drop in between the Reynolds number range 5000 to 20000.

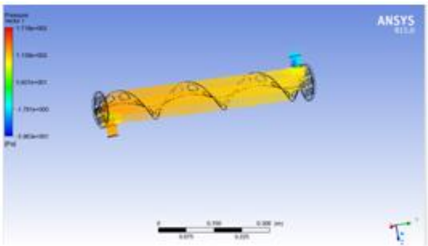


Figure 29 Velocity vectors at both shell and tube side Re = 5555

Here mass flow rate of the fluid flowing in the shell side of the heat exchanger is taken as 0.184kg/s. depending on the mass flow rate the Reynolds number and pressure also changes. The flow of the fluid in the shell side of the heat exchanger is mentioned in the form of flow lines

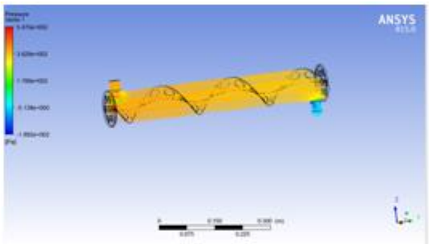


Figure 30 Velocity vectors at both shell and tube side at Re = 10100

Greater pressures may damage the heat exchanger and also decreases the performance of the heat exchanger. The mass flow rate considered in this case is about 0.335 kg/s in tube side and 0.333 kg/s in shell side

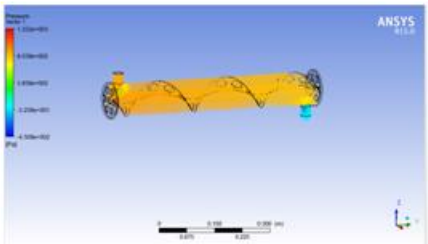


Figure 31Velocity vectors at shell side at Re

The mass flow rate considered here in this case is 0.503 kg/s in tube side and 0.500 kg/s in shell side of the heat exchanger which drastically changes the behaviour of fluid which may leads to change in pressure depends upon mass flow rate.

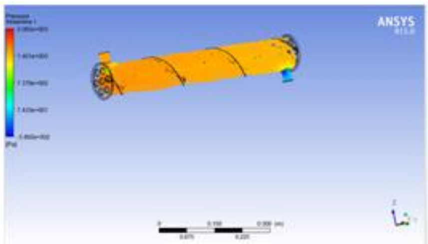
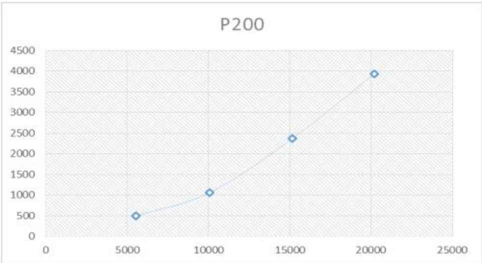


Figure 32Stream Lines at shell side at Re = 20200

Table 8 Pressure drop values at different Reynolds numbers at 200 mm pitch

Reynolds number	5555	10100	15155	20200
Pressure drop (Pascal)	489.9781	1057.252	2364.514	3933.056



Graph – 5 Pressure drop at various Re Numbers

Maximum pressure obtained for mass flow rate of 0.6663 kg/s in shell side of Pitch-200 baffle heat exchanger is about 3933.08 Pascal at 20200 Reynolds number and minimum pressure drop of 489.3 Pascal at 5555 Reynolds number.

Graph – 6 Variation of pressure drop at different Reynolds numbers

4. Conclusion:

CFD investigation were directed on Shell and Tube Heat exchanger with various pitch estimations of helical confuses and to gauge shell side pressure drop. Here we considered helical baffle with 100mm, 125mm, 150mm, 175mm and 200mm and at different Reynolds numbers the shell side pressure drop was ascertained by utilizing FLUENT. It is reasoned that the pressure drop increments with increments of Reynolds number. At various pitches of helical astound the examination are directed at various Reynolds number and at 150 mm pitch helical baffle gives less pressure drop in shell side of shell and tube heat exchanger as appeared in figure.

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