International Journal of Scientific & Engineering Research, Volume 8, Issue 4, April-2017
ISSN 2229-5518

Performance Analysis for the High Mass Flow Rate Double-Pass Solar Air Collector with and Without Porous Media

Dheeraj Kumar, Suraj Bhoyar, S.V. Karankoti

Abstract: The double-pass solar collector with porous media in the lower channel provides a higher outlet temperature compared to the conventional double-pass collector without porous media. As a result, the thermal performance of the solar collector with porous media is higher. To develop a solar air heater of higher mass flow rate a literature survey was conducted to specify the design parameter and based on this survey a collector was designed and the porous media has been inserted to increase the total heat transfer rate and contact area. Moreover, the effect of higher mass flow rate and temperature rise has a higher thermal efficiency for the double-pass solar collector. In addition, pressure drop and heat transfer relationships have been developed for airflow through the porous media. The experiment concluded that for the higher mass flow rate the presence of porous media in the second channel increases the thermal performance of the overall system by 20%.

1 INTRODUCTION

As the population around the world and the demand for energy increases, the consumption of conventional fuels also increases. The limited sources of conventional fuels have directed researchers' attention to renewable energies in recent years. The total solar energy absorbed by the Earth's atmosphere, oceans and land masses is approximately 3,850,000 exajoules (EJ) per year. The amount of solar energy reaching the surface of the planet is about twice as much as will ever be obtained from all of the Earth's non-renewable resources of coal, oil, natural gas, and mined uranium combined. This is a huge source of energy and many applications that use solar energy, which is an abundant, clean and safe source, have been investigated.

Solar technologies can be classified into two groups; passive and active heating.

• *Asst. Professor in Mechanical Engineering Department (Sinhgad College of Engineering, Pune) e-mail: dskumar.scoe@sinhgad.edu*
Asst. Professor in Mechanical Engineering Department
ssbhoyar.scoe@sinhgad.edu
• *S.V. Karankoti, Asst. Prof., SIT, Lonavala.*

Passive solar techniques include designing spaces according to natural circulation, locating buildings with reference to the Sun or selecting high thermal conductive materials. On the other hand, active solar techniques include using solar panels, pumps or fans to convert solar energy into useful outputs.

Solar air heaters are devices that utilize solar radiation for a variety of purposes. These devices are simple and can be constructed inexpensively. Mainly, solar air heaters consist of a transparent glass, an absorber plate having higher thermal diffusivity and insulation material. The air flow enters through the channel that is formed by the absorber plate and the transparent medium. Solar radiation absorbed by the absorber plate. The absorbed heat transferred to the air as it flows along the channel increases its temperature. This heated air can be used in several applications such as removing moisture from agricultural products, heating and air conditioning, water heating and industrial process heating.

There are many advantages of solar air heater systems. They are simple in design and easy to maintain. After the set-up cost, a solar air heater system has no fuel expenditure. There is less leakage and corrosion when compared to the systems that use liquid.

The primary disadvantage of these systems is the low heat transfer coefficient compared to systems that use liquid as the working fluid. Low heat transfer coefficients lead to low thermal efficiency of solar air heaters. For many years researchers have studied the enhancement of heat transfer coefficients of solar air heaters. To increase hthe efficiency of such a system, various configurations and designs have been proposed.

The efficiency of solar air heaters can be affected by various parameters such as collector length, number of channels, depth of channels, type of absorber plate, number and material of glass covers, air inlet temperature and air velocity. All of these parameters and their effects on the efficiency of solar air heaters will be discussed below.

In this paper, porous material was placed in a solar air heater which was constructed for this study and the effect of this material on the collector efficiency and heat transfer was investigated. A literature survey was conducted to specify the design parameters of solar air heaters and based on this survey a collector was designed specially in order to obtain maximum efficiency and minimum heat loss from the collector and its channels. Air circulation is carried out by a radial fan.

2 LITERATURE SURVEY

Ramani et al. theoretically and experimentally investigated the performance of a double pass solar air collector with and without porous. The effects of parameters on the efficiency and on the pressure drop of solar air heater were examined. Two glasses were used with a thickness of 4 mm in the setup. There were two air flow passages. The first one was formed by the gap between the first and second glass while the second air passage was formed by the second glass cover and the absorber. Two solar air collectors were constructed in order to compare the results. One of them had two clear air flow channels while collector's second air flow channel of the other was filled with porous material. Wire mesh was used as porous material. The two passages were of 2100 x 540 x 21 mm in dimension. A black painted aluminum sheet with a thickness of 1mm was used as the absorber in both Solar collectors. It was concluded that using a double pass solar air heater with porous material improved the thermal performance.

Sopian et al. (2009) theoretically and experimentally evaluated the thermal efficiency of a double pass solar collector with porous and nonporous media. The testing facility consisted of a solar collector and a simulator. The simulator used 45 halogen bulbs, with a power of 300W each. Average radiation of 642 W/m2 could be reached with this simulator. The collector was of 2400 x 1200 mm in dimension. The collector consists of a single glass cover and a black painted aluminum plate. The solar collector was operated at varying inlet temperature and radiation conditions. The airflow rate was between 0.03 - 0.07kg/s, the upper channel depth was between 35-105 mm, and the lower channel depth was between 70-140 mm. The presence of porous media in the second channel increased the outlet temperature.

Aldabbagh et al. experimentally investigated the performance of single and double pass solar air heaters with steel wire mesh layers. The set-up was constructed from 20 mm thick, black painted plywood with dimensions of 1500 x 1000 mm. Two normal window glasses with 4 mm thickness were used in the design of solar air heater. The distance between the upper and lower glass covers was 50 mm and the distance between the lower glass and the wire mesh layers was 100 mm. By removing the upper glass cover, the setup could be used as a single pass solar air heater. Ten black painted steel wire meshes were placed in the second channel of the collector. The wires were 2 x 2 mm in cross section and placed with a distance of 10 mm from each other. During the experiments mass flow rate of the air was varied between 0.012 to 0.038 kg/s in order to observe the effect of mass flow rate on the efficiency of the system. It was concluded that by using steel wire mesh layers as the absorber plate, the efficiency of the collector could be enhanced when compared to the conventional type..

As a result of this literature survey, the researches that are investigated in recent years, studied carefully in a detail way. The construction choices, the purposes of other researches', the results of such approaches are determined. Some items about the experimental setups like the dimensions, the materials, the testing procedures are investigated and the experimental setup that has been constructed and tested.

3 EXPERIMENTATION SETUP DESCRIPTION

The experimental set up consists of the following components measuring instruments:

1. Flow straightening duct.
2. Solar collector.
3. Porous media.
4. Exit duct.

5. Suction pipe.
6. Orifice.
7. Control valve to control the mass flow rate.
8. Blower used as suction.
9. Temperature sensors and indicator to measure temperature at various points.
10. Velocity calculator.
11. Control panel.
12. Pressure measuring instruments (U- tube manometer and inclined manometer).

4 EXPERIMENTAL PROCEDURE AND HEAT TRANSFER CALCULATIONS

In this paper, enhancement of heat transfer in a solar air heater by using porous medium is experimentally investigated. A laboratory simulation model is constructed and tested in different conditions to achieve a significant enhancement in heat transfer.

4.1 Experimental Procedure

To carry out the experimental analysis, a double pass solar air collector was constructed which has been shown in figure A2. with the size of length 1m, width 0.5m and the total depth 0.06m. The depth of pre-heater is 2.5cm and the lower channel depth is 3.5cm. The two passages are separated by transparent fibre glass of thickness 3mm. The wall of the collector is made from a compressed asbestos sheet with a profile of 10mm thickness insulated by thermo-col with the thickness of 4cm to prevent the heat loss from the wall. In this setup Aluminium sheet was painted black and used as an absorber. The ordinary transparent glass with a thickness of 5 mm was used as a collector cover. First air was pre-heated in the channel 1 between glass cove and fibre glass and then pass to the next channel over the absorber plate. To draw the ambient air into the collector through the entrance section a blower (0.5hp) with a speed control system was modified as suction to the lower channel. To increase the contact surface area with hot air, porous media where inserted along the length of the collector in the lower channel. In this setup 3 halogen lamp of 1000 watt each was used as a source of heat instead of sun light. Halogen lamp was installed in such a way that the intensity of lamp and the sun intensity for Pune have equal value. From morning to evening the sun intensity is different. To achieve different intensity of light from the lamp a dimmer was used to control the voltage supply. The lamps are installed at a height of 50cm from the glass and intensity was measured at different supply with the help of Lux meter to match the sun intensity.

Temperatures are measured at different points along the length of absorber plate and air at different locations with the help of k -type thermocouple. A digital multi - meter is used to indicate the output of the thermocouples through the selector switch. The temperature was measured four times and it's average values was calculated in 30 minutes A taper duct has been provided at the outlet of the test collectors to stabilize the air flow. The exit section is connected to blower through a PVC pipe. Measurement of mass flow rate of air through collector has been accomplished by 25mm diameter orifice plate.

The test runs to collect relevant heat transfer were conducted under quasi-steady state conditions. The quasi-steady state condition was assumed have to been reached when the temperature at the point does not change for about 10 minutes. When a change in the operating conditions (mass flow rate or intensity of light) is made, it takes about 15 min to reach such a quasi-steady state. Seven values of flow rates were used for each set at a variable heat flux of the test. After each of flow rate, the system is allowed to attain a steady state before the data were recorded. The tests with the solar collector system were carried out for 10 days with different operating condition. The experiments on the efficiency were conducted during which the atmospheric conditions of the lab were almost uniform and data was collected from the efficiency calculation. Values of various input parameters and constants are given in Table 1.
The experiment on efficiency for Collector with porous media (steel wool) and without porous media was conducted and its calculated values are given in Tables 2, 3, 4 and 5.
The following parameters were measured and thermal efficiency where calculated:

1. Temperature of glass plate, fibre plate and absorber.
2. Temperature of air at inlet and outlet of the test section.
3. Pressure difference across orifice.
4. Mass flow rate.
5. Friction factor.

4.2 Heat Transfer Calculations

For the sake of convenience the heat transfer coefficients between the air stream and the covers and between the air stream and the absorber plate are assumed equal and can be calculated as follows:

Performance of Solar Collector Without Porous Media for Variable Mass Flow Rate and Heat Input.

The experiment is conducted by increasing heat flux and mass flow rate for solar air heater without porous media. The temperatures are taken from temperature indicator and for pressure drop the difference of the water level in the manometer.

The volume flow rate of air is measured across orifice by equation 2. The density of air at mean bulk temperature Ta= (Tinlet+Toutlet) / 2 (1)

The air density:

$$\rho_a = \frac{P_a}{RT_a} \quad (2)$$

$$Q = C\frac{\pi}{4}d1^2\sqrt{2g(H1 - H2)} \quad (3)$$

$$C = \frac{c_d}{\sqrt{1-(\frac{d_2}{d_1})^2}} \quad (4)$$

$$m = Q \times \rho_a \quad (5)$$

By using the equation 1 we find the bulk mean temperature of the air =310K. From the air property table at temperature 310K the density of air is ρ_a = 1.1381 kg/ m3.

By using the equation 2, 3 and 4 we get the mass flow rate of the air = 0.193461 kg/sec.

The other properties of air are obtained from the chart.

Kinematic viscosity of air at temperature 310K is:

$$v = \frac{\mu}{\rho} = 0.000016738 \quad (6)$$

Thermal diffusivity of air at temperature 310K is:

$$\alpha = \frac{k}{\rho c_p} = 0.000023046 \quad (7)$$

Prandtl number of air at temperature 310K is:

$$P_r = \frac{v}{\alpha} = 0.72631 \quad (8)$$

Hydraulic diameter of air at temperature 310K is:

$$D_h = \frac{4A_f}{P} = 2D = 0.047619 \quad (9)$$

The Reynolds number is calculated by using equation 10.

$$R_e = \frac{\rho U D_h}{\mu} = \frac{2m}{\mu} \quad (10)$$

Re = 38425.18

Where Q useful is the accumulated energy extracted from the collector during the working period in W, as is collector area in .5 m2.

Useful heat gain for air collector can be expressed as:

$$Q_{useful} = m\,C_p\Delta \quad (11)$$

Where, Cp is the specific heat of the fluid. The Cp of the air at 301K is 1007 J/kg K.

Heat gain by air = 1168.892 W

Heat transfer coefficient:

$$h = \frac{Q_{useful}}{A\,(\Delta T)} \quad (12)$$

Heat transfer coefficient = 46.94347 W/m2K

Convective heat transfer coefficient between any two surfaces is given by:

$$h_{r12} = \frac{\sigma(T_1+T_2)(T_1^2+T_2^2)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1} \quad (13)$$

The calculation of experimental Nusselt number as:

$$Nu = \frac{hD_h}{k} \quad (14)$$

Experimental Nusselt number Nu = 96.36998

Theoretical Nusselt number is calculated by using Dittus- Boltere correlation:

$$Nu = 0.0333\,R_e^{0.8}\,P_r^{0.33} \quad (15)$$

Theoretical Nu = 99.01655.

When the air flows through the channel due to friction there is a pressure drop along the channel. The pressure inside the duct for a given mass flow rate is given by the: [26]

$$p = f\left(\frac{m^3}{\rho}\right)\left(\frac{L^3}{D^3}\right) \quad (16)$$

Where f (friction factor) is

$$f = f_0 + y(\frac{D}{L}) \quad (17)$$

The value of f_0 and y are:

f_0=24/Re, y=0.9 for Laminar flow (Re<2550)

f_0=0.0094, y=2.92Re-0.15for transitional flow (2550<Re<104)

f_0 = 0.059 Re-0.2, y =0.73 for turbulent flow (104<Re<105)

Friction factor = 0.018094

The pressure inside the duct = 176.3044 Pascal

The value of pressure drop (ΔP) across the collector having length as 'L' flow velocity 'V' and hydraulic diameter 'Dh' is determined by using friction factor 'f' as follows:

$$\Delta P_1 = \frac{2fLV^2\rho}{D_h} \quad (18)$$

So far as pressure drop (pumping power) is concerned, the counter flow solar air heater has a U-turn section and extra-length for air passages. Hence the extra pressure drop is introduced by this design. The pressure drop in the u-section can be calculated as:

$$\Delta P_2 = \frac{Km^2}{2\rho D^2} \quad (19)$$

K=1for U-section

$\Delta P_1 + \Delta P_2 = 319.9$ pascal

The pumping power can be calculated as:

$$W = \frac{m(\Delta P_1 + \Delta P_2)}{\rho} \quad (20)$$

The pumping power = 54W

The efficiency of a solar collector is defined as the ratio of useful gain to the incident solar energy, that is:

$$\eta = \frac{\text{heat gain}}{\text{total solar striking collector surface}} = \frac{Q_{useful}}{I \times A_S} \quad (21)$$

The efficiency of a solar collector = 0.22

% Error = (Nu Exp /Nu Th) = 2.672854933

Performance of Solar Collector With Porous Media for Variable Mass Flow Rate and Heat Input.

The experiment is conducted by increasing heat flux and mass flow rate for solar air heater without porous media. The temperatures are taken from temperature indicator and for pressure drop the difference of the water level in the manometer.

The volume flow rate of air is measured across orifice by equation 22. The density of air at mean bulk temperature Ta= (Tinlet+Toutlet) / 2 (22)

The air density:

$$\rho_a = \frac{P_a}{RT_a} \quad (23)$$

$$Q = C\frac{\pi}{4}d1^2\sqrt{2g(H1 - H2)} \quad (24)$$

$$C = \frac{c_d}{\sqrt{1-(\frac{d_2}{d_1})^2}} \quad (25)$$

$$m = Q \times \rho_a \quad (26)$$

By using the equation 22 we find the bulk mean temperature of the air =313K. From the air property table at temperature 313K the density of air is $\rho_a = 1.127$ kg/ m3.

By using the equation 23, 24 and 25 we get the mass flow rate of the air = 0.19252 kg/sec.

The other properties of air are obtained from the chart A.C1.

Kinematic viscosity of air at temperature 313K is:

$$v = \frac{\mu}{\rho} = 0.00001702 \quad (27)$$

Thermal diffusivity of air at temperature 313K is:

$$\alpha = \frac{k}{\rho c_p} = 0.00002346 \quad (28)$$

Prandtl number of air at temperature 313K is:

$$P_r = \frac{v}{\alpha} = 0.7255 \quad (29)$$

Hydraulic diameter is:

$$D_h = \frac{4A_f}{P} = 2D = 0.047619 \quad (30)$$

The Reynolds number is calculated by using equation 31.

$$R_e = \frac{\rho U D_h}{\mu} = \frac{2m}{\mu} \quad (31)$$

Re = 38237.3

Where Q useful is the accumulated energy extracted from the collector during the working period in W, as is collector area in .5 m2.

Useful heat gain for air collector can be expressed as:

$$Q_{useful} = m\,C_p\Delta \quad (32)$$

Where, Cp is the specific heat of the fluid. The Cp of the air at 313K is 1007 J/kg K.

Heat gain by air = 2132.49 W

Heat transfer coefficient:

$$h = \frac{Q_{useful}}{A\,(\Delta T)} \quad (33)$$

Heat transfer coefficient = 46.714 W/m2K

Convective heat transfer coefficient between any two surfaces is given by:

$$h_{r12} = \frac{\sigma(T_1+T_2)(T_1^2+T_2^2)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1} \quad (34)$$

The calculation of experimental Nusselt number as:

$$Nu = \frac{hD_h}{k} \quad (35)$$

Experimental Nusselt number Nu = 95.1558

Theoretical Nusselt number is calculated by using Dittus- Boltere correlation:

$$Nu = 0.0333\, R_e^{0.8}\, P_r^{0.33} \quad (36)$$

Theoretical Nu = 98.4752

When the air flows through the channel due to friction there is a pressure drop along the channel. The pressure inside the duct for a given mass flow rate is given by the: [26]

$$p = f\left(\frac{m^3}{\rho}\right)\left(\frac{L^3}{D^3}\right) \quad (37)$$

Where f (friction factor) is

$$f = f_0 + y(\frac{D}{L}) \quad (38)$$

The value of f_0 and y are:

f_0=24/Re, y=0.9 for Laminar flow (Re<2550)

f_0=0.0094, y=2.92Re-0.15for transitional flow (2550<Re<104)

f_0 = 0.059 Re-0.2, y =0.73 for turbulent flow (104<Re<105)

Friction factor = 0.0181

The pressure inside the duct = 176.373 Pascal

The value of pressure drop (ΔP) across the collector having length as 'L' flow velocity 'V' and hydraulic diameter 'Dh' is determined by using friction factor 'f' as follows:

$$\Delta P_1 = \frac{2fLV^2\rho}{D_h} \quad (39)$$

So far as pressure drop (pumping power) is concerned, the counter flow solar air heater has a U-turn section and extra-length for air passages. Hence the extra pressure drop is introduced by this design. The pressure drop in the u-section can be calculated as:

$$\Delta P_2 = \frac{Km^2}{2\rho D^2}$$

(40)

K=1for U-section

$$\Delta P_1 + \Delta P_2 = 320.01 \text{ Pascal}$$

The pumping power can be calculated as: $W = \frac{m(\Delta P_1+\Delta P_2)}{\rho}$ (41)

The pumping power = 54.6628 W

The efficiency of a solar collector is defined as the ratio of useful gain to the incident solar energy, that is:

$$\eta = \frac{\text{heat gain}}{\text{total solar striking collector surface}} = \frac{Q_{useful}}{I \times A_S} \quad (42)$$

The efficiency of a solar collector = 0.39491

% Error = (NuExp/NuTh) = 3.37078319

Performance of Solar Collector without Porous Media at Constant Mass Flow Rate.

The experiment is conducted by increasing heat flux at constant mass flow rate for solar air heater without porous media. The temperatures are taken from temperature indicator and for pressure drop the difference of the water level in the manometer.

The volume flow rate of air is measured across orifice by equation 44. The density of air at mean bulk temperature Ta= (Tinlet+Toutlet) / 2 (43)

The air density:

$$\rho_a = \frac{P_a}{RT_a}$$

(44)

$$Q = C\frac{\pi}{4}d1^2\sqrt{2g(H1-H2)} \quad (45)$$

$$C = \frac{c_d}{\sqrt{1-(\frac{d_2}{d_1})^2}}$$

(46)

$m = Q \times \rho_a$

(47)

By using the equation 43 we find the bulk mean temperature of the air =311K. From the air property table at temperature 311K the density of air is ρ_a = 1.1344 kg/ m3.

By using the equation 44, 45 and 46 we get the mass flow rate of the air = 0.29255 kg/sec.

The other properties of air are obtained from the chart.

Kinematic viscosity of air at temperature 311K is:

$$v = \frac{\mu}{\rho} = 0.000016832$$

(48)

Thermal diffusivity of air at temperature 311K is:

$$\alpha = \frac{k}{\rho c_p} = 0.000023184 \quad (49)$$

Prandtl number of air at temperature 311K is:

$$P_r = \frac{v}{\alpha} = 0.72604 \quad (50)$$

Hydraulic diameter is:

$$D_h = \frac{4A_f}{P} = 2D = 0.047619 \quad (51)$$

The Reynolds number is calculated by using equation 52.

$$R_e = \frac{\rho U D_h}{\mu} = \frac{2m}{\mu} \quad (52)$$

Re = 58385.5

Where Q useful is the accumulated energy extracted from the collector during the working period in W, as is collector area in 0.5 m2.

Useful heat gain for air collector can be expressed as:

$$Q_{useful} = m\, C_p \Delta T \quad (53)$$

Where, Cp is the specific heat of the fluid. The Cp of the air at 311K is 1007 J/kg K.

Heat gain by air = 3240.54 W

Heat transfer coefficient:

$$h = \frac{Q_{useful}}{A\,(\Delta T)}$$

(54)

Heat transfer coefficient = 70.9865 W/m2K

Convective heat transfer coefficient between any two surfaces is given by:

$$h_{r12} = \frac{\sigma(T_1+T_2)(T_1^2+T_2^2)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}$$

(55)

The calculation of experimental Nusselt number as:

$$Nu = \frac{hD_h}{k}$$

(56)

Experimental Nusselt number Nu = 139.35

Theoretical Nusselt number is calculated by using Dittus- Boltere correlation:

$$Nu = 0.0333\, R_e^{0.8}\, P_r^{0.33}$$

(57)

Theoretical Nu = 138.303

When the air flows through the channel due to friction there is a pressure drop along the channel. The pressure inside the duct for a given mass flow rate is given by the: [26]

$$p = f\left(\frac{m^3}{\rho}\right)\left(\frac{L^3}{D^3}\right)$$

(58)

Where f (friction factor) is

$$f = f_0 + y(\frac{D}{L})$$
(59)

The value of f_0 and y are:
f_0=24/Re, y=0.9 for Laminar flow (Re<2550)
f_0=0.0094, y=2.92Re-0.15for transitional flow (2550<Re<104)
f_0 = 0.059 Re-0.2, y =0.73 for turbulent flow (104<Re<105)
Friction factor = 0.01752
The pressure inside the duct = 391.645 Pascal
The value of pressure drop (ΔP) across the collector having length as 'L' flow velocity 'V' and hydraulic diameter 'Dh' is determined by using friction factor 'f' as follows:

$$\Delta P_1 = \frac{2fLV^2\rho}{D_h}$$
(60)

So far as pressure drop (pumping power) is concerned, the counter flow solar air heater has a U-turn section and extra-length for air passages. Hence the extra pressure drop is introduced by this design. The pressure drop in the u-section can be calculated as:

$$\Delta P_2 = \frac{Km^2}{2\rho D^2}$$
(61)

K=1for U-section
$\Delta P_1 + \Delta P_2 =$ 709.96 Pascal
The pumping power can be calculated as:

$$W = \frac{m(\Delta P_1 + \Delta P_2)}{\rho}$$
(62)

The pumping power = 183.089 W
The efficiency of a solar collector is defined as the ratio of useful gain to the incident solar energy, that is:

$$\eta = \frac{\text{heat gain}}{\text{total solar striking collector surface}} = \frac{Q_{useful}}{I \times A_S}$$
(63)

The efficiency of a solar collector = 0.384
% Error = (NuExp/NuTh) = -0.7569889

Performance of Solar Collector with Porous Media at Constant Mass Flow Rate.
The experiment is conducted by increasing heat flux at constant mass flow rate for solar air heater with porous media. The temperatures are taken from temperature indicator and for pressure drop the difference of the water level in the manometer.
The volume flow rate of air is measured across orifice by equation 65. The density of air at mean bulk temperature Ta= (Tinlet+Toutlet) / 2
(64)
The air density:

$$\rho_a = \frac{P_a}{RT_a}$$
(65)

$$Q = C\frac{\pi}{4}d1^2\sqrt{2g(H1 - H2)} \quad (66)$$

$$C = \frac{c_d}{\sqrt{1-(\frac{d2}{d1})^2}}$$
(67)

$$m = Q \times \rho_a$$
(68)

By using the equation 64 we find the bulk mean temperature of the air =312K. From the air property table at temperature 311K the density of air is ρ_a = 1.1307 kg/ m3.
By using the equation 65, 66 and 67 we get the mass flow rate of the air = 0.29579 kg/sec.
The other properties of air are obtained from the chart A.C1.
Kinematic viscosity of air at temperature 312K is:

$$v = \frac{\mu}{\rho} = 0.000016926$$
(69)

Thermal diffusivity of air at temperature 312K is:

$$\alpha = \frac{k}{\rho c_p} = 0.000023322$$
(70)

Prandtl number of air at temperature 312K is:

$$P_r = \frac{v}{\alpha} = 0.72577$$
(71)

Hydraulic diameter is:

$$D_h = \frac{4A_f}{P} = 2D = 0.047619 \quad (72)$$

The Reynolds number is calculated by using equation 10.

$$R_e = \frac{\rho U D_h}{\mu} = \frac{2m}{\mu}$$
(73)

Re = 58890.8
Where Q useful is the accumulated energy extracted from the collector during the working period in W, as is collector area in 0.5 m2.
Useful heat gain for air collector can be expressed as:

$$Q_{useful} = m\, C_p \Delta$$
(74)

Where, Cp is the specific heat of the fluid. The Cp of the air at 312K is 1007 J/kg K.
Heat gain by air = 4170.04 W
Heat transfer coefficient:

$$h = \frac{Q_{useful}}{A\,(\Delta T)}$$
(75)

Heat transfer coefficient = 71.7735 W/m2K
Convective heat transfer coefficient between any two surfaces is given by:

$$h_{r12} = \frac{\sigma(T_1+T_2)(T_1^2+T_2^2)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}$$
(76)

The calculation of experimental Nusselt number as: $Nu = \frac{hD_h}{k}$
(77)

Experimental Nusselt number Nu = 128.75

Theoretical Nusselt number is calculated by using Dittus- Boltere correlation:

$$Nu = 0.0333\, R_e^{0.8}\, P_r^{0.33}$$
(78)

Theoretical Nu = 139.188

When the air flows through the channel due to friction there is a pressure drop along the channel. The pressure inside the duct for a given mass flow rate is given by the: [26]

$$p = f\left(\frac{m^3}{\rho}\right)\left(\frac{L^3}{D^3}\right)$$
(79)

Where f (friction factor) is

$$f = f_0 + y\left(\frac{D}{L}\right)$$
(80)

The value of f_0 and y are:

f_0=24/Re, y=0.9 for Laminar flow (Re<2550)

f_0=0.0094, y=2.92Re-0.15for transitional flow (2550<Re<104)

f_0 = 0.059 Re-0.2, y =0.73 for turbulent flow (104<Re<105)

Friction factor = 0.01751

The pressure inside the duct = 401.428 Pascal

The value of pressure drop (ΔP) across the collector having length as 'L' flow velocity 'V' and hydraulic diameter 'Dh' is determined by using friction factor 'f' as follows:

$$\Delta P_1 = \frac{2fLV^2\rho}{D_h}$$
(81)

So far as pressure drop (pumping power) is concerned, the counter flow solar air heater has a U-turn section and extra-length for air passages. Hence the extra pressure drop is introduced by this design. The pressure drop in the u-section can be calculated as:

$$\Delta P_2 = \frac{Km^2}{2\rho D^2}$$
(82)

K=1for U-section

$\Delta P_1 + \Delta P_2 =$ 728.35 Pascal

The pumping power can be calculated as:

$$W = \frac{m(\Delta P_1+\Delta P_2)}{\rho}$$
(83)

The pumping power = 190.535 W

The efficiency of a solar collector is defined as the ratio of useful gain to the incident solar energy,

that is:

$$\eta = \frac{\text{heat gain}}{\text{total solar striking collector surface}} = \frac{Q_{useful}}{I\times A_S}$$
(84)

The efficiency of a solar collector = 0.77223

% Error = (NuExp/NuTh) = 7.49935347

RESULTS AND DISCUSSION

The heat transfer enhancement of a solar air heater using porous material was investigated in this study. The reference experiments were performed with the constructed experimental setup described in previous chapters.

Four experimental results will be discussed in this section. The collector was tested while it was clear and assisted with porous material. The graphs for each experiment were drawn for the variation of fiction factor, efficiency, pressure drop and Nusselt number with different Reynolds number.

Effect of Various Parameters on Mass Flow Rate

The variation of efficiency, outlet temperature, pressure drop and heat transfer coefficient with different mass flow rate and heat input for porous and without porous are shown in Figures 1 to 3. With the increase in mass flow rate the efficiency of the system decreases. On the other hand the temperature difference increases with the increase of heat input and mass flow rate. By comparing the values of the efficiency and temperature difference for porous and without porous, it is found that the use of porous media increases the temperature difference by 32°C. This causes an increase in the pressure drop, which means increasing of the cost of the pumping power. Figure.1. shows the heat transfer coefficient of air increases at the different mass flow rate.

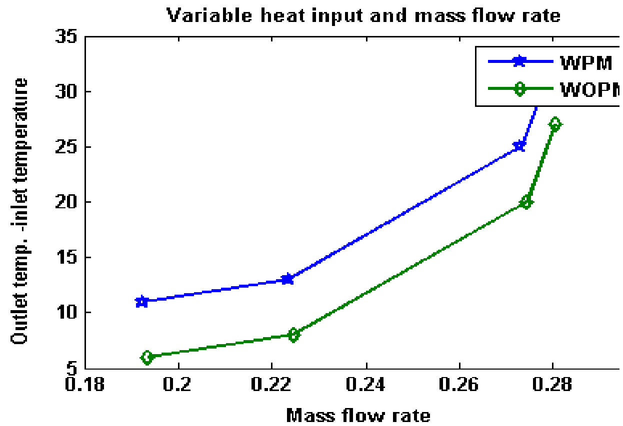

Figure.1: ΔT Vs mass flow rate (variable heat input and mass flow rate)

One should note that this increase in heat transfer coefficient is happening when heat input and mass flow rate increases. It was observed that heat transfer coefficient in the case of porous media is slightly greater than of without porous media at higher mass flow rate of (0.17to 0.27 kg/sec).

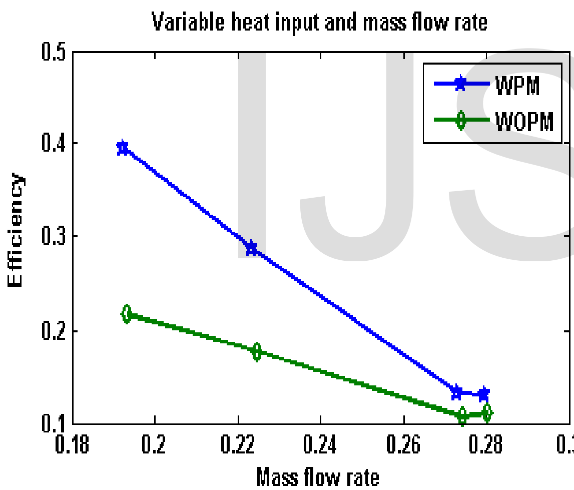

Figure.2: Efficiency Vs mass flow rate (variable heat input and mass flow rate)

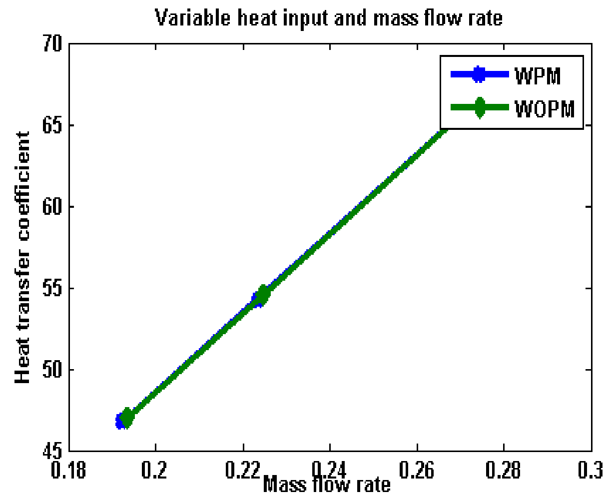

Figure.3: Heat transfer coefficient Vs mass flow rate (variable heat input and mass flow rate)

Effect of various parameters on Reynolds number

The variation of fiction factor, efficiency, pressure drop and Nusselt number with different Reynolds number are shown in figures 4 to 7. With the increase in Reynolds number the efficiency of the system decreases because of higher mass flow rate and also there is a loss of heat because of many factors. By comparing the friction factor and pressure drop for porous and without porous, it is found that as Reynolds number increases, friction factor decreases but the pressure drop increases. But the drop in pressure has no significant effect at higher mass flow rate. Nusselt number increases as Reynolds number increases. By comparing figures 7 and 10, it is found that efficiency of the system increases if porous media is used in the lower channel.

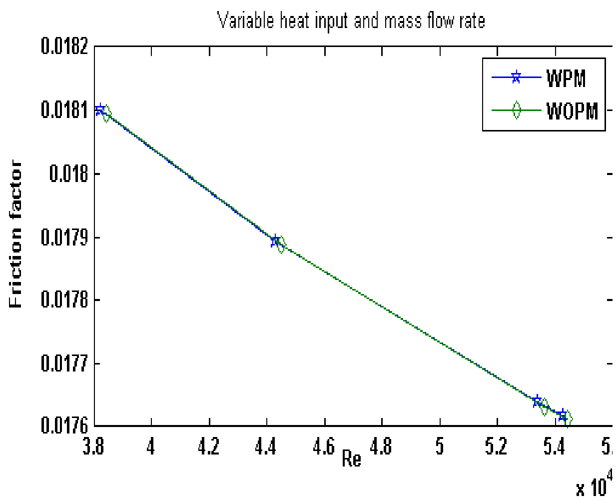

Figure. 4: Friction factor Vs Re (variable heat input and mass flow rate)

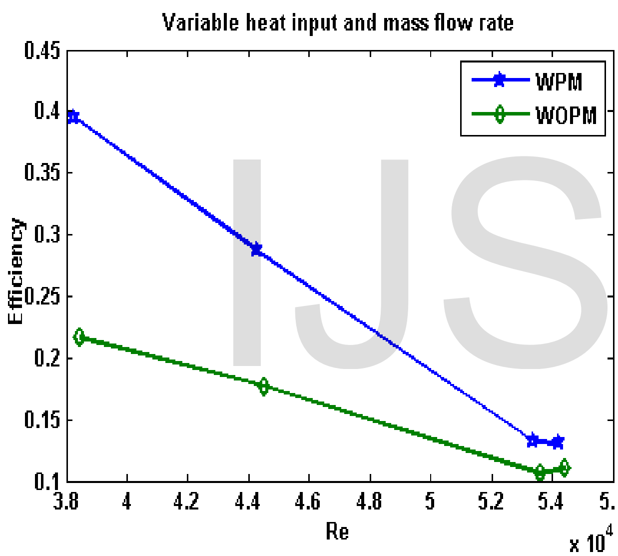

Figure. 5: Efficiency Vs Re (variable heat input and mass flow rate)

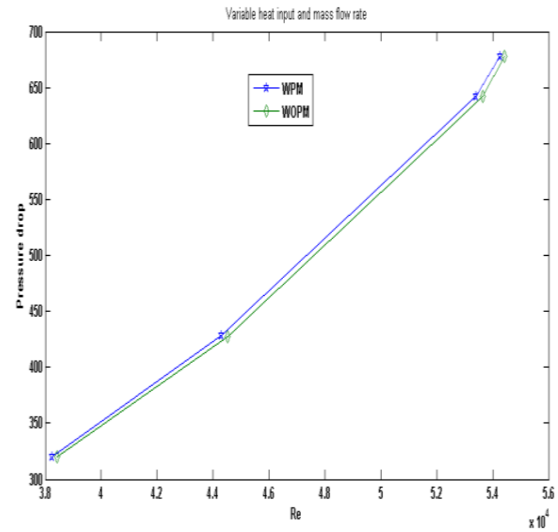

Figure .6: Pressure drop Vs Re (variable heat input and mass flow rate)

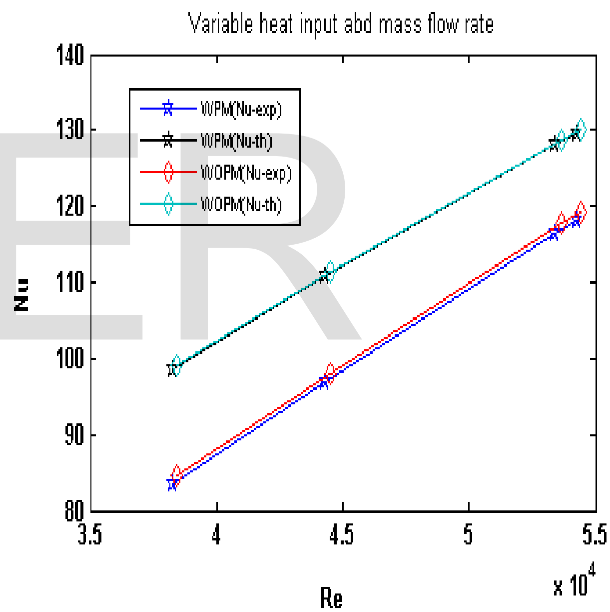

Figure. 7: Nu Vs Re (variable heat input and mass flow rate)

Effect of Temperature on Friction Factor

As the temperature of air inside the collector increases the friction factor decreases which is shown in figure 8. The decrease in friction factor is also due to increase in Reynolds number. The pumping power is directly proportional to the pressure drop and pressure drop is directly proportional to the friction factor. So due to decrease in friction factor the pumping power also decreases.

International Journal of Scientific & Engineering Research, Volume 8, Issue 4, April-2017
ISSN 2229-5518

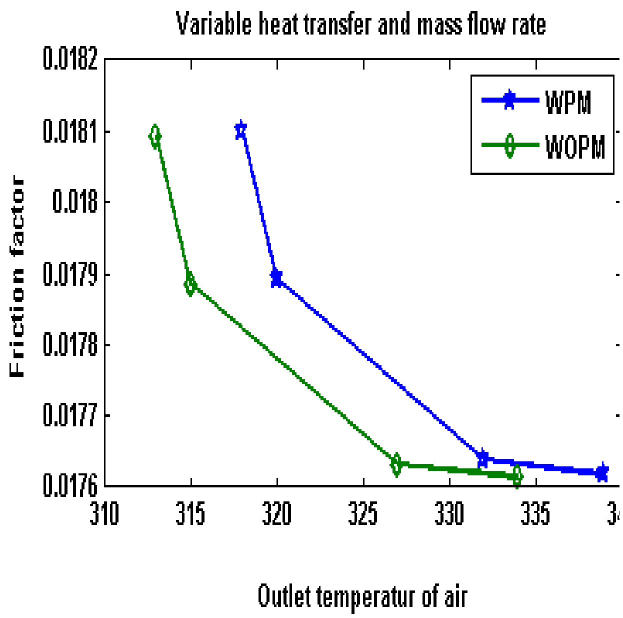

Figure. 8: Friction factor Vs ΔT (variable heat input and mass flow rate)

Effect of Changing Various Parameters on Pressure Drop The variation of pressure drop with different mass flow and light intensity for both solar air heater without porous and with porous media are shown in figures 9, 10, 11 and 12.

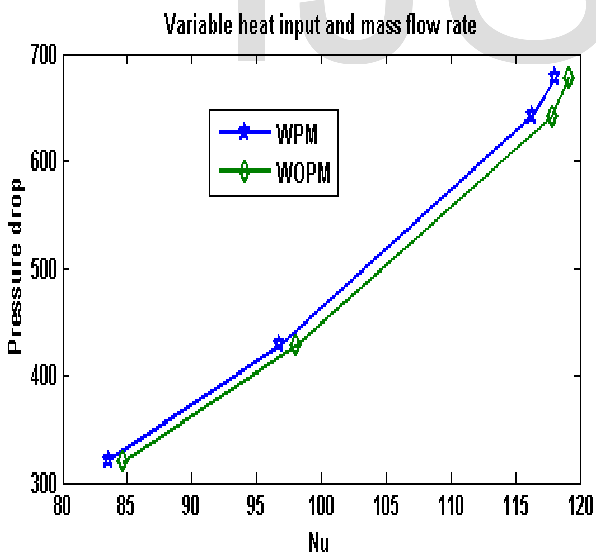

Figure. 9. Variation of Pressure drop Vs Nusselt number.

The increase in pressure drop, Reynolds number and Nusselt number is very high in case of high mass flow rate solar air heater without porous and with porous media. The use of porous media in solar air heater increases the system efficiency and outlet temperature. At lower pressure drop the efficiency of the system is higher. This increase in the pressure drop is higher for solar collector with porous media, which means increasing of the cost of the pumping power expanded in the collector. But this factor has no significant for low flow rates.

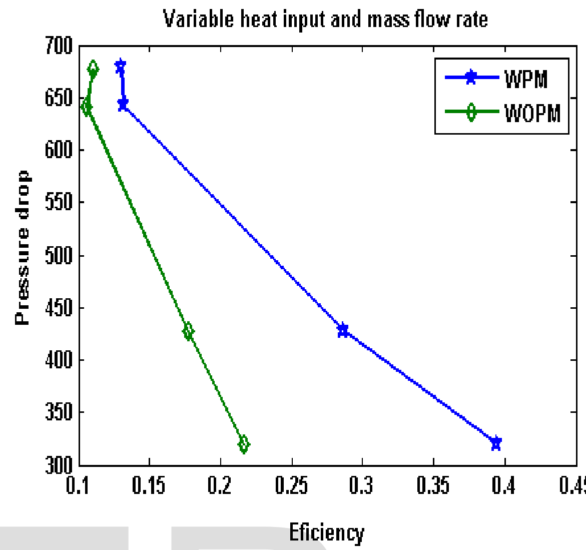

Figure.10. Variation of Pressure drop Vs Efficiency.

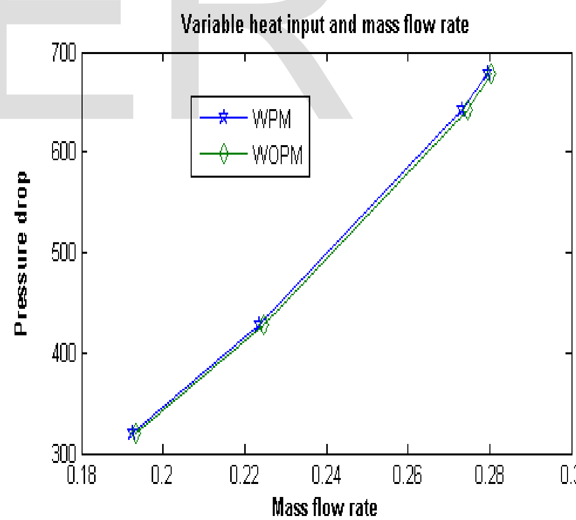

Figure. 11. Variation of Pressure drop Vs Mass flow rate.

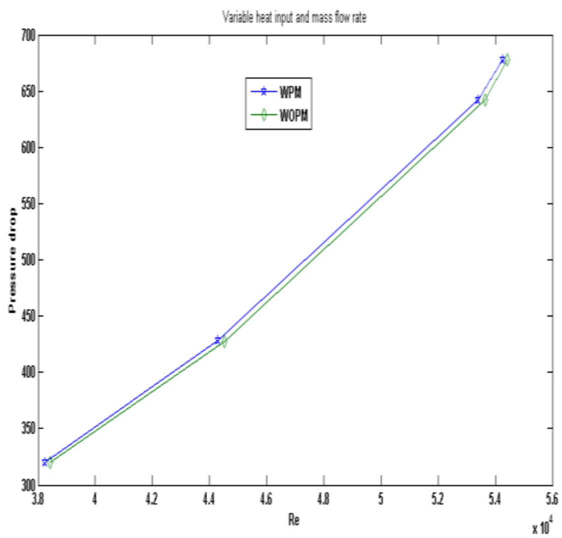

Figure. 12. Variation of Pressure drop Vs Reynolds number.

Effect of Changing Parameters on Efficiency

Figure 13 and 14 shows the variation in thermal efficiency with mass flow rate and Reynolds number. First the efficiency decreases rapidly for mass flow rate 0.18 kg/s to 0.275 kg/s and Reynolds number 38000 to 53000 then it is almost constant. This is because the useful heat gain is directly proportional to the mass flow rate and the thermal efficiency is the ratio of useful gain to the total solar radiation incident on it. The thermal efficiency is maximum for counter flow solar air heater with porous matrix. Adding a porous matrix to a counter flow heater the efficiency decreases drastically, with efficiency above 40% for normal operating range and 15% for high flow rates. The thermal efficiency of solar air heater with porous media is found to increase 5 to 20% higher than solar air heater without porous media.

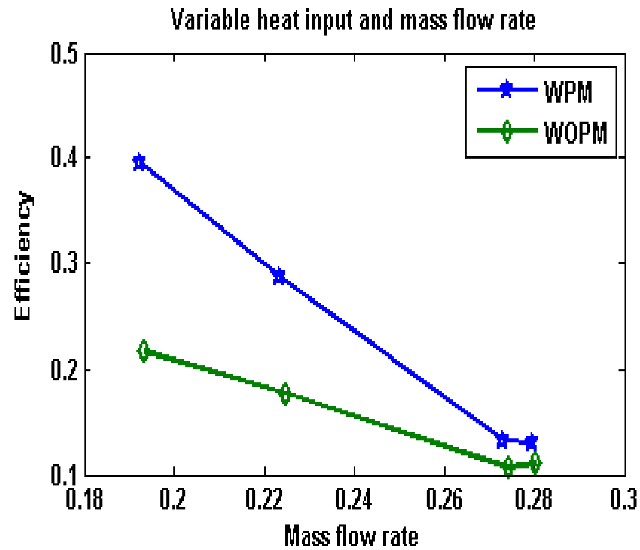

Figure. 13. Variation of Efficiency Vs Mass flow rate.

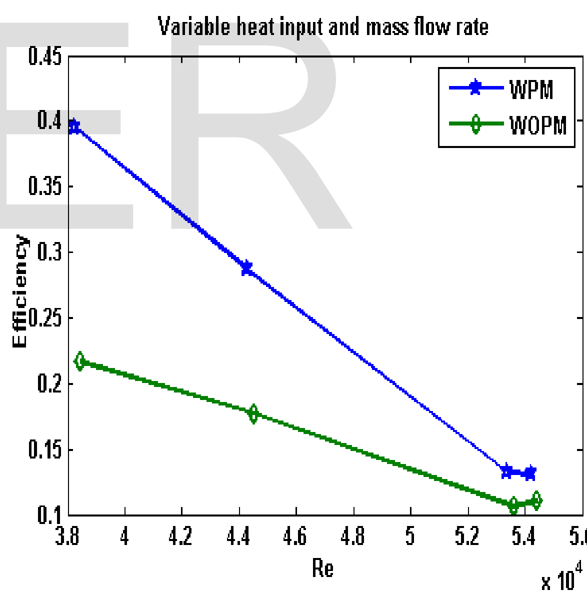

Figure. 14. Variation of Efficiency Vs Re.

Effect of Mass Flow Rate on Efficiency and Pressure Drop

Results obtained from the experiment of solar air heater without porous media and with porous media at constant mass flow rate and variable heat input are under considerations which are tabulated in Table 4 and 5.

In this experiment mass flow rate was 0.29579 kg/sec where as Reynolds number is found to be maximum 58890. The air was heated, however the maximum temperature difference for porous media is 33ºC and for without porous media it was found to be 26°C.The heat retentiveness of the collector was relatively poor as compare to collector having porous media. From Figure 15 it was observed that the efficiency decreases as heat input increases as the mass flow rate is constant. As the heat input increases heat losses also increases as well as heat transfer coefficient of air decreases due to increase in outlet temperature of the air which is shown in Figure 16. There is a decrease in Nusselt number because of decrease in heat transfer coefficient which is shown in Figure 17.

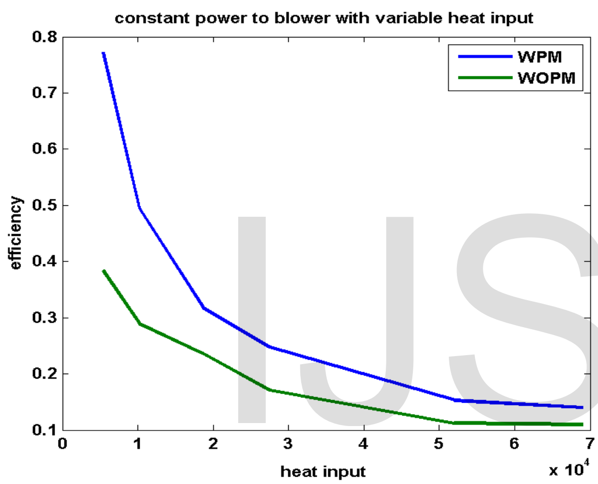

Figure 15: Efficiency Vs heat input at constant mass flow rate.

For the constant mass flow rate, the efficiency of the heater with porous media is higher than of without porous media. The maximum efficiency of the heater with porous is found to be 0.77 while solar collector without porous shows 39% maximum efficiency at same heat input.

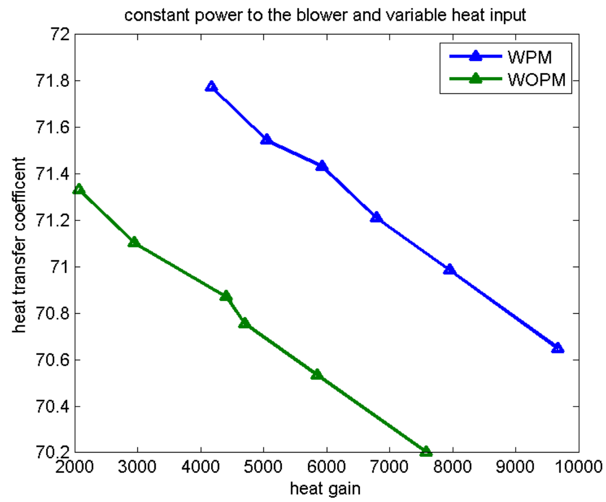

Figure 16: Heat transfer coefficient Vs heat gain at constant mass flow rate.

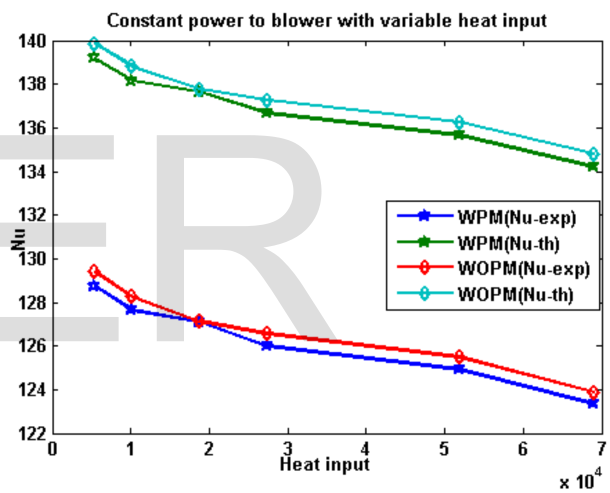

Figure 17: Nu Vs heat input at constant mass flow rate.

Pressure drop versus heat input for this small channel is plotted in figure 18. In general the pressure drop through the proposed solar air heater was not very significant compared with published data for lower mass flow rate. The obtained results shows that increase in heat input at constant mass flow rate increases the pressure drop inside the solar air heater. The difference in pressure drop is not very big, and in general, the pressure drop is higher through solar air heater with porous media.

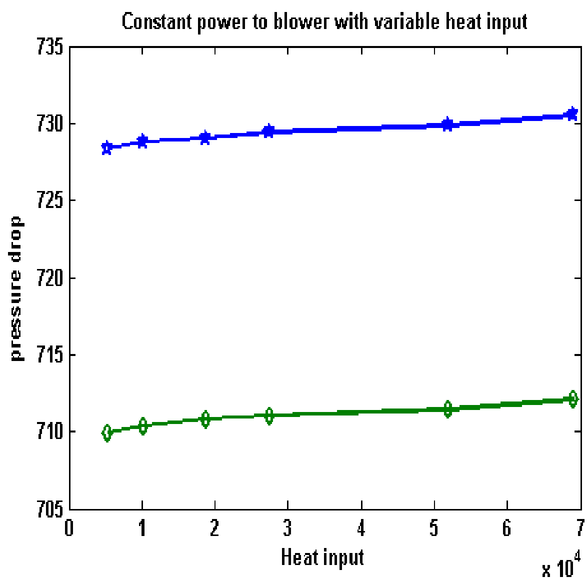

Figure 18: Pressure drop Vs heat input at constant power to the blower.

From Figure 19 heat gain by the air increases as heat input goes on increasing at constant mass flow rate. Heat gain by air with porous media is higher as compared to the solar air heater without porous.

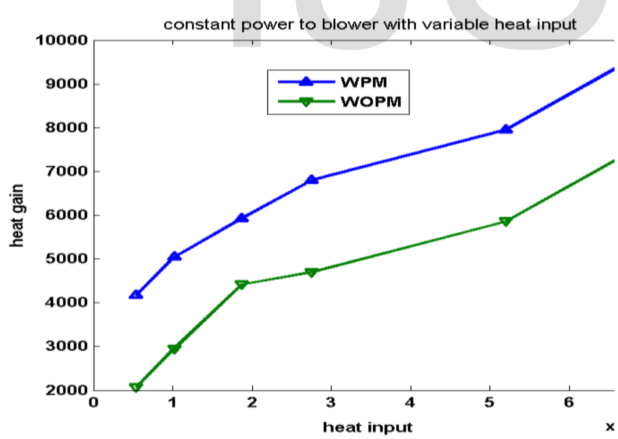

Figure 19: Heat gain Vs heat input at constant mass flow rate.

CONCLUSION

Following conclusion can be made from the present work

1. The most important result is that the absorber surface temperature decreased by 7°C as the porous layers added inside the flow channel.
2. The setup was experimented while it is clear. In this condition, the absorber temperature is at very high values 89°C and the temperature distributed non-uniformly along the depth of the channel..
3. Another benefit of reduced surface temperature is it decreases the heat loss between the surface and surroundings by 25%.
4. After all results examined, it is obvious that the use of porous material inside a solar air heater has a positive effect on the system. The solar air heater with porous media gives 20% higher thermal efficiency that of without porous media.
5. The pressure drop (pumping power) for solar air heater with porous media is higher, but this factor has no significance for high flow rates.
6. The solar air heater with porous media gives 20% higher thermal efficiency than that without porous media. The thermal conductivity of porous media has significant effect on the thermal performance of solar air heater at higher mass flow rate. It was found that the thermal efficiency is maximum at lower mass flow rate. The thermal efficiency of solar air heater with porous media is more than solar air heater without porous media of same construction cost.
7. The major drawback of solar air heater without porous media is the poor heat convection from the absorber to the stream at higher mass flow rate. Whereas, the performances of solar air heater with porous media minimizes heat loss to the atmosphere and maximizes heat transfer to the airstream.

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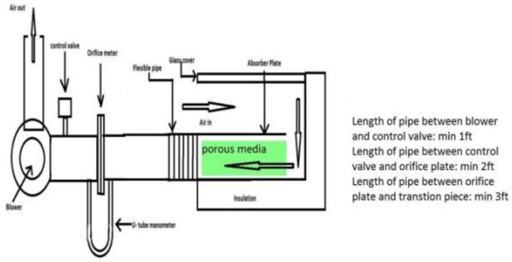

Figure A1.The schematic diagram of the experimental setup

Figure A2. Experimental Setup

TABLE A1 : Various input parameters and constants

S.No.	Input Parameters	Values
1.	Ambient Air Temperature, T_a(K)	304,306
2.	Ambient Air Pressure, P_a(bar)	1
3.	Specific gas constant for air(J/Kg K)	287
4.	Dynamic viscosity for air, μ(kg/ms)	$1.8X10^{-5}$
5.	Specific heat for air, c_p(J/kgK)	1007
6.	Thermal conductivity for air, k_f(W/mk)	0.02624
7.	Thermal conductivity of media, k(W/mK)	386.0
8.	Emissivity of glass covers, ε_c	0.92
9	Emissivity of absorber plate,ε_p	0.92
10	Tranmissivity of glass cover and absorber,τ_c *and* τ_p	0.92
11	Absorptivity of glass cover,α_c	0.06
12	Absorptivity of absorber,α_p	0.92
13	Stephan-Boltzmann constant, σ (W/m^2K^4)	$5.67X10^{-6}$
14	Intensity of solar radiation,(W/m^2)	900-1100
15	Effective thermal conductivity,k_{eff}(W/mK)	0.3
16	Air mass flow rate, m(kg/s)	0.18-0.27
17	**Porous medium(steel wool) porosity**	0.8

TABLE 2. Performance of solar collector without porous media for variable mass flow rate and heat input

Sr. No.	ΔP	ΔT	Heat Gain	h	Nu (Th)	Nu (Exp)	η
	Pascal	c°	W	W/m^2K			
1	319.9	6	1169	46.94	84.681	99.02	0.22
2	427.8	8	1810	54.51	98.062	111.3	0.18
3	605.7	11	2971	65.08	116.43	127.9	0.16
4	623.8	16	4373	65.86	117.17	128.3	0.16
5	641.8	20	5529	66.61	117.86	128.8	0.11
6	677.7	27	7626	68.06	119.13	130.1	0.11

TABLE. 3 Performance of solar collector with porous media for variable mass flow rate and heat input

Sr. No.	ΔP	ΔT	Heat Gain	h	Nu (Th)	Nu (Exp)	η
	Pascal	c°	W	W/m^2K			
1	320.01	11	2132.5	46.714	83.56	98.475	0.3949
2	427.99	13	2926.9	54.251	96.78	110.73	0.2869
3	605.81	16	4308.2	64.882	115.4	127.47	0.2304
4	624.01	21	5712.6	65.549	115.7	127.6	0.2077
5	642.04	25	6878.2	66.295	116.4	128.15	0.1323
6	677.9	32	9008.9	67.838	118.1	129.61	0.1306

TABLE 4. Performance of solar collector without porous media for constant mass flow rate

Sr. No.	ΔP	ΔT	Heat Gain	Nu (Exp)	Nu (Th)	h	H
	Pascal	c°	W			W/m^2K	
1	709.96	7	2072	141.16	139.87	71.33298	0.384
2	710.39	10	2951	139.95	138.82	71.1022	0.289
3	710.82	15	4412	138.75	137.79	70.87067	0.236
4	711.03	16	4698	138.16	137.27	70.75462	0.171
5	711.45	20	5854	137.02	136.27	70.53454	0.113
6	712.07	26	7575	135.32	134.79	70.20313	0.11

TABLE 5. Performance of solar collector with porous media for constant mass flow rate.

Sr. No.	ΔP	ΔT	Heat Gain	h	Nu (Th)	Nu (Exp)	η
	Pascal	c°	W	W/m^2K			
1	728.35	14	4170	71.77	139	128	0.77
2	728.78	17	5047	71.54	138	127	0.49
3	728.99	20	5928	71.43	137	127	0.31
4	729.42	23	6796	71.20	136	126	0.24
5	729.85	27	7953	70.98	135	124	0.15
6	730.49	33	9675	70.64	134	123	0.14