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Study of Forced Convection Heat Transfer in a Channel Partially Filled with Porous Media Connects With Automotive

A Graduation Project is Submitted to the Automobile Engineering Department in Partial Fulfilment of the Requirements for the Degree of Bachelor of Science in Automobile Engineering for the year 2022-2023

by

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بِسْمِ ٱللَّهِ ٱلرَّحْمَٰنِ ٱلرَّحِيمِ

وَفَوْقَ كُلِّ ذِي عِلْمٍ عَلِيمٌ

SUPERVISOR CERTIFICATION

I certify that the preparation of this project entitled **“Study of Forced Convection Heat Transfer in a Channel Partially Filled with Porous Media Connects With Automotive.”** prepared by :

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&

Radhi Mohammed Radhi

Under our supervision in the Automobile Engineering Department of the University of Babylon in partial fulfilment of the requirements for the degree of Bachelor of Science in Automobile Engineering.

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I am deeply grateful to my parents and family members for their support, appreciation, encouragement, and keen interest in my academic achievements.

Dedication

To my dear father

The symbol of affection

To my dear mother

The spring of love

To my brothers

The spirit of life

To my friends

I dedicate this modest effort

Project Students

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Abstract

In this project, a numerical study is conducted to show the heat transfer within a pipe filled with porous media, and investment in this project is to develop the heat transfer efficiency in car radiators. The numerical part includes building a computer simulation code by using ANSYS. Two different configurations of porous media were studied, circular and uncircular porous media and these two cases were investigated at different Reynold numbers (1500, 1800, and 2000), velocity, temperature, and pressure. However, Re effect on its performance and design areas and at fixed heat flux the temperature distribution increases with the decreasing of Reynold number because the increasing Reynolds number yields faster flow through the porous media. Furthermore, the change in porous media configuration between circular and un-circular shape led to the temperature distribution inside un-circular porous media is given a sharper than circular porous media carve which means that un-circular porous media gives less efficient heat transfer phenomena.

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List of symbol

Symbol	Name
$\boldsymbol{F}$	Body force
$\boldsymbol{\eta}$	Dynamic viscosity
$\boldsymbol{u}$	Velocity
$\boldsymbol{\eta}$	Viscosity
$\boldsymbol{\rho}$	Density
$\boldsymbol{T}$	Temperature
$\boldsymbol{k_{eff}}$	Effective heat transfer coefficient

CHAPTER ONE

INTRODUCTION

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INTRODUCTION

1.1 GENERAL

Heat exchangers are defined as equipment that effect the transfer of thermal energy in the form of heat from one fluid to another. The simplest exchangers involve the direct mixing of hot and cold fluids. Most industrial exchangers are those in which the fluids are separated by a wall. The latter type, referred to by some as a recuperator, can range from a simple plane wall between two flowing fluids to more complex configurations involving multiple passes, fins, or baffles. Conductive and convective heat transfer principles are required to describe and design these units; radiation effects are generally neglected [1].

Heat exchangers for the chemical, petrochemical, petroleum, paper, and power industries encompass a wide variety of designs that are available from many manufacturers. Equipment design practice first requires the selection of safe operable equipment. The selection and design process must also seek a cost-effective balance between initial (capital) installation costs, operating costs, and maintenance costs. The proper application of heat exchange principles can significantly minimize both the initial cost of a plant and the daily operating and/or utility costs. Each heat exchange application may be accomplished using many types of heat exchange equipment. To perform these applications, their design and materials of construction must be suitable for the desired operating conditions; the selection of materials of construction is primarily influenced by the operating temperature and the corrosive nature of the fluids being handle [1].

1.2 RADIATOR

A radiator is a heat exchanger which absorb heat from a hot coolant that passes through it and directs the collected heat to the air that blown in by the fan [2]. It is made of many tubes which are mounted in parallel arrangement where the coolant will flow from the inlet to the outlet. Heat is extracted from the tubes by the fins on the radiator surface and transfer it to the air that flows in through the radiator. The amount of heat extraction depends highly on the difference between temperature of the fluid that runs through the radiator and tubes. Radiator is made up of three main portions such as inlet tank, core, and outlet tank. Core mainly consist of two sets of passage which are set of tubes and sets of fins [3]. Tubes will be the flow passage for the coolant while air will flow between the fins. Heat is transmitted to the fins by the tube that obtained it from the hot coolant. Air that flows between the fins will eventually pickup and carries the heat away [4].

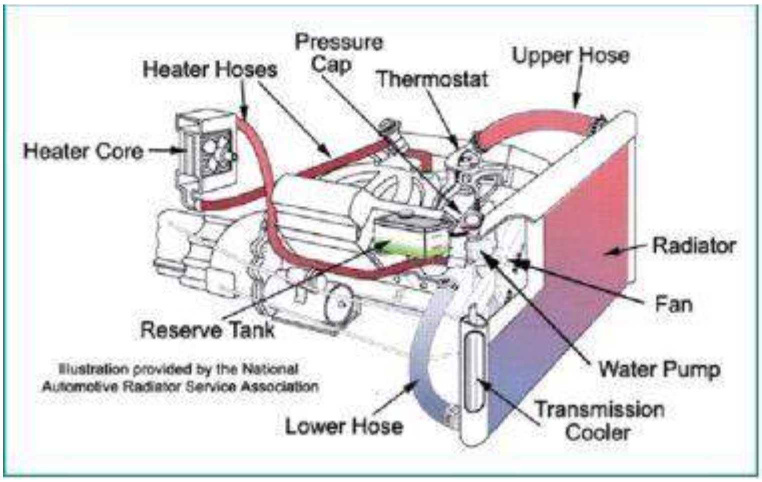

Fig. 1. Parts of cooling system in car [3].

1.3 TYPES OF CAR RADIATORS

Radiators are most commonly made out of aluminum, automobile radiators utilize a cross-flow heat exchanger design. The two working fluids are generally air and coolant

(50-50 mix of water and ethylene glycol). As the air flows through the radiator, the heat is transferred from the coolant to the air. The purpose of the air is to remove heat from the coolant, which causes the coolant to exit the radiator at a lower temperature than it entered at. The benchmark for heat transfer of current radiators is 140 kW of heat at an inlet temperature of 95 °C. The basic radiator has a width of 0.5-0.6 m (20-23“), a height of 0.4-0.7 m (16-27”), and a depth of 0.025-0.038 m (1-1.5”). These dimensions vary depending on the make and model of the automobile [5].

1.3.1 Construction And Material

- There are two materials used for making radiators they are copper and aluminium. Copper is a better conductor of heat than aluminium. But the problem in these radiators is only tubes and fins are made from copper and they are joined together with lead which has very poor heat transfer capabilities. The end tanks are made from brass and the side channels are of steel. These all reduces heat transfer capacity of copper radiator [6].
- The aluminium radiator is 100% aluminium furnace brazed without any insulating solder. This is the reason why aluminium radiators perform better than copper ones and is best material for generator radiators also [6].

1.3.2 Types of Radiator Coolant

Coolant can be defined as a fluid that prevents overheating of a device by flowing through it to gather excessive unwanted heat produced by the device. The collected heat will be transferred to another device that will exploit or disperse it. A coolant with high thermal capacity, low-cost, low viscosity and is chemically inert whereby it does not cause or promotes corrosion of the cooling system is considered as an ideal coolant [24]. Besides that, a coolant is normally chemically combined with a high boiling point liquid to form a compounded fluid. This compounded fluid function as an antifreeze agent against extremely cold conditions and as well as solves the problem of overheating during hot weather. A coolant with relatively high boiling temperature can cool faster as the engine gets hotter. During an operation of an internal combustion engine, about a third of heat energy produced are considered as unwanted heat that ends up in the cooling system. Besides that, it is observed that conventional fluids are unable to meet the increasing demand for cooling in high energy applications including automobile engines. In view of this, a new technique is needed to improve the existing cooling performance of heavy vehicle engines. An engine is prevented from getting damaged when the radiator coolant elevates the boiling point of the water which permits it to transmit more heat away from the engine [7].

- **Water**

Water is a very functional fluid to be used as radiator coolant. It is cheap, possesses good heat-transferring qualities and readily available. It possesses high specific heat capacity which enables it to be an effective thermal transitions medium between engine materials and radiators. This allows water to avoid any thermal overloads resulting from excessive component temperature. Water is categorized as an ideal coolant because of its ability to absorb and release heat efficiently. Apart from that, water is a liquid with low viscosity where it can flow easily. Thus, this characteristic permits water to be used commonly as radiator coolant. However, water has a very low boiling point of 373 K. Since the temperature in a radiator can exceed 373K, this can cause water to vaporize. Loss of coolant can create gas pockets or voids in the water jackets that can cause localized hot spots and implosion. Since water freezes at 273K, this reduces its efficiency in circulating as a coolant in radiator [8]. Thus, water cannot be used as radiator coolant in countries with winter season as it will eventually freeze up and lead to difficulty of starting up the car or causing serious engine damage.

- **Ethylene Glycol**

An organic compound (IUPAC name: ethane-1, 2-diol), Ethylene Glycol is widely used as an automotive antifreeze. It is colourless and odorless in its pure form but Ethylene Glycol is extremely dangerous and any ingestion can result in death. This is mainly due to its high toxic properties. Ethylene Glycol marketed as antifreeze and it can be used during summertime as well as during cold weather because of its higher boiling points . The main drawback of these coolants is that they are very toxic and can be dangerous to humans, animals and the environment [9].

- **Propylene Glycol**

Propylene Glycol is considerably less toxic antifreeze compared to Ethylene Glycol. Propylene Glycol is utilized as antifreeze where the Ethylene Glycol usage would be inappropriate. Any exposure to heat and air causes Propylene Glycol to oxidize. This phenomenon leads to formation of lactic acid [10]. Propylene Glycol is very corrosive; thus, it need to be properly inhibited. In order, to prevent low pH attack on the system metals, Protodin is added to act as a buffer. Protodin helps to avert acid attack that causes corrosion by creating a protective skin inside the tank and pipelines. Propylene Glycol supports the formation of biological fouling that leads to increment of corrosion rate in a radiator system. Corrosion of the radiator system starts after the formation of bacterial slime. Thus, radiator system which applies Propylene Glycol should be maintained on a continuous basis. Regular monitoring should be done on inhibitor level, pH and colour. Nevertheless, routine check-up should be done on biological contamination. Colour changes of Propylene Glycol into red in colour can be used as an indication that it should be replaced [11]. Propylene Glycol are much safer to use around children and animals, and easier to dispose of than the more toxic ethylene products.

1.4 COMPONENTS OF THE INTELLIGENT COOLING SYSTEM

Three electronic cooling system components are used in the investigated intelligent cooling system. These components are an electrical fan, an electrical thermostat, and an electrical water pump. Each of these devices is described in the following section.

1.4.1 Electrical fan

For many years, engine fans operated such that when hot water entered the radiator, the fan simply rotated at maximum speed to cool the hot radiator water without consideration of the rate of cooling demand [12]. With improvements in electrical system control, it was possible to control the fan speed either at multiple fixed speeds or to operate it with continually variable speed. Many modern vehicles, including the BMW E46 and the TOYOTA Adventure 2001 use continually variable speed fans. In this investigation, a variable speed fan with three incremental fixed speeds (off, low speed or 1000 RPM, full speed or 2000 RPM) was considered [13].

1.4.2 Thermostat

In conventional cooling systems, a wax thermostat placed at the entrance to the radiator is used to open or close the coolant route through the radiator. When water temperature rises, the wax melts and expands, forcing the thermostat valve open. The wax usually begins to melt at around 80°C and as a result, the valve may open too soon, resulting in unwanted early cooling. Two thermostat modifications can be introduced to improve the control strategy [13]. These modifications are the introduction of an electrical valve or the introduction of a heated thermostat. In using an electrical valve instead of using a conventional wax thermostat for regulating flow through the radiator, a thermal sensor is used to continuously measure the water temperature. An electrical control unit (ECU) receives these temperature data and then regulates the voltage of a servo-motor electrical valve. The electrical valve throttles the flow of water through the radiator. A heated thermostat is similar to a conventional wax thermostat, but it also contains an electrical heater inside the wax. In the heated thermostat, the wax begins to melt at 120°C instead of 80°C as in the conventional thermostat, and the melting of the wax is controlled by activating the electrical heating element. The use of an electrical valve has several advantages compared to using a heated thermostat. These advantages include less thermal shock, a faster response time, better flow control, less engine fuel consumption, and lower engine emissions. However, in a malfunction situation where the electrical circuit controlling the valve does not operate properly, water temperature can rise and the engine could face catastrophic damage. In this regard, a heated thermostat is advantageous, because if a problem arises with the electronic control of a heated thermostat, the wax will still be melted by hot water (albeit, at a higher temperature than for a conventional wax thermostat) and the coolant will still flow through the radiator, preventing damage to engine. For these reasons (as well as its lower cost) a heated thermostat was chosen for use in this investigation [14].

1.4.3 Electrical water pump

In a conventional water pump, the pump shaft is directly connected to crankshaft, thus the pump's rotational speed depends entirely on the engine rotational speed and is not related to cooling demand. Alternately, water pump rotational speed can be regulated according to cooling demand, and the pump can be driven with an electrical motor. An ECU can be programmed to determine the necessary coolant flow rate, and the water pump speed can be adjusted by supplying the corresponding voltage to the electric motor. In the present work, such an electrical water pump was used [14].

1.5 RADIATOR WORKING PRINCIPLE

A radiator has a inlet and a outlet for fluid to flow. The number of passes (number of times the liquid passes in the tubes or number of pipes passing through the tube) are generally multiple to give the required heat dissipation and for the radiator to work efficiently. The hot fluid enters in the tubes from the inlet channel. The working fluid is generally 50-5- mix of water and ethylene glycol. The liquid is cooled either by natural or forced convection. The flowing air, which moves through the radiator transfers hear from coolant to air. Thus temperature of air increases resulting in reduced temperature of the coolant. This cold liquid is sent for the required application, here, to cool the engine. After cooling the engine, the liquid gains heat and its temperature rises. It is then circulated by a suitable mechanism back to the radiator for cooling and the cycle continues. The tubes are now-a-days finned to increase the surface area for heat dissipation [6].

1.6 ENHANCEMENT OF RADIATOR HEAT TRANSFER RATE

The standard radiator used in any application is finned to increase the area of heat dissipation from the cooling liquid. Efficiency of radiator can be improved by changing its geometrical parameters like diameter of tubes or number of tubes, by varying coolant used or by varying radiator material. This type of radiator is enough or suitable for a general purpose application. With emerging advanced techniques the need of a cheap, more efficient and a economical system is required. This can and is being satisfied by constant study and consequently improving the system for best results. There are a few ways explained to improve the efficiency of a radiator, they are mentioned below in brief [6]:

A. By increasing the air velocity, by changing the fin configuration.
B. Use of nanofluids.
C. S-Shaped Fins
D. Use of carbon-foams fins
E. Increasing the turbulences of coolant
F. Use of porous media

A. By Increasing Air Velocity

Air velocity can be increased by creating a nozzle. The nozzle can be created in various ways, one maybe by using two flat plates. This nozzle helps in increasing velocity and hence pressure decreases. This decreases temperature as pressure is directly proportional to temperature [6].

B. Use of Nano Fluids

Nanofluids are fluids which consist of a base fluid with nano sized particles (1-100 nm) suspended within them. These particles, generally a metal or metal oxide, increase conduction and convection coefficients, allowing for more heat transfer out of the coolant. Nanofluids are more stable and can be used in real world applications, due to the recent progress in the field. These properties would be very beneficial to allow for an increased amount of heat to be removed from the engine. This is useful in such a way that, it may permit the fluid to sustain higher loads for cooling. However, these nanofluids do not show considerable improvement in heat transfer when used with current radiator designs. This is because there are several limitations to current radiator designs [15].

C. S-Shaped Fins

Numerical studies have showed that the fin shape affects the thermal-hydraulic characteristics of the radiator with S-shaped fins. Various fin parameters were studied. Narrower fins produce more heat transfer area per unit volume but worsen the fin efficiency more than the wider fins. It is seen that, the largest heat transfer rate was showed by the narrowest fins. A longer fin length reduces the stream bend and pressure drop that occurs because of the stream bend. Fin roundness at the head and tail edge of the fins minimally affect the heat transfer performance but greatly affect the pressure drop performance. From the real fin shape manufactured by chemical etching, the pressure drop is increased by about 30%. To obtain minimum pressure drop, it would be wise to choose the fin roundness with the minimum value [6].

D. Increasing Turbulence of Coolants

It is observed that higher turbulence will increase the radiator effectiveness. After investigating the heat transfer in plain fin arrangements was, to determine the influence of corner radii of bent metal sheets of the ribs, the Reynolds number range extended from 500 to 3000, and at Re=2000, transition from laminar to turbulent flow was observed. The ducts with the smallest radii resulted in the highest Nusselt number for a given Reynolds number. However, a comparison of the investigated geometries in terms of the volume [6].

E. Use of Carbon Foam Fins

Replacing aluminium fins with carbon foam channels, is a technique which can be employed for improving the radiator working. Due to the thermal properties of carbon foam (k = 175-180 W/mK for carbon foam with 70% porosity), along with increasing the amount of heat rejected, we will be able to reduce the overall size of the radiator while simultaneously increasing the surface area exposed to the air, thus reducing the air side resistance. The carbon foam has channels in a corrugated pattern. This corrugation channels air into the slots and forces the air through the carbon foam.

Many parallelly arranged tubes, provide support for the carbon foam as well as contain the necessary volume of coolant. The end caps are made out of aluminium and also provide structural support and mounting locations. Overall, this design concept is a simple design which will meet most of our customer requirements [6].

D. Use of porous media

The industries are always looking for ways to increase the heat transfer rate. With better heat transfer rate, the heat exchanger size can be reduced, and it makes the heat exchanger more economical to manufacture or use. For this reason, many researchers have tended to investigate this issue. For making the heat transfer process more efficient, there are a lot of techniques that can be generally categorized as passive or active methods. Passive methods do not need any external energy source while active methods do. Due to the complexity of active methods and their limitation for not being functional without an external energy source, passive methods are more common. Implementing fins, fluid additives, swirls flow devices, and porous mediums can be placed in this category. Implementing porous media expanded surface such as planes or fins make the heat transfer better due to enlarging the effective surface [16].

1.7 THE SCOPE OF PRESENT STUDY

The Aims of present study are:

1- Study the types of heat exchangers and radiator used in different car.
2- Investigate the ability of porous media to increase heat transfer rate.
3- Presents the previous study on coolant system and how to improve the efficiency of heat exchanger.

CHAPTER TWO

PREVIOUS STUDY AND LITERATURE REVIEW

CHAPTER TWO

PREVIOUS STUDY AND LITERATURE REVIEW

1.2 INTRODUCTION

The porous material is a present promising technique for a thermal system. The tortuous path of the porous media has large contact with fluids giving rise to heat transfer and flow mixing enhanced. Flows through a porous medium in a channel has been practical applications such as LED backlight module cooling system, heat exchangers, electronic component cooling system, thermal insulation, nuclear reactors, drying processes, and so on [17].

2.2 THEORETICAL AND EXPERIMENTAL STUDY FOR CONVECTION HEAT TRANSFER IN POROUS MEDIA

M. K. ALKAM and M. A. AL-NIMRt [18], in this paper a numerical simulation is presented for the transient forced convection in the developing region of a cylindrical channel partially filled with a porous substrate. The porous substrate is attached to the inner side of the cylinder wall, which is exposed to a sudden change in temperature. The flow within the porous domain is modelled by the Brinkman-Forchheimer-extended Darcy model. The effects of several parameters on the hydrodynamic and thermal characteristics of the present problem are studied. These parameters include the porous substrate thickness, Darcy number and Forchheimer coefficient. Results of the current model show that the existence of the porous substrate may improve the Nusselt number at the fully developed region by a factor of 8. However, there is an optimum thickness of the porous substrate beyond which no significant improvement in the Nusselt number is achieved. Also, in this work, the macroscopic inertial term in the porous domain momentum equation is included due to its significant effect. It is found that the steady state time increases as the substrate thickness increases up to a certain limit and then the steady state time decreases upon further increase in the substrate thickness. Also, increasing the Forchheimer coefficient and decreasing the Darcy number increase the steady state time. Fig (2.1), shows Schematic diagram of the problem studied in this research.

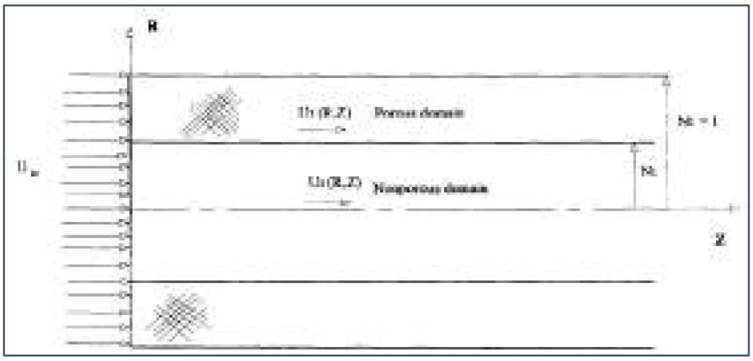

Fig. (2.1): Schematic diagram of the problem studied in this research [18].

Horng-Wen Wu and Ren-Hung Wang [17] performed a numerical study made of the unsteady flow and convection heat transfer for a heated square porous cylinder in a channel. The general Darcy–Brinkman–Forchheimer model is adopted for the porous region. The parameters studies include porosity, Darcy number, and Reynolds number on heat transfer performance. The results indicate that the average local Nusselt number is augmented as the Darcy number increases. The average local Nusselt number increases as Reynolds number increases; in particular, the increase is more obvious at a higher Darcy number. In contrast, the porosity has slight influence on heat transfer.

Kem et al., [19] study a force convection in circular tube with partially filed porous media. The Brinkman–Forchheimer of Darcy model was used to analyze the temperature distribution. This study involved the modeling of two porous media (1) a layer attaches to the tube wall extending inward towards the centerline and (2) a layer at a centerline extending outward. The effect of several parameters has been studied such as Darcy number, effective viscosity, effective thermal conductivity, inertia parameters and the effect of geometric parameters. Fig (2.1) illustrates the schematic of the duct and the test section.

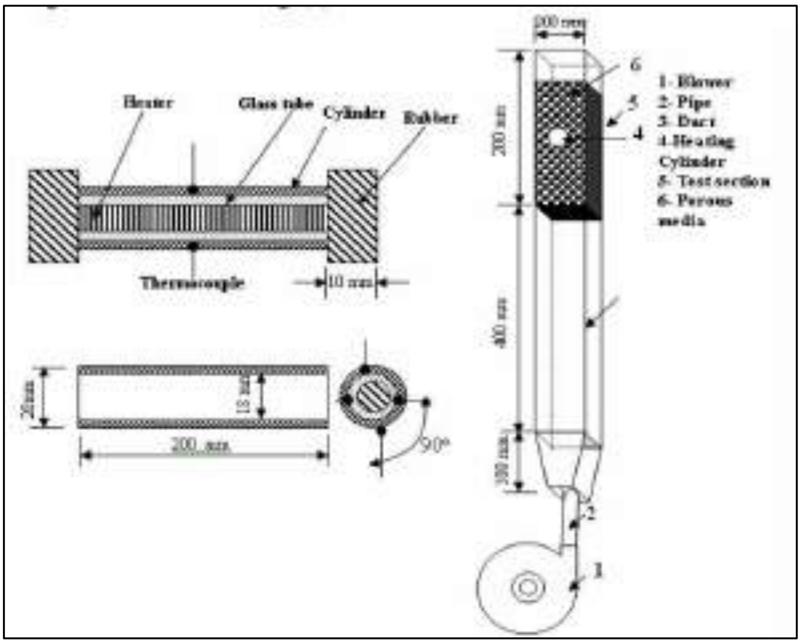

Fig. (2.1): The schematic of the duct and the test section [18].

Tarawnch et al., [20] had experimentally investigate the two-phase laminar forced convection in a single porous tube heat exchanger is presented. The effect of Darcy, Reynolds, and Prandtl numbers on the performance of this heat exchanger during the condensation process of carbon dioxide at different test conditions were investigated. Gravel sand with different porosities is used as a porous medium. The flow in the porous medium is modeled using the Brinkman–Forchheimer-extended Darcy model. Parametric studies are also conducted to evaluate the effects of porosity, Reynolds and Prandtl numbers on the heat transfer coefficient and the friction factor. The study predicted the combined effect of the Reynolds number, Darcy number, porosity, and Prandtl number on the heat transfer and pressure drop of carbon dioxide during the condensation process in a porous medium.

Ghorab [21] study the mixing convection flow inside a convergent horizontal channel partially filled with porous material and a clear channel are investigated numerically in the present study. Four discrete heat sources with uniform heat flux have been applied on the bottom surface of the channel. This study carried out the effect of the channel exit height, Richardson number, Reynolds number, Darcy number and Prandtl number on the flow-field, the Nusselt number and the overall heat transfer performance. The Brinkman–Forchheimer–extended Darcy model is used to solve the governing equations of the fluid in the porous medium. The results reveal that the boundary layer thickness and flow velocity increase at high Richardson number for both porous and clear channels. The overall Nusselt number increases significantly for further increase in Darcy number. Furthermore, Richardson number has a small significant effect on overall Nusselt number and heat transfer performance at low Prandtl number.

N. Guerroudj and H. Kahalerras [22] in this work a numerical simulation of laminar mixed convective in a two-dimensional parallel plate channel provided with porous blocks of various shapes. The upper plate is thermally insulated while the blocks, heated from below, are attached on the lower one. The Brinkman–Forchheimer extended Darcy model with the Boussinesq approximation is adopted for the flow in the porous regions. The governing equations with the appropriate boundary conditions are solved by the control volume method. The influence of the buoyancy force intensity, the porous blocks shape going from the rectangular shape to the triangular shape, their height, the porous medium permeability, the Reynolds number and the thermal conductivity ratio is analysed. The results reveal essentially that the shape of the blocks can alter substantially the flow and heat transfer characteristics. Fig (2.2) shows Streamlines for Re, Da and Hp in this study.

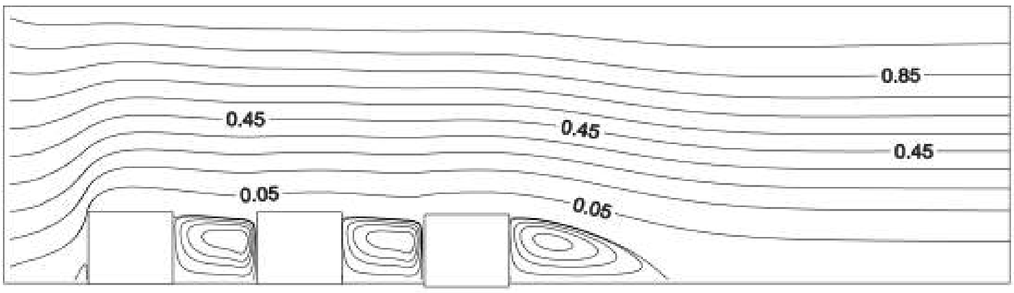

Fig. (2.2): Streamlines for Re = 500, Da = 10?5 and Hp = 0.25.

Ghorab [23], this study investigates numerically forced convection heat transfer and flow analyses of a passive heat exchanger for nonporous and partially filled porous channels with varying exit height (1, 0.5, and 0.25). Four discrete heat sources with uniform heat flux are simulated on the channel bottom wall. The partially filled porous channels are tested at two different porous block heights (0.5 and 1). The flow field and thermal analyses inside the channels are investigated across a wide range of Reynolds and Darcy numbers for Prandtl number of 0.71. The results reveal that the porous block and the exit height affect substantially the flow and heat transfer characteristics inside the tested channels. The Nusselt number is enhanced by 20–40% for the partially filled porous convergent channel (exit height = 0.25 and porous block height = 1) compared to the nonporous channel. Consequently, the heat exchanger size can be reduced by 37.5%. Moreover, the overall heat transfer performance parameter is enhanced with further increase in Darcy number at low Reynolds number.

Kelestani et al., [24] the present work investigates analytically the problem of forced convection heat transfer of a pulsating flow, in a channel filled with a porous medium under local thermal non-equilibrium condition. Internal heat generation is considered in the porous medium, and the channel walls are subjected to constant heat flux boundary condition. Exact solutions are obtained for velocity, Nusselt number and temperature distributions of the fluid and solid phases in the porous medium. The influence of pertinent parameters, including Biot number, Darcy number, fluid-to-solid effective thermal conductivity ratio and Prandtl number are discussed. The applied pressure gradient is considered in a sinusoidal waveform. The effect of dimensionless frequency and coefficient of the pressure amplitude on the system's velocity and temperature fields are discussed. Results show that the amplitudes of the unsteady temperatures for the fluid and solid phases decrease with the increase in Biot number or thermal conductivity ratio. For large Biot numbers, dimensionless temperatures of the solid and fluid phases are similar and are close to their steady counterparts. Results for the Nusselt number indicate that increasing Biot number or thermal conductivity ratio decreases the amplitude of Nusselt number. Increase in the internal heat generation in the solid phase does not have a significant influence on the ratio of amplitude-to-mean value of the Nusselt number, while internal heat generation in the fluid phase enhances this ratio.

Dickson [25], this study investigated theoretically the generation of entropy and transfer of heat during forced convection of a nanofluid through a partially filled porous channel. The problem includes a fully developed flow in a channel with a central porous insert and under constant heat flux boundary condition. The system is assumed to be under local thermal non-equilibrium and the solid and nanofluid phases can feature internal heat generations. Darcy–Brinkman model of momentum transfer along with the two-equation thermal energy transport and two different fundamental porous-fluid interface models are utilised to analyse the heat transfer problem. Analytical expressions are developed for the temperature fields, Nusselt number and, the local and total entropy generations. The subsequent parametric study reveals the strong influences of the pertinent parameters and the utilised porous-nanofluid interface models. The results show considerable increases in the Nusselt number with increasing the concentration of nanoparticles. Internal heat generations are demonstrated to have major effects on the heat transfer and entropy generation characterises of the system. Further, the existence of internal heat sources signifies the role of nanoparticles concentration in the thermal and entropic behaviours of the system. Fig (2.3) shows the configuration of the channel partially filled with a porous material.

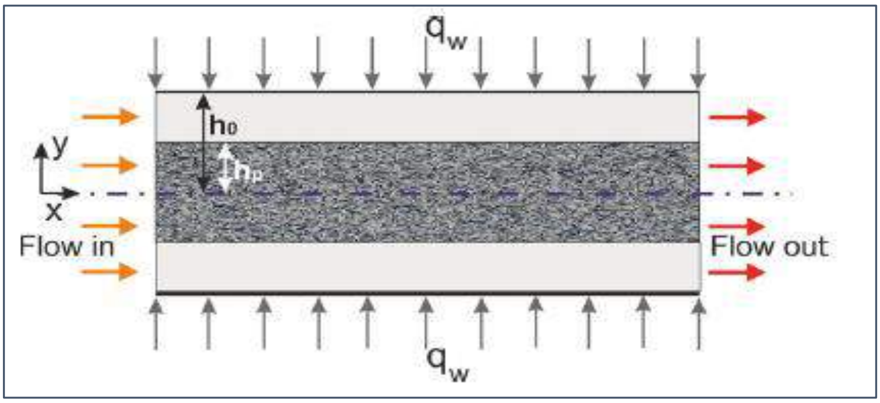

Fig. (2.3): Configuration of the channel partially filled with a porous material.

CHAPTER THREE

MATHEMATICAL MODELLING

CHAPTER THREE

MATHEMATICAL MODELLING

3.1 INTRODUCTON

One of the main components of a cooling system of an engine is a radiator. Vehicle radiators. In the last two decades, convection heat transfer in porous media has been widely studied because it has a wide range of applications in a variety of fields, such as geothermal energy, chemical engineering [26], microscale engineering [27], etc. Actually, porous media has brought significant changes to heat transfer enhancement [28]. Due to the strong competition in the automotive industry, radiators with better performance (higher cooling capacity, less hydrodynamic loss, less weight, etc.) have been desired. A common tool for the determination of thermal characteristics of vehicle radiators is experimental testing. However, experimental testing may not be feasible considering the cost and labor-time. Basic understanding of the past experimental data and analytical/computational modelling can significantly enhance the effectiveness of the design and development phase. There are techniques available to analyze HXs such a

s log mean temperature difference (LMTD) and effectiveness-NTU (ε-NTU). However, these techniques require some parameters known *a priori* such as overall heat transfer coefficients and/or NTU relations for a given HX. There are no general expressions for overall heat transfer coefficients and/or ε-NTU relations valid for any HX [29].

3.2MOMENTUM BALANCES FOR PURE FLUID (WATER)

Incompressible Navier-Stokes equation

$$-\vec{\nabla} \cdot \eta\left(\vec{\nabla}\vec{u} + (\vec{\nabla}\vec{u})^T\right) + \rho(\vec{u} \cdot \vec{\nabla})\vec{u} + \vec{\nabla} \cdot p = \overrightarrow{F} \quad (3.1)$$

$$\vec{\nabla} \cdot \vec{u} = 0 \quad (3.2)$$

Where F = Body force

η = Dynamic viscosity
u = Velocity
ρ = Density of fluid
p = Pressure

General heat transfer equation

To model heat conduction and convection through a fluid, the energy equation is used.

$$\rho C_p \frac{\partial T}{\partial t} + \vec{\nabla} \cdot \left(-k\vec{\nabla}T + \rho C_p T\vec{u}\right) = Q_s \qquad (3.3)$$

3.3 POROUS MEDIA MODELING

The other fundamental set of equations that govern fluid flow are derived from Newton's second law (the conservation of momentum). The equations are called the Darcy-equilibrium equations, and for an incompressible fluid, the full instantaneous equations take the form [31]:

$$\frac{1}{\varepsilon^2}\left[u\frac{\partial u}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial w}{\partial z}\right] = -\frac{1}{\rho f}\frac{\partial p}{\partial x} + \frac{1}{\varepsilon^2}v_{eff}\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) - \left[\frac{v_f}{k} + \frac{\varepsilon C}{\sqrt{K}}|\vec{V}|\right] \qquad (3.4)$$

$$\frac{1}{\varepsilon^2}\left[u\frac{\partial u}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial w}{\partial z}\right] = -\frac{1}{\rho f}\frac{\partial p}{\partial y} + \frac{1}{\varepsilon^2}v_{eff}\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) - \left[\frac{v_f}{k} + \frac{\varepsilon C}{\sqrt{K}}|\vec{V}|\right]v \qquad (3.5)$$

As shown in Fig. (3.1), the flow of pure fluid in tube of radius r is assumed to change in r and z direction, assume that the fluid is incompressible and no tangential flow ($v_\theta = 0$).

$$\frac{1}{\varepsilon^2}\left[u\frac{\partial u}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial w}{\partial z}\right] = -\frac{1}{\rho f}\frac{\partial p}{\partial z} + \frac{1}{\varepsilon^2}v_{eff}\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) - \left[\frac{v_f}{k} + \frac{\varepsilon C}{\sqrt{K}}|\vec{V}|\right] \qquad (3.6)$$

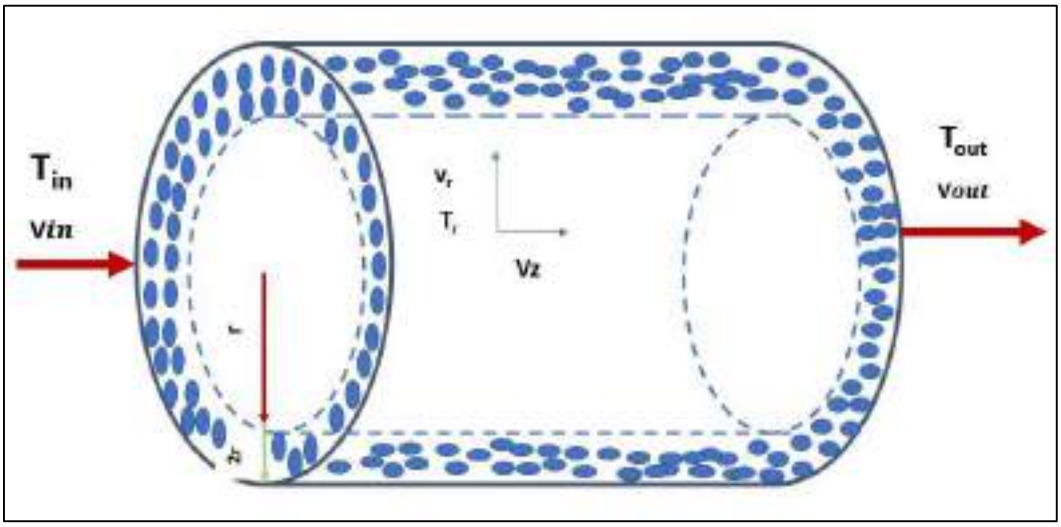

Fig. (3.1): The schematic diagram of system.

3.3.1 THE EQUATION OF ENERGY FOR POROUS NEWTONIAN FLUIDS

Assumption

- Convection heat transfer accrue in r direction only.
- The fluid had constant physical properties (density ρ and thermal conductivity *k*).

  Therefore, the equation of energy for studied system is.

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z} = \frac{k_{eff}}{(\rho C_P)f}\left(\frac{\partial^2 T}{\partial x^2}\right) + \left(\frac{\partial^2 T}{\partial y^2}\right) + \left(\frac{\partial^2 T}{\partial Z^2}\right) \qquad (3.7)$$

3.4 Boundary conditions

At z=0 (Inlet)

$$v=0 \quad (3.8)$$

$$T=T_{in} \quad (3.9)$$

$$P=P_{\circ} \quad (3.10)$$

At z=L (Outlet)

$$\frac{\partial v}{\partial z} = 0 \; or \; u = constant \quad (3.11)$$

$$T= T_{porous,} \quad (3.12)$$

$$P= P_{L} \quad (3.13)$$

At the interface

$$v_{porous}=v_{pure} \quad (3.14)$$

$$T_{porous}=T_{pure} \quad (3.15)$$

At the internal wall

$$v = 0 \quad (3.16)$$

$$T=T_{Wall} \quad (3.17)$$

3.5 Brinkman equation COMSOL 6.1.

There are two methods for solving the pure fluid momentum equations in ANSYS. One method is to use the Darcy momentum equation and the other is the usage of Brinkman momentum equations. Darcy's equations are based on this approach. When the momentum transport is dominated by shear stresses, then Brinkman equations are required to describe the problem. *"Brinkman model extends Darcy's law to include a term that accounts for the viscous transport, in the momentum balance".* The steady state, constant viscosity Brinkman equations used by

$$-\vec{\nabla}\cdot\eta\left(\vec{\nabla}\vec{u}+(\vec{\nabla}\vec{u})^T\right)+\frac{\eta}{k}\vec{u}+\vec{\nabla}p=\vec{F} \qquad (3.18)$$

In (4.3), the viscous (Brinkman) term. This term is only required to satisfy the no-slip condition on the wall. Rewriting (5.3) exclusively for pressure gradient,

$$\vec{\nabla}p=-\vec{\nabla}\cdot\eta\left(\vec{\nabla}\vec{u}+(\vec{\nabla}\vec{u})^T\right)+\frac{\eta}{k}\vec{u}+\vec{F} \qquad (3.19)$$

To accommodate the effects of form, drag in (4.2) the body force, F, is equated to.

$$F=-C_F K^{-\frac{1}{2}}\rho_f|\vec{u}|\vec{u} \qquad (3.20)$$

Therefore, the porous medium equation becomes.

$$\nabla p=-\frac{\eta}{k}\vec{u}+\vec{\nabla}\cdot\eta\left(\vec{\nabla}\vec{u}+(\vec{\nabla}\vec{u})^T\right)-C_F K^{-\frac{1}{2}}\rho_f|\vec{u}|\vec{u} \qquad (3.21)$$

CHAPTER FOUR

Results and Discussion

Chapter Four

Results and Discussion

4.1 Introduction

The problem that was defined in chapter three is solved using ANSYS software. The model is divided into two major subdomains. One such domain is where the fluid flows and porous material fills the other domain. The pure fluid is governed by the equations of the Incompressible Navier-Stokes and Brinkman application modes, respectively. the velocity was studied at Re of 1500, 1800, and 2000. A general heat transfer application mode encompasses both the fluid and porous regions.

4.2 Geometry and general description of the Pipe

The numerically studied pipe is shown schematically and photographically in Figures (4.1) and (4.2), respectively. the pipe has a Radius = 0.1 m and a length (Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m

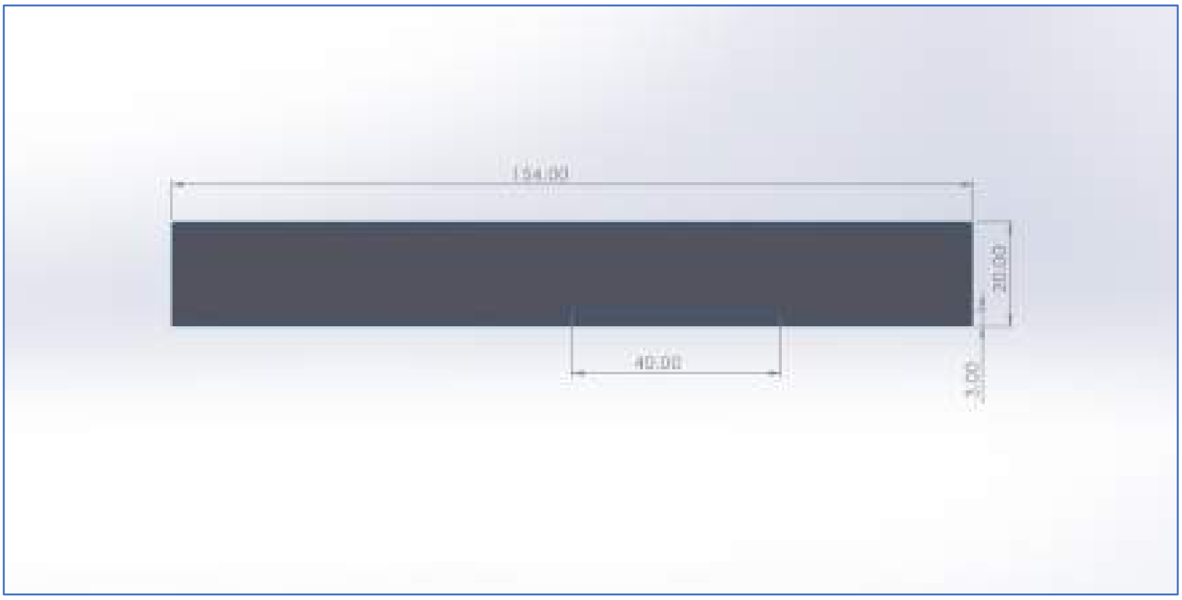

Fig. (4.1): Schematic diagram of pipe.

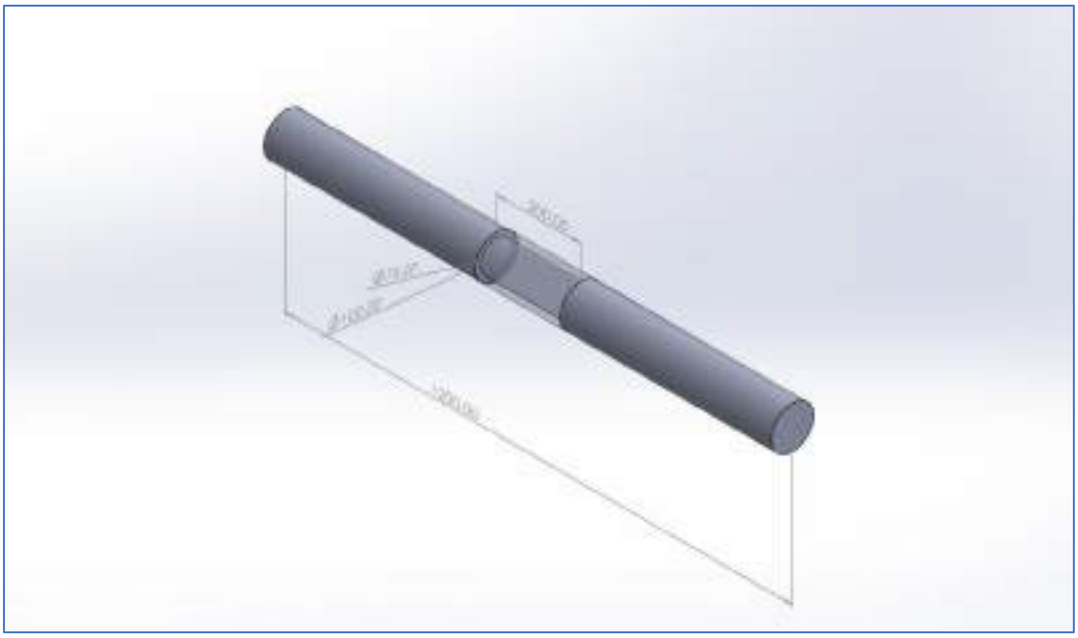

Fig. (4.2): The 3D plot of the studied pipe.

4.3 Boundary condition and initial condition

V_{in} = 0.015 m/s, Re= 1500

q^0= 10^5 w/m^2

T_{in} =297 K

Porosity = 0.6

Interoperability = 1098701.9 m^{-2}

4.4 Modelling details

For the above problem, uniform inlet velocity is specified in the x direction. The Reynolds number, based on the hydraulic diameter of the channel, is small enough for the flow to be maintained laminar (Re_{Dh} <2000). The hydraulic diameter of the 2-D channel is 2H. This application mode is used exclusively to solve energy equations. The results have been discussed for two systems (pipe with circular porous media and the other pipe for un-circular porous media.

4.5 Effect of parameter for pipe with circular porous media for 2D system

4.5.1Effect of Reynold number

The variation of the local surface temperature along the pipe for different Reynolds numbers is shown in figures (4.3 a), (4.3 b), and (4.3 c). This figure presented the lowest pipe heat flux q = 296.996 W/m2. In this figure, the local surface temperature distribution T_x demonstrated that for fixed heat flux the temperature distribution increases with the decreasing of Reynold number because the increasing Reynolds number yields faster flow through the porous media.

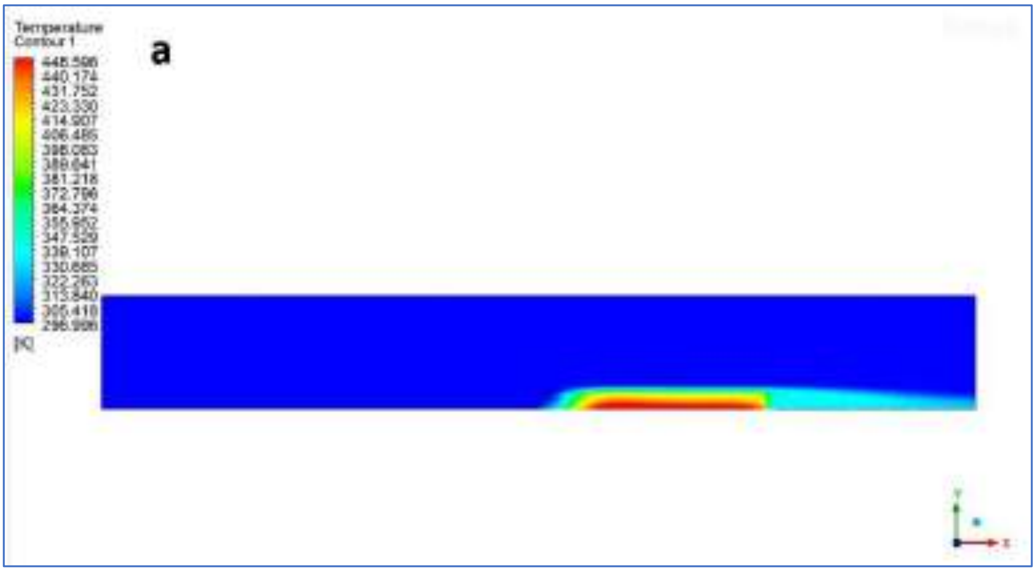

Fig (4.3 a). Temperature distribution for the 2D system with circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m

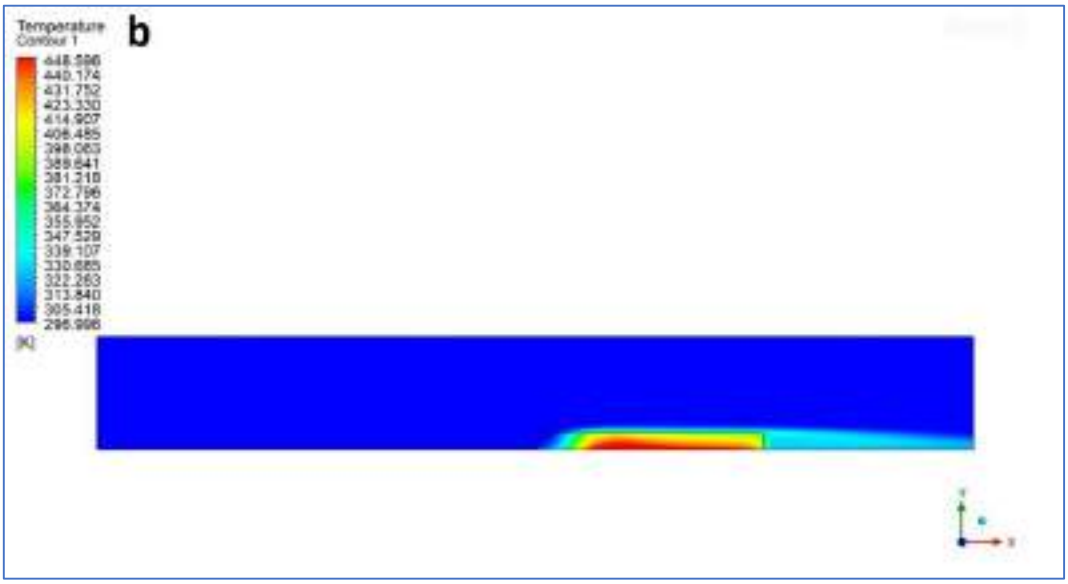

Fig (4.3 b). Temperature distribution for the 2D system with circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m

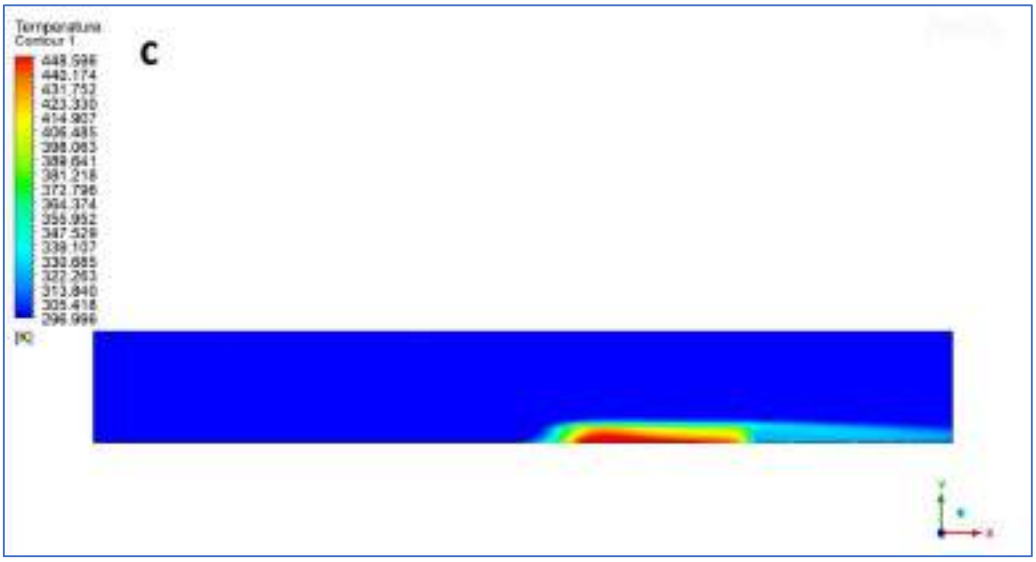

Fig (4.3 c). Temperature distribution for the 2D system with circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m

4.5.2 Effect of velocity

The variation of the local velocity along the pipe for different Reynolds numbers is shown in figures (4.4 a), (4.4 b), and (4.4 c). This figure presented the velocity increase with Reynold number and lower give more stable velocity profile. Because the increasing Reynolds number yields faster flow through the porous media and hence increases the turbulency.

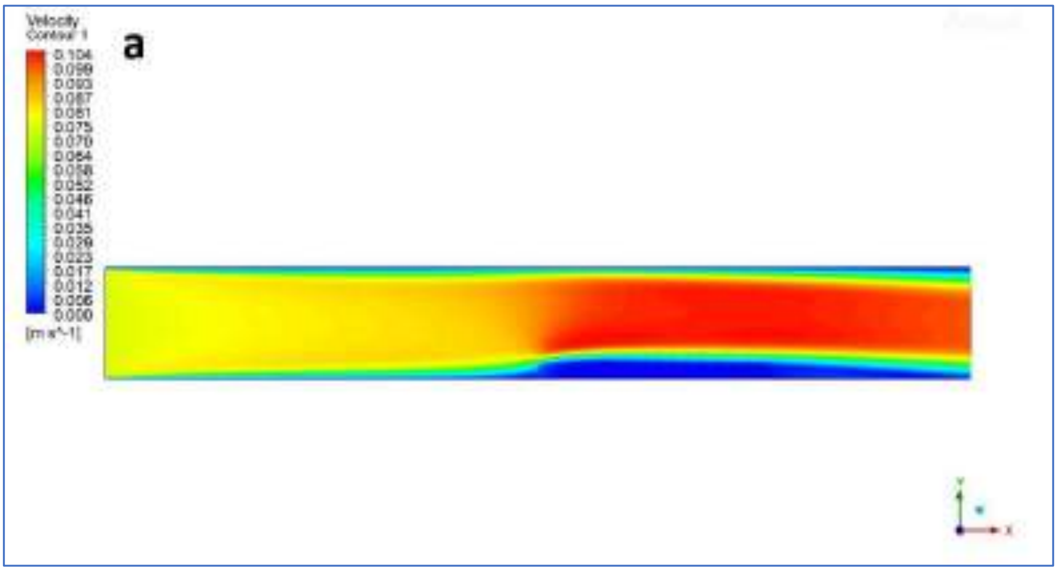

Fig (4.4 a). velocity contour for the 2D system with circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m

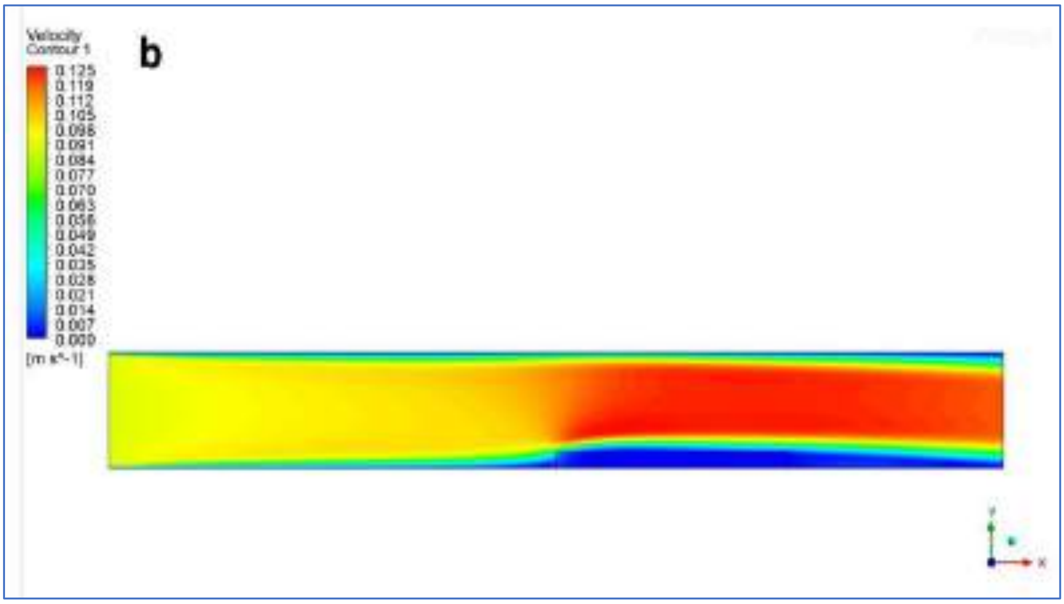

Fig (4.4 b). velocity contour for the 2D system with circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m

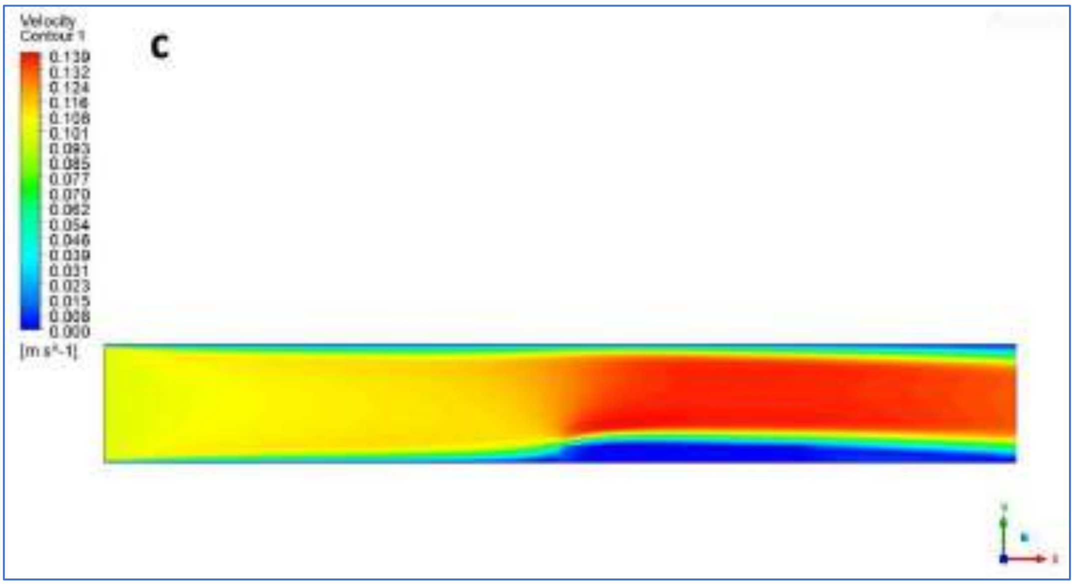

Fig (4.4 c). velocity contour for the 2D system with circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

4.6 Effect of parameter for pipe with un-circular porous media

5.6.1 Effect of Reynold number on temperature

The variation of the local surface temperature along the pipe for different Reynolds numbers is shown in figures (4.5 a), (4.5 b), and (4.5 c). This figure presented the lowest pipe heat flux q = 296.996 W/m2. In this figure, the local surface temperature distribution T_x demonstrated that for fixed heat flux the temperature distribution increases with the decreasing of Reynold number because the increasing Reynolds number yields faster flow through the porous media.

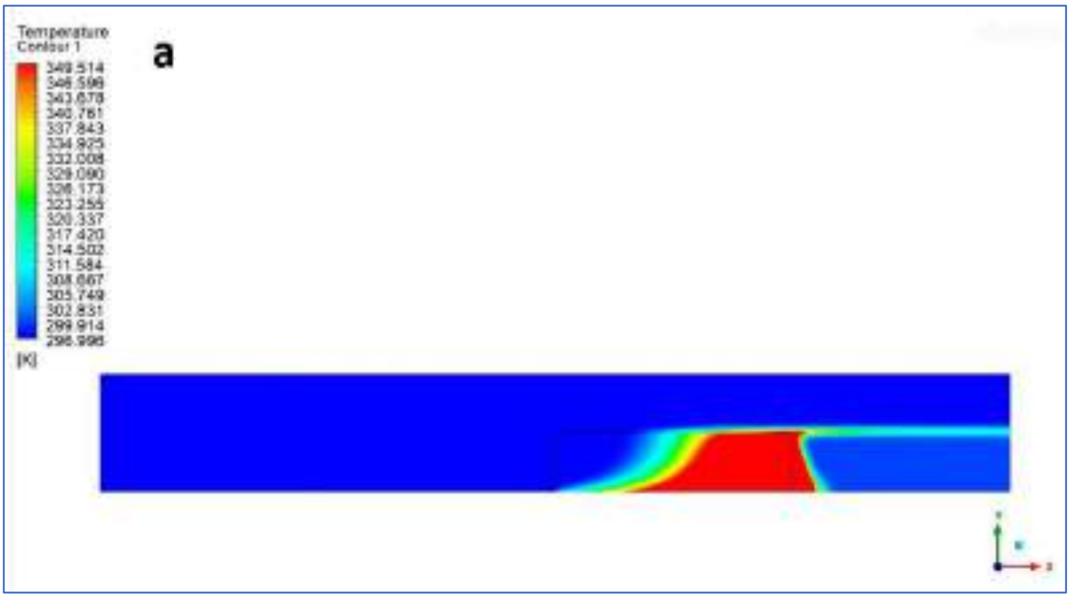

Fig (4.5 a). Temperature distribution for the 2D system with un-circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

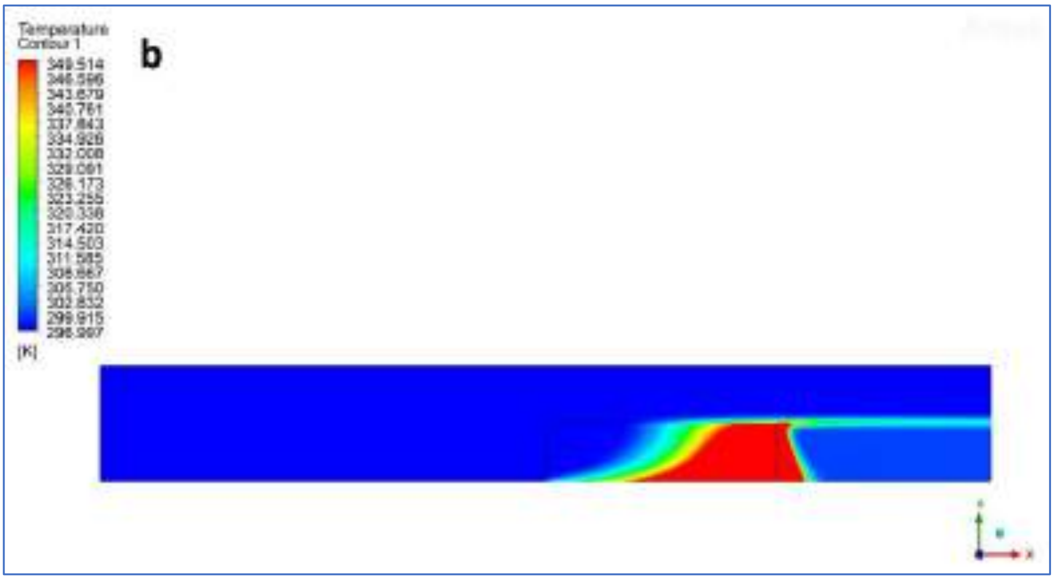

Fig (4.5 b). Temperature distribution for the 2D system with un-circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m

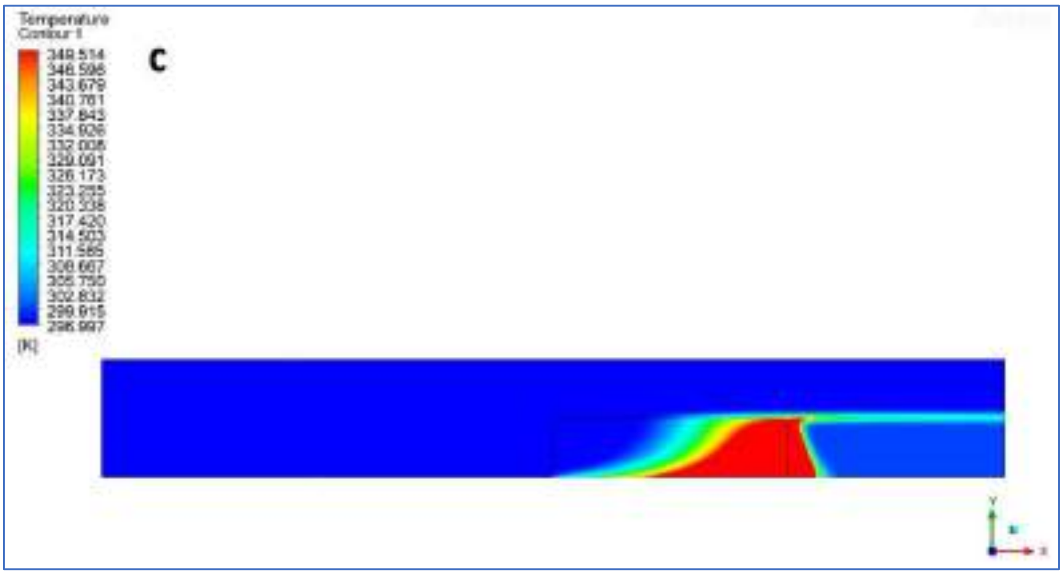

Fig (4.5 c). Temperature distribution for the 2D system with uncircular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

4.6.2. Effect of velocity

The variation of the local velocity along the pipe with un-circular porous media for different Reynolds numbers is shown in figures (4.6 a), (4.6 b), and (4.6 c). This graph showed an increase in velocity with a lower Reynold number and a more consistent velocity profile. because a higher Reynolds number results in a faster flow through porous medium, which raises turbulence.

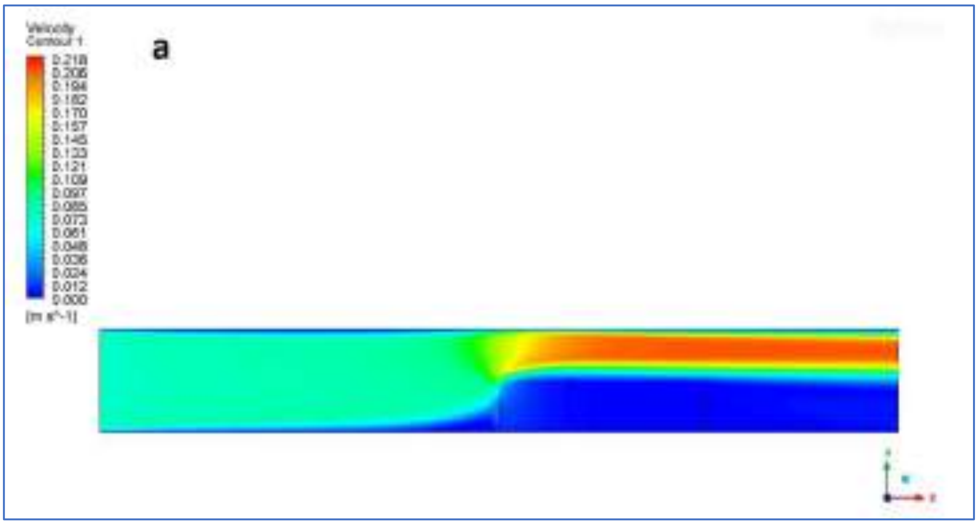

Fig (4.6 a). velocity contour for the 2D system with circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

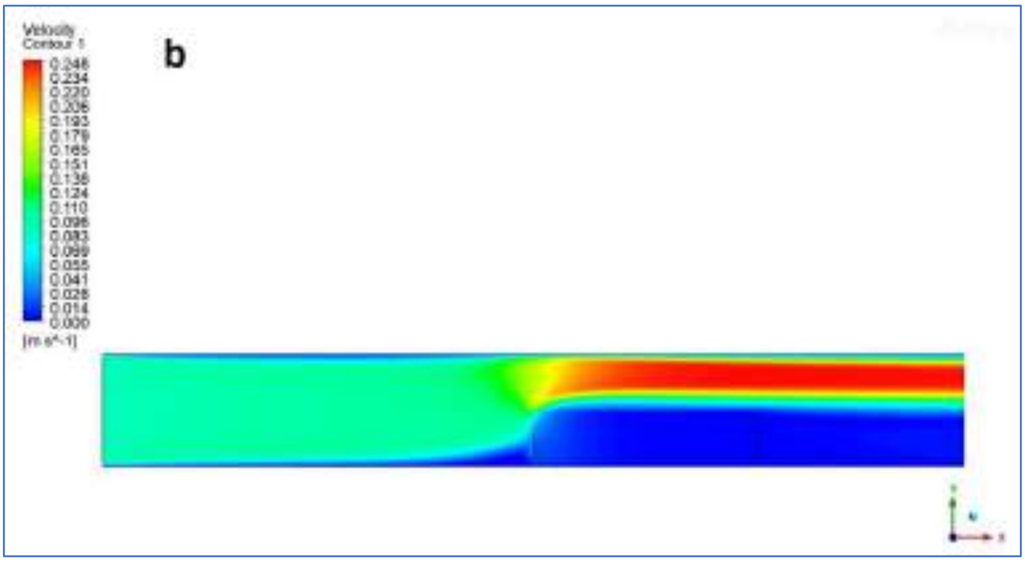

Fig (4.6 b). velocity contour for the 2D system with circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m

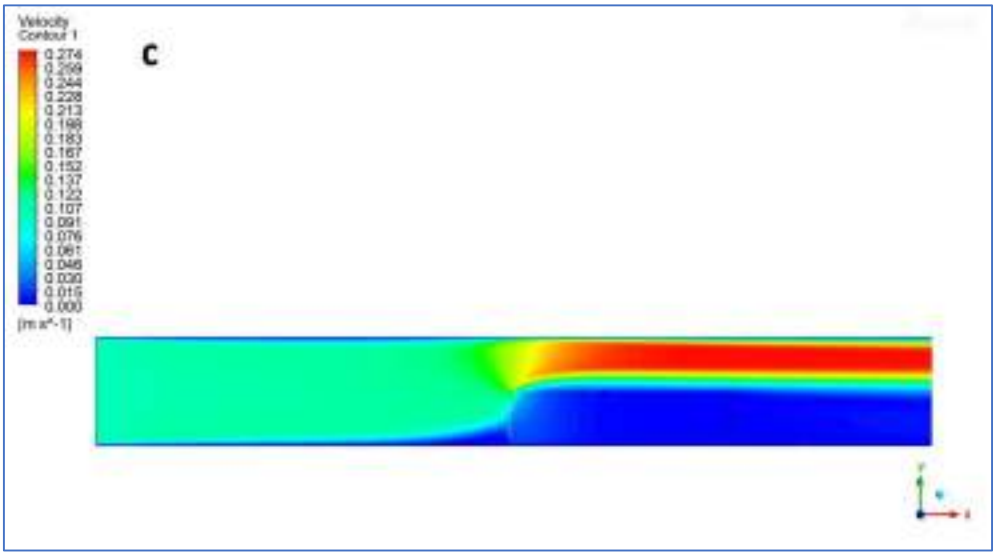

Fig (4.6 c). velocity contour for the 2D system with circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

4.6.3. Effect of pressure

It can note from this figure (4.7 a), (4.7 b), (4.7 c), that the pressure difference increased linearly with increases in the Reynold number because increasing the Reynold number means increasing the fluid velocity which increases the frictional forces, caused by the resistance to flow.

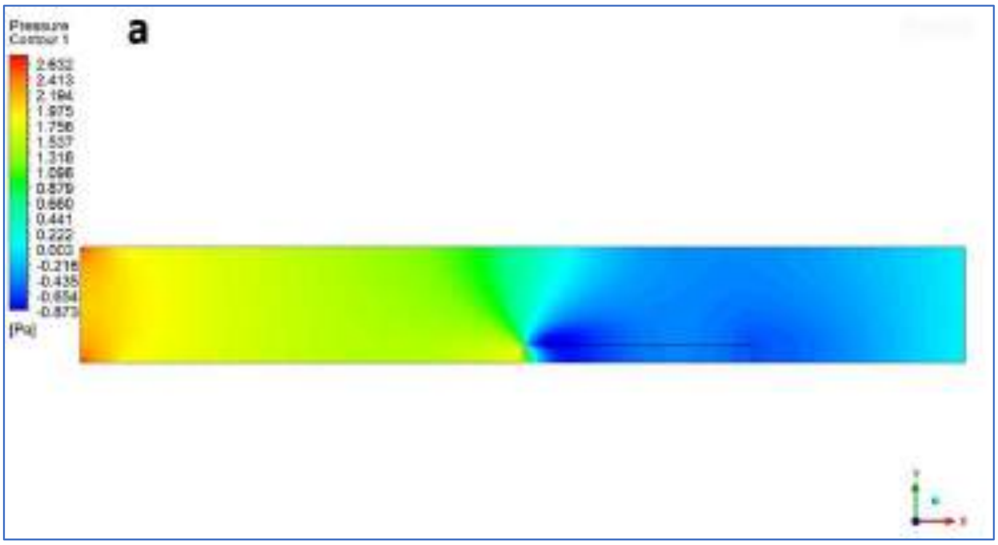

Fig (4.7 a). Pressure contour for 2D system with circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

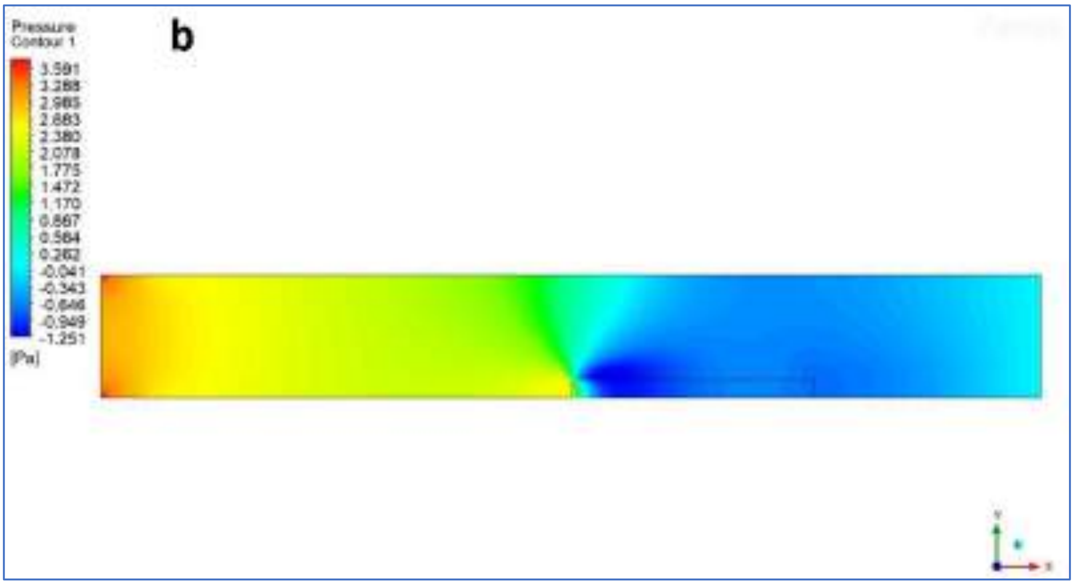

Fig (4.7 b). Pressure contour for 2D system with circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

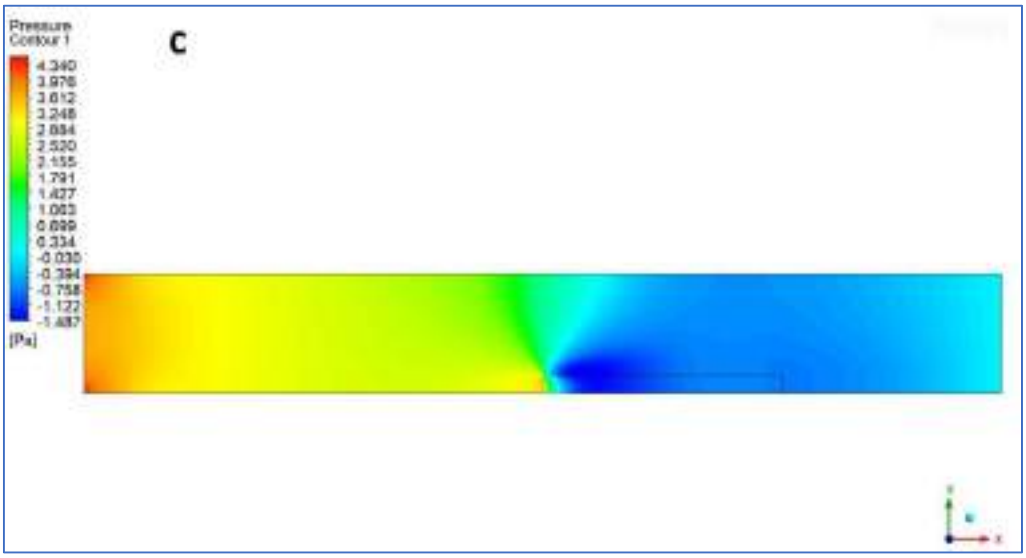

Fig (4.7 c). Pressure contour for 2D system with circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

Moreover, the increase in pressure difference with increases in the Reynold number becomes more obvious in the case of using un-circular porous media as shown in figure (4.8 a), (4.8 b), (4.8 c).

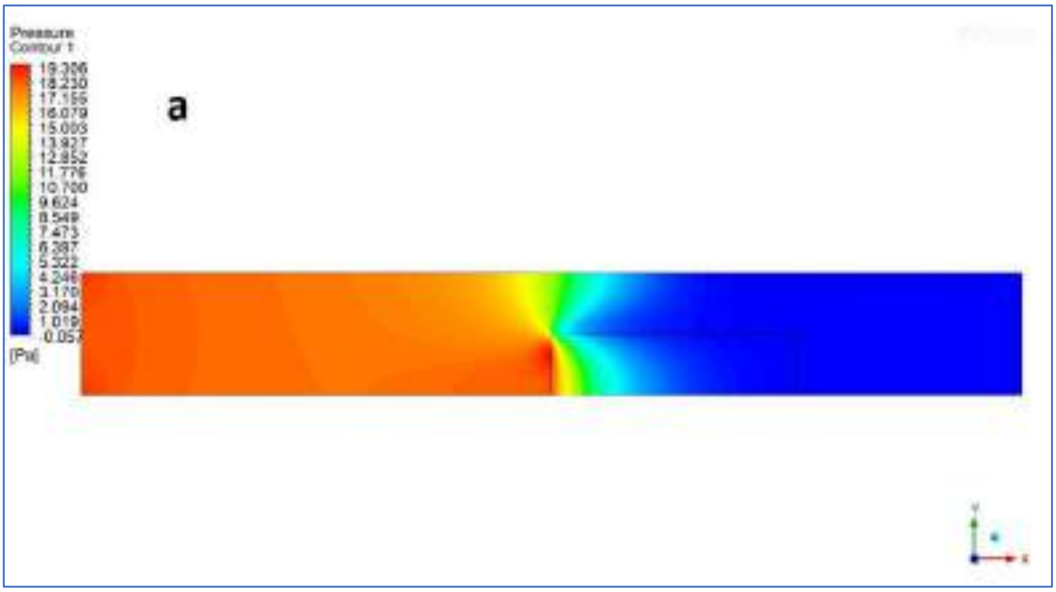

Fig (4.8 a). Pressure contour for 2D system with un-circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

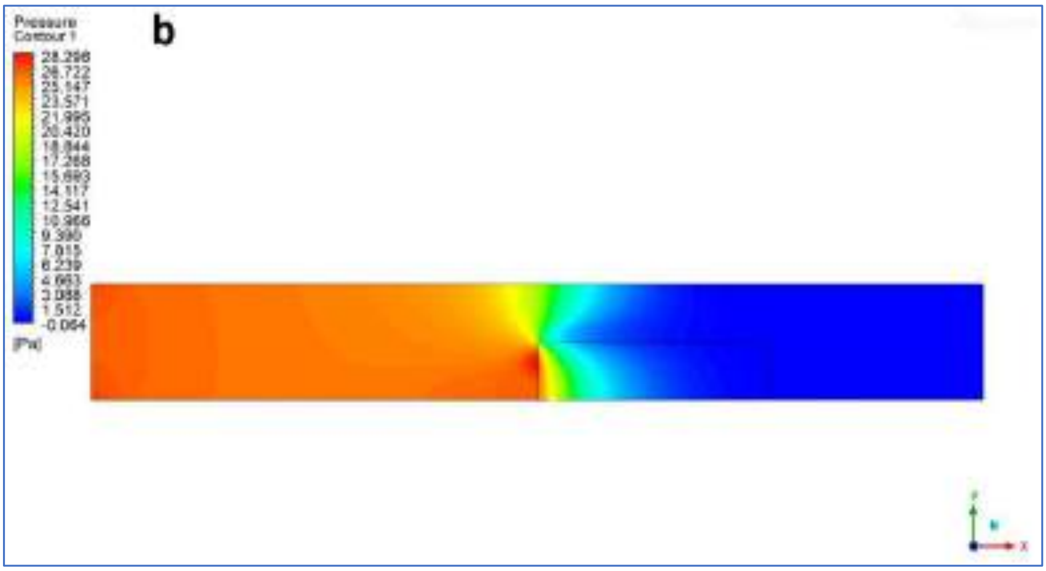

Fig (4.8 b). Pressure contour for 2D system with un-circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

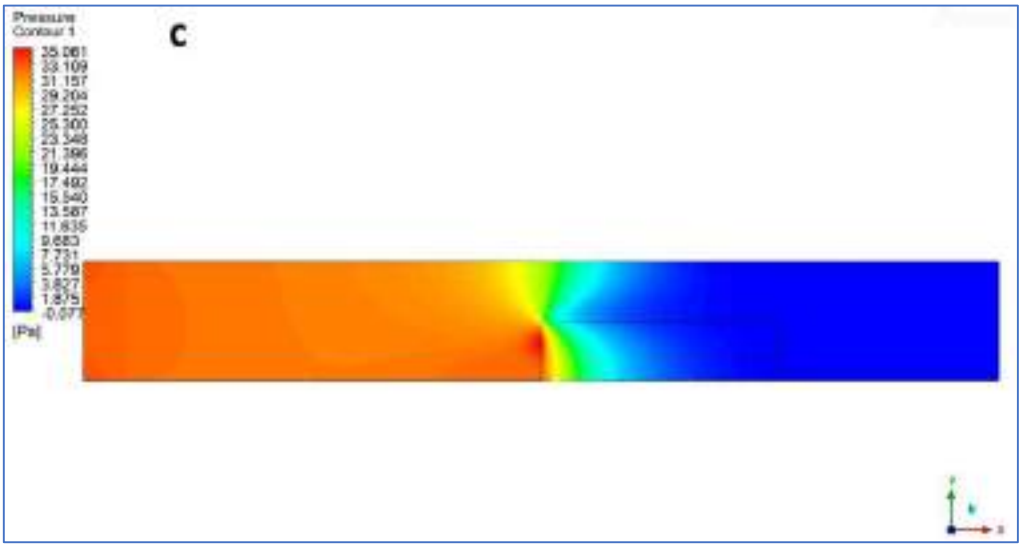

Fig (4.8 c). Pressure contour for 2D system with un-circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

4.7 Comparison between temperature profile for circular and un-circular pipe

As shown in Figure (4.9) the temperature gradients at low Re (1500) are approximately stable along the porous section and temperature changes rapidly with increasing Re (1800, and 2000 respectively).

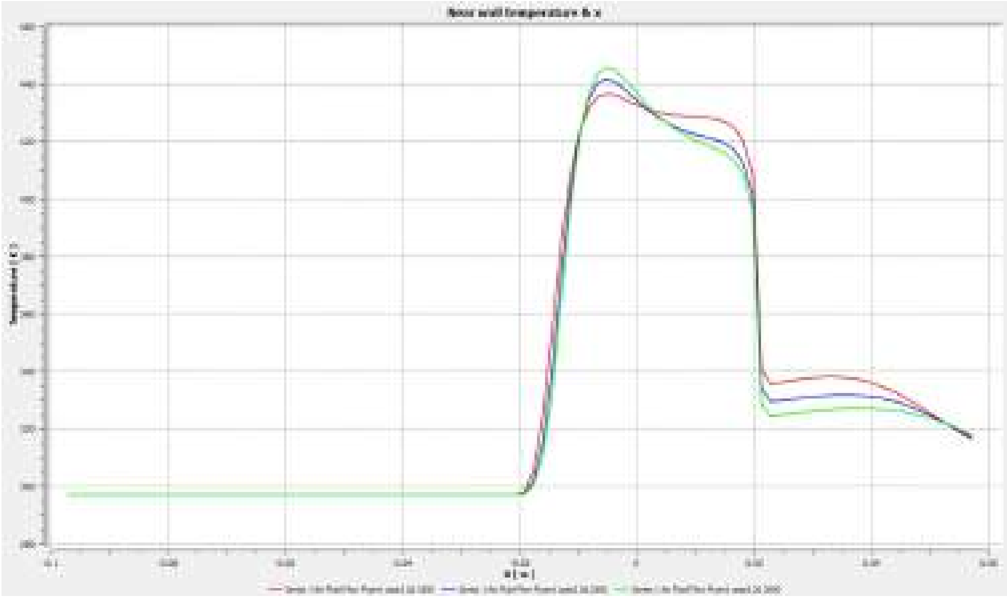

Fig (4.9). change of temperature with x at different Re numbers.

As illustrated in Figures (4.10) the temperature distribution inside un-circular porous media is given ad sharper than circular porous media carve which means that un-circular porous media gives less efficient heat transfer phenomena.

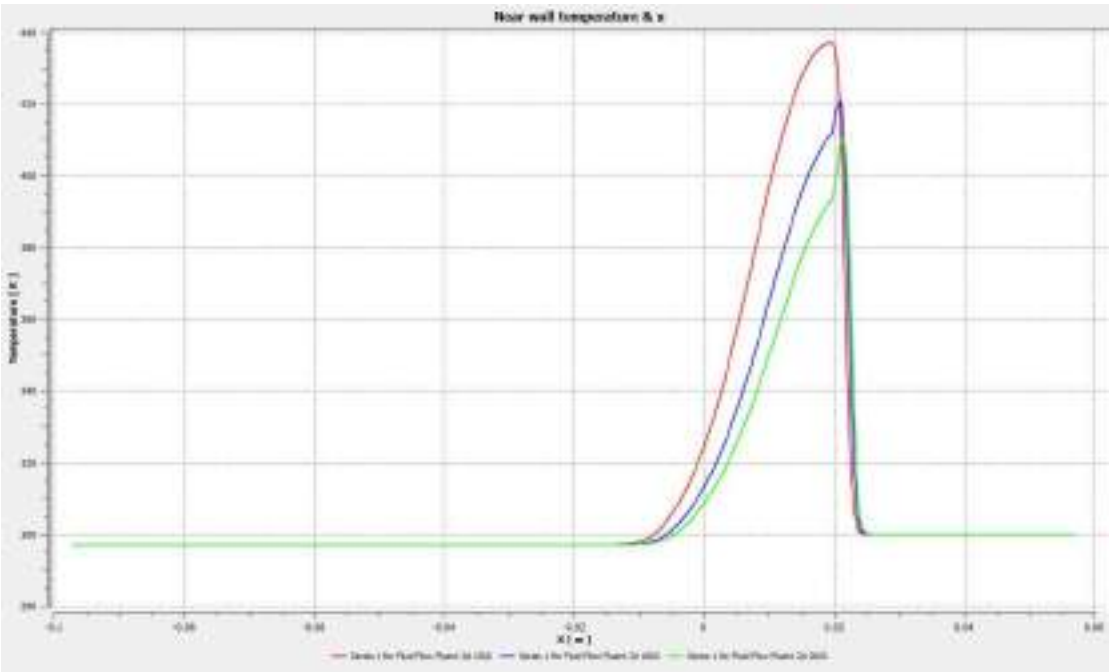

Fig (4.10). change of temperature with x at different Re numbers for un-circular porous media.

4.8 Effect of parameter for pipe with porous media for 3D system

4.8.1. Effect of Reynold's number

The effect of Reynold number on the temperature distribution in a 3D system is shown in figures (4.11 a), (4.11 b), and (4.11 c). The temperature distribution in 3D system is more stable than 2D system (see Figure (4.13)). In this figure, the local surface temperature distribution T_x demonstrated that for fixed heat flux the temperature distribution increases with the decreasing of Reynold number because the increasing Reynolds number yields faster flow through the porous media.

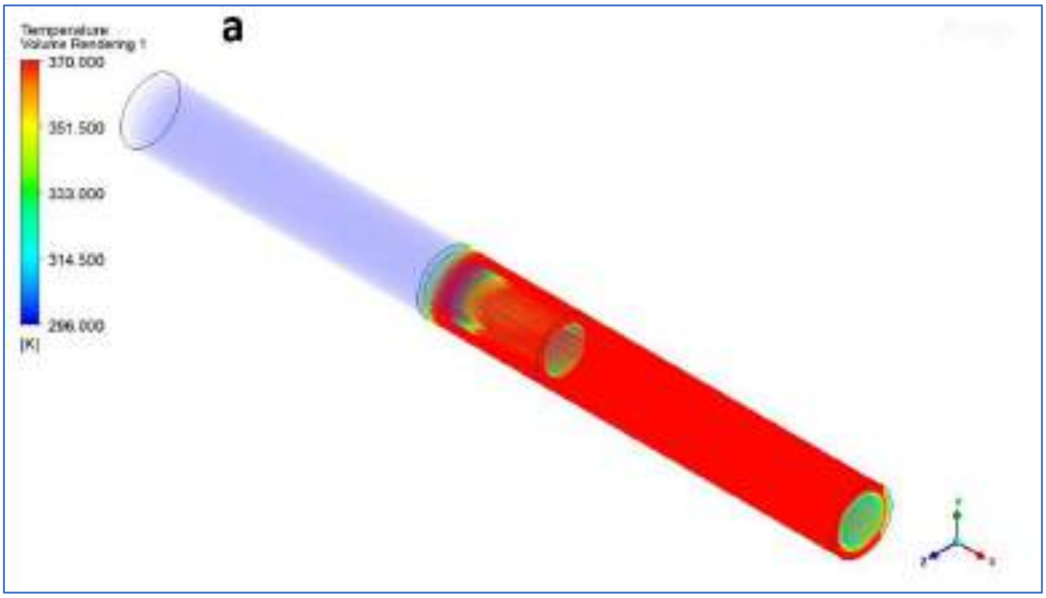

Fig (4.11 c). Temperature distribution for 3D system with circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

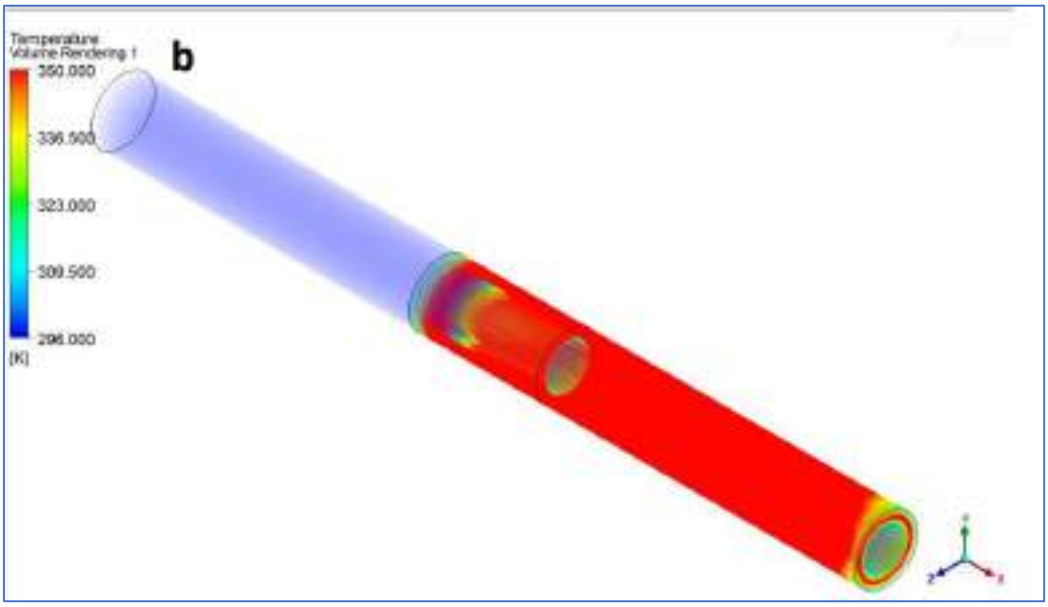

Fig (4.11 b). Temperature distribution for 3D system with circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

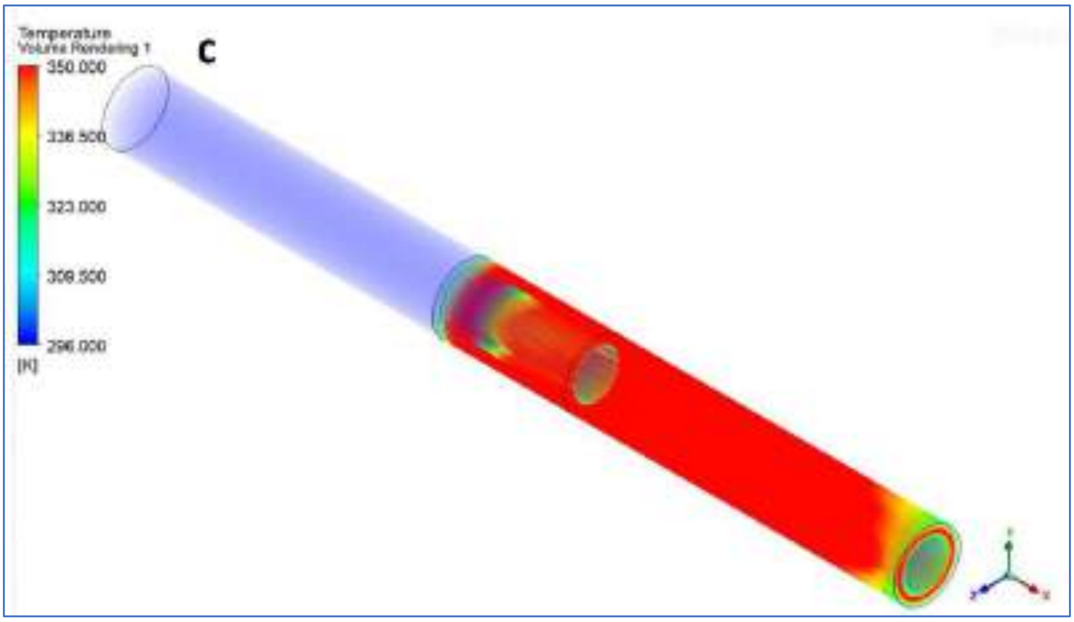

Fig (4.11 c). Temperature distribution for 3D system with circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

When the 3D system was studied for un-circular porous media the velocity and temperature distribution will appear as in figures (4.12 a), (4.12 b), (4.12 c). whereas the circular porous media gives better heat transfer efficiency.

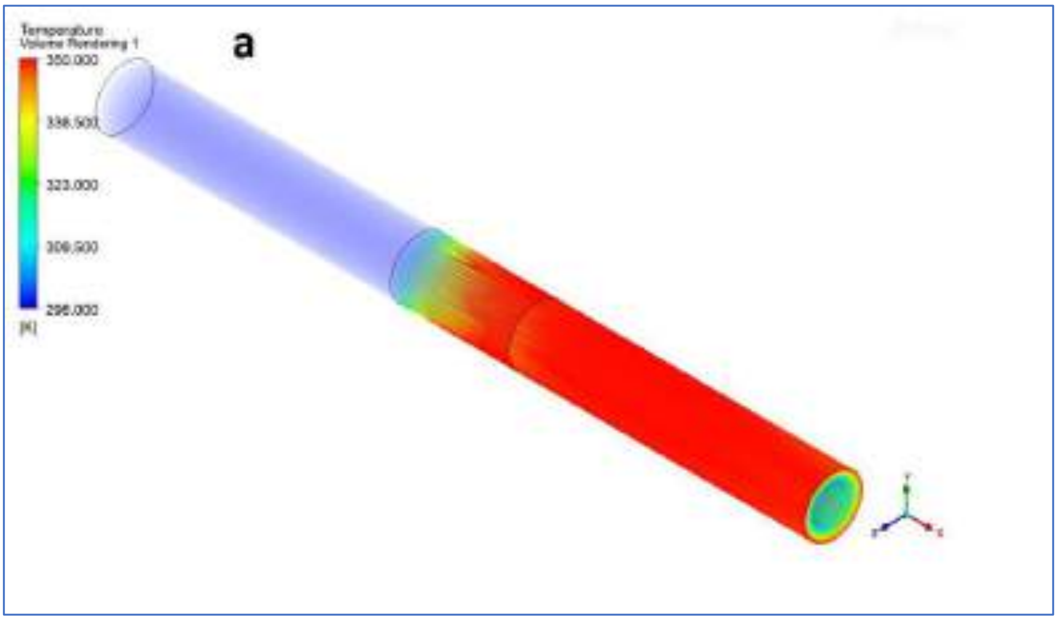

Fig (4.12 a). Temperature distribution for 3D system with un-circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

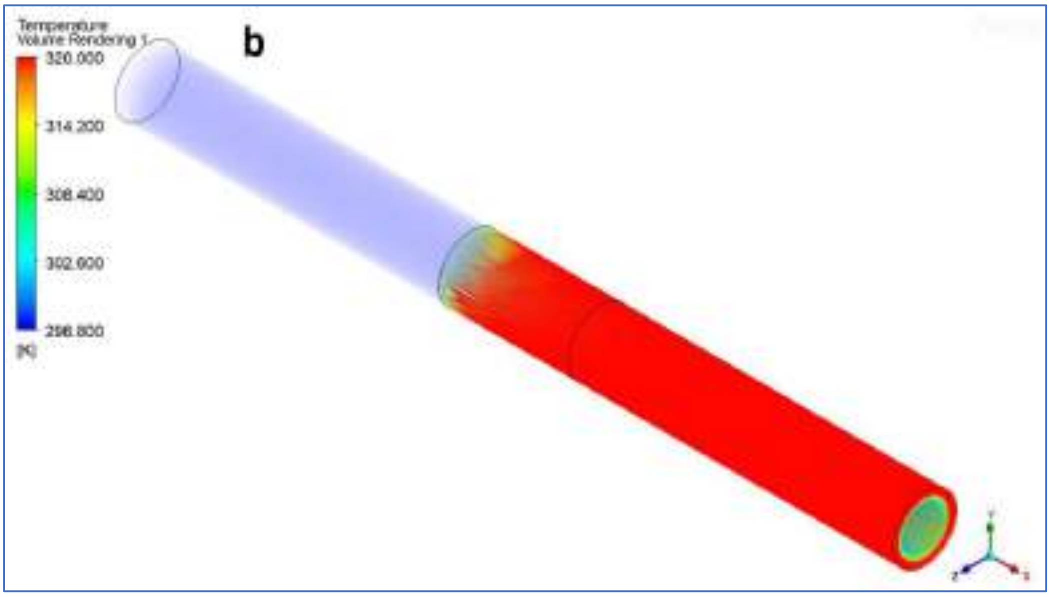

Fig (4.12 b). Temperature distribution for 3D system with un-circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

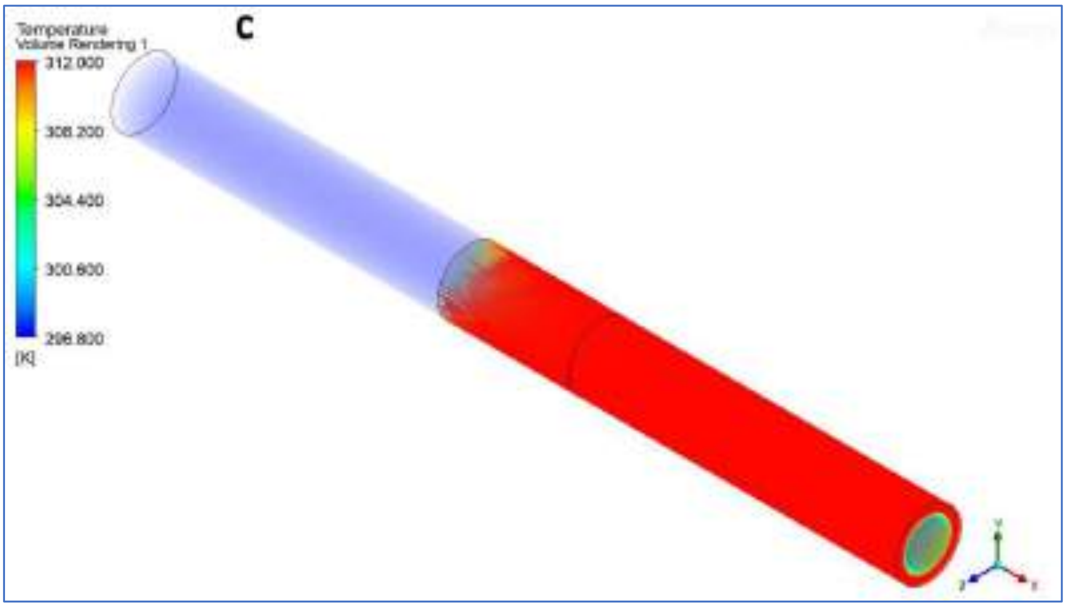

Fig (4.12 c). Temperature distribution for 3D system with un-circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

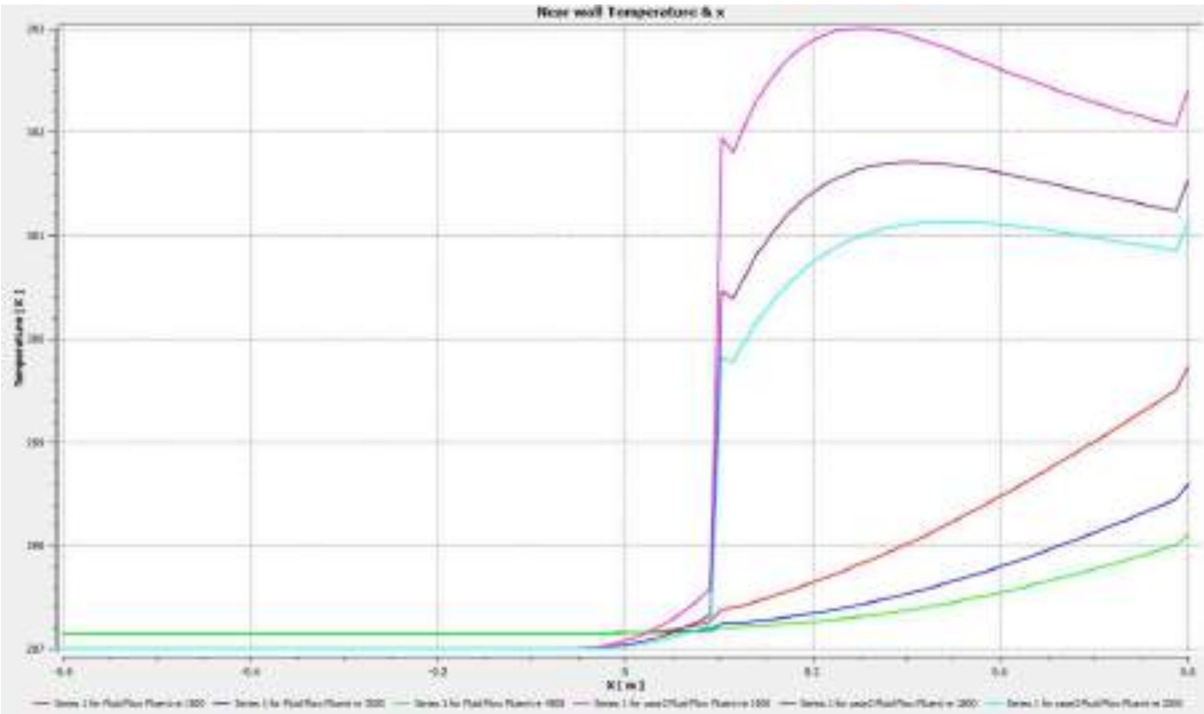

Fig (4.13). change of temperature with x at different Re numbers for circular and un-circular porous media.

4.8.2. Effect of velocity

The variation of the local velocity in 3D for different Reynolds numbers is shown in figures (4.15), and (4.15). This figure presented the velocity increase with Reynold number and lower give more stable velocity profile. Because the increasing Reynolds number yields faster flow through the porous media and hence increases the turbulency.

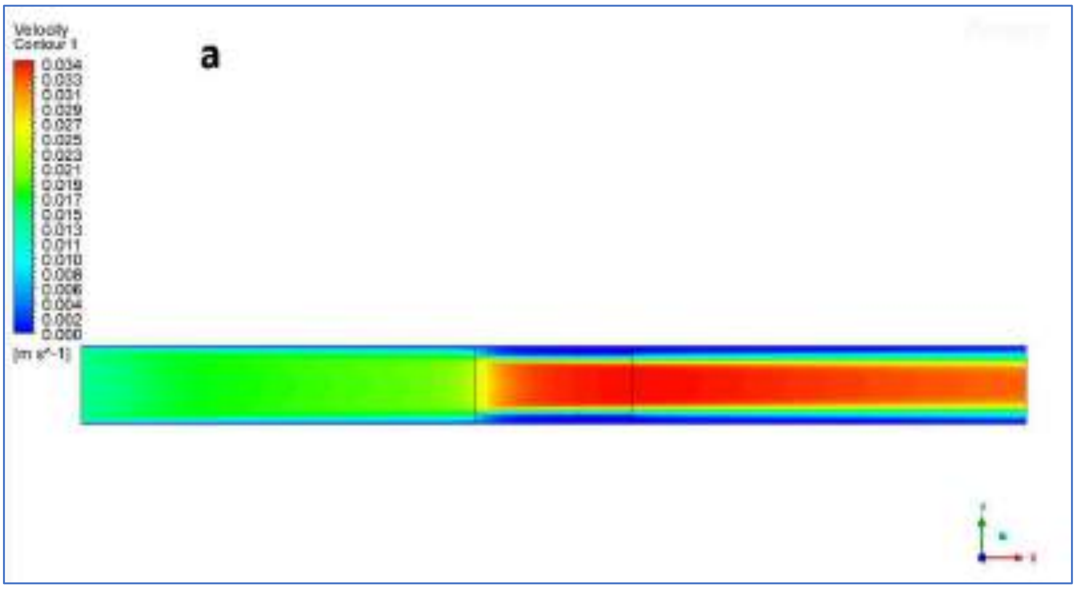

Fig (4.14 a). Velocity gradient for 3D system with circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

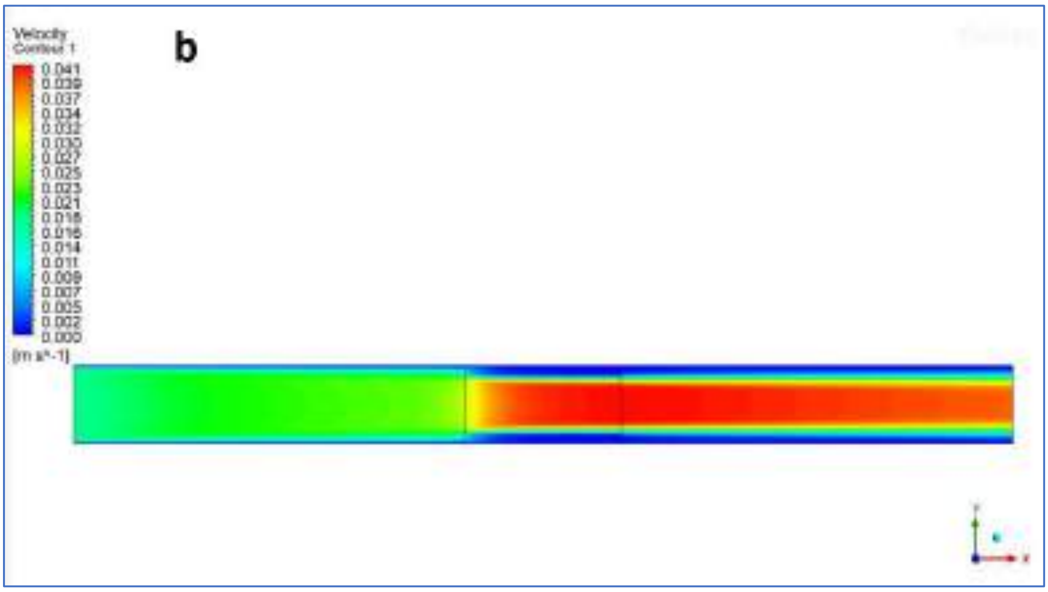

Fig (4.14 b). Velocity gradient for 3D system with circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

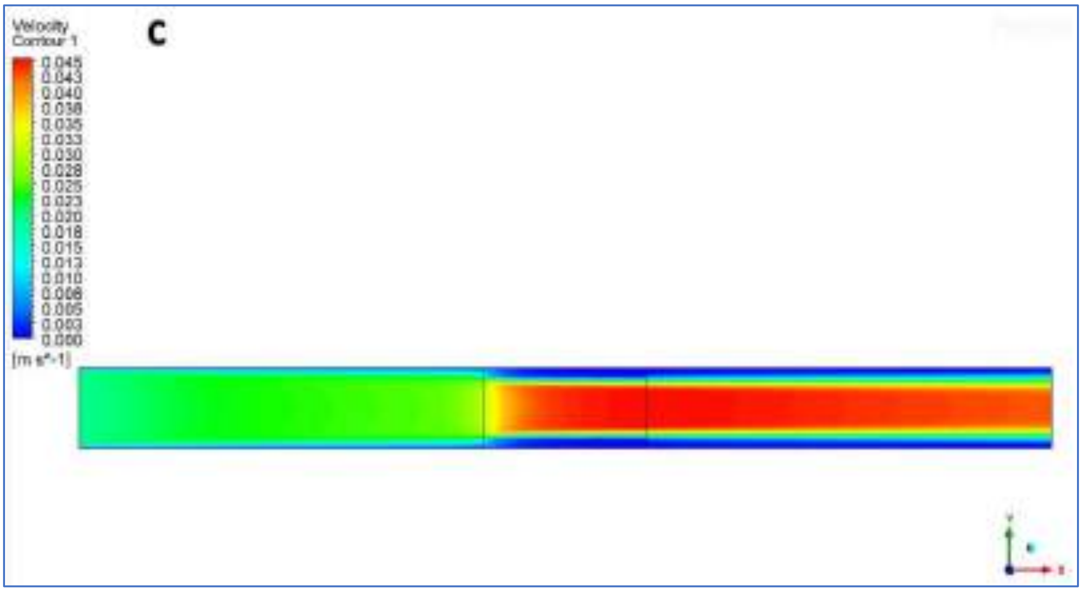

Fig (4.14 c). Velocity gradient for 3D system with circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

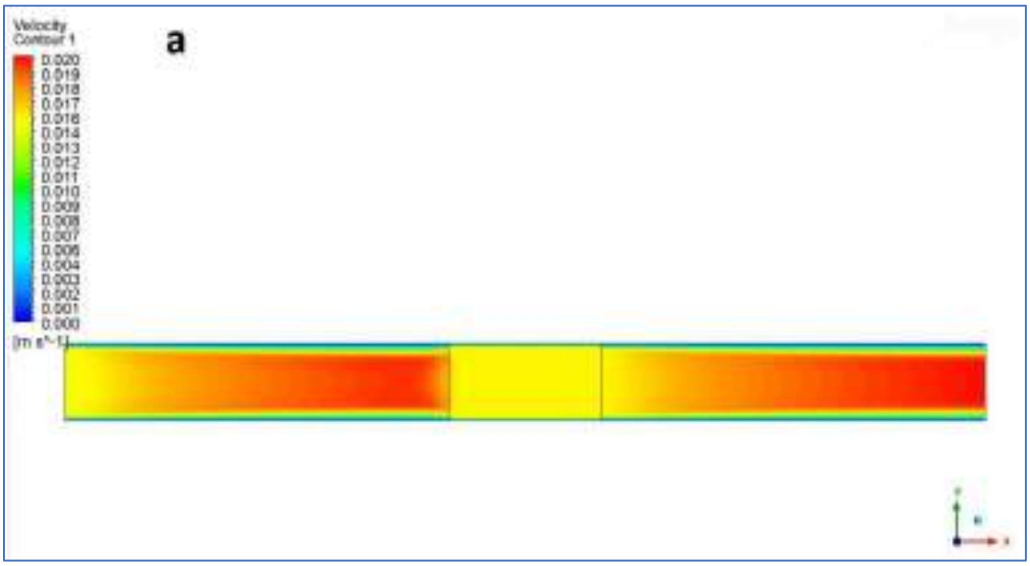

Fig (4.15 b). Velocity gradient for 3D system with un-circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

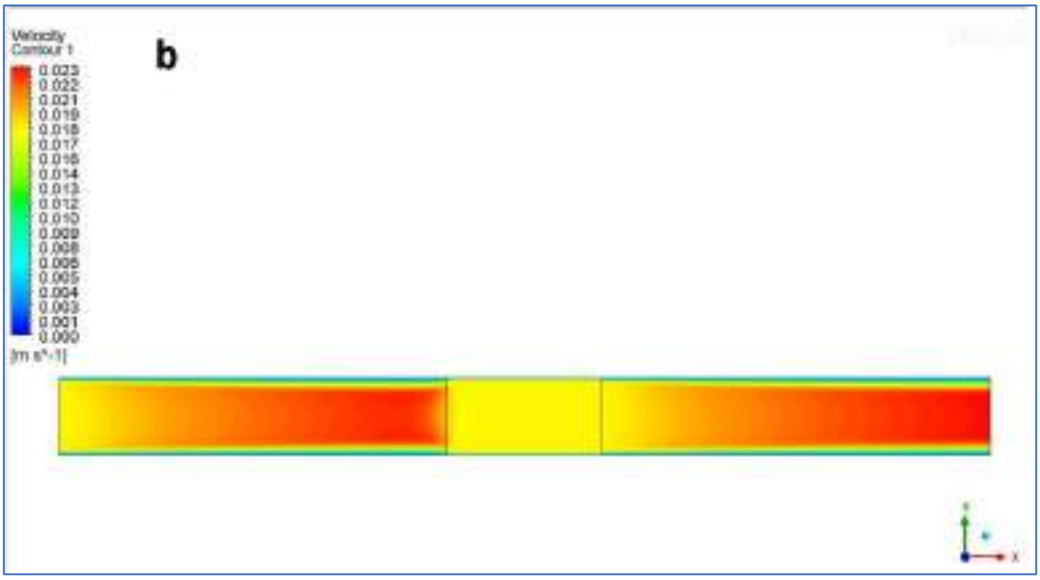

Fig (4.15 b). Velocity gradient for 3D system with un-circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

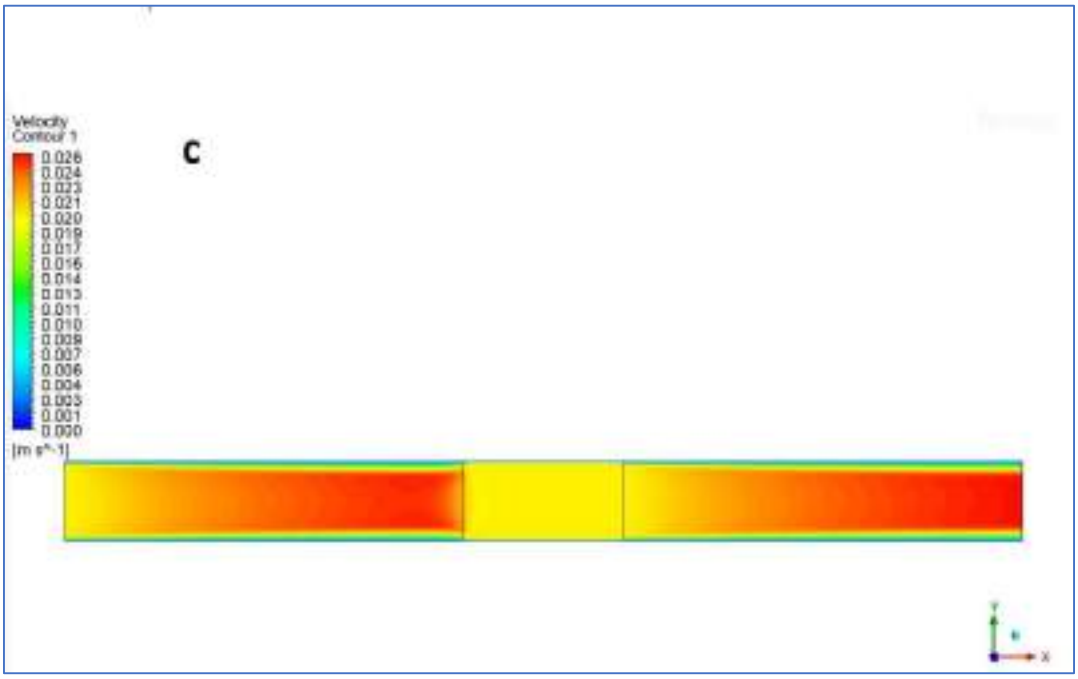

Fig (4.15 c). Velocity gradient for 3D system with un-circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

4.8.2. Effect of pressure

For 3d systems (circular and un-circular) the relationship between pressure and Reynold number has the same behaviour for 2d systems but with less pressure difference. See figures (4.15), (4.16).

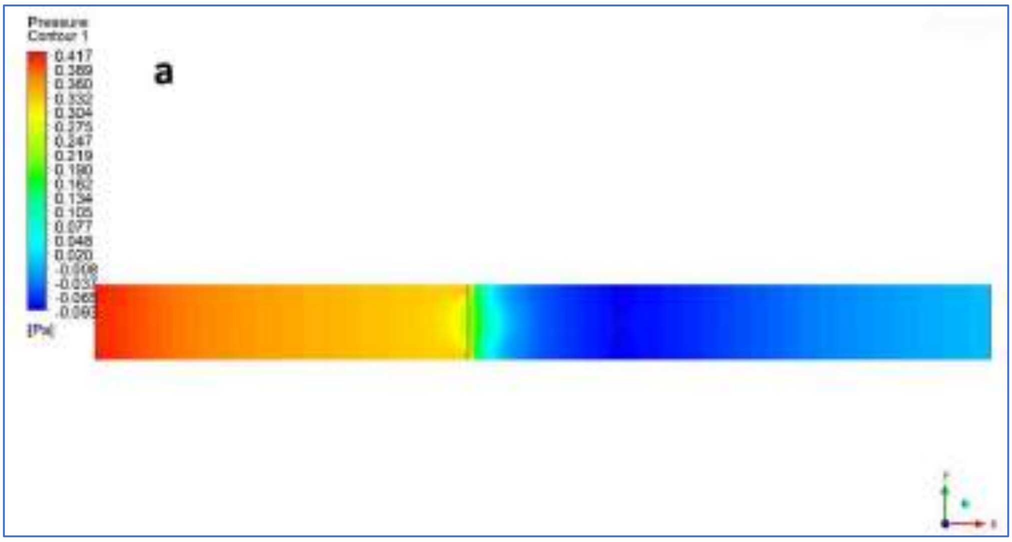

Fig (4.16a). Pressure contour for 3D system with circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

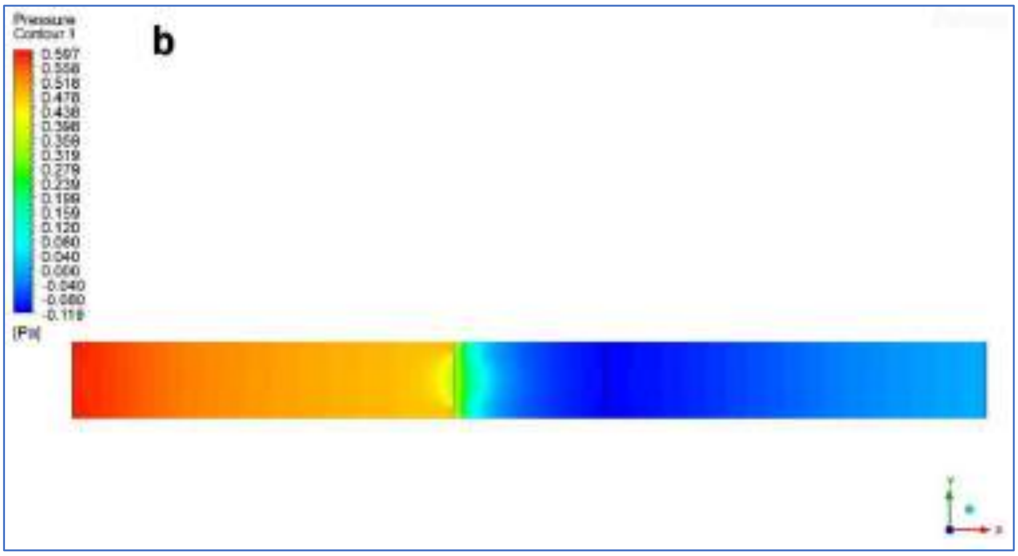

Fig (4.16 b). Pressure contour for 3D system with circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

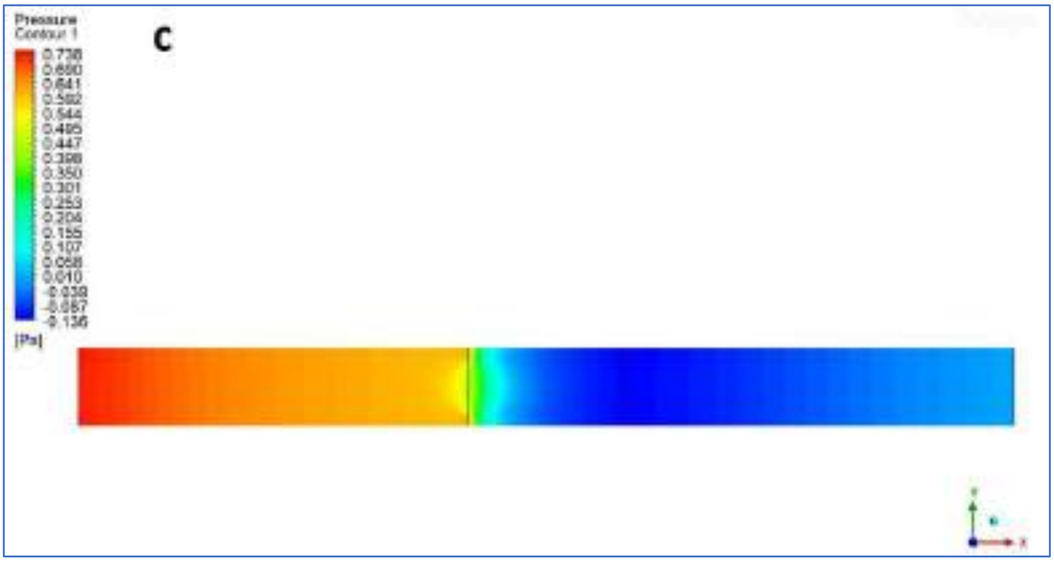

Fig (4.16 c). Pressure contour for 3D system with circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

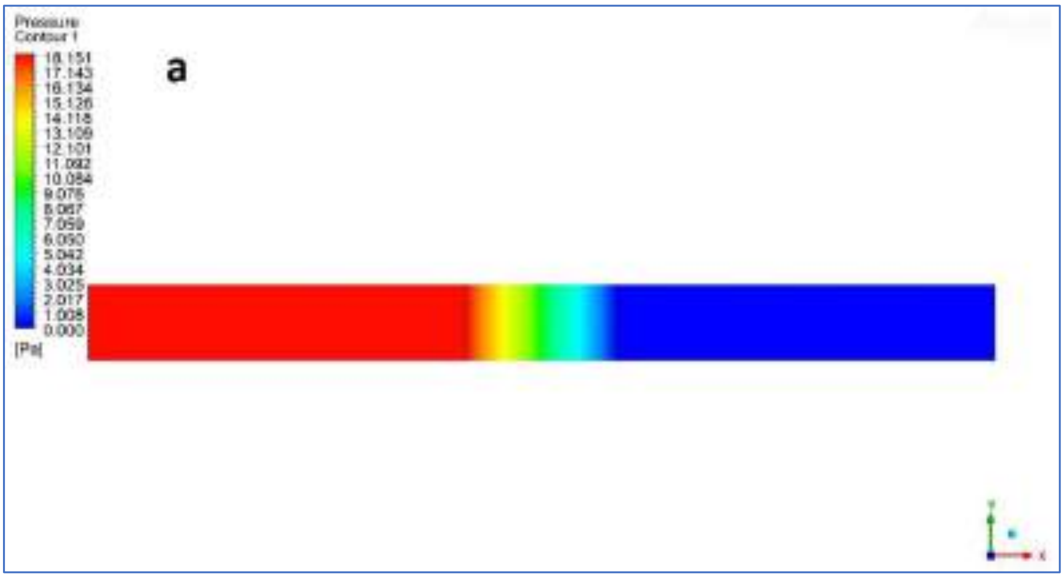

Fig (4.17 a). Pressure contour for 3D system with un-circular porous media at Re 1500, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

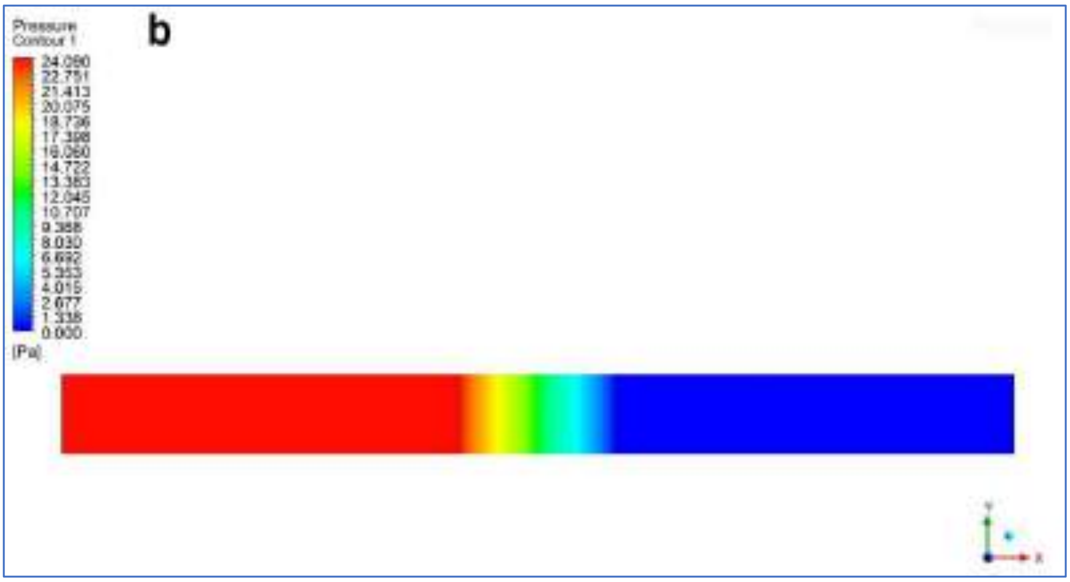

Fig (4.17 b). Pressure contour for 3D system with circular porous media at Re 1800, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

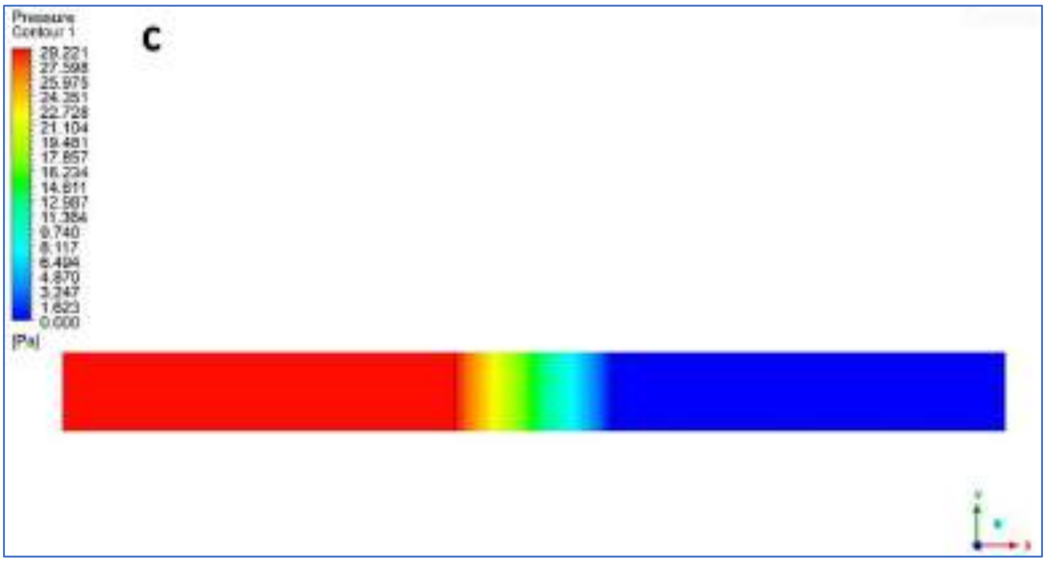

Fig (4.17 c). Pressure contour for 3D system with circular porous media at Re 2000, Radius = 0.1 m, Z= 1.2 m with a porous section of 0.2 m and porous section thickness of 0.75 m.

Chapter Five

CONCLUSIONS AND RECOMMENDATIONS

CHAPTER FIVE
CONCLUSIONS AND RECOMMENDATIONS

In this study, a forced convective heat transfer inside a circular pipe fully filled with different configurations of porous media materials (circular and un-circular) and subject to a constant heat flux has been performed theoretically using ANSYS software. It can be analysed the Automobile radiator cooling system is very important in an internal combustion engine. From different findings are concluded the following conclusions and recommendations for future work can be summarized as follows:

5.1 Conclusions

The following conclusions can be obtained for the present study:

1) The temperature distribution profiles were very affected by the parameters: Reynold number, and porous media shape.
2) The temperature distribution profiles become more stable when the circular porous media had been used.
3) temperature distribution inside un-circular porous media is given and sharper than circular porous media carve which means that un-circular porous media gives less efficient heat transfer phenomena.
4) The local surface temperature distribution T_x demonstrated that for fixed heat flux the temperature distribution increases with the decrease of Reynold's number from (2000 to 1500) for both systems. because the increasing Reynolds number yields faster flow through the porous media.
5) pressure difference increased linearly with increases in the Reynold number.
6) The 3d system with circular porous media gives the less pressure difference than other systems.

5.2 Recommendation

The following can be suggested as topics for further work:

1) Using different working fluids in this model such as nanofluid, oil, air …...etc.

2) Using different porous medium types, different diameter, and shape with different incline angles.

4) Show the effect of turbulence flow on forced convection heat transfer theoretically and experimentally.

5) Study different pipe lengths, diameters and geometry.

6) In conclusion, the use of porous media in radiator is effective and improve heat transfer efficiency without increasing the overall size of radiator

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