

Novel Design of a Portable Heat Energy Storage Device Adopting a Phase Change Material for CHP and Solar Energy Applications

K. Trapani

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Project Supervisor: Dr. Dean Millar

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Abstract

Phase change materials (PCMs) explore an effective method for thermal energy storage, providing advantages of high-energy storage densities and an isothermal storage process. A device composed of a series of straight fins was developed for use as a portable thermal store adopting PCMs within it. This paper highlights the performance of the device through simulation and testing. Consequently analysis of the thermal energy storage system was investigated for different applications, primarily for micro-CHP. The integration of the device into a domestic central heating system was also examined.

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Nomenclature

<i>A</i>	area (m ²)	<i>Subscripts</i>	
<i>c</i>	specific heat capacity (J/kgK)	<i>g</i>	gas
<i>k</i>	thermal conductivity (W/mK)	<i>l</i>	liquid
<i>l</i>	length (m)	<i>s</i>	solid
<i>L</i>	latent heat of fusion (J/kg)	<i>v</i>	volume
<i>m</i>	mass of heat storage material (m)		
<i>P</i>	power (W)	<i>Greek letters</i>	
<i>Q</i>	quantity of heat stored (J)	<i>α</i>	thermal expansion coefficient (K ⁻¹)
<i>t</i>	time (s)	<i>η</i>	efficiency (%)
<i>T</i>	temperature (°C)	<i>ρ</i>	density (kg/m ³)
<i>V</i>	volume (m ³)		

1. Introduction

The continual increases in fuel prices and warnings of global warming, has instigated efforts to utilise renewable energy sources more effectively and to enhance usage of waste energy.

Waste heat energy is typically a by-product from engine processes; the recovery of this energy could be beneficial in decreasing domestic heating costs, reduce consumption of non-renewable resources and enhance the efficiency of the engine. The surplus heat from motive engines cannot be recovered in a conventional manner, thus the heat would have to be stored onboard and delivered to destination as required (Millar & Huang, 2009). The concept of a portable thermal store was thus inspired.

For a thermal store to be portable, it has to be able to contain relatively high energy within a device which is quite light and not bulky in volume. The thermal energy storage can be either a sensible heat storage (SHS), a latent heat storage (LHS) or a combination of both (Sharma et al 2009). In a SHS, the energy is stored by increase in temperature, while for a LHS the energy stored is dependent upon the energy absorbed/released at phase change of the material within the store.

Total energy which can be stored in a LHS:

$$Q = mc_s\Delta T + mL_{sl} + mc_l\Delta T + mL_{lg} + mc_g\Delta T \quad [1]$$

The thermal cycle involved in a LHS compared to that in a SHS ensures a higher energy storage capacity, due to the added energy from the latent heat at phase change. This makes the use of a phase change material (PCM) ideal in a portable thermal store as they can “store 5-14 times more

heat per unit volume than sensible heat storage materials such as water, masonry, or rock” (Sharma et al, 2009).

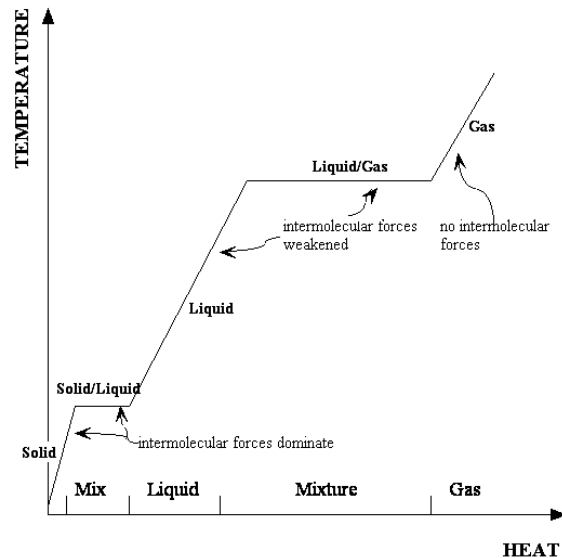


Figure 1.1 - Phase change diagram of materials

Hence by using such a device the heat could be stored at the source and then delivered to the where the heat demand is. The heat demand explored in this research was for the domestic sector, since it is a major heating load consumer.

Expanding upon the requirements of such a system, a device has been designed for this particular application. This research paper will consequently determine its suitability through assessment of its performance, by simulation and testing of the device.

1.1 Literature review

PCMs have been investigated in the past years for their suitability in thermal storages and an in-depth review, outlining more than 150 materials, has been conducted by Zalba et al (2003).

Various encapsulation methods for extraction and storage of heat within a PCM have been explored,

amongst which are: packed bed PCM capsules (Regin et al, 2007), basic cylindrical and rectangular stores (Zivkovic & Fujii, 2000), cylindrical finned PCM thermal store (Groulx & Ogoh, 2009), and wire finned PCM heat exchangers (Bach & Haije, 2001).

Most of the applications for which LHS systems were explored, was for space heating/cooling in buildings, as stationary or integrated units. This can be indicated from the numerous reports published for this sector – amongst which are those written by Ravikumar & Srivinasar (2008), Ip & Gates (1999), Butala & Stritih (2009), and Zalba, Jose, Cabeza, & Mehling (2003). The main heat sources outlined in these papers were either from surplus solar thermal resources or from ventilation heat exchange in a building.

Automotive engines' surplus heat has been explored as mainly a source of heat supply for thermoelectric generators (Hsu et al, 2010) and (Thacher et al, 2007). It was first explored as the source for a thermal store in the document '*domestic air space and hot water heating using waste heat from automotive engines—the mobile micro-CHP concept*' (Millar & Huang, 2009).

Following upon this concept an actual design for a portable thermal store was investigated. Subsequently its performance will be assessed and hence its viability as a portable heat storage device.

2. Applications for the device

The device was mainly targeted as an incorporated unit within a domestic heating system, in which it displaces some of the heating load with the heat energy stored within it. Thus the heat sources from which the device takes up the heat have to be

linked to the domestic residence. The major heat losses/surplus sources identified were:

- Micro CHP (in the form of automotive vehicle engines)
- Surplus solar thermal heat
- Ventilation exhaust
- Drainage discharge

Hence the thermal store could be applied to any of the above applications; although micro CHP and solar heat energy would be the ideal candidates since in general have a relatively high grade heat source. Micro CHP is though the application which greatly requires the particular feature of portability in the thermal store, since the heat cannot be extracted in conventional ways as could be for the other applications.

The advantage that the portable thermal store has over a conventional one, apart from it being portable, is that it could be relatively quickly "charged" and "discharged". This property would be advantageous for some of the other applications and would thus consider the device viable for integration with other particular applications.

3. PCM for the portable LHS

3.1 Review of LHS materials

For an effective thermal store the PCM chosen for the device would ideally have (Ravikumar & Srivinasar, 2008):

- High latent heat of fusion per unit mass
- High specific heat capacity
- High thermal conductivity
- High density

Through reviewing a list of phase changing materials, the suitable PCM could be selected for the heat store.

3.1.1 Phase transition for PCM

There are four possible phase transition in a material, from gas to liquid, solid to gas, solid to liquid or solid to solid. By comparing the latent heat of fusion and volume change properties of each phase transition change, Table 3.1, the most appropriate phase transition for the device was determined.

Table 3.1 - Phase transition properties

	Latent Heat of Fusion (L)	Volume Changes (ΔV)
Gas – Liquid	↑	↑
Solid – Gas	↑	↑
Solid – Liquid	↑	↓
Solid – Solid	↓	↓

↑ High L High ΔV
 ↓ Low L Low ΔV

The appropriate phase transition would be one which has a high heat of fusion, thus can store a large amount of thermal energy, and one with low volume changes on phase change – hence not requiring excess volumes for expansion. On this basis the solid – liquid phase transition was selected as being suitable for the required purpose.

3.1.2 PCM adopted for device

As can be seen from the figure below there are two main categories within the solid-liquid transition.

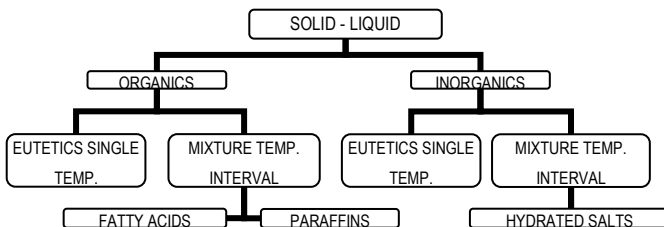


Figure 3.1 - Classification of solid - liquid PCMs
(Sharma et al, 2009)

The main advantage of inorganics over organics is that they have a greater phase change enthalpy.

While the advantage of organics is that they are not corrosive materials, have low or no undercooling and are chemically and thermally stable (Zalba et al, 2003). Due to the nature of the LHS, it being portable, priority has to be given to a material which is chemically and thermally stable. Ideally, also, it should not be corrosive and inflammable. Thus it would seem that organic materials have the appropriate thermal, physical and chemical properties required from the PCM material.

When “selecting a PCM for a particular application, the operating temperature of the heating or cooling should be matched to the transition temperature”, and so the PCM selected should have a melting temperature around 40°C - 60°C (the temperature required for space heating). Thus due to this temperature range, its high heat of fusion, stability in heat cycling and economic reasons the material selected for the thermal store was paraffin wax.

3.2 Thermophysical properties of paraffin wax

In order to establish the actual properties of the device's PCM, a sample of the paraffin wax selected was tested. The sample of paraffin which was utilised for the testing was a batch of ordinary commercial paraffin wax used in candle making.

The PCM was tested for its performance with a heating load, in order to be able to establish its basic properties. The governing equations highlighting the heat transfer in the PCM are:

$$Q_s = mc_s \Delta T \quad [2]$$

$$Q_{sl} = mL_{sl} \quad [3]$$

$$Q_l = mc_l \Delta T \quad [4]$$

These describe the relation, between the increase in temperature and the change in enthalpy of the material - or the heat stored in the PCM at those temperatures.

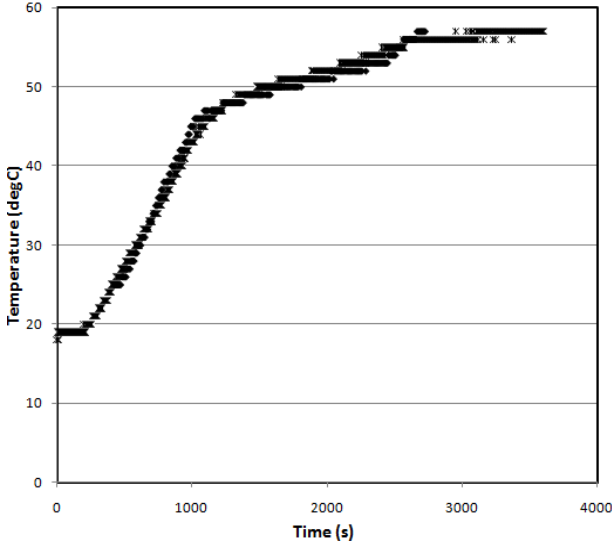


Figure 3.2 – Heating characteristics of PCM

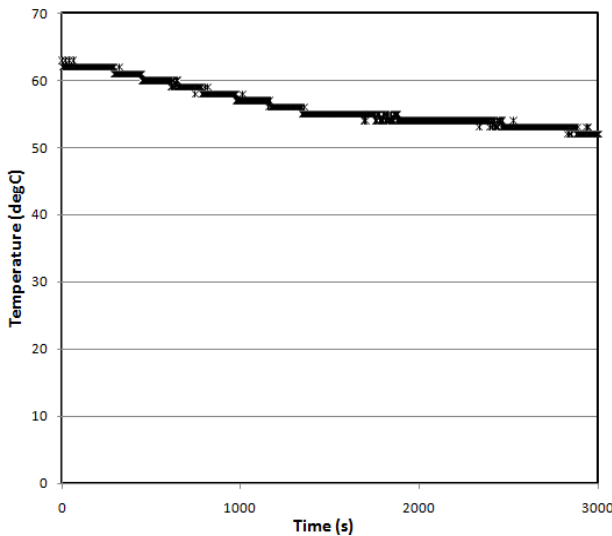


Figure 3.3 - Cooling characteristics of PCM

The heat stored, Q was calculated through the monitoring of the power input, of the heating element.

$$Q = P.t \quad [5]$$

Thus the latent heat of fusion and specific heat capacity, at solid and liquid heating/cooling could be calculated for both the heating and cooling of the

PCM. These values were then normalised through iterations between the different cycles, and so the thermal properties of the PCM could be derived.

Table 3.2 - Thermophysical properties of paraffin wax

Density, ρ (solid)	896 kg/m ³
Density, ρ (liquid)	820 kg/m ³
Melting temperatures, T	328K – 330K
Specific heat capacity, c_s	3412 J/kgK
Specific heat capacity, c_l	4466 J/kgK
Latent heat of fusion, L	197 kJ/kg

The thermal conductivity was taken to be 0.29W/mK and 0.21W/mK (Buddhi, 2001), for the solid and liquid phase respectively. While thermal expansion, α_v , was taken to be 0.00011K⁻¹ (Delta Dirac Consultants Limited, 2010).

4. Design of device

4.1 Requirements of LHS device

The thermal storage device was designed as a rectangular modular unit, so to allow increased portability. Also this would enable the modular units to be easily stacked to any thermal heat storage capacity required.

The “charging” and “discharging” period of the device would have to be optimized to enable fast heating and cooling of the PCM. This is so to ensure that the device would be able to give out the heat almost immediately and also provided that the heat source is large enough, it would be able to store the heat quickly.

Lastly the PCM containment design has to be kept as light and compact as possible, in order to meet its portability requirements. Each modular unit was designed to be 15kg in weight, ensuring that each modular unit is portable.

4.2 Geometry

The specific design studied for the portable LHS was composed of a series of parallel aluminium fins with copper piping running through it. It has basic dimensions of 54mm, 98mm and 310mm, in width, height and depth respectively (excluding the volume occupied for insulation). The copper piping running through it has a diameter of 4mm, while the aluminium fins (64 in number) are 0.5mm thick and they are spaced at 37mm from each other, as can be seen in Figure 4.1. A notch was cut through the aluminium fins, at the bottom centre, to facilitate the addition of the PCM into the device. This model designed is a 1:3 scale model of the 15kg modular unit.

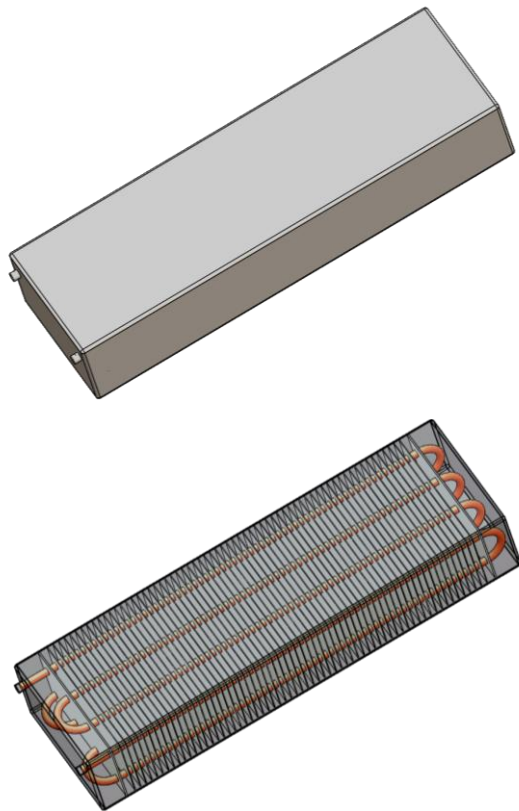


Figure 4.1 - Thermal storage device

The rate of heat transfer is dependent on a number of factors, as can be suggested in the following equation.

$$\frac{\delta Q}{\delta t} = -kA \frac{\Delta T}{l} \quad [6]$$

Thus this would imply that the rate of heat transfer is directly proportional to the area and inversely proportional to the thickness of the fins. Hence by maximising the area and minimising the thickness of the fins, an increase in heat transfer would be expected - this was investigated and confirmed by Groulx and Ogoh (2009). Thus the device was designed to reflect these characteristics.

4.2.1 Device components

The device comprises of a number components, them being:

- Internal piping
- Internal fins
- PCM containment
- Insulation
- External casing

All of the materials selected for the device had to have a low density, so to keep the weight of the device at a minimum. Thermal conductivity of the materials was another important feature in the material selection process, since the piping and fins had to be highly conductive while the PCM containment, insulation and casing had to ideally be non conductive.

Table 4.1 - Components specifications of model

Component	Material	Density <i>kg/m³</i>	Volume <i>m³</i>	Mass <i>kg</i>
PCM	Paraffin wax	986	0.001309	1.173
Piping	Copper	8816	0.000049	0.437
Fins	Aluminium	2558	0.000164	0.421
Internal case	Steel	7795	0.000110	0.859
Insulation	Reflective foam	150	0.000765	0.115
External case	Plastic	1200	0.000244	0.293

Therefore the total mass of the model would be equal to $\approx 3.3\text{kg}$ and have a volume of $\approx 0.017\text{m}^3$ (including the insulation layer and external case).

When sizing up the prototype model to the modular unit, which weighs, 15kg the mass of the external materials would be less compared to the internal. Since the unit would be essentially the summation of a number of smaller internal units, with one common external casing and insulation layer.

5. Computational modelling of the thermal store

The purpose of the simulation was to ensure that the device unit performed as it was designed to and to highlight the heat transfer within the device, ensuring that no stagnation regions occurred. Also by producing temperature graphs of different points within the device, calculations could be made for the power put in and out of the unit, and eventually its theoretical efficiency could be estimated.

5.1 Use of COSMOS

COSMOS is a flow simulation application within the Solidworks modelling software which was utilised for analysing both the thermal and computational fluid dynamic performance of the device.

5.1.1 Enthalpy formulation of simulation's PCM

The simulation was based upon the following assumptions, that:

- 1) the paraffin wax is homogenous and isotropic
- 2) the heat is transferred only by conduction
- 3) the simulation is time dependent

The simulation software did not account for thermal absorption/release due to phase change. Thus the

paraffin wax material designed in Solidworks had to consist of a series of sensible heating stages, as indicated in Figure 5.1.

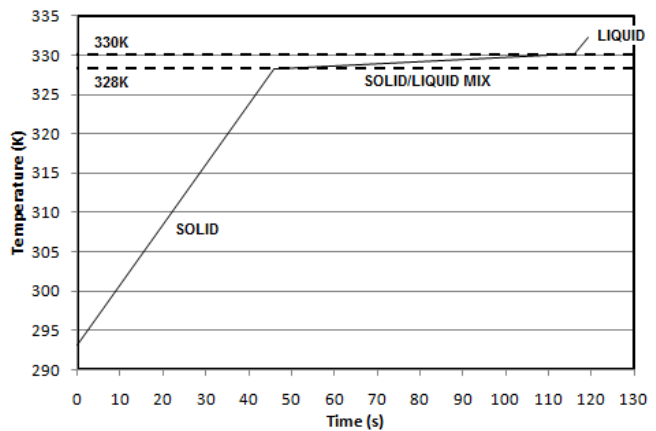


Figure 5.1 - Modelled heating process for the paraffin wax

The specific heat capacities of the paraffin wax were modified in the following ways:

$$c_s = 3412 \text{ J/kgK} \quad \text{for } T < 328\text{K}$$

$$c_{sl} = 98587 \text{ J/kgK} \quad \text{for } 328\text{K} < T < 330\text{K}$$

$$c_l = 4466 \text{ J/kgK} \quad \text{for } T > 330\text{K}$$

Although this does not follow the normal laws applying to phase change (that is phase change at constant temperature), it does provide the same input and output of heat energy in the material. In practice, as could be seen in Figure 3.2, the actual change experienced during phase change was not over one single temperature. Hence the simulation would transfer heat at the periods in which actual transfer would be expected.

The melting interface of the paraffin wax was accounted for in the simulation, by setting the modelled software with a melting temperature of 328K.

5.1.2 Initial and boundary conditions

The flow simulation was taken as being a transient analysis, so accounting for the time dependency of the simulation, and importantly, heat transfer was allowed between solids. With regards to the simulation's mediums, water and copper were selected as the fluid and solid. This implied that the water flowed through a body which was contained by copper, i.e. the copper piping. The outer walls of the system were taken as being adiabatic; thus the system was an ideal one since it did not have any losses.

For the device “charging” simulation the boundary conditions were taken as being:

- Fluid inlet of 6.5l/min or 0.108kg/m³ at 333.2K
- Fluid outlet subjected to normal environmental conditions

The initial conditions for the simulation were taken to be the same as the normal environmental conditions (293.2K and 101325Pa).

5.2 Simulation results

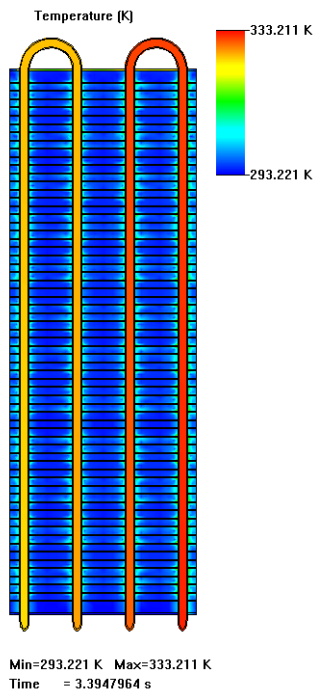
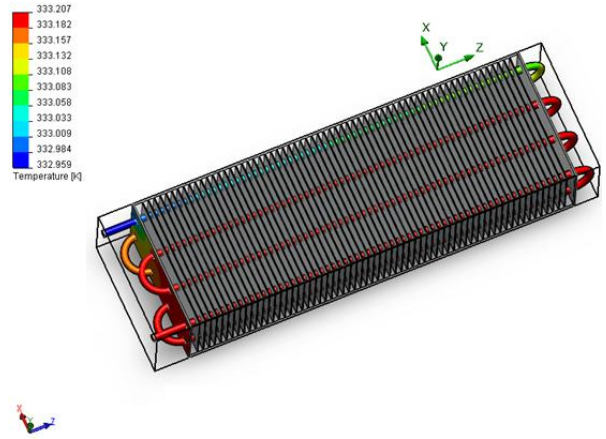


Figure 5.2 - Initial heat transfer on start of simulation



Local parameters

Parameter	Minimum	Maximum	Average	Bulk Average	Volume [m ³]
Solid Temperature [K]	332.985	333.201	333.172	333.172	0.00130864

Figure 5.3 - Thermal performance of device after simulation

The maximum temperature of the PCM model device was simulated to be at 332.172K, while the maximum temperature of the inlet fluid was at 333.2K. The time required for the device to reach saturation temperature was 119.106s. From these simulated values, the maximum efficiency of the device as a heat exchanger could be calculated.

$$\eta = \frac{Q_{stored}}{Q_{delivered}} \quad [7]$$

After t = 119.1s, Q_{stored} = 381718.8J from eq. [1]

Q_{delivered} = 387104.1J from eq. [4]

Thus, this implies that the theoretical “charging” efficiency of the device is 98.6%. Assuming that the device has the same efficiency during “discharging”, then the overall theoretical efficiency of the device would be 97.2%.

Therefore, the maximum stored power in the model is of 381718.8J, equivalent to 106W. Thus the power per m³ and per kg is 66kW and 35W respectively.

6. Experimental testing of device

An experimental set-up for the analysis of the heat transfer of the device was installed as shown in Figure 6.1 & Figure 6.2.



Figure 6.1 - Experimentation test rig

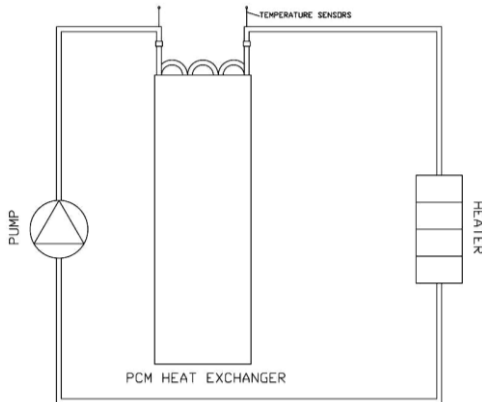


Figure 6.2 - Schematic diagram of test rig

Two experiments which were conducted consisted in the analysis of the heating and cooling of the model – so to be able to determine the actual heat which would be extractable from an eventual system.

The data monitoring the performance of the device consisted of a series of thermocouples at the inlet, outlet and inside the device (within the paraffin

wax), also the power supplied to the system was recorded with the temperatures. This data acquisition was performed through the utilisation of an 8 input port data logger, using thermocouples.

6.1 Device “charging” test

The “charging” test refers to the heating cycle of the device, in which it takes up waste or surplus heat (it being from CHP or solar sources). The grade of heat, does not impact the device’s performance unless it is below the melting temperatures of the PCM or it is too high – thus creating stress on the device from thermal expansion of the PCM.

The inlet from the heater was set at a constant 65°C. a temperature which would be close to the characteristics of the CHP and solar heat sources. While the mass flow rate of the water was 6.5l/min or equivalently 0.108kg/s, according to the pumping capacity of the pump used.

6.1.1 “Charging” test results

The heater power input was controlled by a thermostat, thus the power was turning on and off as necessary. Using a power consumption monitor, the power put in the device could be measured – $P_{in} = 0.1655\text{kWh}$, as calculated from Figure 6.3.

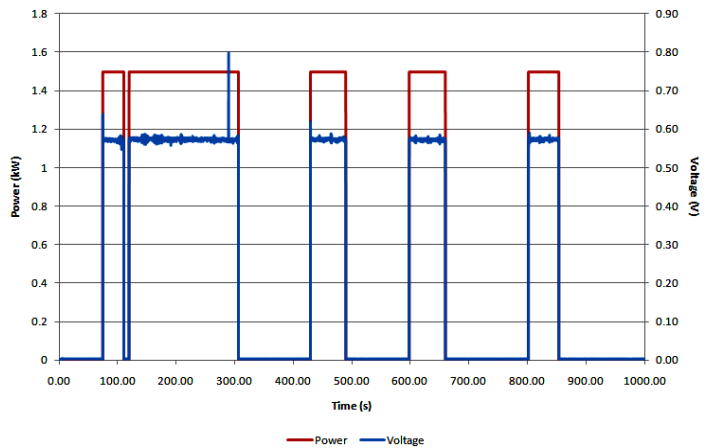


Figure 6.3 - Power consumption monitor

The water was preheated to 65degC before the experimentation was initialised, thus the water had some energy stored within it already. The testing was ended with the water still at that temperature, so it would be correct to assumed that the actual energy delivered to the device is the extra energy supplied during the testing, that of 0.1655kWh.

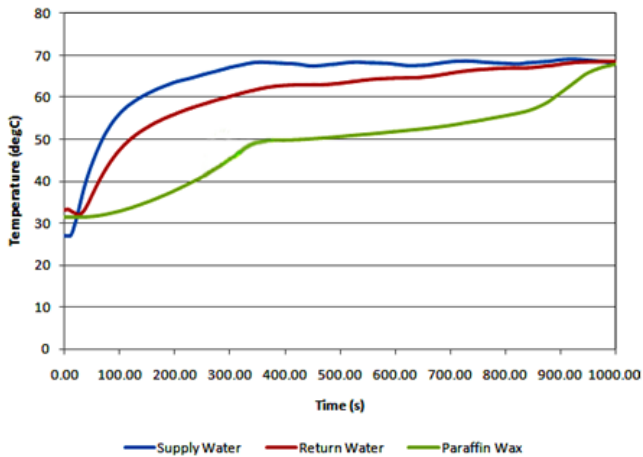


Figure 6.4 - Graph of "charging" temperatures

The total time taken for the charging of the device to 68.4degC was 1018.3s, at an average room temperature of 21.3deg C. The total heat contained within the store is 318.7kJ, eq. [1] and the heat put in is 595.8kJ (or 0.1655kWh). Hence the "charging" efficiency, using eq. [7], is 64.1%.

6.2 Device "discharging" test

The "discharging" test was conducted to establish the practical potential energy extractable from such a device. This was done by considering a closed system in which water, initially cold, was pumped through the device till the device's temperature and that of the water were approximately the same.

6.2.1 "Discharging" test results

Cold water was pumped through the device in order to extract the thermal energy within the heat store.

The water was initially inputted at 28degC and the time taken for the water reach its saturation temperature is 1037s at a temperature of 33.9degC. The quantity of water which was pumped around the device was equivalent to 10 litres.

Thus this implies that the power extracted from the device is 247033J, eq. [4], which can be converted into 0.0686kWh. So the efficiency of the whole system could thus be calculated to be 41.5%, eq. [7], while the "discharging" efficiency is 64.7%.

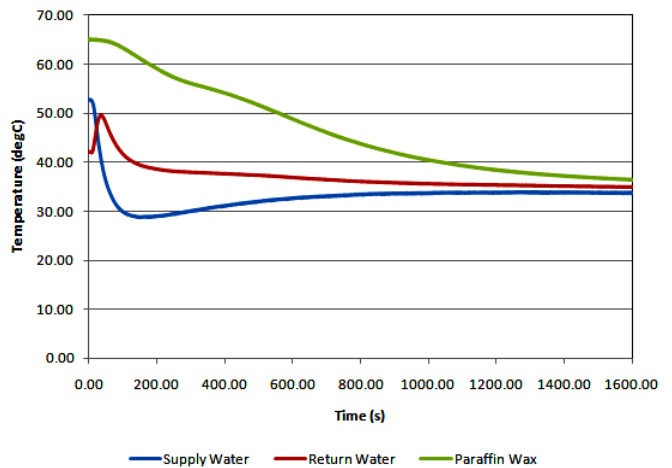


Figure 6.5 – Graph of "discharging" temperatures

7. Integration of the device into a domestic central heating system

For heat energy to be extracted from the portable device it has to be connected to a heating system which utilises fluid transfer. Since the thermal store relies upon heat transfer it requires a medium into which to transfer the heat. Such an ideal system within the domestic sector is the water central heating system (WCHS).

A WCHS typically involves the circulation of heated water (from a boiler) to radiators through pipes, as can be seen in the following schematic.

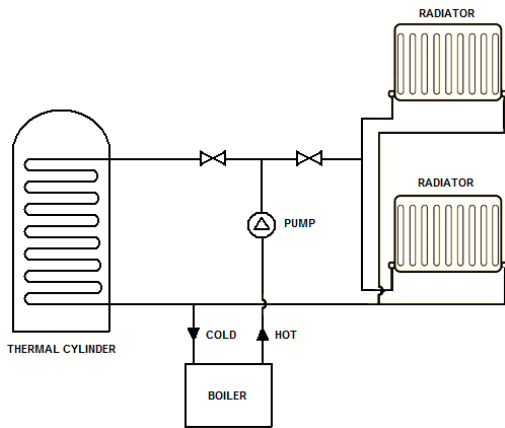


Figure 7.1 - Simplified schematic of domestic central heating system

Most systems are pressurized at between 1 and 1.5 bar, thus the integrating system (between the device and the WCHS) has to incur minimal pressure losses to the system. The WCHS, as shown in Figure 7.1, already has a circulating pump within it, thus the addition of the device would not require its own pump. Effectively the addition of the device could be considered as having the same effects as the inclusion of a small radiator onto the system.

The heating device within a typical WCHS is a boiler which is able to monitor the system's temperature and keep it constant. This monitoring generally consists of a couple of thermocouples at the outlet and inlet of the boiler. Thus ideally the connection of the device to the system would be just prior the boiler's water inlet. At this point any increases to the water temperature at the inlet would be accounted for and the heating load would be adjusted by reducing the heat put in the system to maintain a constant temperature at the outlet.

7.1 Components required for the integrated system

Since the system is a pressurized one, it is important that the input and output are capable of

maintaining the pressure. This could be maintained by installing appropriate fittings at the device's inlet and outlet.

7.1.1 Input and output pipe fitting

The fittings designed to fit the input and output, consist of a K-part and P-part as indicated in the designs below. (A variant of the below design, is available commercially for industrial purposes which is not well suited for the domestic application required).

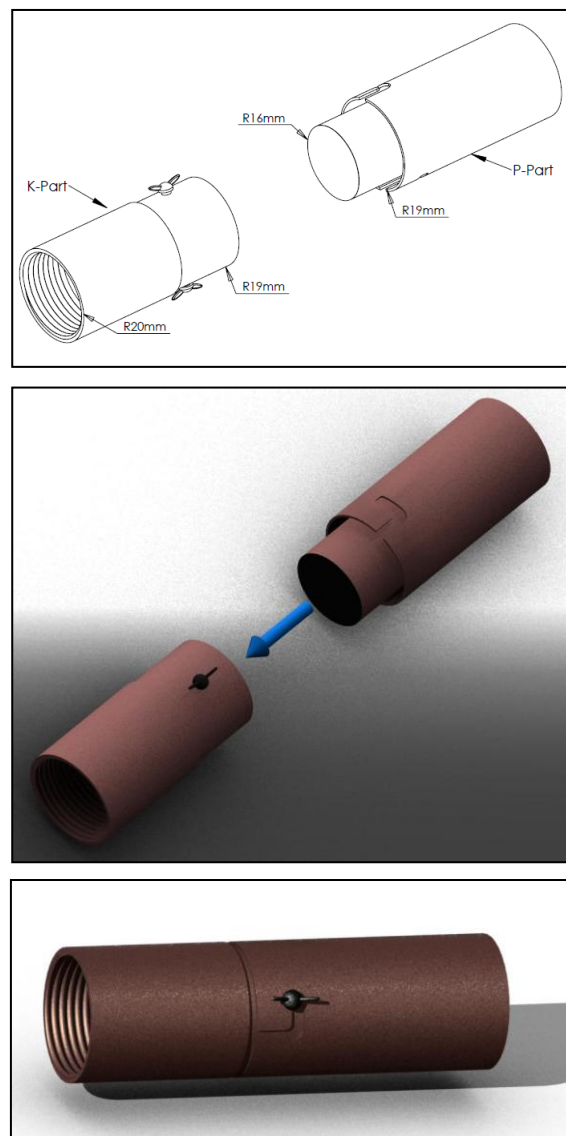


Figure 7.2 - K-Part and P-Part models of components

The parts were designed for easy fit with existing pipes. As components, they would be easy to connect together and also they would ensure minimum water and pressure losses. The P-part was designed to connect at the device's inlet and the systems return pipe, while the K-part would connect at the device's outlet and the systems input to the device.

The fittings maintain the water within through a flap which is held by a spring to oppose the flow of water in that piping section. The fitting would work as shown in Figure 7.2. The factor which would have to be considered in depth, for manufacturing of the components, is stiffness constant of both springs attached to the flaps. For the K-part the stiffness constant would have to match the manual force applied by the P-part on connection, while for the P-part it would have to match the pressure of the water in the WCHS.

7.1.2 Connection piping

Provided that the "charging" and "discharging" location of the device is kept the same, then the retrofitting would be quite simple and similar to common household plumbing work. If the location was a changing one, then the connection would have to be one which could adapt to this and so it would have to be a flexible connection. Insulated silicon piping would be ideal for this application, since the material is flexible and also it has a low thermal conductivity. Whichever connecting pipe is used for the integrated system suitable fittings would have to be installed at the inlets and outlets of the connecting pipe to the device. The fittings would so allow it to easily connect to the device or itself (in order to complete the loop of fluid flow).

8. Heat extraction from micro-CHP (automotive engine)

From the actual fuel which is combusted in the automotive engine, only a small percentage is converted into useful automotive power. The rest of the energy is mostly converted into different kinds of heat losses; this breakdown of the fuel combustion process is highlighted in the following figure.

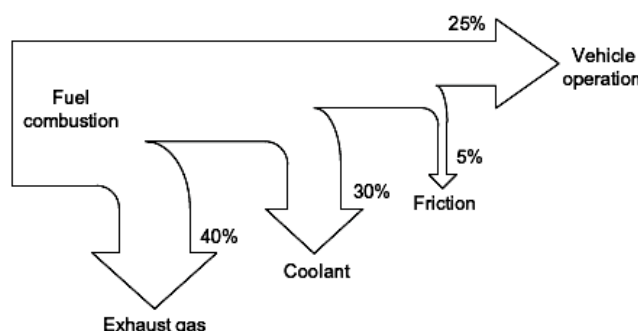


Figure 8.1 - Energy flow in an internal automotive combustion engine

(Yu & Chau, 2009)

The cooling process although contains a lower grade heat, compared to the energy of the exhaust gas, it has almost all the infrastructure in place for connection to the thermal storage. If the heat was to be extracted from the exhaust pipe a heat exchanger would have to be installed. This is unless the flue gas is used as the medium through which heat transfer occurs.

Due to the nature of the gas, it being a post combustion gas, this would be highly inadvisable. This would lead to deposition of flue particles onto the internal piping of the device and possible clogging up of the entire device. Also the flue deposited could contaminate the water within the water system once it is connected to the WCHS.

Hence on the cooling process is the heat source to be considered in the application of automotive micro-CHP.

8.1 Integration with an automotive

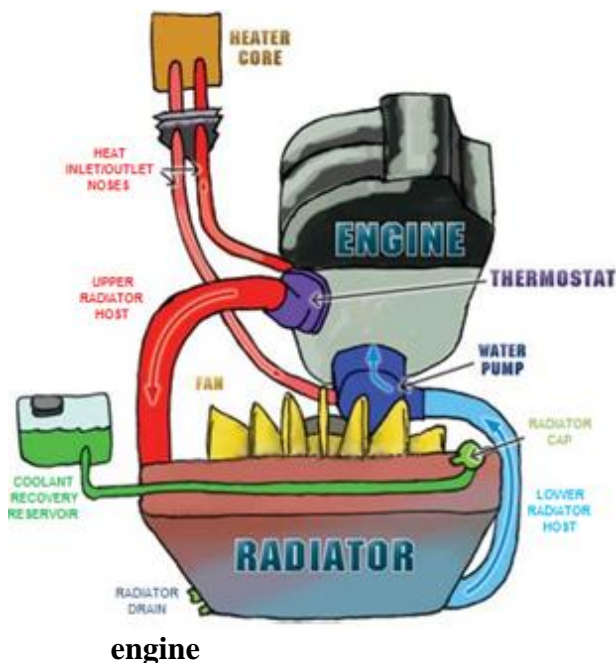


Figure 8.2 - Cooling process in an automotive engine
(Duae Manus, 2009)

The maximum temperature in the cooling process, as demonstrated above, is at the water inlet to the radiator. This would suggest that the maximum heat which could be extracted from that system is at that same point. Thus the retrofitting pipe would have to be connected to the device at that point, while the outlet of the device would have to be connected to the input of the radiator. The piping which could be used for this retrofitting purpose could be made of the same material used for the radiator inlet and outlet pipes, so ensuring that the retrofit would be able to withstand the high temperatures.

The space requirements of the device would be considerable and these could only be

accommodated by utilisation of the empty space at the rear of the car in the boot. Thus the connecting pipes would have to be laid underneath the vehicle and through the boot (and back again to complete the loop for fluid flow).

8.2 Heat storage capacity extraction from a typical automotive vehicle

The fuel consumption of an automotive engine is dependent upon the distance which the car is required to travel and the weight which it has to carry. The EU target in 2005, for gasoline cars fuel consumption was at 6.8litres per 100km (Schipper, 2007), or alternatively 0.068litres of gasoline per km travelled. The energy content of gasoline is 48.5MJ/kg (H A & S, 2004), equivalent to 35.4MJ/litre. Thus the typical energy consumption per km is 2.4MJ.

Of the initial 2.4MJ required for combustion, 30% is transferred to the cooling water, implying that 722.16kJ/km is available for “charging” the thermal heat store. For a modular 15kg unit the mass of paraffin wax would be 6kg and approximately 40% of the total mass of the device. Therefore the heat storage capacity would be approximately 1600kJ, eq. [1].

From the device testing, the “charging” efficiency was calculated to be 64.1%, thus the energy inputted would be 2500kJ. Hence the distance which the vehicle is required to travel to store that amount of energy is 3.5km. Thus this would be the optimal travelling distance for the “charging” of a 15kg unit. A further distance would have to “charge” larger devices (containing more modular units) as can be demonstrated in Figure 8.3.

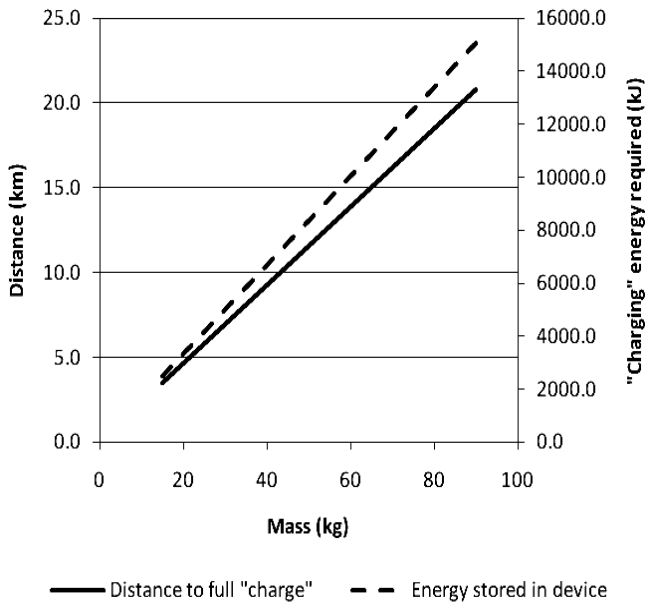


Figure 8.3 - "Charging" time variation with increase in device weight

8.3 Analysis of the portable thermal store for micro-CHP

8.3.1 Cost of the portable thermal store

Prior to establishing any cost analysis for systems which include a portable thermal store, the costing of the actual device has to be estimated. This was done using commercial values as highlighted in the following table.

Table 8.1 - Costing of device

Component	Material	Mass kg	Material Cost £/kg	Material Cost
				£
PCM	Paraffin wax	5.865	1.00	5.87
Piping	Copper	2.185	8.30	18.14
Fins	Aluminium	2.105	9.00	18.95
Internal case	Steel	4.295	3.08	13.23
Insulation	Reflective foam	0.575	/	1.00
External case	Plastic	1.465	/	5.00
Labour Cost				£20.00
Total Cost				£82.17

If the device was to be manufactured commercially, its cost would be considerably less. Thus a retail cost of £41.09 (half the price calculated in the costing table), is taken to be the cost for a 15kg modular unit.

8.3.2 Economical analysis of the system

An economic analysis essentially models the cashflow expected annually from a project, after taking into consideration the revenue of the project, operating costs and the capital expenditure.

The revenue from this system would be in terms of kWh saved from the domestic heating load. Since the heating is generally attributed to a gas or electric boiler, this would be in terms of units of gas or electricity saved. The operating costs are the costs associated with the transporting of the device and some general maintenance (the cost of pumping is not regarded as being an operating cost since water is being circulated around the system either ways).

The transporting costs depend on the extra fuel consumption for the extra load over a typical commuting distance, which in the UK is around 30km daily (Independent, 2010). For which the increased fuel consumption is on average around 0.007litres/km for each 100kg increase (Reynolds & Kandlikar, 2007). This could be considered as being the operating cost associated with the portable device.

Hence the operating margin for such a system would be the heat savings less the extra costs required to transport the thermal store. As shown in the following cashflow model.

Table 8.2 - Cashflow model for a portable thermal store integrated to micro-CHP and displacing gas heating

Life of Project		20 year		No of Units		1			
Revenue				CAPEX					
Energy stored per commute		1607	kJ	Cost of Device		41.09	£		
Energy delivered per commute		1040	kJ	Domestic Integration		150	£		
Commutes yearly		260		Micro-CHP Integration		100	£		
Yearly	Energy saved	75.1	kWh	Total				291.09	£
	Cost per kWh	4.1	p/kWh						
	Cost Savings	3.1	£						
Increase fuel consumption		0.0021	litres/commute						
Cost of fuel		1.19	£/litre						
O&M Cost		0.6	£/annum						

End of Year	0	1	2	3	4	5	6	7	8	9	10
Revenue		3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1
Operating Costs		0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Operating Margin		2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4

Capital Expenditure	291.09											10.00
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Cashflow	-291.09	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3
Cumulative Cashflow		-288.7	-286.2	-283.8	-281.4	-278.9	-276.5	-274.1	-271.7	-269.2	-276.8

End of Year	11	12	13	14	15	16	17	18	19	20
Revenue	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1
Operating Costs	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Operating Margin	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4

Capital Expenditure											
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Cashflow	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3
Cumulative Cashflow	-274.4	-271.9	-269.5	-267.1	-264.6	-262.2	-259.8	-257.4	-254.9	-252.5

Payback Period	93.9	years
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Using the above model, a sequence of economic models were compiled for the two different scenarios – scenario 1: gas heating displacement and scenario 2: units of electricity displacement, with a varying number of modular units being utilised.

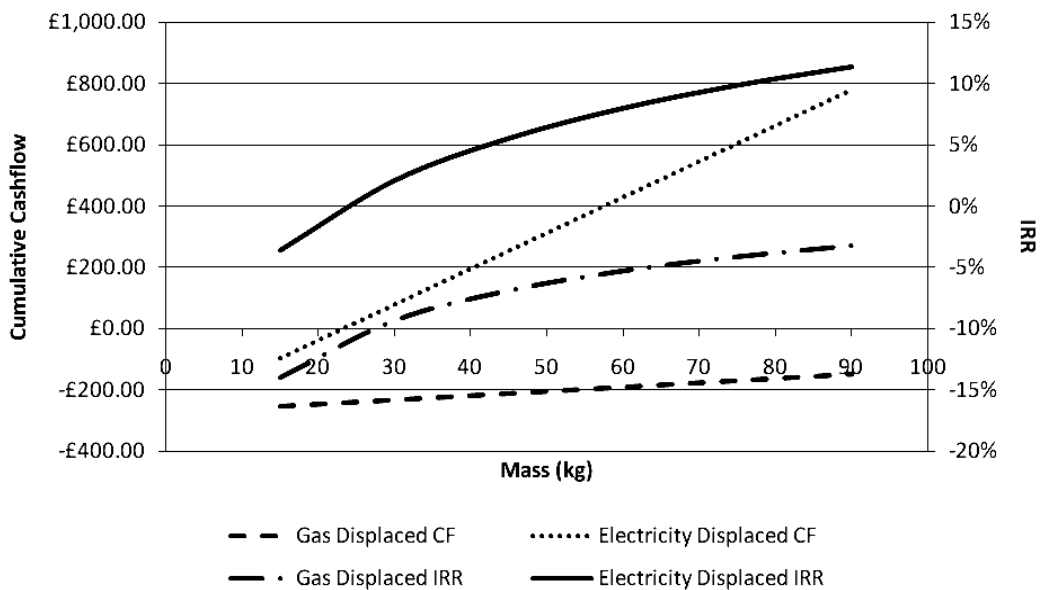


Figure 8.4 - Graph of IRR and Cashflow for the two scenarios at different device masses

As can be noticed from Figure 8.4, at the mass of the thermal storage device increases the project appears to become more economical with higher cumulative cashflows and internal rate of return (IRR) being recorded. This is due to the capital expenditure (CAPEX) to energy saved ratio decreases with an increase in mass, consequently implying that the higher the mass of the storage the greater is the potential for savings. Only up to six units were considered in the study, since this is the maximum amount of units which could be stored easily in the rear of a typical vehicle.

The data recorded for scenario 1 show that the IRR would not break even with any combination of the modular units, highlighting a negative cashflow throughout the series and a minimum payback period of 26.9 years. The lifetime of the device and the project was estimated at 20 years, so not even with 6 modular units contributing to the domestic load would the project be able to pay its initial cost back in savings. With regards to scenario 2, the economics appear more attractive with an IRR varying from -4% to 11% for the 15kg and 90kg devices respectively, with corresponding payback periods ranging from 26.9 to 7.7 years.

The economic models depend upon on a constant price for either the gas or the electricity and do not reflect the trend in increasing cost prices. With an increase in the fuel prices, the revenue would be somewhat greater while the operating costs would also reflect this increase in fuel price.

8.3.3 Carbon benefits of the system

The main function of the device is to displace a fraction of the heating load. Therefore dependent

on the source of the heating load, the respective carbon dioxide displacement will be made – provided that the source originates from a fossil fuel. The displaced heating load in either case is 169.5kWh per 15kg modular unit.

Table 8.3 - Annual carbon savings through a portable thermal store integrated with micro-CHP

No. Of Units	Natural Gas		Electricity	
	CO2 Emissions kg/kWh	Carbon Savings kg	CO2 Emissions kg/kWh	Carbon Savings kg
1	0.194	32.9	0.527	89.33
2	0.194	65.8	0.527	178.65
3	0.194	98.6	0.527	267.98
4	0.194	131.5	0.527	357.31
5	0.194	164.4	0.527	446.63
6	0.194	197.3	0.527	535.96

9. Discussion

The scope of the research, conducted for the design of a portable thermal store, was to establish its heat storage performance and on this basis determine potential energy savings mainly from micro-CHP. The potential savings were considered in terms of heat consumption savings within the domestic sector, commercial sectors could possibly be also considered.

The crucial component within the design of the device, apart from the actual design, was the PCM contained within it since this is where the energy is essentially stored. For this device, the paraffin wax was chosen primarily because of its economic properties although thermally it suites the requirements of the thermal store. Commercial phase change materials with a higher latent heat of fusion would provide a greater energy storage capacity, although they might be economically less viable.

During the paraffin wax testing the phase change diagram obtained for the heat curve, Figure 3.2, did not show a distinct phase change transition as could be noticed from Figure 6.4. This reflects the performance of the heat transfer between the two testing procedures. Implying that due to the increased thermal conductivity between the heat source and the paraffin wax a better characteristic curve could be obtained in the thermal store device. The latent heat of fusion obtained from the paraffin wax on analysis of the results for the “charging” of the device was c.220kJ, which implies an error of approximately 20kJ when compared to the calculated values in Table 3.2. Thus an error of $\pm 5\%$ can be taken to apply to the calculated thermophysical properties of the PCM.

From the simulation conducted, the potential energy which could be extracted from the device was calculated and the timed performance of the “charging” and “discharging” was estimated. Further simulations could be conducted with a varying number of fins to optimise the heat transfer according to the time availability for heat transfer (in the case of micro-CHP applications, it is the average time for each commute). Also for better simulation results, modelling of the simulation software would have to be conducted to overcome issues related with phase change boundaries.

From the testing results obtained for the model, the “charging” and “discharging” efficiencies were approximately the same. This indicates that the result obtained from the testing are constant and also that the same rate of heat transfer applies for either heating or cooling cycle of the device. This confirms eq. 6 in practice, that is $Q = \eta \times kA\Delta T/l$ –

where η is the efficiency for either the “charging” or “discharging” of the device. The calculated energy extractable from a thermal heat store is 22.9Wh per kg, with an overall energy conversion of 41.5%. It was noted that the device was not properly insulated from the top and bottom, if these sections had been properly insulating the heat losses could be expected to decrease by 1/9th (on basis of the surface area of these section compared to the rest of the sections). Hence the expected overall energy conversion in a commercial device would be expected to be approximately 48%, delivering 26.5Wh per kg.

The device is designed to be integrated within a domestic water heating, such a system is ideal because it relies upon a fluid to transfer the heat and also it contains the suitable network (and equipment - i.e. the pump) to deliver the heat. Since the network is made up of ordinary piping, the retrofit which would apply would be relatively simple plumbing. The actual connection of the device to the system might be the most problematic if the appropriate fittings are not used. That is fittings which ensure that the system’s pressure is kept constant and also that no water is lost from the system.

For this purpose the K and P-part fittings were designed. Other similar fittings are commercially available on the market, but these fittings are not designed to allow flow the relatively low pressure of a domestic system. Further study would have to be conducted mainly on the stiffness constant of the springs within the fittings, to ensure that they perform their function well while not depressurising the system or leaking. Without these fittings the

device would not be able to be connected as a portable device to any system. Possibly commercial fittings could be modified to suit the requirements of the domestic heating system.

The highlighted applications for the device were for micro-CHP and solar system. Since a solar system would be stationary and generally connected onto the domestic heating system, the integration of the device would be the same as that for a conventional stationary thermal store. With the application of micro-CHP the integration is somewhat more complicated due to the mobility of the application.

The size and consequently the mass of the device is somewhat an issue when maximising the number of units coupled for heat extraction in a micro-CHP application. This is due to the fact that the space in a vehicle is limited and approximately 0.5m^3 would be available for storing of the LHS. Hence up to six units could be fitted in the vehicle, each unit occupying c. 0.075m^3 . The two heat energy saving scenarios which were considered, when determining the economical viability of the device, was with a gas and electrical boiler integrated in the domestic heating system. The operating costs of the device were taken as being the extra costs associated with the transportation of the extra mass within the vehicle. While the lifetime of the project was estimated to be 20 years, this would be a reasonable period for the device to function in, but the main issue would be the lifetime of the actual vehicle. If the vehicle was to break down before the lifetime of the device, the economical model would change since either the lifetime of the project would be shortened or another retrofitting expense would have to be considered.

The only difference between the economical models of the two scenarios considered is the price per kWh saved for the displaced heat energy. The price per kWh for gas is considerably less than for electricity and this majorly affects the economics of the system. Hence, unless the price of gas was to increase in the future years, integration of the portable thermal store with a gas heated domestic boiler is not financially feasible. For the electrical heated boiler, the economics are more favourable and the project breaks even for a device weighing c. 22kg. In both scenarios considered, as highlighted in Figure 8.4, an increase in mass of the thermal storage implies an improved economic outcome from the system.

The economical feasibility could be improved by either increasing revenue, decreasing the capital expenditure or by decreasing the operating costs. An increase in revenue could be achieved by optimisation of the device to contain more thermal energy per kg. In the proposed design the material containing the thermal energy occupied 40% of the total weight, optimisation of the finings (the minimum fins which would allow full heat transfer in the average 30km of commute rather than the 20.8km to 3.5km which would be required to "charge" the six to one modular units) could increase the ratio of the PCM's weight to the device's total weight. The external casing of the device accounts for almost 30% of the weight, other methods such as vacuum containments could be explored to maximise this ratio. Simulations could be done to optimise the device as necessary. Changes to the design would consequently change manufacturing costs and also the eventual operating costs.

The device has shown the potential to contribute to essential energy and carbon savings, as well as increase the efficiency of the applications which it is fitted to. The increase in efficiency is due to the minimisation of the waste/surplus heat in these applications. More research would have to put in the device to optimise the various components and the heat transfer rate according to requirements of the application exploited.

10. Conclusion

This research evolves upon the concept investigated by Millar & Huang (2009), with a specific design which allows the heat transfer primarily from micro-CHP by a portable heat storage.

The research conducted highlights how the energy could be stored within a thermal store and integrated into a domestic system using applications such as the heat waste from an automotive engine to “charge” the device.

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Consequent simulation and testing of the device allowed the determination of the device's performance. For establishment of its commercial viability, an economic analysis of a system integrated with micro-CHP was conducted. From the results obtained it appeared that the design did not provide suitable economical advantages which would make it viable in a commercial market, especially when integrated with a domestic system which utilises a gas boiler.

Further research would have to be conducted upon the optimisation of the design, in order to maximise the ratio of the PCM's weight to that of the whole device. Hence increasing its heat storage capacity and making it economically more attractive.

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