EFFECT OF NANOFLUID ON HEAT TRANSFER CHARACTERISTICS OF SHELL AND TUBE HEAT EXCHANGERS: EFFECT OF ALUNIMIUM OXIDE NANOFLUID

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***Abstract*— Shell and tube heat exchangers represent the most widely used vehicle for the transfer of heat in industrial process applications. Shell and tube heat exchangers have the ability to transfer large amounts of heat in relatively low cost, serviceable designs. They can provide large amounts of effective tube surface while minimizing the requirements of floor space, liquid volume and weight. A decade ago, with the rapid development of modern nanotechnology, particles of nanometre-size (normally less than 100 nm) are used instead of micrometre-size for dispersing in base liquids, and they are called nanofluids.** In this thesis, **analytical investigations have been done on the shell and tube heat exchanger, forced convective heat transfer and flow characteristics of a nano fluid consisting of mixing water and different volume concentrations of Al_2O_3 nanofluid (0.03,0.054,0.067and 0.135)% flowing under turbulent flow conditions.**

Introduction

Heat exchangers are devices in which heat is transfer from one fluid to another. The most commonly used type of heat exchanger is a shell-and-tube heat exchanger. Shell-and-tube heat exchangers are used extensively in engineering applications like power generations, refrigeration and air-conditioning, petrochemical industries etc. These heat exchangers can be designed for almost any capacity. The main purpose in the heat exchanger design is given task for heat transfer measurement to govern the overall cost of the heat exchanger.

The heat exchanger was introduced in the early 1900s to execute the needs in power plants for large heat exchanger surfaces as condensers and feed water heaters capable of operating under relatively high pressures. Both of these original applications of shell-and-tube heat exchangers continued to be used; but the design have become highly sophisticated and specialized, subject to various specific codes and practices. The broad industrial use of shell-and-tube heat exchangers known today also started in the 1900s to accommodate the demands of emerging oil industry.

The steadily increasing use of shell-and-tube heat exchangers and greater demands on accuracy of performance prediction for a growing variety of process conditions resulted in the explosion of research activities. These included not only shell side flow but also, equally important, calculations of true mean temperature difference and strength calculations of construction elements, in particular tube sheets.

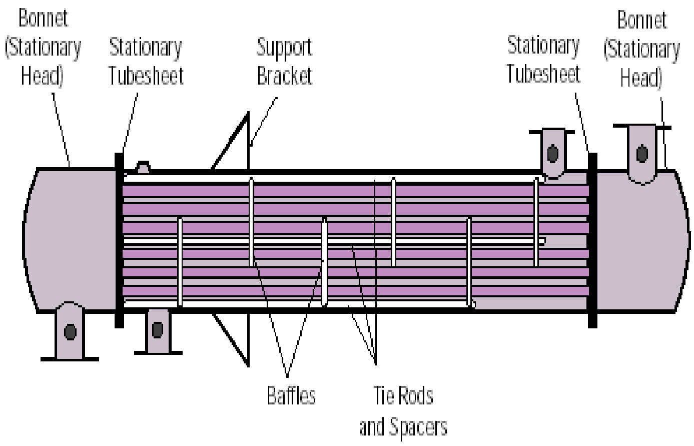
The objective of the thesis is to formulate the design algorithm and optimization procedure for a shell-and-tube exchanger in which exchanger geometry is determined from required performance for fixed pressure drops. First step in the effective consideration of allowable pressure drops is to establish a quantitative relationship between velocity, friction factors, pressure drop of the stream and number of transfer units. The solution of this equation provides the core of such an algorithm.

SHELL AND TUBE HEAT EXCHANGER

By far the most common type of heat exchangers to be encountered in the thermal applications is shell-and-tube heat exchangers. These are available in a variety of configurations with numerous construction features and with differing materials for specific applications. This chapter explains the basics of exchanger thermal design, covering such topics as: shell-and-tube heat exchanger components; classification of shell-and-tube heat exchangers according to constructions.

Constructional Details of Shell and Tube Heat Exchanger

It is essential for the designer to have a good knowledge of the mechanical features of shell-and-tube heat exchangers and how they influence thermal design. The principal components of shell-and-tube heat exchangers are: Shell, shell cover, tubes, channel, channel cover, tube sheet, nozzles, baffles, Other components include tie-rods and spacers, pass partition plates, impingement Plate, longitudinal baffles, sealing strips, supports, and foundation. The Tubular Exchanger Manufacturer is Association, TEMA, has introduced a standardized nomenclature for shell-and-tube heat exchangers. A three-letter code has been used to designate the overall configurations. The three important elements of any shell-and-tube heat exchangers are front head, the shell and rear head design respectively. The Standards of Tubular Exchanger Manufacturers Association (TEMA) [15] describes the various components of various class of shell-and-tube heat exchanger in detail.



Fixed Tube Sheet Shell and Tube Heat Exchanger

Use of multiple tubes because of that is increasing the heat transfer area. Reason of increasing heat transfer area is increase the velocity of fluid and lower effective *ΔT.* There is different type of shell available, all shell are identified regarding the diameter .Basically sizes of the shell are 8, 10, 12 inches. We find 2 inches of increment every step start from 13 inches to 25.From 25 to 39 we find 2 inches increment and after 39 to 72 we find 3 inches increment in shell. Tube size, which type of materials and array are primary criteria of designing of tube and shell type of mat exchanger. After done this step hydraulic design will be done on automatically. Small tube gives less cost with good thermal conductivity and Use of multiple tubes because of that is increasing the heat transfer area. Reason of increasing heat transfer area is increase the velocity of fluid and lower effective *ΔT.* It will create less shell area and size. Normally two arrays are available, triangular array produces the more tube with lower cost for particular heat transfer unit. We can control the pressure difference in square type of array so it is more preferable rather than the triangular type of array. When the cleaning require because of mechanical work, on that time square type of array are preferable. Wide pitch is used in this type of array and 60^o and 90^o arrays have a tendency to create a channeled flow. So that way fluid have a tendency to pass between two row of tube so there si not need to complete the full round of flow. This is happen in each tube so it is big gain for evaporators and condensers for vapor distributions.

Design Methods of Shell and Tube Heat Exchangers

First step in designing of heat exchanger, there is two way to design heat exchanger.

1. LMTD
2. NTU Method.

General equation of heat exchanger is

$$Q = UA_0 F_T \Delta T_{LM}$$

Where ΔT is the Temperature difference between hot and cold fluid

In terms of energy flow for heat exchanger, we can use this equation for hot fluid,

$$Q = -\dot{M} C_p \Delta T_h$$

Where ΔT is the Temperature difference between hot fluids

In terms of energy flow for heat exchanger, we can use this equation for cold fluid,

$$Q = \dot{M}\, C_p \Delta T_c$$

Where ΔT is the Temperature difference between hot fluids

Log Mean Temperature Difference Method

Heat flows between the hot and cold streams due to the temperature difference across the tube acting as a driving force. The difference will vary with axial location. Average temperature or effective temperature difference for either parallel or counter flow may be written as:

$$\Delta T_{LM} = LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

Normal practice is to calculate the LMTD for counter flow and to apply a correction factor F_T, such that

$$\Delta T_{LM} = F_T . \Delta T_{LM,CF}$$

The correction factors, F_T, can be found theoretically and presented in analytical form. The equation given below has been shown to be accurate for any arrangement having 2, 4, 6… 2n tube passes per shell pass.

$$F_{1-2} = \frac{\left[\frac{\sqrt{R^2+1}}{R-1}\right]\ln\left(\frac{1-P}{1-PR}\right)}{\ln\left[\frac{2/P - 1 - R + \sqrt{R^2+1}}{2/P - 1 - R - \sqrt{R^2+1}}\right]}$$

Where the capacity ratio, R, is defined as:

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

The parameter *P* may be given by the equation:

$$P = \frac{1 - X^{1/N_{SHELL}}}{R - X^{1/N_{SHELL}}}$$

Provided that $R \neq 1$ in the case that $R = 1$, the effectiveness is given by:

$$P = \frac{P_0}{N_{Shell} - P_0.(N_{Shell} - 1)}$$

$$P_0 = \frac{t_2 - t_1}{T_1 - t_1} \qquad X = \frac{P_0.R - 1}{P_0 - 1}$$

Gulyani and Mohanty [18] give alternate equations for the calculation of temperature correction factors. They have derived linear equations for the same and established that the factor is below 0.5 % error. They are given in Table 3.1.

N_{Shell}	F_T
1	1.208 G + 0.8037
2	0.237 G + 0.961
3	0.1202 G + 0.9835
4	0.0661 G + 0.991
5	0.0429 G + 0.994

Linear Equations for F_T [18]

Note: $G = 1 - P_0(1 + R)$

Effectiveness-NTU Method

In the thermal analysis of shell-and-tube heat exchangers by the LMTD method, an equation (3.1) has been used. This equation is simple and can be used when all the terminal temperatures are known. The difficulty arises if the temperatures of the fluids leaving the exchanger are not known. In such cases, it is preferably to utilize an altogether different method known as the effectiveness-NTU method. Effectiveness of shell-and-tube heat exchanger is defined as:

$$\varepsilon = \frac{C_S\left(T_{Si} - T_{So}\right)}{C_{\min}\left(T_{Si} - T_{Ti}\right)} = \frac{C_T\left(T_{To} - T_{Ti}\right)}{C_{\min}\left(T_{Si} - T_{Ti}\right)}$$

The group $\frac{UA}{C_{\min}}$ is called number of transfer units, NTU.

Effectiveness for shell-and-tube heat exchanger can also be expressed as:

$$\varepsilon = \varepsilon\left(\frac{UA}{C_{\min}}, \frac{C_{\min}}{C_{\max}}\right)$$

Where $\frac{C_{\min}}{C_{\max}} = \frac{C_S}{C_T} or \frac{C_T}{C_S}$ (depending upon their relative magnitudes).

Kays and London have given expressions for shell-and-tube heat exchangers. Some of their relationships for effectiveness are given below:

For one shell pass, 2, 4, 6 tube passes

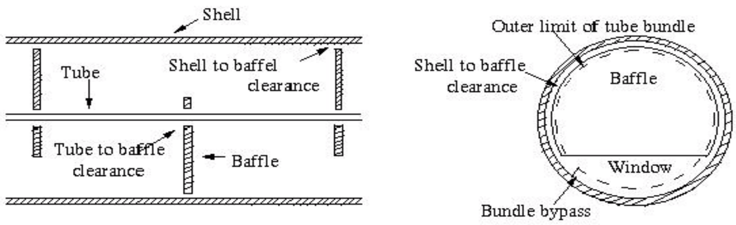
$$\varepsilon_1 = 2\left\{1 + C_{\min} + \sqrt{1 + C_{\min}^{\ 2}}\ \frac{1 + \exp\left[-NTU\left(1 + \sqrt{1 + C_{\min}^{\ 2}}\right)\right]}{1 - \exp\left[-NTU\left(1 + \sqrt{1 + C_{\min}^{\ 2}}\right)\right]}\right\}$$

For two shell pass, any multiple of 4 tubes

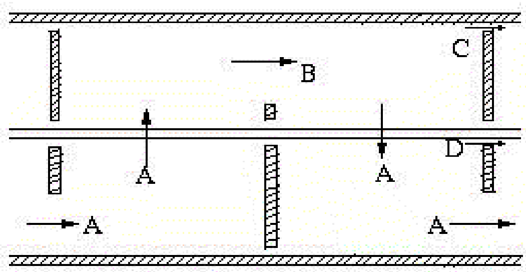
$$\varepsilon_2 = \left[\left(\frac{1 - \varepsilon_1 C_{\min}}{1 - \varepsilon_1}\right)^2 - 1\right]\left[\left(\frac{1 - \varepsilon_1 C_{\min}}{1 - \varepsilon_1}\right)^2 - C_{\min}\right]^{-1}$$

Calculation of Heat Transfer Coefficient and Pressure Drops

Flow across banks of tubes is, from both constructional and physical considerations, one of the most effective means of heat transfer. However, it is recognized quite early that ideal tube bank correlations, if applied to shell-and-tube heat exchangers, needed substantial corrections. In 1951, Tinker presented what has become a classical paper on flow through the tube bundles of shell-and-tube heat exchanger. He pointed out that a number of differing paths existed for flow and argued that the assumption that all of the fluid passed through the whole of the bundle was false. This was clearly demonstrated by his observations of the performance of exchangers handling highly viscous oils. He then proceeded to propose a flow model based on variety of flow paths cross flow, bundle bypass, tube-baffle leakage and shell-baffle leakage.



Mechanical Clearances in Shell and Tube Heat Exchanger [8]



Flow Paths on Shell Side, A Cross Flow; B Window; C Shell-Baffle

Heat Transfer Efficient

The Bell's Delaware method uses ideal tube bank j_h and f factors and then corrects directly the resulting h_i and ΔP_i for derivations caused by the various split streams. The ideal tube bank factor j and f is given as:

$$j_h = a_1\left(\frac{1.33}{L_{tp}/D_t}\right)^{\frac{a_3}{1+0.14\,\mathrm{Re}^{a_4}}} \mathrm{Re}^{a_2}$$

$$f = b_1\left(\frac{1.33}{L_{tp}/D_t}\right)^{\frac{b_3}{1+0.14\,\mathrm{Re}^{b_4}}} \mathrm{Re}^{b_2}$$

For possible computer applications, a simple set of constants is given in [20] for the curve fit of the above form.

The ideal heat transfer coefficient on shell side is defined as:

$$h_{iS} = jC_p \dot{M} \Pr^{2/3}$$

The shell side actual heat transfer coefficient is given in equation:

$$h_S = h_{iS} j_b j_c j_l j_s j_r$$

j_c is the correction factor for baffle cut given by:

$$j_c = 1.27 - 1.44\left(\frac{\cos^{-1}\left(\frac{D_S}{D_S - L_{bb} - D_t}\left(1 - \frac{B_c}{50}\right)\right)}{180} - \frac{\sin\left(2\cos^{-1}\left(\frac{D_S}{D_S - L_{bb} - D_t}\left(1 - \frac{B_c}{50}\right)\right)\right)}{2\pi}\right)$$

j_b is the correction factor for bundle bypass flow is given by:

$$j_b = \exp\left(-1.25\frac{(L_{bb} + 0.5D_t)}{L_{bb}L_{tp,eff} + (D_S - L_{bb} - D_t)(L_{tp} - D_t)}\left(1 - \sqrt{\frac{100N_{ss}L_{pp}}{50 - B_c}}\right)\right)$$

(3.21)

j_l is the correction factor for baffle leakage flow is given as:

$$j_l = 0.44(1 - r_s) + [1 - 0.44(1 - r_s)]e^{-2.2r_{lm}}$$

(3.22)

$$r_s = \frac{S_{sb}}{S_{sb} + S_{tb}} \qquad r_s = \frac{S_{sb} + S_{tb}}{S_m}$$

Where S_m is the cross flow area at the bundle centerline, is given by

$$S_m = L_{bc}\left[L_{bb} + \frac{D_S - L_{bb} - D_t}{L_{tp,eff}}(L_{tp} - D_t)\right]$$

S_{sb} is the shell-to-baffle leakage area, given by

$$S_{sb} = 0.00436 D_S L_{sb}\left(360 - 2\cos^{-1}\left[1 - \frac{B_c}{50}\right]\right)$$

S_{tb} is the tube-to-baffle hole leakage area, is given by

$$S_{tb} = \left\{\frac{\pi}{4}\left[(D_t + L_{tb})^2 - D_t^{\,2}\right]\right\}(N_{tt})(1 - F_w)$$

j_s is the correction factor for variable baffle spacing is presented as:

$$j_s = \frac{N_b - 1 + \left(\frac{L_{bi}}{L_{bc}}\right)^{0.4} + \left(\frac{L_{bo}}{L_{bc}}\right)}{N_b - 1 + \frac{L_{bi}}{L_{bc}} + \frac{L_{bo}}{L_{bc}}}$$

j_r is the correction factor for adverse temperature gradient, which is given as:

For $\mathrm{Re}_S \le 20$ $\qquad j_r = (j_r)_r = \frac{1.51}{N_c^{\,0.18}}$

For $20 \le \mathrm{Re}_S \le 100$

$$j_r = (j_r)_r + \left(\frac{20 - \mathrm{Re}_S}{80}\right)[(j_r)_r - 1]$$

For $\mathrm{Re}_S \ge 100$ $\qquad j_r = 1$

N_c is the total number of tube rows crossed in the entire heat exchanger:

$$N_c = (N_{tcc} + N_{tcw})(N_b - 1)$$

In addition, the shell side heat transfer coefficient is given by the following Nusselt number correlation:

$$\frac{h_S D_e}{k_S} = 0.36 \mathrm{Re}_S^{\,0.55} \mathrm{Pr}_S^{\,1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

Equation (3.34) is given Kern and Krauss [21, 22]. Various correction factors for heat transfer coefficient for shell side flow are calculated as per suggested in the Delaware method. The correlations for tube side Nusselt number are:

For $\mathrm{Re}_T \le 2100$

$$\frac{h_T D_{ti}}{k_T} = 1.86 \left(\mathrm{Re}_T . \mathrm{Pr}_{T.} \frac{D_{ti}}{L}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

For $2100 < \mathrm{Re}_T \le 10000$

$$\frac{h_T D_{ti}}{k_T} = 0.116 \left(\mathrm{Re}_T^{\,2/3} - 125\right) \left(\mathrm{Pr}_T\right)^{1/3} \left(1 + \left(\frac{D_{ti}}{L}\right)^{2/3}\right) \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

For $\mathrm{Re}_T > 10000$

$$\frac{h_T D_e}{k_T} = 0.023 \mathrm{Re}_T^{\,0.8} \mathrm{Pr}_T^{\,1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

Pressure Drop

The shell side pressure drop [20] is calculated as a summation of the pressure drops for the inlet and exit sections (ΔP_e), the internal cross flow sections (ΔP_c), and the window sections (ΔP_w). For a shell-and-tube exchanger, the combined pressure drop is given as:

$$\Delta P_S = \Delta P_c + \Delta P_w + \Delta P_e$$

The zonal pressure drops are calculated from ideal pressure drop correlations and correlation factors, which take account of bundle bypassing and leakage effects. The baffled cross flow pressure drop is given by:

$$\Delta P_c = (N_b - 1) \Delta P_{ci} R_b R_l$$

The end zone pressure drop is given by:

$$\Delta P_e = 2 \Delta P_{ci} R_b \left(1 + \frac{N_{tcw}}{N_c}\right)$$

and, the window pressure drop by:

$$\Delta P_w = N_b \Delta P_{wi} R_l$$

The correction factors for shell side pressure drop are given as:

$$R_l = \exp\left[-1.33 (1 + r_s)(r_{lm})^{-0.15(1+r_s)+0.8}\right] \tag{3.41}$$

$$R_s = \left(\frac{L_{bc}}{L_{bo}}\right)^{2-n} + \left(\frac{L_{bc}}{L_{bi}}\right)^{2-n}$$

$$R_b = \exp\left[-3.7 \frac{S_b}{S_m} \left(1 - \sqrt[3]{\frac{100 N_{ss} L_{pp}}{50 - B_c}}\right)\right]$$

According to Kern and Krauss [22], the shell side pressure drop is given by the following expression:

$$\Delta P_S = \frac{2 f_S G_S^{\,2} D_S (N_b - 1)(NS)}{D_e \rho_S (\mu/\mu_w)^{0.14}}$$

The shell side friction factor correlation is of the form: $f_S = 0.4475 \mathrm{Re}_S^{\,-0.19}$

The tube side pressure drop is given by:

$$\Delta P_T = \frac{2 f_T G_T^{\ 2} L(N_{tp})(NS)}{D_{ti}\rho_T (\mu/\mu_w)^{0.14}} + \frac{1.25 G_T^{\ 2}(N_{tp})(NS)}{\rho_T}$$

The first term is due to friction and the second term is due to return losses. Most of the pressure drop is due to surface friction inside the exchanger in an attempt to increase the heat transfer. Therefore, only the straight tube pressure is considered. For smooth pipes, the correlations for friction factor are of the form: $f_T = K_T \,\mathrm{Re}_T^{\ -mt}$

Note that $K_T = 16$, $mt = $ -1 for $\mathrm{Re}_T \le 2100$, whereas $K_T = 0.046$, $mt =$ -0.2 for $\mathrm{Re}_T > 2100$

The overall heat transfer coefficient (*U*) is related to individual heat transfer coefficient as:

$$\frac{1}{U} = \frac{1}{h_S} + \frac{1}{h_T}\frac{D_t}{D_{ti}} + \frac{D_t}{2k_T}\ln\left(\frac{D_t}{D_{ti}}\right) + R_f$$

It is essential that the designer of shell-and-tube heat exchangers becomes familiar with the principles of the various correlations and methods in numerous publications, their advantages and disadvantages, limitations and degrees of sophistication versus probable accuracy and other related aspects. All the published methods can be logically divided into several groups:

1. The early developments based on flow over ideal tube banks or even single tubes.
2. The "integral" approach, which recognizes baffled cross flow modified by the presence of window, but treats the problem on an overall basis without considerations of the modified effects of leakage and bypass.
3. The "analytical" approach based on Tinker's multistream model and his simplified method.
4. The "stream analysis method", which utilizes a rigorous reiterative approach based on Tinker's model.
5. The Delaware method, which uses the principles of the Tinker's model but interprets them on an overall basis, that is, without reiterations.
6. Numerical prediction methods.

Nanofluids clearly exhibit enhanced thermal conductivity, which goes up with increasing volumetric fraction of nanoparticles. The current review does concentrate on this relatively new class of fluids and not on colloids which are nanofluids because the latter have been used for a long time. Review of experimental studies clearly showed a lack of consistency in the reported results of different research groups regarding thermal properties. The effects of several important factors such as particle size and shapes, clustering of particles, temperature of the fluid, and dissociation of surfactant on the effective thermal conductivity of nanofluids have not been studied adequately. It is important to do more research so as to ascertain the effects of these factors on the thermal conductivity of wide range of nanofluids. Classical models cannot be used to explain adequately the observed enhanced thermal conductivity of nanofluids. Recently most developed models only include one or two postulated mechanisms of nanofluids heat transfer. For instance, there has not been much fundamental work reported on the determination of the effective thermal diffusivity of nanofluids nor heat transfer coefficients for nanofluids in natural convection. There is a growth is the use of colloids which are nanofluids in the biomedical industry for sensing and imaging purposes. This is directly related to the ability to design novel materials at the nanoscale level alongside recent innovations in analytical and imaging technologies for measuring and manipulating nanomaterials. This has led to the fast development of commercial applications which use a wide variety of manufactured nanoparticles. The production, use and disposal of manufactured nanoparticles will lead to discharges to air, soils and water systems. Negative effects are likely and quantification and minimization of these effects on environmental health is necessary. True knowledge of concentration and physicochemical properties of manufactured nanoparticles under realistic conditions is important to predicting their fate, behavior and toxicity in the natural aquatic environment. The aquatic colloid and atmospheric ultrafine particle literature both offer evidence as to the likely behavior and impacts of manufactured nanoparticles, and there is no pretense that a review duplicating similar literature about the use of colloids which are also nanofluids is attempted in the current review. Owing to their enhanced properties as thermal transfer fluids for instance, nanofluids can be used in a plethora of engineering applications ranging from use in the automotive industry to the medical arena to use in power plant cooling systems as well as computers.

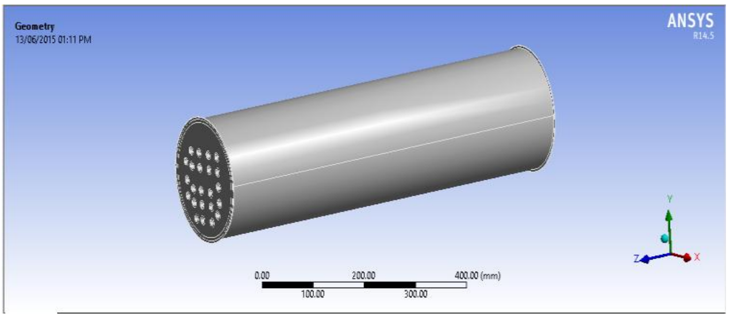
THERMAL ANALYSIS OF SHELL AND TUBE HEAT EXCHANGER

ALUMINUM

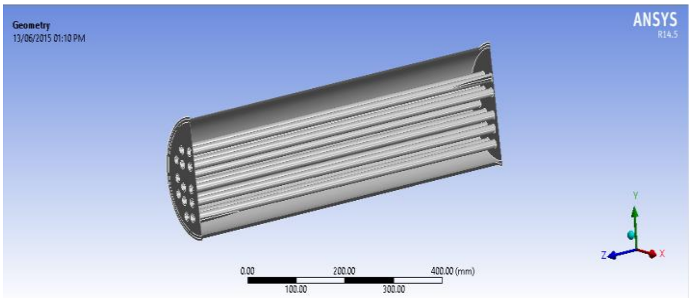
CASE 1 - VOLUME FRACTION OF 0.03

Open work bench 14.5>select **steady state thermal** in analysis systems>select geometry>right click on the geometry>import geometry>select **IGES** file>open

Imported model



Section view



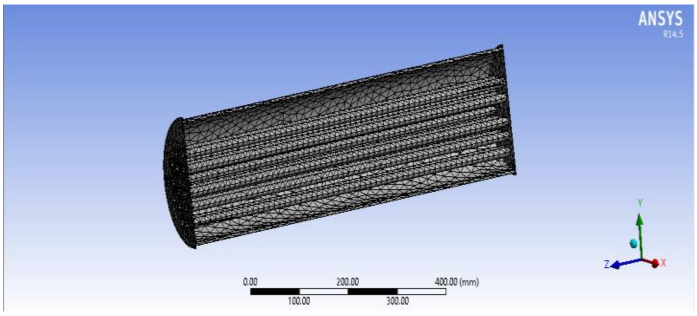
Aluminum material properties

Thermal conductivity of aluminum = 35w/mk

Specific heat =896 j/kg k

Density = 0.0000027kg/mm^3

Model >right click>edit>select generate mesh



Nodes	17361
Elements	2691

Boundary conditions

T_1 =303 k

T_2 =353 k

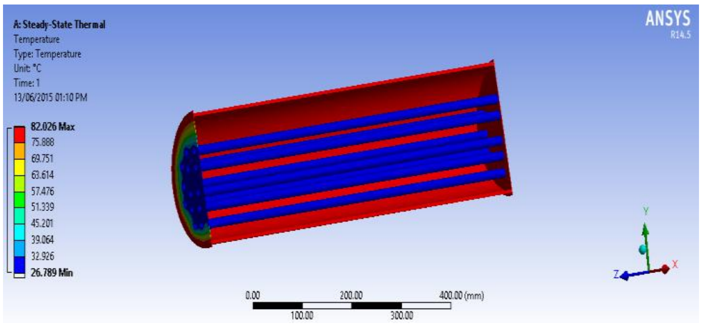
Select steady state thermal >right click>insert>select convection

Select steady state thermal >right click>insert>select heat flux

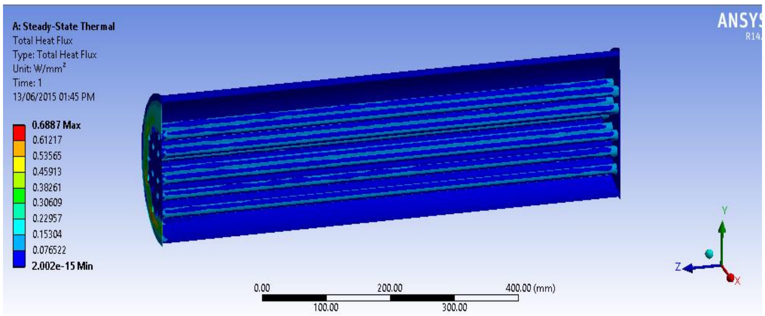
Select steady state thermal >right click>solve

Solution>right click on solution>insert>select temperature

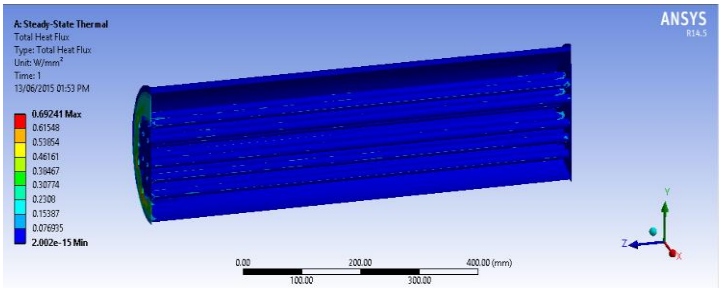
Temperature



Heat flux

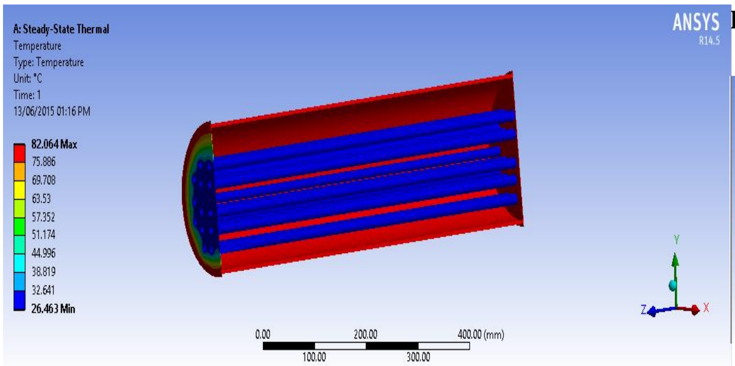


Heat flux



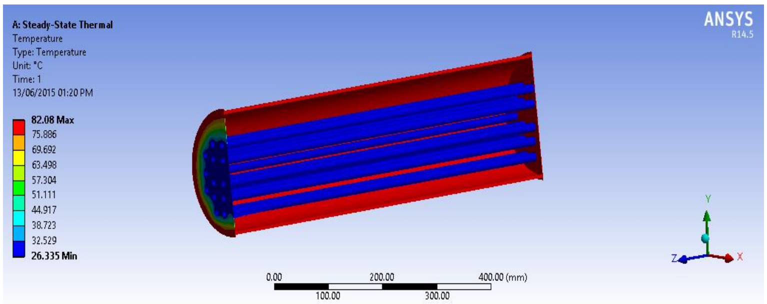
CASE 2 - VOLUME FRACTION OF 0.054

Temperature

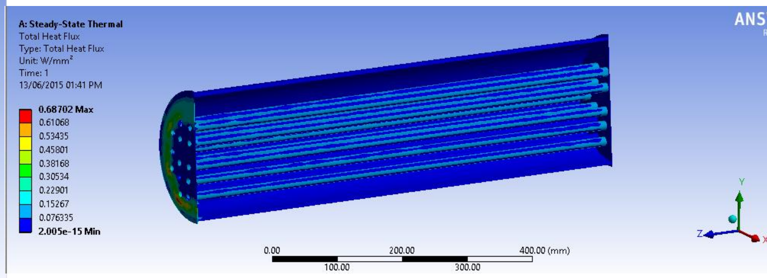


CASE 3 - VOLUME FRACTION OF 0.067

Temperature

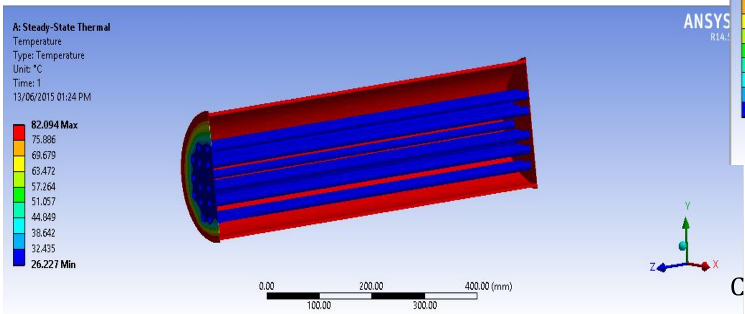


Heat flux

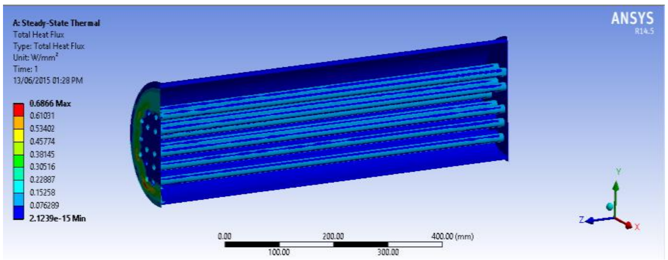


CASE 4 - VOLUME FRACTION OF 0.135

Temperature



Heat flux



COPPER

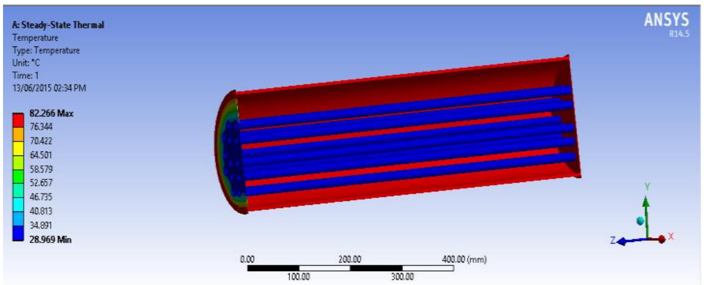
Material properties

Thermal conductivity of copper = 385w/mk

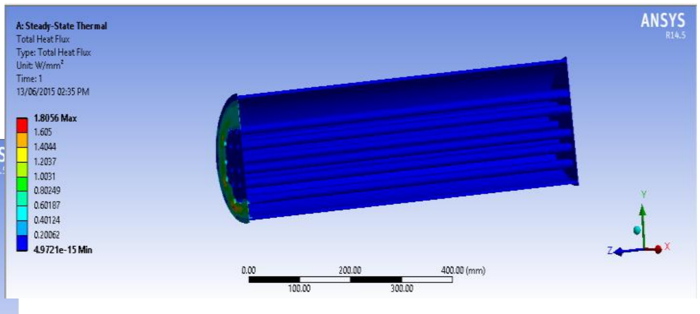
Specific heat =385 J/kgk

CASE 1 - **VOLUME FRACTION OF 0.03**

Temperature

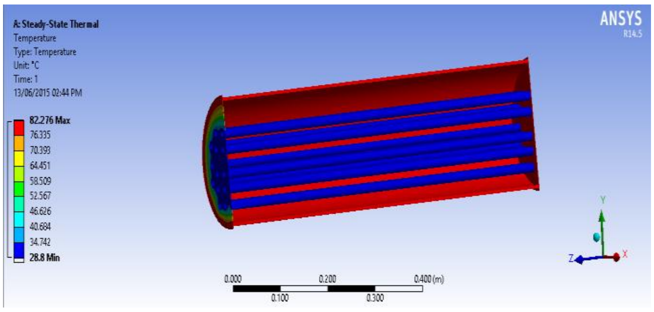


Heat flux

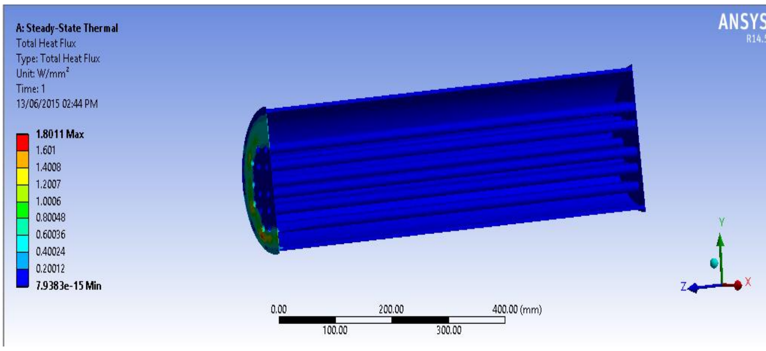


CASE 2 - **VOLUME FRACTION OF 0.054**

Temperature

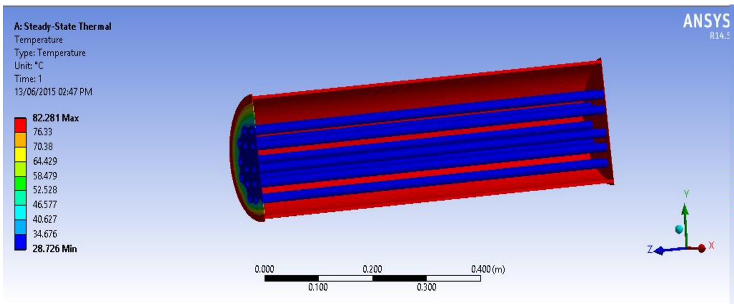


Heat flux

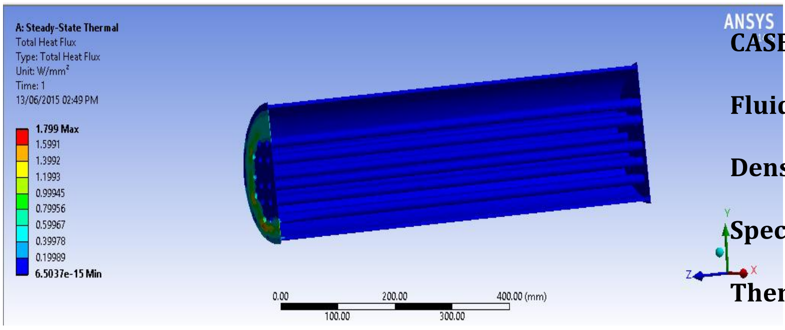


CASE 3 - **VOLUME FRACTION OF 0.067**

Temperature

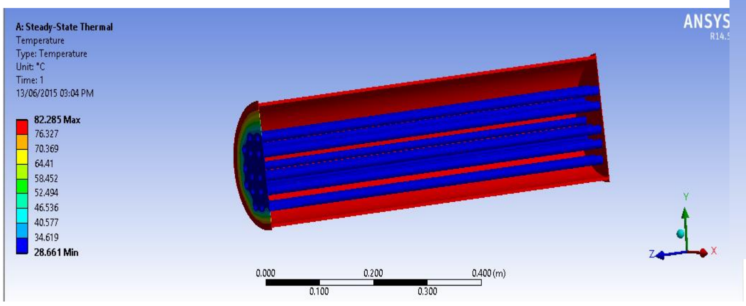


Heat flux

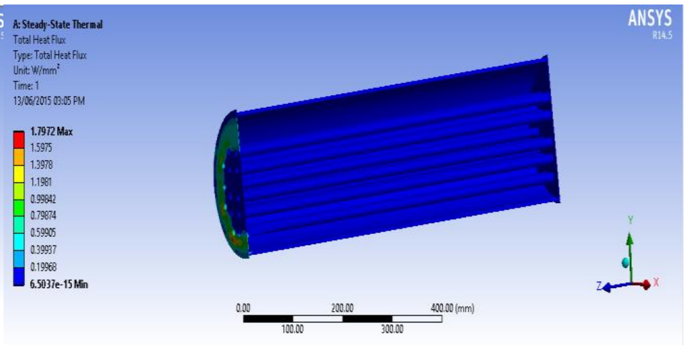


case 4 - **volume fraction of 0.135**

Temperature



Heat flux



CFD ANALYSIS

CASE 1 - VOLUME FRACTION OF 0.03

Fluid properties

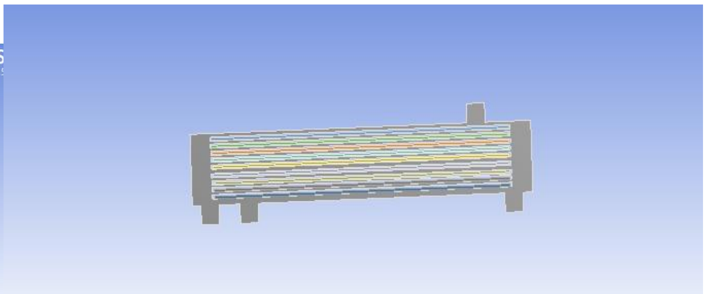
Density =1083 kg/mm³

Specific heat = 4182.95 J/Kgk

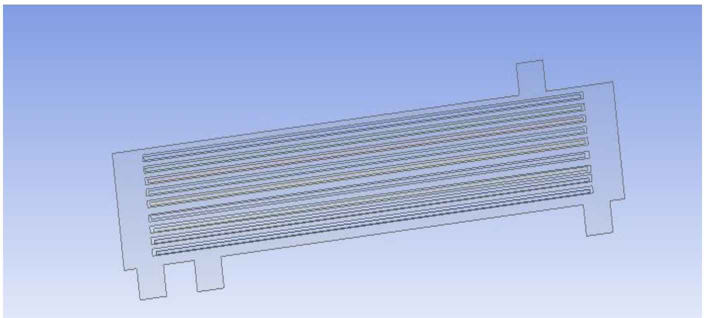
Thermal conductivity =0.6285(w/mk)

Viscosity = 0.0961 Kg/ms

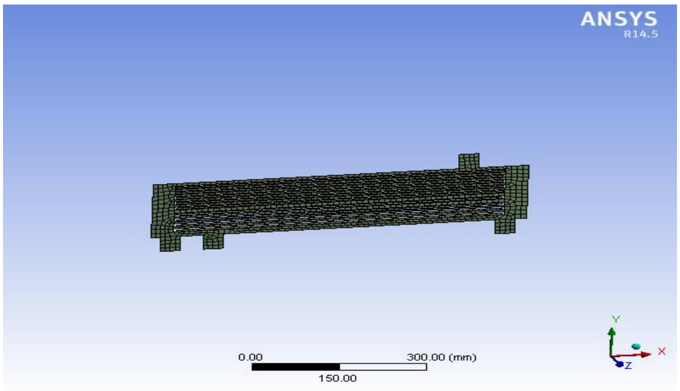
GEOMETRY



Fluid geometry



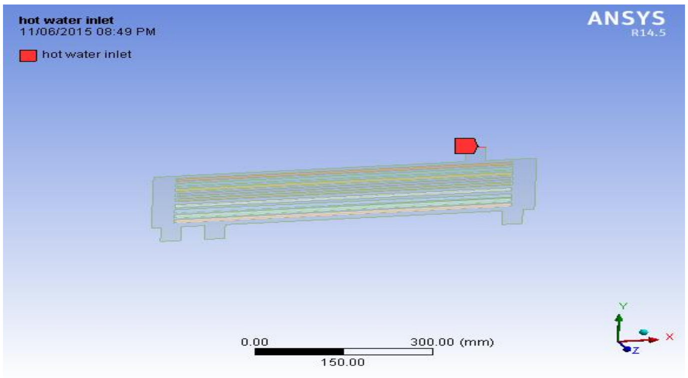
Meshed model



Select faces → right click → create named section → enter name → cold fluid inlet

Select faces → right click → create named section → enter name → cold fluid outlet

Select faces → right click → create named section → enter name → hot water inlet

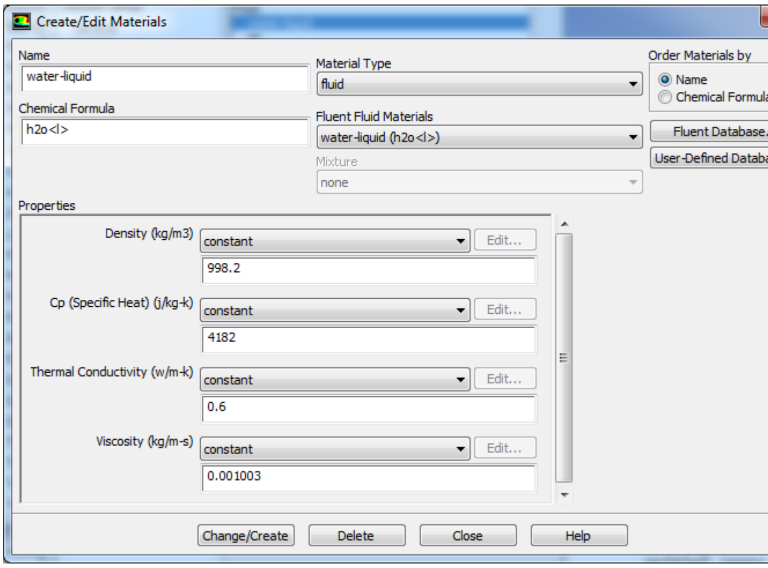


Select faces → right click → create named section → enter name →hot water outlet

Update project>setup>edit>model>select>energy equation (on)>ok

Materials> Materials > new >create or edit >specify fluid material or specify properties > ok

Select fluid



Boundary conditions

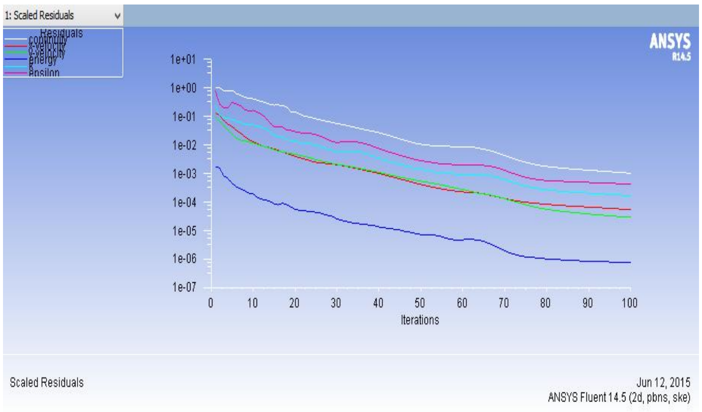
Inlet temperatures(T)	*303k,353 K*
Inlet pressure(P)	*101325 Pa*
Inlet velocity(V)	*1.4412 m/s*

Solution > Solution Initialization > Hybrid Initialization >done

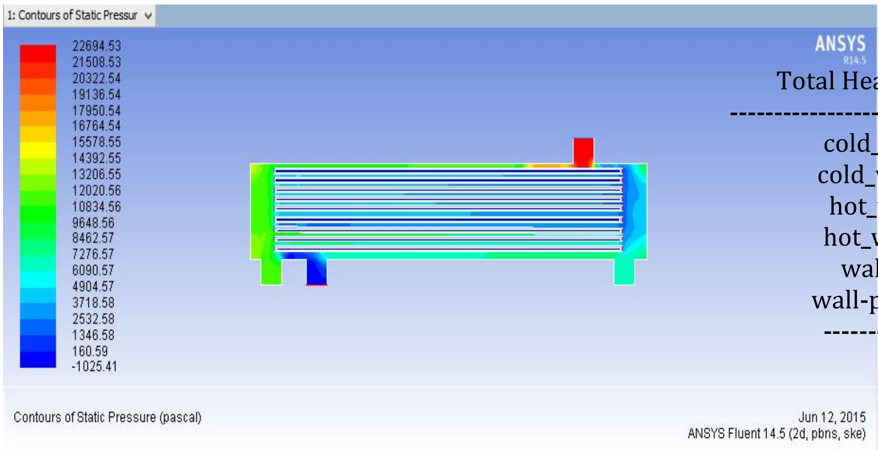
Run calculations > no of iterations = 100> calculate > calculation complete>ok

Results>edit>select contours>ok>select location (inlet, outlet, wall.etc)>select pressure>apply

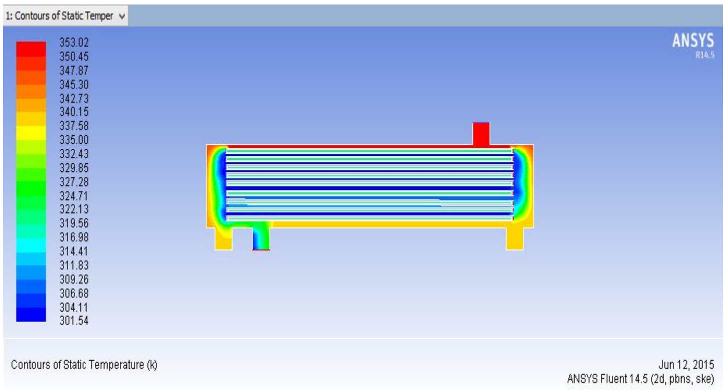
Iterations



Pressure



Temperature



Mass flow rate

"Flux Report"

Mass Flow Rate	(kg/s)
cold_water_inlet	103.57964
cold_water_outlet	-108.75418
hot_water_inlet	51.458931
hot_water_outlet	-46.267403
interior-_trm_srf	-501.7912
interior-part-_trm_srf	0
wall-_trm_srf	0
wall-part-_trm_srf	0
Net	0.016983032

Heat transfer rate

"Flux Report"

Total Heat Transfer Rate	(w)
cold_water_inlet	2098665
cold_water_outlet	-8188493.5
hot_water_inlet	11803789
hot_water_outlet	-5711336
wall-_trm_srf	0
wall-part-_trm_srf	0
Net	2624.5

CASE 2 - VOLUME FRACTION OF 0.054

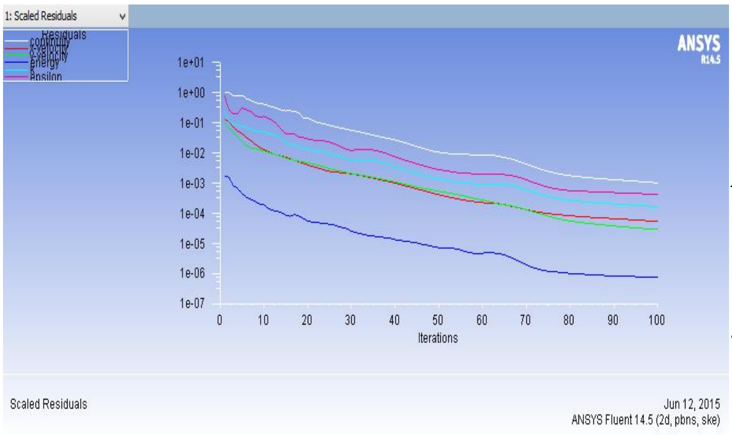
Fluid properties

Density =1156.06 kg/mm^3 **Specific heat =** 4180.519 J/Kg K

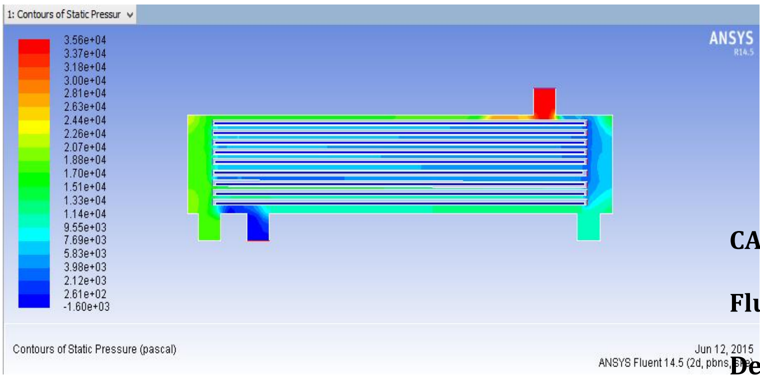
Thermal conductivity =0.67511(W/mm K)

Viscosity = 0.1014 Kg/ms

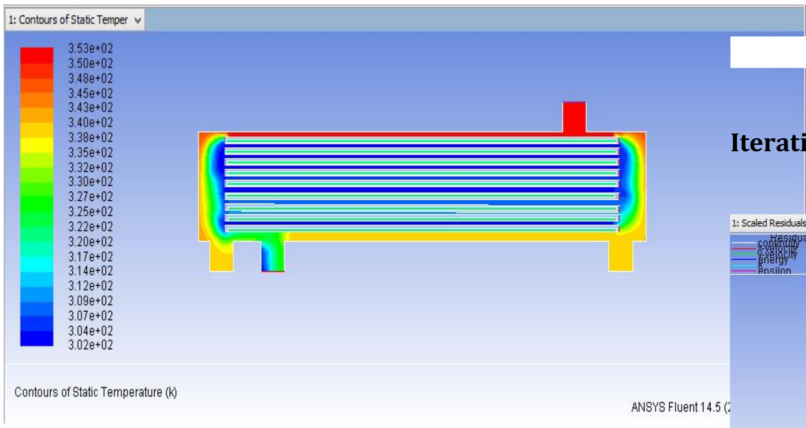
Iterations



Pressure



Temperature



Mass flow rate

"Flux Report"

Mass Flow Rate (kg/s)
-------------------------------- --------------------
cold_water_inlet 140.86598
cold_water_outlet -147.56003
hot_water_inlet 69.983002
hot_water_outlet -63.28318
interior-_trm_srf -677.32806
interior-part-_trm_srf 0
wall-_trm_srf 0
wall-part-_trm_srf 0
---------------- --------------------
Net 0.0057754517

Heat transfer rate

"Flux Report"

Total Heat Transfer Rate (w)
-------------------------------- --------------------
cold_water_inlet 3059035.5
cold_water_outlet -11737987
hot_water_inlet 17199524
hot_water_outlet -8518247
wall-_trm_srf 0
wall-part-_trm_srf 0
---------------- --------------------
Net 2325.5

CASE 3 - VOLUME FRACTION OF 0.067

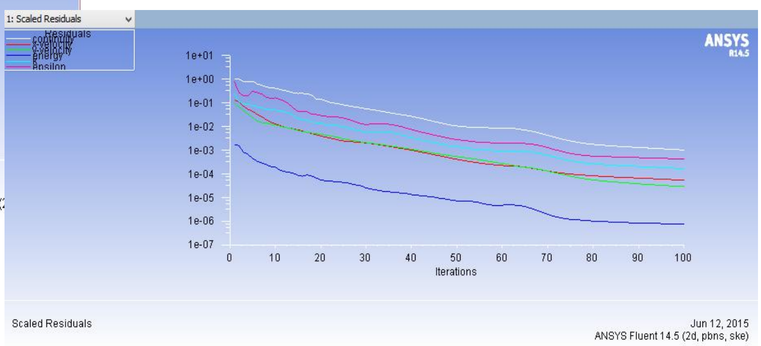
Fluid properties

Density =1193 kg/mm^3 **Specific heat =** 4179.19 J/Kg J

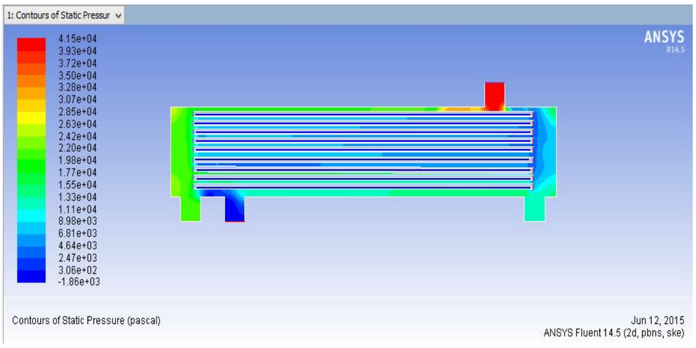
Thermal conductivity =0.69934(W/mm K)

Viscosity = 0.1043 Kg/ ms

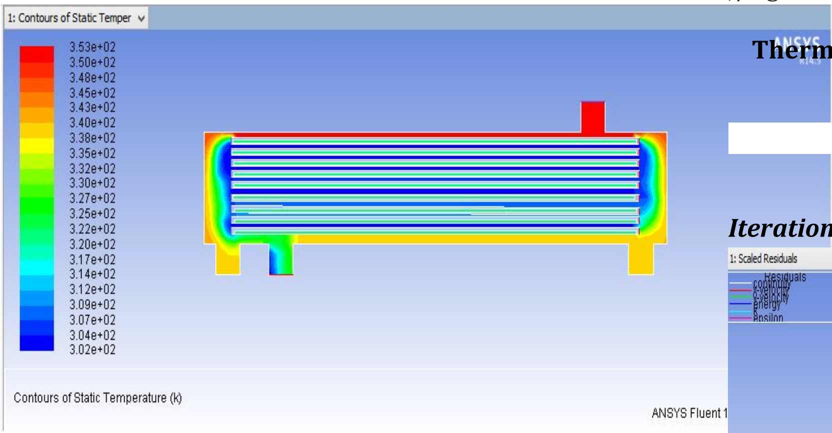
Iterations



Pressure



Temperature



Mass flow rate

"Flux Report"

Mass Flow Rate	(kg/s)
cold_water_inlet	150.92784
cold_water_outlet	-158.0211
hot_water_inlet	74.981781
hot_water_outlet	-67.882912
interior-_trm_srf	-733.61743
interior-part-_trm_srf	0
wall-_trm_srf	0
wall-part-_trm_srf	0
Net	0.005607605

Heat transfer rate

Total Heat Transfer Rate	(w)
cold_water_inlet	2854924
cold_water_outlet	-10983589
hot_water_inlet	16052890
hot_water_outlet	-7921921
wall-_trm_srf	0
wall-part-_trm_srf	0
Net	2304

CASE 4 - VOLUME FRACTION OF 0.135

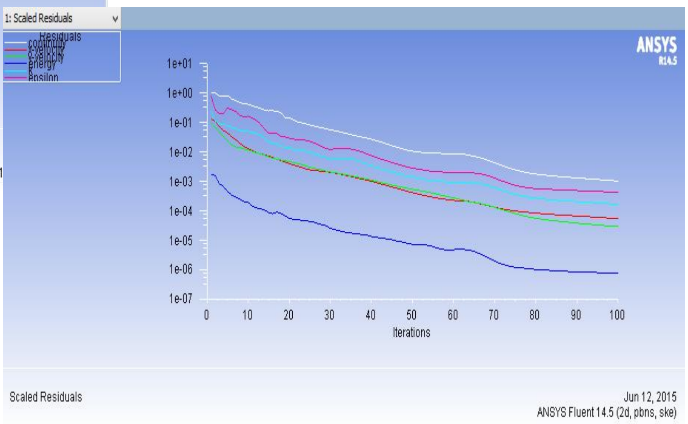
Fluid properties

Density =1390 kg/mm^3 **Specific heat =** 4172.29 J/Kg K

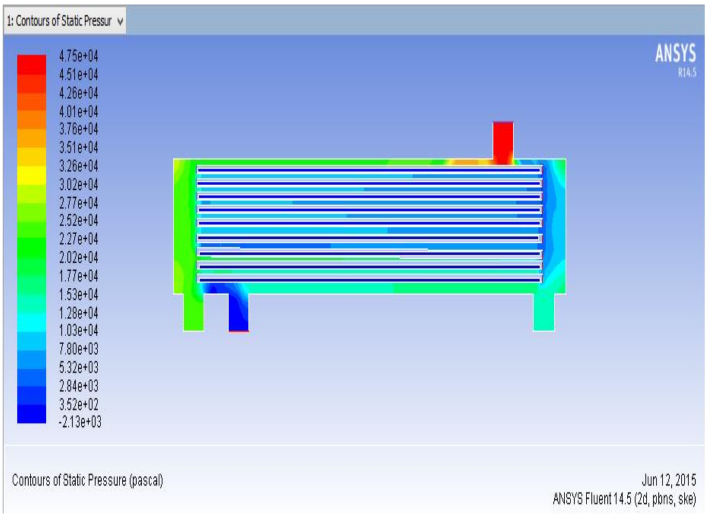
Thermal conductivity =0.88913(W/mm K)

Viscosity = 0.1195 Kg/ ms

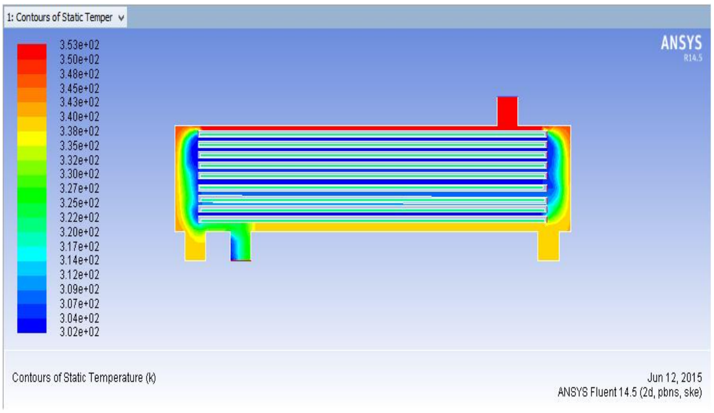
Iterations



Pressure



Temperature



Mass flow rate

```
                "Flux Report"

             Mass Flow Rate              (kg/s)
-------------------------------- --------------------
            cold_water_inlet         130.22917
           cold_water_outlet         -136.50139
             hot_water_inlet         64.69857
            hot_water_outlet         -58.42131
           interior-_trm_srf        -618.79193
      interior-part-_trm_srf                 0
               wall-_trm_srf            0
          wall-part-_trm_srf               0
            ---------------- --------------------
                   Net        0.0050430298
```

Heat transfer rate

```
                "Flux Report"

       Total Heat Transfer Rate                 (w)
-------------------------------- --------------------
            cold_water_inlet         2639156.8
           cold_water_outlet          -10185357
             hot_water_inlet         14840733
            hot_water_outlet         -7292392.5
               wall-_trm_srf            0
          wall-part-_trm_srf               0
            ---------------- --------------------
                   Net           2140.25
```

ALUMINUM

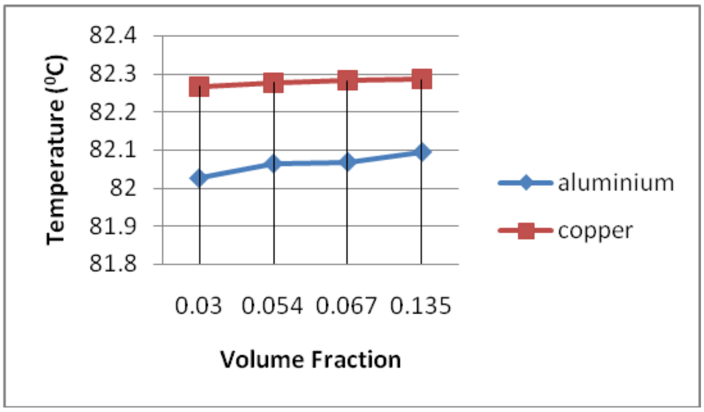
Volume fraction	***Temperature (^0C)***	***Heat flux(W/mm^2)***
0.03	*82.026*	*0.69241*
0.054	*82.064*	*0.6887*
0.067	*82.068*	*0.68702*
0.135	*82.094*	*0.6866*

COPPER

Volume fraction	***Temperature (^0C)***	***Heat flux(W/mm^2)***
0.03	*82.266*	*1.8056*
0.054	*82.276*	*1.8011*
0.067	*82.281*	*1.799*
0.135	*82.285*	*1.797*

GRAPHS

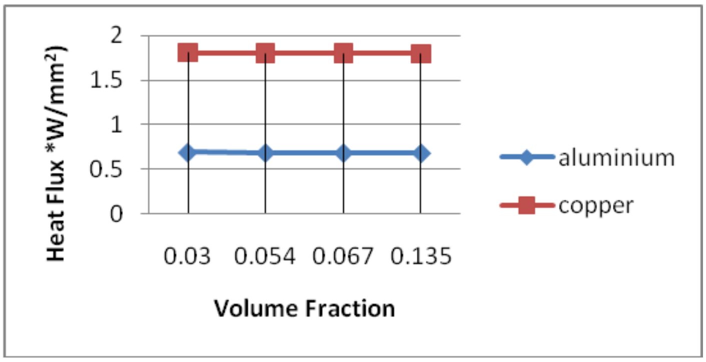
Temperature



THERMAL RESULTS TABLE

GRAPHS

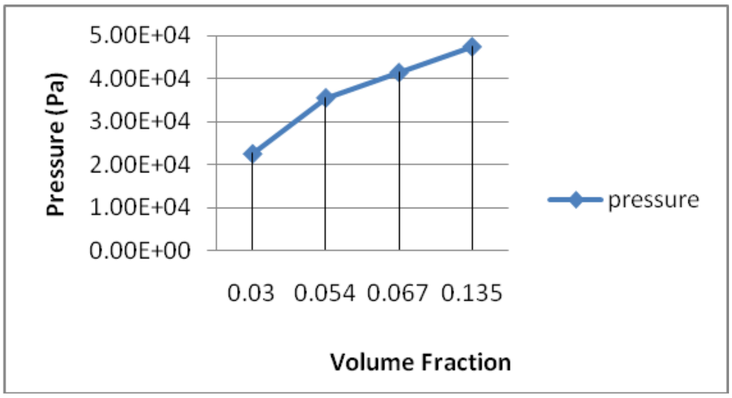
Heat flux

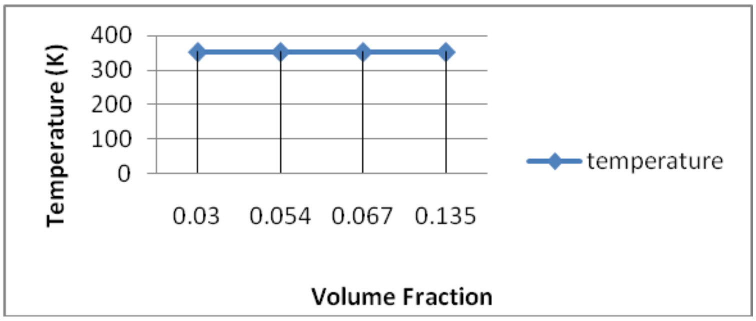


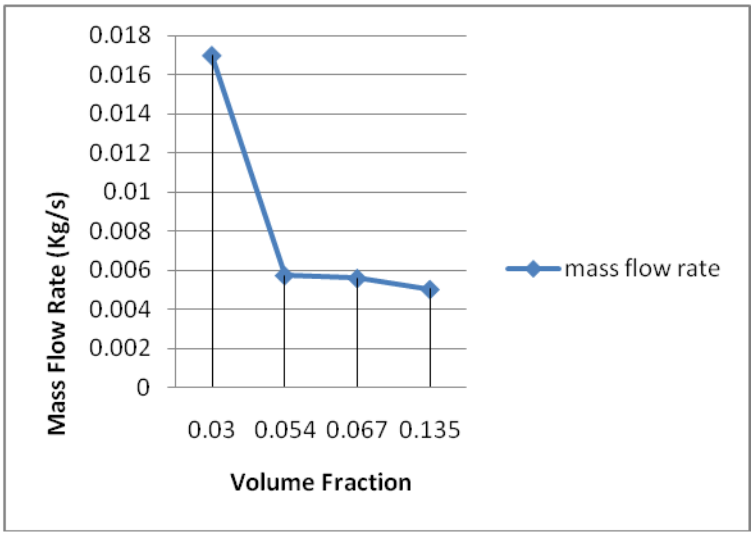
By observing the results, the variation of temperature distribution and heat flux is very less using nanofluids with different volume fractions.

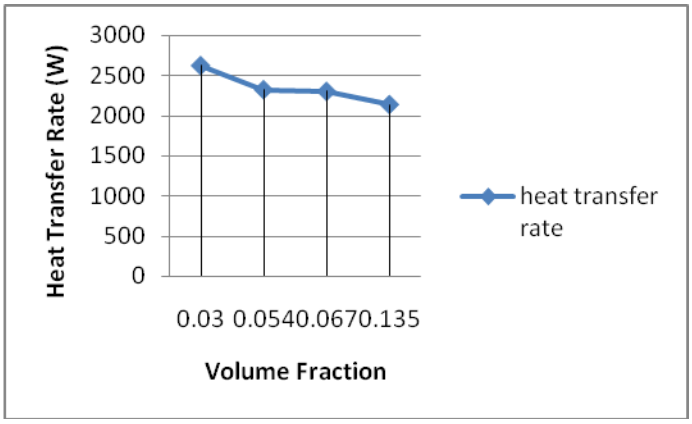
CFD RESULT TABLE

Volume fraction	***Pressure(pa)***	***Velocity (m/s)***	***Temperature (k)***	***Mass flow rate(kg /sec)***	***Heat transfer rate (W)***
0.03	*2.26e+04*	*3.88*	*353*	*0.01698*	*2624.5*
0.054	*3.56e+04*	*4.86*	*353*	*0.005751*	*2325*
0.067	*4.15e+04*	*5.26*	*353*	*0.005607*	*2304*
0.135	*4.75e+04*	*5.63*	*353*	*0.005043*	*2140*









By observing the CFD analysis results, the pressure drop is increasing by increasing the volume fraction, mass flow rate and heat transfer rate are decreasing by increasing the volume fraction.

CONCLUSION

In this thesis, analytical investigations have been done on the shell and tube heat exchanger, forced convective heat transfer and flow characteristics of a nanofluid consisting of water and different volume concentrations of Al_2O_3 nanofluid (0.03,0.054,0.067and 0.135)% flowing under turbulent flow conditions. The properties of the nanofluid with different volume fractions are calculated using theoretical calculations. Thermal and CFD analysis is done in Ansys. *By observing the thermal analysis results, the variation of temperature distribution and heat flux is very less using nanofluids with different volume fractions.* By observing the CFD analysis results, the pressure drop is increasing by increasing the volume fraction, mass flow rate and heat transfer rate are decreasing by increasing the volume fraction.

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BIOGRAPHIES

I ELUMAGANDLA SURENDAR, working as Assistant Professor and H O D in mechanical engineering department in Warangal Institute of Technology and Science,warangal, Telangana. I have been completed Master of Science in mechanical engineering from University Of Norway in April 2008, and completed Master Of Technology in Thermal Engineering in 2014. I got nearly 10+ years of teaching experience and carried different kind of projects during this period.also attended many National level conferences and technical workshops organised in different colleges or universities.