OSCILLATOR FIN AS A NOVEL HEAT TRANSFER AUGMENTATION DEVICE FOR GAS TURBINE COOLING APPLICATIONS

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ABSTRACT

A new concept for enhanced turbulent transport of heat in internal coolant passages of gas turbine blades is introduced. The new heat transfer augmentation component called “oscillator fin” is based on an unsteady flow system using the interaction of multiple unsteady jets and wakes generated downstream of a fluidic oscillator. Incompressible, unsteady and two dimensional solutions of Reynolds Averaged Navier-Stokes equations are obtained both for an oscillator fin and for an equivalent cylindrical pin fin and the results are compared. Preliminary results show that a significant increase in the turbulent kinetic energy level occur in the wake region of the oscillator fin with respect to the cylinder with similar level of aerodynamic penalty. The new concept does not require additional components or power to sustain its oscillations and its manufacturing is as easy as a conventional pin fin. The present study makes use of an unsteady numerical simulation of mass, momentum, turbulent kinetic energy and dissipation rate conservation equations for flow visualization downstream of the new oscillator fin and an equivalent cylinder. Relative enhancements of turbulent kinetic energy and comparisons of the total pressure field from transient simulations qualitatively suggest that the oscillator fin has excellent potential in enhancing local heat transfer in internal cooling passages without significant aerodynamic penalty.

NOMENCLATURE

D	cylinder diameter, oscillator fin width
f	frequency
k	turbulent kinetic energy
k*	k/u_0^2 , non-dimensional turbulent kinetic energy
p	static pressure
p*	$(p-p_{ref})/\rho u_0^2$, non-dimensional static pressure
p_t	$p+\rho(u_x^2+u_y^2)/2$, total pressure
p_t*	$(p_t - p_{ref})/\rho u_0^2$, non-dimensional total pressure coefficient
Re	$\rho u_0 D/\mu_0$, Reynolds Number
St	fD/u_0 , Strouhal Number
t	time
t*	tu_0/D , non-dimensional time
u_0	inlet velocity
u_i	i'th component of the velocity vector
u_i*	u_i/u_0 , non-dimensional velocity
x_i	i'th component of spatial coordinate
x_i*	x_i/D , non-dimensional coordinate
Δp_t*	$p_t^*{}_{inl} - p_t^*$, non-dimensional total pressure loss
ε	viscous dissipation rate of turbulent kinetic energy
ε*	$\varepsilon D/u_0^3$, non-dimensional viscous dissipation rate of turbulent kinetic energy
Φ	viscous dissipation function
Φ*	$\Phi D^2/u_0^2$, non-dimensional viscous dissipation function
μ_0	absolute viscosity
μ_t	turbulent viscosity
μ_t*	$\mu_t/(\rho u_0 D)$, non-dimensional turbulent viscosity
μ*	$1+\mu_t/\mu_0$, non-dimensional viscosity
ρ	density

INTRODUCTION

In order to increase the efficiency of gas turbine engines, effective cooling of high pressure turbine blades is necessary. The enhancement of the heat transfer in internal coolant passages of gas turbine blades can be achieved by increasing the turbulence levels and unsteadiness of the coolant flow while keeping the pressure losses as low as possible. The devices used in internal coolant passages to increase turbulence levels and unsteadiness come with various geometrical sizes and shapes. The most common ones are pin fins and ribs (trip strips).

The trip strips work by tripping the boundary layer periodically and causing a repeating flow pattern within the passage which leads to high turbulence levels in the core flow. Boyle (1984), Han (1988), Abuaf and Kercher (1994), Ekkad and Han (1997) investigated the heat transfer and friction characteristics of channels with ribbed turbulators. Different rib shapes and orientations are investigated by Chandra et al. (1988), Han et al. (1991), Taslim et al. (1996) and Liou et al. (1996). Various rib types like perforated ribs (Hwang and Liou, 1995; Kukreja and Lau, 1996), ribbed-grooved wall combinations (Zhang et al., 1994), rib-vortex generator combinations (Myrum et al., 1996) have also been investigated.

Pin fin banks are arrays of short cylinders and they are used as heat transfer augmentation devices by increasing the internal wetted surface and the passage flow turbulence. In-line and staggered arrays of pin fins are generally used in flow passages. Considerable amount of study has been done on pin fin research. The effects of various parameters on heat transfer and pressure loss has been investigated. Different geometrical parameters as pin height and pin spacing (Van Fossen, 1982), the local and array averaged heat transfer (Simoneau and Van Fossen, 1984, Metzger et al. 1986), pin fin channels with trailing edge ejection holes (Lau et al., 1989), wall contribution to heat transfer (Al Dabagh and Andrews, 1992), perpendicular flow entry (Chyu and Natarajan, 1992) are some of the research topics in the previous years. The heat transfer characteristics of split pin fin arrays are also investigated by Mwangi and Kim (1994). Other than pin-fins and trip strips, some other heat transfer augmentation devices like hemispherical cavities (Schukin et al., 1995), baffles (Habib et al., 1994) and vortex chambers (Glezer et al., 1996) are also investigated during the past years.

The objective of this study is to introduce a new concept for turbulent heat transfer augmentation in gas turbine blades. This new cooling component is called an "oscillator fin" and is based on using a fluidic oscillator instead of conventional cylindrical pin fins. Fluidic oscillators have been used for mass flow rate measurements (Bauer 1980, Bauer 1981). However their application as a turbulent heat transfer augmentation device has never been investigated before.

The present research is a proof of concept study in which two dimensional, incompressible and unsteady solutions of Reynolds Averaged Navier Stokes equations are obtained for an oscillator fin and for an equivalent conventional cylindrical pin fin and the results are compared. The simulations are performed for a Reynolds number of 30000 which lies in the operating regime associated with turbine blade cooling (Armstrong and Winstanley, 1988). This Reynolds number is based on the cylinder diameter for the pin fin case and on the fin width for the oscillator fin case which are taken to be equal. A standard k-ε turbulence model coupled with an Algebraic Reynolds Stress Model is used for turbulence modeling. The turbulent kinetic energy levels in the wakes of the two bodies and the total pressure loss levels are compared using the results of the computer simulations. The present paper summarizes the initial phase of an extensive program that includes fluid mechanics and heat transfer experiments designed as a result of the current proof of concept study.

OSCILLATOR FIN CONCEPT

Oscillatory motion of an impinging jet over a concave wall is the main physical phenomenon in many past flowmeter designs. The present study incorporates this known fluid instability into the proposed oscillator fin designed for gas turbine heat transfer augmentation. The device consists of three separate members (Fig. 1). Two of the members which are of elliptical cross-section are placed

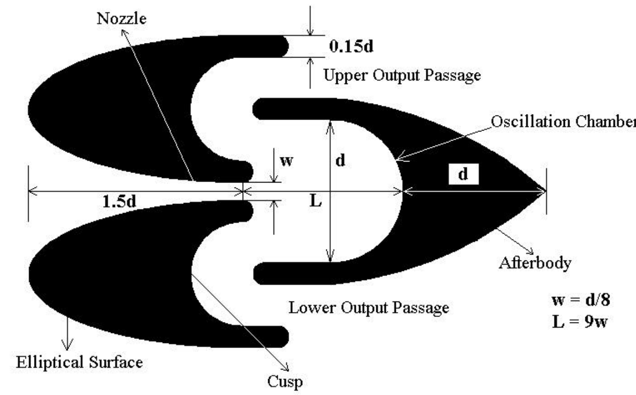


Figure 1. Oscillator Fin Geometry

transversely across the passage with the major axis of the ellipse parallel to the flow direction. These two members form a tapering nozzle which is used to create a jet between them. The downstream ends of the members are defined as downstream facing cusps which will be used for directing the oscillating jet into the flow. The third member which receives the jet from the nozzle is called the oscillation chamber and has an upstream facing U shaped geometry. The relative dimensions of the oscillator fin are also given in Fig. 1.

Water table flow visualization experiments have been instrumental during the conceptual development phase of the oscillator fin in deciding about the final aerodynamic shape described in Fig.1. During the operation of the oscillator fin, the jet coming from the nozzle impinges against the wall of the oscillation chamber and splits into two oppositely directed flows. This split-off process produces two vortices on either side of the jet (Fig.2a) which is a highly unstable condition. This unstable balance between two vortices causes increase in one of the reverse flow loops and one of the vortices tends to get stronger and moves closer to the center of the chamber (Fig. 2b). This movement of the stronger vortex deflects the incoming jet towards the side of the weaker vortex and directs the incoming flow along its periphery to out of the chamber. In the mean time weaker vortex is forced to get closer to the output passage and almost blocks the passage (Fig. 2c). At this moment most of the incoming flow is directed out from one of the outflow passages while the other outflow passage is almost blocked. As the weaker vortex gets closer to the outflow passage it starts to move closer to the nozzle and starts to receive fluid at much higher velocities. As a result of this, the weaker vortex begins spinning faster, its dominance increases among the two vortices and starts to move closer to the center of the chamber (Fig. 2d, Fig. 2e). The cycle is complete when the two vortices achieve side-by-side positions on the opposite side of the incoming jet (Fig. 2a).

As the jet created between the two front members is swept back and forth inside the oscillation chamber by the mechanism explained above, alternating flow pulses are formed and directed by cusps into the main flow. The effect of mixing of these alternating flow pulses coming from the oscillator fin with the vortices shed from the downstream part of the oscillation chamber is expected to increase the turbulence levels, mixing and unsteadiness in the wake region which will in turn increase the heat transfer. The effect of increased heat transfer due to increased unsteadiness created by the vortex shedding

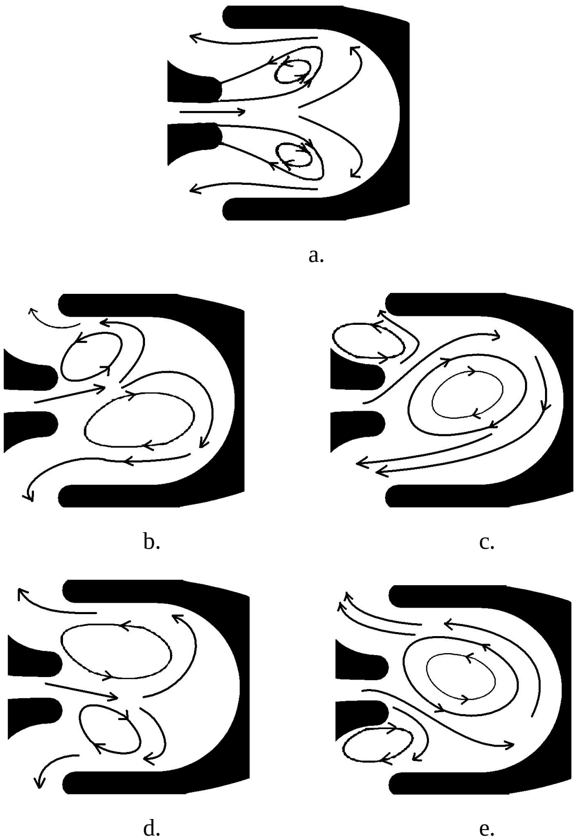


Figure 2. Jet Oscillation Mechanism Inside the Oscillation Chamber

from an immersed body in a channel flow has been studied by Suzuki and Suzuki (1994) and Valencia (1995). Xie and Wroblewski (1997) investigated the effect of vortex shedding from a cylinder on heat transfer in a turbulent boundary layer and concluded that the large scale periodic fluctuations may contribute to the mixing and wall heat transfer enhancement in the wake region of the circular cylinder. Therefore in this study the vortex shedding, turbulence levels and pressure loss levels in the wake region of a circular cylinder and an oscillator fin will be compared because these are the mechanisms that determine the heat transfer rates in the domain. Furthermore, due to the specific geometry of the oscillator fin, the wetted area is also increased. This increase in the wetted area and the periodic sweeping of the jet inside the oscillation chamber are also expected to increase the heat transfer on the oscillator fin itself.

SIMULATION TECHNIQUE

Governing Equations

The computational simulations of the flow fields for the oscillator fin and the pin fin cylinder are obtained by solving two dimensional, incompressible, unsteady solutions of Reynolds Averaged Navier Stokes equations. A two equation standard k-ε turbulence model is used for the simulation of the turbulent flow field. Hence the governing equations for the flow field are,

$$u^*_{i,i} = 0 \tag{1}$$

$$\frac{\partial u_i^*}{\partial t^*} + u_j^* u_{i,j}^* = -p_{,i}^* + \frac{1}{Re}\left[\mu^*\left(u_{i,j}^* + u_{j,i}^*\right)\right]_{,j} \tag{2}$$

$$\frac{\partial k^*}{\partial t^*} + u_j^* k_{,j}^* = \left(\frac{\mu_t^*}{\sigma_k} k_{,j}^*\right)_{,j} - \varepsilon^* + \mu_t^* \Phi^* \tag{3}$$

$$\frac{\partial \varepsilon^*}{\partial t^*} + u_j^* \varepsilon_{,j}^* = \left(\frac{\mu_t^*}{\sigma_k} \varepsilon_{,j}^*\right)_{,j} + c_1\left(\frac{\varepsilon^*}{k^*}\right)\mu_t^* \Phi^* - c_2 \frac{\varepsilon^{*^2}}{k^*} \tag{4}$$

For the calculation of the Reynolds stresses , instead of using Boussinesq's well established eddy viscosity model which lacks the ability to predict the anisotropic structure of turbulence, the eddy viscosity model proposed by Launder (1993) is used. This model represents the Reynolds stress tensor as a cubic function of the strain rate tensor and this provides the necessary mechanism for predicting turbulence anisotropy effects. This modified model is needed in order to accurately capture the turbulent structure of the complex wakes of the cylinder and the oscillator fin.

Solution Method

A finite element based fluid dynamics analysis package FIDAP (1993) is used to solve the governing equations. The flow domain is discretized by using nine-node quadrilateral elements which give a biquadratic velocity and bilinear pressure variation within each element.

Implicit backward Euler temporal formulation with a fixed time increment is used for time integration. At each time step a segregated solver is used to solve the non-linear system of equations. This implicit solver avoids the direct formation of a global system matrix which includes all the unknown degrees of freedom associated with the discretized problem as in fully coupled solvers. Instead it decomposes this matrix into smaller sub-matrices each governing the nodal unknowns associated with only one conservation equation. These smaller matrices are then solved in a sequential manner using different schemes. The use of the segregated solver is preferred because the storage requirements are substantially reduced compared to a fully coupled solver.

Boundary and Initial Conditions

The computations are performed for $Re = 30000$ based on the diameter (or oscillator fin width) and the inlet velocity. Velocity components are specified as zero on the walls and on the bodies in order to satisfy the no-slip condition. At the inlet the x component of the velocity is specified as a uniform steady profile and the y component is specified as zero. Values for the turbulent kinetic energy and for the dissipation rate of turbulent kinetic energy corresponding to a 0.045 % turbulence intensity level are also specified at the inlet. A low inlet turbulence level was chosen to illuminate the main unsteady fluid mechanics and turbulence generation character of the flow field during this proof of concept study. The implications of higher free stream turbulence at the inlet will be investigated during the ongoing experimental research program in the near future. No boundary conditions are explicitly imposed for velocity components, turbulent kinetic energy and the dissipation rate of turbulent kinetic energy at the outflow boundary. The specific form of the finite element solution procedure results in zero streamwise gradients for these variables at the exit plane. The unsteady solutions for the cylinder and for the oscillator fin are started from initial flow fields obtained by a steady solution of the governing equations.

SIMULATION RESULTS AND DISCUSSION

Physical Interpretation of Flowfield for the Cylinder

The simulation of the flowfield around the cylinder is performed for a Reynolds number of 30000 which is based on the inlet velocity and the cylinder diameter. Figure 3 shows the computational domain which starts at 5 diameters upstream and goes upto 8 diameters downstream of the cylinder. The channel walls are 2.5 diameter away from the centerline of the channel.

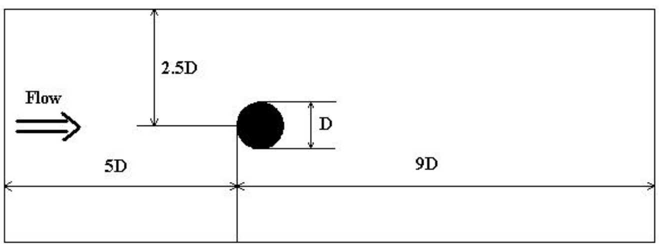


Figure 3. Computational Domain Definition - Cylinder

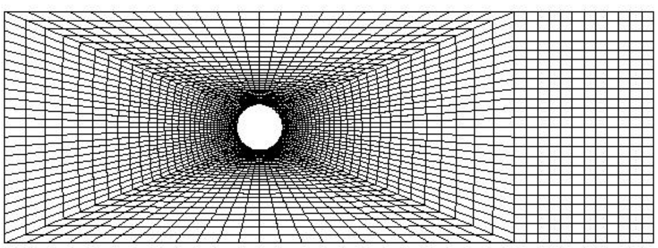
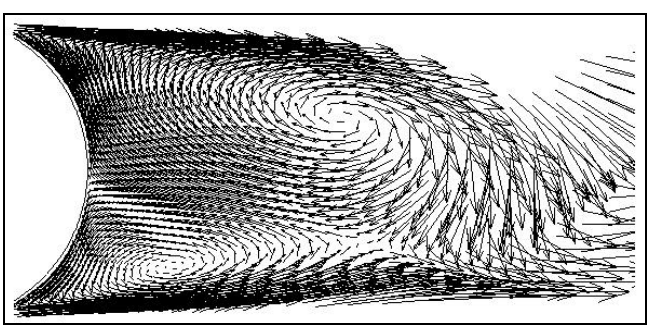
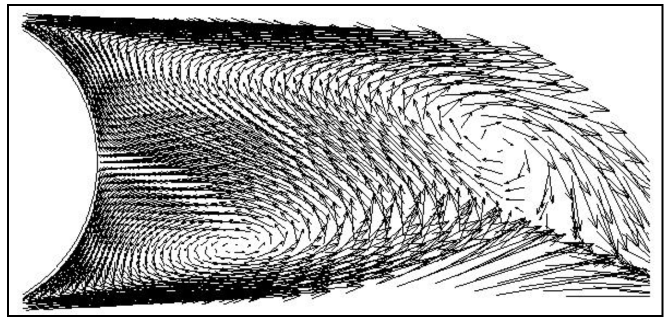


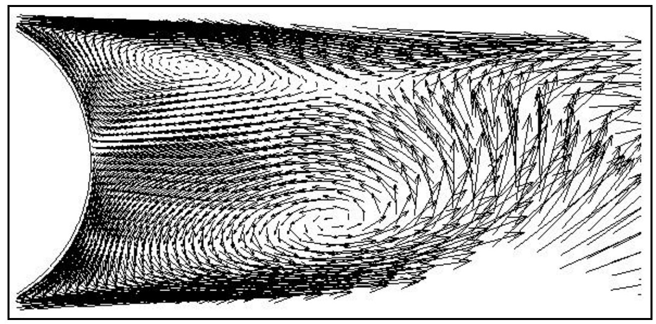
Figure 4. Computational Mesh - Cylinder



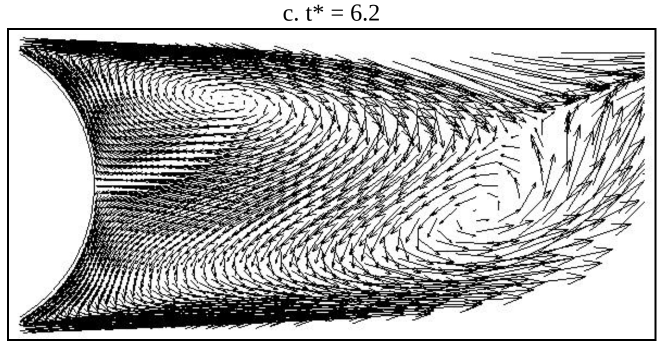
a. t* = 4.6



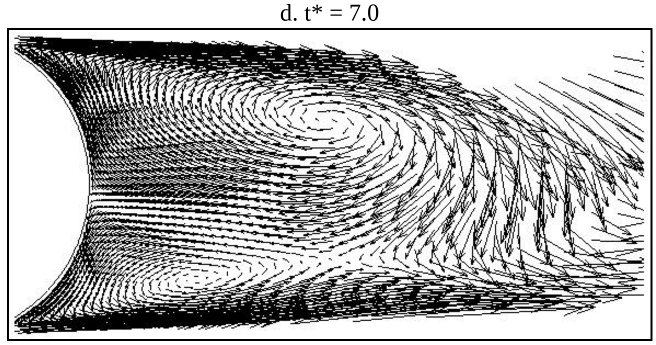
b. t* = 5.4



c. t* = 6.2



d. t* = 7.0



e. t* = 7.8

Figure 5. Vortex Shedding from the Cylinder

Nine-node quadrilateral elements are used to discretize the domain. For the cylinder case 3744 second order finite elements are created which resulted in 14424 number of nodes. The computational mesh is illustrated in Fig. 4.

Figure 5 shows the velocity vectors in the near wake region of the cylinder at different time steps. The periodic vortex shedding process is clearly seen from these figures. Alternating vortices are generated from the separation points on the upper and lower sides of the cylinder and convected downstream inside the wake region.

Time history of the static pressure in the wake region at one diameter downstream of the cylinder is shown in Fig. 6. The periodic variation due to the periodic vortex shedding can be seen from the figure. The Fast Fourier transform is applied to obtain the frequency content of this variation and the result is presented in Fig. 7 as a power spectrum plot. The value of the dominant non-dimensional frequency, or the Strouhal Number, is found to be 0.27. The experimental Strouhal number for a circular cylinder and for a Reynolds number of 30000 is known to be around 0.2 (Schlichting, 1955).

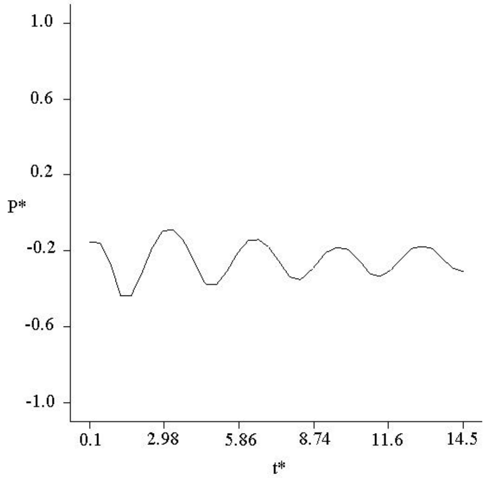


Figure 6. Time History of Static Pressure

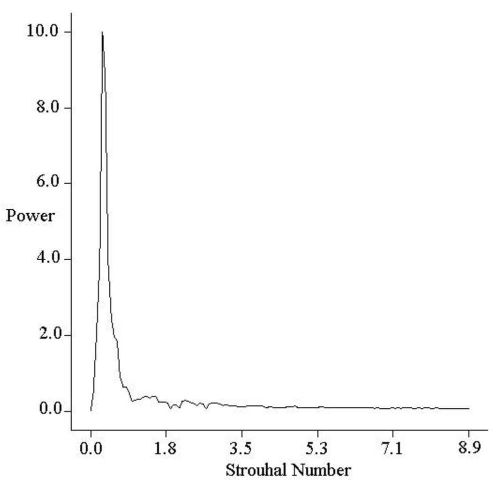


Figure 7. Power Spectrum of Static Pressure

Physical Interpretation of Flowfield for the Oscillator Fin

The solution of the governing equations for the flow field around the oscillator fin is also performed for a Reynolds number of 30000 which is based on the width of the oscillator fin and the inlet velocity. The width of the oscillator fin is the same as the diameter of the cylinder. The computational domain (Fig. 8) is also the same as in the cylinder case which starts at 5D upstream and ends at 8D downstream of the oscillator fin The domain is discretized by using nine-node second order quadrilateral elements and 3258 elements are created as a result of the discretization process (Fig.9). Total number of nodes is 12200.

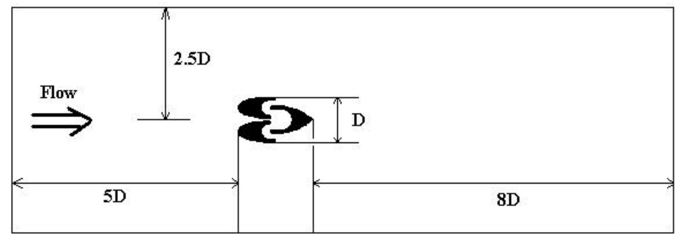


Figure 8. Computational Domain Definition - Oscillator Fin

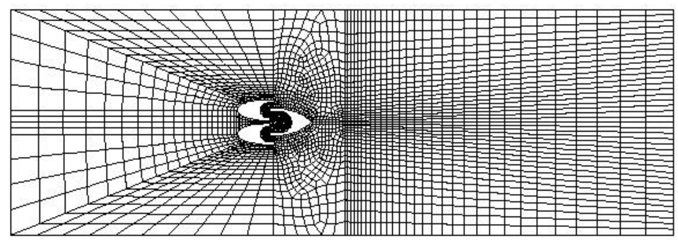


Figure 9. Computational Mesh - Oscillator Fin

The results presented for the oscillator fin are the results between the time at which the jet impinging on the concave wall of the oscillation chamber is not deflected (symmetrical with respect to the centerline) and the time at which the jet has its maximum deflection upwards. Figure 10 shows the jet deflection starting from the centerline and deflecting towards the upper lip of the oscillation chamber. The generation of the vortices inside the chamber and their respective movements as described in the conceptual explanation part is seen from the figure. As the jet moves upwards the lower vortex (L) gets bigger and situates at the center position while the upper vortex (U) is pushed towards the upper output passage and blocks the passage. In the mean time the jet coming from the nozzle is deflected upwards between these two vortices and most of the incoming flow is going out through the lower output passage when the jet is at its maximum deflected position. The impingement point of the jet on the concave wall of the oscillation chamber continuously oscillates in the current design. One complete period for the jet inside the chamber is approximately 6 times larger than the period of wake oscillation in an equivalent cylindrical pin fin. This jet oscillation is also observed in water table flow visualization experiments.

Figure 11 shows the vortex shedding from the downstream part of the oscillation chamber as the jet in the oscillation chamber is deflected from centerline towards the upper lip. When the jet is at the

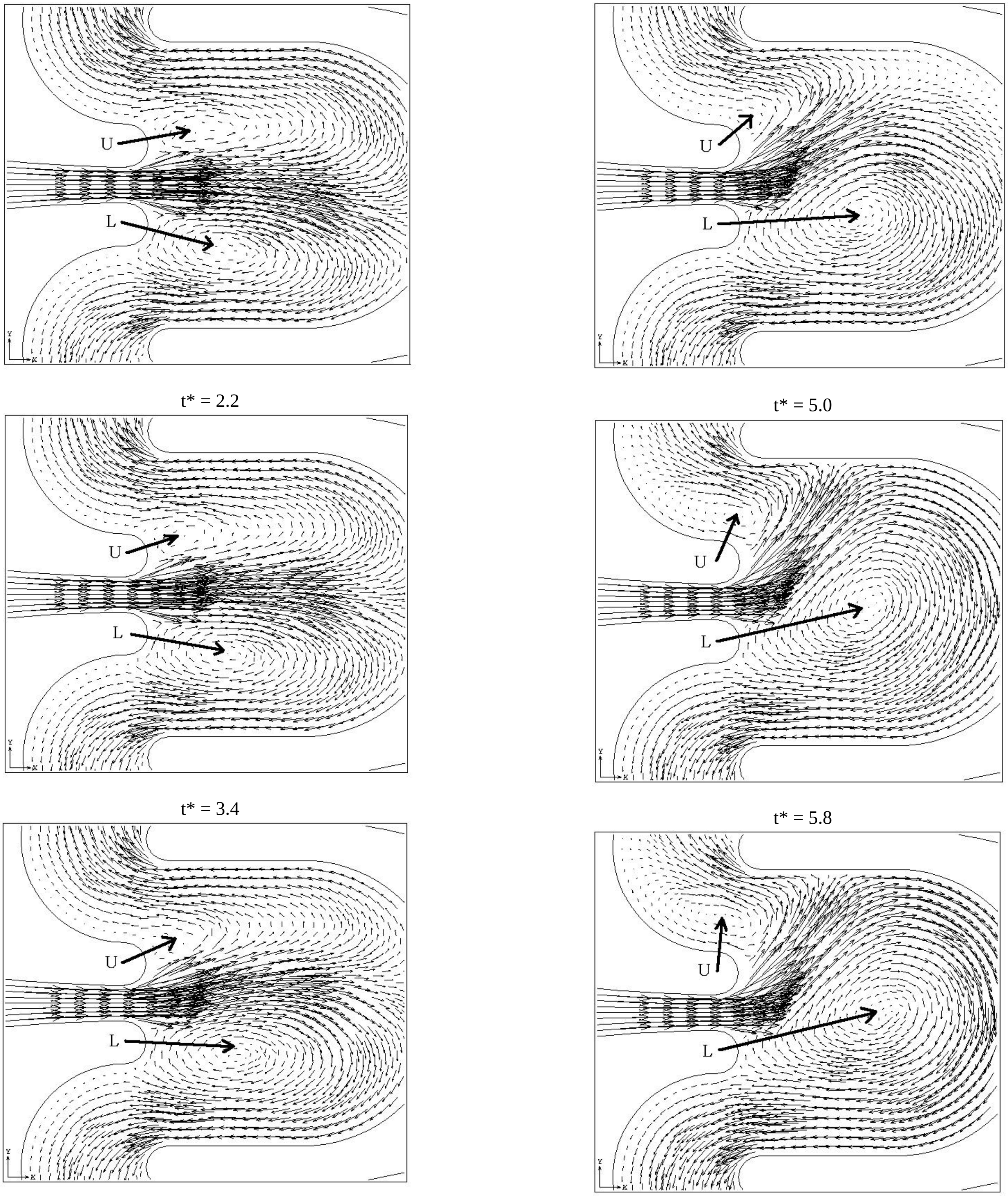
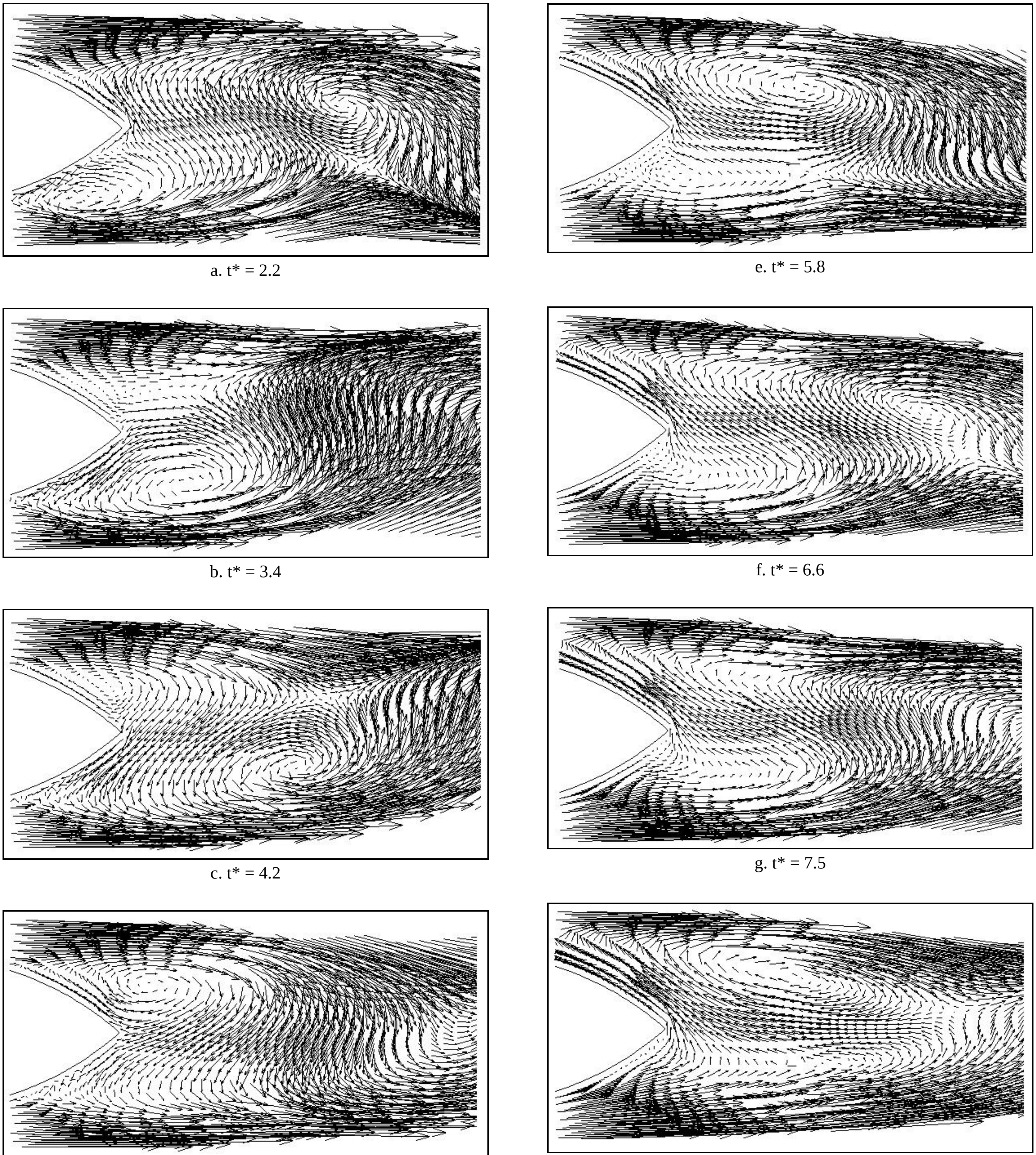


Figure 10. Jet Oscillation Inside the Oscillation Chamber



a. t* = 2.2

e. t* = 5.8

b. t* = 3.4

f. t* = 6.6

c. t* = 4.2

g. t* = 7.5

d. t* = 5.0

h. t* = 8.3

Figure 11. Vortex Shedding From the Oscillator Fin

symmetrical position, the separation points on the afterbody are almost symmetrical. While alternating vortices are shed from the chamber and convected downstream (Fig. 11a-11d), the jet inside the chamber is being deflected upwards and the momentum of the flow coming from the lower output passage of the oscillator fin is slowly increasing. That's due to the fact that the upper flow passage is blocked by the weaker vortex and the stronger vortex delivers the incoming jet through the lower output passage. As a result, the lower separation point is slowly pushed further downstream and at the same time the upper separation point is pulled towards the upper output passage. The lower vortex which would be created and shed if the jet was symmetrical, is now swept away by the flow coming from the lower output passage and mixes with the flow. At the same time, another big vortex is now being formed, again on the upper side, due to very early separation and is about to be convected downstream (Fig. 11e-11h). When the jet inside the oscillation chamber starts to return to its original position, the upper and lower separation points once again will start to move and eventually they will return their original symmetrical position. As soon as the jet starts to deflect downwards, this time the separation points will start to move in the opposite way that they did before. Namely, the upper separation point will be pushed backwards now and the lower separation point will be pulled towards the lower output passage.

Comparison of the Cylinder with the Oscillator Fin

The turbulent kinetic energy level in the flow domain is an indication of the generated turbulence which in turn determines the heat transfer characteristics of the system. Therefore an increase in the turbulence level will directly affect the heat transfer rate. However an increase in the turbulence level is generally accompanied with an increase in total pressure loss level. For these reasons the turbulent kinetic energy and the total pressure loss levels for the cylinder and for the oscillator fin are compared.

In order to compare the turbulent kinetic energy level in the flow domain, an integrated mean value for the whole domain is obtained using,

$$k^*_{AV} = \frac{\int_A k^* dA}{\int_A dA} \qquad (5)$$

where A is the area of the computational domain. The time history of the mean turbulent kinetic energy is presented in Fig.12. It can be seen that the mean turbulent kinetic energy almost remains constant as time progresses both for the cylinder and for the oscillator fin. However the turbulent kinetic energy level of the oscillator fin is almost 65 % higher than that of the cylinder. This increase in the turbulent kinetic energy level may be due to the mixing of the free stream with the highly unsteady flow coming out of the upper and lower output passages. The turbulent kinetic energy contours for the whole domain both for the cylinder and the oscillator fin are also shown in Fig. 13 for various time steps. Unsteady sweeping motion of the jets coming out of the output passages interact with the core flow between the oscillator fin and sidewalls and wake flow downstream of the afterbody. Because the streamwise momentum of the unsteady

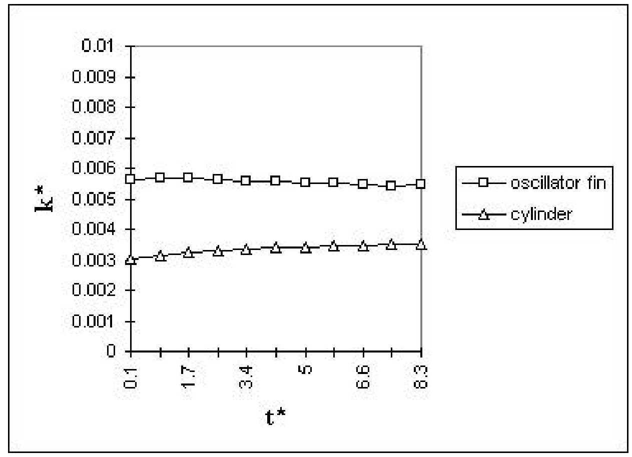


Figure 12. Time History of the Mean Turbulent Kinetic Energy over the Whole Domain

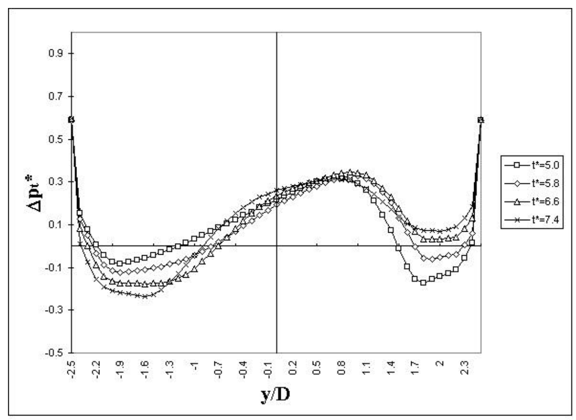


Figure 14. Non-dimensional Total Pressure Loss Distribution for Various Time Steps at the Exit Plane for Cylinder

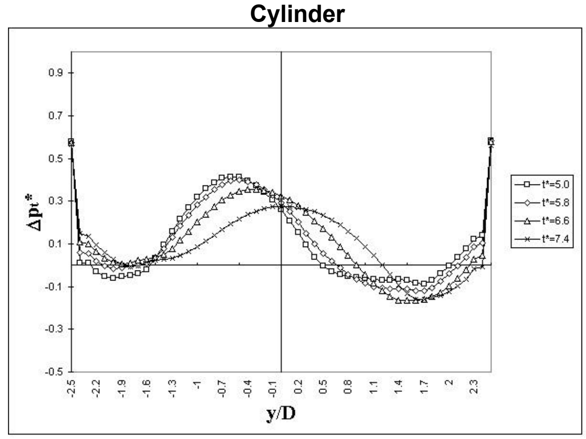
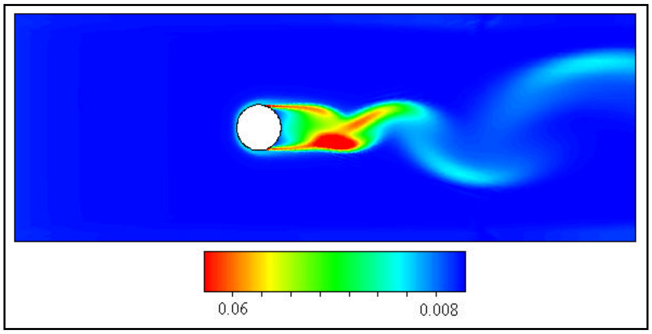
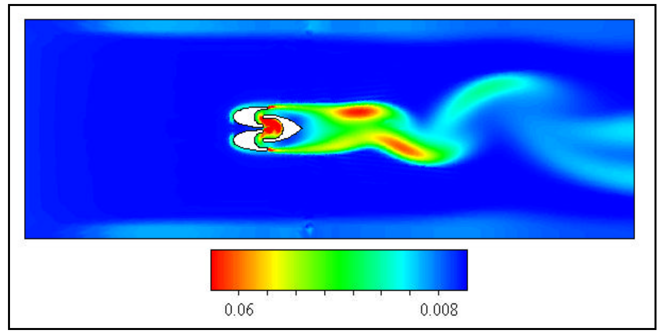


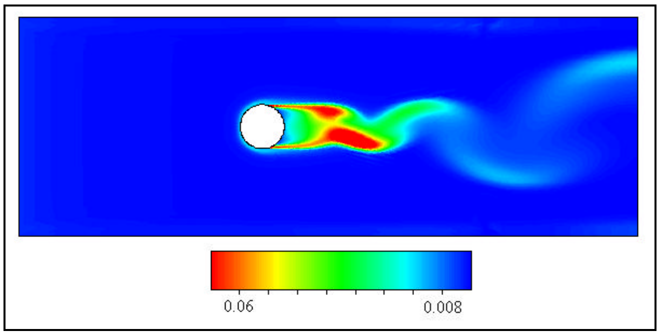
Figure 15. Non-dimensional Total Pressure Loss Distribution for Various Time Steps at the Exit Plane for Oscillator Fin



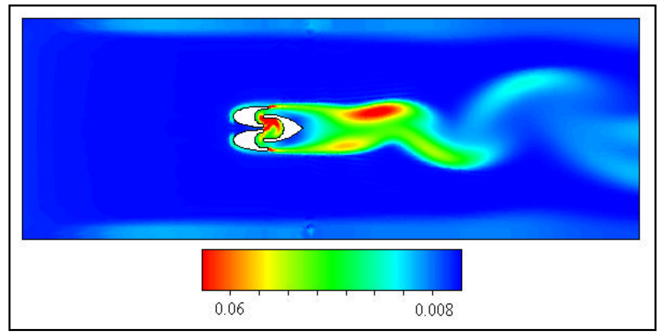
t* = 5.0



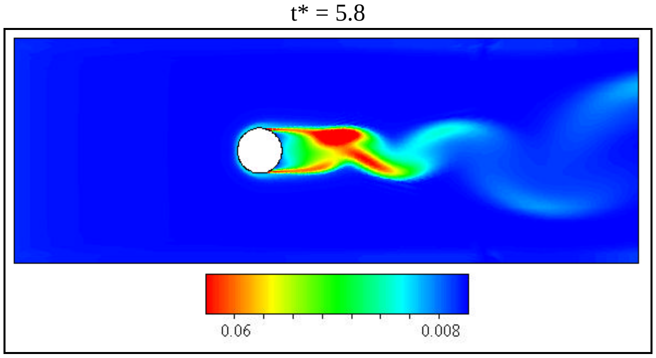
t* = 5.0



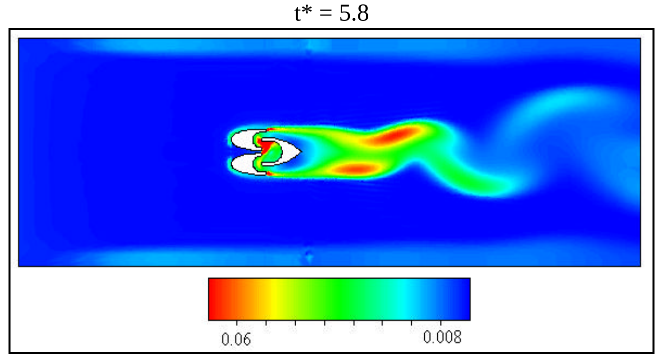
t* = 5.8



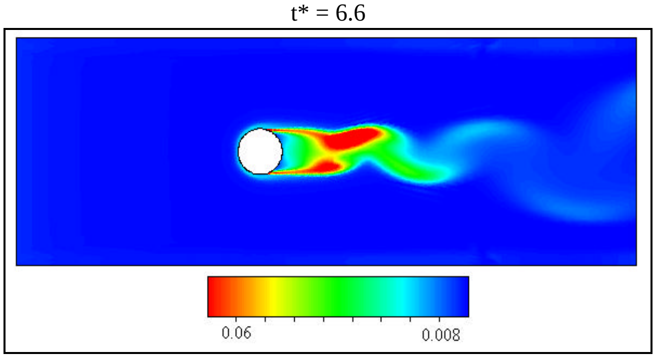
t* = 5.8



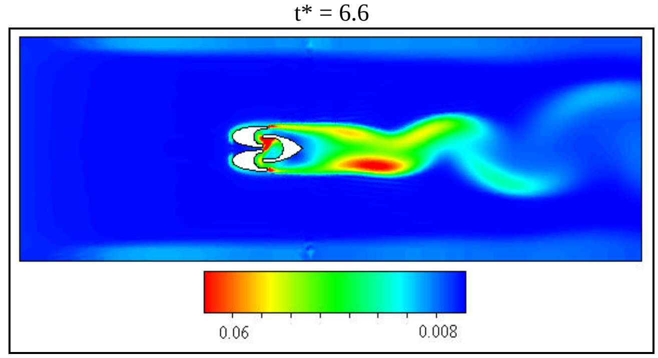
t* = 6.6



t* = 6.6



t* = 7.4



t* = 7.4

- = 7.4

Figure 13. Turbulent Kinetic Energy Contours, k*

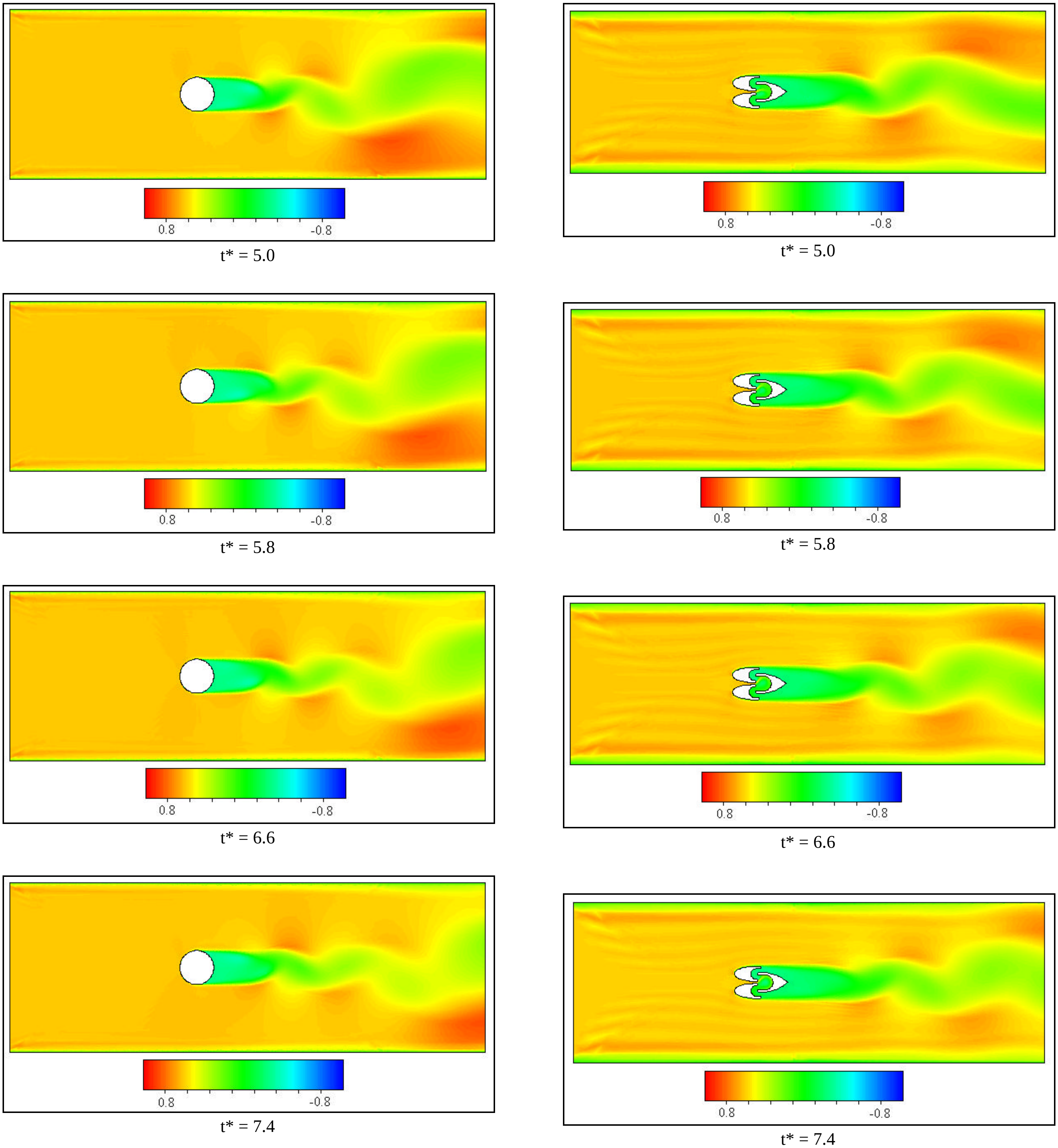


Figure 16. Total Pressure Coefficient Contours, p_t*

jets from the output passages vary in time significantly, the interaction of the core flow with the wake flow is very strong. Local turbulence enhancements are also carried into a wider downstream area via unsteady sweeping nature of the present flowfield. The major turbulence enhancement area occupies a much longer region downstream of the oscillator fin as indicated by red, yellow and green zones in Fig. 13.

The distribution of the non-dimensional total pressure loss at the exit plane for various time steps is given in Fig. 14 for the cylinder and Fig.15 for the oscillator fin. The total pressure contours for the whole computational domain are presented in Fig. 16 for the same time steps. The qualitative numerical results show that the aerodynamic penalty level is at the same order of magnitude both for the oscillator fin and the cylinder. Δp_t* values shown in Fig. 17 are area averaged only at the exit plane for comparison. Detailed experimental investigation of total pressure loss in order to get some quantitative results is under progress.

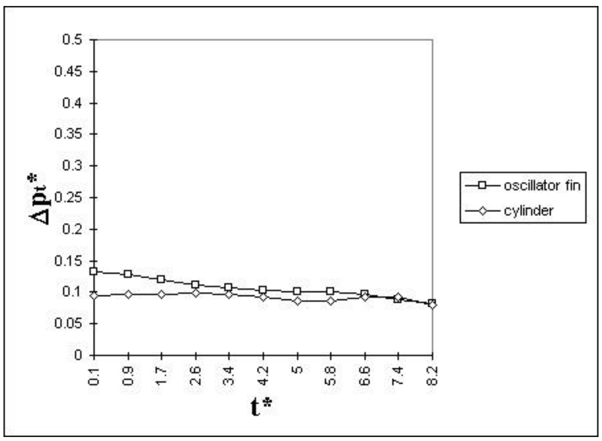


Figure 17. Time History of the Area Averaged Non-Dimensional Total Pressure Loss at the Exit Plane

CONCLUSIONS

A new concept, oscillator fin, is introduced for the enhancement of turbulent heat transfer in the coolant passages of gas turbine blades. The flow fields for the oscillator fin and for the cylinder are obtained by the solution of the Reynolds Averaged Navier-Stokes equations. The computational results are used to compare the turbulent kinetic energy and total pressure loss levels in the flow domains of the two configurations. The results show that the oscillator fin has a great potential in terms of increasing the turbulence levels in the flow when compared to the cylinder. Also the aerodynamic penalty levels are at the same order of magnitude both for the oscillator fin and the cylinder. Due to the alternating pulses of flow coming out of the oscillator fin, the unsteadiness levels in the flow domain are expected to increase. Increased deterministic unsteadiness will significantly enhance the heat transfer over conventional pin fin levels.

This study is the initial phase of an extensive convective heat transfer experimental research program including turbulent flow investigations. This program is currently under progress in parallel to present simulations in the Turbomachinery Heat Transfer Laboratory for analyzing the flowfield of the oscillator fin in detail. The results obtained in this study is mainly used for flow visualization purposes in order to understand the overall features of the unsteady flowfield and to introduce the potential in using the fluidic oscillators as heat transfer enhancement devices. Different oscillator fin geometries are also being investigated as the one shown in Fig.18 in which the fluidic oscillator concept is embedded in a cylinder with exit jets inclined from the axial direction.

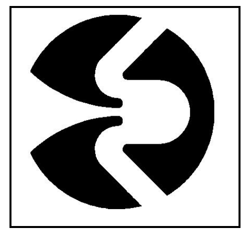


Figure 18. Pac Fin Geometry

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