Numerical Analysis of the Effects of Baffle orientation and shape factor on Pressure Drop and Heat Transfer Coefficient in tube Heat Exchanger

Alok Kumar[1], Dr Shyam Birla[2]

[1]Mtech Scholar, dept of mechanical engineering, Oriental College of Technology Bhopal, Bhopal
[2]Dr Shyam Birla, dept of mechanical engineering, Oriental College of Technology Bhopal

---***---

Abstract- *Heat exchangers can have flat or corrugated plates for the wall material, as well as cylindrical or variable section ducts with baffles of varying shapes and sizes. The purpose of this study is to demonstrate the effectiveness of baffles in creating recirculation zones in a channel by causing a disturbance in the fluid flow and, consequently, the convective heat exchange instability. The use of hydrogen and forced convection creates a turbulence in the flow. The baffles are installed on the vertical and horizontal sides of a rectangular channel at angles of 30 degrees, 45 degrees, and 60 degrees, respectively. Dispersion patterns of the Nusselt number, along with velocity fields and axial velocity profiles, are displayed. Use of baffles results in the creation of recirculation zones, which in turn disrupt the natural flow of convective heat exchange. Sloped at an angle of 30 degrees, the slanted section of the baffle increases the Nusselt number by nearly 26%.*

Key words: Convective instability, Forced convection, rectangular duct, Baffle, Flow disturbance, Hydrogen.

Introduction:

A heat exchanger is a device that allows a transfer of thermal energy between at least two bodies (solids, liquids or gases). Most of the time, these are fluids. These fluids may be in indirect or direct contact, i.e. separated or not by a generally metallic wall. However, this definition is not general because there are a multitude of types of exchangers, with geometries, configurations and even modes of operation which can be very different. Thus, the selection of a heat exchanger for a particular application depends on a number of factors, including the fluids' temperature and pressure range, their thermophysical characteristics, maintenance requirements, and congestion. It is obvious that having a well-adapted, well-sized, well-made and well-used heat exchanger improves the thermal efficiency of this device.

The improvement of the performance of heat exchangers is aimed at the mechanisms of intensification of convective transfers, as evidenced by the increasing number of studies carried out on the effect of inserting the elements into a tubular space which would be of appreciable contribution to new designs of thermal appliances or to optimise the economic in the operation of heat transfer networks.

The most commonly used intensification techniques are the so-called "passive" ones, they consist in the majority of cases in increasing the heat exchange surfaces in order to intensify heat transfers. One of the most important methods is to use extensive elements such as baffles inside the heat exchanger channel.

Literature review

The various sizes, positions, and orientations of obstructions inside exchanger tubes have been the focus of several scientific works, according to the literature. In order to better comprehend convective flow in a rectangular pipe:

Pethkool et al. [1], For a Reynolds number ranging from 5500 to 60000, it was examined how blocking factor (e/D) a corrugated helical tube with monophasic turbulent water flow that was influenced convective heat transfer. The findings demonstrate that the performance factor significantly rises with an increase in blocking factor and reaches a value of about 232 percent when compared to a smooth tube.[10], the evaluation of the Nusselt number is carried out by the following formula:

$$Nu = 1{,}579 Re^{0.639} Pr^{0.3} (e|D_H)^{0.46} (P|D_H)^{0.35} \qquad (I.1)$$

Fig -1 Tube corrugated

Promvonge's experimental work. [2], undertaken to assess the forced convection heat transfer for turbulent airflow in a channel with a 60°-V multiple deflector turbulator., the Reynolds number is between 5000 and 25000, and the blocking factor varies between 0.1 and 0.3. The Nusselt number is evaluated by the following empirical formula:

$$No = 0.147Re^{0.763}Pr^{0.4}(1 - e|\,H)^{-1,793}(PR + 1)^{-0.42} \qquad (I.2)$$

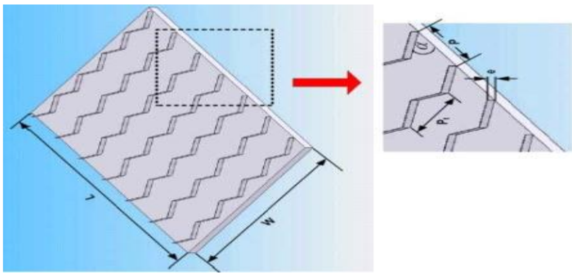


Fig -2 Twisted bands

Demartini et al. [3], We undertaken an intriguing computational and experimental analysis of flat transverse baffles for a turbulent airflow regime in a two-dimensional heat exchange channel. Additionally, they discovered that this issue is crucial in the area of heat exchangers since it affects flow characterisation, pressure distribution, and the potential existence and growth of recirculation's. A reasonably dense recirculation zone is visible above each channel's facets and is moving in the direction of swallow on the profiles and axial velocity distribution. The impact of the flow field's distortion increases as the flow approaches the first channel. The biggest disruption is seen upstream of the second channel, they also demonstrated.

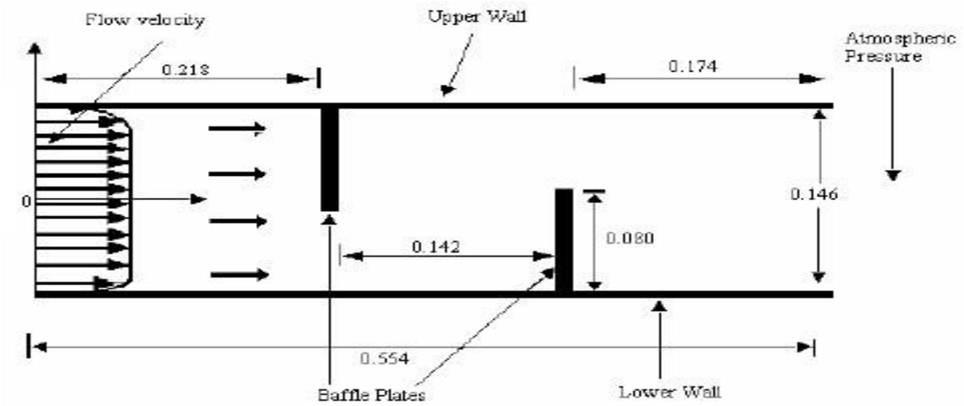
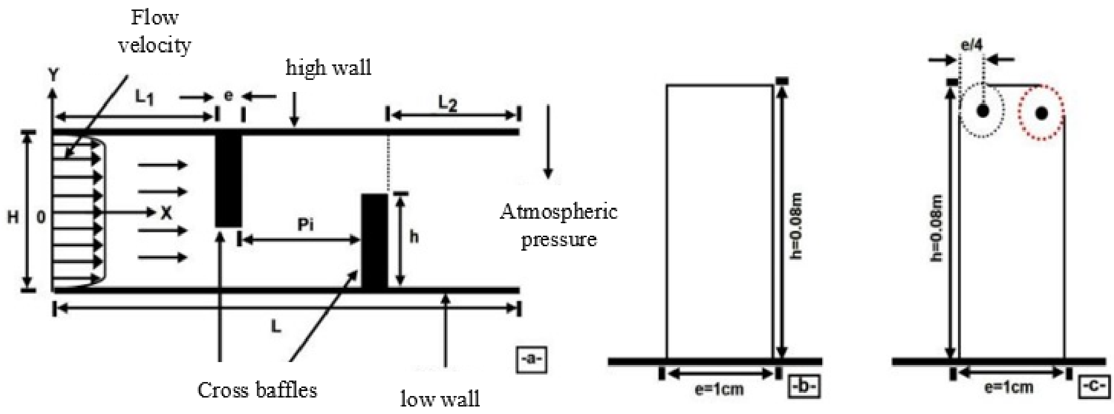


Fig -3 Channel configuration studied by Demartini et al, [3]

Bensenouci et al. [4], Forced convection heat transfer in a rectangular pipe with baffles was investigated by numerical modelling. Convective heat exchange is promoted by the requirement to insert rows of fins and baffles in the vein of the emolument in heat exchangers. Research has revealed that the vertical portions nearest to the baffles are better heated than the distant vertical parts. The distribution of the temperature field in the channel further supports this observation.

Menni et al. [5], numerically studied the comparison between two different forms of fins and transverse baffles. The first is rectangular with a sharp tip, and the second rectangular is with a rounded tip. In a horizontal, two-dimensional pipe with a rectangular cross-section, they are placed in overlap. The fluid (air) is of the Newtonian type, incompressible with constant thermo-physical properties. The flow regime is considered permanent and purely turbulent. For a Reynolds number of between 5000 and 20000, calculations are conducted. The examination of the data showed that the addition of rectangular type baffles without rounding provides a significant increase in speed and improves the intensity of heat transmission when used with rectangular chianes.



b-Rectangular wing with a rounded tip **Fig -4** Test section

Tsay et al. [6], The improvement of heat transmission of a flow in a channel with a vertical channel was quantitatively examined. In the 100–500 Reynolds number range, the impact of channel size and back coatings on the flow structure is thoroughly investigated. They observed that by adding a baffle to the flow, the typical Nusselt number could be increased by 190%. They also noticed that the channel's position affects the thermal and dynamic properties of the flow.

Bilen et al. [7], conducted an experimental study on the thermal and hydraulic characteristics of turbulent airflow in tubes with different groove forms (circular, trapezoidal, and rectangular), with Reynolds numbers ranging from 10000 to 30000. When comparing the findings to a smooth tube, the circular groove configuration increases heat transmission by 63%, the trapezoidal groove configuration by 58%, and the rectangular grooved tube by 47%. The authors attributed the relatively poor thermal performance of rectangular grooves to the appearance of stagnation or recirculation zones in this configuration. The relatively similar thermal performance of trapezoidal and circular grooves, despite a lower number of trapezoidal grooves, is explained by greater flow disturbance in this configuration.

Methodology

- **The Fluent tool:**

Simulations based on computational fluid dynamics (CFD) are used to simulate, visualise, and analyse fluid flows and heat fluxes. It enables users to optimise the performance of new concepts, while reducing the marketing cycle, associated risks and costs.

Simulation of the thermal and dynamic behaviour of the rectangular exchanger channel menu:

- **Import Geometry (msh):**

The dimension of the geometry is in 2D, for this the choice of 2D seems the most appropriate for our simulation, so it is chosen as follows:

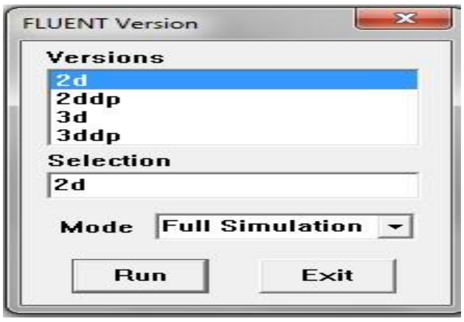


Fig -5 Open the Fluent Version

- **Import geometry:**

File Read Case

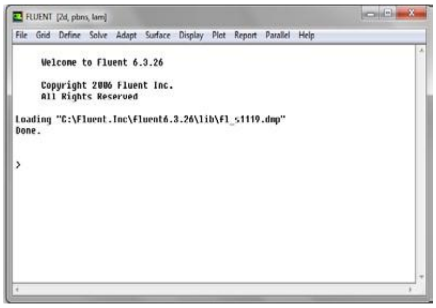


Fig -6 Importing Geometry

To start the simulation, you must import the file (*.msh) generated under Gambit.

- **Checking the imported mesh: Grid Check**

This checks whether the imported mesh contains errors.

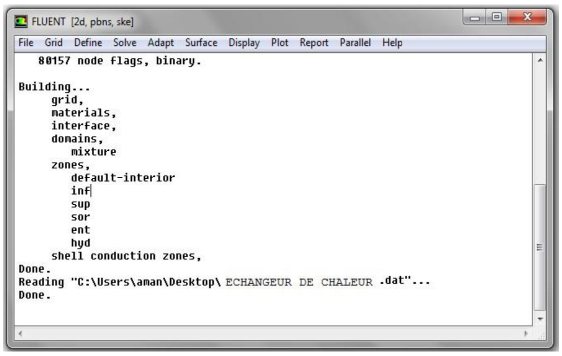


Fig -7 Checking the mesh under Fluent

- **Verification of the scale:**

Grid Scale

Always check that the dimensions displayed correspond to the physical dimensions of the problem.

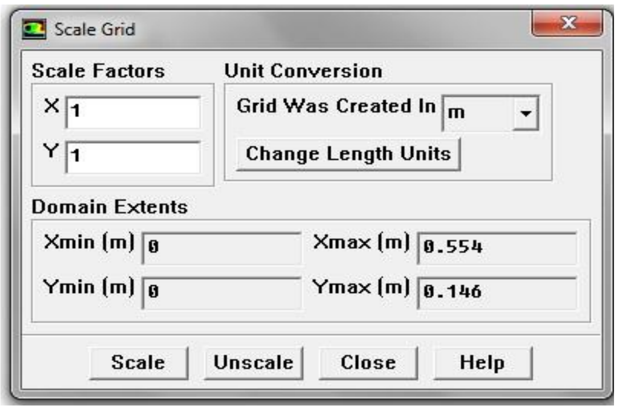


Fig -8 Unit Verification

- **Choice of solver:**

Define Models Solver…

Segregated Solver: is the most suitable for incompressible flows (fans, pumps...)

Coupled Solvers, the "coupled implicit" and "coupled explicit" solvers, are rather reserved for compressible flows at high speed.

This is also where the flow regime is chosen; permanent or unsteady

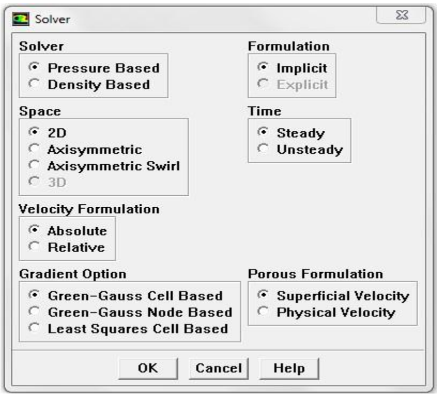


Fig -9 Choosing the Solver in Fluent

- **Grid display:**

Display Grid

You can view the mesh and it is very good to check the boundary conditions defined beforehand in Gambit

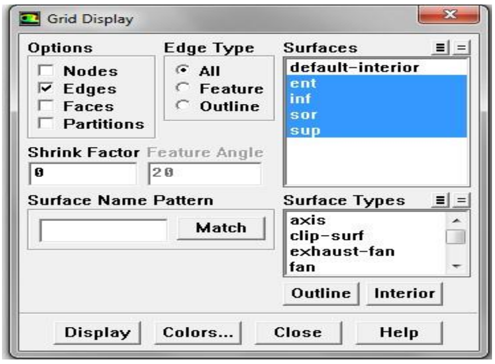


Fig -10 Displaying the Grid and Checking conditions

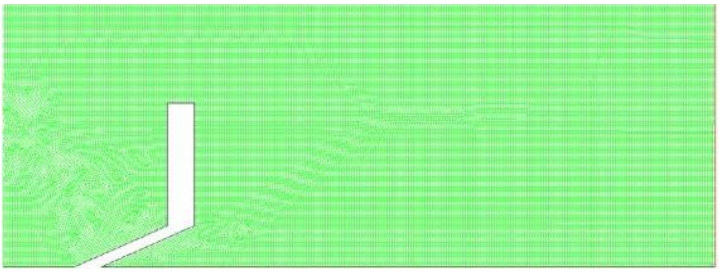


Fig -11 Simulation Domain Display

- **Choice of turbulence model:**

Define Models Viscous

Fluent offers different models of turbulent flow. Among which non-viscous, laminar, turbulent flows ... etc.

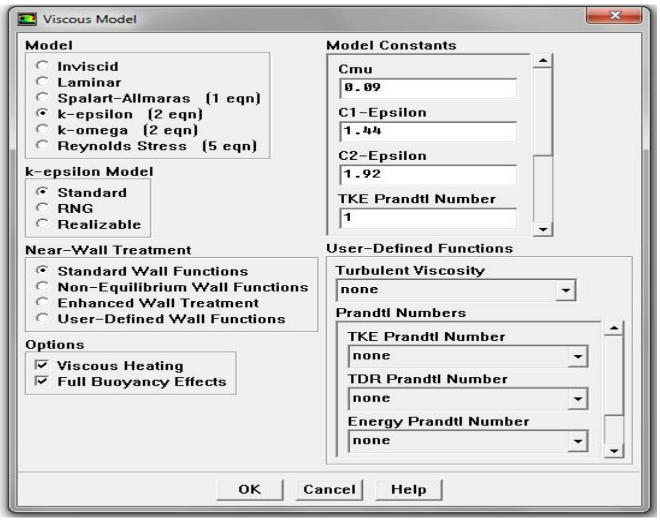


Fig -12 Choice of turbulence model

- **Energy equations:**

Define Models

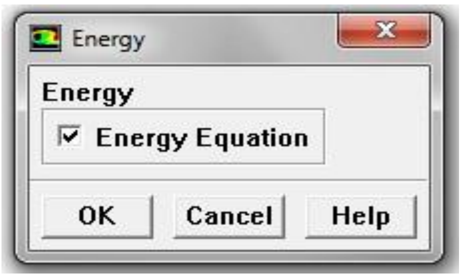


Fig -13 Establishing the Energy Equations

- **Definition of material characteristics: Define Materials**

Fluid characteristics are loaded from Fluent's data library.

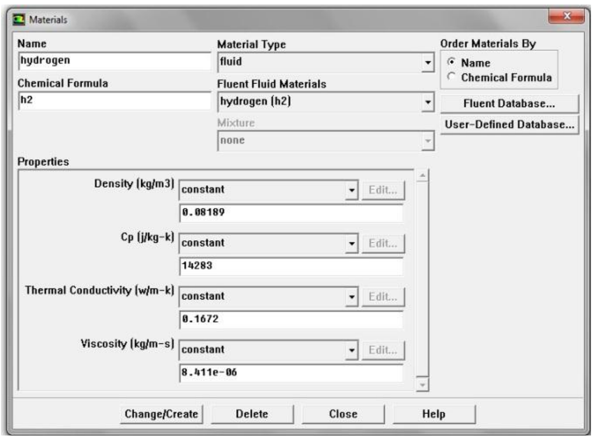


Fig -14 Material Characteristics

- **Usual boundary conditions:**

Define Boundary Conditions:

The values of the boundary conditions shall be fixed as follows:

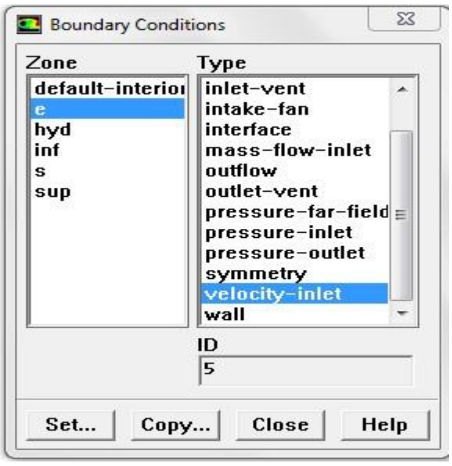


Fig -15 Operation Boundary Conditions Values

- **Choice of solution:**

Solve controls solution

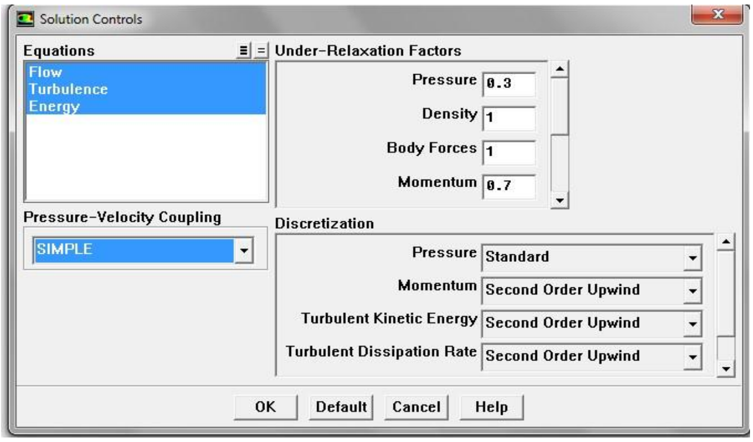


Fig -16 Choice of solution

- **Start of the calculation:**

Solve Iterate…

To start the calculations, you must first choose the number of iterations

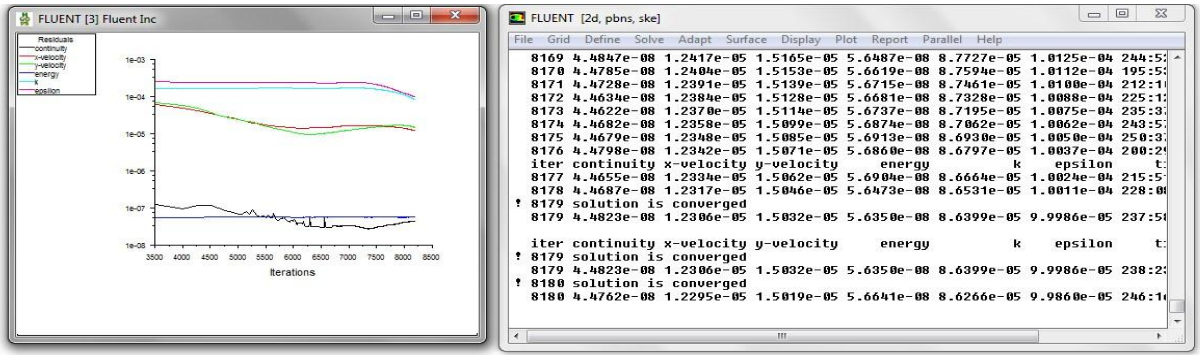
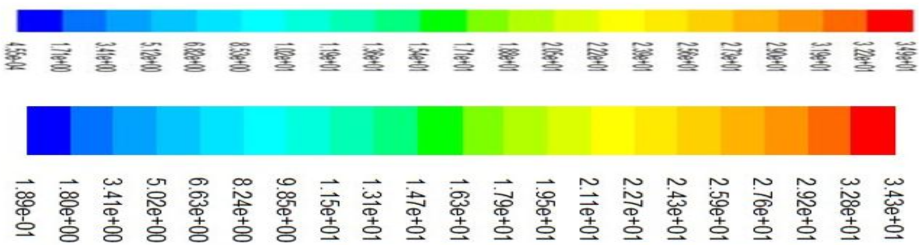


Fig -17 Rates of evolution of the design residues

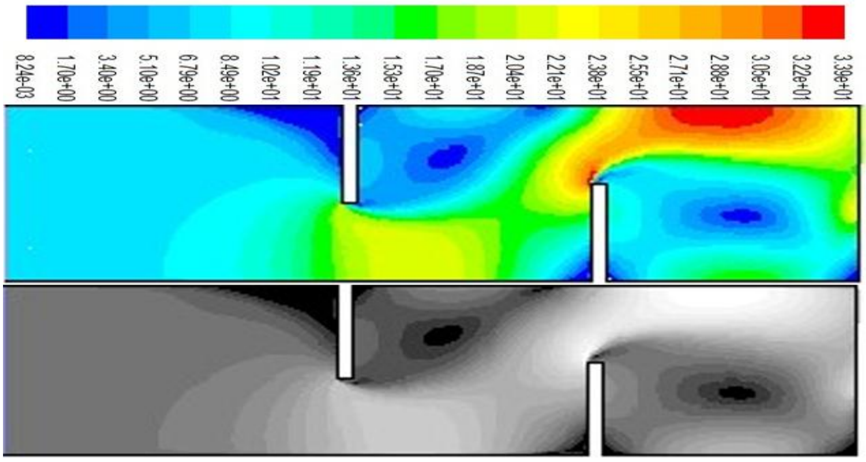
Result

IV.9. Effect of the shape of the baffles:

IV.9.1. Inclined channel θ=30° and chicflat ane θ= 90°:



a. Inclined channel θ=30°



b. Channel plane $_{\theta}$= 90°

Fig -18 Velocity vectors for cases a and b

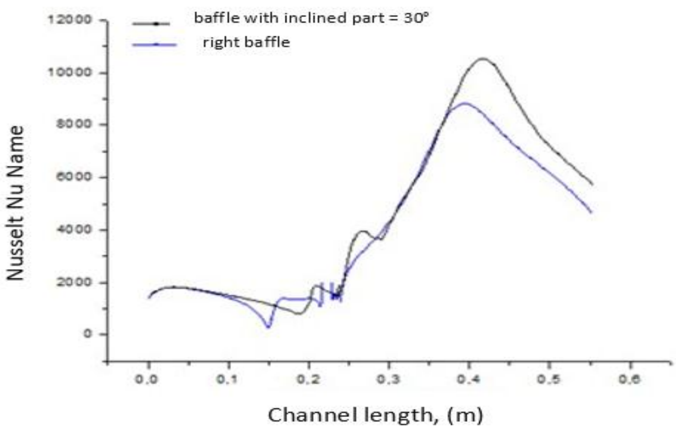
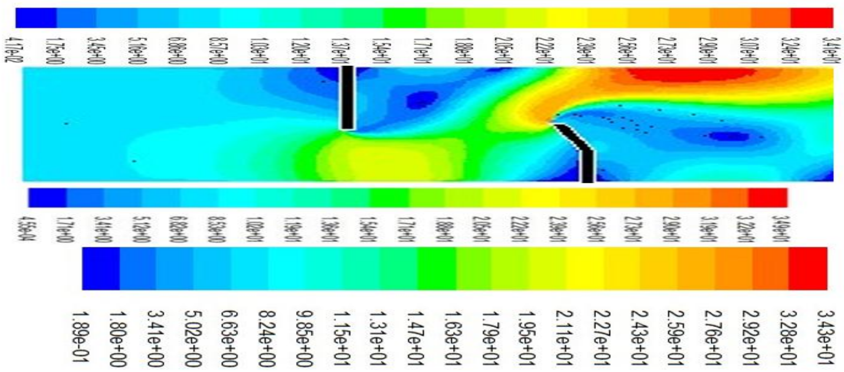


Fig -19 Effect of the shape of the baffles on the Nusselt number for cases a and b.

IV.9.2. Inclined channel θ= 30° and inclined channel θ= 60°



a. Inclined channels ***θ=*** 30° c. Inclined channel ***θ=*** 60° attached to the lower wall of the channel

Fig -20 Velocity vectors for cases a and c.

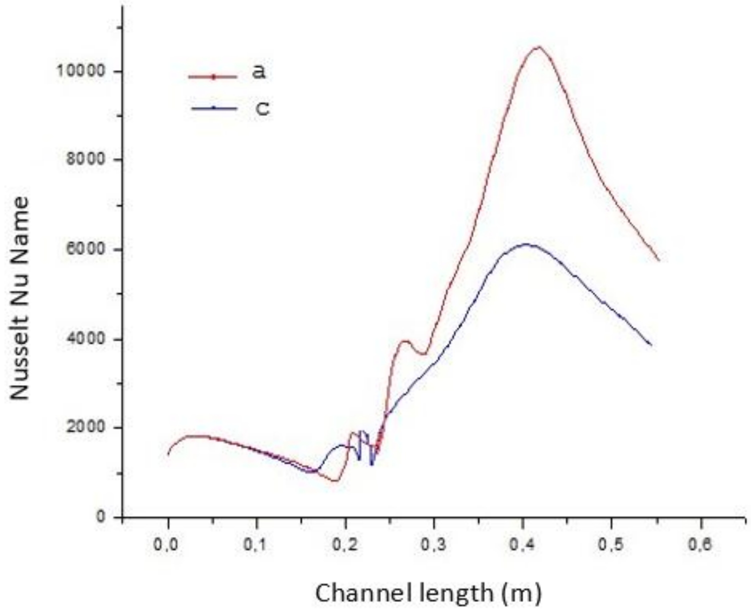
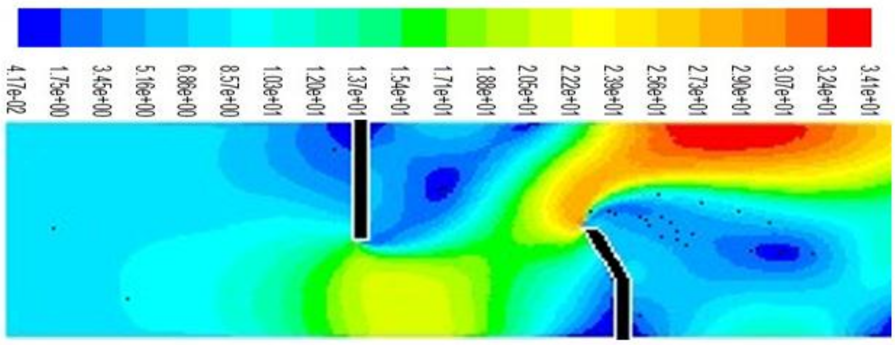
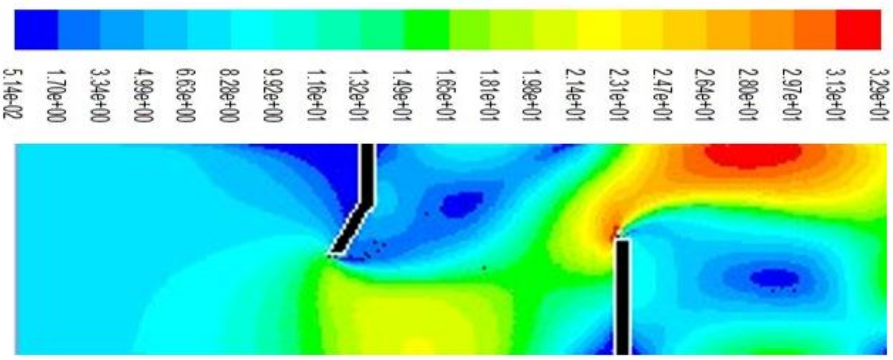


Fig -21 Effect of the shape of the baffles on the Nusselt number for cases a and c.

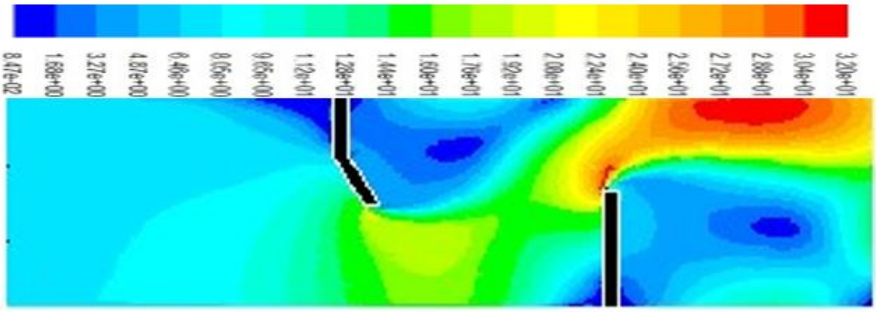
IV.10. Effect of the location and orientation of the angle of inclination



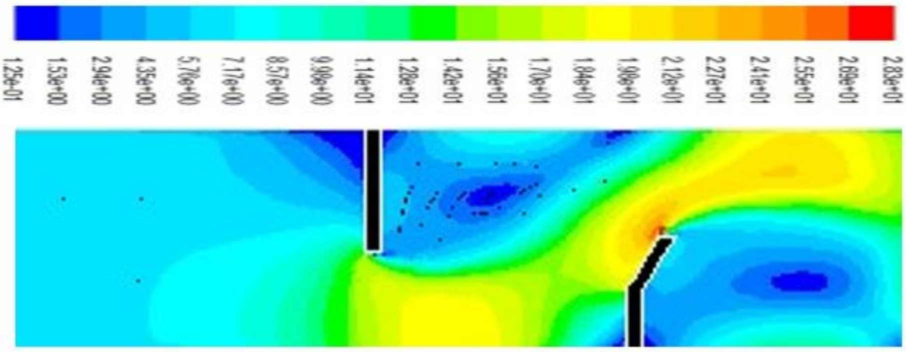
c. Orientation in the opposite direction of flow



d. Orientation in the opposite direction of flow



e. Orientation in the direction of flow



f. Orientation in the direction of flow

Fig -22 Velocity vectors for cases c, d, e and f

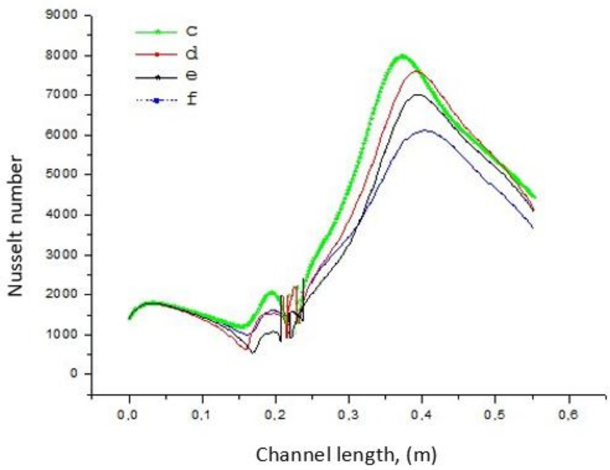


Fig -23 Effect of the location and orientation of the angle of inclination on the Nusselt number for cases c, d, e and f

Fig -23 shows that the appearance of the Nusselt variation is identical in all cases and that the profile is independent of the direction of orientation of the inclined part of the baffles, so the highest Nusselt number corresponds to case c, which allows us to conclude that the orientation effect of the baffles is effectively contributed to the improvement of heat exchange rates.

VI.11. Average Nusselt number:

IV.11.1. Effect of the angle of inclination on the average Nusselt number

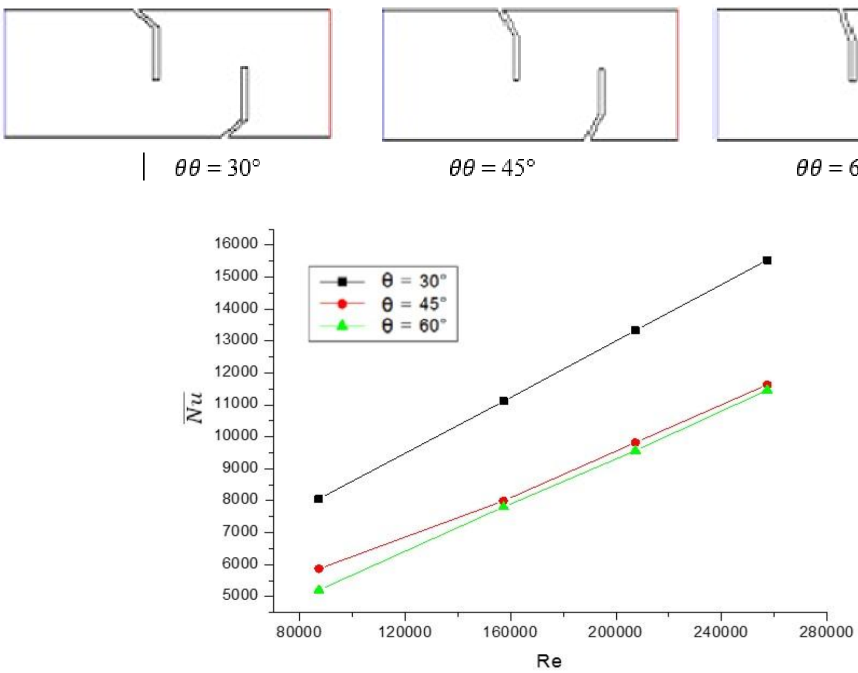


Fig -24 Effect of the angle of inclination θ on the mean Nusselt number NNNN

IV.11.2. Effect of the blocking factor on the average Nusselt number:

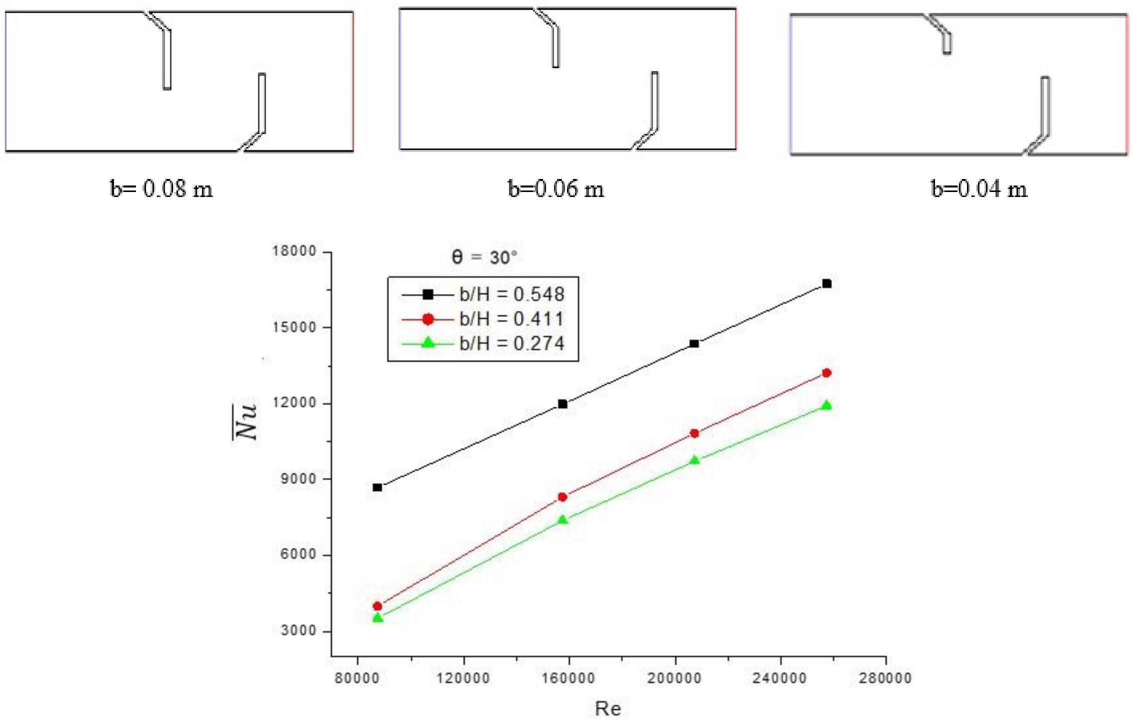


Fig -25 Effect of blocking factor b/H on the mean Nusselt number Nu

Fig -24 and 25 depict the variation of the mean Nusselt number NNNN as a function of the Reynolds number RRRR for various values of the disruptor's angle of inclination and the blocking factor b/H. The figures demonstrate that the average Nusselt number NNNN rises with decreasing disruptor inclination and rising blocking factor b/H.

It is determined that the rise in the mean Nusselt number NNNN values is only caused by changes in the disruptor's and the blocking factor's inclination angles.

Tables 1 and 2, give a comparison between the increase difference of NNNN as a function of the angle of inclination θ and the blocking factor b/H respectively.

Table -1 NNNN increase deviation for different values of θ

⍰	60° - 45°	60° - 30°
Δ (%)	7,34	29 ,89

Table -2 NNNN increment deviation for different b/H values

b/H	0 ,274 - 0,411	0 ,274 - 0,548
Δ (%)	8,60	40,32

IV.11.3. Effect of the channel form on the average Nusselt number

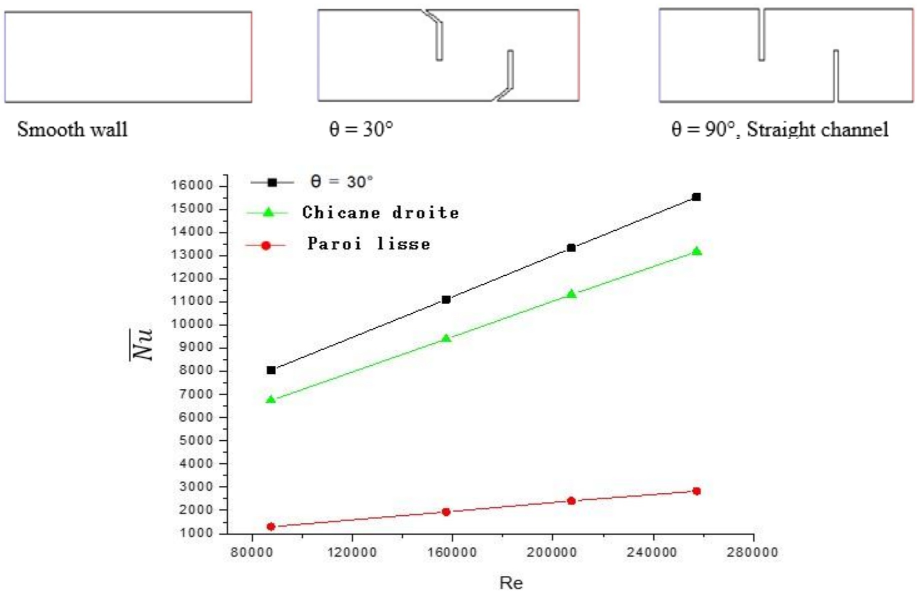


Fig -26 Effect of the channel shape on the mean Nusselt number

Fig -26, shows the effect of the channel shape on the mean Nusselt number Naked, a comparison was made between three different cases of the channel, the first is smooth, the other two have a straight channel θ = 90° and a channel has an inclined part θ = 30° respectively nt, the comparison results clearly state that the channel effect with the inclined part is dominant than the other cases, Table 3.

Case A: Channel with inclined part θ = 30° - Without channel (smooth wall)

Case B: Straight channel θ = 90° - Without channel (smooth wall)

Case C: Channel with inclined part of θ = 30° - Straight channel θ = 90°

Table -3 Deviation from NNNN elevation for different channel shapes

case	A	B	C
Δ (%)	79,4	76,03	14,02

IV.11.4. Ratio of the average Nusselt number:

IV.11.4.1. Effect of the angle of inclination on the ratio of the average Nusselt number:

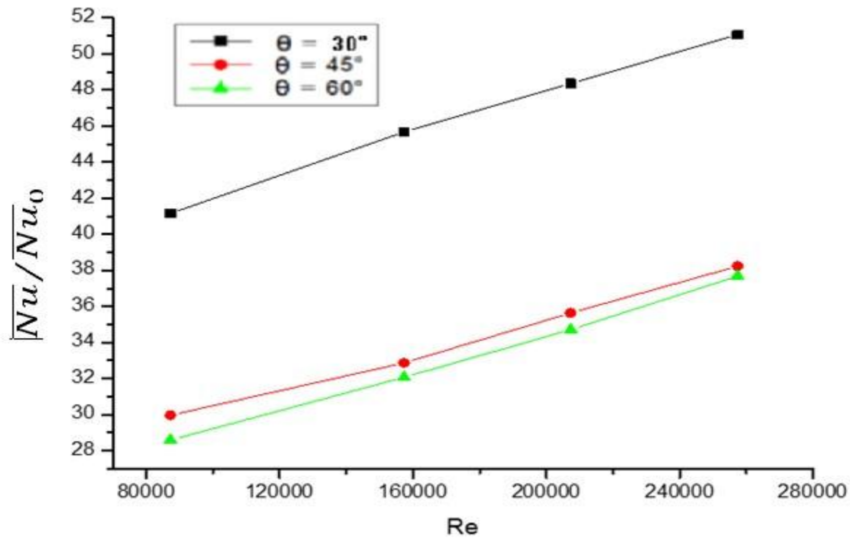


Fig -27 Effect of the angle of inclination θ on the ratio of the mean Nusselt number Nu/Nu_0

IV.11.4.2. Effect of the blocking factor on the ratio of the average Nusselt number:

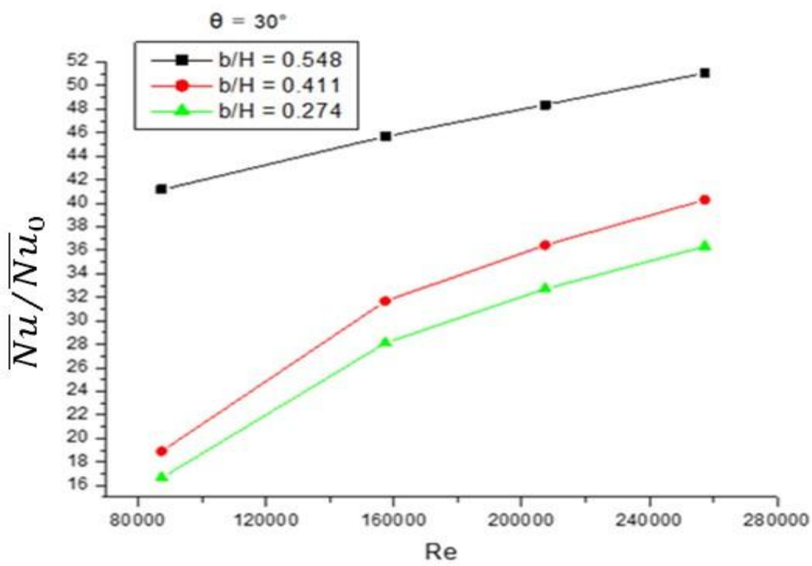


Fig -28 Effect of blocking factor b/H on the ratio of the mean Nusselt number $NNNN/NNNN_0$

The effect of the blocking factor b/H and the inclination rule θ on the mean Nusselt ratio NNNN/$NNNN_0$ is shown in Fig -27 and 28. According to the diagram, as the angle of inclination is reduced, the ratio of the average Nusselt number rises, and as the blocking factor value is reduced, the ratio values of the average Nusselt number decline.

IV.11.5. Coefficient of friction:

IV.11.5.1. Effect of the angle of inclination on the coefficient of friction:

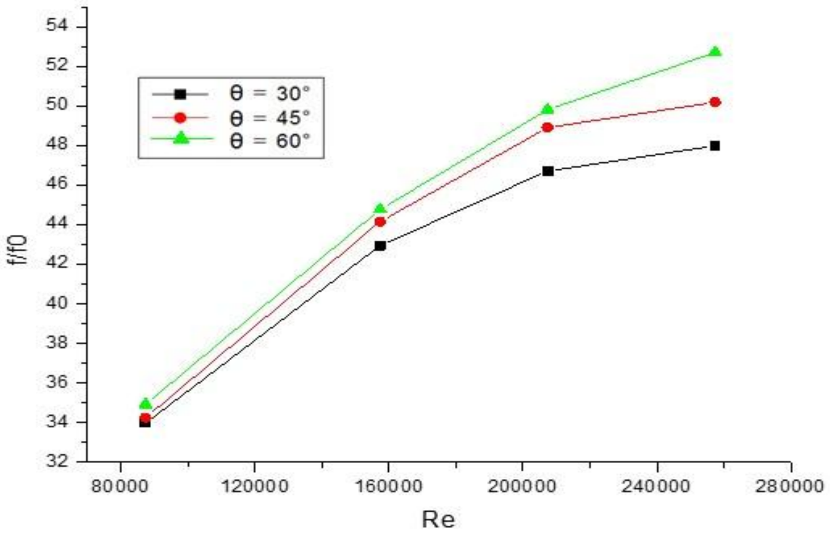


Fig -29 Effect of the angle of inclination θ on the coefficient of friction f/f_0

IV.11.5.2. Effect of the blocking factor on the coefficient of friction:

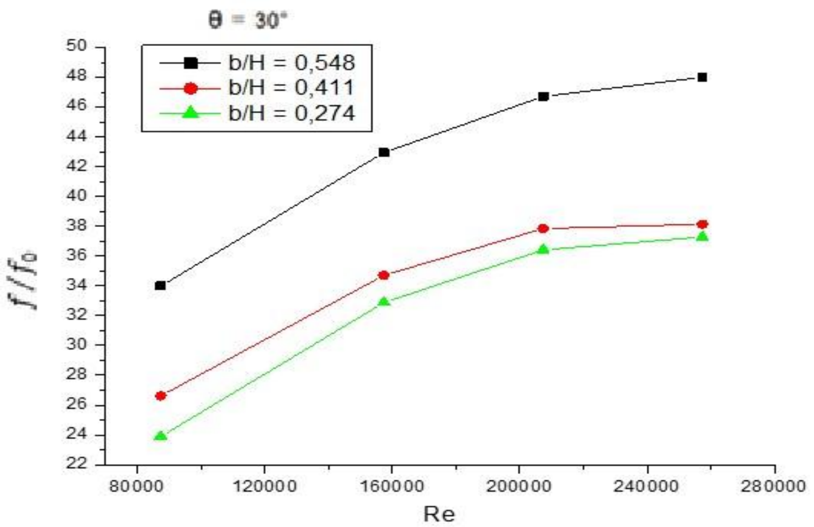


Fig -30 Influence of the blocking factor bb/HH on the friction coefficient ff0

For various values of the angle and the blocking factor bb/HH, the change of the coefficient of friction ff0 as a function of the Reynolds number RRRR.is shown in Fig -29 and 30, it is noted that the fall in the coefficient of friction is displayed during the decrease of the angle of inclination and that the increase of this coefficient is accomplished by the growth of the block factorage.

IV.11.5.3. Effect of the channel shape on the coefficient of friction

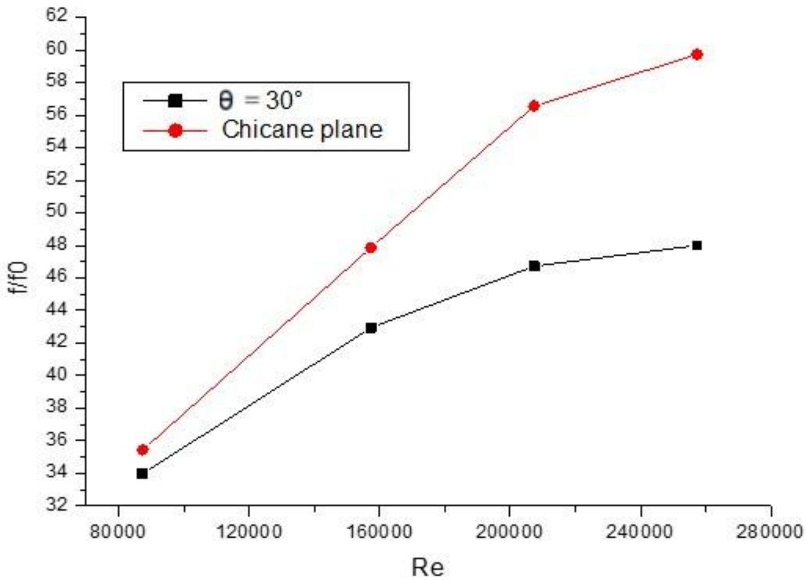


Fig -31 Effect of the channel shape on the coefficient of friction f/f_0

Fig -31, shows the results of comparison between two different cases of the channel shape, one is a flat channel θ = 90° and the other is a channel with an inclined part θ = 30°. The comparison shows that the right channel has a strong effect on the coefficient of friction ff/ff_0.

IV.12. Thermal improvement factor

IV.12.1. Effect of the angle of inclination on the thermal improvement factor

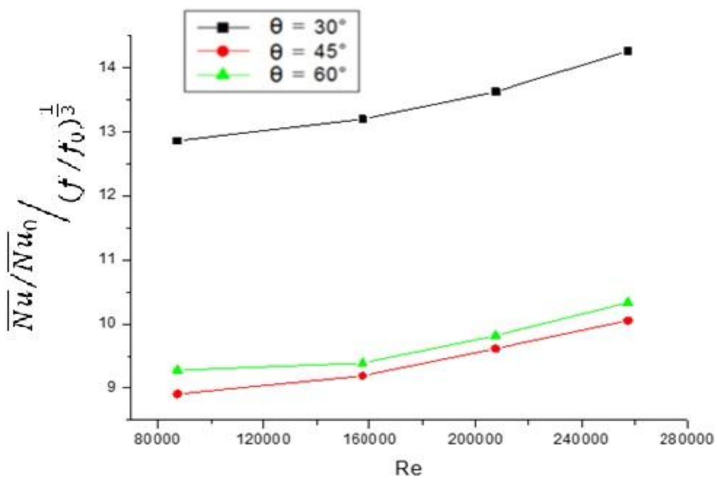


Fig -32: Effect of the angle of inclination θ on the thermal improvement factor ηη

The thermal improvement factor ηη = NNNN/NNNN0 1 is shown in Fig -32, (□□/□□$_0$)3_ this factor increases with the reduction of the angle of inclination

IV.13. Average convective exchange coefficient:

IV.13.1. Effect of the angle of inclination on the average convective exchange coefficient:

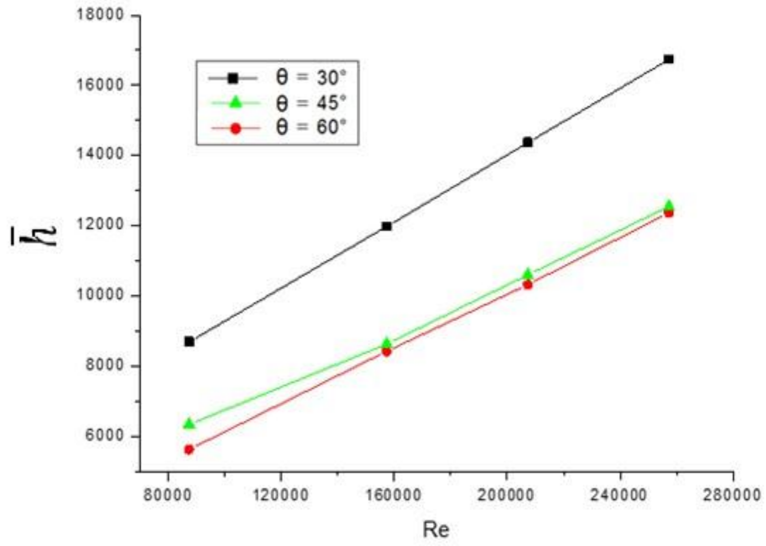


Fig - 33: Effect of the angle of inclination on the mean convective exchange coefficient h

IV.13.2. Effect of the blcage factor on the average convective exchange coefficient:

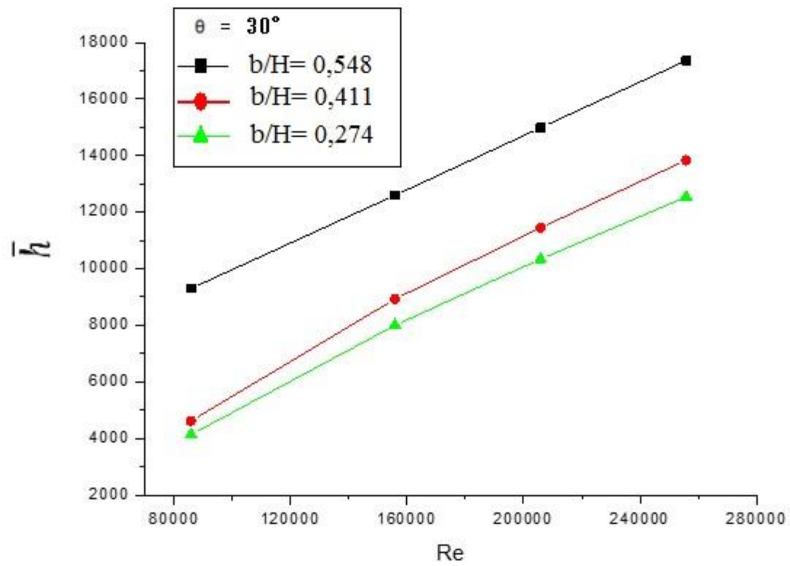


Fig –34 Effect of blocking factor b/H on the mean convective exchange coefficient h

Fig -33 and 34 Show how the average surface exchange coefficient h varies with respect to the Reynolds number when the tilt angle and blocking factor are in play. As can be observed in Fig - 33, the small angles of inclination correlate to the maximum values of the mean convective heat transfer coefficient h. As the slope is dropped, the intensity of the thermal exchange increases. The blocking factor, shown in Fig - IV, has a direct impact on the coefficient of heat transfer via average convection, or h. 35 demonstrates that an increase in the average surface exchange coefficient h, or around 46.33 percent, is implied by an increase in the channels' height.

Conclusion-

The k-numerical model's findings are validated and given in order to evaluate the influence of geometry on the performance of heat exchangers for turbulent fluid flow in forced convection in a rectangular channel with angled baffles. The instability of heat exchanges throughout the channel is greatly impacted by the existence of recirculation zones, which are generated when baffles are present. Negative and positive values of velocities, indicated as an abrupt shift in flow direction, are characteristic of the profile fluctuation of velocities caused by these recirculation zones. Heat transfer efficiency is maximised when the channel's inclined section has a shallow angle of inclination, and the channel's orientation has a significant impact on how quickly heat is transferred.

References

1. Pethkool S, Kwankaomeng, S, Promvonge P. Turbulent heat transfer enhancement in heat exchange using helically corrugated tube. Int Communications Heat Mass Transfer, 2011; 38:340–347.

2. Promvonge P. Heat transfer and pressure drop in a channel with multiple 60° V-Baffles. International Communications in Heat and Mass Transfer, 2010; 37: 335-840.

3. Demartini L. C, Vielmo H. A, Moller S.V . Numeric and experimental analysis of the turbulent flow through a channel with baffle plates. Journal of the Brazilian Society of Mechanical Sciences and Engineering, 2004; 26 (2): 153-159.

4. Bensenouci D, Aliane K, Sari-Hassoun Z. Thermal study of forced convection in a rectangular pipeequipped with baffles, 2nd International Conference of Renewable Energies CIER-2014. International Journal of Scientific Research & Engineering Technology (IJSET), 2015; 3: 123-127.

5. Menni Y, Azzi A, Zidani C. Comparative numerical study between two types of baffles and fins, rectangular and rounded rectangular, used to improve the performance of flat air solar collectors, Renewable Energies, 2015; 18 (3): 347 – 361.

6. Tsay Y L, Chang T S, Cheng J C. Heat transfer enhancement of backward-facing step flow in a channel by using baffle installed on the channel wall, Acta Mech,. 2005; 174: 63–76.

7. Bilen K, Cetin M, Gul H, Balta T. The investigation of groove geometry effect on heat transfer for internally grooved tubes. Applied Thermal Engineering, 2009; 29 (4): 761–769.

8. Nasiruddin M.H. Siddiqui K. Heat transfer augmentation in a heat exchanger tube using a baffle. International Journal of Heat and Fluid Flow, 2007; 28: 318–328.

9. J. Li, S. Wang, J. Chen. Numerical study on a slit fin and tube heat exchanger with longitudinal vortex generators. International journal of heat and mas transfer, 2011; 54: 1743-1751.

10. Wang Y, Liu Z, Huang S. Experimental investigation of shell and tube heat exchanger with a new type of baffles, heat mass transfer, 2011; 47: 833-839.