An Investigation into the Use of Solar Heat Energy to Drive an Absorption Cycle Refrigerator

Nicholas Craig
Executive Summary

The vapour compression cycle type refrigerator is the most commonly used in the refrigeration industry today. This is a work operated cycle which uses a compressor to drive the system. An alternative system is the hydrogen driven absorption cycle which is driven by the pressure differentials within the cycle. The only energy input for this system is the heat energy at the generator. The heat energy required to run an absorption cycle is greater than the energy required to operate a similar sized vapour compression system. However if readily available or waste heat can be utilized then refrigeration can be provided without the need to generate more energy. It has been calculated that the sun provides 1.3 kW/m² of energy which is largely unutilized as an energy source, (Kopp & Lean 2011).

This investigation has been conducted to determine a design for a prototype solar heat driven absorption cycle refrigerator. Tests were performed on an existing absorption refrigerator to evaluate the effectiveness of design changes at counteracting the disadvantages associated with using solar heat energy. These tests found that increasing the insulation of the refrigerator and including a thermal mass inside the refrigerated volume helped keep the refrigerator temperature constant when cooling wasn’t being applied.

To collect and deliver the solar heat energy to the generator of the cycle an investigation and calculations were conducted to determine the best system. The investigation found that a design including an evacuated tube collector to harvest the heat and a copper –water heat pipe to deliver the heat to the generator would be a viable system. Further investigation is required to optimise the reservoir required to transfer heat from the solar collector to the heat pipe.

Acknowledgements:

I would like to thank Justin Hyde for his assistance as my academic supervisor throughout the course of this investigation. I would also like to thank Pat O’Grady for his assistance in sourcing the equipment I needed for my experiments and solving issues we encountered with the equipment. I would also like to thank Fons Nouwens for the loan of his absorption refrigerator for the experiments and his assistance in the development of the experimental process.

Declaration

I certify that the main text of this thesis is entirely my own work and that such work has not been previously submitted as a requirement for the award of a degree at Central Queensland University (Australia) or any other institution of higher education.

Nicholas Craig s0154902
31/05/2012
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**Abbreviations**

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>COP</td>
<td>Coefficient of performance</td>
</tr>
<tr>
<td>W</td>
<td>Watts</td>
</tr>
<tr>
<td>K</td>
<td>Kelvin</td>
</tr>
<tr>
<td>C</td>
<td>Celsius</td>
</tr>
<tr>
<td>m</td>
<td>Metres</td>
</tr>
<tr>
<td>PVC</td>
<td>Polyvinyl chloride</td>
</tr>
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<td>L</td>
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1.0 Introduction

In modern society there is an increasing emphasis on the environmental impact of human energy consumption. This along with mankind’s increasing need for energy creates a distinct need for the development of more environmentally friendly technologies. One such technology that could be made more efficient is refrigeration. The aim of this investigation was to determine the design requirements for a prototype refrigeration system that can be operated using a solar heat source.

The refrigeration industry standard is to use a vapour compression cycle to provide refrigeration; this system requires energy input to drive the compressor which runs the cycle. An alternative system is hydrogen driven absorption cycle refrigeration; in this system if enough heat energy can be provided at the generator no extra energy is needed to run the system. The hydrogen driven absorption refrigeration cycle is driven by pressure differences within the system, so the only energy input required is the heat energy at the generator, (Moran & Sharpiro 2008). This means that if a freely available heat source is used the system can be run without any extra energy input.

The sun has the potential to provide vast amounts of energy if it can be tapped. It is calculated that the sun delivers approximately 1.3 kW/m² of usable energy, (Kopp & Lean 2011). If sufficient heat energy can be harvested from the sun it could potentially be used to provide the drive for an absorption cycle refrigerator, thus provide a refrigeration system run on a renewable heat source. Such a system would also have benefits in isolated areas where electricity is not available.

This report explores this potential and calculates the design properties needed for the development of a prototype system for further testing. To do this testing was done on an existing absorption cycle refrigerator to evaluate the effectiveness of different design changes at improving the refrigerator performance during times of reduced cooling. Calculations were also conducted to quantify the benefits of the changes and to design and size the solar heat collection and delivery system required.
2.0 Literature Review

2.1 Refrigeration Systems

The vapour compression cycle is the most commonly used cycle in refrigeration applications. The most common refrigerant used in these cycles is refrigerant R143a. The vapour compression cycle uses a compressor to increase the pressure and temperature of the refrigerant vapour. This causes the vapour to move to the condenser. In the condenser heat energy is removed from the vapour to turn the refrigerant into a liquid. The liquid is then passed through an expansion valve which lowers the pressure and allows the refrigerant to vaporise. The lower pressure vaporisation draws heat from the refrigerated area. From the evaporator the low pressure vapour is passed back to the compressor where the cycle starts again, (Figure 1). The vapour compression cycle is the commonly used because it is reliable and cost effective to build. However it requires a significant amount of energy to drive the compressor, (Cengel & Boles 2005).

Figure 1: Vapour Compression Cycle, (Cengel & Boles 2005)

The absorption cycle is one of the earlier types of refrigeration cycles first developed in 1860, (Kharagpur n.d.). Unlike the vapour compression type, this system uses a heat source and a pump to drive the system. The five main components are the evaporator, absorber, generator, condenser and pump. In an ammonia water cycle the absorber fluid (water) is used to collect the refrigerant vapour (ammonia) from the evaporator. This removes the heat from the ammonia allowing the water to hold the ammonia at a higher concentration. The mixture is then pumped to the generator where heat energy is absorbed into the system. The heat source needs to provide a temperature between 100°C and 170°C, (Cengel & Boles 2005). This removes the ammonia vapour from the solution allowing the water to return to the absorber via a heat exchanger which increases the temperature.
of the ammonia carrying solution before it enters the generator. The vapour continues to the condenser where the ammonia is cooled to a liquid form by removing some of the heat energy. The liquid ammonia then passes through an expansion valve causing a drop in pressure which vaporises the fluid. This process draws heat energy from the refrigerated area thus creating the desired refrigeration effect. The vapour then passes back to the absorber where the process starts again, (Figure 2), (Eastop & McConkey 1993).

![Figure 2: Absorption Pump Refrigeration Cycle, (Eastop & McConkey 1993)](image)

A modification can be made to the absorption cycle system so that the pump is not needed to drive the cycle. In this type of system a hydrogen gas setup is situated between the evaporator and the absorber, (Figure 3). In this case the ammonia vapour evaporates into the hydrogen gas at a low temperature due to its low partial pressure. The hydrogen and ammonia mixture moves to the absorber where the ammonia is absorbed into the water and the hydrogen passes back to the evaporator to repeat the process, this drives the cycle through the differences in the pressure and eliminates the need for the pump, (Eastop & McConkey 1993). This means that the only energy input to the cycle is the heat energy introduced at the generator.
The effectiveness of a refrigeration cycle is given as the coefficient of performance (COP). There is a higher amount of heat energy required to run an absorption cycle therefore the COP is lower than that of the vapour compression cycle, (Moran & Sharpiro 2008). The COP is equal to the energy removed from the refrigerative area \( Q_{\text{ref}} \) divided by the energy used to run the cycle \( Q_{\text{input}} \), (Tora & El-Halwagi 2010).

\[
COP = \frac{Q_{\text{ref}}}{Q_{\text{input}}}
\]

By using a freely available heat source the apparent COP of the system is increased. The waste heat from industrial processes is a common heat source used for this type of refrigeration, (Cengel & Boles 2005). An alternative would be to use the solar heat energy to operate the cycle.

### 2.2 Prior research

Solar energy has been suggested to drive an absorption cycle refrigerator in the past, (Guangming & Hihara 1999). Such systems have been proposed for different applications with a number of techniques employed to overcome the problems associated with using a solar heat source.

Three main issues have been identified with using a solar heat source to drive an absorption cycle refrigeration system. These are: low coefficient of performance; poor quality and quantity of heat supplied; and the difficulty of keeping the system at a steady state throughout the day, (Guangming & Hihara 1999).

One method used to combat these issues was to use a combination of solar, excess process heat and fossil fuels to provide the heat for the generator. These separate heat sources would supplement each other to ensure a consistent heat supply for the system. It was suggested that the heat generated for this system be accumulated in a heat storage tank. The use of the tank ensures that...
the heat can be delivered from a constant and reliable source throughout the day despite changes in the solar heat input, (Figure 4), (Tora & El-Halwagi 2010).

Another suggested method of improving the reliability and performance was to use a cycle that was co-driven by solar heat and an electrical compressor, (Figure 5). In this system the compressor draws some of the vapour leaving the generator and runs it through a vapour compression cycle. This compressor draws a different amount of vapour depending on short fall from the solar heat source. This ensures that the system can provide a constant refrigeration capacity while using less energy, (Guangming & Hihara 1999).

The analysis of the above system showed it not only overcame the issues associated with an unsteady heat supply but also had a greater coefficient of performance when compared to a traditional absorption cycle, (Figure 6). In this figure COP is the COP for the traditional cycle, COP is the new system under perfect conditions and COP is the new system with efficiency taken into account.
2.3 Heat Collection and Transfer

To collect solar heat energy an effective solar heat collector is needed. The evacuated tube collector is far better at delivering temperatures more than 50°C above the ambient temperature. This type of collector also works much better than the flat plate absorber in overcast and cooler conditions, (Huang 2010). A comparison of the performance shows the higher performance achieved by the evacuated tube collector against the flat plate type in delivering the higher temperatures, (Figure 7).

The evacuated tube collector consists of a number of evacuated tubes on an absorber plate backing. Each tube contains a small amount of working fluid. When the tube absorbs heat from the sun it evaporates the fluid. The vapour then moves via convection to the top end of the tube where a heat exchanger manifold removes the heat from the vapour. The condensed working fluid then moves back down the tubes under the force of gravity to start the process again, (Figure 8 & 9). As a lot of
the heat energy is carried as latent heat and moved via convection the heat transfer rate is very effective. It is this property that makes this type of collector more effective, (Azad 2008).

Figure 8: Evacuated Tube Schematic, (Azad 2008)

Figure 9: Evacuated Tube Collector Manifold, (Azad 2008)

Evacuated tubes can also be used individually to transfer heat efficiently these are also known as heat pipes. In a heat pipe the heat is applied at one end which causes the working fluid to evaporate and move to the other end where the heat is dissipated. A heat pipe can be constructed so that the heat transfer can occur in either direction and does not rely on gravity to retrieve the condensed fluid. In this case the fluid is returned via the porous tube lining through capillary action, (Figure 10), (Tucker n.d.). The heat transfer rate provided by a heat pipe is much higher than the best solid conductor. A common type of heat pipe is a copper tube with water as the working fluid. This type of heat pipe has an operating temperature range of between 5 °C to 230 °C, (Holman 1981). This operating temperature range is ideal for providing the delivery temperature required in an ammonia water absorption cycle.
This literature review has identified that it is possible to use solar heat energy to power an absorption cycle refrigerator. The aim of this investigation is to determine the design requirement for a solar collection system for an existing absorption refrigerator. This design along with changes to the fridge itself will be designed to overcome the shortfalls identified in the use of solar heat energy. The resultant design can be used to build a prototype for testing the design in a practical application.
3.0 Refrigerator Characteristics

The absorption cycle that is present in the refrigerator used for the experiments is the same cycle as the hydrogen driven system discussed in the literature review, (Chapter 2.1). The Figure below shows the back of the fridge with each component labelled, (Figure 11). The evaporator continues through to the refrigerated area where it can provide the cooling, (Figure 12).

![Absorption Cycle](image1)

![Absorption Refrigerator](image2)
The refrigerated volume has a length of 440 mm a width of 240 mm and a depth of 320 mm, giving it a volume of 33.8 L, (Figure 13).

Figure 13: Refrigerated Area Volume

The walls of the refrigerator consist of 1 mm steel plate on the outside and 2mm of PVC plastic on the inside and polyurethane foam has been used to fill the void between these surfaces. The walls, lid and base varying thicknesses thus giving the surfaces a different amount of polyurethane foam, (Figure 14).

Figure 14: Wall Cross-sections
4.0 Proposed Design

The proposed system design will consist of an evacuated tube collector, thermal reservoir, heat pipe, extra insulation and a thermal mass inside the fridge, (Figure 15). This system will be able to collect and deliver enough heat to drive the refrigeration cycle while maintaining consistent temperature within the refrigerated area when cooling is not being provided.

![Figure 15: Proposed Design Schematic](image)

The heat will be collected from the sun by the evacuated tube collector before being retrieved at the condenser end in the thermal reservoir. The heat collected at the thermal reservoir will be transferred to the generator via the heat pipe, (Figure 16).

![Figure 16: Heat Transfer Schematic](image)

The following specifications were determined through a series of characteristic performance tests, research and theoretical calculations. The components involved in the heat transfer process are detailed further in chapter 5.0.

The research into solar heat collectors found that a system containing 2 GL70 type evacuated tubes with the capacity to produce 155 to 170 W of heat energy could provide the required energy most efficiently. The heat generated from this solar collector will be collected at the condenser end in a thermal reservoir, which can hold the heat at the temperature required to drive the cycle. From the thermal reservoir the heat will be absorbed by the evaporator end of a 6 mm diameter copper-water heat pipe. This insulated heat pipe will deliver the heat to the generator section of the absorption cycle. This heat pipe will have a wick structure so that the heat can flow against gravity. This system
is capable of delivering the required 150 W of heat energy required to run the fridge at maximum setting. Further design needs to be conducted on the thermal reservoir to fit the selected pieces.

To assist the solar driven design it is proposed that the insulation on the 3 side walls and the lid be increased with 40mm of polystyrene. This insulation will reduce the heat gained through the walls from 9.6 W to 6.9 W during a 30 degree temperature difference. This will increase the effectiveness of the refrigeration cycle while in operation as well as reducing the rate at which the fridge reheats when cycle isn’t operating.

A thermal mass of 0.6 kg of ice is suggested to act as a thermal mass inside the refrigerated area. This mass of ice has been sized to be enough to accommodate for the theoretical heat losses through the wall over a 12 hour period. This will assist in keeping the refrigerative compartment at a constant temperature during time of inconsistent heat supply. The methodology taken to find these specifications is detailed in chapter 5.0.
5.0 Results & Discussion

To make improvements to the fridge a number of tests were run to see if the improvements would be significant enough to warrant change. The methodologies and raw data for these experiments can be found in Appendices 1 to 6. The conclusions drawn from the results in the experiments are outlined below. Calculations were also conducted to determine the size of the solar collection and delivery system, these can be found in Appendices 6 -10.

To effectively determine the benefit of the design changes a standard test was conducted on the refrigerator. This test involved running the empty fridge for 12 hours before switching it off. During this time temperature recording were taken of the fridge, the inside lid surface and the ambient temperature, (Figure 17). This figure demonstrates the temperature change characteristics of the refrigerator without any design changes made to it. This figure will provide a comparison to the other tests to see if changes to the temperature curves can be observed.

![Figure 17: Standard Test](image)

5.1 Inclusion of a Thermal Mass

The first consideration in this investigation was to determine improvements that could be made to the existing refrigerator to improve the performance. The inclusion of a thermal mass in the refrigerated area was considered as a method to combat the issue of inconsistent heat provided by the sun. This thermal mass would be able to provide cooling through the release of latent heat when the refrigeration effect is reduced. To test the effectiveness of a thermal mass, characteristic performance tests were conducted for comparison purposes, (Appendix 2). Theoretical calculations were performed to determine the optimum size, (Appendix 8).

To test the effectiveness of the thermal mass a comparison was made between the standard test graph, (Figure 17), and the thermal mass test graph, (Figure 18). From this comparison it was clearly seen that the thermal mass reduced the rate of reheat when the refrigerator was switched off from 0.25 °C/min to 0.15 °C/min and then to $1.3 \times 10^{-2}$ °C/min during the plateau period, (Appendices 1 & 2).
These tests confirmed that thermal mass was effective in reducing the reheat cycle of the refrigerator. However to see if the thermal mass could slow the reheat of the refrigerated area for a longer time a further test was conducted with a longer cooling period. This would allow for the production of more ice and therefore more latent heat release, (Figure 19).
This test showed that the temperature could be controlled by a thermal mass while the refrigerative cycle isn’t working. In this test the reheat rate during the initial period was reduced to $6.5 \times 10^2$ °C/min. After this the temperature started to drop at a rate of $1.6 \times 10^2$ °C/min. This drop in temperature can be explained by the increase in the amount of latent heat release. The temperature then rose again to 7.5 °C and stayed at this level for the last 500 minutes of the test, (Appendix 5).

To effectively size the thermal mass the amount of heat gained through the walls over a 12 hour period was calculated by evaluating the thermal resistivity of the refrigerator walls and calculating the rate of heat transfer through these surfaces. Using this value it was calculated that 0.6 kg of ice would have enough latent heat to counter the heat gain over this period, (Appendix 8).

The latent heat release from the thermal mass is what helps keep the fridge temperature consistent. This is because large amounts of heat energy are being absorbed by the ice without any change in temperature. By correctly sizing the thermal mass, the refrigerator could keep the contents cold when the cooling power is interrupted. This thermal mass will reduce the refrigerative volume available for use. 0.6 kg of ice has a volume of approximately 0.652 L which accounts for approximately 2.0% of the refrigerated volume. This is an acceptable amount considering the entire space is not normally utilised during use.

### 5.2 Extra Insulation

Another consideration made for improving the refrigerator performance was to increase the insulation around the refrigerator. By increasing the thermal resistance of the walls it will decrease the refrigerator’s susceptibility to ambient temperature change. To test the effectiveness of the extra insulation characteristic performance test were conducted for comparison, (Appendix 3). Theoretical calculations were conducted to calculate the expected reduction in the heat gained by the refrigerator, (Appendix 7). These calculations found that there was a significant change in the rate at which the heat transferred through the walls.

For extra insulation 40 mm of polystyrene was attached to the outside of the three side walls as well as the lid of the refrigerator, (Figure 20). The thermal insulation test, (Figure 21), was compared to the standard test, (Figure 17), to check the effectiveness of the extra insulation. The comparison showed that the insulation had the effect of improving the refrigerator performance due to the reduction in heat transfer.
This test showed that the refrigerator was able to maintain a lower internal temperature than the standard test. This is because the extra insulation reduces the rate at which heat enters the refrigerator thus allowing a lower temperature to be maintained with the same refrigerative capacity. However there was little apparent change in the reheat of the refrigerator, (Appendices 1 & 3). To see if there is an advantage during the reheat process as well a test was performed with a thermal mass included so that the any change would be apparent. This test showed the plateau period of the reheat cycle increase from 50 minutes, without extra insulation, to 90 minutes with the insulation, (Appendix 5). The test also demonstrated the reduced temperature change by the end of the reheat cycle, (Figure 22).
The reduction in heat gained by the fridge with the 40 mm polystyrene sheeting when the temperature difference was 30 degrees was calculated, (Appendix 7). These calculations found the heat gained through the walls without extra insulation was 9.6 W. When the polystyrene was added to the walls it was found that the heat gained through the walls was reduced to 6.9 W. These calculations were made only considering the thermal conductive gains of the refrigerator, the solar radiance gains were not calculated due to the inaccuracy in calculating this gain and also because these values would be the same for both cases.

5.3 Selection of Solar Collector

The electrical analysis indicated that the refrigerator requires 155.1 W of electrical power to provide heat energy to the generator, (Appendix 6). This value was used to size the solar heat collector to ensure that enough heat is captured to operate the refrigerator.

Through a literature review it was found that the most efficient type of solar thermal collector at temperatures 50 °C above the ambient temperature was the evacuated tube collector. The evacuated tube collector has been proven to provide superior performance at these higher temperatures, (Huang 2010). It was therefore determined that an evacuated tube collector would be best suited to this application.

To meet these requirements a 2 tube GreenLand Systems’ GL70 type evaporated tube collector was selected. This type of collector is not a standard size but is a reduction in the size of an existing system. By recalibrating the specifications, the specifications of the solar collector are as follows, (Table1).
Table 1: Solar Collector Specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of tubes</td>
<td>2</td>
</tr>
<tr>
<td>Tube Dimensions ( \phi \times ) length (mm)</td>
<td>70 x 1750</td>
</tr>
<tr>
<td>Collector Dimensions length x width (mm)</td>
<td>1900 x 200</td>
</tr>
<tr>
<td>Mass (kg)</td>
<td>7.4</td>
</tr>
<tr>
<td>Heating Capacity (W)</td>
<td>155 - 170</td>
</tr>
</tbody>
</table>

This type of solar collector will be able to collect the heat energy required to operate the prototype refrigerator. To collect the heat at the condenser end a thermal reservoir will be needed to collect the heat from both tubes and to deliver it to the heat pipe.

5.4 Selection of Thermal Reservoir

To collect the heat energy from the evacuated tube collector a thermal reservoir is needed at the condenser end. The thermal reservoir will serve as a transfer point between the evacuated tube collector and the delivery heat pipe. This reservoir will have to be insulated against thermal leakage and also for personal protection. The thermal reservoir wasn’t investigated further because design will need to meet the physical requirements of the solar collector and the heat pipe. It would therefore be better to design the reservoir when the physical requirements are known. This is an area that will need to be further developed before a successful prototype can be built.

5.5 Selection of Heat Pipe Type

To transfer the heat energy to the generator a heat pipe will be utilised. The literature review found that heat pipes are highly efficient conduits of heat. It was also found that the best type of heat pipe for the application was the copper tube type with water as the working fluid. This type of heat pipe fitted with a wick structure can operate against gravity which is essential for this application.

Using the specifications for this type of heat pipe and the heat transfer rate required, the size of the heat pipe was calculated, (Appendix 10). These calculations found that the pipe had to be 6 mm in diameter and that 6.8 mm of the length needs to be exposed to the heat source to achieve the required heat transfer rate. The specifications of the chosen heat pipe are as follows, (Table 2).

Table 2: Heat Pipe Specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter (mm)</td>
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</tr>
<tr>
<td>Working Fluid</td>
<td>Water</td>
</tr>
<tr>
<td>Vessel Material</td>
<td>Copper</td>
</tr>
<tr>
<td>Temperature Range (°C)</td>
<td>5 to 230</td>
</tr>
<tr>
<td>Condenser/evaporator length (mm)</td>
<td>6.8</td>
</tr>
<tr>
<td>Axial Heat Flux (W/cm²)</td>
<td>670</td>
</tr>
<tr>
<td>Surface Heat Flux (W/cm²)</td>
<td>146</td>
</tr>
</tbody>
</table>

The heat pipe will need to be suitably insulated along its length to reduce the losses to the atmosphere and improve conduction. Also the generator of the fridge will have to be modified to allow the heat pipe to deliver its heat energy.
6.0 Conclusion
As a result of this investigation the design requirements for a solar thermal driven absorption refrigerator prototype have been determined. This design incorporates change to the original refrigerator design to reduce the effects of inconsistent heat supply associated with solar heat. The solar collection and delivery system was also designed to provide the required amount of heat energy in the most efficient way.

To improve the refrigerator performance 40 mm of extra insulation was added to reduce the heat absorbed by the fridge from 9.6 W to 6.9 W. It was also calculated that a 0.6 kg ice mass could be utilized within the refrigerated area to compensate for times when the refrigeration could not be provided at its full potential.

To provide the heat to drive the cycle it was found that a system of 2 GL70 evacuated tubes connected to thermal reservoir could provide enough heat energy. A 6 mm diameter copper heat pipe with water as the working fluid and a wick structure has been found to provide the required heat transfer rate to the generator. The thermal reservoir will have to be designed to best fit the solar collector and delivery system. This investigation has therefore been able to determine the design requirements needed for a prototype solar heat driven absorption cycle refrigerator.
7.0 Reference List


Appendix 1: Standard Test

Objectives
The objective of this experiment was to develop a characteristic performance graph of the fridge without any modifications to act as a comparison for other tests.

Equipment
- CHESCOLD F40 – F60 fridge freezer
- TX55i 12 channel temperature data logger
- 2 GByte SD card
- 3 x 2 metre thermocouples

Procedure
Location: Fully shaded area out of the weather with access to a standard Australian AC power supply.

Test 1 – Standard test
1. The thermocouples were attached to read the temperature of the fridge, inside lid and surrounding air
2. The temperature recorder was set to take readings every 30 seconds
3. The fridge and recorder were turned on in the morning
4. At sun set the fridge was switched off and recorder left running
5. The test was stopped in the morning
6. The data was extracted from the SD card and temperature recording were plotted against time.

Results and Discussion
This graph shows the temperature recordings of the “Lid”, “Fridge” and “Ambient” temperature against time.

The internal lid temperature was 4-5 °C hotter than the fridge during steady state operation. This is to be expected because the warmer air in the fridge would rise to the top. Also the lid is 18 mm thinner than the walls and would therefore have a lower thermal resistance. This would allow more heat to enter through the lid thus heating the air underneath the lid.

The graph above shows that it didn’t take long for the fridge to gain heat and reach that of the ambient temperature. Again this is to be expected because there was only air inside the fridge to hold the temperature and as air doesn’t have a high thermal capacity it takes a small amount of energy to increase its temperature.

\[
y = 0.2476x - 145.54
\]

The graph above shows first 20 minutes after the refrigerator was switched off. The average rate of temperature gain of over this period is approximately 0.25 °C/min. This will give a reference value for further tests.

These results identify a clear area for improvement of the existing fridge setup. That is to increase the thermal capacitance of the fridge contents and to reduce the thermal leakage through the walls and lid. These areas of improvement will be investigated in Appendix 2 and 3 respectively.
Appendix 2: Thermal Mass Test

Objectives
The objective of this experiment was to test the effectiveness of including a thermal mass in the refrigerated area on the performance of the fridge.

Equipment
- CHESCOLD F40 – F60 fridge freezer
- TX55i 12 channel temperature data logger
- 2 GByte SD card
- 4 x 2 metre thermocouples
- 4 L water

Procedure
Location: Fully shaded area out of the weather with access to a standard Australian AC power supply.

Test 1 – Standard test

1. The thermocouples were attached to read the temperature of the fridge, inside lid, surrounding air and the thermal mass
2. The temperature recorder was set to take readings every 30 seconds
3. The fridge and recorder were turned on in the morning
4. At sunset the fridge was switched off and recorder left running
5. The test was stopped in the morning
6. The data was extracted from the SD card and temperature recording were plotted against time.

Results and Discussion
The graph shows the “Thermal mass”, “Ambient”, “Fridge” and “Lid” temperatures with respect to time.

With the additional 4 Litres of water the fridge takes a lot longer to reach a steady state. The graph appears to only reach a steady state a short time before the fridge was switched off. The water falls short of freezing temperature however given that there is a large amount of water in the fridge it is possible that the parts of the water had begun to freeze. This idea is supported by the slowing of the reheating process a short time after the fridge was switched off. The graph also shows that thermal mass prevents the fridge temperature from reaching the ambient temperature by the end of the test.

This above graph of the reheat shows that the rate of reheat is 0.15 °C/min over the first 20 minutes which is significantly lower the rate observed in the standard test. This is to be expected because the thermal capacity of the fridge is increase due to the presence of the extra water. After this period there is a stage of slower heat increase for about 50 minutes this can be attributed to the latent heat release from the ice that was able to form during the cooling process. The rate of temperature increase during this period was found to be 1.3 x 10^{-2} °C/min. If this rate can be maintained then the fridge temperature can be kept at an appropriate level.

The use of thermal mass has reduced the temperature gain of the fridge while it is switched off as expected from the standard test. As the rate of temperature increase was greatly reduced during the latent heat release it can deduced that if more ice were present this period would last longer.
Appendix 3: Insulation Test

Objectives
The objective of this experiment was to test the effectiveness of extra insulation on the performance of the fridge.

Equipment
- CHES Cold F40 – F60 fridge freezer
- TX5Si 12 channel temperature data logger
- 2 GByte SD card
- 3 x 2 metre thermocouples

Procedure
Location: Fully shaded area out of the weather with access to a standard Australian AC power supply.

Test 1 – Standard test
1. The thermocouples were attached to read the temperature of the fridge, inside lid, surrounding air and the thermal mass
2. The temperature recorder was set to take readings every 30 seconds
3. The fridge and recorder were turned on in the morning
4. At sunset the fridge was switched off and recorder left running
5. The test was stopped in the morning
6. The data was extracted from the SD card and temperature recording were plotted against time.

Results and Discussion

![Insulation Test Graph](image-url)
The resultant graph shows the behaviour of the empty fridge with extra insulation. As in the other tests the “Lid”, “Fridge” and “Ambient” temperatures are shown against time.

The temperatures reached by the fridge and the lid while running are about 5 degrees lower than those reached in standard test. This would indicate that the extra insulation is reducing the heat gained by the fridge and thus allowing it to reach a lower temperature. This is because with the higher thermal resistance of the walls a larger temperature difference is needed to achieve the same level of heat transfer through the walls. This means that less refrigerative power is needed to maintain the same temperature difference. This test has shown that the extra insulation has improved the performance of the refrigerator.

The extra insulation did not appear to change the rate at which the fridge took to reach ambient temperature after being switched off. The above graph shows the first 20 minutes of the reheat and the rate of temperature increase is shown to be approximately 0.29 °C/min. This experiment was only conducted with an empty refrigerative compartment which doesn’t have a high specific heat capacity. This meant that no noticeable change was observed during the reheat stage of the test. Also the lower temperature reached inside the fridge increased the temperature difference between the inside and outside which will increase the rate of heat transfer. With the thermal mass in place there should be a more noticeable change in the rate at which the fridge reheat. This will be investigated in Appendix 4.
Appendix 4: Insulated Thermal Mass Test

Objectives
The objective of this experiment was to test the effectiveness of extra insulation on the reheat curve by including the thermal mass to increase the thermal capacity in the fridge.

Equipment
- CHESCOLD F40 – F60 fridge freezer
- TX5Si 12 channel temperature data logger
- 2 GByte SD card
- 4 x 2 metre thermocouples

Procedure
Location: Fully shaded area out of the weather with access to a standard Australian AC power supply.

Test 1 – Standard test
1. The thermocouples were attached to read the temperature of the fridge, inside lid, surrounding air and the thermal mass.
2. The temperature recorder was set to take readings every 30 seconds.
3. The fridge and recorder were turned on in the morning.
4. At sun set the fridge was switched off and recorder left running.
5. The test was stopped in the morning.
6. The data was extracted from the SD card and temperature recording were plotted against time.

Results and Discussion

![Graph of Insulated Thermal Mass Test](image-url)
The resultant graph shows the behaviour of the extra insulated fridge with 4 litres of water over one day. As in the thermal mass test the “Lid”, “Fridge”, “Ambient” and “Thermal mass” temperatures are shown against time.

Although the day time ambient temperature was higher than in thermal mass test the cooling curves are very similar. This suggests that the extra insulation improves the ability of the fridge to handle large fluctuations in the ambient temperature without hindering its performance. Also the temperature of the fridge and the thermal mass is a couple of degrees lower than those seen at the end of the thermal mass test.

During the reheat phase it can be seen that there are some differences. The period of latent heat release shortly after the fridge is switched off last longer than the test without the extra insulation. The rate of temperature increase during the latent heat release is about the same as that seen in the thermal mass test, 1.2 x 10^{-2} °C/min, however this period lasts for about 90 minutes instead of the 50 minutes observed in the thermal mass test. This can be attributed to a combination of 2 factors: more ice was formed during the cooling process due to the reduction in the leakage through the walls; and the release of latent heat lasts longer because of the reduction in heat gain through the walls.

These results suggest that the period of latent heat release could be increased through the freezing of more ice. This should increase the plateau period observed. If the amount of ice can be optimised the temperature of the fridge could theoretically be held at a constant low temperature during the night. A longer cooling period will be tested in the longer insulated thermal mass test.
Appendix 5: Longer Insulated Thermal Mass Test

Objectives
The objective of this experiment was to test the improvement in the performance due to the thermal mass if it was allowed to freeze for a longer period.

Equipment
- CHESCOLD F40 – F60 fridge freezer
- TX55i 12 channel temperature data logger
- 2 GByte SD card
- 4 x 2 metre thermocouples

Procedure
Location: Fully shaded area out of the weather with access to a standard Australian AC power supply.

Test 1 – Standard test

1. The thermocouples were attached to read the temperature of the fridge, inside lid, surrounding air and the thermal mass
2. The temperature recorder was set to take readings every 30 seconds
3. The fridge and recorder were turned on in the morning
4. At sun set the next day the fridge was switched off and recorder left running
5. The test was stopped in the morning
6. The data was extracted from the SD card and temperature recording were plotted against time.

Results and Discussion

![Graph of Longer Insulated Thermal Mass Test](image-url)
This graph shows the “Lid”, “Fridge”, “Ambient” and “Thermal Mass” taken over the course of this test. The cooling period is approximately 3 times longer than in the previous tests.

The spike that can be seen during the cooling period is when I opened the fridge to check on the progress of the ice formation. It can be seen that the thermal mass had a significant amount of time to form the ice, and from inspection I found that a reasonable amount of the water had frozen.

After the period of latent heat release shown in the graph below the fridge temperature stayed at a constant temperature of about 7.5 degrees for the next 500 minutes until the end of the test. The rate of temperature change through this period was calculated to be \(-9.0 \times 10^{-4}\) degrees per minute. Also the thermal mass had only risen to 1.5 °C by the end of the test.

![Long Thermal Test Reheat](image)

After the fridge was switched off the temperature increased at a rate of \(6.5 \times 10^{-2}\) °C/min for a period of 50 minutes. This initial temperature rise is at a rate less than half that observed in the other tests. After this it can be seen that the latent heat release actually caused a drop in the fridge temperature at a rate of \(1.6 \times 10^{-2}\) degrees per minute for a period of 100 minutes. The temperature then rises again before settling at a constant temperature of about 7.5°C for the remainder of the test.

This has shown that a constant temperature can be achieved by using an effective thermal mass. An appropriate sized mass can be calculated by calculating an approximate amount of heat gain during the night. This test has also shown that food items stored in the fridge would be kept at a suitable temperature during the night. This is because any items in the fridge would also act as a thermal mass and this test showed an increase to only 1.5 °C which is within the appropriate range for a fridge.
Appendix 6: Electrical Energy Consumption

With the fridge set to its maximum setting measurements of voltage and current was taken at 2 millisecond intervals for 6 hours. The power consumption of the fridge was constant during this time. It was also found that the data logging equipment drew some power while the fridge was off. Using the data taken during this test the true power consumption of the fridge could be calculated.

During fridge operation

\[ V_{\text{max}} = 334.09V \quad V_{\text{RMS}} = \frac{334.09}{\sqrt{2}} = 236.2373V \]
\[ I_{\text{max}} = 0.9542A \quad I_{\text{RMS}} = \frac{0.9542}{\sqrt{2}} = 0.674721A \]

Therefore the total power consumption of the fridge can be calculated

\[ P_t = I_{\text{RMS}} V_{\text{RMS}} \]
\[ = 0.674721 \times 236.2373 \]
\[ \approx 159.4W \]

Readings taken by while the fridge was off

\[ V_{\text{max}} = 334.09V \quad V_{\text{RMS}} = \frac{334.09}{\sqrt{2}} = 236.2373V \]
\[ I_{\text{max}} = 0.0257A \quad I_{\text{RMS}} = \frac{0.0257}{\sqrt{2}} = 0.018173A \]

Therefore power consumed by the data logger can be calculated

\[ P_c = I_{\text{RMS}} V_{\text{RMS}} \]
\[ = 0.018173 \times 236.2373 \]
\[ \approx 4.3W \]

Therefore the true max power consumption of the fridge is:

\[ P_f = P_t - P_c \]
\[ = 159.4 - 4.3 \]
\[ = 155.1W \]
Appendix 7: Insulation Calculations
The heat gained through the walls can be calculated by determining the thermal resistivity of all the surfaces. The rate of heat gain can be reduced by adding extra insulation which will increase the thermal resistivity. The calculation will be made by considering a temperature difference of 30 degrees between the inside of the fridge and the outside temperature. The drawing below shows the makeup of the fridge walls, lid and base and the dimensions of the surfaces. An extra insulating layer of 40mm of polystyrene will be added to the outside of the walls for the second set of calculations.

Fridge volume

Calculations for the walls as they are:

\[ A_{walls} = 2 \times (0.24 \times 0.32) + 2 \times (0.32 \times 0.44) = 0.4352 \, m^2 \]
Wall material thermal properties, (The Engineering Toolbox n.d.).

\[ t_{steel} = 0.001m \quad k_{steel} = 43W/mK \]
\[ t_{poly} = 0.065m \quad k_{poly} = 0.03W/mK \]
\[ t_{pvc} = 0.002m \quad k_{pvc} = 0.19W/mK \]

Surface air resistivity

\[ R_{so} = 0.029K/W \quad R_{si} = 0.125K/W \]

Resistivity of layers

\[ R_{steel} = \frac{t_{steel}}{k_{steel}A_{walls}} = \frac{0.001}{43 \times 0.4352} \approx 5.344 \times 10^{-5} K/W \]
\[ R_{poly} = \frac{t_{poly}}{k_{poly}A_{walls}} = \frac{0.065}{0.03 \times 0.4352} \approx 4.979 K/W \]
\[ R_{pvc} = \frac{t_{pvc}}{k_{pvc}A_{walls}} = \frac{0.002}{0.19 \times 0.4352} \approx 2.419 \times 10^{-2} K/W \]

Total wall resistivity as it is:

\[ R_{total} = R_{so} + R_{steel} + R_{poly} + R_{pvc} + R_{si} \]
\[ = 0.029 + 5.344 \times 10^{-5} + 4.979 + 2.419 \times 10^{-2} + 0.125 \]
\[ = 4.849 K/W \]

Assuming a 30 degree temperature difference between the outside and the inside then the heat gained through the walls as they are is:

\[ \dot{Q}_{loss} = \frac{\Delta t}{R_{walls}} = \frac{30}{4.849} \]
\[ = 6.19W \]

For the lid as it is:

\[ A_{lid} = 0.44 \times 0.24 = 0.1056m^2 \]
\[ t_{poly} = 0.047 \]
Resistivity of each layer

\[ R_{\text{steel}} = \frac{t_{\text{steel}}}{k_{\text{steel}}A_{\text{walls}}} = \frac{0.001}{43 \times 0.1056} \approx 2.202 \times 10^{-4} \text{ } K/ W \]

\[ R_{\text{poly}} = \frac{t_{\text{poly}}}{k_{\text{poly}}A_{\text{walls}}} = \frac{0.047}{0.03 \times 0.1056} \approx 14.836 \text{ } K/ W \]

\[ R_{\text{pvc}} = \frac{t_{\text{pvc}}}{k_{\text{pvc}}A_{\text{walls}}} = \frac{0.002}{0.19 \times 0.1056} \approx 9.968 \times 10^{-2} \text{ } K/ W \]

Total lid resistivity

\[ R_{\text{total}} = R_{\text{as}} + R_{\text{steel}} + R_{\text{poly}} + R_{\text{pvc}} + R_{\text{st}} \]
\[ = 0.029 + 2.202 \times 10^{-4} + 14.836 + 9.968 \times 10^{-2} + 0.125 \]
\[ = 15.09 \text{ } K/ W \]

The heat gained through the lid with a 30 degree temperature difference is:

\[ \dot{Q}_{\text{loss}} = \frac{\Delta t}{R_{\text{walls}}} = \frac{30}{15.09} \]
\[ = 1.99 \text{ } W \]

Resistivity of the base

\[ A_{\text{base}} = 0.24 \times 0.44 = 0.1056 \text{ } m^2 \]

\[ R_{\text{steel}} = \frac{t_{\text{steel}}}{k_{\text{steel}}A_{\text{base}}} = \frac{0.001}{43 \times 0.1056} = 2.202 \times 10^{-2} \text{ } K/ W \]

\[ R_{\text{poly}} = \frac{t_{\text{poly}}}{k_{\text{poly}}A_{\text{base}}} = \frac{0.067}{0.03 \times 0.1056} = 21.147 \text{ } K/ W \]

\[ R_{\text{pvc}} = \frac{t_{\text{pvc}}}{k_{\text{pvc}}A_{\text{base}}} = \frac{0.002}{0.19 \times 0.1056} = 9.9681 \times 10^{-2} \text{ } K/ W \]

Total resistivity

\[ R_{\text{base}} = 0.029 + 0.125 + 2.202 \times 10^{-4} + 21.147 + 9.9681 \times 10^{-2} \]
\[ = 21.401 \text{ } K/ W \]

The heat gained through the base with a 30 degree temperature difference is:

\[ \dot{Q}_{\text{loss}} = \frac{\Delta t}{R_{\text{walls}}} = \frac{30}{21.401} \]
\[ = 1.40 \text{ } W \]

Total heat gained into the fridge with current insulation and 30°C temperature difference:

\[ \dot{Q}_{\text{total}} = 6.19 + 1.99 + 1.40 = 9.58 \text{ } W \approx 9.6 \text{ } W \]
With insulation

With 40mm of polystyrene added to the 3 facing walls and the lid the thermal gains can be reduced.

For the walls

\[
A_{inwalls} = 2 \times (0.24 \times 0.32) + (0.32 \times 0.44) = 0.2944
\]

\[
t_{styrene} = 0.04 \text{m} \quad k_{styrene} = 0.03 \text{W/mK}
\]

\[
k_{styrene} = \frac{t_{styrene}}{k_{styrene}A_{walls}} = \frac{0.04}{0.03 \times 0.2944} \approx 4.529 \text{K/W}
\]

\[
R_{steel} = \frac{t_{steel}}{k_{steel}A_{walls}} = \frac{0.001}{43 \times 0.2944} \approx 9.899 \times 10^{-5} \text{K/W}
\]

\[
R_{poly} = \frac{t_{poly}}{k_{poly}A_{walls}} = \frac{0.065}{0.03 \times 0.2944} \approx 7.360 \text{K/W}
\]

\[
R_{pvc} = \frac{t_{pvc}}{k_{pvc}A_{walls}} = \frac{0.002}{0.19 \times 0.2944} \approx 3.576 \times 10^{-2} \text{K/W}
\]

The new total resistivity for the walls with the polystyrene

\[
R_{walls} = 0.029 + 0.125 + 4.529 + 9.899 \times 10^{-5} + 7.360 + 3.576 \times 10^{-2} = 12.079 \text{K/W}
\]

The heat gained through the walls with the polystyrene

\[
\dot{Q} = \frac{\Delta T}{R_{walls}} = \frac{30}{12.079} = 2.48 \text{W}
\]

For the lid

\[
R_{styrene} = \frac{t_{styrene}}{k_{styrene}A_{lid}} = \frac{0.04}{0.03 \times 0.1056} = 12.626 \text{K/W}
\]

The new thermal resistivity for the lid is

\[
R_{lid} = 15.09 + 12.626 = 27.716 \text{K/W}
\]

Therefor the heat gained through the lid with the polystyrene is:

\[
\dot{Q} = \frac{\Delta T}{R_{lid}} = \frac{30}{27.626} = 1.08 \text{W}
\]
Resistivity of the rear wall

\[ A_{\text{rear}} = 0.44 \times 0.32 = 0.1408 \text{m}^2 \]

\[ R_{\text{steel}} = \frac{\frac{t_{\text{steel}}}{k_{\text{steel}} A_{\text{rear}}}}{0.001} = \frac{0.001}{43 \times 0.1408} = 1.65 \times 10^{-4} \text{K/W} \]

\[ R_{\text{poly}} = \frac{\frac{t_{\text{poly}}}{k_{\text{poly}} A_{\text{rear}}}}{0.065} = \frac{0.065}{0.03 \times 0.1408} = 15.3883 \text{K/W} \]

\[ R_{\text{pvc}} = \frac{\frac{t_{\text{pvc}}}{k_{\text{pvc}} A_{\text{rear}}}}{0.002} = \frac{0.002}{0.19 \times 0.1408} = 7.4761 \times 10^{-2} \text{K/W} \]

Total resistivity

\[ R_{\text{rear}} = 0.029 + 0.125 + 1.65 \times 10^{-4} + 15.3883 + 7.4761 \times 10^{-2} \]

\[ = 15.617 \text{K/W} \]

Therefor the heat gained through the rear wall is:

\[ \dot{Q} = \frac{\Delta T}{R_{\text{lid}}} = \frac{30}{15.617} \]

\[ \dot{Q} = 1.92 \text{W} \]

Heat gained through the base remains the same at 1.40 W therefore total heat gains with extra insulation is:

\[ \dot{Q}_{\text{total}} = 2.48 + 1.08 + 1.92 + 1.40 = 6.88 \]

\[ \approx 6.9 \text{W} \]
Appendix 8: Thermal Mass Sizing

For calculating the size of a thermal mass in the fridge it will be assumed that the shading coefficient is zero at night, i.e. the solar heat absorption will be 0kJ. This means that the heat gained will only be through the walls. The heat gained will be calculated over a 12 hour period with an assumed constant temperature difference of 20 degrees.

The thermal resistivity for all the surfaces will be the same as calculated in extra insulation part of appendix 8.

Resistivity of insulated walls:

\[ R_{\text{walls}} = 12.079 \text{K/W} \]

Resistivity of lid:

\[ R_{\text{lid}} = 27.716 \text{K/W} \]

Resistivity of the rear wall

\[ R_{\text{rear}} = 15.617 \text{K/W} \]

Resistivity of the base

\[ R_{\text{base}} = 21.401 \text{K/W} \]

The total heat absorption in Watts can be calculated

\[
\dot{Q}_{\text{loss}} = \frac{\Delta T}{R_{\text{walls}}} + \frac{\Delta T}{R_{\text{lid}}} + \frac{\Delta T}{R_{\text{rear}}} + \frac{\Delta T}{R_{\text{base}}} = \frac{20}{12.079} + \frac{20}{27.716} + \frac{20}{15.617} + \frac{20}{21.401} \\
= 1.66 + 0.72 + 1.28 + 0.93 \\
= 4.59 \text{W}
\]

Total energy gained over a 12 hours period with a constant 20 degree temperature difference.

\[
\dot{Q}_{\text{gain}} = 4.59 \times 12 \times 3600 \times 10^{-3} \\
\approx 198 \text{kJ}
\]

Therefore the thermal mass needs to have 200kJ of latent heat available to counter the losses through the night. The latent heat of water of melting is: 334kJ/kg , (The Engineering Toolbox n.d.).

Therefore the mass of ice water needed to offset this can be calculated:

\[
m_{\text{ice}} = \frac{\dot{Q}_{\text{loss}}}{s_p} = \frac{198}{334} \\
= 0.593 \approx 0.6 \text{kg of ice}
\]
Appendix 9: Solar Collector Sizing

Most commercial evacuated tube solar heat collectors are designed for water heating and industrial purposes and begin in size at approximately 1 kW which is a lot larger than what is needed for this sized refrigerator. To determine the size required for this application the specifications for a larger commercial collector will be used and resized.

GreenLand Systems Evacuated Tube Solar Collector

The GreenLand Systems’ GL70 type collector comes in sets of 20 and 40 tubes. The tubes are 70mm in diameter and 1750mm in length. The set of 20 has a rated heat capacity of 1550 – 1700 W which is approximately 10 times the capacity needed to run the refrigerator, (GreenLand Systems 2008-2011). This means a system with 2 heat pipes would be able to produce 155 – 170 W of heat energy. This higher supply of heat energy will be needed to accommodate for losses during transfer. Also the heat pipe will have a design maximum limit of how much heat it can transfer due to four factors; the Sonic Limit, Entrainment Limit, Capillary Limit and Boiling Limit, (Enerton 2001). A two pipe system would have an approximate area of 200 x 1900.
Appendix 10: Heat Pipe Sizing
The heat pipe needs to be able to deliver the energy to the generator up at a rate of 150 W.

Using water as the working fluid inside a copper vessel the axial heat flux \( (c_{axial}) \) is \( 670 \text{ W} / \text{cm}^2 \), \( \text{Holman 1981} \).

Determining the diameter to the heat pipe

\[
155.1W = c_{axial} \frac{\pi}{4} d^2
\]

\[
= 670 \frac{\pi}{4} d^2
\]

\[
d = 0.5429 \text{ cm}
\]

\[
\approx 5.43 \text{ mm}
\]

The closest standard sized pipe is 6 mm; also the larger diameter will allow the heat pipe to carry more heat to allow for losses during transfer.

The evaporator and the condenser ends of the heat pipe have to provide enough surface area for heat energy to be absorbed into and release from the heat pipe at the required rate. The surface heat flux \( (c_{surface}) \) is \( 146 \text{ W} / \text{cm}^2 \), \( \text{Enerton 2001} \).

Determining the length of heat pipe incorporated into the condenser and evaporator ends:

\[
155.1W = c_{surface} \times \pi d \ell
\]

\[
= 146 \times \pi \times 0.5 \times \ell
\]

\[
\ell = 0.676 \text{ cm}
\]

\[
\approx 6.8 \text{ mm}
\]

Therefore a minimum length of 6.6 mm needs to be exposed to both the heat source and the absorption cycle generator.
Appendix 11: Refrigerator Specifications

THIS FRIDGE FREEZER MUST BE LEVEL IN BOTH DIRECTIONS AT ALL TIMES WHEN OPERATING. DO NOT OPERATE ELECTRIC AND GAS TOGETHER.

TO OPERATE ON 240 VOLT AC
1. ENSURE 12V & LP GAS SUPPLY IS TURNED ‘OFF’
2. PLUG 3 PIN LEAD INTO POWER POINT AND SWITCH ‘ON’.
3. TURN THERMOSTAT KNOB TO EITHER ‘FREEZE’ OR ‘REFRIGERATE’ POSITION.
4. AFTER 20 MINUTES CHECK THAT REAR TOP SLOTTED PANEL IS GETTING WARM.

TO OPERATE ON 12 VOLT DC (CAR BATTERY)
1. ENSURE 240V & LP GAS SUPPLY ARE TURNED ‘OFF’
2. CONNECT 12 VOLT LEAD TO CAR BATTERY, USE PLUG AS SUPPLIED.
3. ENSURE THAT A MINIMUM 5mm WIRE IS RUN DIRECT TO CAR BATTERY NOT TO AN AUXILIARY CIRCUIT ON CAR.
4. ENSURE THAT A 15AMP FUSE IS FITTED TO 5mm WIRE CLOSE TO BATTERY.
5. AFTER 20 MINUTES CHECK THAT REAR TOP SLOTTED PANEL IS WARM.

TO OPERATE ON LP GAS
1. ENSURE 240V & 12V SUPPLY ARE TURNED ‘OFF’.
2. TURN THERMOSTAT KNOB TO EITHER ‘FREEZE’ OR ‘REFRIGERATE’ POSITION.
3. CONNECT FLEXIBLE HOSE FROM REAR OF FRIDGE TO GAS CYLINDER AND ENSURE GAS CONNECTION TO FRIDGE IS TIGHT (USE A SPANNER).
4. A REGULATOR MUST BE FITTED TO CYLINDER (BETWEEN CYLINDER AND FRIDGE).
5. DEPRESS SAFETY VALVE ON BACK OF FRIDGE (SIDE ON F40) AND AT THE SAME TIME PRESS THE PIEZO IGNITER BUTTON SEVERAL TIMES IN QUICK SUCCESSION UNTIL GAS BURNER IS ALIGHT.
6. HOLD GAS SAFETY VALVE PRESSED FOR FURTHER 10 SECONDS.
7. IF BURNER FAILS TO STAY ALIGHT REPEAT 5 AND 6.

WARNING!! DO NOT OPERATE THIS APPLIANCE IN ANY UNVENTED ENCLOSED AREAS SUCH AS CLOSED TENTS OR CLOSED MOTOR VEHICLES. FRESH AIR CIRCULATION MUST BE AVAILABLE TO THE UNIT AT ALL TIMES WHILST OPERATING.

MODEL F40 – F60

PRODUCTION No. 1 – 885

VOLTS 240 VOLT AC – 150 WATTS 50 Hz
VOLTS 12 VOLT DC – 150 WATTS 12.5 AMPS

GAS PROPANE 2.75kPa

CONSUMPTION 9MJ

S.E.C. APPROVAL No. 02882 – F40 – F60

MANUFACTURED BY CHESCO PTY. LTD. BILSEN ROAD, GEEBUNG, BRISBANE, AUSTRALIA

S/N – 86 No. 8727