AIR FLOW INDUCTION BY NATURAL CONVECTION IN A CONVERGENT VERTICAL CHANNEL

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***Abstract**. Experiments were conducted to analyses the effects of thermal power, aspect ratio and convergence angle of one of the walls of a vertical channel with an asymmetrically heated flat plate heat exchanger, fixed at discrete points of the inclined wall, on the velocity of the air flow induced by natural convection at the entrance region of the channel. The results of the experiments were correlated as functions of Nusselt number (Nu), Rayleigh number (Ra) and Richard number ($Ri = Gr/Re^2$). The study of the variation of Ri with the aspect ratio synthesize the analysis of the parameters used in the research, and it confirmed his relevance in analyses of the phenomena governed by the natural convection. Comparing with the values obtained to the parallel wall configuration, an enhancement of the order of the 12% of the air flow velocity was identified when the channel's wall adopted the convergent configuration, showing the effect of the improvement of the heat exchange process. Such results are important to the cooling of electronic equipments, grain drying or thermal comfort.*

***Keywords**: Natural convection, vertical channel, convergent walls, air flow.*

1. INTRODUCTION

During the 1960's, part of the researches on natural convection was concentrated on the early problems of energy dissipation inside small and narrow spaces, as a consequence of the necessity of the integration of electronic components. The results of those researches were directly applied to the next problem of electronic industry – the heat transfer inside personal computers. Despite the proportions, some results of the researches concerned about the heat transfer inside the computers can be applied to the problem of heat transfer in channels and associated with the concerns of the Architecture on thermal comfort. Other applications are the industrial ventilation and the grain dehumidification to preserve the agricultural crops.

In low latitudes countries, the solar energy is the main responsible for the thermal charge inside the habitations. In Brazil, country of intense agricultural activity and high levels of solar radiation, the natural convection could play an important rule on heat transfer processes, especially on those related with grain storage, thermal comfort and ventilation. In any of these situations, natural convection could be used to promote energy efficiency.

The ventilation induced by natural convection is applied to the removal of gases generated in industrial process, as well as in Architecture, which deals with the problem of energy efficiency, in-door air quality and human comfort. Even with its low efficiency, the natural convection is also an important alternative to the dehumidification of products of Agriculture developed by the peasants located in regions where there is no electricity (Dalpasquale *et al*. 1987).

Almost the totality of the researches about applied natural convection is concentrated on the problem of the increase of the heat transfer coefficient ("h"). Usually, the experimental investigations are made in wind tunnels, where the air flow is controlled to the analysis of the influences of parameters such as surface conditions, geometry of the surfaces, inclination and thermal heating power on the h-coefficient. There is a very small number of papers concerned with the problem of air flow induction.

Huang *et al*. (1995) and Gau *et al.* (1996), in experimental investigations, analyzed the influence of the reverse flow on the heat transfer by mixing convection (natural and forced). Muhanna and Said (1992), in a numeric work, consider the patterns of velocity and temperature of air flow induced by natural convection in convergent and divergent vertical channels. Huang *et al*. (1995) found that the stability of the air flow inside a vertical channel is dependent of the distortions caused by the influence of heating power, probably caused by the reverse flow.

There is a lot of works that suggest that the channel inclination could be the key to avoid reverse flow (Oliveira Neto and Marinho, 2002). In the present work, we analyses the influences of the asymmetric inclination in a convergent channel, submitted to asymmetric heating at discrete points, on the velocity of the air induced by natural convection at the entrance region of the channel. This configuration was choosing due to its similitude with a channel (or chimney) heated by solar energy and applied to ventilation by natural convective air flow induction in habitations.

2. METODOLOGY

As the analysis of natural convection is a complex phenomenon, we follow the recommendations of Marinho and Celere (1997) to the experimental procedures. The vertical channel was assembled in a chamber to guarantee an environment which had thermal stabilization. The walls of the channel were built with compensated plywood, with thickness of 30 mm. In the figures 1 and 2 can be seen the other dimensions of the channel.

As the objective of the study was the investigation of the influence of the geometry and the inclination of one of the walls of the channel on flow velocity, four angle of inclination and three positions of the heat exchanger relative to the walls were considered, resulting in twelve aspect ratio configurations, i.e., $0.74 < AR < 4.65$. The junctions of the walls were blocked with 3M tape, to guarantee no leak of air. Ultrasonic equipment was used to prove the efficiency of the method.

Fig. 1. Vertical flow channel: (a) heat exchanger; (b) thermocouples; (c) wood plates; (d) Thermometers; (e) duct to the flow meter.

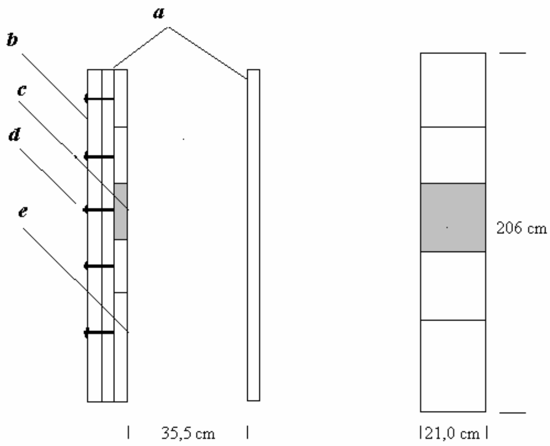


Fig. 2. Vertical flow channel dimensions and materials: (a) Compensated wood plate; (b) Wood; (c) Heat exchanger; (d) Nylon screw-nut; (e) Flow stabilization plate.

In table 1 are presented the data of the fixation point (Y) of the heat exchanger on the channel wall, the inclination angles (θ) and the aspect ratio (AR) considered in this study. The aspect ratio was determinate by the follow relation:

$$AR = (Y/L)/S_i/S_o) \quad (1)$$

Where: AR = aspect ratio
Y = fixation point of the heat exchanger (cm)
S_i = in-section of the channel (cm^2)
S_o = out-section of the channel (cm^2)

Tab. 1. Parameters of the channel wall inclination and heat exchanger positions.

θ	Y1 = 165 cm			Y2 = 135 cm			Y3 = 105 cm		
	Y/L	Si/So	AR	Y/L	Si/So	AR	Y/L	Si/So	AR
0º (*)	4.65	1.00	4.65	3.80	1.00	3.80	2.96	1.00	2.96
3.1º	6.60	1.43	4.33	4.78	1.43	3.35	3.52	1.43	2.46
5.8º	11.0	2.35	3.74	6.19	2.35	2.63	4.22	2.35	1.79
8.7º	33.0	7.26	2.16	8.90	7.26	1.23	5.36	7.26	0.74

(*) parallel walls.

The heat exchanger was built with a stainless steel plate. An electric resistance (50.0 Ω ± 0.50Ω, chrome–nickel alloy, electrically insulated with alumina) was fixed inside the plate to promote warming by Joule effect.

The surface of the plate was polished and three thermocouples were fixed with special cement. In figure 3 can be seen the heat exchanger, its dimensions and the arrangement of the thermocouples.

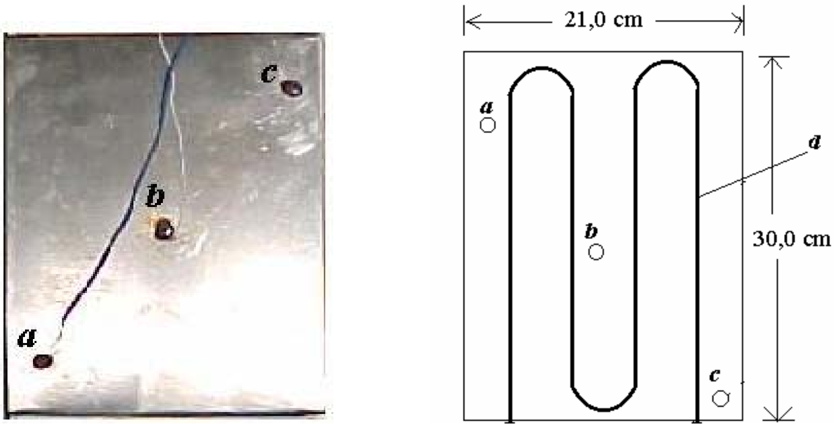


Fig. 3. Heat exchanger with thermocouples (a, b, c) and electrical resistance (d).

The electrical resistance of the heat exchanger was heated by a stabilized DC power supply, which has an automatic cross-over system to compensation the variations of the energy and guarantee the linearity of the relation between current and voltage.

The temperature of the heat exchanger and environment (entrance and exit regions of the channel, and stabilization chamber) were measured with T-type thermocouples (copper-constantan). The thermocouples were connected to a data acquisition system, which made the conversion A/D of the signal of the probes to the computer.

In table 2 are presented the values of U (V), I (A), thermal power (W) and the correspondent mean temperature (°C) reached by the surface of the heat exchanger.

Tab. 2. Values of DC-power supply and heat exchanger temperature.

Voltage (V)	Current (A)	Thermal power (W)	Surface mean temperature (°C)
30.6	0.67	20.5	55.4
34.0	0.74	25.2	60.7
38.0	0.83	31.5	67.6

To the measurement of the air flow, it was used a hot wire anemometer, with calibration certificate and an uncertainty of ± 1.5 cm/s, which has an telescopic probe that smoothed the traverse between the control room and the vertical channel.

During the preliminary experiments, it was determinate a time of 2 hours of heating of the heat exchanger to the stabilization of the natural convective flow in the channel. However, to guarantee the accuracy of the measurements, it was adopt a time of 3.0 hours to the air flow stabilization.

3. RESULTS AND ANALYSES

In table 3 can be observed the symbology of the configurations considered in the analyses, presenting the angle of the bended wall and the position of the heat exchanger in the wall. For example, the code A0Y1 represents the 0° angle and the “1” position, corresponding to the configuration of the channel which the walls are parallel and the heat exchanger was fix in the upper position (165 cm).

Tab. 3. Codes of the configurations analyzed.

Code	Angle (°)	Heat exchanger position (cm)
A0Y1	0	
A1Y1	3.1	165
A2Y1	5.8	
A3Y1	8.7	
A0Y2	0	
A1Y2	3.1	135
A2Y2	5.8	
A3Y2	8.7	
A0Y3	0	
A1Y3	3.1	105
A2Y3	5.8	
A3Y3	8.7	

3.1. Variation of the air flow as a function of the position

In figure 5 are presented the variation of the air flow as a function of the position of the heat exchanger.

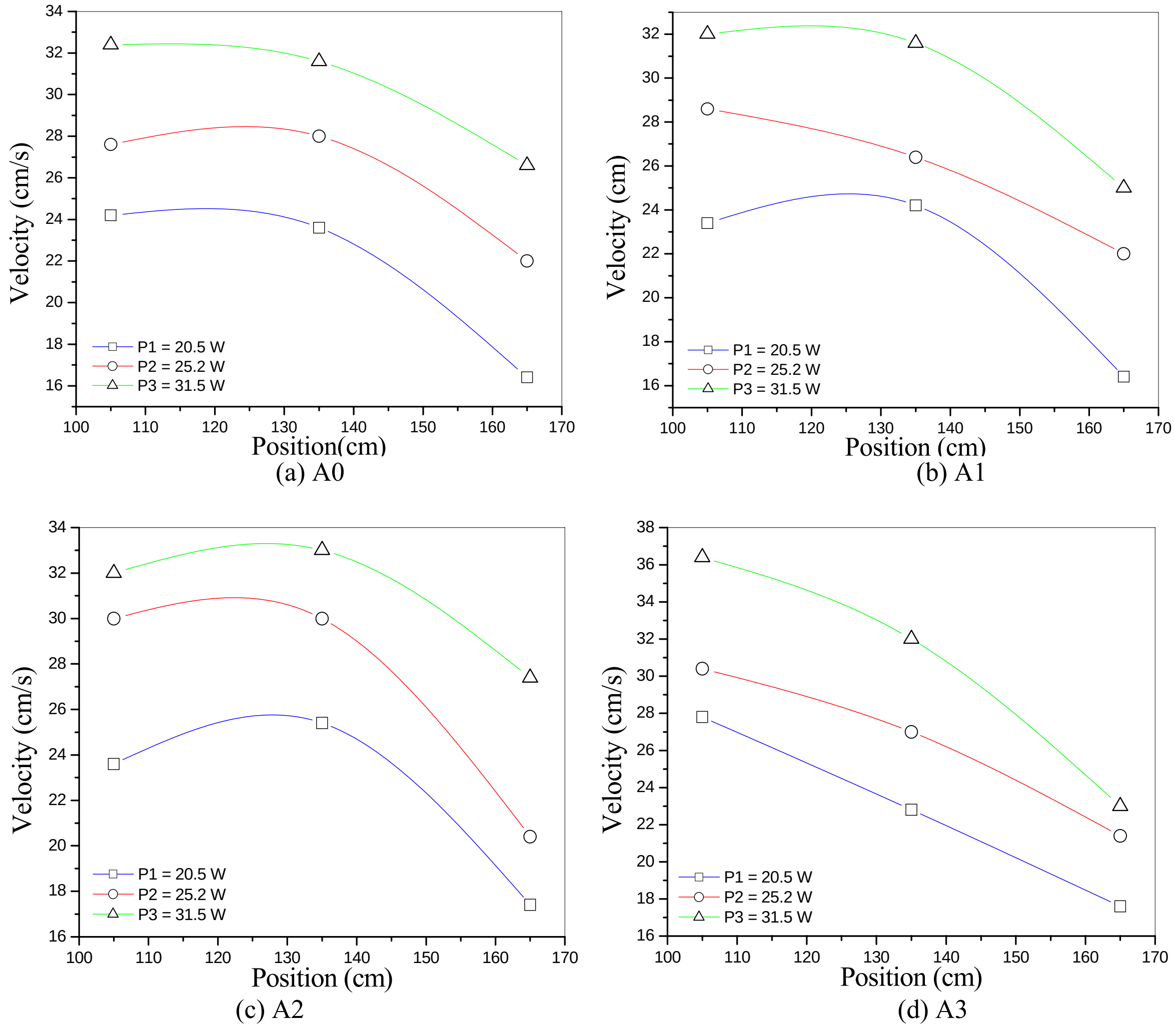


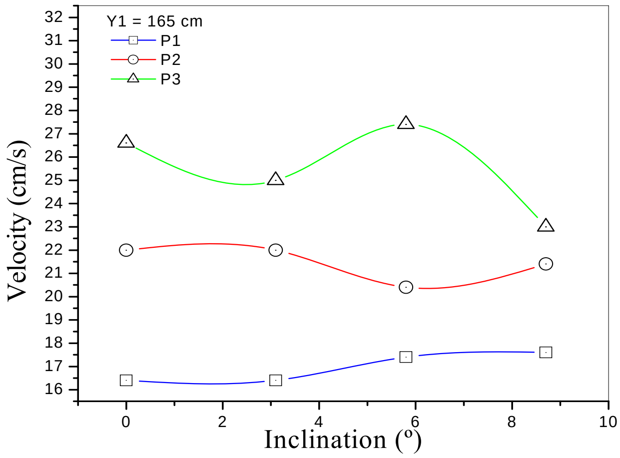
Fig. 5. Variation of the air flow as a function of the heat exchanger position.

Except for the angle A3 (8.7°), in all configurations there was identified a tendency of increasing of the velocity when the heat exchanger was fixed at the intermediate position Y2 (135 cm).

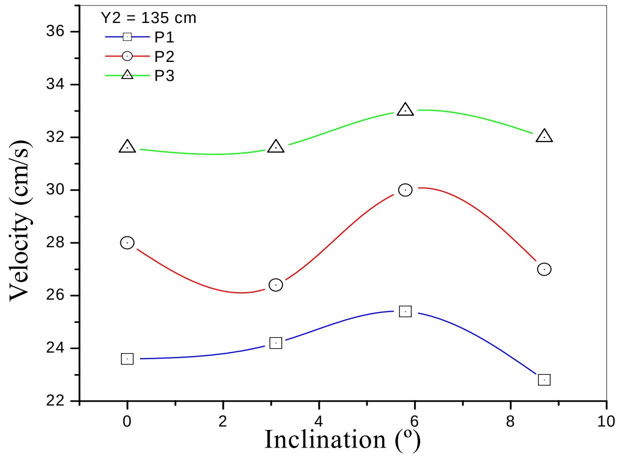
3.2. Variation of the air flow as a function of the convergence angle

In figure 6 are presented the variation of the air flow as a function of the angle of the bended wall. It was identified a tendency of decreasing of the air flow as the angle changes from A0 (0°) to A1 (3.1°).

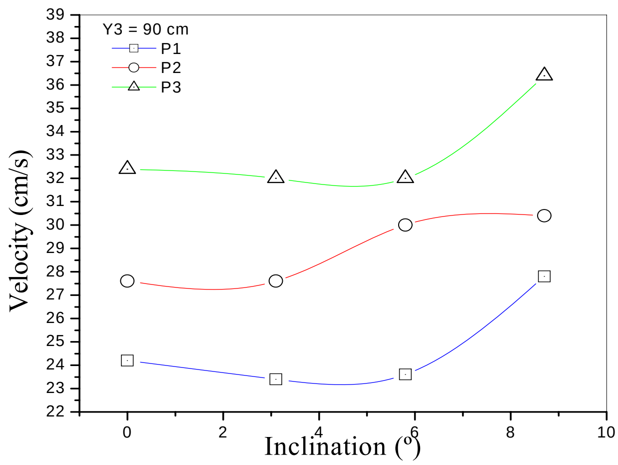
When the angle changed from A1 (3.1°) to A2 (5.8°) and to A3 (8.7°), there was some ambiguity about the results: in same cases there were increasing and in others there were decreasing of the air flow. This ambiguity can be explained as a consequence of the combination of the “strangle” effect (due to the increasing of the thermal boundary layer) and the reduction of the heat transfer due to down-ward position of the plate. If some heated surface is positioned facing the down-ward direction, there is a decreasing of the buoyancy effect, as it was pointed by Incropera and DeWitt (1990).



(a) Y1



(b) Y2



(c) Y3

Fig. 6. Variation of the air flow as a function of the angle of the bended wall.

3.3. Nu – Ra relation

In table 4 are presented the relations between the Rayleigh and Nusselt numbers and the resultant coefficients of the linearization ("c" and "n"), to all twelve configurations, as it appears in the generalized equation to natural convection process, showed in equation (2), (Jakob and Hawinks, 1957):

$$Nu = c.Ra^n \tag{2}$$

Tab. 4 – Coefficients of the Nu – Ra relation

Configuration	Log Ra	Log Nu	c	n
A0Y1	7.57 7.61 7.67	1.61 1.62 1.63	0.57	0.245
A1Y1	7.56 7.61 7.66	1.61 1.62 1.63	0.57	0.245
A2Y1	7.58 7.62 7.68	1.61 1.62 1.63	0.57	0.245
A3Y1	7.55 7.60 7.65	1.60 1.62 1.63	0.55	0.247
A0Y2	7.57 7.61 7.65	1.61 1.62 1.63	0.57	0.245
A1Y2	7.57 7.61 7.66	1.61 1.62 1.63	0.56	0.246
A2Y2	7.57 7.61 7.66	1.61 1.62 1.63	0.57	0.245
A3Y2	7.55 7.60 7.67	1.61 1.62 1.63	0.56	0.246
A0Y3	7.56 7.61 7.66	1.61 1.62 1.63	0.55	0.247
A1Y3	7.56 7.61 7.67	1.61 1.62 1.63	0.57	0.245
A2Y3	7.56 7.60 7.66	1.61 1.62 1.63	0.56	0.246
A3Y3	7.55 7.60 7.65	1.60 1.61 1.63	0.56	0.246

In figure 7 are shown some examples of the linearization made to obtain the coefficients.

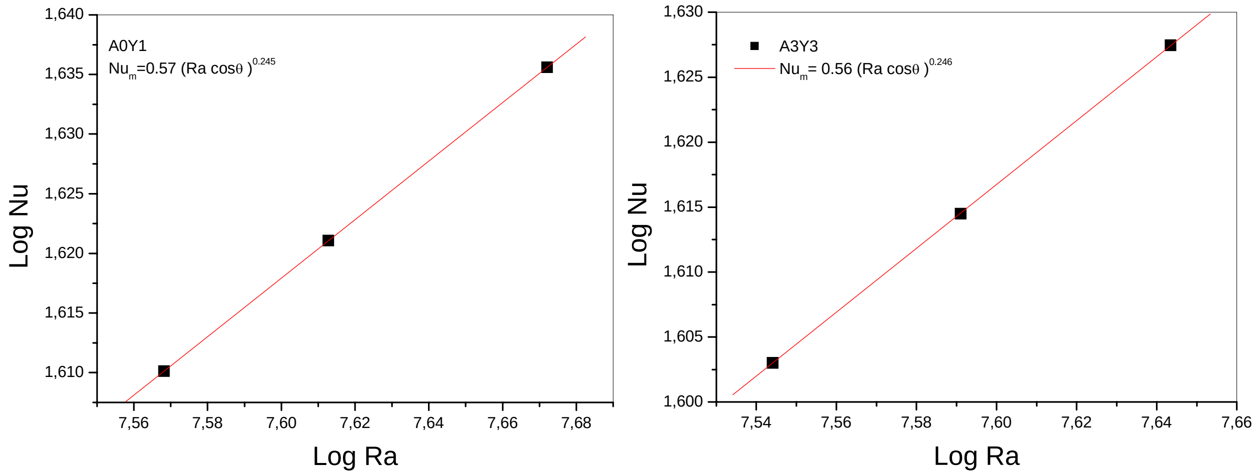


Fig. 7. Nusselt – Rayleigh relation.

It was observed that, to all configurations considered in this study, the linearization of relation between Nusselt and Rayleigh is possible to the mean values expression:

$$Nu_m = 0.56(Racos\theta)^{0.25} \tag{2}$$

The relation expressed by equation (2) is in accordance with those find in the literature, as the result related by Keyhani *et al*. (1988), Webb and Hill (1989), La Pica *et al*. (1993), Marinho and Celere (1997) and Onur and Aktas (1998).

3.4. Variation of AR as a function of Ri

In figure 8 are presented the variation of the aspect ratio parameter as a function of Richardson number, expressed by:

$$Ri = Gr/Re^2 \tag{3}$$

Where: Gr is the number of Grashoff and Re is the number of Reynolds

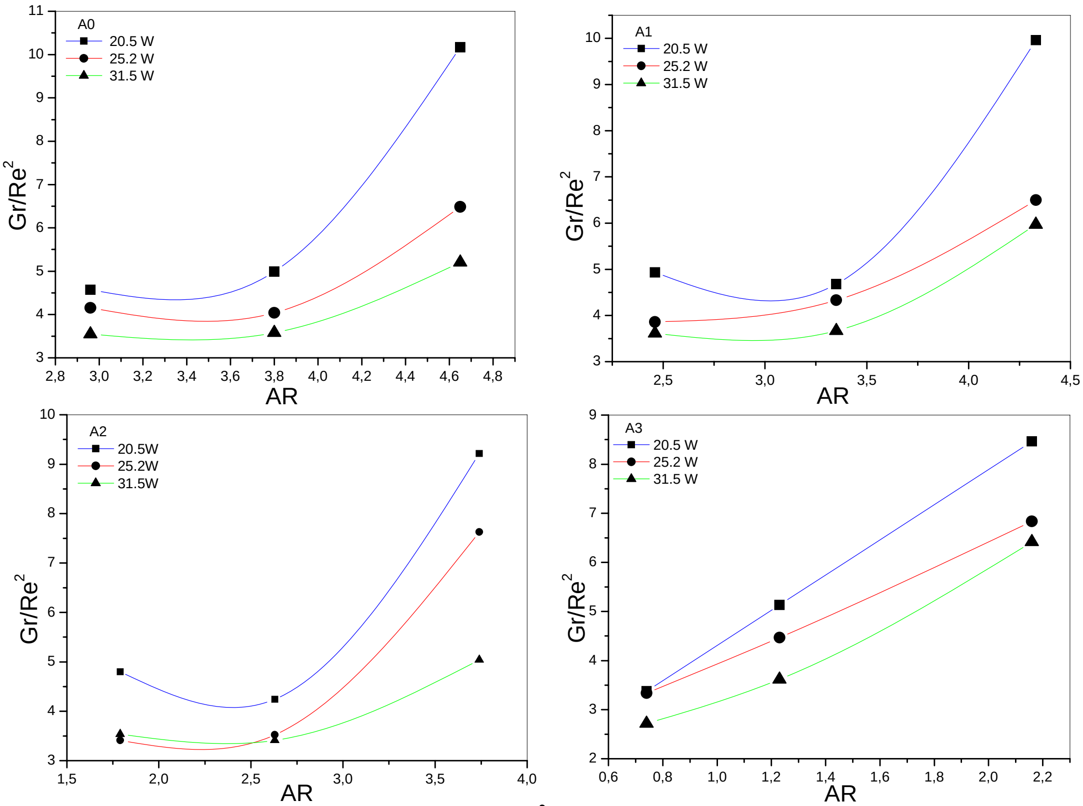


Fig.8. Relation between Gr/Re^2 parameter and the aspect ratio.

It was observed an accentuated increasing of the Ri number as the AR increases. As the increasing of Ri is associated to the decreasing of Reynolds number (Re), those results shows that the variation of the air flow is conditioned by the shift of the heat exchanger inside the channel from the upper position to the lower position (where the air flow is smaller).

4. CONCLUSIONS

Base on the results, we conclude that:

I – the asymmetry of the channel contributes to the increasing of air flow; compared with the situation where the walls are symmetric (parallel), the convergence situation represents an increasing of the velocity as higher as 12.3%;

II – there is a optimum combination of the angle of the wall and the position of the heat exchanger; in those situations, the reduction of the air flow (due to the down-ward position of the heated face of the plate) can be overcome by the increasing of the flow as a consequence of the convergence effect (when the heat exchanger are in the lower position);

III – the Gr/Re^2 parameter (i.e., the Richardson number) has a significant influence in the relation between the thermal power and the aspect ratio that is responsible to the increasing of air flow;

IV – the Nu-Ra relations obtained to all twelve situations are in agreement with the results presented in the literature;

V – the results obtained can be applied to improvement of natural convection process as those where the natural ventilation can be considered as an option to minimize the effects of thermal charge due to solar radiation in habitations located in low latitude regions, as the northeast region of Brazil.

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