"Effects Of Heat Source Location On Natural Convection In A Square Cavity"

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Abstract

Natural convection in a closed square cavity has occupied the centre stage in many fundamental heat transfer analysis which is of prime importance in certain technological applications. Infact buoyancy driven convection in a sealed cavity with differentially heated isothermal walls is a prototype of many industrial applications such as energy efficient buildings, operation and safety of nuclear reactor and convective heat transfer associated with electronic cooling equipment. The internal flow problems are considerably more complex than external ones.

In electronic systems normally the heat generating bodies exist within the cavity. The effect of the presence of heat source on the mass flow rate and heat transfer is considered in present case for investigation. In order to verify the methodology of using fluent, the commercial software, the available problem in the literature is verified for parametric study on the location of heat source and its strength is considered for study. In present work, the given source is split into two parts and its effect on the flow rate and heat transfer is studied. An attempt is made for the best location of the heat source in the cavity so that it can be used in the electronic equipment generating heat.

Keywords – Electronic equipment**,** Heat Source**,** Natural Convection**,** Square Cavity**,** Stream function

Nomenclatures

AR = Aspect ratios, H/L, Gr = Grashoff Number

g = Acceleration due to gravity (m/s^2), Ra = Rayleigh number

H = Height of the cavity (m), Pr = Prandtl number

h = Convective heat transfer coefficient (W/m^{2}K)

k = Thermal conductivity (W/m.K), Nu = Nusselt number

L = Length of the cavity (m), T = Temperature (K)

q = Heat flux (W/m^2)

Greek Symbols

α = Thermal diffusivity (m^2/s)

β = Volume expansion coefficient (K^{-1})

ρ = Fluid Density (kg/m^3)

ν = Kinematic viscosity (m^2/s)

θ = Dimensionless temperature

Subscript

b = Bottom wall   s = Side wall

I. INTRODUCTION

One of the objectives of present day electronics industry is to reduce the size of an electronic device as small as possible with a simultaneous increase in the processor speed. This has made the PCBs to get cramped with miniscule electronic components like transistors, capacitors etc. in a large number on to a miniaturized PCB board. The repercussion of this is the excessive heat generation from the PCB boards which has to be driven away from the source, if not; the processor performance will be affected.

The problem of convective heat transfer in an enclosure has been studied extensively because of the wide application of such process. The convection in enclosures for various different temperature boundary conditions and natural convective cooling of heat-dissipating electronic components, located in square enclosure, and cooled by an isothermal side wall has been studied by Tanmay and Basak [1]. Davis and Jones [2] studied the pure natural convection with uniformly heated walls. Papanicolaou and Gopalakrishna [3] simulated the natural convection in a horizontal, enclosed air layer due to a discrete, constant heat flux source at the bottom surface. The parameters studied are the overall aspect ratio (length/height of the layer), the ratio of source length to total length, and the Rayleigh number. Nguyen and Prudhomme [4] studied the free convection flow in a horizontal rectangular cavity submitted to a uniform heat flux at the horizontal as well as vertical walls. Based on the analytical solutions for the flow amplitude in terms of the Rayleigh numbers, the onset and development of convective flow are shown in details. The results indicate that flow patterns generally consist of isothermal side walls and uniform and non uniform heating of bottom wall because of buoyancy forces induced. Numerical solutions were obtained for Pr = 0.71 to 10 in the range of Ra = 10^3 to 10^5. They got smooth solution in terms of Stream functions and isotherm contours for wide ranges of Pr and Ra with uniform and non-uniform heating of the bottom wall. Iwatsu et al. [5] performed numerical studies for the viscous flow of a thermally stratified fluid in a square container.

II. SIMULATION MODEL

The present study is based on the configuration of Tanmay and Basak [1] and Aydin and Yang [15] where the isothermal side wall and heat source at the bottom wall is replaced with a constant flux heat source for the entire length as well as at different locations of the bottom wall. Then the heat source was split into two parts and the effects of Rayleigh Number and location of heat source at different length of the bottom surface on natural convection heat transfer was studied, which is physically more realistic.

A cavity as illustrated in Figs 2.1 to 2.6 was chosen for simulating natural convective flow and heat transfer characteristics. This problem was selected since the majority of work in the literature is dealing with convection in enclosures restricted to the cases of uniform temperature heating, varying temperature heating and heat flux in a fixed location of different geometry, like rectangular, square, cylindrical, and spherical cavities. In the present work heat source at different location is considered and analysis is carried out for Rayleigh number 10^3, 10^4, and 10^5 to identify the effective location of heat source in the enclosure. Also, sufficient number of publications are available for the comprehensive analysis on flow and heat transfer in closed cavities for Ra = 10^5 and for a single location of heat source. But in the present work the heat source is split into two halves and the analysis is carried out for Ra = 10^3, 10^4, and 10^5. The cavity of length (*L*), and height (*H*), has a hot bottom wall. The top wall is assumed to be adiabatic as shown in the Figs 2.1 to 2.6; the gravitational force is acting downwards. A buoyant flow develops inside the cavity because of thermally induced density gradient. Heat is transferred from the hot wall to cold wall.

The parts excluding the heat source in the bottom wall and the entire upper wall are assumed to be adiabatic. The enclosure represents a practical system such as an air-cooled electronic device. The mass flow rate in the cavity is induced by the buoyancy force resulting from the heat source at the bottom wall. In the present work, the flow and heat transfer phenomena in the cavity are investigated for a series of Rayleigh numbers and effects of different locations of heat source. Symmetric placement of the heat source on either ends of the cavity as well as at the centre is also investigated. Also, efforts are made to identify the best location of the heat source among the different cases studied.

This study includes additional computations for various locations of heat source of length L over the entire length of bottom wall and at lengths of L/2, L/4, and L/8 located at the centre and split at the bottom with the length of L/4 and L/8 located towards the left and the right corners of the wall and their effects on the heat transfer process is analyzed. The study was also conducted to identify the preferred location of the heat source. The results are presented in terms of the Local Nusselt number and maximum temperature at the heat source surface.

A penalty finite Volume method with bi-quadratic rectangular elements has been used to solve the governing mass, momentum and energy equations. The numerical procedure adopted in the present study yields consistent performance over a wide range of parameters (Rayleigh number Ra=10^3, 10^4, and 10^5 while Prandtle number Pr=0.7).

III. MATHEMATICAL MODEL

Cavities as illustrated in **Figs 2.1 to 2.6** were chosen for simulating natural convective flow and heat transfer characteristics. The cavity of length (*L*), and height (*H*), has a hot bottom wall with a uniform heat source of length L / L/2, L/4, and L/8 located at the centre and two constant heat fluxes of length L/4 and L/8 towards the left and right corner of the wall. The two cold vertical walls are at constant temperature *Tc* and the top wall is adiabatic. The gravitational force is acting downwards as shown in the **Figs.2.1 to 2.6**

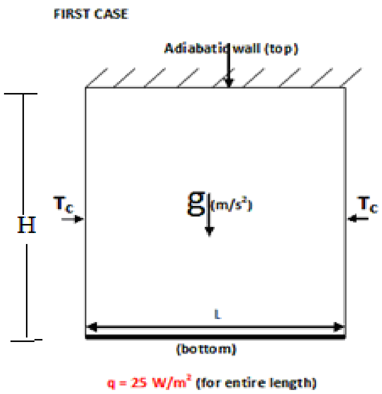

Fig 2.1 case I (along the hole bottom length)

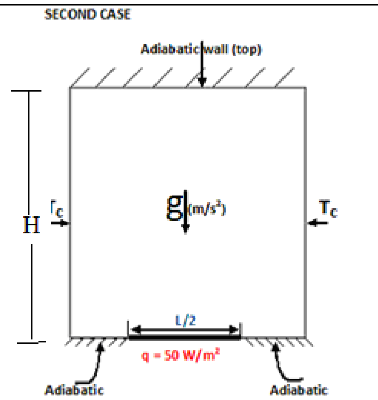

Fig 2.2 case II (half of the length at centre)

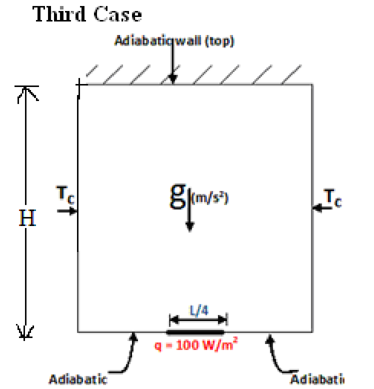

Fig 2.3 Case III (Quarter of the length at the centre)

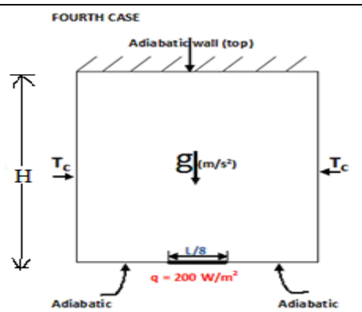

Fig 2.4 Case IV (one eigth of the length at the centre)

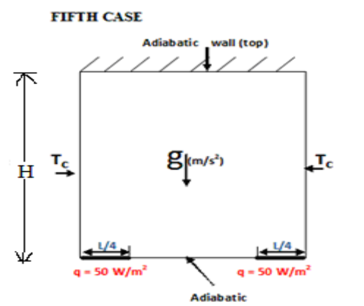

Fig 2.5 Case V (Quarter of the length split on either end)

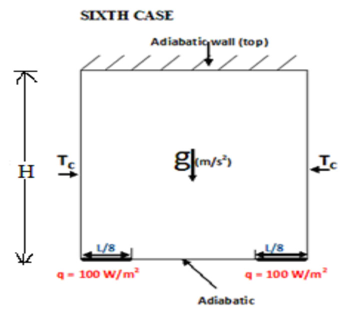

Fig 2.6 Case VI (one eighth of the length split on either end)

A buoyant flow develops because of thermally induced density gradient. Heat is transferred from the hot wall to cold wall

The governing equations for natural convection flow are conservation of mass, momentum and energy equations and they are written as

Continuity equation: $\dfrac{\partial u}{\partial x}+\dfrac{\partial v}{\partial y}=0$ (3.1)

X-momentumequation:

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\gamma\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \quad (3.2)$$

Y-momentumequation:

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\gamma\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+g\beta(T-T_c)$$

(3.3)

Energyequation: $$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)$$

(3.4)

Where, x and y are the dimensional co-ordinates along the horizontal and the vertical directions respectively: u and v are x and y components of velocity respectively.

No-slip boundary conditions are specified at all walls, $u = v = 0$

3.1 **Temperature and Heat flux Boundary Conditions**

Following are the different cases considered for study

For Bottom wall:
Case I: Constant Heat flux for entire length of

bottom wall
Case II: Heat source of length L/2 at centre.
Case III: Heat source of length L/4 at centre.
Case IV: Heat source of length L/8 at centre.
Case V: Heat source of length L/4 on either end of the Bottom wall
Case VI: Heat source of length L/8 on either end of the Bottom wall

For Top wall: $\frac{\partial T}{\partial y}(x, H) = 0,$ (3.5)

Sidewalls: $T(0, y) = T(L, y) = T_c$

The changes of variables are as follows:

$$\theta = \frac{T - T_c}{T_h - T_c}, \quad \Pr = \frac{\nu}{\alpha}, \quad Ra = \frac{g\beta(T_h - T_c)L^3 \Pr}{\nu^2} = \frac{g\beta q'' L^4 \Pr}{k\nu^2} \qquad (3.6)$$

In the present investigation, the geometries have been created and discretized using Gambit 2.4. Fluent 6.3 CFD package is used to simulate the natural convection of air in cavities. The effect of various temperature and heat flux boundary conditions at the bottom wall (as mentioned above) is studied. The side walls are subjected to constant temperature and the top wall is adiabatic. The study was carriedout for various Raleigh numbers.

3.2 Stream Function and Nusselt Number

3.2.1 Stream Function

The fluid motion is displayed using the stream function Ψ obtained from velocity components u and v. The relationship between stream function, Ψ and velocity components for two dimensional flows are given by Batchelor

$$u = \frac{\partial \psi}{\partial y} \text{ and } v = -\frac{\partial \psi}{\partial x} \qquad (3.7)$$

This leads to a single equation:

$$\frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} = \frac{\partial u}{\partial y} - \frac{\partial v}{\partial x} \qquad (3.8)$$

Using the above definition of the stream function used for identifying the mass flow rate in the cavity, the positive sign of Ψ denotes anticlockwise circulation and the negative sign of Ψ clockwise circulation.

3.2.2 Nusselt Number

In order to determine the local Nusselt number, the temperature profiles are fit with quadratic (three nodal points near the wall are considered), cubic and bi-quadratic polynomials and their gradients at the walls are determined. It has been observed that the temperature gradients at the surface are almost the same for all the polynomials considered. Hence only a quadratic fit is made for the temperature profiles to extract the local gradients at the walls to calculate the local heat transfer coefficients from which the local Nusselt numbers are obtained. Integrating the local Nusselt number over each side, the average Nusselt number for each side is obtained as

$$\overline{\mathrm{Nu}_b} = \int_0^L \mathrm{Nu}_b \, dx \ (3.9), \quad \overline{\mathrm{Nu}_s} = \int_0^H \mathrm{Nu}_s \, dy \qquad (3.10)$$

IV. RESULTS AND DISCUSSION

4.1 Verification of Present Methodology.

Computations are carried out for Raleigh numbers from 10^3 to 10^5 and Pr = 0.71. Basak et al. [1] have studied for 10^3 to 10^5 with square cavity, for the cases of constant temperature and sinusoidally varying temperature at the bottom wall. However, in the present case, the study is extended to Ra = 10^3, 10^4 and 10^5 and Pr = 0.7 for different locations of heat source of different intensity. A constant bottom wall heat flux case is also included. The Local Nusselt numbers are computed by the present methodology for the values of Ra ranging from 10^3 to 10^5 with Pr = 0.71 and are compared for different locations (cases) of heat source. The heat source location which provides minimum contour temperature is identified as the preferable location. Fig.4 shows streamlines and temperature profiles for the Case I and for remaining cases results are tabulated in Table.1. The agreement is found to be excellent for the first case which validates the present computations of Basak and Tanmay [1].

When Ra = 10^3, Pr = 0.71, q" = 25 W/m^2 (constant for entire bottom wall).
As seen from the Fig 4 (a) and Table 1, the local Nusselt number is minimum at the centre while it is maximum at the side walls. This is due to the conduction dominance at the centre and

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convection is dominance at the edges. For this case, Nusselt number is less because of conduction dominance. In Fig 4(d), the Contour plot shows that temperature is maximum at the centre of the cavity and reduces towards the side wall. The maximum temperature is 388 K. This is due to the heat concentration at the centre. Also, contour lines of temperature are symmetric about the centre axis of the cavity. This is again due to the heat flux being uniform along the bottom wall. Also, the stream function (mass flow /second) is 3.90 x 10^{-6} kg/s as shown in Fig 4(g). Mass flow rate is minimum, since conduction is dominant, which is confirmed from the less value of Nusselt number. Contours of stream function lines are symmetric about the central axis of the cavity as shown in the Fig 4 (g).

When Ra = 10^4, Pr = 0.71, q" = 25 W/m^2 (constant for entire bottom wall).

As seen from the Fig 4 (b) and Table 1, maximum Nusselt number is 37.92, but still conduction is dominant in this case also. Also, as seen from the Fig 4 (b) and Table.1, the local Nusselt number is minimum at the centre and is maximum at the side walls. This is due to the dominance of conduction at the centre, while convection is dominant at the edges. In Fig 4 (e), the contour plot indicates that temperature is maximum at the centre and reduces towards the side wall. The maximum temperature is 386 K. This is again due to the concentration of heat source at the centre. Also, contours of temperature lines are laterally shifted due to convection. As seen from the Fig 4 (h), mass flow rate increases to 5.67x10^{-5} kg/sec this is more as compared to the results for case (i).

When Ra = 10^5, Pr = 0.71, q" = 25 W/m^2 (constant for entire bottom wall). As seen in the fig 4 (c) and Table 1, the maximum Nu is 45.73. This indicates that convection is more dominant in this case. Nusselt number at the edges increases due to convection while decreases at centre due to conduction. As seen from the Fig 4 (f), the maximum temperature in the cavity is 355 K. Temperature decreases when Ra = 10^5 which is due to the high Rayleigh number. As Ra increases Nu increases and convection will dominate. As seen from the Fig 4 (i), Stream function value is 3.13x10^4 kg/sec. As Ra increases mass flow rate also increases.

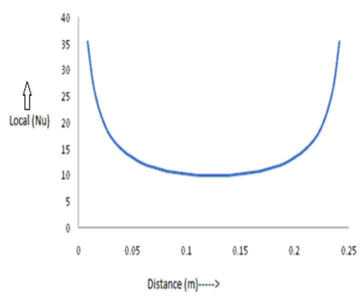
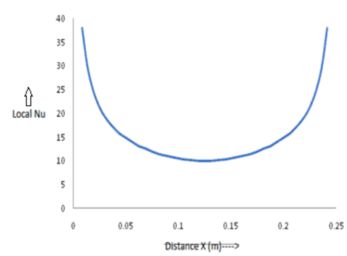


Fig 4 (a) Local Nusselt No V/s Distance (m) when Ra= 10^3 for 1st case



Fig' 4 (b) Local Nusselt No v/s Distance when Ra=10^4 for 1st case

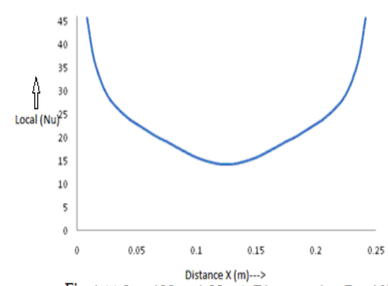


Fig 4 (c) Local Nusselt Nu v/s Distance when Ra=10^5 for case 1st

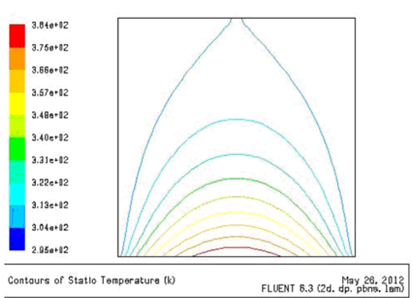


Fig 4 (d) Temperature contour plot when Ra= 10^3 for 1st case

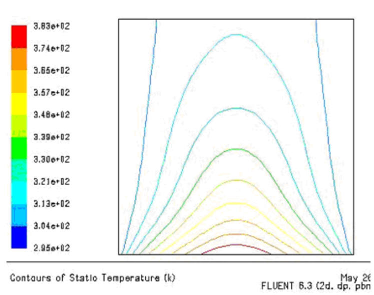


Fig 4 (e)Temperature contour when Ra=10^4 for case 1st

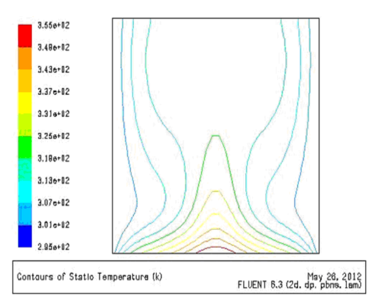


Fig. 4 (f) Contours of Static Temperature when Ra=10^5 for case 1st

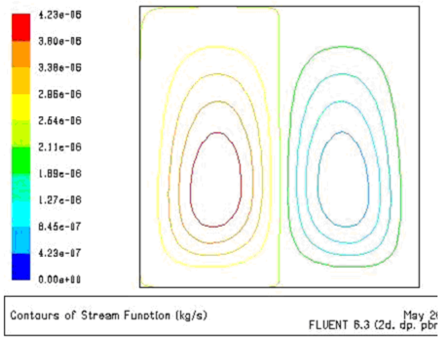


Fig 4(g) Contours of Stream function when Ra=10^3 for 1st case

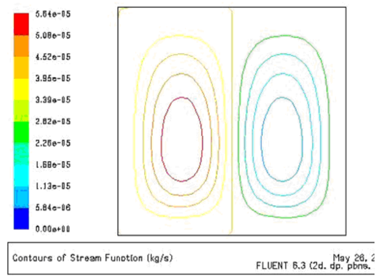


Fig 4 (h) Contours of Stream Function when Ra = 10^4 for case 1st

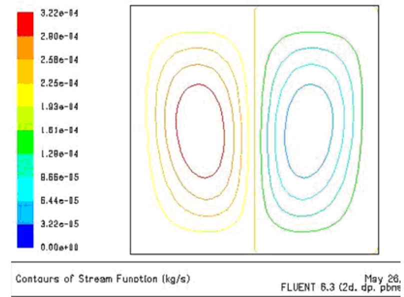


Fig 4(i) Contours of Stream Function when Ra= 10^5 for case 1st

Fig. 4 (a), (b) and (c) variation of Local Nu with varying Ra, **fig 4 (d), (e) and (f)** Temperature contours, **Fig 4 (g), (h) and (i)** Stream function values fordifferent values of Ra.

Similar analysis is carried out for the remaining cases for different heat source locations with Ra values (10^4 and 10^5) while Pr = 0.71 Results are tabulated in the Table 1.

Table 1 Summarized of Results for all six cases

Case	Ra No	Max Temp(contour) (K)	Max Stream function (Kg/s)	Max Nusselt No
Case I	10^3	388 K	3.90E-06	35.452
	10^4	386.8536 K	5.67E-05	37.9295
	10^5	358 K	0.00031338	45.5734
Case II	10^3	441.156	3.01E-06	17.28
	10^4	439.26	3.95E-05	18.8911
	10^5	395	2.57E-04	34.5177
Case III	10^3	494.5859	1.73E-06	22.1662
	10^4	493.5846	2.025883-5	22.6065
	10^5	447.3779	0.000177553	33.2992
Case IV	10^3	547.4462	9.69E-07	33.0485
	10^4	547.1213	9.62E-06	33.1515
	10^5	514.2134	0.000109647	39.2608
Case V	10^3	365.0072	9.66E-07	65.0914
	10^4	365.0071	8.80E-06	65.3258
	10^5	357.4418	0.000158296	69.78
Case VI	10^3	362.9018	2.28E-07	83.8274
	10^4	362.9072	1.72E-06	83.848
	10^5	362.8526	2.61E-05	84.1389

As seen from Table 1, of all the cases minimum temperature in the Square cavity is in the case I and V for Ra = 10^5. These are the preferred locations for the heat source. From Table 1, in case V and VI, the heat sources were split into two halves and this has resulted in reduction in maximum temperature attained in the cavity as compared to the rest of the cases. This indicates that by splitting the heat source, one can improve the heat transfer rate; hence it is preferable to always split the heat source in the cavity. Also from the above observation it is seen that as Ra increases Nusselt number and mass flow rate also increased, promoting heat transfer by convection.

V. CONCLUSION

The effects of various locations of heat source at varied Rayleigh number and constant Prandtl number for the constant heat flux at the bottom wall, for cases i) for the entire length ii) for half iii) for quarter iv) for $1/8^{th}$ of the total length with appropriate heat source strength was investigated. Also, splitting the heat source into two parts over the length of L/4, L/8 on each side of the bottom wall was investigated. The top wall was assumed adiabatic and side walls were assumed to be at constant temperature. The following conclusions have been drawn. The study showed that the flow pattern and heat transfer mechanism were significantly affected by the Rayleigh number, location of heat source and length of the heat source.

1. It was observed that for $Ra \leq 10^4$ conduction heat transfer was dominant for uniform heating of the entire length of the bottom wall.
2. The mass flow rate of the medium was proportional to the length of the bottom wall
3. The contours of stream function and isotherm were observed to be symmetric about the vertical axis for all the cases provided the
4. Nusselt Number increases monotonically with increase in Ra suggesting the dominance of convection mode of heat transfer.
5. Local Nusselt number for the case of splitting the heat source into two parts at the bottom wall was more than that of cases where sources located at single place for all Rayleigh numbers.
6. Rayleigh number influences mass flow rate which in turn influences heat transfer.
7. By splitting the heat source, the rate of heat transfer increases.

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