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The Effect of Non-uniform Energy Generation on Entropy Generation in a Plate being Cooled in a Fluid Medium

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Abstract

The prime objective of the present numerical investigation is to examine the effect of non-uniform internal energy generation on entropy generation rates in a plate dissipating heat into its surrounding stream of fluid. Employing second-order accurate finite difference schemes, the partial differential equation governing the temperature distribution in the plate is solved along with the partial differential equations governing the flow and thermal fields in the fluid by satisfying the continuity of temperature and heat flux at the solid-fluid interface. Numerical results are presented and discussed for wide range of values of aspect ratio of the plate, conduction-convection parameter, total energy generation parameter, and flow Reynolds number. Finally, it is concluded that the assumption of uniform energy generation results in erroneous prediction of entropy generation rates. Further, it is found that error in prediction of global entropy generation rate increases with increase in conduction-convection parameter and flow Reynolds number, while it decreases with increase in aspect ratio of the plate and total energy generation parameter.

Keywords: *non-uniform energy generation; entropy generation; conjugate heat transfer; finite difference method*

Nomenclature

A_r	aspect ratio of the plate
b	width of fluid domain (m)
c_p	specific heat of fluid at constant pressure, (J/kgK)
H	height of the plate (m)
k	thermal conductivity (W/mK)
l_o	distance of the outflow boundary after trailing edge (m)
N_{cc}	conduction-convection parameter
Pr	fluid Prandtl number
q'''	volumetric energy generation (W/m^3)
Q	dimensionless volumetric energy generation function
Q_t	total energy generation parameter
Re_H	flow Reynolds number
S'''_{gen}	local entropy generation rate (W/m^{3}K)
S_l	dimensionless local entropy generation rate
S_g	dimensionless global entropy generation rate
T	temperature (K)
T_o	maximum allowable plate temperature in the plate (K)
u	axial velocity component (m/s)
U	dimensionless axial velocity component
v	transverse velocity component (m/s)
V	dimensionless transverse velocity component
W	half width of the plate (m)
x	axial coordinate (m)
X	dimensionless coordinate in axial direction
y	transverse coordinate (m)
Y	dimensionless transverse coordinate

Greek symbols

θ	dimensionless temperature
θ_∞	dimensionless temperature parameter
μ	dynamic viscosity (kg/ms)
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
Ψ	dimensionless stream function
Ω	dimensionless vorticity

Subscripts

f	fluid

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max	maximum	∞	free stream
s	solid		

1. Introduction

Rectangular plates having non-uniform internal energy generation find application in many thermal systems such as fuel elements of nuclear reactors [1]. The energy generated due to fission within the fuel element is first conducted within itself and eventually dissipated from its lateral surfaces to the surrounding stream of coolant so as to maintain the maximum temperature within the fuel element well within certain permissible limit [2]. As heat transfer process is irreversible, it results in entropy generation. This entropy generated has to be minimized since it is directly proportional to the lost available work [3]. Owing to the preceding facts, quite good number of researchers during the past four decades has paid their attention to the studies on minimization of entropy generation in thermal systems of different geometry. A brief review of the literature relevant to the present study is presented below.

Bejan [4] analytically investigated the problem of entropy generation associated with a heat exchanger assuming uniform heat flux at the heat transfer surfaces. Poulikakos and Bejan [5] analytically investigated the problem of entropy generation minimization in fins of different geometries by assuming one-dimensional axial conduction within the fins and average heat transfer coefficient over their surfaces. San et al. [6] analytically investigated the problem of entropy generation arising out of heat and mass transfer in a parallel plate channel. The problem of entropy generation due to two-dimensional laminar mixed convection flow in a vertical channel with transverse fins attached on its hotter wall was numerically studied by Cheng et al. [7] and the effects of physical and geometrical parameters on distribution of entropy generation were presented. Ruocco [8] numerically investigated the problem of entropy generation associated with conjugate heat transfer from a plate having discrete heat sources. Shuja et al. [9] numerically studied the problem of entropy generation associated with conjugate conduction-forced convection heat transfer from a rectangular block with uniform volumetric heat source. They concluded that entropy generation in the coolant is negligible as compared to that in the block. Ibanez et al. [10] analytically studied the problem of entropy generation associated with steady state one-dimensional conduction in a plate with uniform volumetric energy generation by assuming average heat transfer co-efficient over its surfaces. Bautista et al. [11] analytically studied the problem of entropy generation associated with unsteady state one-dimensional conduction in a slab having uniform volumetric energy generation by assuming average heat transfer co-efficient over its surfaces. Varol et al. [12] numerically studied the problem of entropy generation arising due to conjugate natural convection in a differentially heated rectangular enclosure bounded by two vertical walls of different thicknesses. Mukhopadyay [13] numerically analyzed the problem of entropy generation associated with natural convection heat transfer occurring in square enclosures having discrete heat sources. Aziz and Khan [14] analytically as well as numerically investigated the problem of entropy generation associated with steady state conduction in a plane wall, a hollow cylinder and a hollow sphere having uniform volumetric heat generation. El Haj Assad [15] analytically studied the problem of entropy generation associated with steady state one-dimensional conduction in a slab with non-uniform internal heat generation by assuming average heat transfer co-efficient over its surfaces. Chen et al. [16] performed a numerical study on entropy generation associated with steady, laminar and fully developed mixed convection flow with viscous dissipation in a vertical parallel plate channel. Basak et al. [17] numerically analyzed the problem of entropy generation arising out of natural convection in inclined square cavities by employing finite element method. Torabi and Zhang [18] analytically studied entropy generation rates in composite walls having temperature dependent internal energy generation by assuming steady state one-dimensional conduction within the slab and convective along with radiative conditions over its heat dissipating surfaces.

An up-to-date review of the literature pertinent to entropy generation clearly reveals that with an exception of El Haj Assad [15], and Torabi and Zhang [18] all the investigators have paid their attention to entropy generation studies either with uniform internal energy generation or without internal energy generation. While El Haj Assad [15] assumed average heat transfer co-efficient at the heat dissipating surfaces, unrealistic convective along with radiative boundary conditions were imposed by Torabi and Zhang [18]. Moreover, these studies too are based on the assumption of steady, one-dimensional heat conduction within the solid. Deriving motivation from some of these shortcomings of the previous investigations, the present numerical study aims at examining the effect of non-uniform internal energy generation on entropy generation arising out of conjugate conduction-forced convection heat transfer from a rectangular plate to its surrounding fluid medium.

2. Mathematical Formulation

Figure 1 depicts an energy generating plate of height H, thickness $2W$ and thermal conductivity k_s dissipating heat into the surrounding stream of fluid having density ρ_f, dynamic viscosity μ_f, specific heat c_p, and thermal conductivity k_f. The velocity U_∞ and temperature T_∞ of the fluid at

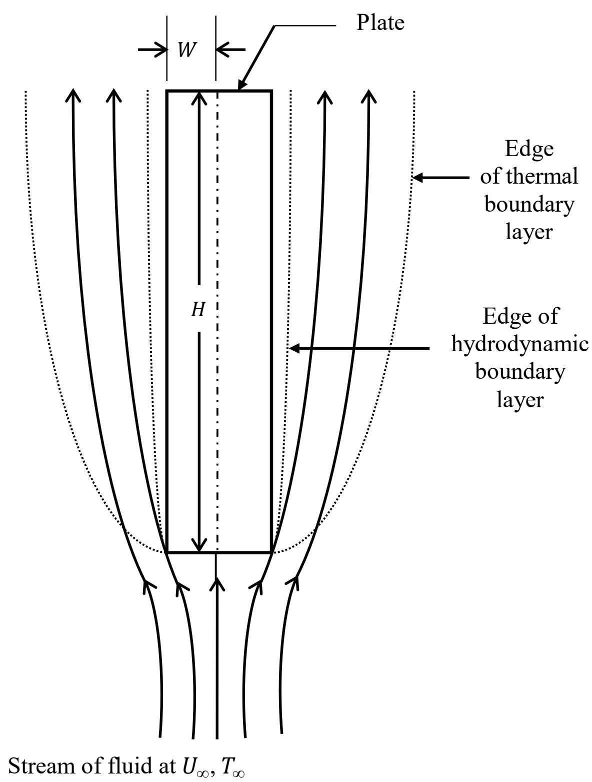

Figure 1. Physical model

the upstream location are taken to be uniform. Under steady state operating conditions, the energy generated within the plate is first conducted to its lateral surfaces and finally dissipated to the surrounding stream of fluid. As a result, entropy is generated both in the plate as well as in the fluid flowing over it. However, the contribution of entropy generation in viscous fluid flow is found to be somewhat insignificant as compared to that in the solid [9]. For transforming the preceding stated physics of the problem into an appropriate mathematical model, the following additional approximations and assumptions are introduced:

(i) The plate material is homogenous and isotropic.
(ii) The thermo-physical properties of the plate material and fluid are constant.
(iii) The heat conduction in the plate is two-dimensional.
(iv) The fluid flow is incompressible, laminar, Newtonian, viscous and two-dimensional.

The conjugate heat transfer problem stated above suggests that temperature distribution in the plate as well as flow and thermal fields in the fluid would be symmetric about the vertical axis of the plate. Therefore, either right or left half of the solution domain is needed to be considered as the computational domain. Figure 2 illustrates such a computational domain with relevant boundary conditions in dimensionless form indicated thereon.

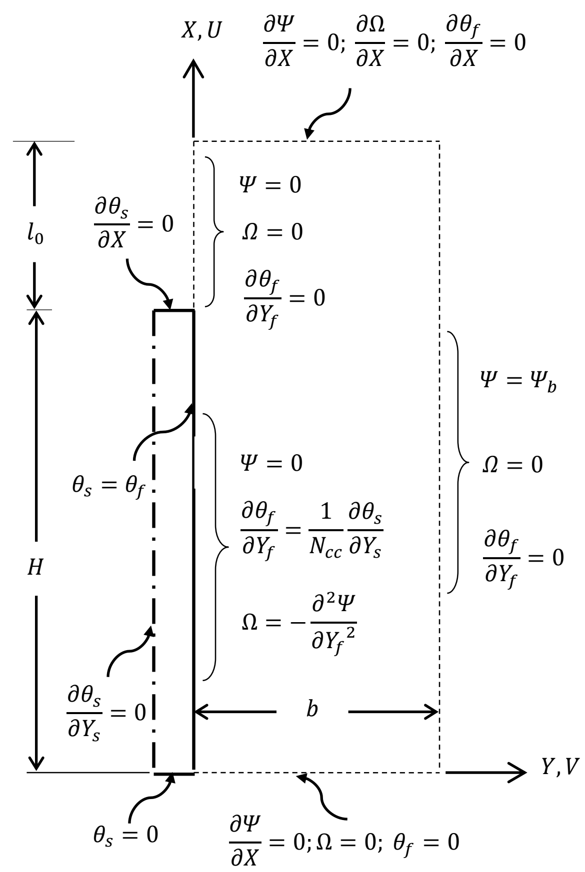

Figure 2. Computational domain

Introducing the assumptions and approximations stated above and by employing first law of thermodynamics, the dimensionless equation governing the two-dimensional steady state temperature distribution in the plate can be derived as:

$$\frac{\partial^2\theta_s}{\partial X^2} + 4{A_r}^2\left(\frac{\partial^2\theta_s}{\partial {Y_s}^2} + Q\right) = 0 \quad (1)$$

It is worth emphasizing here that internal energy generation in the fuel elements of nuclear reactors is non-uniform and it is expressed in terms of cosine function of the axial coordinate [19]. For the present study, the dimensionless volumetric energy generation function Q appearing in Equation (1) is expressed as [20]:

$$Q = Q_{max}\, cos\pi\left(\frac{1}{2} - X\right) \quad (2)$$

In order to compare entropy generation rates in the plate having non-uniform internal energy generation with those of uniform ones on equitable terms, total energy generation parameter Q_t is used as a common input parameter which is essentially obtained by integrating Q over the volume of the plate [20]. Thus, total energy generation parameter Q_t is expressed in terms of maximum dimensionless energy generation rate Q_{max} as:

$$Q_t = \frac{2}{\pi}Q_{max} \quad (3)$$

The dimensionless equations governing the flow and thermal fields in the fluid can be expressed as:
Stream function:

$$\frac{\partial^2\Psi}{\partial X^2} + \frac{\partial^2\Psi}{\partial {Y_f}^2} = -\Omega \quad (4)$$

Vorticity transport:

$$U\frac{\partial\Omega}{\partial X} + V\frac{\partial\Omega}{\partial Y_f} = \frac{1}{Re_H}\left(\frac{\partial^2\Omega}{\partial X^2} + \frac{\partial^2\Omega}{\partial {Y_f}^2}\right) \quad (5)$$

Energy:

$$U\frac{\partial\theta_f}{\partial X} + V\frac{\partial\theta_f}{\partial Y_f} = \frac{1}{Re_H Pr}\left(\frac{\partial^2\theta_f}{\partial X^2} + \frac{\partial^2\theta_f}{\partial {Y_f}^2}\right) \quad (6)$$

Where, the dimensionless stream function, Ψ and dimensionless vorticity, Ω appearing in Equations (4) and (5) are defined as:

$$U = \frac{\partial\Psi}{\partial Y_f}, \quad V = -\frac{\partial\Psi}{\partial X} \text{ and } \Omega = \frac{\partial V}{\partial X} - \frac{\partial U}{\partial Y_f} \quad (7)$$

The dimensionless parameters and variables present in Equations (1) - (7) are defined as:

$$X = \frac{x}{H}, \qquad Y_s = \frac{y}{W}, \qquad Y_f = \frac{y}{H}, \quad U = \frac{u}{U_\infty}, \qquad V = \frac{v}{U_\infty}, \qquad \theta = \frac{T - T_\infty}{T_0 - T_\infty}$$

$$A_r = \frac{H}{2W}, \qquad N_{cc} = \frac{k_f}{k_s}\left[\frac{W}{H}\right],$$

$$Pr = \frac{\mu_f c_p}{k_f}, \qquad Q = \frac{q'''W^2}{k_s\,(T_0 - T_\infty)}, \qquad Re_H = \frac{U_\infty\, H}{\nu_f} \qquad (8)$$

Local entropy generation rate S_l in the plate can be computed from the temperature distribution using the following equation:

$$S_l = \frac{1}{(\theta_s + \theta_\infty)^2}\left[\frac{1}{4{A_r}^2}\left(\frac{\partial \theta_s}{\partial X}\right)^2 + \left(\frac{\partial \theta_s}{\partial Y_s}\right)^2\right] \qquad (9)$$

Where, symbols S_l and θ_∞ present in Equation (9) are defined as $S_l = \frac{S'''_{gen} W^2}{k_s}$ and $\theta_\infty = \frac{T_\infty}{T_0 - T_\infty}$, respectively. Once, the values of S_l in the plate is obtained, the global entropy generation rate S_g in the plate can be computed by employing the following integral equation:

$$S_g = 2\int_0^{-1}\int_0^{1} S_l(X, Y_s)\, dX\, dY_s \qquad (10)$$

3. Numerical Solution

Equations (1), (4), (5) and (6) are coupled partial differential equations and therefore, these equations have to be solved numerically in an iterative manner. While Equations (1) and (4) are discretized using central difference schemes and the resulting system of linear algebraic equations are solved using Line-by-Line Gauss-Seidel iterative solution procedure, pseudo-transient forms of Equations (5) and (6) are discretized using ADI finite difference scheme and the resulting system of linear algebraic equations are solved iteratively by employing Thomas Algorithm. Once the converged values of temperature field in the plate is obtained, local and global entropy generation rates are computed using Equations (9) and (10), respectively.

The numerical results presented in this paper are computed using an indigenously developed computer code which takes care of different kinds of boundary conditions merely by an artefact of computer programming. This code, which is essentially developed for computing steady, two-dimensional temperature distribution in an energy generating plate and steady, two-dimensional flow and thermal fields in the fluid, can also generate numerical results for conjugate conduction-forced convection in a rectangular fin. Figure 3 illustrates a comparison of temperature distribution in a rectangular fin along solid-fluid interface obtained using the present code with those of Sunden [21] which can be seen to be in good agreement. The details of the grid convergence tests performed are not presented for the sake of brevity.

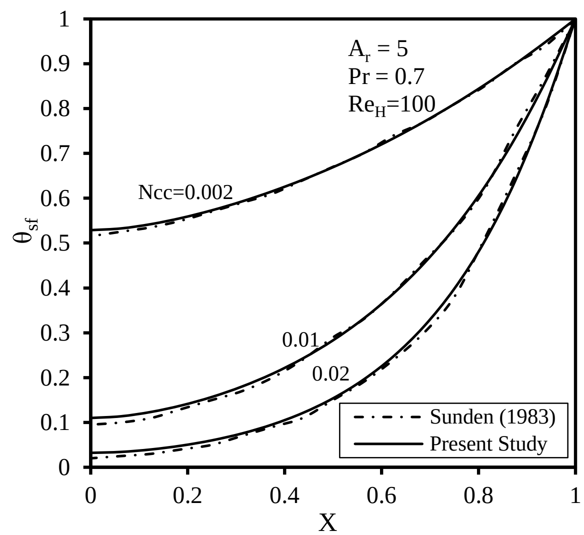

Figure 3. Comparison of solid-fluid interface temperature profile with that of Sunden [21] for different values of N_{cc}

4. Results and Discussion

The prime objective of the present numerical study is to examine the effect of non-uniform internal energy generation on entropy generation rates in a plate dissipating heat into surrounding fluid medium by forced convection. Keeping fluid Prandtl number Pr and dimensionless temperature parameter θ_∞ as fixed at 0.005 and 0.4 respectively, numerical results in the form of transverse profiles of dimensionless plate temperature θ_s and local entropy generation rate S_l, and in the form of variation of global entropy generation rate S_g with involved thermo-geometric parameters such as aspect ratio of the plate A_r, conduction-convection parameter N_{cc}, total energy generation parameter Q_t and flow Reynolds number Re_H are presented and discussed in detail.

Figure 4 depicts the effect of non-uniform volumetric energy generation on transverse variation of θ_s in the plate at two distinct axial locations X = 0.25 and X = 0.75, while the values of A_r, N_{cc}, Q_t and Re_H are being kept constant at 10, 0.50, 0.50 and 2500 respectively. It is worth noticing from this figure that uniform internal energy generation assumption results in significant underestimation of θ_s, which becomes more and more pronounced towards the central line of the plate. Further, it can be noted that the underestimation of θ_s due to the assumption of uniform energy generation decreases towards the trailing edge of the plate and it even results in slight overestimation of θ_s in the vicinity of solid-fluid interface near the trailing edge of the plate. Precisely, error in prediction of θ_s in the vicinity of the central line of the plate decreases from 11.07% at X = 0.25 to 4.52 % at X = 0.75.

Figure 5 exhibits the effect of non-uniform internal energy generation on transverse variation of S_l in the plate at two different axial locations, while the values of A_r, N_{cc}, Q_t and Re_H are being kept constant at 10, 0.50, 0.50 and 2500 respectively. It is abundantly clear from this figure that, for both uniform and non-uniform energy generation cases, S_l takes its maximum value in the vicinity of the solid-fluid interface and it keeps on decreasing to its minimum value along the central line of the plate. Further, it is worth noticing from this figure that the assumption of uniform internal energy generation results in underestimation of S_l except in the region very close to the central line of the plate. Furthermore, it can be clearly noticed from this figure that underestimation of S_l due to the assumption of uniform energy generation decreases towards the trailing edge of the plate. Precisely, it can be noted that error in prediction of S_l in the vicinity of the lateral surface of the plate decreases from 20.13% at X = 0.25 to 5.06 % at X = 0.75.

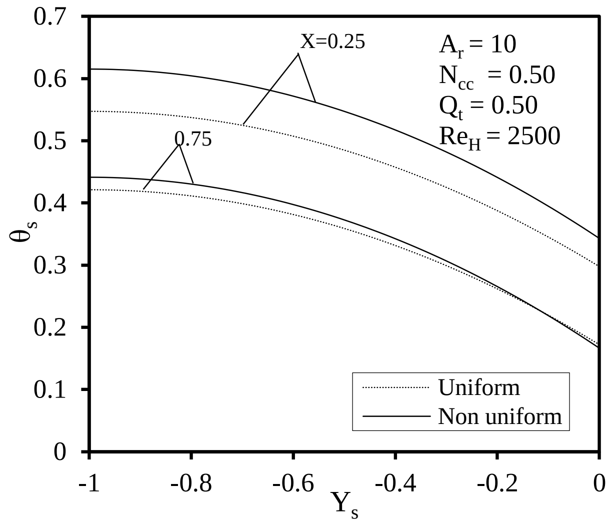

Figure 4. Comparison of transverse temperature profiles between uniform and non-uniform energy generation cases

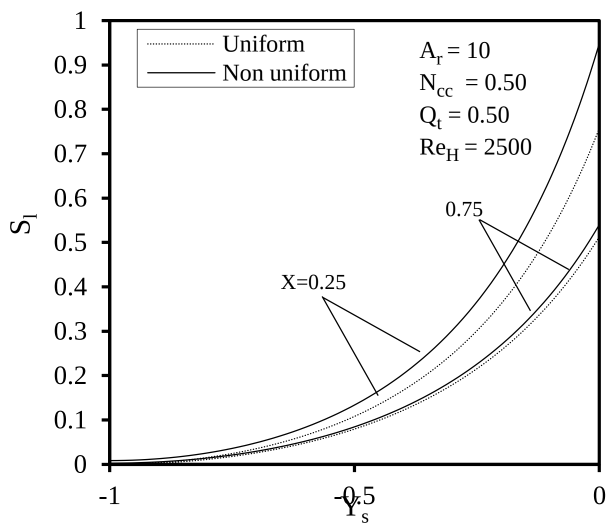

Figure 5. Comparison of transverse local entropy generation rate profiles between uniform and non-uniform energy generation cases

Figure 6 illustrates the variation of S_g with A_r for two distinct values of N_{cc}, while Q_t and Re_H are being kept constant at 0.50 and 2500, respectively. It can be easily noticed from this figure that the assumption of uniform internal energy generation results in under prediction of S_g which is more noticeable for smaller values of A_r as compared to its larger values. Further, it is quite visible from this figure that for any particular value of A_r, the error in prediction of S_g increases with increase in N_{cc}. To be very precise, for A_r= 2.5, the percentage error in the prediction of S_g increases from 8.66% to 10.22% as N_{cc} increases from 0.40 to 0.70.

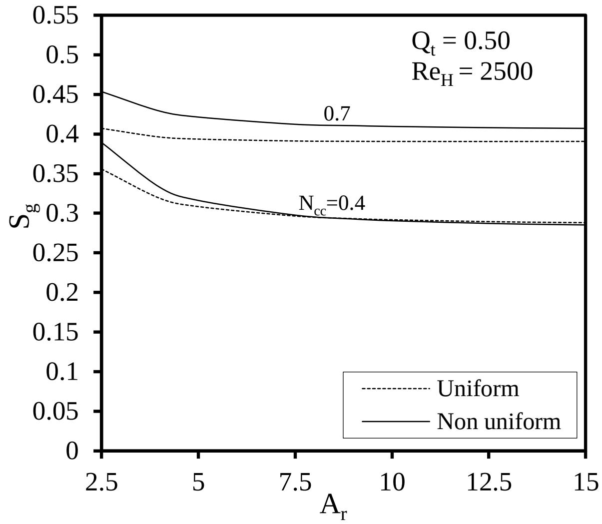

Figure 6. The effect of non-uniform energy generation on the variation of S_g with A_r for different values of N_{cc}

Figure 7 presents the variation of S_g with A_r for two different values of Re_H while N_{cc} = 0.50 and Q_t = 0.50 are being kept constant. It is worth noticing from this figure that S_g decreases with increase in A_r for both uniform and non-uniform energy generation cases. Further, it can be noted from this figure that error in prediction of S_g due to the assumption of uniform energy generation decreases as A_r takes its higher and higher values. To be very precise, for Re_H = 3500, the under prediction of S_g due to the assumption of uniform energy generation decreases from 9.83% to 3.07% as A_r increases from 2.5 to 15. Furthermore, it is interesting to note that, for Re_H = 1500, the assumption of uniform energy generation results in slight overestimation of S_g for all values of $A_r \geq 7.5$.

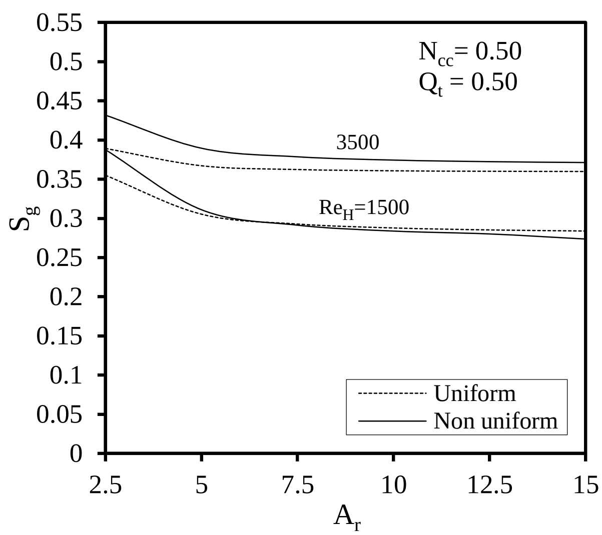

Figure 7. The effect of non-uniform energy generation on the variation of S_g with A_r for different values of Re_H

Figure 8 shows the effect of non-uniform internal energy generation on the variation of S_g with N_{cc} for two distinct values of Re_H while the values of A_r and Q_t are being kept constant at 10 and 0.50, respectively. It is quite clear from this figure that for both uniform and non-uniform energy generation cases, S_g keeps on increasing with increase in N_{cc}. It is also evident that, the assumption of uniform energy generation results in erroneous prediction of S_g which is more noticeable for larger values of N_{cc} and Re_H. Precisely, for Re_H = 3500, underestimation of S_g due to the assumption of uniform energy generation increases gradually from 0.50% to 6.29% as N_{cc} increases from 0.35 to 0.75.

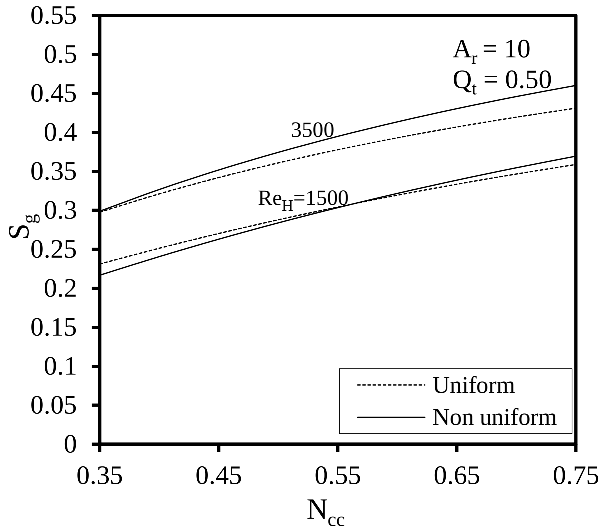

Figure 8. The effect of non-uniform energy generation on the variation of S_g with N_{cc} for different values of Re_H

Figure 9 depicts the effect of non-uniform internal energy generation on the variation of S_g with Q_t for two different values of N_{cc} while A_r = 10 and Re_H = 2500 are being kept constant. It is worth noticing from this figure that S_g increases appreciably with increase in Q_t for both cases. It can be also noted from this figure that, the error in prediction of S_g due to the assumption of uniform energy generation increases with increase in the value of N_{cc}. Precisely, at Q_t= 0.25, underestimation of S_g due to the uniform energy generation assumption increases gradually from 7.32% to 10.66% as N_{cc} increases from 0.40 to 0.70. Further, it is worth mentioning here that, error in prediction of S_g due to the assumption of uniform energy generation decreases as Q_t takes its larger values. Furthermore, it is interesting to note that, for N_{cc}= 0.40, the assumption of uniform energy generation results in slight overestimation of S_g for all values of $Q_t \geq 0.50$.

Figure 10 depicts the effect of non-uniform internal energy generation on the variation of S_g with Re_H for two distinct values of Q_t while A_r = 10 and N_{cc} = 0.50 are being kept constant. It is abundantly clear from this figure that S_g increases with increase in Re_H for both cases. It is also evident from this figure that underestimation in S_g due to the assumption of uniform energy generation increases with increase in Re_H. To be very precise, for $Q_t = 0.25$, the percentage error in prediction of S_g increases from 6.76% to 9.95% as Re_H increases from 1500 to 3500.

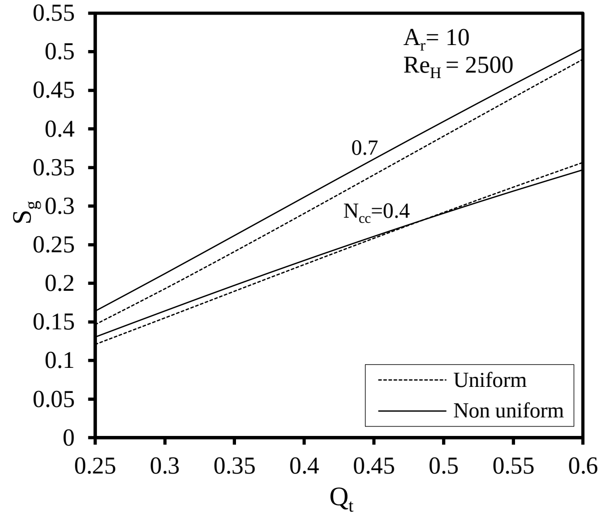

Figure 9. The effect of non-uniform energy generation on the variation of S_g with Q_t for different values of N_{cc}

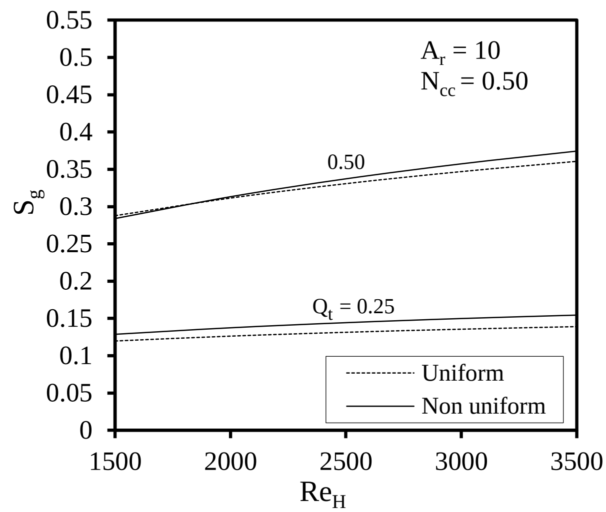

Figure 10. The effect of non-uniform energy generation on the variation of S_g with Re_H for different values of Q_t

5. Conclusions

The main objective of the present investigation is examining the effect of non-uniform internal energy generation on local and global entropy generation rates in a plate dissipating heat into its surrounding fluid medium by conjugate forced convection. Keeping fluid Prandtl number and dimensionless temperature parameter as fixed, numerical results are obtained for wide range of values of aspect ratio of the plate A_r, conduction-convection parameter N_{cc} , total energy generation parameter Q_t and flow Reynolds number Re_H. On the basis of discussion of the results, it is concluded that idealistic uniform internal energy generation results in erroneous prediction of local and global entropy generation rates. Further, it is found that under prediction of global entropy generation rate S_g in the plate increases considerably with increase in N_{cc} and Re_H. Furthermore, it is found that error in prediction of S_g due to uniform energy generation assumption slightly decreases with increase in A_r and Q_t.

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