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Computational Heat Transfer and Entropy Generation Analysis of Natural Convection in a Trapezoidal Cavity

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Abstract
A numerical study is carried out to investigate natural convective heat transfer process in a water-filled trapezoidal cavity heated from below using finite volume method in solving the governing equations. The study examined the effects of Rayleigh number (Ra = 10^1-10^6), heat source length (L_s = 0.1-0.9), angle of inclination (ϕ = 0º-75º) on the Nusselt number (Nu) and entropy generation of the trapezoidal cavity. The results of the investigation indicate that Nu increases when the source length increases, and the angle of inclination decreases. Also, Nu also increases with Ra when the source length and inclination angle are fixed. The total entropy is dominated by thermal entropy

generation ($S_{thermal}$) and is seen to increase with source length. The results presented in isotherms and streamlines indicate that viscous and buoyancy forces are significant in the trapezoidal cavity.
Keywords: Finite volume method. Entropy generation. Natural convection. Rayleigh's number. Trapezoidal cavity. Nusselt number.

Nomenclature

Symbols	Description
g	gravitational acceleration, m/s^2
H	Dimensionless enclosure height
k	Thermal conductivity (W/mK)
L	Dimensionless length
L_s	Dimensionless heat source length
Nu	Nusselt number
n	Normal
P	Pressure (Pa)
Pr	Prandtl number
$\tilde{q}$	Dimensionless heat transfer rate
$\tilde{q}''$	Dimensionless heat flux,
Ra	Rayleigh number
S	Dimensionless entropy generation rate
T	Temperature (°C)
TC	Trapezoidal cavity
u, v	Velocity component rate (m/s)
U, V	Dimensionless velocity component
x, y	Cartesian coordinates (m)
β	Coefficient of volumetric thermal
∇	Differential operator
ε	Convergence criterion
μ	Viscosity (kg/m.s)
ρ	Density (kg/m^3)
φ	Inclination angle
θ	Dimensionless temperature
Subscripts	
c	Cold
f	Fluid
h	Hot
i	Mesh iteration index

1. Introduction

Natural convection happens to be one of the top areas of concentration on research work and practical applications in thermal science and engineering. Studies have revealed how free convective heat transfer is applicable, thus, connects to every aspect of the society such as the health sector, the oil and gas industry, manufacturing industry, energy, piping, rooms and buildings, cooling and heating (Silva *et al.*, 2012, Silva *et al.*, 2010). Researchers have further carried out numerical studies as well as experimental studies to investigate natural convective heat transfer process in cavities having different configurations, form and structure. These cavities or enclosures are in different configurations and shapes such as square, rectangular, ellipsoidal, triangular, cylindrical, trapezoidal (Roslan *et al.*, 2011; Abdulkadhim *et al.*, 2021; Mustafa *et al.*, 2020; Abdelkarim & Djedid, 2019; Sadeghi *et al.*; 2022; Fetuga *et al.*, 2024). The free convection in trapezoidal cavities (TCs) is centered on two phenomena; enclosures heated from below, which is the focus of the current study and those heated from the side.

A lot of research works referred to the occurrence of natural convection that is present inside an enclosure although new modification such as introduction of heat source, the thermal boundary conditions of the cavities, the orientation of the cavities, cavities inclined at various angles, and so on have also been introduced (Horimek and Nekag, 2020). Moreover, the configuration of the shape and geometry is an important factor that influences the mechanism of the convection (Sivarami Reddy *et al.*, 2021). Shirani and Toghraie, 2021 numerically and experimentally studied free convection in TCs having modified and improved structure with the aim of optimizing and achieving the best results.

Gholizadeh and Nikbakhti (2018) investigated a TC with heat sources at different heating sections in the cavity containing fluids inclined at different angles. It was revealed in their results that the position of the source of heat has a remarkable impact on the flow pattern and also the way heat is transferred in the cavity. Gowda *et al.* (2019) examined the heat transfer analysis on a side heated of TC opening and considering an angle orientation. The results found that *Nu* increases as *Ra* increases but reduced with the angle of inclination (ϕ). Kuyper and Hoogendoorn (1995) studied the laminar free convective heat transfer flow in trapezoidal enclosure with various ϕ ranging from $\phi = 0^o - 45^o$ and at $Ra = 10^6$. It was indicated in their findings that the fluid flow and the heat transfer at the edges of the enclosures depend energetically on the intersection angle between the adiabatic walls and the isothermal walls. Venkatadri *et al.* (2019) simulated free convective heat transfer in a 2-D TC. The results indicated that for a top corner of the cavity hot wall, *Nu* reduces gradually from bottom to the top of the hot wall and enhances as *Ra* is increased. Selimefendigil *et al.* (2016) worked on the convective heat transfer process in a Ferro fluid TC having magnetic dipole source. The results found that *Nu* decreases while the value of the magnetic dipole strength increases. It was also reported that increasing *Ra* led to a higher heat transfer rate. Also, Brito *et al.* (2024) and Ojuro *et al*. (2024) numerically analyzed the importance of thermal contact resistance in the thermal management of cooling systems.

Acharya and Moukalled (2000) numerically investigated the natural convention in TCs having baffles with summerlike and winterlike conditions thrust to the upper inclined surfaces. The summerlike conditions resulted in a heat transfer rate decrease as the baffles height and baffles' location increased. However, for winterlike conditions, the heat transfer rate is enhanced with an increased distance from the location of the baffles. Furthermore, Mamun *et al.* (2011) studied the natural convection in a lid driven cavity with flow within, and a tilted enclosure. The focus was on an enclosure driven by lid. It was indicated in their results that for an inclined TC with different aspect ratio of heat source, the elevation in the heat source length augments *Nu*. Also *Nu* decreases as the rotational angle increases. Reddy *et al.* (2021) investigated porous TCs that are inclined consisting of a cross flow which is due to free convection. It was concluded in their study that the trapezoidal cavity inclined affects the intensity of the fluid that circulates in the TC and thus reduced the temperature gradient. Mebarek-Oudina *et al.* (2021) analyzed entropy generation and energy flow that is due to free convection in a TC having both isothermal and non-isothermal heated wall bottom and is at an inclined angle. It was revealed in their findings that a rise in *Nu* occurs as a result of the reduction in the heat source size. In addition, it was also reported that the inclined angle resulted in a low heat transfer coefficient and a low entropy generation. Boussaid *et al.* (2003)

together with Dong and Ebadian (1994) studied heat transfer analysis in trapezoidal cavity focusing on the laminar flow regime, while Van Der Eyden *et al.* (1998) considered heat transfer investigation relating to turbulence flow. Natarajan *et al.* (2007) carried out a numerical study on free convective heat transfer flow in a TC having vertical and an insulated top which is heated uniformly from below. From their results, it was reported that the local *Nu* showed a revert of heat flow at the sides of the walls. In addition, the total rate of heat transfer at the wall bottom is greater for both cases that is when it is heated at the wall left and cooled at the wall right. Kumar (2004) studied the convective heat transfer in a box-type solar cooker having a TC configuration, he compared the result of the study with rectangular geometry and concluded that the trapezoidal cavity had a higher heat transfer coefficient. In the laminar flow regime, Mustafa and Ghani (2012) conducted a numerical analysis of natural convection when they partially heated the base and side of trapezoidal enclosure at varying heat source lengths 0.2-0.8 for $Ra = 10^3$-10^5 and air as cooling fluid. This result of this study revealed that *Nu* increased with source length.

This present investigation is an extension of Mustafa and Ghani (2012) by expanding inclination angle from $\varphi = 0° - 75°$, source lengths $Ls = 0.1 - 0.9$ and $Ra = 10^1$ - 10^6. Also, the working fluid is water for better and effect cooling. For $\phi = 0^o$ is a special case when the trapezoidal cavity becomes a square cavity. The convective heat transfer and entropy generation on thermal performance in a water filled trapezoidal cavity at different *Ra* and φ was also investigated in this study.

2. Numerical methodology

2.1 Physical Problem

The physical problem that is currently under study is illustrated in Figure 1 as suggested by of Mustafa and Ghani (2012). The trapezoidal cavity has walls on the left and right sides which have a uniform and constant cold temperature Tc. The bottom wall consists of a source length Lswhich is located midway of the cavity. It is heated below with a uniform and constant temperature known as Th. The rest of the bottom wall asides the source length is adiabatic as well as the top wall which is adiabatic. The trapezoidal cavity has a base width equal to the height ($L = H$), with an angle inclined φat the left and right walls. Water is used as the working fluid and the properties of the fluid are in dimensionless quantities. The density effect with the Boussinesq approximation due to the natural convection is applied. Also, the fluid is assuming to be steady, laminar and incompressible and two-dimensional flow.

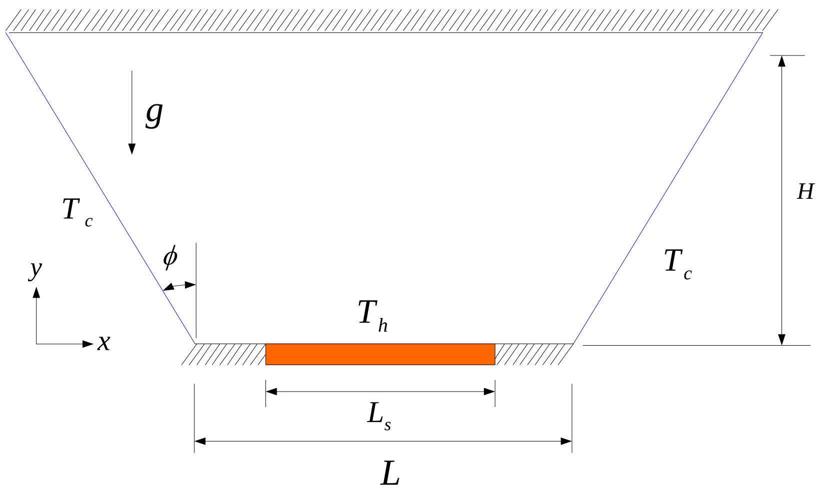

Figure 1 - Schematic of the trapezoidal cavity with heat source

2.2 Governing Equations

The problem considered is centered on the continuity, and momentum and energy equations. Taking cognizance of the assumptions mentioned earlier, the 2-D equations can be expressed as discussed by Teamah and Shehata (2016) as:

$$\frac{du}{dx}+\frac{dv}{dy}=0 \tag{1}$$

$$\rho\left(u\frac{du}{dx}+v\frac{du}{dy}\right)=-\frac{dp}{dx}+\mu\nabla^2 u \tag{2}$$

$$\rho\left(u\frac{du}{dx}+v\frac{du}{dy}\right)=-\frac{dp}{dy}+\mu\nabla^2 v+\rho g\beta(T-T_\infty) \tag{3}$$

$$U\frac{dT}{dx}+V\frac{dT}{dy}=\alpha\nabla^2 T \tag{4}$$

The nondimensionalized form of the variables considered in this are given as:

$$X=\frac{x}{H};\ Y=\frac{y}{H};\ U=\frac{uH}{\alpha};\ V=\frac{vH}{\alpha};\ \theta=\frac{T-T_c}{T_h-T_c};\ P=\frac{pH^2}{\rho\alpha^2};\ Pr=\frac{v}{\alpha};\ Ra=\frac{g\beta(T_h-T_C)H^3}{v\alpha} \tag{5}$$

Employing all these dimensionless variables from Equation (5), the dimensional governing equations (1) to (4) reduced to non-dimensional forms as:

$$\frac{dU}{dX}+\frac{dV}{dY}=0 \tag{6}$$

$$\left(U\frac{dU}{dX}+V\frac{dV}{dY}\right)=-\frac{dP}{dX}+Pr\,\nabla^2\,U \tag{7}$$

$$\left(U\frac{dU}{dX}+V\frac{dV}{dY}\right)=-\frac{dP}{dY}+Pr\,\nabla^2\,V+Ra\,Pr\,\theta \tag{8}$$

$$\left(U\frac{d\theta}{dX}+V\frac{d\theta}{dY}\right)=\nabla^2\theta \tag{9}$$

2.3 The Boundary Conditions

The dimensionless boundary conditions applied to the trapezoidal cavity are summarized:

- Wall bottom with L_s, at T_h, $U=V=0;\ \theta=\theta_h=1$
- Wall inclined T_C, $U=V;\ \theta=\theta_c=0$
- other walls are insulated, $q''=U=V=\frac{dT}{dX}=0$

2.4 Heat transfer performance

The average Nusselt numbers as a measure of performance is expressed as:

$$Nu_{avg}=\frac{\tilde{q}}{(\theta_h-\theta_c)} \tag{10}$$

2.5 Dimensionless Entropy Generation

The $S_{thermal}$, and $S_{friction}$ are expressed in Equations (11) and (12), respectively.

$$S_{thermal} = k\left(\frac{\Delta T}{LT_o}\right)^2\left[\left(\frac{d\theta}{dX}\right)^2 + \left(\frac{d\theta}{dY}\right)^2\right] \quad (11)$$

$$S_{friction} = \frac{\mu}{T_o}\left(\frac{\alpha}{L^2}\right)^2\left[2\left[\left(\frac{dU}{dX}\right)^2 + \left(\frac{dV}{dY}\right)^2\right] + \left(\frac{\partial U}{\partial Y} + \frac{\partial V}{\partial X}\right)^2\right] \quad (12)$$

Therefore, the *dimensionless* total entropy generation rate, S_{Total} is expressed as:

$$S_{Total} = S_{thermal} + S_{friction} \quad (13)$$

2.7 Solution Procedure and Numerical Accuracy

The numerical solution of the dimensionless governing equations and the boundary conditions above was attained using a commercial package ANSYS FLUENT. The information about the numerical procedures is provided by Patankar (2018). The semi-implicit method for pressure linked equations (SIMPLE) scheme algorithm was set for the pressure-velocity coupling. A second-order upwind scheme was employed to discretize the combined convection and diffusion terms in governing equations.

The solution is presumed to converge when the modified equations residuals less than 10^{-10}. A grid dependency test consisting of several mesh refinements was carried out during this course to determine and give a better accuracy of the numerical result and a more accurate value to use for the entire project simulation.

Table 1 shows the grid independence test performed to guarantee accuracy in the simulation results. The convergence of the solution is met when Equation (14) criteria is satisfied. Therefore, for this study, node size 78936 is used for all computations since any additional increase in the number of nodes has an insignificant effect on the numerical result. The computational mesh for the 2-D trapezoidal cavity used for study is as seen in Figure 2.

$$\varepsilon = \frac{|(\tilde{q})i - (\tilde{q})i - 1|}{|(\tilde{q})i|} \leq 10^{-4} \quad (14)$$

Table 1 - Grid independence test for water at $Ls = 0.8$, angle of inclination at$30°$.

Nodes	$\tilde{q}$	ε
72,322	2.69682	-
75,499	2.69703	0.00007679
78,936	2.69712	0.00003393
82,397	2.69728	0.00006136

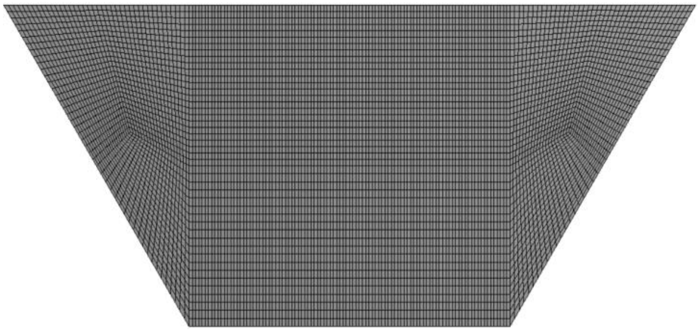

Figure 2 - Computational grid/mesh diagram of a trapezoidal cavity

2.8 Code Validation

Code validation is a crucial part of a numerical study, hence the need for comparison is highly required for the purpose of the validation. This numerical computation of the free convective heat transfer for the TC was validated with the work of Mustafa and Ghani (2012). In the validation, air with a *Pr* = 0.76 was used as the working. This is done to ensure that the numerical computation agrees with literature results presented. The TC has been validated for an inclined angle $\varphi = 30°$at the left and right walls. Figure 3 shows the isotherm and streamlines comparison of the numerical study while Figure 4 shows the graphical validation of the present study with that of experimental work by Mustafa and Ghani (2012). The comparison and validation show similar patterns and the results agreed with a percentage error of 2%.

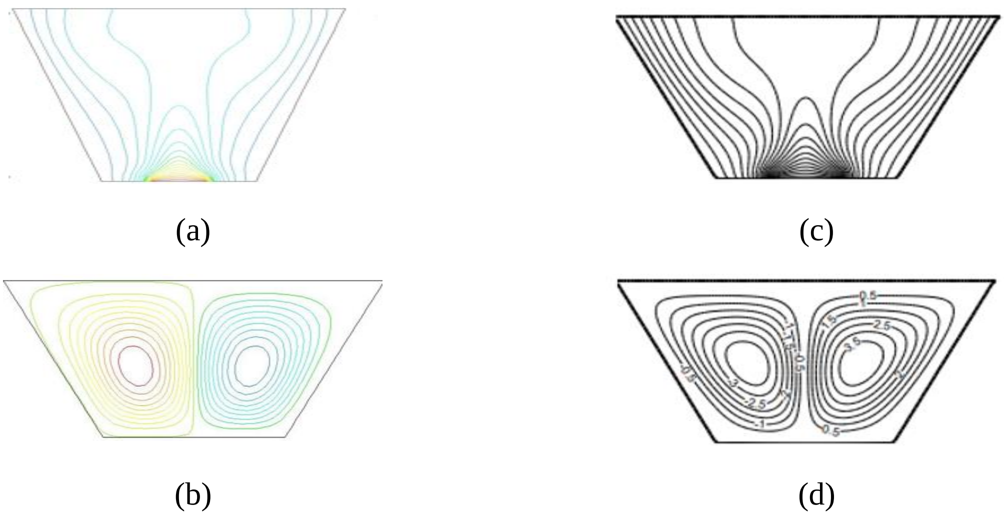

Figure 3 - Comparison of the present study (a) isotherm and (b) streamlines with the (c) isotherm and (d) streamlines in the work of Mustafa and Ghani (2012) for L_s = 0.8 and *Ra* = 10^4.

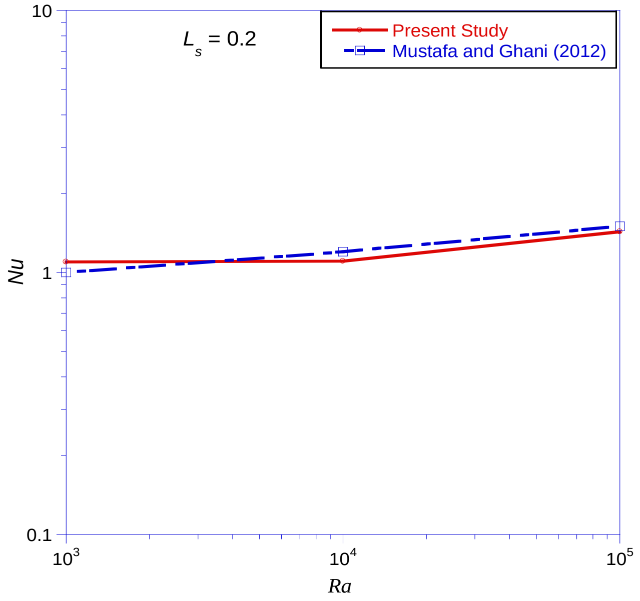

Figure 4 - The comparison of *Nu* as a function *Ra* in the present study with that of Mustafa and Ghani (2012) for L_s = 0.2

3. Results and discussion

This subsection describes the numerical outputs for the isotherms, streamlines for three parameters which are *Ra* = 10^1-10^6, dimensionless source lengths, L_s = 0.1-0.9 and angle of inclination, ϕ = 0º-75º. The dimensionless height and length of the cavity equal are fixed to unity (L= H =1).

3.1 Ra versus Nu

Figure 5 reveals that for each increasing heat source length L_s = 0.1, 0.6 and 0.9, with an increasing *Ra* there is a significant increase in *Nu*. Although across all three source lengths, when the *Ra* increases from 10^3 to 10^4, there is a small change in *Nu* which indicates a small change in heat transfer rate. The influence of *Ra* (when *Ra* = 10^4-10^6) on *Nu* is seen in this Figure 5 as it indicates an increase in heat transfer. This implies that there is a significant amount of high heat transfer rate from *Ra* = 10^4-10^6 at source lengths 0.1 and 0.9 although the effect of *Ra* = 10^3-10^5 on *Nu* at L_s = 0.6 is not as significant as compared to L_s of 0.1 and 0.9.

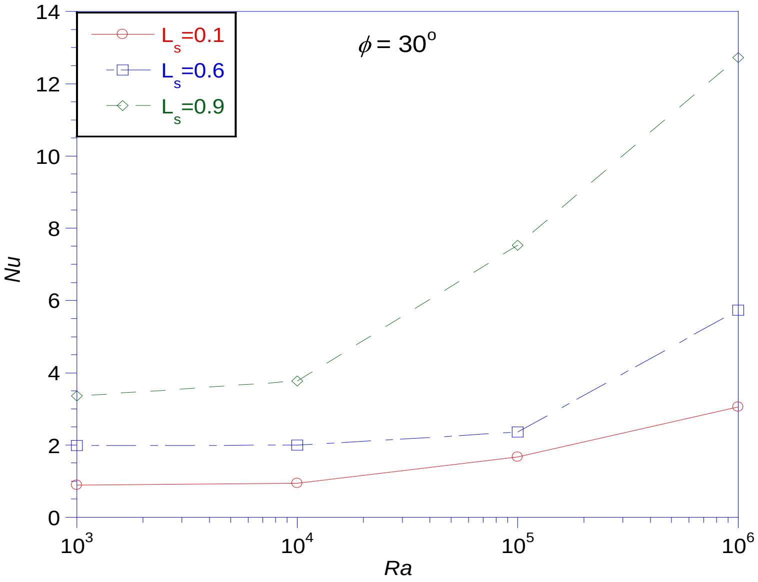

Figure 5 - *Nu* as a function of *Ra* at different L_s for fixed ϕ = 30°

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3.2 The Influence of Heat Source Lengths (L_s) on Nu

Figure 6 shows results of *Nu* as a function of L_s for *Ra* = 10^3-10^6 and fixed ϕ = 30°. When the L_s was varied, the result shows that *Nu* increases with the L_s. Furthermore, increasing the *Ra* from 10^3 to 10^4 did not cause any significant change in *Nu* results. A marginal increase in *Nu* was obtained when the *Ra* was increased 10^5. For instance, at the L_s of 0.3, *Nu* increased by 14.23 % when the *Ra* changed from 10^4 to 10^5. A further increase in the L_s to 0.7 for the Ra 10^4 to 10^5 caused *Nu* to increase by 15.86 %. Whereas *Nu* increased substantially when the *Ra* was increased to 10^6. At the L_s of 0.4 *Ra* was increased from 10^5 to 10^6, *Nu* was increased by 144 %. Further increase in L_s to 0.8 for the *Ra* = 10^5 to 10^6, shows that *Nu* increased by 103 % indicating a strong heat transfer enhancement. In conclusion, for an increasing source length and fixed *Ra*, the heat transfer rate increases.

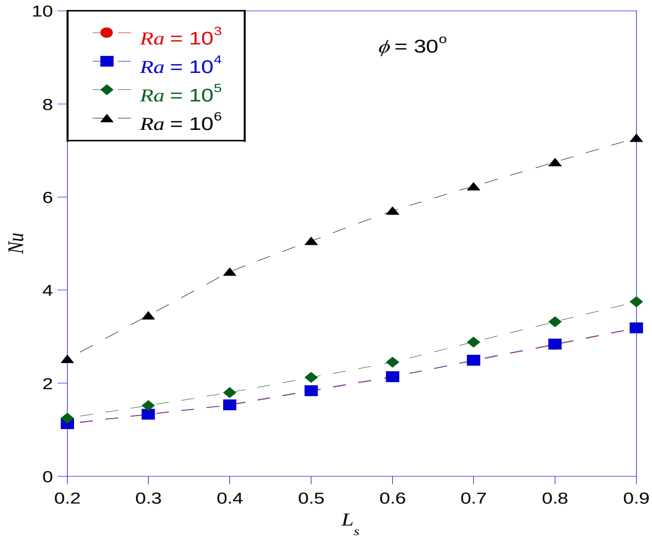


Figure 6 - *Nu* as a function of L_s for *Ra* = 10^3-10^6 at fixed ϕ = 30°

3.3 The Influence of inclination angle (ϕ) and *Ra on Nusselt Number*

Figure 7 shows the influence of ϕ and *Ra* on *Nu*. Figure 7(a) depicts a special case for ϕ = 0º when the trapezoidal cavity becomes a square cavity and Figure 7(b) when ϕ = 75º. The results reveal that elevating the ϕ leads to a decrease in *Nu* across all *Ra*. In Figure 7(a), *Nu* increases with both the *Ra* and L_s increase at the inclination angle of 0º. Also, Figure 7(a) explains that from source lengths L_s = 0.1-0.9, increasing the *Ra* from 10^1-10^4 did not in general increase *Nu* for each L_s. It can be concluded that *Nu* is constant for *Ra* range from 10^1-10^4 but a sharp increment happens across all source length when *Ra* increases from 10^4-10^6. Similar result was obtained in Figure 7(b). *Nu* increases in the cavity system, when ϕ = 75º. It is also observed that *Nu* for each L_s is constant across the *Ra* = 10^1-10^5 but gives a sharp increment from 10^5-10^6. Meanwhile, the source lengths L_s = 0.2-0.8 reveal an irregularity in the distribution of the heat transfer change in the TC at the *Ra* = 10^4. This observation is that Nu is seen to be constant from *Ra* = 10^1-10^3 but increases between *Ra* = 10^3-10^4. *Nu* further decreases from *Ra* = 10^4-10^5 and then finally gives a sharp heat addition from *Ra* = 10^5-10^6. However, when Figure 7(a) is compared with 7(b), the results disclose that elevating the ϕ leads to *Nu* decreases across all *Ra*.

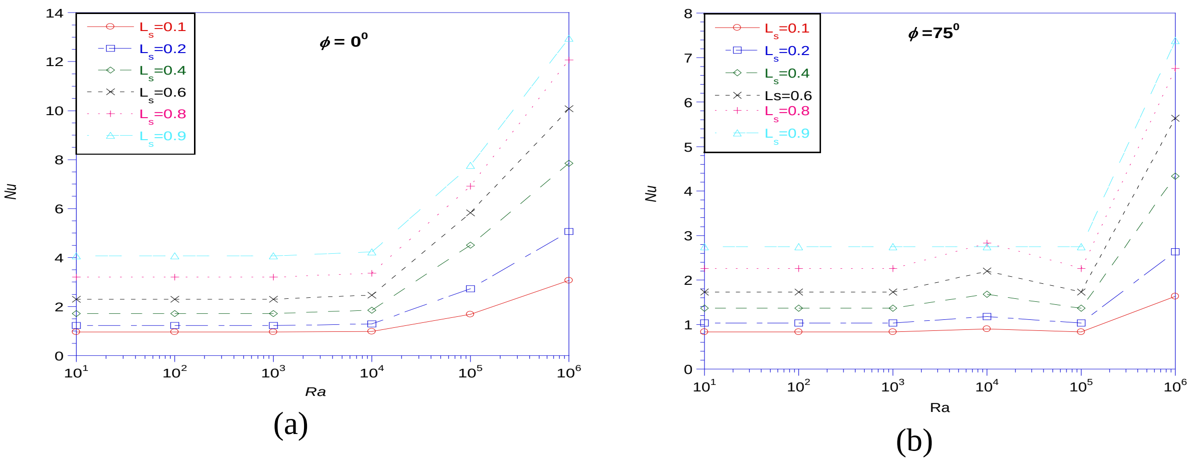


Figure 7 - *Ra* versus *Nu* at different L_s and fixed (a) ϕ = 0º and (b) ϕ = 75º

Figure 8 shows that at a constant heat source length 0.8, considering for *Ra* = 10^1-10^3, there is a reduction in the values of *Nu* across all angles of inclination. At *Ra* = 10^4, there is a constant decrease in *Nu* but reasonable and modest increase in heat transfer takes place from 45º angle of inclination to 60º. The *Ra* = 10^5 shows a sequential decrease in the value of *Nu* when the ϕ is varied from 15º to 45º and then gives a steep increase from 45º to 60º. At *Ra* = 10^6, *Nu* reduces linearly when ϕ increases from 15º to 45º but increases sharply from 45º to 60º.

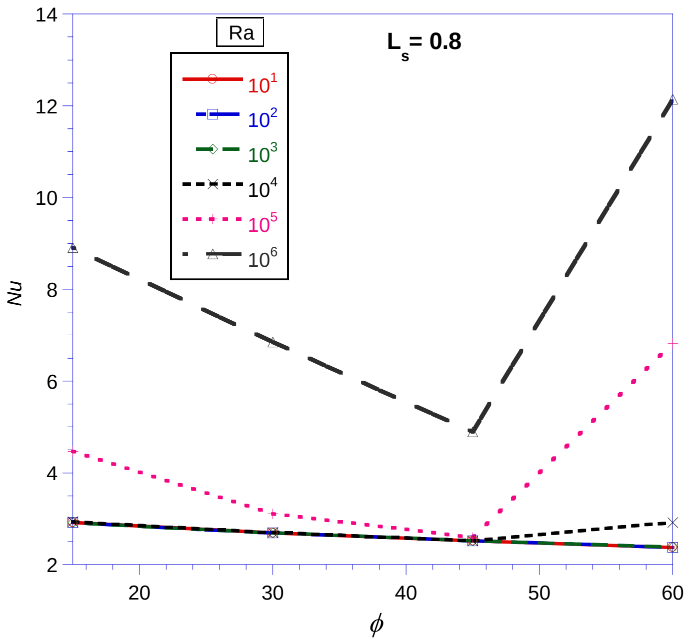


Figure 8 - Effect of ϕ on *Nu* at different *Ra* fixed L_s = 0.8.

3.4 The influence of inclination angles on dimensionless entropy generation

Figure 9 shows the graphical results for the $S_{thermal}$, $S_{friction}$ and S_{Total}. Figure 9(a) shows the $S_{thermal}$, $S_{friction}$ and S_{Total} as a function of ϕ at fixed $Ra = 10^3$ and $L_s = 0.9$. $S_{thermal}$ and S_{Total} reduces as the ϕ increases, while the $S_{friction}$ increase with ϕ. The dominance of the $S_{thermal}$ still reveals that the S_{Total} is dominated by heat transfer and not by pressure drop and this as a result of the influence of the ϕ. Figure 9(b) shows that at fixed ϕ = 15º and $L_s = 0.1$, the $S_{thermal}$, $S_{friction}$ and S_{Total} increases as *Ra* increases.

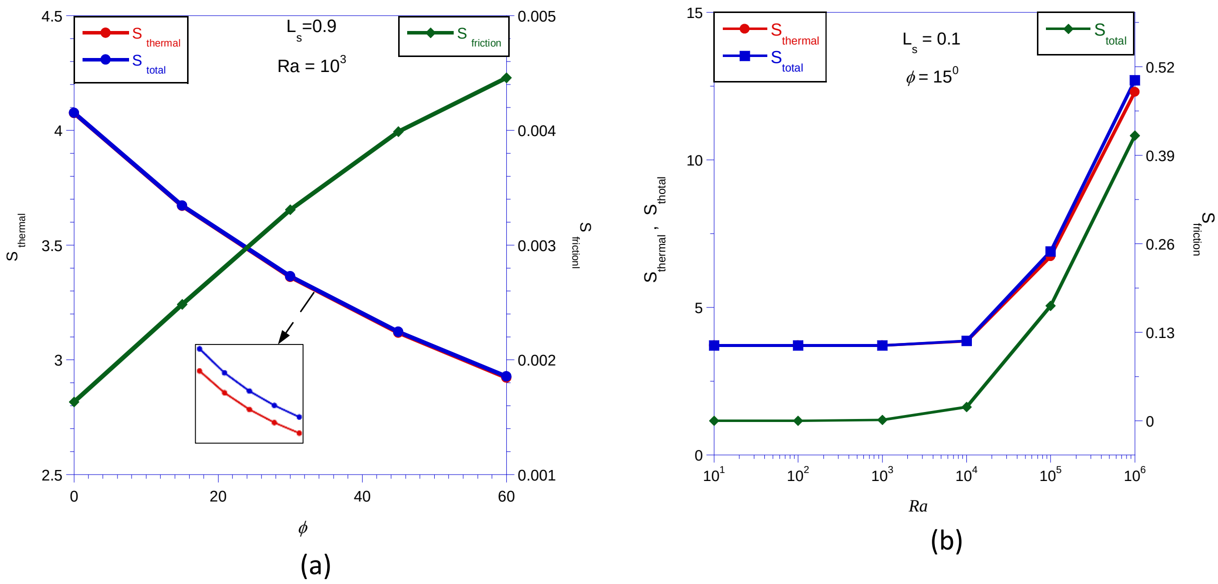
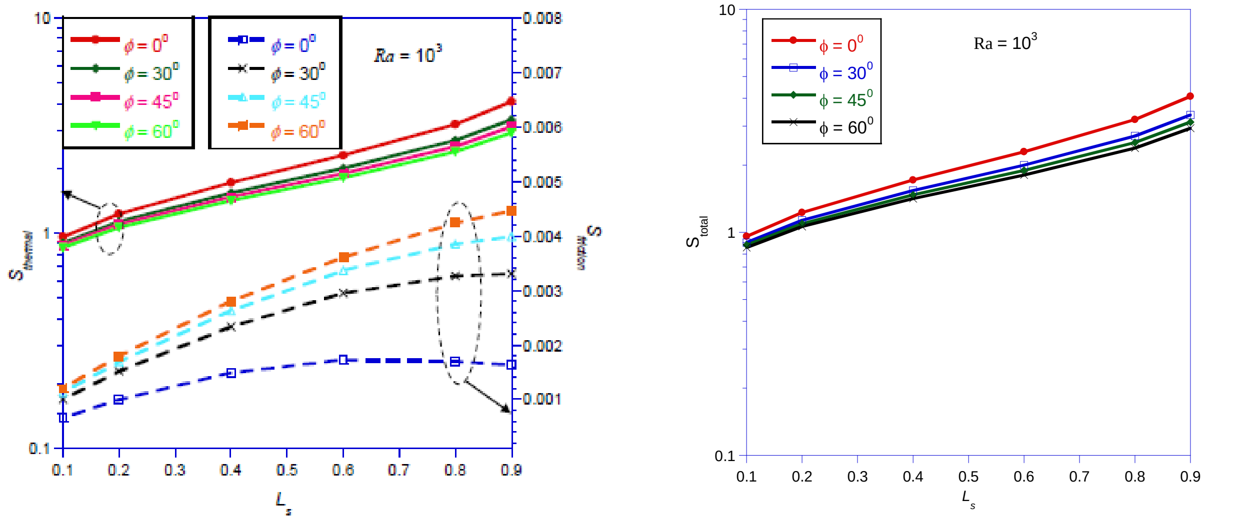


Figure 9 - $S_{thermal}$, $S_{friction}$ and S_{total} vs (a) ϕ at L_s = 0.9 and *Ra* = 10^3 and (b) *Ra* at L_s = 0.1 and ϕ= 15º

Figure 10 shows the graphical results for the $S_{thermal}$, $S_{friction}$ and S_{Total} as functions of L_s for varying ϕ and fixed Ra = 10^3. In Figure 10 (a) the $S_{thermal}$ and $S_{friction}$ are seen to increase with L_s whereas The $S_{thermal}$ reduces while $S_{friction}$ increases as the ϕ increases. Similarly, in Figure 10 (b), the S_{Total} increases with L_s and decreases as ϕ increases. This could be attributed to the minor losses introduced by the increase in the angles. The entropy generated in the trapezoidal cavity is due to the thermal or heat transfer irreversibilites, which makes conduction dominant. This is because the magnitude of $S_{friction}$ is lower compared with the $S_{thermal}$.



(a) (b)

Figure 10 - Effect of L_s on (a) $S_{thermal}$ and $S_{friction}$ and (b) S_{total} at different ϕ *and fixed Ra* = 10^3

4. Conclusion

Natural convection heat transfer with water inside a TC was studied numerically. The commercial CFD (ANSYS FLUENT) code based on finite-volume method was employed to determine the influence of L_s, ϕ and Ra on the heat transfer (Nu) and entropy generation rates in the TC. The φvaried from $\varphi = 0° - 75°$, $Ra = 10^1 - 10^6$ and $Ls = 0.1 - 0.9$. The results and effect of these input parameters on the natural convective flow are clearly presented. From the results achieved, it is seen that for a special case at inclined at $\varphi = 0°$, the trapezoidal cavity is configured as a square cavity that enhance performance. Increase in the L_s, implies that convective heat transfer is improved in the trapezoidal cavity as the L_s is enhanced. Also, as φ increases, Nu decreases across all Ra, which implies that increasing the φ would have influence in the cavity by reducing the heat transfer as a result of its dependency on the φ. The entropy generation is said to be dominated by heat transfer and not necessarily by the friction induced by the inclination angles. The $S_{thermal}$ increases with the heat source length and reduces with the φ.

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