Experimental Investigation of Heat Transfer Enhancement by Longitudinal Rectangular Vortex Generator

Hrishikesh S. Kulkarni*
Research scholar
G.S.Moze COE,Pune,(MS) India

Sandeep S. Kore
Associate professor
Sinhgad Academy of Engg.,Pune

Abstract— The vortex generators are in the form of a rectangular wing is used in present study.. These vortex generators can be mounted on the fin surfaces by either welding, punching or embossing. These vortex generators induce stream wise longitudinal vortices. These vortices disrupt the growth of the thermal boundary layer and serves to bring about heat transfer enhancement between the fluid and the fin surfaces. The geometrical configuration considered in this study is representative of a plate-fin heat exchanger. Air is taken as the working fluid. The flow regime is assumed to be turbulent because, usually the fin spacing is small and the mean velocity is such that the Reynolds numbers of interest are below the critical Reynolds number. The constant heat flux boundary condition is used. This research work gives a performance data for a rectangular wing in a plate-fin heat exchanger. In order to evaluate the performance, bulk temperature and average Nusselt number are calculated. The heat transfer enhancement is observed with the use of vortex generator. It is observed that the heat transfer increases with the increase in the angle of attack and Reynolds number. The experiments are also performed varying the geometrical size and the angle of attack. For a fixed area, increase in the length gives more heat transfer enhancement then an increase in the height of the wing. Vortex generators with aspect ratios of A = 1, 1.33, 2 are studied at attack angles of 20°,30°and 40°. The results indicate that the stream wise vortices generated by a wing can enhance local Nusselt numbers by more than 200% in a developing channel flow. Under some conditions, the average Nusselt number nearly doubled for a heat transfer area that was 37 to 63 times the wing area.

Keywords— Heat transfer Enhancement, Friction factor, delta wing, turbulator

I. INTRODUCTION

Extensive studies have been conducted on the heat transfer enhancement using turbulators in simulated heat exchangers, turbine blades and so on [1–14]. The turbulator usually generates high turbulence vertical motion in uniform flows, which may change the mean velocity fields, modify the flow turbulence properties and the structures of the near wall layers in the velocity boundary layer. These, in turn, cause Modifications in the heat transfer characteristics. Edwards and Alker [1], investigated convective heat transfer augmentation using three-dimensional surface protrusions in the form of cubes and vortex generators. The cube was found to produce the highest local improvement, but the vortex generator effect extended further downstream. There has been a considerable amount of research in the area of heat transfer enhancement available both in the form of experimental results and as predictions of the numerical investigation. In this paper, a summary of these findings is presented in order to put the present problem in the right perspective.
Lachmann et.al [23] reported that the first recorded study of boundary layer control was performed in 1904 by Prandtl. Immediately following the development of boundary layer theory, Prandalt proceeded to demonstrate how separation could be prevented by removing fluid from the boundary layer. According to Kuethe and Chow [22], Taylor and Bruynes invented the first vortex generators for aircraft wing applications. Research by Wentz and Kohl man [24] has shown that vortex size and strength along the upper surface of a delta wing increases with angle of attack. Regarding the use of wings as a heat transfer enhancement mechanism, the downwash velocity (behind the wing) thins the thermal boundary, resulting in an increase in heat transfer.

Edwards and Alker [4] investigated the use of cubes and delta winglets as vortex generators. Experiments were performed with various generator sizes and spacings at a constant Reynolds number of 61000 (based on generator height). The results indicated that cubes provided a maximum local heat transfer enhancement of 76 percent over the unenhanced plate. The highest enhancement achieved with delta winglets was 42 percent which occulted for a counter-rotating vortex pair.

Russell *et al.[5]* used vortex generators to enhance the performance of a finned-tube heat exchanger. The authors found that, rectangular winglets placed in two staggered
rows gave the best overall performance. A full-scale finned flat-tube heat exchanger was tested with rectangular winglets at a 20· angle of attack. At a Reynolds number of 1000, the j factor was increased approximately 50 percent while the f factor increased about 20 percent.

In 1986, Turk and Junkman [6] evaluated the impact of varying the aspect ratio and angle of attack of rectangular-winglet vortex generators on local heat transfer on flat plates. During the experiments, a known, constant heat flux was applied to the flat plate downstream of the vortex generators. The local surface and air temperatures were measured using thermocouples, and from these measurements, the local heat transfer coefficients were deduced. The authors considered flows with a zero or favorable pressure gradient and reported that, in general, the enhancement increased with a favorable pressure' gradient. Local span wise-averaged heat transfer enhancements as large as 250 percent were reported.
Fiebig *et al.* [7] studied the heat transfer enhancement of delta wings and winglets in flat plate channels for Reynolds numbers based on plate spacing between 1360 and 2270.

In 1989, Biswas *et al.* [8] performed, to the author's knowledge, the fU'St numerical work on vortex-induced heat exchanger enhancement by investigating the impact on mixed convection 'in a rectangular channel. They evaluated the impact of a single delta wing with an aspect ratio of one and angles of attack of 20° and 26°.
In later work, Biswas and Chattopadhyay [9] included the effect of the hole under the delta-wing. For a long channel at Reynolds number of 500, the initial computation without the hole beneath the wing gave an average Nusselt number increase' of 34 percent while the friction factor increased 79 percent. Calculated Nusselt numbers were compared to independent experiments and found to agree within 15 percent.

Numerical study was performed by Brockermeier *et al.* [10] to evaluate the impact of vortex generation in forced convection between Parallel plates. Delta wings and winglets were considered, and the impact of the hole under the wing was included. A delta wing with an aspect ratio of one was considered for attack angles varying from 10° to 50°, while the Reynolds number varied from 1000 to 4000. The computations predicted maximum cross flow velocities in the vortex on the same order as the mean axial velocity. With the delta winglets at a 30° angle of attack and a Reynolds number of 4000, an average increase in the Nusselt number of 84 percent was predicted. .

Another very similar paper by Fiebig *et al.* [11] extended [10] by reporting that there is an axial velocity defect in the vortex core, and this defect allows the vortex to stable at delta-wing angles of attack exceeding *So.*
In 1991, Fiebig *et al.* [12] extended their earlier work: [7] by evaluating the impact of vortex generation in channel flows. Delta and rectangular wings and winglets were evaluated using the unsteady, liquid crystal thermography technique for aspect ratios varying from 0.8 to 2.0, angles of attack varying from 10° to 60°, and Reynolds numbers varying from 1000 to 2000.

Using the same flow visualization and unsteady liquid crystal thermography techniques, this research was further extended to include two aligned rows of delta winglet [13]. The authors reported that the, "qualitative flow structure, the number of developing vortices per vortex generator, and their stream wise development were found to be nearly independent of the oncoming flow of the vortex generator, (uniform or vortical)." For a Reynolds number of 5600, local heat transfer enhancements of several hundred percent were reported, and the average heat transfer was increased 77 percent by two aligned rows of vortex generators. No pressure drop data' were recorded.

Tiggel beck extended this multiple row vortex generation work in 1993[14] by including staggered vortex generators and pressure drop experiments. Again, the qualitative flow structure, number of vortices per generator, and stream wise development were reported to be nearly independent of upstream flow conditions.The inline arrangement gave slightly higher heat transfer enhancements than the staggered arrangement, but the inline pressure drop was also higher than the staggered arrangement. At a Reynolds number of 6000, the average heat transfer was increased 80 percent for an angle of attack of 45°.Fiebig *et al.[15,* 16] evaluated the impact of a delta-winglet pair on heat transfer in channel with a tube Experiments were conducted for Reynolds numbers ranging from 2000 to 5000, and the local heat transfer coefficient was reported to increase by 200 percent when compared to heat transfer without the vortex generators. The average heat transfer coefficient increased by up to 20 percent, while the drag decreased by as much as 10 percent. According to the authors, "The reduction in flow losses can be explained by the delayed separation on the tube due to the strong counter rotating longitudinal vortices generated by the delta-winglet pair which introduced high momentum fluid into the region behind the cylinder." delta-winglet vortex generators studied by Fiebig *et aI.[15,* 16].

The Fiebig and Sanchez [17] conducted a numerical study of the geometry and reported that at a Reynolds number of 1200, the fin with the delta-winglet pair performed the same as a fin without generators at Reynolds number of 2000. For a fin-tube element at a Reynolds number of 2000, the winglets can allow a reduction in pumping power of 80 percent for a constant heat duty or an increase in heat transfer of 25 percent for fixed pumping power.

Similar numerical study by *Biswas et al.* [18] provided a comparison. Numerical results from this study were compared to the experimental work of Valencia *et al.* [19]. Nusselt numbers that the authors explained were caused by the simulated vortex strength being higher than that of the experiment. The effect of vortex generation. With multiple tubes in a channel has also been investigated [19. 20].

Using delta-winglet pairs in the wake region of the tube. The heat transfer was increased 55-65 percent for inline tubes and approximately 10 percent for staggered tubes. The friction factor for the inline tubes increased 20 percent at a Reynolds number of 600 and it increased 44 percent at a Reynolds number of 2600. Fiebig *et al.* [21] extended this work to evaluate the impact of placing delta-winglet vortex generators near flat tubes. Experiments were conducted over a range of Reynolds numbers from 600 to 3000. The overall heat transfer enhancement for flat-tubes with delta-winglet vortex generators was reported to be approximately 100 percent, with the corresponding pressure drop penalty also equal to 100 percent.

II. Experimental Set-Up

The test apparatus is an open air flow loop type Consist of centrifugal blower, flow control valve, orifice meter (for flow measurement), an entrance section, the test section and mixing chamber (exit section). The duct is of size 1060mm * 100mm* 25mm and is constructed from epoxy resin of 5mm thickness. The test section is of length 600mm. The entry and exit section length are 400mm and 250mm respectively and is made up of 5mm thick acrylic sheet. The exit section of 250mm is used after the test section in order to reduce the end effect, and to get uniform temperature across the duct. A 400mm acrylic entrance section provides hydro-dynamically fully developed flow at the test section entrance. To connect rectangular cross section to circular cross section plenum chamber is used, in between exit section and orifice meter.

To provide uniform heat flux electric heater of size 1000mm*100mm maintained wattage of 125 Watt is fabricated from galvanizing iron sheet of 0.5mm thickness of 1 amp and 230 volts. The heater is located at the bottom of plate and is covered by the asbestos sheet of size 600mm*100mm*5mm to avoid the heat loss from bottom side of heater. Heat input to the heater is controlled by the use of dimmer range of 0 to 230 volts. The voltage across heater is measured by digital voltmeter.

The whole periphery of the test section is covered by glass wool layer of 6cm thickness and aluminum foil, in order to minimize the heat loss to the surrounding by radiation. The mass flow rate of air is measured by means of an orifice meter and the flow is controlled by the gate valve provided in line. Pressure drop across the orifice meter was measured by a U- tube manometer with water as a manometer fluid. The pressure drop across test section is measured by Differential U- tube manometer, with double reservoir filled with benzyl alcohol and water.
For temperature measurement 22 T-Type thermocouple (24 SWG) are used at inlet and outlet and the test plate. Out of 22 thermocouples one thermocouple measure the inlet temperature, three used for exit temperature measurement and eighteen thermocouples are embedded in test plate through 2mm drilled hole of depth 5mm from side wall of plate through 2mm drilled hole of depth 5mm from side wall of plate. Among ten, six thermocouples embedded in plate from one side and remaining from other side of the test plate to measure average surface temperature. Two thermocouples are used at inlet and outlet of test duct to measure air temperature. A digital mili-voltmeter is used for measurement of thermocouple output through the selector switch.

III Experimental Procedure

The test section is assembled in test bracket and checked for air leakage. The blower was switched on to let a predetermined rate of air flow through the duct. A constant heat flux is applied to the dimpled surface. The net heat flux and the average test surface to bulk mean air temperature difference was determined over test section. To collect relevant heat transfer and flow friction data test runs were conducted under steady state condition which is assumed to be reached when the temperature at a point does not change for about 20 minutes. After starting it from cold it took around 45 to 60 minutes to gain the system steady state.
Six valves of flow rates were used for each set at same or fixed uniform heat uniform heat flux. At each valve of flow rate and the corresponding heat flux, system was allowed to attain a steady state before the temperature data were recorded. The pressure drop in test section is measured in cold state when heater is off.

III Results

During Experimentation the following parameters were measured:
1. Pressure difference across the orifice meter.
2. Temperature of the heated surface and temperature of air at inlet and outlet of the test section.
3. Pressure drop across the test section using differential manometer.

IV **Validation of Experimental Set up**

The basic purpose of validation is to serve as a basis of comparison of result of heat transfer and friction factor from correlation available. Before the actual experimentation flat plate is tasted.The experimental data of Nusselt number and friction factor for presented in following Table 1

Table No-1 Temperature difference at various Re

Reynolds No,(R_e)	Temp at inlet (T_{in})^{0}C	Temp at outlet (T_{out})^{0}C	Temp at plate (T_{plate})^{0}C
9000	33.78	38.87	101.00
12000	34.3	38.59	88.26
15000	34.4	38.32	81.49
18000	34.5	38.10	76.83
21000	34.6	37.78	73.06
24000	34.7	37.98	70.49

The thermo physical properties of air used in calculation of heat transfer are taken from air table corresponding to bulk mean temperature.The Dittus Boelter equation for Nusselt number is valid for the Reynolds number range above 5000.

$$Nu_{th}=0.023\ Re^{0.8}\ Pr^{0.4}$$

Table No:2 Experimental results of wings

Re	$T_{in}\,^0C$	$T_{out}\,^0C$	ΔT_1	T_{plate}	T_{bm}	ΔT_2= T_p-T_{bm}	m	Q= m*Cp* ΔT_1	h= Q/(A*dT_2)
9000	33.8	38.87	5.12	101	36.7	63.6	0.01	60.51	17.8
12000	34.3	38.59	4.29	88.26	36.4	51.82	0.02	67.08	21.6
15000	34.4	38.33	3.92	81.49	36.4	45.13	0.02	76.72	28.3
18000	34.5	38.1	3.6	76.83	36.3	40.53	0.02	84.44	34.7
21000	34.6	37.79	3.18	73.06	36.2	36.87	0.03	87.14	39.4
24000	34.7	37.98	3.28	70.49	36.3	34.15	0.03	102.58	50.1

Table No-3 Comparison of theoretical and experimental Nu

R_e	Nu_{expt}	Nu_{th}
9000	26.36	30.40
12000	31.96	38.27
15000	41.97	45.75
18000	51.44	52.94
21000	58.36	59.89
24000	74.16	66.64

Table No 4 Friction factot at various Re

R_e	f_0
9000	0.000862
12000	0.000803
15000	0.000759
18000	0.000725
21000	0.000698
24000	0.000675

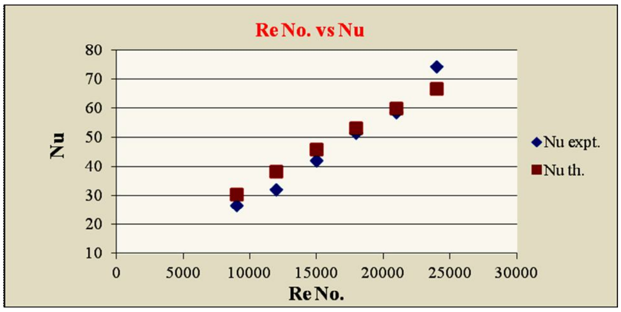


Fig.1: Nusselt Number Vs Reynolds Number for flat plate

v. **Friction factor validation**

Friction Factor is successfully measured by the U –tube differential manometer. Friction Factor is validated using Blasius equation.

$$f = 0.085\ Re^{-0.25}$$

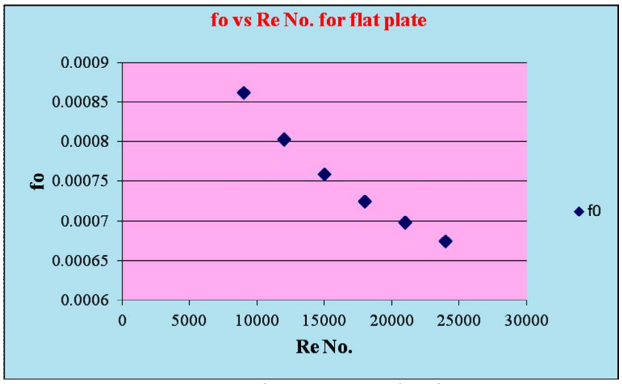


Fig.2: Friction factor vs Re for flat plate

VI **Comparison of Heat transfer performance for different aspect ratios for the each angle of attack**

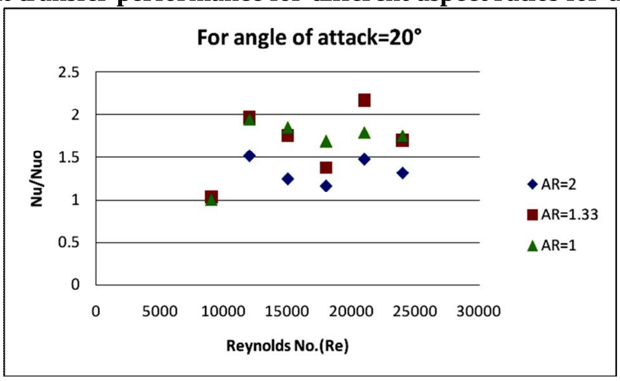


Fig.3: Variation of Nu function of Re at angle of attack=20^0

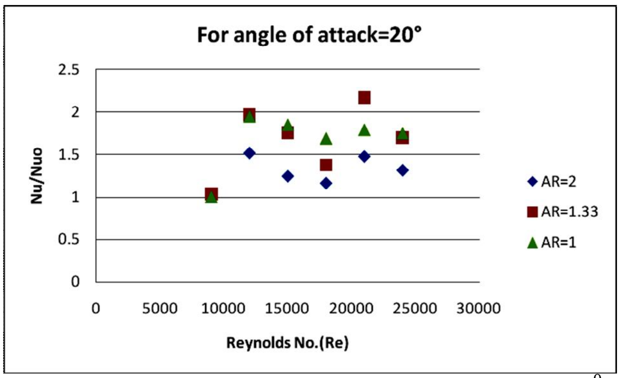


Fig.4: Variation of Nu vs Re at angle of attack=30^0

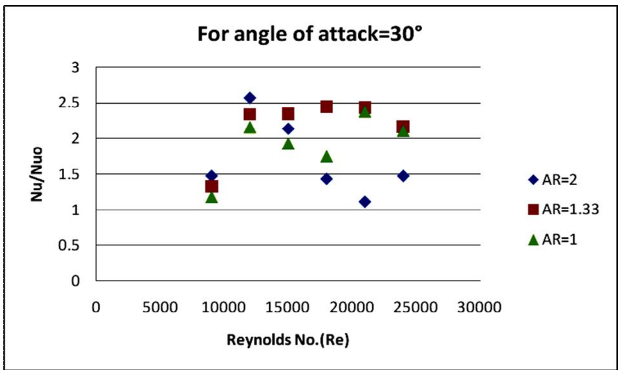


Fig.5: Variation of Nu vs Re at angle of attack=45^0

Similarly for different angle of attacks different results are observed.

1) For angle of attack 30^0, Nu/Nu_o is 1.22, 1.3 and 1.4 for AR=1, 1.33, 2.

2) For angle of attack 40^0 , Nu/Nu_o is 1.10,1 and 0.90 for AR=1, 1.33, 2.
3) For angle of attack 20^0, Nu/Nu_o is 0.98, 0.99 and 1.03 for AR=1, 1.33, 2.

VII. **Comparison of Heat transfer performance for different angle of attack for the each aspect ratios**

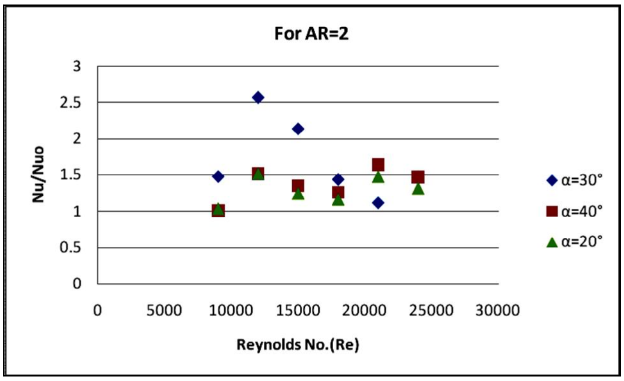


Fig.6: Variation of Nu as function of Re for AR=2

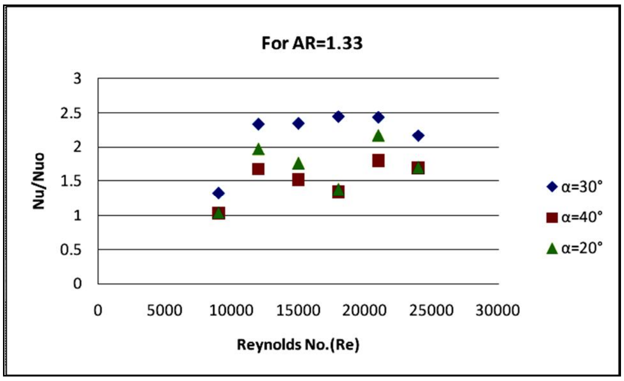


Fig.7: Variation of Nu function of Re for AR=1.33

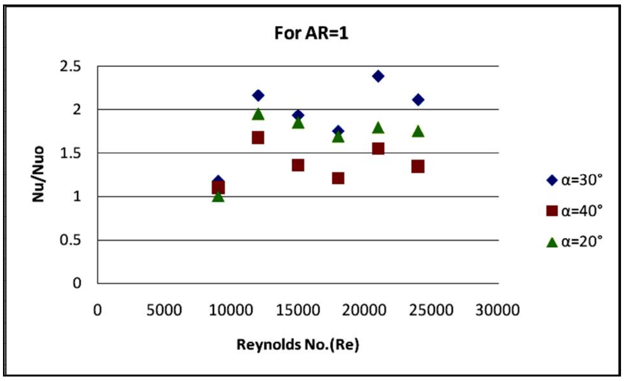


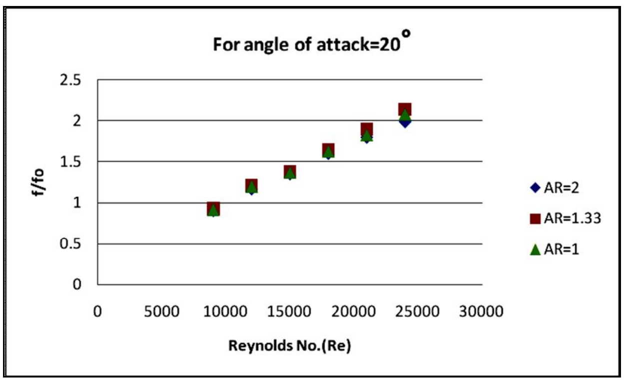
Fig.8: Variation of Nusselt Number as function of Reynolds Number for aspect ratio=1

From above graphs following results are obtained.
1) For AR=1, Nu/Nu_o is 2.4
2) For AR=2, Nu/Nu_o is 2.6
3) For AR=1.33, Nu/Nu_o is 2.5

Similarly,

1) For AR=1, Nu/Nu_o is 1, 1.2 and 1.3 with respect to angle of attack 20^0, 30^0 and 40^0
2) For AR=1.33, Nu/Nu_o is 2, 1.8 and 2.5 with respect to angle of attack 20^0, 30^0 and 40^0
3) For AR=2, Nu/ is 1.5, 2.6 and1.6 with respect to angle of attack 20^0, 30^0 and 40^0

VII. **Comparison of Friction factor performance for different aspect ratios for the each angle of attack**



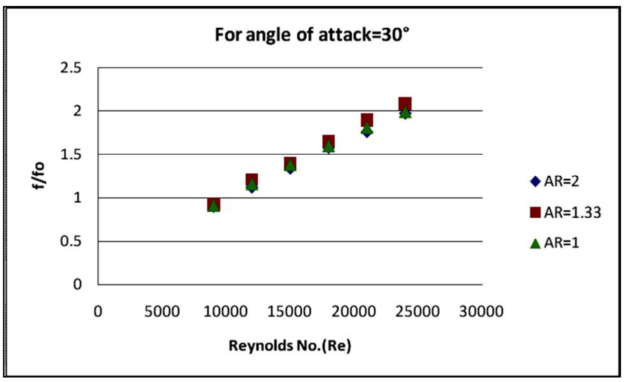


Fig.9: Variation of friction factor as function of Re for angle of attack=30^0

Fig.10: Variation of Friction factor as function of Re for angle of attack=20^0

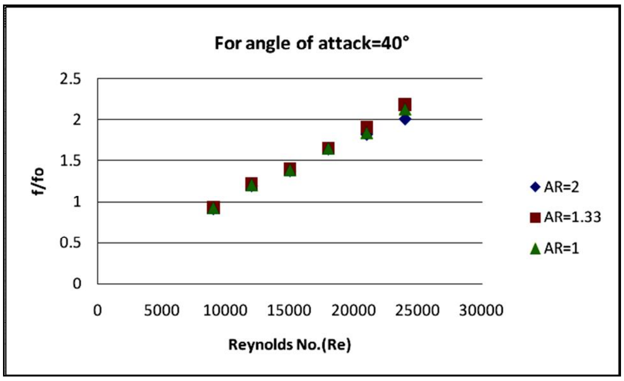


Fig.11: Variation of Friction factor as function of Re for angle of attack=40^0

From above Figures following results are observed.
1) For 20^0, 30^0, 40^0 angle of attack friction factor is higher than flat plate.
2) Lowest friction factor is observed for AR=1 or 2
Similarly,
1) The normalized friction factor for 30^0 angle of attack is 1.3, 1.4 & 1.6 for AR=1, 1.33 and 2 respectively.
2) The normalized friction factor for 40^0 angle of attack is 1.6, 1.7 and 1.5 for AR=1, 1.33 and 2 respectively.
3) The normalized friction factor for 20^0 angle of attack is 1, 1.5 and 1.4 for AR=1, 1.33 and 2 respectively.

VIII **Comparison of Heat transfer performance for variation of friction factor as a function of Re for different AR**

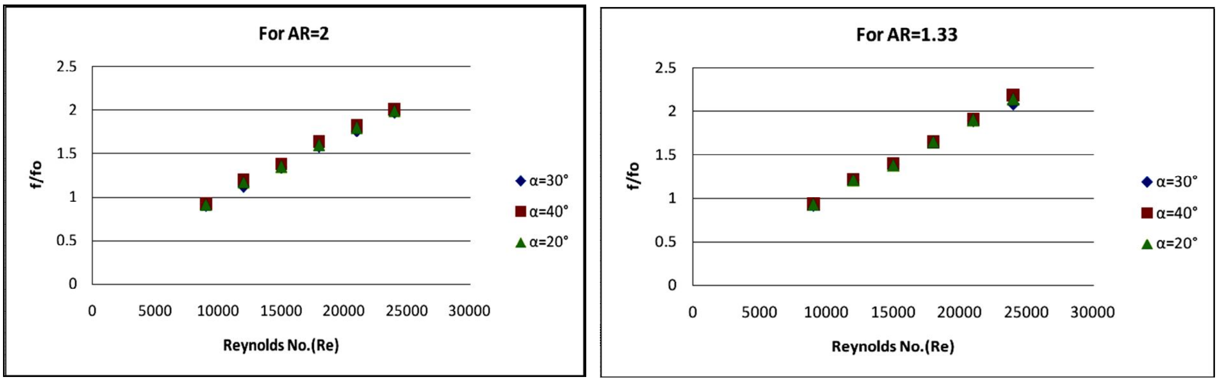


Fig. 13: Variation of Friction factor as function of Re for AR=1.33 and AR= 1.33

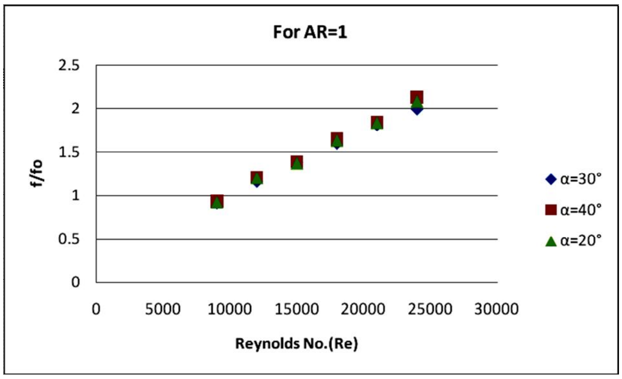


Fig.14: Variation of Friction factor as function of Re for AR=1

From above Figures following results are observed.
1) The friction factor ratio for AR=1 is 1.5
2) The friction factor ratio for AR=1.33 is 1.6
3) The friction factor ratio for AR=2 is 1.8 hence it is higher for AR=2
Similarly,
1) With respect to aspect ratio 1 the f/f_o for $\alpha=20^0$, 30^0, and 40^0 is 1.5, 1.6 and 1.7 respectively.
2) With respect to aspect ratio 1.33 the f/f_o for $\alpha=20^0$, 30^0, and 40^0 is 1.6, 1.7 and 1.5 respectively.
3) With respect to aspect ratio 1 the f/f_o for $\alpha=20^0$, 30^0, and 40^0 is 1, 1.5 and 1.4 respectively.

IX. CONCLUSION

An experimental investigation of air flow in rectangular channel with the wings i.e. system as a longitudinal vortex generator with uniform
heat input has been performed with different aspect ratios and different angle of attacks.
Investigation is done for 9 different conditions of aspect ratios and angle of attacks.

1. As R_e increases Nusselt Number also increases for all geometries.
2. The Nu & Nu/N_{uo} is higher for all geometries compared to flat plate.
3. If three wings of AR=1, 1.33, 2 are compared AR=2 are having higher Nu/Nuo of 1.45.
4. The Nu/N_{uo} is 1.3, 1.4, 1.5 for AR=1, 1.33, 2 respectively.
5. The Nu/N_{uo} with respect to angle of attack is 2, 2.1, 3 for 20^0,30^0,40^0 angle of attack.
6. The thermal performance is higher for $\alpha=30^0$ with AR=2
7. As far as angle of attack is concerned for 30^0 best results obtained.
8. Heat transfer enhancement is about 30 to 40% average compared to flat plate.

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