Gujarat Technological University

"CFD analysis of packed bed solar air heater"

A Detailed Report to be submitted for
Project – II (2181909) Semester VIII
In
Bachelor of Engineering (Mechanical)

Guided by
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CERTIFICATE

This is to certify that Project – II entitled "CFD Analysis of packed bed solar air heater" which is being submitted by Dhruv Girish Apte and Mohit Dhoriya for Project II (2181909) Semester VIII in Bachelor of Engineering (Mechanical) to Gujarat Technological University is a record of the candidates" own work carried out by them under my supervision and Guidance , during the period of December 2018- April 2019.

Prof. SukritindraSoni	Dr.Darshak Desai	Dr. HemantThakkar
(Faculty Guide)	(Head of Department)	(Project Coordinator)

Patent Search and Analysis Report (PSAR) Reports

Submitted as a part of the

PROJECT REPORT

"CFD Analysis of packed bed solar air heater"

Submitted by

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In partial fulfilment for the award of the degree
Of

BACHELOR OF ENGINEERING
In

MECHANICAL ENGINEERING
G.H.PATEL COLLEGE OF ENGINEERING & TECHNOLOGY

Gujarat Technological University
Ahmedabad
April, 2019

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DECLARATION

We hereby declare that the PSAR Reports submitted along with the Project Report for the project entitled "CFD Analysis of Packed Bed solar Air Heater" submitted in partial fulfilment for the degree of Bachelor of Engineering in **MECHANICAL ENGINEERING** to Gujarat Technological University, Ahmedabad, is a bonafide record of the project work carried out at **G.H.PATEL COLLEGE OF ENGINEERING & TECHNOLOGY** under the supervision of **Prof. Sukritindra Soni** and that no part of any of these PSAR reports has been directly copied from any students' reports or taken from any other source, without providing due reference.

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CERTIFICATE

This is to certify that the PSAR reports, submitted along with the project entitled **"CFD Analysis of Packed bed solar air heater"** has been carried out by

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Under my guidance in partial fulfilment for the degree of: **Bachelor of Engineering** in **MECHANICAL ENGINEERING 8th Semester** of Gujarat Technological University, Ahmedabad during the academic year 2018 - 19. These students have successfully completed PSAR activity under my guidance.

Prof. Sukritindra Soni	Dr.Darshak Desai	Dr. Hemant Thakkar
(Faculty Guide)	(Head of Department)	(Project Coordinator)

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Annexure 2

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We hereby certify that we are the sole authors of this UDP project report and that neither any part of this UDP project report nor the whole of the UDP Project report has been submitted for a degree by other student(s) to any other University or Institution.

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ABSTRACT

Solar Air Heater is a simple device that utilizes solar energy in drying mainly agricultural products. It is also used as a low temperature energy-source [1]. They usually consist of an absorber plate with a parallel set of plates for air flow. A glass cover is fixed above the absorber plate. To store the thermal energy gained from the solar heat, packed beds are used.

CFD analysis was carried out to investigate the performance of a solar air heater with packed bed. The porosity of the porous medium was kept the same and the distance between the two porous mediums was varied to observe its effect on the performance of solar air heater. To validate the results the computational model was compared with the experimental investigations done on a simple solar air heater without packed bed.

Keywords: Packed bed, solar air heater, porosity

4. Introduction

Solar energy, radiant light and heat from the sun, has been harnessed by humans since ancient times using a range of ever-evolving technologies. Before 1970, some research and development was carried out in a few countries to exploit solar energy more efficiently, but most of this work remained mainly theoretical and academic. After the dramatic rise in oil prices in the 1970s, several countries began to formulate extensive research and development programs to exploit solar energy.

Packed beds are volumes of porous media obtained by packing particles in a duct. Due to their high heat transfer rates, they make solar air heaters more efficient than conventional heaters. The packed bed can have several materials, normally semitransparent to increase the thermo-hydraulic efficiency. Various approaches have been made regarding packed bed structure, each yielding different results which, will be discussed in the next section.

5. Literature Review

Artificial roughness is a well-known method to increase heat transfer from a surface to roughen the surface either randomly with a sand grain or by use of regular geometric roughness elements on the surface. However, the increase in heat transfer is accompanied by an increase in the resistance to fluid flow. Several investigators have attempted to design an artificially roughened rectangular duct which can enhance the heat transfer with minimum pumping losses. Many investigators have studied this problem in an attempt to develop accurate predictions of the behavior of a given roughness geometry and to define a geometry which gives the best transfer performance for a given flow friction. A lot of studies have been reported in the literature on artificially roughened surfaces for heat transfer enhancement but most of the studies were carried out with two opposite or all the four walls roughened. An early study of the effect of roughness on friction factor and velocity distribution was performed by Nikuradse[2], who conducted a series of experiments with pipes roughened by sand grains and since then many experimental investigations were carried out on the application of artificial roughness in the areas of gas turbine airfoil cooling system, gas cooled nuclear reactors, cooling of electronic equipment, shipping machineries, combustion chamber liners and re-entry vehicles etc.

The roughness elements have to be considered only on one wall, which is the only heated wall comprising the absorber plate. These applications make the fluid flow and heat-transfer characteristics distinctly different from those found in case of two roughened walls and four heated wall duct. In the case of solar air heater, only one wall of the rectangular air passage is subjected to uniform heat flux while the remaining three walls are insulated. It is well known that the heat transfer coefficient between the absorber plate and air of solar air heater is generally poor and this result in lower efficiency. The effectiveness of solar air heater can be improved by using artificial roughness in the form of different types of repeated ribs on the absorber plate. It has been found that the artificial roughness applied to the absorber plate of a solar air heater, penetrates the viscous sublayer to promote turbulence that, in turn, increases the heat transfer from the surface as compared to smooth solar air heater. This increase in heat transfer is accompanied by a rise in frictional loss and hence greater pumping power requirements for air through the duct. In order to keep the friction losses at a low level, the turbulence must be created only in the region very close to the duct surface, i.e., in the laminar sublayer

Initial approaches include slit and expended aluminum foil matrix [3][4] followed by hollow spheres and crushed glass matrix [5][6]. Hasatani et el. [7] filled the packed bed with semi-transparent materials like glass beads, porcelain beads and glass tubes. They concluded that both the theoretical and experimental efficiencies increase approximately 15-20 % in comparison to a smooth collector. Prasad and Saini [8] used single pass air heaters packed with wire mesh matrix in unidirectional and cross flow arrangement and found out that the latter arrangement is less efficient than the former. Their study also concluded that thermal performance is a function of porosity, extinction coefficient, heat transfer area and density other than the orientation of packing. Ahmad et. al[9] used cross-flow arrangement to perform experimental investigations with iron-screen matrices while varying the parameters.
Their study concluded that the thermohydraulic efficiency of the packed bed air heater decreases with increase in bed depth to element size value ratio and porosity.

Later, Sopian K et. al[10] used steel wool to conduct experiments as a means of porous media in the second channel on the double pass solar air heater. The collector had only one glass cover and a blackened metal absorber.

Sopian et al studied various features like effects of mass flow rate, solar radiation and temperature rise on the collector's thermal performance. Typical thermal efficiency of the collector with porous media was recorded as 60-70%, about 30% higher than collector without porous media. Ramadan et. al[11] carried further investigations both experimentally and theoretically. In the analytical solution, the thermal conductivity of the porous media was considered negligible. Since the researchers used low conductivity materials like gravel and limestone, the assumption would have been posited true. The results stated thermal performance with the packed bed material gravel was higher than other materials.

In the numerical investigations, several configurations were studied to analyze the heater performance. It was inferred that increasing the outlet temperature after sunset,packed bed materials with higher masses and low porosities must be used. It was reported that thermal efficiency with gravel is 22-27% more than without packed bed. These modifications also result in an increase in pressure drop. Another interesting outcome was that the thermal performance with upper channel packed bed was higher as compared to lower channel packed bed thermal performance. This strengthened the claims of Sopian et. al[10].

Dhiman et. al[12] investigated a double glass double pass solar air heater packed with iron scraps above the absorber plate under parallel air flow mode theoretically and experimentally. The proposed model's thermal performance was found to be 10-20% increased as compared to conventional flat plate solar air heater.

A similar analysis by the same research group[13] with wire mesh instead of iron scraps in upper duct saw a thermal efficiency of 50-88% in parallel flow and 50-90% in counter-flow. At differential mass rates, the thermohydraulic efficiency was 10% higher for the parallel flow system than the counter-flow system.

One of the most influential studies that affect this project also were conducted by Bhagoria et. al[14][15]. The study used mild steel chips as a packed bed material and compared the thermal performance with marble chips packed bed solar air heater. The first paper referred to the experimental analysis of an artificially roughened solar air heater encompassing the Reynolds Number 3000 to 18000; the relative roughness height of 0.015 to 0.033 and the relative roughness pitch of $60.17\phi^{-1.0264}<p/e<12.12$ where ϕ is wedge angle and p/e is relative roughness pitch. Among the numerous conclusions concluded by the study, the most crucial one was the relation between friction factor and relative roughness pitch with the optimum Nusselt Number at a wedge angle of 10°; the friction factor kept decreasing as the relative roughness pitch increased. In the other paper[15], a numerical investigation was carried for Reynolds Number range 3800 to 18000 with twelve different equilateral triangular-sectioned rib used as roughness element.

The study included a two-dimensional CFD model to predict the heat transfer and flow friction characteristics. The major conclusions of the study were:

1) The average Nusselt number tends to increase as the Reynolds number increases in all cases. The average Nusselt number tends to decrease as the relative roughness pitch increases for a fixed value of relative roughness height and it also tends to increase as the relative roughness height increases for a fixed value of relative roughness pitch.
2) The maximum enhancement in the Nusselt number has been found to be 3.073 times over the smooth duct corresponds to relative roughness height (e/D) of 0.042 and relative roughness pitch (P/e) of 7.14 at Reynolds number (Re) of 15,000 in the range of parameters investigated.

The average friction factor tends to decrease as the Reynolds number increases in all cases. The average friction factor tends to decrease as the relative roughness pitch increases for a fixed value of relative roughness height and it tends to increase as relative roughness height increases for a given value of relative roughness pitch

6. Introduction to CFD

Computational Fluid Dynamics is an amalgamation of fluid mechanics, mathematics (mostly linear algebra) and computer programming to study fluid flow problems and interactions between fluids on the surfaces defined by boundary conditions. The research field of CFD has been studied in distinct fields like aerodynamics, combustion analysis, weather simulation, biological engineering and heat testing of vessels.

CFD is essentially around the Navier-Stokes equations, which describe the behaviour of viscous fluids.

$$\rho \frac{D\vec{V}}{Dt} = -\nabla p + \rho \vec{g} + \mu \nabla^2 \vec{V}$$

The energy equation is another important equation in this study and hence is a topic of discussion:

$$\frac{D_V T}{Dt} = \frac{\partial T}{\partial t} + \vec{V}.\nabla T$$

In case of turbulent flows, the aim is to build a model that can study the flow within the desired accuracy level and not compromising the computational possibility. Several turbulence models have thus been studied with varying powers of accuracy and computational cost. The computational cost corresponds to the range of scales modelled vs. resolved. The more the scales are modelled, the lower is the computational cost but this also results in a decrease in accuracy. Thus, a CFD engineer has to find a suitable balance between the accuracy limit and computational cost.

7.Computational Domain

To account for the credibility of the analysis, a few CFD analysis were done to compare the results with the experimental analysis done by J.L Baghoria[14].

A 2-dimensional analysis was done of a smooth duct without any artificial geometry to compare the results. The domain is shown in figure 2.

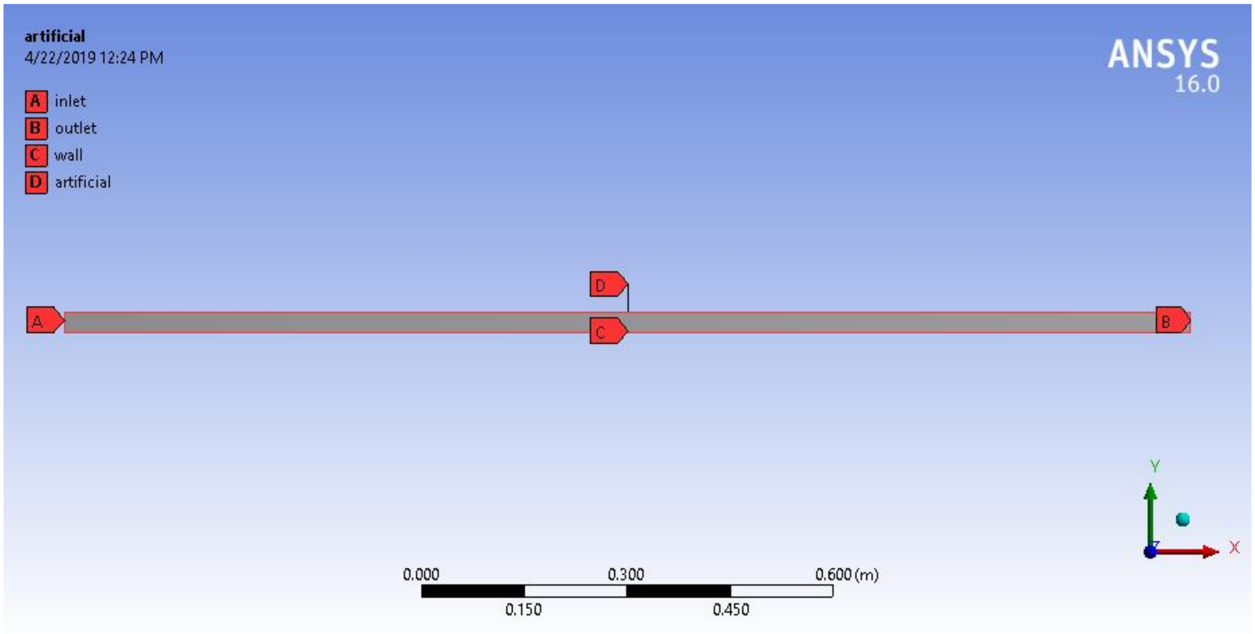

Figure 1. Computational domain for smooth duct.

K-epsilon turbulence model with RNG was used to take into account the turbulence.The Re-Normalization Group (RNG) k-ε model is a turbulence model developed [16] to renormalize the Navier-Stokes equations to account for the effects of smaller scales of motion. The RNG approach results in a form that is accountable for different motion scales through changes in production term.

There are a number of ways to write the transport equations for k and ϵ, a simple interpretation where buoyancy is neglected is

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+P_k-\rho\epsilon$$

$$\frac{\partial}{\partial t}(\rho \epsilon)+\frac{\partial}{\partial x_i}(\rho \epsilon u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\epsilon}\right)\frac{\partial \epsilon}{\partial x_j}\right]+C_{1\epsilon}\frac{\epsilon}{k}P_k-C_{2\epsilon}^{*}\rho\frac{\epsilon^2}{k}$$

where $C_{2\epsilon}^{*}=C_{2\epsilon}+\frac{C_\mu \eta^3(1-\eta/\eta_0)}{1+\beta\eta^3}$

and $\eta=Sk/\epsilon$ and $S=(2S_{ij}S_{ij})^{1/2}$

With the turbulent viscosity being calculated in the same manner as with the standard k-epsilon model.

The length of the domain is 1.65m and the height is 0.03m. A constant heat flux 1500 W/m^2 was given on the boundary named artificial. The reynolds number was varied from 3000 to 18000 and the nusselt no. at the outlet was calculated.

Nusselt number = $\frac{Heat\ flux}{T_w-T_m}$

Where T_w = maximum temperature along the outlet

T_m= mean temperature at the outlet , given by

$$T_m=\frac{\int_{r=0}^{R}(Velocity\ U\ \cdot Temperature\cdot r)dr}{\int(Velocity\ U.r)dr}$$

Reynolds number	computational results of nusselt number	experimental results of nusselt number
3000	12.84	12
4000	16.771	16
6000	21.44	20
8000	30.11	28
14000	47.47	44
18000	56.1742	52

Table 1. Observation table of experimental and computational results

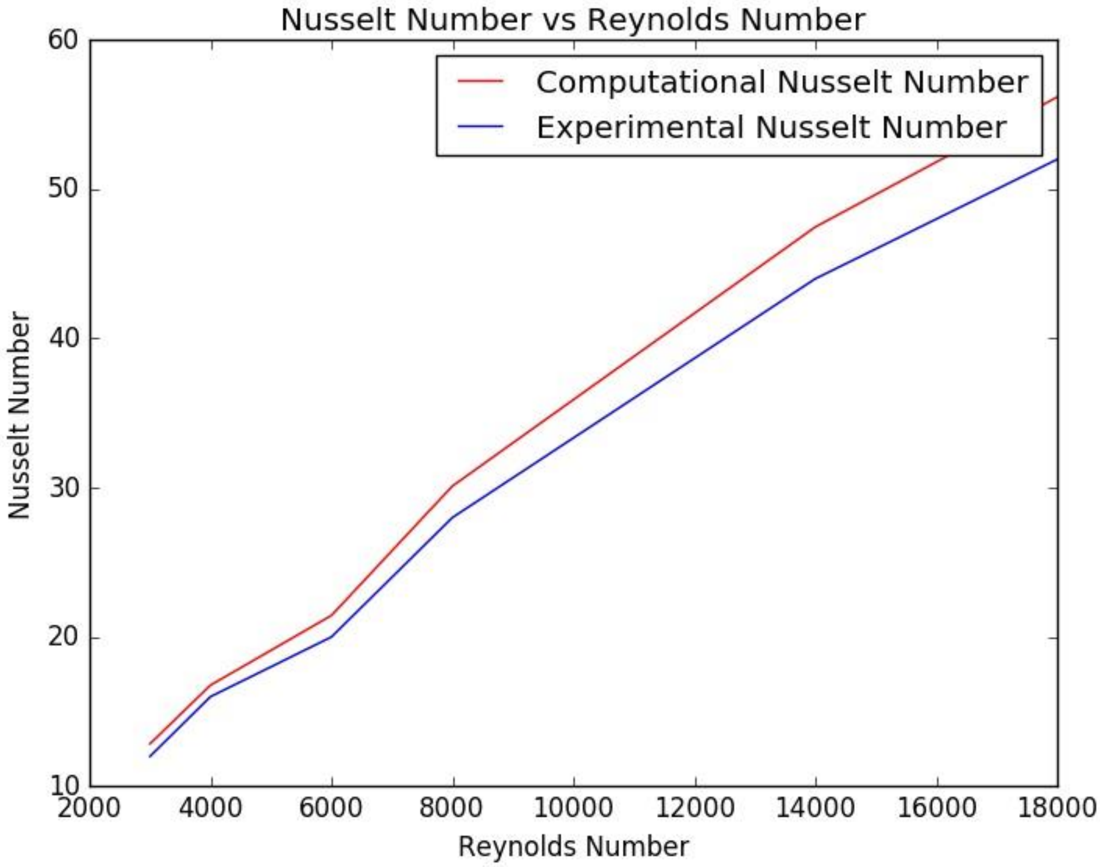

Figure 2. Nusselt number VS Reynolds number

The results obtained were close to the experimental results. The computational model was 96.53% accurate.

The next step would be to introduce porous medium.

8. Introduction of porous medium

Two porous medium of porosity 0.75 were introduced in the domain. The length of each medium was 100mm and it was kept constant. The entry length and exit length were each 420mm. The height of both mediums was varied from 25mm to 12mm. The Reynolds number of the flow was 14000 and was kept constant. After finding the optimum height of the porous material , a V-Shape porous medium was introduced and its angle was varied from $20° \; to \; 40°$ to observe its effect on temperature and pressure.

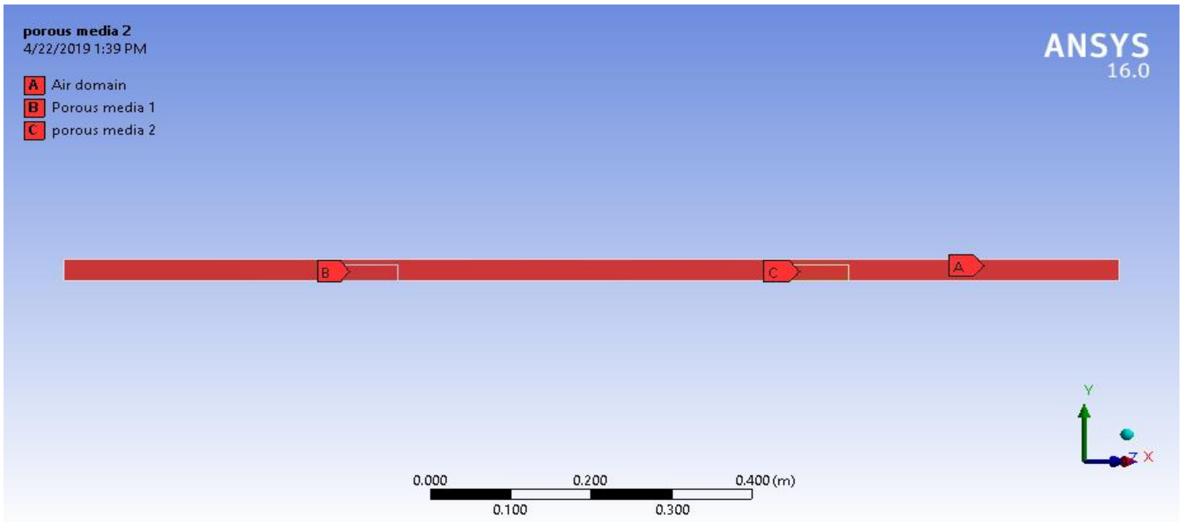

Figure 3. Computational domain with porous mediums (height 25mm)

A zoomed in picture of mesh for height of porous medium (15mm) is shown in figure 3.Inflation was given at the boundary to get accurate results for the boundary layer

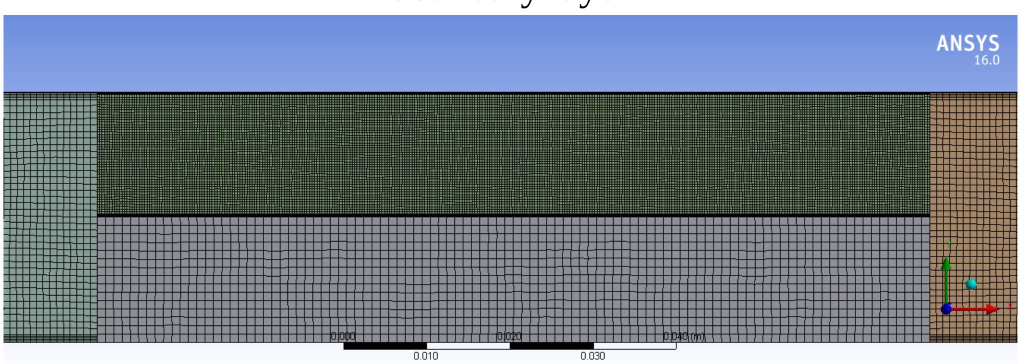

Figure 4. Meshed model of porous medium with height 15mm

About porous media and assumptions:

The porous media model incorporates an empirically determined flow resistance in a region of your model defined as "porous". In essence, the porous media model is nothing more than an added momentum sink in the governing momentum equations. As such, the following modeling assumptions and limitations should be readily recognized:

- The effect of the porous medium on the turbulence field is only approximated.
- In general, the **ANSYS FLUENT** porous medium model, for both single phase and multiphase, assumes the porosity is isotropic, and it can vary with space and time.

The inputs required for the porous media are:

- Porosity
- Viscous resistance coefficient
- Inertial resistance coefficient

Since the modeling involves laminar flow through a packed bed, we will be using the Blake-Kozeny Equation [17]:

$$\frac{|\Delta p|}{L} = \frac{150\mu}{D_p^2}\frac{(1-e)^2}{\epsilon^3}v_\infty + \frac{1.75\rho}{D_p}\frac{(1-e)}{\epsilon^3}v_\infty^2$$

Where μ is the viscosity, D_p is the mean particle diameter, L is the bed depth and ε is void fraction defined as the volume of voids divided by the volume of packed bed region.

Thus, the permeability and inertial loss coefficient in each component direction can be determined as:

$$\alpha = \frac{D_p^2}{150}\frac{\epsilon^3}{(1-\epsilon)^2}$$

And the viscous resistance coefficient can be determined by

$$\vartheta = \frac{1.75\rho}{D_p}\frac{(1-e)}{\epsilon^3}$$

The viscous resistance coefficient was taken equal to 19000000 [1/m^2] and inertial resistance coefficient was taken as 1920 [1/m]

9. Results and Discussion

CFD Analysis was carried out to study the packed bed of solar air heater. RNG k-ε turbulence modelling was carried out to study the geometrical parameters. Initially previous study models were emulated [14] [15] with the same geometrical parameters. Then, with increase in height of porous media the Nusselt number kept on increasing until 25mm height. This was due to the increase in the residence time of the air between the two porous medium and also due to the turbulence created on the upper side of the medium. After 25mm, the effect of turbulence started decreasing so the Nusselt number also decreased to about 66.67 for 30mm height which was the maximum possible height of the porous medium. However, the pressure drops kept on increasing from 12 mm height to 30mm height.Therefore, 25mm height was the optimum height for which the mean temperature at outlet, 314.088 K, was maximum.

Height [mm]	Nusselt number	T_m [K]	Pressure drop [Pa]
12	77.08	307.552	31.379
15	150.598	309.673	58.53
18	155.572	311.458	88
20	164	312.466	147.823
22	173.681	313.391	226.521
25	181.282	314.088	475.335
27	130.197	307.895	911.62
30	66.67	304.596	1670.22

Table 2: Mean temperature and Nusselt Number for various height

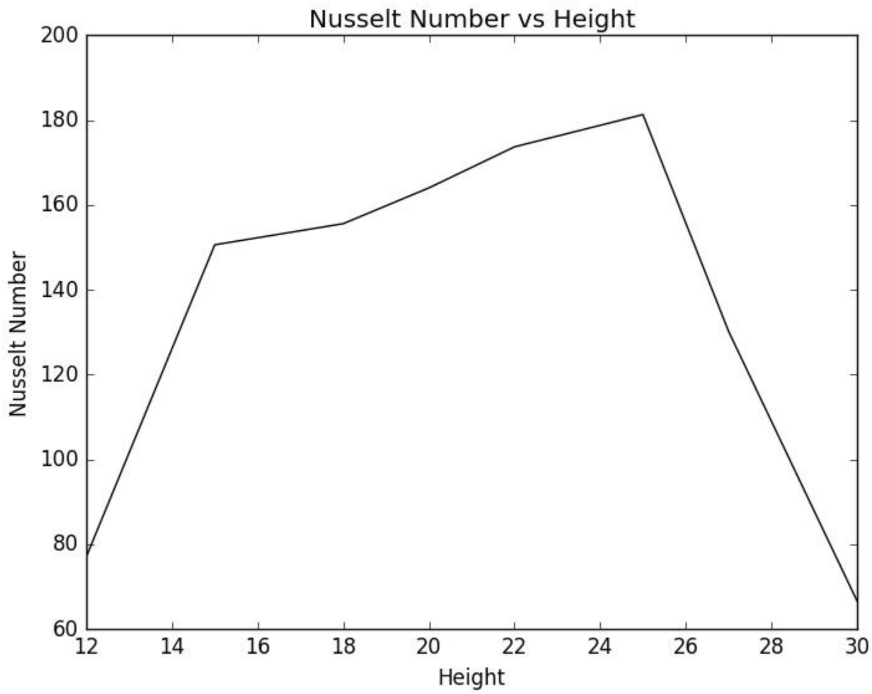

Figure 5. Nusselt number vs Height (mm) of porous medium

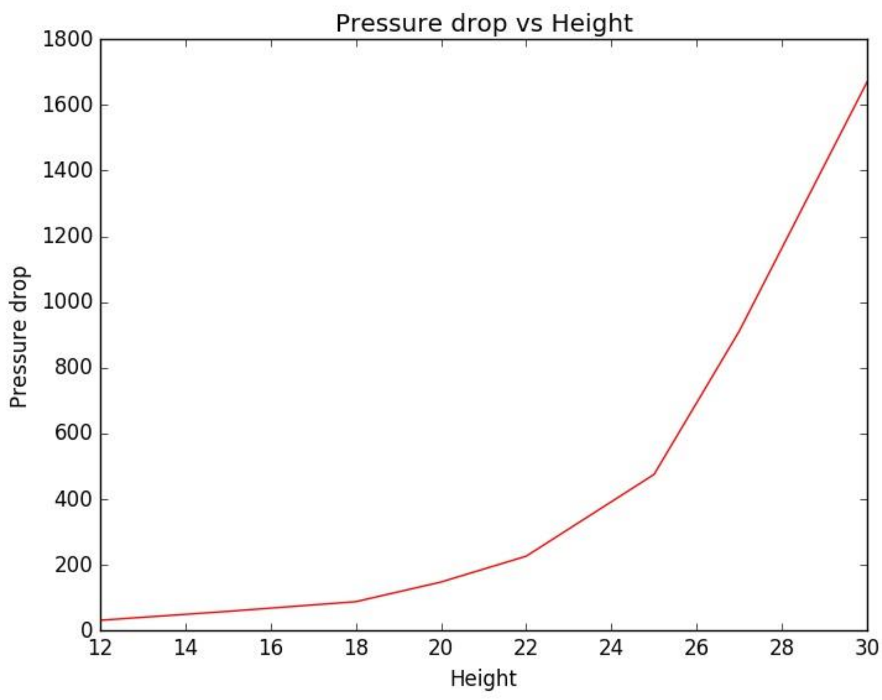

Figure 6. Pressure Drop (Pa) vs Height (mm) of porous medium

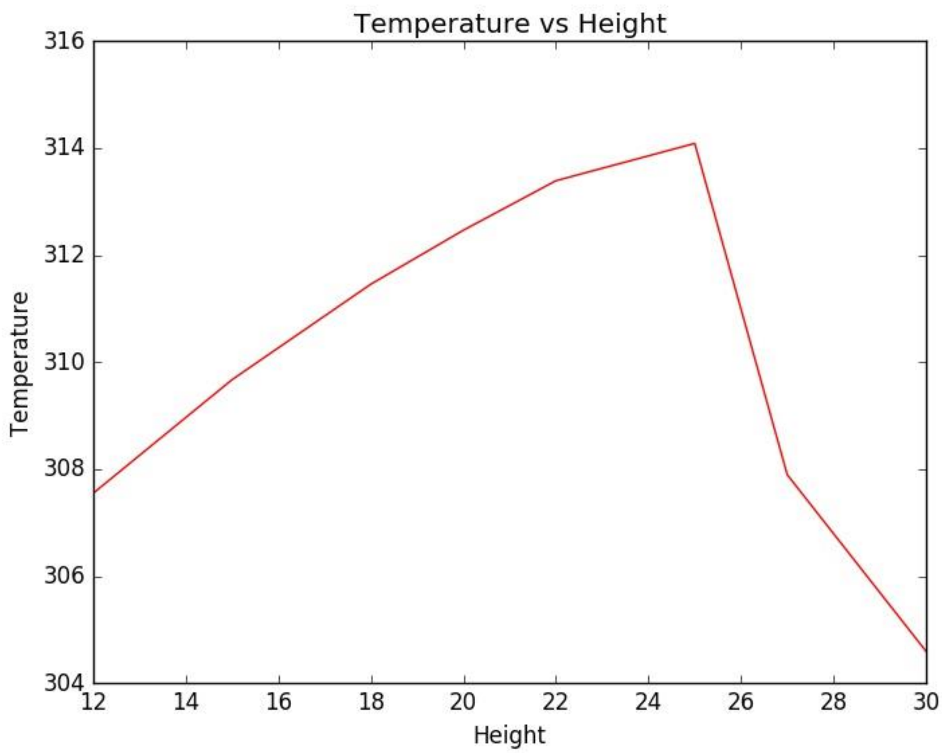

Figure 7: Temperature(K) vs Height(mm)

Until here , the porous medium were in a shape of rectangular blocks. The shape was changed to V- shape blocks of height 25mm and the angle was varied from 25° to 40°.

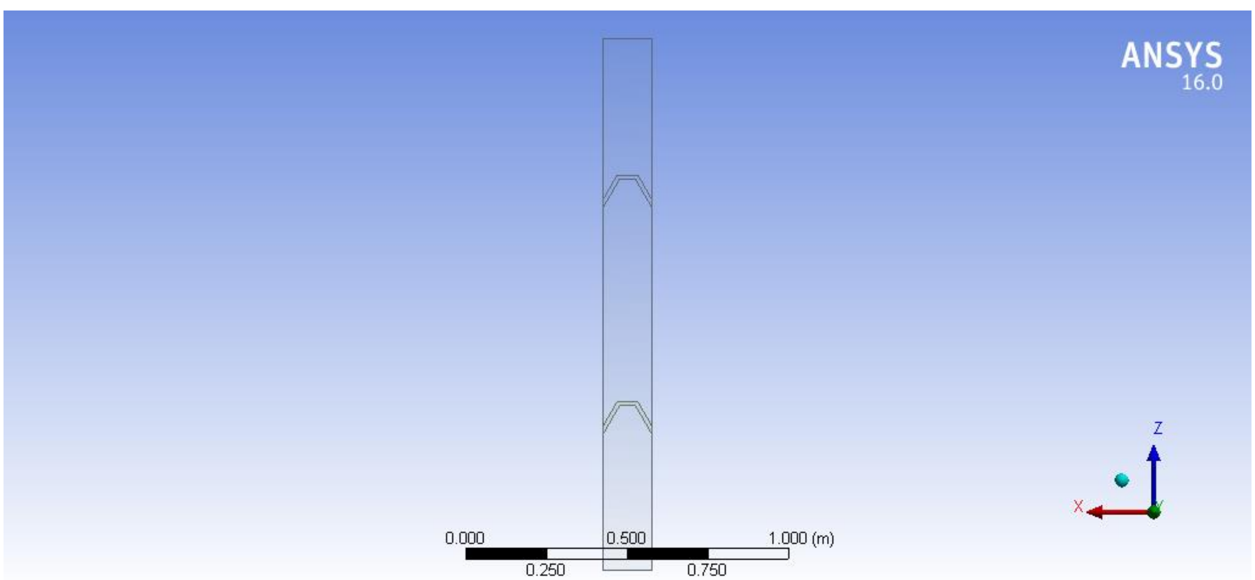

Figure 8. Geometry For V-Block with angle 30°

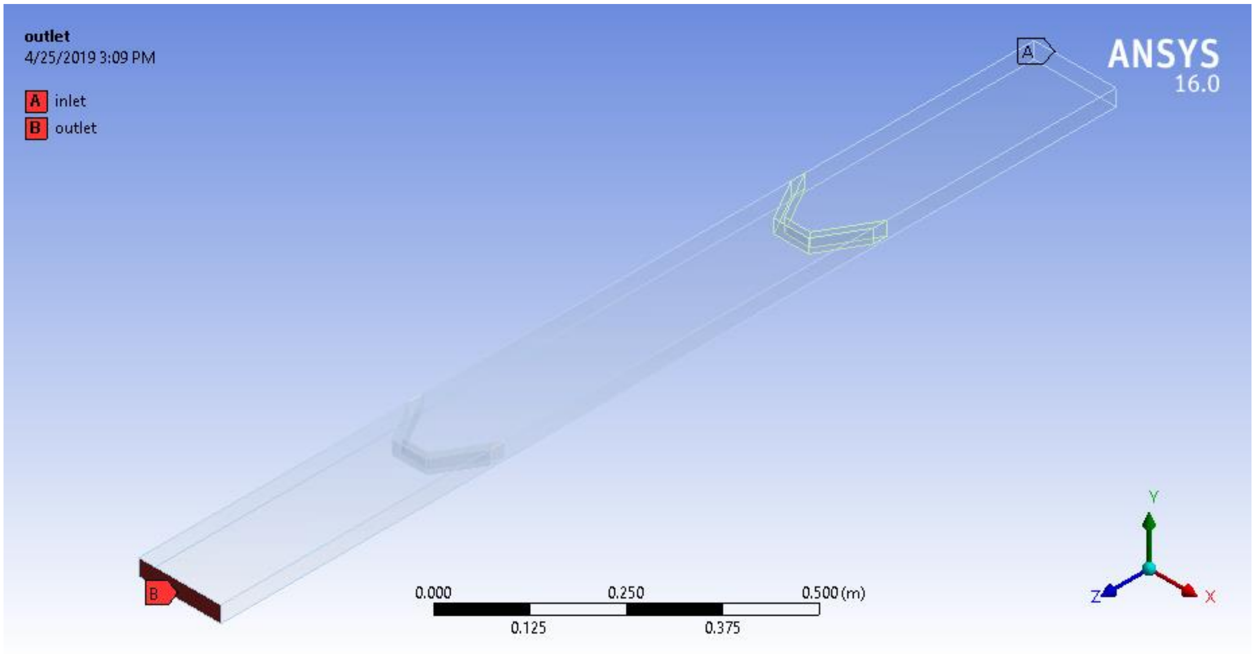

Figure 9 Inlet and outlet Boundaries

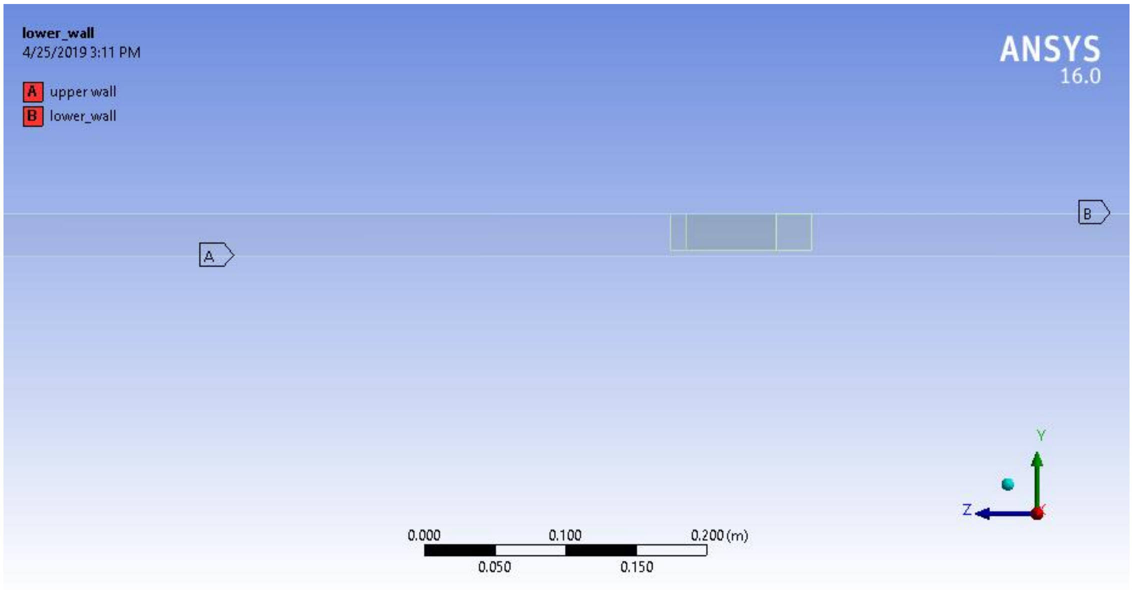

Figure 9 Upper and lower walls

A constant heat flux of 1500 was provided on the upper wall. With the increased in angle from $20° \; to \; 40°$ the mean temperature was maximum for 30°. However, it had the lower most heat transfer coefficient as indicated by Nusselt number. As the blockage provided to air increased with the degree the pressure drop also increased but it was far less compared to the rectangular geometry.

Angle	Nusselt no.	Tm [K]	Pressure drop [Pa]
20	67.34	313.343	55.189
30	54.223	314.105	78.369
40	70.94	313.231	95.818

Table 3: Mean Temperature and Nusselt Number at various angles

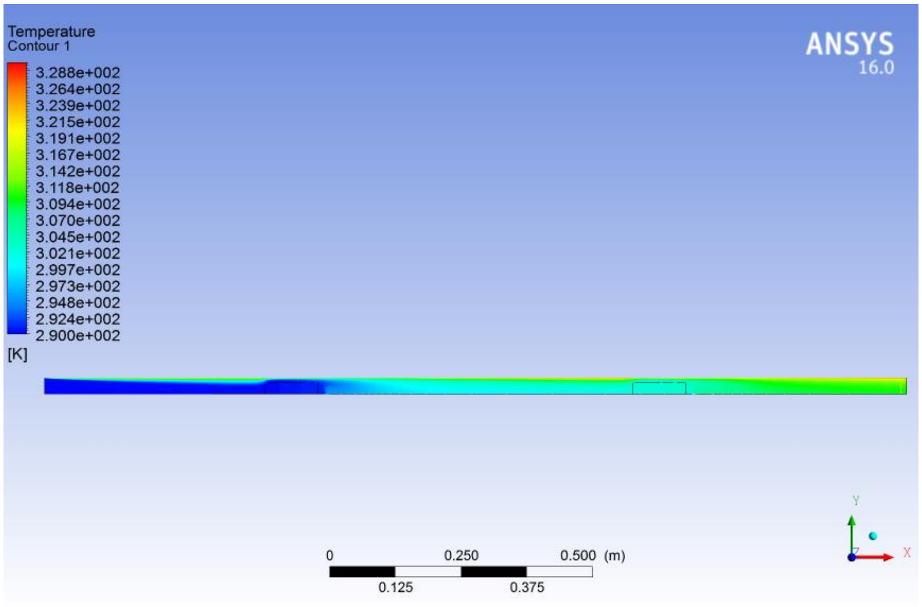

Figure 10 Temperature Contour for porous wall height 22 mm

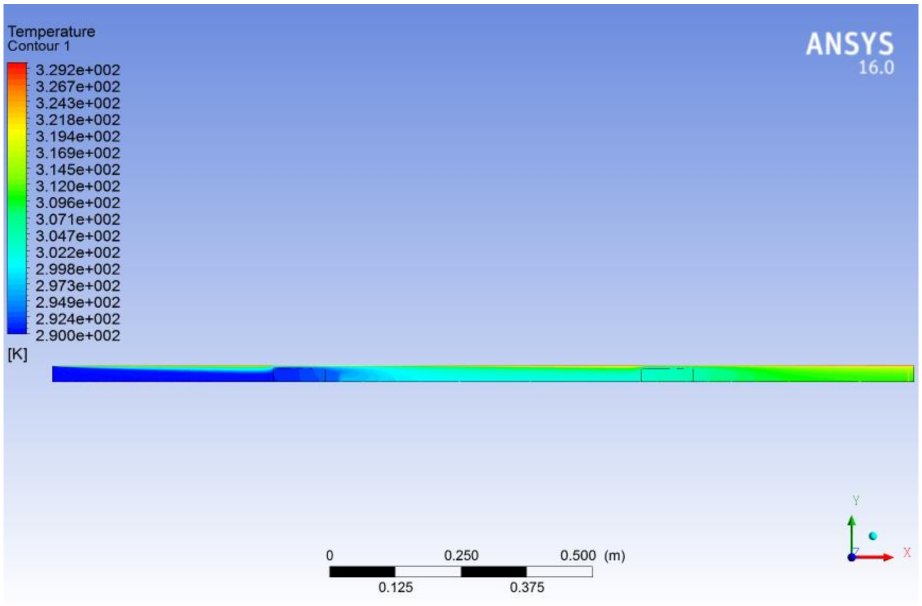

Figure 11 Temperature Contour for porous wall height 25 mm

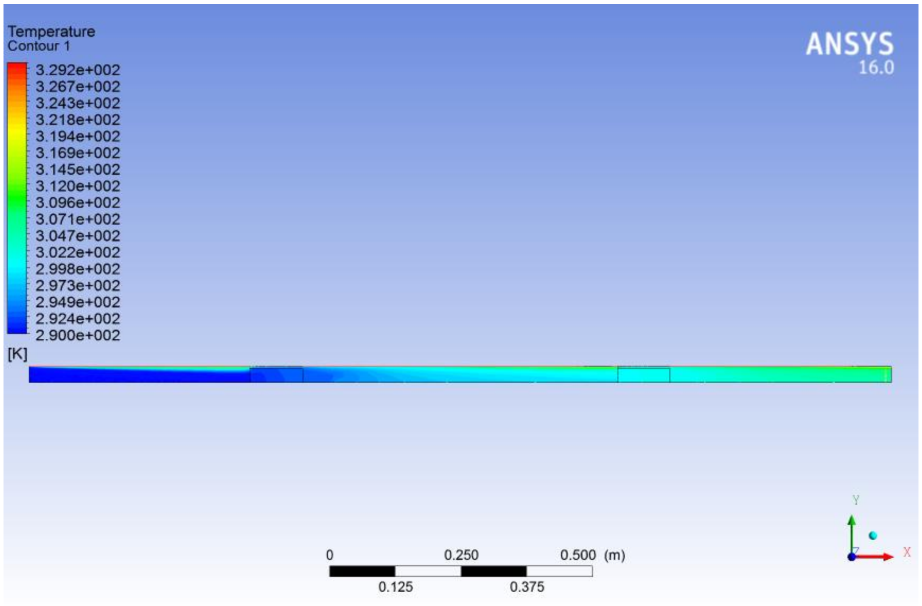

Figure 12 Temperature Contour for porous wall height 27 mm

10. Conclusions and future work

After conducting various CFD simulations of packed bed solar air heater with varying height and different geometries the results showed that for a rectangular type packed bed, the optimum height was 25mm for which the mean temperature at outlet was 314.088. However, corresponding to that model the pumping losses were high as the pressure drop was equal to 475.335 Pa. For a V-shape packed bed , almost same outlet mean temperature as obtained for angle 30° also the pressure drop was comparatively low and equal to 78.369 Pa. Therefore, the V-shape geometry with angle 30°was the best possible geometry out of those analysed.

Similar analysis can be done by varying the porosity of the medium, using different materials for the porous medium. Various other geometry configurations can be analysed instead of V-shape geometry. A combination of porous medium and artificial roughness geometry can also be introduced.

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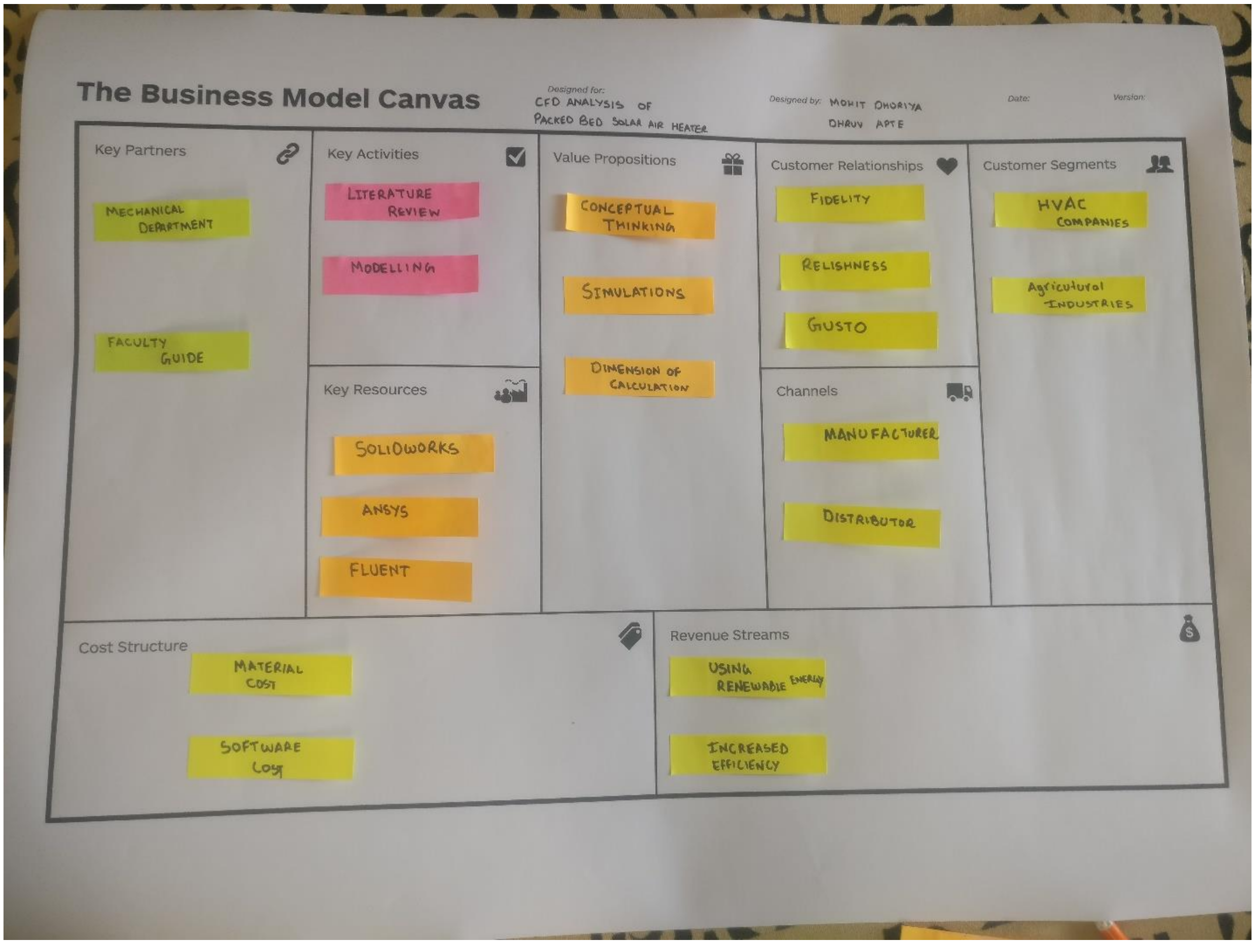
12.BMR Canvas



13. Plagiarism Report