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Genetic algorithm technique for optimization of heat transfer enhancement from a flat plate

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Abstract

In this study, a numerical approach has been implemented to find the optimal shape of a two dimensional turbulator above an isothermal flat plate and parallel to the flow. The turbulent boundary layer over the plate was disrupted at various distances by inserting a quadrilateral bar where the boundary layer thickness was grown to the three times greater than the insert's height. As a result, the overall wall heat transfer was altered. Heat transfer coefficient was defines as a cost function. The single objective Genetic Algorithm (GA) with the modified Teach-t code were employed to optimize variables such as the shape and distance of the obstacle from the plate. Applying the optimal geometry demonstrated that the mean heat transfer coefficient is enhanced by more than 51% over the affected area comparing with the case of undisturbed boundary layer. Downstream flow comprises of the stagnation point which formed on the frontal face of the body and accounts for the slight decrease in the heat transfer coefficient. The jet underneath the obstacle followed by the developing boundary layer is responsible for the sudden peak of the heat transfer coefficient. However, the downstream wake zone makes the main contribution.

Keywords: Heat transfer enhancement, Optimization, Genetic Algorithm, Heat transfer coefficient, flat plate

Nomenclature

c_p	Specific heat
$C_\mu, \sigma_k, \sigma_\varepsilon, C_{1\varepsilon}, C_{2\varepsilon}$	Constants of $\kappa - \varepsilon$ equation, as defined in the content
$h_{c,x}, \overline{h_c}$	local and averaged Heat transfer coefficient
H	Step height
k	Conduction heat transfer coefficient
p	Static pressure
q''	Heat flux
S	Mesh expansion factor
T	Temperature
U	Main stream flow velocity
$x,\ y$	Longitudinal and transversal coordinates, respectively

Greek Symbols

μ	Dynamic viscosity
ρ	Mass density
δ_v	Velocity boundary layer thickness
δ_t	Thermal boundary layer thickness
τ	Shear stress
κ	Turbulent kinetic energy

Subscripts

i	x-wise properties of the nodes
j	y-wise properties of the nodes
a, b, c, d	Coordinates of a quadrilateral obstacle

Exponents

$-$	Average value of a parameter
‘	Fluctuation component of a parameter

Notations

GA	Genetic Algorithm
$\partial/\partial t$	Partial derivative

Non-dimensional Numbers

Re	Reynolds number
y^+	Wall coordinates , y/δ_v
u^+	Wall coordinate, $\overline{u}/u_\tau$

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1. Introduction

Thermal energy is transferred to the main flow primarily by means of the boundary layer. Altering the boundary layer structure has determining effects not only on heat but also momentum transfer. Subsequently, parameters such as heat transfer coefficient and skin friction are affected by the boundary layer features. In this research, the relationship between heat transfer enhancement and the boundary layer features has been studied. The GA is used to introduce the thermally optimal profile of a blunt body that yields the maximum overall heat transfer coefficient. The optimal geometry is limited to a number of constraints applied to the optimization process.

Bhavnani and Bergles [1] studied the effects of laminar boundary layer disruption considering a free convection case. They conducted an interferometer study on a vertical plate kept at constant temperature. Two types of transversal elements were mounted on a 127mm wide, 178mm long aluminum plate (including the ribs and horizontal steps) and local measurements were performed by applying the Mach-Zehnder interferometer. It was found that the transverse ribs, decreased the overall heat transfer rate by creating stagnation zones on both upstream and downstream sides of the ribs. It was also shown that the stepped obstacles slightly enhanced the heat transfer from the plate.

Thermal effects of cylindrical inserts located above the flat plate and normal to the flow were treated in the earlier experimental studies performed by Kawaguchi et al. [2] and Marumo et al. [3]. Kawaguchi experimentally studied the thermal characteristics of the flow disturbed by an array of cylinders mounted inside the boundary layer and parallel to the flow and the plate. A variety of traversal distances from the wall was considered. The results indicated a 40% increase of the heat transfer coefficient over the affected area. The dissimilarity between heat and momentum transfer for turbulent flow disturbed by a cylindrical insert was experimentally investigated by Suzuki et al. [4]. In a similar investigation Inaoka et al. [5] considered a square rod above the plate and conducted a numerical investigation, and then demonstrated it through an experiment.

In a similar study heat transfer enhancement was also achieved by the insertion of a square rod in the forced turbulent boundary layer [6].

The presence of a stagnation point at the frontal face of the obstacle, the jet of fluid underneath and the recirculation zone behind the insert are the common flow feature observed. It has also been shown that the heat transfer augmentation rate is a function of size and location of the insert and parameters of upstream flow. Different parts of the flow are characterized by various effects on the local heat transfer coefficient. Heat transfer initially decreases because of the stagnation point and then rapidly increases as the boundary layer is washed away by the subsequent fluid jet. Afterwards the reestablishment of the boundary layer depreciates the heat transfer coefficient. However it is followed by an increase due to the positive influence of the wakes form behind the body downstream of the separation point, where the fluid particles gain their minimal velocity. The recirculation zone accounts for the main contribution to the wall heat transfer enhancement. The wake induced augmentation endurance depends on the size of the recirculation zone. The present work is based on these outcomes.

The flow features elaborated earlier suggest that a bigger circulation zone results in a higher overall enhancement. Depending on the shape and the location of the obstacle it is possible to achieve better heat transfer enhancement which have been ignored in the reviewed literature. It is obvious that an optimal geometry can be found for each combination of geometrical parameters and flow features.

This paper is aimed to find a quadrilateral insert profile that comes up with the maximum enhancement. In this connection, a 14m/s parallel flow is considered to stream over an isothermal wall heated up to 70°C. The temperature difference between the plate and the main stream maintained at 50°C. The boundary layer is disturbed by a quadrilateral bar and the local Reynolds number is considered based on the streamwise distance from the leading edge. It is favorable in this work to look for the optimized design for a manipulator that yields maximum heat transfer over the affected area. Affected area is defined to be the distance along the plate over which the wall heat transfer coefficient is ±5% different from its identical value for an undisturbed flow. A schematic diagram of the problem is shown in Fig. 1.

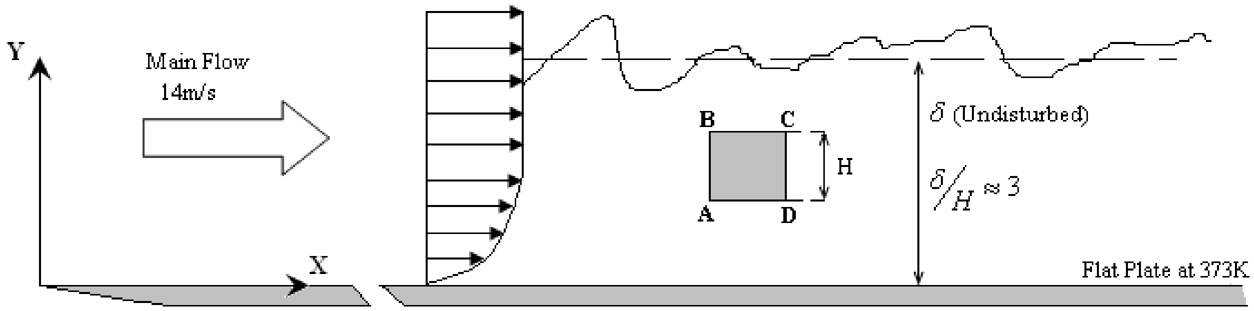

Fig. 1. Schematic diagram of the flow arrangement

2. Governing equations

Flow is considered to be two dimensional, turbulent and the Mach number is low enough to assure the incompressibility. At a certain distance not far from the plate the flow condition can be treated as steady since the beating vortices do not shed far apart from the insert. Therefore, the steady state continuity, momentum and energy equations are solved in their general form.

Continuity:

$$\frac{\partial(\rho u_i)}{\partial x_i}=0 \tag{1}$$

Momentum:

$$\frac{\partial \rho u_i u_j}{\partial x_j}=\frac{\partial \tau_{ij}}{\partial x_j}-\frac{\partial p}{\partial x_i} \tag{2}$$

Herein, neither viscous nor turbulent transforms can be ignored in the flow field. Thus both turbulent and viscous diffusivities should be counted in the governing equations. So,

$$\tau_{ij}=\mu\frac{\partial \overline{u}}{\partial y}-\rho\overline{u_i'u_j'} \tag{3}$$

Where:

$$-\rho\overline{u_i'u_j'}=\mu_t\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\frac{2}{3}\rho k\,\delta_{ij} \tag{4}$$

Likewise the momentum, the total heat transfer is also altered by both molecular and turbulent transport effects:

$$q''=-k\frac{\partial \overline{T}}{\partial y}+\rho\,\mathrm{C}_p\overline{v'T'} \tag{5}$$

However, in dealing with low speed and low turbulent intensity flows turbulent heat transport is not of significant value. Thus, viscous and turbulent dissipations are neglected throughout the numerical calculations. Next the energy equation is simplified as Eq. 6.

$$\frac{\partial(\rho u_i T)}{\partial x_i}=-\frac{\partial q_i}{\partial x_i} \tag{6}$$

Considering the number of cases that needs to be solved to achieve the optimal layout, different simplification is considered to reduce the overall computational cost and time. The turbulent flow is solved by applying the two-equation standard $k-\varepsilon$ model, Eq. 7.

$$\frac{\partial(\rho k)}{\partial t}+div(\rho kU)=div\left[\frac{\mu_t}{\sigma_k}grad\,k\right]+G-\rho\varepsilon$$
$$\frac{\partial(\rho\varepsilon)}{\partial t}+div(\rho\varepsilon U)=div\left[\frac{\mu_t}{\sigma_\varepsilon}grad\,\varepsilon\right]+ C_{1\varepsilon}\frac{\varepsilon}{k}G-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{7}$$

The constants used in these equations are:

$$C_\mu=0.09;\sigma_k=1.00;\sigma_\varepsilon=1.30;$$
$$C_{1\varepsilon}=1.44;C_{2\varepsilon}=1.92$$

Using the high Reynolds number model makes it possible to acquire the near wall solution without increasing the number of near wall nodes which demands less computational effort. The $k-\varepsilon$ model is chosen as it leads to stable and also time-effective solutions.

The mesh is generated frequently for every different case suggested by (GA) throughout the optimization. The developed computer code is verified by evaluating the reattachment length downstream of the obstacle to ascertain a mesh independent solution [6]. The mesh expansion factor, S=1.08 is chosen in the grid generated for each proposed geometry meanwhile the optimization.

3. Optimization technique

GA is an accepted technique in the multidisciplinary optimization problems and has been used in fluid mechanics and heat transfer for many years. McCormack [7] optimized the suction distribution over the flat plate to delay the separation and minimize the friction drag. In another study the plane design was optimized by GA for lift and drag forces [8]. Herein, GA is employed as an optimization technique. Heat transfer coefficient is defined as a cost function and the purpose is to maximize its overall value. The average heat transfer coefficient is defined by Eq. 8 and it is applied to the individual profiles through the optimization.

$$\overline{h_c}=\frac{1}{\Delta x}\int_{x_1}^{x_2}h_{cx}\,dx \tag{8}$$

where:

$$h_{cx}=-k\frac{\left(\dfrac{T_{i,2}-T_w}{y_{i,2}}\right)}{(T_w-T_\infty)}$$

The eight coordinates of the quadrilateral profile are defined as geometrical variables of the fitness functions. The coordinates of the insert are limited in the certain range. Furthermore, to avoid disproportionate geometries a set of logical relations are applied to the proposed designs. These geometrical features are listed in Tables 1 and 2.The optimized geometry is determined according to these criteria. A schematic of the coordinate system and the sequence of the blunt body corners as points A, B. C. D are shown in Fig. 1.

Table 1. Limitations of the geometrical parameters in GA

Parameter	Geometrical limits (all dimensions are in mm)
Point A	$1400\le x_a\le 1410$ $1.4\le y_a\le 9.8$
Point B	$1400\le x_b\le 1414$ $2.2\le y_b\le 12$
Point C	$1406\le x_c\le 1420$ $2.2\le y_c\le 12$
Point D	$1406\le x_d\le 1420$ $1.4\le y_d\le 9.8$

Table 2. constraints of the geometrical features in GA

Condition	Geometrical constraint
1	$x_a < x_d$
2	$x_b < x_c$
3	$y_a < y_b, y_a < y_c$
4	$y_a = y_d$

A FORTRAN code is developed based on the flow chart depicted in Fig. 2. To improve the convergence of the GA, the initial population is chosen based on the results of the code for sample runs. Additionally, the design variables and the fitness functions are saved at the end of each run. Then, new geometrical variables are checked with the database before being processed by the code to debar recapitulation.

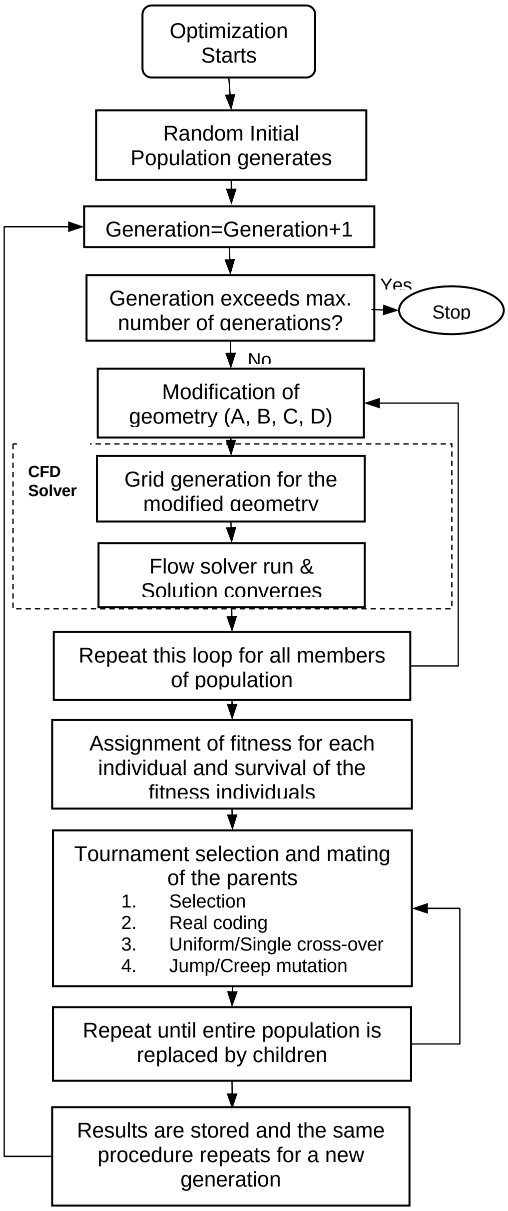

Fig. 2. Flowchart of optimization procedure

3. Results and discussions

The optimal blunt body design is the one that yields maximum heat transfer coefficient behind the obstacle compared to the rest of the population. The parameters applied in the optimization of the heat transfer coefficient are listed in Table 3.

For generations with small population of about 20 to 40 members, better performance of GA were obtained by employing high and low cross-over and mutation rates respectively and mutation rates higher than 0.05 are not suggested. In the authors experience mutation rates in the range of 0.006 to 0.011 save the optimization time. Herein, the probability of mutation is considered to be 0.009 while the probability of cross-over rate is assumed to be 0.88. The chromosomes lengths are real coded.

Table 3. Parameters used in single objective GA optimization of heat transfer coefficient

Number of population	*20*	Selection method	*Ranking*
Initial population generation method	*Random*	Mutation method	*Structured*
Total number of generations	*100*	Crossover method	*Two-point*
Stop criterion	*100^{th} generation*		

The optimization took 500 hours on a system with a 2.8GHz processor and 4GB RAM. The best and average values of the fitness function for each generation are shown in Fig. 3. After the 15^{th} generation the maximum heat transfer coefficient reached to an approximately constant value of 49W/m^{2}K. However, the average value of the fitness function goes up to its maximum value.

The geometrical features of the optimal layout are presented in 4.

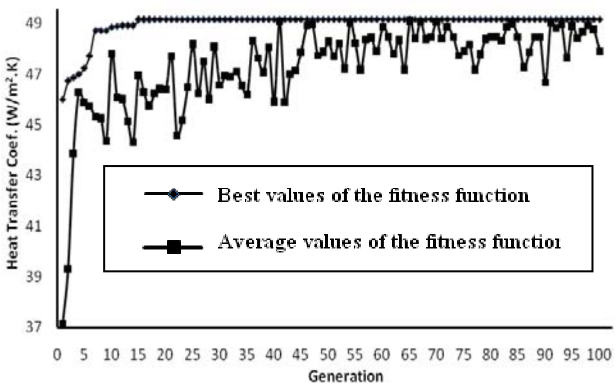

Fig. 3. Variation of the fitness function through different generations

The heat transfer coefficients along the wall of undisturbed and disturbed boundary layer are plotted in Fig. 4. Fluctuations of the heat transfer coefficient can be explained by the variation in the velocity gradients near the wall and the recirculation zone behind the obstacle. The streamlines around the optimal geometry are demonstrated in Fig. 5. The boundary layer disappears underneath the manipulator as the consequence of the presence of the fluid jet which leads to the increase of heat transfer.

Table 4. Coordinates of the suggested optimal profile for the maximum overall heat transfer coefficient

	Point A	Point B	Point C	Point D
x	1.405m	1.405m	1.407m	1.420m
y	1.4mm	12mm	2.2mm	1.4mm

The first vortex generates as the high velocity flow leaves the gap and low pressure zone is formed behind the insert. The downwash flow from the obstacle forms a bigger vortex further downstream. These two vortices rotate in the same direction which results in very low speed flow field near the wall. Consequently the temperature difference between the flow and the wall decreases. Therefore, the heat transfer coefficient drops to its minimum value at this point. The second vortex contributes to the overall heat transfer coefficient behind the obstacle as it is indicated in Fig.4.

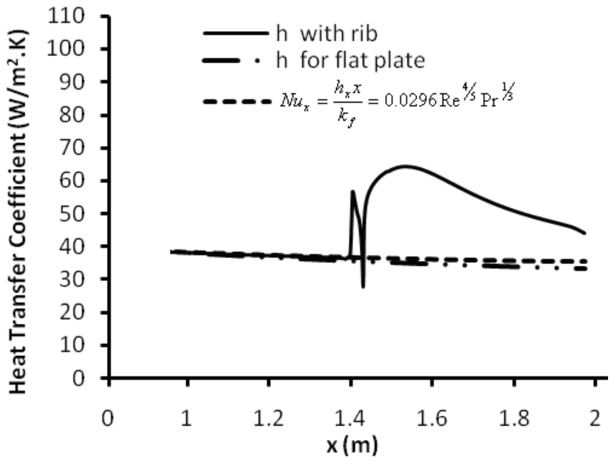

Fig. 3. Variation of heat transfer coefficient along the streamwise distance in the presence and absence of the obstacle

From Eq. 9 the predicted overall heat transfer coefficient is equal to 35.8 $W/m^2.K$ for the undisrupted boundary layer. The identical value achieved by inserting the optimal profile is 54.2 $W/m^2.K$. Comparing these values shows a 51.17% increase in heat transfer coefficient as a result of adopting the proposed geometry.

$$Nu_x = \frac{h_x x}{k_f} = 0.0296\,\mathrm{Re}^{4/5}\,\mathrm{Pr}^{1/3} \quad (9)$$

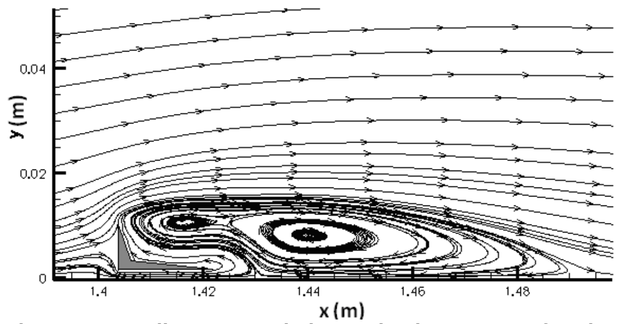

Fig. 2. Streamlines around the optimal geometry for the highest heat transfer coefficient

4. Conclusion

The complexity of the flow behind the obstacle and the number of different parameters involved in the calculation of the heat transfer coefficient, make the optimization of the flow structure a difficult and lengthy process. An optimization method based on the GA and modified (Teach-t) code was developed to investigate the geometry of the optimal insert shape. The optimization was done through the single objective algorithm both for the heat transfer and skin friction coefficients.

The numerical results indicate that the overall heat transfer coefficient depend on the height of the obstacle. And there is an inverse relation between the overall heat transfer coefficient and the obstacle vertical positioning from the plate. The jet formed beneath of the manipulator increases the heat transfer coefficient and has an adverse effect on the wall skin friction by washing out the boundary layer.

As the result of implementing the optimized geometry the overall heat transfer enhancement of about 51% over the affected area was achieved. The overall enhancement mainly counts for the vortices appear downstream of the blunt body. Flows from the vortex core to wall decrease the boundary layer thickness and increase the heat transfer. Moreover, upwards flows from wall to vortices core augment heat transfer by increasing the mixing rate.

References

[1] Bhavnani, S.H. and A.E. Bergles, *Interferometric study of laminar natural convection from an isothermal vertical plate with tranverse roughness elements*, in *8th National Heat and Mass Transfer* 1986, Hemisphere publishing corporation and Springer-Verlag: Visakhapatnam, India.

[2] Kawaguchi, Y., K. Suzuki, and T. Sato, *Heat transfer promotion with a cylinder array located near the wall.* International Journal of Heat and Fluid Flow, 1985. **6**(4): p. 249--255.

[3] Marumo, E., K. Suzuki, and T. Sato, *Turbulent heat transfer in a flat plate boundary layer disturbed by a cylinder.* International Journal of Heat and Fluid Flow, 1985. **6**(4): p. 241--248.

[4] Suzuki, H., K. Suzuki, and T. Sato, *Dissimilarity between heat and momentum transfer in a turbulent boundary layer disturbed by a cylinder.* International Journal of Heat and Mass Transfer, 1988. **31**(2): p. 259--265.

[5] Inaoka, K., J. Yamamoto, and K. Suzuki, *Dissimilarity between heat transfer and momentum transfer in a disturbed turbulent boundary layer with insertion of a rod - modelling and numerical simulation.* International Journal of Heat and Fluid Flow, 1999. **20**(3): p. 290--301.

[6] Kahrom, M., S. Farievar, and A. Haidari, *The effect of square splittered and unsplittered rods on flat plate heat transfer* IJE Transactions A:Basics, 2007. **20**.

[7] McCormack, W., O.R. Tutty, and E. Rogers, *Stochastic optimization based control of boundary layer transition.* Control Engineering Practice, 2002. **10**: p. 243-260.

[8] Ng, T.T.H. and G.S.B. Leng, *Application of Genetic Algorithms to conceptual design of a micro-air vehicle.* Engineering Application of Artificial Intelligence, 2002. **15**: p. 439-445.