Performance Enhancement of Air-cooled Condensers

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Abstract: Heat transfer by convection in air cooled condensers is studied and improved in this work. In order to enhance the performance of air cooled condensers, it is important to take into consideration both of condensation inside condenser tubes and convection outside, where the enhancement in convection side is the dominant one. Aluminum extruded micro-channel flat tubes improve the performance of condensation more than conventional circular tubes but still has potential for air side improving. So the enhancement of convective heat transfer in air side is achieved in this study by inclination of the flat tubes by a certain angle with respect to horizontal in two cases. The first proposed case is to make convergent and divergent channels for air flow (case 1), while the second case is tilting all tubes in parallel to each other (case 2). A parametric study is performed to investigate the optimum inclination angle (β) and aspect ratio (Ar). Mathematical modeling for air cooled condensers was applied to aluminum flat tubes to study and evaluate these proposed two cases. A computational fluid dynamic software (CFD) is used to solve the problem. Theoretical results show that the optimum angle for the proposed two cases is about 4 deg. With corresponding aspect ratio of 0.58. This leads to enhancement of heat transfer coefficient by factor (Kh) of 1.469 and 1.46 against increase in pressure drop factor (KP) of 2.12 and 1.95 for case 1 and case 2 respectively.

1 Introduction

Air-cooled finned-tube condensers are widely used in refrigeration and air-conditioning applications. For the same amount of heat transfer, the operation of air cooled condensers is more economic as compared with water cooled condensers [1]. Typically air-cooled condensers are of the round tube and fin type. To improve the performance of air-cooled condensers multiple techniques can be achieved such as enhancements on inner pipe surface, changing the tube geometry from round to flat shape and external fins.

A micro-channel flat tubes heat exchanger is one of the potential alternatives for replacing the conventional finned tube heat exchangers. This kind of heat exchangers is made of a flat tube with several independent passages in the cross-section, and formed into a serpentine or a parallel flow arrangement. In these heat exchangers, a multitude of corrugated fins with louvers are inserted into the gaps between flat tubes. The flat tube design offers higher thermal performance and lower pressure drop than the finned-round tube heat exchangers [2]. Brazed aluminum heat exchanger is made from micro-channel flat tubes in parallel to each other which is called parallel flow heat exchanger (PFHE). As a result of its superior performance, some companies in heating, ventilating and air conditioning are considering the flat tube heat exchanger as a high efficiency alternative in order to save electricity when used in window and split type air conditioners which consume large amounts of electricity and contribute to the severe electricity shortage in the peak period. The key advantage of the brazed aluminum design is smaller size and lower weight than finned-round tube condensers. The heat capacity of a parallel-flow heat exchanger (PFHE) is 150-200% larger than that of the conventional heat exchanger [3]. This high heat capacity of the PFHE can meet the requirements of compactness and lightness. Oval and flat cross-sectional tube for finned tube heat exchangers provides a higher heat transfer performance as compared to those formed with round tube geometry as mentioned by Chang et. al. [1]. The effect of tube profile change from round to flat shape on condensation has been investigated experimentally by Wilson et. al [4].They used horizontal copper tubes were initially round with 9.52 mm outer diameter and 8.91 mm inner base diameter. The tubes were successfully flattened into an oblong shape with inside heights of 5.74, 4.15, 2.57, and 0.974 mm. Refrigerants R-134a and R-410A were investigated over a mass flux range from 75 to 400 $kg/m^2 . S$, and quality range from approximately 10-80%. They summarized the following results:

1 For a given mass flow rate, there is a significant reduction in refrigerant charge due to flattened tubes.

2 The pressure drop increases as the tube profile is flattened at a given mass flux and quality.

3 There is enhancement of condensation heat transfer coefficient as the tube profile is flattened.

4 Heat transfer enhancement is dependent on the mass flux, quality, and tube profile.

The condensation of refrigerant in multi-port micro-channel extruded tubes has been investigated by many authors [5-7]. All of them concluded that the micro-channel flat tube enhance the inside heat transfer many times than conventional round one. So the present work is mainly concentrated on air side heat transfer from flat tube condensers which is the dominant one.

Although, the PFHE has the above mentioned good thermal performance, but there is still a lot of potentials for improving the air side convective heat transfer which is the dominant one. Therefore, the present study is directed to enhance the convection side heat transfer by inclination of its flat tubes, one is inclined towards clockwise and the next in counter clockwise direction by angles up to 16 deg with respect to horizontal to make convergent and divergent channels for air flow (case 1). Furthermore, without the need of replacing any equipment of production line that producing PFHE, another construction for inclination of all tubes by the same angle range (0:16 deg.) but all tubes are kept in parallel with each other (case 2) is also included in the present study. Finally the effect of aspect ratio (Ar) has been investigated at the optimum inclination angle (β). The best choice for correct range of inclination angles from 0:16 deg that leads to enhancement was obtained from the researches [8-9].

Nomenclature

h	air-side heat transfer coefficient, $W/m^2 . K$
Dh	hydraulic diameter, mm
Vf	Air face velocity, m/ s
H	transverse pitch of parallel tubes. mm
L	Width of flat tube cross section, mm
Ar	Aspect ratio =H/L
P β	pressure drop for case of inclined flat tubes by angle β, Pa.
P0	pressure drop for case of parallel flat tubes with β=0 deg.
β	inclination angle of flat tubes with respect to horizontal, deg.
Re	Reynolds number, dimensionless
ΔP	pressure drop, Pa
PFHE	parallel flow heat exchanger (Aluminum Brazed heat exchanger = PFHE or serpentine flow heat exchanger)
η	overall performance =Kh/KP
Kh	Enhancement factor of h = hβ/h0
KP	pressure drop increase factor= Pβ/P0

<u>*Subscripts*</u>

av average

2 Mathematical Model

Many industrial applications, such as air cooling in the coil of an air conditioner, can be modeled as two-dimensional heat flow. All pre-generated meshes for the studied cases were prepared first by GAMBIT software. Then modeled as bank of tubes in cross-flow, and the air outside flow is classified as turbulent and steady.

The model is used to predict the Flow and temperature fields that result from convective heat transfer. Due to symmetry of the tube bank, only a portion of the geometry was modeled in FLUENT. Domain is discretized into a finite set of control volumes or cells. General transport equations for mass, momentum and energy are applied to *each* cell and discretized. The governing equations are solved to the studied flow field. The numerical solution was conducted to investigate the influence of inclination angle (β) and aspect ratio (Ar) on the performance of air cooled condensers.

The following values which applicable to window and split air conditioning systems, are used as input data for solving the studied problem:

1 Air flow is steady, 2 dimensional and turbulent

2 Air face velocity (Vf)=2.5, 5 and 7.5 m/s

3 The condenser saturation temperature of refrigerant=323 K. Hence study is based on constant wall temperature=323 K

4 Ambient air temperature=308 K

5 The flat tube condenser configurations: tube height (b)=1.8 mm, tube width (L)=18 mm, tubes transverse pitch=10.4 mm.

2.1 Numerical Technique

Flow and heat transfer characteristics is obtained for forced convection of air flow across flat tubes at different operating parameters. By using CFD software, the flat tubes condensers shown in Fig. 2a has been studied first, which is called parallel flow heat exchanger (PFHE). Then the proposed modifications in the following sequence: Case 1: construction of convergent and divergent channels for air flow through inclination of flat tubes by angle up to 16 deg. With respect to horizontal, one is inclined towards clockwise and the next in counter clockwise as shown in Fig. 2b.

Case 2: Tilting of all tubes in parallel to each other by angle up to 16 deg with respect to horizontal) either forward or backward as illustrated in Fig. 2c.

3 Results and Discussions

In order to study the performance of the proposed two cases, the obtained results are presented relative to those of parallel flat horizontal tubes at the same operating conditions. Contour lines for temperature and velocity in axial direction are shown in Figs. 3 and 4 for flat horizontal tubes, convergent divergent (case 1), and tilted sections of tubes (case 2). Generally, it is observed from Fig. 3 that there is a decrease in fluid temperature towards the centre between pipes in flow direction as the flow is developing. Also, it is found from Fig. 4 for the case of convergent divergent passage, the velocity increases in convergent passes and decreases in divergent passage.

Figure 5 shows the contour of local surface heat transfer coefficient (h) for the same studied cases. It is clear from this figure that the local values of heat transfer coefficient (h) for the studied two cases are higher than those of horizontal tubes The effect of inclination angle β on the performance of flat tube air cooled condenser is illustrated in Figs. 6a and 6b for the studied two cases compared with flat horizontal tubes. As shown in Fig. 6a the increase in ΔP is small in the first part up to 8 deg. then ΔP increases sharply. Therefore it is preferable to operate in this first part. Also, it is found that there is a peak value at β =4 deg. Also, there is a higher values for both of hav, ΔP in the second part of the curve which is not preferable practically. In the other hand, for case 2 presented in Fig. 6b with increasing β up to 12 deg there is a continuous increase in both of hav, Δp. Then for β more than 12 deg. Leads to decreasing in hav and ΔP.

To compare between the two cases and to choose the optimum β, it is important to evaluate the enhancement process as a whole. Therefore the effectiveness of the process (η) (which is defined as η=(Kh/Kp) is plotted against β for the studied cases in Fig. 7b at different values of air face velocities (Vf). It is clear from Fig. 7a and Fig. 7b that for different velocities (Vf), the optimum β for both cases is 4 deg. Also to study the effect of the tested face velocities (Vf), it is clear from Fig. 7b that varying Vf from 2.5 up to 5 m/s leads to considerable change in the performance as a whole (η). But with increasing Vf from 5 to 7.5 m/s, there is a small change in the performance as the two curves are nearly coincident. So, the value of 5 m/s is considered as the max. limit for operation (practical value).

Then for the investigated optimum β, the effect of aspect ratio (Ar) on the performance is shown in Fig. 8. Easily the optimum (Ar) is chosen at value of 0.58 which corresponds to H=10.4 mm.

Finally, to verify the obtained results, a comparison with similar researches that investigated experimentally are shown in Figs. 9a and 9b.

For case 1: The convective heat transfer to air flow in converging-diverging tubes were studied experimentally by Ariad et. al. [8]. Their study was based on constant wall temperature at different values of β from 0 up to 16 deg., which is similar to the proposed studied cases. They reported that the obtained enhancement comparing to equivalent straight tube at the same mean diameter is Kh=1.45 against KP of 2.2 value. The corresponding values (at 4 deg., Vf=5 m/s) for the present proposed cases are, Kh=1.469, 1.46 against KP=2.12, 1.9 for, convergent –divergent and tilting case respectively. Also the experimental optimum β was 5,30' is agreed with the present one obtained theoretically (4 deg). As shown in Fig. 9a both of present theoretical results and experimental results [4] for Nu are plotted against β. which demonstrates a good agreement.

For case 2: The effect of inclination angle on the performance of aluminum brazed heat exchanger was investigated experimentally by Kim M. H. [10]. From Fig. 10b the comparison of results is showing acceptable agreement. Also they reported that there is enhancement in hav. With increasing β up to 12 deg which agreed with the present results.

Conclusion

1 Using the proposed convergent divergent construction of heat exchanger with optimum angle of 4 deg offers the best enhancement in heat transfer coefficient. For one row coil which is used in car air condition, the enhancement factor is about Kh= 1.467 with increase in pressure drop (KP factor=2.12).

2 To keep the production line that manufacturing the PFHE, the proposed construction of tilting the all tubes in parallel by 4 deg with respect to horizontal is recommended. This leads to enhancement factor of Kh=1.46 with increase in pressure drop of KP=1.9.

3 This proposed heat exchanger is the strong candidate for use in industrial applications, which is named 'convergent-divergent flow heat exchanger'.

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Figure 1
Micro-channel fat tube cross-sectional view (Lxb)

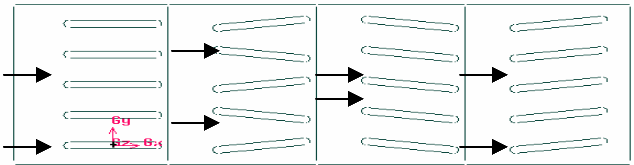
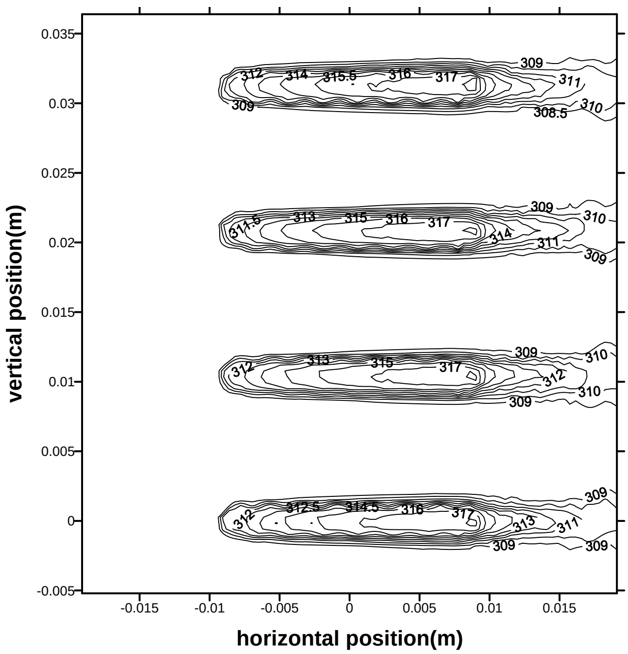


Fig.(2 -a) PFHE
Fig.(2 -b) **Case 1** Convergent-divergent. Construction of tubes channels
Fig.(2 -c) **Case 2** Tilted tubes
tilted backward
tilted forward

Figure 2
Layout of flat horizontal tubes (PFHE) and the proposed two cases of modifications



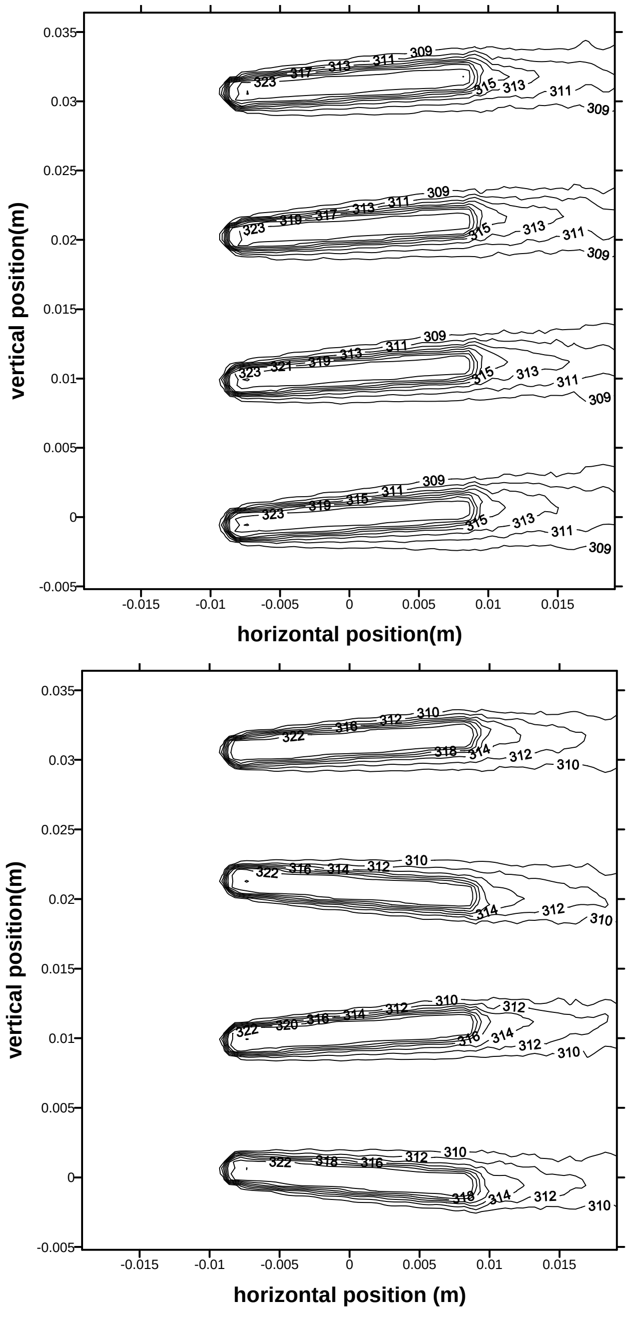
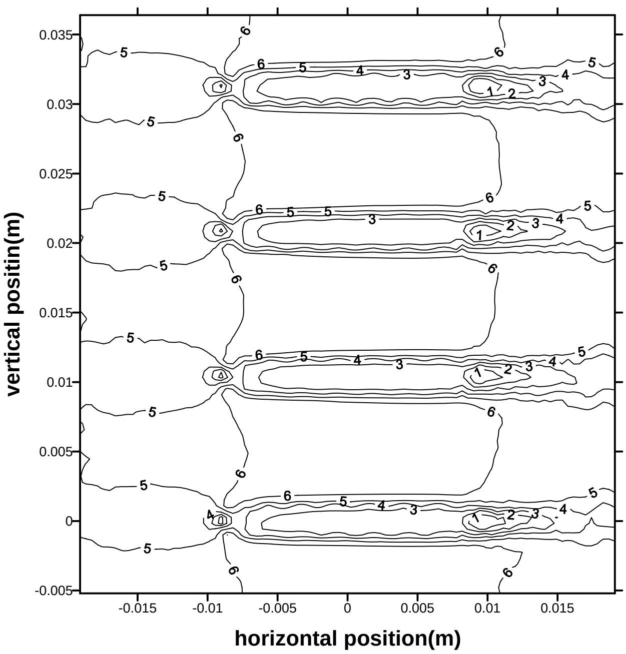
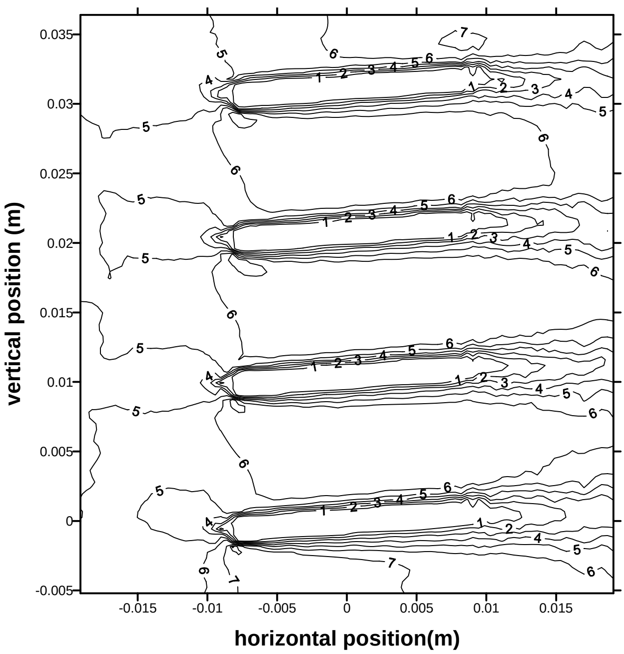


Figure 3

Temperature contour for the studied two cases compared with horizontal flat tubes case


vertical positin(m)
horizontal position(m)


vertical position (m)
horizontal position(m)

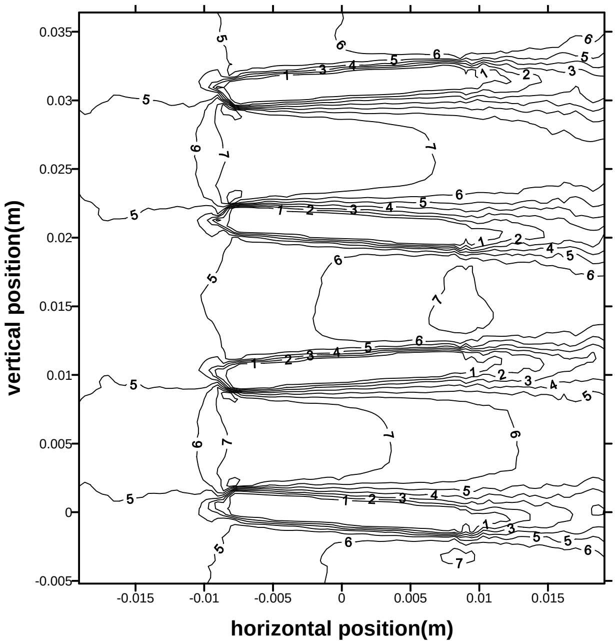
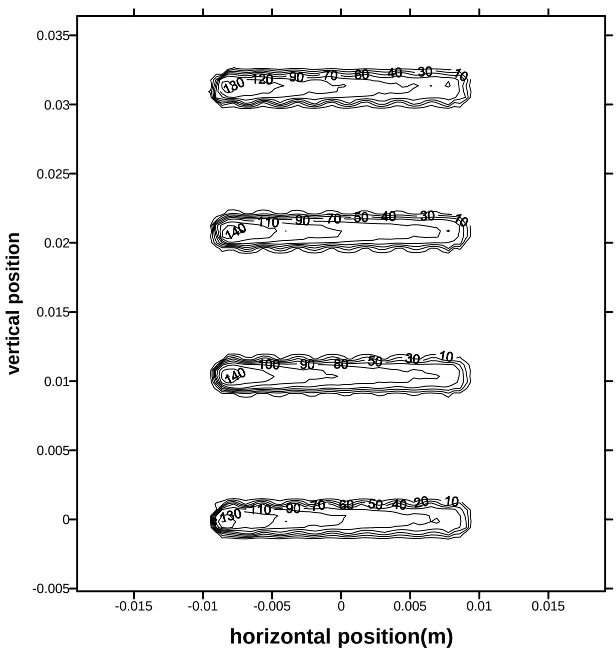


Figure 4

Velocity contour for the studied two cases compared with horizontal flat tubes case



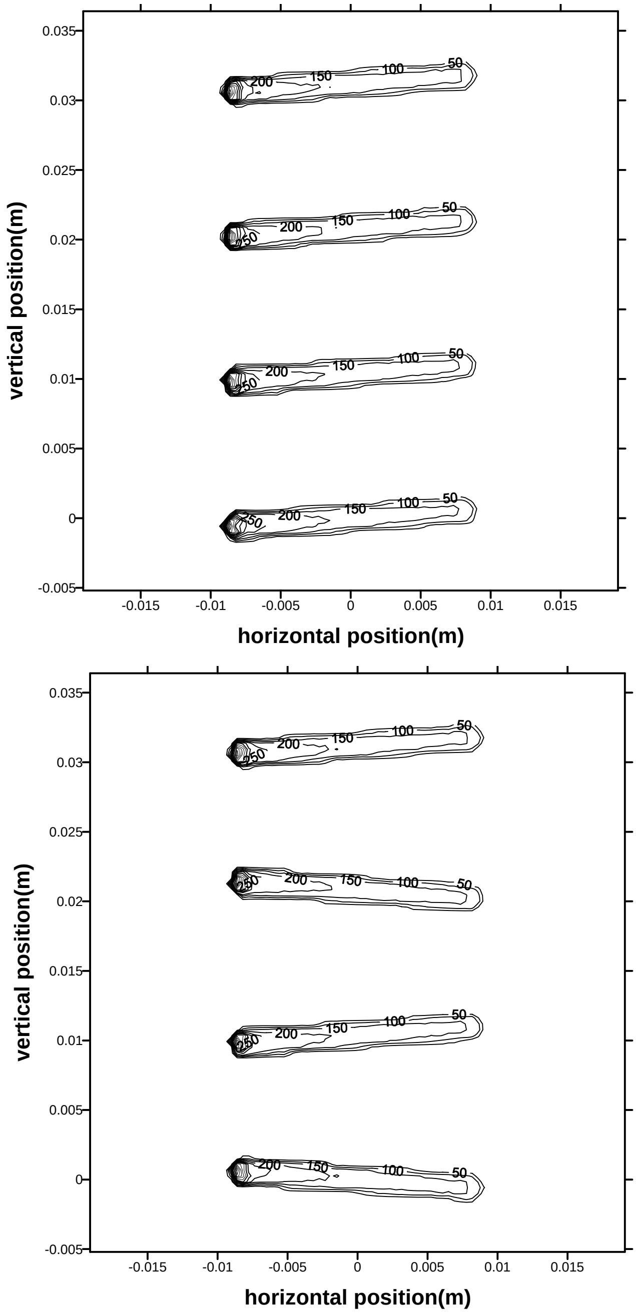


Figure 5

Local heat transfer coefficient contour for the studied two cases compared with horizontal flat tubes case

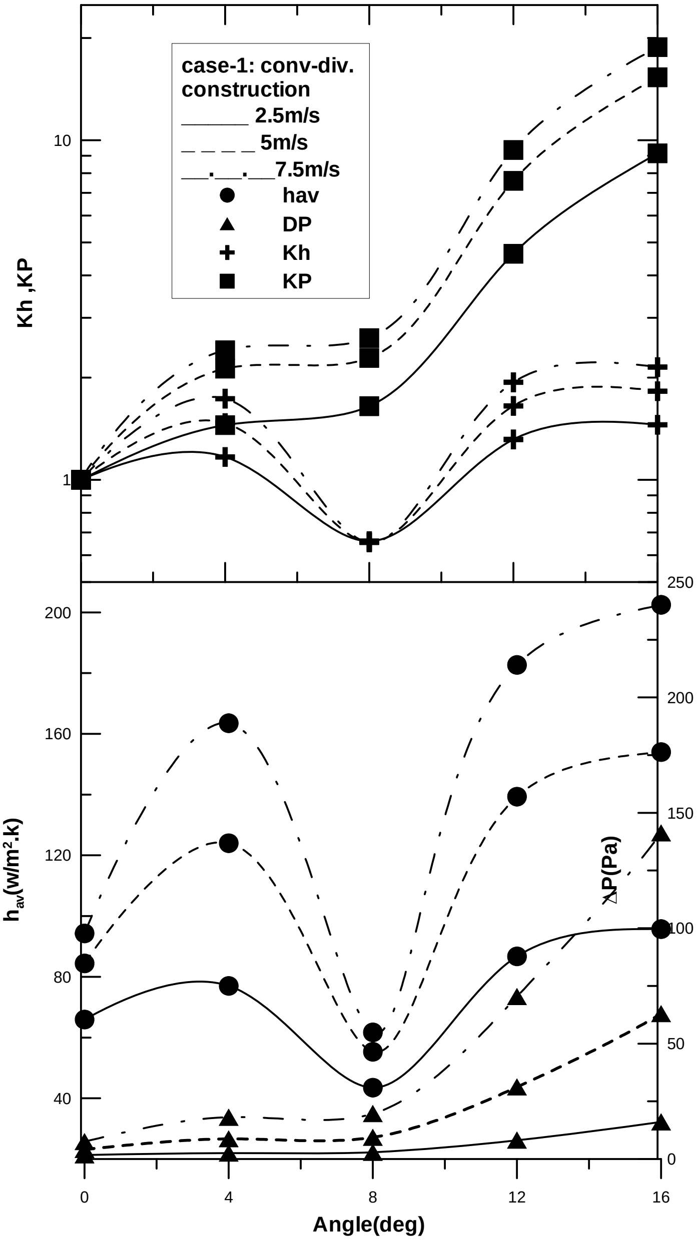


Figure 6a

Variation of performance against β for case 1 (convergent – divergent)

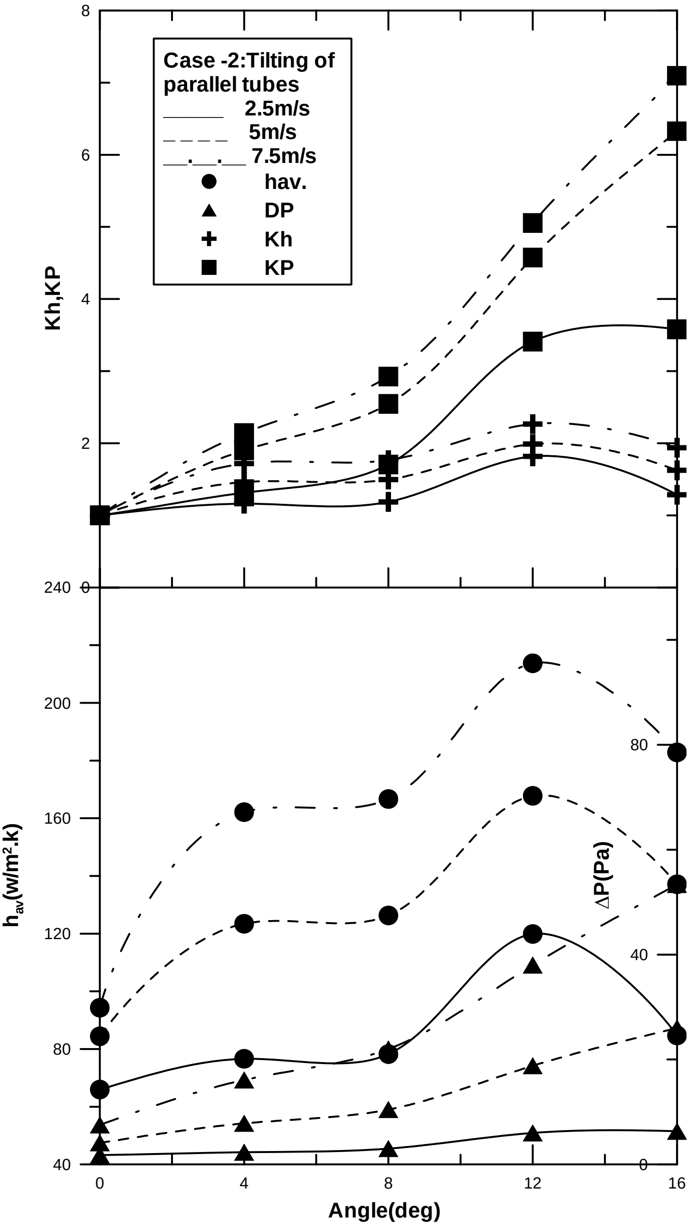
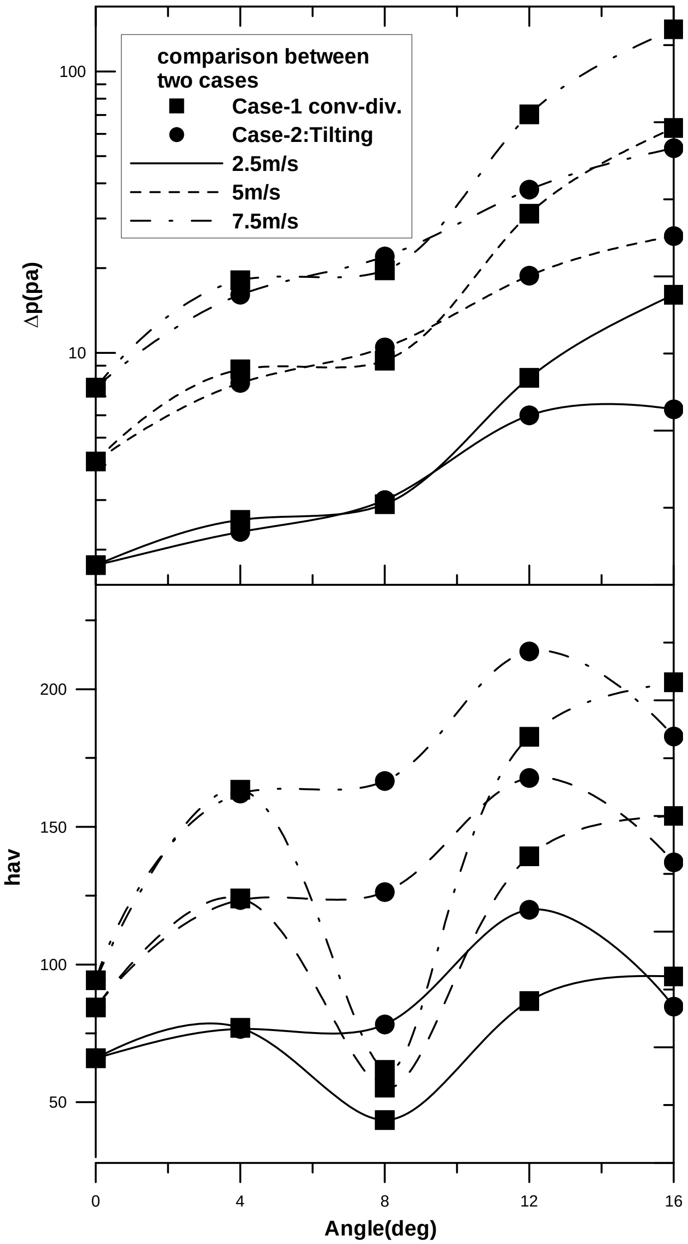


Figure 6b
Variation of performance against β for case 2 (tilting of tubes)



comparison between two cases
Case-1 conv-div.
Case-2:Tilting
2.5m/s
5m/s
7.5m/s
Δp(pa)
100
10
hav
200
150
100
50
0
4
8
12
16
Angle(deg)

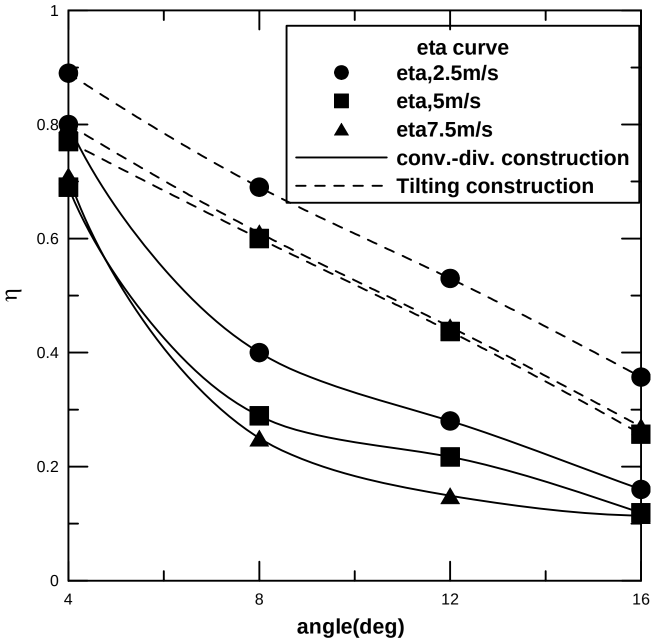


Figure 7
Comparison between performance of the proposed two cases

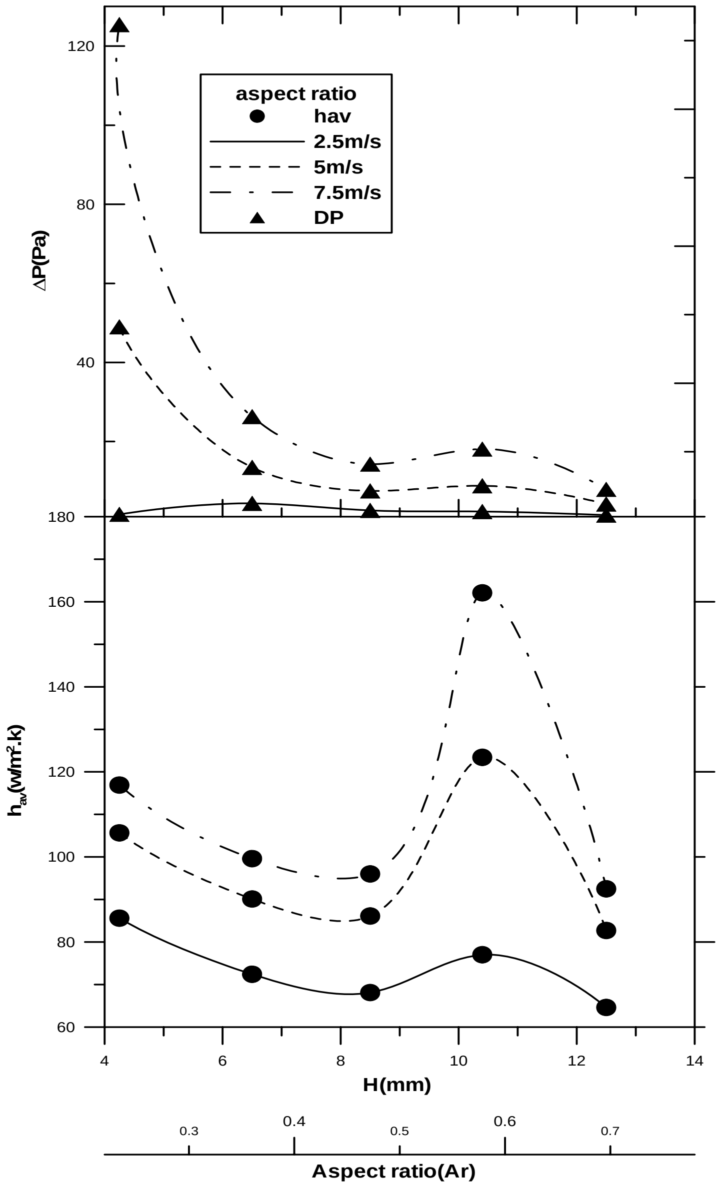


Figure 8

Performance variation against aspect ratio

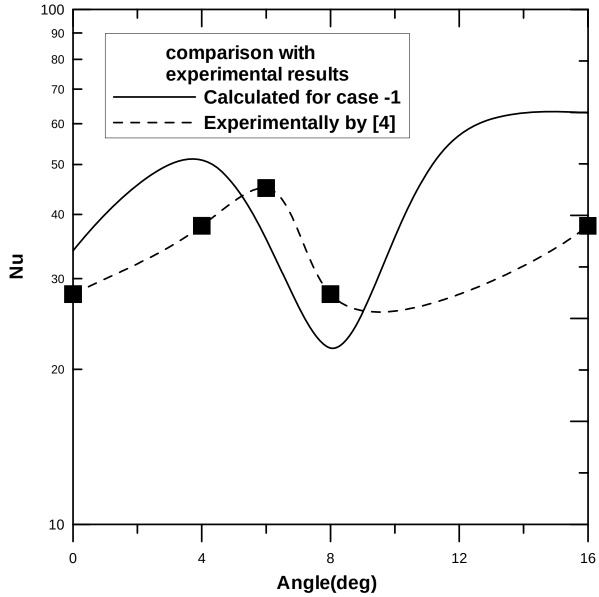


Figure 9a
Comparison with experimental results for case 1

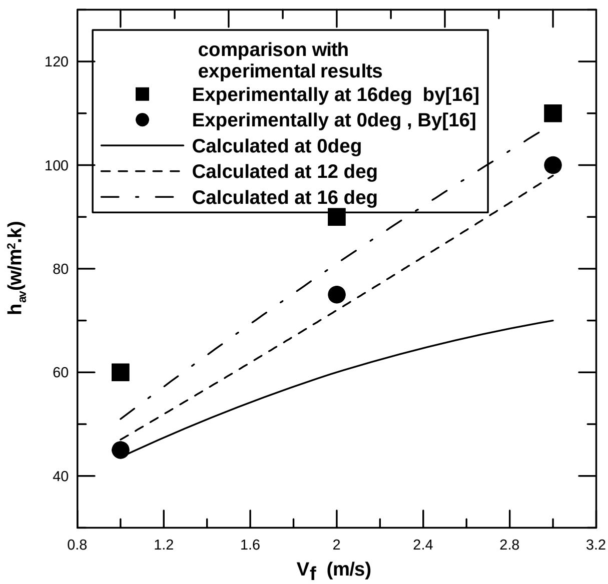


Figure 9b
Comparison with experimental results for case 2