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SIMULATION OF TWO DIMENSIONAL FLOW AND CONJUGATE HEAT TRANSFER PROBLEM IN COOLED GAS TURBINE NOZZLE GUIDE VANE

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ABSTRACT

The coupled treatment (conjugate numerical methodology) allows the simultaneous solution of the external flow (steady, two dimensional, compressible and turbulent flow) and conduction within the metal (steady, two dimensional) of gas turbine nozzle guide vane (with and without internal convection cooling). Validation of the developed conjugate capability is investigated in the present work.

The numerical results were compared with experimental results for steady, two dimensional, compressible and turbulent flow through the gas turbine nozzle cooling guide vane type (NASA-C3X), and the results were found to be in good agreement with experiments by **(Hylton 1983)**.

The study shows that the (conjugate numerical methodology) gives good and more accurate results than the un-coupled treatment. It also shows that the cooling of the vane reduced the thermal stresses which are focused in the trailing edge of the vane for being thin. Moreover, the cooling flow inside the passages of the vane reduced the temperature of vane body and that gives longer life to the vane for the same Turbine Inlet Temperature (TIT) and efficiency. Otherwise, it gives higher Turbine Inlet Temperature and high efficiency if one chose to keep same life of the vane.

Finally, the present study shows that the conjugate heat transfer simulation is a good tool in gas turbine design, and it serves as base future work with more complex geometries and cooling schemes for turbine blade.

الخلاصة:

تم في هذا البحث استخدام (الطريقة العددية المزدوجة) لتحقيق الازدواجية بين حقل الجريان الخارجي(للحالة المستقرة الثنائية الابعاد والانضغاطية وللجريان المضطرب) و الجزءالصلب(للحالة المستقرة الثنائية الابعاد) لريشة التوربين الموجهة للجريان المبردة داخليا بالحمل وغير المبردة. ان النتائج العددية لهذة الدراسة قد قورنت مع نتائج عملية للحالة المستقرة الثنائية الابعاد والانضغاطية وللجريان المضطرب خلال ريشة التوربين الموجهة للجريان والمبردة ومن نوع (NASA-C3X) ، وقد وجدت النتائج بانها ذات توافق جيد مع الحالة العملية بواسطة **(Hylton 1983)**.

لقد بينت الدراسة بان (الطريقة العددية المزدوجة) تعطي نتائج اكثر دقة من المعالجة غير المزدوجة وكذلك بينت الدراسة بان التبريد للريشة يقلل من الاجهادت الحرارية المتمركزة في نهاية الريشة لكون ريشة التوربين في هذه المنطقة تكون نحيفة،الى جانب ذلك فأن الجريان الداخلي المبرد للريشة يسبب نقصان في درجة حرارة جسم الريشة وهذا ما يؤدي الى زيادة عمر الريشة

ولنفس درجة حرارة الدخول الكلية للريشة و كفاءة المحرك التوربيني، وكذلك من الممكن زيادة درجة حرارة الدخول الكلية للريشة وبالتالي زيادة كفاءة المحرك التوربيني و لنفس عمر الريشة.

واخيرا، بينت الدراسة الحالية بان (الطريقة العددية المزدوجة) هي اداة جيدة في تصميم المحرك التوربيني، وهي تخدم العمل المستقبلي مع الاشكال المعقدة وطرق التبريد الاخرى لريش التوربين.

INTRODUCTION

Advanced gas turbine engines operate at high temperatures (1200-1400°C) to improve thermal efficiency and power output. As the Turbine Inlet Temperature (TIT) increases, the heat transferred to the turbine blades also increase. The level and variation in the temperature within the blade material (which causes thermal stresses) must be limited to achieve reasonable durability goals. The operating temperatures are far above the permissible metal temperatures. Therefore, there is a need to cool the blades for safe operation. The blades are cooled by extracted air from the compressor of the engine. Since this extraction incurs a penalty to the thermal efficiency, it is necessary to understand and optimize the cooling technique, operating conditions, and turbine blade configuration.

The continuous increase in turbine inlet pressure and temperature definitely require reliable and accurate predictions of the main stream aerothermal characteristics and of the heat loads imposed to the blades so that a good design from a thermal point of view might allow a higher inlet temperature, less cooling air or a lighter design, thus increasing the performance or efficiency of the turbine and resulting in a longer engine life **(Qingluan Xue 2005)**.

In several industrial applications it is becoming necessary to accompany the computation of flow and associated heat transfer in the fluid with the heat conduction within the adjacent solid. The coupling of these two models of heat transfer has been identified by the name “conjugate heat transfer” in the relevant literature. Typical applications where conjugate heat transfer effects can become important are, among others, the cooling of turbine blade, and cooling of electronics.

A conjugate numerical methodology used to predict the metal temperature of two or three-dimensional gas turbine stator blade (vane) or rotor blade. The conjugate heat transfer approach allows the simultaneous solution of the external flow, internal convection, and conduction within the metal vane or blade, eliminating the need for multiple, decoupled solutions, which are time-consuming and inherently less accurate when combined **(Canelli et al 2003)**.

AIM OF WORK

Predict the flow and the metal temperature through gas turbine nozzle guide vane (with and without cooling) of an axial flow turbine stator by using the fully-coupled approach (conjugate numerical methodology) by FLUENT code.

MATHEMATICAL MODEL

Assumptions

In order to adopt an applicable computational method, some assumptions are made:

- The entering fluid flow is a perfect gas, Newtonian, uniform, turbulent, and compressible.
- The fluid field is steady, adiabatic, ir-rotational, single-phase, and shock free.
- Cartesian coordinate system is used.

<u>Domain Description</u>

The computational domain and the boundary conditions (B.C) that considered in the present work are shown in figure (1):

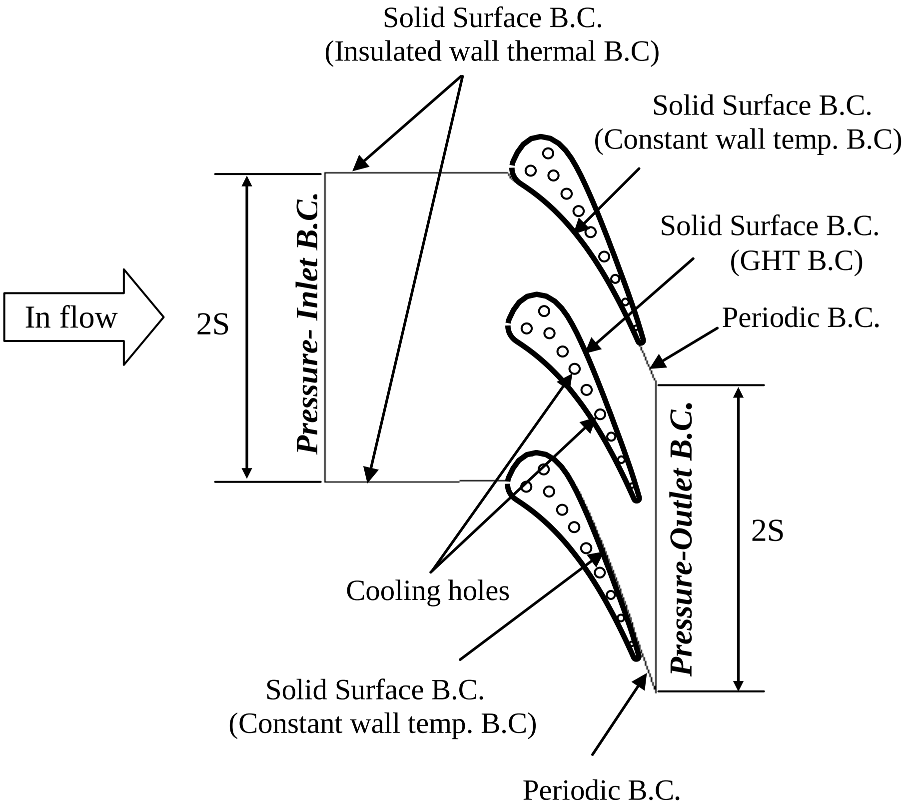

Fig. (1): Computational domain and the boundary conditions of the flow through gas turbine nozzle guide vane with cooling.

The compressible turbulent flow through gas turbine nozzle guide vane with and without cooling was predicted by solving numerically the two dimensional continuity and Navier-Stokes equations. Unstructured (triangle element) grid generation is adopted by using (GAMBIT code). Finite volume method with differencing scheme for diffusion term and up-wind scheme for convection term is employed to discretize the flow domain in the relevant transport equations. A SIMPLE algorithm pressure based method with collocated grid arrangement and **(Rhie and Chow 1983)** interpolation are employed to find the coupling of the pressure-velocity field. The Cartesian velocity components are used as the main dependent variables for the momentum equations. The (k-ε) turbulence model is used to close up the system of the momentum differential equations. In order to account for the compressibility effects, an equation of state and up-winding scheme is used to interpolate the density at control volume faces. Discretizing the governing equations (continuity, momentum, and energy) of fluid flow results in a system of linear algebraic equations, which are solved by Algebraic Multi-Grid Solver (AMGS). The fully-coupled approach between the flow and the solid vane was used. Conjugate heat transfer B.C for the external wall surface (for the solid vane) and constant wall temperature for internal convection cooling holes was used, and another type of B.C for the computational domain is shown in fig. (1). The Fourier equation for heat diffusion was solved in the solid zone (solid vane). The present simulations were run using the FLUENTTM 6.1.18 code from FLUENT INC., Inc.

Governing Equations

The general form of three-dimensional instantaneous governing equations of mass and momentum for unsteady, viscous, and compressible flows written in tensor conservation form expressed in Cartesian coordinates system **(Wang and Komori 1998)**.

Continuity Equation:

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_j}\left(\rho u_j\right)=0 \text{ -------------------------------- } (1)$$

Momentum Equation:

$$\frac{\partial}{\partial t}\left(\rho u_i\right)+\frac{\partial}{\partial x_j}\left(\rho u_i u_j\right)=-\frac{\partial p}{\partial x_i}+\frac{\partial t_{ij}}{\partial x_j} \text{ ------------------------ } (2)$$

Where t_{ij} is the viscous stress tensor, defined by: -

$$t_{ij}=\mu\left[\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\frac{2}{3}\frac{\partial \overline{u_k}}{\partial x_k}\delta_{ij}\right] \text{ -------------------------- } (3)$$

The Kronecker delta (δ_{ij} =1 if i=j and δ_{ij} =0 if i≠j)

Energy Equation:

$$\frac{\partial}{\partial x_j}\left(\rho u_j\left(h+\frac{u_i^2}{2}\right)-p\right)=\frac{\partial}{\partial x_j}\left(u_i t_{ij}\right)-\frac{\partial}{\partial x_j}\left(ql_j\right) \text{ ------------------ } (4)$$

Where, h is the specific enthalpy. ql_j is the laminar heat flux vector, which is obtained from Fourier's law. This is:-

$$ql_j=-K\left(\frac{\partial T}{\partial x_j}\right)=-\left(\frac{\mu C_p}{\Pr_l}\right)\frac{\partial T}{\partial x_j} \text{ -------------------------- } (5)$$

K is the thermal conductivity, and C_p is the specific heat capacity at constant pressure. $\Pr_l$, is the molecular Prandtle number defined by:-

$$\Pr_l=C_p\frac{\mu}{K} \text{ -- } (6)$$

State Equation:

$$P=\rho RT \text{ --- } (7)$$

Reynolds-Averaged Navier-Stokes Equations

The turbulence model used in this study is based on averaging of the instantaneous equations. In order to time and mass average (Reynolds averaged) the conservation equations the various flow properties are decomposed as follows: -

$$\left.\begin{aligned} u_i &= \overline{u_i} + u_i' \\ u_j &= \overline{u_j} + u_j' \\ p &= \overline{p} + p' \\ \rho &= \overline{\rho} + \rho' \\ T &= \overline{T} + T' \end{aligned}\right\} \text{------------------------------------} (8a)$$

Neglect the fluctuating terms the turbulence structure remains unchanged, so Equation (8a) become **(Wang and Komori 1998)**:

$$\left.\begin{aligned} u_i &= \overline{u_i} + u_i' \\ u_j &= \overline{u_j} + u_j' \\ p &= \overline{p} \\ \rho &= \overline{\rho} \\ T &= \overline{T} \end{aligned}\right\} \text{------------------------------------} (8b)$$

Substituting equations (8b) into equations (1), (2), (4) and (7).The time/ mass averaged mean equations of continuity, momentum, energy, and state can be written as **(Wang and Komori 1998)**:-

Continuity Equation

$$\frac{\partial \overline{\rho}}{\partial t} + \frac{\partial}{\partial x_j}\left(\overline{\rho u_j}\right) = 0 \text{---------------------------------} (9)$$

Momentum Equations

$$\frac{\partial}{\partial t}\left(\overline{\rho u_i}\right) + \frac{\partial}{\partial x_j}\left(\overline{\rho u_i u_j}\right) = -\frac{\partial \overline{p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\overline{t_{ij}} + \overline{\tau_{ij}}\right) \text{------------------} (10)$$

Where $\overline{\tau_{ij}}$ is the averaged Reynolds stress tensor, defined as: -

$$\overline{\tau_{ij}} = -\overline{\rho u_i' u_j'} \text{-----------------------------------} (11)$$

This will be determined from the Boussinesq Eddy Viscosity Model using the (*k-ε)* turbulence model **[Jones and Launder (1972)].** The Reynolds stress tensor will be: -

$$\overline{\tau_{ij}} = \mu_t \left[\left(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i}\right) - \frac{2}{3}\frac{\partial \overline{u_k}}{\partial x_k}\delta_{ij}\right] - \frac{2}{3}\overline{\rho k}\delta_{ij} \text{------------------} (12)$$

Where $\overline{k}$ is the averaged turbulent kinetic energy defined as: -

$$\bar{k} = \frac{1}{2}\overline{u_i' u_i'} \quad (13)$$

And μ_t is the turbulent eddy viscosity expressed as: -

$$\mu_t = \frac{C_\mu \overline{\rho k^2}}{\bar{\varepsilon}} \quad (14)$$

Where C_μ is constant ($C_\mu = 0.09$) **[Launder and Spalding (1974)]**. $\bar{\varepsilon}$ is the averaged dissipation rate of turbulent kinetic energy, defined as:-

$$\bar{\varepsilon} = \upsilon \overline{\frac{\partial u_i'}{\partial x_j}\frac{\partial u_i'}{\partial x_j}} \quad (15)$$

Energy Equation

$$\frac{\partial}{\partial x_j}\left(\bar{\rho} u_j'(H') - \bar{p}\right) = \frac{\partial}{\partial x_j}\left(u_i'(\bar{t}_{ij} + \tau_{ij})\right) + \frac{\partial}{\partial x_j}\left(-ql_j - qt_j + \left(\mu + \frac{\mu_t}{\sigma_k}\right)\bar{k}\right) \quad (16a)$$

Where:

$$H' = C_p \bar{T} + \frac{1}{2}\left(\overline{u_i' u_i'}\right) + \bar{k} \quad (16b)$$

qt is the turbulant heat flux vector defined as: -

$$qt = -\left(\frac{\mu_t C_p}{\Pr_t}\right)\frac{\partial \bar{T}}{\partial x_j} \quad (16c)$$

$\Pr_t$, is the turbulent prandtl number that has different value for different flow. For gases, most common values suggested is 0.9 in the case of the boundary layer **[Wang and Komori, (1998)]**.

State Equation

$$\bar{p} = R\overline{\rho T} \quad (17)$$

Turbulence Model (k-ε) Equations

The transport equations of k and ε are formulated and modeled as in **[Wang and Komori (1998)]**:

Equation of Turbulent Kinetic Energy *(k)*:

$$\frac{\partial}{\partial t}\left(\overline{\rho k}\right) + \frac{\partial}{\partial x_j}\left(\overline{\rho k u_j}\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \bar{k}}{\partial x_j}\right] + G_k - \overline{\rho\varepsilon} \quad (18)$$

Where (σ_k) is constant with value ($\sigma_k = 1$), G_k is the production of turbulent kinetic energy (generation term) defined as: -

$$G_k = \overline{\tau_{ij}} \frac{\partial \overline{u_i}}{\partial x_j} \text{------------------------------------} (19)$$

Substitution of ($\overline{\tau_{ij}}$) from equation (12) into equation (19) gives:

$$G_k = \mu_t \left[\left(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i} \right) \right] \frac{\partial \overline{u_i}}{\partial x_j} - \frac{2}{3} \frac{\partial \overline{u_i}}{\partial x_j} \delta_{ij} \left[\mu_t \frac{\partial \overline{u_k}}{\partial x_k} + \overline{\rho k} \right] \text{--------------} (20)$$

Equation of Dissipation of Turbulent Kinetic Energy *(ε)*:

$$\frac{\partial}{\partial t}\left(\overline{\rho\varepsilon}\right) + \frac{\partial}{\partial x_j} \overline{\rho\varepsilon u_j} = C_{\varepsilon 1} \frac{\overline{\varepsilon}}{k} G_k + \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \overline{\varepsilon}}{\partial x_j}\right] - C_{\varepsilon 2} \overline{\rho} \frac{\overline{\varepsilon^2}}{k} \text{--------} (21)$$

Where ($C_{\varepsilon 1}$), ($C_{\varepsilon 2}$) and (σ_ε) are empirical constants and they having the following values, [**Launder and Spalding (1974)**]: -

$C_{\varepsilon 1} = 1.44,$ $C_{\varepsilon 2} = 1.92,$ $\sigma_\varepsilon = 1.3$

Fourier equation for heat diffusion (solid vane)

$$ql_j = -K\left(\frac{\partial T}{\partial x_j}\right) \text{--------------------------------} (22)$$

Boundary Conditions Equations

Boundary conditions specify the flow and thermal variables on the boundaries of the physical model. The boundary conditions types that used in the present work are shown in figure (1) and the equations of these boundary conditions are formulated and modeled as in **(Veresteeg & Malalasekera, 1995)** and **(Launder and Spalding, 1974)**.

RESULTS AND DISCUSSIONS

The first view of geometry configuration and grid distributions are shown in Figs. (2) and (3). The computational domain has been discretized with 4362 nodes for fluid zone and 344 nodes for solid zone for gas turbine nozzle guide vane without cooling as shown in fig. (2), and 4362 nodes for fluid zone and 990 nodes for solid zone for gas turbine nozzle guide vane with internal convection cooling as shown in fig. (3), with unstructured grid generation, triangle elements by GAMBIT code from FLUENT.

The main stream operating condition was simulated for subsonic flow and the B.C set to give the same condition as in experimental work by **(Hylton 1983)**, where, Hylton reported average Mach number at the vane trailing edge plane at the midspan, and that Mach number was converted to average static pressure by using isentropic flow relations. The operating condition and the B.C for the simulation of the flow through gas turbine nozzle guide vane is shown in Table (1).The vane material properties is shown in Table (2). The thermal boundary on each internal cooling channel is explicitly specified temperature values based on experimental data by **(Hylton 1983)**. Table (3) lists the diameters and temperatures of each cooling hole. The hub and tip walls are moduled with no-slip and adiabatic conditions.

The velocity and Mach contour is shown in figure (4) and figure (5). In this figure (4), the flow speed sees high along the suction side near the leading edge. The maximum velocity in the vane passage is at the position about (25-30 %) from the suction side distance from the leading edge. In the pressure side, the velocity is low until (30-35 %) from the axial chord distance from the leading edge, and then the flow is accelerated toward the trailing edge. The contour of pressure is shown in figure (6). Figure (7) shows the temperature contour of the flow through gas turbine nozzle guide vane and inside the solid guide vane with cooling.

For the mainstream condition, the flow field is examined and then heat transfer. Fig. (8) and Fig. (9) shows the pressure distribution at wall of the vane midspan. On the suction side, the pressure falls rapidly from the stagnation line at the leading edge and then reaches to the minimum value at (25-30 %) from the axial chord length, and after that the pressure is increase to the trailing edge but in small range. On the pressure side the flow is still near to the total pressure (P°) from the leading edge until reached to (40-50 %) from the axial chord and then decrease toward the trailing edge. Fig. (10) and Fig. (11) shows the temperatures distribution at wall of the vane midspan (without and with cooling).

The results show a good agreement with experimental data by **(Hylton 1983)** as shown in figure (12) and figure (13)**.**

The heat transfer is investigated with and without cooling van. The temperature dimensionless distribution (T_{wall}/T^{o}) on the external surface at the midspan of the vane is plotted as shown in figure (12). This figure contains the comparison of the temperature distribution at the external surface of the vane with cooling that resulted by using (The Conjugate Methodology) with the experimental value by **(Hylton 1983)**. The results show a good agreement with experimental data.

Figure (13) shows the comparison between the coupled and decoupled treatment of the static pressure on the suction and pressure side on the wall of gas turbine nozzle guide vane with cooling, with the experiment values by **(Hylton, 1983)**.In this figure, the coupled treatment is more accurate than the decoupled treatment and that can be see clearly in suction side.

CONCLUSIONS

- Using unstructured grid generation, triangle elements by GAMBIT code from FLUENT with Cartesian velocities as the dependent variables and collocated grid arrangement gave an easy solution to the problem.
- Introducing the concept of collocated grid arrangement by **(Rhie and Chow 1983)**, with contravariant velocities implicitly in the discretized momentum equations simplifies the discretization of the flow equations.
- Conjugate numerical method is a good method and gives a good accuracy in prediction the temperature distribution over and inside gas turbine nozzle guide vane (NASA-C3X) without and with cooling where, the predicted mid-span temperature distribution on the vane external surface was in reasonable agreement with experimental data by **(Hylton 1983)**.

- The cooling method of the gas turbine nozzle guide vane by the internal convection circular cooling passages in the present work gives more life to the vane materials by decreasing the thermal stresses and for the same design life it gives more efficiency for the gas turbine by increasing the temperature at the inlet of guide vane (TIT).

- The metal temperature everywhere was much closer to the temperature in the passage free stream than the coolant temperature. This was due to the fact that the thermal resistance due to internal convection was much greater than the resistance of the external convection or the conduction within the metal.
- The results were found to be in a reasonable agreement with experiment by **(Hylton 1983)**.

NOMENCLATURE

Latin Symbols

Symbol	Description
Cp	specific heat capacity at constant pressure (J/kg K)
$C_\mu, C_{\varepsilon 1}, C_{\varepsilon 2}$	Constants in the k-ε model
G_k	Production term of kinetic energy
h	Enthalpy (J)
K	thermal conductivity
k	Turbulent kinetic energy
L_c	Characteristic length of turbulence
M	Mass flow rate (kg/s)
P	Pressure (Pa)
P°	Total Pressure (Pa)
$\Pr_l$	Molecular Prandtle number
$\Pr_t$	Turbulent prandtl number
ql_j	Laminar heat flux vector
qt	Turbulant heat flux vector
R	Gas constant (J/kg.K)
R_e	Reynolds number
S	Pitch or vane spacing
t_{ij}	Viscous stress tensor
T	Temperature (K)
T°	Total Temperature (K)
T_{wall}	Wall Temperature (K)
T_u	Turbulent intensity
X	Axial coordinate in the physical domain
Y	Pitchwise coordinate in the physical domain

Greek Symbols

Symbol	Description
ε	Dissipation rate of turbulent kinetic energy
μ	Laminar viscosity
μ_T	Turbulent eddy viscosity
ρ	Density
$\sigma_k, \sigma_\varepsilon$	Effective Prandtl numbers
τ_{ij}	Reynold's stress tensor

Subscripts

Symbol	Description
i,j	Index counter in computational plane
s	Static
x,y	Partial derivative in the physical plane

Superscripts

Symbol	Description
$'$	Fluctuating quantity in time- Mass averaging or correction quantity
$\overline{}$	Averaged quantity
$\circ$	Total

Abbreviations

Symbol	Description
AMGS	Algebraic Multi-Grid Solver
LE	Leading Edge
SIMPLE	Simi-Implicit Method For Pressure Linked Equations
TE	Tailing Edge
TIT	Turbine Inlet Temperature

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Table (1) Operating and Boundary Conditions

Case	Ma_{LE}	Ma_{TE}	M (kg/s)	Re	T° (K)	P° (bar)	$P_{s\ exit}$ (bar)	T_u (%)	L_c (m)	No. of Iteration to converge	Approx. Time to converge (minute)
1	0.17	0.88	1.15	1.9×10^{6}	796	3.2	0.88	6.5	0.32	4320	45
2	0.15	0.76	1.15	1.9×10^{6}	796	3.2	0.88	6.5	0.32	4000	40
Case 1: Turbulent compressible flow through turbine vane with internal convection cooling. Case 2: Turbulent compressible flow through turbine vane without cooling.											

Table (2) Property of the solid vane (ASTM type 310 stainless steel)

Property of the solid vane	The values
density (ρ)	7900 kg/m3
specific heat (Cp)	586.15 J/ (kg. K)
thermal conductivity (K)	17.0 W/(m. K)

Table (3) Cooling Channel, Diameter, and Temperatures (K)

Holes No.	1	2	3	4	5	6	7	8	9	10
Diameter(cm)	0.63	0.63	0.63	0.63	0.63	0.63	0.63	0.31	0.31	0.198
Temperatures (K)	549.6	549.6	538.5	525.7	556.3	559.6	567.4	590.7	629.7	645.3

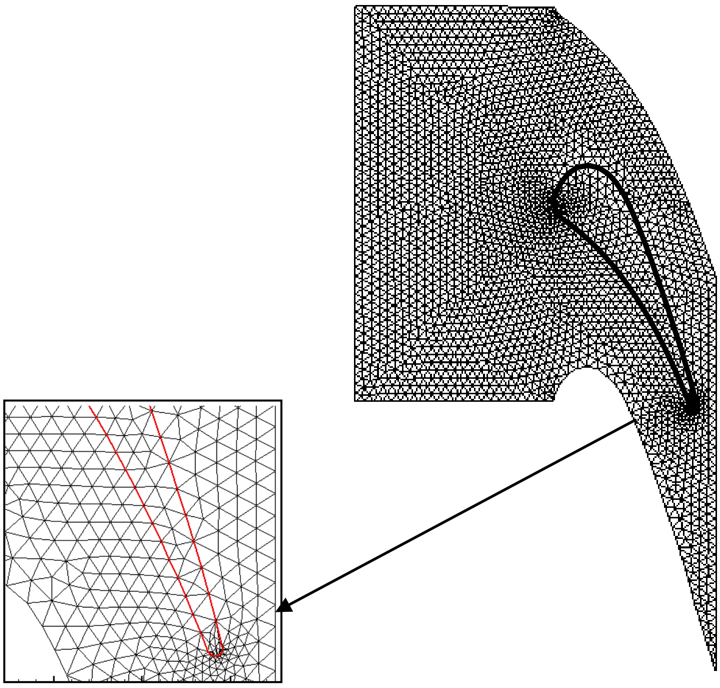

Figure (2): Flow through gas turbine nozzle guide vane without cooling, triangle elements, and computational grid system =4362 grid for fluid zone and 344 grids for solid zone (GAMBIT code by FLUENT).

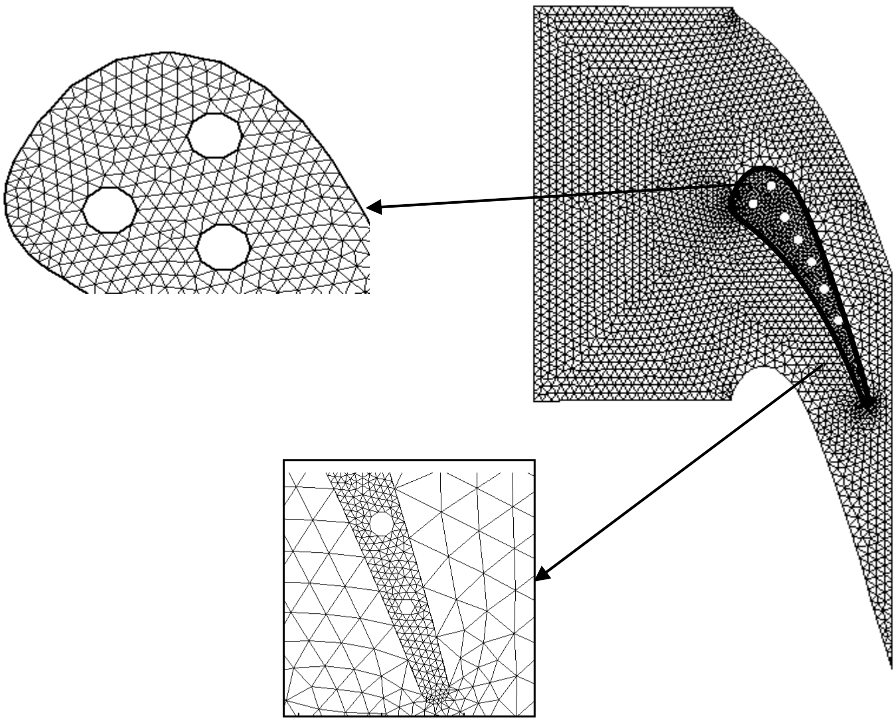

Figure (3): Flow through gas turbine nozzle guide vane with cooling by ten circular holes, triangle elements, and computational grid system =4362 grid for fluid zone and 990 grids for solid zone (GAMBIT code by FLUENT).

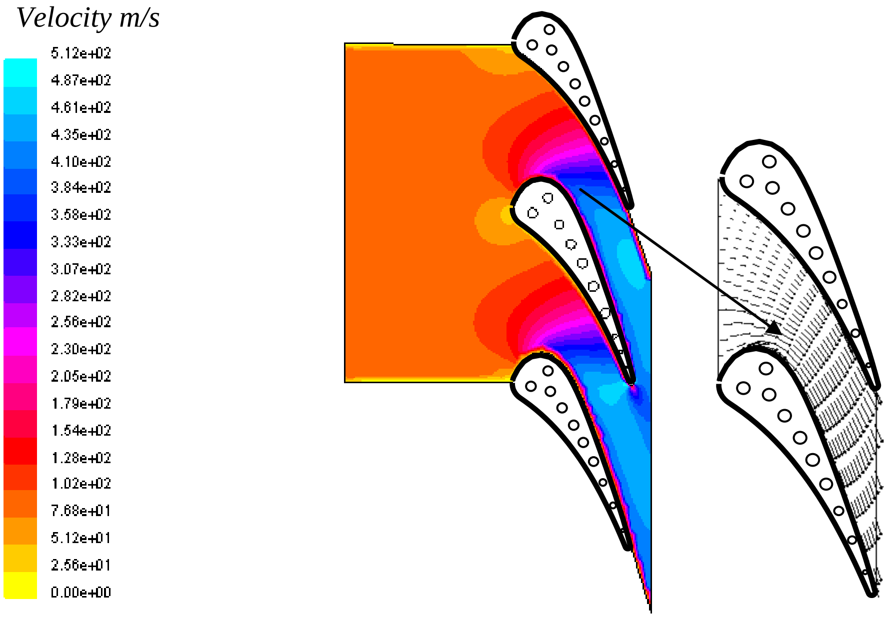

Figure (4): Velocity contour and velocity vector for steady, 2-D, turbulent, and compressible flow through gas turbine nozzle guide vane at midspan (with cooling).

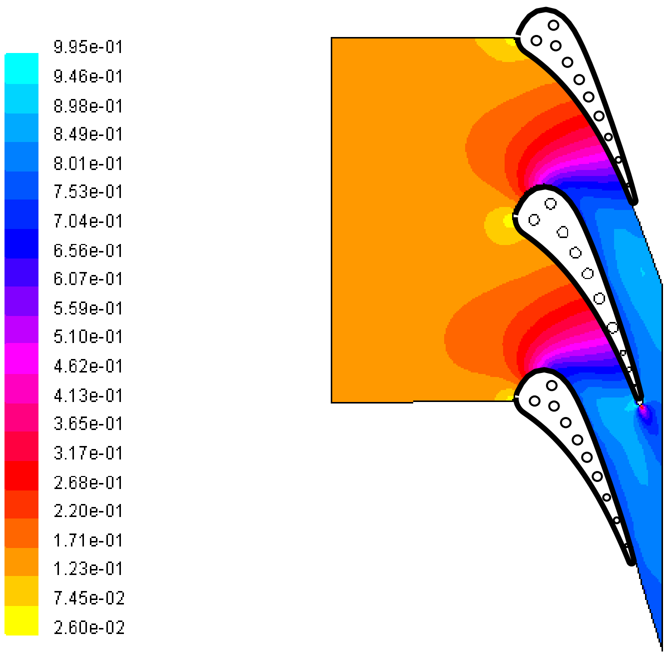

Figure (5): Mach contour for steady, 2-D, turbulent, and compressible flow through gas turbine nozzle guide vane at midspan (with cooling).

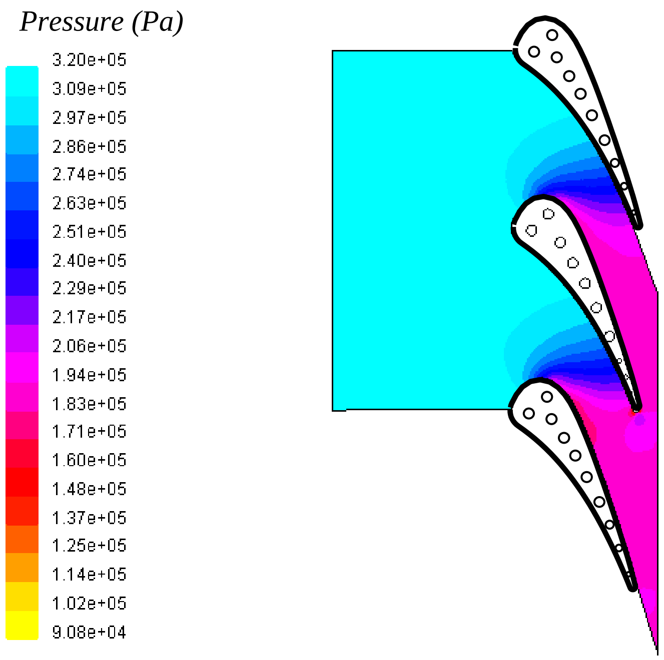

Figure (6): Pressure contour for steady, 2-D, turbulent, and compressible flow through gas turbine nozzle guide vane at midspan (with cooling).

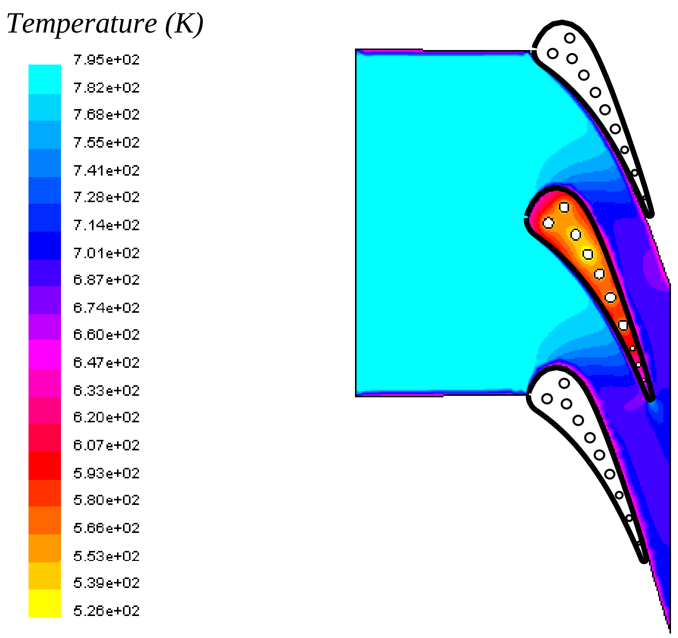

Figure (7): Temperature contour for steady, 2-D, turbulent, and compressible flow through gas turbine nozzle guide vane at midspan (with cooling).

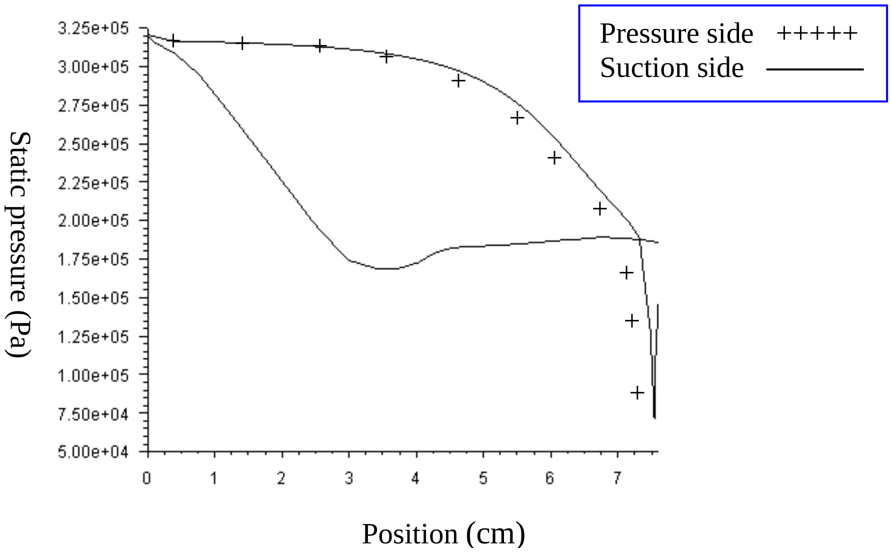

Figure (8): Pressure distribution for steady, 2-D, turbulent, and compressible flow at gas turbine nozzle guide vane wall at the midspan (without cooling).

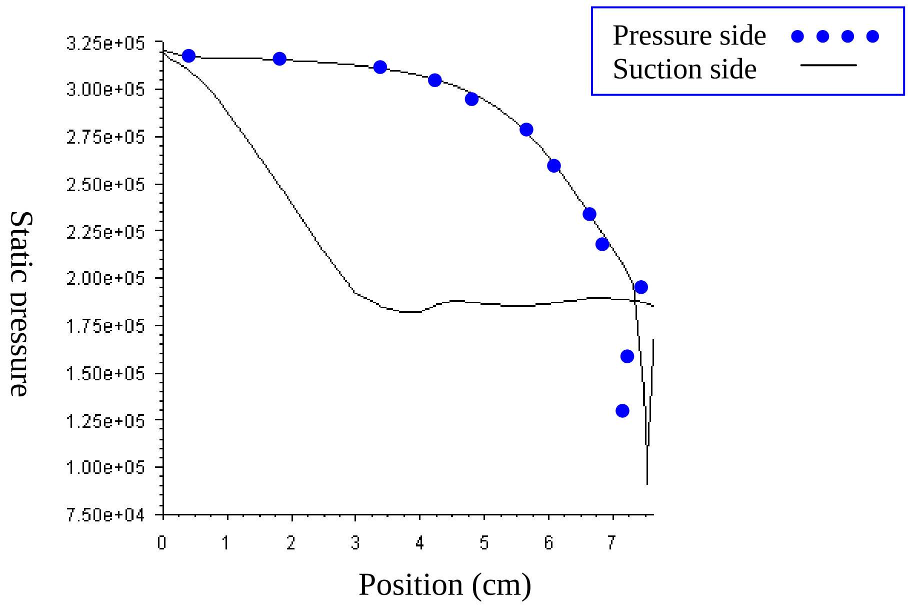

Figure (9) Pressure distribution for steady, 2-D, turbulent, and compressible flow at gas turbine nozzle guide vane wall at the midspan (with cooling).

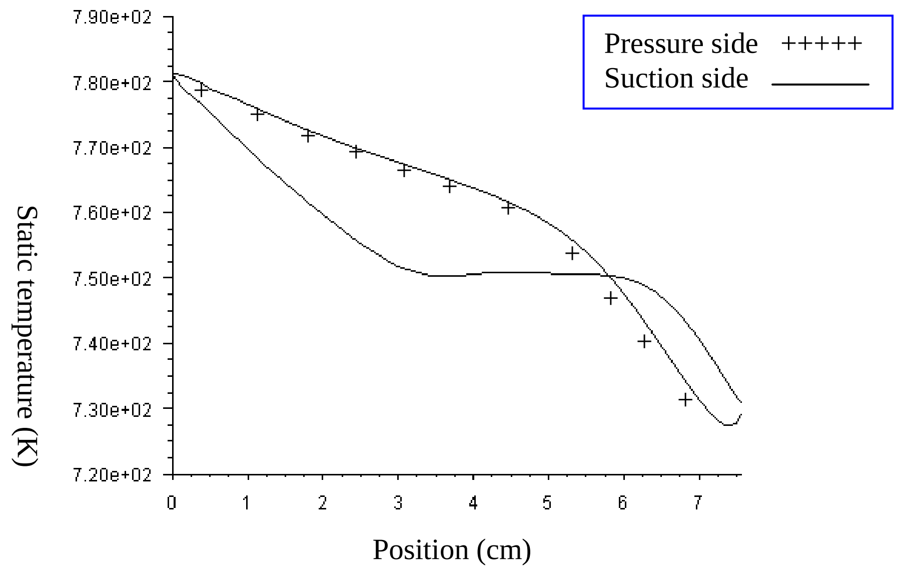

Figure (10): Temperature distribution for steady, 2-D, turbulent, and compressible flow at gas turbine nozzle guide vane wall at the midspan (without cooling).

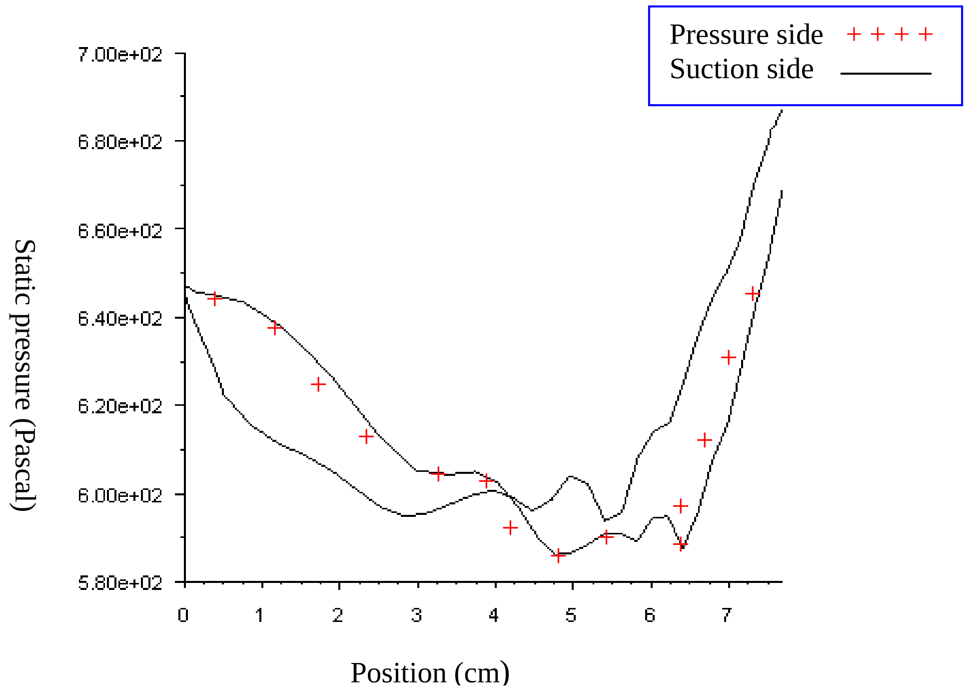

Figure (11) Temperature distribution for steady, 2-D, turbulent, and compressible flow at gas turbine nozzle guide vane wall at the midspan (with cooling).

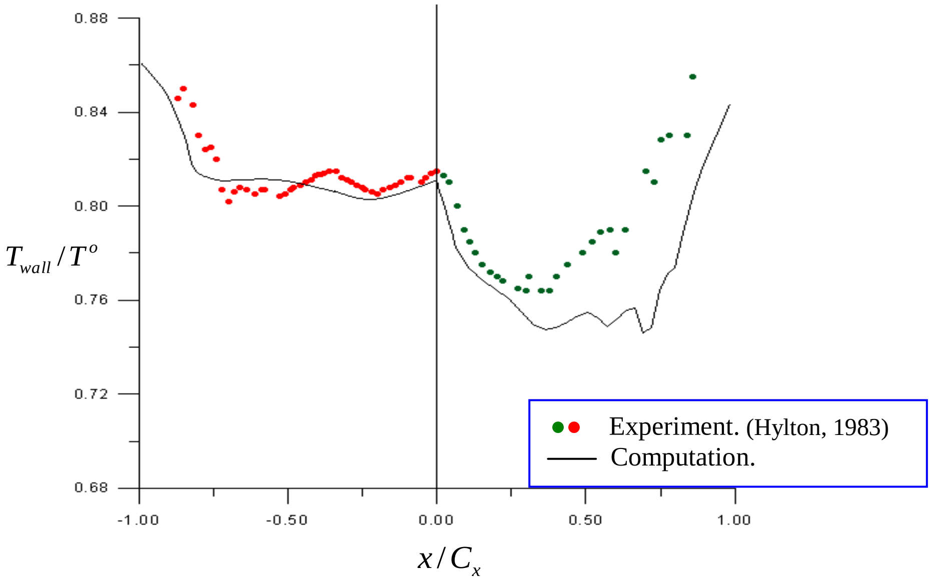

Figure (12): Experimental and predicted temperature distribution (T_{wall}/T^o) for steady, 2-D, turbulent, and compressible flow at gas turbine nozzle guide vane wall at midspan (with cooling).

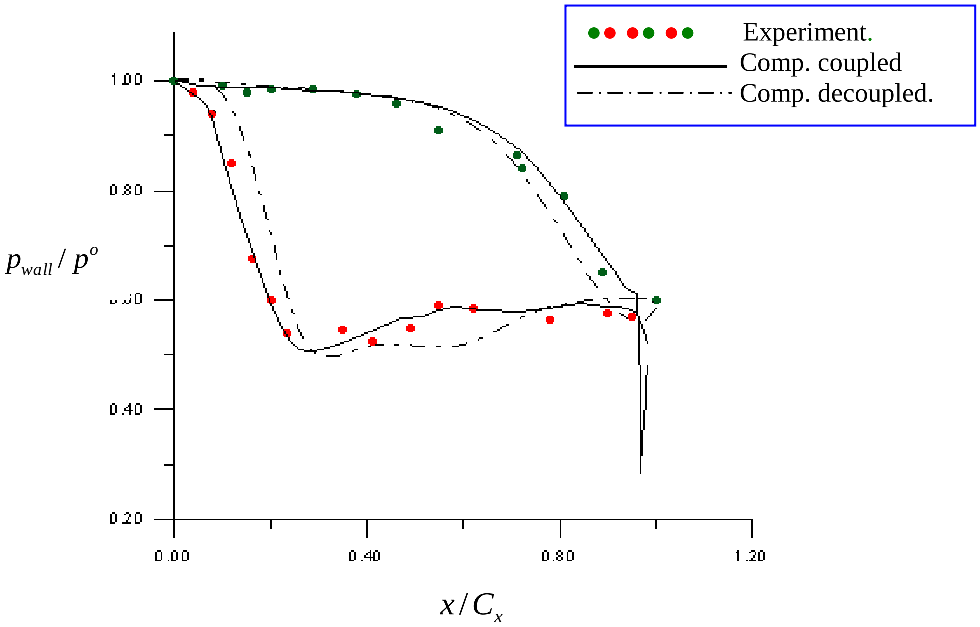

Figure (13) Experimental and predicted pressure distribution (p_{wall}/p^o) for steady, 2-D, turbulent, and compressible flow at gas turbine nozzle guide vane wall at midspan with coupled and decoupled treatment (with cooling).