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L-SHAPED ENCLOSURES TO IMPROVE HEAT TRANSFER WITH INTEGRATED NATURAL CONVECTION-FLUID-STRUCTURE INTERACTION: INCORPORATING A FLEXIBLE FIN AND AN ELASTIC WALL

IZBOLJŠAVA PRENOSA TOPLOTE V OHIŠJU L OBLIKE Z VKLJUČITVIJO NARAVNE KONVEKCIJE – STRUKTURNA INTERAKCIJA TEKOČINE: VGRADNJA GIBLJIVE LOPUTE IN ELASTIČNE STENE

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A numerical study was conducted to investigate the flow and heat transfer characteristics inside an L-shaped enclosure with a flexible fin and elastic wall, using the finite element method. A fluid-structure interaction model was applied to capture the interaction between the fluid and the solid structure. Due to the periodically pulsating temperature on the warm walls, a pulsating flow was induced by the thermal boundary condition, and natural convection contributed to the oscillation of the flexible fin. The results show that the fluid flow within the enclosure is significantly affected by the periodic oscillations of both the flexible fin and the elastic wall.

Keywords: fluid-structure interaction, elastic wall, enclosure, pulsating temperature, flexible fin

Avtorji v članku opisujejo numerično študijo, izdelano s pomočjo metode končnih elementov, s katero so analizirali toplotni tok in lastnosti prenosa toplote v ohišju z obliko črke L z vgrajeno gibljivo loputo in elastično steno. Avtorji so uporabili model Strukturne Interakcije Tekočine (angl.: Fluid-Structure Interaction) zato, da so lahko ugotovili interakcijo med tekočino in ohišjem z izbrano geometrijo. Simulirali so periodično pulziranje temperature na ogrevanih stenah s pulziranjem termičnih robnih pogojev. To je vzbudilo pulziranje toplotnega toka in vzbudilo nastanek naravne konvekcije. Ta pa je posledično povzročila osciliranje gibljive lopute. V članku predstavljeni rezultati so pokazali, da tok je tekočine v ohišju v celoti povzročen s periodično oscilacijo obeh; to je: gibljive lopute in elastične stene. Rezultati študije so torej potrdili predpostavko, da je tok tekočine v ohišju popovzročen s periodično oscilacijo gibljive lopute in elastične stene.

Keywords: strukturna interakcija tekočine, elastična stena, ohišje, pulziranje temperature, gibljiva loputa

1 INTRODUCTION

Heat transfer is a crucial aspect of many engineering applications, such as electronics cooling, energy conversion, and thermal management systems. Enhancing heat transfer rates can lead to improved system performance, which is of great interest to engineers and researchers. Natural convection is a heat transfer mechanism that occurs due to the density differences caused by temperature gradients in fluids.[1] It is an important mechanism for heat transfer in many engineering applications, especially in enclosed spaces where forced convection is not feasible or practical. Fluid-structure interaction (FSI) is another important aspect of engineering systems that involves the interaction between fluid flows and solid structures.[2] FSI is particularly important in systems where deformation of the solid structure can have a significant impact on the fluid flow field.[3] Enclosed cavities play a significant role in various engineering application including thermal management of solar panels, cooling of electronic components, thermal insulation, and fuel cells.[4–6]

Currently, there is a growing interest in studying cavities that contain internal obstacles of complex geometries, such as cylinders or irregular shapes. Besides, few studies have investigated the influence of fluid-structure interaction on the rate of heat transfer and flow domain inside enclosures with natural convection.[8] However, the intricate physics involved and the time consuming calculations have made the numerical analysis of these problems challenging.[9] Ghalambaz et al. conducted a numerical investigation of natural convection flow through a flexible baffle in an L-shaped enclosure.[13] Their study employed the arbitrary Lagrangian-Eulerian moving mesh framework and finite element method to solve the governing equations and boundary conditions. They investigated the influence of dimensionless parameters such as the Rayleigh number and elasticity modulus on

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flow and heat transfer. The findings revealed that a stiffer baffle tends to resist the fluid flow, while a larger size of the baffle, at higher Rayleigh numbers, increases resistance to and stress on the baffle. Some fluid-solid interaction investigations examined how the flexibility of a thin plate affects natural convection within an enclosure.[10,14] It was assumed that the plate had a higher temperature, whereas the side walls were colder, while the top and bottom walls were thermally insulated. Various parameters such as the plate height, length, elasticity modulus, Rayleigh number, and Prandtl number were varied, and their effects were examined.[11,12] These studies found that increased plate flexibility led to a decreased Nusselt number and flow strength. Besides, the plate flexibility was found to significantly impact the flow and heat transfer, depending on the flow regime and other parameters. Jamesahar et al. conducted a study to investigate the impact of two oscillating fins on heat transfer and flow characteristics within a square enclosure filled with a nanofluid.[15] In their work, both fins were attached to the hot walls and oscillated at the same frequencies and amplitudes. The results indicated that the oscillation of the fins increased the heat transfer rate. Khanafer et al. conducted a numerical investigation on the characteristics of natural convective heat transfer in a cavity that was filled with a porous medium, focusing on its relevance to the solar collector.[16] Their study examined how a porous medium, obstructed object, and flexible wall affected the heat transfer performance within the thermal system. Different variables including the Rayleigh number, location of the obstructed cylinder, effective thermal conductivity, and wall flexibility were taken into account. The findings indicated that employing a flexible wall alongside a circular cylinder did not significantly improve natural convective heat transfer as compared to a rigid wall. Moreover, placing obstructed objects inside enclosures was found to be more effective in enhancing the heat transfer. Saleh et al. conducted a study on transient free convection in an open cavity with two elastic thin fins, formulating the governing equations for continuity, momentum, and heat transfer in the fluid, as well as for the structural domains using the Arbitrary Lagrangian-Eulerian method.[17] Parameters including the direction, amplitude, and length of the oscillating fins, as well as flexibility values and aperture sizes, were considered. Their results indicated that the maximum heat transfer enhancement occurred at optimal parameter combinations between the aperture size and the oscillating direction of the fins. Sabbar et al. conducted a study on unsteady mixed convection in a cavity-channel assembly with deformable walls.[18] A discrete heat source was positioned at the cavity's bottom, while the other walls were thermally insulated. The upper surface of the cavity was subjected to the channel flow. The impacts of elasticity modulus, buoyancy-to-viscous-forces ratio, and inertia-to-viscous-forces ratio were examined across various ranges of Cauchy, Richardson, and Reynolds numbers. The study found that the utilization of elastic walls resulted in a 17-% enhancement of the heat transfer rate compared to rigid walls. The influence of the elastic wall was particularly pronounced when $R_i = 10$.

Shahabadi et al. conducted a study on the flow and heat transfer of a power-law non-Newtonian fluid in a cavity.[19] An elastic fin was mounted onto the hot wall, and the fluid circulated inside the cavity due to buoyancy forces. The study involved fluid-structure interaction, coupling the non-Newtonian flow with the deformation of the fin. Their results showed that the deflection of the fin was greater when subjected to a dilatant non-Newtonian fluid in comparison to pseudoplastic and Newtonian fluids. The non-Newtonian effects had minimal impact on the heat transfer rate. Additionally, stiff fins experienced higher maximum stress compared to soft fins. Mehryan et al. conducted a study on heat transfer through natural convection in a cavity with a square shape including a thin flexible heater plate inside.[20] The results highlighted the significance of the inclination angle and the position of the fixed point for the hot plate. The plate experienced high stress when mounted horizontally, with the highest stress occurring at an inclination angle of 40° when fixed at the top. In contrast, the lowest stress was observed when the plate was positioned vertically. Akbal et al. conducted a study on the melting process in an annular closed space filled with a phase change material using flexible fins.[21] The research compared cases with different numbers of flexible fins to a case without fins, focusing on phase change dynamics and employing a two-way fluid structure interaction analysis. Their results showed that the use of elastic fins significantly accelerated the melting process as the dynamics of the melting process were influenced by the deflection of the elastic fins, the number of fins, and their elastic modulus. Chattopadhyay et al. studied how the movement of walls in a wavy chamber influenced mixed convection and thermal performance.[22] Different wall movement cases were considered, and it was found that the direction of moving lids and wavy surface undulations affected the fluid flow and heat transfer. Optimal configurations were recommended based on heat transfer rate and entropy generation.

Shahrestani et al. performed a numerical study on the natural convection of a Newtonian fluid within a partitioned circular enclosure with a flexible wall.[23] The governing equations included continuity, momentum, and energy equations for the fluid, as well as energy and elastodynamic equations for the movable plate. Their results revealed that the deformation of the plate directly depended on the forces exerted by the fluid. At low Rayleigh numbers, plate distortion and vortex power in the cavity halves were negligible, with heat transfer primarily occurring through conduction. Increasing the Rayleigh number from 10^4 to 10^7 resulted in more than a 5-fold increase in the average Nusselt number. Alhashash et al. numerically simulated transient free convection in a composite enclosure with a cold flexible plate and a hot rigid plate.[24] Their study incorporated a hyper-elastic flexible plate and a porous layer attached to the rigid plate, filled with water. The fluid flow was governed by the Navier-Stokes equations, while the flow within the porous layer followed the Brinkman-Forchheimer extended Darcy model. Their study examined the development of convective flow through initial, transitional, and stationary states, with variations in the Darcy and Rayleigh numbers impacting the state intervals.

In this research, the combined effect of natural convection and FSI on the heat transfer rates inside L-shaped enclosures is investigated. Specifically, the impacts of flexible fins and elastic walls on heat transfer rates are studied. Flexible fins can enhance heat transfer rates by inducing fluid mixing for heat transfer, and elastic walls can also impact the fluid flow field by inducing deformation, altering the flow patterns. The objective of this study is to provide a comprehensive understanding of the combined effect of natural convection and FSI on the heat transfer rates inside L-shaped enclosures, including the effects of a flexible fin and an elastic wall. The results of this study can contribute to the design of engineering systems where heat transfer enhancement is a crucial factor.

2 PHYSICAL MODEL AND GOVERNING EQUATIONS

2.1 Model description and boundary condition

Schematics of the geometries designed for the present investigation are shown in **Figure 1**. An enclosure with a flexible fin and an elastic wall was considered. The inclusion of a flexible fin introduces an additional dynamic element to the enclosure. The impact of the deformation of the fin under the influence of fluid flow and temperature gradients on the convective heat transfer within the enclosure is examined. The flexibility of the fin enables changes in its geometry, affecting the flow field and heat transfer rates. Similarly, the elastic wall adds another level of complexity to the system. The influence of wall elasticity on the convective heat transfer process is investigated by taking into account the deformations resulting from the fluid flow and thermal effects. The elastic deformation of the wall can alter the flow patterns, turbulence levels, and heat transfer characteristics within the enclosure. The effects of simultaneous use of the flexible fin and elastic wall on the hydrodynamics and heat transfer are investigated.

The problem under investigation involves an L-shaped enclosure. The enclosure has a height and width denoted as L and D, respectively. Both L and D are set to 10 mm. As seen below, the L-shaped enclosure is composed of six walls. The lower horizontal wall and the left vertical wall are considered the hot walls, the

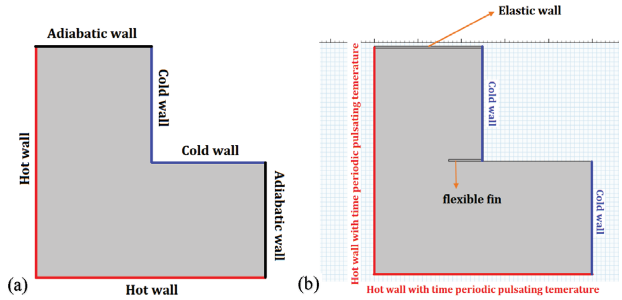

Figure 1: Schematics of the studied problem: a) without an elastic wall and flexible fin; b) with both a flexible fin and an elastic wall

temperature of the hot walls is subjected to a time-periodic pulsating profile, as shown in Equation (1):

$$T\,h(t) = 325\,(1 + 0.15\sin(6\pi t)) \quad (1)$$

The two other walls of the L-shaped enclosure are designated as the cold walls, maintained at a temperature of T_c = 273 K. The remaining walls of the L-shaped enclosure are considered adiabatic.

The Rayleigh number is 1.225×10^6. **Figure 1** shows schematics of the studied problem without an elastic wall and flexible fin, and with both a flexible fin and an elastic wall.

The height of the flexible fins is 3 mm. The effect of an elastic wall is also studied, as seen in **Figure 1**. The cyclic deformations induced by the elastic wall can affect the flow patterns, heat transfer characteristics, and overall thermal behavior within the enclosure. Moreover, different values of the elasticity modulus of the elastic wall are considered. These varying elasticity modulus values allow the examination of the wall's response to different levels of stiffness and its impact on convective heat transfer within the enclosure.

The aim of this study is to investigate the flow and heat transfer characteristics inside an L-shaped enclosure with a flexible fin and an elastic wall. The purpose of the study is to understand the behavior of the fluid flow and heat transfer in such a system, considering the interaction between the fluid and the solid structure. This research is motivated by the following factors: (1) Practical relevance: L-shaped enclosures with flexible fins and elastic walls are encountered in various engineering applications, such as electronic cooling systems, heat exchangers, and microfluidic devices. Understanding the flow and heat transfer behavior in these systems is important for optimizing their performance and enhancing their efficiency. (2) Fluid-structure interaction: The study focuses on the interaction between the fluid and the solid structure. Fluid-structure interaction is commonly observed in many engineering systems, and studying it contributes to improving the design and performance of such systems. (3) Heat transfer enhancement: The investigation aims to explore the effects of a flexible fin and an elastic wall on heat transfer within an enclosure. This information can be valuable for developing heat transfer enhancement techniques in practical applications. (4)

Design optimization: The results of the study, which demonstrate the influence of a flexible fin and elastic wall on fluid flow and heat transfer, can provide insights for optimizing the design of L-shaped enclosures. By understanding the factors that affect the Nusselt number, such as the elastic modulus of the wall, engineers and researchers can design more efficient and effective systems.

The findings of this research hold relevance in various fields, including: (1) Thermal management in electronics: The insights gained from this study can be applied to enhance the thermal management of electronic devices. By optimizing the design of flexible fins and understanding the impact of oscillating forces on heat transfer, the cooling efficiency of electronic components can be improved, leading to better device performance and reliability. (2) Thermal energy storage: The investigation of an L-shaped enclosure with flexible fins and an elastic wall has implications for thermal energy storage systems. (3) Industrial processes: The findings of this study can be applied to various industrial processes that involve heat transfer and fluid flow. For example, in chemical processing or manufacturing, optimizing the design of enclosures with flexible fins and elastic walls can enhance heat transfer, leading to improved process efficiency and productivity.

2.2 Governing equations

The governing equations for the elastic solid domain can be expressed as Equation (2) and (3): [25,30]

$$\partial T_{\mathrm{s}} = a_{\mathrm{s}} \nabla^2 T_{\mathrm{s}} \tag{2}$$

$$P_{\mathrm{s}} \mathrm{d}^2 d_{\mathrm{s}} \nabla \sigma = F_{\mathrm{v}} \tag{3}$$

Density of the solid × acceleration of the solid structure – surface force per unit volume applied to the solid structure = body force per unit volume acting on the solid

Within the given equations, the variables are defined as follows: T_s represents the temperature of the flexible partition, t represents the time, d_s represents the movement vector of the movable partition, α_s represents the thermal diffusion coefficient of the partition, ρ_s represents the density of the flexible partition material, σ represents the stress tensor, and F_v is the force per unit volume derived from the subsequent Equation (4).

$$F_{\mathrm{v}} = m_{\mathrm{s}} g = p\zeta g \tag{4}$$

In this equation, m_s denotes the mass of the partition, ζ represents the volume of the partition, and g signifies the gravitational acceleration. Assuming that the partition functions as an elastic object, the stress tensor can be mathematically represented in the following manner, Equation (5).

$$s = J^{-1} FSF^{\mathrm{T}} \tag{5}$$

J is the determinant of the matrix F, which is defined as $F = I + \nabla \mathrm{d}s$. The variable S represents the partial derivative of the strain energy density function, W_s, with respect to the strain, ε. The subscript T indicates the transpose of the matrix F. The expressions for the strain energy density function, W_s, and the strain ε can be given by Equation (6) and Equation (7):

$$W_{\mathrm{s}} = \frac{1}{2}\mu_1 (J^{-1} I_1 - 3) - \mu_1 \ln(J) + \frac{1}{2}\lambda(\ln(J))^2 \tag{6}$$

$$e = \frac{1}{2}(\nabla d_{\mathrm{s}} + \nabla d_{\mathrm{s}}^{\mathrm{T}} + \nabla d_{\mathrm{s}}^{\mathrm{T}} \nabla d_{\mathrm{s}}) \tag{7}$$

In Equation (6), λ and μ_l are individually known as the Lame's first and second parameters, respectively, and they are calculated by $\mu_l = E/(2(1 + v))$ and $\lambda = vE/[(1 + v)(2v - 1)]$. Additionally, I_1 symbolizes the right Cauchy–Green deformation tensor's first invariant. The following is the format in which the continuity, momentum, and energy equations that control fluid flow are displayed.[25,30,31]

$$\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y} = 0 \tag{8}$$

$$\rho_{\mathrm{f}}\left(\frac{\partial u}{\partial t} + (V - w)\nabla u\right) = -\frac{\partial P}{\partial x}\mu_{\mathrm{f}} \nabla^2 u \tag{9}$$

$$\rho_{\mathrm{f}}\left(\frac{\partial u}{\partial t} + (V - w)\nabla u\right) = -\frac{\partial P}{\partial y}\mu_{\mathrm{f}} \nabla^2 v + \rho_{\mathrm{f}} g \beta_{\mathrm{f}} (T_{\mathrm{f}} - T_{\mathrm{c}}) \tag{10}$$

$$\frac{\partial T_{\mathrm{f}}}{\partial t} + (V - w)\nabla T_{\mathrm{f}} = \alpha_{\mathrm{f}} \nabla^2 T_{\mathrm{f}} \tag{11}$$

In the given equations, Equations (8)–(11), V represents the velocity vector, while u and v denote the horizontal and vertical components of the velocity, respectively. The computational grid velocity is represented by w. The density is denoted by ρ_f, the dynamic viscosity of the fluid by μ_f and the thermal diffusion by α_f.

The local Nusselt numbers over the vertical and horizontal hot walls are defined in Equation (12):

$$Nu_{\mathrm{n,t}} = -\frac{L}{T_{\mathrm{h}} - T_{\mathrm{c}}}\left(\frac{\partial T}{\partial n}\right)_{\text{hot surface}} \tag{12}$$

Here, $\partial T/\partial n$ is the value of the temperature gradient over the hot surface. For the vertical hot wall, $\partial T/\partial n$ is $\partial T/\partial x$ and for the horizontal wall, $\partial T/\partial n$ is $\partial T/\partial y$.

The spatially averaged Nusselt number can be obtained by integrating the local Nusselt number along the hot wall using Equation (13):

$$Nu_{\mathrm{t}} = \frac{1}{L}\int_0^L Nu_{\mathrm{n,t}} \mathrm{d}x \tag{13}$$

The time-averaged Nusselt number can be obtained by integrating the local Nusselt number over a given time period using Equation (14):

$$Nu_{\mathrm{n}} = \frac{1}{t}\int_0^t Nu_{\mathrm{n,t}} \mathrm{d}t \tag{14}$$

2.3 Numerical solution

The numerical solution in this study involves the application of the Galerkin finite element method (FEM) combined with the arbitrary Lagrangian-Eulerian (ALE) method to solve the interdependent, complex, and nonlinear governing equations. The Galerkin FEM is preferred for its ability to handle complex geometries and irregular shapes, accommodating different material properties, and easily incorporating boundary conditions. The ALE method is employed to approximate the governing equations in both moving and fixed boundary domains. By hybridizing the Lagrangian and Eulerian methods, the ALE procedure allows the computational mesh nodes to move, be fixed, or move in an arbitrary way, effectively capturing the interaction between the deformable wall and the fluid. The Galerkin FEM discretizes the weak form of the fluid field equations, while the deformation of the solid domain is computed using the Lagrangian method. Regarding the stability analysis, a fully coupled method is employed to solve all equations simultaneously, as the segregated solver approach failed to converge and maintain stability. Although fully coupled equations require increased computational memory, they ensure a stable solution. An automatic time step scheme with free steps based on the backward Euler method is implemented to monitor time steps and prevent divergence of the solution. This adaptive time stepping scheme helps maintain stability and accuracy throughout the simulation. The fully coupled approach, along with the adaptive time stepping scheme, ensures stability and accuracy of the numerical solution. The convergence criterion is set at $1e^{-6}$, ensuring that the numerical calculations reach a sufficiently stable state.

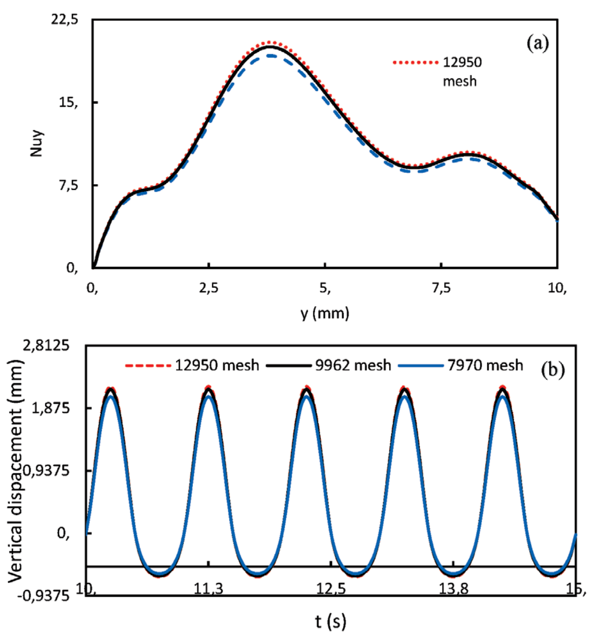

Figure 2: Effect of grid number on: a) temporal variations in the time averaged local Nusselt number along the length of the vertical hot wall and b) the vertical displacement of the flexible fin

2.4 Numerical method validation

Figure 2 shows the effect of grid number on (a) temporal variations in the time averaged local Nusselt number along the length of the vertical hot wall and (b) the vertical displacement of the flexible fin. It presents the variations in the maximum vertical displacement of the elastic vortex generator (flexible fin) in terms of time for three different grid numbers: 7970, 9962, and 12,950. As seen, with the increase in the grid numbers from 9962 to 12,950, the time averaged local Nusselt number along the length of the vertical hot wall and the temporal vertical displacement of the flexible fin does not change considerably. Thus, the grid with a mesh number of 9962 is used as our benchmark grid.

To check the validity of the numerical method used in the present study, the obtained results are compared with those obtained by Turek et al., which consist of a laminar incompressible enclosure flow around an elastic object that contributes to self-induced oscillations of the structure.[26] As seen in **Figure 3**, the variations in the lift force on the elastic wall obtained with the present numerical method are in good agreement with the above.[26] To validate the used FSI method, the study conducted by Kuttler and Wall was examined.[27] Their study examined a square cavity with a flexible bottom wall. The top wall of the cavity was subjected to a velocity of (1 –

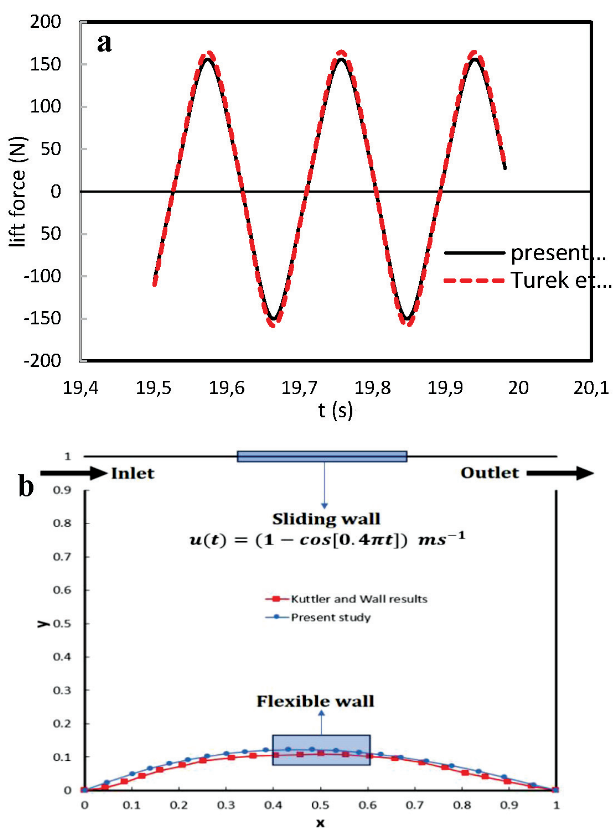

Figure 3: a) Variations in the lift force on elastic wall obtained with the present numerical method in comparison with those obtained by Turek et al.[26], b) comparison between the results based on the present numerical method and those obtained by Kuttler and Wal[27]

cos(0.4πt)) m/s, while two open gaps were positioned at the top of the vertical walls. The oscillation of the lid caused fluid motion within the enclosure, generating a vortex. Consequently, the interaction between the fluid and the flexible lower wall resulted in changes to the bottom wall's shape. These changes were computed over time. Comparison is done between the results based on the present numerical method and those regarding the deformation of the flexible wall after t = 7.5 s.[28] The comparison demonstrates a favorable agreement between the results.

It should be added that in the work done by Turek et al,[26] the fully implicit monolithic ALE-FEM method with a fully coupled multigrid solver was used for modeling the laminar incompressible enclosure flow around an elastic object which contributes to self-induced oscillations of the structure, as shown in **Figure 3a**. Besides, Kuttler and Wal[27] used a fixed-point FSI solver based on Dirichlet–Neumann partitioning to examine their problem, as shown in **Figure 3b**. However, our numerical study involves the application of the Galerkin finite element method (FEM) combined with the arbitrary Lagrangian-Eulerian (ALE) method to solve the interdependent, complex, and nonlinear governing equations.

3 RESULTS AND DISCUSSION

The effect of the elastic fin and elastic wall inside the L-shaped enclosure is studied by placing a fin at the corner where the two legs of the L meet, at a height of 3 mm. The fin is installed horizontally. This specific location is chosen to maximize its influence on the fluid flow and heat transfer characteristics within the enclosure. The purpose is to assess how the presence of the fin and the elastic wall influences the fluid flow and heat transfer characteristics within the enclosure.[28] The elastic nature of the fin allows it to deform and respond to the fluid flow and temperature gradients present in the enclosure. This deformation can significantly impact the flow patterns, turbulence, heat transfer rates, and overall thermal performance of the system. The fluid flow is affected by the collision of the flow with the fin, which influences the hydrodynamics and heat transfer of the flow. The interaction between the fluid flow and the fin affects the hydrodynamic characteristics of the system.[29] The fin can act as an obstacle, obstructing and diverting the flow. This can lead to changes in the flow velocity, turbulence levels, and pressure distribution within the enclosure. The altered flow patterns caused by the fin affect the heat transfer between the fluid and the enclosure walls.

Figure 4 represents the enclosure without a flexible fin and elastic wall at (a) time = 1 s, (b) time = 3 s and (c) time = 6 s; as well as (d) the time periodic pulsating temperature of the hot walls. The hot walls are exposed to the pulsating temperature as shown in Equation (15):

$$T_{\mathrm{h}}(t) = 325(1 + 0.15\sin(6\pi t)) \qquad (15)$$

The combination of the flexible fin and elastic wall introduces additional complexities to the flow dynamics. The flexible fin, with its specified elastic modulus and applied force, contributes to the fluid motion and alters the flow patterns within the enclosure.[30] Simultaneously, the oscillation of the elastic wall, driven by the distributed force, further influences the flow field and heat transfer characteristics. Due to the periodic pulsating temperature of the hot walls, a pulsating flow is induced

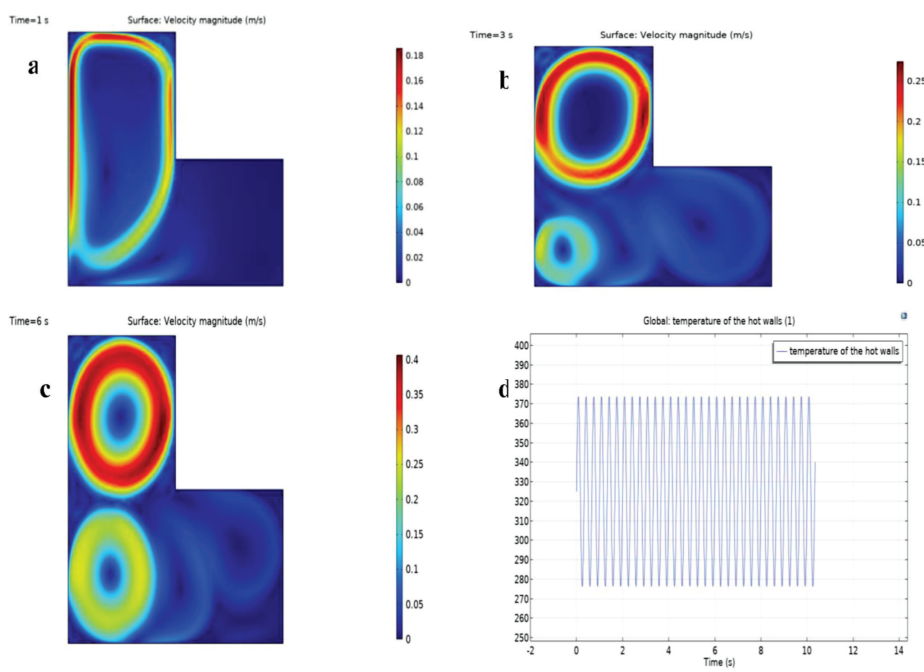

Figure 4: Variations in the contour of the velocity magnitude inside the enclosure at different times: a) time = 1 s, b) time = 3 s, and c) time = 6 s for the enclosure without a fin and elastic wall; d) time periodic pulsating temperature of the hot walls

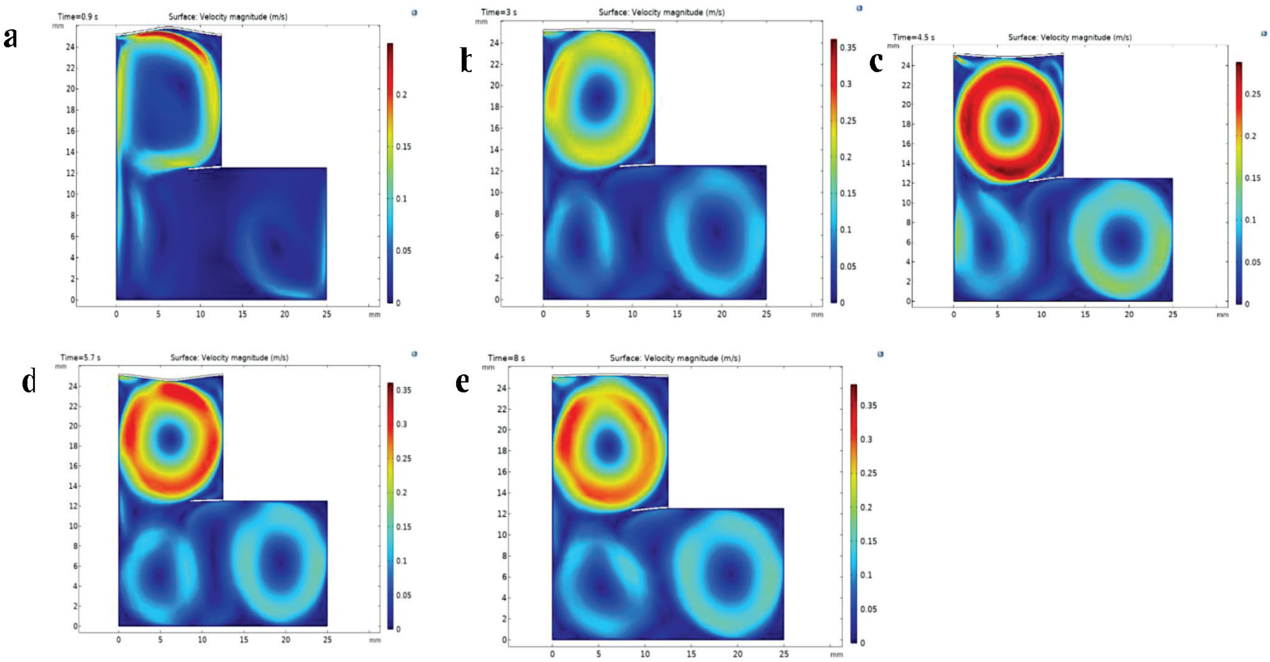

Figure 5: Variations in the contour of velocity magnitude inside the enclosure with a fin and an elastic wall at different times

caused by the effects of the pulsating thermal boundary condition and the effects of natural convection, which contributes to the oscillation of the flexible fin. The oscillation of the elastic fin may result in an enhancement of the heat transfer inside the L-shaped enclosure due to the increase in the turbulence inside the enclosure, as shown in **Figure 5**.

Figure 6 shows the time averaged variations in the local Nusselt number along the length of the vertical and horizontal hot walls for the enclosure with no elastic fin and no elastic wall, and for the enclosure including both flexible fin (E_{FF} = 1e8 Pa) and elastic wall (E = 5e6 Pa). The flexible fin, with its special properties, introduces additional fluid motion and enhances mixing within the enclosure. This increased fluid mixing leads to improved heat transfer, promoting the exchange of thermal energy between the hot wall and the fluid. These perturbations disrupt the boundary layer near the hot wall, reducing its thickness and promoting enhanced convective heat transfer. Combining the flexible fin and elastic wall creates synergistic effects that further enhance the heat transfer coefficient. The fluid motion induced by the flexible fin interacts with the oscillating elastic wall, leading to intensified flow mixing and enhanced heat transfer. This synergy between the two components amplifies the convective heat transfer performance compared to individual effects. **Table 1** presents a comparison between the current study and other, previously mentioned studies.

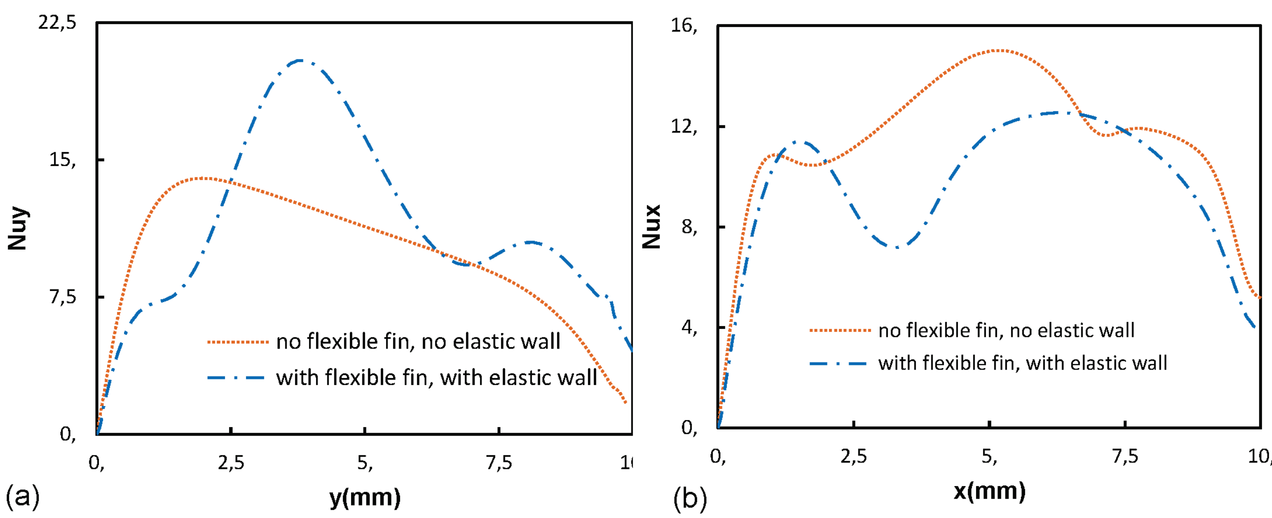

Figure 6: Time averaged variations in the local Nusselt number of the fluid flow for different cases along the length of the a) vertical hot wall, b) horizontal hot wall

Table 1: Comparison of studies

Study	Heat Transfer Enhancement	Enclosure	Flexible Fin	Elastic Wall	Fluid-Structure Interaction
Ghalambaz *et al.*[13]	Yes	L-shaped	Yes	No	Yes
Hashem Zadeh *et al*.[14]	Yes	Square	No	Yes	Yes
Jamesahar *et al*.[15]	Yes	Square	Yes	No	No
Khanafer *et al.*[16]	Yes	Square	No	Yes	No
Saleh *et al.*[17]	Yes	Open cavity	Yes	No	Yes
Sabbar *et al.* [18]	Yes	Square	No	Yes	No
Shahabadi *et al.* [19]	Yes	Square	Yes	No	Yes
Mehrian *et al.*[20]	Yes	Square	Yes	No	No
Akbal *et al*.[21]	Yes	Annular space	Yes	No	Yes
Chattopadhyay *et al.*[22]	Yes	Wavy chamber	No	Yes	No
Shahrestani *et al*. [23]	Yes	Circular	No	Yes	Yes
Alhashash *et al.* [24]	Yes	Square	No	Yes	Yes
Proposed study	Yes	L-shaped	Yes	Yes	Yes

4 CONCLUSION

Using the finite element method, a numerical investigation was conducted to analyze the flow and heat transfer characteristics inside an L-shaped enclosure including a flexible fin and an elastic wall. A fluid-structure interaction (FSI) model was utilized to capture the interaction between the fluid and the solid structure. The hot wall was subjected to the periodically pulsating temperature. A pulsating flow was induced by the pulsating thermal boundary condition and natural convection, contributing to the oscillation of the flexible fin. The oscillation of the elastic fin may have enhanced heat transfer within the L-shaped enclosure by increasing the turbulence inside the enclosure. The flexible fin and elastic wall strongly influenced the Nusselt number along the horizontal and vertical hot walls of the L-shaped enclosure. Additionally, the Nusselt number along the hot vertical wall increased. Furthermore, the combination of the flexible fin and elastic wall contributed to a further enhancement of the Nusselt number along the hot vertical wall of the L-shaped enclosure.

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