A NUMERICAL STUDY ON A CERAMIC HEAT EXCHANGER WITH STACKS OF DIFFERENT CROSS SECTIONS

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ABSTRACT

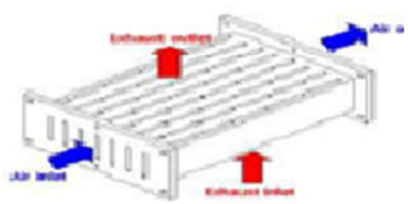
Energy is one of the most important components of economic growth. But day by day the existing energy resources are depleting at alarming rate. Hence it is necessary to introduce alternate techniques to conserve the energy effectively. With this an attempt has been made to introduce a ceramic heat exchanger with different cross sections.

In this project Ceramic heat exchanger of varying tubes were simulated by computational fluid dynamics method (CFD). The multi shaped structure was imported in to ANSYS fluent 15.0 versions as a physical model. A ceramic monolith heat exchanger is designed to find out the performance and effectiveness of heat transfer. The numerical computation was performed throughout the domain including fluid region in exhaust gas side, ceramic core and fluid region in air side. The entire computation was carried out by using different cross sections viz., Rectangular, Elliptical and Cylindrical duct of 6mm and 8mm dia with air and exhaust in cross flow direction. After comparison of theoretical and numerical computation it is observed that the estimated heat transfer rate in numerical analysis is 15% more than the theoretical analysis

INTRODUCTION

A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. Heat exchangers are important engineering devices in many process industries since the efficiency and economy of the process largely depend on the performance of the heat exchangers. The fluids may be separated by a solid wall to prevent mixing or they may be in direct contact. They are widely used in space heating, refrigeration, air conditioning, power stations, chemical plants, petrochemical plants, petroleum refineries, natural-gas processing, and sewage treatment. The classic example of a heat exchanger is found in an internal combustion engine in which a circulating fluid known as coolant flows through radiator coils and air flows past the coils, which cools the coolant and heats the incoming air. Due to the many variables involved, selecting optimal heat exchangers is challenging. Hand calculations are possible, but many iterations are typically needed. As such, heat exchangers are most often selected via computer programs, either by system designers, who are typically engineers, or by equipment vendors. To select an appropriate heat exchanger, the system designers (or equipment vendors) would firstly consider the design limitations for each heat exchanger type. Though cost is often the primary criterion, several other selection criteria are important, High/low pressure limits, Thermal performance, Temperature ranges, Product mix (liquid/liquid, particulates or high-solids liquid), Pressure drops across the exchanger, Fluid flow capacity, Cleanability, maintenance and repair, Materials required for construction, Ability and ease of future expansion Material selection, such as copper, aluminum, carbon steel, stainless steel, nickel alloys, ceramic, polymer, and titanium. Small-diameter coil technologies are becoming more popular in modern air conditioning and refrigeration systems because they have better rates of heat transfer than conventional sized condenser and evaporator coils with round copper tubes and aluminium or copper fin that have been the standard in the HVAC industry. Heat exchangers can be classified in a number of ways, depending on their construction or on how fluid move relative to each other through the device. There are various heat exchangers in which some heat exchangers are, Double pipe heat exchanger Shell and tube heat exchanger, Compact heat exchanger, Cross flow heat exchanger and Ceramics

The word "ceramic" came from Greek word it means A ceramic is an inorganic, nonmetal or metalloid atoms primarily held in ionic and covalent bonds pottery.

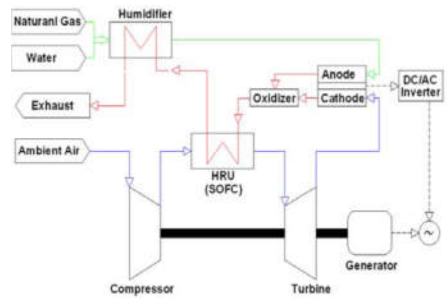


Low temperature - Although ceramic heat exchangers lend themselves to medium and high temperature applications, they can be used in the 500°F to 1400°F range. If the flue gas is corrosive and/or abrasive, the machine is a practical piece of equipment to preheat combustion air for dryers and other similar processes

Medium temperature - Most ceramic exchangers are designed for the 1400°F to 2200°F range, and Heat Transfer International has several full size exchangers operating successfully in these ranges.

High temperature - Special ceramics can be incorporated into the Heat Transfer

In this study, the ceramic heat exchanger of 3 pass recuperator was analyzed to predict the performance, for example, heat transfer rate, effectiveness, and pressure drop, and so on since the ceramic heat exchanger has characteristics of cheap material cost, but low thermal efficiency compared to metallic heat exchangers.



OBJECTIVES OF PRESENT STUDY

The present work include following objectives

1. Designing of Rectangular, circular and elliptical shapes using CREO-PARAMETRIC 3.0

2. Simulating the designs with ANSYS FLUENT 15.

3. To enhance heat transfer coefficient of Rectangular, circular and elliptical tubes.

4. Calculating the heat transfer rate.

5. Calculation of heat transfer coefficient for Rectangular, circular and elliptical shape tubes

6. Comparing heat transfer coefficient between optimized

General design considerations

- For high temperature heat exchangers, the thermal stresses during the startup, shutdown and load fluctuations can be significant. Heat exchanger must be designed accordingly

- The thermal capacitance ("thermal mass") should be reduced for high temperature heat exchangers for shorter startup time.
- High temperature heat exchangers require costly materials contributing to the high cost of balance of power plant. Heat exchanger cost increases significantly with temperature above about 675°C.

Selection of materials for HTHEs

Three major classes of high-temperature materials are promising candidates for different applications:

High-temperature nickel-based alloys (e.g., Hastelloy). Good material compatibility potential for helium and molten salts up to temperatures in the range of 750°C. Also a candidate material for sulfuric acid thermal decomposition. Limited capability under fusion neutron irradiation.

High-temperature ferritic steels (particularly oxide- dispersion ferritic steels). Good performance under fusion and fission neutron irradiation, to temperatures around 750 °C Good potential for compatibility with lead/bismuth under appropriate chemistry control. Demonstrated compatibility with molten salts would have substantial value for the fusion application. Silica bearing steels provide a candidate material for sulfuric acid thermal decomposition.

Advanced carbon and silicon carbide composites. With excellent mechanical strength to temperatures exceeding 1000°C, these are now used for high temperature rocket nozzles to eliminate the need for nozzle cooling and for thermal protection of the space shuttle nose and wing leading edges. Many options are available that trade fabrication flexibility and cost, neutron irradiation performance, and coolant compatibility. These materials can potentially be used with helium and molten salt coolants. Silicon carbide is also compatible with sulfur-iodine thermochemical hydrogen production. Major opportunities and research challenges exist to apply these materials to high-temperature heat transport applications.

The best material available seems to be a SiCp/Al2O3, (particles reinforcing phase-based material), from a US manufacturer; no European manufacturer could supply Ceramic Matrix Composites (CMC) bayonet tubes.

4. THERMAL DESIGN

In the work of Fischedick et al (2007), the thermal design of the HTHE was conducted by using correlations for the Colburn and friction factors for offset strip fins. These correlations were obtained from experiments by Manglik and Bergles (1995).

The present work uses CFD simulations for the thermal design task. This choice can be justified by the possibility of considering the heat conduction in the ceramic material coupled with the convective heat transfer, technique known as conjugate heat transfer. This type of simulation is particularly important since it can provide the temperature distribution in the ceramic material as a result that can be used as input to the structural design.

The number of transfer units (NTU) method is used for the present analysis and design. The theory related to NTU can be found in many texts from literature. It states that the effectiveness ε depends on the number of heat transfer units NTU and the heat capacity rates of the hot and cold flows Cr. This dependence is summarized by Eq. (1).

$$\varepsilon = \varepsilon(NTU, Cr) \quad (1)$$

The number of transfer units (NTU) and the ratio between heat capacity rates are given by Eq. (2) and (3), respectively.

NTU =UA/Cmin (2)

Cr =Cmin/Cmax (3)

Here, U is the overall heat transfer coefficient, A is the heat transfer area, and Cmin and Cmax are the minimum and maximum capacity rates.

MODELLING AND SIMULATION:

The heat exchanger is formed by taking a rectangular ceramic block of 315×202.5×241.5 mm . Ducts for the flows are cut in rectangular, elliptical and cylindrical shapes as per the designed dimensions. The whole model is created in CREO 3.0.

COMPUTATIONAL FLUID DYNAMICS

Computational Fluid Dynamics (CFD) is the science of predicting fluid flow, heat transfer, mass transfer, chemical reaction (e.g., combustion), and related phenomena by solving the mathematical equations that govern these processes using a numerical algorithm on a computer. The technique is very powerful and spans a wide range of industrial and non-industrial application areas.

4.1 GOVERNING EQUATIONS OF FLUID FLOW:

The governing equations of fluid flow represent mathematical statements of the conservation laws of physics. Each individual governing equation represents a conservation principle. The fundamental equations of fluid dynamics are based on the following universal laws of conservation. They are,

- Conservation of mass
- Conservation of momentum
- Conservation of energy

4.1.1 Continuity Equation:

The equation based on the principle of conservation of mass is called continuity equation. The conservation of mass law applied to a fluid passing through an infinitesimal, fixed control volume yields the following equation of continuity,

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0$$

$$\frac{\partial \rho}{\partial t}+div(\rho V)=0$$

Where 'ρ' is the fluid density, u, v, and w is the fluid velocity vectors. For an incompressible flow, the density of each fluid element remains constant.

4.1.3 Energy Equation:

This equation is based on the principle of conservation of energy; the energy equation is derived from first law of thermodynamics which states that the rate change of energy of a fluid particle is equal to the rate of heat addition to the fluid particle plus the rate of work done on particle, which is

$\rho \frac{DE}{Dt}$ = The rate of change energy of a fluid particle

E = Internal energy + kinetic energy + gravitational energy

$$E = i + ½\left(u^2+v^2+w^2\right) + g$$

$$-\frac{\partial q_x}{\partial x}-\frac{\partial q_y}{\partial y}-\frac{\partial q_z}{\partial z}=-div$$

q = The rate of heat addition to the fluid particle.

$$q_x, q_y, q_z = -k\frac{\partial T}{\partial x}, -k\frac{\partial T}{\partial y}, -k\frac{\partial T}{\partial z}$$

= Heat flux components

$$\left[\frac{\partial[u(-p+\tau_{xx})]}{\partial x}+\frac{\partial(u\tau_{yx})}{\partial y}+\frac{\partial(u\tau_{zx})}{\partial z}\right]\delta x\,\delta y\,\delta z$$

= Net rate of work done

by force in 'x' direction (4.1)

(4.2)

Energy equation in conservative form:

$$\frac{\partial}{\partial t}\left[\rho\left(e+\frac{V^2}{2}\right)\right]+\nabla\cdot\left[\rho\left(e+\frac{V^2}{2}\right)V\right]=\rho\dot{q}+\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)-\frac{\partial(up)}{\partial x}-\frac{\partial(vp)}{\partial y}-\frac{\partial(wp)}{\partial z}$$

$$+\frac{\partial(u\tau_{xx})}{\partial x}+\frac{\partial(u\tau_{yx})}{\partial y}+\frac{\partial(u\tau_{zx})}{\partial z}+\frac{\partial(v\tau_{xy})}{\partial x}+\frac{\partial(v\tau_{yy})}{\partial y}+\frac{\partial(v\tau_{zy})}{\partial z}+\frac{\partial(w\tau_{xz})}{\partial x}+\frac{\partial(w\tau_{yz})}{\partial y}+\frac{\partial(w\tau_{zz})}{\partial z}+pf.V$$

(4.15)

OVERALL HEAT TRANSFER COEFFICIENT OF THE CERAMIC HEAT EXCHANGER

ξ-NTU METHOD

The thermal performance of the ceramic heat exchanger was calculated by theoretical equation of ξ-NTU method for which the effectiveness(ξ) is expressed as Eq. (3) in unmixed fluid flow condition, and then compared to that by the numerical computation.by minimum heat capacity(C_{min}), where C_{min} is the lower heat capacity($C_{min}= \dot{m} \times c_{p\,min}$) and C_{max} is the higher heat capacity($C_{max}= \dot{m} \times c_{p\,max}$) of the two fluids with $\dot{m}$ and c_p are mass flow rate and specific heat of the hot and cold fluids, respectively. Then, the rate of heat transfer from hot fluid to cold fluid can be computed as Eq. (4).

$$\xi = 1 - \exp \frac{NTU^{0.22}}{C}[\exp(-CNTU^{0.78}) - 1] \quad (3)$$

Here, C is the ratio of heat capacity(C_{min}/C_{max}). NTU is defined by the total conductance(UA) divided

Reference	Correlation	Condition		Range of validity
		Geometry	Flow regime	
Kay and Crawford[2]	$Nu_{fd} = 8.235(1-1.883/\alpha+3.767/\alpha^2 -5.814/\alpha^3+5.361/\alpha^4-2/\alpha^5)$	Rectangular	Fully developed	$Re < 2200$
Sieder-Tate[3]	$Nu = 1.86(RePrD/L)^{1/3}(\frac{\mu_f}{\mu_w})^{0.14}$	Circular	Simultaneously developing	$Re < 2200$
Stephan and preußer[4]	$Nu = 4.364 + \frac{0.086(RePrD/L)^{1.33}}{1+0.1Pr(ReD/L)^{0.83}}$	Circular	Simultaneously developing (constant wall heat flux)	0.7<Pr<7 or $RePrD/L < 33$ (for Pr>7)
Shah and London[5]	$Nu = \begin{cases} 1.953(RePr\frac{D}{L})^{1/3} & (RePr\frac{D}{L}) \geq 33.3 \\ 4.364 + 0.0722RePr\frac{D}{L} & (RePr\frac{D}{L}) < 33.3 \end{cases}$	Circular	Thermally developing laminar (constant wall heat flux)	-

THEORITICAL ANALYSIS OF HEAT EXCHANGER

Calculation

1) Total Flow Area,

Air Side At = (W × H) × Number of Channels = 0.052 × 0.0065 × 7 = 0.002366 m2

Exhaust Side at = (W × H) × Number of Channels = 0.052 × 0.0065 × 8 = 0.02704 m2

2) Hydraulic Diameter, Dh = 4A/P = (4 × (0.052 × 0.0065))/(2 × (0.052 + 0.0065)) = 0.01156 m2

3) Velocity,

Air side Va = m/ρA = 0.001983/(0.391 × 0.002366) = 2.14358m/s

Exhaust Side Va = m/ρA = 0.001983/ (0.34 × 0.02704) = 0.1875 m/s

4) Reynolds Number (Re.No.)

Air Side, Re.No. = ρVD"h"/μ = (0.391 × 4.287 × "0.01156")/(39.925 × 10-6) = 585

Exhaust Side, Re.No. = ρVD"h"/μ = (0.34 × 0.863 × "0.01156")/(43.00 × 10-6) = 79

5) Nusselt Number and Convective Heat Transfer Coefficient:

Air Side,

Kays and Crawford Correlation,

Nu = 8.235(1 – 1.883⁄α+3.767⁄(α"2" –5.814 ⁄ (α"3" + 5.361⁄(α"4" -2⁄α"5"))))

Aspect Ratio, α = (0.052/0.0065) = 8

Nu = 8.235(1 – 1.883⁄8 + 3.767⁄(8"2"–5.814⁄ (8"3" + 5.361⁄(8"4"–2⁄8"5"))))

h"air" = Nu × k/D"h" = 6.7 × (0.062 × 0.01156) = 35.93w/m^r k

h"gas" = Nu × k/D"h" = 6.7 × (0.0701 × 0.01156) = 40.057 W/m2 K

Sieder-Tate correlation,

Nu = 1.86(RePrD"h"⁄L)"0.33" (μ"f"⁄μ"w")"0.14"

Nu = 1.86((585 × 0.7011 × 0.01156) ⁄ 0.335)"0.33" (0.00003875⁄0.00004305)"0.14"

Nu = 4.395

h"air" = Nu × k/D"h" = 4.394 × (0.06362 × 0.01156) = 23.56 W/m2 K

h"gas" = Nu × k/D"h" = 4.394 × (0.0701 × 0.01156) = 26.64 W/m2 K

Stephan Correlation,

Nu = 4.364 + 0.086(RePrD"h"⁄L)"1.33"/(1 + 0.1Pr (ReD"h"⁄L)"0.83")

Nu = 4.364 + 0.086((585 × 0.7011 × 0.01156) ⁄ 0.335)"1.33"/(1 + 0.1 × 0.7011((585 × 0.01156) ⁄ 0.335)"0.83")

Nu = 5.942

h"Air" = Nu × k/D"h" = 5.942 × (0.06362 × 0.01156)=31.869

h"air" = Nu × k/D"h" = 5.942 × (0.06362 ×0.1156) = 35.98 W/m2 K

Shah and London Correlation,

Nu = 4.364 + 0.0722(RePrD"h"⁄L)

Nu = 4.364 + 0.0722((585 × 0.7011 × 0.01156)⁄0.335)

Nu = 5.385

h"aIr" = Nu × k/D"h" = 5.385 × (0.06362 × 0.01156) = 28.86 W/m2 K h”gas” = Nu × k/D”h” = 5.385 × (0.07012 × 0.01156) = 32.6 W/m2 K

6) Overall Heat Transfer Coefficient:
U = 1/(1/h”air” + ΔX/k + A”air”/(η”t”A”gas”h”gas”)) = 1/(1/35.93 + 0.0065/77.5 + 0.03675/(0.75 × 0.07922 × 40.57))
U = 23.145W/m”2” K
7) Effectiveness (NTU- Method):
NTU = (UA/C”min”) = ((23.145 × 0.03675)/4.441) = 0.87118
C = Cmin/Cmax = 2.2040/2.2580 = 0.9763
ε = 1-exp {NTU”0.22”/C [exp(–CNTU”0.78”)–1]} × 100%
ε = 1–exp {0.87118»0.22»/0.984 [exp (–0.984 × 0.87118”0.78”) – 1]} × 100%
ε = 38.5%
8) Total Heat Transfer Rate:
q = ε × C»min» (T»gas_in»-T”air_in”) =0.38*2.20450(850-560)=247.677W

For elliptical:
Calculation of C for cold fluid :
C= Mass flow rate * specific heat
C=mc
=0.001983*1111.7
C= 2.20450W/K
Calculation of C for hot fluid :
C= Mass flow rate * specific heat
C=mc
=0.001983*1138.7
C=2.2580W/K
From this Cmin= 2.20450 and Cmax= 2.2580
C= Cmin/Cmax
= 2.20450/2.2580

A = 2(L*W+L*H+W*H)

2(315\1000*6.5/1000+315/1000*52/1000+6.5/1000*52/1000)
A = 0.03675m2
Surface area of circular
A = 2πr2 +2πrh
=2*π*(3.25/1000)2+2* π ∗ 3.25/1000*315/1000
A=6.49*10-3m26)
Overall Heat Transfer Coefficient:
U = 1/(1/h”air” + ΔX/k + A”air”/(η”t”A”gas”h”gas”)) = 1/(1/35.93 + 0.0065/77.5 +6.49*10^(-3)/(0.75 × 0.07922 × 40.57))=32.67W/m”2” K
7) Effectiveness (NTU- Method):
NTU = (UA/C”min”) = ((32.67 × 6.49*10^(-3))/2.20450) = 0.096
ε = 1-exp {NTU”0.22”/C [exp(–CNTU”0.78”)–1]} × 100%
C=0.9763
Surface area of a rectangle (A)

U = 1/(1/h”air” + ΔX/k + A”air”/(η”t”A”gas”h”gas”)) = 1/(1/35.93 + 0.0065/77.5 +6.49*10^(-3)/(0.75 × 0.07922 × 40.57))=32.67W/m”2” K
7) Effectiveness (NTU- Method):
NTU = (UA/C”min”) = ((32.67 × 6.49*10^(-3))/2.20450) = 0.096
ε = 1-exp {NTU”0.22”/C [exp(–CNTU”0.78”)–1]} × 100%
ε = 1–exp {0.096»0.22»/0.9763 [exp (–0.9763 × 0.096”0.78”) – 1]} × 100%
ε = 69.9%
8) Total Heat Transfer Rate:
q = ε × C»min» (T»gas_in»-T”air_in”) =0.699*2.20450(850-560)=446.874W

Surface area of elliptical
A = πab
=π*0.315*6.5=6.43*10^(-3)m2
Overall Heat Transfer Coefficient:
U = 1/(1/h”air” + ΔX/k + A”air”/(η”t”A”gas”h”gas”)) = 1/(1/35.93 + 0.0065/77.5 +6.43*10^(-3)/(0.75 × 0.07922 × 40.57))=32.64W/m”2” K
7) Effectiveness (NTU- Method):
NTU = (UA/C”min”) = ((32.64 × 6.49*10^(-3))/2.20450) = 0.096
ε = 1-exp {NTU”0.22”/C [exp(–CNTU”0.78”)–1]} × 100%
ε = 1–exp {0.096»0.22»/0.9763 [exp (–0.9763 × 0.096”0.78”) – 1]} × 100%
ε = 68.9%
8) Total Heat Transfer Rate:
q = ε × C»min» (T»gas_in»-T”air_in”) =0.68*2.20450(850-560)=434.72W

ε = 1–exp {0.096»0.22»/0.9763 [exp (–0.9763 × 0.096”0.78”) – 1]} × 100%
ε = 69.9%
8) Total Heat Transfer Rate:
q = ε × C»min» (T»gas_in»-T”air_in”) =0.699*2.20450(850-560)=446.874W

Surface area of elliptical
A = πab
=π*0.315*6.5=6.43*10^(-3)m2
Overall Heat Transfer Coefficient:
U = 1/(1/h”air” + ΔX/k + A”air”/(η”t”A”gas”h”gas”)) = 1/(1/35.93 + 0.0065/77.5 +6.43*10^(-3)/(0.75 × 0.07922 × 40.57))=32.64W/m”2” K

7) Effectiveness (NTU- Method):
NTU = (UA/C”min”) = ((32.64 × 6.49*10^(-3))/2.20450) = 0.096

ε = 1-exp {NTU”0.22”/C [exp(–CNTU”0.78”)–1]} × 100%

ε = 1–exp {0.096»0.22»/0.9763 [exp (–0.9763 × 0.096”0.78”) – 1]} × 100%

ε = 68.9%

8) Total Heat Transfer Rate:

q = ε × C»min» (T»gas_in»-T”air_in”) =0.68*2.20450(850-560)=434.72W

NUMERICAL ANALYSIS OF HEAT EXCHANGER

Geometrical Model of rectangular tube heat exchanger;

The geometric model for the rectangular tube Heat Exchanger is as shown in the Fig. 5.2.1

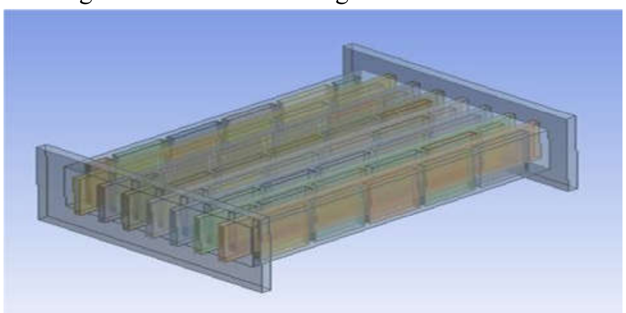


Fig.5.2.1 Fig shows Geometric Model of rectangular tube heat exchanger.

5.2.2. Meshing module of rectangular tube heat exchanger;

The meshing module for the rectangular tube Heat Exchanger is as shown in the Fig. 5.2.2

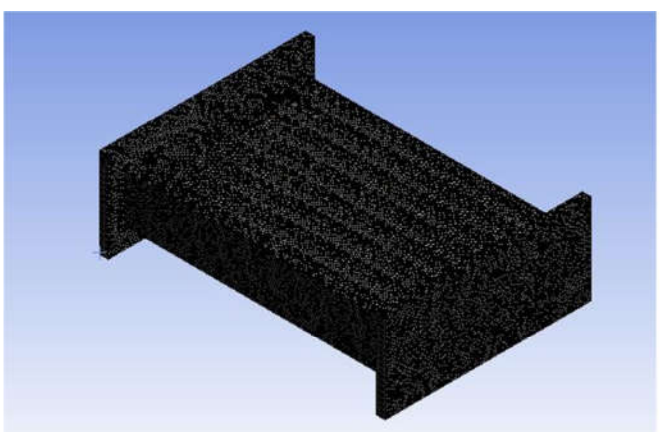
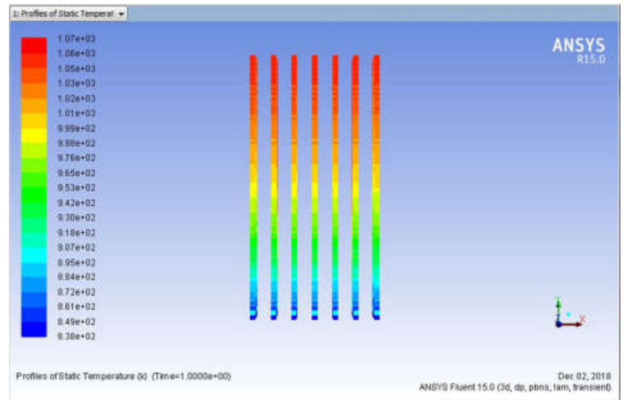
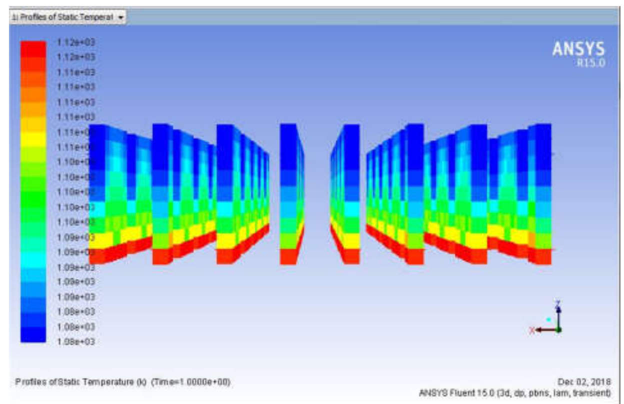


Fig.5.2.2 shows meshing module of rectangular tube heat exchanger.

TEMPERATURE, PRESSURE AND VELOCITY DISTRIBUTIONS OF RECTANGULAR STRUCTURE:



Contours Of temperature for rectangular tube heat exchanger at air side



Contours Of temperature for rectangular tube heat exchanger at exhaust side

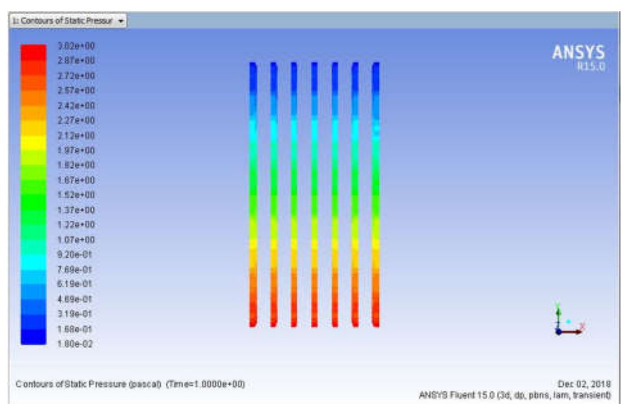


Fig. 5.2.6 Shows Contours Of static pressure for rectangular tube heat exchanger at air side

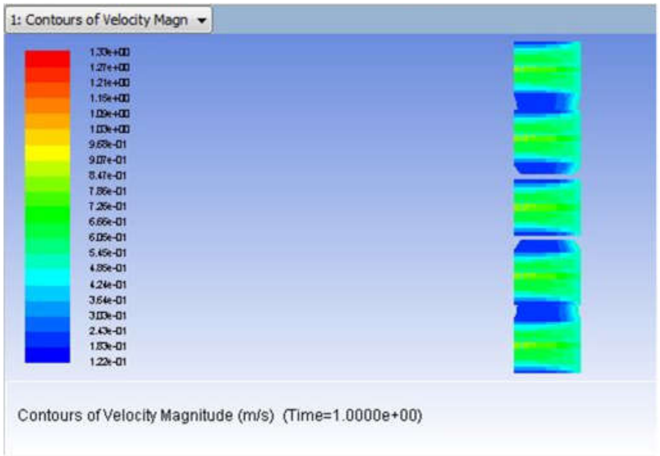
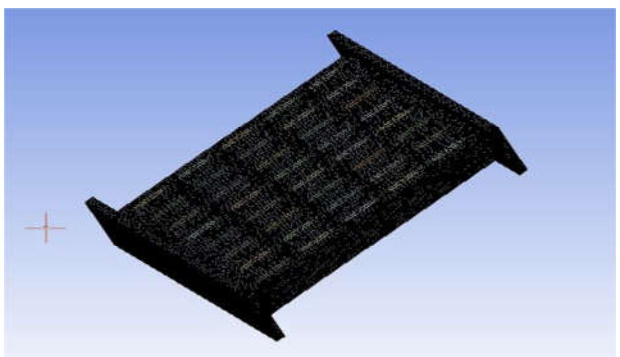


Fig. 5.2.7. Shows Contours Of velocity for rectangular tube heat exchanger at exhaust side

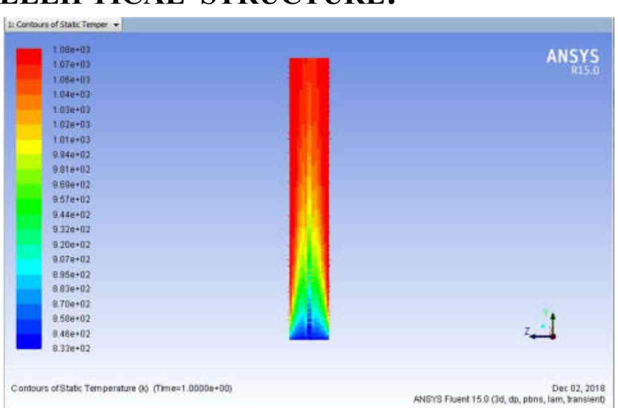
Geometrical Model of elliptical tube heat exchanger;

The geometric model for the elliptical tube Heat Exchanger is as shown in the Fig. 5.3.1

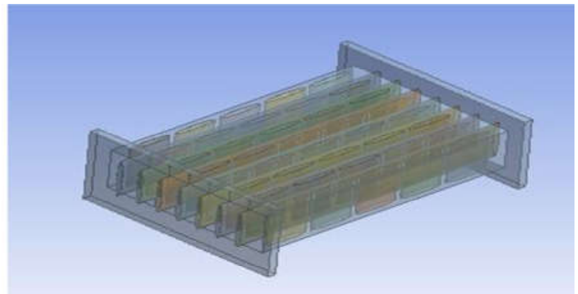
meshing module of elliptical tube heat exchanger.



TEMPERATURE, PRESSURE AND VELOCITY DISTRIBUTIONS OF ELLIPTICAL STRUCTURE:



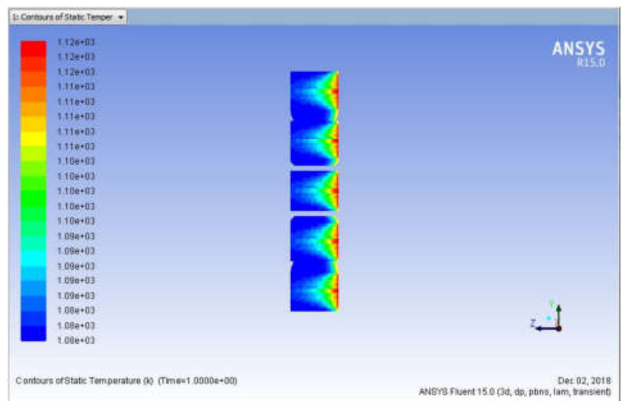
Contours Of temperature for elliptical tube heat exchanger at air side



Geometrical Model of elliptical tube heat exchanger.

Meshing module of elliptical tube heat exchanger;

The meshing module for the elliptical tube Heat Exchanger is as shown in the Fig below

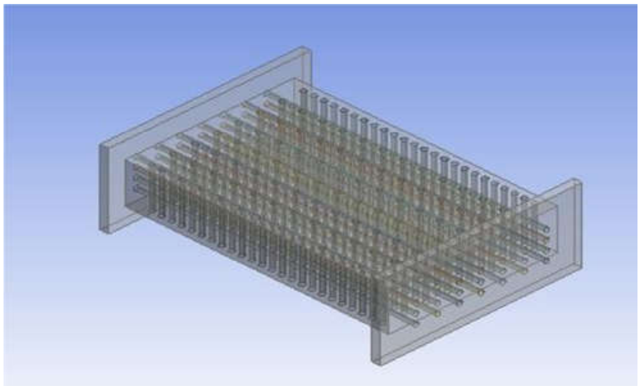


Contours Of temperature for elliptical tube heat exchanger at exhaust side

TEMPERATURE, PRESSURE AND VELOCITY DISTRIBUTIONS OF CYLINDRICAL STRUCTURE:

Geometrical Model of cylindrical tube heat exchanger;

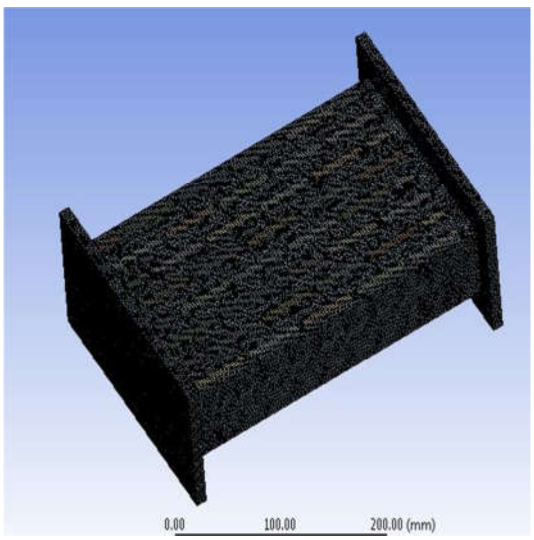
The geometric model for the cylindrical tube Heat Exchanger is as shown in the Fig. 5.4.1



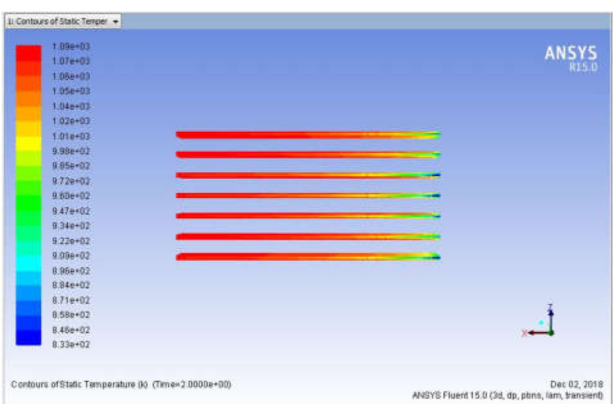
Geometrical Model of cylindrical tube heat exchanger 8mm dia

Meshing module of cylindrical tube heat exchanger;

The meshing module for the cylindrical tube Heat Exchanger is as shown in the Fig below



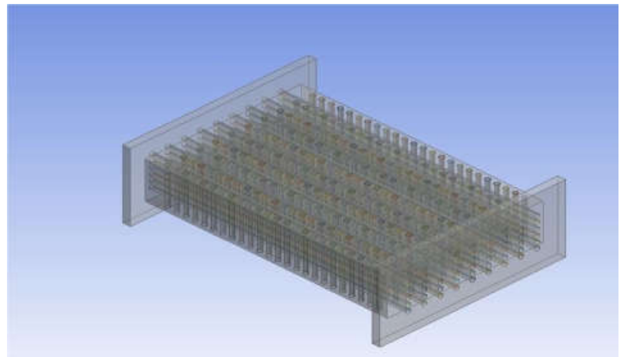
meshing module of cylindrical tube heat exchanger.



Contours Of temperature for cylindrical tube heat exchanger at air side

Geometrical Model of cylindrical tube heat exchanger;

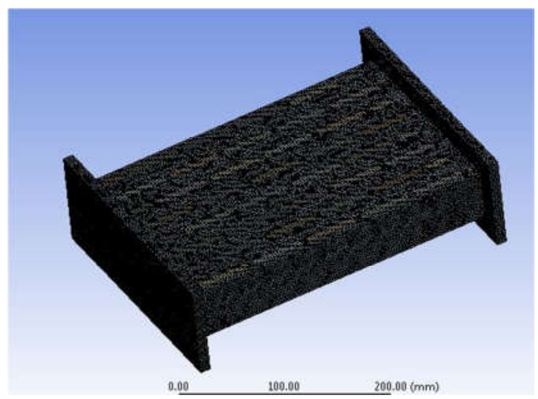
The geometric model for the cylindrical tube Heat Exchanger is as shown in the Fig. 5.4.1



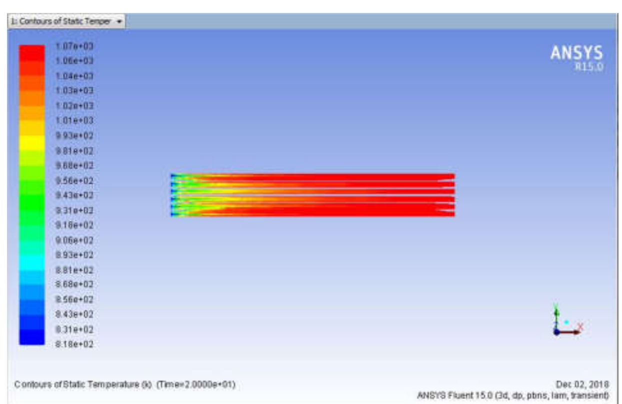
Geometrical Model of cylindrical tube heat exchanger 8mm dia

Meshing module of cylindrical tube heat exchanger;

The meshing module for the cylindrical tube Heat Exchanger is as shown in the Fig below



meshing module of cylindrical tube heat exchanger.

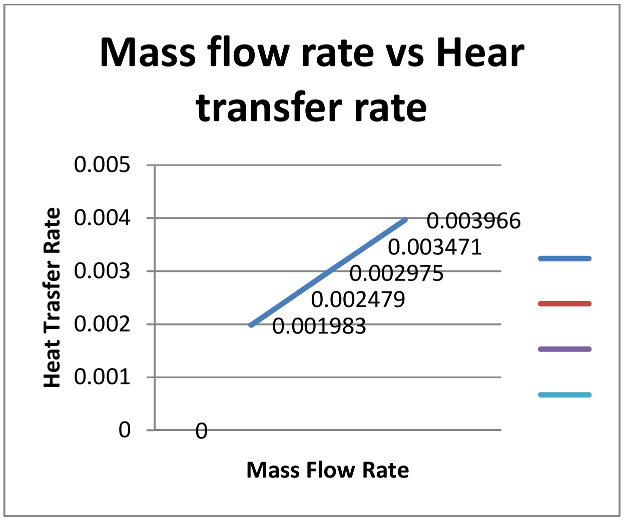


Contours Of temperature for cylindrical tube heat exchanger at air side

Results &Discussion

GRAPHICAL REPRESENTATION

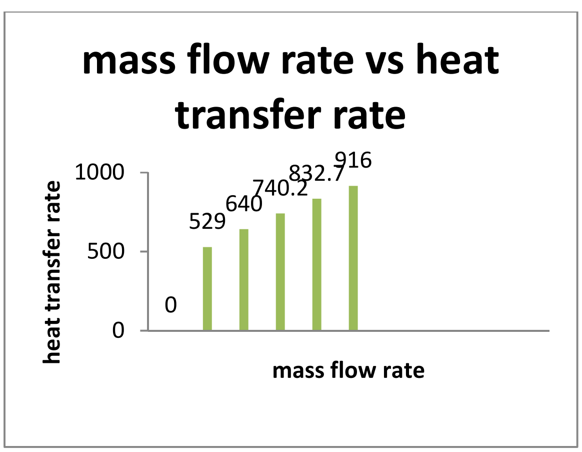
mass flow rate vs heat transfer rate rectangular tube:



Fmass flow rate VS heat transfer rate

The above Fig.6.1 shows the variation of heat transfer rate with the mass flow rate. As the mass flow rate is increases heat transfer rate is also increases With respect to the mass flow rate. Mass flow rate will changes heat transfer rate either rise or fall down and then maximum heat transfer rate will be obtained at 0.003966kg/s.

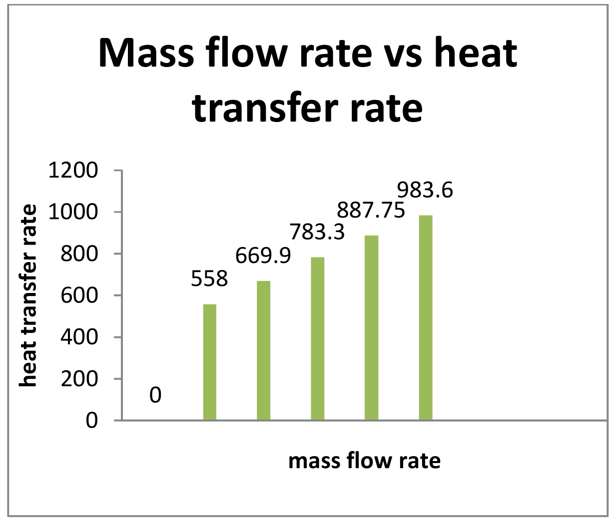
mass flow rate vs heat transfer rate of an elliptical tube:



mass flow rate VS heat transfer rate

The above Fig shows the variation of heat transfer rate with the mass flow rate. As the mass flow rate is increases heat transfer rate is also increases With respect to the mass flow rate. Mass flow rate will changes heat transfer rate either rise or fall down and then maximum heat transfer rate will be obtained at 0.003966kg/s of an elliptical tube.

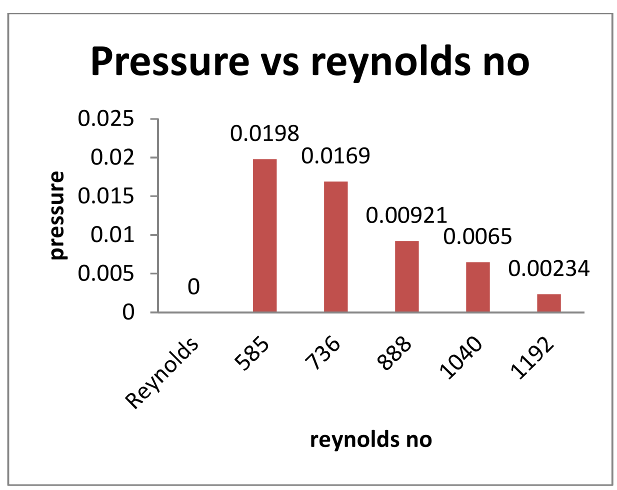
mass flow rate vs heat transfer rate of cylindrical tube:



mass flow rate VS heat transfer rate

The above Fig shows the variation of mass flow rate with heat transfer rate. As the mass flow rate is increases the heat transfer rate is also increases. The above Fig.6.1 shows the variation of heat transfer rate with the mass flow rate. As the mass flow rate is increases heat transfer rate is also increases With respect to the mass flow rate. Mass flow rate will changes heat transfer rate either rise or fall down and then maximum heat transfer rate will be obtained at 0.003966kg/s.

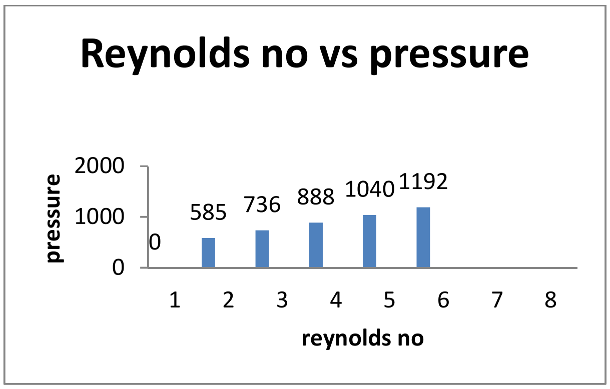
Reynolds no vs pressure of rectangular tube:



reynolds no vs pressure

The above Fig.6.4 shows the variation of Reynolds no with pressure. As the Reynolds no is increases pressure is decreases at outlet. The above Fig shows the variation of pressure with the Reynolds no. As the Reynolds no is increases pressure is also decreases With respect to the Reynolds no. It will changes pressure rate either rise or fall down and then maximum pressure will be obtained at Reynolds no at 585and minimum at 1192.

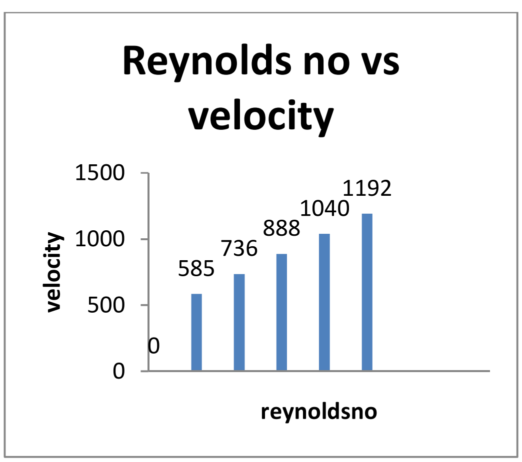
Reynolds no vs pressure of elliptical tube:



reynolds no vs pressure

The above Fig shows the variation of Reynolds no with pressure. As the Reynolds no is increases pressure is also increases With respect to the Reynolds no. It will changes pressure rate either rise or fall down and then maximum pressure will be obtained at Reynolds no at 1192and minimum at 585.

Reynolds no vs velocity :



reynolds no vs velocity

The above Fig.6.6 shows the variation of Reynolds no with velocity. As the Reynolds no is increases velocity is also increases at outlet With respect to the Reynolds no. It will changes velocity either rise or fall down and the maximum velocity will be obtained at Reynolds no at 1192 and minimum at 585.

FIG.6.7 Reynolds no vs Temperature.

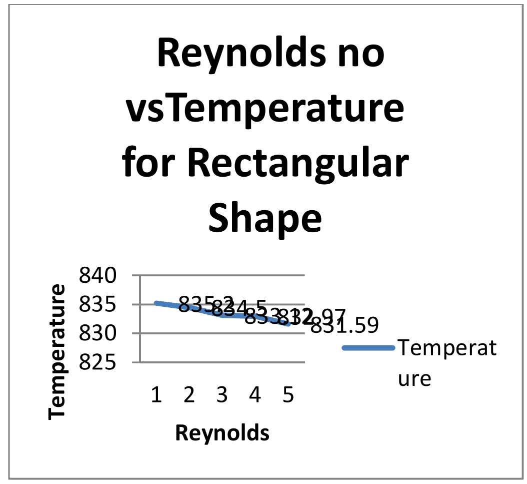


FIG.6.7 Reynolds no vs Temperature

The above Fig.6.7 shows the variation of temperature with the Reynolds no. As the Reynolds no is increases pressure is also decreases With respect to the Reynolds no. It will changes pressure rate either rise or fall down and then maximum pressure will be obtained at Reynolds no at 585and minimum at 1192.

FIG.6.8 Reynolds no vs velocity of an elliptical shape:

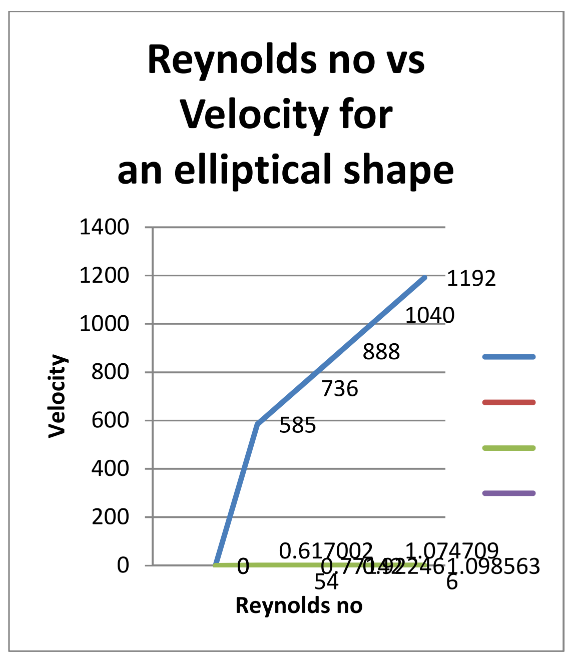


FIG.6.8 Reynolds no vs velocity

The above Fig.6.8 shows the variation of velocity with the Reynolds no. As the Reynolds no is increases pressure is also increases With respect to the Reynolds no. It will changes velocity rate either rise or fall down and then maximum velocity will be obtained at Reynolds no at 1192 and minimum at 585..

FIG.6.9 Reynolds no vs temperature of an elliptical shape:

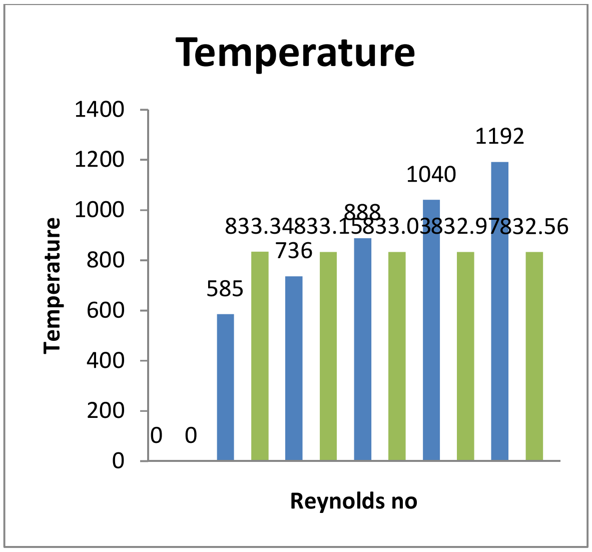


FIG.6.9 Reynolds no vs temperature

The above Fig.6.9 shows the variation of temperature with the Reynolds no. As the Reynolds no is temperature is also decreases With respect to the Reynolds no. It will changes temperature either rise or fall down and the maximum temperature will be obtained at Reynolds no at 585and minimum at 1192.

FIG.6.10 Reynolds no vs pressure of an cylindrical shape:

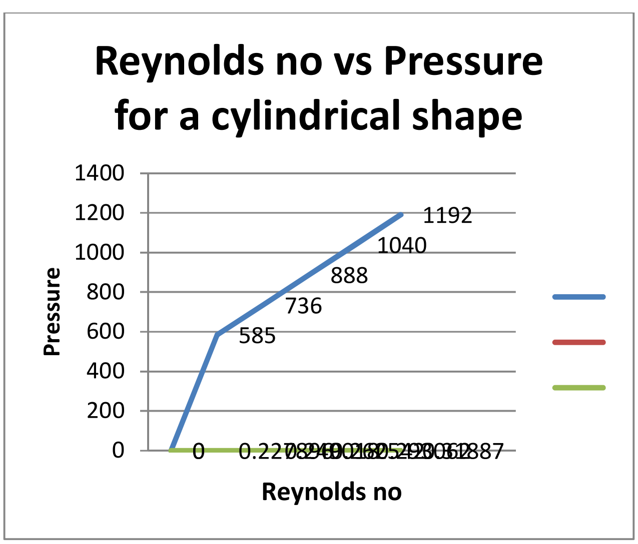


FIG.6.10 Reynolds no vs pressure

The above Fig.6.10 shows the variation of pressure with respect to Reynolds no. As the Reynolds no is increases pressure is also increases With respect to the Reynolds no. It will changes pressure rate either rise or fall down and the maximum pressure will be obtained at Reynolds no at 1192 and minimum at 585.

FIG.6.11 Reynolds no vs velocity of an cylindrical shape:

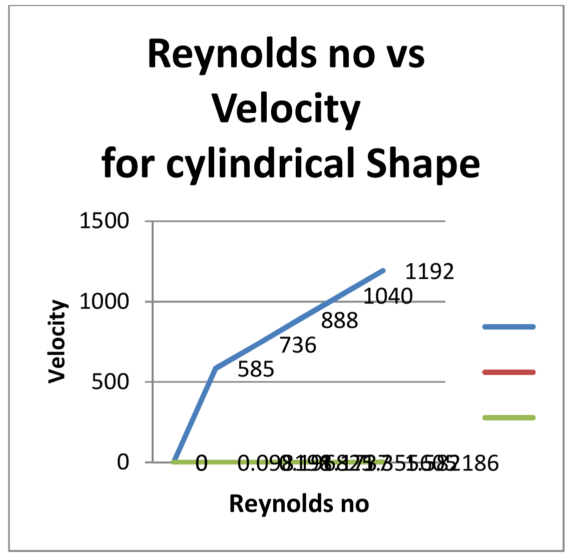


FIG.6.11 Reynolds no vs velocity:

The above Fig.6.11 shows the variation of velocity with the Reynolds no. As the Reynolds no is increases velocity is also increases With respect to the Reynolds no. It will changes velocity rate either rise or fall down and the maximum velocity will be obtained at Reynolds no at 1192 and minimum at 585..

FIG.6.13 Effectiveness between rectangular shape and elliptical shape

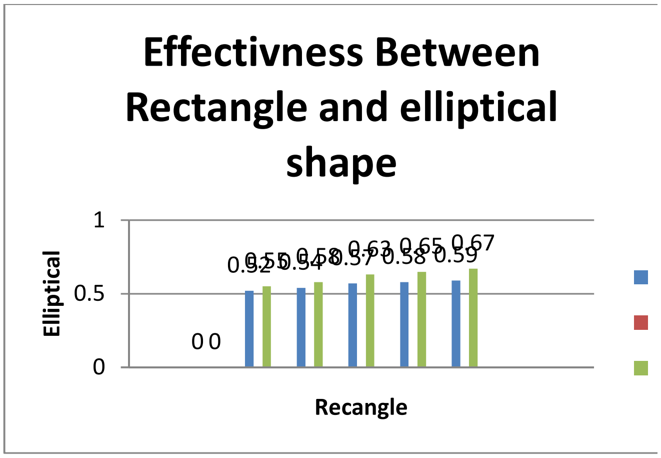


FIG.6.13 Rectangular tube vs elliptical tube:

The above Fig.6.13 shows the variation of effectiveness between rectangular and elliptical tube with respect to Reynolds no. Elliptical tubes possess more amount of heat transfer rate compared to rectangular tube. It possesses maximum efficiency at Reynolds no 1192 and minimum at 585.

FIG.6.14 Effectiveness between rectangular shape and cylindrical shape

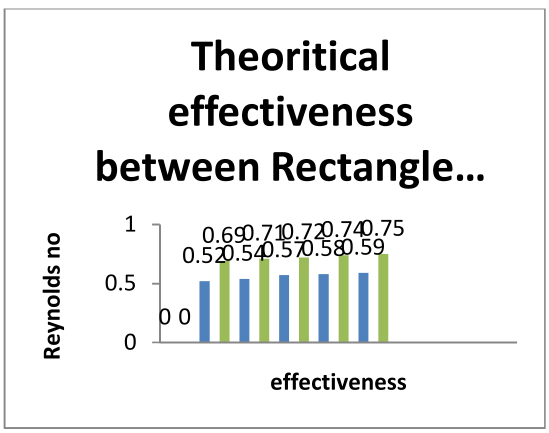
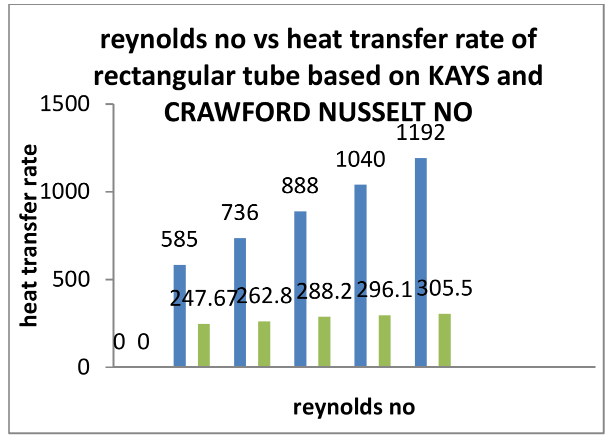


FIG.6.14 Rectangular tube vs cylindrical tube:

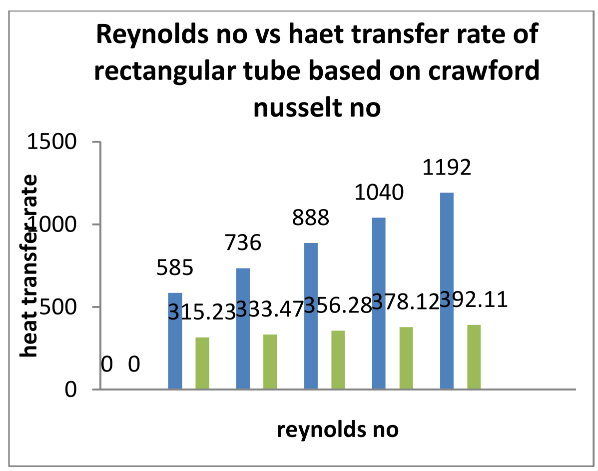
The above Fig.6.14 shows the variation of effectiveness between rectangular and cylindrical tube with respect to Reynolds no. Cylindrical tubes possess more amount of heat transfer rate compared to rectangular tube. It possesses maximum efficiency at Reynolds no 1192 and minimum at 585.

FIG.6.15 heat transfer rate of rectangular tube based on Kays and Crawford Nusselt no:



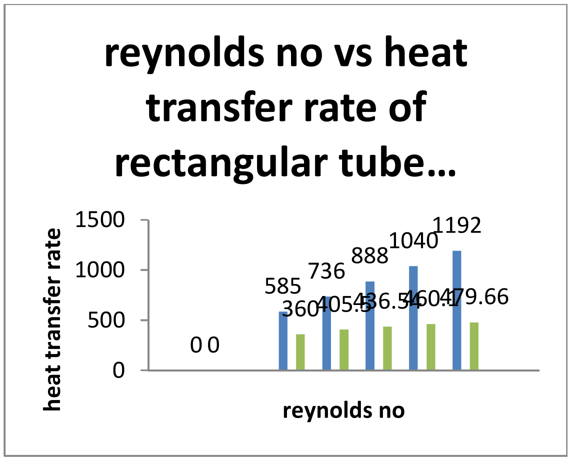
The above Fig.6.15 shows the variation of heat transfer rate between rectangular tubes based on Kay's and Crawford relation with respect to Reynolds no. Rectangular tubes possess more amount of heat transfer rate at Reynolds no 1192 and minimum at 585. Heat transfer rate is increases with increase in Reynolds no.

FIG.6.16 heat transfer rate of rectangular tube based on Crawford Nusselt no:



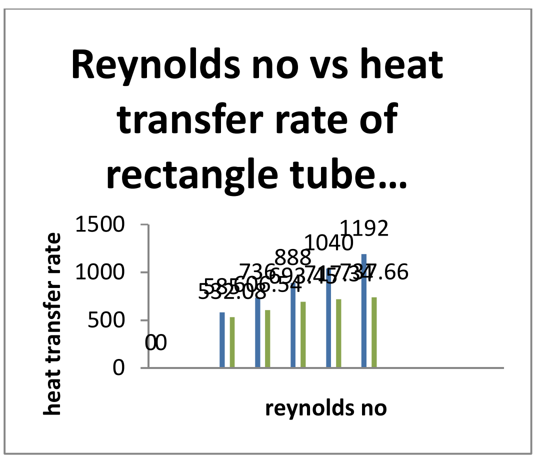
The above Fig.6.16 shows the variation of heat transfer rate between rectangular tubes based on Crawford Nusselt no relation with respect to Reynolds no. Rectangular tubes possess more amount of heat transfer rate at Reynolds no 1192 and minimum at 585. Heat transfer rate is increases with increase in Reynolds no.

FIG.6.17 heat transfer rate of rectangular tube based on Sieder- tate Nusselt no:



The above Fig.6.17 shows the variation of heat transfer rate between rectangular tubes based on sieder tate Nusselt no relation with respect to Reynolds no. Rectangular tubes possess more amount of heat transfer rate at Reynolds no 1192 and minimum at 585. Heat transfer rate is increases with increase in Reynolds no

FIG.6.18 Heat transfer rate of rectangular tube based on Shah and London relation:



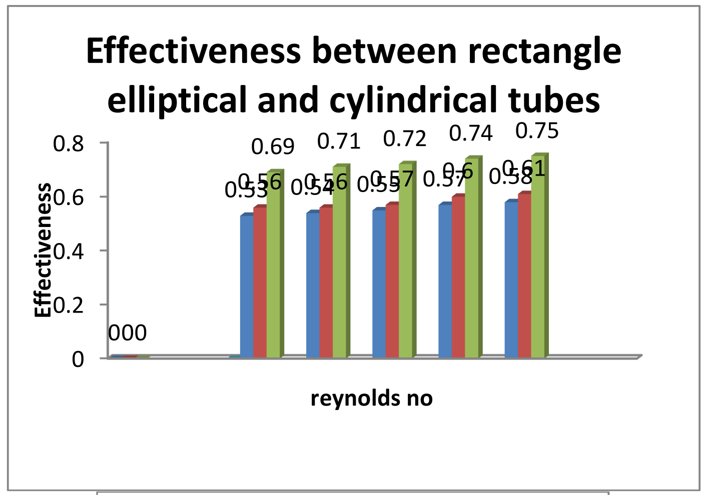
The above Fig.6.18 shows the variation of heat transfer rate between rectangular tubes based on shah and London relation with respect to Reynolds no. Rectangular tubes possess more amount of heat transfer rate at Reynolds no 1192 and minimum at 585. Heat transfer rate is increases with increase in Reynolds no

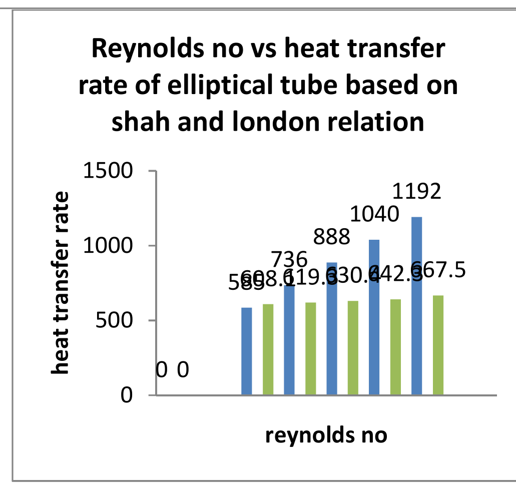
FIG.6.19 Effectiveness between rectangle, elliptical and cylindrical tubes:

The above Fig.6.19 shows the variation of effectiveness between rectangular, elliptical and cylindrical tubes with respect to Reynolds no. Rectangular tubes possess less amount of heat transfer rate when compared to elliptical and elliptical possess less compared to cylindrical at Reynolds no 585 and maximum at Reynolds no 1192. Heat transfer rate is

and posses more efficiency with respect to increase in Reynolds no.

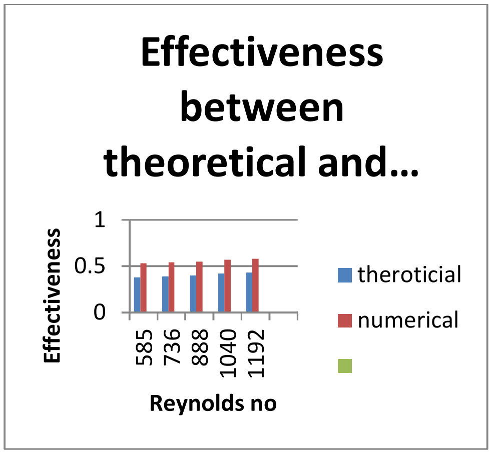
FIG.6.20 Heat transfer rate of elliptical tube based on Shah and London relation:





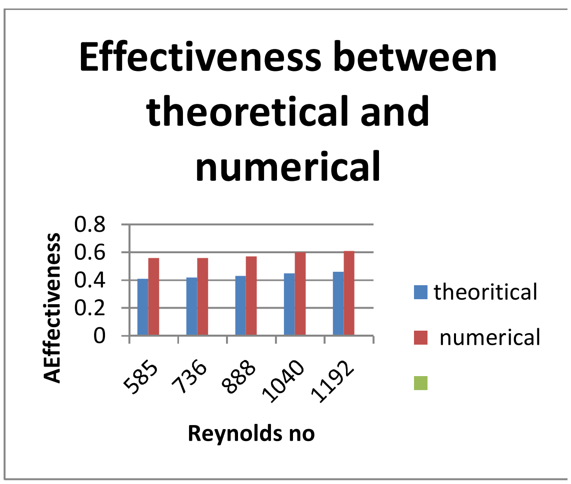
The above Fig.6.20 shows the variation of heat transfer rate between elliptical tubes based on shah and London relation with respect to Reynolds no. Elliptical tubes possess more amount of heat transfer rate at Reynolds no 1192 and minimum at 585. Heat transfer rate is increases with increase in Reynolds no.

FIG.6.21 Effectiveness between theoretical and numerical analysis of rectangular tube:



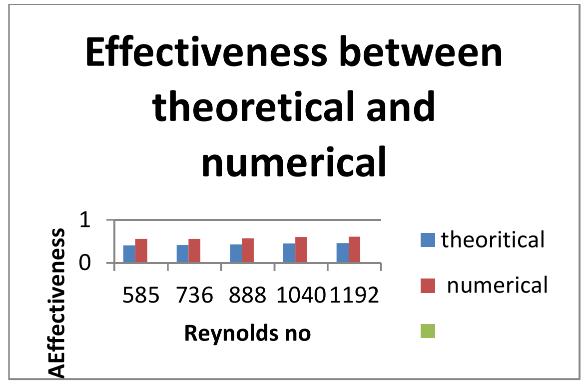
The above Fig.6.21 shows the variation of effectiveness between theoretical and numerical analysis with respect to Reynolds no. Rectangular tubes possess less amount of heat transfer rate in theoretical when compared to numerical analysis. Heat transfer rate is increases with increase in Reynolds and possess 15% more efficiency then theoretical.

FIG.6.22 Effectiveness between theoretical and numerical analysis of elliptical tube:



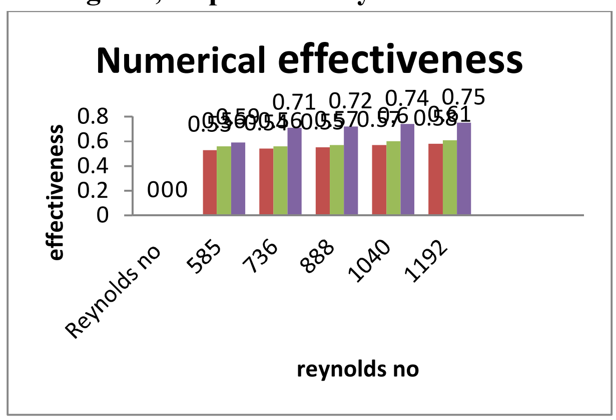
The above Fig.6.22 shows the variation of effectiveness between theoretical and numerical analysis with respect to Reynolds no. Elliptical tubes possess less amount of heat transfer rate in theoretical when compared to numerical analysis. Heat transfer rate is increases with increase in Reynolds and possess 15% more efficiency then theoretical.

FIG.6.23 Effectiveness between theoretical and numerical analysis of cylindrical tube:



The above Fig.6.23 shows the variation of effectiveness between theoretical and numerical analysis with respect to Reynolds no. Cylindrical tubes possess less amount of heat transfer rate in theoretical when compared to numerical analysis. Heat transfer rate is increases with increase in Reynolds and possess 15% more efficiency then theoretical

FIG.6.24 Numerical effectiveness between rectangular, elliptical and cylindrical tubes:



The above Fig.6.24 shows the variation of numerical effectiveness between rectangle, elliptical and cylindrical tubes with respect to Reynolds no. Cylindrical tubes possess more amount of effectiveness compared to rectangle and elliptical tubes. Rectangular tubes possess 52%, Elliptical tube possesses 55% and cylindrical tubes possess 62% of effectiveness.

6.6 RESULT TABULAR

Comparison between Reynolds Number & Correlations

Reynolds No		585	736	888	1040	1192
Rectangle	Pressure	0.0198	0.0169	0.00921	0.0065	0.00234
	Velocity	6.231	4.934	2.421	1.0747	0.0543
	Temperature	835.2	834.5	833.12	832.97	831.59
Elliptical	Pressure	0.018064	0.016901	0.013238	0.006509	0.00356
	Velocity	0.617003	0.77143	0.92246	1.07471	1.098563
	Temperature	833.34	833.15	833.03	832.97	832.56
Cylinder 8mm	Pressure	0.227896	0.249018	0.262542	0.293062	0.31887
	Velocity	0.098198	0.196825	1.1737	1.355605	1.582187
	temperature	832.34	831.01	830.12	829.46	829.16
Cylinder 8mm	Pressure	0.24752	0.25962	0.27548	0.29652	0.326923
	Velocity	0.15319	0.203841	1.26716	1.38465	0.60493
	temperature	817.82	817.43	817.05	816.94	816

Effectiveness between rectangle, elliptical and cylindrical shapes from numerical analysis

Comparison between theoretical and numerical effectiveness						
Reynolds no		585	736	888	1040	1192
Effectiveness(Theoretical)	Elliptical	0.41	0.42	0.43	0.45	0.46
	Cylindrical 8mm	0.54	0.56	0.57	0.59	0.6
	Cylindrical 6mm	0.557	0.564	0.58	0.6	0.62
	Rectangle	0.53	0.54	0.55	0.57	0.58
Effectiveness(Numerical)	Elliptical	0.56	0.56	0.57	0.6	0.61
	Cylindrical 8mm	0.69	0.71	0.72	0.74	0.75
	Cylindrical 6mm	0.7	0.716	0.729	0.746	0.76

Based on Shah &London correlations						
Reynolds no		585	736	888	1040	1192
	U	20.57	20.96	21.73	21.87	22.71
Rectangle	Є	0.37	0.38	0.39	0.41	0.42
	Q(W)	532.08	606.54	693.45	717.34	737.66

Elliptical	U	27.2	27.2	27.2	27.2	27.2
	€	0.4	0.41	0.42	0.44	0.45
	Q(W)	608.1	619.3	630.4	642.3	667.5
Cylindrical	U	29.62	29.62	29.62	29.62	29.62
	€	0.52	0.54	0.55	0.57	0.59
	Q(W)	745.6	756.8	772.1	801.4	843.1

CONCLUSION

In current research the ceramic monolith heat exchanger performance was compared between theoretical and numerical analysis. The theoretical calculations have been executed for the exhaust gas as well as cold air for a measuring domain of 600-1000c. The total heat transfer rate and effectiveness were estimated for theoretical and numerical analysis. The calculations have been executed utilizing NTU method considering numerous Nusselt number correlations taken from the literature.

(1) The performance of the heat transfer by numerical computation was 15% more than theoretical analysis. Among the Nusselt number correlations, the total heat transfer by NTU method with Stephen correlation is more closer to numerical method compared to remaining correlations.

(2) The estimated Effectiveness for rectangular tube heat exchanger, elliptical tube heat exchanger and cylindrical tube heat exchanger of 8mm dia and 6mm diameter are 52%, 55% and 62% and 65% respectively.

FUTURE SCOPE:

In this project the entire work is carried out using ceramic materials of Ni-Cr-Al, NiCrAl + $MgZrO_3$ and $MgZrO_3$. In the same manner it can be carried out by make use of advanced ceramic materials those are high temperature resistant in nature.

Apart from the regular cross sections like Rectangular, Elliptical and Cylindrical, other cross sections can be used by maintaining proper L/D ratio and without varying any mass distribution.

Also instead of simply releasing smoke stack exhaust, it can be used to heat air in the pre heater enabling the pre heater to operate at a lower, energy saving temperature, provided the parameters for a particular application have been established.

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