Simulation and optimization of de-Icing two-phase closed Thermosyphon based on CO_2 work fluid

by

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FOREWORD

Earth-to-surface heat transfer as a reliable and renewable energy resource for various purposes including generating electricity and heating homes has been the subject of research by scientists around the world for many years. The results of many of these studies in recent years have led to the widespread use of this resource and its replacement with fossil fuels.

What is studied in this article is the transfer of this heat to the earth's surface to melt the snow, which challenges many regions in winter and creates many problems, especially in the field of their transportation system.

This paper uses a two-phase closed thermosyphon with carbon dioxide as work fluid to absorb the heat from the ground in the lower part of the pipe and transferring it to the upper part of the pipe based on the phase changing and increasing its latent heat to complete the heat transfer cycle. The main challenge in this field is first the practical hypothesis of heat transfer through this system and then finding the appropriate dimensions of the thermosyphon to achieve the maximum heat transfer rate. This issue has been evaluated by numerical simulation of a thermosyphon and applying the governing relations on it in Matlab.

In this research, an attempt has also been made to use the obtained results for the optimized thermosyphon dimensions, as the database for numerical analysis and data structures (Computational fluid dynamics Analyze - CFD) to evaluate and solve the problem that involves fluid flows (CO_2) and finally compare the results with the numerical simulations performed to make sure that the results are as accurate as possible.

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Simulation et optimisation du thermosyphon fermé biphasé de dégivrage à base de fluide de travail CO_2

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Résumé

Lors de la conception d'un thermosiphon pour transférer la chaleur du sol vers la surface afin de faire fondre la neige et la glace, de nombreux paramètres jouent un rôle dans son efficacité et son efficacité.
La recherche de dimensions géométriques appropriées pour un thermosiphon capable de transférer suffisamment de chaleur pour faire fondre la neige et l'étude du comportement du fluide à l'intérieur en fonction des conditions aux limites, fait l'objet de cette recherche.
En simulant un thermosiphon dans MATLAB basé sur la formation d'un réseau de résistances, on a tenté d'étudier la quantité de ce transfert de chaleur en fonction de ses différentes dimensions. Ensuite, en utilisant des méthodes mathématiques, une tentative a été faite pour trouver les dimensions optimales pour obtenir la quantité maximale de taux de transfert de chaleur.
Enfin, en analysant le CFD de l'une des valeurs optimales calculées, une tentative est faite pour comparer les résultats du transfert de chaleur et afficher le comportement du fluide dans le système.

Mots-clés: Thermosiphon, TPCT, optimisation, simulation Matlab,, analyse CFD

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ABSTRACT

Two phase closed Thermosyphon (TPCT) are subcategory of heat pipes based on natural circulation loop (NCL) with high performance heat transfer capacity and variety applications which are used to transfer a large range of heat through different temperatures. In general, heat pipes are a sealed closed pipe with high length-to-diameter ratio, which operate based on latent heat of evaporation and condensation to transfer heat. The system equipped with capillary wick structure to facilitate the fluid transport where the gravity act as opposition or insufficient force through fluid movement. Two phase closed Thermosyphon (TPCT) is a type of heat pipe, which has no capillary wick structure and the fluid transport by gravity as a directional force.

Heat pipes technology has proven their high performance and efficiency by considering all aspects of their related parameters and practical in many industries from cooling nuclear reactors to transferring heat from computer chipsets to the fan. Proper heat transfer capability with the ability to work with large range of temperature differences at both ends of the system has made the analysis of this technology very important.

What is very important when using thermosyphon is how to design and consider all the parameters and variables of the system. In addition to design complexities for different applications and considering the wicks system, optimizing and increasing the heat transfer capacity of a TPCT depends on the analysis and selection of many related parameters, including its geometric dimensions and the type and amount of work fluid needed to operate confidently. These researches could have more value when it aim is to design a system that can use and transfer geothermal energy as a clean energy source.

This research is based on the simulation a thermosyphon buried in the earth receives the heat energy needed to evaporate its work fluid from the geothermal energy and transfers this energy to the earth's surface in order to melt snow in winter through a thermosyphon mechanism. In addition, by applying numerical optimization on the dimensions and work fluid ratio inside the thermosyphon, in this article, an attempt has been made to present the results of this analysis in different categories by considering its different applications. Finally, by applying CFD analysis on the designed thermosyphon, the fluid behavior in thermosyphon is assessed under mentioned environmental conditions.

Keywords: Two phase close Thermosyphon, TPCT, Heat pipe, CO_2 work fluid, Optimization, Simulation, CFD analyze

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LIST OF ABREVIATIONS

a	Adiabatic	K_p	dimensionless pressure parameter
e	Evaporator	Z	resistance
c	Condenser	f	friction factor
am	amplitude	F	liquid fill fraction
an	Annual	g	Gravity- m/s^2
V	Volume – m3	h	Enthalpy - J/kg
C	Constant value	K	Kinematic Viscosity – m^2/s
Co	Copper	me	merit number
s	Soil	$\bar{Q}$	Heat transfer rate - W
cmt	Cement	P	Pressure - Pa
l	Liquid	Re	Reynolds Number
v	vapour	SF	Shape Factor - m
si	Sink	SL	Specific Latent Heat – J/Kg
so	Source	ω	angular frequency
i	inner	$\phi 1$	Figure of merit for condensation
o	outer	$\phi 2$	Figure of merit for nucleated boiling
Ave	Average	Cp	Specific heat capacity - (J/kg-K)
A	Cross sectional area – m^2	Ve	Velocity – m/s
Bo	Bond number - mg/γL	α	Ground thermal diffusivity - (m^2/s)
Tic	Thickness - m	λ	Thermal conductivity coefficient - W/(m·K)
D	Diameter - m	μ	Dynamic Viscosity - Pa.s
H	Height -m	ρ	Density – kg/m^3
L	Length - m	σ	Surface tension - N/m
T	Temperature - K	φ	Phase
t	Time - s	EL	Effective length - m
S	Surface Area	z	Depth - m

INTRODUCTION

Increasing use of Heat pipes in recent years has led to much research to develop their performance in cases where the need for heat transfer required. Proper performance and high efficiency of this system in comparison to the other conventional heat exchanger method created a special place in the heat transfer industry.

Depending on the function and application of the heat pipes, their dimensions can be vary from less than a millimeter up to more than one meter in diameter and about 0.1 up to 450 for length to diameter ratio which provide different heat transferring rates. [1] However, the performance of each heat pipe is effected by several limitations, which restrict the heat-transferring rate and should be considered seriously in calculation and designing process.

Determining the appropriate dimensions and work fluid filled rate, which can guarantee both demanding heat transfer rate and possibility of production with feasible cost function, are subjects of most recent research in this area.

Heat pipes have always been of interest to scientists in various fields and it is still the subject of an average of more than 250 scientific articles per year. The period between 1960 and 2005 can be considered as a leap forward in this technology. Description and formulation of its general process, new kinds of wick structures to speed up the fluid transfer process (heat transfer rate) for conventional and flat heat pipes were introduced and new working fluids were examined. In this period, the scientist highlighted the development of new types of loop heat pipes (LHP and CPL), micro heat pipes. Solar collector systems, high-temperature heat pipes, and space applications, which are all the fundamental sources for recent researches. This growing trend has continued so that for the years between 2012 and 2014, for example, the number of studies reached 800 articles. [2]

Two phase closed Thermosyphon (TPCT) as a subcategory of the heat pipe, which has no capillary wick structure and no additional power input to circulate the work fluid has received more attention in recent years. Their simple operation (no complicated calculations required for capillary wick structure), high heat transfer capacity and their high reliability is part of the reasons that much of the recent research on heat transfer area has been devoted to it. Fundamental heat transfer theory indicates that any heat transferring is based on difference temperature and larger the temperature difference ($T_{so} - T_{si}$), the higher heat transferring rate. However, in our project, the difference temperature is not too much (ground temperature – cold day's average weather temperature). Therefore, to solve this problem, a process should be used that, despite the low temperature difference between its two ends, has the ability to transmit high heat, which this is one of the characteristics of thermosyphons. One of the most significant advantages of thermosyphones is the vapour inside of it carries large amount of latent heat from evaporator to condenser which could be another acceptable reason to be used in my project.

This thesis is based on TPCT operated with CO_2 work fluid in order to provide sufficient heat for melting snow cover the surfaces during the winters and de-icing. Many cities pay annually a lot of money to melt snow that reaches several meters (especially in the cold seasons of the year) in some areas. Despite these costs, these operations are often inefficient and will have far-reaching consequences for cleaned environments for a very long time. Reducing the asphalt resistance of roads and destroying their texture in the long time, severe pollution caused by salt sprays to melt snow and severe erosion due to their acidic properties, timeliness of the process and the limited access to all areas are just some of the problems associated with current conventional methods of clearing snow and ice from the surface of roads and highways. The city of Montreal, for example, sees an average of 2.10 meters of snowfall annually, often with very low temperatures, which also causes frost. It costs an average of about $ 192 million in 2018 to deal with this amount of snow removal from the city, which, of course, is always accompanied by public dissatisfaction with the slow pace of work. [3] The idea of using thermosyphones to transfer heat from the earth to the surface to keep the temperature constant in a range has been studied by scientists for years and has sometimes been evaluated in practice. Therefore, measuring the implementation of this technology to melt snow accumulated on the surface can not only be important for many countries with heavy snowfall, but can also be much more important for sensitive areas such as fire stations, airports and military areas where accessing to snowless areas in their work environment is an essential.

Besides many studies which focused on the performance of thermosyphones work with water as the common work fluid, and due to the very small temperature difference between the inside of the earth as heat source energy and its surface as heat sink source, this study focus on carbon-dioxide as the work fluid. CO_2 with acceptable thermo-physical properties such as density, viscosity, suitable surface tension, latent heat of evaporation, non-flammable, non-toxic gas that has received a lot of attention in recent research as work fluid.

In the second chapter of this thesis, the result of simulation is demonstrate and examined by some initial inputs to get a better assumption of the thermosyphon size and heat transfer rate. In the following, the effect of each of the variables on the overall rate of heat transfer is briefly described and a solution has been proposed to increase its heat capacity.

In the next chapter, the simulation formulation has converted to the standard optimization format and according to the different classifications; the best values with the aim of transferring the maximum amount of heat of these parameters by numerical methods are calculated.

Finally, in the last chapter of this study, an attempt has been made to provide a CFD analyze of the performance of one of the optimal dimensions calculated in ANSYS software to be able to compare with the simulated values.

CHAPTER 1

PROBLEM STATEMENT

In this section, the brief of the problem, which are more considered in this thesis, are presented and the effect of the problems on results is formally defined. In the next section, the procedure of simulation and the method for optimization the parameters in the problem-solving process are described.

1.1 Problem Definition

The use and performance of heat pumps are strongly influenced by the design methods and various environmental factors. The length of the different parts of a Heat Pipe(HP) ,as the premier factor of choice for its design, has a direct impact both on the amount of heat absorbed by the heat source (in this study, Geothermal energy in evaporator section) and its heat transferring rate to the heat sink (snow-covered surface on top of condenser section).

In the next step, selection the type and amount of work fluid, which has the highest heat transfer capacity, proportional to the temperature difference between the two ends of the heat pipe, should be considered to achieve the highest efficiency. Selection of a working fluid is directly connected to the properties of fluids. Compatibility with wick and wall materials, good thermal stability, high latent heat and thermal conductivity and low viscosity are some of the most important factors that should be considered for choosing the best work fluid. [4]

Furthermore, a number of constrains that limit the amount of heat transfer rate at specified intervals must also be taken into account in the optimal design of a heat pipe such as: Boiling limit, Sonic limit, Counter current flow limit, Dry out limitation and Vapour temperature limitation which will be described in detail in next sections.

The calculation and challenge of choosing the appropriate parameters for a Two phase closed Thermosyphon (TPCT) as a subcategory of the heat pipes, not only affect the aspect of heat transfer needed for melting snow (as the design goal of TPCT in this research), but can also greatly influence the justification of the project for cost. Certainly, the technology and science of using heat pipes and thermocouples to extract heat from the earth and transferring it to the surface is the continuation of all the research that scientists have been doing for many years to learn more about them and discover the equations that govern them which briefly mention to its history in the next section.

1.2 Historical Development and background of Heat pipe and Thermosyphon

The Pre-template model of the heat pipe was developed firstly by Richard S. Gaugler of the General Motors Corporation in 1942 and was registered as a patent in America. [5]. Gaugler's definition of a heat pipe was in fact a simple description of a low-capacity heat transfer device. Due to the industry's lack of sense of need at the time, research and subsequent advances in heat pipes postponed for about 10 years until 1962 by Trefethen [6], and in 1963 Wyatt's patent for a new application of his heat pipe, a new period of the challenge began [7] .Exactly one year later, this emerging phenomenon, which was at the beginning of its path of cognition, was designed by George Grove in Los Alamos National Laboratory in a similar program related to space research, and for the first time took the name of “heat pipe”. [8] Yale Eastman, a member of the Board of Directors and a shareholder of Advanced Cooling Technologies, Inc. (ACT) some years later described the theory and history of the heat pipe technology in an article published in Scientific American in May 1968 and tried to reclaim the importance of this advancing technology, which had been rejected by many factories. [9]

In the 1970s, Bienert et al. tried to theorize the theory of the transfer of a finite range of heat through heat pipes based on gravitational forces to Earth's surface to limit the amount of surface freezing. [10] Later, the theory was tested by placing heat pipes in small concrete cubes in some highways in Virginia and the technical capability of this process was later published in several reports. (Cheng & Zarlin 1993) In the next years, the development of heat pipes has entered a new phase with the introduction of thermosyphones as an effective method of heat transferring when the heat source is at the bottom and the heat sink place at the top. What distinguish heat pipes from thermosyphons are the capillary properties that are created by placing wick system inside the pipe and have the ability to transfer heat-containing vapour in the opposite direction to the earth's gravitational force. The two-phase closed thermosyphon

(TPCT) has a very high heat transfer capacity and is widely used in various industries and engineering applications such as heat exchangers, cooling of electronic components, solar energy conversion systems, Nuclear power plants, spacecraft thermal control, cooling of gas turbine rotor blades, etc. [11]

The heat transfer performance of a TPCT is affected mostly by different parameters, such as its physical geometry, inclined angle, vapor temperature and pressure provided before and within the operation, filling ratio (F), and the thermophysical properties of the working fluid using to transfer the heat inside the pipe. The heat transfer of the TPCT has been widely studied experimentally by several researchers in the past few decades. Through the complexity of heat transfer in liquid pools, the calculation has mostly been based on empirical correlations.

El-Genk and Saber collected more than 700 heat transfer data-points of various researchers for heated liquid pools of water, ethanol, methanol, Dowtherm-A, R-11, and R-113. They compared these data by superimposing the natural convection and nucleated boiling heat transfer correlations by using a power-law approach. [12] .Imura et al. [13] examined water- and ethanol-filled in TPCT and stated heat transfer correlations according to their results. In 1981, Shiraishi et al. [14] developed Imura et al.'s correlation using a corrected operating pressure. Groß [15] collected and classified the heat transfer data of various researchers and investigated more than 2,500 data-points, which included 11 different working fluids.

Despite the high number of heat transfer correlation in thermosyphons based on different work fluids they filled with, the lack of TPCT analysis that works with carbon dioxide has always been seen over the years. In the next chapter, I will refer to a part of its history.

1.3 General function of Heat pipe and Thermosyphones

A Heat pipe is a sealed pipe or tube included of three different sections. Evaporator where heat is absorbed from the proximity heat source and transmitted to the work fluid, Condenser where the vapour fluid condensate and transfer its latent heat to the heat sink with a lower temperature than the heat source and adiabatic section which act as separator of evaporator and condenser parts. (Figure 1.1).

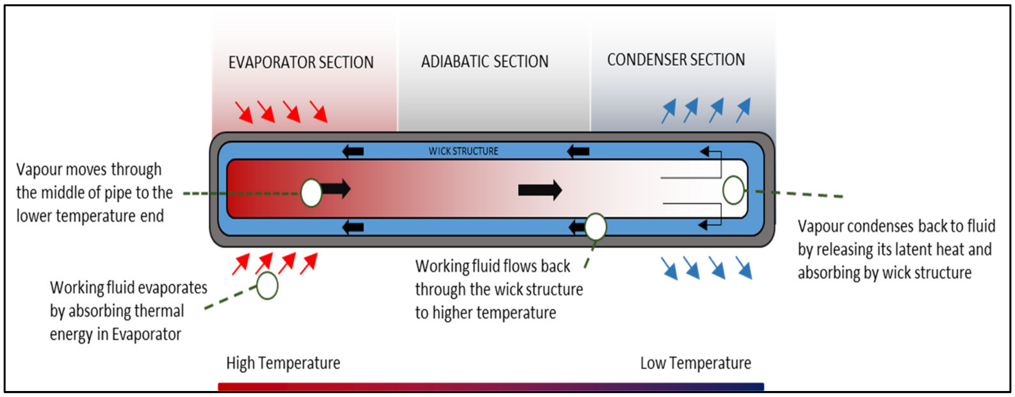

Figure 1.1 Parts of a Heat pipe

Taken from "Overview of heat pipe studies" India, (January 2016)

The system also includes the capillary wick system where by compensating the pressure difference between the two ends of the pipe causes the fluid to return from the low-pressure part in the condenser to the high-pressure part in the evaporator and a fluid in a partial vacuum to complete the heat transfer cycle. Selecting the type and amount of work fluid depending on the temperature conditions and the amount of heat transfer required plays an important role in the efficiency of the heat pipes, which are discussed by Per Wallin in 2012. [4]

Heat pipes mostly act as passive devices that require no external power and force or control to perform their function. These specifications make heat pipe to be considered as a high thermal conductivity device, regarding Fourier's law, as the effective thermal conductivity along the direction of heat transport with generally at least four to five orders of magnitude greater than the thermal conductivity of copper. Heat pipes are based on both the principles of thermal conduction and phase transition to transfer heat between two solid surfaces with different temperatures. In the evaporator section, where is in contact with a thermally conductive surface, the work fluid absorbs the heat and turns into the vapor. The vapor then move along the heat pipe to the colder section (Condenser), release its latent heat, and back to the hot section in liquid through the heat pipe by either capillary action, gravity or centrifugal force. This cycle repeat until the heat source's temperature decrease, which cannot steam the work fluid anymore or the heat sink temperature increase where the work fluid cannot condense.

1.4 Two phase closed Thermosyphones

A Two phase closed Thermosyphon (TPCT) is a kind of heat transfer device which performs the same function as a heat pipe, but it does not rely on capillary forces to circulate the work fluid through the pipe. In a TPCT, the condenser placed higher than the evaporator section in order to use the gravity force to drive the condensate liquid from condenser to evaporator for completing the heat transferring cycle. This property will dramatically decrease the expenses of designing and avoiding installing wicks into the pipes or using the conventional pumps. However, the operation will be limited on projects where it is possible to place the source of heat at or below the heat sink. The angle of inclination should be 5 to 90 degree to horizontal (Figure 1.2). [16]

In a TPCT system, same as the heat pipes, the input power is provided through the evaporator walls to the work fluid and as the result, the liquid inside the pool begin to evaporate. The evaporation temperature depends on heat source temperature, the pressure inside the pipe and

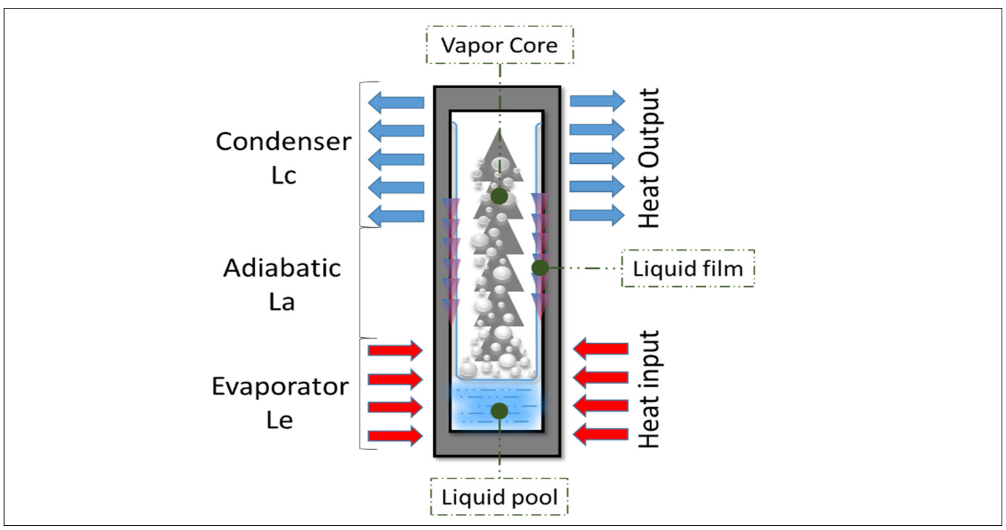

Figure 1.2 Schematic of a thermosiphon and its various components
Taken from Heat pipes – General information on their use, operation and design, (Heat Transfer Group of ESDU, 2008)

the work fluid's saturation temperature. Vapor flows upwards inside the pipe to the condenser where transfers the latent heat to the heat sink. In the condenser inner pipe wall, the fluid starts to condensing because of losing its latent heat and creating a film condensation of work fluid and flow back to the down part of the pipe. The direction of film flow is opposite to steam flow and in the direction of gravitational force towards the evaporator section. The vapor always flows through the core region of the thermosyphon and as the result; the film thickness hardly can affects the dynamic of vapor flow.

The use of heat pipes and thermosyphones has been expanding rapidly in recent decades, which may be due to their unique properties: [17]

1. High heat transfer capacity using latent heat property.
2. Passive heat transfer device.
3. Heat transfer capability in a wide range of temperature differences depending to different applications.
4. Heat transfer in only one direction (Thermal diodes).
5. Separation of hot and cold environments, which is an essential requirement of work in many industries.
6. Ability to transfer heat over long distances.
7. Constant temperature control capability.
8. Heat transfer capability despite low temperature difference between heat and cooling source.

However, the reason for choosing TPCT for heat transferring in this project is that it has more advantages than heat pipe. Thermosyphones have lower thermal resistance and wider operating limits (the integrity of the wick materials inside the heat pipes may not hold at the high temperature). Economically, thermosyphones lack the wick system and have less geometric complexity, which significantly saves on reducing design, construction and maintenance costs. Moreover, of course, the biggest feature of thermosyphones is that they can be turned off (out of orbit) when they do not need to transfer heat, compared to the heat pipe systems which cannot be turn off! This feature is very important for the purpose of design in this article, which is the melting of ice in the cold days of a year, so in hot seasons, the system can be turn off, which reduces dramatically the maintenance costs.

CHAPTER 2

LITERATURE REVIEW

In this chapter, the basics of a two phase closed Thermosyphon (TPCT) simulation based on the recent research results will discussed. Then this simulation strategy will be used for analysis the TPCT by CO_2 as work fluid and specific environmental conditions includes Geothermal energy as the heat source of the system and the Montreal city temperature in winter as the cold source of the system (heat transfer destination).

2.1 History of thermosyphon analysis and simulation

The complexity of the mathematical description and numerical simulation of each process can go a long way in enlightening how the overall performance of the system works and determining the range of expected results from the system. However, it can also be seen as a weakness for affecting the practical physical analysis of a system from the point of design engineers' view.

Over the years, many attempts have been made to simulate and analyze the thermosyphon system to better understanding of its performance, although it is a very difficult task due to the details and complexity of the system. The first universal study of the performance of a thermosyphon can be attributed to Lee and Mital (1972) [18]. Their ideas were followed later by extensive analysis of the system, including Chen et al. (1991) and Gunnerson and Sanderlin (1994) and many others whose results were discussed in various scientific circles. [17]

Various aspects of thermosyphon have been studied in detail in different scientific researches around the world. From horizontal, sloping or vertical thermosyphones, cross-sections with different geometries, from very long thermosyphones to very small sizes with different applications. Constant/variable heat flux analysis, constant temperature or forced convective heating /cooling, different work fluid with individual specification and different fill ratio.

Given the thermosyphon analysis, many articles have attempted to focus on analyzing only part of the overall system process such as condensation process in condenser by Gross (1992) and flooding in condenser. (Gross1992; Peterson and Bage 1991). [17]

However, the first experience of analyzing thermosyphons as an integrated system was by Shiraishi et al. in 1981. [14] This simulation was later developed by Dorban in 1989 based on conservation of mass, energy and momentum. [19] Simultaneously, Reed and Tien (1987) have tried to present one-dimensional model of steady-state performance of two-phase closed thermosyphon and provided it as his research result. [20] This simulation was more complete by Harley and Faghri (1994) research [21] by a transient two-dimension model including conjugated heat transfer through the wall of the pipe and falling of liquid film, which was condensate in condenser and based on laminar flow regime.

Research on the types of working fluids used in thermosyphones and its effect on heat transfer rate, depending on their unique characteristics, has been another topic studied in this field.

Carbon dioxide gas (CO_2) with the unique properties as a working fluid is also an important part of these studies in the field of heat pipes and TPCT. The main advantage of carbon dioxide is very positive environmental properties with negligible global warming potential (GWP). The heat transfer coefficient of carbon dioxide (CO_2) is two to three times higher than that of conventional refrigerants at the same saturation temperature, while the pressure drop in the two-phase is significantly smaller. [22] However, the principal drawbacks of carbon dioxide are its high critical pressure and rather low critical temperature. The constraints resulting from the applications with high operating pressure make it difficult to design and produce the system. As the first dedicated research, Schmidt [23] and Hahne [24] studied vertical tubes between 1960 and 1965, which were filled with carbon dioxide as work fluid. The condensation of carbon dioxide and its heat transfer in the uncritical region and the relationship between Condensation Condition and Heat Transfer was the subject in which Ishihara et al focus their investigations on it. [25] .

The use of geothermal energy as a source of endless and cheap energy has gradually led scientists to extract and transmit it using heat pipes and thermosyphons. Due to the low temperature of the earth in its shallow layers (suitable for drilling and embedding equipment), the use of carbon dioxide as a working fluid became more important. In recent years, research on the performance of thermosyphons has continued at a rapid pace. Johann-Cristoph et al. [26] In 2016 provides a dynamic simulation to validate a TPCT for geothermal application. The flow modeling in his research was based on Nusselet theory and film condensation and tried to make comparison of the experimental data with numerical simulation. Rieberer studied ground-coupled heat pumps based on the thermosyphon system using carbon dioxide as a working fluid. The system is equipped with two 65 meter boreholes. [27] Experimental investigation of CO_2 thermosyphon in supercritical region is the other research provided by Lin Chen et al. in 2013 for investigating the performance of TPCT in near-critical CO_2 as work fluid in high temperature/pressure condition. [28] Ochsner published analyses with a high-grade steel corrugated pipe system, in which a carbon dioxide heat pipe was used in combination with a ground source heat pump. These studies concentrated mainly on the recovery of deep ground-source energy. [29] The amount of thermosyphon filling rate by the working fluid is another important topic that was analyzed by Hamidreza Shabgard and Bin Xiao in 2013 for having better understanding of different variables of closed thermosyphones. [30] In this regard, this chapter has tried to provide the necessary conditions for the initial modeling and finally optimization of the system by stating the basic principles of simulating thermosyphons and formulating the governing relations.

2.2 Initial design of Two Phase Closed Thermosyphon

The performance of a TPCT can be simplified by simulating to an overall thermal resistance network, Z_{total}, includes all resistors refer to different parts of TPCT which can affect the amount of total heat transfer rate. (Figure 2.1). Having the total resistance can determine the overall Heat transfer rate when the differential temperature between the heat source and heat sink is known:

$$\dot{Q} = \frac{\Delta T}{Z_{total}} \tag{2-1}$$

Where:

$\dot{Q}$ = Total heat-transferring rate

ΔT = Differential temperature between the heat source and heat sink

Z_{total} = Overall thermal resistance network

2.3 Thermal resistance network in TPCT

As it was mentioned in previous section, for simulation of a TPCT, the system simplified to a network of thermal resistance where affected by two main temperature related to a heat source (T_{so}) and a heat sinks (T_{si}) respectively.

TPCT thermal resistances can be simulated by considering the process starting with the transfer of heat from the heat source followed by the process of evaporation of the work fluid in the evaporator and then condensing on the condenser and transferring the latent heat to the heat sink includes ten resistances corresponding to its different parts and functions. Given the impact of each of these thermal resistances on the rate of heat transfer, some of these values can be omitted in the final calculations, which will be discussed separately.

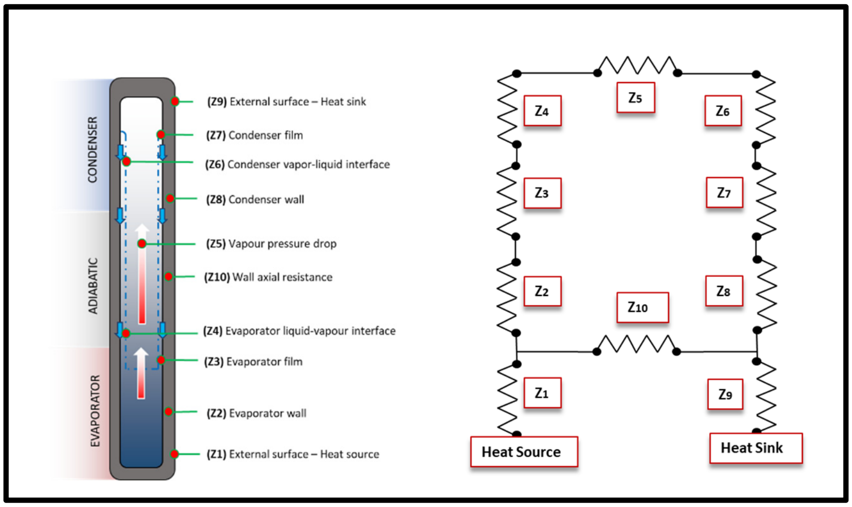

Figure 2.1 Thermal resistance network for TPCT

Taken from "Heat pipes – performance of Two phase closed thermosyphones," Engineering Sciences Data Unit - ESDU, (November 1983)

Z_1 - The thermal resistance between heat source and evaporator external surface

According to the cylindrical shape of the pipe located in the ground, the resistance level due to the surface of the pipe with the surrounding environment (soil in this research) is calculated through the following formula [31]:

$$Z_1 = \frac{1}{SF_e . \lambda_s} \tag{2-2}$$

Where:

SF_e = Conduction Shape Factor of evaporator (m)

λ_s = Soil thermal conductivity W/ (m·K)

The Conduction shape factor related to the evaporator section also will be calculated according the following equation:

$$SF_e = \frac{2\pi L_e}{\log(\frac{4L_e}{D_o})} \tag{2-3}$$

Where:

L_e = Length of Evaporator (m)

D_o = External diameter of TPCT (m)

Z_2 - Resistance along the thickness of the tube wall in the evaporator section. [32]

$$Z_2 = \frac{\ln(\frac{D_o}{D_i})}{2\pi L_e \lambda_{co}} \tag{2-4}$$

Where:

D_i = Internal diameter of TPCT - m

λ_{co} = Thermal conductivity coefficient of pipe (Copper) -W/ (m.K)

Z_4 and Z_6 – Internal thermal resistance of Liquid-vapour interface in evaporator and condenser respectively. Given the nature of this type of resistance and complexity in the calculations, as well as their very small amount in related to other thermosyphon thermal resistances and the minimum heat transfer rate, we neglect the value of these two parameters in our calculations.

Z_5 - The nature of this resistance is based on the change in saturation temperature of the work fluid during transfer from the evaporator to the condenser. Due to the vapor pressure dropping through the movement from the heat source in lower part of pipe to the heat sink in the upper part, the saturation temperature of the fluid, which determines the amount of fluid condensation, will change. [32] These changes can be considered equivalent to the thermal resistance against the thermosyphon heat transferring rate, which, given its small amount, can be neglected in the calculations.

Z_8 - Resistance along the thickness of the tube wall in the condenser section

$$Z_8 = \frac{\ln(\frac{D_o}{D_i})}{2\pi L_c \lambda_{co}} \tag{2-5}$$

Where:

L_c = Length of Condenser - m

Z_9 - Thermal resistance between condenser external surface and heat sink

$$Z_9 = \frac{1}{SF_c . \lambda_{cmt}} \tag{2-6}$$

Where:

SF_c = Conduction Shape Factor of Condenser - m

λ_{cmt} = Cement thermal conductivity -W/ (m·K)

Z_{10} – The axial thermal resistance of the pipe wall in TPCT

In addition to the thermal resistance along the thickness of the tube walls in the thermosyphon, the thermal resistance can also be defined along the axis of the container (longitudinal direction of the pipe).

$$Z_{10} = \frac{(0.5L_e + L_a + 0.5L_c)}{A\lambda_{co}} \tag{2-7}$$

Where:

A = Cross-sectional area of the wall of the container - m^2

L_a = Adiabatic length - m

Z_3 – Internal thermal resistance of the work fluid boiling in evaporator

This parameter is a Multi-part equation depends on fluid properties, dimensional aspects of the TPCT pipe and rate of heat transfer rate. For calculation of Z_3, First, the Reynolds number should be calculated:

$$Re_f = \frac{4\dot{Q}}{SL\mu_l \pi D} \tag{2-8}$$

Where:

SL = Specific latent of Operating fluid at vapour temperature

μ_l = Dynamic viscosity of operating fluid at vapour temperature

According to calculated (equation 5) or initial forecast of Reynolds number, if $50 \le Re_f \le 1300$, then internal resistance refer to liquid film and internal resistance refers to nucleate boiling in the pool equations are presented as follows respectively:

$$Z_{3f} = \frac{C\dot{Q}^{\frac{1}{3}}}{D^{\frac{4}{3}} g^{\frac{1}{3}} L_e \Phi_2^{\frac{4}{3}}} \tag{2-9}$$

$$Z_{3p} = \frac{1}{\Phi_3 g^{0.2} \dot{Q}^{0.4} (\pi D L_e)^{0.6}} \tag{2-10}$$

Where:

$C = \frac{1}{4}(\frac{3}{\pi})^{\frac{4}{3}} = 0.235$

Φ_2 = Figure of Merit for condensation [33] - $(\frac{L\lambda_l^3 \rho_l^2}{\mu_l})^{0.25}$

Φ_3 = Figure of Merit for nucleate boiling

g = Gravitational acceleration

For other Reynolds number range according to reference [34] , the internal thermal resistance could be calculated according to specific correction coefficients.

If $Z_{3p} \le Z_{3f}$, then $Z_3 = Z_{3p}$, otherwise the mean of internal thermal resistance of the work fluid boiling in evaporator, Z_3 should be calculated by interpolation:

$$Z_3 = Z_{3p} F + Z_{3f}(1 - F) \tag{2-11}$$

Where:

F = liquid fill (the fraction of the work fluid volume in the pipe to the volume of the evaporator section) – m

Z_7 – Internal thermal resistance of the work fluid condensing in condenser

For calculating the Thermal resistance of the condensing film of liquid, according to the references [35] , Nusselt theory of film wise condensation can be use:

$$Z_7 = \frac{C\dot{Q}^{\frac{1}{3}}}{D^{\frac{3}{4}} g^{\frac{1}{3}} L_c \Phi_2^{(\frac{4}{3})}} \tag{2-12}$$

Z_{Total} – Total system resistance

According to Equation 2-1, the total heat transfer rate in a thermosyphon is equal to the ratio of the temperature difference between its two ends to the total resistance equated for the system, which is the function of the individual resistors calculated for different parts of the system.

Under normal thermosyphon operation, due to the very small amount of some resistances and their small effect on the heat transfer rate, if the condition 2-13 is satisfied, the total resistance can be calculated from Equation 2-14:

If

$$Z_{10} / (Z_2 + Z_3 + Z_5 + Z_7 + Z_8) > 20 \tag{2-13}$$

Then:

$$Z_{Total} = Z_1 + Z_2 + Z_3 + Z_5 + Z_7 + Z_8 + Z_9 \tag{2-14}$$

Otherwise,

$$Z_{Total} = Z_1 + [(Z2 + Z_3 + Z_5 + Z_7 + Z_8)^{-1} + 1/Z_{10}]^{-1} + Z_9 \tag{2-15}$$

2.4 Maximum heat transferring rate limitation in TPCT

As the operation begins on the thermosyphon, the heat transfer rate increases because of the temperature difference between the two ends. As more fluid evaporates and condenses in it, more heat is transferred up to its highest level. Due to the different characteristics of the thermosyphon and related work fluid, as well as the temperature difference in the system, different constraints affect the maximum amount of heat transferring rate. The importance of considering these limitations is crucial for the optimal design of a TPCT with the highest heat transfer capacity that we will discuss later in Chapter 4 of this article.

These constraint can be due to various parameters such as severe reduction of vapor pressure, decrease in vapor velocity, dryness of the evaporator wall and insufficient work fluid layer, limitation of boiling in the evaporator walls, and finally the limitation of the opposite current, which will be briefly described below.

2.4.1 Maximum heat transfer rate - Boiling limit

When a steady film of vapor forms continuously between the work fluid liquid and the heated wall in the evaporator section, it cause limitation in maximum heat transferring rate by limiting and preventing heat transfer from the heat source to the liquid state fluid in the thermosyphon pool. Considering this boiling limitation gives the maximum heat flux as [36]:

$$\frac{\dot{Q}_{MAX\,(1)}}{S_{i\,e}} = 0.12\ SL\,(\rho_v)^{0.5}[\sigma g\,(\rho_l - \rho_v)]^{0.25} \tag{2-16}$$

Where:

$S_{i\,e}$ = is the internal surface area of the evaporator
σ = is the surface intension of the liquid
ρ_v = Density refer to vapour

ρ_l = Density refer to liquid

2.4.2 Maximum heat transfer rate - Sonic limit

Working in low-pressure operating condition may cause great velocity for vapour in comparison with sonic velocity in the vapour. This limitation similar to the viscosity limitation is typically encountered during the startup or over-power transient but mostly does not present in failure of the system. [17] The sonic limit can be estimated through the reference [32] where the maximum axial vapour mass is mentioned as:

$$\frac{\dot{Q}_{MAX\,(2)}}{A.SL} = 0.5(p_v\,\rho_v)^{0.5} \tag{2-17}$$

2.4.3 Maximum heat transfer rate - Counter current flow limit

This limitation occurs when the speed rate of absorption of liquids by vapour prevents the downward flow of liquid (This could happen when thermosyphones transfer high level of heat and as the result, the mass flow rate increase dramatically. The interfacial shear force overcomes the gravitational force in liquid film and the liquid film may be reversed). The Counter current flow limit (sometimes known as flooding" or "entrainment limitation") is independent from drying out limitation, which is related to the amount of liquid filled in Thermosyphon.

$$\frac{\dot{Q}_{MAX\,(3)}}{A\,.\,SL} = f_1 f_2 f_3 (\rho_v)^{0.5} [g(\rho_l - \rho_v)\sigma]^{0.25} \qquad (2\text{-}18)$$

Where:

f_1, is a function of Bond number:

$$Bo = D\,[\frac{g(\rho_l - \rho_v)}{\sigma}]^{0.5} \qquad (2\text{-}19)$$

When Bo be greater than 11, then f_1 = 8.2, Otherwise it should be read from Bond number figure. (Figure 2.2) [37]

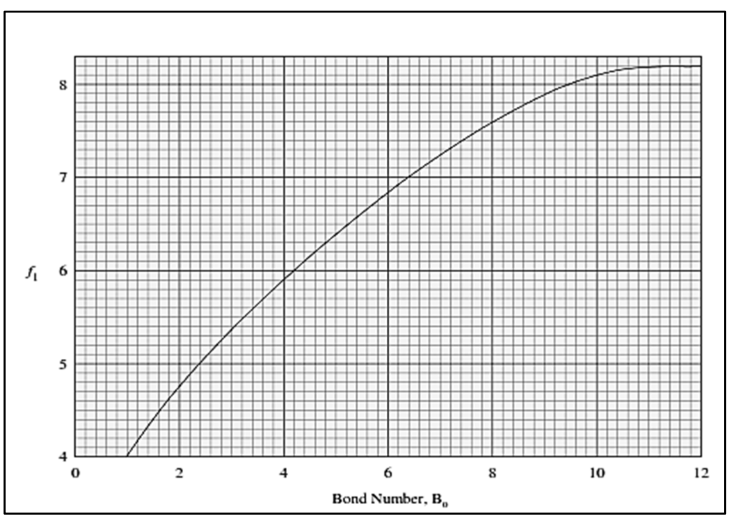

Figure 2.2 Bond number

Taken from Heat pipes – performance of Two phase closed thermosyphones, Engineering Sciences Data Unit – ESDU , (November 1983)

The factor f_2 is a function of the dimensionless pressure parameter, K_p, which is defined as:

$$K_p = \frac{P_v}{[g(\rho_l - \rho_v)\sigma]^{0.5}} \tag{2-20}$$

$$\text{And} \begin{Bmatrix} f_2 = K_p^{-0.17} & if & K_p \le 4\times10^4 \\ f_2 = 0.165 & if & K_p \ge 4\times10^4 \end{Bmatrix} \tag{2-21}$$

The factor f_3 is a function of the inclination of the pipe. When the pipe is vertical, f_3 = 1.

2.4.4 Liquid fill limitation (Dry out limitation)

This phenomenon happens when the volume of filled liquid is not enough to cover all inner side of the pipe above the pool with the film of liquid. Specially in the vertical pipes, where the most portion of falling film of liquid will evaporated before reaching to the pool, leaves some parts of inner walls of Thermosyphon dry. For eliminating liquid fill limitation, it is recommended to:

$$\begin{Bmatrix} i.\ 0.5 \le F \\ ii.\ \ V_l \ge .001D(L_e + L_a + L_c) \end{Bmatrix} \tag{2-22}$$

Where:

L_{total} = is the total length of thermosyphon (L_e+L_c+L_a)

$F = \frac{V_l}{AL_e}$ Is the liquid fill fraction.

2.4.5 Vapour pressure limitation

This limitation is related to the vapour pressure and will be considered whenever the thermosyphon operate at the pressure below atmospheric. The vapor pressure significantly drops in the end of condenser compared with the pressure in the end of evaporator.

$$\dot{Q} = \frac{D^2 h_v P_v \rho_v}{64\mu_v EL} \tag{2-23}$$

Where:

h_v = Enthalpy of vaporization

μ_v = Vapor dynamic viscosity

EL = Effective TPCT length ($0.5L_e + L_a + 0.5L_c$)

2.4.6 Thermosyphon length limitation

The allowable length of a thermosyphon for transferring heat between the heat sources to the heat sink largely depends on the vapor pressure of work fluid inside the system. This value in fact determines the range at which the cold source can be designed in projects within the allowable distance from the heat source. This distance is directly related to the operating temperature in the system and the thermal properties of the work fluid inside the thermosyphon, especially the vapor pressure.

The allowable total length of the thermosyphon can be calculated by equation (23):

$$V = \frac{\mathrm{Re}.K}{D_i} \tag{2-24}$$

$$f = \frac{64}{\mathrm{Re}} \tag{2-25}$$

$$L_{allow} = \frac{2P_v D_i}{4f\rho_v Ve^2} \tag{2-26}$$

Where:

L_{allow} = allowable total length of the thermosyphon

Ve = Velocity of flow - (m/s)

K = Kinematic viscosity - (m^2/s)

P_v = Pressure of vapour calculation at vapour temperature - (Pa)

f = friction factor for laminar flow

ρ_v = Density of vapour at vapour temperature – (kg/m^3)

2.5 Average of Earth's temperature

As described in the overall operation of a thermosyphon, the temperature difference in the two ends of the tube containing the fluid is the major factor in the heat transfer rate in the system. The heat required for changing the work fluid phase in the thermosyphon analyzed in this article is provided by the ground heat (Geothermal Energy). This means that the bottom of the thermosyphon is placed inside the ground and the geothermal energy as heat transferred to the evaporator section of the TPCT.

Certainly, the deeper the pipe, the more heat is transferred into the system. Given the possible limitations, the subject of optimum amount for the depth will be considered in the following chapters, but at this point, a function must be defined to calculate this heat as the heat source of the system. The heat energy emitted by the sun transmits continuously to the earth's surface, which affects the Earth's temperature at depths near its surface. The heat absorbed by the sun at different depths has been studied for many years. Soil temperature at different depths can be defined based on a harmonic equation and its expansion based on different temperature properties of the environment as follows: [38]

$$T_{soil}(z,t) = T_{an} - T_{am} e^{\left(-z\sqrt{\frac{\omega}{2\alpha}}\right)} \cos\left(\omega t - \varphi - z\sqrt{\frac{\omega}{2\alpha}}\right) \tag{2-27}$$

Where:

z = Depth (m)

t = Time coordinate (s)

T_{an} = Annual average temperature of soil

T_{am} = Amplitude (Co)

ω = Angular Frequency (rad/s)

α = Ground Thermal diffusivity (m^2/s)

ϕ = Phase (rad)

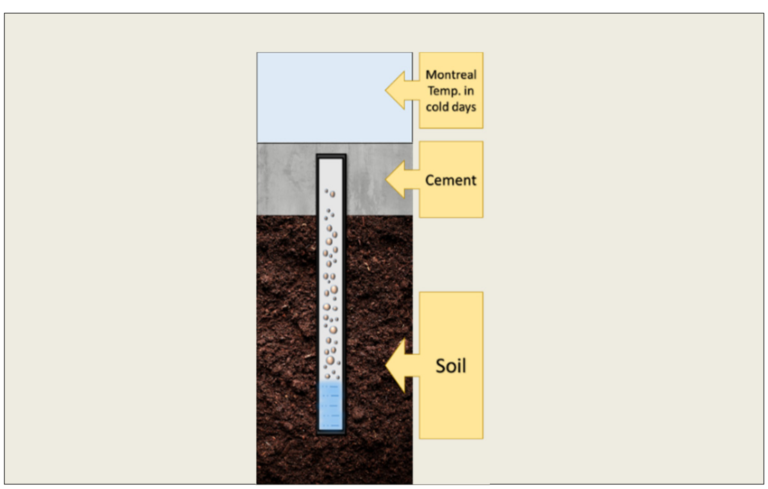

Figure 2.3 Environmental conditions of thermosiphon

2.6 Carbon dioxide as the TPCT work fluid

Choosing the right fluid that is capable of transferring heat through a thermosyphon is one of the biggest challenges of its designing. Charging a thermosyphon requires a specific condition where molten work fluid under vacuum is transferred to the evacuated thermosyphon pipe. Pure fluid without condensable gases is an essential factor for a thermosyphon for proper operation. Various factors can influence the choice of the work fluid, including environmental factors, material of tube containing fluid, the amount of heat needed to transfer from the hot source to the cold source, and the degree available for the evaporation and condensation of the fluid.

One of the most important criteria for selecting a work fluid in a heat pipe is the evaporation and condensation condition of the fluid, which should be capable to provide the phase changing in the temperature difference range in the evaporator and condenser. The working fluid should have the melting temperature below the operating temperature of the thermosyphon and the critical temperature above the operating temperature range. If the operating temperature is very high, it may exceed the critical temperature of the work fluid and cause the vapour will not condense. On the other hand, in comparison to the work fluid melting temperature, if the operating temperature be too low, the liquid will not evaporate properly.

Due to the different range of thermo-physical properties for different materials, many indicators have been presented for selecting work fluid within Thermosyphon in previous research. One of the most important is the comparison of the number of merits related to different materials, which was suggested by Chi [39]:

$$me_l = \left(\frac{SL\lambda_l^{3}\rho_l^{2}}{\mu_l}\right)^{0.25} \tag{2-28}$$

For each work fluid, the higher merits number means the greater ability to transfer heat in a thermosyphon. High liquid density and high latent heat will reduce the fluid flow required to transport a given power, while high surface tension increase the pumping capability. A low liquid viscosity reduced pressure drop for a given power [40]. The Merit number comparison to different temperature operation for some work fluids demonstrates in figure 2.4.

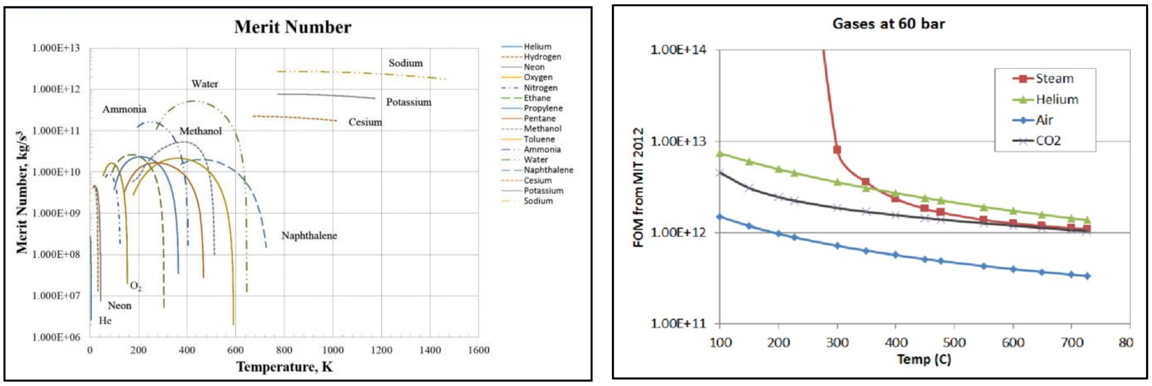

Figure 2.4 Merit number of different work fluids

Taken from G. Rice, "Heat pipe," DOI: 10.1615/AtoZ.h.heat_pipes, (2011)

Since the temperature difference of the thermosyphon investigated in this study is limited to the range of -10 to 10 Celsius, the choice of carbon dioxide gas was made according to the specification (table 2.1) of this material and result of previous related studies. [41]

Table 2.1 Physical properties of CO_2

Chemical formula	**CO_2**
Molecular weight	44,01 g.mol-1
Melting point (at a pressure of 0.5 MPa)	-56.6 ºC
Boiling point (at a pressure of 101.235 kPa)	-78.5 ºC
The critical temperature	31.01 ºC
The critical pressure	7.386 MPa
Gas density (0 ° C, 101.325 kPa)	1.965 kg/m3
Liquid density (-56.6 º C, 0.52 MPa)	1.178 kg/dm3
Heat of vaporization	571.08 kJ/kg

Moreover, CO_2 has some other advantages as work fluid, which is described briefly here according to reference [42]

- "For energy exploitation from heat sources with low-grade temperature associated with a low temperature heat sink, CO_2 transcritical power cycle is suitable."
- "It is inexpensive, non-explosive, non-flammable and abundant in the nature. In addition, it has no ozone depleting potential (ODP) and low global warming potential (GWP). As a pure working fluid, it also has better heat transfer characteristics than fluid mixtures. Moreover, due to its relatively high working pressure, the carbon dioxide system is more compact than the system operating with other working fluids."

CHAPTER 3

NUMERICAL ANALYSIS OF TWO PHASE CLOSED THERMOSYPHON

In this chapter, considering the requirements and assumptions mentioned for thermosyphon simulation in the previous chapter, first the results of simulation for different variables will be examined and then I will optimize the relevant variables to transfer the maximum amount of heat.

3.1 Simulation steps of Two Phase Closed Thermosyphon

Computer simulation is the method used in this research for mathematical modeling of the system to predict and calculate the behavior of the physical phenomenon or overall system performance by computer. This method of analysis is particularly useful and reliable, especially for those whose evaluation is very complex and difficult, and enables to analyze the behavior of the modeled system under different conditions.
The general algorithm of simulation and analysis of thermosyphon is performed according to the following diagram (Figure 3.1). As illustrated in the figure, the system is first simulated according to the previous chapter's description in the MATLAB program, and then the results with different initial values are recorded as the initial results. Then, by comparing with theoretical values, we tried to correct these values using mathematical optimization methods.

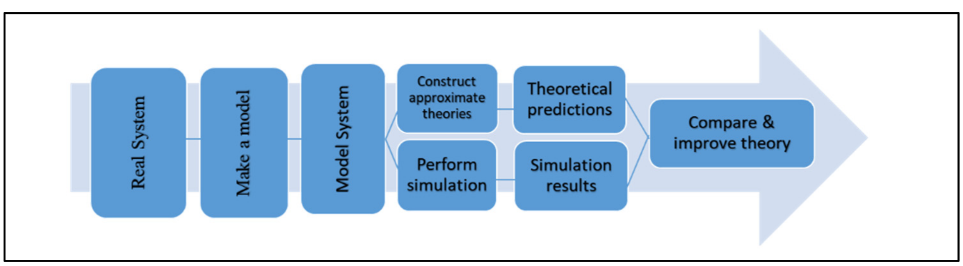

Figure 3.1 Algorithm of Mathematical simulation

3.2 Geothermal energy as heat source

Considering geothermal energy as a hot source and supplying the heat needed for evaporation of work fluid within the pipe, the system is independent of the supply of an external heat source in the evaporator section, which reduces cost significantly and benefits an unlimited and reliable energy source.

To calculate the amount of this heat, we have to enter the input values of Equation 2-27 according to the characteristics of the study area to obtain the average temperature at the required depth of the Earth.

Annual average temperature of soil for Montreal as my target city in three different depth 2, 3 and 4 meter is respectively equal to 6.3, 6.32 and 6.36C^{o}, calculated by analyzing the hourly weather recorded data (8760 data) for a year. By applying the variables in equation, the temperature gradient according to Montreal weather condition in depth of 4 meter, for instance, (As an initial estimation of approximate total length of Thermosyphon) illustrates in figure 3.2 and 3.3.

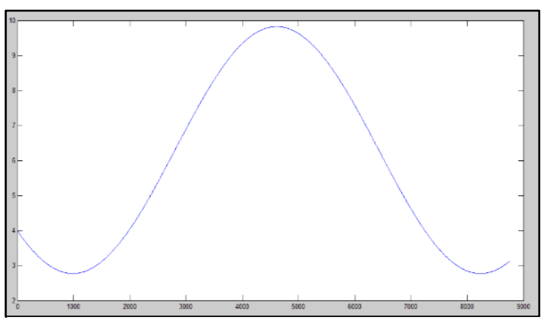

Figure 3.2 Annual ground temperature at a depth of 4 m in Montreal

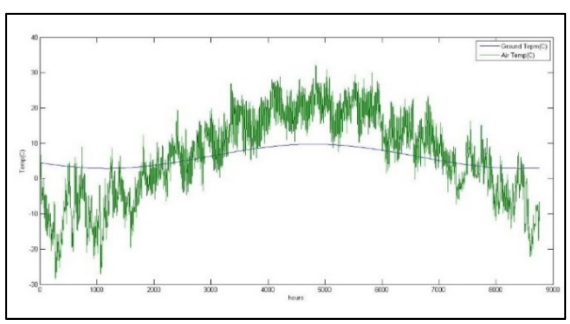

Figure 3.3 Ground temperature comparison to weather temperature - Montreal

In this project, the focus is on cold months (winter and fall) when the probability of snowing is higher. According to the Montreal weather data, the average temperature for this period is -5.19 C° and by using the equation 2-27, the soil average temperature will consider equal to 6.3 C°, which are used in further calculations.

3.3 Thermosyphon simulation

To simulate a Two Phase Closed Thermosyphon (TPCT), Due to the various basic conditions and different types of thermosyphon, the characteristics of the TPCT analyzed in this article include:

i. Circular tubes with uniform cross-section through all the sections.
ii. A single-component working fluid and non-condensable gas.
iii. There should be no wick insert in the different sections to improve liquid contact with the internal tube wall.
iv. 90° angle of inclination to the horizontal, with the evaporator at the lower end and condenser in the upper side of the thermosyphon.
v. Only the gravity forces drive the liquid from the condenser to the evaporator.
vi. Simulation based on the assumption that the evaporator part is inside the soil and the condenser part is placed on a layer of cement to transfer heat to the surface.
vii. (vii) Heat sink temperatures are assumed to be equal to the average months temperatures in Montreal.
viii. Thermosyphon tube is made of copper and the work fluid used is carbon dioxide gas.

In the simulation of the thermosyphon, it is assumed that the bottom portion is inside the soil (evaporator section), the primary portion is inside the cement (condenser section) and the two parts are separated by the middle section (adiabatic section).

The tube is made of copper and, as mentioned, CO_2 is selected as the work fluid in the tube to transfer the heat through the phase change process caused by the absorption of heat from the earth and transferring to the upper section of the thermosyphon. (Figure 3.4)

The temperature of the end of the pipe within the soil varies with different length of the pipe, which will calculate for each value corresponding to equation 2-27. The temperature of the condenser section is also considered as -5.19 degrees Celsius, according to the average temperature in Montreal's cold months (fall and winter).

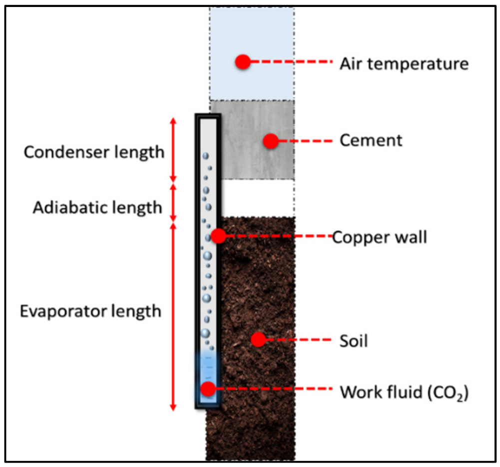

Figure 3.4 Environmental conditions of thermosyphon under study

The simulation is carried out according to the properties mentioned in Table 3.1 and the assumptions mentioned in the previous section.

Simulation performed by MATLAB and based on the trial-and-error method, because the heat transfer rate ($\bar{Q}$) is not known.

Table 3.1 Initial parameters for simulation

Item	Symbol	Quantity	unit
Slope degree	S	90	degree
Thermal conductivity of pipe - Copper	λ_{co}	400	W/(m.K)
Atmospheric pressure	Pa	$1.013e^5$	Pascal
Hours of the year of the minimum surface temperature	T_{shift}	0	hr
Soil specific heat capacity	CP_s	730	J/(Kg·C)
Density of soil	ρ_s	1400	Kg/m3
Density of cement	ρ_{cmt}	2500	Kg/m3
Thermal conductivity of Soil	λ_s	1.8	W/(m·K)
Thermal conductivity of Cement	λ_{cmt}	2.33	W/(m·K)
Average temperature of Montreal	T_{ave}	6.91	C°

The basis of the simulation is based on Equation 2-1 where the heat-transferring rate is related to the network of resistances that are replaced with different barriers against heat transferring and determining the temperature of the condenser and evaporator as well. Then, the first approximation to solve the problem can be made without considering the internal thermal resistance Z_3, Z_4, Z_5, Z_6 and Z_7. Later by achieving the initial value of heat, all the neglected resistances will be calculated according to heat transfer rate and this cycle will be repeat until the calculated Heat transferring rate will not change to a tangible amount.

As the algorithm in Figure 3.5 shows, by entering the values of the temperature of the two ends of the pipe, the heat transfer rate is finally determined by an acceptable approximation. These calculations can be done for each TPCT with its specific characteristics (including evaporator length, adiabatic length and condenser length, the amount of work fluid in the tube, inner diameter and thickness of pipe and the angle of the pipe relative to the horizon), which will be calculated and compared in the next section for several different tubes.

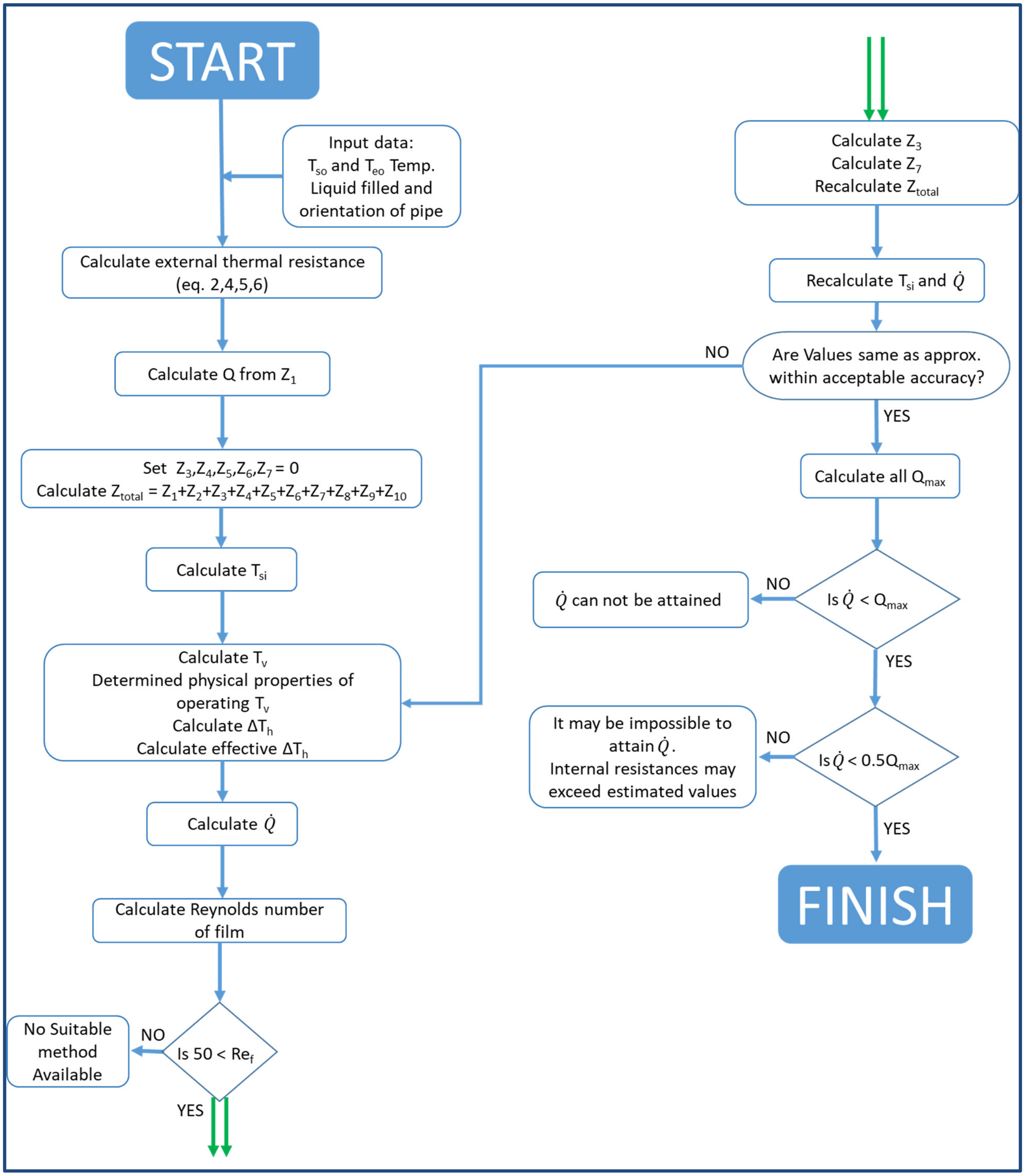

Figure 3.5 Algorithem for heat transferring rate calculation

Taken from Heat pipes – performance of Two phase closed thermosyphones, Engineering Sciences Data Unit - ESDU, (November 1983.)

3.4 Simulation results for a sample TPCT

There are an unlimited number of responses to this simulation, with five inputs to determine the thermosyphon heat transfer's performance. Evaporator length as a connection section to the heat source, condenser as section where is connected to heat sink ,adiabatic length, work fluid content inside the thermosyphon , thickness and radius of pipe are all variables that determine how simulated thermosyphon works and how efficient it is.

In this section, let us look at the performance of a TPCT with the following specifications:

- $\mathbf{L_e}$ (evaporator length) = 3 m
- $\mathbf{L_c}$ (condenser length) = 0.5 m
- $\mathbf{L_a}$ (adiabatic length) = 1 m
- $\mathbf{z_l}$ (depth of work fluid filled) = 1.2 m
- $\mathbf{D_o}$ (tube external Diameter) = 0.05 m
- **Tic** (tube thickness) = 0.003 m

The amount of resistances calculated according to the values mentioned for the dimensions of the pipe will be equal to:

$\mathbf{Z_1}$ = 0.1615 K/W	$\mathbf{Z_6}$ = neglected
$\mathbf{Z_2}$ = 1.6954e-5 K/W	$\mathbf{Z_7}$ = 0.0027 K/W
$\mathbf{Z_3}$ = 0.0154 K/W	$\mathbf{Z_8}$ = 1.017e-4 K/W
$\mathbf{Z_4}$ = neglected	$\mathbf{Z_9}$ = 0.5040 K/W
$\mathbf{Z_5}$ = neglected	$\mathbf{Z_{10}}$ = 14.5892 K/W

For calculating the total resistance of the system, first we should consider the equation 2-13 for choosing the appropriate equation:

$Z_{10} / (Z_2 + Z_3 + Z_5 + Z_7 + Z_8) = 799.6793 > 20$

According to the result, the total resistance will be calculated by equation 2-14 as follow:

$Z_{total} = Z_1+Z_2+Z_3+Z_7+Z_8+Z_9 = 0.6837$ K/W

Due to the high amount of resistance calculated, the heat transfer rate is extremely low and will be equal to:

$$\dot{Q} = \frac{\Delta T}{Z_{total}} = \frac{11.57}{0.6028} = 16.9982 \quad W$$

3.5 The effect of different TPCT sections length on heat transfer rate

Given that each TPCT consists of three main parts (evaporator, condenser and adiabatic) with different functions, determining the length effect of each mentioned parameters on the total heat transfer rate of the system can play an essential role in its optimal design.

Changes in the length of the evaporator due to the use of geothermal energy can directly affect the rate of heat received and ultimately increase the efficiency of the system. The length of the condenser can also cause significant changes in the heat transfer process due to the role of heat transferring to the heat sink part. The adiabatic section also plays a role in creating temperature differences between the two TPCT heads due to the task of separating the previous two sections.

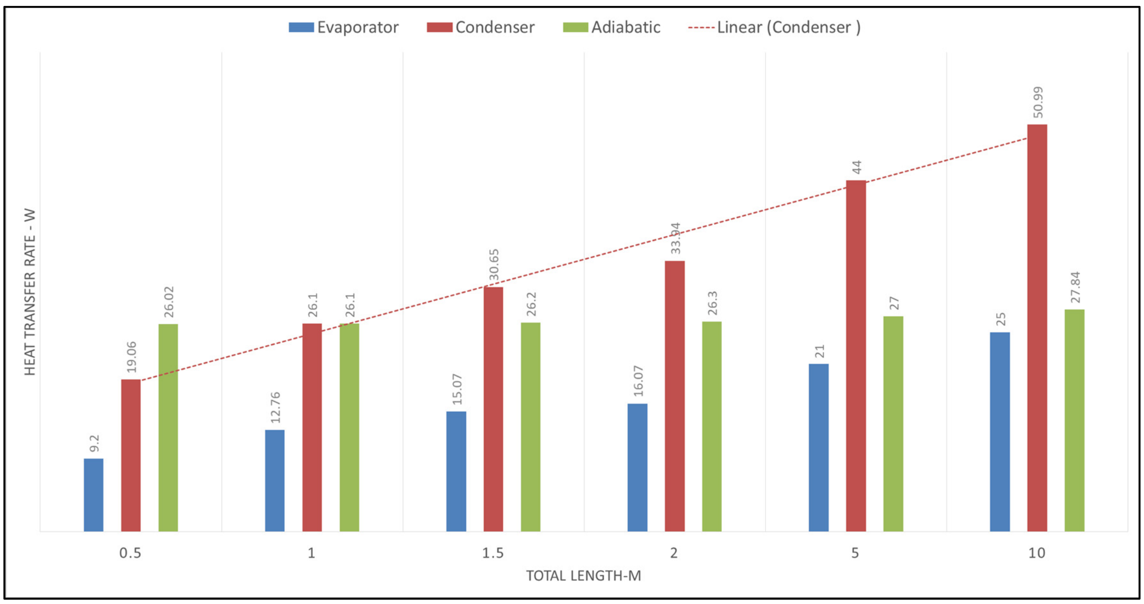

Figure 3.6 The effect of changing the length of different parts of thermosyphon on heat transfer rate

Calculations have been performed for each variable, assuming that the values of the other variables are as follows:

$L_e = 3$ m	$L_c = 0.5$ m	$L_a = 1$ m	$z_l = 0.5*Le$ m	Tic = 0.003 m	Do = 0.05 m

Looking at the diagram, what is most obvious is the effect of the changes in the length of the condenser on the rate of heat transfer compared to the other two parts. Increasing the length of the condenser by increasing its length or even increasing the contact area by equipping the system with a fin grid can greatly increase heat transfer. This conclusion, of course, has been applied in practice by using a network of pipes embedded in ground surface in Bad Waldsee, Germany to prevent the freezing of asphalt in the cold seasons of the year. [43] Increasing the length of the evaporator is also not a cost-effective way to increase heat transfer due to the slight changes in the earth's temperature at greater depths and due to the high cost of creating and examining holes with high depth inside the earth. In the next chapter, we will try to calculate the best pipe length by considering the optimal amount of heat transfer by using optimal solutions.

Another important point in this case is the error "incomplete wetting of the heated surface" for all the results except the last two values for the condenser length. This error is due to the low Reynolds number in the calculations, which will later be applied as one of the conditions and constraints for calculating the optimal dimensions of the thermosyphon in the calculations.

3.6 The effect of different TPCT Diameters and thicknesses on heat transfer rate

The effect of these two parameters on the heat-transferring rate is reversed, so that with increasing pipe outlet diameter, naturally more fluid has the capacity to be transferred from the thermosyphon hot part to the cold section and thus higher heat transfer capacity.

Unlike the outer diameter, as the thickness of the tube increases, the rate of heat transfer decreases, which is inversely related to the increase in resistance due to the thickness of the tube.

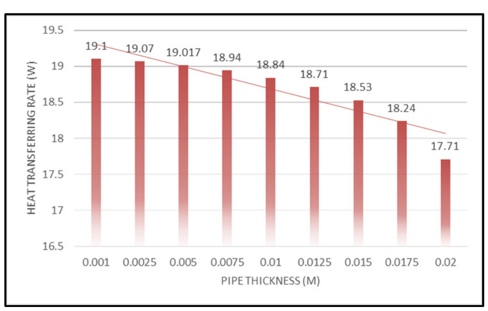

Figure 3.7 The effect of Thickness change on heat transfer rate

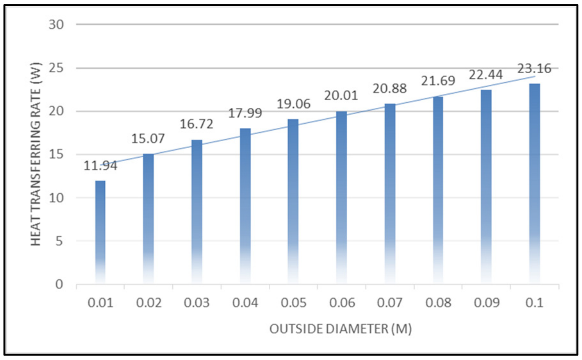

Figure 3.8 The effect of external diameter change on heat transfer rate

The relationship between the change in values of these two parameters and the amount of heat transfer in the diagram is shown in figures 3.7 and 3.8. The point to consider in this section is the limitations that exist in the selection of the above values in practical design.

Despite the direct relationship between increasing pipe outlet diameter and heat transfer rate, increasing this parameter can have a negative impact on the ability of work fluid vapor in evaporators, Reynolds number, prevent fluid return from condenser etc., which should be considered in designing and calculating optimal values.

In calculating the thickness of the pipe, reducing this amount with the aim of increasing the heat transfer capacity is subject to design restrictions and environmental conditions too.

3.7 The effect of work fluid filled length on heat transfer rate in TPCT

In this section, we look at the effect of the amount of work fluid on the heat transfer range in thermosyphones. The thermosyphon analyzed in this section has the fixed geometric dimensions (L_e = 3m, L_a = 2m, L_c = 1m, D_o = 0.1m, Tic = 0.0005)

The rate of heat transfer rate in eight different heights of the work fluid is displayed the Figure 3.9. According to the proposed range of 40% to 60% of the working fluid height relative to the total length of the thermosyphon [44], these coefficients are selected as $0.3L_e$ to L_e to illustrate the effect of changing on liquid filled height beyond the appropriate range.

As can be seen in Figure 3.9, the amount of heat transfer decreases as the working fluid inside it increases due to the selected geometric values for the thermosyphon. Therefore, the lowest permissible fluid level (40% of the total length) must be used to achieve the highest heat transfer rate. This relationship, of course, can behave differently depending on the different geometric characteristics of the thermosyphon, which must be examined in proportion to each geometry.

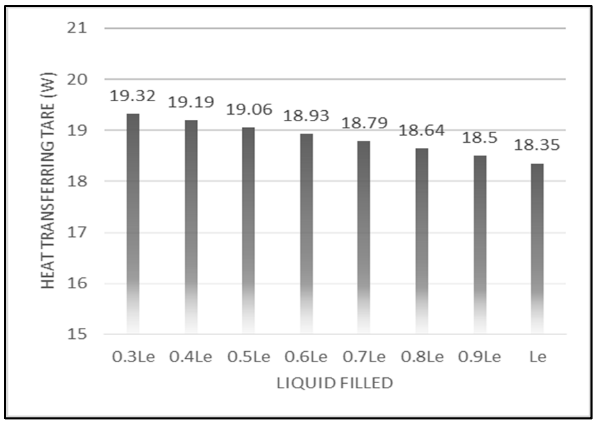

Figure 3.9 Heat transferring rate base on liquid filled height

CHAPTER 4

OPTTIMAZATION OF TWO PHASE CLOSED THERMOSYPHON

In this section, we try to determine the appropriate method for finding the most optimal values for its design variables according to the characteristics of the equations governing the rate of heat transfer in a thermosyphon and calculate the values considering the existing constraints.

4.1 Introduction the Optimization of a Design

From the point of mechanical science view, any system, regardless of its functionality, consists of a network of passive subsystems, which by combining the performance of each of them, the final performance of the whole can be analyzed. In the industry, these systems, taking into account their constituent factors, form a complex network. The analysis of these networks has always been a very slow process, and examining their performance is time consuming, costly, and requires access to many human and scientific resources.

The goal of optimizing a design is to simulate these complex systems in mathematical language and then try to improve its performance through the mathematical relationships that govern these equations, even if they are not the best answers. In this growing industry, computer programming, especially in recent decades, has been of great help to engineers, so that it is possible to analyze and optimize a network of complex systems in a short time.

In terms of optimization, it is important to identify all variables and define their relationship to the system output. Simulating a system and formulating its performance, which is similar to its practical results, may take years, but it can certainly have a better outcome in analyzing the optimization of its variables.

In this section, due to the existence of different variables and their different effects on heat transfer rate, an attempt has been made to convert the performance of a thermosyphon in the standard form defined for optimization and finally calculate the optimal values.

4.2 Optimum Design Problem

To achieve the standard form for using the optimization methods, we need to categories all the equations in two main types:

A. **Main equation (Cost function)**
B. **Constraint equations (Restrictor equations)**

The main condition for determining the optimal values of the input variables of a system is that we can make the relationships and equations governing it in the form of the two equations mentioned and then use it as the standard form in programming. (Equation 4-1)

$$\begin{aligned} &Minimize - x \qquad F(x) \\ &Subject\ to: \quad \begin{pmatrix} g_i(x) \leq 0 & i = 1,.....,m \\ h_j(x) = 0 & j = 1,.....,p \end{pmatrix} \end{aligned} \tag{4-1}$$

A. **Main equation (Cost function) – F(x):** The main function of optimization operations is to find the minimum (maximum) of this equation. In short, the final evaluation of the performance of a system is the result of this equation. The answer to this equation is the value that is used because of optimizing the input variables in the main function in the calculations.
   This function must include all input variables directly or indirectly and can be linear or non-linear, which will have its own solution depending on the type. Numerical methods of optimization are usually in the form of trying to find the minimum of a cost function for the desired inputs, but by placing the negative value of this function, its maximum values can be calculated.

B. **Constraint equations (limiting equations) – $g_i(x)$ and $h_j(x)$:** These equations and constraints limit the area in which the variables or results of the equations must be defined or calculated.

The definition of these areas can be in the form of a point, a line, a surface, or even a three-dimensional area in which the variables or results must be defined. The equations of constraints can be in the form of linear or nonlinear equations, equal to or unequal, or a combination of these two types of equations, depending on the type of solution.

What is important in defining the equations of constraints is that these equations must be independent of each other, and the sum of all these equations must together form a feasible region. (Page 25 of reference [45]).

4.3 Formulation

In this section, I am going to turn the heat transfer process in a Thermosyphon into a standard form according to the simulation done last season, and then calculate the optimization operation for system input variables with the goal of finding the maximum amount of heat transfer.Therefore, first, we define the functions and relationships governing this process systematically and then by finding the best optimization method using MATLAB program, we determine the amount of these values.

Step 1: Project/problem description

Given that the goal of optimal thermosyphon design in this research is to find the maximum amount of heat transfer, so the main function (cost function) in this system is in accordance with equation (2-1) in this thesis, where the heat transfer capacity is defined according to its equivalent resistors and the temperature difference between the two ends:

$$\dot{Q} = \frac{\Delta T}{Z_{total}}$$

Step 2: Data and information collection

All variables and parameters in the equations are divided into two main categories:

1. Design variables.
2. Fixed variables.

Design variables include variables whose their values are not known and are used as final response indicators. The goal of system optimization is to find their values that ultimately minimize (maximize) the cost equation. Fixed variables are values that have a constant value in each case and their value does not change due to a change in status.

As stated in the first part of this thesis, the thermosyphon under study is of two-phase type and its body material is copper. The horizontal angle of the thermosyphon is constant at 90 degrees. The lower part of the system is on the ground and the upper part is covered with cement. The work fluid inside the pipe is carbon dioxide, the system lacks the wick system for transferring the condensed fluid to the evaporator and instead, this process is done through the gravity of the earth. Fixed variables in designed thermosyphon are shown in the table 4.1.

Table 4.1 Initial data for optimazation

Item	Symbol	Quantity	unit
Slope degree	S	90	degree
Thermal conductivity of pipe - Copper	λ_{co}	400	W/(m.K)
Atmospheric pressure	Pa	$1.013e^5$	Pascal
Hours of the year of the minimum surface temperature	T_{shift}	0	hr
Soil specific heat capacity	CP_s	730	J/(Kg·C)
Density of soil	ρ_s	1400	Kg/m3
Density of cement	ρ_{cmt}	2500	Kg/m3
Thermal conductivity of Soil	λ_s	1.8	W/(m·K)
Thermal conductivity of Cement	λ_{cmt}	2.33	W/(m·K)
Average temperature of Montreal	T_{ave}	6.91	C°

Step 3: Definition of design variables

The design variables for the cost function defined in Section 4..31 include all dimensions of the pipe as well as the height of the work fluid filled inside the pipe.

$\mathbf{L_e}$ = Evaporator section (meter)

$\mathbf{L_c}$ = Condenser section length (meter)

$\mathbf{L_a}$ = Adiabatic section length (meter)

$\mathbf{D_o}$ = Outside diameter length (meter)

Tic = Thickness of pipe (meter)

$\mathbf{z_l}$ = Depth of work fluid filled in pipe (meter)

Step 4: Optimization criterion

Now that the Cost function and Design variables for the proposed problem have been determined, the question can be expressed in the standard form of optimization:

Minimization of:

$$\dot{Q}(L_e, L_c, L_a, Tic, D_o, z_l) = -\frac{\Delta T}{Z_{total}} \qquad (4\text{-}2)$$

Now to get to the standard form (Equation 4-1), it is enough to specify the Constraints equations and choose the method that fits the type of existing equations.

Step 5: Formulation of constraints

Determining constraint equations is very important in optimizing a system. These equations should determine all the limitations that directly or indirectly (subsystem outputs) affect the design variables. In practice, in the case of the thermosyphon under study, in addition to the physical limitations, a function including evaluation and limitation of project costs should also be applied to the system, which is beyond the scope of this thesis but can be applied in the future.

In this thesis, to limit the values of the Design variables directly, only an attempt has been made to understand the logic of the dimensions of thermosyphon pipes according to the patterns available in the market. Therefore, the following restrictions have been applied to them:

Table 4.2 Thermosyphon geometry constraints

Constraint number	Item	Symbol	unit
(C1)	Length of evaporator	0.1≤Le≤3	meter
(C2)	Length of condenser	0.1≤Lc≤1	meter
(C3)	Length of Adiabatic	0.1≤La≤3	meter
(C4)	External Diameter	0.001≤Do≤0.50	meter
(C5)	Thickness of pipe	0.001≤Tic≤0.0001	meter
(C6)	Depth of work fluid filled	0.1≤DF≤3	meter

The second phase of the constraint equations is related to the allowable values of the maximum heat transfer rate that was mentioned in the second chapter. (Equations 2-16 to 2-26) .In applying optimization operations, we must also consider the condition that the heat transfer values do not exceed these values in standard for as follow:

$\dot{Q} - \dot{Q}_{\max(1)} \leq 0$ Boiling limit (C7)

$\dot{Q} - \dot{Q}_{\max(2)} \leq 0$ Sonic limitation (C8)

$\dot{Q} - \dot{Q}_{\max(3)} \leq 0$ Counter flow limit (C9)

Another constraint equation is related to limit the amount of work fluid can be inside the tube. This measure, as stated in Section 2.4.4, can be calculated through equation 2.22 and its standard form for optimization is as follows:

$0.5 \leq F$ Counter flow limit (C10)

$ii.\ \ V_l \geq .001 D (L_e + L_a + L_c)$ Counter flow limit (C11)

Where:

V_l is the volume of Work fluid filled in the pipe.

The next constraints determine the range that vapour Temperature of the work fluid (CO_2) can be defined:

$T_{V\,(K)} - 304.13 \leq 0$ Vapour Temperature (C12)

$216.59 - T_{V\,(K)} \leq 0$ Vapour Temperature (C13)

Finally, the allowable total length of the thermosyphon according to equation 2.26 will be define as:

$T_{total} - T_{allow} \leq 0$ Total length limitation (C14)

The common denominator of all mentioned constraint equations (C1 to C14) will be the area that includes points that can maximize or minimize the cost function. The purpose of optimizing a function is to actually find the area, calculate and compare the value of the cost function for these points. These points are often located at the boundary points of the constraint functions and may not be unique.

Due to the non-linear nature of the constraint functions, the Sequential Quadratic Programming (SQP) method is selected to solve the optimization of the desired function.

4.4 Sequential quadratic programming (SQP) method for optimization

Briefly, Sequential quadratic programming (SQP) is an iterative process (mathematical procedure which uses an initial guess to provide a sequence of improving approximate solutions) for constrained nonlinear optimization.

The general form of the method is as follows [45]:

$Minimize - x \qquad F(x) \ as \ quadratic \ \ problem$

$F(x) = C^T x + 0.5 x^T Hx$

$Subject\ to: \begin{pmatrix} A^T x \leq b & x \geq 0 \\ N^T x = e & x \geq 0 \end{pmatrix}$

And the Lagrangian for this problem is:

$$L(x, \lambda, \mu, \zeta) = C^T x + 0.5 x^T Hx + \lambda^T (N^T x - e) + \mu^T (A^T x + s - b) - \zeta^T x \qquad (4\text{-}3)$$

Where:

C = Gradient Matrix of Cost Function - ∇ F(x)

H = Hessian Matrix of Cost Function - ∇^2 F(x)

T = Matrix of transpose

A = Matrix of coefficients of inequality functions

N = Matrix of coefficients of equality functions

λ = Lagrange multiplier for equality

μ = Lagrange multiplier for inequality

ζ = Lagrange multiplier associated with $x \geq 0$

S = Slack variable

By applying the First Order Necessary condition of optimizing ($\nabla L = 0$), and solving the resulting equations, x values are finally calculated.

4.5 Optimization results of a TPCT

In this section, we will discuss the results of optimizing the cost function under study (the rate of heat transfer in the thermosyphon) with Matlab programming under the conditions of the constraint functions mentioned in the previous section.

The calculated values for the input variables according to the defined intervals are as follows:

$\mathbf{L_e}$ = 3 meter

$\mathbf{L_c}$ = 1 meter

$\mathbf{L_a}$ = 3 meter

$\mathbf{D_o}$ = 0.50 meter

Tic = 0.0001 meter

$\mathbf{z_l}$ = 1.5 meter

As expected, most of the above values are calculated in the boundary areas of the defined ranges. These values were calculated by counting seven iterations and the following output in MATLAB program:

Iterations: 17	lambda :
Func-Count: 126	eq-lin: [0x1 double]
Algorithm: 'sequential quadratic programming'	eq-non-lin: [0x1 double]
Message: [1x777 char]	in-eq-lin: [0x1 double]
Constr-violation: 0	lower: [6x1 double]
Step-size: 1	upper: [6x1 double]
First-order-opt: [1x1 double]	in-eq-non-lin: [9x1 double]

Given the optimal values calculated, we once again calculate the average heat-transferring rate, considering that the constant values in Table 4.1 are also true for these calculations.

$$\dot{Q} = \frac{\Delta T}{Z_{total}} = \frac{11.70}{0.2387} = 49.02 \quad W$$

The calculated value for the heat transfer rate seems to have increased in comparison with the last calculated value for the thermosyphon with same dimension limitation. What is not considered in these optimal calculations is the “Reynolds number”. The Reynolds number for the last calculation could make “incomplete wetting of the heated surface” according to reference [32].

4.6 Reynolds number calculation for the designed TPCT

The Reynolds number plays an important role in preventing dryness inside the pipe as well as in estimating the calculated resistance. The following equation can be used to calculate the Reynolds number in a thermosyphon:

$$\mathrm{Re}_f = \frac{4\dot{Q}}{SL\,\mu_l \pi D_i} \tag{4-4}$$

Where:

SL = Specific latent heat of vaporization

μ_l = Dynamic viscosity of the liquid

D_i = Inner Diameter of pipe

As it is mentioned in reference [32], the overall thermal resistance of the system may be appreciably greater than the amount was calculated in this thesis, especially for the internal thermal resistance of the work fluid boiling in evaporator (Z_3), when the Reynolds number of the liquid film is less than 50 in the adiabatic section. For the optimal dimensions calculated for the designed Thermosyphon, the Reynolds number according to equation 4-4 will be equal to 5.75, which is not satisfies this condition for preventing dryness inside the tube through the boiling in evaporator.

To apply the mentioned limitation, the condition that the Reynolds number should be greater than 50 must be added to the constraint functions in the optimization operation, which will affect the acceptable thermosyphon dimensions and its related heat transferring capacity.

$50 - \mathrm{Re} \le 0$ Reynolds limitation (C15)

4.8 Optimized thermosyphon classification based on total length

One of the biggest challenges in designing heat pipes and thermosyphons, whose their heat source is from geothermal energy, is to determine its total length in order to obtain the desired heat transfer range, taking into account the cost of drilling the earth in order to place the pipe in it. For this purpose, in this section, I have divided the optimal geometric values of thermosyphon according its total length and the result of the heat transfer rate for each one with the proposed dimensions.

Categorized thermosyphons are analyzed in the following five proposed categories:

1) TPCT with total length of less than 1 meter
2) TPCT with total length of length up to 3 meters
3) TPCT with total length from 3 up to 6 meters
4) TPCT with total length of 10 meters and less
5) and finally, TPCT with total length of 15 meters and less.

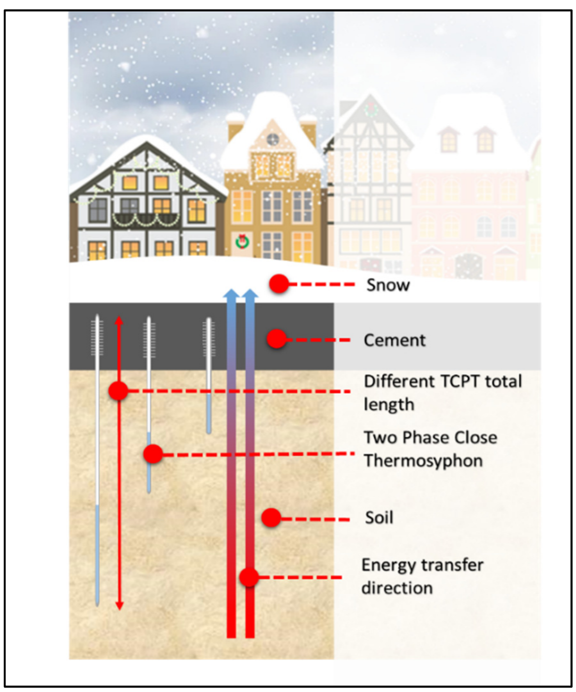

Figure 4.1 Different TPCT comparison

Due to the limitation for the length of the condenser in this project (asphalt infrastructure up to the surface), the length of the condenser is equal to one meter or less in all the assumed steps.

As shown in Table 4.3, the heat transfer values increase with increasing the total length of thermosyphon. According to the results, it seems that due to the limitation of condenser's length, increasing the length of the evaporator (increase the contact area of the heat source and work fluid) is the only factor, which can increase the heat transfer rate.

Table 4.3 Optimal values of thermosyphon geometric based on its total length

Total length(m)	Le(m)	Lc(m)	La(m)	Do(m)	Tic(m)	Df(m)	Q(W)	Reynolds
Under 1 meter	0.50	0.4	0.1	0.03	0.00154	0.25	4.84	58.16
1 to 3 meter	1.23	0.87	0.1	0.023	0.009	0.97	11.61	235.06
3 to 6 meter	4.9	1	0.1	0.034	0.0001	2.45	24.8	51.14
6 to 10 meter	8.9	1	0.1	0.03	0.0001	4.45	28.18	60.41
10 to 15 meter	13.9	1	0.1	0.03	0.0001	6.95	30.30	66.67

Increasing the total length, of course, is accompanied by a gentle slope rate of heat transfer rate and is moving towards a constant value with increasing thermosyphon length. (Figure 4.2)

The logarithmic trend-line equation for heat-transferring rate through the increasing of total length is equal to:

$$Y = 17.105\ln(X) + 3.567 \quad (4\text{-}5)$$

Where:

Y = heat-transferring rate

X = Total length of thermosyphon.

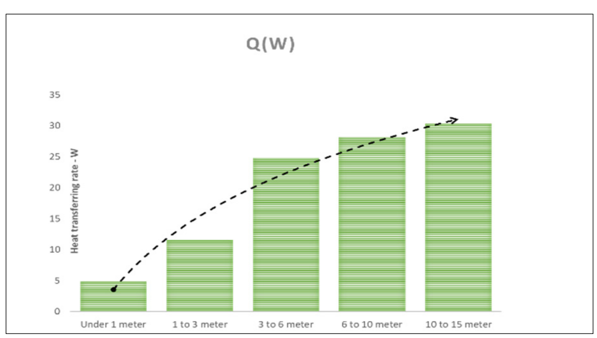

Figure 4.2 Heat transfer rate trend line

CHAPTER 5

CFD ANALYZING OF OPTIMIZED THERMOSYPHON

In this section, I have tried to provide a 2D analyze for one of the optimized thermosyphons in Ansys software to measure the performance of the fluid in the analytical conditions mentioned in the previous sections. The purpose of the analysis and modeling in this section is to verify the behavior of the system according to the assumptions considered in the thermocouple simulation section and verification of the results calculated in MATLAB.

5.1 Preliminary settings for Thermosyphon modeling

In order to analyze the work fluid behavior inside the thermosyphon in this project, Ansys 19.2 and fluent section have been used. This design includes four main parts includes: body geometric design, component meshing, definition of analysis type and boundary conditions and finally the results of analysis, each of which is briefly described in this section.

5.1.1 Geometric design of thermosyphon

The dimensions of the thermosyphon analyzed are randomly selected from one of the calculated optimized designs (Table 5.1), which its dimensions are briefly listed in the following table:

Table 5.1 Geometric properties of thermosyphon analyzed by CFD

Parameter	Description	Value
S	Slope degree	90
Le	length of evaporator in meter	4.9
Lc	length of condenser in meter	1
La	length of Adiabatic in meter	0.1
Do	external Diameter in meter	0.03
Tic	Thickness in meter	0.0001
Df	the depth of fluid filled in meter	2.4

The thermosyphon designed includes five main parts: 1.Evaporator section at the bottom of the system 2. Adiabatic section is located in the middle of the thermosyphon and is insulated. 3. Condenser in the highest part of thermosyphon 4. Pipe with a thickness of 0.5 mm and copper material 5. The inner part of the pipe for work fluid flow. (Figures 5.1 and Table 5.1)

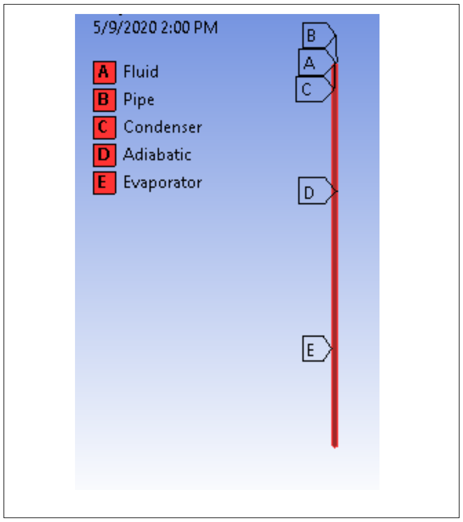

Figure 5.1 Different designing sections for CFD analyze

5.1.2 Meshing the designed Thermosyphon

The system meshing is considered according to the details of changing the fluid state during the process and its behavior during the phase interaction. The mesh size is considered smaller than the rest of the pipe due to the sensitivity of heat transfer in the walls and fluid density on both sides of the pipe. (Figure 5.2)

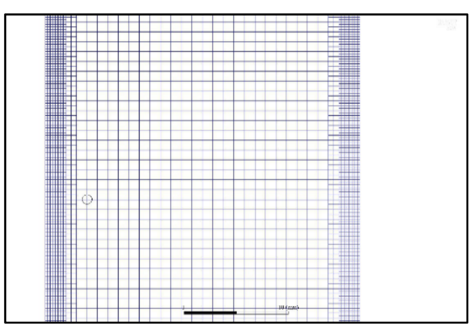

Figure 5.2 Meshing of TPCT for CFD analyze

5.1.3 Analyze settings and boundary conditions

For CFD analysis of TPCT, the setting of calculation is based on activation the “Multiphase model” (Volume of Fluid) due to the existence of two different phases of fluid in the process, “Energy model” due to the transfer of energy during the process of fluid phase changing and finally the “Viscous model” in Laminar condition according to the Reynolds calculated in previous section. The working fluid selected for the system (according to the project subject) is carbon dioxide, which the characteristics are defined for both phases, fluid and gas, at its evaporation temperature (phase changing point).

The boundary conditions defined for the system (Figure 5.3) are divided into the following sections according to the subject of the project:

A. Evaporator section with a constant external wall temperature of 280 Kelvin (The temperature inside the earth)
B. Adiabatic section with zero heat transfer flux in the walls (insulated thermosyphon section)
C. The condenser section with constant wall temperature set in 268 Kelvin (The average temperature of the cold seasons in Montreal)
D. Copper pipe material with the characteristics defined in the software.

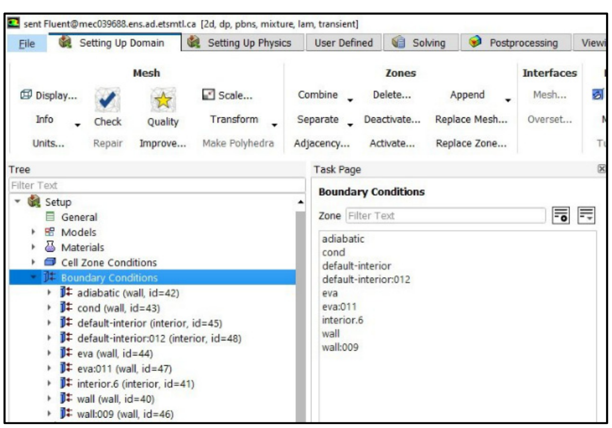

Figure 5.3 Boundary condition of TPCT analyze

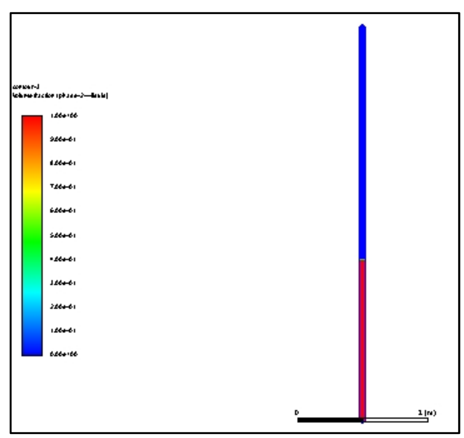

Figure 5.4 Definition of liquid region in Thermosyphon

In the last step of boundary condition definition, the fluid inside the thermosyphon is determined by defining a region with 2.45 meters height above the bottom of the pipe and is completed by activation the acceleration of the Earth's gravity by -9.8 meters per second of the boundary design. Finally, the system run by 500-time step with 1e-05 step size (s) and 100 max iterations for each step calculation.

5.2 Work fluid behavior through CFD analysis

As shown in Figure 5.5, the liquid reaches its critical point due to the heat supplied by the ground, which is carried by the walls of the tube containing the fluid, and the phase changing happen. The ratio of CO_2 fluid velocity in the inner walls of the tube to the center of the tube is quite clear in the Figure 5.5 .This behavior, as predicted, is caused by the pressure inside the pipe, which reduces the evaporation temperature of the fluid to a temperature below the Earth's internal temperature. Figure 5.6 shows how carbon dioxide in gas format is formed by the fluid vapor process.

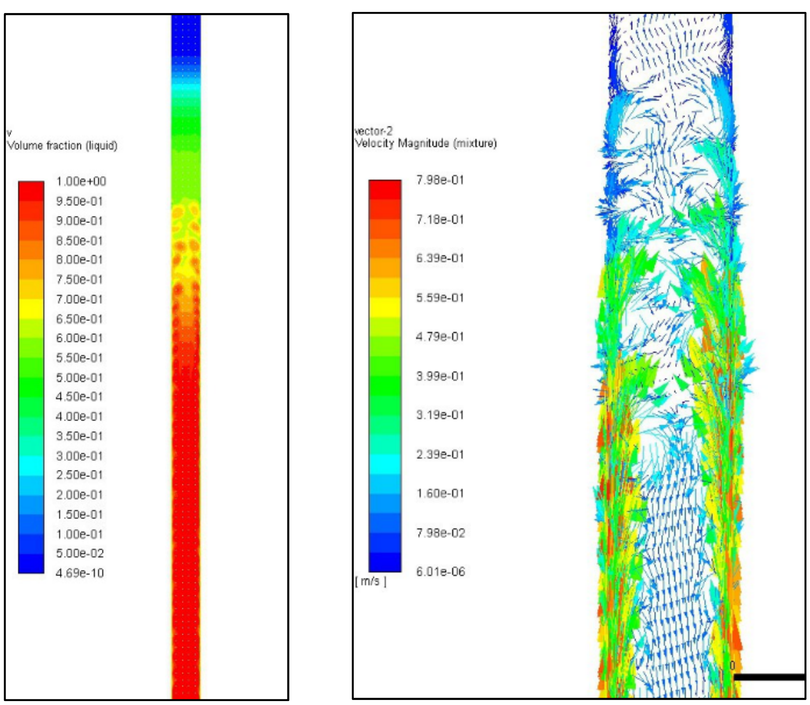

Figure 5.5 - Evaporation of the fluid inside the thermosyphon

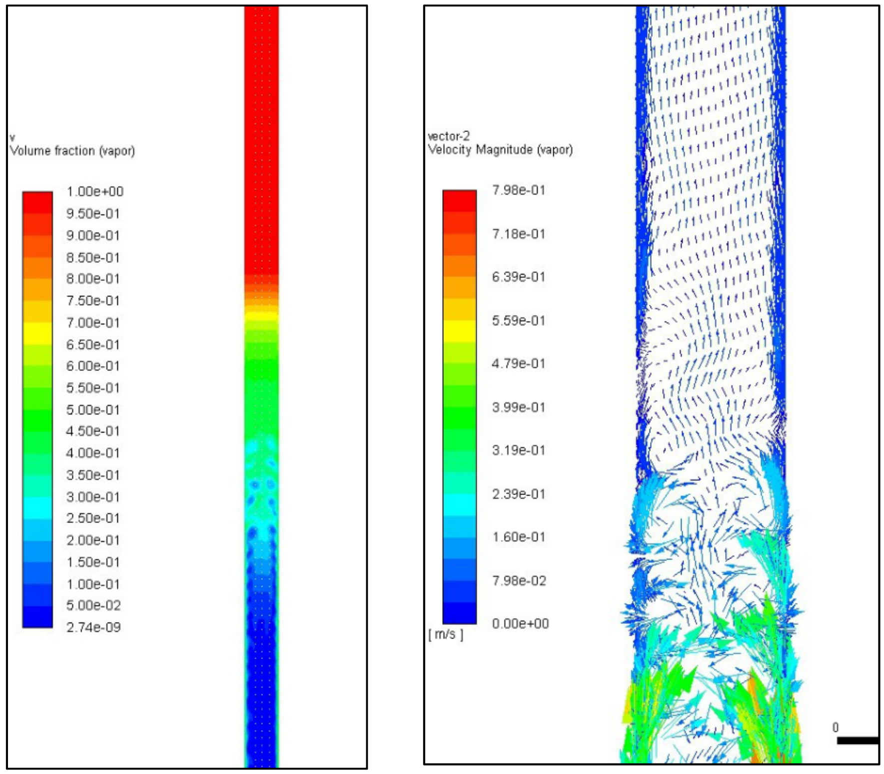

Figure 5.6 Forming of the gas format of CO_2 in Condenser

Figure 5.7 shows the pressure distribution inside the pipe. The maximum pressure is at the bottom of the pipe (pool pressure) which causes the fluid to reach the point of phase change at a lower temperature. The pressure difference between the two ends of the pipe, as shown in the figure 5.7, causes the flow of fluid vapor to the area with less pressure (the upper part of the pipe).

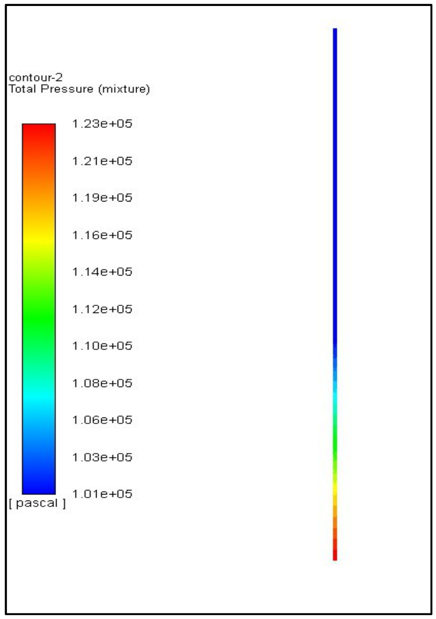

Figure 5.7 Pressure distribution through the Thermosyphon

The velocity vector diagram according to the velocity in the fluid and vapor is illustrates in Figure 5.8. As mentioned earlier, due to the transfer of heat by the evaporator walls to the work fluid, the velocities of the fluid adjacent to the sidewalls show a greater value. This amount is less intense by moving from both sides to the center of the pipe.

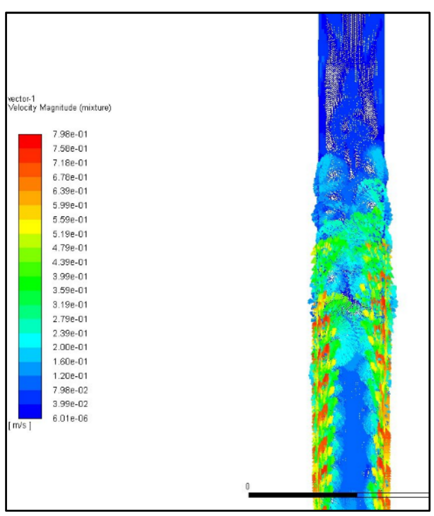

Figure 5.8 Velocity vector at Thermosyphon

The behavior of velocity vectors for carbon dioxide gas is shown in Figure 5.9 and 5.10. As predicted, condensed vapor in condenser is moving down the pipe using the sidewalls of the pipe, and the high-pressure vapor is moving upward from the evaporator through the middle parts of the pipe. The behavior of the fluid at the end of the condenser is quite clear where by hitting the surface with a lower temperature; condensation occurs and moves downward from the two sidewalls.

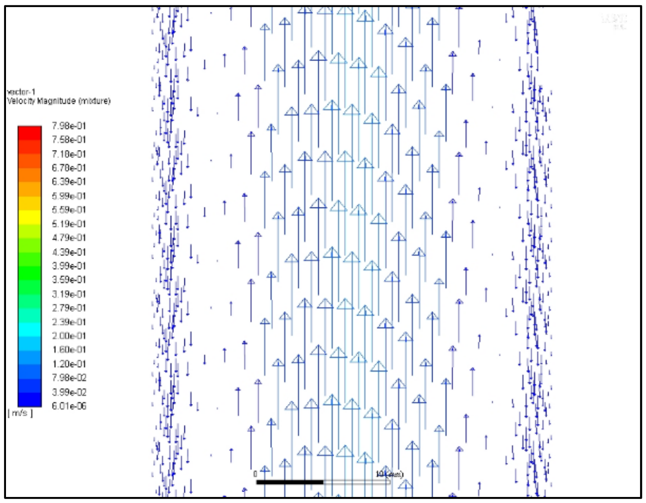

Figure 5.9 Work Fluid flow behavior in the process

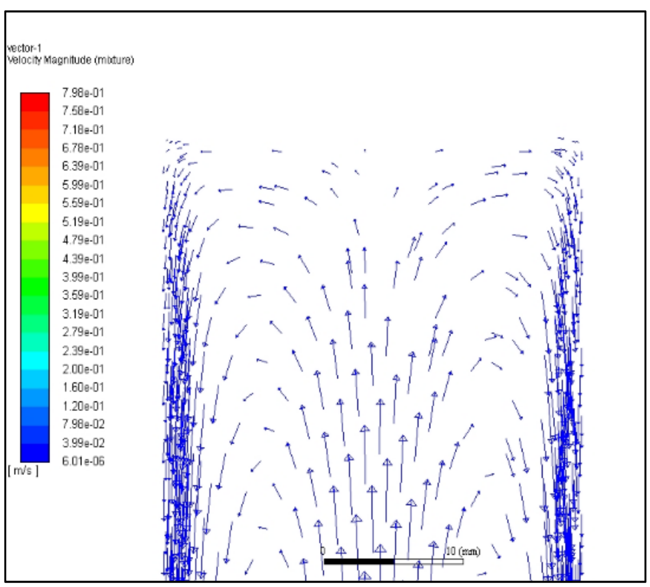

Figure 5.10 Work Fluid flow behavior at the top of the condenser

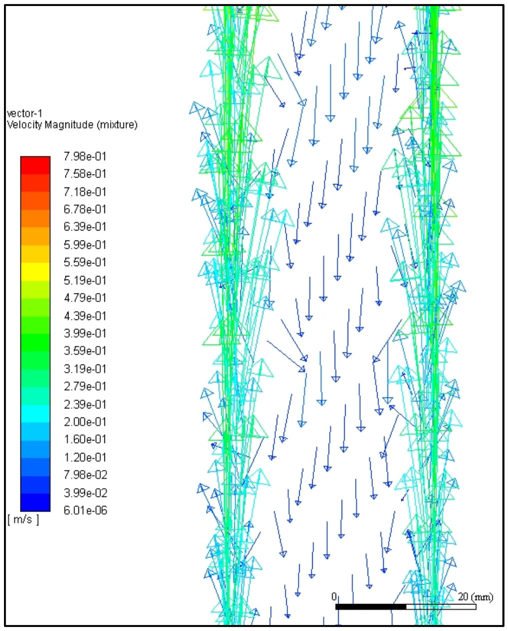

Figure 5.11 Work Fluid flow behavior at the evaporator

5.3 Heat transfer rate through CFD analysis

After analyzing the fluid behavior in the previous section and comparing it with what was predicted, in this section, we would compare the amount of heat transfer through the desire boundary conditions in the thermosyphon with the dimensions mentioned by the CFD analysis. The amount of heat transfer rate in this analysis is equal to 22.876 Watts per square meter, which is 1.92 Watts less than the amount of heat transfer calculated by the MATLAB (24.8 Watts per square meter).

The difference in this value can be mainly due to the more accurate analysis of Ansys Fluent on fluids behavior. In the simulation by MATLAB, the effect of some resistances and the formation of the fluid layer in the walls on the amount of heat transfer were ignored while these values were all taken into account in the calculation by Ansys.

```
"Volume Integral Report"

                 Mixture
          Mass-Weighted Average
        Total Surface Heat Flux          (w/m2)
------------------------------- --------------------
                    Gas          16.135868
                  Liquid          23.155097
            ---------------- --------------------
                    Net          22.876703
```

Figure 5.12 Total heat transfer rate – W/m2

5.4 Heat transfer evaluation for fin-equipped thermosyphon through CFD analysis

The calculated amount of heat transfer is very small and is unjustifiable for practical use in energy transferring. As can be seen, the biggest reason for the low amount of heat transfer is due to the resistance of Z_1 and Z_9 (resistances through external wall of the evaporator/condenser - hot source/heat sink). To solve this problem, we need to look at the formula for calculating these two resistors to determine which factors can be modified.

Due to the limited selection of heat transfer materials covered the TPCT, increasing the contact surface can be a solution for increasing the efficiency. Therefore, by adding a network of fins in the outer walls of the evaporator and condenser, the contact surface area will increase and as a result, the capacity of heat absorbed in the evaporator and dissipated in the condenser is likely to increase. According to the "Handbook for transversely finned tube heat exchanger design" [46], the blades are considered with the following geometric features:

Table 5.2 Fin specification (m)

H_{fin}	**Fin height (m)**	**0.02**	**D_{fin}**	**Fin Diameter**	**$D_o + 2L_f$**
Tic_{fin}	**Average fin Thickness (m)**	**0.001**	**L_{fin-e}**	**Length of finned segment evaporator**	**$0.9L_e$**
Sp_{fin}	**Fin spacing (m)**	**0.002**	**L_{fin-c}**	**Length of finned segment condenser**	**0.9Lc**

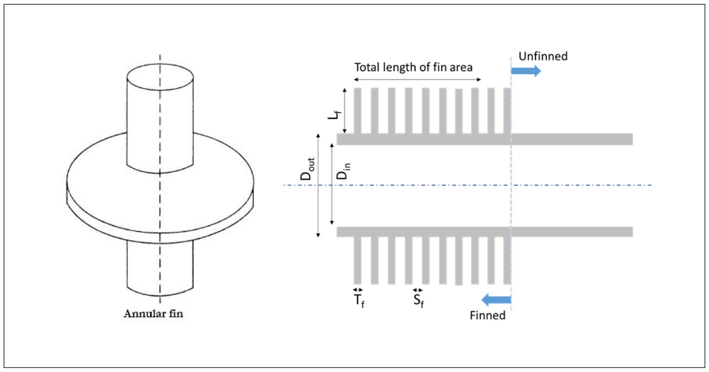

Figure 5.13 Equip the thermosyphon to the fins
to increase heat transfer capability

Increasing the area of contact of the thermosyphon with the hot / cold environment can increase the heat absorption/dissipation capacity of the system with its surroundings. In calculating the amount of equivalent resistance, equipping the system by fins would directly affect the amount of conduction shape factors (SF) in both evaporator and condenser, which leads to decreasing the resistances Z1 and Z9.

To evaluate the effect of the fins on the heat transfer rate by Ansys Fluent software, I equipped 90% of the total length of the evaporator and condenser with blades with the characteristics mentioned in Table 5.2 and Figure 5.13. The images of a fin-equipped tube are shown in Figure 5.14, 5.15 and 5.16.

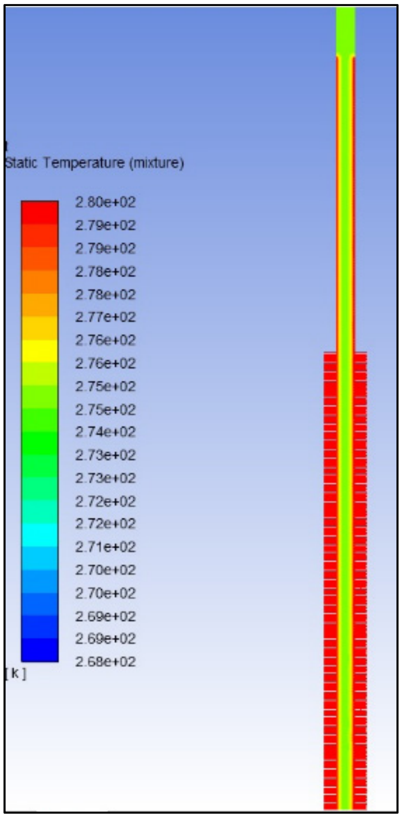

Figure 5.14 Evaporator equipped with fins

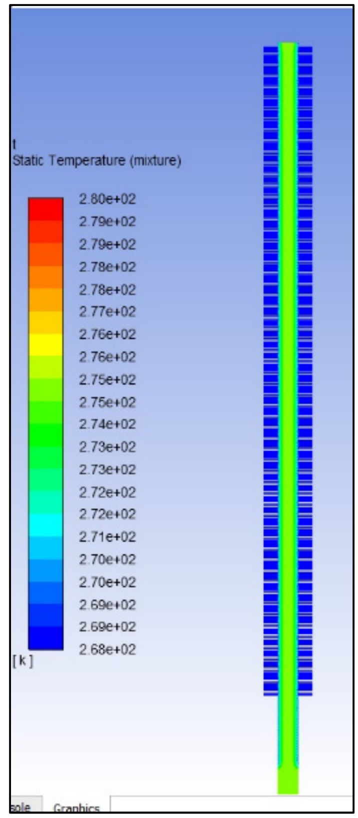

Figure 5.15 Condenser equipped with fins

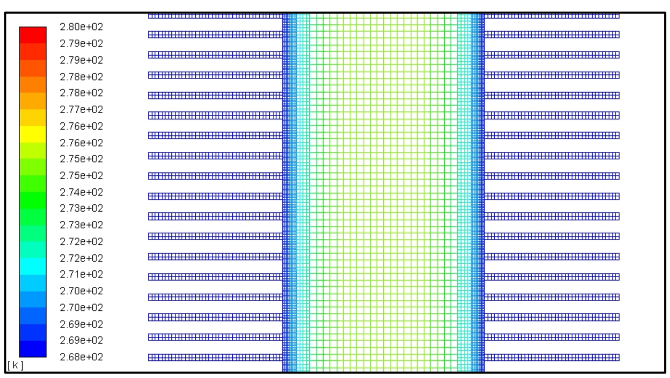

Figure 5.16 View of blade geometry in thermosyphon

As can be seen in Figures 5.17 and 5.18, in thermosyphon analysis equipped with fins, the work fluid behavior is similar to the previous pattern. The liquid CO_2 evaporates by absorbing heat from the walls of the evaporator (heat absorption capacity is increased by adding the fin network) and is transferred to the upper part of the tube, where it loses its latent heat to the condenser (equipped with a fins with higher capacity absorption and heat transfer) returns to the bottom of the pipe.

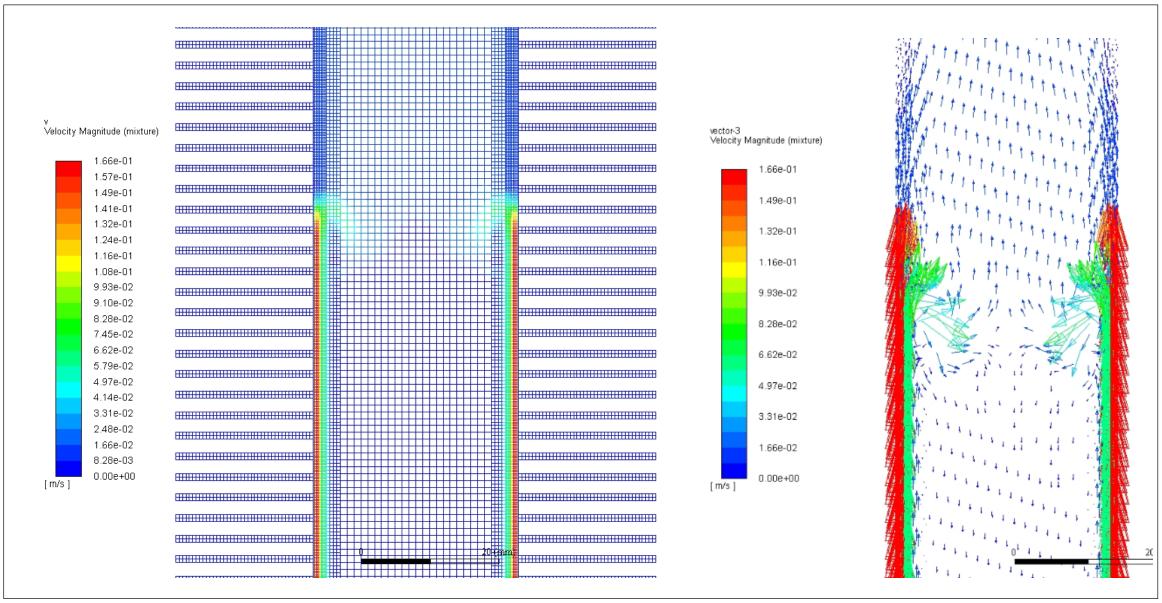

Figure 5.17 Fluid velocity at the time of evaporation

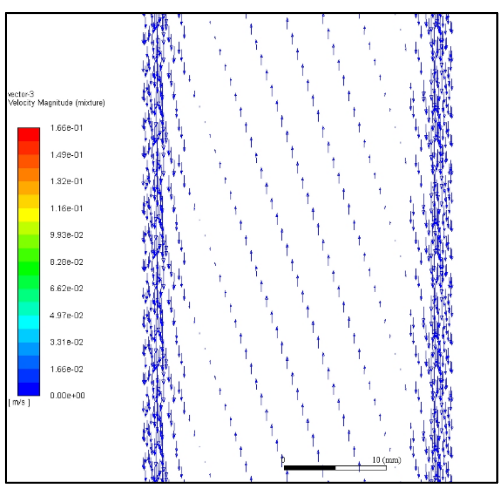

Figure 5.18 Work fluid behavior
through the pipe

The rate of heat transfer in the thermosiphon equipped with fins (to compare the heat transfer capacity) with the dimensions mentioned in Table 5.2 is equal to 59,987 Watts per square meter.

This rate of heat transfer in comparison with thermosyphon with similar dimensions and without fins is increased by 2.5 times, which can be a considerable result.

```
Mixture
        Mass-Weighted Average
      Total Surface Heat Flux           (w/m2)
-------------------------------- --------------------
                      Gas          18.449418
                    Liquid          59.987204
            ---------------- --------------------
                      Net          59.781973
```

Figure 5.19 Total heat transferring rate for Thermosiphon equipped with fins

CONCLUSION

In the analysis of two-phase closed thermosyphons with the aim of transferring heat from the ground heat source to the surface, what is very important is to consider all the parameters that can played a key role in heat transferring rate. What was attempted in the first part of this article was to simulate a two-phase closed thermosyphon in the MATLAB programming space. Considering some of the parameters in the defined equations and its relations, we finally get the amount of heat transfer rate in the pipe. Then, by mathematically optimizing these values to obtain the highest heat transfer rate according to the different dimensions of the thermosyphon and possible limitations, the optimized dimensions obtained for maximum rate of heat transferring which were briefly displayed in Table 4.3.

Whether this amount of heat is enough to melt the snow on the street surface is beyond the title of this article. However, what is important in the analysis of the obtained values is that by analyzing the numerical values of the heat transfer rate for the optimal values of the thermosyphon dimensions, it is observed that:

First, this method of heat transferring with carbon dioxide as the work fluid should be limited to applications where the calculated heat-transferring rate should be bigger than the heat required. What is certain is that the recent results are based on a constant consideration of soil and air temperatures (average temperature in the cold seasons of the year), which will certainly vary in reality.

Secondly, by analyzing Table 4.3, what is observed is that with increasing the length of the thermosyphon, the trend of increasing the rate of heat transfer goes to a constant value and its slope goes to zero. Therefore, increasing the length of the thermosyphon in order to increase the rate of heat transfer cannot be a good solution (especially for high amounts of heat transferring required). However, according to the result, by equipping the thermosyphon to different kind of the fin networks (considering the geometric limitation), we can multiply the rate of heat transferring.

According to the results of thermosyphon CFD analysis and comparison with the amount obtained from the simulation by MATLAB and the difference of 1.92 Watts between the results for heat transferring rate, we can be optimistic about the accuracy of the results of two-phase thermosyphon simulation. As mentioned before, according to the more specialized analysis of Ansys software, the smaller amount of heat transfer resulted by this software, considering its realistic analyzing fluid conditions during the process than the simulation in MATLAB, is justifiable.

RECOMMENDATIONS

In thermosyphon analysis to melt the ice on the roads, in addition to the function of the thermosyphon and the capacity of the heat transfer rate from the ground to its surface, what is very important is the behavior of the snow-containing surface during heat transfer. This is important because it determines the heat required to melt snow (ice) depending on its thickness and the weather condition. In this analysis, what was not evaluated is the behavior of the heat absorbed after transfer to the condenser until absorption by the snow-covered surface.

For a more comprehensive analysis of the performance of the thermosyphon designed in this study (assuming the desired dimensions are known) can be evaluated simultaneously in a complete heat transfer cycle from the time the heat is absorbed by the ground to the surface to melt the ice on it.

It is also possible with the financial analysis of the dimensions and performance of the thermosyphon, considering the heat transfer capacity of thermosyphons with different dimensions, the most economically possible state to balance the imposed costs due to the need to change the dimensions of the thermos to receive the transfer rate and maximum heat transferring capacity.

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