

AERODYNAMIC COMPARISON AND VALIDATION OF RANS,  
K- $\omega$ , STANDARD K- $\epsilon$  AND URANS SIMULATIONS OF FLAT  
PLATE FILM-COOLING

OSAMA.H.ABDULGUAD

This project report presented in partial  
fulfillment of the requirements for the award of  
the Degree of Master of Mechanical and Manufacturing Engineering

Faculty of Mechanical and Manufacturing Engineering  
University Tun Hussein Onn Malaysia

JUNE, 2014

To my beloved and lovely parent and wife for care and support all the times. My beloved sisters, and all my friends



PTTA UTHM  
PERPUSTAKAAN TUN TUN AMINAH

## ACKNOWLEDGEMENTS

First of all, I deeply thank to Allah SWA for His guidance and blessing.

I thank and acknowledge my advisor; Dr. Mohammad bin Abdullah, whose help, advice and supervision was invaluable.

I am also indebted to Universiti Tun Hussein Onn Malaysia (UTHM) for providing the facilities for my study and research, Technicians and staff at the Mechanical and Manufacturing Engineering for their assistance in this study.

My sincere appreciation also extends to all my colleagues and others who have provided assistance at various occasions. Their views and tips are useful indeed.

May Allah SWA bless all of us!



PTTA UTHM  
PERPUSTAKAAN TUN AMINAH

## ABSTRACT

Film cooling has been extensively used to provide thermal protection for the external surfaces of gas turbine components. For the past 40 years, numerous number of film cooling hole designs and arrangements have been introduced. Due to broad designs and arrangements of film cooling, numerical investigation has been utilized to provide initial insight on the aerodynamics and thermal performance of the new film cooling designs or arrangements. The present work focuses on the numerical investigation of RANS and URANS analyses on a flat plate film cooling. The investigation aims to provide comparison between various turbulent models available for the Reynolds Average Navier Stokes (RANS) analyses and extended to unsteady Reynolds Average Navier Stokes (URANS). The numerical investigations make use of ANSYS CFX ver. 14 and were carried out at Reynolds Number,  $Re = 7,000$  based on the hole diameter at blowing ratio,  $BR = 0.5$ . The results of the RANS analyses show significant influence of the turbulent models on the predicted aerodynamics and thermal performance of the film cooling. Qualitative comparison between the simulation and experimental results shows that standard  $k-\epsilon$  produces more accurate results from the other considered turbulent models. In addition to that, results of URANS indicate limitation of RANS analyses to provide details on the eddied and vortices formation in film cooling flow structure.

## CONTENTS

<b>TITLE</b>	<b>ii</b>
<b>DECLARATION</b>	<b>iii</b>
<b>DEDICATION</b>	<b>iv</b>
<b>ACKNOWLEDGEMENT</b>	<b>v</b>
<b>ABSTRACT</b>	<b>vi</b>
<b>CONTENTS</b>	<b>vii</b>
<b>LIST OF TABLES</b>	<b>ix</b>
<b>LIST OF FIGURES</b>	<b>x</b>
<b>NOMENCLATURE</b>	<b>xii</b>
<b>LIST OF EQUATIONS</b>	<b>xiii</b>
<b>CHAPTER 1 INTRODUCTION</b>	<b>1</b>
1.1 Background of Study	2
1.2 Problem statement	4
1.3 Objectives of Study	5
1.4 Scope of study	6
<b>CHAPTER 2 LITERATURE REVIEWS</b>	
2.1 Introduction	7
2.2 The Jet in Cross Flow	7
2.3 Turbulence Models	9
2.3.1 The SST Model	9
2.3.2 The Reynold's Stress Model by (SSG)	11

2.3.3	RNG K- $\epsilon$ Model	13
2.3.4	K- $\omega$ Model (Wilcox's)	14
2.3.5	Standard K- $\epsilon$ Model	15
2.3.6	The Unsteady RANS (URANS) Model	17
<b>CHAPTER 3 METHODOLOGY</b>		18
3.1	Introduction	18
3.2	ANSYS CFX	18
3.3	Methodology Flowchart	19
3.4	Computational Fluid Dynamics (CFD) Simulation	20
3.4.1	Geometry	20
3.4.2	Meshing	23
3.4.3	Boundary Conditions	26
3.5	Unsteady Case	27
3.6	Counter Rotating Vortex Pair	28
3.7	Inside Hole Flow Structure	29
3.8	Performance Indicator Film Cooling Effectiveness	30
<b>CHAPTER 4 RESULTS AND DISCUSSION</b>		32
4.1	Grid dependency test	32
4.2	Flow Structure	34
4.3	Counter rotating vortex pair.	40
4.4	Inside hole flow structure	44
4.5	Film cooling effectiveness	46
<b>CHAPTER 5 CONCLUSIONS</b>		48
<b>REFERENCES</b>		49

**LIST OF TABLES**

2.1	Typical Values for the Constants in the Equations	11
2.2	Typical Values for Constants in the Equations for SSG	13
2.3	The Coefficients of Rodi	14
2.4	The Difference in the Values taken by some of the Variables	15
2.5	The Fine Constants Contained in the Equation	16
3.1	The Mesh Sizing Nod Parameters	25
3.2	The Boundary Condition Parameters	26
3.3	The Vortices Parameters	28
3.4	The Film Cooling Effectiveness Parameters	31



## LIST OF FIGURES

1.1.	An Open-Cycle Gas-Turbine Engine	1
1.2	Industrial Gas Turbine	2
1.3	An Closed-Cycle Gas-Turbine Engine	3
1.4	Temperature–Entropy plot of ideal Brayton cycle	4
2.1	Main Vortical Structures as Proposed by Fric and Roshko	8
3.1	Methodology Flowchart	19
3.2	The Steady Simulation	20
3.2a	The Steady Simulation Side View	21
3.2b	The Steady Simulation Top View	21
3.3	The Unsteady Simulation	21
3.3a	The Unsteady Simulation Side View	22
3.3b	The Unsteady Simulation Top View	22
3.4	Meshing Model for Steady Simulation	23
3.5	Hybrid Type of Meshing for Steady Simulation	24
3.6	Meshing Model for Unsteady Simulation	24
3.7	Hybrid Type of Meshing for Unsteady Simulation	25
3.8	Boundary Conditions for Steady Simulation	26
3.9	Boundary Conditions for Unsteady Simulation	27
3.10	Counter-rotating vortex pair	28
3.11	Flow structure inside cooling hole	29
3.12	Film cooling region	30
4.1	$u/U_\infty$ in the hole exit plane and Centre Plane with distribution for Coarse, Medium and fine	33



4.2	Film Cooling Effectiveness Diagram	34
4.3	$u/U_\infty$ in the hole exit plane for SST, SSG, $k-\omega$ , $k-\epsilon$ , RNG and URANS	35
4.4	$u/U_\infty$ in the plane $X/D=1$ for all models	36
4.5	$u/U_\infty$ in the plane $X/D=2$ for all models	37
4.6	$u/U_\infty$ in the plane $X/D=5$ for all models	38
4.7	$u/U_\infty$ in the plane $X/D=10$ for all models	39
4.8	Vortices x direction contour in the plane $X/D=1$ for all models	40
4.9	Vortices x direction contour in the plane $X/D=2$ for all models	41
4.10	Vortices x direction contour in the plane $X/D=5$ for all models	42
4.11	Vortices x direction contour in the plane $X/D=10$ for all models	43
4.12	Center Plane Vector Plot and Measurement Location	44
4.13	Inside Hole Normalized u-Velocity Contour of all the Model	45
4.14	Vector Plot on the Inside Hole Middle Plane for all the Model	45
4.15	The Comparison of the cooling effectiveness	46
4.16	The laterally average FCE distribution	47



## NOMENCLATURE

K	Turbulent Kinetic Energy [ $m^2s^{-2}$ ]
M	Blowing Ratio $\frac{\rho_c U_c}{\rho_\infty U_\infty}$
$R_{ed}$	Reynold's Number inside the Cooling Holes
$R_{ex}$	Reynold's Number of Flat Plate at Distance x from the Upstream Edge
u	Velocity Fluctuation in x-direction [ $ms^{-1}$ ]
U	Velocity in x-direction [ $ms^{-1}$ ]
v	Velocity fluctuation in y-direction [ $ms^{-1}$ ]
V	Velocity in y-direction [ $ms^{-1}$ ]
w	Velocity fluctuation in z-direction [ $ms^{-1}$ ]
$\delta$	Boundary Layer thickness [m]
$\eta$	Cooling Efficiency



## LIST OF EQUATIONS

EQUATION NO	TITLE	PAGE
1.1	The Thermal Efficiency of the Brayton's Cycle	3
2.1	Kinematic Eddy Viscosity	10
2.2	Turbulence Kinetic Energy	10
2.3	Specific Dissipation Rate	10
2.4	$F_2$ Coefficient	10
2.5	$P_K$ Coefficient	10
2.6	$F_1$ Coefficient	10
2.7	$CD_{K\omega}$ Coefficient	10
2.8	A blend of an Inner and Outer Constants	11
2.9	The Transport Equation of the Reynold's Stresses	11
2.10	The Production Term of the Reynold's Stress Model	11
2.11	Molecular Diffusion of the Reynold's Stress Model	11
2.12	The Model for the Turbulent Diffusion	11
2.13	A General Form for the Pressure-Strain Term	12
2.14	The Stress Anisotropy Tensor	12
2.15	Strain Rate Tensor	12
2.16	The Rotation Tensor	12
2.17	Reduction of the Second Moment Equation	12
2.18	A Transport Equation for the Dissipation	12
2.19	The Production of Dissipation	13
2.20	Turbulent Diffusion Term of the Dissipation Equation	13
2.21	K-Equation	14
2.22	$\varepsilon$ -Equation	14
2.23	K-Equation	14

2.24	$\omega$ -Equation	15
2.25	The Turbulent Eddy Viscosity	15
2.26	K-Equation	16
2.27	$\varepsilon$ -Equation	16
2.28	The Turbulent Eddy Viscosity	16
2.29	Continuity of URANS	17
2.30	Momentum of URANS	17
2.31	The Boussinesque Approximation	17
2.32	The Eddy Viscosity	17
2.33	K-Equation	17
2.34	$\varepsilon$ -Equation	17
3.1	Film Cooling Effectiveness	30





## CHAPTER 1

### INTRODUCTION

This chapter will provide description on gas turbine and film cooling technologies. The brief description aims to provide sufficient background of the topics for further discussion of the present work. Attention will also be paid to existing researches on film cooling technologies to develop the state of art awareness on the technologies itself.

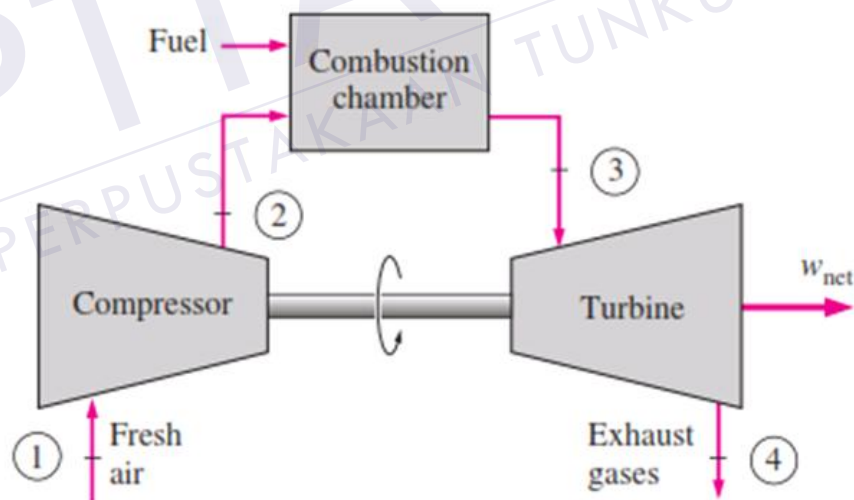


Figure 1.1: Open-cycle gas turbine engine

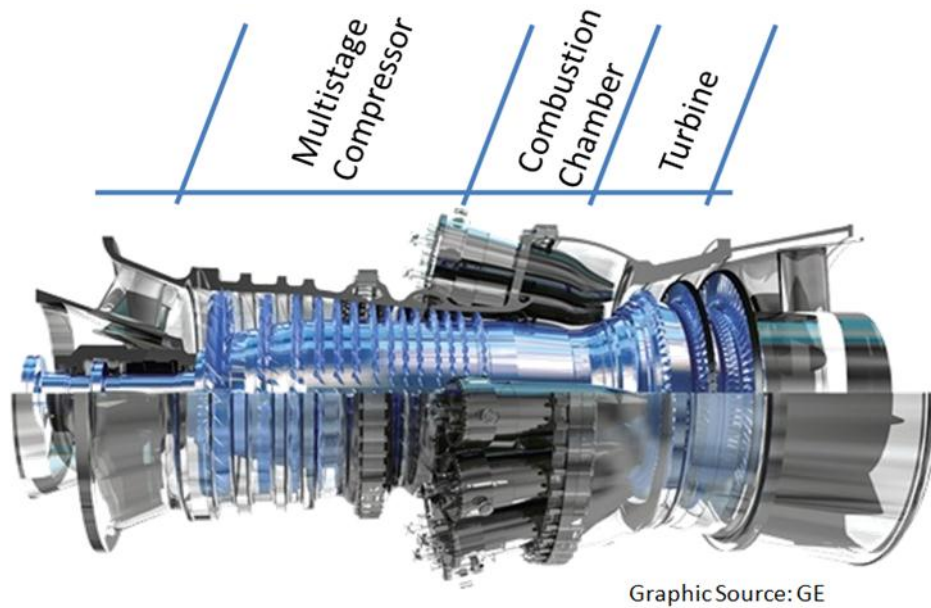


Figure 1.2: Industrial gas turbine

### 1.1 Background

The interest on gas turbine appeared approximately a century and half ago. But, the real success on the technology was only achieved in the year 1930 when Frank Whittle got a patent award on the jet engine. The static test of the pattern has been successfully carried out in 1937. Two years later in the year 1939, a jet engine powered flight was demonstrated by Hans von Ohain in Germany. Since then, gas turbines have been widely used in the transportation and power utility industries. Industrial gas turbine and jet engine operate in an open cycle as shown in Figure 1.1. The fresh air from the ambient will be compressed before being mixed with the fuel and burned in the combustion chamber. The high pressure and temperature flue gas will then be supplied into the turbine for power extraction process. The turbine is functioning as energy converter, converting high flow energy of the flue gas to mechanical energy. The mechanical energy (shaft rotation), will later be converted to desired form of energy. The flue gas will later release back to the ambient to complete the process. Figure 1.2 shows the real picture of an industrial gas turbine.

The working cycle of gas turbine system has been based on the ideal Brayton cycle (Figure 1.3 and Figure 1.4). The close loop of ideal Brayton cycle consists of four processes main process; a) isentropic compression (compressor), b) constant pressure heat addition (combustion chamber), c) isentropic expansion (turbine), d) constant pressure heat rejection (heat exchanger). The overall thermal efficiency of the Brayton's cycle can be presented by equation:

$$\eta_{th,Brayton} = \frac{W_{net}}{Q_{in}} = 1 - \frac{T_4}{T_3} \quad (1.1)$$

where  $W_{net}$ ,  $Q_{in}$ ,  $T_3$ , and  $T_4$  are the net work produce by the cycle [W], the heat supplied to the cycle [W], inlet temperature of the turbine [K], and the exit temperature of the turbine [K], respectively.

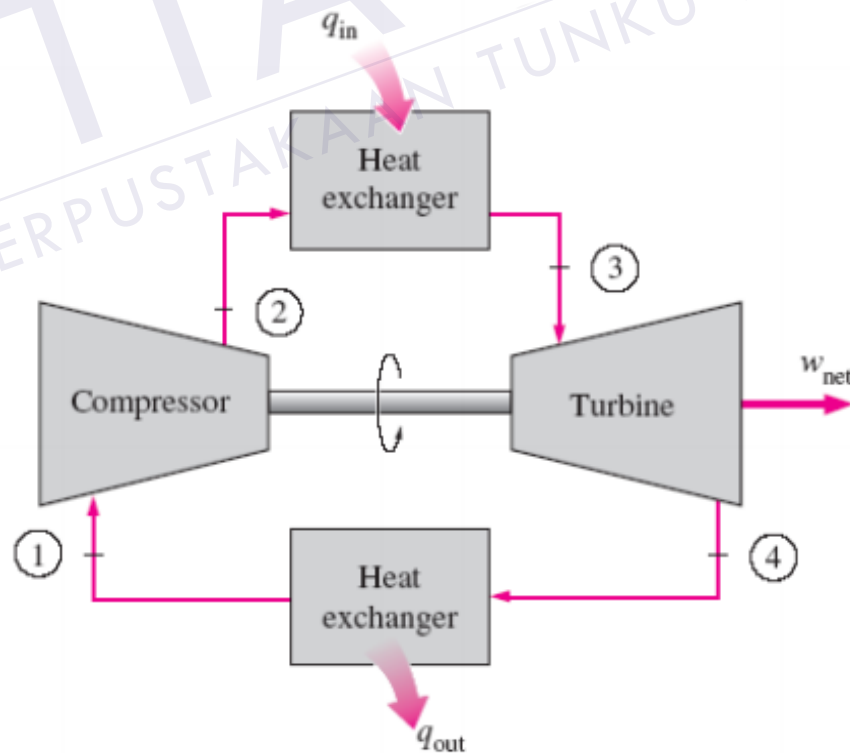


Figure1.3: Closed cycle gas turbine engine



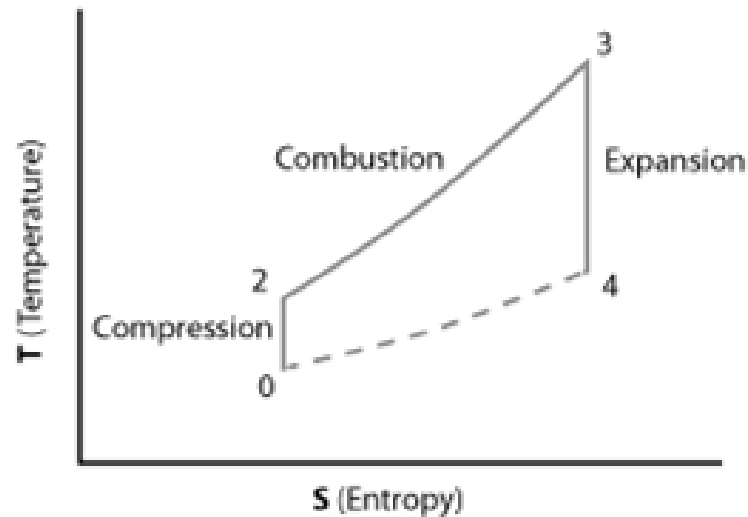


Figure 1.4: Temperature–Entropy plot of ideal Brayton cycle

Eq. (1.1) indicates higher turbine inlet temperatures produces higher overall thermal efficiency of the cycle. The increasing demand for better performance gas turbine provokes the increase of turbine inlet temperature. The modern gas turbine nowadays works at the temperature range around 1800K- 2000K, which is higher than the melting temperature of the turbine components materials. Such increasing of the turbine inlet temperature became possible because of application of cooling scheme on the turbine components. One of the cooling methods involved is film cooling technique. The working concept of film cooling lays on the injection of cold air to provide a layer of cool fluid between the hot gases and the blade surface, reducing temperature of the surface.

## 1.2 Problem Statement

Enormous numbers of researches have been done to improve film cooling performance. The researches focus on eliminating the counter rotating vortices effects through new hole design, holes arrangements, and introduction of passive and active devices. Due to the broad design space of film cooling, computational fluid dynamics

has been used to provide preliminary ideas on the newly proposed concept of film cooling. Therefore, it is important to acknowledge the discrepancy that could be influenced by the application of different turbulent models in the steady Reynolds Average Navier Stokes (RANS) simulation. In addition to that, it is also important to acknowledge the incapability of RANS in providing accurate prediction of the formation of vortices in comparison with the unsteady Reynolds Average Navier Stokes simulation.

For the present study, six turbulence models are used to simulate a film cooling case. The model accuracy will be compared with experimental results data that are already available. In addition, further discussion will also be made available on the inside hole flow phenomena predicted by all turbulent models.

### 1.3 Objectives Of Study

The present study aims to evaluate the capability of different turbulence models to provide the accurate prediction of aerodynamics and thermal performance of the film cooling phenomena. The objectives of the project are:

- a) To compare the predicted results of different turbulent model of RANS analysis with the available experimental results;
- b) To proposed the best available turbulent models of RANS analysis for film cooling study;
- c) To compare the predicted results of URANS with RANS and available experimental results.

#### 1.4 Scope Of Study

The scopes of the present study are:

- i) the simulation will be carried out using commercial computational fluid dynamics package of ANSYS CFX software.
- ii) the steady RANS analyses will involve six different turbulent models; SST, SSG, RNG k- $\epsilon$ , k- $\omega$ , and standard k- $\epsilon$ .
- iii) the simulation will be run at Reynolds number based on hole diameter,  $Re = 7,000$  and blowing ratio,  $BR = 0.5$



PTTA UTHM  
PERPUSTAKAAN TUNKU TUN AMINAH

## CHAPTER 2

### LITERATURE REVIEW

#### 2.1 Introduction

In this chapter will discuss the computational models that will be used for simulation of the flow field. This is theoretical part of the work, dealt with the crucial characteristics of a different models using nowadays. The equations will be review of every model and outline their main advantages and disadvantages.

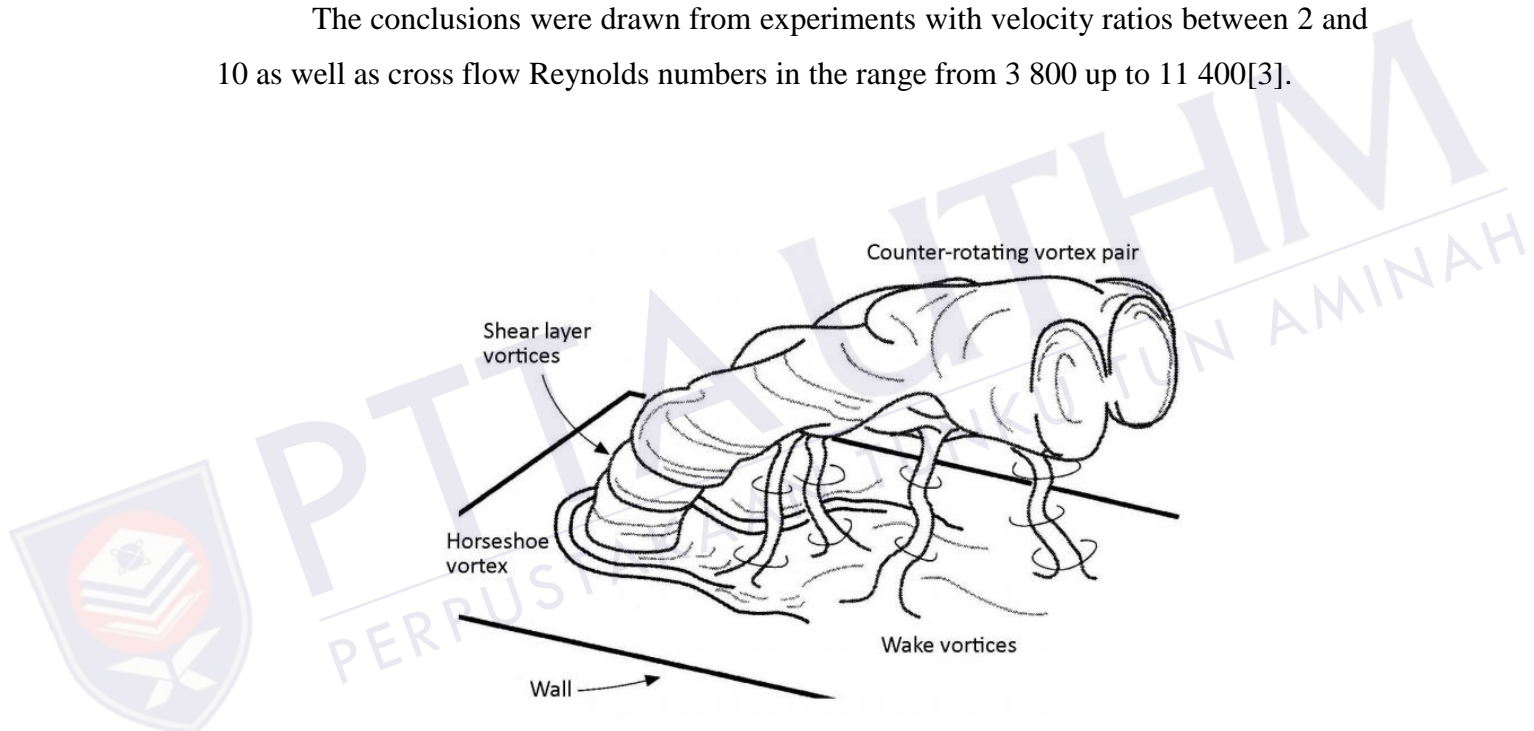
#### 2.2 The Jet In Cross Flow

A numerous studies have been undertaken both experimentally and computationally to investigate the cooling efficiency, evolution of cooling film or influences of different geometries on the performance. Important work from experimental side has been done by Andreopoulos and Rodi [1] and Fric and Roshko [2], who characterized the evolution and main features of jet in cross flow configurations. Figure (2.1) shows a sketch of a jet in cross flow as established by Fric and Roshko [2]

Containing four principal types:

- Shear layer vortices
- Counter-rotating vortex pair
- Horseshoe vortex
- Wake vortices

The conclusions were drawn from experiments with velocity ratios between 2 and 10 as well as cross flow Reynolds numbers in the range from 3 800 up to 11 400[3].



**Figure 2.1: Main vortical structures as proposed by Fric & Roshko [2]**

A lot of experimental work has been done by the group of Crawford in the field of the application of jets in cross flow for film cooling applications [4-8]. They measured film cooling effectiveness for full-coverage film cooling configurations [4-6], as well as aerodynamic measurements [7] and film cooling effectiveness [8] for single row configurations. Goldstein [9] gives an introduction to film cooling, which includes several cooling methods and explains important parameters in film cooling flows. He also

gives an overview of work done in the field during the 1950s and 60s. A bibliography done in CFD of film cooling flows is given by Kercher [10]. It includes references from 1971 to 1996. Bacchi and Facchini [11] used a modified  $k-\epsilon$  model to simulate flat-plate film cooling in order to overcome the typical errors of standard two-equation models in film cooling. These errors are an underpredicted lateral jet spreading and an overpredicted jet penetration into the mainstream. These systematical errors arise from the assumption of isotropic turbulence. Yavuzkurt [12-13] use the standard, RNG and realizable  $k-\epsilon$  model and the  $k-\omega$  model for film cooling from inclined holes. Their results show that all used models give acceptable results for a low blowing ratio of  $M=0.5$  but completely fail to predict the cooling effectiveness for a large blowing ratio of  $M=1.5$ , especially in the area just downstream of the cooling hole ( $x/D < 8$ ). Furthermore their studies show that the results strongly depend on the used mesh. For a hexahedral mesh the results are clearly better than for a mesh which is partly meshed tetrahedral and partly hexahedral. Harrison and Bogard [14] used three turbulence models, namely the standard  $k-\epsilon$ , the standard  $k-\omega$  and a Reynolds Stress model (RSM) to simulate film cooling from a 30 degree inclined hole. Their results show that the standard  $k-\omega$  model gives best results for the laterally averaged cooling effectiveness, but at the same time gives the worst agreement with the experiments for the centerline effectiveness.

## 2.3 Turbulence Models

### 2.3.1 The SST Model

The SST  $k-\omega$  turbulence model [Menter 1993] is a two- equation eddy-viscosity model which has become very popular. The shear stress transport (SST) formulation combines the best of two worlds. The use of a  $k-\omega$  formulation in the inner parts of the boundary layer makes the model directly usable all the way down to the wall through the viscous sub-layer; hence the SST  $k-\omega$  model can be used as a Low-Re turbulence model without any extra damping functions. The SST formulation also switches to a  $k-\epsilon$  behavior in the free-stream and thereby avoids the common  $k-\omega$  problem that the model is too sensitive to the inlet free-stream turbulence properties. Authors who use the SST  $k-\omega$

model often merit it for its good behavior in adverse pressure gradients and separating flow. The SST k- $\omega$  model does produce a bit too large turbulence levels in regions with large normal strain, like stagnation regions and regions with strong acceleration. This tendency is much less pronounced than with a normal k- $\epsilon$  model though [15].

### Kinematic Eddy Viscosity

$$v_T = \frac{a_1 k}{\max(a_1 \omega, SF_2)} \quad (2.1)$$

### Turbulence Kinetic Energy

$$\frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} = P_k - \beta^* k \omega + \frac{\partial}{\partial x_i} \left[ (v + \sigma_k v_T) \frac{\partial k}{\partial x_i} \right] \quad (2.2)$$

### Specific Dissipation Rate

$$\frac{\partial \omega}{\partial t} + U_j \frac{\partial \omega}{\partial x_j} = \alpha S^2 - \beta \omega^2 + \frac{\partial}{\partial x_j} \left[ (v + \sigma_\omega v_T) \frac{\partial \omega}{\partial x_j} \right] + 2(1 - F_1) \sigma_{\omega 2} \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_i} \quad (2.3)$$

### Closure Coefficients and Auxiliary Relations

$$F_2 = \tanh \left[ \left[ \max \left( \frac{2\sqrt{k}}{\beta^* \omega y}, \frac{500 v}{y^2 \omega} \right) \right]^2 \right] \quad (2.4)$$

$$P_k = \min \left( \tau_{ij} \frac{\partial U_i}{\partial x_j}, 10 \beta^* k \omega \right) \quad (2.5)$$

$$F_1 = \tanh \left\{ \left\{ \min \left[ \max \left( \frac{\sqrt{k}}{\beta^* \omega y}, \frac{500 v}{y^2 \omega} \right), \frac{4 \sigma_{\omega 2} k}{CD_{K\omega} y^2} \right] \right\}^4 \right\} \quad (2.6)$$

$$CD_{K\omega} = \max \left( 2 \rho \sigma_{\omega 2} \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_i}, 10^{-10} \right) \quad (2.7)$$

Each of the constants is a blend of an inner (1) and outer (2) constant, blended via:

$$\phi = \phi_1 F_1 + \phi_2 (1 - F_1) \quad (2.8)$$

Table 2.1: Typical values for the constants in the equations

$\alpha_1$	$\alpha_2$	$\beta_1$	$\beta_2$	$\beta^*$	$\sigma_{k1}$	$\sigma_{k2}$	$\sigma_{\omega 1}$	$\sigma_{\omega 2}$
5/9	0.44	-3/40	0.0828	9/100	0.85	1	0.5	0.856

### 2.3.2 The Reynolds Stress Model by Speziale, Sarkar and Gatski (SSG)

The transport equation of the Reynolds stresses can be derived by multiplying the momentum equation with the fluctuating velocity components and time averaging the product (Wilcox, 1993):

$$\frac{\partial \overline{u'_i u'_j}}{\partial t} + \overline{U}_k \frac{\partial \overline{u'_i u'_j}}{\partial x_k} = P_{ij} + \frac{\partial}{\partial x_k} (D_{ijk}^v) + \frac{\partial}{\partial x_k} (D_{ijk}^t) + \Pi_{ij} - \varepsilon_{ij} \quad (2.9)$$

The production term is defined using the exact form

$$P_{ij} = - \left[ \overline{u'_j u'_k} \frac{\partial \overline{u}_i}{\partial x_k} + \overline{u'_i u'_k} \frac{\partial \overline{u}_j}{\partial x_k} \right] \quad (2.10)$$

Molecular diffusion also follows the exact definition

$$D_{ijk}^v = \nu \frac{\partial}{\partial x_k} (\overline{u'_i u'_j}) \quad (2.11)$$

The model for the turbulent diffusion

$$D_{ijk}^t = C_s \frac{k}{\varepsilon} \left[ \overline{u'_k u'_j} \frac{\partial \overline{u'_i u'_j}}{\partial x_i} \right] \quad (2.12)$$

Many models for the pressure - strain term have been developed since the pioneering work of Launder, Reece, and Rodi. A general form is given by



## References

- [1] Andreopoulos, J., Rodi, W. (1984). *Experimental Investigation of Jets in a Crossflow*. J. of Fluid Mechanics, 138, pp. 93-127.
- [2] Fric, T.F., Roshko, A. (1994). *Vortical Structure in the Wake of Transverse Jet*. J. of Fluid Mechanics, 279, pp. 1-47.
- [3] A. Sau, T. Sheu, R. Hwang, and W. Yang (2004). *Three-Dimensional Simulation of Square Jets in Cross-Flows*. Physical Review E, 69.
- [4] Crawford, M.E., Kays, W.M., Moffat, R.J. (1980). *“Full-Coverage Film-Cooling on Flat, Isothermal Surfaces: A Summary Report on Data and Predictions”*. NASA Paper No. CR3219.
- [5] Crawford, M.E., Kays, W.M., Moffat, R.J. (1980). *“Full-Coverage Film Cooling; Part 1 Comparison of Heat Transfer Data for Three Injection Angles”*, ASME Paper No. 80/GT/43.
- [6] Crawford, M.E., Kays, W.M., Moffat, R.J. (1980). *“Full-Coverage Film Cooling; Part 2 Heat Transfer Data and Numerical Simulation”*, ASME Paper No. 80/GT/44.
- [7] Pietrzyk, J.R., Crawford, M.E., Bogard, D.G. (1988). *“Hydrodynamic Measurements of Jets in Crossflow for Gas Turbine Film Cooling Applications”*, ASME Paper No. 88/GT/174.
- [8] Sinha, A.K., Bogard, D.G., Crawford, M.E. (1990). *“Film Cooling Effectiveness Downstream of a Single Row of Holes with Variable Density Ratio”*, ASME Paper No. 90/GT/43.
- [9] Goldstein, R.J. (1971). *“Film Cooling”*, Advances in Heat Transfer, 7, pp. 321-380.

- [10] Kercher, D.M. (1998). "*A Film-Cooling CFD Bibliography*", Int. J. of Rotating Machinery (1971-1996), 4, No. 1, pp. 61-72.
- [11] Bacci, A., Facchini B. (2007). "*Turbulence Modeling for the Numerical Simulation of Film and Effusion Flows*". ASME TurboExpo, Montreal, Canada. Paper No. GT2007-27182.
- [12] Yavuzkurt, S., Hassan, J. (2007). "*Evaluation of Two-Equation Models of Turbulence in Predicting Film Cooling Performance Under High Free Stream Turbulence*". ASME TurboExpo, Montreal, Canada. Paper No. GT2007-27184.
- [13] Yavuzkurt, S., Habte, M. (2008). "*Effect of Computational Grid on Performance of Two-Equation Models of Turbulence for Film Cooling Applications*". ASME TurboExpo, Berlin, Germany. Paper No. GT2008-50153.
- [14] Harrison, K.L., Bogard, D.G. (2008). "*Comparison of RANS Turbulence Models for Prediction of Film Cooling Performance*". ASME TurboExpo, Berlin, Germany. Paper No. GT2008-51423.
- [15] Menter, F. R. (August 1994). "*Two-Equation Eddy-Viscosity Turbulence Models for Engineering Applications*". AIAA Journal, Vol. 32, No. 8, pp. 1598-1605.
- [16] Journal of Physics: Conference Series 318(2011)042031, 13<sup>th</sup> European Turbulence Conference (ETC 13).
- [17] V. C. PATEL, W. RODI, and G. SCHEUERER. (1985). "*Turbulence models for near-wall and low Reynolds number flows - A review*". AIAA Journal, Vol. 23, No. 9, pp. 1308-1319.
- [18] DAVID C. WILCOX. "*Comparison of two-equation turbulence models for boundary layers with pressure gradient*". AIAA Journal, Vol. 31, No. 8 (1993). pp. 1414-1421.
- [19] Versteeg, H.K., Malalasekera, W. (1995). "*The Introduction to Computational Fluid Dynamics. The Finite Volume Method*". Longman Scientific and Technical, New York, United States.
- [20] Asif Hoda, B.S. (1997). "*Turbulence Modeling for Film Cooling Flows*". Birla Institute of Technology, India.

[21] Pietrzyk, J.R. (1989). "*Experimental Study of the Interaction of Dense Jets with a Crossflow for Gas Turbine Applications*", PhD Thesis, University of Texas at Austin.

