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Computational Analysis of Fluid Dynamics and Heat Transfer Characteristics of a Vibrating Heated Plate

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Abstract

The introduction of vibrations to a horizontal plate can induce turbulence in the flow field adjacent to the plate under certain combinations of amplitudes and frequency. It is also known that beyond a threshold level of heating, the convective flow field over a heated plate will transition to turbulence. The characterization of the flow field with turbulence in the domain is of paramount importance to ensure a realistic simulation of the flow physics. In the present study, a computational analysis is carried out to characterize the flow regime over a transversely vibrating flat plate (unheated and heated) into laminar or turbulent. The range of frequency and amplitudes of vibrations considered for this analysis are 0 – 150 Hz and 0 – 2 mm respectively. Three different models *viz.* laminar; Reynolds Averaged Navier-Stokes (RANS) approach with k-ω SST model; and Large-Eddy Simulations (LES) approach with dynamic Smagorinsky-Lilly model are employed, and the results of local, time and space averaged wall shear stress, and the Nusselt number are compared. It has been found that under conditions with unheated or heated vibrating plate, the wall shear stress predictions by all three models are in good agreement with each other with a maximum deviation of 9.5 %. However, when the predictions of local Nusselt number on the heated vibrating plate are compared, it is found that the laminar and LES predictions are in good agreement with each other; the k-ω SST model predictions deviate significantly from the other two models.

Introduction

Oscillatory flows enhance performance of heat transfer devices, and intensify mixing in thermo-chemical processes. An understanding of the nature of velocity and temperature fields in an oscillating fluid medium, by considering the parameters that govern the heat transfer performance in such flows can help design low-cost and better performing heat transfer devices.

One of the earlier studies on the effects of heat transfer on vibration is due to Lemlich [1], who defined a parameter called vibrational Reynolds number to correlate the effect of amplitude and frequency of vibration on the heat transfer. Fand [2] provided the validity of various hypotheses to explain the mechanics of interaction between vibration and heat transfer. Prasad and Ramanathan [3] studied free convection heat transfer from a longitudinally vibrating vertical plate, and reported a maximum of 33 % enhancement in heat transfer. Additionally, they reported that the extent of enhancement depends on the ratio of vibrational Reynolds number to Grashof number. Zhang et al. [4] performed a numerical study on a longitudinally vibrating vertical plate, and reported an increase in heat transfer performance with oscillating frequency, amplitude and the Prandtl number but a decrease with Grashof number. Gomaa and Al Taweel [5] carried out an analytical study to identify the effect of longitudinal vibrations applied to vertical flat surfaces on heat transfer. The results reported in their study showed good agreement with experimental data in the literature. They also found that the heat transfer performance increases with increase in axial temperature gradient in the presence of oscillatory motion.

Miller [6] experimentally measured the local heat transfer coefficients in a fully established oscillating turbulent boundary layer over a flat plate, and reported a small or no-measurable effect on heat transfer with oscillations that are considerably smaller than the free stream mean velocity. The study [7] performed with flows having higher oscillating velocities compared to the mean flow velocity reported almost a 5-fold increase in heat transfer from that of steady flow conditions.

Although many studies have been performed to study the effects of vibration on heat transfer there still seems to be an ambiguity in understanding the fundamental mechanisms involved with the effect of vibration on heat transfer performance. Fand [2], Drummond and Lyman [8] and Ralph [9] provided various theories that are inconsistent with each other. There is also lack of proper guidelines for the selection of critical non-dimensional parameters governing the mechanism of the problem for effective scale-up methodologies. Many studies have used appropriately defined vibrational Reynolds to study the effect of vibration on heat transfer but no work has reported threshold values for vibrational Reynolds number to categorize flow fields into laminar, transitional and turbulent.

The scope of the present work is to evaluate the suitability of computational models that could be used to realistically model the laminar and/or turbulent flow field, to understand the flow field and heat transfer characteristics associated with a vibrating heated plate subjected to sinusoidal vibration for a range of frequencies and amplitudes. Present analysis forms the ground-work for an exhaustive study dealing with the effects of vibration on thermo-fluidic characteristics to address the short comings mentioned in the previous paragraph. The simulation parameters are chosen by considering the range of parameters obtained from the literature review, besides accommodating a broad range for a planned experimental work in future.

Problem formulation and computational methodology

The geometry and the computational domain considered in this study are shown in figure 1. A horizontal flat plate of length l, situated between two stationary plates each of length b, is set to vibrate at a known frequency f and amplitude a in ambient surrounding temperature, T_∞ = 298.16 K.

The stationary and vibrating plates have no-slip boundary condition applied on them. All other boundaries of the domain are set to be at ambient conditions of air. For simulations involving heat transfer, additionally an isothermal boundary condition is applied on the vibrating plate and adiabatic condition on stationary plates. Air properties are kept constant (at T_∞ =

298.16 K) because the wall temperature considered is not high enough to cause any significant change in the properties of air except the density, when the vibrating plate is heated. The Boussinesq approximation is also used.

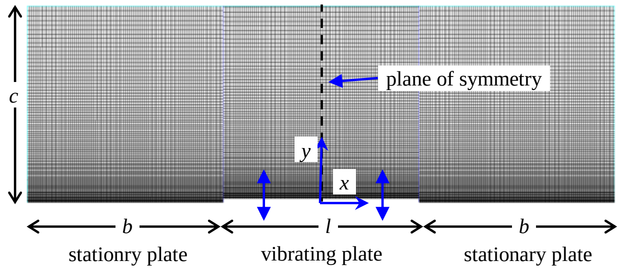


Figure 1. Physical geometry and computational domain.

The length scale of the geometry in the direction perpendicular to the plane of figure 1 is much larger than the other length scales in the domain, and hence the problem is assumed to be two-dimensional. However, for the LES simulations, a three-dimensional domain is developed extending into the plane of figure 1, up to a length of 4 times the maximum amplitude of vibration considered in this study. This length is chosen to effectively capture the flow features generated during the vibration of the plate, which are expected to be more or less of the same order as the amplitude of vibration. The boundary conditions on the vibrating plate and rest of the boundaries of the LES domain are similar to that of the two-dimensional simulation. Moreover, a periodic boundary condition is applied on the front and the back faces of the domain, essentially making the LES a quasi-two-dimensional simulation.

The two-dimensional mesh used is a fully-structured, spatially varying non-uniform mesh with rectangular elements, shown in figure 1. The three-dimensional mesh used in LES is a similar mesh with hexahedron elements. Both the meshes have finer cells close to the moving boundary (vibrating plate) to sufficiently capture the near wall effects. As the region in the close vicinity of the moving boundary is of high importance to obtain physically sensible results, dynamic meshing is achieved in such a way that mesh close to the moving boundary remain undisturbed.

Methodology

The purpose of vibrating the flat plate is to generate oscillations in the fluid medium surrounding it, which may help in the enhancement of heat transfer. There is no published information about the nature of the flow field - laminar, transitional or turbulent - about such a vibrating plate for any range of frequency and amplitude of vibration. Hence, it is not known *apriori* which computational model better suits the problem. In lieu of this, the problem is modelled using (i) a laminar model, (ii) RANS approach (*k*-ω Shear Stress Transport (SST)) model [10] and, (iii) a LES approach (dynamic Smagorinsky-Lilly Model) [11]. This is done to compare the results from the above three models and to see the variation, if any from one another to determine the nature of flow field, i.e. laminar, transitional or turbulent.

As the information on boundary layer velocities and temperature profiles are crucial for the problem at hand, the models with wall function approach are ruled out because of their sensitivity and their tendency to predict inaccurate results at finer grids closer to the wall. The *k-ω* SST model with low-Reynolds number formulation seemed appropriate as it can effectively resolve the viscous sub-layer that is close to a boundary including the log-law region. Its ability to switch to *k-ε* model formulation in the free stream, which is known to accurately capture the free stream turbulence made the *k-ω* SST model a choice for the problem at hand. Although, other two-equation models with enhanced wall treatment may be used, the *k*-ω SST model is selected because of its accurate and robust near-wall treatment [12]. For the LES, dynamic Smagorinsky-Lilly sub-grid scale model is chosen because accurate near-wall treatment is critical for the problem considered because of the possibility of laminar-turbulent transition in the boundary layer. For the accurate modelling of such flows, Mentor [13] recommends the dynamic Smagorinsky-Lilly model because it is formulated in such a way that it returns zero eddy-viscosity in laminar shear flows and because of its accurate near wall treatment without the use of any near wall damping functions.

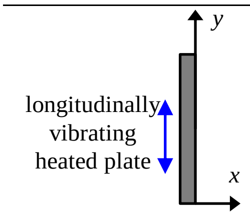
Above models are solved using the finite-volume solver available in the academic version of ANSYS FLUENT 14.0. The pressure-velocity coupling is achieved using the SIMPLE method. Second order upwind differencing scheme is used for solving pressure, momentum and energy equations. First-order implicit transient formulation is employed. The time step for each case is dependent on the frequency of vibration prescribed for the corresponding case.

Grid-Independence study

The grid sensitivity test has been performed by successively refining the mesh and comparing the relative variation in the wall shear stress. The mesh configurations chosen are with 150×50, 300×100, 500×150, 600×200 and 750×250 elements in the x and y direction respectively. It is found that further refinement in mesh beyond the mesh size of 300 × 100 resulted in a relative variation of less than 0.1 % in averaged wall shear stress.

Validation

The present computational methodology and dynamic meshing approach is validated by comparing the predicted results against the experimental data from Prasad and Ramanathan [3]., who studied the effect of longitudinal vibrations of a heated plate on heat transfer performance. A case is developed to see the effects of longitudinal vibration on heat transfer under similar operating conditions as the experimental study [3].



T_w	350.84 K
T_∞	298.16 K
a	3.125×10^{-3} m
f	10 Hz
Grashof number(Gr)	5.049×10^{6}

Table 1. Operating conditions of Prasad and Ramanathan [3] considered for validation

For the operating condition as shown in Table 1, Prasad and Ramanathan [3] experimentally observed and reported that the flow field is laminar and hence a laminar model is used to validate the results from simulations.

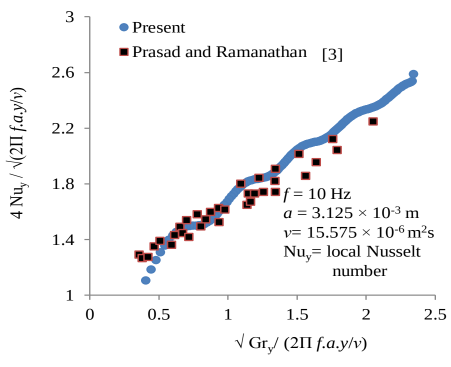


Figure 2. Comparison of computational results with the experimental results of Prasad and Ramanathan[3].

Figure 2 shows the comparison of non-dimensional parameters proposed by Sparrow [14], and adopted by [3] enabling validation of predictions from the present simulation. The results obtained using the present computational methodology is found to be in good agreement with the experimental values. Since the maximum deviation from the experiments is found to be less than 8%, the computational methodology is extended to carry out the present work.

Results and Discussion

The ranges of operating parameters considered for the present study are indicated in Table-2. As shown in Table-2, the laminar and *k*-ω SST simulations are performed for all the combinations of frequencies and amplitudes; LES cases are restricted to only two cases: (i) $f = 50$ Hz, $a = 0.001$ m & (ii) $f = 150$ Hz, $a = 0.002$. For heat transfer calculations, a wall temperature, $T_w = 323.16$ K is employed.

	units	Laminar	*k*-ω SST	LES
a	$\times 10^{-3}$ m	0.5, 1, 1.5, 2	0.5, 1, 1.5, 2	1, 2
f	Hz	5, 50,100,150	5, 50, 100,150	50, 150
l	m	0.05		
T_∞	K	298.16		
T_w	K	323.16		

Table 2. Operating parameters considered for the present study

Comparison of fluid dynamic characteristics predicted by the models

Figure 3 (a-h) shows the velocity vectors in the region adjacent to the vibrating plate at different stages of a cycle of vibration, obtained using LES for a representative set of frequency and amplitude of 150 Hz and 0.002 m respectively. In the figure, the stationary plate is showed in black and the vibrating plate is showed in red. Only one half about the plane of symmetry is shown.

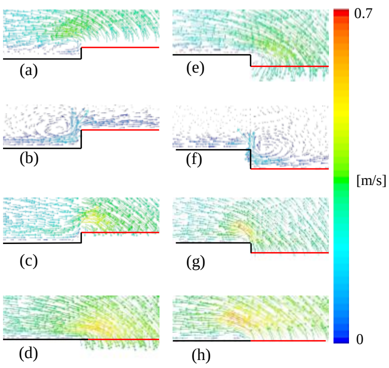


Figure 3. Velocity vectors (coloured by velocity magnitude) in the vicinity of the vibrating plate at different phases of a cycle of vibration.(a) 12.5 % (b) 25 %; (c) 37.5 %; (d) 50 %; (e) 62.5 %; (f) 75 %; (g) 87.5 %; (h) 100%.

The qualitative trends obtained from *k*-ω SST and laminar simulations are similar and hence omitted for brevity. As it is seen from the figure that as the plate moves upwards in the first quarter of the cycle it imparts momentum to the fluid adjacent to it, thereby resulting in a flow directed away from the plane of symmetry. As it goes down in the second quarter of the cycle a relatively low pressure zone is formed on the top of the vibrating surface, and this low pressure causes the fluid (air) to change direction and the cyclic repetition of this leads to the formation of rotational velocity components close to the edges of the vibrating plate which can be seen near the edge of the vibrating plate from figure 3. Similarly after the 75% of the cycle the vibrating plate changes direction and because of no slip condition the fluid also changes direction and as explained above cyclic repetition of this forms vortex like flow features around the edge of the plate. The size of these vortex-like structures is found to be dependent on the frequency and amplitude of vibrations.

Figure 4 illustrates the distribution of wall shear stress on the surface of the vibrating plate. The left hand side of the plot corresponds to vibrational parameters, $f = 50$ Hz and $a = 0.001$ mm, and the right hand side of the plot corresponds to $f = 100$ Hz and $a = 0.001$ mm. It can be seen that the local wall shear stress increases as we move away from the plane of symmetry towards the edges. This trend was also noticed as the cycle of vibration progressed.

It is also seen from the comparison of the left and right hand sides of the plot that for the same amplitude and increase in frequency the local wall shear stress also increase, particularly more so towards to the edge of the plate. Similar increase in the instantaneous local wall shear stress is observed when moving from the centre of the vibrating plate to its edge when comparing cases having same frequency but with different amplitudes.

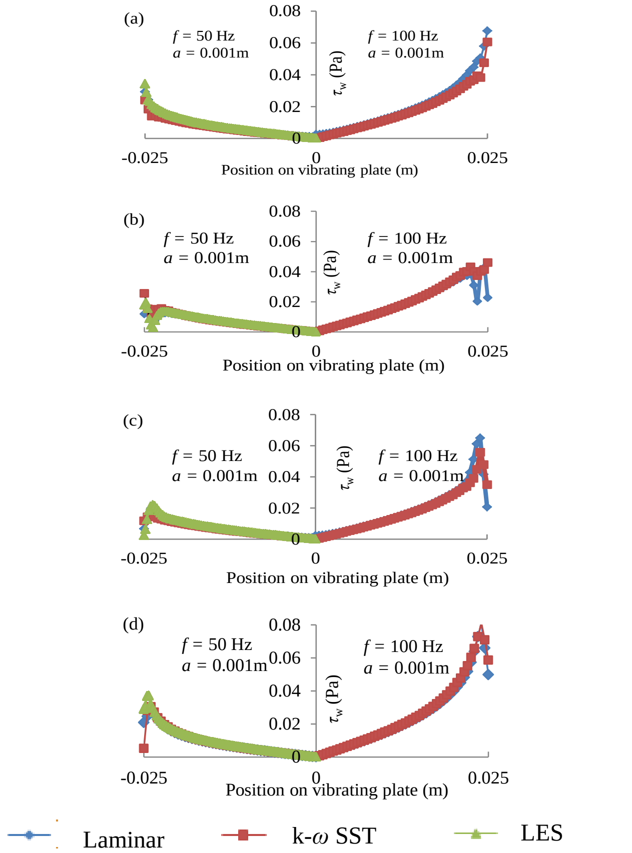


Figure 4. Local wall shear stress (instantaneous) illustrated at various phases of the cycle of vibration (a) 25 %; (b) 50 %; (c) 75 %; (d) 100 %.

It is also observed that all the three models are in good agreement with each other, with maximum deviation found towards the edge of the plate. There appears a sudden decrease and increase in wall shear stress at the edge of the plate almost consistent after the first quarter of the cycle. This can be seen as a consequence of the low pressure created on the vibrating plate, and the subsequent formation of recirculation zones as explained above. As the recirculation appears to occur close to the edge of the plate, it is noted that care must be taken to increase the accuracy of modelling this effect.

As the sudden decrease in wall shear stress appear to occur within 5% of the length of the plate from the edge of plate only 90% of the total length of the vibrating plate is considered for the estimation of the time and space averaged wall shear stress, for all the vibration controlling parameters considered for the study.

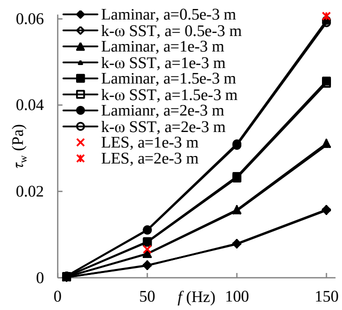


Figure 5. Comparison of the time and space averaged wall shear stress.

Figure 5 shows the variation in the predicted magnitude of wall shear stress (time and space-averaged) obtained using the laminar, k-ω SST and LES models, with a variation in the frequency of vibration, for different values of the amplitude of surface vibration. While it may appear that the magnitude of wall shear stress is very small, it is pointed out that it is of the same order of magnitude as seen in natural convection heat transfer over a horizontal plate presented by Samanta and Guha [15]. It is also seen from figure 5 that the wall shear stress increases concomitantly with an increase in both the frequency as well as the amplitude of applied surface vibration. It is seen that there is negligible difference in wall shear stress predicted by the three models with a minimum and maximum relative deviation of 0.9 and 9% respectively. This suggests that the flow field is laminar over the entire range of operating vibration parameters considered for the present study. This is also corroborated by the time and space averaged ratio of turbulent eddy viscosity to molecular viscosity which is found to be as low as 0.01.

Comparison of thermo-fluid characteristics predicted by the models

A case was developed (T_w = 323.16 K; T_∞ = 298.16 K; f = 50 Hz; a = 0.001 m; Gr = 4.2620 $\times$ 10^5) to study the effect of transverse vibrations on heat transfer. Simulations have been carried out using the laminar, k-ω SST, and LES models and with isothermal wall boundary condition on the vibrating plate. The local and time-averaged Nusselt numbers were obtained.

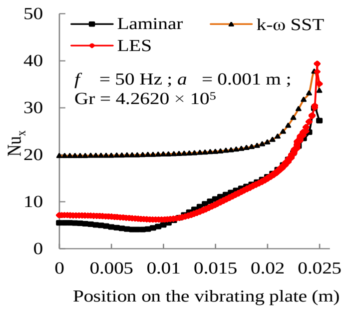


Figure 6. Time averaged profile of local Nusselt number over the vibrating heated plate.

Figure 6 shows the time averaged Nusselt number distribution over one half of the plate about the plane of symmetry. The time averaged local Nusselt number profile obtained from laminar and LES simulations appear similar in trend and values, but the k-ω SST results differ in the values and trend compared to laminar and LES results. In contrast to Nusselt number predictions, the local wall shear stress data from all three models seem to be in good agreement with each other. As the wall shear stress predicted by the three models in cases with and without heat transfer are of the same order of magnitude, the flow for the present case with f = 50 Hz , a = 0.001 m is also characterized to be laminar.

Conclusions

A computational analysis of the fluid dynamics and heat transfer characteristics of a vibrating surface for a wide range of vibration controlling parameters is performed. Three computational modelling approaches *viz.* laminar; RANS with k-ω SST model; and LES with dynamic Smagorinsky-Lilly model are employed to study the thermo-fluidic characteristics of a vibrating surface. The key findings are: the wall shear stress increases with increase in frequency and amplitude of vibration, and the flow field is laminar for the vibrational parameters considered for the vibrating plate. The flow field is characterized as laminar, and the laminar modelling approach can be safely used for the entire range of parameters considered for the study. For a case with f = 50 Hz, a = 0.001 m, and a wall temperature of 323.16 K, the flow field is also characterized as laminar. Furthermore, a more exhaustive set of vibration controlling parameters are being investigated to establish the threshold values of vibrational Reynolds number to characterize the flow over a vibrating heated surface into laminar, transitional and turbulent regimes.

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References

[1] Lemlich, R., 1955, “Effect of Vibration on Natural Convective Heat Transfer,” *Ind. Eng. Chem.*, **47** (6), 1175–1180.

[2] Fand, R.M., 1962,“Mechanism of Interaction between Vibrations and Heat Transfer,” *J. Acoust. Soc. Am.*, **34** **(**12), 1886.

[3] Prasad, K.K., and Ramanathan, V., 1972, “Heat transfer by free convection from a longitudinally vibrating vertical plate,” *Int. J. Heat. Mass. Tran.*, **15** (6), 1213–1223.

[4] Zhang, X.R., Maruyama, S. and Sakai, S., 2004, “Numerical investigation of laminar natural convection on a heated vertical plate subjected to a periodic oscillation,” *Int. J. Heat. Mass. Tran.*, **47** (19-20), 4439–4448.

[5] Gomaa, H. and Al Taweel, A.M., 2005, “Effect of oscillatory motion on heat transfer at vertical flat surfaces,” *Int. J. Heat. Mass. Tran.*, **48** (8), 1494–1504.

[6] Miller, J.A., 2011, “Heat Transfer in the Oscillating Turbulent Boundary Layer,” *J. Eng. Gas. Turb. Power*, **91** (4), 239.

[7] Dec, J.E., Keller, J.O., and Arpaci., V.S. 1992, “Heat transfer enhancement in the oscillating turbulent flow of a pulse combustor tail pipe,” *Int. J. Heat. Mass. Tran*, **35** (9), 2311–2325.

[8] Drummond, C.K., and Lyman, F.A., 1990, “Mass transfer from a sphere in an oscillating flow with zero mean velocity,” *Comput. Mech.*, **6** (4), 315–326.

[9] Ralph, M.E., 2006, “Oscillatory flows in wavy-walled tubes,” *J. Fluid. Mech.*, **168** (1), 515.

[10] Menter, F.R., 1993, “Zonal Two Equation k-ω Turbulence Models for Aerodynamic Flows,” *AIAA.*, **93**, 2906.

[11] Lilly, D.K., 1992, “A proposed modification of the Germano subgrid-scale closure method,” *Phys. Fluids. A- Fluid.*, **4** (3), AIP, 633.

[12] Menter, F.R., Kuntz, M., Langtry, R., 2003, “Ten years of industrial experience with SST turbulence model.”, *Proceedings of the fourth International Symposium on Turbulence, Heat and Mass Transfer*.

[13] Menter, F.R., 2012, "Best Practice: Scale-Resolving Simulations in ANSYS CFD", *Ansys Resource Library*.

[14] Sparrow, E.M., Eichhorn, R., and Gregg, J.L., 1959, “Combined Forced and Free Convection in a Boundary Layer Flow,” Phys. Fluids, **2** (3), 319.

[15] Samanta, S., and Guha. A., 2012, "A similarity theory for natural convection from a horizontal plate for prescribed heat flux or wall temperature", *Int. J. Heat. Mass. Tran*, **55**, 3857-3868.