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DEVELOPMENT OF TEMPERATURE MEASUREMENT METHOD FOR GAS TURBINE COOLING APPLICATION

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DEVELOPMENT OF TEMPERATURE MEASUREMENT METHOD FOR GAS TURBINE COOLING APPLICATION

By

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Thesis Submitted to the School of Graduate Studies, Universiti Putra Malaysia, in Fulfilment of the Requirements for the Degree of Doctor of Philosophy

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DEDICATION

Mohd Saiah Bahaudin and Zaiton Jaafar

Abstract of thesis presented to the Senate of Universiti Putra Malaysia in fulfilment of the requirement for the degree of Doctor of Philosophy

DEVELOPMENT OF TEMPERATURE MEASUREMENT METHOD FOR GAS TURBINE COOLING APPLICATION

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Chair : Associate Professor Azmin Shakrine Mohd Rafie, PhD
Faculty : Engineering

Temperature measurement are one of the essential part in gas turbine cooling research. The resulting heat transfer coefficient and adiabatic wall temperature are two of the important information analysed from the temperature data. One dimensional semi-infinite heat transfer solution is widely used to solve for the heat transfer coefficient and adiabatic wall temperature. However, the experimental time for this solution was limited resulting in less temperature data for analysis. There is an issue regarding longer experimental time is needed to accurately calculate the heat transfer coefficient and the adiabatic wall temperature. A temperature measurement method was investigated to solve this issue. A test rig was designed to have similar test area to the wheel space area for a representative single stage gas turbine rig. Crank Nicolson finite difference method was proposed to solve for the internal temperatures of the test plate. In this work, the solution was designed to have two different back face boundary condition. First, an adiabatic back face boundary condition to simulate the one dimensional semi-infinite heat transfer condition. Second, a conduction-convection back face boundary condition to solve the time limitation issue. The resultant heat transfer coefficient from adiabatic back face boundary condition had an average of 2.5% difference and the adiabatic wall temperature had an average of 2% difference when compared to reference values. Duration for heat transfer experiments were longer for the conduction-convection back face boundary condition, at Fo = 0.7 rather than Fo = 0.1. This results in an increase of 40% more temperature data range for the heat transfer analysis. For these experiments, the conduction-convection back face boundary condition had an average of 5% difference in heat transfer coefficient and 3.5% difference in adiabatic wall temperature. Meanwhile, the adiabatic back face boundary condition had an average of 11.3% difference in heat transfer coefficient and 4.9% difference in adiabatic wall temperature when compared to reference values. Crank Nicolson solution method with conduction-convection back face boundary condition allowed more temperature data for analysis and provide more accurate heat transfer coefficient and adiabatic wall temperature values.

Abstrak tesis yang dikemukakan kepada Senat Universiti Putra Malaysia sebagai memenuhi keperluan untuk ijazah Doktor Falsafah

PEMBANGUNAN KAEDAH PENGUKURAN SUHU BAGI APLIKASI PENYEJUKAN GAS TURBIN

Oleh

HELMEY RAMDHANEY BIN MOHD SAIAH

Januari 2021

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Pengukuran suhu adalah salah satu keperluan di dalam kajian penyejukan gas turbin. Pekali pemindahan haba dan suhu dinding adiabatik yang terhasil adalah dua maklumat penting yang dianalisis daripada data suhu. Kaedah penyelesaian pemindahan haba satu dimensi separa tak terhingga digunakan secra meluas bagi menyelesaikan pekali pemindahan haba dan suhu dinding adiabatik. Tetapi, masa eksperimen bagi kaedah penyelesaian ini adalah terhad dan menyebabkan kekurangan data suhu untuk analisis. Terdapat isu berkenaan durasi eksperimen yang lebih lama diperlukan bagi pengiraan pekali pemindahan haba dan suhu dinding adiabatik yang lebih jitu. Satu siasatan kaedah pengukuran suhu telah dilakukan bagi menyelesaikan isu ini. Sebuah pelantar ujian yang mempunyai kawasan ujian yang sama kepada kawasan ruang roda wakil gas turbin satu tahap telah direka. Kaedah perbezaan terhingga Crank Nicolson telah diusulkan bagi menyelesaikan suhu dalaman plat ujian. Di dalam kerja ini, kaedah penyelesaian tersebut telah direka supaya terdapat dua keadaan sempadan muka belakang. Pertama adalah keadaan muka belakang adiabatik bagi menyamai keadaan pemindahan haba satu dimensi separa tak terhingga. Kedua adalah keadaan sempadan muka belakang pengkonduksian-perolakan bagi menyelesaikan isu masa yang terhad. Pekali pemindahan haba yang terhasil daripada keadaan sempadan muka belakang adiabatic mempunyai purata perbezaan sebanyak 2.5% dan suhu dinding adiabatik pula mempunyai purata perbezaan sebanyak 2% apabila dibandingkan kepada nilai rujukan. Masa eksperimen pemindahan haba bagi keadaan sempadan muka belakang pengkonduksian-perolakan adalah lebih lama iaitu pada Fo = 0.7 berbanding pada Fo = 0.1. Ini menghasilkan peningkatan julat data suhu sebanyak 40% bagi analisis pemindahan haba. Bagi eksperimen ini, keadaan sempadan muka belakang pengkonduksian-perolakan menghasilkan purata perbezaan sebanyak 5% bagi pekali pemindahan haba, dan perbezaan sebanyak 3.5% bagi suhu dinding adiabatik. Sementara itu, keadaan sempadan muka belakang adiabatik menghasilkan purata perbezaan sebanyak 11.3% bagi pekali pemindahan haba, dan perbezaan sebanyak 4.9% bagi suhu dinding adiabatik apabuila

dibandingkan kepada nilai rujukan. Kaedah penyelesaian Crank Nicolson keadaan sempadan muka belakang pengkonduksian-perolakan membenarkan lebih banyak data suhu unutk dianalisis dan menghasilkan pekali pemindahan haba dan suhu dinding adiabatik yang lebih jitu.

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This thesis was submitted to the Senate of Universiti Putra Malaysia and has been accepted as fulfilment of the requirement for the degree of Doctor of Philosophy. The members of the Supervisory Committee were as follows:

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NOMENCLATURES

Symbols		Unit
A	Area	m^2
Bi	Biot number	
c_p	Specific heat	kJ/kgK
Fo	Fourier number	
h	Heat transfer coefficient	W/m^2K
k	Thermal conductivity	W/mK
L	Plate thickness	m
$\dot{m}$	Mass flow rate	kg/s
Nu	Nusselt number	
Pr	Prandtl number	
q	Surface heat flux	W/m^2
R	Uncertainty term	
Re	Reynolds number	
t	Time	s
T	Temperature	ºC / K
U	Velocity	m/s
w	width	m
x	Axial location	m
X	Product of Bi and Fo	
y	Axis location	m
z	Axis location	m

Greek

α	Thermal diffusivity	m^2/s
β	Non-dimensional distance	
δ	Boundary layer thickness	m
ρ	Density	kg/m^3
μ	Dynamic viscosity	kg/ms
η	Efficiency	
Θ	Temperature ratio	

Subscript

ad	Adiabatic
amb	Ambient
FP	Flate plate
g	Working air
i	Initial, Nodal location in Crank Nicolson equation
L	Plate thickness
n	Newton Raphson iteration
s	surface
th	Thermal
x	Axial location
∞	Freestream

Superscript

n	Time step in Crank Nicolson equation

Abbreviations

1DSIHT	One dimensional semi-infinite heat transfer
ABF	Adiabatic back face
BF	Blockage factor
CCBF	Conduction-convection back face
CFD	Computational fluid dynamics
CN	Crank Nicolson
erfc	Complimentary error function
exp	Exponential function
FPHT	Flat plate heat transfer
PIV	Particle image velocimetry
PSP	Pressure sensitive paints
RPM	Revolution per minute
TLC	Thermochromic liquid crystals
TSM	Taylor series method

CHAPTER 1

INTRODUCTION

1.1 Chapter overview

The introduction chapter discussed the need for accurate heat transfer experiment and data analysing technique. Beginning with the basic operation of gas turbine engines, the need of increasing its turbine inlet temperature and the importance of integrating cooling technique into high temperature regions. Investigation on increasing the turbine inlet temperature leads to the introduction of cooling techniques in the turbine section. These investigation depend highly on the heat transfer analysis. The usual approach on the heat transfer experiment was utilising the one dimensional semi-infinite heat transfer condition and analysing method. However, this technique has its limitations that could be improvised. The current research work revolves around this issue. The objectives for the thesis were defined at the end of this chapter.

1.2 The basic of gas turbine engines

The gas turbine engine concept started a long way back in the early 1790's. Since then theories began being defined, prototypes being designed, and ultimately, a real working gas turbine engine was being invented. Be it land, sea, or air, the gas turbine engine proved to be a beneficial power generator. The basis for a gas turbine engine is based on the Brayton thermodynamics cycle. The thermodynamic cycle involves isentropic compression, constant pressure combustion, isentropic expansion, and constant heat rejection. These four theoretical processes are achieved by the three core components of a gas turbine engine, the compressor, combustion chamber, and the turbine.

The compressor section draws and compresses ambient air where the pressure and consequently the temperature of the intake air are increased. Majority of the compressed air is then directed to the combustion chamber where fuel is added in a specific ratio to the compressed air. The air-fuel mixture is then combusted at a constant pressure condition. The resultant combustion product, which possesses extremely high pressure, enters the turbine stage and expand isentropically. Power will be extracted from this particular stage to either drive the compressor, powers a mechanical drive application, driving speed reduction gears for generators, and/or produces thrust for aero-engines. Heat rejection at a constant pressure takes place in the atmosphere for an open cycle, while for a closed cycle the heat rejection process takes place in a heat exchanger unit.

The thermal efficiency, η_{th}, for this thermodynamic process operating at an optimal compression ratio is given by, Kadambi (2003);

$$\eta_{th} = 1 - \left(\frac{T_{sink}}{T_{source}}\right)^{1/2} \tag{1.1}$$

T_{sink} is the temperature for energy rejection, usually taken as the ambient atmospheric temperature. T_{source} is the maximum operating temperature in the Brayton cycle, which can be represented by the turbine stage entry temperature. Therefore, in order to increase the thermal efficiency, T_{sink} should be as low as possible and T_{source} should be as high as possible. For an open Brayton cycle, T_{sink} can be as low as the atmospheric temperature. This is why flying in higher altitude where the temperature is significantly low is preferable. T_{source} on the other hand, could be set to be high as possible only to be limited by the materials temperature limitations. In an ideal condition, the combustion process should be in a stoichiometric temperature of around 2000 °C where complete combustion occurs.

Specific fuel consumption of a gas turbine can also be related to the thermal efficiency of the Brayton cycle. Higher turbine entry temperature would indicate that the combustion temperature is near the stoichiometric temperature. This is where optimal combustion process occurs and fuel is being combusted at its peak level. Engine designers tend to push the turbine entry temperature as high as possible in order to gain more power and optimize fuel consumption. This aim however, will jeopardize the functionality, and lifespan for the turbine stage components. The turbine stage, having components moving at high speed and being exposed to high temperatures can be regarded as the most crucial section in the gas turbine engine. Aiming for higher turbine entry temperature would eventually mean exceeding the temperature limitation for the turbine stage components material. Utilizing high temperature resistance materials such as nickel-based alloys and the implementation of turbine cooling systems allow the combustion gases to yield higher temperatures well exceeding the materials temperature limit.

1.3 Gas turbine cooling technologies

The variation of gas turbine engine can be classified into two major variations, the industrial gas turbines and aero-derivative gas turbine. Despite these variations, the gas turbine manufacturers share the same aim that is to increase the efficiency of the gas turbine engine. One of the ways to increase the efficiency of a gas turbine engine is to increase the turbine entry temperature. However, without utilising high temperature resistance materials and advanced cooling schemes, increasing the turbine entry temperature will only jeopardise the life span of the engine components. Development of high temperature resistance materials and advanced cooling schemes were introduced to increase the turbine entry temperature. Figure 1.1 shows higher turbine entry temperature achievable with the advancement of cooling technologies to gas turbine engine. The maximum allowable turbine entry temperature for uncooled turbines was in the temperature region of 1200 K, but with advanced cooling technologies, the turbine entry temperature may be increased further to 2000 K.

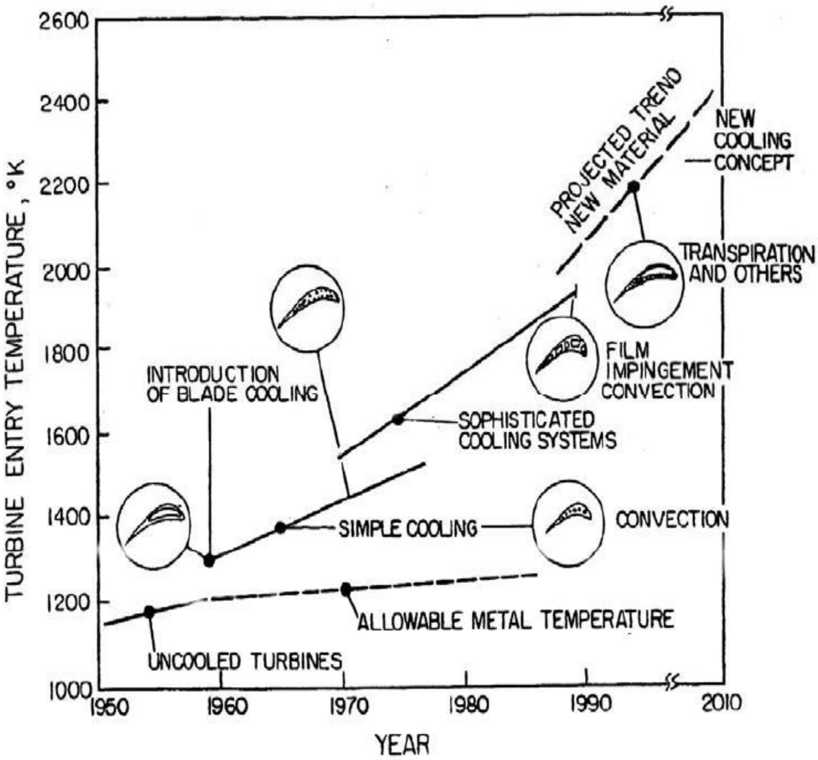

Figure 1.1: Advancement of cooling technology in gas turbine engine
(Source: Han et al., 2013)

The first involvement of gas turbine cooling technique was the introduction of turbine blade cooling. In the early stage of turbine blade cooling, cooling air, usually bled from the compressor stage, passes through internal passages in the turbine blade body to cool the turbine blade. This technique is well known as internal convection cooling technique. Promoting turbulence in the cooling passages enhances the heat transfer process. Further enhancement in the cooling technologies led the cooling air from the internal convective cooling arrangement to impingement cooling before ejecting them through small openings on the surface of the turbine blades. Impingement cooling involves impinging the coolant directly to the inner surface of the turbine blades. The basis of this technique is a flow with a lower temperature coming out from an orifice and cools the adjacent heated surface.

Air from the impingement cooling scheme, will be ejected through small openings on the surface of the turbine blades. Parts of these small openings are called film cooling holes. The ejected coolant through these film cooling holes will act as a buffer between the hot mainstream gas flow and the outer surface of the turbine blade. This thin layer of cooling air protects the turbine blades from a direct contact to the hot combustion gases. Arrangements of the film cooling holes determine the coverage area and the film cooling thickness. Another portion of the small openings for the ejected coolant air to the mainstream gas path is for transpiration cooling scheme. These holes are much smaller compared to the film cooling holes. This technique involves the idea of having the cooling air penetrating the turbine blade surface through porous walls. Even though known as one of the most efficient cooling technique, due to the very small size of the transpiration holes, there is a high possibility of the holes getting blocked which limits its applicability.

Research concerning cooling technologies is highly dependent on the heat transfer aspect. The effectiveness of a cooling technique is governed by the interaction between the cooling air and the hot gas stream. 1% addition of bled compressor stage air to the cooling channels may require 11K rise in turbine entry temperature to maintain the same level of power output from the gas turbine engine, Nikolaidis et al. (2020). An ideal cooling technique is a technique which utilises the least amount of bled air from the compressor and produces optimal cooling. Typically, up to 20% of the total engine mass flow was used for cooling purposes, Rolls Royce (1996). The interaction between the cooling air and the hot gas stream can be predicted based on reliable heat transfer experiments. In a heat transfer experiment, the useful parameters are the heat transfer coefficient and the adiabatic wall temperature.

1.4 Gas turbine heat transfer

The heat transfer coefficient shows the rate of heat energy being transferred from one state to the other. Heat will always flow from a high temperature region to a low temperature region. High heat transfer coefficient indicates large amount of heat transfer process. In the case of cooling a hot material, high heat transfer coefficient indicates better cooling process. Adiabatic means no heat transfer, thus an adiabatic wall temperature indicates a specific surface temperature when the temperatures between fluid and solid surface are in thermal equilibrium and there is no heat transfer process.

In actual gas turbine application, knowing the adiabatic wall temperature is important so that the turbine components would not operate over its metal temperature limit. Thus, adiabatic wall temperature could also signify the temperature limit for the turbine components. Under-prediction of adiabatic wall temperature may cause the components to work beyond its metal temperature limit. An increase of approximately 11 ˚C of metal temperature limit may reduce the lifespan of the components into half, Glezer (2003). Over-prediction of adiabatic wall temperature may keep the components well below the metal temperature limit but this also means excessive usage of cooling air. An effective cooling technique should allow the turbine components working within the metal temperature limit while utilising minimal amount of bled cooling air from the compressor.

Heat transfer experiments can be categorised into two conditions: steady state and transient. In steady state heat transfer experiments, the interaction between the hot and cold components can be said to be in a thermal equilibrium state. A condition where there will be no heat transfer process between the two components. Before reaching the thermal equilibrium, the heat transfer process is a function of time. This condition is called transient heat transfer. In transient heat transfer experiments, the important measurement will be the surface temperature history. From the surface temperature history data, the heat transfer coefficient and the adiabatic wall temperature can be calculated. Fourier one-dimensional heat equation is often used together with the semi-infinite solid condition to analyse the surface temperature history data to obtain the heat transfer coefficient and the adiabatic wall temperature.

The semi-infinite solid condition assumes heat conduction in one direction normal to the front surface of a solid without having significant heat loss through the back surface of the solid. If the temperature at the back surface changes more than 1% that of the front surface temperature, the semi-infinite solid condition became invalid. The heat transfer coefficient and adiabatic wall temperature calculated from this invalid condition will therefore be inaccurate and not reliable.

1.5 Problem statement

The general issue for researchers is to safely increase the turbine entry temperature without compromising the functionality of the components in the turbine stage. Introduction to the turbine cooling technique have made the increase in turbine entry temperature possible. The need for efficient cooling system in gas turbine engines is therefore crucial. An effective cooling system is achieved by conducting reliable heat transfer experiments which represent the real environment inside a gas turbine engine. These heat transfer experiments utilised reliable measurement methods using suitable sensing devices. The resultant experimental data will usually be analysed and validated based on standard correlation or previously published work by other researchers. Therefore, in order to conduct experiments concerning an effective cooling system, it is important to closely simulate the conditions of a gas turbine, utilising a reliable measurement method, and properly analyse the experimental data to obtain the necessary heat transfer parameter, the heat transfer coefficient and the adiabatic wall temperature.

In transient heat transfer experiments, researchers often analyse the temperature history data using the semi-infinite solid condition. The solution of Fourier one-dimensional heat equation requires temperature history data as the input. The problem arises when only part of the surface temperature history data can be used as an input to calculate the heat transfer coefficient and adiabatic wall temperature due to the considerably short experimental time. This is to keep the semi-infinite solid condition valid. Assume an insulation type material with a thermal diffusivity of α and a thickness of L, the experimental time limit, t_L, at which the back surface temperature changes by 1% that of the front surface temperature is given by (Shultz and Jones 1973);

$$t_L = \frac{L^2}{9\alpha} \tag{1.2}$$

Based on the material properties of 15 mm thick polycarbonate, the experimental time limit is approximately 188s before the back surface temperature changes more than 1% of its front surface temperature. The surface temperature history up to 160s will then be used to calculate the heat flux data using the boundary condition at $x = 0$. Due to the insufficient input, the typical heat flux versus surface temperature relationship graph would not lead to $q = 0$. Therefore, an extrapolation of the linear q versus T_s relationship based on curve fitting tool is necessary to estimate the T_{ad} at $q = 0$. However, this extrapolation is an error prone process. Therefore, an approach that could complement the time limitation and superposition of data will be proposed to accurately obtain the heat transfer coefficient and adiabatic wall temperature.

1.6 Objectives

The current field of research involves heat transfer studies in a gas turbine engine. The research work will focus on the investigation of techniques to obtain accurate result of heat transfer coefficient and adiabatic wall temperature for transient heat transfer case. A test rig must be designed to perform the experimentations. Properly calibrated temperature sensors are also crucial in order to perform the experimentations. Experiments involved will include both steady state and transient heat transfer conditions. Improvisations to the current experimental and analysis techniques will result in a more accurate calculation of heat transfer coefficient and adiabatic wall temperature.

The specific objectives are as follows:

a. To build a test rig for the heat transfer experiments and temperature sensor calibration.
b. To perform one dimensional semi-infinite heat transfer experiments to calibrate the functionality of the test rig.
c. To propose experimental and analysis method to provide more surface temperature data for accurate heat transfer analysis

1.7 Novelty of research work

Based on the proposed research objectives, the research novelty would be the temperature measurement and analysing method. This method will allow users to bypass the one dimensional semi-infinite heat transfer time limitation and subsequently acquiring a more accurate heat transfer coefficient and adiabatic wall temperature for gas turbine cooling technique application.

1.8 Scope of research

The scope of study in this work is based on a representative single stage gas turbine rig. The test area of the test rig will have the same dimensions as the wheel space area of the representative rig. Polycarbonate will be used as the target plate. Even though hard plastic type material has slower surface temperature response, their smooth surface condition excludes surface roughness effect in the boundary layer analysis for the test rig. Foam type material such as Rohacell has faster surface temperature response, but their rough surfaces may affect the fluid and thermal behaviour on the target surface. The blower used in the experiments will have a range of 1500 – 2500 because at lower and higher RPMs, the blower output will be unstable. This setting is also configured for the power supply to the mesh heater. The maximum power supply set is at 90A to avoid damaging the mesh heater. The temperature measurement method chosen for this work is a point measurement approach. Infrared sensors would be used to acquire the full surface temperature history from the wetted surface and the back surface of the target plate. This method was also chosen because it is suitable for the chosen alternative solution, the Crank Nicolson finite difference method.

The condition for the heat transfer process in this work would still involve part of the one dimensional semi-infinite heat transfer condition. The heat transfer process is assumed to be one dimensional, but the semi-infinite solid assumption would be neglected. The lateral heat transfer process at the back surface will not be considered in this work, as it will need thermal imaging or full surface area temperature sensor to monitor the heat transfer process at the back surface. This will result in additional analysing technique incorporated to the suggested Crank Nicolson solution.

1.9 Thesis layout

Chapter 1 – Introduction to gas turbine engine. The need of increasing turbine inlet temperatures to increase performance. Introduction to cooling techniques to support the continuous increase of turbine inlet temperature throughout the year. The importance of heat transfer experiments and analysing technique for the investigation of cooling techniques. Thesis problem statement and objectives.

Chapter 2 – Previous work related to heat transfer experiments. Includes the selection of temperature sensors involved in heat transfer experiments. Methods of other researchers in introducing heated air to induced the heat transfer experiments. Analysing techniques used by other researcher in obtaining the heat transfer coefficient and adiabatic wall temperature. The motivation for conducting the current research work.

Chapter 3 – The methodology of the experimental research work. Detailed explanation on the experimental test rig, selected temperature sensors and the data acquisition system. Elaboration on the heat transfer solution techniques that were involved in the current research work.

Chapter 4 – Results and discussions on the experimental findings. Commissioning test results on the characteristic of the blower, stainless steel mesh heater, and boundary layer analysis. The effect of having narrow channels in heat transfer experiments were presented. Justification on the capability of the test rig to conduct proper heat transfer experiments by conducting simple heat transfer experiments. Validating the proposed solution method against commonly used one dimensional semi-infinite heat transfer solution. Heat transfer results for the Crank Nicolson solution with modified back face boundary condition to allow longer experimental time.

Chapter 5 – The summary of the findings in the research work, the contribution gained from the results, and the possible future work that could be conducted to extend the current work.

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APPENDICES

Appendix A

Matlab coding for Crank Nicolson solution with adiabatic back face boundary condition

```
clearvars; clc; format long;
tic
% Define material properties - Polycarbonate

rho = 1200;        % density
k = 0.2;           % thermal conductivity
cp = 1250;         % specific heat

alpha = k/(rho*cp); % thermal diffusivity

L = 0.015; % test plate thickness

% Reads excel file for experimental data input

dn = xlsread('IR_temp_data_poly','Data','A5');
% experimental data time step interval

To = xlsread('IR_temp_data_poly','Data','B4');
% initial temperature, Ti = temp at node i

Tn = xlsread('IR_temp_data_poly','Data','B4:B1603');
% range of Ts values, Fo = 0.1
%Tn = xlsread('IR_temp_data_poly','Data','B4:B5903');
% range of Ts values, Fo = 0.35

N = length(Tn)/(1/dn); % end time @ experimental time  (s)

di = 0.001 %input('Input spatial interval step (m): ');
   % value of spatial step size in meters
   % 0.0001 --- d = 1.3333
   % 0.0003 --- d = 0.1481
   % 0.0005 --- d = 0.0533
   % 0.001  --- d = 0.0133

   % according to Ex 7.2-2, diffusion number must be <0.5

I = (L/di);        % total number of spatial step, nondimensional value
Icount = I-1;      % to satisfy known condition when n=1,
                   % initial boundary condition

d = (alpha*dn)/(di^2); % diffusion number for CN equations
```

```
Tilist = ones(I,1)*To; % define initial condition for all Ti when n=1 @ To
% define matrix and CN solver

step1 = [d     2*(1-d)       d]; % RHS equation

step2 = [-d    2*(1+d)       -d]; % LHS equation

mat1 = zeros(Icount,I); % set matrix size for RHS

mat2 = zeros(Icount,I); % set matrix size for LHS


for icount = 1:Icount
   % inserting the parameters for Eq C2 and C3 in the matrix
   % possibly becasue for 1-14, the number starts from 0-14

      if icount == Icount

         mat1(end,end) = 1-d;
         mat1(end,end-1) = d;

         mat2(end,end) = 1+d;
         mat2(end,end-1) = -d;

      else
         mat1(icount,icount:icount+2) = step1;
         mat2(icount,icount:icount+2) = step2;
         % here started at icount+1,
         % so that at 1, the values are left unchanged

      end

end

Tilistsave = Tilist;

for Tcount = 2:length(Tn);

   sol1 = mat1*Tilist;

   Tilist(1,1)=Tn(Tcount,1);

   Tilist(end,1)=To;

   Tilist(2,1)=quant((sol1(1,1)-((-
d)*Tilist(1,1))+(d*Tilist(3,1)))/(2*(1+d)),0.000001);

   Tilist(3,1)=quant((sol1(2,1)-((-
d)*Tilist(2,1))+(d*Tilist(4,1)))/(2*(1+d)),0.000001);
```

```
    Tilist(4,1)=quant((sol1(3,1)-((-
d)*Tilist(3,1))+(d*Tilist(5,1)))/(2*(1+d)),0.000001);

    Tilist(5,1)=quant((sol1(4,1)-((-
d)*Tilist(4,1))+(d*Tilist(6,1)))/(2*(1+d)),0.000001);

    Tilist(6,1)=quant((sol1(5,1)-((-
d)*Tilist(5,1))+(d*Tilist(7,1)))/(2*(1+d)),0.000001);

    Tilist(7,1)=quant((sol1(6,1)-((-
d)*Tilist(6,1))+(d*Tilist(8,1)))/(2*(1+d)),0.000001);

    Tilist(8,1)=quant((sol1(7,1)-((-
d)*Tilist(7,1))+(d*Tilist(9,1)))/(2*(1+d)),0.000001);

    Tilist(9,1)=quant((sol1(8,1)-((-
d)*Tilist(8,1))+(d*Tilist(10,1)))/(2*(1+d)),0.000001);

    Tilist(10,1)=quant((sol1(9,1)-((-
d)*Tilist(9,1))+(d*Tilist(11,1)))/(2*(1+d)),0.000001);

    Tilist(11,1)=quant((sol1(10,1)-((-
d)*Tilist(10,1))+(d*Tilist(12,1)))/(2*(1+d)),0.000001);

    Tilist(12,1)=quant((sol1(11,1)-((-
d)*Tilist(11,1))+(d*Tilist(13,1)))/(2*(1+d)),0.000001);

    Tilist(13,1)=quant((sol1(12,1)-((-
d)*Tilist(12,1))+(d*Tilist(14,1)))/(2*(1+d)),0.000001);

    Tilist(14,1)=quant((sol1(13,1)-((-
d)*Tilist(13,1))+(d*Tilist(15,1)))/(2*(1+d)),0.000001);

    Tilistsave(:,Tcount)=Tilist;

end

Tilistsave;
```

Appendix B

Matlab coding for Crank Nicolson solution with conduction-convection back face boundary condition.

```
clearvars; clc; format compact;
tic
% Define material properties - Polycarbonate

rho = 1200;       % density
k = 0.2;          % thermal conductivity
cp = 1250;        % specific heat

alpha = k/(rho*cp); % thermal diffus ivity

L = 0.015; % test plate thickness

% Reads excel file for experimental data input

dn = xlsread('IR_temp_data_poly','Data','A5');
% experimental data time step interval

To = xlsread('IR_temp_data_poly','Data','B4');
% initial temperature, Ti = temp at node i
Tob = xlsread('IR_temp_data_poly','Data','C4');

%Tn = xlsread('IR_temp_data_poly','Data','B4:B1603');
% range of Ts values, Fo = 0.1
%Tb = xlsread('IR_temp_data_poly','Data','C4:C1603');
% range of Ts values, Fo = 0.1

%Tn = xlsread('IR_temp_data_poly','Data','B4:B5904');
% range of Ts values, Fo = 0.35
%Tb = xlsread('IR_temp_data_poly','Data','C4:C5904');
% range of Ts values, Fo = 0.35

%Tn = xlsread('IR_temp_data_poly','Data','B4:B8404');
% range of Ts values, Fo = 0.5
%Tb = xlsread('IR_temp_data_poly','Data','C4:C8404');
% range of Ts values, Fo = 0.5

Tn = xlsread('IR_temp_data_poly','Data','B4:B11804');
% range of Ts values, Fo = 0.7
Tb = xlsread('IR_temp_data_poly','Data','C4:C11804');
% range of Ts values, Fo = 0.7

%Butterworth and Median Filtering
%For BF, the input data needs to start from 0.
%so all the data will be -To at first
```

```
[b,a] = butter(1,0.35); %set butterworth filter

Tnf1 = filter(b,a,Tn-To); %Tn Butterworth filtered

Tnf2 = medfilt1(Tnf1,1000); %Tn median filtered

Tnf3 = Tnf2+To;

Tbf1 = filter(b,a,Tb-Tob); %Tb Butterworth filtered

Tbf2 = medfilt1(Tbf1,1000); %Tb median filtered

Tbf3 = Tbf2+To;

N = length(Tn)/(1/dn); % end time @ experimental time  (s)

di = 0.001 %input('Input spatial interval step (m): ');
   % value of spatial step size in meters
   % 0.0001 --- d = 1.3333
   % 0.0003 --- d = 0.1481
   % 0.0005 --- d = 0.0533
   % 0.001  --- d = 0.0133

   % diffusion number must be <0.5

I = (L/di);          % total number of spatial step, nondimensional value
Icount = I-1;        % to satisfy known condition when n=1,
                     % initial boundary condition

d = (alpha*dn)/(di^2); % diffusion number for CN equations

Tilist = ones(I,1)*To; % define initial condition for all Ti when n=1 @ To
% define matrix and CN solver

step1 = [d      2*(1-d)       d]; % RHS equation

step2 = [-d     2*(1+d)      -d]; % LHS equation

mat1 = zeros(Icount,I); % set matrix size for RHS

mat2 = zeros(Icount,I); % set matrix size for LHS


for icount = 1:Icount
   % inserting the parameters for Eq C2 and C3 in the matrix

      if icount == Icount

         mat1(end,end) = 1-d;
         mat1(end,end-1) = d;
```

```
            mat2(end,end) = 1+d;
            mat2(end,end-1) = -d;

        else
            mat1(icount,icount:icount+2) = step1;
            mat2(icount,icount:icount+2) = step2;
            % here we started at icount+1,
            % so that at 1, the values are left unchanged

        end

    end

    Tilistsave = Tilist;

    for Tcount = 2:length(Tnf3);

        sol1 = mat1*Tilist;

        Tilist(1,1)=Tnf3(Tcount,1);

        Tilist(end,1)=Tbf3(Tcount,1); %needs to be read from Tb data

        Tilist(2,1)=quant((sol1(1,1)-((-
d)*Tilist(1,1))+(d*Tilist(3,1)))/(2*(1+d)),0.000001);

        Tilist(3,1)=quant((sol1(2,1)-((-
d)*Tilist(2,1))+(d*Tilist(4,1)))/(2*(1+d)),0.000001);

        Tilist(4,1)=quant((sol1(3,1)-((-
d)*Tilist(3,1))+(d*Tilist(5,1)))/(2*(1+d)),0.000001);

        Tilist(5,1)=quant((sol1(4,1)-((-
d)*Tilist(4,1))+(d*Tilist(6,1)))/(2*(1+d)),0.000001);

        Tilist(6,1)=quant((sol1(5,1)-((-
d)*Tilist(5,1))+(d*Tilist(7,1)))/(2*(1+d)),0.000001);

        Tilist(7,1)=quant((sol1(6,1)-((-
d)*Tilist(6,1))+(d*Tilist(8,1)))/(2*(1+d)),0.000001);

        Tilist(8,1)=quant((sol1(7,1)-((-
d)*Tilist(7,1))+(d*Tilist(9,1)))/(2*(1+d)),0.000001);

        Tilist(9,1)=quant((sol1(8,1)-((-
d)*Tilist(8,1))+(d*Tilist(10,1)))/(2*(1+d)),0.000001);

        Tilist(10,1)=quant((sol1(9,1)-((-
d)*Tilist(9,1))+(d*Tilist(11,1)))/(2*(1+d)),0.000001);
```

```
   Tilist(11,1)=quant((sol1(10,1)-((-
d)*Tilist(10,1))+(d*Tilist(12,1)))/(2*(1+d)),0.000001);

   Tilist(12,1)=quant((sol1(11,1)-((-
d)*Tilist(11,1))+(d*Tilist(13,1)))/(2*(1+d)),0.000001);

   Tilist(13,1)=quant((sol1(12,1)-((-
d)*Tilist(12,1))+(d*Tilist(14,1)))/(2*(1+d)),0.000001);

   Tilist(14,1)=quant((sol1(13,1)-((-
d)*Tilist(13,1))+(d*Tilist(15,1)))/(2*(1+d)),0.000001);

   Tilistsave(:,Tcount)=Tilist;

end

Tilistsave;
```

BIODATA OF STUDENT

Helmey Ramdhaney Mohd Saiah was born in Malacca, Malaysia on 18th June 1983. He has a bachelor's degree and a master's degree, both in aerospace engineering from Universiti Putra Malaysia. He is currently pursuing PhD degree in aerospace engineering at Universiti Putra Malaysia. His research interest is in gas turbine cooling technologies, mainly involving experimental techniques. He did turbine blade film cooling research during his bachelor's degree final year project and turbine blade impingement cooling research for his master's degree. His PhD research on the other hand, involved methodology of acquiring better experimental data and analysis of temperature data for gas turbine cooling heat transfer experiments. He worked as a tutor in the Department of Aerospace Engineering, Universiti Putra Malaysia from 2007 until 2020. His work scope involved assisting lecturers in lectures and laboratories for thermo-fluid, heat transfer, propulsion, and aerodynamics classes.

LIST OF PUBLICATIONS

Mohd Saiah, H.R., Mohd Rafie, A.S., & Romli, F.I. (2018). Freestream velocity correction in narrow channels. *Journal of Mechanical Engineering*, Vol SI 5(2), 54-66, ISSN 1823-5514.

Mohd Saiah, H.R., Mohd Rafie, A.S., Romli, F.I., & Mohd Harithuddin, A.S. (2018). Semi-infinite solid heat transfer limitation. *International Journal of Engineering & Technology*, 7(4.13), 146-150, DOI 10.14419/ijet.v7i4.13.21347

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