A study of entropy generation minimization in an inclined channel

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Abstract: An analysis of a mixed convection and entropy generation for a fluid flow through porous medium, between two inclined parallel plates has been performed. The entropy generation is estimated via an analytical solution of the temperature and velocity profiles obtained from the mass, momentum and energy equations governing the flow and heat transfer of the problem, with the given conditions. The optimum values of the involved parameters at which the entropy generation assumes its minimum is calculated. The Bejan number is also obtained and discussed.

Key–Words: Entropy Generation, Mixed Convection Parameter, Porous Medium, Bejan number.

1 Introduction

The study of flow through a saturated porous media in channels and over parallel plates have been the focus of many research papers over the last decades. This configuration is encountered in many of energy-related applications, such as solar power collectors, geothermal energy systems, conventional flat plate collectors or the cooling of modern electronic systems. The vertical parallel plates configuration is applicable in the design of electronic equipment cooling systems. A huge amount of papers related to flow and heat transfer through porous materials were written by researchers such that Baytas and Pop [2], Pilevne and Misirlioglu [17] and Theeran *et al.* [19]. Also, studies in capillary porous media (see Farjad *et al.* [10]), in channels partially filled by a porous material (Keyhani *et al.* [14]) or of thermal instability in the presence of magnetic field (Hanadi *et al.* [11]), were recently reported.

The process of studying the entropy generation in porous media is comparatively harder than the clear fluid case partly due to the increased number of variables present in the governing equations. Also, different models for viscous dissipation, that lead to different fluid friction irreversibility terms are available. Moreover, the complexity of the problem becomes clearer when one observes that, numerical or theoretical, solutions addressing the second law analysis of natural, forced or mixed convection in porous ducts, are mostly restricted to circular tubes or parallel plate channels where the geometry allows analytical solution of closed form.

Minimization of entropy generation is a method for modeling and optimizing of energy systems (see Bejan [4]). In earlier studies related to the natural convection, only the first-law of thermodynamics was used. However, the method of entropy generation combines from the start the most important parameters of thermodynamics, heat transfer and fluid mechanics. To improve the heat transfer performance is a chief task in heat exchanger designs (see Ingham and Pop [13]). Owing to the fact that the heat transfer enhancement is always achieved at the expense of the increase of friction loss, the optimal trade-off by selecting the most appropriate configuration and the best flow conditions has become the critical challenge for the design work. The analysis of the energy utilization and the entropy generation has become one of the primary objectives in designing a thermal system. Bejan [3], has described the systematic methodology of computing entropy generation through heat and fluid flow in heat exchangers. Fundamentals of entropy generation are also presented by Narusawa [15] and Rosen [18]. Recent papers related to this subject were written by Hooman and Ejlali [12] and Nejma *et al.* [16].

The present paper studies the entropy generation production for a problem of a mixed convection flow of a fluid saturated porous medium through an inclined channel with uniform heated walls, see Cimpean *et al.* [8]. Another mixed convection problem but in different conditions, for a non-Newtonian fluid flow over a permeable wedge embedded in a porous medium, was treated recently by Chamkha [5].

2 Problem formulation

Consider the mixed convection flow in an inclined infinitely long two-dimensional channel bounded by parallel plane walls and filed with a fluid-saturated porous medium, see Cimpean *et al.* [8]. The x axis is considered up lengthways and the y axis is oriented into the channel, see Figure 1. The fluid and porous media properties are constant except for the variation of density in the buoyancy term of the Darcy equation. The fluid has an uniform upward streamwise velocity distribution at the channel entrance and the walls are at uniform heat flux q. Under these assumptions, and with the use of the Darcy's law and the Boussinesq approximation, the governing equations are written as follows

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

$$\frac{\partial u}{\partial y} - \frac{\partial v}{\partial x} = \frac{g\beta K}{\nu}\left(\frac{\partial T}{\partial y}\sin\gamma - \frac{\partial T}{\partial x}\cos\gamma\right) \tag{2}$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha_m\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{3}$$

where u and v are the cartesian velocity components, T is the fluid temperature. The coefficients are β the fluid thermal expansion, K the specific permeability of the medium, ν the kinematic viscosity and α_m the effective fluid thermal diffusivity. Also, the tilt angle, measured counterclockwise from the horizontal is denoted by γ in the considered equations.
The equations (1)-(3) have to be solved subject to the boundary conditions:

$$\begin{aligned} v = 0,\ \tfrac{\partial T}{\partial y} &= -\tfrac{q}{k} \text{ on } y = 0 \\ v = 0,\ \tfrac{\partial T}{\partial y} &= \tfrac{q}{k} \text{ on } y = D \end{aligned} \tag{4}$$

where q is the heat flux to the wall, D is the channel width and k is the thermal conductivity of the fluid.

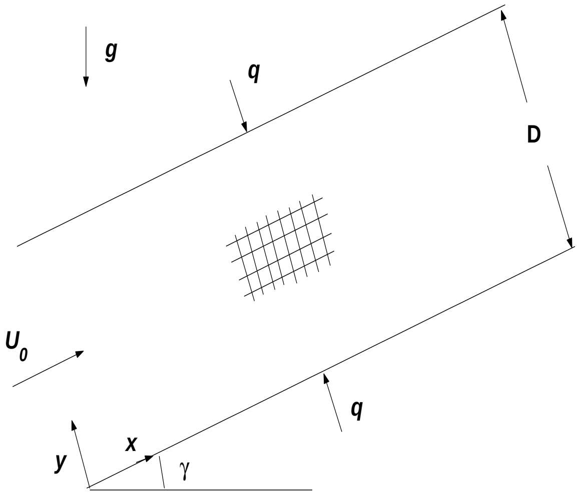

Fig.1 Channel configuration.

We introduce the following non-dimensional variables:

$$X = \frac{x}{D},\quad Y = \frac{y}{D},\quad U = \frac{u}{U_0},\quad \theta = \frac{T - T_0}{qD/k} \tag{5}$$

Further, we consider

$$U = U(Y),\ \theta(X, Y) = C_1 X + F(Y) \tag{6}$$

and following the paper by Cimpean *et al.* [8], from the given conditions we have $C_1 = 2/Pe$ and a third order ordinary differential equation is obtained:

$$\frac{d^3F}{dY^3} - (2\lambda\sin\gamma)\frac{dF}{dY} + \frac{4\lambda\cos\gamma}{Pe} = 0 \tag{7}$$

which has to be solved, subject to the boundary conditions:

$$\frac{dF}{dY} = -1 \text{ at } Y = 0;\ \frac{dF}{dY} = 1 \text{ at } Y = 1 \tag{8}$$

In the Eq. (7) the parameters are

$$\lambda = \frac{g\beta KqD}{U_0\nu k}, Pe = \frac{U_0 D}{\alpha_m} \tag{9}$$

the mixed convection parameter and the $P\acute{e}clet$ number, respectively.

Another important parameter to consider in our results, is the Rayleigh number, defined as $Ra = \lambda Pe$.

It is assumed that the $P\acute{e}clet$ number is $Pe > 0$ throughout. Only channels inclined in an upward direction are considered, then, we can limit γ to the range $0 \leq \gamma \leq \pi/2$. Hence, in Eq. (7) we have the terms $\sin\gamma \geq 0$ and $\cos\gamma \geq 0$, for $\gamma = 0$ for horizontal channels.

2.1 General case $\gamma > 0$

Further, we will take into account the general case of the inclination of the channel ($\gamma > 0$) and the analytical solutions for this case is given by:

$$\frac{dF}{dY} = \frac{2\cot\gamma}{Pe} + \left(1 - \frac{2\cot\gamma}{Pe}\right)\frac{\sinh\alpha Y}{\sinh\alpha} - \left(1 + \frac{2\cot\gamma}{Pe}\right)\frac{\sinh\alpha(1-Y)}{\sinh\alpha} \quad (10)$$

where $\alpha = \sqrt{2\lambda\sin\gamma} > 0$.

The velocity profile is

$$U(Y) = \frac{\alpha}{2\sinh\alpha}\left[\left(1 + \frac{2\cot\gamma}{Pe}\right)\cosh\alpha(1-Y)\right] + \frac{\alpha}{2\sinh\alpha}\left[\left(1 - \frac{2\cot\gamma}{Pe}\right)\cosh\alpha Y\right] \quad (11)$$

On integrating Expression (10) and applying the condition

$$\int_0^1 F(Y)U(Y)dY = 0 \quad (12)$$

we find that

$$F(Y) = \frac{1}{\alpha\sinh\alpha}\left[\left(1 + \frac{2\cot\gamma}{Pe}\right)\cosh\alpha(1-Y)\right] + \frac{1}{\alpha\sinh\alpha}\left[\left(1 - \frac{2\cot\gamma}{Pe}\right)\cosh\alpha Y\right] + \frac{2\cot\gamma}{Pe}Y + A_0 \quad (13)$$

where the constant A_0 is given by

$$A_0 = -\frac{(\cosh\alpha + 1)(\sinh\alpha + \alpha)}{2\alpha\sinh^2\alpha} - \frac{2\cot^2\gamma(3\sinh\alpha - \alpha)(\cosh\alpha - 1)}{Pe^2\alpha\sinh^2\alpha} - \frac{\cot\gamma}{Pe} - \frac{2\cot^2\gamma}{Pe^2} \quad (14)$$

2.2 Vertical channel $\gamma = \pi/2$

For the vertical channel, the solutions from Expressions (10) and (11) become:

$$\frac{dF}{dY} = \frac{\sinh\alpha Y}{\sinh\alpha} - \frac{\sinh\alpha(1-Y)}{\sinh\alpha} \quad (15)$$

$$U(Y) = \frac{\alpha}{2\sinh\alpha}\left[\cosh\alpha Y + \cosh\alpha(1-Y)\right] \quad (16)$$

The analytical results of both, the general case and the vertical channel case, will be useful, from now on, to observe the entropy generation into the channel.

3 Entropy generation minimization

3.1 The volumetric entropy generation

The entropy generation is caused by the non-equilibrium state of the fluid, resulting from the thermal gradient between the two media. For the problem involved, the exchange of energy and momentum within the fluid-saturated porous medium and at the solid boundaries, give the non-equilibrium conditions which cause the entropy generation in the flow field of the channel. This entropy generation is due to the irreversible nature of heat transfer and viscosity effects, within the fluid and at the solid boundaries. From the known temperature and velocity fields, volumetric entropy generation can be calculated by the equation (see Baytas [1] and Bejan [3]):

$$S_g = \frac{k}{T_0^2}(\nabla T)^2 + \frac{\mu}{KT_0}\left(u^2 + v^2\right) \quad (17)$$

The first term on the right-hand side is the local entropy generation due to heat transfer across a finite temperature difference and is denoted by $(S_g)_{heat}$ and the second term is the local entropy generation due to fluid friction, denoted by $(S_g)_{fric}$.

The volumetric entropy generation is written as follows:

$$S_g = (S_g)_{heat} + (S_g)_{fric} \quad (18)$$

3.2 The characteristic entropy generation rate

It is important to describe the dimensionless number for the local entropy generation rate. This number is defined by dividing the local volumetric entropy generation rate to a characteristic entropy generation rate. Then, first, we have to define the characteristic entropy generation rate (see Yazdi *et al.* [22]):

$$S_{g0} = \frac{k\triangle T^2}{D^2T_0^2} \quad (19)$$

Here, k is the thermal conductivity of the fluid and $\triangle T$ is given from the boundary conditions (8), such that $\triangle T^2 = (q/k)^2$.

3.3 The entropy generation number

To obtain the dimensionless entropy generation number, we use the formula:

$$N = \frac{S_g}{S_{g_0}} \tag{20}$$

then we obtain

$$N = \frac{4}{Pe^2} + \left(\frac{dF}{dY}\right)^2 + \Phi U^2 \tag{21}$$

Here Pe is the $Péclet$ number defined in (9) and Φ is called the irreversibility distribution ratio (see Baytas [1]), defined as

$$\Phi = \frac{\mu T_0}{k}\left[\frac{\alpha_m^2}{K(\Delta T)^2}\right] \tag{22}$$

Also, we have to notice that, by using the expressions (5) in the Eq. (17) the formula for entropy generation number, N, is obtained (see Cimpean and Pop [9]).

The total local entropy generation number can be written as a summation of the local entropy generation due to heat transfer, denoted by N_{heat} and the local entropy generation due to fluid friction, given as N_{fric}, as follows

$$N = N_{heat} + N_{fric} \tag{23}$$

The last expression gives us the possibility to calculate these terms separately and then compare them to notice which entropy generation mechanism dominates. In the convection problems, both, fluid friction and heat transfer, contribute to the rate of entropy generation. From (21) and (23) N_{heat} and N_{fric} are obtained separately

$$N_{heat} = \frac{4}{Pe^2} + \left(\frac{dF}{dY}\right)^2, N_{fric} = \Phi U^2 \tag{24}$$

3.4 The Bejan number

The entropy in a system is associated with the presence of irreversibility. We have to notice that the contribution of the heat transfer entropy generation, N_{heat}, to the overall entropy generation rate, is needed in many engineering applications.

The Bejan number, Be, is an alternative irreversibility distribution parameter and it represents the ratio between the heat transfer irreversibility, N_{heat} and the total irreversibility due to heat transfer and fluid friction, N. It is defined by

$$Be = \frac{N_{heat}}{N} = \frac{N_{heat}}{N_{heat} + N_{fric}} \tag{25}$$

The Bejan number takes the values between 0 and 1, see Cimpean *et al.* [7]. The value of $Be = 1$ is the limit at which the heat transfer irreversibility dominates, $Be = 0$ is the opposite limit at which the irreversibility is dominated by fluid friction effects and $Be = 1/2$ is the case in which the heat transfer and fluid friction entropy production rates are equal (see Varol *et al.* [20], [21]). Further, the behavior of the Bejan number is studied for the optimum values of the parameters at which the entropy generation takes its minimum.

4 Results and discussion

First, the case of a vertical channel, $\gamma = \pi/2$ is considered to study the entropy generation. It is observed from the expressions (15) and (16) that the solutions is independent of $Péclet$ number, but the entropy generation number N, given by Eq. (21) still depends on it. There is no reversed flow since $U(Y) > 0$ for $0 \leq Y \leq$.

Figure 2 illustrates the entropy generation number N versus Y for equal values of the $Péclet$ and Rayleigh parameters, $Pe = Ra$. For Figure 2a, the vertical channel is considered and, as for the solutions presented in Cimpean *et al.* [8], the entropy profile is symmetric about the center line of the channel $Y = 1/2$. For high $Péclet$ numbers, the entropy generation number, N, take minimum values. As Pe number increases from 10, no important change is seen. A similar behavior is obtained for a different inclination angle of the channel (see Figure 2b for $\gamma = \pi/4$).

Figure 3 represents the total local entropy generation number N for a high mixed convection parameter $\lambda = 100$, and different $Péclet$ numbers, $Pe = 1, 10, 100$. This results that, the Rayleigh number takes the values $Ra = 10^2, 10^3, 10^4$. The entropy number has a maximum in the middle of the channel for $Pe = 1$. Also, for larger values of λ, could be seen that the boundary layers develop near the walls, in which there are increased flow rates. In the central part of the channel is a very low flow rate, so the entropy generation number takes there smaller values for the mixed convection parameter λ.

Figures 4a and 4b compare the entropy generation number obtained by (21) with the entropy generation due to heat transfer, N_{heat}, given by (24), for small and great Pe numbers. The shape of N_{heat} follows the N shape for all λ values. For Figure 4a, a minimum is observed close to the lower wall ($Y = 0$) for the inclination angle $\gamma = \pi/4$. The heat transfer irreversibility reach the zero value for $Y \in (0, 0.1)$ and high λ numbers and a maximum value to the middle of the channel. A different behavior is observed at high Pe numbers (see Figure 4b), where the entropy for both N and N_{heat} take the minimum for $Y = 1/2$. Then, the best choice for the parameters contains a great Pe number and small λ value for obtaining the optimum entropy generation number.

The fluid friction irreversibility and entropy generation number are plotted versus Y in Figures 5a and 5b. The influence from the fluid friction seems to be important near the walls of the channel, greater for small Pe number to the lower wall.

In order to observe the entropy generation behavior to the walls, the solutions from the expressions (10) and (11) for $Y = 0$ (lower wall) and $Y = 1$ (upper wall) are considered. Also, for a noticeable effect on the flow and heat transfer, a moderate values of $Péclet$ number is required, as observed from the first figures. Provided that, $\cot\gamma$ is not too small i.e. the channel is not inclined at a smaller angle to the horizontal, we observe from Eqs. (10) - (13) that the result approaches the symmetric solution (for $\gamma = \pi/2$) in the limit as $Pe \to \infty$ (this is equivalent to setting $\cot\gamma = 0$ in these equations.

Thus, from Eq. 11, we obtain:

$$U(0) = \frac{\alpha}{2\sinh\alpha}\left[\cosh\alpha + 1 + \frac{2\cot\gamma}{Pe}(\cosh\alpha - 1)\right] \tag{26}$$

$$U(1) = \frac{\alpha}{2\sinh\alpha}\left[\cosh\alpha + 1 - \frac{2\cot\gamma}{Pe}(\cosh\alpha - 1)\right] \tag{27}$$

By following the results by Cimpean *et al.* [8] and from the Eqs. (26) and (27), $U(0) > 0$ for all parameter values. If

$$Pe < \frac{2(\cosh\alpha - 1)}{\cosh\alpha + 1}\cot\gamma \tag{28}$$

then $U(1) < 0$ and we can expect a reversed flow region within the channel.

Figures 6-9 present the entropy generation behavior to the walls as functions of the important parameters involved in the problem. Figures 6a and 6b show the entropy generation numbers to the lower wall ($Y = 0$) and upper wall ($Y = 1$), respectively, versus the inclination angle of the channel $\gamma \in (0, \pi)$. Symmetrical profiles to the vertical ($\gamma = \pi/2$) is seen for $N(0)$ and $N(1)$. The entropy numbers to the walls increase as the mixed convection parameter λ increases, having a maximum for $\gamma \in (\pi/4, \pi/2)$ and a minimum for $\gamma \in (\pi/2, 3\pi/4)$ at the lower wall and reverse for the upper wall, respectively. Only the values for $\lambda = 1$ are excepted from this rule.

The influence of the fluid friction irreversibility to the walls is important only for the great λ values and small inclination of the channel, see Figures 7 and 8. The variation of the entropy generation is given at the walls by the fluid friction, the influence from the heat transfer remains constant for unchanged Pe number, according to the condition of the problem.

On the walls, the influence of the heat transfer to entropy generation is greater than of the fluid friction and varies only with Pe number. The entropy generations, to both walls, yield to constant values for $Pe > 10$, see Figure 9.

The Bejan number given by (25), shows if the irreversibility is dominated by fluid friction or heat transfer. From the figure 10, it is obvious that the heat transfer irreversibility dominates between the walls of the channel. The Bejan number takes the value 1 for $\lambda = 50$ into the middle of the channel. The influence of the fluid friction irreversibility could be seen better for the lower wall and $\gamma = \pi/4$ (see Figure 10b), confirming the obtained conclusions.

If Pe varies from 1 to 10, the Bejan number yields to 1 for higher λ numbers. The minimum value is $Be = 0.82$ for $Pe = 10$, see Figure 11. The entropy generation production is dominated by the heat transfer.

5 Conclusion

The present paper investigates the minimization of entropy generation for a mixed convection flow of a fluid saturated porous medium, between two inclined parallel heated plates. The analytical results obtained for the velocity and temperature profiles are used in order to obtain the entropy generation production. The non-dimensional entropy generation number is calculated for the problem involved. The results of heat transfer irreversibility and fluid friction irreversibility are studied for different values of the important parameters of the problem, namely the $Péclet$ number, Pe, the mixed convection parameter, λ, and the inclination angle of the channel from the horizontal, γ. The Bejan number is also calculated to conclude that, the influence of the heat transfer dominates the entropy generation mechanism. The results and discussion show the optimum values of the parameters at which the entropy generation number takes its minimum.

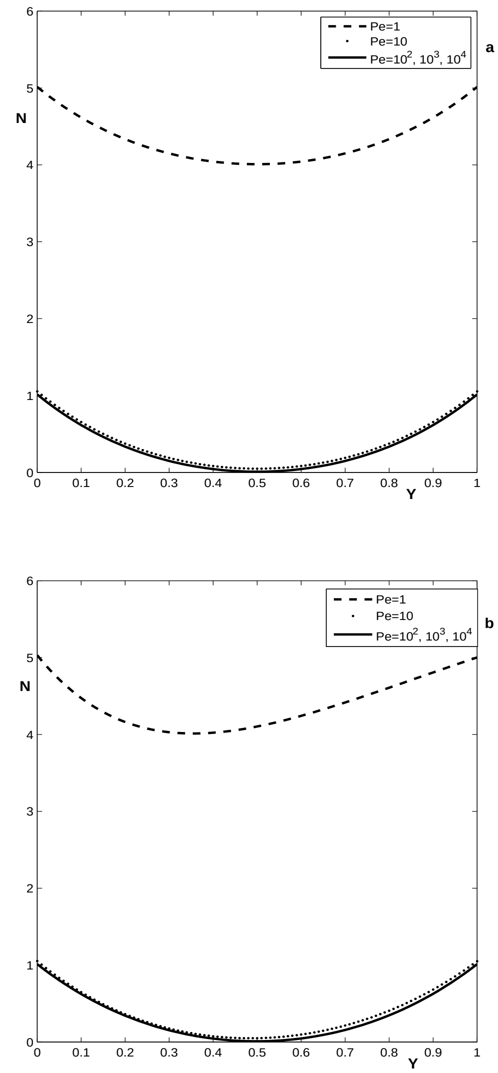

Fig.2 Entropy generation number N versus Y for $\lambda = 1$, $\gamma = \pi/2$ (a) and for $\gamma = \pi/4$ (b).

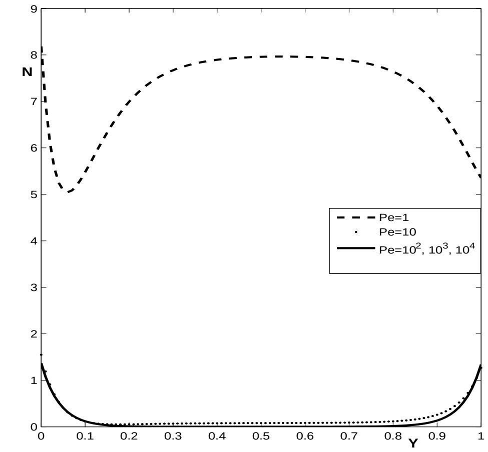

Fig.3 Entropy generation number N versus Y for $\lambda = 100$ and $\gamma = \pi/4$.

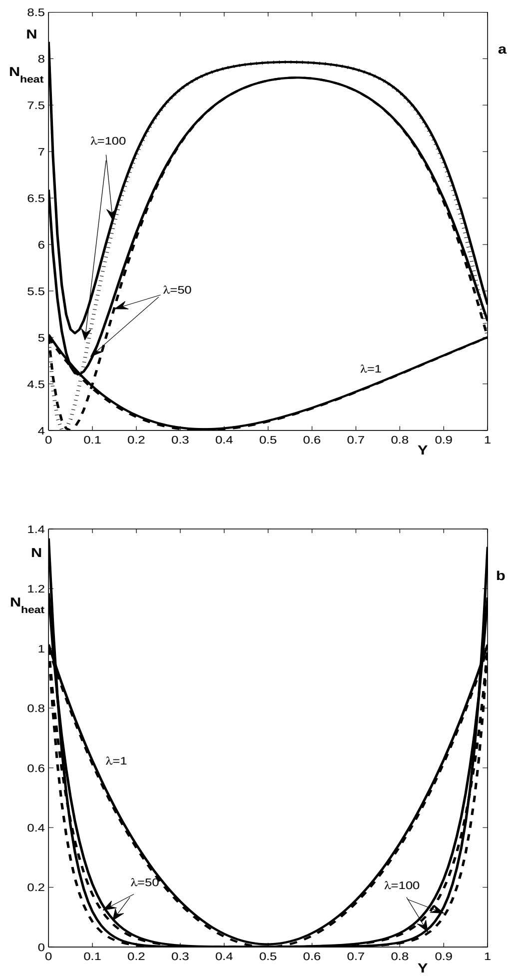

Fig.4 The entropy generation numbers N (shown by line) and N_{heat} (shown by broken line) versus Y for $\gamma = \pi/4$ for $Pe = 1$ (a) and $Pe = 100$ (b).

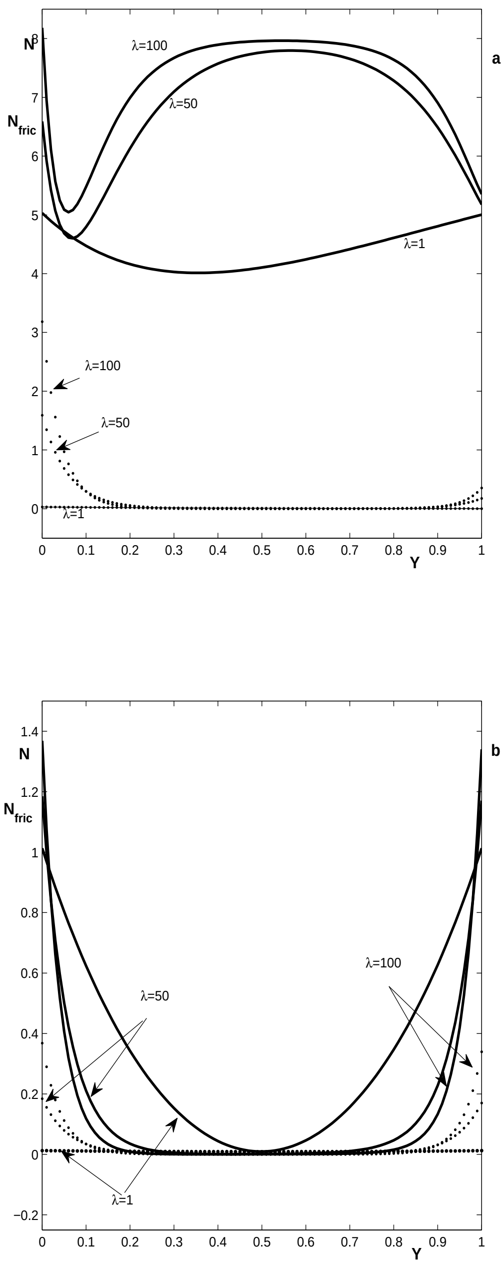

Fig.5 The entropy generation number, N, (shown by line) and fluid friction irreversibility, N_{fric}, (shown by dots) versus Y, for $\gamma = \pi/4$ and for the $Péclet$ numbers $Pe = 1$ (a) and $Pe = 100$ (b).

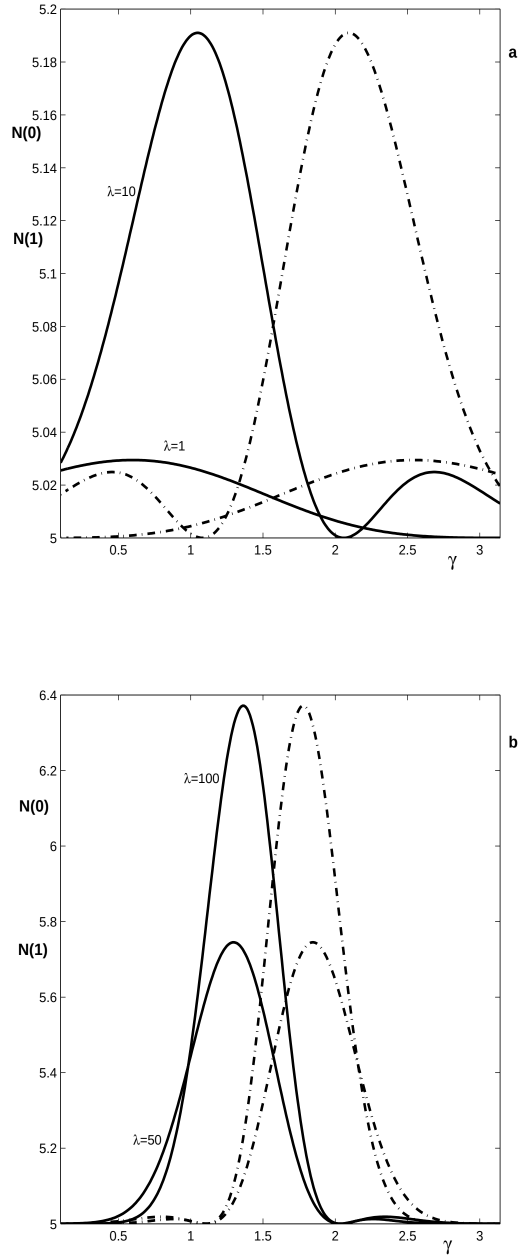

Fig.6 The entropy generation number to the lower wall, $N(0)$ (shown by line) and the entropy generation number to the upper wall, $N(1)$, (shown by broken line), versus the inclination angle of the channel, γ, for different mixed convection parameters $\lambda = 1, 10$ (a) and $\lambda = 50, 100$ (b).

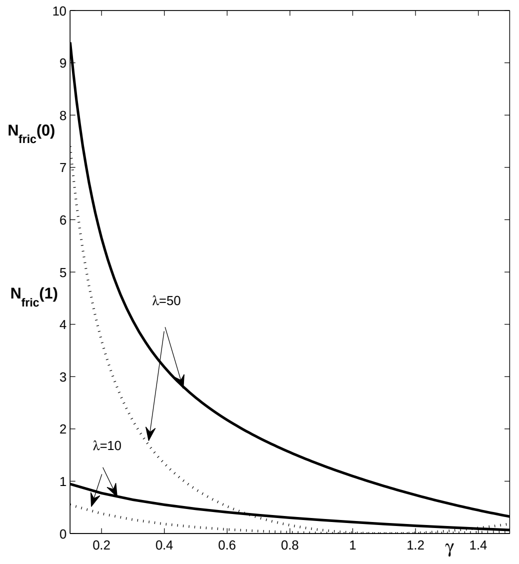

Fig.7 The entropy generation due to fluid friction on the walls, $N_{fric}(0)$ (shown by line) and $N_{fric}(1)$ (shown by broken line) versus γ for $Pe = 1$.

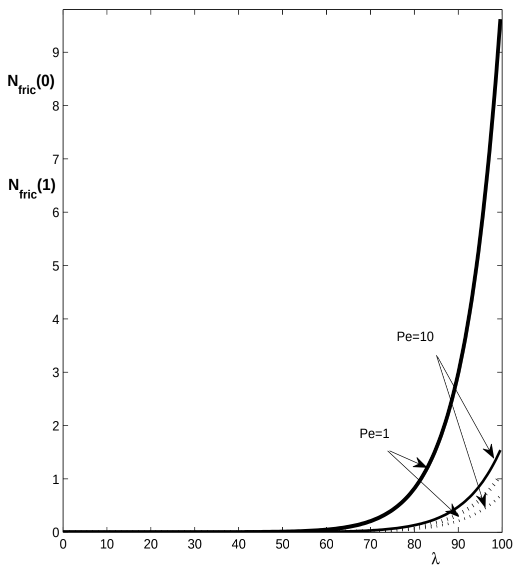

Fig.8 The fluid friction irreversibility and heat transfer irreversibility, on the walls, $N_{fric}(0)$ (shown by line) and $N_{fric}(1)$ (shown by broken line) versus λ.

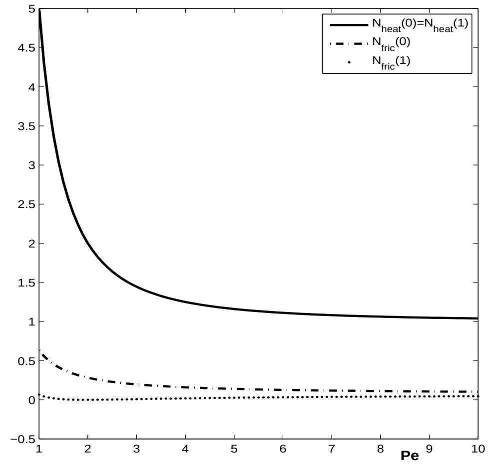

Fig.9 The entropy generation numbers given by heat transfer and fluid friction on the walls versus Pe.

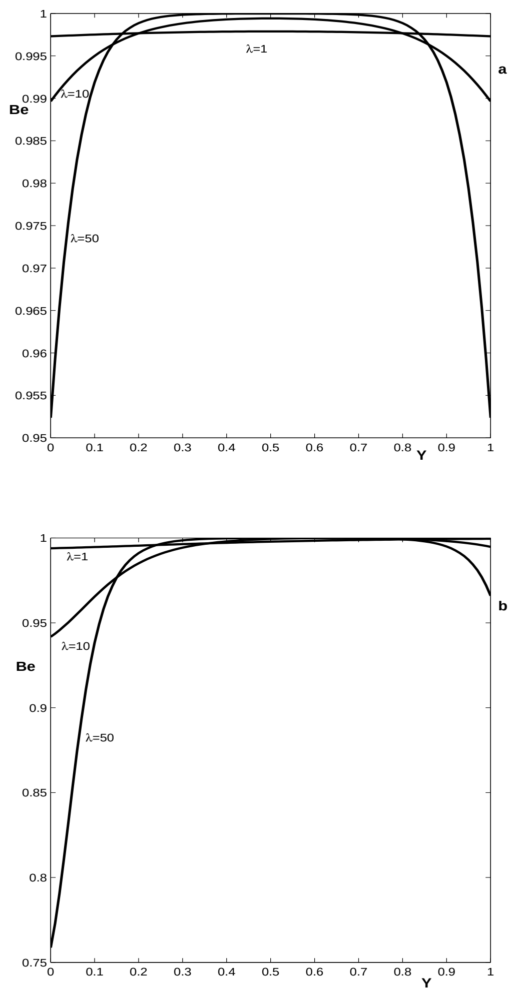

Fig.10 Be number for $\gamma = \pi/2$ (a), $\gamma = \pi/4$ (b).

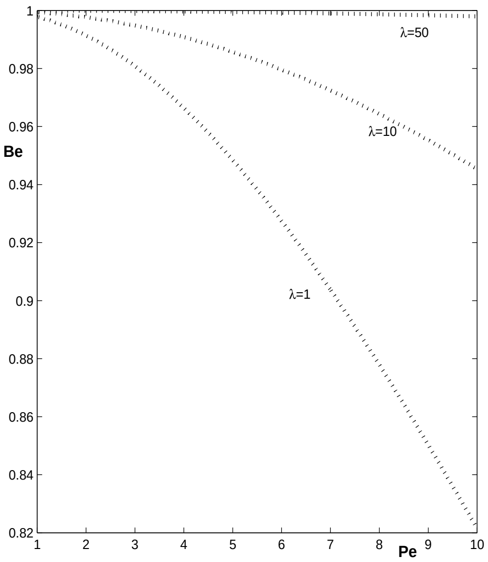

Fig.11 The Bejan number versus Pe for different values of the mixed convection parameter λ.

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