MASTER'S THESIS

Master of Science in Offshore Technology
(Marine and Subsea)

Validation of Heat Transfer Coefficients in Pipes and Deck Element without Ice Glazing

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Spring Semester, 2016

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Abstract

In recent years, there has been unprecedented interest shown in the Arctic region by the industry as it has become increasingly accessible for exploration. It has become quite common to have oil & gas field developments in such areas, which till a few decades ago posed serious challenges, one of the ongoing challenge is, how to minimize the heat loss from the piping system and deck elements with effective design and insulation. Engineering research in heat transfer studies and design of material suitable for low ambient temperature has progressed in right direction to instill confidence in operators that energy loss can be minimized.

This thesis tries to answer some of these queries by undertaking comprehensive study of the heat transfer phenomenon in horizontal pipes and deck elements. Detailed review of the available literature on heat transfer coefficients for pipes and plates subjected to cross flow wind were carried out to understand the current industry standards and establish a test methodology to determine heat transfer coefficients through experiments. A jig was designed for accommodating multiple pipes and carrying out the experiments at the climate laboratory capable of simulating subzero temperatures and cross flow wind, which was controlled and constantly monitored. Deck element for testing was free issued by GMC. In this thesis, cross flow wind of 5 m/s, 10 m/s and 15 m/s blowing over several single pipe and multiple pipe configurations of diameter 25 mm and 50 mm steel pipes with and without insulation were examined. The joint experiment with (Kvamme, 2016) involved more than 380 hours of testing at the climate laboratory. Detailed calculations were performed both manually and using programming code for theoretical and experimental readings to determine the effect of cross flow wind and insulation on the heat transfer coefficients.

A thorough comparison of the heat transfer coefficients determined experimentally and through theoretical methods using existing heat transfer correlations such as the Hilpert, Fand and Keswani, Morgan, Žukauskas, Whitaker and Churchill-Bernstein for horizontal pipes under cross-flow wind conditions showed that the values were in good agreement for the insulated pipes with the deviation in the range 0.5 - 2.82 % for diameter 50 mm insulated pipe and 12 -14 % for diameter 25 mm insulated pipe. Comparison of diameter 50 mm uninsulated and insulated pipe showed that the reduction in heat transfer coefficient is in the range of 400 - 4000 % with the usage of insulation material having low thermal conductivity.

However, in the case of uninsulated pipe and deck element, the values were substantially higher for experimental heat transfer coefficient values compared to theoretical results. The values were in the range 72 - 88 % and 17- 90 % respectively. Time to freeze results for diameter 25 mm and diameter 50 mm uninsulated and insulated pipes showed increase in time to freeze by 27 % and by 52 % with the usage of 10 mm and 25 mm insulation respectively in the case of diameter 25 mm pipe. For diameter 50 mm pipe, the time to freeze increased by 22 % and 47 % respectively for similar increase in insulation thicknesses. Based on the governing criteria and experimental findings, the Churchill-Bernstein correlation was suggested as the best method for use by the industry.

Keywords: Heat transfer correlations, overall heat transfer coefficient, cross-flow wind, flat plate, heat transfer, heat loss, convective heat transfer, insulated pipe, flat plate

Preface

This Master Thesis has been written during the spring semester at the University of Stavanger, 2016. The thesis has been submitted in partial fulfillment of the requirement for completing the degree of Master of Science, and has been executed at the Faculty of Science and Technology, Department of Mechanical and Structural Engineering and Materials Science in Stavanger, Norway. The process of writing this thesis has been extremely fascinating and academically challenging. The most challenging aspect of writing this thesis has been the innumerable hours spent on the experiment at the cooling laboratory in extremely cold condition in addition to the countless hours spent on designing and constructing the testing equipment required for performing the experiments. This has been a multi discipline thesis involving usage of mechanical engineering, electrical engineering, computer programming, 3D designing etc. for understanding and completion of the assigned scope of work. The entire journey has been quite demanding with its fair share of ups and downs. Despite the amount of effort and time required to finish this thesis, I am extremely satisfied with my choice of topic and the final outcome.

Jino Peechanatt, Stavanger, 14th June, 2016

Acknowledgement

This thesis would have been impossible without the support of some crucial people who guided and motivated me throughout the course of this extremely challenging project involving research, designing, experimentation in extremely cold condition, calculation etc. First of all, I would like to thank my partner Mr. Bjarte Odin Kvamme, who had collaborated with me for the designing of the test jig and conducting experiments at GMC's climate laboratory. His sound knowledge base, extremely friendly nature and willingness to accept any challenge gave me immense confidence and motivation to complete the tasks at hand.

I would like to thank GMC Maritime AS for allowing us to use the climate laboratory at their yard and helping us with the procurement of necessary materials. Mr. Oddbjørn Vassbø Hølland, Mr. Øystein Aasheim, Mr. Trond Spande and Mr. Knut Espen Solberg from GMC Maritime AS helped us from the beginning in clearly explaining the scope of work and the way forward when we faced numerous challenges throughout the course of this thesis.

Also, I am thankful to Professor Ove Tobias Gudmestad for considering me worthy enough to undertake this daunting task and for providing us with continuous guidance and support during all stages of the thesis especially his visit to the climate laboratory to motivate us and provide his valuable comments.

I am indebted to Mr. Yaseen Amith, Univeristy from the Department of Mechanical and Structural Engineering and Materials Science at University of Stavanger, who facilitated in the procurement of pipes and allowed full access to the mechanical workshop and 3D printer for completing our task and Mr. Romuald Bernacki from the Department of Electrical Engineering and Computer Science at University of Stavanger, who helped us in understanding the various aspects of designing and building the electrical circuits. Finally, I am grateful to the engineers and technicians at the mechanical laboratory and electronics laboratory, University of Stavanger, who explained the working of various machines and equipment to us which helped us to design and construct the testing jig. The guidance and support of all the above people has helped me during the writing of this report.

Table of Contents

List of Figures

List of Tables

Abbreviations

A	=	Area, m^2
A1	=	Inside area of the composite section, m^2
A2	=	Outer area of the composite section, m^2
bbl	=	Barrels
BOE	=	Barrels of oil equivalent
c_p	=	Specific heat at constant pressure, $J/kg \cdot K$
d/D	=	Diameter, m
D_o	=	Outer diameter, m
D_i	=	Inner diameter, m
g	=	Gravitational acceleration, m/s^2
GMC	=	GMC Maritime AS
h	=	Convective heat transfer coefficient, $W/m^2 \cdot K$
I	=	Electrical current, A
ID	=	Inner diameter, m
k	=	Thermal conductivity, $W/m \cdot K$
K	=	Degrees kelvin, unit of measurement
m	=	Mass, kg
NGL	=	Natural gas liquids
Nu_D	=	Nusselt number, dimensionless
OD	=	Outer diameter, m
p	=	Pressure, N/m^2
Pr	=	Prandtl number, dimensionless
q	=	Heat transfer rate, W
q'	=	Heat transfer rate per unit length, W/m
q''	=	Heat transfer rate per unit area, W/m^2
r_i	=	Inner radius, m
r_o	=	Outer radius, m
R_e	=	Electrical resistance, Ω

R_{th}	=	Thermal resistance, W/K
Re_D	=	Reynolds number, dimensionless
$Re_{x,c}$	=	Critical Reynolds number, 5×10^5
T_f	=	Film temperature, K
T_i	=	Internal temperature, K
T_∞	=	Ambient temperature, K
Ts	=	Surface temperature, K
t	=	Time, s
t_w	=	Wall Thickness, m
t_{ins}	=	Insulation Thickness, m
Tcf	=	Trillion cubic feet
u_s	=	Set wind velocity, m/s
u_m	=	Measured velocity, m/s
u_∞	=	Free-stream velocity, m/s
U	=	Overall heat transfer coefficient, W/m2 · K
V	=	Electrical voltage, V
YTF	=	Yet to find
α	=	Thermal diffusivity, m^2/s
δ	=	Hydrodynamic boundary layer thickness, m
δt	=	Thermal boundary layer thickness, m
ε	=	Emissivity, dimensionless
μ	=	Dynamic viscosity, kg/s · m
ν	=	kinematic viscosity, m^2/s
σ	=	Stefan-Boltzman constant, 5.6704×10^{-8} W/m^2K^4
η	=	Power efficiency, dimensionless
ρ	=	Density, kg/m^3

1 Introduction

1.1 General

Arctic Region is considered to be one of most important emerging frontiers of the oil and gas industry even though it is amongst the least understood in terms of familiar parameters. The Arctic region refers to a portion of the Earth which is above 66.5° N latitude. It encompasses approximately 6% of the globe's surface. The Arctic region consists of 1/3rd land, 1/3rd continental shelf, and 1/3rd waters which is deeper than 500 m (Budzik, 2009). The Arctic has shares of eight countries: Canada, Denmark (Greenland), Finland, Iceland, Norway, Russia, Sweden, and the United States as shown in Figure 1-1 below. There is no other region on Earth which is this large and has largely remain unexplored in terms of exploration and development to produce hydrocarbons. Governments and international operators have been initiating a lot of new exploration activities in the Arctic region over the years, due to declining production from mature oil fields worldwide and growing demand. (Spath, 2013)

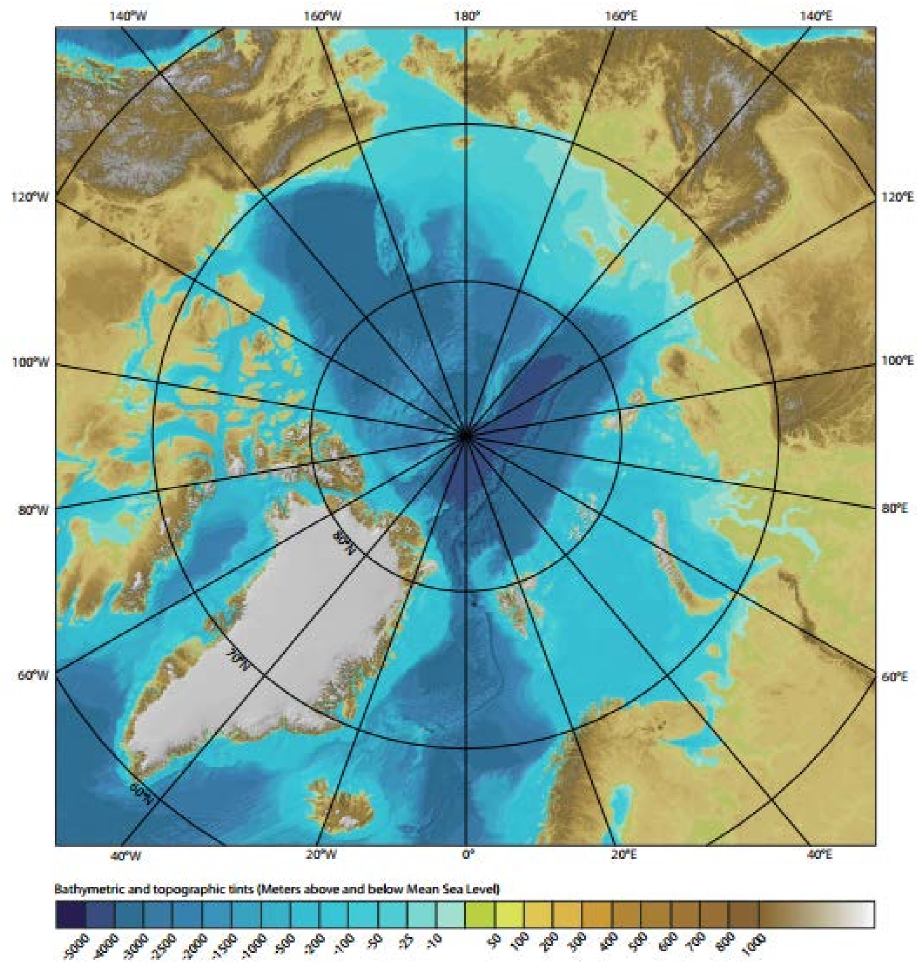

Figure 1-1 Region within the Arctic Circle
(North America is to the left and Eurasia is to the right)(National Geophysical Data Center, 2012)

There is considerable uncertainty surrounding the estimate of Arctic hydrocarbon resources due to the restricted amount of data from wells drilled throughout this huge region. As per the Circum-Arctic Resource Appraisal (CARA) performed by the US Geological Survey in 2008 using a probabilistic methodology of geological analysis and analog modeling, total undiscovered conventional hydrocarbon resources of 90 billion bbl of oil, 1,669 Tcf of natural gas, and 44 billion bbl of NGL i.e. a total of 412 billion BOE is yet to be found in the Arctic which constitutes vast 30% of the world's undiscovered gas and 13% of the undiscovered oil (Bishop et al., 2011)

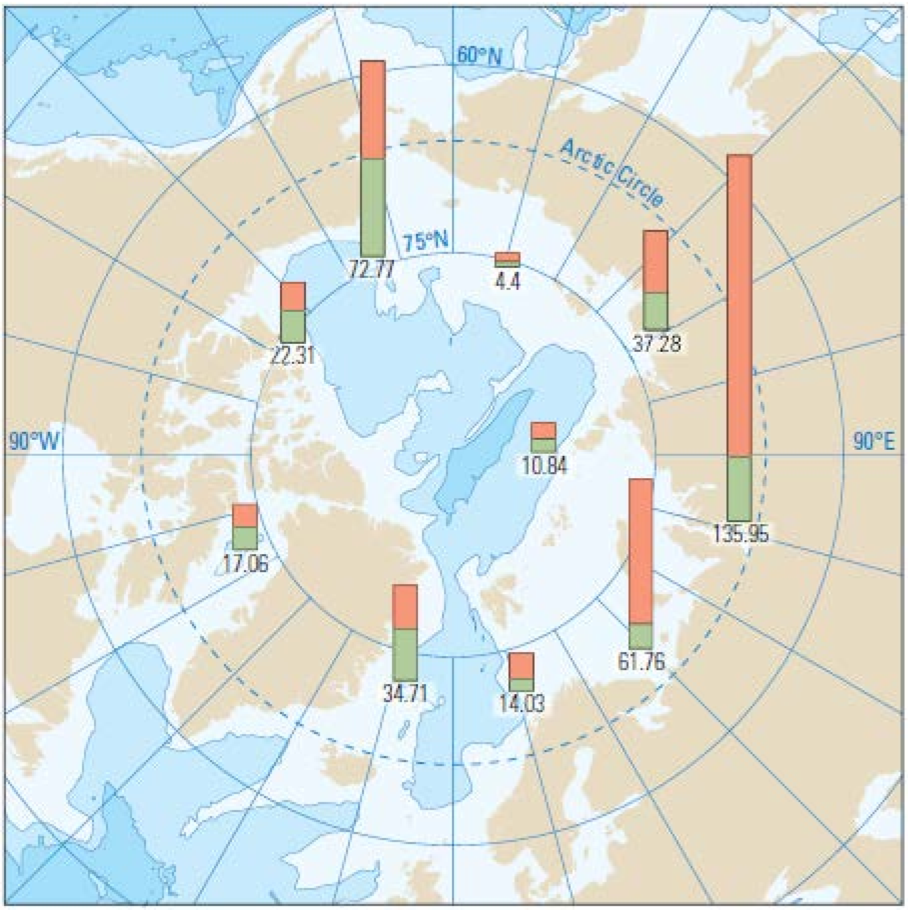

Figure 1-2 Yet-To-Find Arctic Resources in Billion Barrels of Oil Equivalent (Bird et al., 2008)

The map in Figure 1-2 shows the most promising areas for finding yet-to-find (YTF) or undiscovered conventional hydrocarbon resources. The height of each columns represents the volume of YTF resources i.e. red for gas and green for oil in billions of BOE. It is evident from the data that most of these undiscovered resources consist of natural gas in Russia. See Figure 1-3 which shows percentage of worldwide hydrocarbon resources in Arctic region .(Bishop et al., 2011)

However, only a few of the large Arctic fields which were discovered in the 1970s and 1980s have been developed until now, mainly because of high costs, major technical, environmental, and logistical challenges. One of the most important challenge is design an equipment to withstand extremely cold temperatures, strong wind, and severe ice conditions besides constant changes in weather which primarily interferes with the work schedules. Arctic region usually has a very short operating season of about 3 months per year. (Spath, 2013)

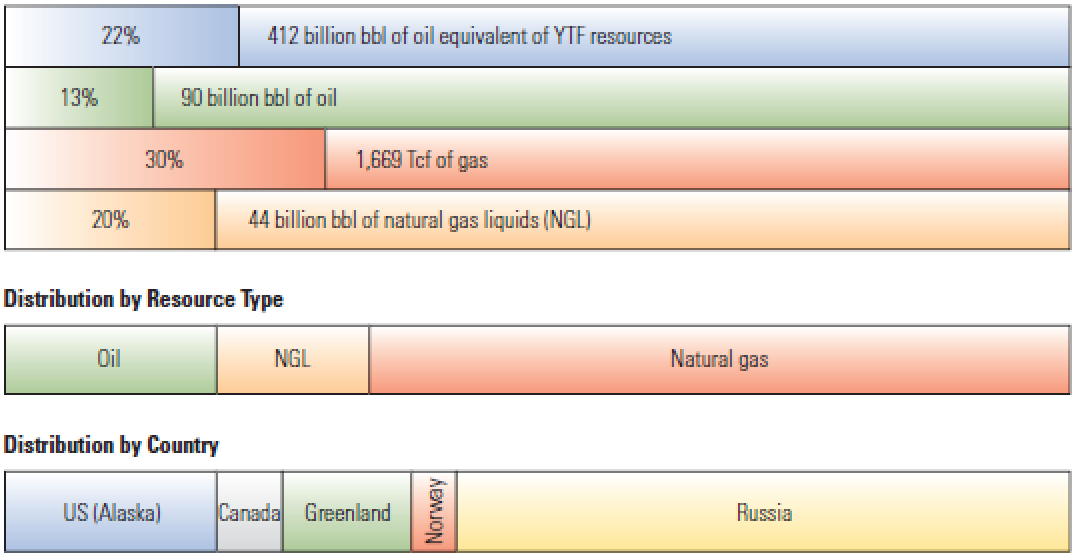

Figure 1-3 Arctic Region: Percentage of Worldwide Hydrocarbon Resources (Bird et al., 2008)

With the climate change rendering the Arctic region increasingly accessible to human intervention, there has been a significant increase in industry's interest in the region, whatever be the ultimate hydrocarbon reserves, it is evident that Arctic resources are adequate enough to attract enhanced exploration and development; it is estimated that over $100 billion could be invested in the development of Arctic over the next decade and Energy companies and service companies will be at the forefront of this investment.(Eurasia Group, 2014)

Development in the Arctic requires costly, customized technologies as well as precautions necessary for the extreme climatic conditions, it represents the final frontier in the conventional hydrocarbon development field. Finding these resources and bringing them to the customer could require another 20 years or more based on the current understanding. It is forgone conclusion that substantial investment and extensive exploration activity will be required to line up these resources as the next significant source of energy supply after the shale oil and shale gas. The best practices from countries like Norway and Russia have to be derived to overcome the technical and environmental challenges as they have been successful in exploration and development activity in their Arctic territories. Though, Norway cannot be truly classified as "Arctic" because of the absence of pack ice and permafrost. (Eurasia Group, 2014)

One of the crucial challenge in the Arctic is minimizing the heat loss occurring from pipes and deck elements because of the environmental conditions. The measures taken to minimize the energy loss from these elements play a significant role in the overall cost escalation and thus, are driving the research and development towards studies to find an optimized solution for this issue. Success in overcoming this challenge especially in these remote areas will solely depend on proper selection of best existing technologies and efforts in the development of more efficient ones. Also, the Arctic resource base largely contains natural gas and natural gas liquids, which are more challenging and expensive to transport than oil over long distances. Major development in liquefied natural gas (LNG) technologies has made natural gas increasingly available in markets far away from these regions. But, the advantage has so far primarily been realized by LNG plants which are built in low and middle latitude regions. (Budzik, 2009)

1.2 Tasks

1. Assess the relevant theoretical methods and industry standards used for describing the heat transfer from heated deck elements and for pipes exposed to a cross-flow wind arrangement. For pipes, insulation and heat transfer bridge (e.g. pipe supports) must be included in the methodology.

2. Based on the findings in Task 1, suggest the best method for use by the industry for describing the heat transfer from pipes and decks, and document the argumentation behind. The arguments below must be taking into consideration.
    a. Ease of use
    b. Range of validity
    c. Accuracy

3. Develop a test methodology for testing the heat transfer from the pipes and heated deck elements, conforming to industrial usage scenarios and perform experiments to validate the findings in Task 1. Heated deck elements for testing shall be obtained from GMC. The testing rig for the heat transfer from pipes needs to be designed, procured and assembled.

4. Define the deviation between the theoretical and experimental approaches for each case.

5. Develop tables describing the required time to freeze for different diameters and different degrees of insulation based on the theoretical approach, with correctional factors (if required) from the experimentation.

6. Based on findings from the theoretical and experimental approaches:
    a. Defined key elements to be considered for an optimal design of the deck elements
    b. Recommend a design that fulfils industry requirements

1.3 Scope of this Report

Oil and gas offshore production facilities, ships and LNG carriers operating in extreme cold climate and Arctic conditions require numerous design considerations and operational preparedness for intended purpose. Offshore winterization of equipment is considered to be one of the crucial aspects for ensuring 100 percent that a facility is fully capable of and appropriately prepared for the operations in Arctic condition and cold climates. During operational mode, the facility which is located in the cold temperature needs to have the piping equipment and deck required for safe working and commercial operation functional all the time and must be adequately designed to minimize risk of hazards against freezing, icing, and material properties (Conachey et al., 2007, IMO, 2016, DNV GL, 2015, Lee and Dasch, 2015)

This thesis will investigate the winterization issues on piping system and deck equipment surface based on the present relevant theories and industry standards pertaining to heat transfer and measures to reduce heat loss from pipes using insulation which is basically a low conductivity material applied to the pipes and through heat tracing in the case of deck elements. The aim in section 2 will be to compare all the relevant theories and suggest the best method which can be implemented by the industry for maximum output with minimal effort and cost.

The write up in section 3 will discuss about the designing of the testing jig and test methodology developed to study the actual heat transfer in pipes and deck elements including the simulation of the arctic condition in GMC's climate laboratory to get accurate results. Arduino programming code developed to get the surface temperature readings from the pipe surface as part of the test methodology is discussed. The experimental procedure is covered in detail to show the resemblance to the actual conditions. Thus, trying to validate and relate the theoretical and the practical aspect. The calculations based on the actual data obtained from the experiments conducted over the span of 3 months in the test facility is presented in section 4. Detailed calculation for heat transfer in insulated pipe, uninsulated pipe and deck element under strong cross flow wind conditions are part of section 4 of this report.

In Section 5, the results from all the experimental and theoretical calculations performed using python code and Microsoft Excel program are presented along with discussion. Also, tables which will describe the time to freeze for different diameter pipes i.e. 25mm and 50mm with varying thickness of insulation is also covered. Plots and tables comparing the overall heat transfer coefficient for uninsulated pipe, insulated pipe and deck element is part of section 5 while conclusions including recommendation of the best design suited for industrial use and requisite key elements to be considered for optimal design are covered under Section 6.

The objective of the thesis is to cover all the aspects as specified in the task list and identify winterization needs, design considerations and proper safeguards for pipes and deck element, considered to be important for operation and to safety of the personnel, environment and facility.

2 Theory

In order to determine the heat loss from the pipes and deck element under various scenarios, it is extremely important to establish an equation that would take into account all the environmental factors, which is extremely difficult. The calculation of heat loss from the pipe surface or deck element is in general not difficult unless there is a situation which involves wind flowing over the surface, in that case the equation becomes rather complex. In our case, there is wind flowing over the pipe surface and the deck element.

To get started with the process, it is important to establish the constants and calculations that were used and which all assumptions were made. Some concepts and ratios are fundamental to the heat transfer calculations which will later be performed, and a brief introduction is presented here.

2.1 Basic Concept

The basic principle behind this whole experiment revolves around the concept that any substance that is warmer than the surrounding it is placed in, it will transfer energy in the form of heat to the surroundings until the material and surroundings are in equilibrium with each other, this is the result of the temperature difference (Second Law of Thermodynamics). Heat transfer mechanisms are divided into following three types:

1. Conduction
2. Convection
3. Thermal radiation

These different types of heat transfer mechanisms are shown in Figure 2-1.The method of conduction is generally used to describe the heat transfer that happens when a temperature gradient is present in a solid or fluid medium. The method of convection describes the heat transfer that will occur between a surface and a moving fluid when they are at different temperatures. In thermal radiation, electromagnetic waves will transfer energy between different surfaces, unless an obstructing medium is introduced and we are aware that all surfaces that has a temperature, will continue to emit energy to the surroundings in the form of electromagnetic waves. (Incropera et al., 2006)

2.1.1 Conduction

In conduction, there is transfer of energy from higher energy particles of a substance to the adjacent lower energy particles as a result of the interactions between the particles. Conduction can happen in solids, liquids and gases. In the case of liquids and gases, conduction happens because of the collision and diffusion of the molecules during their motion which is random (Çengel, 2006). The property of the material which governs how effective the object will transfer the thermal energy to the adjacent object is thermal conductivity. Metals are considered to be very good conductors of heat. For one dimensional steady state heat conduction, conductive heat transfer is obtained from Fourier's law of heat conduction and presented in (2.1).

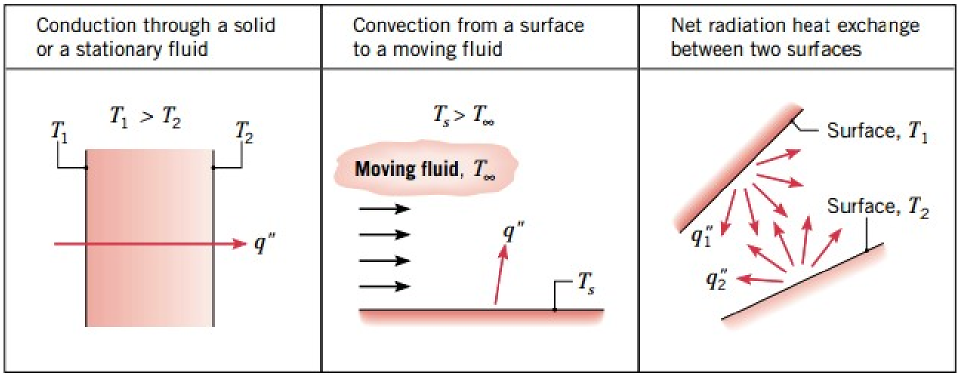

Figure 2-1 Conduction, Convection and Thermal Radiation (Incropera et al., 2006)

$$q_{cond} = -kA\frac{dT}{dx} \tag{2.1}$$

Where, dT /dx is defined as the temperature gradient. Under steady-state conditions, the temperature gradient because of linear temperature distribution can be written as:

$$\frac{dT}{dx} = \frac{T_{2-}T_1}{L} \tag{2.2}$$

Based on equation (2.1) given above, conductive heat transfer through a pipe wall can be formulated. Assuming a pipe with constant thermal conductivity for the pipe wall and no heat propagation through the wall and having the below parameters.

- r_i is the inner radius
- r_o is the outer radius
- L is the length
- Thermal conductivity, k is the Thermal conductivity
- T_i is the internal temperature
- T_∞ is the external temperature

Fourier's law of heat conduction applied to a pipe wall can then be expressed as:

$$q_{cond,cyl} = -kA\frac{dT}{dr} \tag{2.3}$$

Where A = 2πrL is the surface area (heat transfer) at radius r.

Equation (2.3), after rearrangement and integration with respective boundary conditions gives:

$$\int_{r1}^{r2} \frac{q_{cond,cyl}}{A} dr = -\int_{T_i}^{T_\infty} kdT \tag{2.4}$$

Equation (2.4), after inserting formulae for the surface area gives:

$$q_{cond,cyl} = 2\pi Lk \frac{T_{i-}T_\infty}{\ln(r_0/r_i)} \tag{2.5}$$

2.1.2 Convection

In convective heat transfer, there is the transfer of energy by a fluid which is in motion. Convective heat transfer is of two types: Forced convection and natural convection. Forced convection is when an external medium such as a blower, fan, pump or other agent passes air over the surface. Natural convection takes place when there is no fluid movement happening over the surface of the object. The change in temperature of the fluid medium results in the change of the density of the fluid medium, causing circulation effect, due to buoyancy effect as the dense fluid falls, and the light or warm fluid rises. This thesis deals with only forced convection as cross-flow wind is considered. The formulae for convective heat transfer rate is shown in equation (2.6)

$$q = hA\,(T_i - T_\infty) = \frac{T_{i-}T_\infty}{(1/hA)} \tag{2.6}$$

Parameters are:

- h is the convective heat transfer coefficient
- A is the surface area,
- T_i is the internal temperature
- T_∞ is the external temperature

2.1.3 Thermal Radiation

Thermal radiation is the energy which is emitted by any object which is at non-zero temperature (Holman, 2010) The formula for heat transfer rate in radiation is shown below:

$$q = \varepsilon\sigma A\,(T_i^4 - T_\infty^4) \tag{2.7}$$

Parameters are:

- ε is the emissivity and depends on the geometry and properties of the surface.
- σ is Stefan-Boltzmann constant

- A is defined as the surface area,
- Ti is the internal temperature,
- T∞ is the external temperature,

2.1.4 Thermal resistance

The concept of thermal resistance can help to greatly simplify otherwise complex heat transfer problems. Many physical phenomena can be described by the general equation shown below (Serth, 2007).

$$Flowrate = \frac{Driving\ force}{Resistance} \tag{2.8}$$

Ohm's Law in electricity follows this general equation.

$$I = \frac{V}{R_e} \tag{2.9}$$

Heat transfer uses the same principle. In heat transfer, flow rate is the heat. Temperature difference between the object and the surroundings is the driving force, and thermal resistance is the resistance offered to the flow, which is denoted by R_{th}. From this, equation (2.10) is obtained, which is the governing equation in the heat transfer calculations which will be done in the calculation section.

$$q = \frac{dT}{R_{th}} \tag{2.10}$$

It is to be noted that the principle is the same as Ohm's Law of electricity and thus, the thermal resistance can be specified in the same way as electrical resistance.

Therefore, for series arrangement, the total resistance is given by equation (2.11) and equation (2.12) shows the total resistance in parallel,

$$R_{tot} = \sum_i R_i \tag{2.11}$$

$$R_{tot} = \left(\sum_i \left(\frac{1}{R_i}\right)\right)^{-1} \tag{2.12}$$

Figure 2-2 shows the implementation of the same. In this figure, there is a cross section of the composite material having four different materials with different value of thermal resistances,

The total value of thermal resistance is:

$$R_{th,\,tot} = R_A + R_{BC} + R_D$$

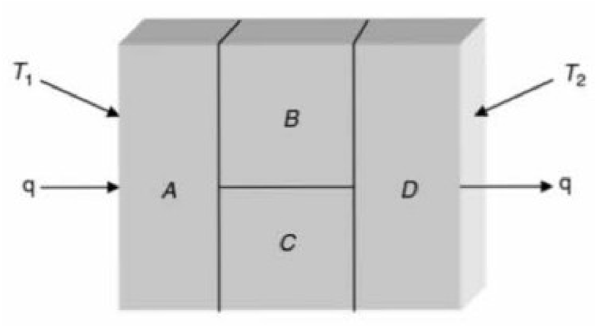

Figure 2-2 Heat transfer through a composite material (Serth, 2007)

Where R_{BC} is:

$$R_{BC} = \left(\frac{1}{R_B} + \frac{1}{R_C}\right)^{-1} = \frac{R_B R_c}{R_B + R_C}$$

Using the principle of thermal resistance, previously explained in equation (2.5) for conduction, can be rewritten as:

$$q_{cond,cyl} = \frac{T_{1-}T_2}{R_{cond,cyl}}$$

Where, $R_{cond,cyl}$ is the thermal resistance for the pipe layer, given as:

$$R_{\ cond,cyl} = \frac{\ln(r_2/r_1)}{2\pi Lk} \tag{2.13}$$

For convection, $R_{conv,cyl}$ is given as:

$$R_{\ conv,,cyl} = \frac{1}{2\pi rLh} \tag{2.14}$$

2.1.5 Distribution of Temperature in a composite cylindrical wall

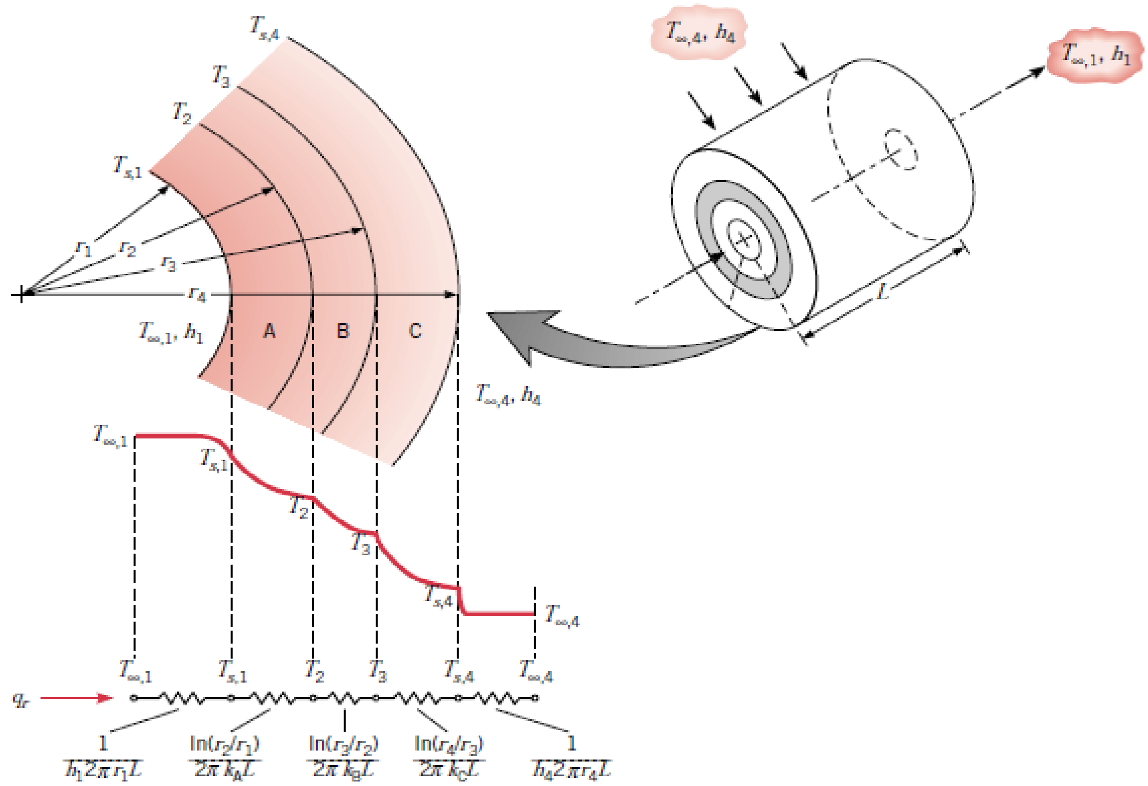

Figure 2-3 Temperature distribution for a composite cylindrical wall (Incropera et al., 2006)

In the case of a composite system having multiple layers, we neglect the interfacial contact resistances and the heat transfer can be expressed as below

$$q_r = \frac{T_{\infty,1} - T_{\infty,4}}{\frac{1}{2\pi r_1 L h_1} + \frac{\ln(r_2/r_1)}{2\pi K_A L} + \frac{\ln(r_3/r_2)}{2\pi K_B L} + \frac{\ln(r_4/r_3)}{2\pi K_c L} + \frac{1}{2\pi r_4 L h_4}} \tag{2.15}$$

The above equation can be presented in terms of the overall heat transfer coefficient form as shown in equation (2.16)

$$q_r = \frac{T_{\infty,1} - T_{\infty,4}}{R_{tot}} = UA(T_{\infty,1} - T_{\infty,4}) \tag{2.16}$$

The overall heat transfer coefficient U can be defined in terms of the inside area of the composite section, $A_1 = 2\pi r_1 L$, equating (2.15) and (2.16) will give (Incropera et al., 2006)

$$U_1 = \frac{1}{\frac{1}{h_1} + \frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{K_B}\ln\frac{r_3}{r_2} + \frac{r_1}{K_c}\ln\frac{r_4}{r_3} + \frac{r_1}{r_4}\frac{1}{h_4}} \tag{2.17}$$

2.1.6 Nusselt number

The Nusselt number is a dimensionless number and provides a measure of the convection coefficient, or the ratio of convection to pure conduction heat transfer. The equation for Nusselt number is shown below, where D is characteristic length of the surface, diameter for pipe. (Kothandaraman, 2006).

$$Nu_D = \frac{hD}{k} \tag{2.18}$$

2.1.7 Prandtl number

The Prandtl number is a dimensionless number and shows the ratio of momentum diffusivity and thermal diffusivity. It provides a measure of the relative effectiveness of momentum and energy transport by diffusion in the velocity and thermal boundary layers. The equation to find the Prandtl number is presented below (Incropera et al., 2006).

$$Pr = \frac{C_p \mu \nu}{k \propto} \tag{2.19}$$

2.1.8 Reynolds number

The Reynolds number is a dimensionless number and shows the ratio of inertia to viscous forces, and can be used to characterize the flows at the boundary layer. The Reynolds number is defined below. (Moran et al., 2003).

$$Re_D = \frac{\rho u_\infty D}{\mu} = \frac{u_\infty D}{\nu} \tag{2.20}$$

As shown in Figure 2-4 presented in section 2.2.2, the transition between laminar and turbulent flow takes place at an arbitrary location x_c. This is important when calculating the behaviour at the boundary layer. This location is found from the critical Reynolds number, $Re_{x,c}$ which varies from 1×10^5 to 3×10^6, depending on the turbulence level of the air and surface roughness, a value of 5×10^5 is frequently used. The formulae for the critical Reynolds number is shown below.(Incropera et al., 2006)

$$Re_{x,c} = \frac{\rho u_\infty x_c}{\mu} \tag{2.21}$$

2.1.9 Film temperature

The term film temperature was formulated by (Çengel, 2006) in order to account for the variation in thermodynamic properties with temperature. It is defined as the average of the surface and ambient temperature. Fluid properties are assumed to be constant during the entire flow when considering the film temperature. The equation is shown below in (2.22).

$$T_f = \frac{T_s + T_\infty}{2} \tag{2.22}$$

2.2 Heat transfer correlations

2.2.1 Forced flow over a cylinder in cross-wind

In order to calculate the convective heat transfer coefficient of a cylinder in cross-flow wind, a correlation is required. There are many correlations that can be used, with wide applicability and accuracy. (Incropera et al., 2006) suggests an accuracy of ±20% using this correlation whereas (Moran et al., 2003) has put the expected accuracy in the range ±25-30%.

There have been many comparisons of the different correlations. (Morgan, 1975) had done a detailed review of the existing literature on convective heat transfer. (Manohar and Ramroop, 2010) carried out a comparison study of five different correlations using experimental findings on inclined pipes at different wind speeds. Later some errors were found in the constants used by them for some of the correlations. (Whitaker, 1972) carried out an elaborate review of different correlations and reviewed them based on comparative plots.

2.2.1.1 Hilpert correlation

This correlation was suggested by (Hilpert, 1933), and provides a good estimate for the average Nusselt number for a pipe in a cross-flow wind arrangement. The Hilpert correlation is presented in equation (2.23) (Çengel, 2006; Incropera et al., 2006; Moran et al., 2003). The constants which were originally proposed by Hilpert are presented in Table 2-1. But, they have been revised based on new and more accurate thermodynamic values which has emerged from research work over time. The constants shown in Table 2-2 are proposed for use by (Çengel, 2006; Incropera et al., 2006; Moran et al., 2003).

$$\overline{Nu_D} = C{Re_D}^m Pr^{1/3} \tag{2.23}$$

$$[Pr \geq 0.7]$$

All properties in Hilpert correlation **are** evaluated at film temperature.

(Fand and Keswani, 1973) proposed different values for the constants used in Hilpert's correlation when more accurate values for thermodynamic properties of air became available over time with further research in heat transfer. All properties for the Hilpert's correlation are evaluated at film temperature. The constants proposed by (Fand and Keswani, 1973) are shown in Table 2-3.

(Morgan, 1975) recommended different values for the constants used in the Hilpert correlation based on a detailed review and analysis of available literature on convective heat transfer. The revised values proposed by Morgan are found in Table 2-4.

Table 2-1 Originally proposed constants by (Hilpert, 1933)

Re_D	C	m
1 - 4	0.891	0.330
4 - 40	0.821	0.385
40 - 4 000	0.615	0.466
4 000 - 40 000	0.174	0.618
40 000 - 400 000	0.0239	0.805

Table 2-2 Revised constants for Hilpert correlation (Çengel, 2006; Incropera, DeWitt, Bergman, & Lavine, 2006; Moran, Shapiro, Munson, & DeWitt, 2003).

Re_D	C	m
0.4 - 4	0.989	0.330
4 - 40	0.911	0.385
40 - 4 000	0.683	0.466
4 000 - 40 000	0.193	0.618
40 000 - 400 000	0.027	0.805

Table 2-3 Proposed values of C and m by (Fand & Keswani, 1973)

Re_D	C	m
1 - 4	0.875	0.313
4 - 40	0.785	0.388
40 - 4 000	0.590	0.467
4 000 - 40 000	0.154	0.627
40 000 - 400 000	0.024	0.898

Table 2-4 Proposed values of C and m, by (Morgan, 1975).

Re_D	C	m
0.0001 - 0.004	0.437	0.0895
0.004 - 0.09	0.565	0.136
0.09 - 1	0.800	0.280
1 - 35	0.795	0.384
35 - 5 000	0.583	0.471
5 000 - 50 000	0.148	0.633
50 000 - 200 000	0.0208	0.814

2.2.1.2 Žukauskas correlation

(Žukauskas, 1972) proposed the correlation shown in equation (2.24). All the properties in this correlation are found at the ambient temperature, except for Prandtl number Pr_s, which is obtained at the surface temperature.

$$\overline{Nu_D} = C Re_D{}^m Pr^{\,n} \left(\frac{Pr}{Pr_s}\right)^{1/4} \tag{2.24}$$

$$\begin{bmatrix} 1 \le Re_D \le 1 \times 10^6 \\ 0.7 \le Pr \le 500 \end{bmatrix}$$

The constants used in the above correlation are presented in Table 2-5 & Table 2-6

Table 2-5 Values of n for different Prandtl numbers by (Žukauskas, 1972)

Pr	n
< 10	0.37
≥ 10	0.36

Table 2-6 Proposed values of C and m by (Žukauskas, 1972)

Re_D	C	m
1 - 40	0.75	0.4
40 - 1 000	0.51	0.5
1 000 - 200 000	0.26	0.6
200 000 - 1 000 000	0.076	0.7

2.2.1.3 Whitaker correlation

(Whitaker, 1972) presented the correlation shown in equation (2.25).

$$\overline{Nu_D} = \left(0.5 Re_D{}^{1/2} + 0.06 Re_D{}^{2/3}\right) Pr^{\,0.4} \frac{\mu_b}{\mu_s}^{1/4} \tag{2.25}$$

$$\begin{bmatrix} 1.00 \le Re \le 1 \times 10^5 \\ 0.67 \le Pr \le 300 \end{bmatrix}$$

Where, μ_b is the fluid viscosity at ambient temperature and μ_s is the fluid viscosity at surface temperature. (Whitaker, 1972) observed that usually this correlation is within ±25% of other correlations, except at lower value of Reynolds numbers, where the Hilpert correlation gives significantly higher values.

2.2.1.4 Churchill-Bernstein correlation

(Churchill and Bernstein, 1977) proposed the correlation shown in equation (2.26) and this provided a single, comprehensive equation for the calculation of heat transfer coefficient of a pipe subjected to cross-flow wind. This is applicable for almost all ranges of Reynolds numbers, and a broad range of Prandtl numbers. There are no look up tables for constants unlike other correlations. All fluid properties in this correlation are evaluated at film temperature.

$$\overline{Nu_D} = 0.3 + \frac{0.62 Re^{1/2} Pr^{1/3}}{\left(1 + (0.4/Pr)^{2/3}\right)^{1/4}} \times \left[1 + (Re/282000)^{5/8}\right]^{4/5} \tag{2.26}$$

$$[Re_D Pr \geq 0.2]$$

2.2.1.5 Discussion

(Incropera et al., 2006) recommends use of the Žukauskas and the Churchill-Bernstein correlations as they are have wider applicability and were developed in recent times compared to other correlations. The Churchill-Bernstein correlation is recommended by (Moran et al., 2003) unless the simplicity of the Hilpert equation is advantageous. (Çengel, 2006) also recommends the use of the Churchill-Bernstein correlation, while (Theodore, 2011) recommends the use of Hilpert correlation, However, it has to be noted that all the correlations have their applicability under some range of Reynolds number and Prandtl number and it is difficult to predict which correlation is more accurate than others. Also, the wind speed experienced in practical cases is much lower than 20 m/s and considering a maximum diameter of 1.0 m for the pipe, it is observed that the Reynolds number will not increase beyond 400,000, which is the maximum applicability limit of the Hilpert's correlation. This means that Morgan's constants cannot be used in the Hilpert correlation as it is applicable only up to Reynolds number of 200,000 besides the Whitaker correlation which has applicability only up to Reynolds number 100,000.

None of the correlations are difficult to implement for practical purposes with the availability of programming code and Microsoft Excel. Some correlations like the Hilpert's and Žukauskas's correlations employ look up tables for the constants which are not required in the case of the Whitaker and the Churchill-Bernstein correlations. So, the choice of correlation depends on specific conditions and the accuracy of the results obtained by using them.

2.2.2 Forced flow over a flat plate

For heat transfer in a flat plate which is subjected to forced flow, it is crucial to understand the development of wind over the surface. Figure 2-4, shows different stages of flow over the surface. Laminar flow is seen during the first stage which will change to a transitional flow prior to becoming b turbulent. The Nusselt number calculation varies for laminar and turbulent flow and the equation is shown below. For laminar flow, equation (2.27) is used and equation (2.28) is used for transitional and turbulent flows (Incropera et al., 2006).

$$\overline{Nu_D} = \frac{\overline{h_D} D}{k} = 0.664 {Re_D}^{1/2} Pr^{1/3} \tag{2.27}$$

$$[Pr \geq 0.6]$$

$$\overline{Nu_D} = (0.037 {Re_D}^{4/5} - A) Pr^{1/3} \tag{2.28}$$

$$\begin{bmatrix} Re_{x,c} \ \leq \ Re_D \ \leq \ 1 \times 10^8 \\ 0.6 \ \leq Pr \leq 60 \end{bmatrix}$$

Where, the value of the constant A is determined by the critical Reynolds number $Re_{x,c}$. The formulae for finding A is shown in equation (2.29). Generally, a value of 5×10^5 is used for $Re_{x,c}$ and the value of A is found to be 867

$$A = \left(0.037 {Re_{x,c}}^{4/5} - 0.664 {Re_{x,c}}^{1/2}\right) \tag{2.29}$$

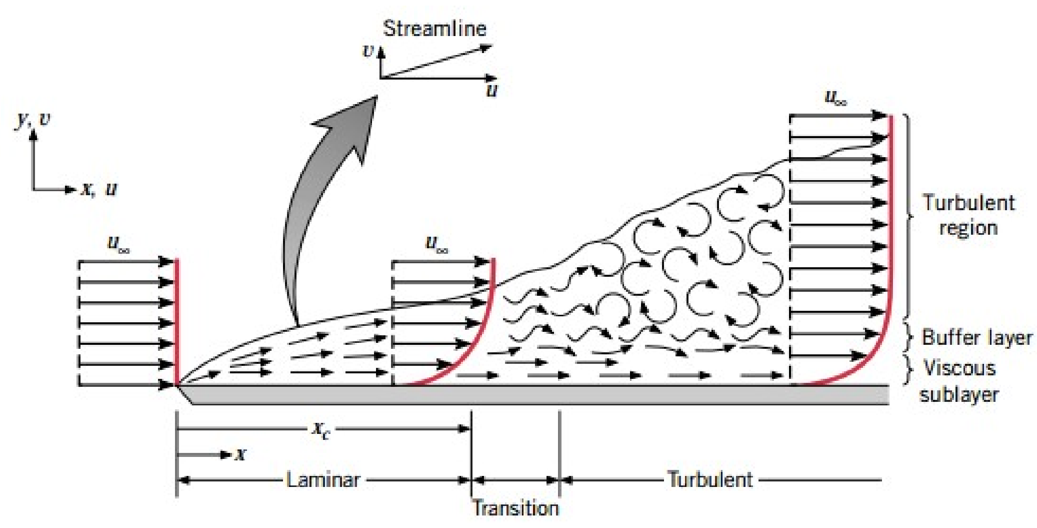

Figure 2-4 Velocity boundary layer development over a flat plate (Incropera et al., 2006)

2.3 Time to freeze

The method used for generating time to freeze tables was taken from chapter (19-20) provided in (ASHRAE, 2010) refrigeration handbook. In general, the method described in the book was for freezing of foods and beverages. But, it was implemented for time to freeze for pipes in the programming code with minor changes. The values obtained showed good agreement to the actual cases as understood from other literatures.

The following process is suggested by (ASHRAE, 2010):

1. Obtain the relevant thermal properties from the tables.
2. Calculate the surface heat transfer coefficient.
3. Calculate characteristic dimensions and ratios along with the Biot, Plank and Stefan numbers from relevant formulas.
4. Compute the freezing time for an infinite slab and equivalent heat transfer.
5. Compute the freezing time

Since, the method is cumbersome, it is directly implemented in code to generate the time to freeze tables for different diameter pipes and varying insulation thickness and manual calculations were not performed. The (ASHRAE, 2010) Refrigeration handbook suggests various methods and correction factors for individual cases. So, it is recommended to confer with the handbook for specific cases.

2.3.1 Biot number

The Biot number is defined as the ratio of the external heat transfer resistance to the internal heat transfer resistance. The formula is shown in equation (2.30)

$$B_i = \frac{hL}{k} \tag{2.30}$$

2.3.2 Plank number

The Plank number is defined as the ratio between the volumetric specific heat of the unfrozen phase and the volumetric enthalpy change. The formula is shown in equation (2.31)

$$Pk = \frac{C_l(T_i - T_f)}{\Delta H} \tag{2.31}$$

2.3.3 Stefan number

The Stefan number is defined as the ratio between the volumetric specific heat of the frozen phase and the volumetric enthalpy change. It is similar to the Plank number. The formula is shown in the below equation (2.32)

$$Ste = \frac{C_s(T_i - T_f)}{\Delta H} \tag{2.32}$$

3 Experiments

3.1 Test Apparatus

A rectangular testing jig was designed and built to experimentally determine the average heat transfer coefficient *h* for circular pipes in cross flow wind arrangement. The apparatus was designed to accommodate multiple circular pipes of varying diameters (50 mm and 25 mm) one behind another as shown in Figure 3-1(a) as one of the main aim of the testing was to find the effect of cross flow wind on the adjacent pipes. The dimension of the jig was 110 cm (L) x 66 cm (W) x 100 cm (H) and the height of the horizontal section, for the placement of the pipes, can be adjusted to allow for the direct impact of the cross flow wind from the tunnel. The wind tunnel for simulating cross flow wind was 110 cm wide and 160 cm long as seen in Figure 3-1 (b) except for the tapered section which was to be connected to the wind turbine via 0.5 m hose to complete the test assembly. The wind tunnel supplied by GMC was assembled in the cooling laboratory as per the height of the testing jig. In this arrangement, as shown in Figure 3-2 the wind flowed transversely across the test specimen. One of the main governing factor behind the design was the portability factor as the jig had to be moved to offshore for testing. So, angle section with predefined holes for nuts and bolts were used for the ease of assembly and it was fixed on to a pallet for the ease of shifting. The climate laboratory at GMC's yard is 3.6 m wide and 11 m long and easily accommodated the testing jig. (Manohar and Ramroop, 2010)

Figure 3-1 a) Test rig mounted on a pallet and b) Wind tunnel

The pipes were held in place using clamps having rubber lining. These clamps were adjustable for fine alteration of height and can be used for a small range of pipe diameters. The steel pipes with diameter 50 mm and 25 mm having wall thickness of 2 mm were procured in 6m length and cut to a length of 120 cm using mechanical saw. See Appendix C showing the purchase order for the steel grade and dimension. The steel grade used was DIN 2394. The 3D printing laboratory in the University of

Stavanger was used to make end caps for the pipes which were designed in OpenSCAD software. Heating elements procured from (RS Components AS, 2016) was 143 cm long and was made with Incoloy (Nickel Iron Chromium Alloy) having power rating of 1000W at 240V. It was used to create a uniform heat flux inside the pipes. See Appendix E for further details of heating element.

Figure 3-2 Testing Arrangement for Pipes and Deck Element

The straight heating elements were permanently installed inside the pipe through the end cap using silicon sealant. The output of the heating element was controlled using a variac as the rated power was much higher than our requirement. A variac is basically a variable transformer which regulates the voltage input and thus, the power output which is proportional to the voltage as the resistance of each heating element is constant. The resistances for each element was measured, and are presented in Table 3-1 below.

Table 3-1 Resistances of heating elements.

Pipe No:	Resistance (Ω)
25mm #1	57.1
25mm #2	58.9
25mm #3	57.6
50mm #1	58.2
50mm #2	57.6
50mm #3	58.6

The measured resistance of each element is used to find out the total resistance of the system based on the pipe combination and Ohms law of resistance applicable for parallel loads as explained in equation (2.12). The actual current and voltage across the heating elements were checked using hand held multimeters and the total heat output wascalculated using the equation.

3.2 Test Specimen

The steel pipes of diameter 50 mm and 25 mm having wall thickness of 2 mm with electric heating element were used for the testing as shown in Figure 3-3 (a). Pipes had insulation with thermal conductivity of 0.033 W/m. K as can be seen in the data sheet for insulation, see Appendix D. The Deck element shown in Figure 3-3 (b) below was company issued and had thermocouples for the temperature readings. But, the infrared camera was available to measure the surface temperatures. The elements were placed in such a way as to have a cross flow impact of wind from the tunnel.

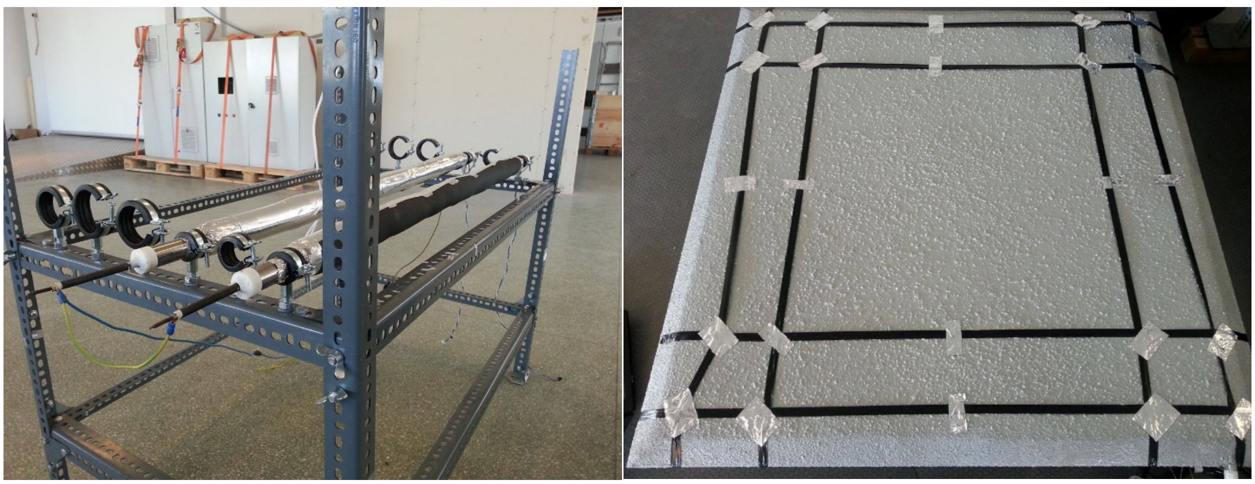

Figure 3-3 Test Specimen a) Steel Pipes with Insulation b) Deck Element

3.3 Temperature Measurement

The pipe surface temperature in the experiment was monitored with the Arduino Uno R3 data logger via maxim integrated DS18B20 sensors (Maxim Integrated, 2015). The code used for temperature logging is presented in Appendix B for reference. The DS18B20 sensors have an accuracy of ±0.5˚C over the temperature range –55 ^{o} C to +85 ^{o} C. The six temperature sensors were strategically attached on the surface of each pipe as show in Figure 3-4. The temperature sensors have an extended range from −55 ◦C and +125 ◦C with much lower accuracy. The resolution is set at 0.0625˚C. Details of Sensor DS18B20 is shown in Appendix F. Ambient temperature and humidity was measured using AM2303 sensor (Aosong (Guangzhou) Electronics Co. Ltd., 2009). In order to check for uniform surface temperature on the pipe and surface temperature stability, preliminary heating tests were carried out to verify the overall test arrangement. Equilibrium conditions were reached within 150 minutes of heating and were verified by monitoring the six thermocouples at 30 seconds time interval for 24 hours. Equilibrium conditions were taken as being established when the variation in temperature readings from the six thermocouples over a 2 ½ hour period were within 0.50°C. In the experiment, thermocouples or thermistors which are more reliable and stable could have been used. But, there was no microcontroller which could accommodate 18 sensors from three pipes. The problem could have been resolved by using multiple microcontrollers which could have led to significant cost escalation beyond the approved budget for the thesis. A plot of one set of temperature readings for 2 x 50 mm diameter pipes with heating element switched on and cross flow wind value of 0 m/s, 5 m/s, 10 m/s and 15 m/s is shown in Figure 3-5.

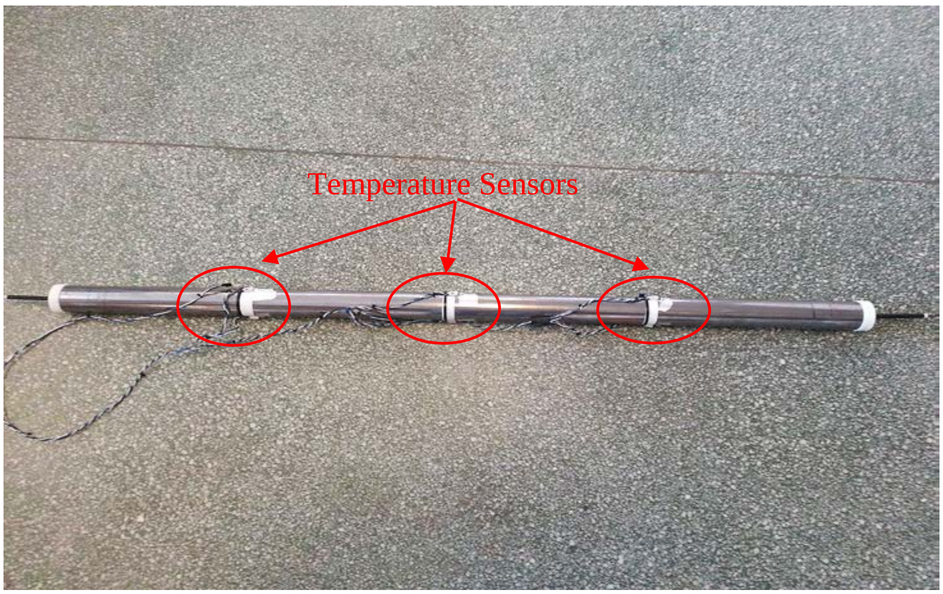

Figure 3-4 Pipe with temperature sensors

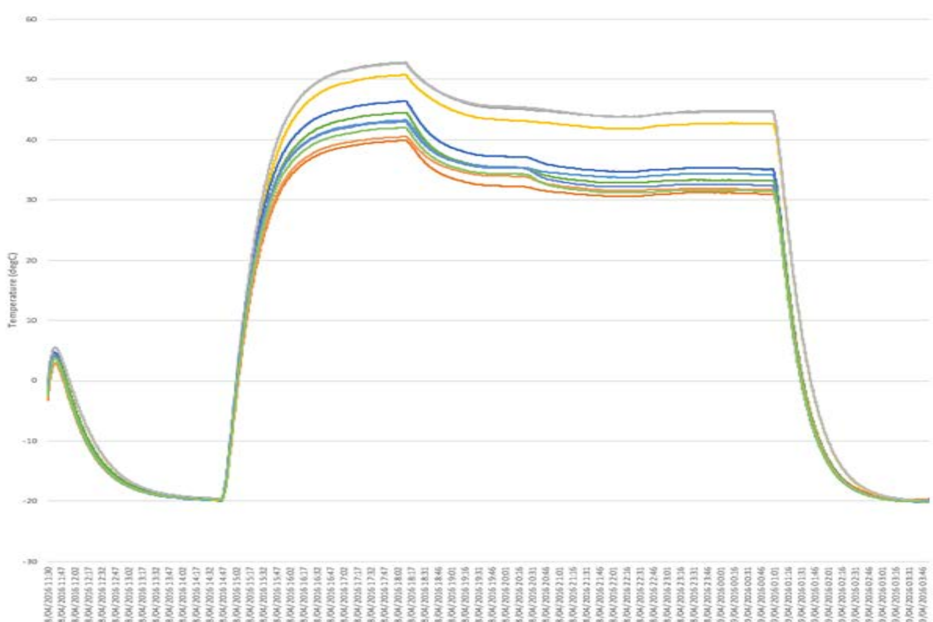

Figure 3-5 Temperature plot for 2 X 50 mm pipe configuration

3.4 Wind Measurement

The wind velocity measurement was done using a hand held anemometer LCA600 which was calibrated. It was necessitated by the fact that the wind sensor was not giving accurate readings as per the set wind speed in the control panel. The hand held anemometer was used to find velocity at right, middle and left section of the wind tunnel to arrive at the actual wind speed as shown in Table 3-2 and it was observed that the values are considerably different from the wind sensor measured velocity displayed on the control panel. A graph was plotted to find the relation between the set wind velocity and the actual value as shown in Figure 3-6

Table 3-2 Wind Velocity readings from Anemometer (LCA6000)

Set Wind Velocity (m/s)	Wind Sensor Readings (m/s)	Anemometer Readings (m/s)			Mean Value (m/s)
		Section 1	Section 2	Section 3	
2.5	3.5	4.56	5.3	4.86	4.91
5	5	6.1	7.1	6.7	6.63
7.5	7.5	9	10.3	9.6	9.63
10	10.1	11.4	13.6	13	12.67
12.5	12.5	13.6	16	14.5	14.70
15	15	17.9	18.6	16.4	17.63

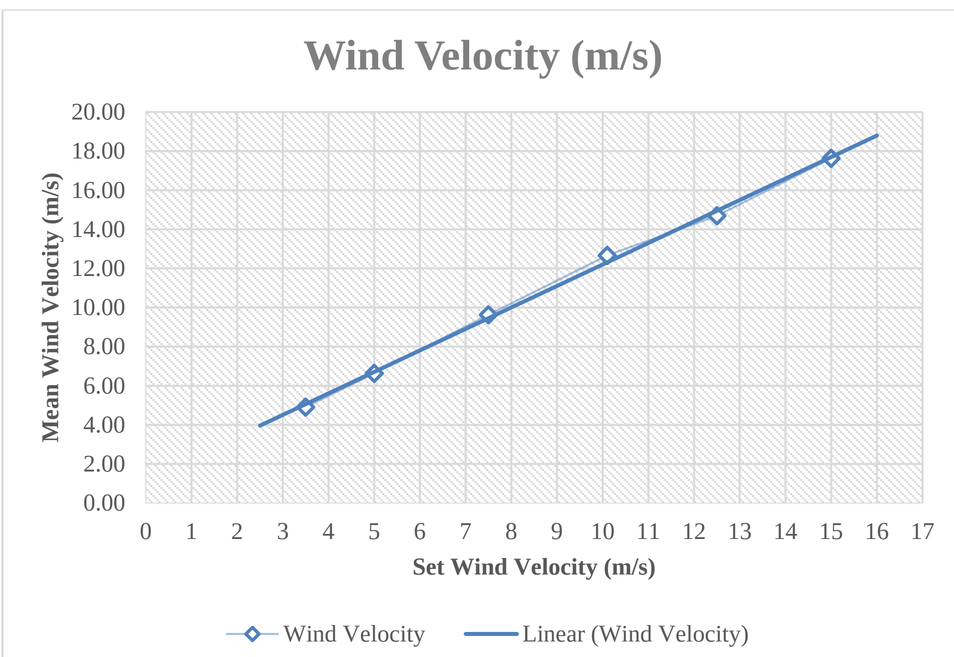

Figure 3-6 Graph showing relation between Set value and Actual value (wind velocity)

3.5 Test Procedure

3.5.1 Test Procedure for Pipes

1. The testing rig was positioned in the climate laboratory, directly in front of the wind tunnel. The height of the jig was adjusted so that the pipes are in the middle of the air flow.
2. The wind speed sensor was connected the junction box.
3. The ambient temperature sensor was positioned and connected to the junction box.
4. Pipe configuration was chosen as per the test schedule and setup was done.
5. The temperature sensors were attached on the pipe at the top and bottom at three different location and they were connected to the junction box.
6. The heating elements were connected with the power source and multimeter was used to measure the voltage
7. The junction box was connected to the Arduino using the data cable and power cable was plugged to the Arduino. The Logging was started and it was confirmed by looking at the flashing LED sensors. The Arduino had memory card for storing the data and it was also connected to the computer for real time monitoring.
8. The doors of the climate laboratory was closed and the temperature was allowed to settle down to the test temperature of -20 ^{o} C. The output voltage of the variac was adjusted to 57.5V on the control panel. This equals 50W with a resistance of 58.5 ohm from the heating element.

The temperature in the climate laboratory and speed of the wind flow from the tunnel was adjusted and monitored using the interface program on the control panel. The power source was also controlled from the same interface as shown in Figure 3-7.

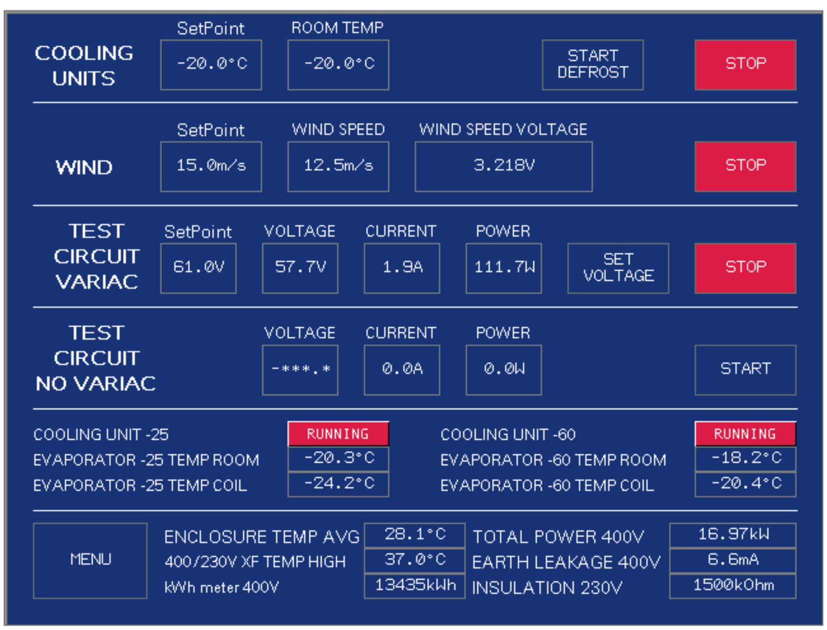

Figure 3-7 Interface Program on the Control Panel

The apparatus was continuously monitored and the programme was configured to record temperature readings in an interval of every 30 seconds. The plots were continuously monitored to determine uniformly heated pipe surface and attainment of equilibrium conditions. After equilibrium, wind speed was increased and the same procedure was repeated. This procedure was repeated three times with same pipe configuration and prior to each run, the heating elements were switched off and the test pipes were allowed to cool to the set temperature. See the Table 3-3 for sample recording sheet

Table 3-3 Sample Recording sheet for Pipe Experiment

	Temp. (°C)	Wind (m/s)	Date / Time Start	Date / Time Stop	Pipe #	Sensors	Current (A)	Voltage (V)
Run #1	**-20**	**0**	06-04-16 17:45	06-04-16 20:20	1	S1-6	1	56.2
	-20	**5**	06-04-16 20:22	06-04-16 23:06	1	S1-6	1	56.2
	-20	**10**	06-04-16 23:12	07-04-16 01:35	1	S1-6	1	56.2
	-20	**15**	07-04-16 01:40	07-04-16 04:51	1	S1-6	1	56.2
Run #2	**-20**	**0**	07-04-16 07:20	07-04-16 10:00	1	S1-6	1	56.2
	-20	**5**	07-04-16 10:02	07-04-16 12:30	1	S1-6	1	56.2
	-20	**10**	07-04-16 12:32	07-04-16 14:39	1	S1-6	1	56.2
	-20	**15**	07-04-16 14:42	07-04-16 16:58	1	S1-6	1	56.2
	-20	**0**	07-04-16 20:45	07-04-16 23:55	1	S1-6	1	56.2
Run #3	**-20**	**5**	07-04-16 23:58	08-04-16 01:45	1	S1-6	1	56.2
	-20	**10**	08-04-16 01:48	08-04-16 04:04	1	S1-6	1	56.2
	-20	**15**	08-04-16 04:06	08-04-16 08:37	1	S1-6	1	56.2

3.5.2 Test Procedure for Deck Elements

1. The deck elements were cooled down to the measured air temperature which is monitored from the data logger and the heating elements were started. The interface used is the same as for pipes and shown earlier in Figure 3-7
2. The heating elements were allowed to stabilize prior to taking readings.
3. Temperatures from data logger and from the thermal imaging camera (See Figure 3-8) were recorded.
4. Voltage, current and power were entered from the data logger.
5. Wind speeds were subsequently increased to 5m/s, 10 m/s and 15 m/s
6. The heating element was stopped and the deck element was allowed to cool down to the test temperature prior to the next run.
7. The test was repeated for -15, -30 and -35 degrees C.

See the Table 3-4 below for sample recording sheet.

Table 3-4 Sample Recording sheet for Deck Element

Temp. (°C)	Wind (m/s)	Date / Time Start	Date / Time Stop	Ambient Temp (ºC)	Surface Min (ºC)	Surface Max (ºC)	Surface Average (ºC)	Current (A)	Voltage (V)	Power (W)
-15	**0**	14.05.2016 10:15	14.05.2016 11:22	-13,95	11,5	17,2	15,1	4,5	221,2	997
-15	**5**	14.05.2016 11:23	14.05.2016 12:38	-13,57	-0,8	7,3	3,7	4,8	222.3	1077
-15	**10**	14.05.2016 12:39	14.05.2016 13:09	-13,06	-4,4	3	-0,6	5	221,7	1104
-15	**15**	14.05.2016 13:10	14.05.2016 13:40	-12,52	-6,1	0,7	-2,6	5,1	222,1	1135

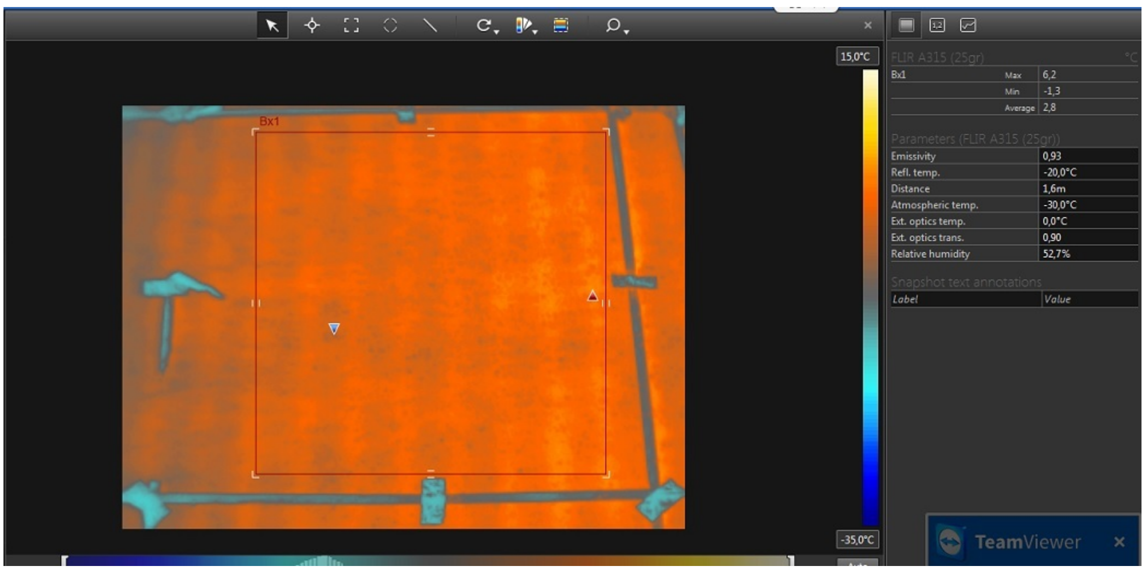

Figure 3-8 Thermal Imaging Camera

3.6 Test Readings/Schedule

The tests conducted in the climate laboratory of GMC with different experimental set up for the pipes and deck element subjected to cross flow wind are presented in the Table 3-5 below, there were total 12 experiments performed which included eleven experiments on different pipe configuration and one experiment on the deck element. Each experiment had different wind speeds and multiple runs were conducted to see the trend. Since, the experiments were jointly conducted with (Kvamme, 2016), the analysis scope was split up with (Kvamme, 2016) concentrating on 6 single pipe configuration and this thesis analyzed the piping arrangement involving multiple pipes (highlighted in Table 3-5) of similar and varying sizes besides the deck element which was analyzed separately. Testing done on single uninsulated pipe of 50 mm diameter (Experiment 11) part of (Kvamme, 2016) was the reference point for comparison of results with other piping configuration.

Table 3-5 Different Experimental Configuration

Experiment No:	Pipe / Deck Element Configuration	Details
1	**O** X X	50mm pipe/ *Free Slot* / *Free Slot*
2	**O** X **O**	50mm pipe/ *Free Slot* / 50mm pipe
3	**O O O**	50mm pipe/ 50mm pipe / 50mm pipe
4	**O** X X	50mm pipe (ice glazing)/ *Free Slot / Free Slot*
5	**O** X X	50mm pipe (ice coating)/ *Free Slot / Free Slot*
6	**O** X X	50mm pipe (roughened surface)/ *Free Slot / Free Slot*
7	**o** X **O**	25mm pipe/ *Free Slot* / 50mm pipe
8	**o** X X	25mm pipe/ *Free Slot / Free Slot*
9	**o** X **o**	25mm pipe/ *Free Slot /* 25mm pipe
10	**O** X **o**	50mm pipe */ Free Slot /* 25mm pipe
11	**O** X X	*50mm pipe (No insulation)/ Free Slot / Free Slot*
12	**-**	*Deck plating with antiskid coating/roughened surface*

3.6.1 Test Readings from Experiment 2

Table 3-6 Readings from Experiment 2 (2 x 50mm)

	Experiment 2		**2x 50mm pipe (O, X, O)**					
	Temp. (°C)	**Wind (m/s)**	**Date / Time Start**	**Date / Time Stop**	**Pipe #**	**Sensors**	**Current**	**Voltage**
Run #1	**-20**	**0**	08.04.2016 14:44	08.04.2016 18:09	1 & 3	S1-6, S13-18	1,9	55,8
	-20	**5**	08.04.2016 18:10	08.04.2016 20:19	1 & 3	S1-6, S13-18	1,9	55,8
	-20	**10**	08.04.2016 20:20	08.04.2016 22:30	1 & 3	S1-6, S13-18	1,9	55,8
	-20	**15**	08.04.2016 22:32	09.04.2016 00:58	1 & 3	S1-6, S13-18	1,9	55,8
Run #2	**-20**	**0**	09.04.2016 03:52	09.04.2016 09:50	1 & 3	S1-6, S13-18	1,9	55,8
	-20	**5**	09.04.2016 09:51	09.04.2016 13:13	1 & 3	S1-6, S13-18	1,9	55,8
	-20	**10**	09.04.2016 13:14	09.04.2016 15:25	1 & 3	S1-6, S13-18	1,9	55,8
	-20	**15**	09.04.2016 15:28	09.04.2016 17:59	1 & 3	S1-6, S13-18	1,9	55,8
Run #3	**-20**	**0**	09.04.2016 21:20	10.04.2016 00:26	1 & 3	S1-6, S13-18	1,9	55,8
	-20	**5**	10.04.2016 00:27	10.04.2016 05:57	1 & 3	S1-6, S13-18	1,9	55,8
	-20	**10**	10.04.2016 05:58	10.04.2016 07:36	1 & 3	S1-6, S13-18	1,9	55,8
	-20	**15**	10.04.2016 07:37	10.04.2016 09:44	1 & 3	S1-6, S13-18	1,9	55,8

3.6.2 Test Readings from Experiment 3

Table 3-7 Readings from Experiment 3 (3 x 50mm)

	Experiment 3		3x 50mm pipe (O, O, O)					
	Temp. (°C)	**Wind (m/s)**	**Date / Time Start**	**Date / Time Stop**	**Pipe #**	**Sensors**	**Current**	**Voltage**
Run #1	**-20**	**0**	04.04.2016 19:57	04.04.2016 23:00	1,2,3	S1-18	2,90	54,6
	-20	**5**	04.04.2016 23:00	05.04.2016 03:00	1,2,3	S1-18	2,90	54,6
	-20	**10**	05.04.2016 03:00	06.04.2016 06:45	1,2,3	S1-18	2,90	54,6
	-20	**15**	05.04.2016 06:45	06.04.2016 09:00	1,2,3	S1-18	2,90	54,6
Run #2	**-20**	**0**	10.04.2016 19:16	10.04.2016 22:00	1,2,3	S1-18	2,9	54,6
	-20	**5**	10.04.2016 22:03	11.04.2016 00:19	1,2,3	S1-18	2,9	54,6
	-20	**10**	11.04.2016 00:20	11.04.2016 06:23	1,2,3	S1-18	2,9	54,6
	-20	**15**	11.04.2016 06:24	11.04.2016 09:23	1,2,3	S1-18	2,9	54,6

3.6.3 Test Readings from Experiment 7

Table 3-8 Readings from Experiment 7 (1 x 25mm and 1 x 50mm)

	Experiment 7		1x 25mm + 1x 50mm pipes (o, x, O)					
	Temp. (°C)	**Wind (m/s)**	**Date / Time Start**	**Date / Time Stop**	**Pipe #**	**Sensors**	**Current**	**Voltage**
Run #1	**-20**	**0**	04.05.2016 11:57	04.05.2016 14:40	1,3	1 (S7-12), 3 (S13-18)	2	56,3
	-20	**5**	04.05.2016 14:43	04.05.2016 16:44	1,3	1 (S7-12), 3 (S13-18)	2	56,3
	-20	**10**	04.05.2016 16:46	04.05.2016 19:20	1,3	1 (S7-12), 3 (S13-18)	2	56,3
	-20	**15**	04.05.2016 19:22	04.05.2016 20:12	1,3	1 (S7-12), 3 (S13-18)	2	56,3
Run #2	**-20**	**0**	05.05.2016 10:47	05.05.2016 13:35	1,3	1 (S7-12), 3 (S13-18)	2	56,3
	-20	**5**	05.05.2016 13:38	05.05.2016 16:07	1,3	1 (S7-12), 3 (S13-18)	2	56,3
	-20	**10**	05.05.2016 16:11	05.05.2016 18:15	1,3	1 (S7-12), 3 (S13-18)	2	56,3
	-20	**15**	05.05.2016 18:18	05.05.2016 19:34	1,3	1 (S7-12), 3 (S13-18)	2	56,3
Run #3	**-20**	**0**	05.05.2016 21:48	06.05.2016 00:27	1,3	1 (S7-12), 3 (S13-18)	2	56,3
	-20	**5**	06.05.2016 00:29	06.05.2016 02:10	1,3	1 (S7-12), 3 (S13-18)	2	56,3
	-20	**10**	06.05.2016 02:12	06.05.2016 06:20	1,3	1 (S7-12), 3 (S13-18)	2	56,3
	-20	**15**	06.05.2016 06:22	06.05.2016 07:51	1,3	1 (S7-12), 3 (S13-18)	2	56,3

3.6.4 Test Readings from Experiment 9

Table 3-9 Readings from Experiment 9 (2 x 25mm)

	Experiment 9		2x 25mm pipe (o, x, o)					
	Temp. (°C)	**Wind (m/s)**	**Date / Time Start**	**Date / Time Stop**	**Pipe #**	**Sensors**	**Current**	**Voltage**
Run #1	**-20**	**0**	11.05.2016 20:46	12.05.2016 00:16	1,3	1 (S7-12), 3 (S13-18)	2	56,30
	-20	**5**	12.05.2016 00:19	12.05.2016 02:39	1,3	1 (S7-12), 3 (S13-18)	2	56,30
	-20	**10**	12.05.2016 02:41	12.05.2016 04:19	1,3	1 (S7-12), 3 (S13-18)	2	56,30
	-20	**15**	12.05.2016 04:21	12.05.2016 06:07	1,3	1 (S7-12), 3 (S13-18)	2	56,30
Run #2	**-20**	**0**	12.05.2016 11:13	12.05.2016 13:53	1,3	1 (S7-12), 3 (S13-18)	2	56,30
	-20	**5**	12.05.2016 13:55	12.05.2016 16:11	1,3	1 (S7-12), 3 (S13-18)	2	56,30
	-20	**10**	12.05.2016 16:13	12.05.2016 18:17	1,3	1 (S7-12), 3 (S13-18)	2	56,30
	-20	**15**	12.05.2016 18:19	12.05.2016 20:16	1,3	1 (S7-12), 3 (S13-18)	2	56,30
Run #3	**-20**	**0**	12.05.2016 22:24	13.05.2016 01:26	1,3	1 (S7-12), 3 (S13-18)	2	56,30
	-20	**5**	13.05.2016 01:28	13.05.2016 03:37	1,3	1 (S7-12), 3 (S13-18)	2	56,30
	-20	**10**	13.05.2016 03:40	13.05.2016 06:27	1,3	1 (S7-12), 3 (S13-18)	2	56,30
	-20	**15**	13.05.2016 06:28	13.05.2016 08:32	1,3	1 (S7-12), 3 (S13-18)	2	56,30

3.6.5 Test Readings from Experiment 10

Table 3-10 Readings from Experiment 10 (1x 50mmand 1x 25 mm)

	Experiment 10		1x 50mm + 1x 25mm (O, x, o)					
	Temp. (°C)	**Wind (m/s)**	**Date / Time Start**	**Date / Time Stop**	**Pipe #**	**Sensors**	**Current**	**Voltage**
Run #1	**-20**	**0**	08.05.2016 09:52	08.05.2016 12:12	3,1	1 (S7-12), 3 (S13-18)	2	56,9
	-20	**5**	08.05.2016 12:13	08.05.2016 15:53	3,1	1 (S7-12), 3 (S13-18)	2	56,9
	-20	**10**	08.05.2016 15:54	08.05.2016 19:01	3,1	1 (S7-12), 3 (S13-18)	2	56,9
	-20	**15**	08.05.2016 19:02	08.05.2016 21:20	3,1	1 (S7-12), 3 (S13-18)	2	56,9
Run #2	**-20**	**0**	08.05.2016 23:53	09.05.2016 02:26	3,1	1 (S7-12), 3 (S13-18)	2	56,9
	-20	**5**	09.05.2016 02:28	09.05.2016 04:30	3,1	1 (S7-12), 3 (S13-18)	2	56,9
	-20	**10**	09.05.2016 04:32	09.05.2016 06:59	3,1	1 (S7-12), 3 (S13-18)	2	56,9
	-20	**15**	09.05.2016 07:00	09.05.2016 10:31	3,1	1 (S7-12), 3 (S13-18)	2	56,9
Run #3	**-20**	**0**	09.05.2016 13:00	09.05.2016 16:16	3,1	1 (S7-12), 3 (S13-18)	2	56,9
	-20	**5**	09.05.2016 16:17	09.05.2016 17:34	3,1	1 (S7-12), 3 (S13-18)	2	56,9
	-20	**10**	09.05.2016 17:35	09.05.2016 19:32	3,1	1 (S7-12), 3 (S13-18)	2	56,9
	-20	**15**	09.05.2016 19:33	10.05.2016 00:23	3,1	1 (S7-12), 3 (S13-18)	2	56,9

3.6.6 Test Readings from Experiment 12

Table 3-11 Readings from Experiment on Deck element

	Experiment 12		**Deck element**										
	Temp. (°C)	**Wind (m/s)**	**Date / Time Start**	**Date / Time Stop**	**Ambient Temp**	**Air Temp**	**Surface Min**	**Surface Max**	**Surface Average**	**Current**	**Voltage**	**Power**
Run #1	**-15**	**0**	14-05-16 10:15	14-05-16 11:22	-13.95	-11.2	11.5	17.2	15.1	4.5	221.2	997
	-15	**5**	14-05-16 11:23	14-05-16 12:38	-13.57	-12.6	-0.8	7.3	3.7	4.8	222.3	1077
	-15	**10**	14-05-16 12:39	14-05-16 13:09	-13.06	-11.9	-4.4	3	-0.6	5	221.7	1104
	-15	**15**	14-05-16 13:10	14-05-16 13:40	-12.52	-11.5	-6.1	0.7	-2.6	5.1	222.1	1135
Run #2	**-15**	**0**	14-05-16 14:55	14-05-16 17:40	-13.82	-11.2	18	26.4	23.9	3.8	225.7	876
	-15	**5**	14-05-16 17:42	14-05-16 19:03	-14.04	-12.7	-9.5	7.6	3.5	4.7	224.8	1073
	-15	**10**	14-05-16 19:05	14-05-16 20:10	-13.74	-12.7	-11.7	1.8	-1.6	5	223.9	1131
	-15	**15**	14-05-16 20:12	14-05-16 21:26	-13.67	-12.3	-7.7	-0.9	-4.1	5.1	224.5	1165
Run #3	**-15**	**0**	14-05-16 23:16	15-05-16 1:02	-13.67	-11.5	16.5	23	20.9	4.1	225.7	935
	-15	**5**	15-05-16 1:04	15-05-16 2:15	-13.97	-13.1	-0.5	8	3.9	4.8	224.7	1078
	-15	**10**	15-05-16 2:17	15-05-16 3:19	-13.64	-12.3	-5.7	1.9	-1.5	5	224.3	1135
	-15	**15**	15-05-16 3:21	15-05-16 4:21	-12.84	-11.5	-7.2	0.3	-2.8	5.1	224.9	1155
Run #1	**-20**	**0**	18-05-16 15:07	18-05-16 17:59	-18.72	-16.8	16.5	24.3	21.9	4.1	224.9	937
	-20	**5**	18-05-16 18:00	18-05-16 22:19	-19.16	-17.7	-6.8	1.7	-2.1	5.2	222.5	1174
	-20	**10**	19-05-16 18:51	19-05-16 21:19	-19.26	-17.6	-11.7	-4.2	-7.8	5.5	224.2	1231
	-20	**15**	19-05-16 21:21	19-05-16 23:45	-18.75	-17.5	-12.9	-6.3	-9.5	5.6	225.6	1264
Run #2	**-20**	**0**	20-05-16 1:15	20-05-16 3:16	-18.88	-17.6	14.8	21.8	19.5	4.3	226.2	972
	-20	**5**	20-05-16 3:18	20-05-16 6:40	-19.02	-17.8	-7	1.8	-2	5.2	223.1	1180
	-20	**10**	20-05-16 6:42	20-05-16 12:45	-18.91	-18.2	-11.1	-3.7	-7	5.5	223.6	1236
	-20	**15**	20-05-16 12:46	20-05-16 16:02	-19.01	-18.3	-13.3	-6.8	-9.9	5.6	226.3	1272
Run #3	**-20**	**0**	20-05-16 17:11	20-05-16 20:27	-19.11	-17.2	17.2	25	22.4	4.1	225.8	933
	-20	**5**	20-05-16 20:29	20-05-16 22:21	-19.37	-18.6	-7.2	2.1	-2	5.2	224.6	1172
	-20	**10**	20-05-16 22:23	20-05-16 23:29	-18.77	-18.3	-10.7	-3.5	-6.8	5.4	226.9	1230
	-20	**15**	20-05-16 23:31	21-05-16 1:45	-18.81	-17.1	-13	-6.6	-9.6	5.6	223.8	1255
Run #1	**-30**	**0**	15-05-16 8:01	15-05-16 10:03	-30.86	-29.6	4.5	12.9	9.7	4.7	225.1	1075
	-30	**5**	15-05-16 10:04	15-05-16 12:54	-28.75	-27.1	-20.5	-10.3	-14.9	5.7	226.5	1292
	-30	**10**	15-05-16 12:56	15-05-16 13:59	-25.43	-24.1	-22.3	-13.8	-17.7	5.8	226.3	1325
	-30	**15**	15-05-16 14:01	15-05-16 14:01	-31.55	-29.8	-28	-19.7	-23.4	5.9	225.6	1361
Run #2	**-30**	**0**	15-05-16 16:17	15-05-16 18:44	-30.98	-29.5	5.7	13.3	10.1	4.7	228.3	1079
	-30	**5**	15-05-16 18:45	15-05-16 19:48	-27.19	-26.9	-18.5	-7.8	-12.3	5	227.6	1158
	-30	**10**	15-05-16 19:51	15-05-16 20:51	-29.83	-28.4	-24	-15.4	-19.2	5.8	227.2	1320
	-30	**15**	15-05-16 20:52	15-05-16 21:52	-25.96	-23.1	-23.4	-16.4	-19.6	6	227.3	1359
Run #3	**-30**	**0**	16-05-16 0:06	16-05-16 1:07	-25.87	-21.9	1.2	12.5	8.7	4.6	227.6	1060
	-30	**5**	16-05-16 1:09	16-05-16 2:07	-27.52	-25.7	-17.3	-7.4	-11.5	5.5	225.2	1244
	-30	**10**	16-05-16 2:11	16-05-16 3:16	-28.31	-27.4	-23.2	-14.1	-18.3	5.8	224.5	1317
	-30	**15**	16-05-16 3:18	16-05-16 4:18	-31.38	-28.7	-28.2	-20.2	-24.1	6	224.8	1366
Run #1	**-35**	**0**	16-05-16 7:17	16-05-16 12:33	-32.83	-31.1	-4.5	8.2	3.7	4.8	227.3	1111
	-35	**5**	16-05-16 12:35	16-05-16 14:36	-25.17	-22.6	-17.7	-8.1	-12.3	5.7	225.3	1296
	-35	**10**	16-05-16 14:38	16-04-16 16:23	-31.4	-28.9	-29	-20.6	-24.6	6.2	225.7	1398
	-35	**15**	16-05-16 16:24	16-05-16 17:42	-28.19	-25.8	-26.9	-20.2	-23.1	6.2	226	1418
Run #2	**-35**	**0**	17-05-16 14:24	17-05-16 16:41	-27.13	-23.9	-3.1	7.8	3.5	5	226.4	1138
	-35	**5**	17-05-16 16:42	17-05-16 17:57	-29.49	-28.9	-18.6	-8.6	-12.9	5.7	226	1287
	-35	**10**	17-05-16 17:59	17-05-16 19:44	-29.7	-27.8	-26.9	-18.8	-22.5	6.1	226	1400
	-35	**15**	17-05-16 19:45	17-05-16 22:10	-25.6	-22	-25.2	-17.2	-21.6	6.2	225.6	1400

4 Calculations

4.1 Experimental Method.

4.1.1 Case 1: Heat Transfer co-efficient calculation for uninsulated pipe

In this case, the convective heat transfer coefficient is calculated for a single pipe which is in direct influence of the wind flow. The values used are from the experimental readings done on uninsulated pipe which were analyzed by (Kvamme, 2016). This will help in the comparison of the heat transfer coefficient with insulated which will be performed later in the section. So, we look into an uninsulated pipe with an outer diameter of 50 mm and internal diameter of 46mm. The pipe has a heating element which is centrally placed in the pipe. The ambient temperature is −20 °C and the pipe is subjected to a cross flow wind of 5 m/s. The values used for the calculation will be the actual ambient temperature and wind velocity obtained from calibrated sensors and anemometer at the time of experimentation. The picture shown in Figure 4-1 depicts the actual setup of the uninsulated pipe with temperature distribution.

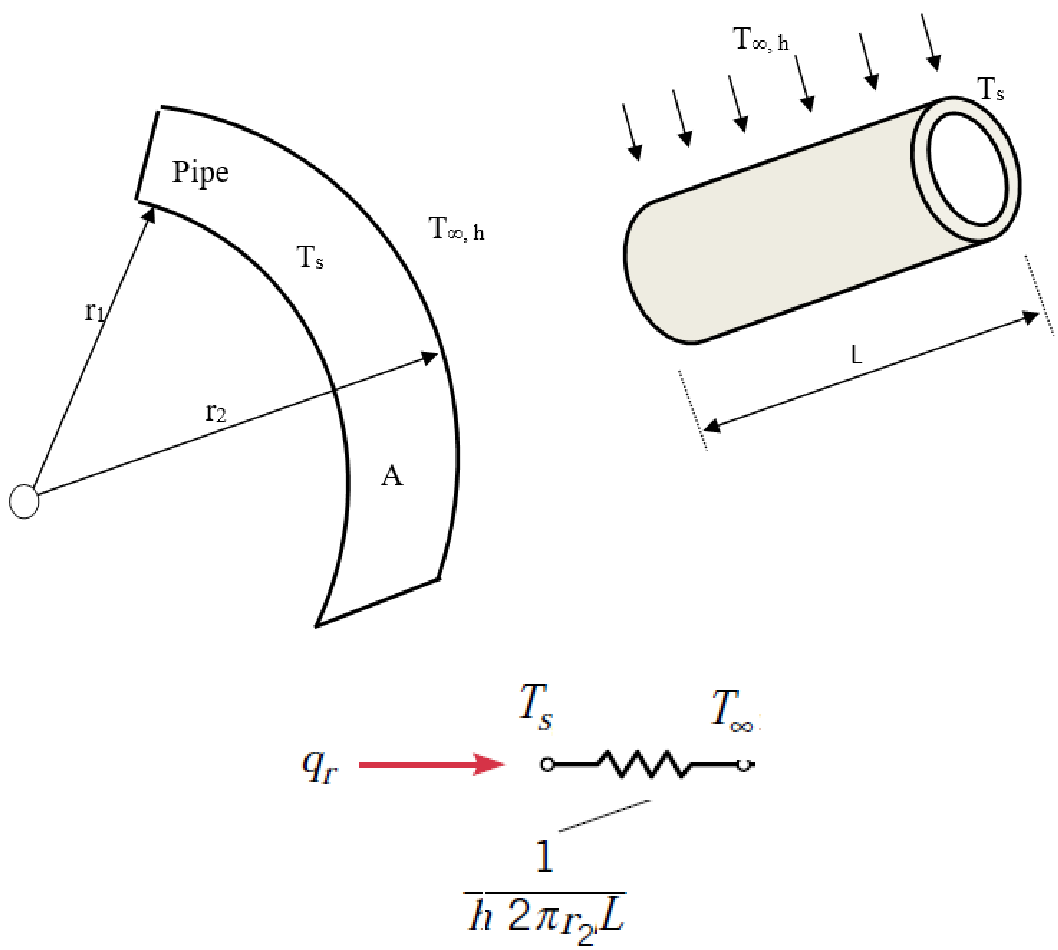

Figure 4-1 Temperature Distribution for the uninsulated pipe

Assumptions:

1. Overall Steady-state conditions.
2. Heat transfer in the radial direction is one-dimensional.
3. Uniform surface temperature for the pipe
4. 15% of the power is lost through the cumulative effect of surface radiation and conduction through the end pieces

All the constants and the variables which are to be used in the calculation of convective heat transfer coefficient for uninsulated pipe are mentioned below

Length (pipe), L_{pipe} *(m)*	=	1.2
Length (heating element), L_{elem} *(m)*	=	1.372
Outer Diameter of uninsulated Pipe, D_o *(m)*	=	0.050
Inner Diameter of uninsulated Pipe, D_i *(m)*	=	0.046
Pipe wall Thickness, t_w *(m)*	=	0.002
Internal pipe radius, r_1 *(m)*	=	0.023
External pipe radius, r_2 *(m)*	=	0.025
Surface area, A (m^2)	=	0.1884
Ambient Temperature, T_∞ (^oC)	=	-19.41
Surface Temperature of pipe, T_s (^oC)	=	-16.63
Voltage, V *(V)*	=	56.2
Current, I *(A)*	=	1.0
Power efficiency, η	=	0.85

Using Equation (2.6) explained earlier,

$$q = hA\,(T_s - T_\infty) = \frac{T_{s-}T_\infty}{(1/hA)}$$

$$q = \eta * V * I = \frac{T_{s-}T_\infty}{(1/hA)} \tag{4.1}$$

Rearranging (4.1),

$$h = \frac{\eta * V * I}{A * (T_{s-}T_{\infty})} \tag{4.2}$$

$$A = \left(2\pi * r_2 * L_{pipe}\right)$$

$$A = (2\pi * 0.025 * 1.2) = 0.1884\, m^2$$

$$h = \frac{[(0.85 * 56.2 * 1.0)/1.372] * 1.2}{0.1884 * \ (256.52 - 253.74)}$$

$$Heat\ \ Transfer\ Coefficient, \qquad h = \frac{41.78}{0.1884 * \ 2.78} \ = \ 79.77\ \ W/m^2.K \tag{4.3}$$

4.1.2 Case 2: Heat Transfer co-efficient calculation for insulated pipe

In this case, we will consider the same the convective heat transfer coefficient is calculated for a single insulated pipe which is in direct influence of the wind flow. The values used are from the experimental readings done on insulated pipe. Here, we consider an insulated pipe with an outer diameter of 50 mm, inner diameter of 46 mm and insulation thickness of 10 mm. The pipe has a heating element which is centrally placed in the pipe. The ambient temperature is $-20\ ^{\circ}\mathrm{C}$ and the pipe is subjected to a cross flow wind of 5 m/s. The values used for the calculation will be the actual ambient temperature and wind velocity obtained from calibrated sensors and anemometer at the time of experimentation. Temperature distribution of an insulated pipe is shown in Figure 4-2.

Assumptions:

1. Overall Steady-state conditions.
2. Heat transfer in the radial direction is one-dimensional.
3. Negligible radiation loss between surroundings and surface.
4. Negligible heat loss through the end caps of the pipe.
5. Uniform surface temperature for the pipe.
6. 15% of the power is lost through the cumulative effect of surface radiation and conduction through the end pieces.
7. Change in thermal conductivity over a small temperature range is considered negligible.
8. Change in thermal diffusivity over a small temperature range is considered negligible.

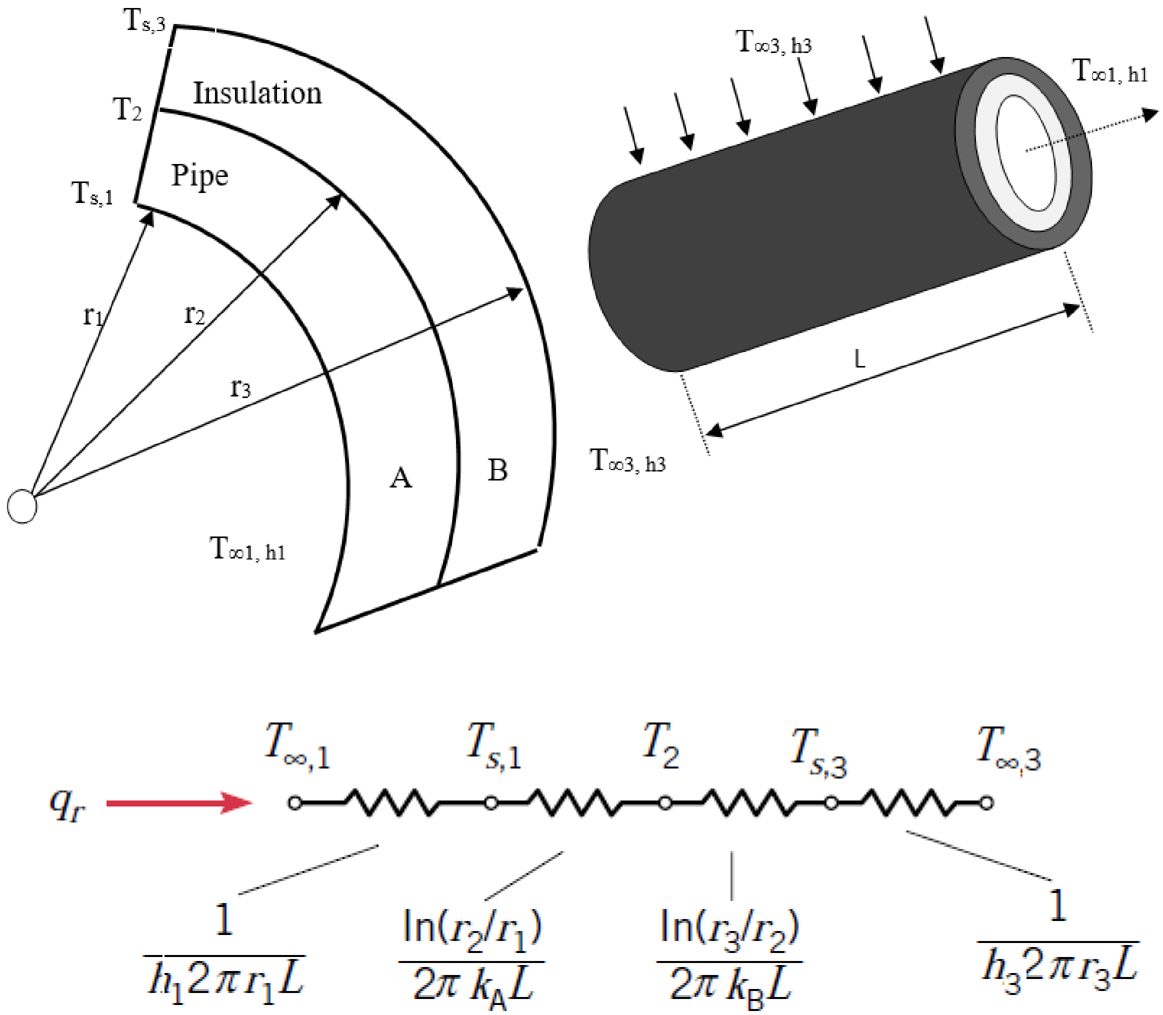

Figure 4-2 Temperature Distribution for the insulated pipe

All the constants and the variables which are to be used in the calculation of heat transfer coefficient for insulated pipe are mentioned below

Length (pipe), L_{pipe} *(m)*	=	1.2
Length (heating element), L_{elem} *(m)*	=	1.372
Length (with insulation), L_{ins} *(m)*	=	1.04
Outer Diameter, D_o (m)	=	0.070
Inner Diameter, D_i (m)	=	0.046
Pipe Wall Thickness, t_w (m)	=	0.002
Insulation Thickness, t_{ins} *(m)*	=	0.10

Internal pipe radius, r_1 *(m)*	=	0.023
External pipe radius, r_2 *(m)*	=	0.025
External insulation radius, r_3 *(m)*	=	0.035
Ambient Temperature (pipe internal), $T_{\infty,1}$ *(°C)*	=	To be calculated
Ambient Temperature, $T_{\infty,3}$ *(°C)*	=	-19.68
Internal Temperature of pipe, $T_{s,1}$ *(°C)*	=	To be calculated
Surface Temperature of pipe, T_2 *(°C)*	=	38.74
Surface Temperature of insulation, $T_{s,3}$ *(°C)*	=	To be calculated
Surface area (pipe internal), A_1 *(m^2)*	=	0.1502
Convective Heat transfer coefficient, *h (W/m^2 . K)*	=	To be calculated
Overall Heat transfer coefficient, U_1 *(W/m^2 . K)*	=	To be calculated
Thermal conductivity of air , k_{air} *(W/m . K) at 256 K*		22.3×10^{-3}
Thermal conductivity of pipe, K_A *(W/m . K)*	=	60.5
Thermal conductivity of insulation, K_B *(W/m . K)*	=	0.033
Voltage, *V (V)*	=	55.8
Current, *I (A)*	=	0.95
Power efficiency, η	=	0.85

Using equation (2.15) and (2.16) explained earlier,

$$q_r = \frac{T_{\infty,1} - T_{\infty,4}}{\frac{1}{2\pi r_1 L h_1} + \frac{\ln(r_2/r_1)}{2\pi K_A L} + \frac{\ln(r_3/r_2)}{2\pi K_B L} + \frac{\ln(r_4/r_3)}{2\pi K_c L} + \frac{1}{2\pi r_4 L h_4}}$$

$$q_r = \frac{T_{\infty,1} - T_{\infty,3}}{R_{tot}} = UA(T_{\infty,1} - T_{\infty,3})$$

The heat transfer rate can be expressed in terms of the temperature difference and resistance associated with each element as shown below.

$$q_r = \frac{T_{s,1} - T_2}{\frac{\ln(r_2/r_1)}{2\pi K_A L}} \tag{4.4}$$

Internal Temperature of pipe (Ts, 1) can be calculated using the below equation obtained from (4.4),

$$\eta * V * I = \frac{T_{s,1} - T_2}{\frac{\ln(r_2/r_1)}{2\pi K_A L}} \tag{4.5}$$

$$[(0.85 * 55.8 * 0.95)/1.372] * 1.04 = \frac{T_{s,1} - (38.74 + 273.15)}{\frac{\ln(0.025/0.023)}{2 * 3.14 * 60.5 * 1.04}}$$

$$34.155 = \frac{T_{s,1} - 311.89}{2.11 * 10^{-4}}$$

$$\text{Internal Temperature of pipe } (T_{s,1}) = 311.89\, K \quad or \quad 38.74\ ℃ \tag{4.6}$$

Similarly, surface temperature of insulation (Ts, 3) can be calculated using the below equation,

$$q_r = \frac{T_2 - T_{s,3}}{\frac{\ln(r_3/r_2)}{2\pi K_B L}} \tag{4.7}$$

$$\eta * V * I = \frac{T_2 - T_{s,3}}{\frac{\ln(r_3/r_2)}{2\pi K_B L}} \tag{4.8}$$

$$[(0.85 * 55.8 * 0.95)/1.372] * 1.04 = \frac{(38.74 + 273.15) - T_{s,3}}{\frac{\ln(0.035/0.025)}{2 * 3.14 * 0.033 * 1.04}}$$

$$34.155 = \frac{311.89 - T_{s,3}}{1.5611}$$

$$\text{Surface Temperature of insulation } (T_{s,3}) = 258.56\,K \quad or \; -14.58\,℃$$

Convective Heat transfer coefficient (outer surface), h_3 can be calculated using the equation,

$$q_r = \frac{T_{s,3} - T_{\infty,3}}{\frac{1}{2\pi r_3 L h_3}} \tag{4.9}$$

$$\eta * V * I = \frac{T_{s,3} - T_{\infty,3}}{\frac{1}{2\pi r_3 L h_3}} \tag{4.10}$$

$$[(0.85 * 55.8 * 0.95)/1.372] * 1.04 = \frac{258.6 - 253.47}{\frac{1}{2 * 3.14 * 0.035 * 1.04 * h_3}} \tag{4.11}$$

$$34.155 = \frac{5.099}{4.374} * h_3$$

$$h_3 = \frac{149.39}{5.099}$$

$$\text{Convective Heat transfer coefficient (outer surface),} \qquad \text{h} = 29.302\ W/m^2.K \tag{4.12}$$

The overall heat transfer coefficient U can be defined in terms of the inside area of the insulated pipe section, $A_1 = 2\pi r_1 L_{pipe}$ using equation (2.16)

$$q_r = \frac{T_{\infty,1} - T_{\infty,3}}{R_{tot}} = UA(T_{\infty,1} - T_{\infty,3})$$

$$U_1 = \frac{q_r}{A_1(T_2 - T_{\infty,3})} \tag{4.13}$$

$$A_1 = (2\pi * r_1 * L_{pipe})$$

$$A_1 = (2 * 3.14 * 0.023 * 1.04) = 0.1502m^2 \tag{4.14}$$

$$U_1 = \frac{\eta * V * I}{A_1(T_2 - T_{\infty,3})}$$

$$U_1 = \frac{[(0.85 * 55.8 * 0.95)/1.372] * 1.2}{0.1502 * (311.89 - 253.47)}$$

$$U_1 = \frac{34.155}{0.1502 * 58.42}$$

$$\text{Overall heat transfer coefficient}, U_1 = 3.892\ W/m^2.K \tag{4.15}$$

4.1.3 Case 3: Heat Transfer co-efficient calculation for deck element (flat plate)

In this section, heat transfer coefficient for deck element will be calculated using the readings obtained during experiment. So, we look into a steel plate with size 1.1 m x 1.1 m with epoxy coating. The thickness of the plate is 3 cm and the bottom surface doesn't have epoxy coating. The plate has heating tracing underneath the coating. The ambient temperature is −20 °C and the pipe is subjected to a cross flow wind of 5 m/s. The values used for the calculation will be the actual ambient temperature and wind velocity obtained from calibrated sensors and anemometer at the time of experimentation.

Assumptions:

1. Overall Steady-state conditions.
2. Uniform heat transfer coefficient.
3. Negligible radiation loss between surroundings and surface.
4. Constant properties.
5. Uniform surface temperature for the plate
6. 15% of the power is lost through the cumulative effect of surface radiation and conduction through the edges

All the constants and the variables which are to be used in the calculation of convective heat transfer coefficient for the deck element are mentioned below.

Length, *L (m)*	=	1.1
Width, *W (m)*	=	1.1
Thickness, t (m)	=	0.03
Ambient Temperature, T_∞ *(^{o}C)*	=	-18.03
Surface Temperature, T_s *(^{o}C)*	=	-2.033
Voltage, *V (V)*	=	223.4
Current, *I (A)*	=	5.2
Power efficiency, η	=	0.85

Using Equation (2.6) explained earlier,

$$q = UA\,(T_s - T_\infty) = \frac{T_{s-}T_\infty}{(1/hA)}$$

$$q = \eta * V * I = \frac{T_{s-}T_\infty}{(1/UA)} \tag{4.16}$$

$$U = \frac{\eta * V * I}{A\,(T_s - T_\infty)} \tag{4.17}$$

$$U = \frac{(0.85 * 223.4 * 5.2)}{\big(2(1.1 * 1.1) + (1.1 * 0.03 * 4)\big) * \ (271.12 - 255.12)}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ , U = \ 24.18\ W/m^2.K \tag{4.18}$$

4.2 Theoretical Method

4.2.1 Case 1: Wind blowing over uninsulated pipe (forced flow scenario)

In this case, we will consider the same uninsulated pipe under direct influence of the wind flow which we used for calculation using experimental data in order to find out the heat transfer coefficient. The OD and ID of the pipe is 50 mm and 46 mm respectively. The pipe has a heating element which is centrally placed in the pipe. The ambient temperature is −20 °C and the pipe is subjected to a cross flow wind of 5 m/s. The values used for the calculation will be the actual ambient temperature and wind velocity obtained from calibrated instruments at the time of experimentation. The picture shown in Figure 4-3 depicts the actual setup of the uninsulated pipe along with temperature distribution.

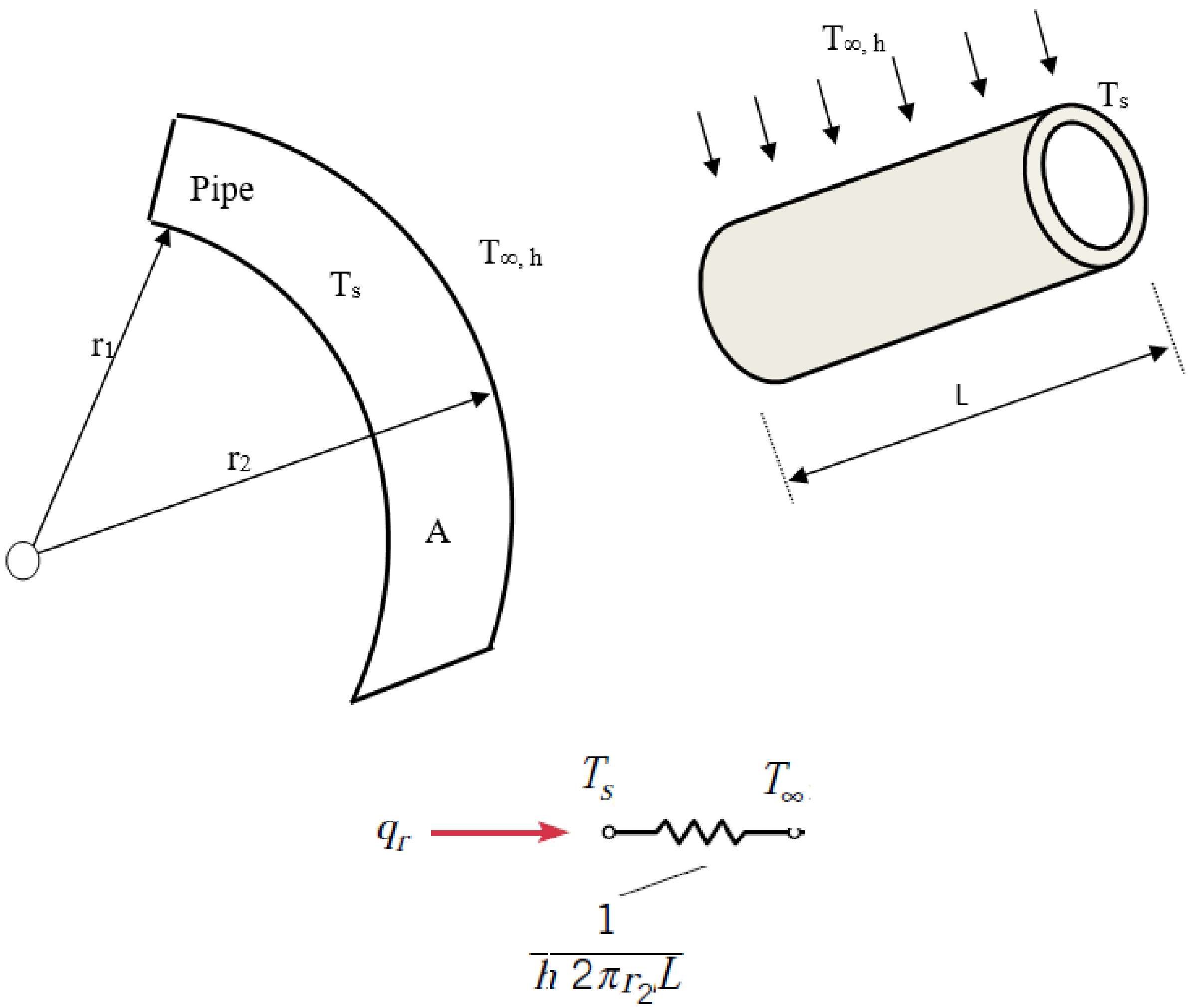

Figure 4-3 Temperature Distribution for the uninsulated pipe (forced flow scenario)

Assumptions:

1. Overall Steady-state conditions.
2. Heat transfer in the radial direction is one-dimensional.
3. Uniform surface temperature for the pipe.
4. 15% of the power is lost through the cumulative effect of surface radiation and conduction through the end pieces.
5. Change in thermal conductivity over a small temperature range is considered negligible.
6. Change in thermal diffusivity over a small temperature range is considered negligible.

All the constants and the variables which are to be used in the calculation of heat transfer coefficient for uninsulated pipe are mentioned below

Length (pipe), L_{pipe} *(m)*	=	1.2
Length (heating element), L_{elem} *(m)*	=	1.372
Outer Diameter, D_o (m)	=	0.070
Inner Diameter, D_i (m)	=	0.046
Pipe Wall Thickness, t_w (m)	=	0.002
Internal pipe radius, r_1 *(m)*	=	0.023
External pipe radius, r_2 *(m)*	=	0.025
Ambient Temperature (pipe internal), $T_{\infty,1}$ *(^oC)*	=	NA
Ambient Temperature, T_∞ *(^oC)*	=	-19.41
Internal Temperature of pipe, $T_{s,1}$ *(^oC)*	=	-16.63
Surface Temperature of pipe, T_s *(^oC)*	=	-16.63
Film Temperature, T_f (^oC)	=	-18.02
Set wind velocity, u_s (m/s)	=	5
Measured wind velocity, u_m (m/s)	=	6.63
Surface area, *A* *(m^2)*	=	0.1884
Convective Heat transfer coefficient, *h* *($W/m^2 . K$)*	=	To be calculated
Overall Heat transfer coefficient, U_1 *($W/m^2 . K$)*	=	To be calculated
Thermal conductivity of air , k_{air} *(W/m . K) at 255K*		22.3×10^{-3}
Thermal conductivity of pipe, K_A *(W/m . K)*	=	60.5
Thermal diffusivity of air, α_{air} (m^2 / s) *at 256K*	=	15.96×10^{-6}

We need thermophysical properties of air at atmospheric pressure and film temperature for calculation of overall heat transfer coefficient with Hilpert correlation, Fand & Keswani constants and Morgan constants using theoretical method.

Using equation (2.22) for film temperature explained earlier,

$$T_f = \frac{T_s + T_\infty}{2}$$

$$T_f = \frac{(-16.63 + 273.15) + (-19.41 + 273.15)}{2}$$

$$T_f = \frac{256.52 + 253.74}{2}$$

$$T_f = 255.13.02\ K\ or\ -18.02\ ℃ \tag{4.19}$$

Table 4-1Thermophysical properties of air at film temperature (Incropera et al., 2006)

Thermal conductivity of air , *k (W/m . K)*	=	22.3×10^{-3}
Thermal diffusivity of air, α (m^2 / s)	=	15.96×10^{-6}
Dynamic viscosity of air, *μ (N. s /m^2)*	=	159.6×10^{-7}
Kinematic viscosity of air, *ν (m^2/s)*	=	11.44×10^{-6}
Density of air, *ρ (kg/m^3)*	=	1.3947

Using equation (8.19) explained earlier for Prandtl Number at film temperature,

$$Pr_f = \frac{\nu}{\propto}$$

$$Pr_f = \frac{11.44 * 10^{-6}}{15.96 * 10^{-6}}$$

$$Prandtl\ Number, Pr_f = 0.716 \tag{4.20}$$

Using equation (2.20) explained earlier for Reynolds Number at film temperature,

$$Re_{D,f} = \frac{\rho u_m D}{\mu}$$

$$Re_{D,f} = \frac{1.3947 * 6.63 * 0.050}{159.6 * \ 10^{-7}}$$

$$\text{Reynolds number}, Re_{D,f} = 28968.86 \tag{4.21}$$

4.2.1.1 Hilpert correlation

Using equation (8.23) explained earlier for Nusselt number,

$$Nu_D = C{Re_{D,f}}^m {Pr_f}^{1/3}$$

$$[Pr_f \geq 0.7]$$

Since, the Prandtl number is above 0.7, we can use the Hilpert correlation.

The overall heat transfer coefficient shall be obtained using original Hilpert constants and the updated Hilpert constants given in Table 2-1 and Table 2-2

Original Hilpert constants

Using Table 2-1, for Reynolds Number between (4,000 - 40,000)

C = 0.174, m = 0.618

Substituting in the below equation to find the Nusselt number,

$$Nu_D = C{Re_{D,f}}^m {Pr_f}^{1/3}$$

$$Nu_D = 0.174 * \ 28968.86^{\ 0.618} * 0.716^{1/3}$$

$$Nusselt\ number, Nu_D = 89.05 \tag{4.22}$$

Using equation (2.18) explained earlier,

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D} \tag{4.23}$$

$$h = \frac{89.05 * 22.3 \text{ x } 10^{-3}}{0.050}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ , \quad h = \ 39.71\ W/m^2.K \tag{4.24}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{r_2}\frac{1}{h_2}} \tag{4.25}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.025} * \left(\frac{1}{39.71}\right)}$$

$$U_1 = \frac{1}{0.02319} \tag{4.26}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ , \quad U_1 = 43.11\ W/m^2.K \tag{4.27}$$

4.2.1.2 Updated Hilpert constants

Using Table 2-2, for Reynolds Number between (4,000 - 40,000),

C = 0.193, m = 0.618

Substituting in the below equation to find the Nusselt number,

$$Nu_D = C{Re_{D,f}}^m {Pr_f}^{1/3}$$

$$Nu_D = 0.193 * \ 28968.86^{\ 0.618} * 0.716^{1/3}$$

$$Nusselt\ number, Nu_D = 98.77 \tag{4.28}$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{98.77 * 22.3 \text{ x } 10^{-3}}{0.050}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ , \quad h = \ 44.05\ W/m^2.K \tag{4.29}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{r_2}\frac{1}{h_2}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.025} * \left(\frac{1}{44.05}\right)}$$

$$U_1 = \frac{1}{0.2091}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ , \quad U_1 = 47.81\ W/m^2.K \tag{4.30}$$

4.2.1.3 Fand and Keswani Reviewed Constants

Using Table 2-3, for Reynolds Number between (4,000 - 40,000)

C = 0.154, m = 0.627

Substituting in the below equation to find the Nusselt number,

$$Nu_D = CRe_{D,f}{}^{m} Pr_f{}^{1/3}$$

$$Nu_D = 0.154 * \ 28968.86^{\ 0.627} * 0.716^{1/3}$$

$$Nusselt\ number, Nu_D = 86.45 \qquad (4.31)$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{86.45 * 22.3 \text{ x } 10^{-3}}{0.050}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ , \quad h = \ 38.55\, W/m^2.K \qquad (4.32)$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A} \ln\frac{r_2}{r_1} + \frac{r_1}{r_2}\frac{1}{h_2}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.025} * \left(\frac{1}{38.55}\right)}$$

$$U_1 = \frac{1}{0.2389}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ ,\quad U_1 = 41.84\ W/m^2.K \tag{4.33}$$

4.2.1.4 Morgan Reviewed Constants

Using Table 2-4, for Reynolds Number between (5,000 - 50,000)

C = 0.148, m = 0.633

Substituting in the below equation to find the Nusselt number,

$$Nu_D = 0.148 * \ 28968.86^{\ 0.633} * 0.716^{1/3}$$

$$Nusselt\ number, Nu_D = 88.36 \tag{4.34}$$

$$Nu_D = C{Re_{D,f}}^m {Pr_f}^{1/3}$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient,

$$h = \frac{Nu_D k}{D}$$

$$= \frac{88.36 * 22.3 \text{ x } 10^{-3}}{0.050}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ ,\quad h = \ 39.41\, W/m^2.K \qquad (4.35)$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{r_2}\frac{1}{h_2}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.025} * \left(\frac{1}{39.41}\right)}$$

$$U_1 = \frac{1}{0.0233}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ ,\quad U_1 = 42.78\ \ W/m^2.K \qquad (4.36)$$

We need thermophysical properties of air at ambient temperature and surface temperature for calculation of overall heat transfer coefficient with the Žukauskas correlation, the Whitaker correlation and the Churchill-Bernstein correlation using theoretical method.

Table 4-2 Thermophysical properties of air at ambient temperature (Incropera et al., 2006)

Thermal conductivity of air , *k (W/m . K)*	=	$22.3 \text{ x } 10^{-3}$
Thermal diffusivity of air, α (m^2 / s)	=	$15.96 \text{ x } 10^{-6}$
Dynamic viscosity of air, *μ (N. s /m^2)*	=	$159.6 \text{ x } 10^{-7}$
Kinematic viscosity of air, *ν (m^2/s)*	=	$11.44\text{x } 10^{-6}$
Density of air, *ρ (kg/m^3)*	=	1.3947

Using equation (2.19) for Prandtl Number at ambient temperature,

$$Pr_a = \frac{\nu}{\propto}$$

$$Pr_a = \frac{11.44 * 10^{-6}}{15.96 * 10^{-6}}$$

$$Prandtl\ Number, Pr_a = 0.716 \tag{4.37}$$

Using equation (2.20) explained earlier for Reynolds Number at ambient temperature,

$$Re_{D,a} = \frac{\rho u_m D}{\mu}$$

$$Re_{D,a} = \frac{1.3947 * 6.63 * 0.050}{159.6 * 10^{-7}}$$

$$\text{Reynolds number}, Re_{D,a} = 28968.86 \tag{4.38}$$

Table 4-3 Thermophysical properties of air at surface temperature (Incropera et al., 2006)

Thermal conductivity of air , *k (W/m . K)*	=	22.3 x 10^{-3}
Thermal diffusivity of air, α (m^2 / s)	=	15.96 x 10^{-6}
Dynamic viscosity of air, *μ (N. s /m^2)*	=	159.6 x 10^{-7}
Kinematic viscosity of air, *ν (m^2/s)*	=	11.44x 10^{-6}
Density of air, *ρ (kg/m^3)*	=	1.3947

Using equation (2.19) for Prandtl Number at surface temperature,

$$Pr_s = \frac{\nu}{\propto}$$

$$Pr_s = \frac{11.44 * 10^{-6}}{15.96 * 10^{-6}}$$

$$Prandtl\ Number, Pr_s = 0.716 \tag{4.39}$$

Using equation (2.20) for Reynolds Number at surface temperature,

$$Re_{D,s} = \frac{\rho u_m D}{\mu}$$

$$Re_{D,s} = \frac{1.3947 * 6.63 * 0.050}{159.6 * 10^{-7}}$$

$$\text{Reynolds number}, Re_{D,s} = 28968.86 \tag{4.40}$$

4.2.1.5 Žukauskas correlation

Using equation (2.24) for Nusselt number,

$$Nu_D = C Re_{D,a}{}^m Pr_a{}^n \left(\frac{Pr_a}{Pr_s}\right)^{1/4}$$

$$\begin{bmatrix} 1 \le Re_{D,a} \le 1 \times 10^6 \\ 0.7 \le Pr_a \le 500 \end{bmatrix}$$

Since the above condition for Prandtl number and Reynolds number is satisfied, we can use the Žukauskas correlation.

Using Table 2-5 and Table 2-6 presented earlier, for Prandtl number <10 and Reynolds Number between (1,000 - 200,000)

C = 0.26, m = 0.6 and n = 0.37

Substituting the values in the above equation,

$$Nu_D = 0.26 * 28968.86^{0.6} * 0.716^{\,0.37} * \left(\frac{0.716}{0.716}\right)^{1/4}$$

$$Nusselt\ number, Nu_D = 109.25 \tag{4.41}$$

Using equation (2.18) explained earlier,

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{109.25 * 22.3 \text{ x } 10^{-3}}{0.050}$$

$$Convective\ Heat\ Transfer\ Coefficient\ , \quad h = 48.72\ W/m^2.K \tag{4.42}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{r_2}\frac{1}{h_2}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.025} * \left(\frac{1}{48.72}\right)}$$

$$U_1 = \frac{1}{0.01891}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ ,\quad U_1 = 52.88\ \ W/m^2.K \tag{4.43}$$

4.2.1.6 Whitaker correlation

Using equation (2.25) for Nusselt number,

$$Nu_D = \left(0.5Re_{D,a}{}^{1/2} + 0.06Re_{D,a}{}^{2/3}\right)Pr_a{}^{0.4}\ \frac{\mu_a}{\mu_s}{}^{1/4}$$

$$\begin{bmatrix} 1.00 \le Re_{D,a} \le 1 \times 10^5 \\ 0.67 \le Pr_a \le 300 \end{bmatrix}$$

Since the above condition for Prandtl number and Reynolds number is satisfied, we can use the Whitaker correlation.

Substituting the values in the above equation,

$$Nu_D = \left(0.5 * 28968.86^{1/2} + 0.06 * 28968.86^{2/3}\right) * 0.716^{\ 0.4} * \left(\frac{159.6 \text{ x } 10^{-7}}{159.6 \text{ x } 10^{-7}}\right)^{1/4}$$

$$Nusselt\ number, Nu_D = \ 123.97 \tag{4.44}$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{123.97 * 22.3 \text{ x } 10^{-3}}{0.050}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ , \quad h = \ 55.29\, W/m^2.K \tag{4.45}$$

Using equation (8.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{r_2}\frac{1}{h_2}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.025} * \left(\frac{1}{55.29}\right)}$$

$$U_1 = \frac{1}{0.0166}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ , \quad U_1 = 59.98\ W/m^2.K \tag{4.46}$$

4.2.1.7 Churchill-Bernstein correlation

Using equation (2.26) for Nusselt number,

$$Nu_D = 0.3 + \frac{0.62 {Re_{D,s}}^{1/2} {Pr_s}^{1/3}}{\left(1 + (0.4/Pr_s)^{2/3}\right)^{1/4}} \times \left[1 + \left(Re_{D,s}/282000\right)^{5/8}\right]^{4/5}$$

Using equation (2.26) for Nusselt number,

$$[Re_D Pr \geq 0.2]$$

Since the above condition $Re_D Pr \geq 0.2$ is satisfied, we can use the Churchill-Bernstein correlation.

Substituting the values in the above equation.

$$Nu_D = 0.3 + \frac{0.62 * 28968.86^{1/2} * 0.716^{1/3}}{(1 + (0.4/0.716)^{2/3})^{1/4}} \times \left[1 + (28968.86/282000)^{5/8}\right]^{4/5}$$

$$Nu_D = 0.3 + 82.94 \times 1.188$$

$$Nusselt\ number, Nu_D = 98.83 \tag{4.47}$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{98.83 * 22.3 \text{ x } 10^{-3}}{0.050}$$

$$Convective\ Heat\ Transfer\ Coefficient, \quad h = 43.48\ W/m^2.K \tag{4.48}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A} \ln\frac{r_2}{r_1} + \frac{r_1}{r_2}\frac{1}{h_2}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.025} * \left(\frac{1}{43.48}\right)}$$

$$U_1 = \frac{1}{0.02119}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ ,\quad U_1 = 47.19\ W/m^2.K \tag{4.49}$$

Table 4-4 Heat transfer coefficient values from different correlations-50 mm uninsulated pipe

Different Heat Transfer Correlations	**Nusselt Number, Nu_D**	**Convective Heat Transfer coefficient, h (W/m^2 .K)**	**Overall Heat Transfer coefficient, U_1 (W/m^2 .K)**
Hilpert Correlation	-		
Original Hilpert Constants	89.05	39.71	43.11
Updated Hilpert Constants	98.77	44.05	47.81
Fand & Keswani Reviewed Constants	86.45	38.55	41.84
Morgan Reviewed Constants	88.36	39.41	42.78
Žukauskas Correlation	109.25	48.72	52.88
Whitaker Correlation	123.97	55.29	59.98
Churchill-Bernstein Correlation	98.33	43.48	47.19

4.2.2 Case 2: Wind blowing over multiple insulated pipes (forced flow scenario)

In this case, we will consider the same insulated pipe under direct influence of the wind flow which we used for calculation using experimental data to find out the heat transfer coefficient. The outer diameter of the pipe is 50 mm, thickness of 2 mm and 10 mm thick insulation. The pipe has a heating element which is centrally placed in the pipe. The ambient temperature is −20 °C and the pipe is subjected to a cross flow wind of 5 m/s. The values used for the calculation will be the actual ambient temperature and wind velocity obtained from calibrated instruments at the time of experimentation. The picture shown in Figure 4-4 depicts the actual setup of the insulated pipe with temperature distribution.

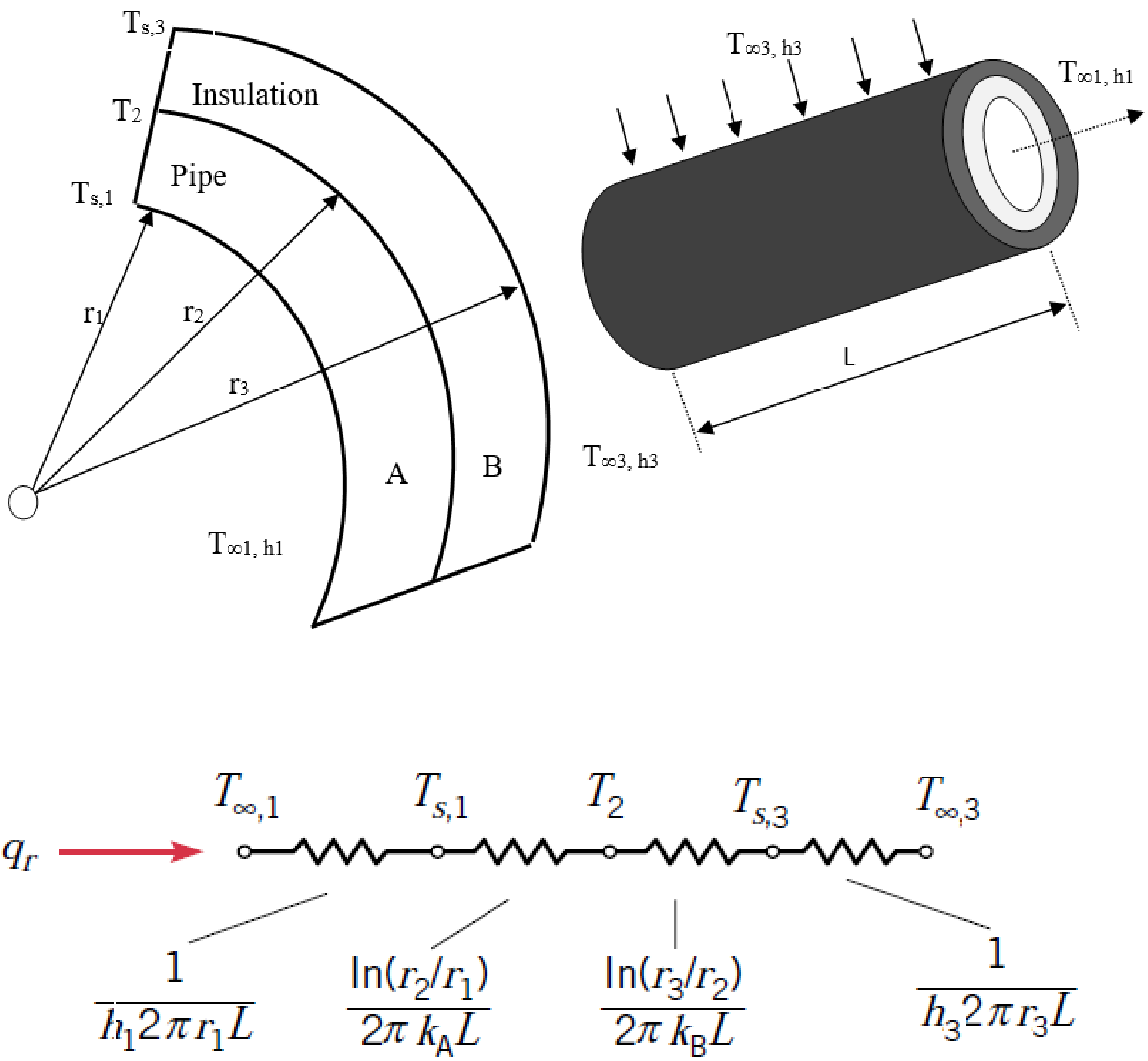

Figure 4-4 Temperature Distribution for the insulated pipe

Assumptions:

1. Overall Steady-state conditions.
2. Heat transfer in the radial direction is one-dimensional.
3. Thermal resistance at the tube wall is negligible.
4. Insulation has constant properties.
5. Negligible radiation loss between surroundings and insulation surface.
6. Negligible heat loss through the end caps of the pipe.
7. Uniform surface temperature for the pipe
8. Change in thermal conductivity over a small temperature range is considered negligible
9. Change in thermal diffusivity over a small temperature range is considered negligible

All the constants and the variables which are to be determined for the insulated pipe heat transfer coefficient calculation are mentioned below.

Length (pipe), L_{pipe} *(m)*	=	1.2
Length (with insulation), L_{ins} *(m)*	=	1.04
Outer Diameter, D_o (m)	=	0.070
Inner Diameter, D_i (m)	=	0.046
Pipe Wall Thickness, t_w (m)	=	0.002
Insulation Thickness, t_{ins} *(m)*	=	0.10
Internal pipe radius, r_1 *(m)*	=	0.023
External pipe radius, r_2 *(m)*	=	0.025
External insulation radius, r_3 *(m)*	=	0.035
Ambient Temperature (pipe internal), $T_{\infty,1}$ *(°C)*	=	NA
Ambient Temperature, $T_{\infty,3}$ *(°C)*	=	-19.68
Internal Temperature of pipe, $T_{s,1}$ *(°C)*	=	38.74
Surface Temperature of pipe, T_2 *(°C)*	=	38.74
Surface Temperature of insulation, $T_{s,3}$ *(°C)*	=	-14.58
Film Temperature, T_f (°C)	=	-17.13
Set wind velocity, u_s (m/s)	=	5
Measured wind velocity, u_m (m/s)	=	6.63
Surface area (pipe internal), A_1 *(m^2)*	=	0.1502
Convective Heat transfer coefficient, h *($W/m^2 \cdot K$)*	=	To be calculated

Overall Heat transfer coefficient, U_1 $(W/m^2 \,.\, K)$	=	To be calculated
Thermal conductivity of air , k_{air} $(W/m \,.\, K)$ *at 256K*		22.3×10^{-3}
Thermal conductivity of pipe, K_A $(W/m \,.\, K)$	=	60.5
Thermal conductivity of insulation, K_B $(W/m \,.\, K)$	=	0.033
Thermal diffusivity of air, α_{air} (m^2/s) *at 256K*	=	15.96×10^{-6}

We need thermophysical properties of air at atmospheric pressure and film temperature for calculation of overall heat transfer coefficient with Hilpert correlation, Fand & Keswani constants and Morgan constants using theoretical method.

Using equation (2.22) for film temperature explained earlier section,

$$T_f = \frac{T_{s,3} + T_{\infty,3}}{2}$$

$$T_f = \frac{(-14.58 + 273.15) + (-19.68 + 273.15)}{2}$$

$$T_f = \frac{258.57 + 253.47}{2}$$

$$T_f = 256.02\, K \; or \; -17.13\,℃ \tag{4.50}$$

Table 4-5 Thermophysical properties of air at film temperature (Incropera et al., 2006)

Thermal conductivity of air , k $(W/m \,.\, K)$	=	22.3×10^{-3}
Thermal diffusivity of air, α (m^2/s)	=	15.96×10^{-6}
Dynamic viscosity of air, μ $(N.\, s/m^2)$	=	159.6×10^{-7}
Kinematic viscosity of air, v (m^2/s)	=	11.44×10^{-6}
Density of air, ρ (kg/m^3)	=	1.3947

Using equation (2.19) explained earlier for Prandtl Number at film temperature,

$$Pr_f = \frac{\nu}{\propto}$$

$$Pr_f = \frac{11.44 * 10^{-6}}{15.96 * 10^{-6}}$$

$$Prandtl\ Number, Pr_f = 0.716 \tag{4.51}$$

Using equation (2.20) explained earlier for Reynolds Number at film temperature,

$$Re_{D,f} = \frac{\rho u_m D}{\mu}$$

$$Re_{D,f} = \frac{1.3947 * 6.63 * 0.070}{159.6 * 10^{-7}}$$

$$\text{Reynolds number}, Re_{D,f} = 40556.40 \tag{4.52}$$

4.2.2.1 Hilpert correlation

Using equation (2.23) explained earlier for Nusselt number,

$$Nu_D = CRe_{D,f}{}^{m}Pr_f{}^{1/3}$$

$$\left[Pr_f \geq 0.7\right]$$

Since the Prandtl number is above 0.7, we can use the Hilpert correlation.

The overall heat transfer coefficient shall be obtained using original Hilpert constants and the updated Hilpert constants given in Table 2-1 and Table 2-2

4.2.2.2 Original Hilpert constants

Using Table 2-1, for Reynolds Number between (40,000-400,000)

C = 0.0239, m = 0.805

Substituting in the below equation to find the Nusselt number,

$$Nu_D = CRe_{D,f}{}^{m} Pr_f{}^{1/3}$$

$$Nu_D = 0.0239 * \ 40556.40^{\ 0.805} * 0.716^{1/3}$$

$$Nusselt\ number, Nu_D = 109.52 \tag{4.53}$$

Using equation (2.18) explained earlier,

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D} \tag{4.54}$$

$$h = \frac{109.52 * 22.3 \text{ x } 10^{-3}}{0.070}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ , \quad h = \ 34.89\ \ W/m^2.K \tag{4.55}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{K_B}\ln\frac{r_3}{r_2} + \frac{r_1}{r_3}\frac{1}{h_3}} \tag{4.56}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.033}\ln\left(\frac{0.035}{0.025}\right) + \frac{0.023}{0.035} * \left(\frac{1}{34.89}\right)}$$

$$U_1 = \frac{1}{0.2533} \tag{4.57}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ , \quad U_1 = 3.947\ \ W/m^2.K \tag{4.58}$$

4.2.2.3 Updated Hilpert constants

Using Table 2-2, for Reynolds Number between (40,000-400,000),

C = 0.027, m = 0.805

Substituting in the below equation to find the Nusselt number,

$$Nu_D = C{Re_{D,f}}^m {Pr_f}^{1/3}$$

$$Nu_D = 0.027 * \ 40556.40^{\ 0.805} * 0.716^{1/3}$$

$$Nusselt\ number, Nu_D = 123.73 \tag{4.59}$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{123.73 * 22.3 \text{ x } 10^{-3}}{0.070}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ , \quad h = \ 39.41\ W/m^2.K \tag{4.60}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{K_B}\ln\frac{r_3}{r_2} + \frac{r_1}{r_3}\frac{1}{h_3}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.033}\ln\left(\frac{0.035}{0.025}\right) + \frac{0.023}{0.035} * \left(\frac{1}{39.41}\right)}$$

$$U_1 = \frac{1}{0.2512}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ ,\quad U_1 = 3.980\ W/m^2.K \tag{4.61}$$

4.2.2.4 Fand and Keswani Reviewed Constants

Using Table 2-3, for Reynolds Number between (40,000-400,000)

C = 0.024, m = 0.898

Substituting in the below equation to find the Nusselt number,

$$Nu_D = CRe_{D,f}{}^m Pr_f{}^{1/3}$$

$$Nu_D = 0.024 *\ 40556.40^{\ 0.898} * 0.716^{1/3}$$

$$Nusselt\ number, Nu_D = 295.04 \tag{4.62}$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{295.04 * 22.3 \text{ x } 10^{-3}}{0.070}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ ,\quad h = \ 93.99\, W/m^2.K \tag{4.63}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{K_B}\ln\frac{r_3}{r_2} + \frac{r_1}{r_3}\frac{1}{h_3}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.033}\ln\left(\frac{0.035}{0.025}\right) + \frac{0.023}{0.035} * \left(\frac{1}{94.26}\right)}$$

$$U_1 = \frac{1}{0.2414}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ ,\quad U_1 = 4.140\ \ W/m^2.K \tag{4.64}$$

4.2.2.5 Morgan Reviewed Constants

Using Table 2-4, for Reynolds Number between (5,000-50,000)

C = 0.0208, m = 0.814

Substituting in the below equation to find the Nusselt number,

$$Nu_D = C{Re_{D,f}}^m {Pr_f}^{1/3}$$

$$Nu_D = 0.0208 * \ 40556.40^{\ 0.814} * 0.716^{1/3}$$

$$Nusselt\ number, Nu_D = 104.87 \tag{4.65}$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient,

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{104.87 * 22.3 \text{ x } 10^{-3}}{0.070}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ , \quad h = \ 33.40\, W/m^2.K \tag{4.66}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{K_B}\ln\frac{r_3}{r_2} + \frac{r_1}{r_3}\frac{1}{h_3}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.033}\ln\left(\frac{0.035}{0.025}\right) + \frac{0.023}{0.035} * \left(\frac{1}{33.50}\right)}$$

$$U_1 = \frac{1}{0.2541}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ , \quad U_1 = 3.935\ \ W/m^2.K \tag{4.67}$$

We need thermophysical properties of air at ambient temperature and surface temperature for calculation of overall heat transfer coefficient with Žukauskas correlation, Whitaker correlation and Churchill-Bernstein correlation using theoretical method.

Table 4-6 Thermophysical properties of air at ambient temperature (Incropera et al., 2006)

Thermal conductivity of air, *k (W/m . K)*	=	22.3×10^{-3}
Thermal diffusivity of air, α (m^2 / s)	=	15.96×10^{-6}
Dynamic viscosity of air, μ *(N. s /m^2)*	=	159.6×10^{-7}
Kinematic viscosity of air, v *(m^2/s)*	=	11.44×10^{-6}
Density of air, ρ *(kg/m^3)*	=	1.3947

Using equation (2.19) for Prandtl Number at ambient temperature,

$$Pr_a = \frac{\nu}{\propto}$$

$$Pr_a = \frac{11.44 * 10^{-6}}{15.96 * 10^{-6}}$$

$$Prandtl\ Number, Pr_a = 0.716 \tag{4.68}$$

Using equation (2.20) explained earlier for Reynolds Number at ambient temperature,

$$Re_{D,a} = \frac{\rho u_m D}{\mu}$$

$$Re_{D,a} = \frac{1.3947 * 6.63 * 0.070}{159.6 * 10^{-7}}$$

$$\text{Reynolds number}, Re_{D,a} = 40556.40 \tag{4.69}$$

Table 4-7 Thermophysical properties of air at surface temperature (Incropera et al., 2006)

Thermal conductivity of air, *k (W/m . K)*	=	22.3×10^{-3}
Thermal diffusivity of air, α (m^2 / s)	=	15.96×10^{-6}
Dynamic viscosity of air, *μ (N. s /m^2)*	=	159.6×10^{-7}
Kinematic viscosity of air, *ν (m^2/s)*	=	11.44×10^{-6}
Density of air, *ρ (kg/m^3)*	=	1.3947

Using equation (2.19) for Prandtl Number at surface temperature,

$$Pr_s = \frac{\nu}{\propto}$$

$$Pr_s = \frac{11.44 * 10^{-6}}{15.96 * 10^{-6}}$$

$$Prandtl\ Number, Pr_s = 0.716 \tag{4.70}$$

Using equation (2.20) for Reynolds Number at surface temperature,

$$Re_{D,s} = \frac{\rho u_m D}{\mu}$$

$$Re_{D,s} = \frac{1.3947 * 6.63 * 0.070}{159.6 * 10^{-7}}$$

$$\text{Reynolds number}, Re_{D,s} = 40556.40 \tag{4.71}$$

4.2.2.6 Žukauskas correlation

Using equation (8.24) for Nusselt number,

$$Nu_D = C Re_{D,a}{}^m Pr_a{}^n \left(\frac{Pr_a}{Pr_s}\right)^{1/4}$$

$$\begin{bmatrix} 1 \leq Re_{D,a} \leq 1 \times 10^6 \\ 0.7 \leq Pr_a \leq 500 \end{bmatrix}$$

Since the above condition for Prandtl number and Reynolds number is satisfied, we can use the Žukauskas correlation.

Using Table 2-5 and Table 2-6 presented earlier, for Prandtl number <10 and Reynolds Number between (1,000 - 200,000)

C = 0.26, m = 0.6 and n = 0.37

Substituting the values in the above equation,

$$Nu_D = 0.26 * 40556.40^{0.6} * 0.716^{\,0.37} * \left(\frac{0.716}{0.716}\right)^{1/4}$$

$$Nusselt\ number, Nu_D = 133.69 \tag{4.72}$$

Using equation (2.18) explained earlier,

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{133.69 * 22.3 \text{ x } 10^{-3}}{0.070}$$

$$Convective\ Heat\ Transfer\ Coefficient, \quad h = 42.59\ W/m^2.K \tag{4.73}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{K_B}\ln\frac{r_3}{r_2} + \frac{r_1}{r_3}\frac{1}{h_3}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.033}\ln\left(\frac{0.035}{0.025}\right) + \frac{0.023}{0.035} * \left(\frac{1}{42.59}\right)}$$

$$U_1 = \frac{1}{0.2499}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ ,\quad U_1 = 4.001\ \ W/m^2.K \tag{4.74}$$

4.2.2.7 Whitaker correlation

Using equation (2.25) for Nusselt number,

$$Nu_D = \left(0.5 Re_{D,a}{}^{1/2} + 0.06 Re_{D,a}{}^{2/3}\right) Pr_a{}^{0.4}\ \frac{\mu_a}{\mu_s}\ {}^{1/4}$$

$$\begin{bmatrix} 1.00\ \le\ Re_{D,a}\ \le\ 1\ \times\ 10^5 \\ 0.67\ \le Pr_a \le 300 \end{bmatrix}$$

Since the above condition for Prandtl number and Reynolds number is satisfied, we can use the Whitaker correlation.

Substituting the values in the above equation,

$$Nu_D = \left(0.5 * 40556.40^{1/2} + 0.06 * 40556.40^{2/3}\right) * 0.716^{\ 0.4} * \left(\frac{159.6 \text{ x } 10^{-7}}{159.6 \text{ x } 10^{-7}}\right)^{1/4}$$

$$Nusselt\ number, Nu_D = \ 150.06 \tag{4.75}$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{150.06 * 22.3 \text{ x } 10^{-3}}{0.070}$$

$$Convective\ Heat\ \ Transfer\ Coefficient\ , \quad h = \ 47.80\ \ W/m^2.K \tag{4.76}$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{K_B}\ln\frac{r_3}{r_2} + \frac{r_1}{r_3}\frac{1}{h_3}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.033}\ln\left(\frac{0.035}{0.025}\right) + \frac{0.023}{0.035} * \left(\frac{1}{47.80}\right)}$$

$$U_1 = \frac{1}{0.2482}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ , \quad U_1 = 4.028\ \ W/m^2.K \tag{4.77}$$

4.2.2.8 Churchill-Bernstein correlation

Using equation (2.26) for Nusselt number,

$$Nu_D = 0.3 + \frac{0.62 {Re_{D,s}}^{1/2} {Pr_s}^{1/3}}{(1 + (0.4/Pr_s)^{2/3})^{1/4}} \times \left[1 + \left(Re_{D,s}/282000\right)^{5/8}\right]^{4/5}$$

$$[Re_D Pr \geq 0.2]$$

Since the above condition $Re_D Pr \geq 0.2$ is satisfied, we can use the Churchill-Bernstein correlation. Substituting the values in the above equation,

$$Nu_D = 0.3 + \frac{0.62 * 40556.40^{1/2} * 0.716^{1/3}}{(1 + (0.4/0.716)^{2/3})^{1/4}} \times \left[1 + (40556.40/282000)^{5/8}\right]^{4/5}$$

$$Nu_D = 0.3 + 98.138 \times 1.379$$

$$Nusselt\ number, Nu_D = 135.63 \quad (4.78)$$

Using equation (2.18),

$$Nu_D = \frac{hD}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{D}$$

$$h = \frac{135.63 * 22.3 \text{ x } 10^{-3}}{0.070}$$

$$Convective\ Heat\ Transfer\ Coefficient, \quad h = 43.20\ W/m^2.K \quad (4.79)$$

Using equation (2.17) for overall heat transfer coefficient w.r.t Area A_1,

$$U_1 = \frac{1}{\frac{r_1}{K_A}\ln\frac{r_2}{r_1} + \frac{r_1}{K_B}\ln\frac{r_3}{r_2} + \frac{r_1}{r_3}\frac{1}{h_3}}$$

Substituting the values in the above equation,

$$U_1 = \frac{1}{\frac{0.023}{60.5}\ln\left(\frac{0.025}{0.023}\right) + \frac{0.023}{0.033}\ln\left(\frac{0.035}{0.025}\right) + \frac{0.023}{0.035} * \left(\frac{1}{43.20}\right)}$$

$$U_1 = \frac{1}{0.2497}$$

$$Overall\ Heat\ \ Transfer\ Coefficient\ ,\quad U_1 = 4.004\ W/m^2.K \tag{4.80}$$

Table 4-8 Heat transfer coefficient values from different correlations-50 mm insulated pipe

Different Heat Transfer Correlations	**Nusselt Number, N_{uD}**	**Convective Heat Transfer coefficient, h (W/m^2 .K)**	**Overall Heat Transfer coefficient, U_1 (W/m^2 .K)**
Hilpert Correlation	-		
Original Hilpert Constants	109.52	34.89	3.947
Updated Hilpert Constants	123.73	39.41	3.980
Fand & Keswani Reviewed Constants	295.04	93.99	4.140
Morgan Reviewed Constants	104.87	33.40	3.935
Žukauskas Correlation	133.69	42.59	4.001
Whitaker Correlation	150.06	47.80	4.028
Churchill-Bernstein Correlation	135.63	43.20	4.004

4.2.3 Case 3: Wind blowing over deck element / flat plate (forced flow scenario)

In this section, heat transfer coefficient for deck element will be calculated using the theoretical method. The same steel deck plate with size 1.1 m x 1.1 m with epoxy coating will be considered here. The plate has heating tracing underneath the coating. The ambient temperature is −20 ˚C and the pipe is subjected to a cross flow wind of 5 m/s. The values used for the calculation will be the actual ambient temperature and wind velocity obtained from calibrated sensors and anemometer at the time of experimentation.

Assumptions:

1. Overall steady-state conditions.
2. Uniform heat transfer coefficient.
3. Constant properties.
4. Uniform surface temperature for the plate.
5. Transition occurs at a critical Reynolds number of $Re_{x,c}$ = of 5 x 10^5.
6. Constant thermal conductivity during one dimensional conduction through the wooden pallet.
7. Change in thermal conductivity of air over a small temperature range is considered negligible
8. Change in thermal diffusivity of air over a small temperature range is considered negligible.
9. 80 % of the wooden pallet area is in touch with the bottom surface of deck element and the rest 20% is exposed to ambient conditions resulting in convective heat transfer.
10. The power lost during the transmission through the cables is 15%.
11. Cross flow wind to the bottom surface of the deck element is obstructed due to the wooden pallet.

All the constants and the variables which are to be used in the calculation overall heat transfer coefficient for the deck element are mentioned below.

Length, *L (m)*	=	1.1
Width, *W (m)*	=	1.1
Thickness, *t (m)*	=	0.03
Length of wooden pallet, L_w *(m)*	=	1.2
Width of wooden pallet, W_w *(m)*	=	0.8
Thickness of wooden pallet contact surface, t_w *(m)*	=	0.03
Surface area of plate surface, *A* (m^2)	=	1.21
Surface area of wooden pallet, *Aw* (m^2)	=	0.96
Ambient Temperature, T_∞ (^oC)	=	-19.18
Surface Temperature, T_s (^oC)	=	-2.033
Set wind velocity, u_s (m/s)	=	5
Measured wind velocity, u_m (m/s)	=	6.63
Convective Heat transfer coefficient, *h* $(W/m^2 . K)$	=	To be calculated

Thermal conductivity of air , k_{air} *(W/m . K) at 256K*	=	22.3 x 10^{-3}
Thermal conductivity of wood, k_{wood} *(W/m. K)*	=	0.15
Thermal diffusivity of air, α_{air} (m^2 / s) *at 256K*	=	15.96 x 10^{-6}
Emissivity , ε	=	0.93
Transition Distance, x_c *(m)*	=	To be calculated
Power efficiency, η	=	0.85

Table 4-9 Thermophysical properties of air at ambient temperature (Incropera et al., 2006)

Thermal conductivity of air , *k (W/m . K)*	=	22.3 x 10^{-3}
Thermal diffusivity of air, α (m^2 / s)	=	15.96 x 10^{-6}
Dynamic viscosity of air, μ *(N. s /m^2)*	=	159.6 x 10^{-7}
Kinematic viscosity of air, ν *(m^2/s)*	=	11.44x 10^{-6}
Density of air, ρ *(kg/m^3)*	=	1.3947

Using equation (2.19) for Prandtl Number at ambient temperature,

$$Pr = \frac{\nu}{\propto}$$

$$Pr = \frac{11.44 * 10^{-6}}{15.96 * 10^{-6}}$$

$$Prandtl\ Number, Pr = 0.716 \tag{4.81}$$

Using equation (2.20) for Reynolds Number at surface temperature,

$$Re_D = \frac{\rho u_m L}{\mu}$$

$$Re_D = \frac{1.3947 * 6.63 * 1.1}{159.6 * 10^{-7}}$$

$$\text{Reynolds number}, Re_D = 637314.98\ or\ 6.373 * 10^5 \tag{4.82}$$

Since, Re_D is larger than the critical Reynolds number ($Re_{x,c}$) of 5 x 10^5, there will be a combination of laminar flow and turbulent flow. There are different equation to calculate the Nusselt number for these flows and they were presented in equation (2.27) and (2.28). They are summarized below,

Nusselt number for Laminar flow,

$$Nu_D = \frac{hD}{k} = 0.664 {Re_D}^{1/2} Pr^{1/3} \qquad [Pr \geq 0.6]$$

Nusselt Number for Turbulent flow

$$Nu_D = (0.037 {Re_D}^{4/5} - A) Pr^{1/3}$$

$$\begin{bmatrix} Re_{x,c} \leq Re_D \leq 1 \times 10^8 \\ 0.6 \leq Pr \leq 60 \end{bmatrix}$$

Where, A is a constant which is determined by the critical Reynolds number $Re_{x,c}$. The equation for A is shown in equation (8.29). For, $Re_{x,c} = 5 \times 10^5$, the value of A = 867 (Incropera et al., 2006)

In order to find the distance x_c, where transition from laminar flow to turbulent flow takes place, the below equation is used.

$$x_c = \frac{\nu Re_{x,c}}{u_m} \tag{4.83}$$

$$x_c = \frac{(11.44 * 10^{-6})}{6.63} * (5.0 * 10^5)$$

$$x_c = 0.8627\, m \tag{4.84}$$

Substituting the values in the above equations,

Nusselt number for Laminar flow,

$$Nu_D = 0.664 * 637314.98^{1/2} * 0.716^{1/3}$$

$$Nusselt\ number\ for\ lamiar\ flow, Nu_D = 474.22 \tag{4.85}$$

Nusselt Number for Turbulent flow

$$Nu_D = (0.037 * 637314.98^{4/5} - 867) * 0.716^{1/3}$$

$$Nusselt\ number\ for\ turbulent\ flow, Nu_D = 680.90 \tag{4.86}$$

Using equation (2.18),

$$Nu_D = \frac{hL}{k}$$

Rearranging for finding convective heat transfer coefficient

$$h = \frac{Nu_D k}{L}$$

Substituting the values to find the convective heat transfer coefficient for laminar flow,

$$h = \frac{474.22 * 22.3 \text{ x } 10^{-3}}{1.1}$$

$$Heat\ \ Transfer\ Coefficient\ for\ laminar\ flow\ , \quad h = \ 9.613\ \ W/m^2.K \tag{4.87}$$

Substituting the values to find the convective heat transfer coefficient for turbulent flow,

$$h = \frac{680.90 * 22.3 \text{ x } 10^{-3}}{1.1}$$

$$Heat\ \ Transfer\ Coefficient\ for\ turbulent\ flow\ , \quad h = \ 13.803\ \ W/m^2.K \tag{4.88}$$

Using Equation (2.6) below, the heat transfer for laminar part and turbulent part is obtained.

$$q = hA\ (T_s - T_\infty) \tag{4.89}$$

Heat Transfer for laminar flow region is found by substituting values in equation (4.89),

$$q = 9.613 * (0.8627 * 1.1) * (271.11 - 253.97)$$

$$Heat\ Transfer\ for\ Laminar\ region, q = 156.42\ W \tag{4.90}$$

Heat Transfer for turbulent flow region is found by substituting values in equation (4.89),

$$q = 13.803 * (0.2373 * 1.1) * (271.11 - 253.97)$$

$$Heat\ Transfer\ for\ Turbulent\ region, q = 61.75\ W \tag{4.91}$$

Heat loss due to thermal radiation can found by equation (2.7)

$$q = \varepsilon\sigma A_t\ (T_s^4 - T_\infty^4)$$

$$q = 0.93 * 5.67 \text{ x } 10^{-8} * \big((1.1 * 1.1 * 2) + (0.03 * 4 * 1.1)\big) * (271.11^4 - 253.97^4)$$

$$Heat\ loss\ due\ to\ thermal\ radiation, q = 167.12\ W \tag{4.92}$$

Heat loss due to conduction from the bottom surface can found by equation (2.1)

$$q_{cond} = -kA_w \frac{(T_2 - T_1)}{t_w}$$

$$q_{cond} = -0.15 * 0.8 * 0.96 * \frac{(253.97 - 271.11)}{0.03}$$

$$Heat\ loss\ due\ to\ conduction, \qquad q = 65.81\ W \tag{4.93}$$

Heat transfer through convection from remaining bottom surface is obtained using Equation (4.89),

$$q = hA\ (T_s - T_\infty)$$

$$q = h\big((A - A_w) + (t * L * 4)\ + (0.2 *\ A_w)\big)\ (T_s - T_\infty)$$

$$q = 9.613 * \big((1.21 - 0.96) + (0.03 * 4 * 1.1) + (0.2 * 0.96)\big) * (271.11 - 253.97)$$

$$Heat\ loss\ \ through\ convection\ from\ bottom\ surface, q = 94.57\ W \tag{4.94}$$

Total heat transfer is found by summation of (4.90), (4.91), (4.92), (4.93) and (4.94),

$$Total\ Heat\ transfer, q = \ q_{lam} + \ q_{turb} + q_{rad} + \ q_{cond} + q_{conv}$$

$$Total\ Heat\ transfer, q = 156.42 + 61.75 + 167.12 + \ 65.81 + 94.57\ W$$

$$Total\ Heat\ transfer, q = 545.68\ W \tag{4.95}$$

The amount of power which needs to be supplied to maintain a constant surface temperature with 85% efficiency for the heat tracing is

$$q_s = \frac{q}{\eta} \tag{4.96}$$

$$q_s = \frac{545.68}{0.85}$$

$$Power\ Needed, \qquad q_s = 641.98\ \ W \tag{4.97}$$

5 Results and Discussion

The results from detailed calculations and analysis are presented in this section. Since, there are many plots and tables from the calculations, it is considered appropriate to discuss and comment on them in the same section for better understanding. As mentioned earlier, the experiments were performed jointly with (Kvamme, 2016) and the scope was subsequently split up for detailed calculation and analysis. The result presented in this section pertains to heat transfer coefficient for multiple pipe configuration which were performed using experimental and theoretical methods. But, some results from analysis of single pipe configuration like uninsulated pipe is used in this thesis to show the effect of insulation on heat transfer coefficient.

The diagram shown in Figure 5-1 illustrates the multiple insulated pipe configuration along with positioning of sensors and the applicable pipe surfaces used for analysis. The overall heat transfer coefficient for the pipe is obtained using temperature from all 6 sensors, whereas, analysis of heat transfer coefficient for top and bottom pipe surface involved usage of readings from only those 3 sensors which were connected to the respective top and bottom surface of the pipe.

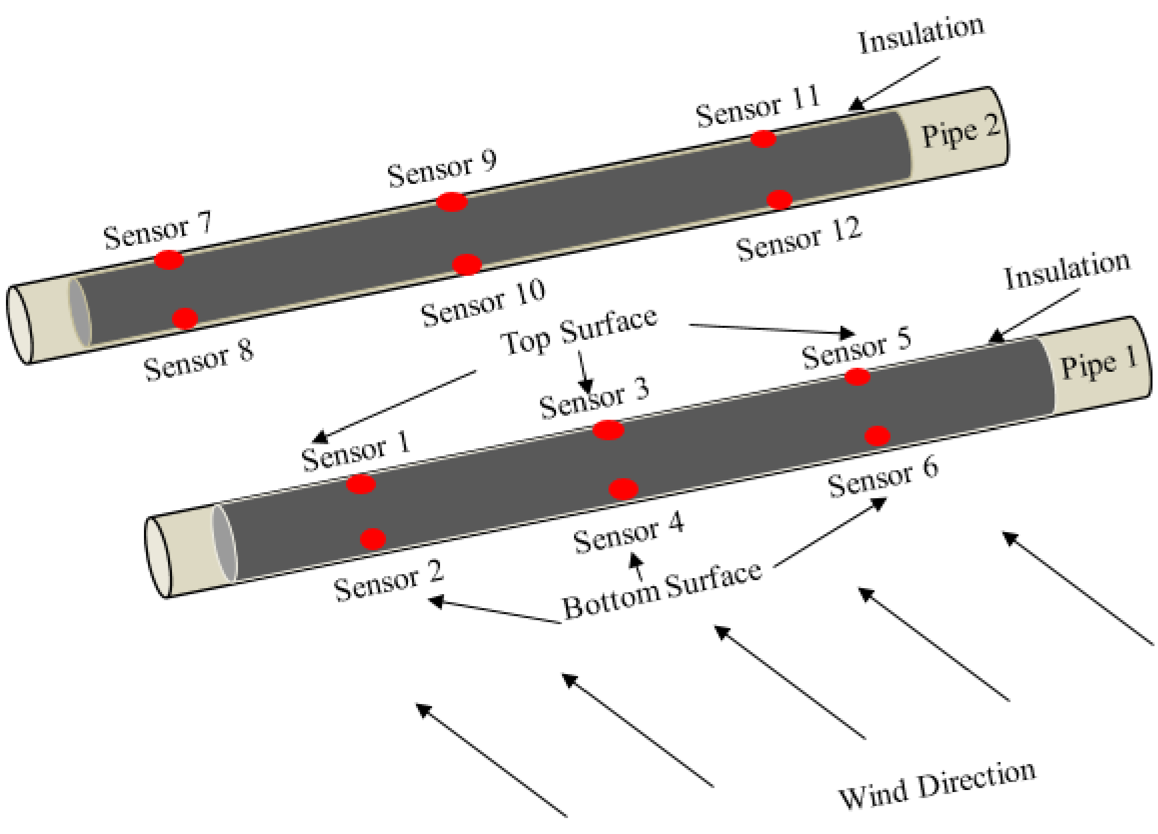

Figure 5-1 Pipe configuration showing top and bottom surface along with temperature sensors

5.1 Results from Experimental Method

5.1.1 Case 1: Heat Transfer co-efficient for uninsulated pipe.

Table 5-1 from Experiment 11 shows the overall heat transfer coefficient and temperature readings for 1 x 50 mm uninsulated pipe. Plots of overall heat transfer coefficient for uninsulated pipe versus wind velocity is shown in Figure 5-2.

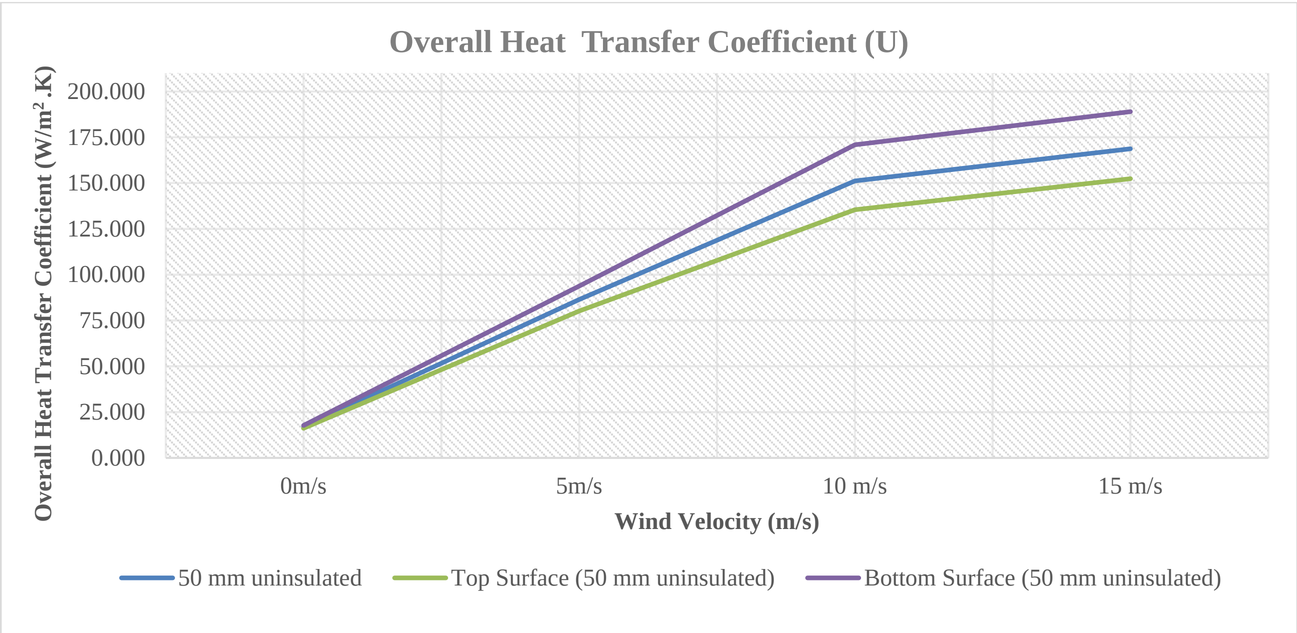

Figure 5-2 Overall Heat Transfer coefficient for a single uninsulated pipe v/s insulated pipe.

The plot from experiment 11 in Figure 5-2 clearly shows that the overall heat transfer coefficient for the uninsulated pipe is very high with an average value of 17 W/m^2. K for 0 m/s wind velocity and increases considerably as the wind velocity is raised keeping the ambient conditions constant. The aim of the experiment was to show the effect of cross flow wind on the heat transfer coefficient and it is observed that the overall heat transfer coefficient increases by 400 % with an increase of 5 m/s in the cross flow wind and touches a value of 169 W/m^2. K at 15 m/s wind speed which corresponds to 894 % of the initial value. These numbers are very significant as they show the rate of heat loss from pipes which are uninsulated and can be seen as the indication of the energy that is lost in transit when hot fluids are circulated through an uninsulated piping system. Additionally, the plot also shows the overall heat transfer coefficient for the top and bottom surface of the uninsulated pipe and how it relates to the average overall heat transfer coefficient value. As expected, the heat transfer coefficient of the bottom surface is higher than the top surface because of the convection inside the pipe. The warm air inside the pipe rises up resulting in the heating up of the top surface. This explains the 30 % rise in the temperature of the top pipe surface compared to the bottom surface for 0 m/s wind speed as evident from the experimental readings shown in Table 5-1. The temperature readings are within 3 % range with increase in wind speed as the circulation of heat helps in balancing the temperature at these surfaces. Also, it can be seen that until 10 m/s wind speed there is a steady increase in the overall heat transfer coefficient. But, its value doesn't show significant change when the wind speed is increased to 15 m/s as the pipe surface temperature is in equilibrium with the ambient temperature (Oosthuizen and Naylor, 1999).

Table 5-1 Experiment 11-Heat Transfer Coefficient and Temperature Readings

Experiment 11		$T_{ambient}$ (°C)	Pipe 1		Pipe 1		Pipe 1	
			T_{avg} (°C)	U_{avg} ($W/m^2.K$)	T_{top} (°C)	U_{top} ($W/m^2.K$)	T_{bottom} (°C)	U_{bottom} ($W/m^2.K$)
0 m/s	Run 1	-19.70	-6.00	17.581	-5.31	16.739	-6.69	18.512
	Run 2	-19.36	-4.78	16.533	-4.10	15.794	-5.46	17.345
	Run 3	-19.44	-4.87	16.543	-4.23	15.840	-5.52	17.311
	Average	-19.50	-5.22	16.871	-4.55	16.113	-5.89	17.705
5 m/s	Run 1	-19.52	-16.45	78.523	-16.25	73.798	-16.64	83.895
	Run 2	-19.42	-16.79	91.651	-16.56	84.285	-17.02	100.428
	Run 3	-19.31	-16.65	90.409	-16.42	83.301	-16.87	98.844
	Average	-19.41	-16.63	86.438	-16.41	80.172	-16.84	93.768
10 m/s	Run 1	-18.80	-17.24	154.586	-17.06	138.566	-17.42	174.794
	Run 2	-18.91	-17.33	152.471	-17.14	136.293	-17.52	173.009
	Run 3	-18.73	-17.08	146.581	-16.90	131.714	-17.27	165.230
	Average	-18.81	-17.22	151.136	-17.04	135.464	-17.40	170.909
15 m/s	Run 1	-17.33	-15.84	161.415	-15.69	146.912	-15.99	179.096
	Run 2	-17.99	-16.58	171.657	-16.43	154.622	-16.74	192.910
	Run 3	-18.04	-16.65	173.664	-16.49	155.934	-16.81	195.943
	Average	-17.79	-16.36	168.738	-16.20	152.384	-16.51	189.025

5.1.2 Case 2: Heat Transfer co-efficient for insulated pipes.

5.1.2.1 Experiment No: 2

Table 5-2 shows the heat transfer coefficient and temperature readings for 2 x 50 mm insulated pipe configuration. Plots of overall heat transfer coefficient versus wind velocity for different surfaces are shown in Figure 5-3, Figure 5-4 and Figure 5-5.

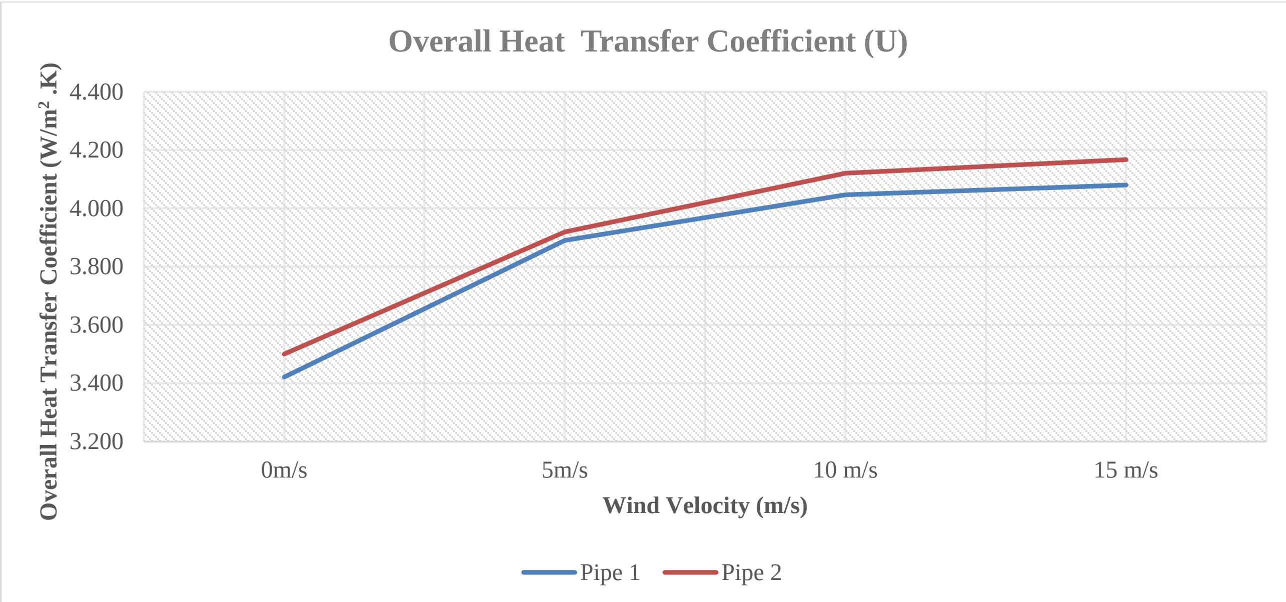

Figure 5-3 Experiment 2- Overall Heat Transfer coefficient for Pipe 1 & Pipe 2

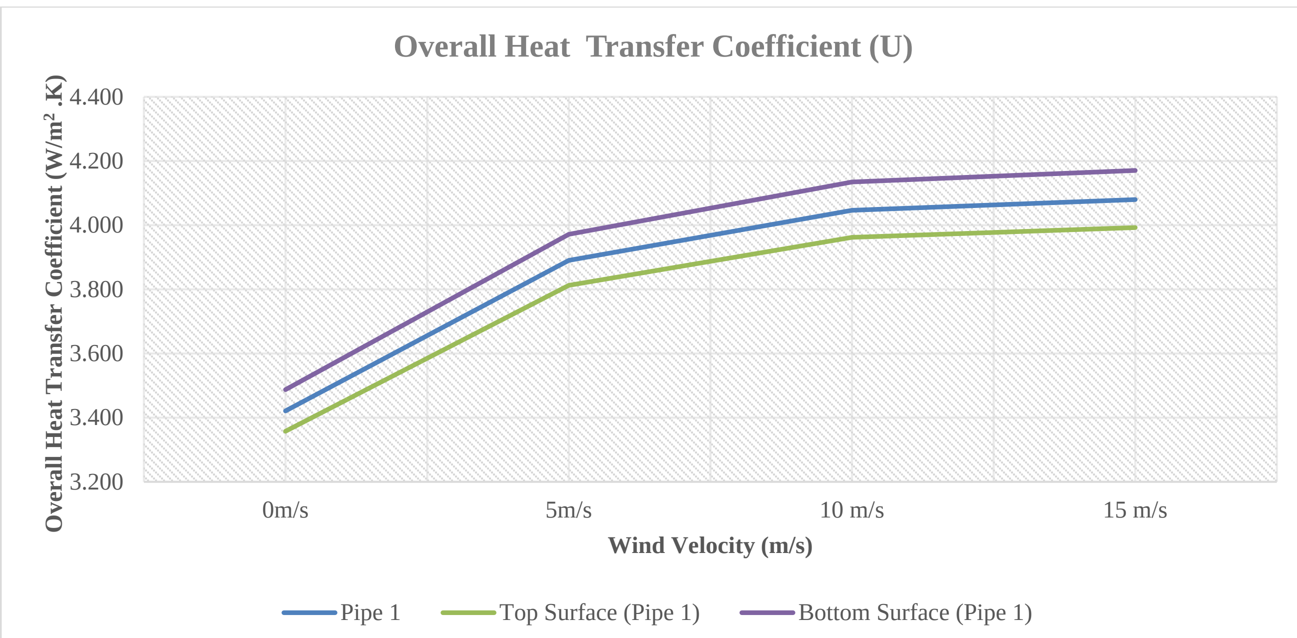

Figure 5-4 Experiment 2- Overall Heat Transfer coefficient for Pipe 1 at different wind velocity

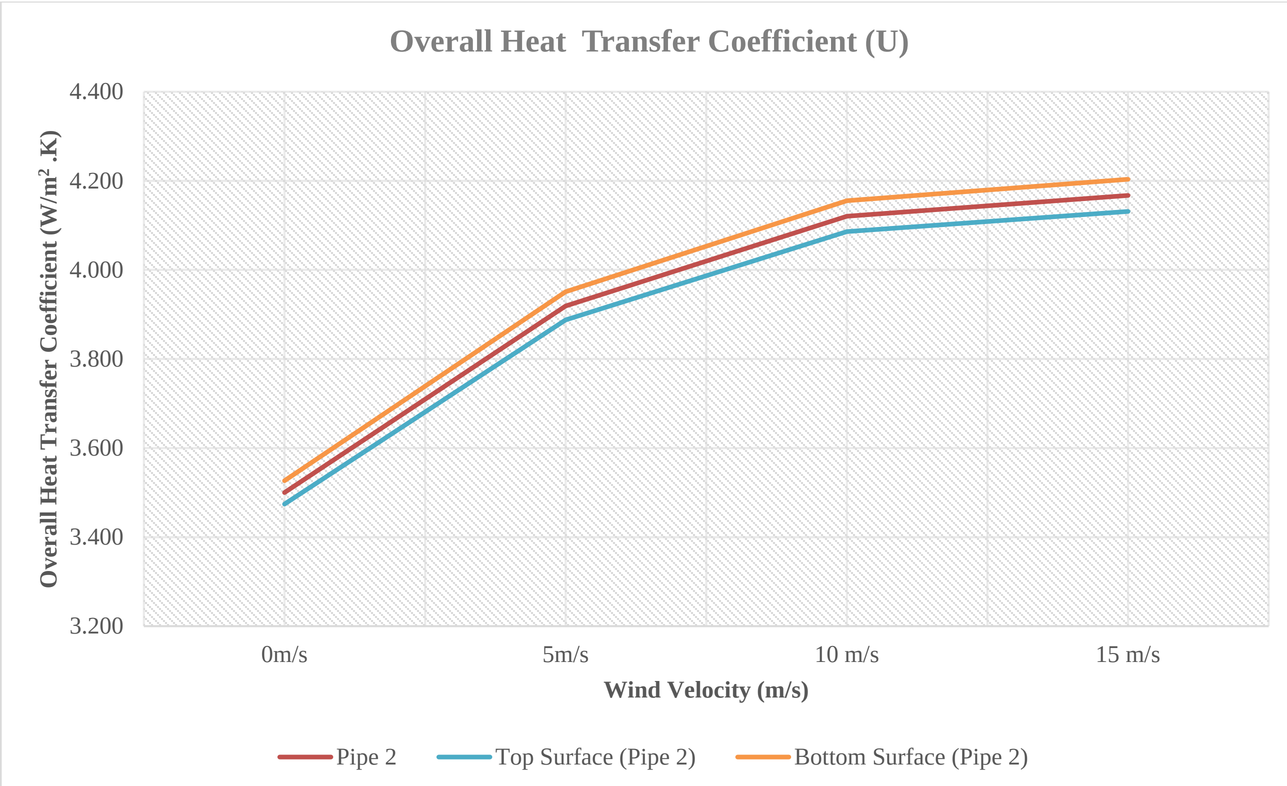

Figure 5-5 Experiment 2- Overall Heat Transfer coefficient for Pipe 2 at different wind velocity

The plot from experiment 2 in Figure 5-3 shows the overall heat transfer coefficient for two numbers diameter 50 mm insulated pipes placed one behind another and subjected to cross flow wind. It is observed that the overall heat transfer coefficients for the insulated pipes are in the range of 3.4 - 4.2 W/m^2. K. The average value of overall heat transfer coefficient increases as the wind velocity is raised while the ambient conditions remains the same. But, the change is not substantial from 10 m/s to 15 m/s as the pipe's surface temperature is in balance with the surroundings as observed in Table 5-2. The main objective of the experiment was to study the effect of cross flow wind on the heat transfer coefficient for the pipe 2 due to the hindrance from pipe 1, and it is observed that the change in overall heat transfer coefficient is miniscule with just 2.3 % increase in the value. It can be seen that the rate of heat transfer from insulated pipe is lower which is advantageous. Furthermore, as expected from theory, the plot of overall heat transfer coefficient for the top and bottom surfaces of the insulated pipe (Figure 5-4 and Figure 5-5) shows that the heat transfer coefficient of the bottom surface is slightly higher than the top surface because of convective heat transfer inside pipe. The overall heat transfer coefficient and temperature readings for pipe 1 and pipe 2 throughout the experiment relate very well with 2-3 % change as can be seen in Table 5-2. The experiment produced expected results with minimal deviation (Faghri et al., 2010).

Table 5-2 Experiment 2-Heat Transfer Coefficient and Temperature Readings

Experiment 2			**Pipe 1**		**Pipe 2**		**Pipe 1**		**Pipe 2**		**Pipe 1**		**Pipe 2**	
		$T_{ambient}$ (°C)	T_{avg} (°C)	U_{avg} ($W/m^2.K$)	T_{avg} (°C)	U_{avg} ($W/m^2.$	T_{top} (°C)	U_{top} ($W/m^2.K$)	T_{top} (°C)	U_{top} ($W/m^2.$	T_{bottom} (°C)	U_{bottom} ($W/m^2.K$	T_{bottom} (°C)	U_{bottom} ($W/m^2.K$)
0 m/s	Run 1	-19.31	46.46	3.455	45.00	3.534	47.70	3.391	45.48	3.508	45.21	3.522	44.52	3.560
	Run 2	-19.32	47.47	3.403	45.93	3.483	48.72	3.340	46.41	3.457	46.21	3.468	45.45	3.509
	Run 3	-19.54	47.19	3.406	45.68	3.485	48.45	3.342	46.17	3.459	45.93	3.472	45.19	3.511
	Average	-19.39	47.04	3.421	45.54	3.500	48.29	3.358	46.02	3.474	45.78	3.487	45.05	3.526
5 m/s	Run 1	-19.72	38.32	3.915	37.87	3.946	39.51	3.837	38.34	3.914	37.13	3.997	37.41	3.978
	Run 2	-19.69	39.47	3.841	39.05	3.869	40.68	3.764	39.53	3.838	38.27	3.921	38.58	3.900
	Run 3	-19.63	38.43	3.915	38.00	3.943	39.61	3.836	38.48	3.911	37.24	3.996	37.53	3.976
	Average	-19.68	38.74	3.890	38.31	3.919	39.94	3.812	38.78	3.888	37.55	3.971	37.84	3.951
10 m/s	Run 1	-19.60	36.53	4.049	35.54	4.121	37.73	3.964	36.00	4.087	35.33	4.137	35.08	4.156
	Run 2	-19.52	36.62	4.048	35.61	4.123	37.82	3.964	36.08	4.088	35.42	4.137	35.14	4.158
	Run 3	-19.43	36.79	4.042	35.77	4.117	37.98	3.959	36.23	4.083	35.59	4.130	35.30	4.152
	Average	-19.52	36.65	4.046	35.64	4.120	37.84	3.962	36.10	4.086	35.45	4.135	35.17	4.155
15 m/s	Run 1	-19.32	37.06	4.031	35.90	4.116	38.29	3.945	36.37	4.081	35.82	4.121	35.42	4.152
	Run 2	-19.26	36.57	4.071	35.40	4.158	37.79	3.984	35.87	4.122	35.36	4.161	34.93	4.194
	Run 3	-19.05	35.87	4.139	34.69	4.229	37.06	4.051	35.16	4.192	34.68	4.230	34.22	4.266
	Average	-19.21	36.50	4.080	35.33	4.167	37.71	3.993	35.80	4.131	35.28	4.171	34.86	4.203

5.1.2.2 Experiment No: 3

Table 5-3 shows the heat transfer coefficient and temperature readings for 3 x 50 mm insulated pipe configuration. Plots of overall heat transfer coefficient versus different wind velocities and surfaces are shown in Figure 5-6, Figure 5-7and Figure 5-8.

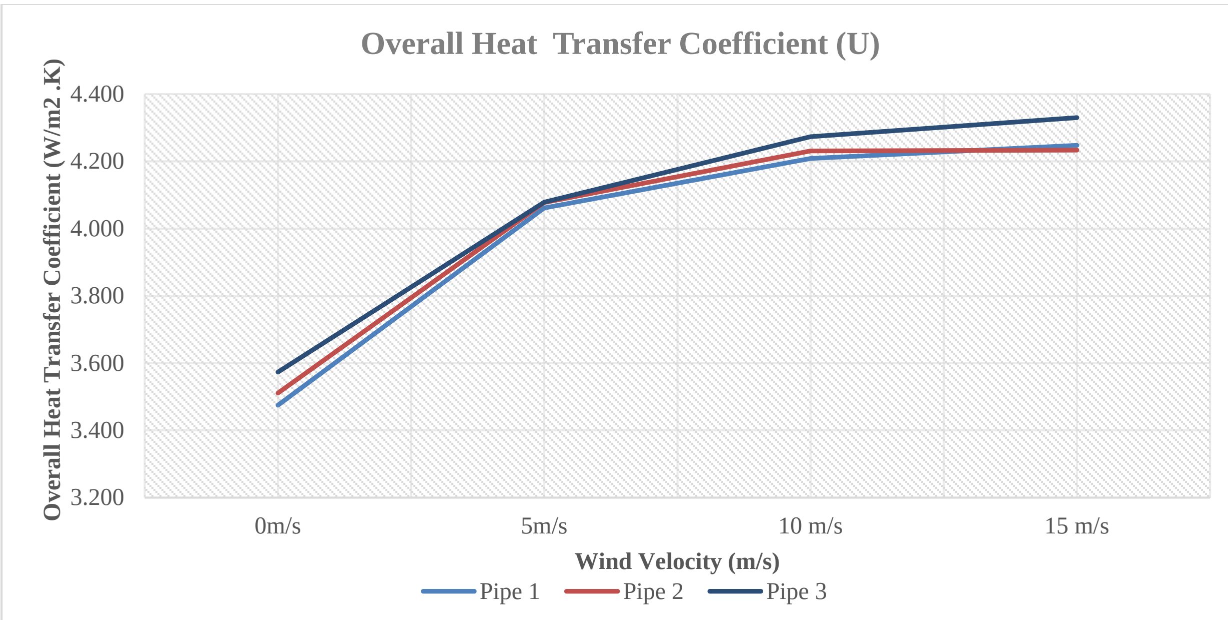

Figure 5-6 Experiment 3- Overall Heat Transfer coefficient for Pipe 1, Pipe 2 & Pipe 3

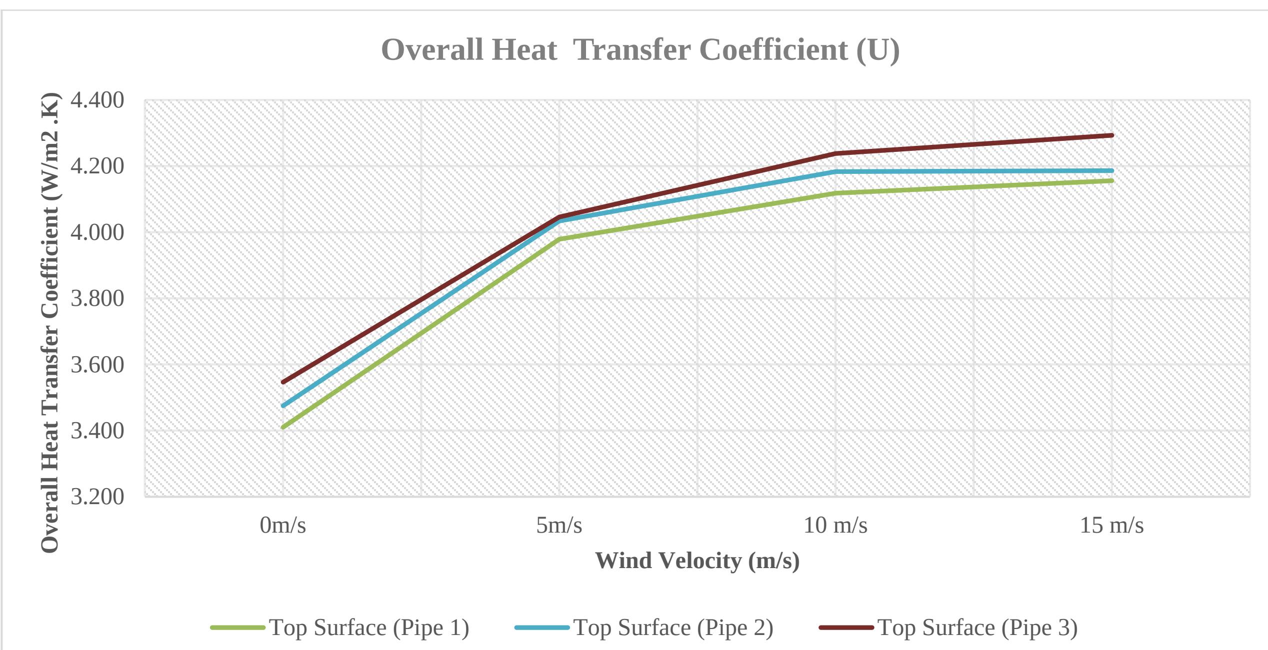

Figure 5-7 Experiment 3- Overall Heat Transfer coefficient for Top Surface

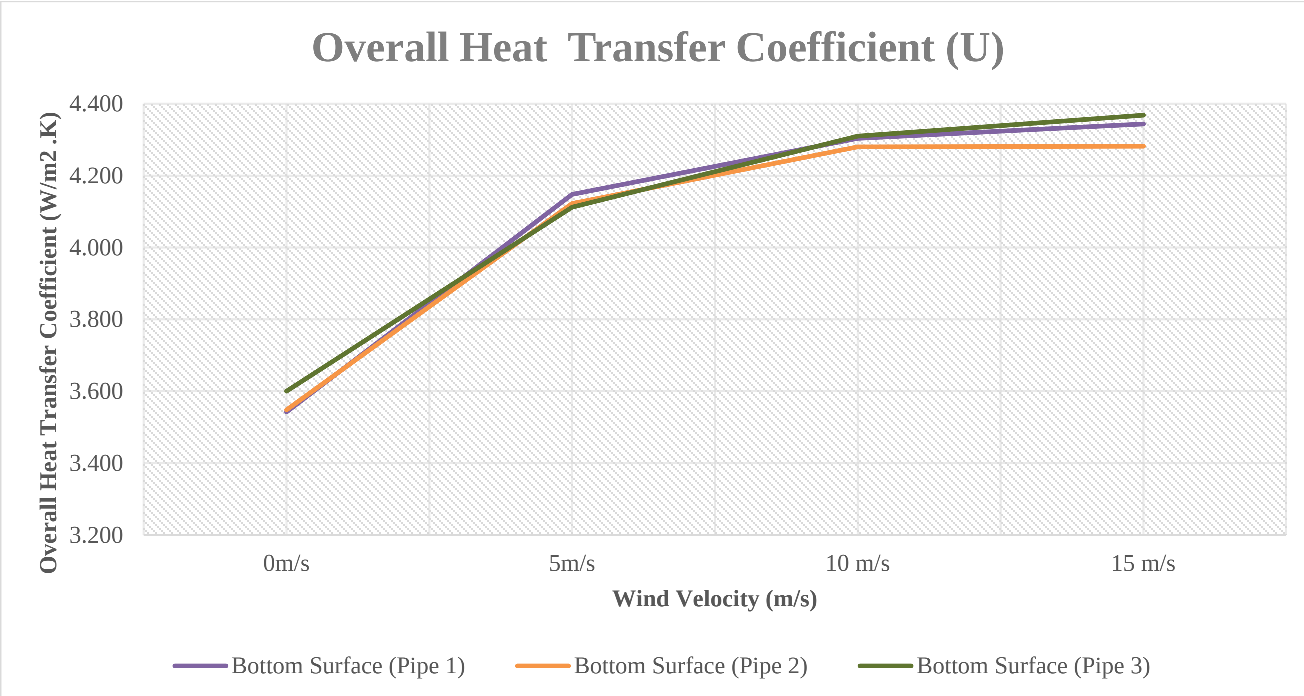

Figure 5-8 Experiment 3- Overall Heat Transfer coefficient for Bottom Surface

The plot from experiment 3 in Figure 5-6 shows the overall heat transfer coefficient for three number diameter 50 mm insulated pipes placed one behind another and subjected to cross flow wind. The results are very different from the earlier 2 x 50 mm case as can be seen clearly in the plots with the overall heat transfer coefficients varying from 3.4 – 4.3 W/m^2. K. Though, the average value of overall heat transfer coefficient increases as the wind velocity is increased with the ambient temperature at approx. -20 °C. But, as seen in experiment 2, the change is not significant from 10 m/s to 15 m/s as the pipe's surface temperature is comparable to the surrounding temperature which can be observed in Table 5-3. The effect on the heat transfer coefficient of pipe 3 and pipe 2 due to pipe 1 is very low with maximum difference of 2% across all the different runs. The plot of overall heat transfer coefficient for the top and bottom surfaces of the insulated pipe (Figure 5-7 and Figure 5-8) shows similar trend as experiment 2 with the heat transfer coefficient of the bottom surface being slightly higher than the top surface for the same reason. The closeness of the overall heat transfer coefficient values and temperature readings for pipe 1, pipe 2 and pipe 3 throughout the experiment can be seen in Table 5-3. We can clearly see that the experiment 3 follows similar trend and addition of diameter 50 mm insulated pipe doesn't have much implication on the values of overall heat transfer coefficient (Kreith et al., 2011).

Table 5-3 Experiment 3-Heat Transfer Coefficient and Temperature Readings

Experiment 3			Pipe 1		Pipe 2		Pipe 3		Pipe 1		Pipe 2		Pipe 3		Pipe 1		Pipe 2		Pipe 3	
		$T_{ambient}$ (°C)	T_{avg} (°C)	U_{avg} ($W/m^2.K$)	T_{avg} (°C)	U_{avg} ($W/m^2.K$)	T_{avg} (°C)	U_{avg} ($W/m^2.K$)	T_{top} (°C)	U_{top} ($W/m^2.K$)	T_{top} (°C)	U_{top} ($W/m^2.K$)	T_{top} (°C)	U_{top} ($W/m^2.K$)	T_{bottom} (°C)	U_{bottom} ($W/m^2.K$)	T_{bottom} (°C)	U_{bottom} ($W/m^2.K$)	T_{bottom} (°C)	U_{bottom} ($W/m^2.K$)
0 m/s	Run 1	-19.22	45.65	3.489	44.85	3.532	43.44	3.611	46.88	3.424	45.53	3.495	43.92	3.584	44.42	3.556	44.16	3.570	42.96	3.639
	Run 2	-19.46	45.91	3.461	45.37	3.490	44.53	3.536	47.15	3.397	46.03	3.455	45.00	3.510	44.67	3.528	44.71	3.526	44.06	3.562
	Average	-19.34	45.78	3.475	45.11	3.511	43.99	3.573	47.01	3.410	45.78	3.475	44.46	3.547	44.54	3.542	44.44	3.548	43.51	3.600
5 m/s	Run 1	-19.62	35.00	4.143	34.82	4.156	34.66	4.169	36.15	4.057	35.42	4.111	35.11	4.134	33.85	4.232	34.23	4.202	34.21	4.204
	Run 2	-19.73	37.08	3.983	36.81	4.002	36.94	3.993	38.25	3.903	37.42	3.959	37.39	3.962	35.91	4.067	36.21	4.045	36.50	4.024
	Average	-19.67	36.04	4.061	35.82	4.078	35.80	4.079	37.20	3.979	36.42	4.034	36.25	4.046	34.88	4.148	35.22	4.122	35.35	4.112
10 m/s	Run 1	-20.02	33.27	4.246	33.14	4.257	32.44	4.314	34.46	4.154	33.75	4.208	32.89	4.277	32.08	4.343	32.53	4.306	31.99	4.351
	Run 2	-19.52	34.72	4.172	34.28	4.206	33.92	4.234	35.90	4.083	34.90	4.158	34.36	4.200	33.53	4.265	33.67	4.254	33.48	4.270
	Average	-19.77	33.99	4.209	33.71	4.231	33.18	4.274	35.18	4.118	34.32	4.183	33.62	4.238	32.81	4.304	33.10	4.280	32.73	4.310
15 m/s	Run 1	-19.33	33.54	4.280	34.08	4.237	32.62	4.356	34.71	4.187	34.67	4.191	33.07	4.318	32.37	4.377	33.49	4.284	32.17	4.394
	Run 2	-19.41	34.27	4.216	34.08	4.230	33.16	4.305	35.45	4.125	34.70	4.182	33.61	4.267	33.08	4.311	33.46	4.279	32.70	4.342
	Average	-19.37	33.90	4.248	34.08	4.233	32.89	4.330	35.08	4.156	34.69	4.186	33.34	4.293	32.72	4.344	33.48	4.282	32.43	4.368

5.1.2.3 Experiment No: 7

Table 5-4 shows the heat transfer coefficient and temperature readings for 1 x 25mm and 1 x 50 mm insulated pipe configuration. Plots of overall heat transfer coefficient versus wind velocity and different surfaces are shown in Figure 5-9 and Figure 5-10.

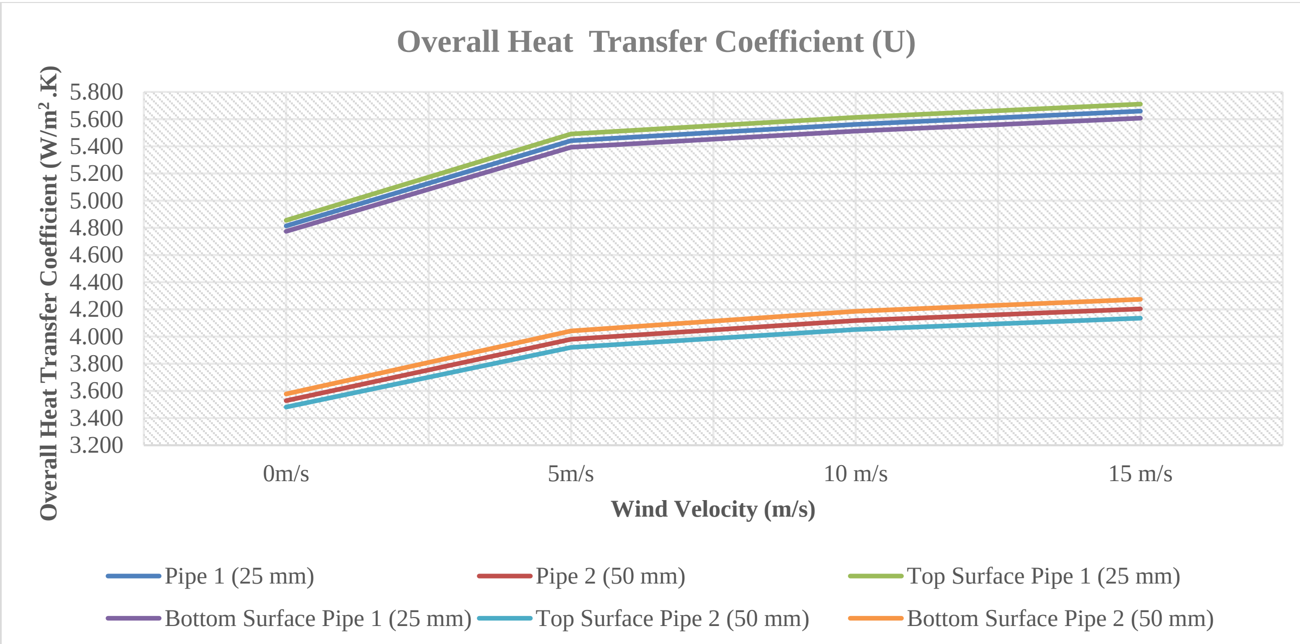

Figure 5-9 Experiment 7- Overall Heat Transfer coefficient for the whole pipe configuration

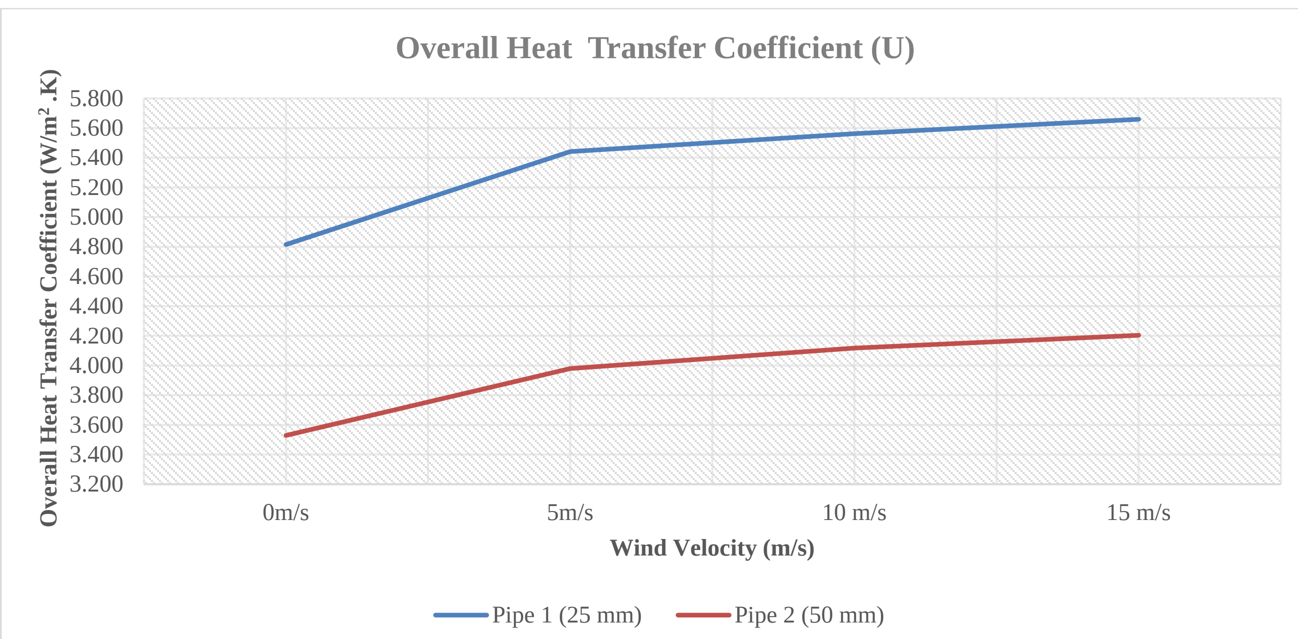

Figure 5-10 Experiment 7- Overall Heat Transfer coefficient for Pipe 1 & Pipe 2

Table 5-4 Experiment 7-Heat Transfer Coefficient and Temperature Readings

Experiment 7			Pipe 1		Pipe 2		Pipe 1		Pipe 2		Pipe 1		Pipe 2	
		$T_{ambient}$ (°C)	T_{avg} (°C)	U_{avg} ($W/m^2.K$)	T_{avg} (°C)	U_{avg} ($W/m^2.K$)	T_{top} (°C)	U_{top} ($W/m^2.K$)	T_{top} (°C)	U_{top} ($W/m^2.K$)	T_{bottom} (°C)	U_{bottom} ($W/m^2.K$)	T_{bottom} (°C)	U_{bottom} ($W/m^2.K$)
0 m/s	Run 1	-19.42	90.43	4.813	49.01	3.527	89.51	4.854	49.91	3.482	91.35	4.773	48.12	3.574
	Run 2	-19.52	89.79	4.837	48.76	3.535	88.88	4.878	49.70	3.487	90.70	4.797	47.81	3.585
	Run 3	-19.52	90.75	4.795	49.00	3.522	89.83	4.835	49.97	3.474	91.67	4.755	48.04	3.573
	Average	-19.48	90.32	4.815	48.92	3.528	89.40	4.855	49.86	3.481	91.24	4.775	47.99	3.577
5 m/s	Run 1	-19.52	77.12	5.471	40.87	3.997	76.26	5.520	41.78	3.937	77.97	5.423	39.96	4.058
	Run 2	-19.54	77.37	5.455	40.91	3.993	76.51	5.504	41.84	3.932	78.23	5.408	39.97	4.055
	Run 3	-19.43	78.48	5.400	41.69	3.949	77.59	5.449	42.63	3.889	79.37	5.351	40.75	4.011
	Average	-19.50	77.66	5.442	41.16	3.979	76.79	5.491	42.08	3.919	78.52	5.394	40.23	4.041
10 m/s	Run 1	-19.43	74.47	5.630	38.28	4.182	73.62	5.682	39.23	4.115	75.32	5.580	37.34	4.252
	Run 2	-19.10	77.06	5.498	40.39	4.057	76.17	5.549	41.35	3.992	77.94	5.448	39.42	4.124
	Run 3	-19.35	75.76	5.559	39.30	4.116	74.89	5.610	40.25	4.050	76.63	5.509	38.35	4.184
	Average	-19.29	75.76	5.562	39.32	4.118	74.90	5.613	40.28	4.052	76.63	5.512	38.37	4.186
15 m/s	Run 1	-16.49	76.34	5.696	40.11	4.265	75.50	5.748	41.06	4.194	77.17	5.645	39.16	4.338
	Run 2	-17.29	77.35	5.587	41.11	4.134	76.49	5.638	42.07	4.067	78.21	5.536	40.15	4.203
	Run 3	-17.55	75.29	5.695	39.72	4.215	74.43	5.748	40.67	4.146	76.15	5.643	38.77	4.286
	Average	-17.11	76.32	5.659	40.31	4.204	75.47	5.711	41.26	4.135	77.18	5.608	39.36	4.275

5.1.2.4 Experiment No: 9

Table 5-5 shows the heat transfer coefficient and temperature readings for 2 x 25 mm insulated pipe configuration. Plots of overall heat transfer coefficient versus different wind velocities and surfaces are shown in Figure 5-11 and Figure 5-12

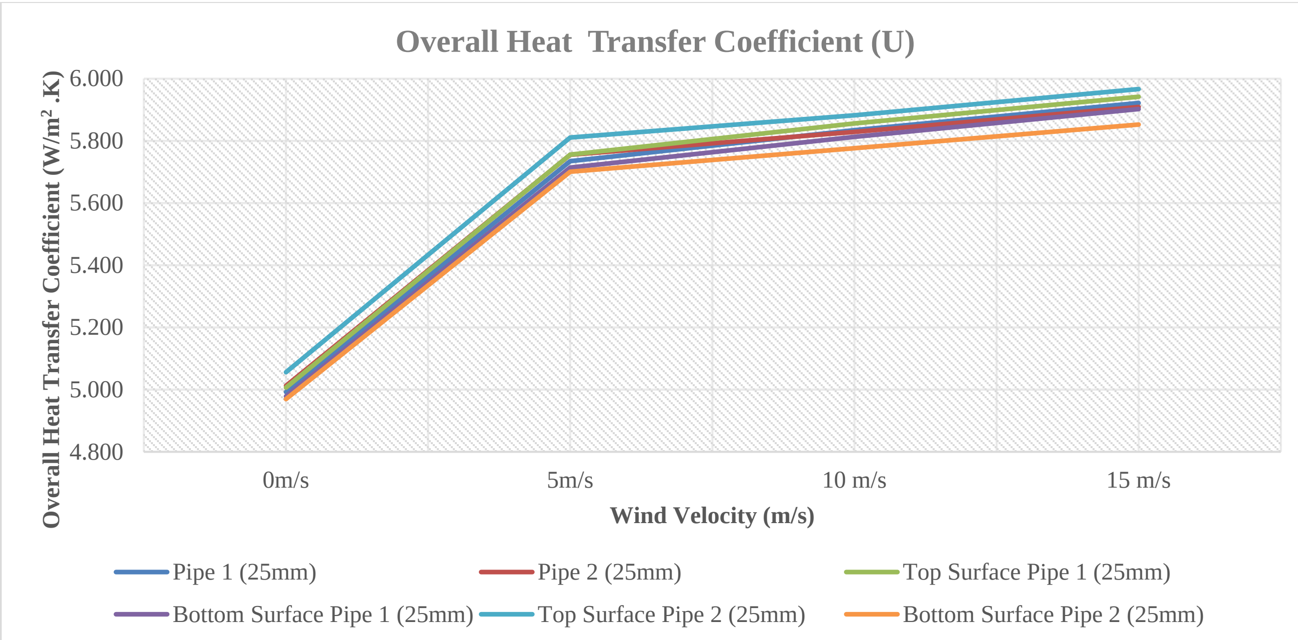

Figure 5-11 Experiment 9- Overall Heat Transfer coefficient for the whole pipe configuration

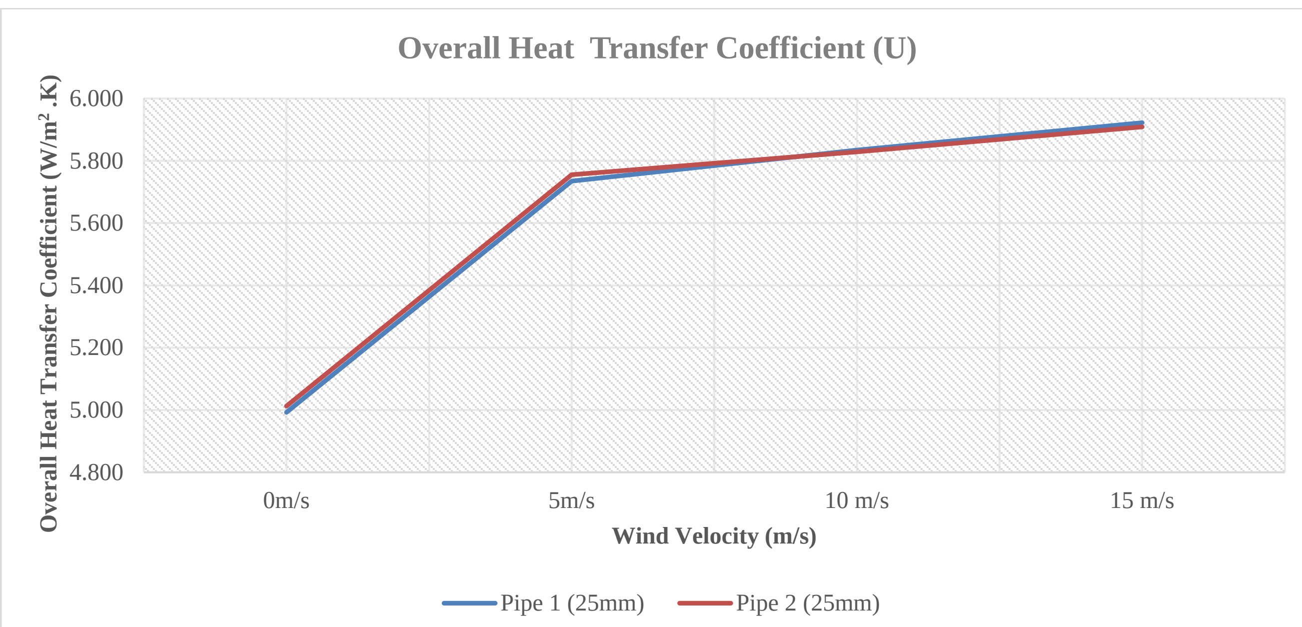

Figure 5-12 Experiment 9- Overall Heat Transfer coefficient for Pipe 1 & Pipe 2

Table 5-5 Experiment 9-Heat Transfer Coefficient and Temperature Readings

Experiment 9		$T_{ambient}$ (°C)	Pipe 1 T_{avg} (°C)	Pipe 1 U_{avg} ($W/m^2.K$)	Pipe 2 T_{avg} (°C)	Pipe 2 U_{avg} ($W/m^2.K$)	Pipe 1 T_{top} (°C)	Pipe 1 U_{top} ($W/m^2.K$)	Pipe 2 T_{top} (°C)	Pipe 2 U_{top} ($W/m^2.K$)	Pipe 1 T_{bottom} (°C)	Pipe 1 U_{bottom} ($W/m^2.K$	Pipe 2 T_{bottom} (°C)	Pipe 2 U_{bottom} ($W/m^2.K$)
0 m/s	Run 1	-19.54	85.13	5.051	85.31	5.042	84.87	5.064	84.42	5.086	85.40	5.038	86.20	5.000
	Run 2	-19.31	86.48	4.998	85.85	5.027	86.17	5.012	84.95	5.071	86.78	4.983	86.76	4.984
	Run 3	-19.22	88.01	4.931	87.18	4.969	87.69	4.945	86.26	5.012	88.32	4.916	88.10	4.926
	Average	-19.36	86.54	4.993	86.12	5.013	86.24	5.007	85.21	5.056	86.84	4.979	87.02	4.970
5 m/s	Run 1	-19.53	73.17	5.703	72.47	5.746	72.84	5.724	71.60	5.802	73.51	5.682	73.35	5.692
	Run 2	-19.22	73.14	5.724	72.98	5.735	72.80	5.746	72.09	5.790	73.49	5.703	73.86	5.680
	Run 3	-19.09	72.41	5.778	72.30	5.785	72.08	5.799	71.43	5.841	72.74	5.758	73.17	5.731
	Average	-19.28	72.91	5.735	72.58	5.755	72.57	5.756	71.70	5.811	73.24	5.714	73.46	5.701
10 m/s	Run 1	-18.91	71.87	5.824	71.79	5.829	71.52	5.846	70.91	5.886	72.21	5.803	72.67	5.773
	Run 2	-18.76	72.07	5.821	72.15	5.816	71.74	5.842	71.43	5.862	72.40	5.800	72.86	5.770
	Run 3	-18.92	71.34	5.858	71.58	5.842	71.01	5.879	70.71	5.899	71.66	5.837	72.46	5.786
	Average	-18.86	71.76	5.834	71.84	5.829	71.42	5.856	71.02	5.882	72.09	5.813	72.66	5.776
15 m/s	Run 1	-17.54	72.39	5.879	72.47	5.874	72.08	5.900	71.58	5.932	72.70	5.859	73.36	5.816
	Run 2	-18.05	70.82	5.949	71.04	5.934	70.55	5.967	70.17	5.993	71.08	5.932	71.91	5.877
	Run 3	-17.83	71.20	5.938	71.49	5.919	70.88	5.960	70.64	5.976	71.52	5.917	72.33	5.864
	Average	-17.81	71.47	5.922	71.67	5.909	71.17	5.942	70.80	5.967	71.77	5.902	72.53	5.852

5.1.2.5 Experiment No: 10

Table 5-6 shows the heat transfer coefficient and temperature readings for a combination of 1 x 50 mm and 1 x 25 mm insulated pipe configuration. Plots of overall heat transfer coefficient versus wind velocity and surfaces are shown in Figure 5-13 and Figure 5-14.

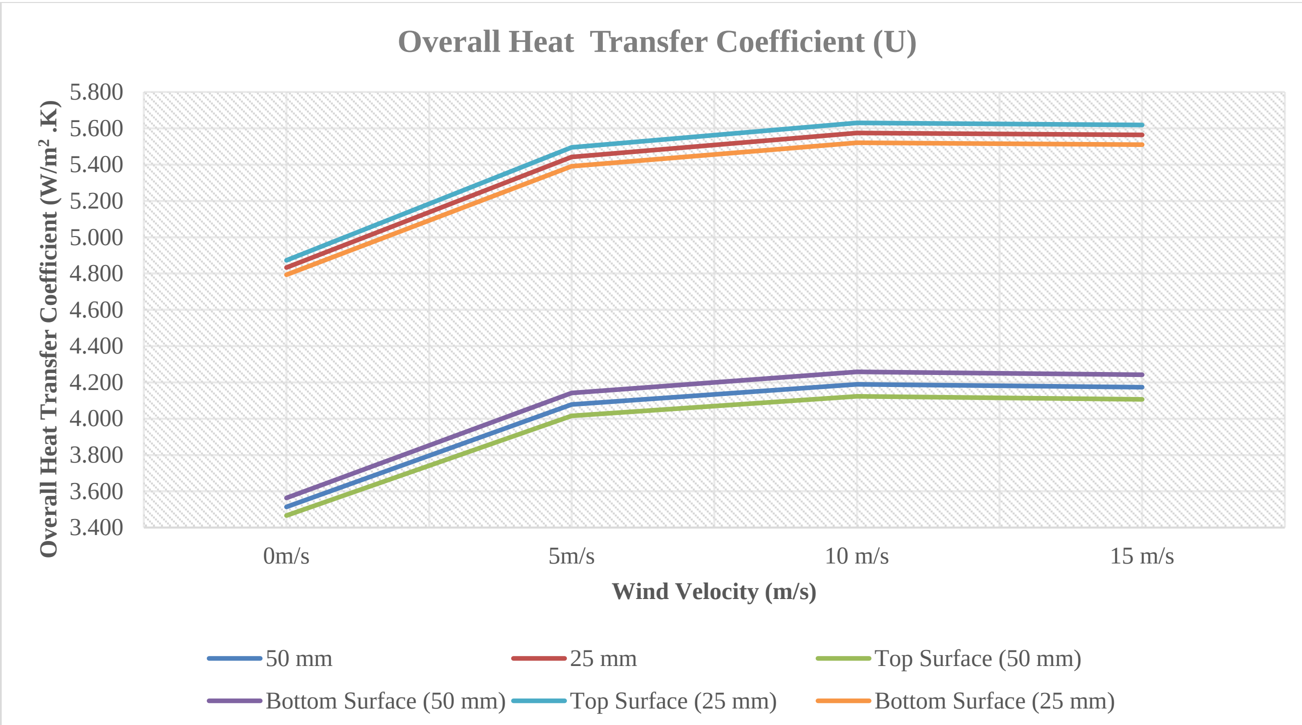

Figure 5-13 Experiment 10-Overall Heat Transfer coefficient for the whole pipe configuration

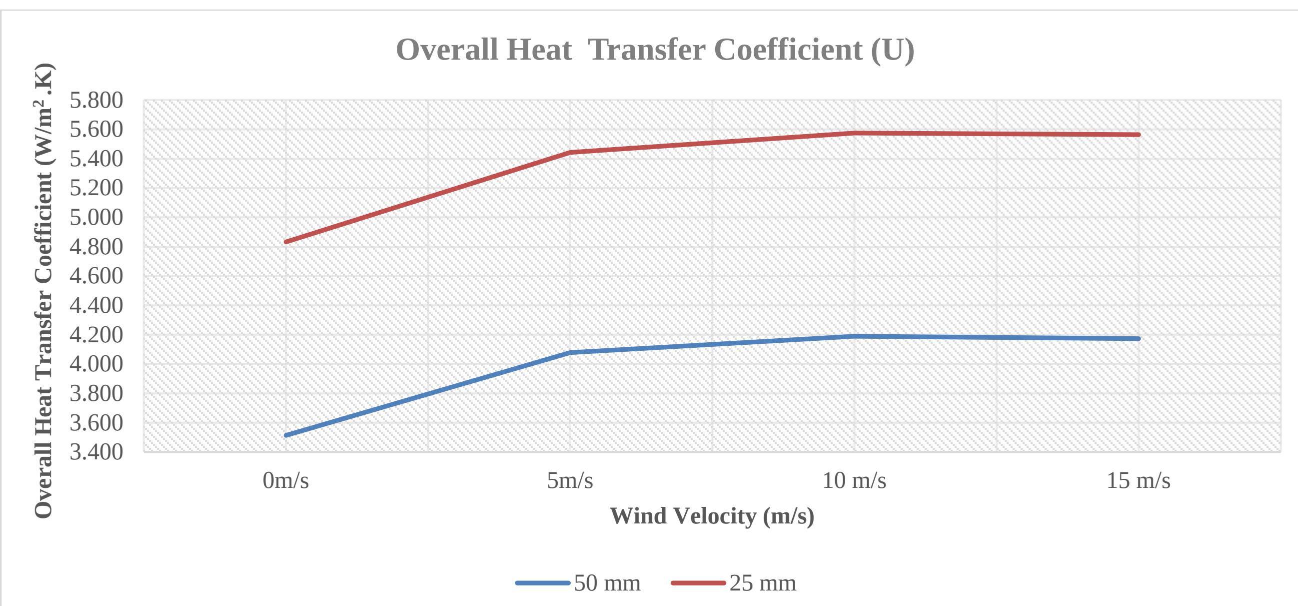

Figure 5-14 Experiment 10- Overall Heat Transfer coefficient for Pipe 1 & Pipe 2

Table 5-6 Experiment 10-Heat Transfer Coefficient and Temperature Readings

Experiment 10		$T_{ambient}$ (°C)	Pipe 1 T_{avg} (°C)	Pipe 1 U_{avg} ($W/m^2.K$)	Pipe 2 T_{avg} (°C)	Pipe 2 ($W/m^2.K$)	Pipe 1 T_{top} (°C)	Pipe 1 U_{top} ($W/m^2.K$)	Pipe 2 T_{top} (°C)	Pipe 2 ($W/m^2.K$)	Pipe 1 T_{bottom} (°C)	Pipe 1 U_{bottom} ($W/m^2.K$)	Pipe 2 T_{bottom} (°C)	Pipe 2 U_{bottom} ($W/m^2.K$)
0 m/s	Run 1	-19.44	48.87	3.571	89.92	4.886	49.84	3.521	89.04	4.926	47.90	3.622	90.80	4.847
	Run 2	-19.25	50.84	3.480	92.43	4.784	51.80	3.433	91.49	4.825	49.88	3.529	93.37	4.744
	Run 3	-19.21	50.64	3.492	91.44	4.829	51.57	3.446	90.51	4.870	49.71	3.539	92.38	4.788
	Average	-19.30	50.12	3.514	91.27	4.833	51.07	3.466	90.35	4.873	49.16	3.563	92.18	4.793
5 m/s	Run 1	-19.33	40.34	4.088	79.01	5.434	41.28	4.025	78.11	5.484	39.40	4.154	79.90	5.385
	Run 2	-19.32	40.32	4.090	78.64	5.455	41.24	4.028	77.68	5.509	39.40	4.155	79.60	5.402
	Run 3	-18.92	41.23	4.055	79.30	5.440	42.14	3.995	78.34	5.493	40.32	4.118	80.26	5.387
	Average	-19.19	40.63	4.078	78.98	5.443	41.55	4.016	78.05	5.495	39.70	4.142	79.92	5.391
10 m/s	Run 1	-18.84	39.66	4.170	77.83	5.528	40.62	4.102	76.88	5.582	38.70	4.240	78.78	5.474
	Run 2	-19.29	38.39	4.229	75.56	5.633	39.31	4.163	74.63	5.689	37.47	4.298	76.49	5.579
	Run 3	-18.72	39.78	4.170	77.31	5.564	40.71	4.105	76.37	5.619	38.85	4.237	78.26	5.510
	Average	-18.95	39.28	4.189	76.90	5.575	40.21	4.123	75.96	5.630	38.34	4.258	77.84	5.520
15 m/s	Run 1	-18.14	41.15	4.114	79.03	5.499	42.12	4.048	78.07	5.554	40.18	4.183	79.99	5.445
	Run 2	-18.23	39.35	4.237	76.39	5.647	40.28	4.169	75.47	5.703	38.41	4.307	77.32	5.592
	Run 3	-18.58	39.92	4.170	77.73	5.548	40.86	4.104	76.79	5.603	38.97	4.238	78.66	5.495
	Average	-18.32	40.14	4.173	77.72	5.564	41.09	4.106	76.77	5.619	39.19	4.242	78.66	5.510

Since, the plots from experiment 7, experiment 9 and experiment 10 present similar trend despite the fact that the pipe configuration were dissimilar in these experiments, it clearly demonstrate that the effect of order of pipes on the overall heat transfer subjected to cross flow wind is not substantial. The overall heat transfer coefficient for both diameter 25 mm and diameter 50 mm insulated pipes irrespective of the order of their placement in the configuration are similar as seen in Figure 5-10 and Figure 5-14. The value of overall heat transfer coefficient varies in the range 4.8 – 5.7 W/m^2. K for diameter 25 mm pipe and from 3.5 – 4.2 W/m^2. K in the case of diameter 50 mm pipe (Table 5-4 and Table 5-6). The difference is because of the lower surface area in the case of diameter 25 mm pipe compared to diameter 50 mm pipe even though same amount of power was supplied for both the pipes regardless of the experiment. From equation (2.16), overall heat transfer coefficient is inversely proportional to the surface area of the pipe. The overall heat transfer coefficient for diameter 25 mm pipe is 33-37 % higher than the diameter 50 mm pipe throughout experiment 7 and experiment 10. In the case of experiment 9, two diameter 25 mm pipes were used instead of a combination of one diameter 25 mm and one diameter 50 mm pipes, the overall heat transfer coefficient shows slight increase of 3.5 – 4.0 % compared to the combination tests. This is within acceptable limits and can be attributed to the slight difference in the measured ambient condition on the day of the experiment (Welty et al., 2008).

Additionally, plots of overall heat transfer coefficient for the top and bottom surface of the insulated pipes are also presented in Figure 5-9, Figure 5-11 and Figure 5-13 along with the average overall heat transfer coefficient values. As anticipated from earlier tests and theory, the heat transfer coefficient of the bottom surface is higher than the top surface because of the convection inside the pipe. The warm air inside the pipe rises up resulting in the heating up of the top surface. This explains the rise in the temperature of the top pipe surface compared to the bottom surface throughout the experimental readings found in Table 5-1, Table 5-5 and Table 5-6. The temperature readings were within 3 % range with increase in wind speed as the circulation of heat helps in balancing the temperature at these surfaces. Also, it can be seen that until 5 m/s wind speed there is a steady increase in the overall heat transfer coefficient. But, its value doesn't show significant change when the wind speed is increased to 10 m/s or 15 m/s as the pipe surface temperature has peaked and reached a point of equilibrium. It can be clearly seen that the rate of heat transfer from insulated pipe is lower. The experiments produced results on expected lines and shows negligible deviation (Bejan and Kraus, 2003).

5.1.3 Case 3: Heat Transfer co-efficient for deck element (flat plate)

5.1.3.1 Experiment 12

Plot of overall heat transfer coefficient versus wind velocity for deck element at different ambient temperature is shown in Figure 5-15. Overall heat transfer coefficient, surface temperature readings and power consumption for deck element at different ambient conditions are tabulated in Table 5-7, Table 5-8, Table 5-9 and Table 5-10.

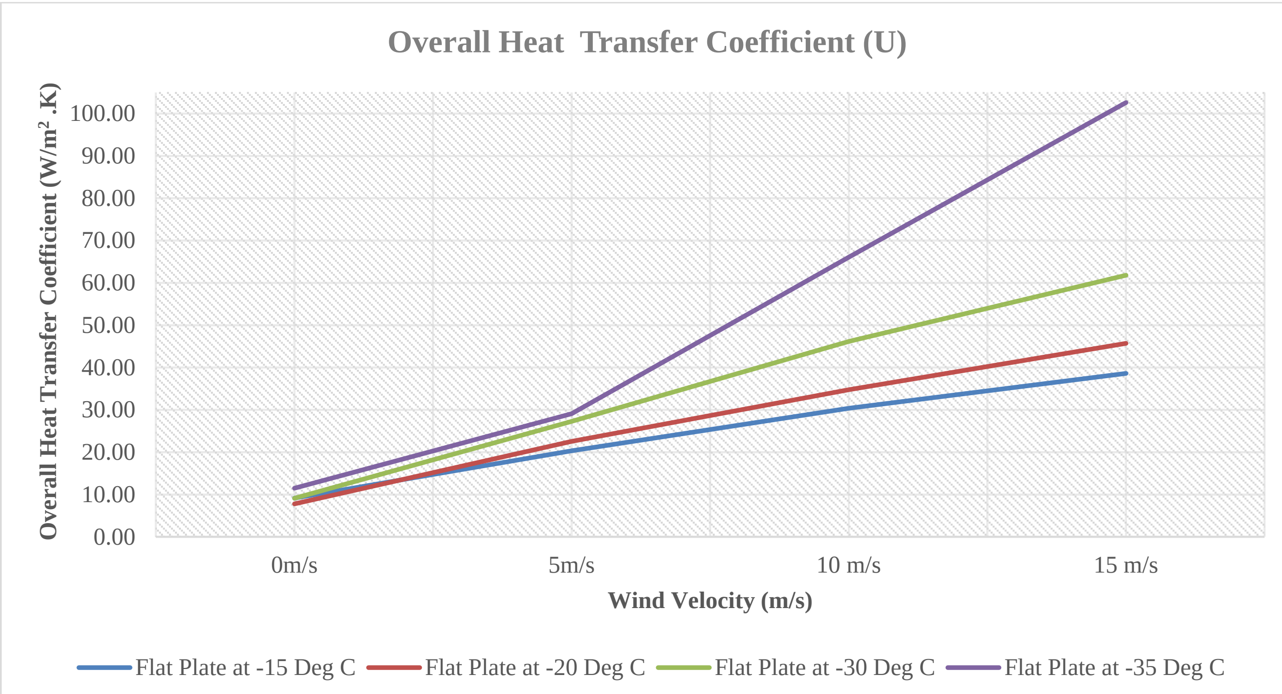

Figure 5-15 Experiment 12-Overall Heat Transfer coefficient for the deck element

The plot from experiment 12 in Figure 5-15 depicts that the overall heat transfer coefficient for the deck element with epoxy coating is very high with an average value of 10 W/m^2. K for 0 m/s wind velocity at different temperatures and increases considerably as the wind velocity and ambient temperature is increased. The overall transfer coefficient of the deck element at -15° C, -20° C and -30° C show similar trend upon increase in the wind velocity except for -35° C condition which can be because of erroneous ambient temperature reading at the time of the experiment. See Table 5-10 for the readings and it appears that the ambient temperature didn't reduce even though the wind velocity was increased to 15 m/s which is not expected. So, the spike in the overall heat transfer coefficient value for the deck element at -35° C and 15 m/s is not justified and can be attributed to this error in the temperature reading. Otherwise, the deck element shows clear trend and could have yielded better results if they were allowed to stabilize for more time. In addition, it can be seen from Table 5-7, Table 5-8, Table 5-9 and Table 5-10 that the power consumption increases with the reduction in ambient temperature and increase in wind velocity as the deck element tries to use maximum capacity to heat up the deck element (Baehr and Stephan, 2011).

Table 5-7 Experiment 12-Heat Transfer Coefficient and Temperature Readings at -15°C

Experiment 12		**DECK ELEMENT**						
Wind Speed (m/s)	***Set Temp (°C)***	***Ambient Temp (°C)***	***Air Temp (°C)***	***Surface Temp (°C)***	***Current (A)***	***Voltage (V)***	***Power (W)***	***U (W/m^2. K)***
0	-15	-13.81	-11.30	19.97	4.13	224.20	936	9.14
5	-15	-13.86	-12.80	3.70	4.77	224.75	1076	20.32
10	-15	-13.48	-12.30	-1.23	5.00	223.30	1123	30.37
15	-15	-13.01	-11.77	-3.17	5.10	223.83	1151	38.63

Table 5-8 Experiment 12-Heat Transfer Coefficient and Temperature Readings at -20°C

Experiment 12		**DECK ELEMENT**						
Wind Speed (m/s)	***Set Temp (°C)***	***Ambient Temp (°C)***	***Air Temp (°C)***	***Surface Temp (°C)***	***Current (A)***	***Voltage (V)***	***Power (W)***	***U (W/m^2. K)***
0	-20	-18.90	-17.20	21.27	4.17	225.63	947	7.80
5	-20	-19.18	-18.03	-2.03	5.20	223.40	1175	22.56
10	-20	-18.98	-18.03	-7.20	5.47	224.90	1232	34.76
15	-20	-18.86	-17.63	-9.67	5.60	225.23	1263	45.71

Table 5-9 Experiment 12-Heat Transfer Coefficient and Temperature Readings at -30°C

Experiment 12		**DECK ELEMENT**						
Wind Speed (m/s)	***Set Temp (°C)***	***Ambient Temp (°C)***	***Air Temp (°C)***	***Surface Temp (°C)***	***Current (A)***	***Voltage (V)***	***Power (W)***	***U (W/m^2. K)***
0	-30	-29.24	-27.00	9.50	4.67	227.00	1071	9.11
5	-30	-27.82	-26.57	-12.90	5.40	226.43	1231	27.30
10	-30	-27.86	-26.63	-18.40	5.80	226.00	1320	46.17
15	-30	-29.63	-27.20	-22.37	5.97	225.90	1362	61.81

Table 5-10 Experiment 12-Heat Transfer Coefficient and Temperature Readings at -35°C

Experiment 12		**DECK ELEMENT**						
Wind Speed (m/s)	***Set Temp (°C)***	***Ambient Temp (°C)***	***Air Temp (°C)***	***Surface Temp (°C)***	***Current (A)***	***Voltage (V)***	***Power (W)***	***U (W/m^2. K)***
0	-35	-27.77	-24.90	4.47	4.90	226.97	1123	11.49
5	-35	-27.33	-25.75	-12.60	5.70	225.65	1291	29.08
10	-35	-30.55	-28.35	-23.55	6.15	225.85	1399	66.09
15	-35	-26.90	-23.90	-22.35	6.20	225.80	1409	102.59

5.2 Results from Theoretical Method

All the tables generated in theoretical analysis were obtained using python code (Kvamme, 2014) in the case of pipe configuration and for the generation of time to freeze tables. Analysis using theoretical method for deck element was done using Microsoft excel. The python code used for the calculation is presented in Appendix A

5.2.1 Case 1: Heat Transfer co-efficient for uninsulated pipe (forced flow scenario)

5.2.1.1 Uninsulated pipe with OD=50 mm (insulation thickness t=0mm)

Table 5-11 shows the overall heat transfer coefficient for 50 mm uninsulated pipe with varying wind velocity using different heat transfer correlations. Plot of overall heat transfer coefficient of 50 mm uninsulated pipe for each correlation versus different wind velocity is shown in Figure 5-16.

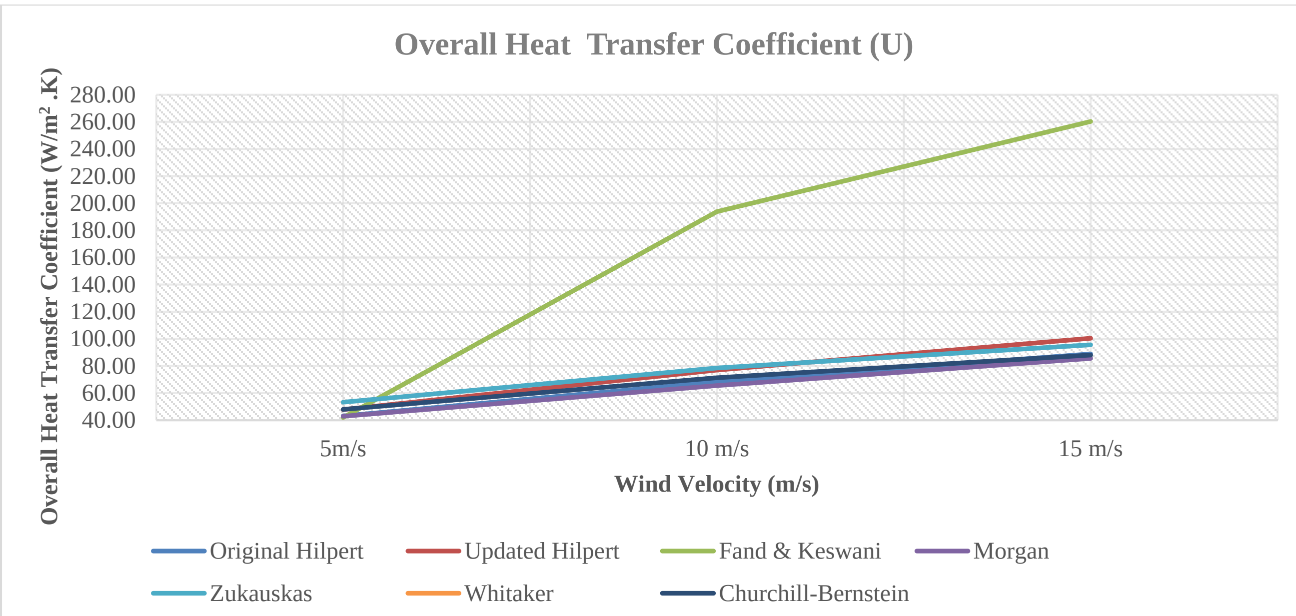

Figure 5-16 Overall Heat Transfer coefficient for 50mm uninsulated pipe using different correlations

The plot from theoretical calculation using similar ambient conditions as were present during experiment is shown in Figure 5-2. All the correlations except for (Fand and Keswani, 1973) show very good accuracy even at higher Reynolds number. (Fand and Keswani, 1973) shows good correlation until Reynolds number of 50,000 and afterwards the values are extremely high. Theoretical calculation shows that that the overall heat transfer coefficient for the uninsulated pipe is very high and increases linearly as the wind velocity is increased keeping the ambient conditions constant. The purpose of the experiment was to show the effect of cross flow wind on the heat transfer coefficient and it is observed that the overall heat transfer coefficient value increases by 100 % in going from 5 m/s to 15 m/s (see Table 5-11). These numbers are important as they show the rate of heat transfer from pipes which are uninsulated and an indicative of the energy that is exchanged when hot fluids are circulated.

Table 5-11 Overall heat transfer coefficient for 50mm uninsulated pipe using different correlations

Pipe OD (mm)	***Wind Velocity u_m(m/s)***	***Internal Temp t (°C)***	***Ambient Temp t (°C)***	***Reynolds Number***	***Nusselt Number (NuD)***	***Convective Heat Transfer coefficient, h (W/m2 .K)***	***Overall Heat Transfer coefficient, U1 (W/m2 .K)***
Original Hilpert Correlation							
50	5	-15	-20	28968.86	89.22	39.79	43.19
	10	-15	-20	55359.80	140.97	62.87	68.19
	15	-15	-20	77031.83	183.91	82.03	88.91
Updated Hilpert Constants							
50	5	-15	-20	28968.86	98.96	44.14	47.90
	10	-15	-20	55359.80	159.25	71.03	77.01
	15	-15	-20	77031.83	207.77	92.66	100.40
Fand & Keswani Constants							
50	5	-15	-20	28968.86	86.62	38.63	41.93
	10	-15	-20	55359.80	402.28	179.42	193.82
	15	-15	-20	77031.83	541.22	241.38	260.21
Morgan Constants							
50	5	-15	-20	28968.86	88.53	39.49	42.86
	10	-15	-20	55359.80	135.35	60.37	65.48
	15	-15	-20	77031.83	177.12	78.99	85.63
Zukauskas Correlation							
50	5	-15	-20	28968.86	109.98	49.05	53.23
	10	-15	-20	55359.80	162.21	72.35	78.44
	15	-15	-20	77031.83	197.77	88.21	95.59
Whitaker Correlation							
50	5	-15	-20	28968.86	119.81	53.43	57.97
	10	-15	-20	55359.80	173.16	77.23	83.72
	15	-15	-20	77031.83	209.18	93.30	101.08
Churchill-Bernstein Correlation							
50	5	-15	-20	28968.86	99.11	44.20	47.97
	10	-15	-20	55359.80	147.39	65.74	71.29
	15	-15	-20	77031.83	182.21	81.27	88.09

5.2.1.2 Uninsulated pipe with OD=25 mm (insulation thickness t=0mm)

Table 5-12 Overall heat transfer coefficient for 25 mm uninsulated pipe using different correlations shows the overall heat transfer coefficient for 25 mm uninsulated pipe with varying wind velocity using different heat transfer correlations. Plot of overall heat transfer coefficient of 25 mm uninsulated pipe for each correlation versus different wind velocity is shown in Figure 5-17.

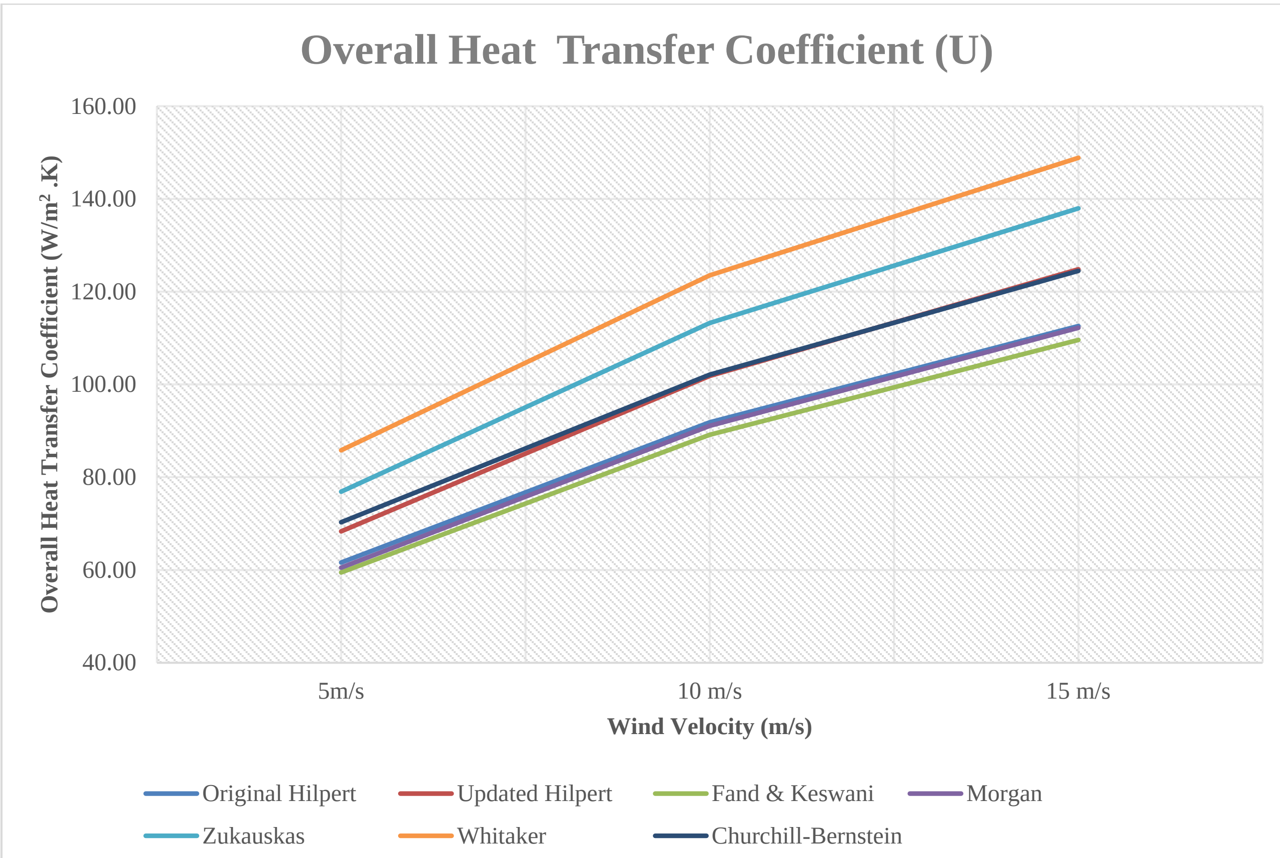

Figure 5-17 Overall Heat Transfer coefficient for 25 mm uninsulated pipe using different correlations

The actual experiment for uninsulated pipe was conducted only for diameter 50 mm pipe. The plot from theoretical calculation for diameter 25 mm uninsulated pipe is shown in Figure 5-16. It is observed that all the correlations including (Fand and Keswani, 1973) are comparable even though there are some cases where the difference between the values obtained from the correlation have a difference in the range of 30-35 % . But, it has to be noted the Reynolds number is significantly lower for diameter 25 mm pipe as it is directly proportional to outside diameter. Theoretical calculation shows that that the overall heat transfer coefficient for the uninsulated pipe increases linearly as the wind velocity is increased keeping the ambient conditions constant. The values are higher than the ones obtained for diameter 50 mm uninsulated pipe. The overall heat transfer coefficient value increases by 75-100 % in going from 5 m/s to 15 m/s as can be observed from Table 5-12. (Baehr and Stephan, 2011)

Table 5-12 Overall heat transfer coefficient for 25 mm uninsulated pipe using different correlations

Pipe OD (mm)	Wind Velocity u_m(m/s)	Internal Temp t (°C)	Ambient Temp t (°C)	Reynolds Number	Nusselt Number (NuD)	Convective Heat Transfer coefficient, h (W/m2 .K)	Overall Heat Transfer coefficient, U1 (W/m2 .K)
Original Hilpert Correlation							
25	5	-15	-20	14484.43	58.13	51.86	61.62
	10	-15	-20	27679.90	86.75	77.38	91.86
	15	-15	-20	38515.92	106.39	94.90	112.60
Updated Hilpert Constants							
25	5	-15	-20	14484.43	64.48	57.52	68.33
	10	-15	-20	27679.90	96.22	85.83	101.86
	15	-15	-20	38515.92	118.01	105.27	124.84
Fand & Keswani Constants							
25	5	-15	-20	14484.43	56.09	50.03	59.45
	10	-15	-20	27679.90	84.18	75.09	89.15
	15	-15	-20	38515.92	103.55	92.37	109.60
Morgan Constants							
25	5	-15	-20	14484.43	57.09	50.92	60.51
	10	-15	-20	27679.90	86.02	76.73	91.09
	15	-15	-20	38515.92	106.03	94.58	112.21
Zukauskas Correlation							
25	5	-15	-20	14484.43	72.56	64.72	76.87
	10	-15	-20	27679.90	107.02	95.46	113.26
	15	-15	-20	38515.92	130.48	116.39	137.98
Whitaker Correlation							
25	5	-15	-20	14484.43	81.03	72.28	85.82
	10	-15	-20	27679.90	116.76	104.15	123.52
	15	-15	-20	38515.92	140.83	125.62	148.87
Churchill-Bernstein Correlation							
25	5	-15	-20	14484.43	66.33	59.16	70.28
	10	-15	-20	27679.90	96.47	86.05	102.12
	15	-15	-20	38515.92	117.67	104.97	124.49

5.2.2 Case 2: Heat Transfer co-efficient for insulated pipe (forced flow scenario)

5.2.2.1 Insulated pipe with OD=50 mm and insulation thickness t=10mm insulation

Table 5-13 shows the overall heat transfer coefficient for 50 mm insulated pipe with varying wind velocity using different heat transfer correlations. Plot of overall heat transfer coefficient of 50 mm insulated pipe for each correlation versus different wind velocity is shown in Figure 5-18.

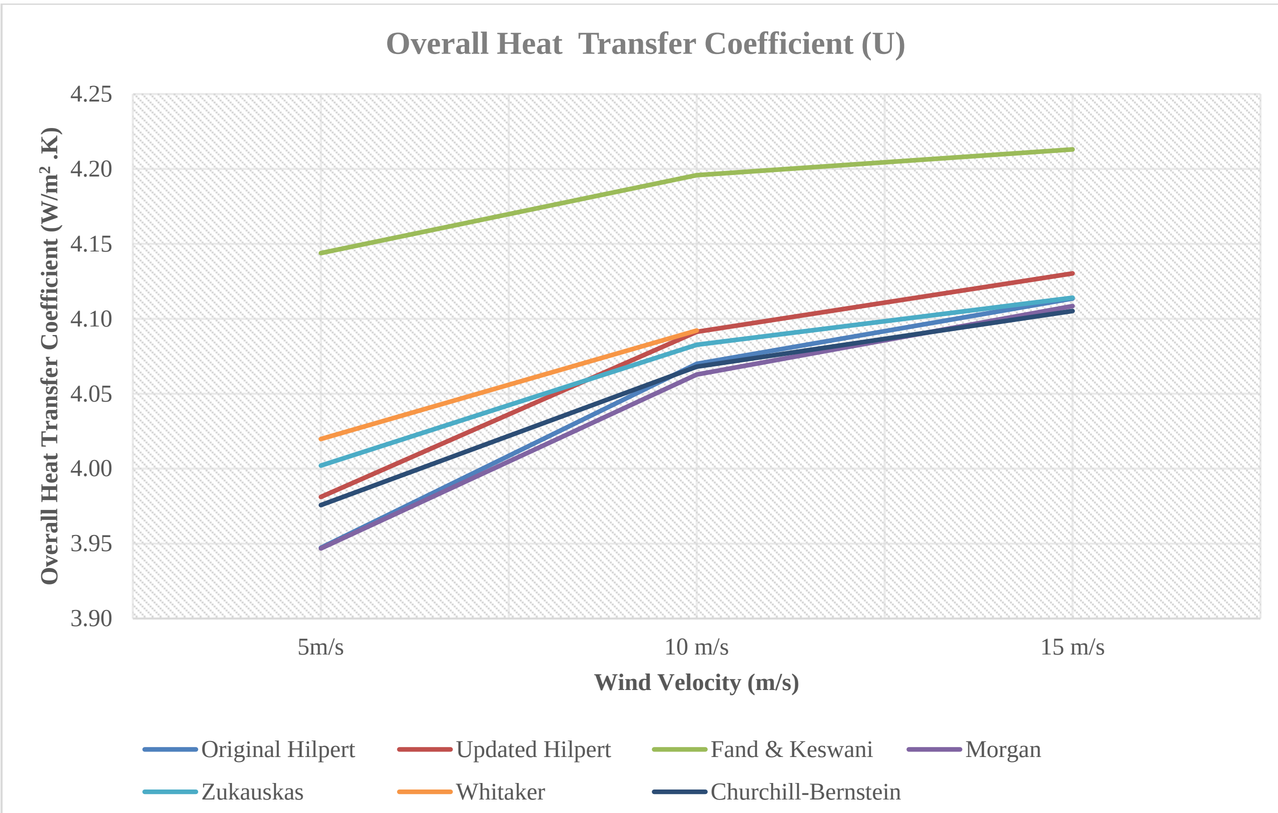

Figure 5-18 Overall Heat Transfer coefficient for 50 mm insulated pipe using different correlations

The theoretical calculation for OD 50 mm insulated pipe are presented in Figure 5-3 and it shows the overall heat transfer coefficients for the insulated pipes are in the range of 3.95 - 4.10 W/m^2. K. The values of overall heat transfer coefficient calculated by (Fand and Keswani, 1973) are in the range 4.15 – 4.20 W/m^2. K whereas rest of the correlations show proximity. The average value of overall heat transfer coefficient increases as the wind velocity is raised while the ambient conditions remains the same. But, the change is not substantial from 10 m/s to 15 m/s as the convective heat transfer has reached almost the threshold value and further increase in the wind velocity doesn't help in increasing the pipe's convective heat transfer value. The main objective of the experiment was to study the effect of cross flow wind on the heat transfer coefficient for the pipe and it is observed that the change in overall heat transfer coefficient is very small with just 3.0 - 4.0 % increase in the value in going from 0 m/s wind velocity to 15 m/s. The overall heat transfer coefficient for a particular case relate very well with 1-3 % change throughout the experiment as can be seen in Table 5-13.

Table 5-13 Overall heat transfer coefficient for 50mm insulated pipe using different correlations

Pipe OD (mm)	***Wind Velocity u_m(m/s)***	***Internal Temp t (°C)***	***Ambient Temp t (°C)***	***Reynolds Number***	***Nusselt Number (NuD)***	***Convective Heat Transfer coefficient, h (W/m^2 .K)***	***Overall Heat Transfer coefficient, U1 (W/m^2 .K)***
Original Hilpert Correlation							
50	5	-15	-20	40556.41	109.73	34.96	3.95
	10	-15	-20	77503.72	184.82	58.88	4.07
	15	-15	-20	107844.57	241.13	76.82	4.11
Updated Hilpert Constants							
50	5	-15	-20	40556.41	123.96	39.49	3.98
	10	-15	-20	77503.72	208.79	66.51	4.09
	15	-15	-20	107844.57	272.40	86.78	4.13
Fand & Keswani Constants							
50	5	-15	-20	40556.41	304.21	96.91	4.14
	10	-15	-20	77503.72	544.19	173.36	4.20
	15	-15	-20	107844.57	732.14	233.24	4.21
Morgan Constants							
50	5	-15	-20	40556.41	109.55	34.90	3.95
	10	-15	-20	77503.72	178.00	56.71	4.06
	15	-15	-20	107844.57	232.92	74.20	4.11
Zukauskas Correlation							
50	5	-15	-20	40556.41	134.59	42.88	4.00
	10	-15	-20	77503.72	198.50	63.24	4.08
	15	-15	-20	107844.57	242.02	77.10	4.11
Whitaker Correlation							
50	5	-15	-20	40556.41	145.03	46.20	4.02
	10	-15	-20	77503.72	209.92	66.87	4.09
	15	-15	-20	107844.57	N/A	N/A	N/A
Churchill-Bernstein Correlation							
50	5	-15	-20	40556.41	121.45	38.69	3.98
	10	-15	-20	77503.72	182.94	58.28	4.07
	15	-15	-20	107844.57	227.85	72.59	4.11

5.2.2.2 Insulated pipe with OD=25 mm and insulation thickness t=10mm insulation

Table 5-14 Overall heat transfer coefficient for 25 mm insulated pipe using different correlations Plot of overall heat transfer coefficient of 25 mm insulated pipe for each correlation versus different wind velocity is shown in Figure 5-19.

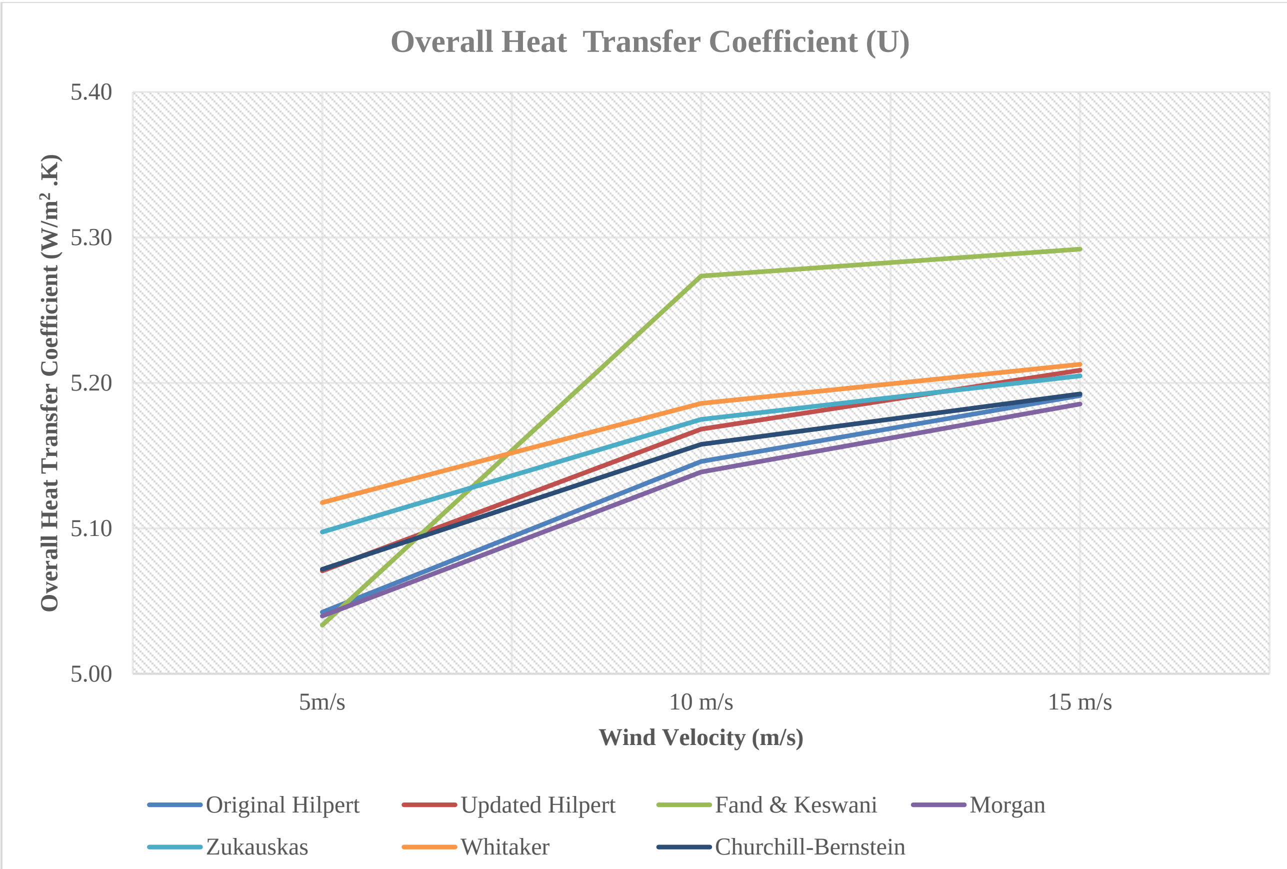

Figure 5-19 Overall Heat Transfer coefficient for 25 mm insulated pipe using different correlations

The theoretical calculation for diameter 25 mm insulated pipe was performed and the output is shown in Figure 5-19 and it demonstrates the overall heat transfer coefficients for the insulated diameter 25 mm pipe is in the range of 5.05 – 5.20 $W/m^2.K$ except for (Fand and Keswani, 1973) values which is within 2.0 % of other correlation. But, doesn't show similar uniform trend compared to other correlations and will not be considered for further study. The average value of overall heat transfer coefficient increases as the wind velocity is raised while the ambient conditions remains the same. But, the change is not substantial from 10 m/s to 15 m/s. But, similar to diameter 50 mm insulated pipe as the convective heat transfer has reached peak value. The overall heat transfer coefficient values are higher because of the lower surface area in the case of diameter 25 mm pipe. From equation (2.16), it can be seen that the overall heat transfer coefficient is inversely proportional to the surface area of the pipe. The overall heat transfer coefficient for diameter 25 mm pipe varies 1.0- 1.5 % for different correlations at lower velocity and almost converges at higher velocities of 15 m/s. as evident from the plot and Table 5-14 (Kutz, 2015)

Table 5-14 Overall heat transfer coefficient for 25 mm insulated pipe using different correlations

Pipe OD (mm)	Wind Velocity u_m(m/s)	Internal Temp t (°C)	Ambient Temp t (°C)	Reynolds Number	Nusselt Number (NuD)	Convective Heat Transfer coefficient, h (W/m2 .K)	Overall Heat Transfer coefficient, U1 (W/m2 .K)
Original Hilpert Correlation							
25	5	-15	-20	26071.98	83.60	41.43	5.04
	10	-15	-20	49823.82	129.50	64.18	5.15
	15	-15	-20	69328.65	168.96	83.73	5.19
Updated Hilpert Constants							
25	5	-15	-20	26071.98	92.73	45.95	5.07
	10	-15	-20	49823.82	146.30	72.50	5.17
	15	-15	-20	69328.65	190.87	94.59	5.21
Fand & Keswani Constants							
25	5	-15	-20	26071.98	81.08	40.18	5.03
	10	-15	-20	49823.82	365.97	181.36	5.27
	15	-15	-20	69328.65	492.36	243.99	5.29
Morgan Constants							
25	5	-15	-20	26071.98	82.82	41.04	5.04
	10	-15	-20	49823.82	124.79	61.84	5.14
	15	-15	-20	69328.65	162.56	80.56	5.19
Zukauskas Correlation							
25	5	-15	-20	26071.98	103.24	51.16	5.10
	10	-15	-20	49823.82	152.27	75.46	5.17
	15	-15	-20	69328.65	185.66	92.00	5.20
Whitaker Correlation							
25	5	-15	-20	26071.98	112.87	55.93	5.12
	10	-15	-20	49823.82	163.06	80.81	5.19
	15	-15	-20	69328.65	196.93	97.59	5.21
Churchill-Bernstein Correlation							
25	5	-15	-20	26071.98	93.11	46.14	5.07
	10	-15	-20	49823.82	137.95	68.36	5.16
	15	-15	-20	69328.65	170.17	84.33	5.19

5.2.3 Case 3: Heat Transfer co-efficient for deck element / flat plate (forced flow scenario)

The theoretical heat transfer coefficient was calculated using Microsoft excel program. Plot of overall heat transfer coefficient for deck element versus wind velocity at different ambient temperatures is shown in Figure 5-20. Overall heat transfer coefficient, surface temperature readings and power consumption for deck element at different ambient conditions are tabulated in Table 5-15, Table5-16, Table 5-17 and Table 5-18.

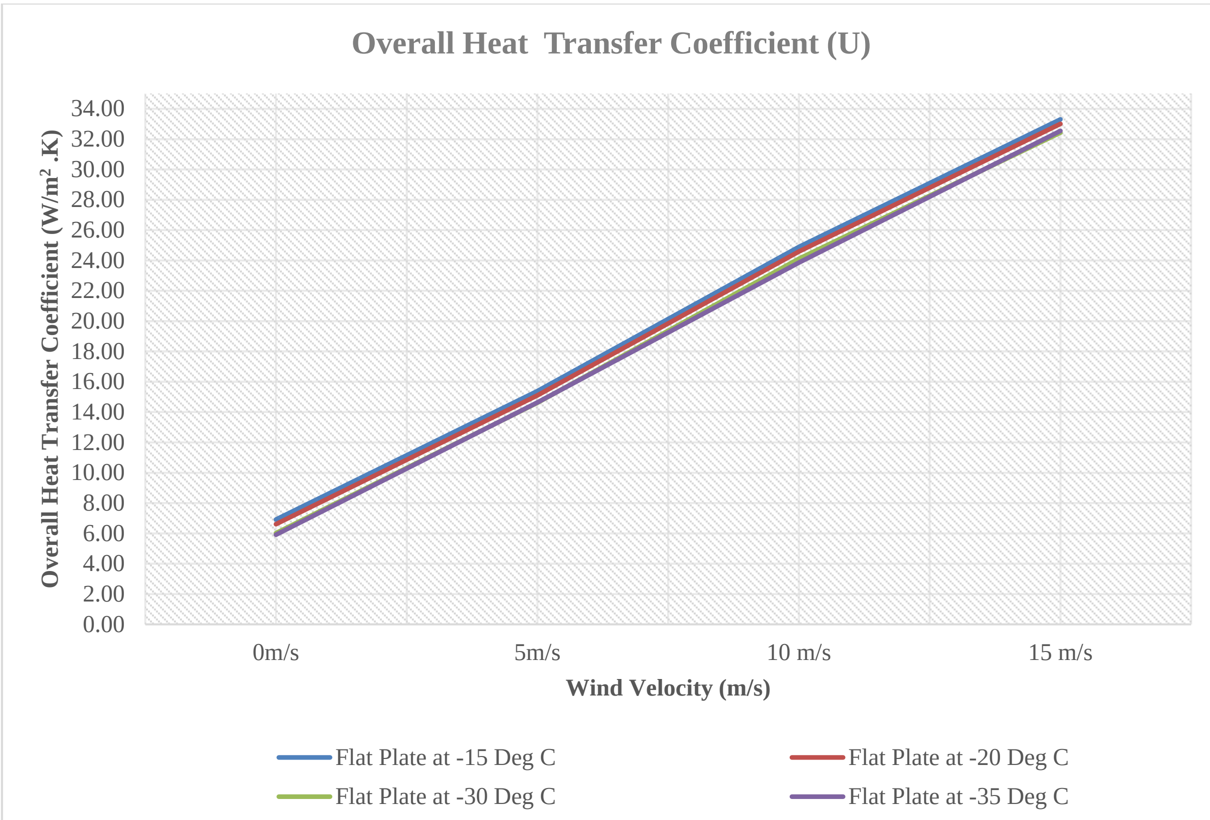

Figure 5-20 Overall Heat Transfer coefficient for the deck element

The average overall heat transfer coefficient for the deck element with epoxy coating is 6.5 W/m^2. K for 0 m/s wind velocity at different temperatures and increases considerably as the wind velocity and ambient temperature is increased. The overall transfer coefficient of the deck element calculated at -15 °C, -20 °C , -30 °C and -35° C show similar trend upon increase in the wind velocity. The increase in the overall heat transfer coefficient from 0 m/s to 15 m/s is linear as seen in the Figure 5-20. In addition, it can be seen from Table 5-15, Table 5-16, Table 5-17 and Table 5-18 that the power consumption increases with the reduction in ambient temperature and increase in wind velocity as the deck element tries to use maximum capacity to heat up the deck element. It can be observed that the effect of the ambient temperature is not very significant on the heat transfer coefficient because of the delta temperature between the surface and the ambient condition being constant for similar wind speeds which governs the overall heat transfer coefficient value.

Table 5-15 Heat Transfer Coefficient and Theoretical Power at -15°C

FLAT PLATE								
Wind Speed (m/s)	***Set Temp (°C)***	***Ambient Temp (°C)***	***Surface Temp (°C)***	***Reynolds Number***	***Heat laminar flow (W)***	***Heat turbulent flow (W)***	***Power (W)***	***Overall Heat Transfer Coefficient U (W/m²K)***
0	-15	-13.81	19.97	0.00	0.00	0.00	521.66	6.05
5	-15	-13.86	3.70	608477.20	160.53	63.30	671.33	14.98
10	-15	-13.48	-1.23	1217915.66	81.01	296.25	755.06	24.16
15	-15	-13.01	-3.17	1694700.32	55.17	410.69	812.96	32.36

Table 5-16 Heat Transfer Coefficient and Theoretical Power at -20°C

FLAT PLATE								
Wind Speed (m/s)	***Set Temp (°C)***	***Ambient Temp (°C)***	***Surface Temp (°C)***	***Reynolds Number***	***Heat laminar flow W)***	***Heat turbulent flow (W)***	***Power (W)***	***Overall Heat Transfer Coefficient U (W/m²K)***
0	-20	-18.90	21.27	0.00	0.00	0.00	715.80	6.98
5	-20	-19.18	-2.03	637314.98	156.79	61.82	641.98	14.67
10	-20	-18.98	-7.20	1217915.66	77.90	284.89	717.20	23.86
15	-20	-18.86	-9.67	1694700.32	51.52	383.56	751.84	32.06

Table 5-17 Heat Transfer Coefficient and Theoretical Power at -30°C

FLAT PLATE								
Wind Speed (m/s)	***Set Temp (°C)***	***Ambient Temp (°C)***	***Surface Temp (°C)***	***Reynolds Number***	***Heat laminar flow (W)***	***Heat turbulent flow (W)***	***Power (W)***	***Overall Heat Transfer Coefficient U (W/m²K)***
0	-30	-29.24	9.50	0.00	0.00	0.00	634.96	6.42
5	-30	-27.82	-12.90	637314.98	136.40	53.78	540.94	14.21
10	-30	-27.86	-18.40	1217915.66	62.56	228.78	559.50	23.18
15	-30	-29.63	-22.37	1694700.32	40.70	303.00	610.05	32.91

Table 5-18 Heat Transfer Coefficient and Theoretical Power at -35°C

FLAT PLATE								
Wind Speed (m/s)	***Set Temp (°C)***	***Ambient Temp (°C)***	***Surface Temp (°C)***	***Reynolds Number***	***Heat laminar flow (W)***	***Heat turbulent flow (W)***	***Power (W)***	***Overall Heat Transfer Coefficient U (W/m^2K)***
0	-35	-27.77	4.47	0.00	0.00	0.00	520.29	6.32
5	-35	-27.33	-12.60	637314.98	134.66	53.10	534.76	14.23
10	-35	-30.55	-23.55	1217915.66	46.29	168.29	413.09	23.12
15	-35	-26.90	-22.35	1694700.32	25.51	189.90	366.46	31.59

5.3 Time to freeze tables for different OD pipes with varying insulation thickness

Time to freeze tables were developed with the help of python code which is presented in Appendix A. The tables were generated for diameter 25 mm and diameter 50 mm pipes having similar wall thickness with insulation thickness of 0 mm, 10 mm and 25 mm at 0, 5, 10 and 15 m/s wind speed to show the effect of insulation and wind speed on the freezing time of water inside the pipe. The code can be used to generate values for any combination. But, cases which are applicable in this thesis are presented in Table 5-19, Table 5-20, Table 5-21, Table 5-22, Table 5-23, and Table 5-24 to show the comparison. It can be clearly observed that the time to freeze for water inside diameter 25 mm and diameter 50 mm pipes having no insulation is significantly lower than that compared with the insulated pipes. (ASHRAE, 2010)

In the case of diameter 25 mm pipe subjected to 0 m/s wind condition, the time to freeze increases by 27 % with 10 mm thick insulation and to 52 % with 25 mm thick insulation. In general, the values given by correlations are comparable and show closeness among themselves with minimal deviation. The time to freeze reduces by 2000 % with the introduction of 5 m/s wind speed. Variation in time to freeze from 5 m/s to 15 m/s is 62 % for uninsulated pipe. But, the same variation drops to less than 1 % with 10 mm and 25 mm thick insulation.

Furthermore, for diameter 50 mm pipe, the time to freeze values are much higher because of larger volume per unit length inside the pipe compared to 25 mm pipe. The other values for diameter 50 mm pipe compare well in general with diameter 25mm pipe with similar percentage of increase or decrease in time to freeze. Lower thermal conductivity of the insulation helps considerably in minimizing the heat loss from the piping system. Among all the correlation Churchill-Bernstein is closest to the average value and can be used. Whitaker correlation is applicable only for Reynolds number upto100, 000 and that can be a drawback when higher velocities are involved as see in Table 5-21, Table 5-23 and Table 5-24

Table 5-19 Time to freeze for OD 25 mm uninsulated pipe

Pipe OD (mm)	Insulation Thickness (mm)	Wind Velocity u_m(m/s)	Internal Temp t (°C)	Ambient Temp t (°C)	Overall Heat Transfer coefficient, U1 (W/m2 .K)	Time To Freeze TTF (Hours)
Original Hilpert Correlation						
25	0	0	15	-20	2.56	13.79
		5	15	-20	61.62	0.68
		10	15	-20	91.86	0.49
		15	15	-20	112.60	0.42
Updated Hilpert Constants						
25	0	0	15	-20	2.84	12.44
		5	15	-20	68.33	0.62
		10	15	-20	101.86	0.45
		15	15	-20	124.84	0.39
Fand & Keswani Constants						
25	0	0	15	-20	2.47	14.29
		5	15	-20	59.45	0.70
		10	15	-20	89.15	0.50
		15	15	-20	109.60	0.43
Morgan Constants						
25	0	0	15	-20	2.47	14.29
		5	15	-20	60.51	0.69
		10	15	-20	91.09	0.49
		15	15	-20	112.21	0.42
Zukauskas Correlation						
25	0	0	15	-20	2.43	14.52
		5	15	-20	76.87	0.56
		10	15	-20	113.26	0.42
		15	15	-20	137.98	0.36
Whitaker Correlation						
25	0	0	15	-20	2.52	14.02
		5	15	-20	85.82	0.52
		10	15	-20	123.52	0.39
		15	15	-20	148.87	0.34
Churchill-Bernstein Correlation						
25	0	0	15	-20	2.75	12.86
		5	15	-20	70.28	0.61
		10	15	-20	102.12	0.45
		15	15	-20	124.49	0.39

Table 5-20 Time to freeze for OD 25mm pipe with 10 mm thick insulation

Pipe OD (mm)	Insulation Thickness (mm)	Wind Velocity u_m(m/s)	Internal Temp t (°C)	Ambient Temp t (°C)	Overall Heat Transfer coefficient, U1 (W/m2 .K)	Time To Freeze TTF (Hours)
Original Hilpert Correlation						
25	10	0	15	-20	2.01	17.60
		5	15	-20	5.04	7.05
		10	15	-20	5.15	6.91
		15	15	-20	5.19	6.85
Updated Hilpert Constants						
25	10	0	15	-20	2.14	16.51
		5	15	-20	5.07	7.01
		10	15	-20	5.17	6.88
		15	15	-20	5.21	6.83
Fand & Keswani Constants						
25	10	0	15	-20	1.96	17.97
		5	15	-20	5.03	7.06
		10	15	-20	5.27	6.74
		15	15	-20	5.29	6.72
Morgan Constants						
25	10	0	15	-20	1.97	17.89
		5	15	-20	5.04	7.05
		10	15	-20	5.14	6.92
		15	15	-20	5.19	6.86
Zukauskas Correlation						
25	10	0	15	-20	1.95	18.08
		5	15	-20	5.10	6.97
		10	15	-20	5.17	6.87
		15	15	-20	5.20	6.83
Whitaker Correlation						
25	10	0	15	-20	2.09	16.88
		5	15	-20	5.12	6.94
		10	15	-20	5.19	6.85
		15	15	-20	5.21	6.82
Churchill-Bernstein Correlation						
25	10	0	15	-20	2.14	16.47
		5	15	-20	5.07	7.01
		10	15	-20	5.16	6.89
		15	15	-20	5.19	6.85

Table 5-21 Time to freeze for OD 25mm pipe with 25mm thick insulation

Pipe OD (mm)	Insulation Thickness (mm)	Wind Velocity u_m(m/s)	Internal Temp t (°C)	Ambient Temp t (°C)	Overall Heat Transfer coefficient, U (W/m^2 .K)	Time To Freeze TTF (Hours)
Original Hilpert Correlation						
25	25	0	15	-20	1.69	20.93
		5	15	-20	2.80	12.64
		10	15	-20	2.82	12.53
		15	15	-20	2.83	12.49
Updated Hilpert Constants						
25	25	0	15	-20	1.76	20.08
		5	15	-20	2.80	12.61
		10	15	-20	2.83	12.51
		15	15	-20	2.83	12.47
Fand & Keswani Constants						
25	25	0	15	-20	1.66	21.26
		5	15	-20	2.84	12.46
		10	15	-20	2.85	12.41
		15	15	-20	2.85	12.40
Morgan Constants						
25	25	0	15	-20	1.66	21.22
		5	15	-20	2.79	12.65
		10	15	-20	2.82	12.53
		15	15	-20	2.83	12.49
Zukauskas Correlation						
25	25	0	15	-20	1.65	21.40
		5	15	-20	2.81	12.59
		10	15	-20	2.82	12.52
		15	15	-20	2.83	12.49
Whitaker Correlation						
25	25	0	15	-20	1.75	20.17
		5	15	-20	2.81	12.58
		10	15	-20	2.83	12.51
		15	15	-20	N/A	N/A
Churchill-Bernstein Correlation						
25	25	0	15	-20	1.75	20.13
		5	15	-20	2.80	12.62
		10	15	-20	2.82	12.53
		15	15	-20	2.83	12.50

Table 5-22 Time to freeze for OD 50mm uninsulated pipe

Pipe OD (mm)	Insulation Thickness (mm)	Wind Velocity u_m(m/s)	Internal Temp t (°C)	Ambient Temp t (°C)	Overall Heat Transfer coefficient, U(W/m^2 .K)	Time To Freeze TTF (Hours)
Original Hilpert Correlation						
50	0	0	15	-20	1.43	49.48
		5	15	-20	39.79	2.29
		10	15	-20	62.87	1.64
		15	15	-20	82.03	1.38
Updated Hilpert Constants						
50	0	0	15	-20	1.73	44.62
		5	15	-20	47.90	2.12
		10	15	-20	77.01	1.52
		15	15	-20	100.40	1.29
Fand & Keswani Constants						
50	0	0	15	-20	1.50	51.36
		5	15	-20	41.93	2.35
		10	15	-20	193.82	0.92
		15	15	-20	260.21	0.82
Morgan Constants						
50	0	0	15	-20	1.50	51.20
		5	15	-20	42.86	2.31
		10	15	-20	65.48	1.69
		15	15	-20	85.63	1.42
Zukauskas Correlation						
50	0	0	15	-20	1.45	52.83
		5	15	-20	53.23	1.96
		10	15	-20	78.44	1.50
		15	15	-20	95.59	1.33
Whitaker Correlation						
50	0	0	15	-20	1.66	46.35
		5	15	-20	57.97	1.85
		10	15	-20	83.72	1.44
		15	15	-20	101.08	1.28
Churchill-Bernstein Correlation						
50	0	0	15	-20	1.72	44.87
		5	15	-20	47.97	2.12
		10	15	-20	71.29	1.60
		15	15	-20	88.09	1.40

Table 5-23 Time to freeze for OD 50mm pipe with 10mm thick insulation

Pipe OD (mm)	***Insulation Thickness (mm)***	***Wind Velocity u_m(m/s)***	***Internal Temp t (°C)***	***Ambient Temp t (°C)***	***Overall Heat Transfer coefficient, U (W/m^2 .K)***	***Time To Freeze TTF (Hours)***
Original Hilpert Correlation						
50	10	0	15	-20	1.27	60.19
		5	15	-20	3.95	19.85
		10	15	-20	4.07	19.27
		15	15	-20	4.11	19.07
Updated Hilpert Constants						
50	10	0	15	-20	1.37	56.03
		5	15	-20	3.98	19.69
		10	15	-20	4.09	19.17
		15	15	-20	4.13	19.00
Fand & Keswani Constants						
50	10	0	15	-20	1.24	61.78
		5	15	-20	4.14	18.94
		10	15	-20	4.20	18.71
		15	15	-20	4.21	18.63
Morgan Constants						
50	10	0	15	-20	1.24	61.58
		5	15	-20	3.95	19.85
		10	15	-20	4.06	19.30
		15	15	-20	4.11	19.09
Zukauskas Correlation						
50	10	0	15	-20	1.23	62.54
		5	15	-20	4.00	19.59
		10	15	-20	4.08	19.21
		15	15	-20	4.11	19.07
Whitaker Correlation						
50	10	0	15	-20	1.35	56.66
		5	15	-20	4.02	19.50
		10	15	-20	4.09	19.17
		15	15	-20	N/A	N/A
Churchill-Bernstein Correlation						
50	10	0	15	-20	1.36	56.28
		5	15	-20	3.98	19.71
		10	15	-20	4.07	19.28
		15	15	-20	4.11	19.11

Table 5-24 Time to freeze for OD 50mm pipe with 25mm thick insulation

Pipe OD (mm)	***Insulation Thickness (mm)***	***Wind Velocity u_m(m/s)***	***Internal Temp t (°C)***	***Ambient Temp t (°C)***	***Overall Heat Transfer coefficient, U (W/m² .K)***	***Time To Freeze TTF (Hours)***
Original Hilpert Correlation						
50	25	0	15	-20	1.05	72.67
		5	15	-20	2.01	38.37
		10	15	-20	2.03	37.94
		15	15	-20	2.04	37.79
Updated Hilpert Constants						
50	25	0	15	-20	1.11	69.14
		5	15	-20	2.02	38.25
		10	15	-20	2.04	37.86
		15	15	-20	2.05	37.73
Fand & Keswani Constants						
50	25	0	15	-20	1.03	74.00
		5	15	-20	2.05	37.67
		10	15	-20	2.06	37.51
		15	15	-20	2.06	37.46
Morgan Constants						
50	25	0	15	-20	1.04	73.79
		5	15	-20	2.01	38.41
		10	15	-20	2.03	37.96
		15	15	-20	2.04	37.80
Zukauskas Correlation						
50	25	0	15	-20	1.03	74.21
		5	15	-20	2.02	38.24
		10	15	-20	2.03	37.94
		15	15	-20	2.04	37.82
Whitaker Correlation						
50	25	0	15	-20	1.11	68.91
		5	15	-20	2.02	38.18
		10	15	-20	N/A	N/A
		15	15	-20	N/A	N/A
Churchill-Bernstein Correlation						
50	25	0	15	-20	1.10	69.32
		5	15	-20	2.01	38.33
		10	15	-20	2.03	37.98
		15	15	-20	2.04	37.84

5.4 Comparison

In order to elaborate the findings of the results better, comparison between the theoretical and experimental analysis for pipes and deck element are shown in the following pages. The plots are taken from experimental and theoretical readings which were presented earlier.

5.4.1 Comparison of overall heat transfer coefficients for uninsulated and insulated pipes

Plot showing the comparison between the overall heat transfer coefficients for diameter 50 mm uninsulated pipe versus diameter 50 mm insulated pipes is presented in Figure 5-21.

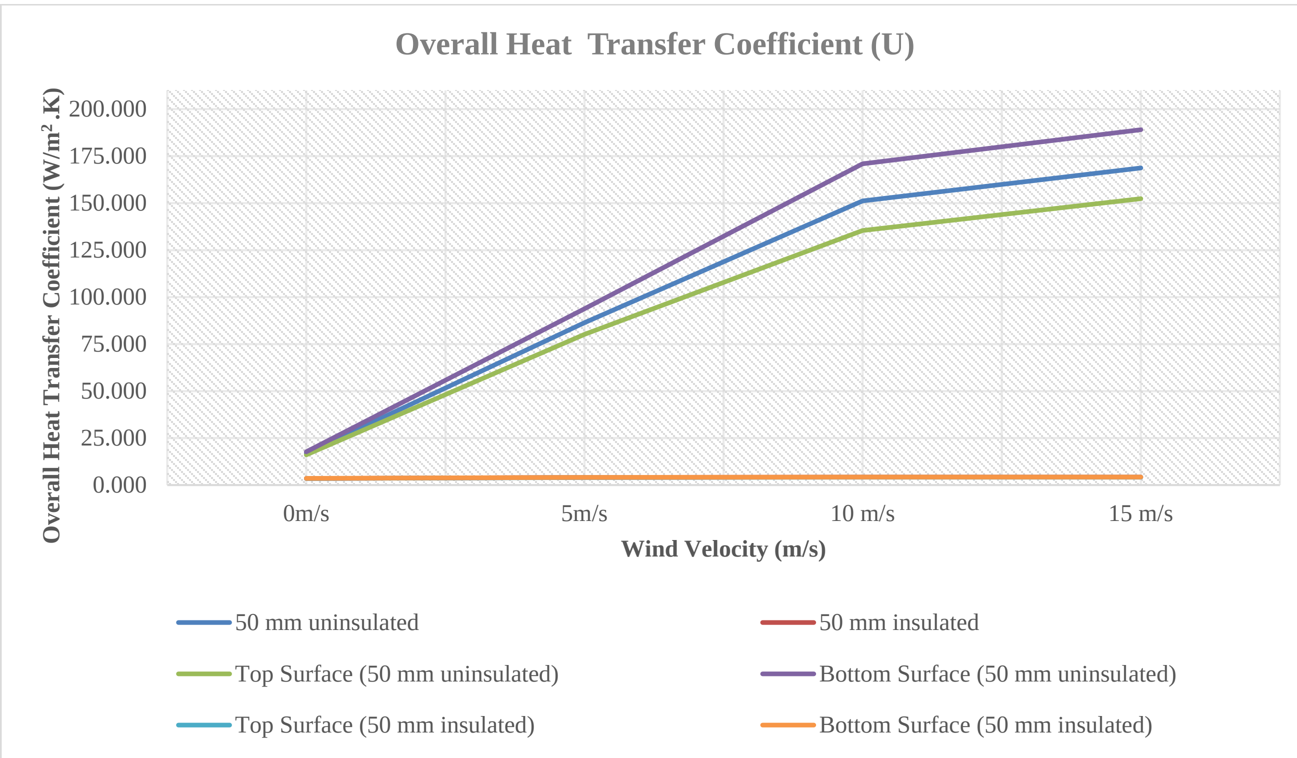

Figure 5-21 Overall Heat Transfer coefficient for a single uninsulated pipe v/s insulated pipe.

This plot is a combination of earlier results which were explained in detail. The aim here is to compare them to see the difference in overall heat transfer coefficients. As observed, the values of uninsulated pipe is extremely high compared to the insulated pipe. The values for uninsulated pipe is in the range of 17-169 W/m^2. K for 0 – 15 m/s wind speed condition. Whereas, for insulated pipe the value ranges from 3.4 – 4 W/m^2. K for 0-15 m/s wind speed condition. The effect of insulation and their role in decreasing the heat loss because of low thermal conductivity is conclusive from these values. The decrease in overall heat transfer coefficient by 400- 4000 % is substantial. These numbers are very significant as they are indicative of the amount of energy that is transferred by not using proper insulation in piping system when transporting fluids from one place to another.

5.4.2 Comparison of theoretical and experimental findings for 50 mm uninsulated pipe.

Plot showing the comparison between the overall heat transfer coefficients obtained from theoretical and experimental method for 50 mm uninsulated pipe is shown in Figure 5-22.

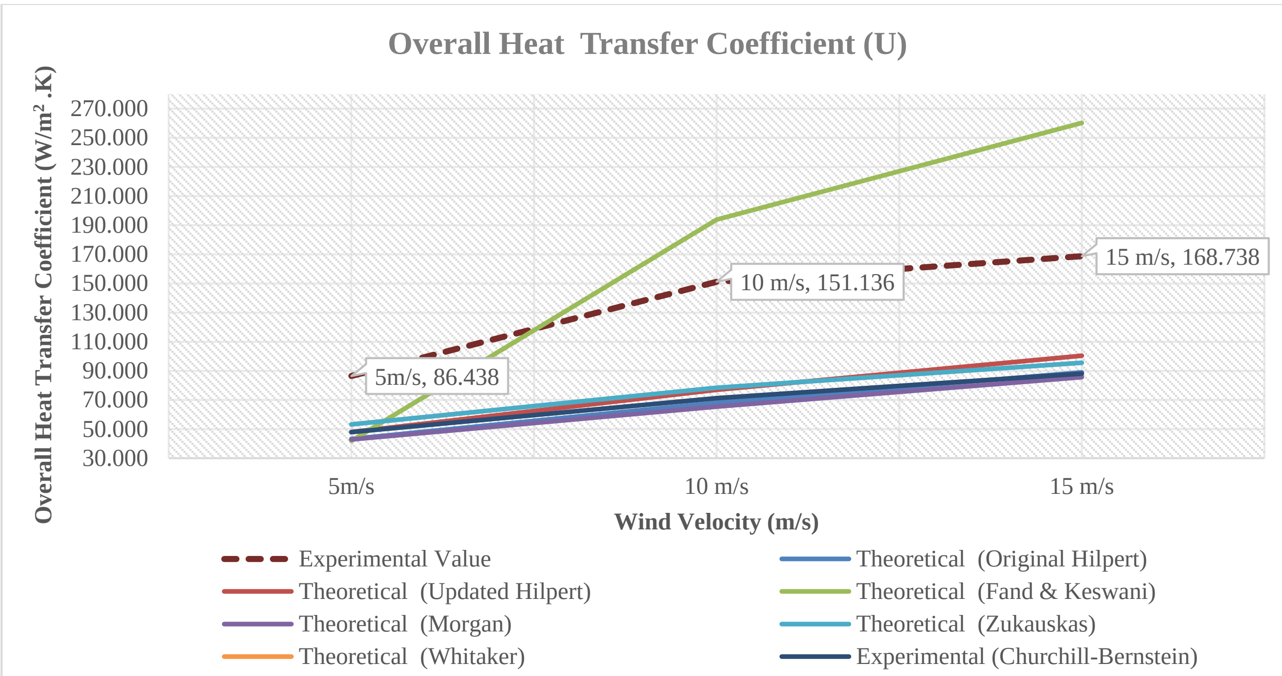

Figure 5-22 Comparison between experimental and theoretical analysis of 50mm uninsulated pipe

The above plot is obtained by including values from experimental and theoretical calculations. This will help in finding the best method for use by the industry in describing the heat transfer from uninsulated pipes. As explained earlier, all the theoretical correlations show agreement and compare well except for (Fand and Keswani, 1973). The deviation between results from experimental and conventional method are in the range of 72-88 %. It has to be noted that this experiment for uninsulated pipe was not one of the stable experiments considering the temperature readings from the bare pipe were influenced by the wind effect and the attached sensors may not be showing the actual surface temperature of the pipe unlike insulated pipe surface. Although, the values from experimental and theoretical calculation show huge difference, it can be clearly seen that the trend is the same and the error could have been due the temperature readings getting influenced. Most of the correlations give good estimation of the heat transfer coefficient values. But, considering the governing factors of ease of use, range of validity and accuracy, some of them can be avoided like (Whitaker, 1972) and (Fand and Keswani, 1973) because of range of validity (Reynolds number up to 100,000) and accuracy. (Morgan, 1975) can also be avoided because of range of validity as it is applicable only up to Reynolds number 200,000. (Žukauskas, 1972) and Churchill-Bernstein are recommended because of their wide range, accuracy and minimal deviation from the experimental values. Another advantage with Churchill-Bernstein is that it is a comprehensive equation and doesn't require look up tables unlike (Hilpert, 1933) and (Žukauskas, 1972)

5.4.3 Comparison of theoretical and experimental findings for 50 mm insulated pipe.

Plot showing the comparison between the overall heat transfer coefficients obtained from theoretical and experimental method for 50 mm insulated pipe with 10 mm thick insulation is shown in Figure 5-23.

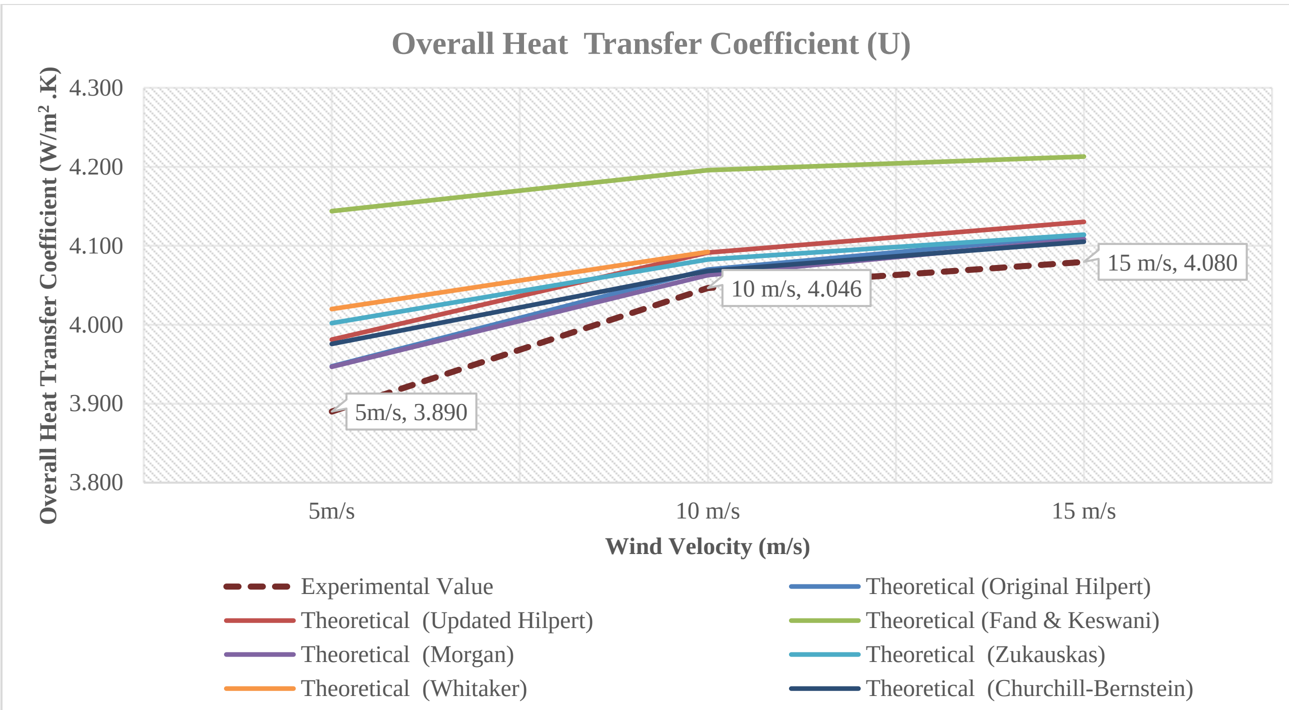

Figure 5-23 Comparison between experimental and theoretical analysis of 50mm insulated pipe

In order to suggest the best method to use in the industry to perform heat transfer calculation for insulated pipe, we need to study this plot carefully. It can be noted that all the heat transfer coefficient calculated using different correlations except for (Fand and Keswani, 1973) show deviation in the range of just 0.5 -2.82 % which is negligible. It is evident that this experiment for insulated pipe was one of the most stable experiments considering the pipe surface temperature readings were not influenced by the wind effect because of the presence of insulation. Although, the values from experimental and theoretical calculation show close proximity, some of the correlations can be omitted based on the governing factors suggested in the criteria like range of validity, ease of use and accuracy. Most of the correlations gave good estimation of the heat transfer coefficient values. (Whitaker, 1972) and (Fand and Keswani, 1973) can be neglected because of similar issue as explained earlier pertaining to range of validity (Reynolds number up to 100,000) and accuracy. Similarly, (Morgan, 1975) can also be avoided because of range of validity criteria as it is applicable only up to Reynolds number 200,000. (Žukauskas, 1972) and Churchill-Bernstein are recommended because of their wide range, accuracy and minimal deviation from the experimental values. One of the major advantage with Churchill-Bernstein is that it is a comprehensive equation and doesn't require look up tables for constants based on Prandtl numbers and Reynolds number unlike (Hilpert, 1933) and (Žukauskas, 1972)

5.4.4 Comparison between theoretical and experimental findings for 25 mm insulated pipe.

Plot showing the comparison between the overall heat transfer coefficients obtained from theoretical and experimental method for 25 mm insulated pipe with 10 mm thick insulation is shown in Figure 5-24

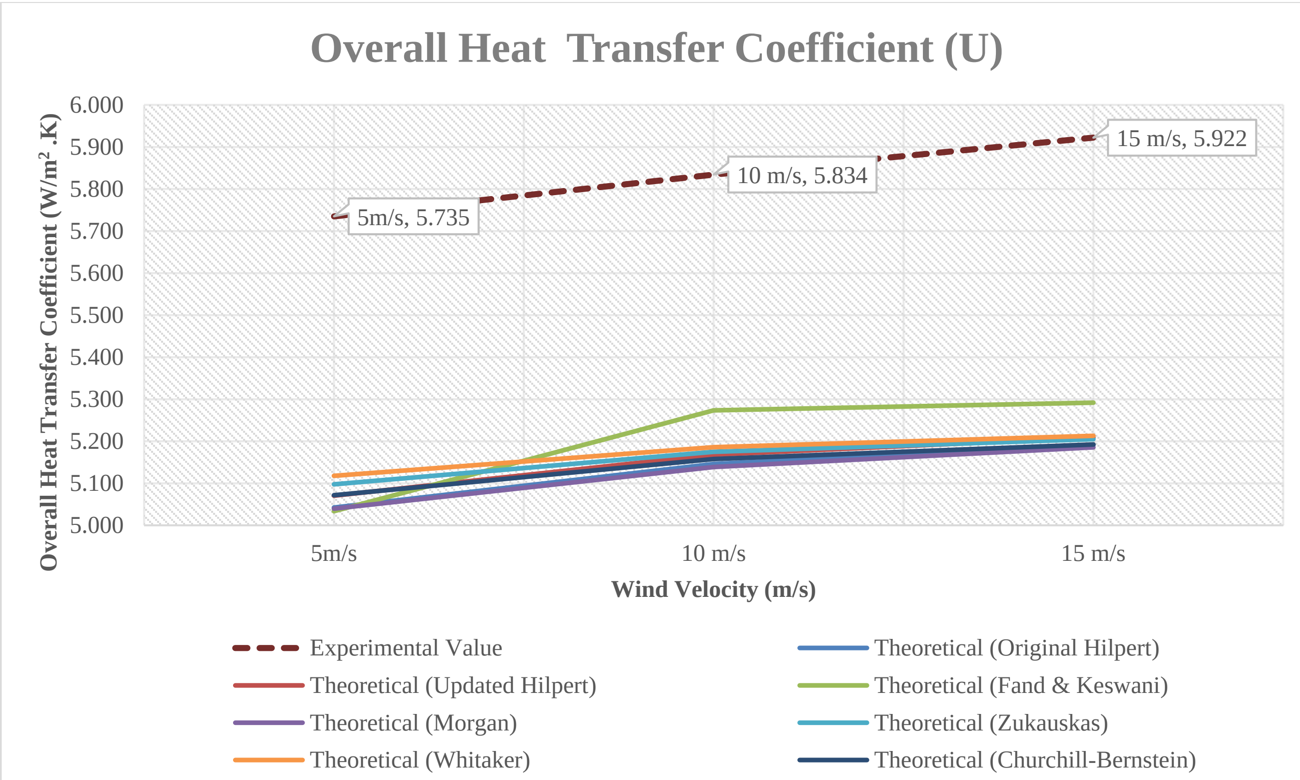

Figure 5-24 Comparison between experimental and theoretical analysis of 25mm insulated pipe

It is observed that all the heat transfer coefficient calculated using different correlations for OD 25 mm pipe except for (Fand and Keswani, 1973) show deviation in the range of 12-14 % compared to experimental values. The experiments were performed in stable condition as can be noted from the trend followed by the experimental plot. Although, the values from experimental and theoretical calculation show slight deviation, it can be neglected as the difference is significant. Like in the case of OD 50 mm insulated pipe some of the correlations can be omitted based on the governing factors suggested in the criteria like range of validity, ease of use and accuracy. (Whitaker, 1972) and (Fand and Keswani, 1973) can be neglected due to its range of validity (Reynolds number up to 100,000) and accuracy issue respectively. (Morgan, 1975) is applicable only up to Reynolds number 200,000 and can be avoided even though the deviations in the overall heat transfer coefficients were found to be comparable. . (Žukauskas, 1972) and Churchill-Bernstein are recommended because of fulfilling the criteria like wide range, accuracy and least deviation from the experimental values. Churchill-Bernstein is more preferred because it is a comprehensive equation and doesn't require look up tables for constants based on Prandtl numbers and Reynolds number unlike (Hilpert, 1933) and (Žukauskas, 1972)

5.4.5 Comparison between theoretical and experimental findings for deck element

Table 5-25 shows the difference between the theoretical and the experimental power requirement for deck element. It also shows the increased power consumption when the wind velocity increases from 0 m/s to higher values in order to maintain the temperature of the deck element. Plot showing the comparison between the overall heat transfer coefficients obtained from theoretical and experimental analysis of the deck element is shown in Figure 5-25

Table 5-25 Comparison of experimental and theoretical power requirement for deck element

FLAT PLATE							
Set Temp (°C)	***Wind Speed (m/s)***	***Surface Temp (°C)***	***Consumed Power (W)***	***Increased Power (W)***	***Power (Theory) (W)***	***Calculated Required Power (W)***	***Delta Power (W)***
-15	0	19.97	936	0	522	0	0
	5	3.70	1076	140	671	224	84
	10	-1.23	1123	187	755	377	190
	15	-3.17	1152	216	813	466	250
-20	0	21.27	947	0	716	0	0
	5	-2.03	1175	228	642	219	-9
	10	-7.20	1232	285	717	363	78
	15	-9.67	1264	316	752	435	119
-30	0	9.50	1071	0	635	0	0
	5	-12.90	1231	160	541	190	30
	10	-18.40	1321	249	559	291	42
	15	-22.37	1362	291	610	344	53
-35	0	4.47	1124	0	520	0	0
	5	-12.60	1292	168	535	188	20
	10	-23.55	1399	275	413	215	-61
	15	-22.35	1409	285	366	215	-70

The consumed power is the value displayed on the control interface during the experiment. This is the power which is consumed by the deck element during the experiment in order to heat up the deck element and the increased power shows the excess power requirement when the speed of the wind is increased from 0 m/s. The heat transfer from the deck element takes place through different modes (refer to sample theoretical calculation shown in section 4.2.3). It can be observed that the delta power which is the summation of the laminar convective heat transfer and turbulent convective heat transfer from the top surface of the deck element is comparable to the increase in power consumption of the deck element to compensate for the wind effect. The values are not consistent throughout the experiment which can be attributed to the fact that the in some experiments of the deck element, fully stabilized condition were not attained and the tests might have been stopped prior to attainment of the equilibrium condition as there was no means to log the temperature readings continuously compared to pipe testing. The heating coils were embedded inside the deck surface and the temperature readings were monitored using the infrared camera which gives the surface temperature and not the temperature gradient of the deck surface. It is important to attain steady state for application of the relevant theoretical methods. The difference between the actual power consumed and theoretical power is very high and not fully comparable.

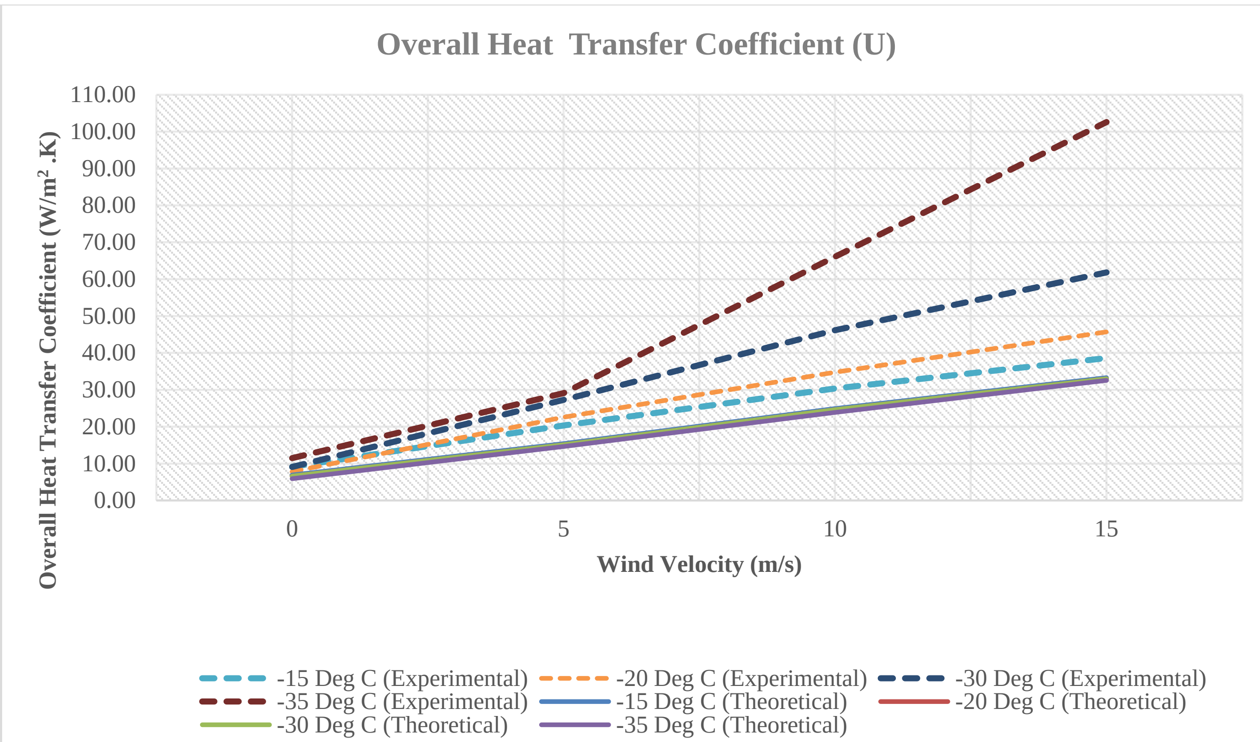

Figure 5-25 Comparison between experimental and theoretical analysis of deck element

From the plot in Figure 5-25, it can be noted that for ambient condition of -15° C and -20° C, and to an extend -30° C shows similar trend for overall heat transfer coefficient as suggested by the theoretical method. Though, values for -30° C are higher at increased velocity. The overall heat transfer coefficient for the deck element with epoxy coating is high. The only exception is -35° C ambient condition which can be because of erroneous ambient temperature reading at the time of the experiment as explained in earlier section. The deck elements are not self-regulating and the governing factor for the increased power supply is the temperature of the cable. Neglecting the last value for -35°C ambient condition, it is observed that the increase in the overall heat transfer coefficient is 30 -90% for 0 m/s wind condition and 17- 87 % for 15 m/s wind condition (excluding -35° C condition) when comparing the theoretical and the experimental values.

6 Conclusions

Extensive review of the available literature on heat transfer from horizontal pipes and flat plates under cross flow wind showed the availability of different heat transfer correlations which have wide range of validity. This thesis, while comparing experimental findings and theoretical calculations, shows that proper selection of heat transfer correlation is extremely critical. Usage of an improper correlation can give erroneous results up to 100% and thus, proper guidance is essential for designers and engineers performing calculations for heat loss from horizontal pipes which are subjected to cross-flow wind. For deck elements, there is only one correlation available for performing the heat transfer calculations unlike for horizontal pipes.

The test methodology developed for testing the heat transfer from the pipes and a heated deck element gave reasonably good results conforming to theoretical calculation for the selected correlation. So, it is recommended for industrial usage to conduct the experiments to validate the findings. The test apparatus designed for determination of the heat transfer coefficient was portable and sturdy; it was capable of accommodating multiple pipes of varying diameters thus providing a wide range of applicability and worked on the principle of energy balance upon reaching equilibrium condition.

The experiments performed using cross flow wind of 5 m/s, 10 m/s and 15 m/s blowing over multiple pipe configurations of diameter 25 mm and 50 mm insulated and uninsulated steel pipes yielded mostly consistent results. Heat transfer correlations such as those suggested by Hilpert, Fand and Keswani, Morgan, Žukauskas, Whitaker and Churchill-Bernstein were used to determine the heat transfer coefficients for horizontal pipes subjected to cross flow wind and the results were compared with the experimental values. The comparison showed that the values of the heat transfer coefficients for the insulated pipes had minimal deviation; i.e. in the range of 0.5 - 2.82 % in the case of diameter 50 mm insulated pipe and 12 -14 % in the case of diameter 25 mm insulated pipe. The most significant finding was the effect of insulation on the reduction of heat loss. Comparison of diameter 50 mm uninsulated and insulated pipes showed that the reduction in heat transfer coefficient is in the range of 400 - 4000 % with the usage of 10 mm thick insulation made of elastomeric foam based on synthetic rubber.

However, in the case of uninsulated pipe and deck element, the heat transfer coefficients values didn’t show very close proximity compared to insulated pipe. Comparison of experimental values and theoretical calculations yielded results which had deviation in the range 72 - 88 % and 17- 90 % respectively. Time to freeze results for diameter 25 mm and diameter 50 mm uninsulated and insulated pipes showed increase in time to freeze. The increase was 27 % and 52 % with the usage of 10 mm and 25 mm insulation, respectively in the case of diameter 25 mm pipe. For diameter 50 mm pipe, the time to freeze increased by 22 % and 47 % respectively for similar increase in insulation thickness.

Based on the governing criteria such as ease of use, range of validity, accuracy and the experimental findings, the Churchill-Bernstein correlation was suggested as the best method for use by the industry.

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APPENDIX

Appendix A -Python code for heat transfer coefficient and time to freeze calculations

```
## Code for calculating heat loss from cylinders
## Written by Bjarte Odin Kvamme


## Import required packages
import numpy as np
import scipy as sp
import matplotlib as mpl
import matplotlib.pyplot as plt

##

from numpy import *
from math import pi
import scipy.linalg
from datetime import datetime
import xlsxwriter

## Define variables
V_infty = [6.63, 12.63, 17.67]                    # Wind Speed values in m/s,
comma delimited
TiC = [10]                                # Initial temperature of pipe, in
degrees Celsius, comma delimited
TeC = [-20]                             # External temperature, in degrees
Celsius, comma delimited
D_tab = [0.025, 0.050]
t_w = 0.002                                       # Wall thickness of pipe
p_atm = 103.1                                     # Air pressure, in
kilopascal (kPa) obtained from Incropera et al., 2006.
Tc = 272.15                                       #Desired temperature of ice

# Create inital array to store the results obtained
results = []
results.append([])
row = 0                                           # Initial loop counter value

## Constants
# Properties of pipe
D_i = 0.046                                       # Inner diameter of pipe (m)
D_o = 0.050                                       # Outer diameter of pipe (m)
t_ice = 0.005                                     # Thickness of external ice
layer (m)
k_pipe = 43                                       # Thermal conductivity of
carbon steel pipe (W/(m K))

# Properties of insulation
t_ins = [0, 0.01]                                 # Thickness of insulation (m)
k_ins = 0.033                                     # Thermal conductivity of
insulation (W/(m K))

# Properties of air
#Pr_air = 0.714                                   # Prandt number for air at 10C
#k_air = 0.02265                                  # Thermal Conductivity of
air (W/(m K))
#v_air = 13.3e-6                                  # Kinematic viscosity of air
(m2/s)
#rho_air = 1.3163                                 # Density of air at -5C
(kg/m3)
#mu_air = 1.76e-5                                 # Dynamic viscosity of air
(kg/m s)
R_air = 0.287                                     # kJ / kg K obtained from
Incropera et al., 2006.

# Properties of water
cp_w = 4217                                       # Specific heat of water at
5C (J/(Kg K)) obtained from Incropera et al., 2006.
Tf_w = 273.15                                     # Freezing temperature of
```

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water (0C) obtained from Incropera et al., 2006.
h_w = 1000                                           # Heat transfer co-efficient of water (W/(m2 K))
rho_w = 1000                                         # Density of water at 0C (kg/m3) obtained from Incropera et al., 2006.
hfs_w = 333.7                                        # Latent heat of fusion for water (J/g) obtained from Incropera et al., 2006.

# Properties of ice
k_ice = 1.88                                         # Thermal conductivity of ice at 0C (W/(m K)) obtained from Incropera et al., 2006.
rho_ice = 920                                        # Density of ice at 0C (kg/m3) obtained from Incropera et al., 2006.
cp_ice = 2.040

# Properties of heat tracing
# ql ht = 50                                         # Applied heat (W/m) from heat tracing

# All functions below assume steady-state conditions.

# Thermodynamic properties of air, obtained from Incropera et al., 2006.
alpha air table = {'100': 2.54E-6, '150': 5.84E-6, '200': 10.3E-6, '250': 15.9E-6, '300': 22.5E-6, '350': 29.9E-6, '400': 38.3E-6, '450': 47.2E-6}
k_air_table = {'100': 9.34E-3, '150': 13.8E-3, '200': 18.1E-3, '250': 22.3E-3, '300': 26.3E-3, '350': 30.0E-3, '400': 33.8E-3, '450': 37.3E-3}
mu_air_table = {'100': 71.1E-7, '150': 103.4E-7, '200': 132.5E-7, '250': 159.6E-7, '300': 184.6E-7, '350': 208.2E-7, '400': 230.1E-7, '450': 250.7E-7}
cp air table = {'100': 1.032, '150': 1.012, '200': 1.007, '250': 1.006, '300': 1.007, '350': 1.009, '400': 1.014, '450': 1.021}
nu_air_table = {'100': 2.00E-6, '150': 4.426E-6, '200': 7.590E-6, '250': 11.44E-6, '300': 15.89E-6, '350': 20.92E-6, '400': 26.41E-6, '450': 32.39E-6}
Pr_air_table = {'100': 0.786, '150': 0.758, '200': 0.737, '250': 0.720, '300': 0.707, '350': 0.700, '400': 0.690, '450': 0.686}
Rho air table = {'100': 3.5562, '150': 2.3364, '200': 1.7458, '250': 1.3947, '300': 1.1614, '350': 0.9950, '400': 0.87711, '450': 0.7740}

# Thermodynamic properties of water, obtained from Incropera et al., 2006.
cp_w_table = {'273.15': 4.217, '275': 4.211, '280': 4.198, '285': 4.189, '290': 4.184, '295': 4.181, '300': 4.179, '305': 4.178, '310': 4.178, '315': 4.179, '320': 4.180, '325': 4.182, '330': 4.184, '335': 4.186, '340': 4.188, '345': 4.191 }

# Hilpert correlation constants
Hilpert_C = {'1-4': 0.891, '4-40': 0.821, '40-4000': 0.615, '4000-40000': 0.174, '40000-400000': 0.0239}
Hilpert_m = {'1-4': 0.330, '4-40': 0.385, '40-4000': 0.466, '4000-40000': 0.618, '40000-400000': 0.805}

# Updated Hilpert correlation constants
UpdatedHilpert_C = {'0.4-4': 0.989, '4-40': 0.911, '40-4000': 0.683, '4000-40000': 0.193, '40000-400000': 0.027}
UpdatedHilpert_m = {'0.4-4': 0.330, '4-40': 0.385, '40-4000': 0.466, '4000-40000': 0.618, '40000-400000': 0.805}

# Updated Hilpert correlation constants, Fand & Keswani (1973)
FandKeswani_C = {'1-4': 0.875, '4-40': 0.785, '40-4000': 0.590, '4000-40000': 0.154, '40000-400000': 0.0247}
FandKeswani_m = {'1-4': 0.313, '4-40': 0.388, '40-4000': 0.467, '4000-40000': 0.627, '40000-400000': 0.898}

# Updated Hilpert correlation constants, Morgan (1975)
Morgan_C = {'0.0001-0.004': 0.437, '0.004-0.09': 0.565, '0.09-1': 0.800, '1-35': 0.795, '35-5000': 0.583, '5000-50000': 0.148, '50000-200000': 0.0208}
Morgan_m = {'0.0001-0.004': 0.0895, '0.004-0.09': 0.136, '0.09-1': 0.280, '1-35': 0.384, '35-5000': 0.471, '5000-50000': 0.633, '50000-200000': 0.814}

# Zukauskas correlation constants, Zukauskas (1972)
Zukauskas_C = {'1-40': 0.75, '40-1000': 0.51, '1000-200000': 0.26, '200000-1000000':
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-2-

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0.076}
Zukauskas_m = {'1-40': 0.4, '40-1000': 0.5, '1000-200000': 0.6, '200000-1000000': 0.7}

# def As ( D ):
    # "This function calculates the surface area per length (m2/m) of pipe"
    # return (pi*D)

# def Vl ( D_i ):
    # "This function calculates the volume per unit length (m3/m) of pipe"
    # return (pi*(D_i/2)**2)

# def Ml ( D_i, rho ):
    # "This function calculates the mass per unit length, based on the diameter of
    the pipe and the density of the contents (kg/m3)"
    # return ((pi*(D_i/2)**2)*rho)

def Re( V, D, rho, mu ):
    "This function calculates the Reynolds number given the wind speed and diameter
    of the pipe"
    return ((rho*V*D)/mu)

# def Pr_air_calc ( nu, alpha ):
    # "This functions calculates the Prandtl number for air, based on the air
    temperature"
    # return ( nu/alpha )

# def rho_air_calc ( T, p ):
    # "This function calculates the density of air at a given temperature"
    # return (p/(R_air * T))

# def mu_air_calc ( T ):
    # "This function return the dynamic viscosity of air at a given temperature"
    # mu_ref = 17.16*10**-6
    # T_ref = 273.15
    # S = 110.4
    # return ( ((mu_ref*(T/T_ref)**(3/2))*((T_ref+S)/(T+S))) )

# def nu_calc ( mu, rho ):
    # "This function calculates the kinematic viscosity of a fluid"
    # return ( mu / rho )

def Pr_air_calc ( T ):
    "This function returns the thermal diffusivity of air at a given temperature"
    if 100 <= T <= 450:
        error = 0
        if 100 <= T < 125:
            Pr = Pr_air_table['100']
        elif 125 <= T < 175:
            Pr = Pr_air_table['150']
        elif 175 <= T < 225:
            Pr = Pr_air_table['200']
        elif 225 <= T < 275:
            Pr = Pr_air_table['250']
        elif 275 <= T < 325:
            Pr = Pr_air_table['300']
        elif 325 <= T < 375:
            Pr = Pr_air_table['350']
        elif 375 <= T < 425:
            Pr = Pr_air_table['400']
        elif 425 <= T <= 450:
            Pr = Pr_air_table['450']
    if error == 0:
        return ( Pr )
    else:
        return ('N/A')

def cp_w_calc ( T ):
    "This function returns the thermal diffusivity of air at a given temperature"
```

-3-

```
    if 273.15 <= T <= 350:
        error = 0
        if 273.15 <= T < 275:
            cp_w = cp_w_table['273.15']
        elif 275 <= T < 280:
            cp_w = cp_w_table['275']
        elif 280 <= T < 285:
            cp_w = cp_w_table['280']
        elif 285 <= T < 290:
            cp_w = cp_w_table['285']
        elif 290 <= T < 295:
            cp_w = cp_w_table['290']
        elif 295 <= T < 300:
            cp_w = cp_w_table['295']
        elif 300 <= T < 305:
            cp_w = cp_w_table['300']
        elif 305 <= T < 310:
            cp_w = cp_w_table['305']
        elif 310 <= T < 315:
            cp_w = cp_w_table['310']
        elif 315 <= T < 320:
            cp_w = cp_w_table['315']
        elif 320 <= T < 325:
            cp_w = cp_w_table['320']
        elif 325 <= T < 330:
            cp_w = cp_w_table['325']
        elif 330 <= T < 335:
            cp_w = cp_w_table['330']
        elif 335 <= T < 340:
            cp_w = cp_w_table['335']
        elif 340 <= T < 345:
            cp_w = cp_w_table['340']
        elif 345 <= T <= 350:
            cp_w = cp_w_table['345']
    if error == 0:
        return ( cp_w )
    else:
        return ('N/A')

def Rho_air_calc ( T ):
    "This function returns the thermal diffusivity of air at a given temperature"
    if 100 <= T <= 450:
        error = 0
        if 100 <= T < 125:
            Rho = Rho_air_table['100']
        elif 125 <= T < 175:
            Rho = Rho_air_table['150']
        elif 175 <= T < 225:
            Rho = Rho_air_table['200']
        elif 225 <= T < 275:
            Rho = Rho_air_table['250']
        elif 275 <= T < 325:
            Rho = Rho_air_table['300']
        elif 325 <= T < 375:
            Rho = Rho_air_table['350']
        elif 375 <= T < 425:
            Rho = Rho_air_table['400']
        elif 425 <= T <= 450:
            Rho = Rho_air_table['450']
    if error == 0:
        return ( Rho )
    else:
        return ('N/A')

def nu_air_calc ( T ):
    "This function returns the thermal diffusivity of air at a given temperature"
    if 100 <= T <= 450:
        error = 0
```

```
        if 100 <= T < 125:
            nu = nu_air_table['100']
        elif 125 <= T < 175:
            nu = nu_air_table['150']
        elif 175 <= T < 225:
            nu = nu_air_table['200']
        elif 225 <= T < 275:
            nu = nu_air_table['250']
        elif 275 <= T < 325:
            nu = nu_air_table['300']
        elif 325 <= T < 375:
            nu = nu_air_table['350']
        elif 375 <= T < 425:
            nu = nu_air_table['400']
        elif 425 <= T <= 450:
            nu = nu_air_table['450']
    if error == 0:
        return ( nu )
    else:
        return ('N/A')

def mu_air_calc ( T ):
    "This function returns the thermal diffusivity of air at a given temperature"
    if 100 <= T <= 450:
        error = 0
        if 100 <= T < 125:
            mu = mu_air_table['100']
        elif 125 <= T < 175:
            mu = mu_air_table['150']
        elif 175 <= T < 225:
            mu = mu_air_table['200']
        elif 225 <= T < 275:
            mu = mu_air_table['250']
        elif 275 <= T < 325:
            mu = mu_air_table['300']
        elif 325 <= T < 375:
            mu = mu_air_table['350']
        elif 375 <= T < 425:
            mu = mu_air_table['400']
        elif 425 <= T <= 450:
            mu = mu_air_table['450']
    if error == 0:
        return ( mu )
    else:
        return ('N/A')

def k_air_calc ( T ):
    "This function returns the thermal conductivity of the air for a given
    temperature"
    if 100 <= T <= 450:
        error = 0
        if 100 <= T < 125:
            k = k_air_table['100']
        elif 125 <= T < 175:
            k = k_air_table['150']
        elif 175 <= T < 225:
            k = k_air_table['200']
        elif 225 <= T < 275:
            k = k_air_table['250']
        elif 275 <= T < 325:
            k = k_air_table['300']
        elif 325 <= T < 375:
            k = k_air_table['350']
        elif 375 <= T < 425:
            k = k_air_table['400']
        elif 425 <= T <= 450:
            k = k_air_table['450']
    if error == 0:
```

-5-

```
        return ( k )
    else:
        return ('N/A')

def alpha_air_calc ( T ):
    "This function returns the thermal diffusivity of air at a given temperature"
    if 100 <= T <= 450:
        error = 0
        if 100 <= T < 125:
            alpha = alpha_air_table['100']
        elif 125 <= T < 175:
            alpha = alpha_air_table['150']
        elif 175 <= T < 225:
            alpha = alpha_air_table['200']
        elif 225 <= T < 275:
            alpha = alpha_air_table['250']
        elif 275 <= T < 325:
            alpha = alpha_air_table['300']
        elif 325 <= T < 375:
            alpha = alpha_air_table['350']
        elif 375 <= T < 425:
            alpha = alpha_air_table['400']
        elif 425 <= T <= 450:
            alpha = alpha_air_table['450']
    if error == 0:
        return ( alpha )
    else:
        return ('N/A')

def T_film ( Ti, Te ):
    "This function calculates the film temperature, to be used for fluid properties"
    return ((Ti + Te)/2)

def Nu_CB ( Re, Pr ):
    "This function calculates the Nusselts number using the Churchill-Bernstein correlation"
    if Re*Pr >= 0.2:
        error = 0
    else:
        error = 1
    if error == 0:
        return 0.3+(0.62*(Re**0.5)*Pr**(1/3)/((1+(0.4/Pr)**(2/3))**(1/4)))*(1+(Re/282000)**(5/8))**(4/5)
    else:
        return ('N/A')
{}
def Nu_Hilpert ( Re, Pr, Corr ):
    if Corr == 'Original':
        if 1 <= Re <= 400000 and Pr >= 0.7:
            error = 0
            if 1 <= Re <= 4:
                C = Hilpert_C['1-4']
                m = Hilpert_m['1-4']
            elif 4 < Re <= 40:
                C = Hilpert_C['4-40']
                m = Hilpert_m['4-40']
            elif 40 < Re <= 4000:
                C = Hilpert_C['40-4000']
                m = Hilpert_m['40-4000']
            elif 4000 < Re <= 40000:
                C = Hilpert_C['4000-40000']
                m = Hilpert_m['4000-40000']
            elif 40000 < Re <= 400000:
                C = Hilpert_C['40000-400000']
                m = Hilpert_m['40000-400000']
        else:
            error = 1
```

-6-

```
        elif Corr == 'UpdatedHilpert':
            if 0.4 <= Re <= 400000 and Pr >= 0.7:
                error = 0
                if 0.4 <= Re <= 4:
                    C = UpdatedHilpert_C['0.4-4']
                    m = UpdatedHilpert_m['0.4-4']
                elif 4 < Re <= 40:
                    C = UpdatedHilpert_C['4-40']
                    m = UpdatedHilpert_m['4-40']
                elif 40 < Re <= 4000:
                    C = UpdatedHilpert_C['40-4000']
                    m = UpdatedHilpert_m['40-4000']
                elif 4000 < Re <= 40000:
                    C = UpdatedHilpert_C['4000-40000']
                    m = UpdatedHilpert_m['4000-40000']
                elif 40000 < Re <= 400000:
                    C = UpdatedHilpert_C['40000-400000']
                    m = UpdatedHilpert_m['40000-400000']
            else:
                error = 1
        elif Corr == 'FandKeswani':
            if 1 <= Re <= 400000 and Pr >= 0.7:
                error = 0
                if 1 <= Re <= 4:
                    C = FandKeswani_C['1-4']
                    m = FandKeswani_m['1-4']
                elif 4 < Re <= 40:
                    C = FandKeswani_C['4-40']
                    m = FandKeswani_m['4-40']
                elif 40 < Re <= 4000:
                    C = FandKeswani_C['40-4000']
                    m = FandKeswani_m['40-4000']
                elif 4000 < Re <= 40000:
                    C = FandKeswani_C['4000-40000']
                    m = FandKeswani_m['4000-40000']
                elif 40000 < Re <= 400000:
                    C = FandKeswani_C['40000-400000']
                    m = FandKeswani_m['40000-400000']
            else:
                error = 1
        elif Corr == 'Morgan':
            if 0.0001 <= Re <= 200000 and Pr >= 0.7:
                error = 0
                if 0.0001 <= Re <= 0.004:
                    C = Morgan_C['0.0001-0.004']
                    m = Morgan_m['0.0001-0.004']
                elif 0.004 < Re <= 0.09:
                    C = Morgan_C['0.04-0.09']
                    m = Morgan_m['0.04-0.09']
                elif 0.09 < Re <= 1:
                    C = Morgan_C['0.09-1']
                    m = Morgan_m['0.09-1']
                elif 1 < Re <= 35:
                    C = Morgan_C['1-35']
                    m = Morgan_m['1-35']
                elif 35 < Re <= 5000:
                    C = Morgan_C['35-5000']
                    m = Morgan_m['35-5000']
                elif 5000 < Re <= 50000:
                    C = Morgan_C['5000-50000']
                    m = Morgan_m['5000-50000']
                elif 50000 < Re <= 200000:
                    C = Morgan_C['50000-200000']
                    m = Morgan_m['50000-200000']
            else:
                error = 1
        if error == 0:
            return (C*(Re**(m))*Pr**(1/3))
```

-7-

```
    else:
        return ('N/A')

def Nu_Zukauskas( Re, Pr, Prs ):
    if 0.7 <= Pr <= 500 and 1 <= Re <= 1000000:
        error = 0
        if Pr < 10:
            n = 0.37
        elif Pr >= 10:
            n = 0.36
        if 1 <= Re <= 40:
            C = Zukauskas_C['1-40']
            m = Zukauskas_m['1-40']
        elif 40 < Re <= 1000:
            C = Zukauskas_C['40-1000']
            m = Zukauskas_m['40-1000']
        elif 1000 < Re <= 200000:
            C = Zukauskas_C['1000-200000']
            m = Zukauskas_m['1000-200000']
        elif 200000 < Re <= 1000000:
            C = Zukauskas_C['200000-1000000']
            m = Zukauskas_m['200000-1000000']
    else: error = 1
    if error == 0:
        return (C*(Re**(m))*Pr**(n)*(Pr/Prs)**(1/4))
    else:
        return ('N/A')

def Nu_Whittaker ( Re, Pr, Ti, Te ):
    mu_b = mu_air_calc(Te)
    mu_s = mu_air_calc(Ti)
    if 1 <= Re <= 100000 and 0.67 <= Pr <= 300:
        error = 0
    else:
        error = 1
    if error == 0:
        return ((0.5*(Re**(1/2))+0.06*(Re**(2/3)))*(Pr**(0.4))*((mu_b/mu_s)**(1/4)))
    else:
        return ('N/A')

def h_conv( Nu, k, D_o ):
    "This function calculates the convective heat transfer co-efficient of an
    external flow over a pipe"
    return ((Nu*k)/D_o)

def U0(h_external, k_pipe, D_i, D_o ):
    print(h_external)
    print(k_pipe)
    print(D_i)
    print(D_o)
    print(((D_o*math.log(D_o/D_i))/(2*k_pipe)))
    print(1/h_external)
    "This function calculates the overall heat transfer co-efficient of an
    uninsulated pipe. h_internal is the internal heat transfer co-efficient,
    h_external is the external heat transfer co-efficient, kp is the thermal
    conductivity of the pipe, ID is the internal diameter of the pipe, OD is the
    outer diameter of the pipe."
    return (1/(((D_o*math.log(D_o/D_i))/(2*k_pipe))+(1/h_external)))

def U1(h_external, k_pipe, k_ins, D_i, D_o, t_ins ):
    D_o_ins = D_o + 2*t_ins
    "This function calculates the overall heat transfer co-efficient of an insulated
    pipe. h_internal is the internal heat transfer co-efficient, h_external is the
    external heat transfer co-efficient, kp is the thermal conductivity of the pipe,
    k_ins is the thermal conductivity of the insulation, ID is the internal diameter
    of the pipe, OD is the outer diameter of the pipe and ti is the thickness of the
    insulation"
    #return
```

-8-

```
        (1/(((D_o_ins*math.log(D_o/D_i))/(2*k_pipe))+((D_o_ins*math.log(D_o_ins/D_o))/(2*k
        _ins))+(1/h_external)))
        return (1/(((D_o_ins*math.log(D_o_ins/D_o))/(2*k_ins))+(1/h_external)))

    def U2(h_external, k_pipe, k_ins, k_ice, D_i, D_o, t_ins, t_ice ):
        D_o_ins = D_o + 2*t_ins
        D_o_ins_ice = D_o_ins + 2*t_ice
        "This function calculates the overall heat transfer co-efficient of an insulated
        pipe with ice. h_internal is the internal heat transfer co-efficient, h_external
        is the external heat transfer co-efficient, kp is the thermal conductivity of
        the pipe, k_ins is the thermal conductivity of the insulation, k_ice is the
        thermal conductivity of ice, ID is the internal diameter of the pipe, OD is the
        outer diameter of the pipe, ti is the thickness of the insulation and tice is
        the thickness of the ice glazing."
        return
        (1/(((D_o_ins_ice*math.log(D_o/D_i))/(2*k_pipe))+((D_o_ins_ice*math.log(D_o_ins/D_
        o))/(2*k_ins))+((D_o_ins_ice*math.log(D_o_ins_ice/D_o_ins))/(2*k_ice))+(1/h_extern
        al)))

    # def ql( U, A, Ti, Te ):
        # "This function calculates the heat loss, or heat flux of a pipe in W/m"
        # return (U*A*(Ti-Te))

    # def tc( Ml, Ti, Te, ql ):
        # "This function calculates the required time (in seconds) to cool water inside
        a unit length of pipe to freezing temperature (0degC)."
        # return ((cp_w*Ml*(Ti-Tf_w))/ql)

    # def tf_w( Ml, ql ):
        # "This function calculates the required time (in seconds) to freeze water
        inside a unit length of pipe."
        # return ((hfs_w*Ml)/ql)

    def TimeToFreeze(h_external, rho_s, rho_l, Hf, c_s, c_l, k_s, Ti, Tamb, Tf, Tc, D, L):
        "This function calculates the time to freeze a cylinder filled with a liquid"
        h_l = Hf + (Ti-Tf)*c_l
        h_s = (Tf-Tc)*c_s
        deltaH = (rho_l*h_l)-(rho_s*h_s)
        Cs = (rho_s*c_s)
        Cl = (rho_l*c_l)
        Beta = (L/D)
        Bi = ((h_external*D)/k_s)
        Pk = ((Cl*(Ti-Tf))/deltaH)
        Ste = ((Cs*(Tf-Tamb))/deltaH)
        deltaT = ((Tf-Tamb)+((((Ti-Tf)**2)*(Cl/2)-(((Tf-Tc)**2)*(Cs/2)))/deltaH))
        U = (deltaT/(Tf-Tamb))
        P = (0.7306-(1.083*Pk)+Ste*((15.4*U)-15.43+(0.01329*(Ste/Bi))))
        R = (0.2079-0.2656*U*Ste)
        theta = (((deltaH*10**3)/deltaT)*(((P*D)/h_external)+((R*(D**2))/k_s)))
        phi = (2.32/(Beta**1.77))
        X = (phi/((Bi**1.34)+phi))
        E2 = ((X/Beta)+((1-X)*(0.5/(Beta**3.69))))
        E = (2+E2)
        theta_shape = ((theta/E)/3600)
        return (theta_shape)


    # Prepare spreadsheet for results
    workbook = xlsxwriter.Workbook('Results.xlsx')
    worksheet = workbook.add_worksheet()
    bold = workbook.add_format({'bold': 1})
    merge_format = workbook.add_format({
        'bold': 1,
        'align': 'center',
        'valign': 'vcenter'})
    #worksheet.set_column(1, 1, 15)
    worksheet.write(2, 0, 'Pipe OD', bold)
    worksheet.write(2, 1, 't_ins', bold)
```

-9-

```
worksheet.write(2, 2, 'V_infty', bold)
worksheet.write(2, 3, 'TiC', bold)
worksheet.write(2, 4, 'TeC', bold)
worksheet.write(2, 5, 'Re', bold)
for g in range(0,3):
    g = 0
    label = ['Nu', 'h', 'U', 'ttf, h']
    for h in range(6,34):
        worksheet.write(2, h, label[g], bold)
        if g < 3:
            g += 1
        elif g == 3:
            g = 0

# Merge headers
worksheet.merge_range(1, 0, 1, 5, 'Common', merge_format)
worksheet.merge_range(1, 6, 1, 9, 'Hilpert Correlation', merge_format)
worksheet.merge_range(1, 10, 1, 13, 'Updated Hilpert', merge_format)
worksheet.merge_range(1, 14, 1, 17, 'Fand & Keswani', merge_format)
worksheet.merge_range(1, 18, 1, 21, 'Morgan', merge_format)
worksheet.merge_range(1, 22, 1, 25, 'Zukauskas', merge_format)
worksheet.merge_range(1, 26, 1, 29, 'Whitaker', merge_format)
worksheet.merge_range(1, 30, 1, 33, 'Churchill-Bernstein', merge_format)


counter_row = 3
counter_column = 0
# Starting main calculation loop

# Calculating fixed variables
# Ml_temp = Ml(D_i, rho_w)
# D_o_ice = D_o+2*t_ice
# D_o_ins = D_o+2*t_ins
# D_o_ins_ice = D_o_ins+2*t_ice

for i in range(len(V_infty)):
    print('Calculating for a wind speed of', V_infty[i], ' m/s')
    for j in range(len(TiC)):
        Ti = TiC[j]+273.15
        print('Calculating for an internal temperature of', Ti, ' degC')
        for k in range(len(TeC)):
            Te = TeC[k]+273.15
            T_film_temp = T_film(Ti, Te)
            # Pr_air_temp = Pr_air_calc( nu_calc(mu_air_calc(T_film(Ti, Te))),alpha_air_calc(T_film(Ti, Te)))
            Pr_air_inf = Pr_air_calc(Te)
            Pr_air_film = Pr_air_calc(T_film_temp)
            Pr_air_surf = Pr_air_calc(Ti)
            print('Calculating for an external temperature of ', Te, ' degC')
            for l in range(len(D_tab)):
                print('Calculating for an Pipe OD of ', D_tab[l], ' m')
                for m in range(len(t_ins)):
                    print('Calculating for a insulation thickness of ', t_ins[m], ' m')
                    D_o = D_tab[l]+2*t_ins[m]
                    D_i = D_tab[l]-2*t_w
                    Re_temp = Re(V_infty[i], D_o, Rho_air_calc(T_film_temp), mu_air_calc(T_film_temp))
                    Re_amb = Re(V_infty[i], D_o, Rho_air_calc(Te), mu_air_calc(Te))
                    worksheet.write(counter_row, counter_column, D_tab[l])
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, t_ins[m])
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, V_infty[i])
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, TiC[j])
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, TeC[k])
```

-10-

```
counter_column += 1
worksheet.write(counter_row, counter_column, Re_temp)
counter_column += 1
for n in range(0,4):
    # Calculate heat loss for uninsulated pipe using Hilpert
    Corr = ['Original', 'UpdatedHilpert', 'FandKeswani', 'Morgan']
    print('Calculating Hilpert', Corr[n])
    Nu_temp = Nu_Hilpert(Re_temp, Pr_air_film, Corr[n])
    if Nu_temp == 'N/A':
        h_temp = 'N/A'
        U_temp = 'N/A'
        ttf = 'N/A'
    else:
        k_air_temp = k_air_calc (T_film_temp)
        h_temp = h_conv(Nu_temp, k_air_temp, D_o)
        if t_ins[m] == 0:
            U_temp = U0(h_temp, k_pipe, D_i, D_tab[l])
        elif t_ins[m] > 0:
            U_temp = U1(h_temp, k_pipe, k_ins, D_i, D_tab[l], t_ins[m])
        ttf = TimeToFreeze(U_temp, rho_ice, rho_w, hfs_w, cp_ice, cp_w_calc(Ti), k_ice, Ti, Te, Tf_w, Tc, D_i, 1)
    worksheet.write(counter_row, counter_column, Nu_temp) # Write Nusselts number to spreadsheet
    counter_column += 1
    worksheet.write(counter_row, counter_column, h_temp) # Write convective heat transfer co-efficient to spreadsheet
    counter_column += 1
    worksheet.write(counter_row, counter_column, U_temp) # Write overall heat transfer co-efficient to spreadsheet
    counter_column += 1
    worksheet.write(counter_row, counter_column, ttf) # Write time to freeze to spreadsheet
    counter_column += 1
# Calculate heat loss for uninsulated pipe using Zukauskas
print('Calculating Zukauskas')
Nu_temp = Nu_Zukauskas(Re_amb, Pr_air_inf, Pr_air_surf)
if Nu_temp == 'N/A':
    h_temp = 'N/A'
    U_temp = 'N/A'
    ttf = 'N/A'
else:
    k_air_temp = k_air_calc (T_film_temp)
    h_temp = h_conv(Nu_temp, k_air_temp, D_o)
    if t_ins[m] == 0:
        U_temp = U0(h_temp, k_pipe, D_i, D_tab[l])
    elif t_ins[m] > 0:
        U_temp = U1(h_temp, k_pipe, k_ins, D_i, D_tab[l], t_ins[m])
    ttf = TimeToFreeze(U_temp, rho_ice, rho_w, hfs_w, cp_ice, cp_w_calc(Ti), k_ice, Ti, Te, Tf_w, Tc, D_i, 1)
worksheet.write(counter_row, counter_column, Nu_temp) # Write Nusselts number to spreadsheet
counter_column += 1
worksheet.write(counter_row, counter_column, h_temp) # Write convective heat transfer co-efficient to spreadsheet
counter_column += 1
worksheet.write(counter_row, counter_column, U_temp) # Write overall heat transfer co-efficient to spreadsheet
counter_column += 1
worksheet.write(counter_row, counter_column, ttf) # Write time to freeze to spreadsheet
counter_column += 1
#Calculate heat loss using Whitaker
print('Calculating Whitaker')
Nu_temp = Nu_Whittaker(Re_temp, Pr_air_film, Ti, Te)
if Nu_temp == 'N/A':
    h_temp = 'N/A'
```

-11-

```
                        U_temp = 'N/A'
                        ttf = 'N/A'
                    else:
                        k_air_temp = k_air_calc (T_film_temp)
                        h_temp = h_conv(Nu_temp, k_air_temp, D_o)
                        if t_ins[m] == 0:
                            U_temp = U0(h_temp, k_pipe, D_i, D_tab[l])
                        elif t_ins[m] > 0:
                            U_temp = U1(h_temp, k_pipe, k_ins, D_i, D_tab[l], t_ins[m])
                        ttf = TimeToFreeze(U_temp, rho_ice, rho_w, hfs_w, cp_ice, cp_w_calc(Ti), k_ice, Ti, Te, Tf_w, Tc, D_i, 1)
                    worksheet.write(counter_row, counter_column, Nu_temp) # Write Nusselts number to spreadsheet
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, h_temp) # Write convective heat transfer co-efficient to spreadsheet
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, U_temp) # Write overall heat transfer co-efficient to spreadsheet
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, ttf) # Write time to freeze to spreadsheet
                    counter_column += 1
                    # Calculate heat loss using Churchill-Bernstein
                    print('Calculating Churchill-Bernstein')
                    Nu_temp = Nu_CB(Re_temp, Pr_air_film)
                    if Nu_temp == 'N/A':
                        h_temp = 'N/A'
                        U_temp = 'N/A'
                        ttf = 'N/A'
                    else:
                        k_air_temp = k_air_calc (T_film_temp)
                        h_temp = h_conv(Nu_temp, k_air_temp, D_o)
                        if t_ins[m] == 0:
                            U_temp = U0(h_temp, k_pipe, D_i, D_tab[l])
                        elif t_ins[m] > 0:
                            U_temp = U1(h_temp, k_pipe, k_ins, D_i, D_tab[l], t_ins[m])
                        ttf = TimeToFreeze(U_temp, rho_ice, rho_w, hfs_w, cp_ice, cp_w_calc(Ti), k_ice, Ti, Te, Tf_w, Tc, D_i, 1)
                    worksheet.write(counter_row, counter_column, Nu_temp) # Write Nusselts number to spreadsheet
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, h_temp) # Write convective heat transfer co-efficient to spreadsheet
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, U_temp) # Write overall heat transfer co-efficient to spreadsheet
                    counter_column += 1
                    worksheet.write(counter_row, counter_column, ttf) # Write time to freeze to spreadsheet
                    counter_column += 1
                    counter_row += 1
                    counter_column = 0
workbook.close()
```

Appendix B – Arduino Code used for Temperature Measurement

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```
// Code for temperature, humidity and wind speed logging
// Written by Bjarte Odin Kvamme

#include "DHT.h" // Load library for the DHT22 Temperature/Humidity sensor
#include <OneWire.h> // Load library for the OneWire protocol
#include <DallasTemperature.h> // Load library for the Maxim/Dallas D18B20 digital temperature sensor
#include <SPI.h> // Load library for the SPI bus, used for accessing the SD card
#include <SD.h> // Load library for interaction with the SD Card
#include <Wire.h> // Load library for interfacing with the RTC sensor
#include "RTClib.h" // Load library for the RTC module

// Define constants for use with the RTC module
RTC_DS1307 RTC;

#define LOG_I 30000 // Define how many milliseconds between grabbing the data and logging it
#define SYNC_I 30000 // Define how often the data should be written to the SD card. Set as the same as LOG_I to write data as soon as it is logged
uint32_t syncTime = 0; // time of last sync()
#define E2S 0 //Toggle whether data should be echoed to the serial port for real time monitoring on a computer
#define L2S 1
#define W2S 0 //Choose whether the Arduino should wait for input in the serial console before starting the logger

// PIN CONFIGURATION
#define LED1 4 //Pin the green LED is connected to
#define LED2 5 // Pin the red LED is connected to
#define DHT_P 2 //Pin the ambient temperature/humidity sensor is connected to
#define OW_P 3 //Pin the D18B20 digital temperature sensors is connected to
int W_P = 0; // Analog pin the Wind Speed sensor is connected to

// Define constants for use with the Dallas temperature sensor
#define TEMP_PRE 12 // Define resolution used for the temperature logging
// Setup a oneWire instance to communicate with any OneWire devices (not just Maxim/Dallas temperature ICs)
OneWire oneWire(OW_P);
// Pass our oneWire reference to Dallas Temperature.
DallasTemperature sensors(&oneWire);
int DevCnt; // Number of temperature devices found
DeviceAddress tmpDevAdd; // Temporary variable for store a device address

// Define constants for the DHT22 digital temperature/humidity sensor
#define DHTTYPE DHT22 // Sensor model
DHT dht(DHT_P, DHTTYPE);

// Define constants for the wind speed measurements
int WVAL = 0;
float WVOLT = 0;
float WSPEED = 0;

File lf;
```

-1-

```

// Define the chip select pin for the SD card
const int cS = 10;

// Error handling code. Will stop the logger and light the red LED to indicate an error.
void err (const char * s) {
  Serial.print("Error: ");
  Serial.println(s);
  // activate the red LED to indicate error
  digitalWrite(LED2, HIGH);
  while(1);
}

// function to print the temperature for a device
void prtTem(DeviceAddress devAdd) {
  float tempC = sensors.getTempC(devAdd);
  Serial.print(tempC);
}

// function to print a device address
void prtAdd(DeviceAddress devAdd) {
  for (uint8_t i = 0; i < 8; i++) {
    if (devAdd[i] < 16) Serial.print( F("0"));
    Serial.print(devAdd[i], HEX);
  }
}

// function to log the temperature for a device
void logTem(DeviceAddress devAdd) {
  float tempC = sensors.getTempC(devAdd);
  lf.print(tempC);
}

// function to log a device address
void logAdd(DeviceAddress devAdd) {
  for (uint8_t i = 0; i < 8; i++) {
    if (devAdd[i] < 16) lf.print( F("0"));
    lf.print(devAdd[i], HEX);
  }
}
void setup() {
  Serial.begin(9600);
  Serial.println();
  pinMode(LED2, OUTPUT); //Set the red LED pin to output
  pinMode(LED1, OUTPUT); //Set the green LED pin to output

  //Check if we should stop and await character from the serial console
  #if W2S
    Serial.println( F("Type any character to start")) ;
    while (!Serial.available());
  #endif //W2S

  // Activate both LEDs and wait for 15 seconds to allow the arduino to settle
```

```
  #if E2S
    Serial.println( F("Waiting for Arduino to settle. Please wait..."));
  #endif //E2S
  digitalWrite(LED1, HIGH);
  digitalWrite(LED2, HIGH);
  delay(5000); //Wait for Arduino to settle before initializing memory card.
  // Deactivate the LEDs
  digitalWrite(LED1, LOW);
  digitalWrite(LED2, LOW);
  Serial.println();
  //check if the SD card is present and can be initialized
  #if E2S
    Serial.print( F("Initializing SD card... "));
  #endif //E2S
  pinMode(cS, OUTPUT); // Set the pin used for the SD card to output
  if (!SD.begin(cS)) {
    err("Card failed or is not present!");
  }
  #if E2S
    Serial.println( F("SD card initialized."));
  #endif //E2S

  //Create a new file to use for logging data
  char fn[] = "LOGGER00.CSV";
  for (uint8_t i = 0; i < 100; i++) {
    fn[6] = i/10 + '0';
    fn[7] = i%10 + '0';
    if (!SD.exists(fn)) {
      //Only open a new file if it does not already exist
      lf = SD.open(fn, FILE_WRITE);
      break; // Leave the loop
    }
  }

  if (! lf) {
    err( "Could not create file on SD card.");
  }

  //Connect to the RTC module
  Wire.begin();
  if (! RTC.isrunning()) {
    Serial.println( F("RTC is NOT running!"));
  }
  if (!RTC.begin()) {
    lf.println( F("RTC failed!"));
    err("RTC failed!");
    #if E2S
      Serial.println( F("RTC failed!"));
    #endif //E2S
  }
  // to re-adjust the RTC clock, uncomment the line below.
 // RTC.adjust(DateTime(__DATE__, __TIME__));

```

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```

   // Log information in lf
   lf.println(
   F("millis,stamp,Date-Time,AmbientT,AmbientH,WindSensorVolt,WindSensorSpeed,Sensor1,Sens
   or2,Sensor3,Sensor4,Sensor5,Sensor6,Sensor7,Sensor8,Sensor9,Sensor10,Sensor11,Sensor12,
   Sensor13,Sensor14,Sensor15,Sensor16,Sensor17,Sensor18"));

  //Start DHT sensor
  dht.begin();
    #if E2S
     Serial.print( F("Logging data to: "));
     Serial.println(fn);
     Serial.println( F("millis,stamp"));
    #endif //E2S
    #if L2S
     Serial.print(
     F("Date-Time,AmbientT,AmbientH,WindSensorVolt,WindSensorSpeed,Sensor1,Sensor2,Sensor3
     ,Sensor4,Sensor5,Sensor6,Sensor7,Sensor8,Sensor9,Sensor10,Sensor11,Sensor12,Sensor13,
     Sensor14,Sensor15,Sensor16,Sensor17,Sensor18"));
     Serial.println();
   #endif //E2S
   float ambh = dht.readHumidity();
   float ambt = dht.readTemperature();
   if (isnan(ambh) || isnan(ambt)) {
     err("Failed to read from DHT sensor!");
     return;
   }

  //Setup D18B20 temperature sensors
  sensors.begin();
  DevCnt = sensors.getDeviceCount();
  #if E2S
   Serial.print( F("Locating D18B20 devices on bus... "));
  #endif //E2S
  if (DevCnt > 0) {
   #if E2S
     Serial.print( F("Found "));
     Serial.print(DevCnt, DEC);
     Serial.print( F(" devices."));
     Serial.println();
   #endif //E2S

   // Log serial numbers of the temperature sensors to the CSV file for future reference.
   lf.print( F("SERIAL ,NUMBERS ,FOR ,SENSORS ,FOLLOWS ,"));
   #if L2S
     Serial.print( F("SERIAL NUMBERS ,FOR SENSORS ,FOLLOWS ,"));
   #endif //L2S
   for (int i=0;i<DevCnt; i++) {
     if (sensors.getAddress(tmpDevAdd, i)) {
       logAdd(tmpDevAdd);
       lf.print(F(","));
       sensors.setResolution(tmpDevAdd, TEMP_PRE);
       #if L2S
```

-4-

```
          prtAdd(tmpDevAdd);
          Serial.print( F(","));
        #endif //L2S
        #if E2S
        Serial.print( F("Found device "));
        Serial.print(i, DEC);
        Serial.print( F(" with address: "));
        prtAdd(tmpDevAdd);
        Serial.println();
        Serial.print( F("Setting resolution to "));
        Serial.println(TEMP_PRE, DEC);
        Serial.print( F("Confirmed sensor resolution: "));
        Serial.print(sensors.getResolution(tmpDevAdd), DEC);
        Serial.println();
        #endif //E2S
        }
        else {
          Serial.print( F("Found ghost device at "));
          Serial.print(i, DEC);
          Serial.print( F(" but could not detect address. Check power and wires"));
        }
     }
    lf.println();
    #if L2S
      Serial.println();
    #endif //L2S
    }
   else {
    err("Did not find any temperature sensors, check the connections.");
   }


  }

  // Start logging loop
  void loop() {
    int cd = 0;
    while (LOG_I-767 > cd) {
      digitalWrite(LED1, HIGH);
      delay(250);
      digitalWrite(LED1, LOW);
      delay(250);
      cd = cd + 500;
    }
    //Delay for the logging interval
    //delay((LOG_I -1) - (millis() % LOG_I));
    DateTime now = RTC.now();
    digitalWrite(LED1, HIGH); //activate the green LED to indicate that logging is active
    // log milliseconds seens starting
    uint32_t m = millis();
    lf.print(m);
    lf.print(F(","));
    #if E2S
```

```
  Serial.print(m);        // milliseconds since start
  Serial.print(F(","));
#endif E2S

//Fetch the time
now = RTC.now();
// log time
lf.print(now.unixtime()); // seconds since 1/1/1970
lf.print(F(","));
lf.print('"');
lf.print(now.year(), DEC);
lf.print(F("/"));
lf.print(now.month(), DEC);
lf.print(F("/"));
lf.print(now.day(), DEC);
lf.print(F(" "));
lf.print(now.hour(), DEC);
lf.print(F(":"));
lf.print(now.minute(), DEC);
lf.print(F(":"));
lf.print(now.second(), DEC);
lf.print('"');
#if E2S
  Serial.print(now.unixtime()); // seconds since 1/1/1970
  Serial.print(F(","));
#endif //E2S
#if L2S
  Serial.print('"');
  Serial.print(now.year(), DEC);
  Serial.print(F("/"));
  Serial.print(now.month(), DEC);
  Serial.print(F("/"));
  Serial.print(now.day(), DEC);
  Serial.print(F(" "));
  Serial.print(now.hour(), DEC);
  Serial.print(F(":"));
  Serial.print(now.minute(), DEC);
  Serial.print(F(":"));
  Serial.print(now.second(), DEC);
  Serial.print('"');
#endif //L2S
// Read ambient temperature and humidity from the DHT22
// Reading temperature or humidity takes about 250 milliseconds!
// Sensor readings may also be up to 2 seconds 'old' (its a very slow sensor)
float ambh = dht.readHumidity();
// Read temperature as Celsius (the default)
float ambt = dht.readTemperature();
// Check if any reads failed and exit early (to try again).
if (isnan(ambh) || isnan(ambt)) {
  // err("Failed to read from DHT sensor!");
  return;
}
lf.print(F(","));
```

```
    lf.print(ambt);
    lf.print(F(","));
    lf.print(ambh);
    #if L2S
      Serial.print(F(","));
      Serial.print(ambt);
      Serial.print(F(","));
      Serial.print(ambh);
    #endif //L2S

      // Record wind speed
   WVAL = analogRead(W_P);
   if (WVAL > 0) {
     WVOLT = 0.005 + (WVAL * 2.5 * 0.004873046875);
   }
   else {
     WVOLT = (WVAL * 2.5 * 0.004873046875);
   }

   if (WVAL > 0) {
     WSPEED = 0.9 + (WVOLT * 4.2806);
   }
   else {
     WSPEED = 0;
   }
    lf.print(",");
    lf.print(WVOLT);
    lf.print(",");
    lf.print(WSPEED);
    #if L2S
      Serial.print(",");
      Serial.print(WVOLT);
      Serial.print(",");
      Serial.print(WSPEED);
    #endif //L2S

    // Read data from the D18B20 temperature sensors
    //Serial.print( F("Requesting temperatures from D18B20 devices... "));
    sensors.requestTemperatures(); // Send command to get temperatures
    //Serial.println( F("DONE"));
    // Loop through each device, print out temperature data
    for(int i=0;i<DevCnt; i++) {
      // Search the wire for address
      if(sensors.getAddress(tmpDevAdd, i)) {
      // Output the device ID
      lf.print(F(","));
      logTem(tmpDevAdd);
      #if L2S
        Serial.print(F(","));
        prtTem(tmpDevAdd);
      #endif L2S
    }
    //else ghost device! Check your power requirements and cabling
```

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```
    }
    lf.println();
    #if L2S
      Serial.println();
    #endif //L2S

    digitalWrite(LED1, LOW);

    // Write data to SD card
    if ((millis() - syncTime) < SYNC_I) return;
    syncTime = millis();
    //flash LED to show that the data is written to the SD card
    digitalWrite(LED2, HIGH);
    lf.flush();
    digitalWrite(LED2, LOW);
  }

```

-8-

Appendix C- Pipe Material Details

TIBNOR

Tibnor AS

E-post: firmapost@tibnor.com
Internet: www.tibnor.no

****

O R D R E B E K R E F T E L S E

****

Postadresse:
UNIVERSITETET I STAVANGER

POSTBOKS 384 ALNABRU
0614 OSLO

Vareadresse:
UNIVERSITETET I STAVANGER
INNGANG VEST - VERSKTEDET
KRISTINE BONNEVIESVEI 22
4036 STAVANGER

Sidenr: 01
Ordrenummer: 518733
Ordredato: 28.01.2016
Lev.dato: 1.02.2016
Salgsavd. Bryne
Kunde-nr.: 926367
Vår referanse: Sigve M. Olsen
Deres ref.: AMITH, YAASEEN
Fri-periode: 00
Kredittdager: 030
Leveringsbetingelse:
CPT adresse + frakt
Leveringsmåte:
Bil

Kommentarer/merknader

Vi takker for Deres ordre av 28012016 og bekrefter for levering som følger:

Bestilt Mengde	Enh	Varetekst	Lengde	Vekt	Pris pr. prisenh.	Beløp
		Runde Pres.Stålrør DIN 2394				
		-------------------------------m/pris pr.			M	
1,00	LGD	25,0 X 2,00 MM	6,000	6,78	19,83	118,98
1,00	LGD	50,0 X 2,00 MM	6,000	14,22	37,32	223,[illegible]
			Totalvekt:	21,00	**Ordretotal**	342,90

Alle priser er ekskl. M.V.A.
Eventuelle Fraktandel / Emballasje/Skrapjernstillegg kommer i tillegg.
Ordren bekreftes i h.h.til våre salgsbetingelser på siste side i lagerprogrammet.

Med vennlig hilsen
TIBNOR AS

Sigve M. Olsen

Appendix D- Insulation Material Details

Tekniske data - AF/Armaflex N

Kort beskrivelse	Høyfleksibel cellegummiisolasjon med lukket cellestruktur, med høy diffusjonsmotstand og lav varmeledningsevne, har innebygget Microban® antimikrobiell beskyttelse.
Materialtype	Cellegummiisolasjon basert på syntetisk gummi (elastomer). Fabrikkprodusert fleksibel cellegummi (FEF) iht. NS-EN 14304.
Spesiell materialinformasjon	Selvklebende tape: trykkfølsom tape på en modifisert akrylatbase med nettstruktur. Dekket med polyetylenfolie. Det kan finnes spor av silikon på dekkpapiret/-folien som er brukt for å beskytte den selvklebende tapen.
Bruksområde	Isolering/beskyttelse av rør, luftkanaler, beholdere (inkl. bend, fittings, flenser, ventiler etc.) innen VVS, air-condition, kjøling og prosessindustri for å hindre kondens og for å spare energi. Reduksjon av strukturstøy i installasjoner for sanitær- og avløpsvann.
Spesielle funksjoner	Slangene har en økende isolasjonstykkelse med økende rørdiameter. Dette sikrer at overflatetemperaturen holdes lik uansett rørdiameter.
Anmerkninger	Ytelseserklæring (Certificate of Conformity) nr. 0550 og 0551 fra GSH (Güteschutzgemeinschaft Hartschaum e.V.), approved Inspection and Certification Body in Germany (No. ÜG049)

Egenskaper	Verdi / vurdering	Test*3	Overvåk kontroll	Spesielle merknader
Temperaturområde				
Temperaturområder	maks. driftstemperatur +110 °C (+85 °C hvis plater eller tape hellimes til objektet (undelaget).) min. driftstemperatur[1] -50 °C	EU 5621 EU 6228		Testet iht. NS-EN 14706, NS-EN 14707 og NS-EN 14304
Varmeledningsevne				
Varmeledningsevne	ϑ_m +/-0 °C λ= Slanger (serie AF-1 til AF-4) λ ≤ 0,033 W/(m · K) [33 + 0,1 · ϑ_m + 0,0008 · ϑ_m^2]/1000 Slanger (serie AF-5) λ ≤ 0,036 W/(m · K) [36 + 0,1 · ϑ_m + 0,0008 · ϑ_m^2]/1000 Plater, tape (AF-03MM til AF-32MM) λ ≤ 0,033 W/(m · K) [33 + 0,1 · ϑ_m + 0,0008 · ϑ_m^2]/1000 Plater (AF-50MM λ ≤ 0,036 W/(m · K) [36 + 0,1 · ϑ_m + 0,0008 · ϑ_m^2]/1000	EU 5621 EU 6228	○	Testet iht. NS-EN 12667 NS-EN ISO 8497
Diffusjonsmotstand				
Relativ fuktmotstand	Plater (AF-03MM til AF-32MM) og slanger (serie F-1 til AF-4) μ ≥ 10.000 Plater (AF-50MM) og slanger (serie AF-5) μ ≥ 7.000	EU 5621 EU 6228	○	Testet ith. NS-EN 12086 NS-EN 13469
Brannegenskaper				
Brannklasse[2]	slanger B_L-s3,d0 plater B-s3,d0 tape B-s3,d0	EU 5621 EU 6228	○	Klassifisert iht. NS-EN 13501-1 Testet iht. NS-EN 13823 NS-EN ISO 11925-2
Brannegenskaper i praksis	Selvslukkende, drypper ikke, ingen flammespredning			
Andre tekniske egenskaper				
Dimensjoner og toleranser	I samsvar med EN 14304, tabell 1	EU 5621 EU 6228		Testet iht. NS-EN 822, NS-EN 823, NS-EN 13467
UV-motstand	Må beskyttes må UV-stråling.Se ansvarsfraskrivelse.			
Lagring og lagringstid	Selvklebende tape, selvklebende plater, selvklebende slanger: 1 år	Kan lagres i tørre, rene rom med normal fuktighet (50 % til 70 %) og omgivelsestе (0 °C–35 °C).		Lagres i tørre og rene rom ved normal relativ fuktighet (50% til 70%) og omgivelsetemperatur (0 °C – 35 °C).
Antimikrobielle egenskaper	Innebygd Microban® antimikrobiell beskyttelse: Soppvekst ikke funnet			

1. Ved temperaturer under -50 °C, vennligst kontakt vår salgsrepresentant for vår tekniske informasjon.
2. Brannklassen gjelder ved metalloverflater.

*1 Flere dokumenter som testsertifikater, godkjenninger o.l. kan bestilles ved å henvise til oppgitte registreringsnr.

*2 ●: Offisiell ekstern overvåking hos uavhengige forskningsinstitutt og/eller testmyndigheter
○: Egen kvalitetskontroll i fabrikken

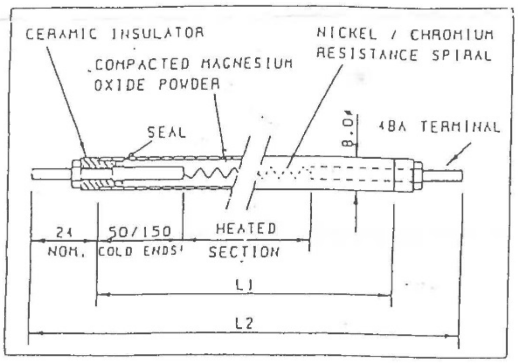
Alle tekniske data og tekniske informasjoner er basert på bruksresultater oppnådd under typiske driftsforhold. I egen interesse bør alle som mottar disse data og opplysninger, avklare med oss i god tid om dette også passer til den ønskede anvendelse som bruker har planlagt. Montasjeveiledning finnes i vår Armaflex montasjemanual. Ikke egnet for utendørsbruk. Armaflex bør beskyttes innen 3 dager, f.eks. med Armafinish maling eller Arma-Chek mantling. Før isolering av rustfritt stål, utføres, vennligst kontakt vår salgsrepresentant for mer informasjon. Noen kuldemedier kan ha en utgangstemperatur på over +110 °C, kontakt vår salgsrepresentant for mer informasjon.

24

Appendix E - Heating Element Details

STRAIGHT LENGTH ELEMENTS

CONSTRUCTION



Terminal: Mild Steel Threaded 4BA

Element Sheath: Incoloy 800 - Nickel/ Chromium alloy for high temperatures up to 800°C.

Seal: Silicone rubber - maximum operating temperature 250°C.

INSTALLATION:

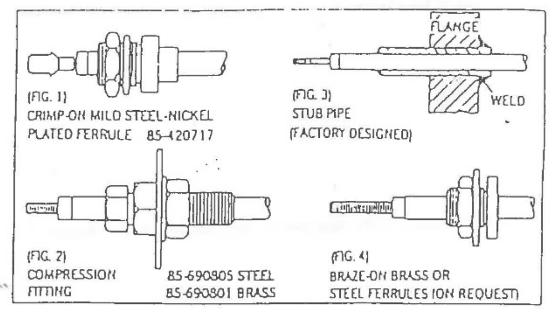
The RS range of straight length elements is suitable for a very wide range of applications. They can be formed into complex shapes with the minimum of tools and can be incorporated into your plant with the minimum of fuss.

All design and installation work must be carried out by competent persons to comply with the Health & Safety At Work Act. Particular attention must be paid to preventing contact of persons with element sheath or electrically live terminal pins.

FORMING:

The element may be bent cold around a roller of minimum radius 13mm. Ensure when forming that the terminal pin end does not fall within the arc of bending.

MOUNTING METHODS



Typical mounting methods are illustrated. Note that brazing, soldering or welding the elements may [illegible] the seal [illegible]. Elements must then be dried out and resealed

RANGE

Stock No	Rating (W) 230/240V	L1 Nominal (mm)	L2 Maximum (mm)
200-1229	918/1000	1372	1444
200-1235	918/1000	1524	1599
200-1241	1837/2000	2134	2215
200-1257	1378/1500	2440	2525
200-1263	1837/2000	2440	2525
200-1279	2296/2500	2440	2525
200-1285	2755/3000	2440	2525
200-1291	2112/2300	2440	2525

All units are packed in 3's.

Statement of conformance with European Harmonised Directives.

Straight length elements are manufactured to BS 7351, and will comply with the Directives if fitted to correctly designed equipments. Electrical design should satisfy BS EN 60335

If automatic switching is utilised design must satisfy the requirements of EN60555-3.

Appendix F- Temperature Sensor DS18B20

DS18B20

Programmable Resolution 1-Wire Digital Thermometer

General Description

The DS18B20 digital thermometer provides 9-bit to 12-bit Celsius temperature measurements and has an alarm function with nonvolatile user-programmable upper and lower trigger points. The DS18B20 communicates over a 1-Wire bus that by definition requires only one data line (and ground) for communication with a central microprocessor. In addition, the DS18B20 can derive power directly from the data line ("parasite power"), eliminating the need for an external power supply.

Each DS18B20 has a unique 64-bit serial code, which allows multiple DS18B20s to function on the same 1-Wire bus. Thus, it is simple to use one microprocessor to control many DS18B20s distributed over a large area. Applications that can benefit from this feature include HVAC environmental controls, temperature monitoring systems inside buildings, equipment, or machinery, and process monitoring and control systems.

Applications

- Thermostatic Controls
- Industrial Systems
- Consumer Products
- Thermometers
- Thermally Sensitive Systems

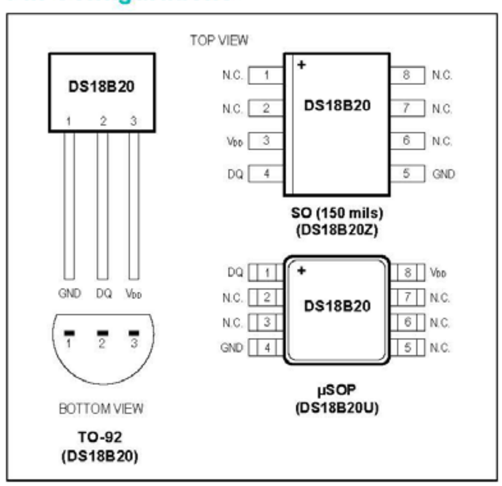
Ordering Information ***appears at end of data sheet.***

1-Wire is a registered trademark of Maxim Integrated Products, Inc.

Benefits and Features

- Unique 1-Wire® Interface Requires Only One Port Pin for Communication
- Reduce Component Count with Integrated Temperature Sensor and EEPROM
  - Measures Temperatures from -55°C to +125°C (-67°F to +257°F)
  - ±0.5°C Accuracy from -10°C to +85°C
  - Programmable Resolution from 9 Bits to 12 Bits
  - No External Components Required
- Parasitic Power Mode Requires Only 2 Pins for Operation (DQ and GND)
- Simplifies Distributed Temperature-Sensing Applications with Multidrop Capability
  - Each Device Has a Unique 64-Bit Serial Code Stored in On-Board ROM
- Flexible User-Definable Nonvolatile (NV) Alarm Settings with Alarm Search Command Identifies Devices with Temperatures Outside Programmed Limits
- Available in 8-Pin SO (150 mils), 8-Pin µSOP, and 3-Pin TO-92 Packages

Pin Configurations



19-7487; Rev 4; 1/15

Absolute Maximum Ratings

Voltage Range on Any Pin Relative to Ground....-0.5V to +6.0V
Operating Temperature Range......................... -55°C to +125°C
Storage Temperature Range............................ -55°C to +125°C
Solder Temperature...............................Refer to the IPC/JEDEC J-STD-020 Specification.

These are stress ratings only and functional operation of the device at these or any other conditions above those indicated in the operation sections of this specification is not implied. Exposure to absolute maximum rating conditions for extended periods of time may affect reliability.

DC Electrical Characteristics

(-55°C to +125°C; V_{DD} = 3.0V to 5.5V)

PARAMETER	SYMBOL	CONDITIONS		MIN	TYP	MAX	UNITS
Supply Voltage	V_{DD}	Local power (Note 1)		+3.0		+5.5	V
Pullup Supply Voltage	V_{PU}	Parasite power	(Notes 1, 2)	+3.0		+5.5	V
		Local power		+3.0		V_{DD}	
Thermometer Error	t_{ERR}	-10°C to +85°C	(Note 3)			±0.5	°C
		-55°C to +125°C				±2	
Input Logic-Low	V_{IL}	(Notes 1, 4, 5)		-0.3		+0.8	V
Input Logic-High	V_{IH}	Local power	(Notes 1,6)	+2.2		The lower of 5.5 or V_{DD} + 0.3	V
		Parasite power		+3.0			
Sink Current	I_L	$V_{I/O}$ = 0.4V		4.0			mA
Standby Current	I_{DDS}	(Notes 7, 8)			750	1000	nA
Active Current	I_{DD}	V_{DD} = 5V (Note 9)			1	1.5	mA
DQ Input Current	I_{DQ}	(Note 10)			5		µA
Drift		(Note 11)			±0.2		°C

Note 1: All voltages are referenced to ground.
Note 2: The Pullup Supply Voltage specification assumes that the pullup device is ideal, and therefore the high level of the pullup is equal to V_{PU}. In order to meet the V_{IH} spec of the DS18B20, the actual supply rail for the strong pullup transistor must include margin for the voltage drop across the transistor when it is turned on; thus: $V_{PU\_ACTUAL} = V_{PU\_IDEAL} + V_{TRANSISTOR}$.
Note 3: See typical performance curve in Figure 1.
Note 4: Logic-low voltages are specified at a sink current of 4mA.
Note 5: To guarantee a presence pulse under low voltage parasite power conditions, V_{ILMAX} may have to be reduced to as low as 0.5V.
Note 6: Logic-high voltages are specified at a source current of 1mA.
Note 7: Standby current specified up to +70°C. Standby current typically is 3µA at +125°C.
Note 8: To minimize I_{DDS}, DQ should be within the following ranges: GND ≤ DQ ≤ GND + 0.3V or V_{DD} – 0.3V ≤ DQ ≤ V_{DD}.
Note 9: Active current refers to supply current during active temperature conversions or EEPROM writes.
Note 10: DQ line is high ("high-Z" state).
Note 11: Drift data is based on a 1000-hour stress test at +125°C with V_{DD} = 5.5V.

AC Electrical Characteristics–NV Memory

(-55°C to +125°C; V_{DD} = 3.0V to 5.5V)

PARAMETER	SYMBOL	CONDITIONS	MIN	TYP	MAX	UNITS
NV Write Cycle Time	t_{WR}			2	10	ms
EEPROM Writes	N_{EEWR}	-55°C to +55°C	50k			writes
EEPROM Data Retention	t_{EEDR}	-55°C to +55°C	10			years

AC Electrical Characteristics

(-55°C to +125°C; V_{DD} = 3.0V to 5.5V)

PARAMETER	SYMBOL	CONDITIONS		MIN	TYP	MAX	UNITS
Temperature Conversion Time	t_{CONV}	9-bit resolution	(Note 12)			93.75	ms
		10-bit resolution				187.5	
		11-bit resolution				375	
		12-bit resolution				750	
Time to Strong Pullup On	t_{SPON}	Start convert T command issued				10	µs
Time Slot	t_{SLOT}	(Note 12)		60		120	µs
Recovery Time	t_{REC}	(Note 12)		1			µs
Write 0 Low Time	t_{LOW0}	(Note 12)		60		120	µs
Write 1 Low Time	t_{LOW1}	(Note 12)		1		15	µs
Read Data Valid	t_{RDV}	(Note 12)				15	µs
Reset Time High	t_{RSTH}	(Note 12)		480			µs
Reset Time Low	t_{RSTL}	(Notes 12, 13)		480			µs
Presence-Detect High	t_{PDHIGH}	(Note 12)		15		60	µs
Presence-Detect Low	t_{PDLOW}	(Note 12)		60		240	µs
Capacitance	$C_{IN/OUT}$					25	pF

Note 12: See the timing diagrams in Figure 2.
Note 13: Under parasite power, if t_{RSTL} > 960µs, a power-on reset can occur.

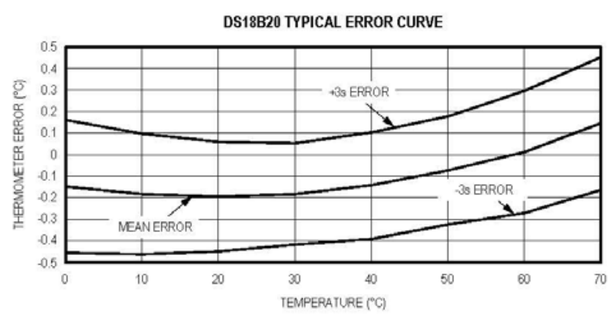


Figure 1. Typical Performance Curve

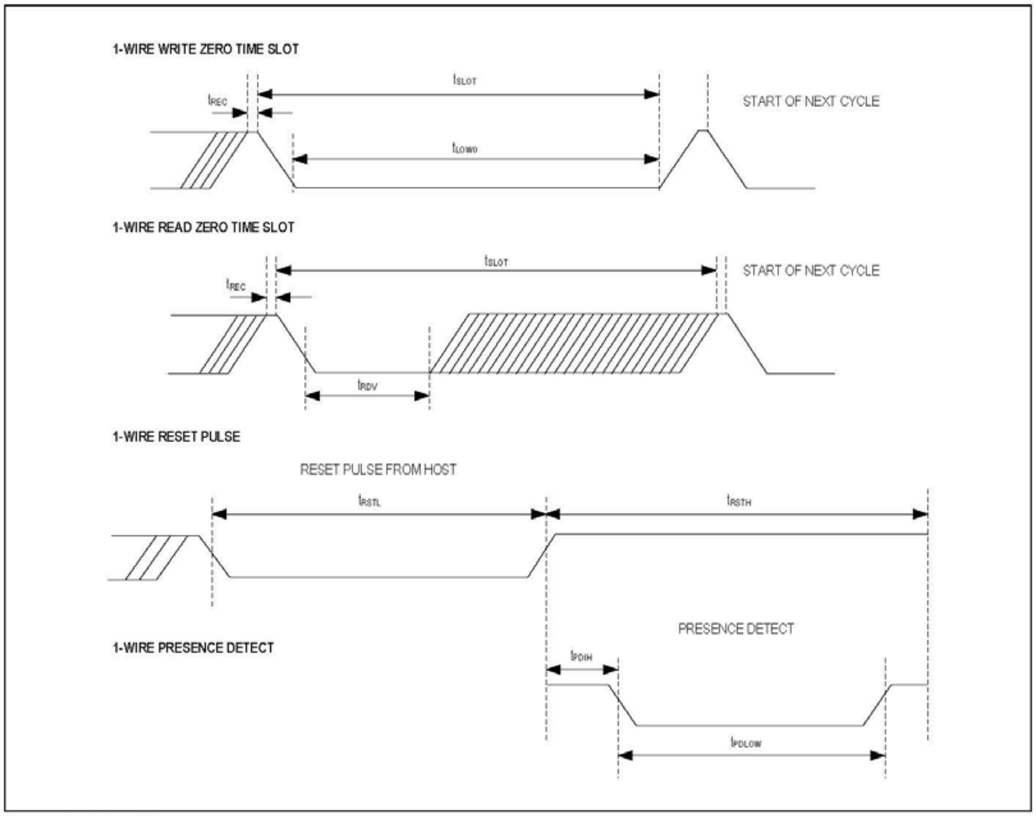


Figure 2. Timing Diagrams

Pin Description

PIN			NAME	FUNCTION
SO	µSOP	TO-92		
1, 2, 6, 7, 8	2, 3, 5, 6, 7	—	N.C.	No Connection
3	8	3	V_{DD}	Optional V_{DD}. V_{DD} must be grounded for operation in parasite power mode.
4	1	2	DQ	Data Input/Output. Open-drain 1-Wire interface pin. Also provides power to the device when used in parasite power mode (see the *Powering the DS18B20* section.)
5	4	1	GND	Ground

Overview

Figure 3 shows a block diagram of the DS18B20, and pin descriptions are given in the *Pin Description* table. The 64-bit ROM stores the device's unique serial code. The scratchpad memory contains the 2-byte temperature register that stores the digital output from the temperature sensor. In addition, the scratchpad provides access to the 1-byte upper and lower alarm trigger registers (T_H and T_L) and the 1-byte configuration register. The configuration register allows the user to set the resolution of the temperature-to-digital conversion to 9, 10, 11, or 12 bits. The T_H, T_L, and configuration registers are nonvolatile (EEPROM), so they will retain data when the device is powered down.

The DS18B20 uses Maxim's exclusive 1-Wire bus protocol that implements bus communication using one control signal. The control line requires a weak pullup resistor since all devices are linked to the bus via a 3-state or open-drain port (the DQ pin in the case of the DS18B20). In this bus system, the microprocessor (the master device) identifies and addresses devices on the bus using each device's unique 64-bit code. Because each device has a unique code, the number of devices that can be addressed on one bus is virtually unlimited. The 1-Wire bus protocol, including detailed explanations of the commands and "time slots," is covered in the *1-Wire Bus System* section.

Another feature of the DS18B20 is the ability to operate without an external power supply. Power is instead supplied through the 1-Wire pullup resistor through the DQ pin when the bus is high. The high bus signal also charges an internal capacitor (C_{PP}), which then supplies power to the device when the bus is low. This method of deriving power from the 1-Wire bus is referred to as "parasite power." As an alternative, the DS18B20 may also be powered by an external supply on V_{DD}.

Operation—Measuring Temperature

The core functionality of the DS18B20 is its direct-to-digital temperature sensor. The resolution of the temperature sensor is user-configurable to 9, 10, 11, or 12 bits, corresponding to increments of 0.5°C, 0.25°C, 0.125°C, and 0.0625°C, respectively. The default resolution at power-up is 12-bit. The DS18B20 powers up in a low-power idle state. To initiate a temperature measurement and A-to-D conversion, the master must issue a Convert T [44h] command. Following the conversion, the resulting thermal data is stored in the 2-byte temperature register in the scratchpad memory and the DS18B20 returns to its idle state. If the DS18B20 is powered by an external supply, the master can issue "read time slots" (see the *1-Wire Bus System* section) after the Convert T command and the DS18B20 will respond by transmitting 0 while the temperature conversion is in progress and 1 when the conversion is done. If the DS18B20 is powered with parasite power, this notification technique cannot be used since the bus must be pulled high by a strong pullup during the entire temperature conversion. The bus requirements for parasite power are explained in detail in the *Powering the DS18B20* section.

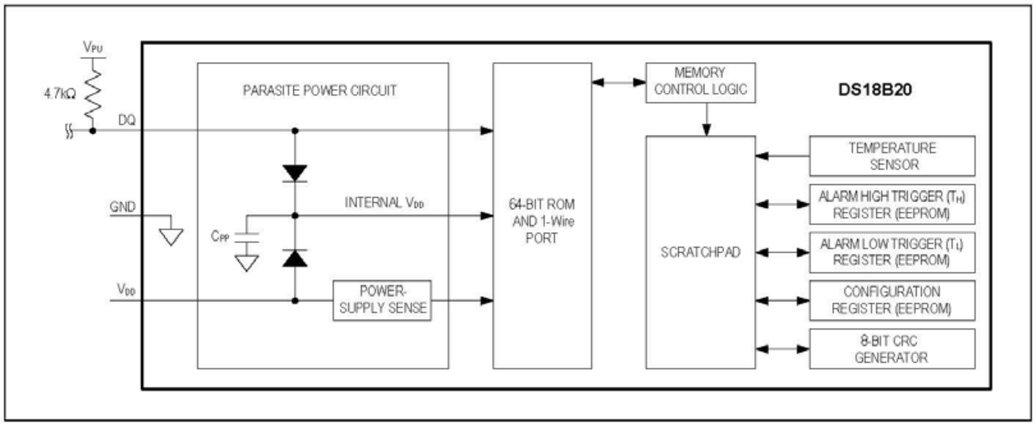


Figure 3. DS18B20 Block Diagram

Appendix G -Wind Sensor Details

KNX/EIB
Wind sensor, Rain sensor

GIRA

Wind sensor, heatable
Order-No. : 0580 00
Rain sensor 0/10V
Order-No. : 0579 00

Operating instructions

1 Safety instructions

Electrical equipment may only be installed and fitted by electrically skilled persons.

Failure to observe the instructions may cause damage to the device and result in fire and other hazards.

Do not operate in the vicinity of chimneys or other exhaust or ventilation systems. Doing so will compromise function.

Do not operate in the vicinity of radio transmitter systems. Doing so will compromise function.

Select the mounting place so that the device will still be accessible for maintenance purposes.

Do not lay sensor cables parallel to mains- or load-transmitting cables. Doing so will compromise function.

These instructions are an integral part of the product, and must remain with the end customer.

2 Function

Intended use

- Sensors for measuring weather data
- Power is supplied to the sensors and the sensor signals are evaluated via additional electronics, e.g. a weather station

Wind sensor (Figure 1):

- Detection of the horizontal wind speed
- Vertical installation in outdoor areas, e.g. on walls of buildings, using the supplied mounting bracket

Rain sensor (Figure 2):

- Detection of precipitation
- Installation in outdoor areas, e.g. on walls of buildings, using the supplied 110° mounting bracket

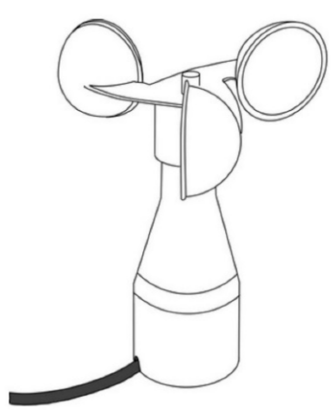


Figure 1: Wind sensor – View

32506122 10499205 I00 20.05.2011 1/4

KNX/EIB
Wind sensor, Rain sensor

GIRA

Product characteristics

Wind sensor

- Measurement of the rotational speed of the anemometer
- Output with analogue output signal 0...10 V
- Maintenance-free
- Operation without additional power supply possible

[i] Recommendation: To avoid dew and condensation, use a separate 24 V AC/DC power supply for heating (see chapter 4.2. Accessories).

[i] For proper function, the anemometer must be able to rotate freely. Heavy fouling, icing or frozen precipitation can jam the anemometer.

Rain sensor

- Measurement of the electrical conductivity on the sensor surface
- Output by means of analogue output signal: 0= dry, 10 V = rain
- Heating of the sensor surface with separate 24 V AC/DC power supply (see chapter 4.2. Accessories)

[i] The sensor signal is reset when the sensor surface has dried out and a run-on time of 4 minutes has elapsed. The heater speeds up the drying and melts snow and ice.

[i] For proper function, clean the rain sensor regularly with a mild cleaning agent.

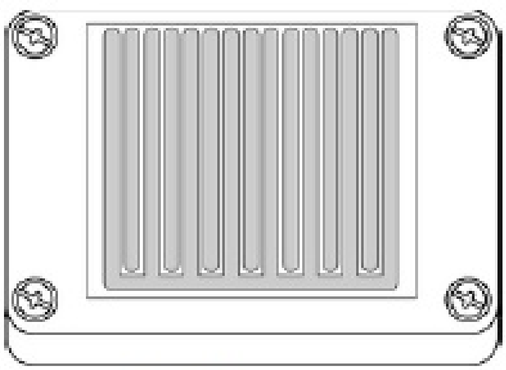


Figure 2: Rain sensor – view

3 Information for electrically skilled persons

3.1 Fitting and electrical connection

DANGER!
Electrical shock on contact with live parts in the installation environment.
Electrical shocks can be fatal.
Before working on the device, disconnect the power supply and cover up live parts in the working environment.

Mounting and connecting the wind sensor

Selecting a suitable installation location. Do not install in wind shadows or locations with strong turbulence, updrafts, etc.

- Mount wind sensor vertically on the building wall using the enclosed mounting bracket.
- Connect wind sensor to an evaluation device, e.g. a weather station.

32506122 10499205 I00 20.05.2011 2/4

brown	Operating voltage 24 V DC
white	Operating voltage earth, GND
green	Sensor signal 0...10 V output
yellow	Sensor signal earth, GND output
grey, pink	Heating connection
green-yellow	Shield, earth connection

Installing and connecting the rain sensor

Select a suitable installation location: rain must be able to reach the sensor in an unobstructed manner. Do not install under projecting roofs.

- Mount rain sensor on wall of building using enclosed 110° mounting bracket.
- Connect rain sensor to an evaluation device, e.g. a weather station.

brown	Operating voltage 24 V DC
green	Sensor signal 0...10 V output
white	Common earth operating voltage/sensor signal, GND
yellow, grey	Heating connection

4 Appendix

4.1 Technical data

Wind sensor, heatable, Order-No. 0580 00

Supply	
Rated voltage	DC 18 ... 32 V SELV
Current consumption	6 ... 12 mA
Heating	
Rated voltage	AC/DC 24 V
Switch-on current	max. 1 A
Ambient conditions	
Ambient temperature	-25 ... +60 °C
Safety class	III
Protection rating	IP 65 (in position for use)
Output signal	
Measuring range	0.9 ... 40 m/s
Load	max. 60 m/s (for short periods)
Output voltage	DC 0 ... 10 V
Load	min. 1.5 kΩ
Connection cable	
Cable type	LiYY 6x0.25 mm²
Cable length	approx. 3 m
Can be extended up to	max. 100 m
Dimensions Ø×H	134×160 mm
Weight	approx. 300 g

Rain sensor 0/10V, Order-No. 0579 00

Supply	
Rated voltage	DC 15 ... 30 V
Current consumption	approx. 10 mA
Heating	
Rated voltage	AC/DC 24 V
Power consumption	max. 4.5 W

KNX/EIB

Wind sensor, Rain sensor

GIRA

Ambient conditions	
Ambient temperature	-30 ... +70 °C
Safety class	III
Protection rating	IP 65
Output signal	
Output voltage	DC 0 / 10 V
Load	min. 1 kΩ
Reaction time	max. 4 min
Connection cable	
Cable type	LiYY 5x0.25 mm²
Cable length	approx. 3 m
Can be extended up to	max. 100 m
Dimensions L×W×H	58×83×17 mm
Weight	approx. 300 g

4.2 Accessories

Power supply Order-No. 1024 00

4.3 Warranty

The warranty is provided in accordance with statutory requirements via the specialist trade.

Please submit or send faulty devices postage paid together with an error description to your responsible salesperson (specialist trade/installation company/electrical specialist trade). They will forward the devices to the Gira Service Center.

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