
A thesis submitted in fulfilment of the requirements for the degree of Doctor of Philosophy

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Declaration

I certify that except where due acknowledgement has been made, the work is that of the author alone; the work has not been submitted previously, in whole or in part, to qualify for any other academic award; the content of the thesis is the result of work which has been carried out since the official commencement date of the approved research program; any editorial work, paid or unpaid, carried out by a third party is acknowledged; and, ethics procedures and guidelines have been followed.

Muhammad Fairuz Bin Remeli

01/07/2015
Acknowledgement

All praises to Allah S.W.T, the Most Gracious and Merciful, for giving me strength and ability to accomplish this thesis. All perfect praise belongs to Allah S.W.T, Lord of the Universe. May this blessing belong upon the Prophet Muhammad S.A.W, his family and companions.

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Muhammad Fairuz Remeli
Abstract

In most of the process industry, more than two thirds of the energy used is lost to the environment as waste heat. The current energy conversion systems are not sufficiently efficient to avoid this thermal energy leakage. With the broad concern about recent rises in the price of oil, society has turned attention to technologies which can reduce fossil fuel consumption and minimise the greenhouse effect. The reuse of industrial waste heat can save money and represents an efficient energy practice for industry. However, most industrial waste heat is of low-grade and at low temperature. This type of waste heat is difficult to recover or to use with a conventional heat engine, for example a steam turbine in a power plant. It is highly desirable for researchers to design a heat transfer device which works passively in order that minimal temperature drop can be achieved when recovering low-temperature waste heat.

There are many types of heat recovery system readily available in the market including the convective-type recuperator, heat wheel, and economizer. Most of these systems need auxiliary forces such as using a pump or compressor to move the heat transfer fluid in their operation. Consequently these systems require additional electrical energy input and frequent maintenance. However, the heat pipe offers a simple solution to these problems because it is a passive heat transfer device. The heat pipe has a compact design, is light in weight and has no moving parts. A heat pipe allows its working fluid to evaporate at low temperature under vacuum pressure. This use of heat pipes can minimise the temperature drop during heat transfer processes and can increase heat recovery effectiveness.

A thermoelectric generator (TEG) is a solid state energy converter which allows the direct conversion of waste heat into electrical power using the Seebeck effect. As with the heat pipe, the TEG operates without moving parts, vibration or noise and is very reliable. It is capable of generating power even with low-temperature difference between the heat source and sink which is highly desirable for the low-grade waste heat case. Although the TEG is known for having low thermal-to-electrical conversion efficiency, many recent studies have been conducted to design more advanced and high-performance TEGs by improving their material properties. The TEG is ideal for waste heat recovery because its operating cost is negligible compared to the TEG module cost as the energy input into the TEG is free. An important part of TEG power generation is optimisation of the heat transfer system to reduce the overall cost-per-watt of the energy production.
The aim of this research work was to develop a passive heat transfer and heat to work conversion system for simultaneous heat recovery and power generation. This system is designated as a heat pipe thermoelectric generator (HP-TEG). The basic concept of the system consists of thermoelectric generators (TEGs) sandwiched between two heat pipes, one connected to the hot side of the TEG, and the other connected to the cold side of the TEG.

The thesis presents a detailed design process, a theoretical model, experimental analysis, and cost-benefit analysis for an actual system. The proposed system has the potential to simultaneously recover waste heat and produce electrical power in an entirely passive system without any auxiliary forces.
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Symbols

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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>U</td>
<td>Voltage (V)</td>
</tr>
<tr>
<td>I</td>
<td>Current (A)</td>
</tr>
<tr>
<td>T</td>
<td>Temperature (°C)</td>
</tr>
<tr>
<td>ΔT</td>
<td>Temperature difference (°C)</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate (kg/s)</td>
</tr>
<tr>
<td>c_p</td>
<td>Specific heat capacity (J/kgK)</td>
</tr>
<tr>
<td>C</td>
<td>Heat capacity rate (W/K)</td>
</tr>
<tr>
<td>R</td>
<td>Thermal resistance (°C/W)</td>
</tr>
<tr>
<td>a</td>
<td>Module thickness (m)</td>
</tr>
<tr>
<td>b</td>
<td>Module width (m)</td>
</tr>
<tr>
<td>(\dot{Q})</td>
<td>Heat transfer rate (W)</td>
</tr>
<tr>
<td>L</td>
<td>Duct length (m)</td>
</tr>
<tr>
<td>P</td>
<td>Electric power (W)</td>
</tr>
<tr>
<td>ZT</td>
<td>TEG figure of merit</td>
</tr>
<tr>
<td>d</td>
<td>Diameter (m)</td>
</tr>
<tr>
<td>k</td>
<td>Conductivity (W/m °C)</td>
</tr>
<tr>
<td>V</td>
<td>Velocity (m/s)</td>
</tr>
<tr>
<td>ΔP</td>
<td>Pressure drop (Pa)</td>
</tr>
<tr>
<td>A</td>
<td>Area (m²)</td>
</tr>
<tr>
<td>I</td>
<td>Current (A)</td>
</tr>
<tr>
<td>t</td>
<td>Thickness (m)</td>
</tr>
<tr>
<td>G</td>
<td>Maximum mass velocity (kg/m²s)</td>
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</tbody>
</table>

Greek Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
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</tr>
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<tbody>
<tr>
<td>(\alpha)</td>
<td>Seebeck coefficient ((\mu)V/K)</td>
</tr>
<tr>
<td>(\varepsilon)</td>
<td>Heat exchanger effectiveness (%)</td>
</tr>
<tr>
<td>(\mu)</td>
<td>Conversion efficiency (%)</td>
</tr>
<tr>
<td>(\rho)</td>
<td>Density (kg/m³)</td>
</tr>
<tr>
<td>(\Omega)</td>
<td>Electric resistance (Ohm)</td>
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Dimensionless Numbers

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<td>Nu</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>(Pr)</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
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Subscripts

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<th>Description</th>
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<tr>
<td>c</td>
<td>Cold</td>
</tr>
<tr>
<td>h</td>
<td>Hot</td>
</tr>
<tr>
<td>i</td>
<td>Inlet</td>
</tr>
<tr>
<td>o</td>
<td>Outlet or Outer</td>
</tr>
<tr>
<td>M</td>
<td>Module</td>
</tr>
<tr>
<td>TIM</td>
<td>Thermal interface material</td>
</tr>
<tr>
<td>TEG</td>
<td>Thermoelectric generator</td>
</tr>
<tr>
<td>v</td>
<td>Vapour</td>
</tr>
<tr>
<td>w</td>
<td>Wick</td>
</tr>
<tr>
<td>---</td>
<td>------</td>
</tr>
<tr>
<td>l</td>
<td>Liquid</td>
</tr>
<tr>
<td>e</td>
<td>Evaporator</td>
</tr>
<tr>
<td>c</td>
<td>Condenser</td>
</tr>
<tr>
<td>hyd</td>
<td>Hydraulic</td>
</tr>
<tr>
<td>eff</td>
<td>Effective</td>
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<td>o</td>
<td>Overall</td>
</tr>
<tr>
<td>fr</td>
<td>Frontal</td>
</tr>
<tr>
<td>s</td>
<td>Surface</td>
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My Research Publications

Journal Papers


Conference Papers

- **M. F Remeli**, A. Date, B. Singh, LP Tan, A. Akbarzadeh, Combined Thermosyphon Thermoelectric Model (TTM) for Waste Heat Recovery And Power Generation, 11th International Heat Pipe Symposium (11th IHSP), Beijing, China, June 9-12, 2013.

1 Introduction

1.1 Motivation

In 2013, the total world energy consumption was approximately 12730 mtoe (millions of tonnes oil equivalent). With the fast industrial growth of developing nations over the last decade, the industrial sector has consumed energy at approximately 2852 mtoe [1]. About 33% of the total energy consumed in industry is rejected as waste heat [2]. Together with the concern for global warming and oil depletion issues, there is a strong incentive to develop more efficient and clean technology for both heat recovery and energy conversion systems using waste heat.

Most of the energy rejected by industry is identified as low-grade waste heat. This type of waste heat has a small work producing potential for temperatures below 230°C, and this implies a low energy density. These characteristics make it almost impossible to recover this waste heat for power generation using a conventional energy conversion system such as steam or gas turbine. There are other suitable low-grade heat to electricity conversion systems that can be chosen for using low-grade waste heat such as the Organic-Rankine Cycle (ORC) and the Kalina cycle [3]. However, these systems are more complex as they require rotary parts which are subject to wear and tear and require a high investment cost, therefore these options are not considered economically viable. Because of the constraints associated with low-grade waste heat, it is desirable to have a passive method to convert this thermal energy into electrical energy.

A heat pipe is one of the best devices for heat recovery applications as it has an extensive range of operating temperature and a high thermal conductance [4]. It is widely used as an efficient air-to-air heat recovery component in commercial and industrial applications [5]. There are many benefits of using heat pipes including no moving parts, compact structure, high effectiveness, no cross contamination, small pressure drop, light weight, economy and reliability [6].

Direct thermal-to-electrical energy conversion of waste heat can also be realized by using a TEG. The TEG can generate electrical power if it is placed between a heat source and heat sink. The basic phenomenon is called the Seebeck effect [7]. The TEG is known as a
solid state energy converter which is suitable to use in a low-grade temperature range. A TEG is small in size and requires no maintenance [8]. Although the current conversion efficiency of the TEG is low (less than 5%), several researchers have claimed greater performance, including the NASA Jet Propulsion Laboratory. They revealed obtaining more than 20% TEG conversion efficiency at a high operating temperature [2] and the results are significant enough to indicate that with the development of new TEG materials, direct power production by utilizing free energy from waste heat is economically viable and practical in the near future.

1.2 Research Aim

The expectation that the recovery of low-grade waste heat from industry can reduce greenhouse emissions and can improve the associated overall energy conversion efficiency has encouraged the author to investigate the potential of utilizing this cost-free energy. There is abundant low temperature (<150°C) heat exhausted to the surroundings from different industrial processes. In the majority of cases, recovery of this low-temperature waste heat is unsustainable and uneconomical. This is because of the requirement for active heat transfer and heat to work conversion devices.

The aim of this project was to develop a passive heat transfer and heat to work conversion system for simultaneous heat recovery and power generation.

The main objectives were:

1) Evaluate the thermal performance of heat pipes for medium heat flux (5kW/m² to 25 kW/m²) and for very low driving temperature differences.
2) Evaluate the thermal and electrical performance of air to air finned heat-pipe heat exchangers coupled with thermoelectric generators.
3) Evaluate the economic feasibility of the heat pipe thermoelectric heat recovery system.

The following research questions were needed to be addressed before achieving the project aims:

1) What would be the performance of a water-copper heat pipe under medium heat flux with very low driving temperature difference?
2) What are the critical parameters that influence the combined performance of a heat pipe – thermoelectric generator?

3) What would be the capital cost of a HP-TEG system $/kWt or $/kWe and the corresponding payback period? (kWt – thermal / kWe – electric)

1.3 Research Scope

The scope of this doctoral study included a detail design process, mathematical modeling, experimental analysis and cost benefit analysis of heat recovery and power generation from industrial waste heat using the system described. The theoretical model was validated against experimental results. The validated model was used as a simulation tool to determine the thermal and electrical performance of different designs addressing the requirements of an actual industrial case study.
2 Literature Review

2.1 Introduction

This chapter presents previous work relating to the objectives of this study. The literature review starts with the definition of waste heat and its availability from the perspective of Australian industry. The existing techniques for recovering waste heat from industry are discussed in this section, and special attention has been given to the passive heat transfer method using heat pipes.

The next section of the literature review discusses various heat pipe technologies and their recent development. Previous methods for design and analysis of heat pipes are also discussed as they are relevant to the theoretical part of this study.

The latter section of this chapter describes the working principles of the thermoelectric power generation method known as the “Seebeck effect”. The section covers the standard material for producing Thermoelectric generators (TEG), the dimensionless parameter known as the figure of merit for evaluating TEG efficiency and the current problems associated with TEG technology.

The last section of this literature survey discusses the previous theoretical and experimental investigations using TEG. These include the technology for recovering heat from various heat sources including renewable energy and industrial waste heat. The discussion also includes active and passive cooling methods for a TEG.

2.2 Waste heat

The rising price of oil has motivated the industry to utilize its waste heat. 20-50% of the fuel energy input to the industrial sector is released to the environment as waste heat. Approximately 33% of total consumed energy has been rejected to the surroundings because of inability to recycle the excess energy [2]. The estimated amount of waste heat rejected by United States manufacturing industries was around 3000 TWh/year (which is equivalent to more than 1.72 billion barrels of oil [2]). In the U.K, approximately 11.4 TWh/year of industrial heat was released to environment which equated to 5% of their total energy consumption [9]. Waste heat is commonly produced by machinery, electrical equipment, and
industrial-generating processes. Waste heat losses are transferred to the surroundings through various processes including conduction, convection and radiation. The heat is released in many forms including hot exhaust gas, cooling water, heated product and hot equipment surfaces [10]. From the Australian industrial prospective, the emission of waste heat increased from 13925 GWh in 2009 to 14411 GWh in 2010 [11]. Table 2.1 shows examples of waste heat sources from Australian industry.

Table 2.1 Examples of waste heat sources from Australian Industry.

<table>
<thead>
<tr>
<th>Company Name</th>
<th>Type of Industry</th>
<th>Source of waste heat</th>
<th>Type of waste heat</th>
<th>Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jac Australia Pty Ltd</td>
<td>Labelling-paper manufacturer [12]</td>
<td>Dryer flue gas</td>
<td>Low temperature</td>
<td>80°C to 200°C</td>
</tr>
<tr>
<td>PBR Australia</td>
<td>Aluminium Foundry [13]</td>
<td>Furnace hot flue gas</td>
<td>High temperature</td>
<td>~700°C</td>
</tr>
<tr>
<td>Arnotts Biscuits Holdings Pty Ltd</td>
<td>Biscuits manufacturer [14]</td>
<td>Boilers steam blow down to remove scale and treatment chemical</td>
<td>Low temperature</td>
<td>~ 170°C</td>
</tr>
<tr>
<td>Buttercup Bakeries</td>
<td>Commercial bread baking[15]</td>
<td>Baking oven's flue gases</td>
<td>Medium temperature</td>
<td>~ 300°C - 350 °C</td>
</tr>
</tbody>
</table>

2.2.1 Benefit of waste heat

There are several benefits of reusing industrial waste heat, for example:

- It is cheap or free energy
- It can reduce energy consumption in equipment (such as in boilers, furnaces and dryers).
- It can improve energy efficiency up to 50%.
- It can be recovered by exchanging energy with other material or fluids using readily available technology (such as recuperators or heat exchangers).
- It can be converted into other forms of energy such as thermal to electrical.

Waste heat can be classified into high, medium, and low temperature ranges which are shown in Table 2.2 [10].

<table>
<thead>
<tr>
<th>Types of Waste Heat</th>
<th>Temperature (°C)</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>High temperature</td>
<td>above 600</td>
<td>Nickel and aluminium refining furnaces, glass melting</td>
</tr>
<tr>
<td>Medium temperature</td>
<td>230-600</td>
<td>Steam boiler exhaust, drying &amp; baking ovens</td>
</tr>
<tr>
<td>Low temperature</td>
<td>Below 230</td>
<td>Process steam condensate, cooling water from bearing</td>
</tr>
</tbody>
</table>

### 2.3 Passive heat recovery system

#### 2.3.1 Heat pipe and thermosyphon

The heat pipe is a device which can transfer heat efficiently over large distances at close to uniform temperature without adding external electrical energy. Despite its simple design; a heat pipe has high thermal conductance and operates without moving parts. A basic concept of a heat pipe consists of a thin tube which has a wick structure lined on the inner surface. A small amount of saturated water is inserted into the tube and acts as a working fluid [17]. The heat pipe can be divided into 3 sections which are an evaporator at one end, a condenser at the other end and an adiabatic section in the middle (Figure 2.1).

The evaporator absorbs heat and vaporizes the working fluid at high temperature. The vapour rises and flows to the cooler condenser effectively transferring the latent heat of vaporization. This process is assisted by buoyancy forces [18]. The vapour then condenses and changes to the liquid phase. Because of the effect of gravity, liquid fluid will flow down through the wick to the evaporator whilst the vapour flows continuously in the opposite direction through the core of the tube.
Both heat pipes and thermosyphons (Figure 2.2) have the same working principle. But, for a thermosyphon, condensate returns to the hot end by gravity whereas for a heat pipe by capillary action of a wick [19]. Therefore for a heat pipe, the location of the evaporator is not restricted and can be placed anywhere, but for a thermosyphon, it must be located below the condenser.

The operating pressure and the type of fluid inside the heat pipe depend mainly on the operating temperature. Normally, water is used as working fluid for moderate temperature ranges, while other fluids are used for high-temperature applications. The property of surface tension should be considered when selecting the working fluid in order to increase the capillary effect and to be compatible with wick substances. Other selection criterions are chemical stability, readily availability, non-toxicity and cheapness [17].

Figure 2.1 Diagram of heat pipe [20].
2.4 Reviews on heat recovery system using heat pipes

A heat pipe was chosen in this research because of its ability to transfer heat using a passive method. Details of heat recovery conducted by other researchers using heat pipes are discussed in this section.

2.4.1 Heat pipe heat recovery from geothermal sources

Heat from geothermal sources can be extracted using a heat pipe turbine and converted into electricity. This type of heat pipe has also been referred to as a Thermosyphon Rankine Engine. T Nguyen et al. [19] have introduced the basic concept of this system (Figure 2.3). It consisted of a closed vertical cylinder which had an evaporator, an insulated section and a condenser. A turbine was placed at the top end of the cylinder for producing electrical power. A plate divided the high pressure area in the evaporator and the low pressure area in the condenser. A nozzle converted the thermal energy into kinetic energy. The mechanical energy developed was converted into electricity through an electrical generator. Nguyen tested 4 different designs of heat pipe turbine. The final heat pipe design was 0.5 m in diameter and 2.8 m in height and it successfully produced 100 W of electricity using 10 kW heat input and running at 6000 rpm turbine speed.
To improve the heat pipe turbine performance, Akbarzadeh et al. [21] introduced their latest turbine design consisting of two “S” shaped pipes attached to a hub (Figure 2.4). Vapour flowed vertically though the hub and changed its direction horizontally when entering the two pipes. The vapour created the turbine reaction torque when it exited at high speed through nozzles at the ends of the pipes. This turbine was expected to produce 3 kW output power from 100 kW heat input. The prototype was designed to work at 55°C evaporator temperature and 25°C condenser temperature based on a geothermal bore specification in Portland, Victoria.
In a commercial scale case, a large scale heat pipe extracted heat from a geothermal well in Kyushu, Japan as reported by Kusaba et al. [22]. The heat pipe was of 150 mm diameter and 150 m length. The heat pipe system was equipped with a liquid feeding tube with a showering nozzle. It was tested in a geothermal well of 70-150 m depth. The system extracted approximately 90kW of heat at 80°C working fluid temperature. A computer simulation was developed to predict the output power available from the heat extraction by considering a turbine to be installed on the top of the heat pipe. The prediction showed that the 0.8 m diameter turbine could generate approximately 7.8 kW of electric power at 3000 rpm.

2.4.2 Heat pipe heat recovery from the bakery industry

The flue gases from bakery ovens have produced a lot of waste heat that can be recovered for the other uses. Lukitobudi et al. [23] constructed a lab scale rig of an air-to-air thermosyphon heat pipe for a bakery system which is illustrated in Figure 2.5. The thermosyphon system is suitable for an application at temperatures below 300°C. Three different configurations of heat pipes were tested including a bare copper tube, a continuous copper plate fin and a circular steel spiral fin. The heat exchanger was operated by recycling the heated air to the counter flow heat exchanger.

The results showed that the effectiveness of the system increased by adding more heat input and increasing the air face velocity. The finned copper thermosyphon produced the best
performance and the highest effectiveness compared with the other configurations. However, it had a lower allowable stress which limited its adiabatic section temperature to less than 200°C for safety reasons. This limitation can be relaxed by using a steel thermosyphon in the real industrial application. The prototype of the system was installed in the Buttercup Bakery in Clayton, Victoria.

![Figure 2.5](image)

Figure 2.5 The thermosyphon heat exchanger test rig [23].

2.4.3 Heat pipe heat recovery from heating ventilation and air-conditioning (HVAC) system.

A study of heat recovery for natural ventilation using a heat pipe heat exchanger (HPHE) was reported by Riffat and Gan [6]. They built a test chamber made from insulated plywood (Figure 2.6). The chamber was divided into two zones using horizontal partitions. The partition had an aperture in the middle area to circulate air from one zone to another. The supply and exhaust ducts were connected to the chamber on one of its vertical walls. Air entered the upper zone from the lower zone through a supply duct. The return air was released through the exhaust duct. The HPHE connected the ducts to exchange heat between
the return and the supply air. Three different heat pipe recovery units were tested in a two-zone chamber separated by a horizontal partition (Figure 2.7). The effectiveness of heat pipes was investigated by varying several parameters including air speed, the shape of fins and pipe arrangement. Riffat found that the effectiveness of this system decreased with increasing air velocity. The two banks of plain finned heat pipes could recover 17% more heat than the one bank system. The plain fin had better effectiveness than the spine fin because of the low thermal contact of the latter. Moreover, the performance of the in-line heat pipe was better than the staggered heat pipe for the natural ventilation case.

Figure 2.6 Schematic of two zone test chamber [6].
In an air conditioning system, heat pipes have been used to reheat energy to control the relative humidity of the air as reported by Xiao Ping et al. [24]. A thermosyphon heat exchanger was installed in a laboratory scale air-conditioning unit to measure the efficiency of such a system. Figure 2.8 shows the concept of an air conditioner equipped with a heat pipe heat exchanger (HPHE). Heated and humidified air was passed through the evaporator section of the HPHE. This process was to pre-cool the air. The air was cooled further by a cooling coil before entering a condenser section of the HPHE. The result showed that the cooling capacity of the air conditioner was improved between 20-33% by installing the HPHE. The condenser of the HPHE could potentially replace the conventional air re-heater for controlling the relative humidity of air of less than 70%.

Figure 2.7 Cross-section of fins used in the tests [6].
Figure 2.8 Concept of air-conditioner with heat pipe heat exchanger [24].

In another case, Mostafa et al. [25] investigated the effectiveness of a heat pipe heat exchanger (HPHE) in an air conditioning system. A fresh air stream was thermally connected to a return air stream using a HPHE (Figure 2.9). A laboratory refrigerator supplied the return cold air on the condenser side of the HPHE. An air blower was installed in the fresh air duct to supply air on the evaporator side. The mass flow rate of fresh air was maintained at 0.4 kg/s and the return air was varied between 0.4 – 0.93 kg/s. The temperature of the inlet fresh air was controlled between 32- 40°C and the inlet return air was maintained at 26°C. When the temperature of the inlet fresh air was raised to 40°C, the effectiveness and heat recovery ratio increased to 48% and 85%, respectively. It was found that the volume flow rate, air velocity and maximum temperature difference were the main parameters influencing the effectiveness of the HPHE.
Figure 2.9 Air ducts and measurement devices [25].

Noie-Baghban and Majideian [26] presented a study of waste heat recovery using a conventional heat pipe for a surgery room in a hospital. This study consisted of designing, constructing and testing an air-to-air heat pipe heat exchanger (HPHE) for a low temperature range application (15-55°C). A computer program was developed to design and manufacture in-house heat pipes. This computer program could determine the important heat pipe characteristics such as the number of wicks, volume and pressure, the figure of merit, priming factor and heat transfer limits of working fluids.

A test rig was built for testing and validating the developed computer model (Figure 2.10). The test rig had 3 rows of heat pipe banks arranged in a staggered equilateral triangle form. The effectiveness of the HPHE obtained was small at 16% because of insufficient fins, high pitch to diameter ratio and high air face velocity used by the authors. To increase the effectiveness of the HPHE, they suggested optimizing the number of fins and rows, and using better insulation and pipe sealing.
2.4.4 Heat pipe heat recovery from the combustion process.

In this study, Habeebullah et al. [27] recovered waste heat from combustion chamber exhaust gas using a heat pipe. The recovered energy from a combustion chimney became the heat input for a modified aqua-ammonia absorption chiller. A conceptual gas turbine was cooled by chilled water from this chiller. The evaporation side of the heat pipe was fitted concentrically to the combustion chimney (Figure 2.11). Flue gas flowed through the inner core. An evaporator jacket was annularly fitted to the outer surface of the chimney. The upper and the lower parts of the jacket were connected to a large vapour exit pipe and a small condensate return, respectively. The condensation section of the heat pipe was attached to the chiller generator. The chilled water that flowed through the inner core of the generator was used to cool the gas turbine engine. From this study, Habeebullah found that the system still supplied cold water although the gas temperature was low at 105°C and the burner was shut down for more than 280 hours. The system successfully recovered between 70-80% of the waste heat.
2.4.5 Heat pipe heat recovery from drying process

Lin et al. [28] presented a Computational Fluid Dynamic (CFD) study of a heat pipe heat exchanger (HPHE) in the drying cycle of domestic appliances. In their heat recovery concept, the heat pipe evaporator and condenser were placed before and after a condensing section in a dehumidification conduit (Figure 2.12). The CFD study proved that the heat recovery system increased the condensate removal by up to 30%. Moreover, the CFD analysis provided an easy method and procedure for predicting the thermal performance of a dehumidification process using a HPHE.
Figure 2.12 Drying cycle of domestic appliances incorporating heat pipe heat recovery [28].

S. Rittidec et al. [29] suggested that more energy could be saved in a drying process by preheating the fresh air using a closed-end oscillating heat pipe (CEOHP). Figure 2.13 illustrates the CEOHP which contains 32 sets of copper capillary tubes with an inner diameter of 2 mm. The hot gas heated the heat pipe evaporator tubes at a temperature range of 60°C - 80°C and velocity of 3.3 m/s. The recovered heat from the hot gas was rejected from the heat pipe condenser to pre-heat fresh air at 30°C. The study concluded that the effectiveness of the CEOHP increased by maximizing the hot gas temperature and by changing the working fluid from water to R123.

Figure 2.13 The CEOHP air-preheater [29].
2.4.6 Heat pipe heat recovery from an automotive engine

Because of thermodynamic limitations, an automotive engine rejects a lot of thermal energy through its exhaust pipe. Yang et al. [30] developed a system for heating a bus cabin by recovering exhaust gas heat using a heat pipe heat exchanger (HPHE). The exhaust gas was passed through the HPHE evaporator and transferred to the HPHE condenser. The rejected heat from the HPHE condenser warmed up the circulated air from the bus cabin (Figure 2.14). The heated air was directed back to the cabin for space heating. The bus had a six cylinder engine of 5.42 L cylinder capacity. The maximum rotational speed of the bus engine was 3000 rpm. The authors found that the theoretical results agreed well with the experiment results and they concluded that the HPHE could be an effective heat transfer device for bus cabin heating.

![Figure 2.14 Schematic of bus heating system [30].](image)

2.4.7 Heat pipe heat recovery from solar energy

The HPHE can also recover the available heat from solar energy. For example, Tundee et al. [31] extracted heat from the lower convective zone (LCZ) of a solar pond using the HPHE. The thermal energy stored in the solar pond was transmitted using the HPHE to heat the cold air from ambient temperature (Figure 2.15). A variable air blower drove the air
through a duct and the air flowed through the HPHE condenser for heat collection. This system extracted approximately 100 W of continuous heat from the solar pond LCZ. The HPHE effectiveness peaked at 43% for a duct air velocity of 1m/s.

![Diagram of solar pond heat pipe heat exchanger]

Figure 2.15 Experimental set-up for solar pond heat pipe heat exchanger [31].

### 2.4.8 Heat pipe recovery in nuclear application

Emergency cooling is one of the most important aspects in nuclear power plant operation. A conceptual study conducted by Mochizuki et al. [32] has shown that heat pipes could possibly cool down the spent fuel of the Fukushima nuclear reactors. Some of the Fukushima nuclear reactors were badly damaged and radioactive material leaked during the Tsunami tragedy in 2011. In this study, the nuclear reactor number 4 was chosen for a detailed thermal analysis to improve its cooling system. The reactor produced 2.4 MW thermal power and 784 MW electrical power. The reactor consisted of 658 spent fuel bundles and it took 1.5 years for the reactor to shut down. After the shutdown, the spent fuels were left about a month in the reactor before transferring to a water cooling pool. Figure 2.16 shows a schematic of the conceptual design of a spent fuel cooling system using a thermal diode heat pipe. The heat pipe evaporators were submerged in a 10 m high cooling pool. The 4.5 m spent fuel bundles were totally submerged in the pool contained 1400 tons of water. The initial water temperature of the pool was 30°C. The 4 m heat pipe condensers were exposed to the surrounding and were cooled by natural convection. Four different cases studies were conducted, however a design that used 1,622 pieces heat pipe was considered to be the most applicable and cost effective. The installation of heat pipes could provide 0.9
MW cooling capacity. It could decrease the water temperature to 68°C after 70 hours and could stabilize the water temperature at 50°C after 2000 hours. The installation cost for such system was estimated at USD 2.16 million.

![Figure 2.16 Concept of a cooling system for spent fuel using a thermal diode heat pipe][32].

HPHE technology also has potential to be applied for nuclear seawater desalination. Jouhara et al. [33] discussed a proposed new concept for a nuclear desalination system integrated with a HPHE. The heat from the steam flow of a nuclear power plant evaporates the water in a seawater feed chamber (Figure 2.17). The steam condenses in a steam chamber and returns to the nuclear power plant. The steam chamber and the seawater feed chamber are separated by the adiabatic section of the HPHE. This section is well insulated and it will not allow any mass transfer. The authors concluded that the proposed system was more economic and efficient than a conventional system such as a shell-and-tube heat exchanger. The system was found to be more secure as it has an additional loop to prevent from direct contact radiation of water and it was free from contamination. The proposed system could also improve the overall thermodynamic efficiency of the desalination process.
2.4.9 Method for analysing and designing heat pipe heat exchanger

Both theoretical and experimental analyses have been used to evaluate the performance of the HPHE. For example, back to 1978, Lee and Bedrossian [34] have investigated the characteristics of heat pipes or thermosyphon heat exchangers by using a simple theoretical model and conducting experimental work. The experimental arrangement consisted of two rectangular air ducts which thermally contacted a bank of heat transfer elements (Figure 2.18). The segmented baffles mixed the air streams in order to achieve a uniform temperature distribution. Both the thermosyphon and heat pipe were tested in this study using a counter flow and a parallel flow arrangement. The effects of inline and staggered arrangements of tube bank geometry were also compared. The heat transfer coefficient involved in each process was estimated in order to develop a heat conductance model (Figure 2.19). From this study, it was found that an increase in the Reynolds and Prandlt numbers increased the overall heat transfer coefficient up to a certain limit. The tube bank geometry with a staggered arrangement produced similar performance to the in-line arrangement. The effect of element orientation to the gravitational field for a thermosyphon is more pronounced than for a heat pipe. This computer model could also predict the optimum number of fins to be used for best performance of the HPHE.
Figure 2.18 Schematic diagram of experiment set-up and the element bundle arrangement [34].

Figure 2.19 Heat conductance model [34].
Azad and Geoola [35] discussed a theoretical design procedure for a gravity-assisted (HPHE). This procedure was used to design a heat recovery system for a solar agriculture dryer that released moist air at 70°C. In order to predict the HPHE performance, a method called effectiveness-number heat transfer units was developed. The overall effectivenesses of the HPHE were compared and plotted against Ce/Cc (the heat capacity ratio between evaporator and condenser) for different cases of the tube spacing, number of fins, thickness of fins, and lengths of evaporator and condenser. The overall effectiveness of the HPHE increased with increasing numbers of fins, fins thickness, and lengths of evaporator and condenser. The overall effectiveness decreased with increasing tube spacing normal to the flow direction.

In 1985, Azad et al. continued conducting an analytical study using a water-to-air heat pipe heat exchanger [36]. The heat from a solar collector was transferred using a gravity assisted HPHE to heat fresh air before entering an agricultural dryer (Figure 2.20). The evaporator section of the HPHE was finned and enclosed in a rectangular box. The condenser section was exposed to hot water from the solar collector. The water in the condenser section was directed to the heat pipe array using a baffle to obtain a uniform temperature distribution. The fluid flows in this study were arranged in a counter flow configuration. Another air-to-air HPHE was connected to this system to preheat the feed air supplied to the dryer. The heat source was the moist exhaust air from a dryer is shown in Figure 2.21. Approximately 40% of waste heat recovery and 50% HPHE effectiveness were achieved when the outlet air temperature increased from 51°C to 58.5°C. In addition, the temperature rise caused a significant reduction in drying time.
Huang and Tsuei [37] developed an analysis method for a HPHE using a conductance model. The specific heat conductance value of a single heat pipe was obtained from a performance test. The result from this test was used to develop a computer model using a finite difference method. This model could predict the performance of a heat pipe heat exchanger (HPHE). An experiment using the HPHE was conducted to validate the analysis method (Figure 2.22). It was found that the theoretical heat transfer rate obtained from the analysis method produced approximately 10% error compared to the experimental results. However this error was considered acceptable in the engineering context.
Azad and Gibbs [38] presented a theoretical analysis of an air-to-water (HPHE) using the effectiveness-NTU method. An axial heat pipe was used because it was cheap and light in weight compared to other heat pipes. The HPHE used consisted of many tubes connected in series with the condenser section (Figure 2.23). The heat from hot exhaust gas was absorbed by the finned evaporator section of the HPHE. The rejected heat was used to increase the temperature of the cold water that circulated over the outer surface of the HPHE condenser. The overall effectiveness of HPHE versus Ce/Cc was plotted for 8, 10 and 12 row cases. It was concluded that the HPHE effectiveness rose with a decrease in the Reynolds numbers. The effectiveness could also be increased by adding more fins to the system.
Figure 2.23 Air-to-water heat pipe heat exchanger [38].

Tan and Liu [39] investigated the performance of HPHE using an analysis based on the effectiveness NTU-method. The method developed did not require any iterative procedure to calculate temperatures at several locations. Also this method was found to consume less computational time. The results of the analysis were close to the theoretical results using an iteration method by [37]. The analysis method was concluded to be easier and more convenient to use compared to the analysis using the conductance method which required a lengthy iterative calculation.

Noie et al. [40] developed an iterative computer program to predict the outlet temperature and the effectiveness of an air-to-air thermosyphon HPHE using the effectiveness-NTU method. An experimental rig was also designed to recycle the heated air in the evaporator section of the thermosyphon (Figure 2.24). The thermosyphon was tested under steady state conditions to observe the effect of velocity and temperature of the input air, the pipe material, and the filling ratio of the working fluid. The study found that the HPHE effectiveness ranged between 37% and 65% depending on the operating conditions. The computer simulation results were found to be close to the experimental results and the agreement improved at higher air velocity.
Figure 2.24 Schematic of experimental rig of thermosyphon heat exchanger [40].

Azad [41] developed an analytical model to predict the temperature distribution in a gas-to-gas thermosyphon HPHE using the effectiveness-NTU method. The model assumed that the pipes were installed with continuous type fins and were arranged in a staggered configuration. This model could also predict the number of heat pipe rows, evaporator and condenser temperatures and the saturation temperature. The heat pipe temperature distribution was calculated by considering the row-by-row approach as shown in Figure 2.25. The simulations showed that the effectiveness of the HPHE could be improved by increasing the number of fins per unit length and value of Ce/Cc (the hot and the cold air heat capacity ratio).
2.5 Reviews of thermoelectric power generation (TEG)

A thermoelectric power generator (TEG) is a device that can convert thermal energy into electricity. It is a solid state heat engine and has no moving parts, no vibration and no noise, is light in weight and very reliable [42]. To generate electrical power, the TEG should be attached between a heat source and a heat sink. Because of the temperature gradient created between the heat source and the heat sink, heat will flow through the module and be rejected to the surroundings through the heat sink. If the temperature gradient is maintained, electrical power will be continuously generated [43].

The thermoelectric module operation is based on the Seebeck effect (Figure 2.26). The Seebeck effect was discovered by Thomas J. Seebeck in 1821 [44]. He found that an electromotive force or potential difference (voltage) could be generated by a circuit made from two different wires if one of the junctions was heated. In a thermoelectric module, when two dissimilar conductor are connected and the two junctions are maintained at temperature $T_H$ and $T_C$ where $T_H > T_C$, an open circuit electromotive force or potential difference (voltage) is developed between this junctions. The electric voltage is proportional to the temperature difference and is given by:

$$U = \alpha(T_H - T_C)$$  \hspace{1cm} (2.1)
Where $\alpha$ is the Seebeck coefficient and is measured in $\mu V/K$.

![Diagram of Seebeck effect](image)

Figure 2.26 Seebeck effect [45].

The materials to produce a thermoelectric (TE) cell made from a n-type and p-type couple. A figure of merit, $ZT$ expresses the efficiency of the TE material. Rowe [42] has categorized thermoelectric materials into established materials and new materials. Today, the most common available thermoelectric materials are Bismuth Telluride (Bi$_2$Te$_3$) - based alloys and PbTe-based alloys which they have $ZT$ values close to unity [8]. Bell [46] reported that a higher TEG efficiency can be achieved if the $ZT$ value is maximized and the temperature differential between the hot and the cold sides of the TEG is maintained as large as possible.

The major challenge of thermoelectric power generation is its low heat-to-electricity conversion efficiency which typically near to 5% [47]. Although the TEG cannot be used widely because of this limitation, there several companies have used TEG for many applications including cooling/heating car seats, laser diode coolers, DNA synthesizers and low-voltage electric generators [46].
2.5.1 Theoretical model studies of the TEG power generation.

A number of researchers have developed theoretical models in order to investigate the performance of the thermoelectric generator. For example, Wu [48] developed a theoretical model of a waste-heat thermoelectric power generator which included analysis of both internal and external irreversibility effects. This analysis method was considered more real because it provided actual generator specific power and efficiency estimation rather than ideal values.

Rowe and Min [49-51] have established an optimization procedure to evaluate the performance of thermoelectric modules for power generation. This analysis correlated the module geometry with the requirements for obtaining maximum power output and conversion efficiency. The analysis also included the effect of thermal and electrical contact resistance. This analysis gave a guide for designing a module geometry which included economic factors such as cost-per-watt, watt-per-area and manufacture quality factor. The authors concluded that the TEG can be shown to be a promising device for harvesting low-temperature waste heat based on the optimization study results of several commercial TEGs.

Esarte et al. [52] have analysed the performance of a thermoelectric generator by studying the effect of heat exchanger geometry, fluid volume flow rate, fluid properties and fluid inlet temperatures. The overall heat transfer coefficient of the system was calculated using the thermal resistance method and an effectiveness-NTU (Number of Transfer Units) method was used to determine the unknown fluid outlet temperatures of the heat exchanger. Three different heat exchanger geometries were compared in this study including spiral, zigzag and straight fins. This method gave a design procedure for determining the optimum operating parameters of a thermoelectric generator.

Chen et al. [53] have performed a heat transfer irreversibility analysis to investigate the performance characteristics of a multi-element thermoelectric system. The Joulean heat effect and heat leak through element leg were also considered in this analysis. It was concluded that beside the heat transfer irreversibility, many other factors affected the performance of a thermoelectric generator such as the number and size of thermoelectric elements, the conductance of heat exchangers, and the internal and external resistances or loads. These parameters must be optimized to maximise power output and efficiency of thermoelectric devices.
Hsiao et al. [54] have presented a theoretical model of a thermoelectric generator (TEG) for the use in an automotive application. An exhaust pipe and a radiator in the automotive were identified as possible locations for applying this system. The theoretical model was developed using a one dimensional thermal resistance method. A basic experiment was conducted to obtain the characteristic coefficient of the thermoelectric cell and the results of the test were used to validate the theoretical model. The model prediction showed a good agreement with the experimental result. It was found that the TEG performed better on the exhaust pipe than on the radiator. This model provided a direct relation between TEG performance and automotive engine speed and coolant temperature.

Han et al. [55] have developed an analysis model of a thermoelectric power generator by considering manufacturing factors, pallet size and different thermal conditions. Initially, the model was analysed using constant thermoelectric properties such as Seebeck coefficient, electrical resistance and thermal conductivity. It was found that the maximum power generated occurred when external electrical resistant equalled the internal electrical resistance. However, the model predicted some performance deviations from experimental results because the internal electrical resistance is strongly temperature dependent. A modified model was created by adding the temperature-dependent internal electrical resistance. As the result, the performance of the new model showed better agreement with the experimental results.

### 2.5.2 Laboratory experimental studies of the TEG

Apart from the theoretical analysis of the TEG; many researchers have constructed in-house test rigs to study TEG characteristics and performances. For example, Casano and Piva [56] evaluated the power output and conversion efficiency of the TEG using different load resistances. Nine TEGs were connected electrically in series and sandwiched between a square aluminium plate and a heat sink. A nickel-chrome resistance heater supplied heat to the top of the TEG surface. The finned heat sink was immersed in a water bath for cooling (Figure 2.27). The power output and the efficiency of the TEG were plotted against temperature difference. The lumped model was found to be in good agreement with the experimental data and it became a useful tool for predicting TEG performance.
Tzeng et al. [57] tested a TEG system including replicating an exhaust system. The metal pin fins were used as a heat absorber and heat sink in this system. They developed 1-D steady heat conduction model which included the Joule heat generation and the Seebeck effect. Figure 2.28 illustrates an electric heater being used to supply heat to the heat absorber in the test section. The heat was then conducted through 4 TEGs elements before it was removed from the heat sink. Air flow from an air blower cooled the heat sink. Plain plate, in-line pin fins and staggered pin fins were tested as the TEG heat sink. The staggered pin fin configuration produced the highest power output compared with other modes. The model agreed well with the experimental data which confirmed its validity.
Figure 2.28 Schematic diagram of the experimental system and decomposition chart of the test section [57].

Woo et al. [58, 59] fabricated a TEG system which was clamped between aluminium plates. Hot water at 95°C flowed at the centre of a cold extrusion container of size 86 x 23 x 350 mm. On the cooling side, 20°C cold water was maintained through both ends of the container. The TEG power and voltage output were presented as a function of temperature difference of the hot and the cold water. The efficiency of TEG increased with the temperature difference.

Rowe et al. [50] sandwiched TEGs in an aluminium plate heat exchanger. A 97°C hot water heat exchanger was sandwiched between two cold water heat exchangers (Figure 2.29). The temperature of the cold water was 14°C. The heat exchanger and the TEG performances were analysed using Fluent CFD software. The temperature-profile across the system, channel aspect ratio, inter channel fin structures and water flow rate and direction were optimized by the simulation. The optimum design of the TEG system was referred to as WATT-100. That consisted of 36 TEG modules and generated approximately 97 W of electricity. This TEG system could compete economically with the other power generation methods if waste heat was used as the heat source. The low TEG efficiency would be less significant when it worked under a passive system or parasitic mode.
Ono and Suzuki [60] designed a TEG system which worked in a multistage heat-exchanger. It contained many air flow channels, separated by the TEG walls (Figure 2.30). The forced turbulent air circulated in counter flow through the channels. The air circulation caused change of air density and reduced the air temperature. This air circulation created the temperature gradient across the TEG walls. The temperature gradient caused heat to conduct through the wall from hotter to the cooler air. This system was designed for low-grade heat applications at temperatures below 600K. The TEG system has produced enough power of 70 µV/K for the proposed system.

Figure 2.29 Schematic of WATT-20 generating system [50].
Ikoma et al. [61] constructed a TEG system using the heat from a combustor bench. The exhaust gas from the combustor was divided into the TEG channel and a bypass channel (Figure 2.31). 72 units of the Si-Ge TEG were sandwiched between an inner and an outer shell of the generator. To improve heat transfer, fins were used at the inner shell which contacted directly with gas flow. Water was circulated inside the inner shell to maintain the cold temperature of the generator. Approximately 36 W of electric power with 0.9% conversion efficiency was produced by this study.
Xing Niu et al. [62] generated electrical power by providing the temperature gradient of fluids from temperature controlled baths. Glycol and water mixtures were circulated through a multilayer heat exchanger (Figure 2.32). 56 TEGs were sandwiched between three cold fluid passages and two hot fluid channels. The system generated 146.5 W of electricity and 4.44% efficiency at a temperature difference of 120°C ($T_{\text{hot}} = 150^\circ\text{C}$ and $T_{\text{cold}} = 30^\circ\text{C}$).
2.5.3 Thermoelectric power generation from vehicle waste heat

A number of researchers have converted the vehicle thermal waste into electricity using the TEG technology. For example, Kim et al. [63] have investigated the thermoelectric power generation by using the hot exhaust gas from a hybrid vehicle. The limited space near a hot exhaust pipe was extended by using 10 sections of heat pipes (Figure 2.33). Moreover, the excess heat from the exhaust gas was transferred efficiently to the hot surface of the TEG using these heat pipes. The other surface of the TEG was cooled by water. The system has produced approximately 350 W of electricity at an evaporator surface temperature of 170°C.

Figure 2.33 Schematic of experiment set-up of thermoelectric generator assisted by heat pipes [63].

Goncalves et al. [64] utilized a variable conductance heat pipe (VCHP) to control high temperature exhaust gas from an engine and to suit the TEG operating temperature. The system was predicted to generate approximately 550 W of electric power for an engine input power of 30 kW.
Kim et al. [65] developed a low-temperature TEG system using heat from an engine water coolant. They claimed that the system could potentially substitute the conventional radiator without adding other mechanical devices. The system consisted of a hot side block, two cold side blocks (Figure 2.34). 128 heat pipes were mounted on the cold side blocks. The engine coolant flowed through the hollow section of the hot block. 72 TEGs were sandwiched between the cold and the hot blocks. The designed system was smaller than the original car radiator. The temperature of the hot-side block was between 90°C and 95°C and the cold-side block was at 70°C during the car idle condition. At 80 km/h speed, the hot side temperature rose from 95°C to 100°C and cold side reduced to 45°C. This system generated around 75 W of electric power which was equivalent to around 2.1% TEG conversion efficiency.

![Figure 2.34 Perspective and side view of the thermoelectric generator (TEG) [65].](image)

### 2.5.4 Thermoelectric power generation from drying process

In Thailand, a biomass dryer was used to preserve agricultural product such as fibres and papers. Maneewan and Chindaruksa [66] installed a TEG system onto a biomass dryer wall which is illustrated in Figure 2.35. The temperature of the wall fluctuated between 218-240°C. The TEG was cooled using a water cooling block. Four ventilation fans in the drying chamber were driven by the generated power using the 12 TEGs installed. Approximately 1 W per module of electric power was produced and the system could provide 9.62 m³/s of air
flow at 80°C drying room temperature. The thermal efficiency of TEC system was calculated at 4.05%.

![Diagram of a drying system](image)

**Figure 2.35** Thermoelectric power generation using waste heat from biomass drying [66].

### 2.5.5 Thermoelectric power generation from a cooking stove

In some rural areas, a wood stove is the main tool for cooking. These areas are often unreached by electricity. Nuwayhid et al. [67, 68] designed, fabricated and tested a high performance and low cost TEG that was fitted to a domestic woodstove. The aim of the study was to provide an off-grid power source for this area. A commercial heat sink cooled the TEG by using a natural convection process (Figure 2.36). It was reported that the TEG system has generated about 4 W of the maximum power at a matching load. The output power of the system decreased with increasing number of TEG because the heat sink was unable to dissipate heat effectively to the surroundings. This caused the cold side temperature to rise, decreased the temperature gradient of the TEG and dropped the power output.
In another study, Champier et al. [69] built laboratory scale biomass cook stoves to improve combustion completeness and to reduce air pollution. In addition, a thermoelectric generator (TEG) was incorporated into the stove for providing light and powering fan to increase the air fuel-ratio of the combustion process. They developed a bench consisting of a 150 W thermo-coax electric heater to continuously heat the TEG. A heat sink was used to cool the TEG cold side. The authors also investigated the performance of TEG using fan air cooling and water cooling. The heat flux and the thermal conductance were estimated using the 2D-COMSOL software. The calculated model was validated by the experimental data and the calculated value of the thermal conductance was used to determine the conversion efficiency of the TEG. The system produced 6 W of electricity which was enough to power a battery, fan and LED light for the woodstove system. The system cost was estimated at 20 Euro per watt.

In a rural village in Malawi, Shaughnessy et al. [70] developed a thermoelectric generator (TEG) which was integrated into a portable biomass cook stove. The power generated by the TEG could charge a single 3.3 V lithium-iron phosphate battery and drive a low powered fan for cooling the heat pipe heat sink at the thermoelectric cold side. The battery was able to charge mobile phones and to power other equipment such as lights and radios. The TEG was installed on the side wall of the stoves (Figure 2.37) and the cold side of
the TEG was cooled by a CPU cooler. The cooler had 4 pieces of heat pipes. A 92 mm (12V) fan impeller was mounted to cold the heat sink. Under laboratory conditions, the stove produced 9 Wh of electric power and 8 Wh of battery storage for an hour long cooking period.

![Diagram of the stove and its components](image)

**Figure 2.37 Integration of generator with Chitetezo Mbaula [70].**

### 2.5.6 Thermoelectric power generation from a gas heater

A common ordinary house is usually equipped with gas or electrical heating for water or space heating during winter. The gas burner is a cheap option for house heating, however, not all of the fuel combustion in the burner is converted into thermal power and part of it is lost to the surroundings. This lost energy from the gas combustion could be used for power generation. This work has been carried out by Allen and Wonsowski [71]. They developed and tested a demonstration unit of a self-powered central heater which was used for thermoelectric power generation. Figure 2.38 depicts an upper section of the unit consisting of a heat exchanger which used water as a working fluid. In the middle section, 18 bismuth-telluride TEGs were concentrically mounted on a hexagonal heat receiver. The heat receivers
were cooled by six water cooled plates. A 15 kW ceramic burner was placed in the central hollow section of the thermoelectric stage.

The system produced around 11 kW of thermal power using 12.9 kW of natural gas combustion at 1.2 m$^3$/h. Approximately 43 % of the heat was transferred to the thermoelectric section and 57 % to the heat exchanger section. The gross electrical power output produced was around 109 W.

Qiu and Hayden [72] developed a fuel-fired system integrated with a 550 W thermoelectric generator (TEG) for a residential heating system. The TEG had a radial configuration and was made from the PbSnTe doped. It had 325 P-N thermocouples and was sealed in a hermetically enclosed space and filled with argon gas to remove air. The inner wall of the TEG was heated to 650°C and the cold surface was cooled by the circulating water from a heat storage tank. A heat conducting fin was installed on the inner wall of the TEG to reduce the large temperature difference between the burner surface and the inner surface of the TEG (Figure 2.39).

The system generated maximum output power of 550 W at a temperature difference of 535°C. The electricity generated could power the entire heating system components such as air blower fan, water pump, igniters, various valves and control panel.
Later, Qiu and Hayden [73] have introduced an advanced version of the TEG system from their previous study [72]. For this study, two radially shaped thermoelectric modules were arranged in tandem in the system (Figure 2.40). Each module consisted of 325 thermoelement made from PbSnTe doped. At the hot side of the TEG, the lower module had an inner flat surface and the upper module was equipped with heat conducting fins. A circulating water jacket maintained the temperature at the cold junction of the TEG. A mathematical model was established for system optimization. It was predicted that the electrical power would increase significantly with increasing burner operating temperature. The model also showed that the electrical efficiency improved by adding the heat recuperation. The experimental results showed that the upper and lower TEGs generated 487 W and 566 W of electrical power, respectively. The lower TEG power was higher because it received more heat radiation from the burner. This integrated system could offer a reliable house heating system and could potentially reduce the electrical power consumption.
M. P. Codecasa et al. [74] have integrated a TEG with an autonomous gas heater for outdoor use. The TEG was equipped with a gas tank and could operate for 17 hours. A small battery was installed to power an electronic starter, a controller and a safety device. 3 elements of TEG (brand: Altec-1060) were installed between a heater collector and heat sinks (Figure 2.41). The external surface of the heat collector had 6 equal flat surfaces. A wax phase change material (PCM), FSF52 and fin heat sinks were used to cool the cold side of the TEG. The system produced approximately 5 W of electricity at the TEG temperature gradient of 122°C.

Figure 2.40 Gas fired thermoelectric power generation system [73].
2.5.7 Thermoelectric power generation using waste heat from power plant and furnace.

Because of thermodynamic limitations, power plants rejects most of their energy in the form of liquids or gases such as through condenser, furnaces, chimneys and boilers. Kyono et al. [75] developed a theoretical model for thermoelectric power generation by using the available heat from a power plant condenser. The temperature difference between vapour that formed in condenser tubes and coolant water from the sea was converted to electrical power using the TEG. The vapour in the condenser temperature was maintained at 40°C during a condensation process. The sea water temperature ranges between 15°C and 19°C. The heat conduction model of the cylindrical wall could predict the output power of the TEG. They analysed the output power and the condenser size as a function of heat transfer coefficient, TEG module thickness and TEG conductivity.

Kaibe et al. [76] tested a TEG system using heat from a carburizing furnace at the Awazu plant of Komatsu (Figure 2.42). The heat rejected by the after-burner of the furnace was estimated between 20 kW and 30 kW. 16 of the Bi-Te TEGs were installed between a heated plate and a water cooled plate. The flame from an ignition burner heated the TEG...
collector plate. The system operated at a hot temperature of 250°C and cold temperature of 40°C. It was reported that the efficiency of the heat collector was only 20% which equated 4 kW out of 20 kW of flame heat. The maximum generated power output was 240 W at 220°C of TEG temperature difference. The electricity generated could charge 4 lead storage batteries and could light up LED lamps in the plant.

![Power system of TEG installed with carburizing equipment at the Awazu plant of Komatsu](image)

**Figure 2.42** Power system of TEG installed with carburizing equipment at the Awazu plant of Komatsu [76].

Rodríguez et al. [77] developed a power generation system using TEG which harnessed waste heat from a biomass power plant (Figure 2.43). The target waste heat temperature in this study was less than 150°C. The study objective was to generate enough power to operate sensors for measuring pressure, temperature, flow and PH. The cold side of the ENERKIT SC-127-10-15 TEGs was cooled by finned dissipaters. Several potential hot spots were identified for installing the TEGs such as at the chimney (at 127°C), the smoke filter (at 127°C), the ignition air flow (at 70°C) and the degasser (at 140°C).
Also a mathematical model was developed as an optimization tool for the TEG design by controlling the input parameters such as the TEG geometry and material, the thermal resistance of the dissipator, the electric load resistance, the temperatures at the cold and hot side of the reservoir and the number of modules. One to four modules were simulated and the results showed that an increase in the number of modules resulted in a rise in electricity production. To reduce cost and space, the model predicted that the system could work better using multiple modules per dissipator rather than one module per dissipator. It was found that the electricity production was not able to feed continuously to a vortex flow meter. As an alternative, a battery system was developed to store the power.

![Scheme of biomass power plant](image)

Figure 2.43 Scheme of biomass power plant [77].

Meng et al. [78] studied recycling hot water heated by blast furnace slag and converted the thermal energy into electricity using a TEG. The study presented the relevant parameters that affected the TEG performance such as the thermoelectric element length, flow passage length and water temperature. This system was estimated to produce 0.93 kW/m² power density, equivalent to 2% TEG conversion efficiency.

### 2.5.8 Thermoelectric power generation using solar energy.

Photovoltaics are well established as a direct energy conversion means from solar energy into electric energy. However, the efficiency of PV cells decreases with rise of cell
temperature. On the other hand, the TEG efficiency keeps increasing with rising hot surface temperature. This has attracted many researchers to use TEG technology for directly converting sun energy into electrical energy.

For example, Zheng et al [79] developed a thermoelectric cogeneration system that used waste heat sourced from the boiler and from solar power. The system consisted of two thermoelectric blocks, a parasitic block and the main block as shown in Figure 2.44. Mains water was directed into a cooling plate in the main block to absorb heat from the TEG cold side for preheating purposes. The water then entered the boiler for further heating. The hot side of the TEG was maintained by the boiler waste heat, solar power and boiler-heated hot water. It was reported that the developed laboratory system produced electric power between 15 W and 50 W of 60°C temperature difference. This system conversion efficiency was around 4%.

Figure 2.44 Thermoelectric cogeneration system using waste heat sources from boiler and solar power [79].

X.F Zheng et al. [80] introduced a small scale thermoelectric cogeneration system (TCS) using a combined heat source of boiler waste heat and solar energy. The
thermoelectric module (TEM) was heated by these heat sources in a heat exchanger which contained heat transfer oil. The cooling system used mains water and the recovered heat from the TEM was recycled to the boiler. The study revealed four detailed analysis aspects of the system including the electrical performance, hydraulic performance, thermal performance and dynamic response performance.

A combination of thermosyphon and TEG was studied by Singh et. al [81]. By using a fully passive system that extracted heat from a solar pond, the system produced 3.2 W from 16 TEG modules while maintaining the surface temperature of the modules at 27°C.

Tan et al. [82] developed a passive cooling system that used thermosyphon heat pipe and phase change material (PCM) thermal storage for operating a concentrated thermoelectric generator (CTEG) (Figure 2.45). This system generated electricity by creating a temperature gradient between the generated heat flux of a Fresnel lens concentrator and a PCM cooling system. Heat pipes transported excess heat effectively from the cold side of the TEG to the heat storage tank. The system simulation showed that 9.5 W of electric power could be produced by maintaining the TEG temperature difference at 152°C.

![Figure 2.45 Schematic diagram of proposed CTEG-PCM Cooling system [82].](image-url)
Miljkovic and Wang [83] studied a hybrid solar thermoelectric system (HSTE) which combined a solar concentrator and an evacuated chamber. Figure 2.46 shows that the cylinder type evacuated tube has 3 layers; the first layer is vacuum glass, second is thermoelectric generator (TEG) and third is a thermosyphon. The reflected heat from the solar concentrator is absorbed in the vacuum glass. The evaporator section of the thermosyphon cools the other side of the TEG. The heat collected by the thermosyphon is transmitted to the condenser section of the thermosyphon to be used for other applications such as water heating and cogeneration. An energy model based on a thermal resistance method was developed to evaluate the overall performance of the system. The model was used to investigate the system efficiency that worked under various conditions such as working temperatures between 300-1200K, solar concentrations of 1-100 suns and different materials of thermoelectrical systems and thermosyphons. It was found that the ideal system efficiency could be achieved up to 52.6% by using 100 suns and at 776K of thermoelectric cold side temperature.

Fan et al. [84] designed, modelled and tested a concentrating thermoelectric generator (CTEG) using available energy from the sun. Figure 2.47 shows that the CTEG system utilises a 1.8 m diameter parabolic dish for concentrating sunlight onto a 260 mm diameter receiver made from copper plate. Four thermoelectric cells (TEC) were placed at the bottom of the receiver to convert solar thermal energy directly into electricity. A water cooling block

Figure 2.46 Schematic of the hybrid solar thermoelectric system (HSTE) [83].
was used to remove heat from the cold side of the TEC. This system also was designed to be able to track the sun using a two axis tracking system. The individual TEC was tested under maximum heat flux at 109°C temperature difference and was able to produce 4.9 W of electrical power which was equivalent to 2.9% conversion efficiency. It was found that at 68°C TEC hot side temperature and 35°C TEC temperature difference, the CTEG system was able to generate 5.9 W of electrical power.

![Diagram](image_url)

Figure 2.47 Schematic of a concentrator thermoelectric generator (CTEG) [84].

He et al. [85, 86] conducted parametrical and experimental studies of an integrated solar heat-pipe thermoelectric generator (SHP-TEG) that can be used for both water heating and power generation applications. Figure 2.48 shows an experimental set-up containing an evacuated glass tube, a thermoelectric generator (TEG) and a heat pipe. The evacuated tube collects thermal energy from solar radiation and transfers heat to the condenser side of the heat pipe. This heat pipe condenser is attached to the hot side of TEG. The cold surface of TEG is cooled by exposing it to a water jacket. An analytical model was developed to measure the thermal and electrical efficiencies of the system relating to its important parameters such as the solar irradiation, the ambient and water temperatures, the heat transfer area of TEG and evacuated tube and the TEG parameter.
The model was operated using experimental data and used to simulate and optimize the different design and operating conditions of the SHP-TEG unit. From the simulation, it was found that the SHP-TEG may produce about 55% thermal efficiency and 1% electrical efficiency at the solar radiation more than 600W/m$^2$ and water temperature at 45°C.

![Diagram of an integrated SHP-TE system](image)

Figure 2.48 The schematic diagram of an integrated SHP-TE system [85].

### 2.6 Chapter summary and research gap

There have been a number of valuable studies of heat recovery using the heat pipe heat exchangers (HPHE) and some are summarized in Table 2.3, all of which represent theoretical and experimental investigations. However, none of these studies provide evidence of simultaneous passive heat recovery and power generation using industrial low-temperature waste heat at less than 150°C. The present research focuses on recovering this type waste heat.

The majority of the TEG power generation studies used active cooling methods. The active cooling methods required auxiliary forces that subject to possible mechanical failure of their rotating parts. Any failure of this cooling system could damage the TEG cells and incur
high cost for their replacement. Therefore, a passive system using heat pipes is developed in this research to overcome these problems.

Moreover, most of the TEG studies have only focused on the power generation performance and none of them have considered recovering the heat released from the TEG cold side which can be used for other thermal applications. This study presents a new concept of recovering waste heat and electricity conversion by sandwiching TEGs between heat pipes. This totally passive heat transfer system not only concentrates on electric power generation but also considers the amount of energy that can be recovered from the rejected heat.

This study will lead to a reduction in the required size and material of the combined heat recovery and power generation unit. This size reduction will also decrease the capital costs and increase the reliability of the system. The study will provide a sustainable power source that can improve the overall thermal efficiency, reduce energy consumption and reduce greenhouse emission from industry. This proposed research will provide a deeper understanding of the thermal and electrical behaviour of heat pipe and thermo-electric combination systems. The experience and knowledge gained from the study will be useful for designing an industrial level waste heat recovery cogeneration system.
Table 2.3 The TEG studies using heat pipe as a cooling method.

<table>
<thead>
<tr>
<th>Author</th>
<th>Heat pipe type</th>
<th>Application</th>
<th>Heat source</th>
<th>Cooling method</th>
<th>TEG hot side (°C)</th>
<th>No of TEG</th>
<th>Power output (W)</th>
<th>Conversion efficiency (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Singh et al. 2011)[81]</td>
<td>Copper water thermosyphon</td>
<td>Power generation</td>
<td>Solar pond</td>
<td>Water cooled (forced convection)</td>
<td>80</td>
<td>16 units</td>
<td>3.2</td>
<td>~1%</td>
</tr>
<tr>
<td>(Goncalves et al. 2010)[64]</td>
<td>Variable conducted type</td>
<td>Power generation</td>
<td>Car Engine</td>
<td>Water cooled (forced convection)</td>
<td>217</td>
<td>30 units</td>
<td>550</td>
<td>1.7%</td>
</tr>
<tr>
<td>(Tan et al. 2012) [87]</td>
<td>Copper water thermosyphon</td>
<td>Power generation and cooling</td>
<td>Solar concentrator</td>
<td>Heat pipe-PCM cooled</td>
<td>&gt;150</td>
<td>2 units</td>
<td>4</td>
<td>1.7%</td>
</tr>
<tr>
<td>(Kim et al. 2011) [63]</td>
<td>Copper water</td>
<td>Power generation</td>
<td>Car engine</td>
<td>Water cooled (forced convection)</td>
<td>170</td>
<td>112 units</td>
<td>350</td>
<td>Not provided.</td>
</tr>
<tr>
<td>(Date et al. 2014) [88]</td>
<td>Copper water</td>
<td>Power generation &amp; water heating</td>
<td>Solar concentrator</td>
<td>Heat pipe – water cooled</td>
<td>&gt;150</td>
<td>1 unit</td>
<td>3.5</td>
<td>2.1%</td>
</tr>
</tbody>
</table>
3 Theoretical Modelling of the Heat Pipe Thermoelectric Generator (HP-TEG)

3.1 Introduction to the proposed concept

In this study, a theoretical model has been developed to predict the waste heat recovery and electrical conversion performances using TEGs. The input parameters in this theoretical model are based on an experimental rig as shown in Figure 3.1 and Figure 3.2. The experiment rig incorporates a TEG sandwiched between two heat pipes to achieve the temperature gradient for thermoelectricity generation. The condenser section of heat pipe 1 is thermally attached at the hot side of the TEG as a heater, and the evaporator section of heat pipe 2 provides the cooling role.

Figure 3.1 Experimental rig for the HP-TEG system.
Figure 3.2 Proposed concept of the HPTEG system.

Figure 3.3 and Figure 3.4 show the actual heat transfer device (heat pipe) in the experimental rig and the assembly of the thermoelectric power device, respectively. The heat transfer device contains of two sets of rectangular finned tube heat pipes that are attached to copper blocks. In order to minimize the interfacial thermal resistance of the heat transfer device and TEG, lead free galvanizing solder is used to thermally bond the heat pipes and copper blocks.
Figure 3.3 Finned heat pipe soldered with copper block.

Figure 3.4 Assembly of the thermoelectric power device.
3.2 Theoretical modelling

3.2.1 Energy balance equations

Cold air enters from the upper part of the duct in the \(-x\) direction and exits to the bottom part of the duct \((+x)\) direction as shown in Figure 3.5. The energy balance can be written as:

\[
-\dot{m} c_{p\,\text{air}} \left( \frac{dT_c}{dx} \right) dx = -\dot{m} c_{p\,\text{air}} \left( \frac{dT_h}{dx} \right) dx
\]  

(3.1)

where \(T_c\) and \(T_h\) are the cold and hot air temperatures, respectively. \(\dot{m}\) is the air mass flow rate, \(c_{p\,\text{air}}\) is the specific heat capacity of air and \(dx\) is the increment of length in the \(x\)-direction. The rate of energy transfer from the lower duct through the HP-TEG system is determined using the follow equations:

\[
-\dot{m} c_{p\,\text{air}} \left( \frac{dT_h}{dx} \right) dx = (T_h - T_c) / R
\]  

(3.2)

\[
R = \left( \frac{ab}{bdx} \right) R_M
\]  

(3.3)

where \(R_M\) is the thermal resistance for a single row of thermoelectric power devices, \(a\) is the TEG module thickness in the \(x\)-direction and \(b\) is the module width.

![Figure 3.5 Theoretical modelling domain for HP-TEG system.](image)
An electrical heater is used to simulate the waste heat input to the HP-TEG system. The energy input is shown as:

\[ \dot{m}c_{p\text{air}}(T_h - T_c) = \dot{Q}_{\text{input}} \]  

(3.4)

where \( \dot{Q}_{\text{input}} \) is the rate of heat added to the HP-TEG system. By solving Equations (3.1), (3.2) and (3.4), the temperature profiles of hot and cold air in the upper and lower ducts can be determined using Equations (3.5) and (3.6).

\[
T_h(x) = \left( -\frac{ax}{a\dot{m}c_{p\text{air}}R_M} \right) + \beta
\]

(3.5)

\[
T_c(x) = T_h(x) - \alpha
\]

(3.6)

where \( \alpha = \dot{Q}_{\text{input}}/\dot{m}c_{p\text{air}} \), \( \beta = T_{ci} + \alpha(1 + \frac{L}{a\dot{m}c_{p\text{air}}}) \), \( L \) is the duct length, and \( T_{ci} \) is the inlet cold air temperature. The total rate of heat transfer through the HP-TEG heat exchanger is finally represented by:

\[ \dot{Q}_M = N\left(\frac{x}{R_M}\right) \]

(3.7)

where \( N = L/a \) is the number of rows and \( R_M \) is the total thermal resistance for a single module of HP-TEG (see Figure 3.6 and Figure 3.7). The resistance is defined as:

\[
R_M = R_{air\,h} + 2R_{hp} + 2R_{solder} + 2R_{cb} + 2R_{TIM} + R_{TEG} + R_{air\,c}
\]

(3.8)

where \( R_{air\,h/c}, R_{hp}, R_{solder}, R_{cb}, R_{TIM} \) and \( R_{TEG} \) are the thermal resistances of hot/cold air convection, heat pipes, soldering material, copper blocks, thermal interface material, and thermoelectric generator, respectively.

The effectiveness of a heat exchanger, \( \varepsilon \), is the ratio rate of the heat transferred by the heat exchanger to the maximum possible rate of heat transfer between the air streams. The effectiveness can be expressed as follows:

\[
\varepsilon = \frac{\dot{m}(T_e - T_{ci})}{\dot{m}_{min}(T_h - T_{ci})}
\]

(3.9)

As the mass flow rate is assumed to be constant \( (\dot{m}_{min} = \dot{m}) \), the equation (3.9) can be simplified as follows:
\[ \varepsilon = \frac{T_c - T_{ci}}{T_h - T_{ci}} \]

(3.10)

The thermoelectric power generation \( P_{TEG} \) can be defined as:

\[ P_{TEG} = \mu_{TEG} \dot{Q}_M \]

(3.11)

where \( \mu_{TEG} \) is the conversion efficiency of the thermoelectric generator. The conversion efficiency of the TEG, \( \mu_{TEG} \) is defined as follows:

\[ \mu_{TEG} = \left( 1 - \frac{T_{c TEG}}{T_{h TEG}} \right) \left( \frac{\sqrt{ZT + 1} - 1}{\sqrt{ZT + 1} + \frac{T_{c TEG}}{T_{h TEG}}} \right) \]

(3.12)

where \( T_{c TEG} \) and \( T_{h TEG} \) are the hot and cold side temperature of the TEG, respectively. \( ZT \) is the figure of merit of the TEG.

Figure 3.6 A single module of the heat pipe thermoelectric generator (HP-TEG) and its equivalent thermal resistance, \( R_M \).
3.2.2 Thermal resistance of heat pipe

The heat pipe used in this study has 75 numbers of grooves with 0.075mm depth. It was filled with 600 fine fibre wicks made of cooper. The heat pipe used water as the working fluid. The detail parameters of the heat pipe are shown in Table 3.1.

Table 3.1 Parameters of installed heat pipe.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, ( L )</td>
<td>300</td>
<td>mm</td>
</tr>
<tr>
<td>Outer Diameter, ( d_o )</td>
<td>8</td>
<td>mm</td>
</tr>
<tr>
<td>Wall thickness, ( t )</td>
<td>0.408</td>
<td>mm</td>
</tr>
<tr>
<td>Maximum heat transfer rate, ( \dot{Q}_{max} )</td>
<td>85</td>
<td>W</td>
</tr>
<tr>
<td>Vapor spacing, ( d_v )</td>
<td>0.709</td>
<td>mm</td>
</tr>
<tr>
<td>Diameter of fine fiber wick, ( d_w )</td>
<td>0.045</td>
<td>mm</td>
</tr>
<tr>
<td>Number of fine fiber wick, ( N_w )</td>
<td>600</td>
<td>-</td>
</tr>
</tbody>
</table>

The thermal resistance of the heat pipe is shown as follows:

\[
R_{heat\ pipe} = R_{p,e} + R_{w,e} + R_{w,c} + R_{p,c} \tag{3.13}
\]
where \( R_{p,e} \) and \( R_{p,c} \) are the radial resistances of the heat pipe wall at the evaporator and condenser sections respectively.

The radial resistance at the heat pipe wall is calculated based on Fourier’s law.

\[
R_p = \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi L_{e,c}k_{hp}} \tag{3.14}
\]

where \( d_o \) and \( d_i \) are the heat pipe outer and inner diameters respectively. Since both heat pipes in the HP-TEG module have similar evaporator and condenser lengths, \( L_{e,c} \) will be used for representing both evaporator and condenser lengths for all heat pipes in the HP-TEG system. \( k_{hp} \) is the heat pipe conductivity and \( R_{w,e} \) and \( R_{w,c} \) are the resistances of the liquid wick combination at evaporator and condenser respectively.

The thermal resistance of the liquid wick combination can be expressed as:

\[
R_w = \frac{\ln\left(\frac{d_i}{d_v}\right)}{2\pi L_{e,c}k_{eff}} \tag{3.15}
\]

where \( d_v \) is the vapour spacing.

The effective thermal conductivity, \( k_{eff} \) for a liquid-saturated wick is calculated using the following expression.

\[
k_{eff} = \varepsilon k_l + k_w (1 - \varepsilon) \tag{3.16}
\]

where \( k_l \) and \( k_w \) are the conductivities of water and wick, respectively.

The wick porosity, \( \varepsilon_w \) is defined as:

\[
\varepsilon_w = 1 - \frac{1}{4} \pi 1.05 N_w d_w \tag{3.17}
\]

where \( N_w \) is the mesh number and \( d_w \) is the wire diameter.

### 3.2.3 Thermal resistance of copper block

Copper blocks are attached to both the hot and cold surfaces of the thermoelectric cells. They provide a larger heat conduction area between the heat pipe and TEG. The thermal resistance for a copper block can be calculated as:
\[ R_{cb} = \frac{t_{cb}}{k_{cb} A_{cb}} \]  


where, \( t_{cb} \), \( k_{cb} \) and \( A_{cb} \) are the thickness, thermal conductivity and area of the copper block.

### 3.2.4 Thermal resistance of forced convection

The expression for convective heat transfer coefficient \( h_{\text{air}} \) from the air stream-to-fins and from fins-to-tube is based on Zukauskas [89]. It is represented using the Nusselt number as follows:

\[
Nu = \frac{h_{\text{air}} D_{\text{hyd}}}{k_{\text{air}}} = 0.9 Re^{0.4} Pr^{0.36} \left( \frac{P_{r}}{P_{s}} \right)^{0.25} 
\]  

(3.19)

where \( k_{\text{air}} \) is the thermal conductance of air, \( P_{r} \) is the Prandtl number and \( \frac{P_{r}}{P_{s}} \approx 1 \) for gas with constant \( Pr \).

The hydraulic diameter, \( D_{\text{hyd}} \) is used for calculating the Reynolds number, \( Re \):

\[
Re = \frac{\rho V_{\text{max}} D_{\text{hyd}}}{\mu} 
\]  

(3.20)

The Reynolds number, \( Re \) is calculated based on the maximum fluid velocity, \( V_{\text{max}} \) occurring within the tube bank. Figure 3.8 shows the aligned arrangement of the tube bank, and for a transverse plane \( A_{t} \), the maximum fluid velocity, \( V_{\text{max}} \) is calculated as [90]:

\[
V_{\text{max}} = \frac{S_{t}}{S_{t-D}} V 
\]  

(3.21)

![Figure 3.8 Aligned arrangement of tube bank.](image)

The thermal resistance of the air flow is calculated using the equation:
\( R_{\text{air}} = \frac{1}{\mu_0 h_{\text{air}} A_t} \) \hspace{1cm} (3.22)

where \( \mu_0 \) is the overall surface efficiency and \( A_t \) is the total surface area.

### 3.2.5 Overall surface efficiency

\[ \mu_0 = 1 - \left( \frac{N_f A_f}{A_t} \right) (1 - \mu_f) \] \hspace{1cm} (3.23)

where \( A_t = N_f A_f + A_b \) is the total surface area, \( A_f \) is the fin area, \( N_f \) is the number of fins in the array, \( A_b \) is the un-finned area.

### 3.2.6 Fin efficiency

For rectangular and straight fins, the fin efficiency can be calculated as follows:

\[ \mu_f = \frac{\tanh m l_f}{m l_f} \] \hspace{1cm} (3.24)

and,

\[ m = \left( \frac{2h_{\text{air}}}{k_f t_f} \right)^{1/2} \] \hspace{1cm} (3.25)

where \( k_f \) is the fin thermal conductivity and \( t_f \) is the fin thickness.

### 3.2.7 Pressure drop associated with flow across finned tube banks

The pressure drop across the finned tube banks may be computed as follows [90, 91]:

\[ \Delta P = \frac{G^2}{2 \rho_i} \left[ 1 + \left( \frac{A_{\text{min}}}{A_{fr}} \right)^2 \left( \frac{\rho_i}{\rho_o} \right) + f \frac{A_{\text{min}}}{A_t} \frac{\rho_i}{\rho_m} \right] \] \hspace{1cm} (3.26)

where \( G \) is the maximum mass velocity, \( \rho_i \) and \( \rho_o \) are the fluid inlet and outlet density, \( f \) is the friction factor, \( A_{\text{min}} \) is the minimum free-flow area of the finned passages, \( A_{fr} \) is the frontal area, and \( A_t \) is the total heat transfer surface area.

### 3.3 Determination of TEG thermal properties

An experiment was conducted to measure the thermal properties of the TEG using a forced convection cooling approach. This experiment was important for determining the following parameters which are required for the mathematical modelling:
a) TEG thermal resistance, $R_{TEG}$

b) TEG thermal-to-electric power conversion efficiency, $\mu_{TEG}$

In the experiment, an electrical heater was used to supply heat to the hot side of the TEG. The cold side was cooled by a water cooling block as shown in Figure 3.9. An electronic load device (150W BK Precision 8540) was connected to the electrical circuit for determining the electrical resistance of the TEG by matching the internal and external electric resistances. All the data including temperature measurements, output voltage and current were recorded by using an electronic data logger (Agilent 34970A).

Figure 3.9 Schematic diagram to determine the properties of the installed TEG.

Figure 3.10 and Figure 3.11 present the experimental results obtained from a single TEG. The tested heating wattage as the heat source to the TEG was from 20 W to 120 W. The physical dimensions of the tested TEG were 40 mm (length) x 40 mm (width) x 4 mm (depth). For each heat load, the electronic load device was used to vary the electrical resistance to obtain the voltage-current characteristics and maximum power point (MPP) of the TEG. The results are shown showed in Figure 3.10 and Figure 3.11, respectively. The MPP occurs when the internal resistance of the TEG matches the external load resistance.
which is determined by the electronic load device. The measured internal resistance of the TEG is approximately 2.5 Ω as shown in Figure 3.12. In Figure 3.11, the highest power output obtained was 1.7 W with 120 W heat source. Hence, the thermal resistance of the TEG, $R_{TEG}$ was determined experimentally using the system described. The $R_{TEG}$ had a small change with a standard deviation of 0.03 °C/W over the heating wattage of 120 W. Therefore, an average value of 0.8 °C/W was considered in this study. The result of this investigation is summarized in Table 3.2.

Table 3.2 Experimental data obtained from the thermoelectric generator (TEG) performance test.

<table>
<thead>
<tr>
<th>Heat Input, (W)</th>
<th>20</th>
<th>40</th>
<th>60</th>
<th>80</th>
<th>100</th>
<th>120</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum power output, $P_{\text{max}}$ (W)</td>
<td>0.06</td>
<td>0.24</td>
<td>0.52</td>
<td>0.84</td>
<td>1.37</td>
<td>1.72</td>
</tr>
<tr>
<td>Current, $I_{\text{max}}$ (A)</td>
<td>0.15</td>
<td>0.32</td>
<td>0.45</td>
<td>0.58</td>
<td>0.79</td>
<td>0.85</td>
</tr>
<tr>
<td>Voltage, $V_{\text{max}}$, U (V)</td>
<td>0.38</td>
<td>0.76</td>
<td>1.14</td>
<td>1.46</td>
<td>1.74</td>
<td>2.02</td>
</tr>
<tr>
<td>Electrical Resistance, $\Omega$ (Ohm)</td>
<td>2.48</td>
<td>2.39</td>
<td>2.51</td>
<td>2.52</td>
<td>2.20</td>
<td>2.37</td>
</tr>
<tr>
<td>Open circuit voltage, $U_{\text{oc}}$ (V)</td>
<td>0.73</td>
<td>1.51</td>
<td>2.32</td>
<td>3.05</td>
<td>3.73</td>
<td>4.41</td>
</tr>
<tr>
<td>TEG hot side temperature, $T_{\text{hot TEG}}$, (°C)</td>
<td>43</td>
<td>67</td>
<td>94</td>
<td>118</td>
<td>143</td>
<td>161</td>
</tr>
<tr>
<td>TEG cold side temperature, $T_{\text{cold TEG}}$, (°C)</td>
<td>27</td>
<td>34</td>
<td>43</td>
<td>51</td>
<td>58</td>
<td>66</td>
</tr>
</tbody>
</table>
Figure 3.10 Current-voltage characteristics of the TEG unit.

Figure 3.11 Output power versus voltage outputs of the TEG.
Figure 3.12 Output power versus electrical resistance at different input heat loads.

Figure 3.13 shows the TEG conversion efficiency versus the TEG temperature gradient. It is observed that the efficiency increased with the rise of temperature difference of the TEG. The highest efficiency found in the experiment was 2.3% at a temperature difference of 85°C.
3.4 Results and discussion of theoretical model

The aim of the theoretical model developed was to predict the performance of the HP-TEG system under different parametric conditions. The key parameters are: number of heat pipes, number of installed TEGs, rate of heat input, and air flow rate. In addition, the optimal parameters and settings for the system can be determined using the developed model.
3.4.1 The effect of mass flow rate

Figure 3.14 Heat exchanger effectiveness against mass flow rate.

In this computer simulation, the temperature of supply air is assumed constant. The air flow rate in the duct was varied between 0.02 kg/s (0.6 m/s) to 0.06 kg/s (1.9 m/s). Figure 3.14 shows that the effectiveness of the HPTEG drops with the rise of air face velocity. The easiest way to explain this behaviour is by analysing a simple heat exchanger model consisting of one thermal resistance, $R$. Based on Figure 3.5, an energy balance of the heat exchanger exposed to two streams of air at different temperatures can be developed as follows:

$$(T_h - T_{ci})/R = \dot{m} c_{p\, air}(T_c - T_{ci})$$

(3.27)

After substituting Equation (3.27) into Equation (3.10), the effectiveness can be rearranged as:

$$\varepsilon = \frac{1}{c_{p\, air} \dot{m} R + 1}$$

(3.28)

Equation (3.28) shows that the effectiveness of the heat exchanger falls with the rise of mass flow rate and when the mass flow rate approaches infinity, the effectiveness approaches zero.
Figure 3.14 also shows the effectiveness increases with the numbers of rows. Apparently, the increasing number of rows has increased the amount of heat transfer that contributes to an increase of the air temperatures. The effect of rows is discussed further in the next section.

### 3.4.2 The effect of the number of rows

Figure 3.15 shows that variation of the heat transfer rate with the number of rows installed in the heat exchanger. The results show that the rate of heat transfer can be increased by installing more rows of HP-TEG module. The heat transfer rate from the hot air to the cold air stream depends on the module thermal resistance. As the rows are arranged in parallel to the direction of heat transfer, the total thermal resistance of the module reduces to $R_M/N$ which results in the rise of heat transfer rate with the installed row increment. Conversely, the heat transfer rate decreases with the rise of mass flow rate. Based on equation 3.6, $\alpha = T_h - T_c = \dot{Q}_{\text{input}}/\dot{m}c_{p\text{air}}$, the temperature gradient between hot and cold stream is inversely correlated with the mass flow rate. In the experimental arrangement, the heat input $\dot{Q}_{\text{input}}$ is maintained constant at 2 kW and as the mass flow rate approaches infinity, the temperature gradient will approach zero. This implies that no heat transfer will occur when there is no temperature gradient between the two streams at a high mass flow rate. The main objective of setting up the experimental rig was to validate the developed theoretical model. For this reason, the number of rows of HP-TEG was limited to 8 units which were suitable for laboratory scale testing.
Figure 3.15 Effect of the number of rows.

Figure 3.16 presents the temperature gradient between hot and cold sides of a TEG for the case of 8 installed rows. As the maximum hot side temperature of TEG specified by the manufacturer is 125°C, Figure 3.16 provides important information in predicting the ideal mass flow rate that maximises the performance of the HP-TEG system within the maximum operating temperature of the TEG. For the current HP-TEG system design, the mass flow rate must be around 0.03kg/s (0.9 m/s) or greater to avoid damage to the installed TEG. The prime damage risk is the melting of soldering materials caused by high cell temperature. However, the drawback of using higher mass flow rates is the resulting lower temperature gradients of the TEG associated with low heat transfer rates as illustrated previously in Figure 3.15.
Figure 3.16 Temperature of TEG surfaces and TEG temperature difference versus air mass flow rate.

Figure 3.17 Heat transfer rate and electrical power output against air mass flow rate.
Figure 3.17 shows the heat transfer rate and electrical power output versus the mass flow rate. For the 8 installed rows at a mass flow rate of 0.03 kg/s (0.9 m/s), the predicted power output is approximately 10 W and 1.6 kW of recovered waste heat by the modules.
4 Experimental Method of Simultaneous Power Generation and Heat Recovery Using the Heat Pipe Thermoelectric Generator (HP-TEG)

4.1 Introduction

An experimental set-up of the heat pipe thermoelectric generator (HP-TEG) system was entirely designed and fabricated in the Renewable Energy Laboratory at RMIT University. Figure 4.1 illustrates the HP-TEG system consisting of several components such as an inlet and outlet air ducts, a U-duct connector, a 2 kW electric duct heater, a PID temperature control, combined heat pipe thermoelectric modules, and an air blower. The inlet and the outlet ducts were made of stainless steel, and they have a similar size of 1500 mm (length) x 160 mm (width) x 170 mm (height).

Eight rows of the HP-TEG modules were placed in these ducts. The modules were arranged in series to the direction of air flow. A 55 mm gap separated each module. The fins parts of the heater were exposed to the air flow through a rectangular hole in the outlet duct wall. The galvanized steel terminal of the heater was screwed in the wall of outlet duct. The heater temperature setting was controlled by a proportional-integral-derivative (PID) controller. A flexible aluminium hose was connected to the air blower and the entrance of the inlet duct for supplying air flow in the duct system. The detail specifications of the components and experiment procedures are discussed in the next sub-chapters.
4.2 Heat pipes module

The heat pipe is one of the main components in this research. It was bought from Fujikura Ltd, Japan. As shown in Figure 4.2 and Figure 4.3, one set of the heat transfer module consists of four heat pipes. The heat pipes were finned with 62 pieces of 50 mm wide x 153 mm long aluminium sheet. The distance from the left edge of the fins to the heat pipe is 15 mm whereas the distance from the right edge to the heat pipe is 9 mm (Appendix C). Although the initially designed fins were symmetry (where the left and right edges have similar distance), the manufacturer suggested using available off-the-shelf fins because of their low price. The bottom end of each heat pipe was soldered into a copper block. 4 mm holes were drilled in six different positions in the copper block for clamping purposes. The detailed specifications of the heat pipe module are displayed in Table 4.1.
Figure 4.2 Front view of the heat pipe module.

Figure 4.3 Side view of the heat pipe module.
Table 4.1 Specifications of the heat pipe module.

<table>
<thead>
<tr>
<th>Heat Pipe</th>
<th>Rectangular Fins</th>
<th>Copper Block</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material of pipe, copper</td>
<td>Material, aluminium</td>
<td>Height, 120 mm.</td>
</tr>
<tr>
<td>Diameter of pipe, 8 mm</td>
<td>Number of fins, 62</td>
<td>Width, 80 mm.</td>
</tr>
<tr>
<td>Length of pipe, 300 mm</td>
<td>Fin height, 153 mm</td>
<td>Thickness, 15 mm.</td>
</tr>
<tr>
<td>Length of evaporator section, 120 mm</td>
<td>Fin length, 153 mm</td>
<td></td>
</tr>
<tr>
<td>Length of condenser section, 150 mm</td>
<td>Fin width, 50 mm</td>
<td></td>
</tr>
<tr>
<td>Type of wick, 600 copper wires</td>
<td>Fin gap, 2 mm</td>
<td></td>
</tr>
<tr>
<td>Working fluid, water</td>
<td>Fin thickness, 0.5 mm</td>
<td></td>
</tr>
<tr>
<td>Heat duty per pipe, 100 W</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

A functional test of all the sixteen heat pipe modules was conducted by immersing them in hot water. Figure 4.4 illustrates that the temperature of the hot water was maintained at 60 °C with the ambient temperature of 25-30°C. Only the copper block section was submerged in the hot water to observe the heat transfer effect from the evaporator side to condenser side (finned section of the module). Six temperature points were measured at the end tip of each heat pipes every 90 seconds. The temperature measurement locations were marked as Thp#1, Thp#2, Thp#3 and Thp#4.

Figure 4.4 Functionality testing of the heat pipe module.
The result from the heat pipe temperature distribution can be seen in Table 4.2. It was found that there were minimal temperature differences between each heat pipe because of the asymmetrical design of the fins.

<table>
<thead>
<tr>
<th>Test No</th>
<th>Temperature point (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Thp#1</td>
</tr>
<tr>
<td>1</td>
<td>51.86</td>
</tr>
<tr>
<td>2</td>
<td>52.13</td>
</tr>
<tr>
<td>3</td>
<td>49.24</td>
</tr>
<tr>
<td>4</td>
<td>51.74</td>
</tr>
<tr>
<td>5</td>
<td>52.48</td>
</tr>
<tr>
<td>6</td>
<td>52.33</td>
</tr>
<tr>
<td>7</td>
<td>51.94</td>
</tr>
<tr>
<td>8</td>
<td>52.28</td>
</tr>
<tr>
<td>9</td>
<td>51.16</td>
</tr>
<tr>
<td>10</td>
<td>52.59</td>
</tr>
<tr>
<td>11</td>
<td>52.68</td>
</tr>
<tr>
<td>12</td>
<td>53.03</td>
</tr>
<tr>
<td>13</td>
<td>52.41</td>
</tr>
<tr>
<td>14</td>
<td>52.76</td>
</tr>
<tr>
<td>15</td>
<td>51.98</td>
</tr>
<tr>
<td>16</td>
<td>52.87</td>
</tr>
</tbody>
</table>

4.3 Thermoelectric generator (TEG)

Bismuth Telluride based thermoelectric generators (TEG) with a size of 40 mm x 40 mm x 4 mm was used in this study (Figure 4.5). According to the manufacturer specifications, the TEG has 127 P-N type thermocouple legs and has the maximum allowable operating temperature of 125°C.
4.4 Silicone thermal paste

A silicone heat transfer compound was used to reduce the thermal contact resistance between the thermoelectric generator (TEG) and the copper block of heat pipe module. The compound has a grease-like silicone material and heavily impregnated with high conductive zinc oxide. This compound usually used for thermo coupling of electronic components and heat sinks. It has a high thermal conductivity of 0.9 W/mK and can be used for temperature range between -200°C and 130°C (Appendix D).
4.5 Electric duct heater

For the experimental purposes, an electric duct heater was used to replicate industrial waste heat as shown in Figure 4.7. Due to safety issues, the electrical heater was set less than 2 kW heating power in order to limit the maximum operating temperature. The detail specifications of the heater are listed in Table 4.3. The details information of the heater is available in Appendix E.

<table>
<thead>
<tr>
<th>Heating capacity</th>
<th>2 kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage (V)</td>
<td>240 single stage</td>
</tr>
<tr>
<td>Fin material</td>
<td>Stainless steel</td>
</tr>
<tr>
<td>Terminal box material</td>
<td>Galvanized steel with mounting flange</td>
</tr>
<tr>
<td>Electrical Insulation</td>
<td>6 mm millboard with mounting face</td>
</tr>
</tbody>
</table>

This electric duct heater can withstand hot air temperature up to 400°C. However, for safety precaution, a proportional-integral-derivative (PID) temperature controller (Figure 4.8) was connected to this heater for adjusting the desired operating temperature. A thermocouple temperature sensor attached to the fin of the electric heater was connected to the PID controller for temperature monitoring. There are two adjustable temperature control panels available; 1) a maximum temperature setting, and 2) a desirable operating temperature. Once the maximum temperature is set, the user cannot work exceeding this temperature point because the PID controller automatically disconnects the electrical power supply to the heater.
4.6 Air duct system

An air duct system was fabricated using 3 mm mild steel. The duct has 160 mm x 170 mm cross section area and 1500 mm long (Figure 4.9). The inlet and outlet duct were connected by using the U-duct connector and fixed on a stand made of hollow bar steels for stability during conducting experiments.
Figure 4.9 Diagram of the U-type air duct system.

### 4.7 AC fan blower

A flow damper was installed at the AC fan blower entrance to control the air flow rate into the duct (Figure 4.10). The AC air blower and the inlet duct were connected by 90 mm aluminium flexible hose (Figure 4.11).
Figure 4.10 AC Fan Blower.

Figure 4.11 Connection detailed between the fan blower and inlet duct.
4.8 Air flow measurement

The stagnation pressure (total pressure) in the air duct was measured using a Pitot tube. The Pitot tube was inserted in an extension pipe of the outlet duct (Figure 4.12 and Figure 4.13) and connected to a liquid column manometer (Figure 4.14) for measuring the dynamic pressure, which is equivalent to the pressure difference between total pressure and static pressure. The air velocity was calculated by taking the pressure difference in the manometer gauge that follows the Bernoulli’s equation.

The standard formula for calculating the velocity from the velocity pressure is

\[ V = 4.05 \sqrt{P_v} \]  \hspace{1cm} (4.1)

where \( P_v \) is the velocity pressure at standard air density of 20°C and 760 mmHg.

For non-standard air conditions, the velocity pressure can be calculated as follows:

\[ V = 4.05 \sqrt{\frac{760 \times \frac{T}{293} \times \frac{10350}{10350 + P_s} \times P_v}{B}} \]  \hspace{1cm} (4.2)

where

\( V \) = velocity, m/s

\( B \) = barometric pressure, mmHg

\( T \) = absolute pressure,°K (t°C+273 where t is the air stream temperature)

\( P_s \) = static pressure, mm H₂O

\( P_v \) = velocity pressure, mm H₂O

It was noted that the expression \( \frac{10350}{10350 + P_s} \) is a correction for the static pressure in the duct and it may be ignored if \( P_s \) is less than 250 mm H₂O.
Figure 4.12 Extension duct length for the measurement of air velocity.

Figure 4.13 Pitot tube installation at the exit of outlet duct.
Figure 4.14 Air Flow Type 4 Test Set for measuring the air velocity in the duct [92].

The ASRAE standard velocity traverse of air ducts (log-Tchebycheff method) was used to get an average air velocity. The position of the measurement points is shown in Figure 4.15. The volume flow rate was calculated by using the averaged air velocity readings.

Figure 4.15 Measuring point locations for traversing a round duct using the log-Tchebycheff method [93].
4.9 Temperature measurement

The temperature measurements of air in the ducts were taken using T-type thermocouples. The PFA Teflon twisted pair thermocouples have strand size of 0.2 mm. The temperature rating of the thermocouples is between -75°C and 260°C. The T-type thermocouples were connected to the Agilent 34970A data logger for data recording. Figure 4.16 illustrates the locations where the temperatures of air were measured. At each location, the average temperature of the air was calculated using the measured results from 9 thermocouples as shown in Figure 4.17.

![Diagram showing measurement locations](image)

| Tci  | cold air inlet temperature. |
| Tco  | cold air outlet temperature. |
| Thi  | hot air inlet temperature.  |
| Tho  | hot air outlet temperature.  |

Figure 4.16 Side view of the air measurement locations.
4.10 Data acquisition system

The Agilent 34970A data logger (Figure 4.18) recorded the measurement data during the experiment. This data logger has capable of measuring and converting 11 different input signals such as temperature with thermocouples, DC/AC volts, 2 and 4 wire resistances, frequency and period, and DC/AC currents. It has 20 channels which each channel individually configurable. The data logger could easily be connected to a computer using USB port. The BenchLink Data Logger Software was provided for configuring and controlling the data measurement. The experiment data were recorded and stored in comma-separated values (CSV) format every 10 seconds throughout the tests. This format has been converted into Excel file for further analysis.
4.11 Installation procedure of the heat pipe thermoelectric module

Figure 4.19 shows the methods that involved for installing a module of the HP-TEG. The installation procedures are as follows:

a) Silicone thermal paste was applied on a copper block surface. The thermal paste was used to eliminate air gaps or to reduce thermal contact resistance between the copper block and TEG surfaces. This thermal paste was evenly distributed over the surfaces using a wooden spreader.

b) Six units of the TEG were electrically connected in series using electric wires. The cold side of the TEGs were attached to the copper block surfaces that were layered with the thermal paste.

c) Another layer of the thermal paste was applied onto the hot side surface of the TEG.

d) The TEGs were sandwiched between two copper blocks by clamping them with 4 mm nuts and bolts.

e) A complete set of the heat pipe thermoelectric module was finally inserted into the air duct and was arranged in the vertical direction.
Figure 4.19 Installation procedure of a HP-TEG module
(a) Layering silicone thermal paste onto copper block surface, (b) Placing thermoelectric generators (TEGs) that electrically connected in series, (c) Layering thermal paste onto TEG surfaces, (d) Sandwiching TEGs with two copper blocks using nuts and bolts, (e) Installing heat pipe thermoelectric module in the air duct.
4.12 Experimental procedure

To begin the test, the air blower starts to drive air into the inlet duct as shown in Figure 4.20. The air velocity was set by adjusting a damper at the air blower inlet. Once the air flow was stable, the 2 kW electric heater was activated for heating the flowing air. The air was heated until its temperature achieved a steady state. At this state, the TEG electric wires were connected to the data logger for measuring the current and voltage outputs. The DC electronic load device was also linked to the TEG electrical connections. The DC electronic load device acted as an external electric resistance to the system. The electronic load resistor was varied between 1000 Ω and 40 Ω to obtain the current and voltage output of the TEGs. The maximum power point (MPP) was determined by multiplying the voltage data and the current data at a matching load. The temperature, voltage, and current measurement data were automatically recorded in a computer connected to the data logger.

Figure 4.20 Schematic diagram of the HPTEG experiment.
5 Experimental Assessment and Theoretical Validation of the Heat Pipe Thermoelectric Generator (HP-TEG) System.

5.1 Introduction

This chapter describes the outcomes obtained from the heat pipe thermoelectric generator (HP-TEG) system experiments. The full-scale prototype of the HP-TEG system complete with insulation and data logging equipment has been installed as shown in Figure 5.1. As a sequel to the theoretical model described in Chapter 3, Section 5.2 explains the effect of air convection on the experiments. The thermal and electrical performances of the HP-TEG are discussed in the latter sections.

![Experimental setup of the HP-TEG installed with insulation and data acquisition system.](image)

Figure 5.1 Experimental setup of the HP-TEG installed with insulation and data acquisition system.

5.2 The effect of air convection to the HP-TEG system.

Figure 5.2 shows how the air thermal resistance, $R_{\text{air}}$, decreases with increasing air face velocity, $V_{\text{air face}}$. The slope of the air thermal resistance against the air face velocity, $dR_{\text{air}}/dV_{\text{air face}}$, is plotted in the same figure. The $dR_{\text{air}}/dV_{\text{air face}}$ curve showed an obvious drop for air face velocities between 0 and 1 m/s. Moreover, the $dR_{\text{air}}/dV_{\text{air face}}$...
curve approached zero when the air face velocity was larger than 1 m/s. Although the air thermal resistance showed a significant change in the lower air face velocity range especially at less than 1 m/s, the minimum allowable air face velocity in the actual experiment was limited to above 1.1 m/s. The reason was that the air temperature became excessive when the air velocity was set at less than 1.1 m/s. As can be seen in Figure 5.3, the theoretical prediction of the hot air inlet temperature, $T_h$, drastically increases to over 125°C when the air face velocity is reduced to less than 1 m/s. The TEG used in this study had a temperature limit of 125°C. Beyond this limit, the TEG material might be damaged which could affect its performance.

The air thermal resistance $R_{air}$ is part of the total thermal resistance, $R_M$, for a single module of the HP-TEG as stated in equation 3.8. The other components of the overall thermal resistance in equation 3.8 including for the heat pipe, $R_{hp}$ and the TEG, $R_{TEG}$ were found constant and independent of air face velocity. The equation 3.7 can be rearranged as $T_h - T_c = \dot{Q}_M R_M$. Using the new arrangement of equation 3.7, it is seen that the decrease of air thermal resistance, $R_{air}$ with increasing air face velocity (Figure 5.2) has decreased the overall thermal resistance, $R_M$. This caused the temperature difference between the hot and the cold sides, $T_h - T_c$ to decrease as shown in Figure 5.4. The drop of the temperature difference, $T_h - T_c$ affected the heat transfer performance of the HP-TEG system as discussed in section 5.4.
Figure 5.2 Thermal resistance of air, $R_{air}$ versus air face velocity, $V_{air\ face}$.

Figure 5.3 Theoretical inlet hot air temperature, $T_{hi}$ versus air face velocity, $V_{air\ face}$.
5.3 Results and discussion of the HP-TEG experiments

This section discusses the results obtained from the experiments, as shown in Figure 5.1. The temperature changes of the cold air inlet, cold air outlet, hot air inlet and hot air outlet versus time for an air speed of 1.4 m/s are presented in Figure 5.5. It can be clearly observed in this figure that the hot air inlet temperature, $T_{hi}$ was the first to rise before reaching a steady state at 82°C. The heat transfer through the HP-TEG module increased the cold air temperature, $T_{ci}$ from 28°C to 48°C before leaving the cold outlet. The hot air exited the system at 57°C. The system achieved a steady state at approximately 3000s (50 minutes).
The actual heat transfer rate from the hot side to the cold side, \( \dot{Q}_{\text{actual}} \), could not be larger than the 2 kW of heat supplied by the electric heater. The equation 5.1 can be rearranged as \( \Delta T_c = \dot{Q}_{\text{actual}} / m_{\text{air}} c_{p \text{air}} \). This equation shows that any increment of the air face velocity (or mass flow rate) could reduce the temperature difference of air at the cold side. This trend was demonstrated by the experimental result shown in Figure 5.6.
Figure 5.6 Temperature gradient created between the inlet and the outlet of the cold air stream.

Figure 5.7 shows the cold air outlet temperature, $T_{co}$ as a function of the air face velocity. It was found that $T_{co}$ decreased with increase of air speed. The highest $T_{co}$ measured was 59°C at 1.1 m/s airspeed. The theoretical predictions indicated similar trends to the experimental findings with an average deviation of 5%. Figure 5.8 shows the hot air outlet temperature, $T_{ho}$ against the air face velocity. The figure shows a trend similar to Figure 5.7 with an average deviation of 18% between the theoretical results and experimental measurements. A higher deviation of the temperature measurement was found in the hotter duct because the location of temperature sensor was comparatively close to the electric heater (Figure 4.16). Therefore heat radiation from the heater has also influenced the temperature measurement. To reduce the effect of heat radiation and improve the heat convection, a copper mesh was installed in between the temperature sensor and the heater.
Figure 5.7 Outlet temperature of cold air, $T_{co}$ versus air face velocity.

Figure 5.8 Outlet temperature of hot air, $T_{ho}$ versus air face velocity.
5.4 Heat transfer performance of the HP-TEG system

The heat transfer performance of the HP-TEG system may be evaluated in terms of its effectiveness. The actual heat transfer rate of the system is determined as follows:

\[ Q_{actua} = \dot{m}_{air} c_{p\ air}(T_{co} - T_{ci}) \]  

where \( \dot{m}_{air} \) is the mass flow rate of the air, \( c_{p\ air} \) is the air heat capacity and \( T_{co} \) and \( T_{ci} \) are the outlet and inlet temperatures of the cold air, respectively.

The effectiveness of the HP-TEG heat exchanger, \( \varepsilon \), is defined as the ratio of the rate of heat recovery to the maximum possible rate of heat transfer between the air streams. It is expressed as follows:

\[ \varepsilon = \frac{\dot{m}_{air} c_{p\ air}(T_{co} - T_{ci})}{\dot{m}_{air\ min} c_{p\ air}(T_{hi} - T_{ci})} \]  

Where \( \dot{m}_{air\ min} \) is the lesser heat capacity of the hot or cold air streams and \( T_{hi} \) is the hot air inlet temperature. For the experimental arrangement shown in Figure 5.1, the mass flow rate of air through the system is considered constant \( (\dot{m}_{air\ min} = \dot{m}_{air}) \) as the cold and the hot ducts are connected. For this reason, the heat capacities for the cold and the hot air streams are assumed to be similar. The heat exchanger effectiveness is therefore simplified as follows:

\[ \varepsilon = \frac{T_{co} - T_{ci}}{T_{ho} - T_{ci}} \]
Figure 5.9 shows that the rate of heat transfer decreases with rising air face velocity. The maximum heat transfer rate of 1079 W was obtained when the air face velocity was at the lowest speed of 1.1 m/s. Conversely, the minimum heat transfer rate of 835 W was obtained at the highest airspeed setting of 1.6 m/s.

It was observed that the temperature of air in the hot duct rose when the air face velocity reduced further. When a very low airspeed was used, the maximum temperature difference was achieved between the hot and cold duct, allowing a higher rate of heat transfer through the heat exchanger. However, when the air velocity was increased, the temperature difference between the hot and the cold ducts dropped and less heat transfer occurred.

Good agreement was observed between the experimental results and the theoretical predictions with an average deviation of approximately 10% as shown in Figure 5.9. The experimental heat transfer rates were lower than the theoretical predictions because of the heat losses through the duct wall and the module. These heat losses were unavoidable, despite the experimental rig being well insulated. The theoretical model was developed using the 1-D
thermal resistance method with no heat loss assumed through the heat exchanger wall. The heat loss explains why the theoretical heat transfer rate was predicted to be higher than the experimental results.

Figure 5.10 The heat exchanger effectiveness against the air face velocity.

Figure 5.10 shows the effectiveness of the HP-TEG system versus the air face velocity. It displays the same trend as Figure 5.9. It was concluded that the system became more efficient when operated at a lower airspeed. The measured effectiveness increased from 35% to 41% when the airspeed was reduced from 1.6 m/s to 1.1 m/s. The theoretical value slightly under-predicted the experimental results. However, the deviation was small, being approximately 5%.
5.5 Output power performance of the HP-TEG system

![Diagram of MPP vs Air Face Velocity](image)

Figure 5.11 Maximum power point (MPP) produced by the eight modules of the HP-TEG system.

![Diagram of Voltage, Current, and Power Characteristics](image)

Figure 5.12 The voltage - current and power characteristics of the HP-TEG for various air face velocities.
Figure 5.11 shows that the maximum power point (MPP) tends to reduce when the air velocity increases. As expected, the highest MPP of 7 W was obtained when the system operated at the lowest air face velocity of 1.1 m/s. The MPP started to drop once the airspeed was increased. The MPP dropped to approximately 4 W when the air flow reached the highest speed of 1.6 m/s. Good agreement could be observed between the theoretical predictions and experimental results with an average deviation of 13%. Figure 5.12 shows the measured voltage-current characteristics and the corresponding power curves for different air face velocities. A peak can be seen in each power curve showing the MPP for each air face velocity setting. The highest MPP peak was produced by the power curve at an air face velocity of 1.1 m/s.

Figure 5.13 shows the MPP generated by the individual modules of the HP-TEG system for an air velocity of 1.1 m/s. The highest power generated by the module no. 1 was around 1.3 W. The module no. 1 was located close to the electrical heater as indicated in Fig. 4.20. The power outputs of the other modules (no. 2 to no. 8) showed a reducing trend from 0.9 W to 0.5 W. In addition, Figure 5.14 presents the open-circuit voltages (OCV) for the individual modules of the HP-TEG system. The OCV showed a declining pattern from the module no. 1 to module no. 8. This pattern was attributed mainly to the reduction of the temperature gradient between the hot and cold copper blocks as shown in Figure 5.15. For a counter flow case, a constant temperature gradient between the two air streams should be achieved. However, in the real case, the temperature gradient was not uniform as heat losses and the pressure drop occured in the system. The temperature gradient dropped from 28°C to 15°C.
Figure 5.13 Maximum Power Point (MPP) produced by each module of the HP-TEG system.

Figure 5.14 Open-circuit voltage produced by each module of the HP-TEG system.
5.6 Pressure drop associated with the air flow across the HP-TEG finned tube banks

Figure 5.16 indicates that the measured pressure drop increases with increasing air face velocity in the duct. However, the pressure drop was lower than 40 Pa over the range of airspeeds used in this study. The previous discussion showed that the proposed system performed better at the lowest air speed of 1.1 m/s. At this speed setting, the air flow developed an even smaller pressure drop of around 20 Pa. This result shows that the finned tube banks arrangement of the HP-TEG did not produce a significant pressure drop that would otherwise have caused a need for additional fan power for the existing air flow system.
Figure 5.16 Pressure drop of air flow across the finned tube bank of the HP-TEG system.
6 Counter Flow Experiment on the HP-TEG

6.1 Introduction

In the previous design of the heat pipe thermoelectric generator (HP-TEG) heat exchanger described in Chapter 5, the cold duct was connected to the hot duct using a U-duct connector. The cold air from the surroundings was preheated in the condenser section of the cold duct before entering the heating section of the hot duct. As a result, a higher temperature of the air was obtained before it entered the evaporator section of the HP-TEG in the hot duct and this resulted in a better heat recovery ratio. However, this type of heat exchanger configuration was practical only for use in the laboratory experiment for validating theoretical models as reported by several authors [23, 24, 94-100].

In the practical industrial situation, the hot flue gas and the cold air from the surroundings are un-mixed and separated. Therefore, the cold air will not be contaminated by flue gas during the heat transfer process. Normally, industrial heat exchangers operate in a counter-flow configuration. The advantages of using a counter-flow heat exchanger (CFHE) (Figure 6.1) are listed as follows [101]:

i) More uniform temperature differences between the two fluids reduces temperature stress in the heat exchanger.

ii) The outlet temperature of the cold fluid can approach the highest temperature of the hot inlet fluid.

iii) A more uniform heat transfer rate can be achieved because of the uniform temperature difference obtained.

In this chapter, study of a counter flow arrangement of the (HP-TEG) is described. The objectives of this study were as follows:

a) To validate using experimental results the theoretical model developed for the counter flow case of the HP-TEG.

b) To conduct a case study by using actual data concerning waste heat from a bakery oven. The data were used as the input parameters for the validated model. The thermal and electrical performances of the bakery HP-TEG system are investigated.

c) To conduct an economic feasibility study of the HP-TEG in a bakery using the validated model.
6.2 Analytical model

The analytical model of the HP-TEG heat exchanger in a counter flow arrangement was developed using the effectiveness-NTU method. The outlet temperature of air in the HP-TEG heat exchanger can be predicted using this method if the inlet temperatures of air are known. The heat transfer rate through the heat exchanger is determined using the overall heat transfer coefficient, $UA$:

$$\dot{Q} = UA_s(T_h - T_c) \quad (6.0)$$

Where $UA_s$, which equals $1/R_M$, is the overall heat transfer coefficient and $R_M$ is the total thermal resistance for a single module of the HPTEG. The calculation of $R_M$ is from Equation 3.8. The thermal performance of the HP-TEG heat exchanger in the counter flow arrangement is assessed by the effectiveness, $\varepsilon$ and is defined as the ratio of the actual heat transfer rate, $q$ to the maximum possible heat transfer rate, $q_{max}$ [90]

$$\varepsilon = \frac{q}{q_{max}} \quad (6.1)$$

and

$$q_{max} = C_{min}(T_{h,i} - T_{c,i}) \quad (6.2)$$

Where $C_{min}$ is the lesser of $C_c$ or $C_h$.

The actual heat transfer rate can be determined from this expression:

$$q = \varepsilon C_{min}(T_{h,i} - T_{c,i}) \quad (6.3)$$
A dimensionless parameter used in analysing the heat exchanger is termed the number of heat transfer units (NTU) and is defined as a ratio of the overall thermal conductance to the smaller heat capacity rate:

$$NTU = \frac{UA}{C_{\text{min}}}$$  \hspace{1cm} (6.4)

It can be shown that $\varepsilon = f(NTU, C_r)$. For a counter flow heat exchanger, the heat exchanger effectiveness can also be written as

$$\varepsilon = \frac{1 - \exp[-NTU(1-C_r)]}{1-(C_r)\exp[-NTU(1-C_r)]}$$  \hspace{1cm} (6.5)

where the heat capacity rate ratio, $C_r = C_{\text{min}}/C_{\text{max}}$, is equal to $C_c/C_h$ or $C_h/C_c$ depending on the relative magnitudes of the hot and the cold heat capacity rates.

### 6.3 Counter-flow arrangement of the HP-TEG experimental setup

An experimental arrangement of the counter flow HP-TEG was developed at the RMIT University, and it is illustrated in Figure 6.2. A schematic diagram of this experimental arrangement is shown in Figure 6.3. This arrangement had two separate air ducts; a) the hot air duct and b) the cold air duct. The length of the hot duct was 3.5 m with the cross-section area of 160 mm x 170 mm. A round PVC pipe of 100 mm length and 72 mm diameter was connected to the outlet of this duct. A Pitot tube was fitted in this pipe to measure air velocity. The cold air duct was of 1.5 m length. Similar to the hot air duct, a round PVC pipe was connected to the cold air outlet for measuring the air velocity. The fresh air was supplied by fan blowers at the entrance of each duct. The industrial waste heat was simulated using an electric heater which was placed before the evaporator section of the hot air duct. The ducts were thermally connected with eight rows of the HP-TEG modules.
6.4 Experimental procedure of the counter-flow HP-TEG

The fans at both duct entrances drove the fresh air into the system. The fan at the hot inlet duct was a direct current (DC) type, and its speed was controlled using a DC power supply. The fan at the entrance to the cold duct was an alternating-current (AC) centrifugal type. The flow rate of air pushed by this fan was manually controlled by adjusting the inlet damper at the fan intake. A Pitot tube, which was connected to a manometer, was used to measure the velocity of air in the system. The procedure to determine the air face velocity in the duct was discussed in Chapter 4. Once the desired air velocities were set, the electrical
heater was turned on to heat the fresh air flowing in the hot air duct. The temperature of the air was measured using T-type thermocouples connected to a data logger. Figure 6.3 shows the locations of the measured temperatures and they are marked as Tci (cold inlet), Tco (cold outlet), Thi (hot inlet) and Tho (hot outlet). At the marked locations, nine thermocouples measured the temperature variations of air over the duct cross section. The temperature readings were monitored using a computer which was linked to a data logger (Figure 6.2). The measurement of the maximum power point (MPP) of the HP-TEG at a steady state was conducted by applying an electronic load as explained in Chapter 4.

6.5 Results and discussion of the counter-flow HP-TEG

To obtain a higher temperature gradient and a higher heat transfer rate between the hot air and the cold air, the hot air inlet temperature should be maximized. On the other hand, the temperature of the cold air should be minimized. For this reason, the air velocity of the hot side, \( V_{\text{hot air}} \) was set at the lowest level of 1.1 m/s. The velocity of the hot air was maintained constant for the entire experiment. On the other hand, the air velocity at the cold side, \( V_{\text{cold air}} \) was varied between 0.9 m/s and 1.8 m/s. The effect of varying the cold air velocity, \( V_{\text{cold air}} \), was studied in this experiment.

The average cold air inlet temperature, \( T_{ci} \), was maintained at 23°C during the experiment. While the hot air inlet temperature, \( T_{ci} \) was maintained at 76°C. Figure 6.4 shows the variation of the cold air outlet temperature, \( T_{co} \) with the air face velocity of the cold side, \( V_{\text{cold air}} \). The graph shows that the cold air outlet temperature, \( T_{co} \) steadily decreases with increasing the air velocity on the cold side, \( V_{\text{cold air}} \). The highest \( T_{co} \) was at approximately 51°C when \( V_{\text{cold air}} = 0.9 \) m/s. \( T_{co} \) dropped gradually below 40 °C when the air velocity was increased to above 1.6 m/s. The theoretical analysis matched well with the experimental data with a deviation of less than 1.3%.
Figure 6.4 Cold air outlet temperature, $T_{co}$ versus the variation of air face velocity at the cold side, $V_{cold\ air}$.

As shown in Figure 6.5, the increase of air velocity on the cold side, $V_{cold\ air}$ did not significantly affect the hot air outlet temperature, $T_{ho}$. $T_{ho}$ decreased by less than 3°C within the air velocity range. A possible explanation could be that the air velocity at the hot side, $V_{hot\ air}$, was set constant at 1.1 m/s during the entire experiment. Consequently the convection effect on the hot side was minimal and caused $T_{ho}$ to be maintained at 45°C. The theoretical $T_{ho}$ also agreed well with the experimental data with a deviation of less than 1.5%.
Figure 6.5 Hot air outlet temperature, $T_{ho}$ versus the variation of air face velocity at the cold side, $V_{cold\ air}$.

To illustrate the thermal performance of the counter flow HP-TEG, Figure 6.6 shows the variation of overall heat exchanger effectiveness, $\varepsilon$ with the air face velocity at the cold side, $V_{cold\ air}$. It is observed that the heat exchanger effectiveness, $\varepsilon$ increased with increasing $V_{cold\ air}$. The theoretical results for the effectiveness were in good agreement with the experimental data. The experimental data deviated by 0.1-2.3% from the theoretical predictions. The effectiveness was calculated from the experimental data according to the following equations [90]:

$$
\varepsilon = \frac{C_h(T_{hi}-T_{ho})}{C_{min}(T_{hi}-T_{ci})} \quad \text{when } C_{min} = C_h
$$

(6.6)

or

$$
\varepsilon = \frac{C_c(T_{co}-T_{ci})}{C_{min}(T_{hi}-T_{ci})} \quad \text{when } C_{min} = C_c
$$

(6.7)
Figure 6.7 shows that the heat capacity rate of the hot air, $C_h$, was always lower than the heat capacity rate of the cold air, $C_c$, indicating that $C_{min} = C_h$. Therefore, Equation 6.6 was chosen for calculating the experimental effectiveness. It is also noted that from Figure 6.8, the temperature difference of air in the hot side, $\Delta T_{hot} = T_{hi} - T_{ho}$ increases with increasing air face velocity. Therefore, the increase of $\Delta T_{hot}$ explains why the effectiveness increased against $V_{cold \, air}$ as shown in Figure 6.6.

![Graph showing heat exchanger effectiveness](image)

Figure 6.6 Heat exchanger effectiveness of the HP-TEG, $\varepsilon$ versus the variation of air face velocity at the cold side, $V_{cold \, air}$. 
Figure 6.7 Heat capacity rate ($C_c$ or $C_h$) versus the variation of air face velocity at the cold side, $V_{cold\ air}$.

Figure 6.8 Temperature difference in the cold and hot side, $\Delta T$ versus the variation of air face velocity at the cold side, $V_{cold\ air}$.
Figure 6.9 presents a comparison between the theoretical predictions and the experimental data for the heat transfer rate of the counter-flow HP-TEG. The theoretical prediction of the heat transfer rate rose steadily with increasing air face velocity on the cold side, $V_{\text{cold air}}$. It is apparent from the graph that the theoretical model was able to closely forecast the thermal behaviour of the system for the airspeed range between 0.9 m/s - 1.6 m/s with a deviation less than 3%. The deviation became larger (about 4.0 - 4.5%) when $V_{\text{cold air}}$ exceeded 1.6 m/s. After this point, the experimental heat transfer rate remained close to constant at approximately 900 W. One of the possible reasons for this behaviour was the contraction at the outlet of cold air duct. The contraction occurred because the outlet of the duct was connected to a 72 mm diameter round pipe. At a higher air velocity, the pressure loss and the back pressure due to the contraction became more dominant and influenced the heat transfer rate. For the chosen air velocity range, it was difficult to measure the air flow in the main rectangular duct because of the low observed pressure drop. Therefore, the measurement was performed in the smaller round pipe and the air flow contraction was unavoidable.

![Graph showing heat transfer rate versus air face velocity](image-url)

**Figure 6.9** Heat transfer rate, $\dot{Q}_{\text{TEG}}$ versus the variation of air face velocity at the cold side, $V_{\text{cold air}}$. 
In Figure 6.10, the maximum power point (MPP) generated by the HP-TEG was plotted against $V_{\text{cold air}}$. It was found that the MPP increased with increasing air face velocity. The theoretical MPP was slightly over predicted compared to the experimental data with a deviation less than 26%. As shown previously in Figure 6.9, the higher heat transfer rate was achieved with rising $V_{\text{cold air}}$. The higher heat transfer rate resulted in higher TEG hot side temperature. Conversely, the temperature of the TEG cold side decreased with increasing $V_{\text{cold air}}$ as shown in Figure 6.8. As the result, the temperature gradient across the TEG increased and caused the MPP to increase.

![Figure 6.10 Maximum power point (MPP) of the HP-TEG, $P_{\text{elec max}}$ versus the variation of air face velocity at the cold side, $V_{\text{cold air}}$.](image)

6.6 Pressure drop across the counter-flow HP-TEG system.

Figure 6.11 shows the pressure drop associated with the air flow across the finned tube banks. The pressure drop increased with increasing air face velocity at the cold side, $V_{\text{cold air}}$. Within the velocity range, the pressure drop developed by the air flow was relatively small and less than 20 Pa. Figure 6.12 shows that the fan power associated with pressure drop was minimal, being less than 1.6 W. Therefore, the pressure drop associated
with the air flow in the HP-TEG system could be neglected as a marginal effect. In the same graph, it is shown that the TEG power output was always higher than the fan power requirement. This indicates that the electricity generated by the TEG could provide additional power to an existing fan system in the case of power shortage. Because of the negligible pressure drop, it was demonstrated that the HP-TEG system is a passive system without need for additional auxiliary forces.

Figure 6.11 The pressure drop across the finned tube banks (cold side).
Figure 6.12 The TEG power output compared to the fan power requirement associated with the pressure drop.
7 Case Study and Economical Evaluation of the Heat Pipe Thermoelectric Generator (HP-TEG) System

7.1 Case study: Bakery heat recovery system

Akbarzadeh et al [102] designed, constructed and tested a heat pipe heat recovery (HPHE) system for a bakery factory in Melbourne. In this study, the recovered waste heat from the bakery main oven was used to heat a proofing oven. Previously, the bakery used a gas fired steam generator to supply heat to the proofing oven. A steam coil heat exchanger was placed at the base of the proofing oven. The boiler injected steam to the proofing oven to maintain its humidity. In the new heat recovery system, waste heat from a chimney of the main baking oven was transferred to the proofing oven using a gravity assisted wickless heat pipe heat exchanger as illustrated in Figure 7.1. The flue gas temperature from the main oven was 270°C. The distance between the main baking oven and the proofing oven was 11 m. The heat exchanged was used to maintain the temperature of the proofing oven at 40°C and the humidity at 95%.

![Figure 7.1 Bakery heat recovery system using gravity assisted thermosyphon heat pipe][103].
The measured data from the bakery heat recovery system are summarized in Table 7.1.

Table 7.1 Actual measured flue gas and fresh air data from the bakery heat recovery system [103].

<table>
<thead>
<tr>
<th>Data of Flue Gas Section</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Temperature (°C)</td>
<td>265</td>
</tr>
<tr>
<td>Face Area of the Flue Gas Duct (m²)</td>
<td>0.342</td>
</tr>
<tr>
<td>Density of Flue (kg/m³)</td>
<td>0.786</td>
</tr>
<tr>
<td>Specific heat of Flue (kJ/kg.K)</td>
<td>1.051</td>
</tr>
<tr>
<td>Air Face Velocity (m/s)</td>
<td>1.51</td>
</tr>
<tr>
<td>Mass Flow rate of Flue (kg/s)</td>
<td>0.405</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Data of Fresh Air Section</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Temperature (°C)</td>
<td>38</td>
</tr>
<tr>
<td>Face Area of the Fresh Air Duct (m²)</td>
<td>0.342</td>
</tr>
<tr>
<td>Density of the Fresh Air (kg/m³)</td>
<td>1.040</td>
</tr>
<tr>
<td>Specific heat of the Fresh Air (kJ/kg.K)</td>
<td>1.009</td>
</tr>
<tr>
<td>Air Face Velocity (m/s)</td>
<td>1.55</td>
</tr>
<tr>
<td>Mass Flow rate of Fresh Air (kg/s)</td>
<td>0.551</td>
</tr>
</tbody>
</table>

7.2 Prediction of the HP-TEG system performance in a bakery using actual waste heat data.

The actual heat recovery data obtained from the bakery in Table 7.1 became input parameters for the validated computer model (Chapter 6). The proposed heat recovery system integrated with the HP-TEG is illustrated in Figure 7.2. A HP-TEG heat exchanger was located between a bypass flue gas duct and an air circulation duct of a proofing oven. In this simulation, the inlet temperature and the air face velocity of the flue gas were set at 265°C and 1.51 m/s, respectively. The face area of the evaporator side followed the real duct size of 0.342 m². The air face velocity and the temperature of the fresh air flowed from the proofing oven were set at 1.55 m/s and 38°C, respectively. The face area of the condenser side was
treated as similar to the evaporator side. The air ducts were assumed to be thermally connected using the HP-TEG modules.

![Diagram](image)

Figure 7.2 The proposed heat recovery system using the HP-TEG in the bakery.

The specifications of the heat pipe and the TEGs used in this case study were different from the computer model described in previous chapter in order to match the bakery waste heat conditions. A 21 mm diameter heat pipe was chosen to fit the larger air duct size of 0.342 m². The maximum number of heat pipes installed in the duct could not exceed 18 units per row of the HP-TEG module because of the limit set by the duct width. A manufacturer pre-tested TEG was selected to estimate the real potential of power generation from the
system. This TEG had higher conversion efficiency than the TEG tested and described in the previous chapter. The specifications of the heat pipes and TEG are listed in Table 7.2.

<table>
<thead>
<tr>
<th>Heat Pipe</th>
<th>TEG</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type: copper-water</td>
<td>Model type: TGM-127-1.4-1.5</td>
</tr>
<tr>
<td>Diameter: 21 mm</td>
<td>Dimension: 40 x 40 mm</td>
</tr>
<tr>
<td>Length: 5.5 m</td>
<td>Height: 3.9 mm</td>
</tr>
<tr>
<td>Number of fins: 100</td>
<td>Thermal conductivity: 2.98 W/mK</td>
</tr>
<tr>
<td>Fin material: aluminium</td>
<td>Electrical resistance: 2.0 Ω</td>
</tr>
<tr>
<td>Fins thickness: 0.5 mm</td>
<td>Thermal resistance: 1.7 K/W</td>
</tr>
<tr>
<td>Fin length: 100 mm</td>
<td></td>
</tr>
<tr>
<td>Fin width: 585 mm</td>
<td></td>
</tr>
</tbody>
</table>

The user parameters of the TEG were provided by the manufacturer. These parameters are shown in Table 7.3 which was used for calculating the TEG conversion efficiency. All the parameters are given for a hot junction temperature of 150°C.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Temperature of cold side, $T_c$ (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage, U (V)</td>
<td>0</td>
</tr>
<tr>
<td>Current, I (A)</td>
<td>3.8</td>
</tr>
<tr>
<td>Power, P (W)</td>
<td>1.9</td>
</tr>
<tr>
<td>Efficiency, %</td>
<td>7.2</td>
</tr>
<tr>
<td></td>
<td>5.8</td>
</tr>
</tbody>
</table>

Using the parameters listed in Table 7.3, the TEG conversion efficiency can be written as a function of the temperature difference between the TEG hot side and the TEG cold side. The conversion efficiency is described as follows:
\[ \mu_{TEG} = 0.0004(\Delta T - 100) + 0.038 \]  

(7.1)

A sensitivity analysis was conducted to identify critical parameters that influencing the thermal and electrical performance of the bakery HP-TEG system. It was found that several parameters including the number of rows of the HP-TEG module, number of heat pipes per row and number of TEG per row needed to be optimised in order to obtain the best performance of the system.

Figure 7.3 shows that the TEG power output peaked at approximately 900 W when 5 rows of HP-TEG module were chosen. For this case, a row of the HP-TEG module contained 300 units of TEG and 18 lengths of heat pipe (where both evaporator and condenser sections of the HP-TEG module had 9 units of heat pipes). The power output decreased when the number of rows was increased to more than 7 rows as the Carnot efficiency of the TEG kept dropping. The Carnot efficiency dropped because the TEG temperature gradient decreased with increasing the number of HP-TEG rows (Figure 7.4).

![Figure 7.3 Output power and Carnot Efficiency versus number of the HP-TEG row.](image-url)
Figure 7.4 TEG temperature difference against number of the HP-TEG row.

Figure 7.5 indicates the thermal and electrical output versus the number of HP-TEG rows. The optimum point of the thermal and electrical output was achieved when 5 rows of HP-TEG were selected. At this point, the system was capable of recovering approximately 47 kW of waste heat from the flue gas and producing 890 W electric power. In total, for 5 rows of HP-TEG to be used, the system needed approximately 90 heat pipes and 1500 TEG elements.
Figure 7.5 The optimal thermal and electrical performances of the HP-TEG.

Figure 7.6 The predicted air temperatures in the HP-TEG-bakery system.
Figure 7.6 shows the predicted inlet and outlet temperatures of the condenser and evaporator of the HP-TEG (Figure 7.1). At the optimum condition, the temperature of the fresh air at 38°C from the proofing oven rose to 116°C before leaving the condenser section. The condenser outlet temperature, $T_{co}$ obtained was sufficiently high to provide the heating capacity required to maintain the temperature of the proofing oven at 40°C.

7.3 Costs benefit analysis of the HP-TEG system in the bakery.

A cost benefits analysis was conducted to evaluate the feasibility of installing the HP-TEG system in the bakery. The analysis was started by calculating the operating cost of the proofing oven without the heat recovery system. The bakery was operated for a minimum of 20 hours a day and for a total of 7280 hours per year. Natural gas was used as the fuel for heating the proofing oven. The consumption of natural gas for the bakery oven was approximately 0.4012 GJ/hour [103]. For calculating the fuel consumption, the latest market price of natural gas was taken as AUD$ 30.38/GJ (Appendix F). From this information, Table 7.4 shows the annual fuel cost for heating the proofing oven to be estimated at AUD$ 88730/year.

The next step in the analysis was to estimate the investment cost for installing the HP-TEG. The primary investment cost was allocated to purchasing the main equipment of the system including the heat pipes and the TEG. For simplicity, the cost of heat pipe was considered as the raw cost of copper pipe per meter length. For this case, the price of copper pipe per meter was taken as approximately AUD$ 25 (Appendix G). The heat pipe length used in this case study was 5.5 m. For 5 rows of the HP-TEG modules, 90 lengths of heat pipe were needed costing approximately AUD12251.

The total number of TEGs needed for the bakery HP-TEG system was 1500 units which each unit costed at approximately AUD$5.5. The total cost of acquiring the TEGs was approximately AUD$ 8250. The other important investment costs were the cost for installing the system air ducts including their material, fittings and insulation. The installation and labour cost were assumed to be approximately 30% of the main equipment total cost.

The next economic study was to predict the cost saving of the heat recovered from the flue gas. The 47 kW recovered heat was compared to the annual cost of natural gas consumed
by the proofing oven. It was found that approximately AUD$ 37142/year of natural gas consumption could be saved if the bakery was fitted with the HP-TEG system.

Moreover, this economic analysis has also revealed the profit generated by the TEG output power. The total amount of electricity production was predicted at approximately 6503 kWh/year. At an electricity price of 0.2816 AUD$/kWh (Appendix H), the annual value of electricity generation was approximately AUD$ 1831. The combined annual profit of thermal and electrical power was approximately AUD$ 38973. This combined profit showed that approximately 43% of the annual operating cost of the proofing oven could be cut if the system was installed.

A simple payback period method was used to estimate the return duration of the total investment cost of the bakery HP-TEG system. The payback period can be calculated as follows:

\[
\text{Payback period (years)} = \frac{\text{Total investment cost}}{\text{Annual profit}}
\]  

(7.2)

This analysis found that the bakery might start to gain profit from its investment after less than a year of operation, or more specifically approximately 0.84 years. Table 7.4 summarizes all the steps involved in carrying out the cost benefits analysis of the HP-TEG system in the bakery.
Table 7.4 Costs benefit analysis of the HP-TEG system in the bakery.

<table>
<thead>
<tr>
<th>Annual Running Cost of the Proofing Oven</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Bakery annual operating hours (20 hours x 7 days x 52 weeks)</td>
<td>7280 hours/year</td>
</tr>
<tr>
<td>Natural gas consumption for the heating of proofing oven</td>
<td>0.4012 GJ/hours</td>
</tr>
<tr>
<td>Yearly natural gas consumption for heating proofing oven</td>
<td>2921 GJ</td>
</tr>
<tr>
<td>Cost of natural gas (AUD)</td>
<td>30.83 AUD $/GJ</td>
</tr>
<tr>
<td>The annual fuel cost for heating the proofing oven</td>
<td>88730 AUD $</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Investment Cost [Co]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Pipe</td>
</tr>
<tr>
<td>Cost of copper pipe per meter</td>
</tr>
<tr>
<td>Length of heat pipe</td>
</tr>
<tr>
<td>Cost per heat pipe</td>
</tr>
<tr>
<td>Number of HP-TEG row</td>
</tr>
<tr>
<td>Number of heat pipe/row</td>
</tr>
<tr>
<td>Total number of heat pipe needed</td>
</tr>
<tr>
<td>Total cost of heat pipe</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>TEG</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cost of the TEG per pieces</td>
</tr>
<tr>
<td>Number of HP-TEG row</td>
</tr>
<tr>
<td>Number of TEG/row</td>
</tr>
<tr>
<td>Total number of TEG needed</td>
</tr>
<tr>
<td>Total cost of TEG</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Air Ducts</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material, fittings and insulation (30% of capital cost)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Labour</th>
</tr>
</thead>
<tbody>
<tr>
<td>Installation (30% of capital cost)</td>
</tr>
<tr>
<td>Total Investment Cost</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Saving Cost from Heat Recovery</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Recovered</td>
</tr>
<tr>
<td>Bakery Operating Hours</td>
</tr>
<tr>
<td>The annual heat recovered</td>
</tr>
<tr>
<td>Description</td>
</tr>
<tr>
<td>---------------------------------------------------------------</td>
</tr>
<tr>
<td>The annual heat recovered</td>
</tr>
<tr>
<td>Cost of natural gas (AUD)</td>
</tr>
<tr>
<td>The annual fuel saving</td>
</tr>
<tr>
<td><strong>Profit from Power Generation</strong></td>
</tr>
<tr>
<td>Power generated</td>
</tr>
<tr>
<td>Bakery Operating Hours</td>
</tr>
<tr>
<td>The annual power generated</td>
</tr>
<tr>
<td>Cost of electricity</td>
</tr>
<tr>
<td>The annual electricity saving</td>
</tr>
<tr>
<td><strong>Annual Profit [B]</strong></td>
</tr>
<tr>
<td>Heat recovery + Power generated</td>
</tr>
<tr>
<td><strong>Simple Pay Back Period [Co/B]</strong></td>
</tr>
</tbody>
</table>
8 Conclusions and Recommendations for Future Work

8.1 Conclusions

This doctoral study explores a new method of recovering waste heat and electricity using a combination of heat pipes and thermoelectric generators (HP-TEG). The HP-TEG system consists of Bismuth Telluride (Bi$_2$Te$_3$) based thermoelectric generators (TEGs), which are sandwiched between two finned heat pipes to achieve a temperature gradient across the TEG for thermoelectricity generation. This system is special because it can simultaneously recover waste heat and generate electricity using an entirely passive method without using any moving parts.

A detailed theoretical model was developed to provide a preliminary performance estimate for such a system before commencing more detailed laboratory work. The theoretical model was derived using the energy balance equation and the thermal resistance method.

An experiment was conducted to determine the thermal and electrical characteristics of the thermoelectric generator (TEG) used. It was found from the experiment that the thermal resistance of the TEG varied with the heat input. The TEG thermal resistance showed a small change (standard deviation of 0.03°C/W) with the increasing heat input over the heating wattage range of 120 W. For simplicity, an average value of 0.8 °C/W was considered in this study. It was also noticed that the TEG thermal-to-electric power conversion efficiency increased with increasing temperature gradient across the TEG. The parameters gathered from this experiment were used in the theoretical model to determine the optimum settings of the prospective full scale HPTEG system.

The simulation results of the theoretical model have shown that the thermal and electrical performance of the HP-TEG system decreased with increasing mass flow. This was because the temperature gradient between the hot and the cold side of the HP-TEG reduced when the air speed increased. The simulation results suggested that the optimum mass flow rate that should be chosen is approximately 0.03 kg/s (or air face velocity of 0.9 m/s) for producing the maximum thermal and electrical performance of the system.

It was noted from the simulation result that reducing the mass flow rate less than 0.03 kg/s could cause the temperature of hot inlet of the HP-TEG to increase over the TEG
temperature limit of 125°C. In addition, an increase in the number of heat pipe row installed to the system could increase the amount of heat transfer rate and the electrical power output. Based on these results, it was suggested that the prospective full scale HP-TEG system should be installed with 8 rows of the HP-TEG modules. Under these conditions, the theoretical model predicted that the HP-TEG system to potentially generate approximately 10 W of electrical power and the rate of heat recovery would be 1.6 kW using 2 kW of heating power input.

A U-type air duct system integrated with the HP-TEG modules was designed and built in the RMIT University, laboratory. The experimental data obtained from this testing was used to validate the theoretical model. The basic concept of the experimental rig consisted of a TEG sandwiched between two heat pipes which they act as an evaporator (heat pipe 1) and a condenser (heat pipe 2). The TEG surfaces were heated and cooled by the evaporator and the condenser, respectively. These processes created a temperature gradient across the TEG and generate electricity. The system configuration was expected to increase the recovery ratio of waste heat because the heat released from the condenser preheats the incoming air and hence increases the temperature of air flow over the evaporator. In the actual experimental bench, eight rows of the HP-TEG modules were installed between these ducts. The modules were arranged in series to the direction of air flow. A 55 mm gap separated each module.

In the experiments, the air face velocity was varied between 1.1 m/s and 1.6 m/s using a fan blower at the system inlet. Due to the high temperature rise over the TEG temperature limit, the lowest air face velocity was limited to 1.1 m/s. The temperature limit of the TEG was set by the manufacturer at 125°C. Over this limit, the solder material of the TEG could be damaged with possible loss of performance.

It was found from the experimental results that the heat transfer rate of the HP-TEG system reduced with an increase of air face velocity. The reduction of the heat transfer rate occurred because of a drop of the temperature gradient between the hot and the cold ducts. The highest heat transfer rate measured was 1079 W at an air face velocity of 1.1 m/s. It was found that the theoretical predictions for the heat transfer rate were slightly higher than the experimental data, being within 10% deviation. This deviation was attributed to the heat loss though the system wall in the actual experiments. For the theoretical model, heat loss did not occur since the wall was considered adiabatic.
The heat recovery performance of the HP-TEG system was assessed by the heat exchanger effectiveness. It was found that the effectiveness of the system improved from 35% to 41% when the air flow was decreased to the minimum velocity. At a low air face velocity, the temperature difference between the hot and the cold air increased, contributing to the increase of the system effectiveness. It was found that the theoretical and the experimental results of the effectiveness deviated by less than 5%.

Similar to the rate of heat transfer case, the power output increased when the air face velocity was reduced to the lowest air blower setting. The maximum electrical power generated was approximately 7 W. It was found that the theoretical power output was in acceptable agreement with the experimental data with an average deviation of 13%.

Experiments on the proposed system have demonstrated that the use of fully passive devices (heat pipes and TEGs) for heat recovery and power generation is an efficient method for reusing industrial waste energy.

In industrial applications, the heat exchanger is commonly installed in the counter-flow arrangement. This is because it offers many advantages compare to the parallel flow heat configuration.

A counter flow configuration of a HP-TEG was also developed to replicate the practical situation of the heat exchanger in industry. The counter flow heat exchanger of the HP-TEG has two separate air ducts. These two ducts were thermally connected using the HP-TEG modules. The condenser section of the HP-TEG modules was installed in the cold duct which carried cold air from the surroundings. The evaporator section of the HP-TEG was placed in the hot duct which carried hot air. Air blowers supplied fresh air into the duct and were installed at the entrance of each duct. The ambient air entering the hot duct was heated by a 2 kW electrical heater before flowing through the evaporator section of the HP-TEG.

A theoretical model was developed to predict the performance of this counter-flow system. This model used the effectiveness-NTU method to estimate the outlet temperatures of air when the air inlet temperature was known. This model also was able to predict the thermal and electrical performance output of the HP-TEG including the rate of heat transfer, the effectiveness of the heat exchanger and the power output.

To assess the system, the effect of changing the air face velocity of the cold side, $V_{\text{cold air}}$ was studied in this experiment. $V_{\text{cold air}}$ was varied between 0.9 m/s and 1.8
m/s. In contrast, the air face velocity of the hot side, $V_{hot\,air}$ was minimized at 1.1 m/s and was maintained at that level for the entire experiment.

The effectiveness of the counter flow HP-TEG increased with increasing $V_{cold\,air}$. The effectiveness increased because of the increase of the temperature difference of air at the hot side. The theoretical effectiveness agreed well with the experimental data with a deviation of less than 2.3%. The heat recovery rate of the counter flow HP-TEG also rose with increasing $V_{cold\,air}$. The model prediction of the heat recovery rate agreed well with the experimental data. The difference was less than 4.5%. The maximum power output was found to rise from 3 W to 4.3 W with increasing $V_{cold\,air}$ because of an increase of heat transfer rate through the HP-TEG module. However, the theoretical model predicted a higher power output than measured experimentally with a deviation less than 26%. The experimental power output was lower than the theoretical result because of the heat loss which occurred during the heat transfer process. The heat loss reduced the hot side temperature of the TEG surfaces and decreased the thermal-to-electric conversion efficiency.

It can be concluded that the effectiveness of the counter flow HP-TEG heat exchanger and its power performance increased with increasing air velocity which opposite with the U-type HP-TEG heat exchanger. The U-type HP-TEG heat exchanger performance was better when operating at low air speed.

A case study was conducted using actual data for waste heat from a bakery oven. The flue gas at 270°C from the main oven was recovered for use in a proofing process. The recovered energy heated incoming fresh air to 38°C before it entered the proofing oven. The study was conducted by applying the actual data to the validated theoretical model of the HP-TEG.

The performance of the HP-TEG system in the bakery was optimized by varying several input variables. The optimum configuration of the input parameters including the number of heat pipes, the number of heat pipe/rows and the number of thermoelectric units/row was investigated. The bakery heat recovery system was found to provide optimum performance when it used 5 rows of HP-TEG modules. Each row of the HP-TEG consisted of 9 heat pipes at both condenser and evaporation sections. In total, 90 heat pipes of 21 mm diameter and 5.5 m length were needed for the system. Approximately 1500 TEG units
should be installed for the entire system. Using this configuration, the system could produce 47 kW of thermal power and 0.9 kW of electric power.

The cost benefits analysis revealed approximately AUD$ 37142/ year of fuel could be saved by the thermal power recovery. This could save approximately 43% of the natural gas usage for heating the proofing oven. The annual profit obtained from TEG power generation was approximately AUD$ 1831. By considering the investment cost including the price of material, installation and the labour to install the HP-TEG in the bakery, it was found that the bakery factory could achieve profit in less than a year of operation.

8.2 Future works

A model simulation using Computational Fluid Dynamic (CFD) should be used in the future to study detail air flow characteristic of the HP-TEG system. The effect of various flows speed could easily manipulated using the CFD. Using the CFD, the effect of different heat pipe and fins configurations could be further studied.

It is recommended for the future research to develop a better HP-TEG system using a high temperature TEG and high ZT materials. For this case, the hot side temperature of the TEG could go beyond 150°C as it is known that the TEG conversion efficiency might increase for higher surface temperature. The similar system also should be tested in a real industrial site to study the long term effect on the thermal stress of the TEG and heat pipes.
Appendix A

References

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Appendix B

Uncertainty analysis

The relative uncertainties related to the equipment used were calculated were based on Kline [104]:

Relative uncertainty = \frac{0.5 \times \text{resolution}}{\text{measuring value}} \quad (B1)

Three repetitions \((N)\) were conducted for all the measurements involved in the HP-TEG experiments. The formulas used for quantifying the set of data, the mean value \(\bar{x}\) and standard deviation \(\sigma\) are shown in Equations B2 and Equation B3 respectively [105]:

\[
\bar{x} = \frac{\sum_{i=1}^{N} x_i}{N} \quad (B2)
\]

\[
\sigma = \sqrt{\frac{1}{N-1} \sum_{i=1}^{N} (x_i - \bar{x})^2} \quad (B3)
\]

According to [104], if \(R\) is a function of the independent variables of \(x_1, x_2, x_3, ..., x_n\), thus

\(R = R(x_1, x_2, x_3, ..., x_n) \quad (B4)\)

The general form of the uncertainty of \(R\) can be represented as follows:

\[
\omega_R = \left[ \left( \frac{\partial R}{\partial x_1} \omega_1 \right)^2 + \left( \frac{\partial R}{\partial x_2} \omega_2 \right)^2 + \cdots + \left( \frac{\partial R}{\partial x_n} \omega_n \right)^2 \right]^{1/2} \quad (B5)
\]

\(R = x_1^{a_1} x_2^{a_2} ... x_n^{a_n} \quad (B6)\)

Taking the partial derivative of the Equation B6, it can be shown that

\[
\frac{\partial R}{\partial x_i} = x_1^{a_1} x_2^{a_2} ... (a_i x_i^{a_i-1}) ... x_n^{a_n} \quad (B7)
\]

When equation B7 is divided by equation B6, it can be shown that

\[
\frac{1}{R} \frac{\partial R}{\partial x_i} = \frac{a_i}{x_i} \quad (B8)
\]

Inserting equation B8 into equation B5 gives
\[
\frac{\omega_R}{R} = \left[ \sum \left( \frac{\omega x_i}{x_1} \right)^2 \right]^{1/2}
\] (B9)

Equation B9 is the general form to find the uncertainties for product functions.

**Uncertainty Measurement for Air Face Velocity**

The air velocity was measured using a Pitot tube that inserted in a 75 mm diameter pipe as explained in Chapter 4. The visual inspection was used to take the measurement on the manometer gauges such as the velocity pressure, \( P_v \) and Barometric pressure, \( B \). The temperature of air, \( T_{air} \) was measured using T-type thermocouple that connected to a data logger. The air velocity in the pipe was calculated using a formula supplied by the manometer supplier:

\[
V_{pipe} = 4.05 \left( \frac{760}{B} \times \frac{T}{293} \times P_v \right)^{1/2} = 6.52 \left( \frac{T P_v}{B} \right)^{1/2}
\] (B10)

where

\( V \) is velocity (m/s).

\( T \) is absolute temperature K* (airstream temperature).

\( B \) is barometric pressure mmHg.

\( P_v \) is velocity pressure in mmH₂O.

The measuring value and the resolution of the equipment for measuring air velocity are listed in Table B1. The relative uncertainty that associated with equipment is not larger than 3% as shown in the Table B1.
Table B1. The list of equipment uncertainty for measuring air velocity.

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Resolution</th>
<th>Measuring range</th>
<th>Relative uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure gauge (Barometric pressure measurement)</td>
<td>2 mmHg</td>
<td>752-754 mmHg</td>
<td>0.13%</td>
</tr>
<tr>
<td>T-Type thermocouple (Temperature measurement)</td>
<td>0.1°C</td>
<td>50-63 °C</td>
<td>0.08-0.1%</td>
</tr>
<tr>
<td>Manometer gauge (Velocity pressure measurement)</td>
<td>0.2 mm</td>
<td>3.4-6.8 mmHg</td>
<td>1.5-2.9%</td>
</tr>
</tbody>
</table>

The uncertainty of air velocity is derived according to Equation B7.

\[
\frac{\partial v}{\partial T} = 0.5x6.52T^{-0.5}P_v^{0.5}B^{-0.5}
\]

(B11a)

\[
\frac{1}{V} \frac{\partial V}{\partial T} = 0.5xT^{-1}
\]

(B11b)

\[
\frac{\partial v}{\partial P_v} = 0.5x6.52T^{0.5}P_v^{-0.5}B^{-0.5}
\]

(B12a)

\[
\frac{1}{V} \frac{\partial V}{P_v} = 0.5xP_v^{-1}
\]

(B12b)

\[
\frac{\partial v}{\partial B} = -0.5x6.52T^{0.5}P_v^{0.5}B^{-1.5}
\]

(B13a)

\[
\frac{1}{V} \frac{\partial V}{B} = -0.5xB^{-1}
\]

(B13b)

Referring to Equation B9, the uncertainty of velocity can be determined as:

\[
\frac{\omega_V}{V} = \left[ \left( \frac{1}{V} \frac{\partial V}{\partial T} \omega_T \right)^2 + \left( \frac{1}{V} \frac{\partial V}{P_v} \omega_{P_v} \right)^2 + \left( \frac{1}{V} \frac{\partial V}{B} \omega_B \right)^2 \right]^{1/2}
\]

(B14)

\[
\frac{\omega_V}{V} = \left[ \left( \frac{1}{T} \omega_T \right)^2 + \left( \frac{1}{P_v} \omega_{P_v} \right)^2 + \left( \frac{1}{B} \omega_B \right)^2 \right]^{1/2}
\]

(B15)
The air velocity uncertainties are tabulated in Table B2-B4. The uncertainty analysis of the air velocities revealed that the errors were controlled to be less than 1% that ensured the reliability of the data measured.

Table B2. Experiment relative uncertainty of the air face velocity (U-type HP-TEG).

<table>
<thead>
<tr>
<th>Air Velocity, V (m/s)</th>
<th>Relative uncertainty of the air velocity, $\frac{\omega V}{V}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.60</td>
<td>0.108136094</td>
</tr>
<tr>
<td>1.50</td>
<td>0.13889198</td>
</tr>
<tr>
<td>1.48</td>
<td>0.148634959</td>
</tr>
<tr>
<td>1.44</td>
<td>0.159440708</td>
</tr>
<tr>
<td>1.34</td>
<td>0.217015322</td>
</tr>
<tr>
<td>1.13</td>
<td>0.432526419</td>
</tr>
</tbody>
</table>

Table B3. Experiment relative uncertainty of the cold air velocity (Counter Flow HP-TEG).

<table>
<thead>
<tr>
<th>Air Velocity, V (m/s)</th>
<th>Relative uncertainty of the air velocity, $\frac{\omega V}{V}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.84</td>
<td>0.061750674</td>
</tr>
<tr>
<td>1.79</td>
<td>0.070880465</td>
</tr>
<tr>
<td>1.72</td>
<td>0.082198214</td>
</tr>
<tr>
<td>1.61</td>
<td>0.102052302</td>
</tr>
<tr>
<td>1.34</td>
<td>0.217017955</td>
</tr>
<tr>
<td>0.92</td>
<td>1.03305832</td>
</tr>
</tbody>
</table>
Table B4. Experiment relative uncertainty of the hot air velocity (Counter Flow HP-TEG).

<table>
<thead>
<tr>
<th>Air Velocity, V (m/s)</th>
<th>Relative uncertainty of the air velocity, $\frac{\omega_V}{V}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.13</td>
<td>0.832987133</td>
</tr>
<tr>
<td>1.13</td>
<td>0.832987116</td>
</tr>
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<td>1.13</td>
<td>0.832987098</td>
</tr>
<tr>
<td>1.13</td>
<td>0.832987102</td>
</tr>
<tr>
<td>1.13</td>
<td>0.832987052</td>
</tr>
<tr>
<td>1.13</td>
<td>0.832986922</td>
</tr>
</tbody>
</table>

**Uncertainty Measurement for Heat Transfer Rate**

The heat transfer rate was calculated using Equation 5.1. The air face velocity, $V$ and the temperature difference of the cold air, $\Delta T_c$, were the important variables needed in determining the heat transfer rate. Steps below show the derivation to determine the measurement relative uncertainty of the heat transfer rate.

$$\dot{Q} = \rho A V c_p \Delta T_c$$  \hspace{1cm} (B16)

$$\dot{Q} = f (V, \Delta T_c)$$  \hspace{1cm} (B17)

$$\frac{\partial \dot{Q}}{\partial V} = \rho A c_p \Delta T_c$$  \hspace{1cm} (B18)

$$\frac{1}{\dot{Q}} \frac{\partial \dot{Q}}{\partial V} = 1/V$$  \hspace{1cm} (B19)

$$\frac{\partial \dot{Q}}{\partial \Delta T_c} = 1/\Delta T_c$$  \hspace{1cm} (B20)

$$\frac{\omega_{\dot{Q}}}{\dot{Q}} = \left[ \left( \frac{\partial \dot{Q}}{\partial V} \omega_V \right)^2 + \left( \frac{\partial \dot{Q}}{\partial \Delta T_c} \omega_{\Delta T_c} \right)^2 \right]^{1/2}$$  \hspace{1cm} (B21)

$$\frac{\omega_{\dot{Q}}}{\dot{Q}} = \left[ (\omega_V/V)^2 + (\omega_{\Delta T_c}/\Delta T_c)^2 \right]^{1/2}$$  \hspace{1cm} (B22)

Table B5 and B6 show the tabulated data of the uncertainty of the heat transfer rate. The maximum uncertainty of the heat transfer rate was identified less than 1% that proved the reliability of the data taken in the experiments.
Table B5. Experiment relative uncertainty of the heat transfer rate (U-type HP-TEG).

<table>
<thead>
<tr>
<th>Heat transfer rate, $\dot{Q}$ (W)</th>
<th>Relative uncertainty of the heat transfer rate, $\frac{\omega \dot{Q}}{\dot{Q}}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>835</td>
<td>0.109709109</td>
</tr>
<tr>
<td>840</td>
<td>0.139828713</td>
</tr>
<tr>
<td>891</td>
<td>0.149291079</td>
</tr>
<tr>
<td>945</td>
<td>0.159872401</td>
</tr>
<tr>
<td>954</td>
<td>0.217242739</td>
</tr>
<tr>
<td>1079</td>
<td>0.432562149</td>
</tr>
</tbody>
</table>

Table B6. Experiment relative uncertainty of the heat transfer rate (Counter Flow HP-TEG).

<table>
<thead>
<tr>
<th>Heat transfer rate, $\dot{Q}$ (W)</th>
<th>Relative uncertainty of the heat transfer rate, $\frac{\omega \dot{Q}}{\dot{Q}}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>902</td>
<td>0.065438965</td>
</tr>
<tr>
<td>894</td>
<td>0.073911948</td>
</tr>
<tr>
<td>893</td>
<td>0.084437979</td>
</tr>
<tr>
<td>904</td>
<td>0.103398698</td>
</tr>
<tr>
<td>905</td>
<td>0.217323908</td>
</tr>
<tr>
<td>800</td>
<td>1.033081836</td>
</tr>
</tbody>
</table>

Uncertainty Measurement for Maximum Power Point (MPP)

The MPP is the multiplication of the voltage, $U_{max}$ and current, $I_{max}$. Using similar method in the previous section, the experiment relative uncertainty of the MPP can be estimated as follows:

$$\frac{\omega P_{max}}{P_{max}} = \left[\left(\frac{\omega U_{max}}{U_{max}}\right)^2 + \left(\frac{\omega I_{max}}{I_{max}}\right)^2\right]^{1/2}$$  \hspace{1cm} (B23)

The measuring range and the resolution of the equipment for measuring current and voltage outputs are listed in Table B7. The relative uncertainty that associated with equipment for measuring the MPP is not larger than 0.25% as shown in the Table B7. Table
B8 and Table B9 display the relative uncertainty of the MPP was not larger than 1.3% that guaranteed the reliability of the variable measurements conducted during the experiments.

Table B7. List of equipment uncertainty.

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Resolution</th>
<th>Measuring range</th>
<th>Relative uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electronic load (current measurement)</td>
<td>1mA</td>
<td>0.19-0.26A</td>
<td>0.19-0.25%</td>
</tr>
<tr>
<td>Electronic load (Voltage measurement)</td>
<td>1mV</td>
<td>20.8-27.1V</td>
<td>0.0018-0.0024%</td>
</tr>
</tbody>
</table>

Table B8. Experiment relative uncertainty of the maximum power output (U-type HP-TEG).

<table>
<thead>
<tr>
<th>Maximum power output, $P_{\text{elec max}}$ (W)</th>
<th>Relative uncertainty of the maximum output power, $\frac{\omega_P}{P}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.13</td>
<td>1.271377649</td>
</tr>
<tr>
<td>4.30</td>
<td>1.239258153</td>
</tr>
<tr>
<td>4.61</td>
<td>1.211225747</td>
</tr>
<tr>
<td>4.96</td>
<td>1.033117056</td>
</tr>
<tr>
<td>5.35</td>
<td>1.072326158</td>
</tr>
<tr>
<td>7.06</td>
<td>0.739013494</td>
</tr>
</tbody>
</table>
Table B9. Experiment relative uncertainty of the maximum power output (Counter Flow HP-TEG).

<table>
<thead>
<tr>
<th>Maximum power output, $P_{\text{elec\ max}}$ (W)</th>
<th>Relative uncertainty of the maximum output power, $\frac{\omega P}{P}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.02</td>
<td>1.259616935</td>
</tr>
<tr>
<td>4.30</td>
<td>1.197679011</td>
</tr>
<tr>
<td>4.04</td>
<td>1.239710429</td>
</tr>
<tr>
<td>4.11</td>
<td>1.332911011</td>
</tr>
<tr>
<td>3.80</td>
<td>1.297839068</td>
</tr>
<tr>
<td>3.08</td>
<td>1.580321876</td>
</tr>
</tbody>
</table>

**Uncertainty measurement for TEG thermal resistance**

Thermal resistance of a TEG was derived from the ration of temperature difference between the hot and cold side of a TEG and the heat flowing through a TEG.

$$R_{th} = \frac{\Delta T_{TEG}}{Q_{in}} = \frac{\Delta T_{TEG}}{P_{elec\ in}} = \frac{\Delta T_{TEG}}{U_{in}I_{in}}$$  \hspace{1cm} (B24)

The TEG hot and the cold side temperatures were measured using the T-type thermocouple having a relative uncertainty of $\pm 0.1\%$. The thermal power (heat flow) through the TEG was varied using a voltage regulator. The current and voltage output of the regulator was measured using the ACV monitor. The relative uncertainty of the current and voltage output were $\pm 0.01\%$ and $\pm 0.0004\%$, respectively.

Relative uncertainty of thermal resistance of the TEG is given as follows:

$$\frac{\omega R_{th}}{R_{th}} = \left[ \left( \frac{\omega \Delta T_{TEG}}{\Delta T_{TEG}} \right)^2 + \left( \frac{\omega U}{U} \right)^2 + \left( \frac{\omega I}{I} \right)^2 \right]^{1/2}$$  \hspace{1cm} (B25)
Table B10. Experiment relative uncertainty of TEG thermal resistance.

<table>
<thead>
<tr>
<th>$R_{th}$ (K/W)</th>
<th>Relative uncertainty of thermal resistance of TEG, $\frac{\Delta R_{th}}{R_{th}}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.79</td>
<td>0.082407</td>
</tr>
<tr>
<td>0.81</td>
<td>0.041097</td>
</tr>
<tr>
<td>0.86</td>
<td>0.02649</td>
</tr>
<tr>
<td>0.85</td>
<td>0.020056</td>
</tr>
<tr>
<td>0.85</td>
<td>0.01617</td>
</tr>
<tr>
<td>0.79</td>
<td>0.013642</td>
</tr>
</tbody>
</table>
Appendix C
Appendix D

Unick Chemical Corp.

Silicon-based compound for thermo coupling of electronic components and heat sinks. High-heat conductive property allows heat to pass freely where heat dissipation and dispersion is important.

Temperature range: -200°C to 130°C
Thermal conductivity: 0.9 W/mK

- The Unick Silicone Heat Transfer Compound is a grease-like silicone material that is heavily impregnated with heat conductive metal oxides (zinc oxide). This combination of base and fillers produces a material that has a high thermal conductivity. Together with low bleed properties, the compound facilitates heat transfer from semiconductors to heat sinks by covering and smoothening pits and lands along the metal surfaces, thereby transferring heat uniformly and efficiently away from those surfaces and avoiding the development of "hotspots".

- Uses and application: The compound is applied to the base and mounting studs of transistors, power diodes, CPUs and/or any component where excess heat is required to be drawn away. Ideally, this application should be done in a thin and uniform film without any break across the surface before coupling to a suitable metal heat sink is made.
- One 10g tube will do up to 30 TO-3 package transistors
- Composition:
  - Dimethyl Polysiloxane (silicone) 28%
  - Zinc Oxide 65%
  - Submicroscopic Pyrogenic Silica 4%
  - Stabilizer 3%
- General Physical and Chemical Properties:
  - Colour: Opaque White
  - Odour: None
  - Consistency: Penetration, worked and measured within 1 minute after working (ASTM D-217) = 260
  - Bleed (after 24 hours @ 200°C): 1.0%
  - Evaporation (after 24 hours @ 200°C): 1.0%
  - Specific Gravity at 25°C: 2.3
  - Thermal Conductivity, K Factor, calcJm²°C/sec cm: 0.0015
  - Dielectric Strength, kV at 0.1mm, ASTM D-149: 1.0
  - Boiling Point: >250°C
  - Melting Point: 1,970°C
  - Flash Point: 230°C
  - Auto-ignition Temperature: 425°C
  - Vapour Pressure: >2hPa
  - Relative Density: 2.1 (@ 20°C)
Appendix E
Appendix F

FACT SHEET

Future Residential Gas Prices*

Gas prices to Tasmanian households are expected to increase moderately over the next four years. Price rises of $5 - $7 per GJ are expected, which is an increase of 5% - 6% per annum.

Over the next four years, two components are expected to increase:
1. Gas itself is expected to increase from $4 per GJ to $8 per GJ, or one dollar per year over the next four years.
2. Transportation charges for shipping gas from Victoria to Tasmania are also expected to rise.

The total price increase over four years is likely to be approximately $5 - $7 per GJ, or $1.25 to $1.75 per GJ per annum.

Gas is a viable and attractive energy option and remains competitive to electricity, LPG and fuel oils.

<table>
<thead>
<tr>
<th>Year</th>
<th>Price Per GJ</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electricity</td>
<td>$68.70</td>
</tr>
<tr>
<td>Average LPG</td>
<td>$54.25</td>
</tr>
<tr>
<td>2014 Hot Water/Heating</td>
<td>$41.40</td>
</tr>
<tr>
<td>Natural Gas (Standard Residential)</td>
<td>$30.38</td>
</tr>
<tr>
<td>Anticipated 2018 Natural Gas (Standard Residential)</td>
<td>$37.38</td>
</tr>
</tbody>
</table>

*All prices are expressed in current 2014 dollars.

For further information or to discuss your requirements please contact Tas Gas on 1800 438 427 or visit www.tasgas.com.au.
Appendix G
Appendix H

Business Prices

Synergy Business Plan (L1) tariff
The Synergy Business Plan® tariff is available to eligible customers where the electricity is used for business purposes at low/medium voltage (240/415V) and the electricity supplied to the customer’s premises is less than 50 megawatt hours per annum (or less than 137 units per day on average).

<table>
<thead>
<tr>
<th></th>
<th>Prices effective 1 July 2014 (incl. carbon)</th>
<th>Prices effective 1 Sept 2014 (excl. carbon)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Supply charge – cents per day</td>
<td>42.8472</td>
<td>42.8472</td>
</tr>
<tr>
<td>Electricity charge – cents per unit</td>
<td></td>
<td></td>
</tr>
<tr>
<td>First 1,650 units per day</td>
<td>30.5658</td>
<td>28.1603</td>
</tr>
<tr>
<td>More than 1,650 units per day</td>
<td>27.8157</td>
<td>25.4102</td>
</tr>
</tbody>
</table>

By law we calculate our prices to four decimal places. Electricity is charged by the ‘unit’. A “unit” is one kilowatt-hour (kWh).

Please see page 17 for other charges that may be applied to your account.
Further details are available by calling 13 13 54 or visiting synergy.net.au

Note: Recent changes to federal laws relating to carbon requires changes to your Synergy invoice. For the period 1 July to 31 August 2014 Synergy invoiced customers using prices inclusive of a carbon component. On and from 1 September 2014 Synergy will invoice customers using prices exclusive of a carbon component. Carbon charges for the period 1 July to 31 August 2014 will be reimbursed to customers in the form of a credit displayed on the first bill on or after 1 September 2014. These changes are subject to amendments to the tariff by-laws occurring on 1 September 2014.
Appendix I

Figure I1 Number of rows versus heat transfer rate