Feasibility of solar pond-assisted isopentane heat pump for sustainable building energy consumption

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SUMMARY

The feasibility of sustainable thermal extraction for indoor heating using a solar pond, combined with a heat pump using isopentane as the refrigerant, was experimentally investigated.

Heated water from the solar pond’s lower convective zone rejects heat to isopentane at a heat pump’s evaporator. After compression, isopentane rejects heat to ambient air flowing through the condenser. The heated air exits, ready for use. Isopentane was selected for its high efficiencies and environmentally-friendly properties. Experimental analyses for the heat exchangers and heat pump was conducted over a range of operation temperatures.

The heat exchangers have high effectiveness and LMTD at temperatures below 46° C, which steadily decrease afterwards. The compressor showed high exergy destruction and 22.7% efficiency, limiting the condenser’s heat rejection. Nonetheless, the heat pump’s coefficient of performance is 5–6.6 over evaporator temperatures 30-80 °C, with lower temperatures producing optimum COPh values. Therefore, the system is confirmed as feasible.

INTRODUCTION

Solar pond

With rising awareness and concerns surrounding global warming, fuel costs and abundantly wasted solar energy, solar ponds offer high potential as an alternative energy source.

The term ‘solar pond’ refers to a body of water that absorbs heat from solar radiation, had has an increasing salt concentration with depth. Having a thin, low saline concentration at the pond surface, called the upper convective zone (UCZ) and near saline saturation at the bottom region, called the lower convective zone (LCZ), this saline density gradient inhibits the free convection of heat through the water (Tabor 1981).

Sensible heat from solar energy is trapped in the LCZ, providing the ability for sustainable and inexpensive heat storage, extraction and recovery. Between the UCZ and LZC is the middle non-convective zone (NCZ), where the saline concentration varies with density, and acts as a thermal insulation between the UCZ and LCZ (Tundee et al. 2010).

A study reported that typical annual temperatures for solar ponds average up to 85° C at the LCZ for heating applications (Gommed and Grossman 1988). This allows solar ponds to be an inexpensive, long term solar collector and thermal energy source, to be used in applications that rely on low grade temperature air or water.

As there are several factors that affect the heat absorption performance and solar pond temperatures, such as solar radiation, salt water thermal properties and underground temperature, this study considers the solar pond temperatures in Melbourne over a year, as reported by Wang and Akbarzadeh (1982).

Use of internal and external heat exchangers

Thermal extraction from solar ponds is often carried out using an internal heat exchanger. Numerous studies over the years, for example, by Sabetta et al. (1985) and Jaefarzadeh (2006), conduct thermal extraction via an internal heat exchanger located at the LCZ of the solar pond, which circulates a heated fluid ready for use as a thermal source, often to an external heat exchanger. The fluid rejects heat to another working fluid at the external heat exchanger, before being pumped back to the pond base.

Counterflow arrangements favour better heat exchange between working fluids in a heat exchanger, and the Log Mean Temperature Difference (LMTD) is a method of evaluation of their mean temperature difference. Increasing LMTD values indicate higher heat transfers between the two fluids (Lienhard IV and Lienhard V 2017).

A useful indicator for heat absorbed from one fluid to another is called the heat rate (Dincer 2017). Effectiveness is another important parameter used for heat exchanger performance analysis, which is the ratio of the heat transfer rate occurring to the maximum possible heat transfer rate of the exchange (Serth and Lestina 2007).

Combing a vapour-compression heat pump

Heat pumps are systems that pump heat from a thermal source and reject it at a higher temperature to a thermal sink, using external energy sources. Utilising heat pumps substitute the need for purchased thermal energy (Berghmans 1983), but there are further benefits.

A study by Shah et al. (1981) combined a solar pond as a thermal source for a heat pump used in greenhouse heating. By incorporating this brine-sourced electric-driven heat pump, a larger temperature cycle was reported through the solar pond, concluding that the combination improved the performance of both the solar pond and heat pump.

Furthermore, heat-actuated heat pumps, as opposed to work actuated heat pumps, utilise overlooked low-grade heat,
such as that present in solar ponds. This provides an attractive method for sustainable energy consumption and abating environmental concerns.

An exergy analysis of a heat pump is a useful method of evaluating the associated energies and offering comparison. Exergy destruction, especially, vital to exposing information relevant to optimising the main components affecting the heat pump’s performance and efficiency (Baldi and Leoncini 2014).

The Coefficient of Performance for a heat pump (COPh) is a performance evaluation ratio of useful heating delivered per energy input required. Therefore, for a COPh greater than one, the heat pump offers a method of energy and cost conservation compared to other conventional heating methods. Most heat pumps used in heating applications typically have a COP between 2 and 6 (Meyer 2011).

**Assessment of isopentane as a working fluid**

An important factor in refrigerant selection is its environmental impact, to mitigate global warming and harmful effects previously caused by chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). Similarly, its use should be efficient to decrease wasted energy contributions, especially as stricter regulations on minimum equipment efficiencies are implemented.

Table 1 shows the summarised thermal and environmental properties of R-600a and R-601a. Both refrigerants display environmentally friendly properties of zero ozone depletion (ODP), and negligible global warming potential (GWP).

**Table 1. Thermal and environmental properties R-601a**

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>R-601a</th>
</tr>
</thead>
<tbody>
<tr>
<td>Molar mass (kg/kmol)</td>
<td>72.15</td>
</tr>
<tr>
<td>Boiling Temperature (°C)</td>
<td>27.8</td>
</tr>
<tr>
<td>Critical Temperature (°C)</td>
<td>187.2</td>
</tr>
<tr>
<td>Critical Pressure (MPa)</td>
<td>3.39</td>
</tr>
<tr>
<td>ODP</td>
<td>0</td>
</tr>
<tr>
<td>GWP (100yr)</td>
<td>11</td>
</tr>
</tbody>
</table>

Moreover, a theoretical analysis of the energy and exergy efficiencies of pure hydrocarbon refrigerants, such as butane, isobutane, pentane and isopentane, it was concluded that isopentane reaches a maximum efficiency as butane, so is the recommended alternative to R-22 and R-134a (Bayrakçi and Özgür 2009).

**Aim and purpose of the design**

An experimental rig and vapour compression heat pump has been designed and built in this project with the aim of analysing solar ponds as a main heat source for sustainable air heating applications in buildings, while also decreasing fossil-fuel based energy in conventional electrical or oil heaters.

This study aims to harvest thermal energy to be used as a long-term, environmentally-friendly approach for indoor heating and comfort applications. This study’s method experimentally investigates the feasibility of extracting thermal energy from a solar pond combined with a vapour compression heat pump, using isopentane as its working fluid.

The thermal energy extracted from the solar pond can be transferred to the heat pump, where it is amplified and rejected to ambient air flowing through the heat pump's condenser. This heated air exits the condenser ready for use in low grade heating, such as indoor comfort applications.

**METHODS**

**Schematic design**

A schematic of the study's design is shown in Figure 1. This study uses two closed loop systems for the heat extraction and transfer from the solar pond to the heat pump, and a third open loop for heat rejection to the air.

![Figure 1 - Schematic design of experimental setup](image)

The solar pond acts as the heat source through its collection and storage of heat from solar radiation. In the first loop, an internal heat exchanger situated at the pond’s base, containing water as its working fluid, circulates the heat extracted from the pond’s LCZ to an external heat exchanger at the heat pump. The cooled water is pumped back to the solar pond for heating.

The heat pump, or second loop, consists of four main components – an evaporator, a compressor, a condenser and a throttling valve. The evaporator acts as an external heat exchanger, where heat from the first loop is rejected to the isopentane working fluid in the second loop. The heated, vaporised isopentane passes through the compressor where its temperature and pressure is increased, and is pumped to the condenser.

The third, open loop is the heat sink, where ambient air flows into the system at the condenser, and absorbs heat from the heated isopentane as it flows over the condenser coils, and exits the system ready for application. The isopentane vapour condenses into liquid as it leaves the condenser, and passes to the liquid receiver where excess refrigerant fluid not in circulation is stored.
The liquid is passed through the liquid filter dryer, where impurities, such as air moisture, are eliminated. Finally, the fluid passes through nozzle of the thermal expansion valve to return into vapour, where the cycle repeats as the vapour enters the evaporator and loops.

**Experimental setup used**

In the first loop, to model the solar pond, an adjustable hot water supply was used from the laboratory to replicate constant water flow rates at varying temperatures. To model the solar pond temperatures range in Melbourne over a year, the range was from 35 to 70º C with a fixed flow rate of 2 L/min. A variable flow meter with ± 5% FS accuracy was used to monitor and control the water flow rate.

The second loop consists of a vapour compression heat pump, using isopentane as working fluid. The heat pump utilises a flat plate heat exchanger evaporator, a compressor, finned condenser, liquid receiver, filter dryer, refrigerant flow meter, thermal expansion valve, and a sight glass.

The evaporator consists of two brazed flat plate heat exchangers connected in series, where the two working fluids of water and isopentane operate in countercflow. It has five plates on each side, originally designed for R134a and water, having a total area of 0.23 m² and a nominal heat load of 2.8 kW. The heat pump compressor is hermetically sealed and rated 0.74kW. Originally designed for R-404a, it has a return gas of 20º C with a 10.33 cm³ displacement volume.

The condenser comprises of an aluminium-finned copper tube with 40 raw refrigerant-to-air tubes of 9.52 mm outer diameter, 8.72 mm inner diameter and 15.4 m long. The liquid receiver is installed with a 0.75 L volumetric capacity after the condenser, connected via high pressure flexible hoses designed to withstand up to 3 MPa, whilst other tubes in the system are of annealed copper.

A variable area flow metre is installed on the liquid line after filter dryer to measure the refrigerant flow rate with accuracy of ± 5%. Full Scale (FS). The thermal expansion valve is used to adjust the superheat temperature, which is also set for R-404a.

In the air loop, a variable speed fan supplies a constant 1.5 L/sec air flow rate, where the air flowing over the condenser coil absorbs the heat from isopentane. A manometer with ± 2% accuracy was used to measure the air flow rate.

**Measurement methods**

The pressure measurement was conducted manually using pipe-fitted pressure gauges permissible error limits of ± 1%. The pressure was measured at pipe boundaries before and after the compressor, as well as before and after the condenser.

The temperature of water, isopentane, and air was measured by a type-T thermocouple with ± 0.7 error limit. For the refrigerant and water temperatures, the thermocouples were fitted on the pipe surface and insulated against heat variances from the surroundings or radiation. For the air temperatures, the thermocouples were distributed within the duct to attain reliable average temperature readings.

The COP of the heat pump (COPh) was determined from properties of isopentane at operating pressure and temperature, with the maximum COP found from temperature at condenser and evaporator (NIST 2015). The compressor’s electrical consumption was measured in terms of voltage and current to determine the power indicator.

**RESULTS**

**Heat exchangers LMTD**

Figure 2 shows water temperature versus isopentane temperature and LMTD. The results indicate that as water temperature increases, the isopentane temperature in the evaporator increases almost linearly. An increasing LMTD value is also seen with the proportional temperature increase.

**Effectiveness of evaporator**

Figure 3 shows the evaporator temperature and effectiveness versus water temperature. A parabolic increase in effectiveness is seen, which peaks at 46º C. After stagnating over 46 to 50º C, it continuously decreases thus forth.
Compressor power

Figure 4 shows the supplied and used compressor power by the heat pump over a range of evaporator temperatures. As the evaporator temperature increases, the compressor’s required power also increases.

![Supplied and used compressor power vs. evaporator temperature](image)

Figure 4 - Supplied and used compressor power vs. evaporator temperature

Heat Rate of compressor, evaporator and condenser

Figure 5 shows the condenser, evaporator and compressor heat rate versus the condenser temperature. The experimental results show a linear relationship between the components, as heat in the evaporator increases, the compression and condenser heat rate increases.

![Heat pump heat rate vs. Condenser temperature](image)

Figure 5 - Experimental heat rate for condenser, evaporator and compressor vs. condenser temperature

Exergy performance of heat pump

Table 2 shows the exergy destruction for the compressor, condenser and evaporator of the heat pump, and the exergy efficiency for each COPh and evaporator temperature. The system has an overall exergy efficiency between 30-35%.

<table>
<thead>
<tr>
<th>$T_{Evap}$ (°C)</th>
<th>COPh</th>
<th>$EX_{Comp}$</th>
<th>$EX_{Evap}$</th>
<th>$EX_{Cond}$</th>
<th>$EX_{Eff}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>61</td>
<td>5.1</td>
<td>46.52</td>
<td>0.73</td>
<td>2.54</td>
<td>34%</td>
</tr>
<tr>
<td>59</td>
<td>4.8</td>
<td>45.36</td>
<td>0.08</td>
<td>0.71</td>
<td>33%</td>
</tr>
<tr>
<td>52</td>
<td>5.1</td>
<td>37.47</td>
<td>0.13</td>
<td>0.69</td>
<td>34%</td>
</tr>
<tr>
<td>48</td>
<td>5.3</td>
<td>34.08</td>
<td>0.13</td>
<td>0.78</td>
<td>33%</td>
</tr>
<tr>
<td>44</td>
<td>5.6</td>
<td>30.04</td>
<td>0.10</td>
<td>0.78</td>
<td>33%</td>
</tr>
<tr>
<td>36</td>
<td>6.2</td>
<td>22.18</td>
<td>0.10</td>
<td>0.70</td>
<td>31%</td>
</tr>
<tr>
<td>32</td>
<td>6.4</td>
<td>21.10</td>
<td>0.10</td>
<td>0.64</td>
<td>30%</td>
</tr>
<tr>
<td>34</td>
<td>6.6</td>
<td>20.08</td>
<td>0.09</td>
<td>0.59</td>
<td>30%</td>
</tr>
<tr>
<td>39</td>
<td>5.9</td>
<td>25.96</td>
<td>0.12</td>
<td>0.85</td>
<td>32%</td>
</tr>
<tr>
<td>43</td>
<td>6.1</td>
<td>26.86</td>
<td>0.12</td>
<td>0.71</td>
<td>31%</td>
</tr>
<tr>
<td>46</td>
<td>6.2</td>
<td>20.28</td>
<td>0.24</td>
<td>7.81</td>
<td>37%</td>
</tr>
<tr>
<td>50</td>
<td>5.6</td>
<td>32.91</td>
<td>0.24</td>
<td>1.29</td>
<td>33%</td>
</tr>
<tr>
<td>58</td>
<td>5.0</td>
<td>35.54</td>
<td>0.12</td>
<td>0.55</td>
<td>34%</td>
</tr>
</tbody>
</table>

Coefficient of performance of the heat pump (COPh)

Figure 6 shows theoretical and experimental COPh versus evaporator temperature. The results reveal that the theoretical COPh is almost four times the actual COPh, with the experimental COPh of 5 - 6.6 achieved for heat source temperature range of 30-80°C.

![Heat pump coefficient of performance vs. evaporator temperature](image)

Figure 6 - Experimental and theoretical COPh vs evaporator temperature

DISCUSSION

Heat exchanger evaluation

The increasingly large LMTD value between the heat exchangers from Figure 2 shows favourable heat transfer between the fluids. This is confirmed by the curve where, for example, the water outlet temperature of 38°C corresponds to isopentane inlet temperature of 36°C, and the trend continues for 43 and 41°C, 46 and 44°C etc., showing excellent rates of heat transfer between the fluids.
However, after 46° C, an increase is seen in the temperature difference to 55 and 51° C, and eventually, to 75 and 68° C. This decrease in heat transfer is validated by the decreasing value of the evaporator effectiveness from Figure 3. Though the temperature values are still close, it can be suggested that this low effectiveness may be due to a starved evaporator, which means the mixture of isopentane has higher value of liquid compare to unexpanded refrigerant. As a result, the heat exchanger effectiveness is reduced, resulting in lower heat transfer between the fluids.

**Heat pump evaluation**

Analysis of the theoretical and experimental compressor performance results from Figure 4 yields a poor compressor efficiency of 22.7%. These inefficiencies are indicative of excessive friction and heat loss present in the reciprocating compressor, as well as a sparse discharge to suction ratio, called the pressure ratio.

The system’s energy balance is exhibited in Figure 5, where the heat rate in the evaporator and compression is seen to provide the heat out at the condenser. However, the compressor’s low efficiency leads to poor compression relative to the heat rate into the system, which limits the condenser’s heat rejection capability.

A low heating absorption capacity by the evaporator is also seen compared to the water’s heat supply by water, for the same reasons discussed that caused low heat exchanger effectiveness. This creates a limitation to isopentane’s heat absorption capacity rate in this system. Therefore, decreasing the mass flow rate of isopentane or water might allow higher heat transfer time, in efforts to enhance the heat rate in the evaporator.

The exergy analysis results in Table 2 also indicate that the compressor has the highest exergy destruction amongst all the components, over all evaporator temperatures. In addition to the low efficiency and compression, this confirms that to optimise the heat pump’s overall exergy efficiency, the compressor needs to be optimised.

Despite the compressor’s limitations, Figure 6 shows noticeably high COPh values for the heat pump. The heat pump has an experimental COPh range of 5 – 6.6, with the highest COPh of 6.6 is recorded at lower evaporator and condenser temperature, while at higher condenser and high evaporator temperatures COPh reduces to its minimum value of 5.

As stated in the introduction section, a heat pump has a typical value between 2 and 6, showing that this study’s system has a high COPh. To benefit from the maximum energy conservation benefits and cost-efficiency, the system can be used over lower condenser and evaporator temperatures where the highest COPh, LMTD and heat exchanger effectiveness is seen.

Since indoor heating applications do not require excessively high temperatures, and air temperatures above 46° C is not suited for comfort, the system can be confirmed as feasible for its intended aim and applications.

**CONCLUSIONS**

In this experimental study, the feasibility of using heated water from a solar pond as a thermal energy source for a heat pump with isopentane as the working fluid was investigated.

An internal heat exchanger situated at the lower convective zone of a solar pond circulates hot water to the evaporator of the heat pump, which acts as an external heat exchanger. Isopentane absorbs heat in the evaporator, undergoes compression in the compressor, and flows to the condenser, where it rejects heat to the ambient air flowing over the condenser coils. The heated air exits the heat pump ready for use.

The suitability of using isopentane was confirmed based on its environmental friendly properties of zero ozone depletion and negligible global warming, as well as an energy and exergy analysis that showed an acceptable efficiency.

At lower condenser and evaporator temperatures, a high heat exchanger effectiveness was seen, with high heat transfer and LMTD between the fluids. However, at temperatures above 46° C, the effectiveness decreased, leading to lower heat absorption of the isopentane.

The compressor yielded a poor compression efficiency of 22.7% due to excess friction and heat losses, which limits the condenser’s ability for heat rejection. An exergy analysis identified the compressor as having the highest exergy destruction, confirming that, to increase the exergy efficiency of heat pump, the compression process would require optimisation.

Experimental results for the coefficient of performance of the heat pump (COPh) showed that values between 5 - 6.6 can be expected over evaporator temperatures of 30-80° C. A higher COPh of 6.6 is seen for lower heat source and sink temperature, and lower values approaching 5 at higher temperatures.

As indoor heating and comfort applications do not typically use air temperatures higher than 46° C, this confirms that the system of a heat pump using isopentane as working fluid for a heat pump integrated with solar pond for heating air is feasible for its aim and application.

**REFERENCES**


NIST 2015. U.S Department of Commerce.


