Engine preheating before ignition

Ríkey Huld Magnúsdóttir

FACULTY OF ENGINEERING
UNIVERSITY OF ICELAND

Engine preheating before ignition

A thesis by

Ríkey Huld Magnúsdóttir

submitted to the Faculty of Engineering in partial
fulfillment of the requirements for the degree of
Magister Scientiarum.

University of Iceland 2007

University of Iceland
Faculty of Engineering
VR-II, Hjarðarhaga 2-6, IS-107 Reykjavik, Iceland
Phone +354 525 4648, Fax +354 525 4632
verk@hi.is
www.hi.is

Abstract

Air pollution is a major problem in the urban world and a large part of it comes from the emissions from automobiles. When igniting an automobile on a cold day the engine emits more polluting chemicals than on a hot day. There are ways to decrease the emissions from automobiles and one of them is to preheat the engine before ignition.

The objective of this study is to investigate feasibility of pumping hot coolant through the engine to heat it before ignition. Experiments were carried out by pumping hot water through an engine and monitoring how the engine responded. The data was used to fit unknown parameters of a heat transfer model, which simulated the heat transfer from the hot water to the engine. The results provide good expectations supporting further work on the use of a hot coolant to preheat an engine sufficiently to reduce the emissions from it when ignited in a cold environment.

Útdráttur

Loftmengun hefur aukist undanfarin misseri og er hægt að rekja stóran hluta af þeirri aukningu til útblásturs bifreiða. Þegar bílvél er ræst á köldum degi þá gefur hún frá sér meira af mengandi útblæstri en á heitum degi. Það eru til aðferðir til að draga úr þessum útblæstri og er ein þeirra að forhita vélina fyrir ræsingu.

Markmið þessa verkefnis er að athuga hvort hægt er að nota heitan kælivökva til þess að forhita vélina fyrir ræsingu. Tilraunir voru gerðar með því að dæla heitu vatni í gegnum bílvél og fylgjast með hvernig vélin brást við. Gögnin úr mælingunum voru síðan notuð til að meta óþekkta stika í varmaflutningslíkani, sem hermdi eftir varmaflutningi frá heita vatninu yfir í vélina. Niðurstöðurnar gáfu góðar vonir um að hægt sé að nota heitan kælivökva til að forhita bílvél fyrir ræsingu nægjanlega mikið til að draga úr mengandi útblæstri á köldum degi.

Contents

List of Tables

List of Figures

Nomenclature

ΔT	temperature difference	$[^\circ C]$
ΔT	temperature difference	$[^\circ C]$
c	specific heat	$[kJ/kg^\circ C]$
E	energy	$[J]$
M	mass of hot water	$[kg]$
m	mass of engine	$[kg]$
$\dot{m}$	mass flow rate	$[kg/s]$
q	heat transfer	$[W]$
T	temperature	$[^\circ C]$
UA	heat transfer coefficient	$[W/^\circ C]$

Subscripts

s	subscript referring to engine block
a	subscript referring to cylinder head
c	subscript referring to conditions between inlet and outlet
e	subscript referring to outlet conditions
i	subscript referring to inlet conditions

1 Introduction

1.1 Motivation and background

Nowadays people are getting more aware of the environmental problems the world is facing. Air pollution is the most serious and common damage which automobiles cause to the environment and to peoples health. Much of it is caused by the exhaust gases, that are emitted when an automobile engine is started. The emissions contain carbon monoxide (CO), unburnt hydrocarbons (HC) and various oxides of nitrogen (NO_x) (19). The amount of these chemicals in the emissions depend on the temperature of the engine when started. If the engine is cold when started then the fuel mixture has to be richer in order to let the engine start and run easily, this contributes even more to the high emission levels, particularly of CO and HC (23). A cold start is defined as a start in which the temperature of the engine and catalytic converters are the same as the surroundings. Cool start is defined as a start in which the engine temperature is minimum $+20\,^{\circ}\mathrm{C}$ and the catalytic converter has cooled to below the reaction temperature (+250-300°C) (10).

The catalytic converter reduces the pollution and the most common one is the three-way catalytic converter. It is called a three-way catalyst because it removes all three pollutants CO, HC and NO_x simultaneously. It converts the pollutants into CO_2, H_2O and N_2 (1; 19). But the problem with a catalytic converter is that it is ineffective until its temperature has risen above $250-300\,^{\circ}\mathrm{C}$ (9), which is not reached until the exhaust from the engine itself has heated up the catalytic converter (22). During this warm up period the engine runs without any effective emission control. Therefore it is important that the engine heats up as rapidly as possible in order for the catalytic converter to reach its optimum temperature. During winter the problem is exaggerated since the fuel mixture must be richer and it takes the engine as well as the catalytic converter longer time to warm up (23). Calculations of the percentage of cold-start emissions in total emissions indicate that they account for 83 % of CO, 84 % of HC and 51 % of NO_x (1).

Engine heaters can shorten the time it takes to heat up the engine and the catalytic converter. They are generally based on heating the liquid of the engine cooling system. The benefits of using an engine heater are threefold, it reduces the emission from the engine, the fuel consumption decreases and the engine wear is greatly reduced (7; 15; 21). When an automobile engine is started the oil pump needs to get the lubricant quickly and safely to all friction areas in the engine. In a cold engine the lubricant is more viscous and the engine will therefore start more heavily which causes wear on bearings, camshaft and pistons.

For cars equipped with catalytic converters, regular use of engine heaters and fewer cold starts would give far greater environmental benefits than lower speed limits or reduced driving distances. When the emission of two vehicles, one driven five times longer distance than the other in one year, is compared it can be seen that there is not so much difference in the emissions. This indicates that it is important to have engine heater for short distance drives in order to decrease the emissions (1).

In Scandinavia the use of electric engine heaters is common. In 1995 about one third of Sweden's three million cars were equipped with engine heaters, which are mostly used in the winter. One reason that electric engine heaters have not yet become standard equipment in Swedish cars is the lack of connection facilities (1; 23). In Sweden electric engine heaters can achieve considerable emission reduction without significant overall energy consumption, since about 95 % of Swedish electricity is generated with hydro- or nuclear-power. For countries which use fossil fuels as the primary source for energy production, it would be of particular advantage to use engine heaters that are not based on electricity from the electric system, as the aim is to decrease pollution. This shows the importance of preheating the engine coolant as an effective way to reduce the emission from cold starts (3).

In Iceland it is obligated by the pollution control regulation, for vehicles to be equipped with a catalytic converter and has been so since 1995. Engine heaters are not commonly used in Iceland despite efforts by Landsvikjun, Reykjavik Energy, The Icelandic Automobile Association, Icelandic National Energy Authority and others. Their attempt to show good example in using electric engine heaters and the discount given for those who did use it, did not have the expected effects on the public.

The capital of Iceland, Reykjavik, with 116,000 inhabitants (8), is a city where nearly everyone has a car and a small proportion uses other transportation means. In 2002 85 % of the citizens used their private car to drive to and from work (18). The CO_2 emission in Reykjavik has been increasing over the past years and transportation is the cause for 95 % of it (14). At the end of 2006 there were 96,000 passenger cars in Reykjavik, which is about 49 % of all passenger cars in Iceland (20). In 2004 the fuel consumption of automobiles in Iceland was 149,000 tons of petrol (16). This suggests that in Reykjavik the consumption is 73,000 tons of petrol which gives rise to about 600 tons of CO_2. The mean distance traveled by passenger cars in Reykjavik is 5.04 km and according to numerical simulation model it will be 5.82 km in the year 2024 (11). The Swedish Road and Traffic Research Institute, VTI, states that the fuel consumption increases by 0.2 liters per cold start and the average motorist makes about 300-500 cold starts per year. By using an engine heater in 500 cold starts the emission of CO_2 per car might decrease as much as 250 kg per year, which is about 10 % of the total emission based on 15,000 km driving (23).

In Reykjavik the main source of NO_x is from automobiles and therefore its concentration is higher near heavily-trafficked streets. The amount of NO_x has been measured in Reykjavik since 1990. In 1995 the amount of NO_x in the air in Reykjavik started to decrease and that is attributed to the usage of catalytic converters. The amount did increase again in 2003-2004 and that was most likely caused by increased number of cars on the streets and that cars are getting bigger, heavier and more powerful in addition to increased total amount of kilometers driven (6; Böðvarsdóttir).

The emission can be decreased by preheating the engine with some means, one being the use of heat storage. Experiments have been made with heat exchangers, which use phase change materials (PCM), to preheat engines before starting. Schatz Thermo Engineering, Munich, Germany has developed a heat storage device that stores waste latent heat energy from the engine exhaust for use in later applications. This device, called the Schatz Heat Battery, stores the heat energy under vacuum in a molten salt ($Ba(OH)_2 8H_2O$). The salt releases heat energy to the cold engine coolant which is pumped through a canister containing the packaged molten salt. The coolant is warmed by contact with the salt containing packages and may be pumped to various locations within the vehicle. In testing of the Schatz Heat Battery two different types of fuel were used and they both gave good results concerning HC and CO emission reductions. The reduction of HC and CO with a 60 second preheating period was 69-85 % and 76-83 % respectively, depending on which fuel was used (17).

Other successful experiments have been made with heat exchangers for diesel motors, which have particular problems in cold starts because of increased frictional forces at low temperatures and therefore need several times more power to start than a warm engine. These heat exchangers are based on the waste heat produced while the engine is still running and phase change materials (13), like the Schatz Heat Battery. Similar experiments have been made for internal combustion engines (4; 21). From these experiments it has been calculated that the weight of the heat storage installation should not exceed 1-2 % of the total weight of the vehicle in order not to decrease considerably the useful weight. The shape and volume of the heat storage installation are determined by the free space available (13).

In this thesis the preheating of the engine by using coolant storage will be considered as a way to reduce the emission from cold starts. The possibility of heating the engine by pumping hot water through it will be investigated. The engine needs to heat up to +20 °C before starting so it will no longer be considered as cold nor cool start, as defined above.

1.2 Project description

The water cooling system is usually a single loop where a water pump pumps coolant to the engine block and then to the cylinder head. The coolant then flows to a radiator or heat exchanger and back to the pump. In this project, a new method will be investigated where stored engine coolant is used for preheating instead of the classical method of preheating with electricity. An insulated heat storage will be inserted into the water cooling system which will store a certain amount of hot coolant that can be pumped through the engine before starting it next time it is used. Minor adjustments will have to be made to the coolant circuit to install the heat storage under the bonnet of the car.

This thesis is organized as follows:

In chapter two the energy calculations and the theoretical modeling are shown and explained.

In chapter three the measuring equipment and the setup are explained. The experiments made to gain the needed data and the data obtained are explained.

Chapter four shows the results. Firstly the results from the experiments described in chapter 3 and secondly the theoretical results from the model described in chapter 2 are presented.

In chapter five the results are discussed and explained.

Chapter six is about what came out of this project and the future work.

2 Theory

In this chapter the energy calculations and theoretical model are introduced. The model simulates the heat transfer from hot water, pumped through an engine, to the engine parts being heated.

2.1 Energy calculation

To evaluate the heat transfer by convection resulting from a hot fluid flowing inside a cold pipe, where the fluids temperature consequently falls as it flows from the inlet conditions at T_i to exit conditions at T_e, the energy balance of the fluid is

$$q = \dot{m}(h_i - h_e) \tag{2.1}$$

where $\dot{m}$ is the mass flow rate, h_i is the enthalpy at the inlet and h_e is the enthalpy at the exit. For many single phase liquids operating over reasonable temperature ranges $\Delta h = \mathrm{c}\Delta T$ where

$$\Delta T = (T_i - T_e) \tag{2.2}$$

which gives

$$q = \dot{m}c(T_i - T_e) \tag{2.3}$$

where c is the specific heat capacity of the fluid (12).

The total amount of energy (E) transferred through heat transfer by convection is

$$E = \Delta T \dot{m} c \Delta t \tag{2.4}$$

where Δt is the time difference between measured points.

2.2 The model

A body whose interior temperature remains essentially uniform at all times during a heat transfer process can be observed to behave like a "lump". The temperature of such bodies can be taken to be a function of time only, T(t) (2). Heat transfer analysis that utilizes this idealization is known as lumped system analysis and will be used here. To calculate how much heat is transferred from hot water that is pumped through a cold engine, a heat transfer model was built, see figure 2.1. This model is similar to heat exchanger as the hot water gives heat to the cold engine. The heat transfer for heat exchangers is calculated with (12)

$$q = UA\Delta T \tag{2.5}$$

where U is the overall heat transfer coefficient, A is the surface area for heat transfer consistent with U and ΔT is the temperature difference across the heat exchanger. The engine is thought of as two masses, one being the engine block made of steel and the other one being the cylinder head made of aluminum. Therefore the total heat transfer from the water to the engine is divided between the engine block(q_s) and cylinder head(q_a). When the water flows from the engine its temperature has decreased to T_e as the temperatures of the engine block(T_s) and the cylinder head(T_a) have risen.

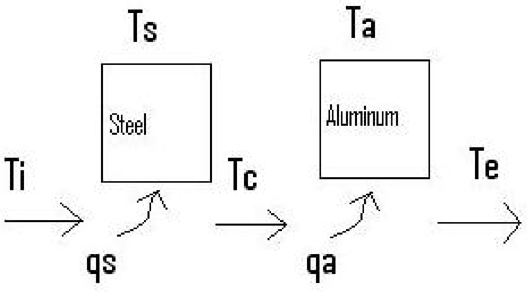

Figure 2.1: Heat transfer model

Calculating the heat transferred from the hot water to the engine block (2.6) is used and (2.7) is used to find out the temperature of the water after it has flowed past the steel part of the engine, which is T_c.

$$q_s = U_s A_s \left(\frac{T_i + T_c}{2} - T_s \right) \tag{2.6}$$

$$q_s = \dot{m} c (T_i - T_c) \tag{2.7}$$

??eqn:aluminumheatgain) is used to calculate the heat that the cylinder head gains from the water and (2.9) is used to find the temperature of the water, T_e, as it flows out of the engine.

$$q_a = U_a A_a \left(\frac{T_c + T_e}{2} - T_a \right) \tag{2.8}$$

$$q_a = \dot{m} c (T_c - T_e) \tag{2.9}$$

The temperatures of the engine block and the cylinder head change with time and are calculated by the following equations respectively

$$m_s c_s \frac{\delta T s}{\delta t} = q_s \tag{2.10}$$

$$m_a c_a \frac{\delta T a}{\delta t} = q_a \tag{2.11}$$

Substituting (2.6) into (2.10) gives:

$$m_s c_s \frac{\delta T s}{\delta t} = U_s A_s \left(\frac{T_i + T_c}{2} - T_s \right) \tag{2.12}$$

and substituting (2.8) into (2.11) gives:

$$m_a c_a \frac{\delta T a}{\delta t} = U_a A_a \left(\frac{T_c + T_e}{2} - T_a \right) \tag{2.13}$$

where T_c and T_e are found by substituting (2.6) into (2.7) and by substituting (2.8) into (2.9)

$$T_c = \frac{(2\dot{m}c - U_s A_s) T_i + 2 U_s A_s T_s}{2\dot{m}c + U_s A_s} \tag{2.14}$$

$$T_e = \frac{(2\dot{m}c - U_a A_a) T_c + 2 U_a A_a T_a}{2\dot{m}c + U_a A_a} \tag{2.15}$$

To solve (2.12) and (2.13) they are set up as the following matrix:

$$\begin{bmatrix} 2m_s c_s & 0 \\ 0 & 2m_a c_a \end{bmatrix} \begin{bmatrix} \dot{T}_s \\ \dot{T}_a \end{bmatrix} = \begin{bmatrix} \frac{2B^2}{A+B} - 2B & 0 \\ \frac{2BC}{A+B} + \frac{2ABC-2BC^2}{A^2+AC+AB+BC} & \frac{2AC^2+2BC^2}{A^2+AC+AB+BC} - 2C \end{bmatrix} \begin{bmatrix} T_s \\ T_a \end{bmatrix}$$

$$+ \begin{bmatrix} (B + \frac{AB-B^2}{A+B})T_i \\ (\frac{AC-BC}{A+B} + \frac{A^2C-ABC-AC^2+BC^2}{A^2+AC+AB+BC})T_i \end{bmatrix}$$

where
A = $2\dot{m}c$
B = U$_s$A$_s$
C = U$_a$A$_a$

The temperature change in the engine block ($\dot{T}_s$) and cylinder head ($\dot{T}_a$) over the pumping time are the solution from the differential equations 2.12 and 2.13. They are solved with Runge-Kutta, which is a method of numerically integrating ordinary differential equations by using a trial step at the midpoint of an interval to cancel out lower-order error terms.

3 Data and Measurements

In order to preheat an engine with a coolant, it is necessary first to obtain data on the engine response as hot water is pumped through it. No such data were accessible and therefore experiments were made to gather the needed data. A four valve water cooled Ford engine was used for the experiments which were conducted at the facilities of the Engineering Department of the University of Iceland. The setup of the engine and the measurement equipment is shown in figure 3.1 and as well in figure 2 in Appendix A. Hot water, 70 °C, from the district heating system was pumped through the engine and the temperature at four different positions was measured. The water was pumped through the same passageways as the cooling fluid flows and in the same direction. The water was pumped through water hoses with diameter of 12.7 mm.

3.1 The measuring equipment and setup

The measuring equipment setup used in the measurements is shown in figure 3.2. It consist of thermocouples (Type K), thermocouple amplifiers (AD595 from Analog Devices, see figure 1 in Appendix A), an analog-digital converter (ni6210 from National Instruments) and a computer.

The thermocouples were inserted into the engine to measure the heat from the hot water that was pumped through it. One was at the water intake, two were mounted on the cylinder head of cylinders 2 and 4 (see figure 3.2 for cylinder numbering) and the last one was at the water outflow. The temperature measured at these locations will from here on be referred to as T_i, T_{v2}, T_{v4} and T_e respectively. Monolithic thermocouple amplifiers with cold junction compensation, pretrimmed for Type K thermocouples, were used to produce a high level (10 mV/°C) output directly from the thermocouple signals. The output was then converted by the analog-digital converter and logged by a computer code programmed in C.

A circuit was made to connect the thermocouples to the amplifiers. Each amplifier has 14 pins and they were connected to the circuit for dual supply operation. Two 9 volt batteries were used as power supply for the circuit in order to obtain negative supply so the amplifiers could respond to temperatures below 0 °C. The thermocouple wires were connected to pins 1 and 14 on the amplifier, see figure 3.1. As the alarm output at pin 13 was not used it was connected to common. Pins 1 and 4 were also connected to common. Pin 7 was connected to the to the -V supply and pin 11 to the +V supply. The pre calibrated feedback network at pin 8 was tied to the output at pin 9 to provide the 10 mV/°C nominal temperature transfer characteristic.

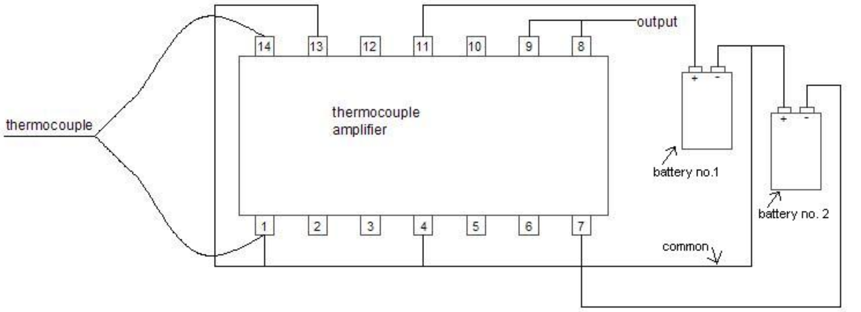

Figure 3.1: Diagram of an AD 595 thermocouple amplifier

The performance of the measurement setup was evaluated by running a few tests. First the circuit was tested to see if it would react to minus degrees and heat. This was done by inserting the thermocouples into a freezer to see if the output would give negative temperature. A voltmeter was used to read the output directly from the amplifiers by connecting it to pins 4 and 9 on each amplifier. The reading from the voltmeter was compared to the results from the computer code and the findings did agree. This procedure was also carried out for boiling water which showed that the circuit did react as expected to heat. The amplifiers did not have to be calibrated as they were precalibrated by manufacturer. Secondly, the amplifiers were tested one at a time, all using the same thermocouple. The thermocouple was first inserted to boiling water along with a thermometer and then into ice water. The voltmeter was again used to read the output from the amplifiers along with the computer code. Three of the amplifiers, number 1, 2 and 3 (see figure 3.1), gave correct results while amplifier number 4 did show 1 °C too low. All measurement results from amplifier number 4 were therefore corrected by +1 °C. This test did confirm that the amplifiers were working correctly and within margin of error, which is ± 3 °C. Thirdly, the thermocouples were tested individually by connecting them all to the same amplifier and inserting them into boiling water as well as ice water. Again a thermometer was inserted to the water and a voltmeter along with the computer code were used to read the output from the amplifiers. These tests did indicate that the maximum difference between the thermocouples was 0.5 °C.

When the measurements started, disturbances in the measured values were detected and were believed to be caused by conduction between the thermocouples. The ends of the thermocouples were therefore insulated and the conduction between them eliminated so the only disturbances remaining in the experimental setup were those from the house electricity.

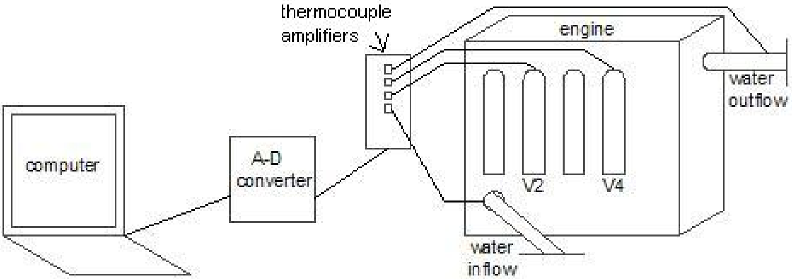

Figure 3.2: The measurement setup

3.2 The engine experiments

The experiments were conducted with three different initial engine temperatures, T_o = -4 °C, 6 °C and 20 °C. They were to represent the outdoor temperatures at which the engine would start. In order for the data obtained from the experiments to be comparable, the variables that were changed were the mass of hot water (M) and the mass flow ($\dot{m}$). The mass and the mass flow of the hot water had two values in each experiment, M = 5 kg and 10 kg, and $\dot{m}$ = 0.05 kg/s (slow) and 0.2 kg/s (fast). Each experiment lasted around 20 minutes, although the pumping time was only 25-200 seconds depending on the mass of hot water pumped and the mass flow rate. The purpose of having the experiment time this much longer than the pumping time was to observe how the engine responded after the pumping of the hot water stopped. The measurement frequency was 100 measurements per second, which gave 120,000 measured values for each thermocouple in each measurement. The initial engine temperature T_o = -4 °C was reached by filling up the engine cooling passageways with antifreeze and the passageways were closed on both ends to prevent the antifreeze to leak out. The engine was then put outside overnight during a cold spell, where the temperatures were below zero degrees for several days. This was done for each of the T_o = -4 °C measurements. To reach the other initial engine temperatures T_o = 6 °C and T_o = 20 °C, water of the desired temperature was pumped through the engine until a stable temperature was reached.

The output from these experiments provided comprehensive data on the response of the engine temperature indicators under cold ambient start up conditions and hot water flow through the coolant path. To evaluate the data, a computer code was programmed in Matlab to filter the data and to discard peaks caused by interferences from the house electricity on the measurement devices. To be able to filter the data the largest peaks had to be diminished. That was done for each series of measured values by comparing each point with its neighboring points. If the values did differ by more than 5 °C then the point got a new value, which was the mean value of the neighboring points values. There were only few points in each experiment whose value did differ more than 5 °C and they were caused by interference of house electricity and are shown in figure 3.3. The data was subsequently filtered in both the forward and reverse directions with a Butterworth digital filter. After filtering the data these peaks did vanish and the curves became smoother, as can be seen in figure 3.4.

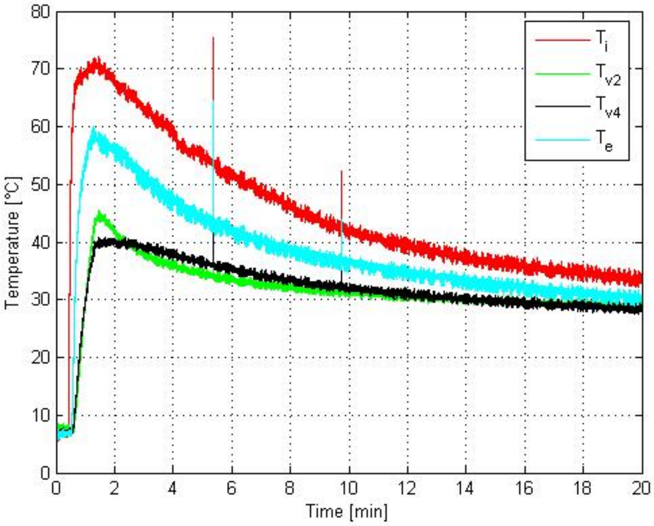

Figure 3.3: The raw data from a measurement with M = 10 kg, $\dot{m}$ = 0.2 kg/s and To = 6° C.

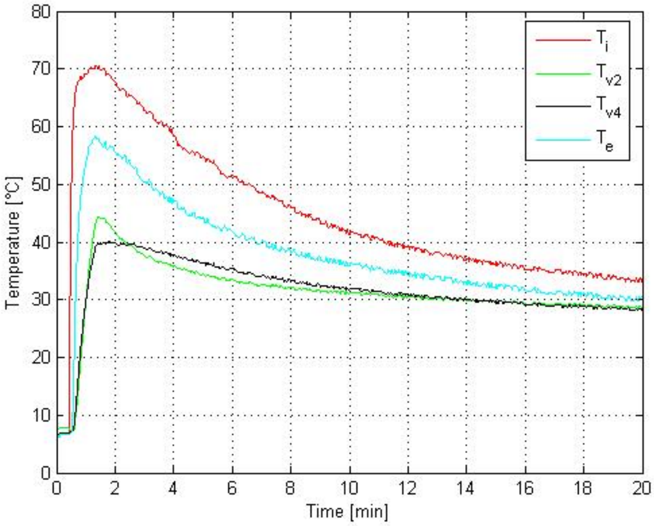

Figure 3.4: Filtered data from a measurement with $M = 10\,kg$, $\dot{m} = 0.2\,kg/s$ *and* $To = 6\,^\circ C$.

Figure 3.3 shows experimental data obtained for $M = 10\,\text{kg}$ of hot water, $\dot{m} = 0.2\,\text{kg/s}$ and $T_o = 6\,^\circ\text{C}$. The peaks caused by interferences of the house electricity can be seen, in this experiment, at about 5.4 and 9.7 minutes.

Figures 3.3 and 3.4 show the experimental raw data and the filtered data on temperature variations with time. They illustrate that the temperatures rise very rapidly after pumping of the hot water starts. After the pumping stops the temperatures begin to fall. The temperatures of the water (T_i and T_e) decrease faster than the temperatures of the engine (T_{v2} and T_{v4}) but they all slowly reach the ambient temperature. The temperature at the water inlet and exit do not rise simultaneously, T_e starts to rise 6-25 seconds after T_i. This is the time it takes the hot water to flow through the engine, which depends on the mass of the hot water and the mass flow rate. The temperature change over the pumping time is shown in figure 3.5 for an experiment with $M = 10\,\text{kg}$, $\dot{m} = 0.2\,\text{kg/s}$, $T_o = 6\,^\circ\text{C}$ and pumping time of 50 seconds. When examining how high the cylinder head temperatures rise, it can be seen by comparing figure 3.5 to figure 3.6, that it has not reached its maximum temperature when the pumping time is over. The cylinder heads continue to gain temperature, 3-4 °C for 13 seconds before it start falling again. The temperature of the water reaches its maximum before the pumping time is over and reaches steady state before it starts to fall again.

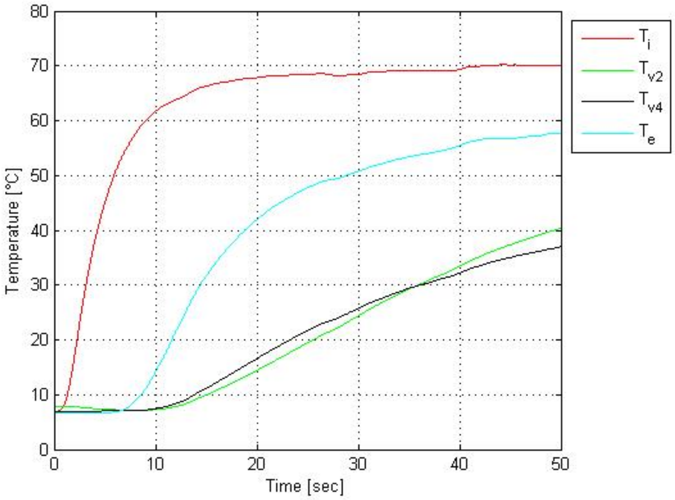

Figure 3.5: Temperature over 50 second pumping time.

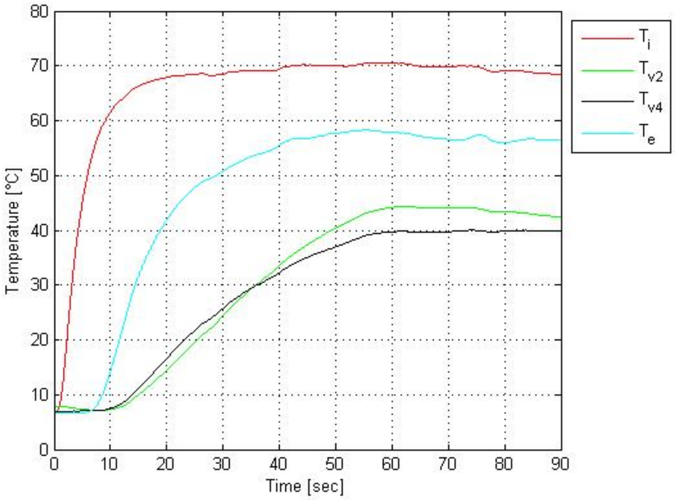

Figure 3.6: Temperature over 90 second time interval.

3.2.1 Additional experiments

After the predetermined experiments were completed and the results analyzed it was decided to carry out few additional measurements. They were only performed with initial engine temperature $T_o = 6\,^\circ\mathrm{C}$ and only with $M = 10\,\mathrm{kg}$ of hot water. The mass flow was the only variable that did change. The mass flows that were to be used were $\dot{m} = 0.033\,\mathrm{kg/s}$, $0.067\,\mathrm{kg/s}$, $0.140\,\mathrm{kg/s}$ and $0.333\,\mathrm{kg/s}$. Measurements with the first three mass flows were successful but trying to reach the fastest one was not accomplished. The fastest possible mass flow that could be reached was $\dot{m} = 0.260\,\mathrm{kg/s}$. The hot water mass flow was limited by the diameter of the water hose and the pressure in the hot water system at the experiment location. As in the predetermined experiments the total measurement time was 20 minutes and the frequency was 100 measurements per second. Data processing was the same as for the first set of experiments.

Figures 3.7 and 3.8 show filtered data from two of the additional measurements as temperature over the whole measurement time. In figure 3.7 the pumping time was 5 minutes while the pumping time in figure 3.8 was only 38.45 seconds. These figures show clearly that the fast and the slow mass flows have different effect on the heat stream to the engine. With slow mass flow the engine does not gain as much heat as when the hot water is pumped fast through it. The response time is also much longer with slower mass flow for all the temperature monitored locations.

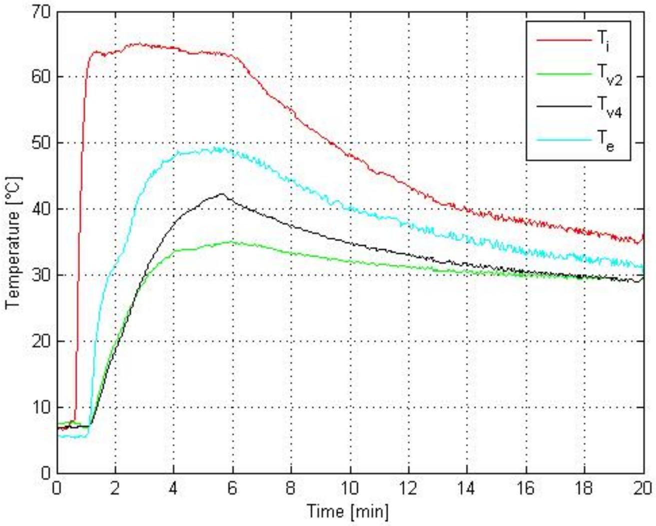

Figure 3.7: Filtered data from a measurement with $M = 10\,kg$, $\dot{m} = 0.033\,kg/s$ *and* $T_o = 6°C$.

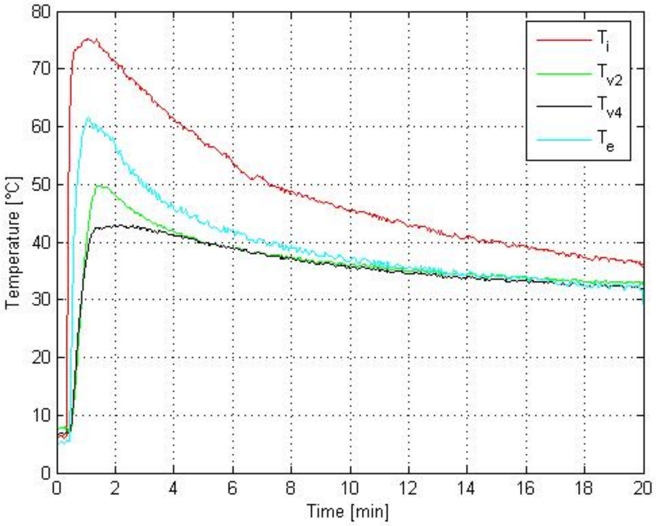

Figure 3.8: Filtered data from a measurement with $M = 10\,kg$, $\dot{m} = 0.260\,kg/s$ *and* $T_o = 6°C$.

4 Result

In this chapter results will be shown in two parts. Firstly in section 4.1 the results from the experiments described in chapter 3 and secondly in section 4.2 the results from the theoretical model described in chapter 2 are presented.

4.1 Experimental results

After the data from the experiments had been processed, they were analyzed and the results evaluated. For each experiment the data comprise about 120,000 measured values which span 20 minutes. The pumping time was from 25 to 200 seconds and therefore that will be the periods examined in the results rather than the whole experiment time. Exception are the additional experiments were the pumping time did extend to 300 seconds.

One of the temperature measurement positions was at the cylinder head, inside the combustion areas, see figure 3.2. Table 4.1 shows the maximum temperature at the cylinder head during experiments with different initial engine temperatures (T_o) and different mass flow rates ($\dot{m}$), but all with the same mass, $M = 5\,\mathrm{kg}$. There is a temperature difference between the combustion areas in each measurement. It is of course preferable that the cylinder head has uniform temperature when the engine is started in order for a proper combustion to occur. Table 4.2 shows the maximum temperature at the cylinder head for experiments with $M = 10\,\mathrm{kg}$. A temperature difference between the combustion areas is also observed here. The average temperature difference between the combustion areas is about 2.7 °C with 5 kg of hot water pumped but 4.4 °C when 10 kg of hot water is pumped.

The temperature difference between experiments with $M = 5\,\text{kg}$ and $M = 10\,\text{kg}$ is noticeable and is significantly higher with greater mass of hot water. That is the cylinder head does reach higher temperatures when 10 kg of hot water is pumped through the engine. The mass flow rate seems to have some affect as the temperatures for the experiments with $\dot{m} = 0.2\,\text{kg/s}$ reach higher than those for $\dot{m} = 0.05\,\text{kg/s}$ in most of the measurements. This indicates that the more turbulent the flow is the more heat is conducted into the engine parts. These result give good expectancy that it should be possible to use hot engine coolant to heat up the engine in a relatively short time before igniting it.

Table 4.1: Temperatures of the combustion areas with $M = 5\,\text{kg}$.

T_o [°C]	0.05 kg/s		0.20 kg/s	
	$T_{v2}[°C]$	$T_{v4}[°C]$	$T_{v2}[°C]$	$T_{v4}[°C]$
-4	31.35	28.61	33.30	28.95
6	30.61	32.78	36.36	33.77
20	39.50	41.81	42.59	40.65

Table 4.2: Temperatures of the combustion areas with $M = 10\,\text{kg}$.

T_o [°C]	0.05 kg/s		0.20 kg/s	
	$T_{v2}[°C]$	$T_{v4}[°C]$	$T_{v2}[°C]$	$T_{v4}[°C]$
-4	37.72	37.76	43.81	38.75
6	39.87	47.47	44.43	40.07
20	42.97	48.53	55.49	51.67

Table 4.3: Temperatures of the combustion areas with $M = 10\,\text{kg}$ and $T_o = 6\,°\text{C}$.

$\dot{m}[kg/s]$	$T_{v2}[°C]$	$T_{v4}[°C]$
0.033	35.09	42.31
0.067	43.70	47.47
0.140	46.48	42.23
0.260	49.76	42.97

The temperature differences between the combustion areas in the additional experiments, where the mass flow rate was the only variable changing, are shown in table 4.3. The maximum cylinder head temperature increases with increasing mass flow rate, at combustion area number 2 (T_{v2}) but not so much at combustion area number 4 (T_{v4}). Despite the difference in the mass flow the maximum temperature at the combustion areas is not much higher than the maximum temperatures seen in table 4.2 at initial engine temperature $T_o = 6\,°\text{C}$. This indicates that the heat conduction from the water to the engine parts being heated does not increase significantly after a certain mass flow rate is reached.

To evaluate the gains from pumping hot water through a cold engine, the energy (E) flow from the hot water to the engine was calculated using equation 2.4. The total amount of energy gained in each experiment is shown in table 4.4. The highest energy gain is $E_{max} = 1.22\,\text{MJ}$, which is reached when the initial engine temperature is -4 °C and 10 kg of hot water is slowly pumped through the engine. In this experiment the engine temperature changed from -4 °C to 37.7 °C, a temperature gain of 41.7 °C. Starting the engine at these temperatures would no longer be a cold start nor a cool start, according to definition in chapter 1, and the engine would emit far less polluting chemicals. For all three initial engine temperatures there is more to gain in pumping 10 kg then 5 kg. How fast the hot water should be pumped seems to be related to the initial engine temperature. The results in table 4.4 indicate that when the engine initial temperatures are low there seems to be greater energy gain if the water is pumped slowly, but when the engine initial temperature increases to 20 °C then higher the mass flow rate becomes efficient.

Table 4.5 shows the total amount of energy gained in each of the additional experiments. These results are shown graphically in figures 4.1- 4.5. For the additional measurements, which all had $M = 10\,\text{kg}$ and $T_o = 6\,°\text{C}$ but varying mass flow rates, the maximum energy gained was 1.138 MJ, see table 4.5.

Most energy is gained from the hot water when pumped through the engine with initial temperature of T_o = -4 °C than with T_o = 6 °C or T_o = 20 °C. The mass of hot water also affects the energy gain. More energy is gained with the higher mass, that is with $M = 10\,\text{kg}$ of hot water than with $M = 5\,\text{kg}$, for all of the initial engine temperatures.

The energy gained in MJ, by pumping 5 kg of hot water through the engine, for different initial engine temperatures, is shown in figures 4.1 and 4.2. The difference in the amount of energy gained by pumping the 5 kg slowly ($\dot{m} = 0.05\,\text{kg/s}$) or fast ($\dot{m} = 0.20\,\text{kg/s}$) is not significant, it varies from 0.8 kJ to 15.7 kJ. The largest difference is for a initial engine temperature $T_o = -4\,°\text{C}$ and it is only 2% of the maximum energy gained. The greatest difference between these results is in the pumping time. For slow mass flow the pumping takes 100 seconds while it only takes 25 seconds for the fast mass flow.

The energy gained by pumping 10 kg is similarly shown in figures 4.3 and 4.4. These figures show a noticeable difference in energy for different mass flow rates. Here the difference varies from 50 kJ to 179 kJ. In this case the greatest energy gain is also for initial engine temperature To = -4 °C. From figures 4.1- 4.4 it can be seen that there is considerably more energy to gain from pumping 10 kg than 5 kg of hot water through the engine to heat it up. The smallest energy difference is for

Table 4.4: Energy gained in each pumping cycle.

$T_o[°C]$	$M[kg]$	$\dot{m}[kg/s]$	$E[MJ]$
-4	5	0.05	0.7316
-4	5	0.2	0.7159
-4	10	0.05	1.2200
-4	10	0.2	1.0410
6	5	0.05	0.7177
6	5	0.2	0.7184
6	10	0.05	1.0320
6	10	0.2	0.9821
20	5	0.05	0.5899
20	5	0.2	0.5893
20	10	0.05	0.7245
20	10	0.2	0.8002

Table 4.5: Energy gained in each pumping cycle in the additional measurements where $M = 10\,\mathrm{kg}$ and $T_o = 6\,°\mathrm{C}$.

$\dot{m}[kg/s]$	$E[MJ]$
0.033	0.9855
0.067	1.1030
0.140	1.1380
0.260	1.1210

$T_o = 6\,°\mathrm{C}$ both in measurements with $M = 5\,\mathrm{kg}$ and $M = 10\,\mathrm{kg}$, although more energy is gained in experiments with $M = 10\,\mathrm{kg}$. Because of this little difference it was examined whether the mass flow rate alone had any effect on the energy gained. These were the additional measurements with $M = 10\,\mathrm{kg}$ and $T_o = 6\,°\mathrm{C}$. The results from these experiments show that the slower the mass flow is the less energy is gained, see figure 4.5. 1.12 MJ of energy is gained for $\dot{m} = 0.260\,\mathrm{kg/s}$ and 1.14 MJ of energy is gained for $\dot{m} = 0.140\,\mathrm{kg/s}$. Accordingly it seems to be more efficient to use more mass and pump it rapidly to get the largest amount of energy to heat up the engine.

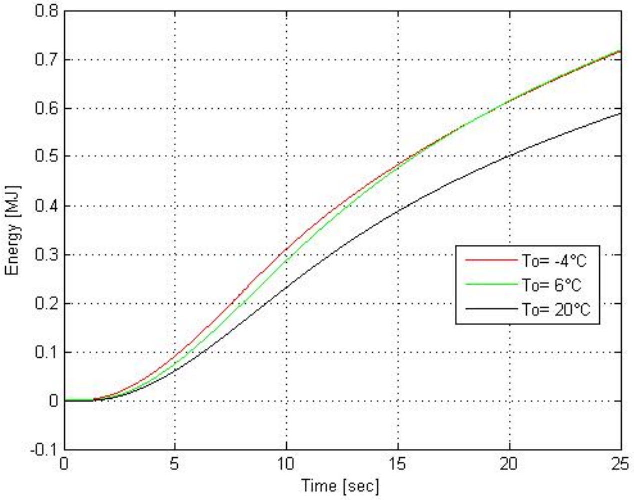

Figure 4.1: Energy in measurements with $M = 5\,\mathrm{kg}$ *and* $\dot{m} = 0.2\,kg/s$

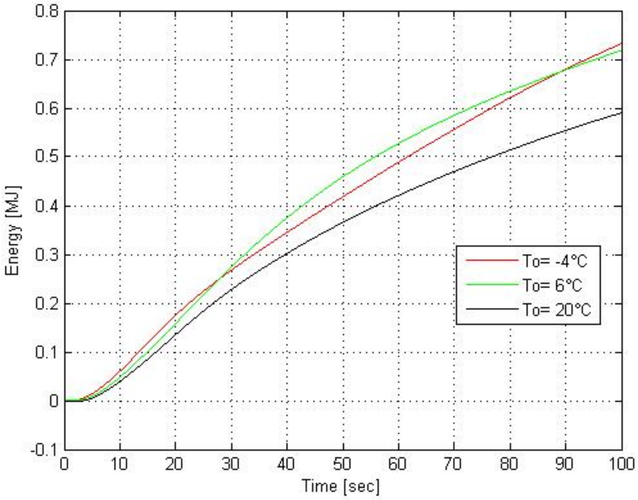

Figure 4.2: Energy in measurements with $M = 5\,\mathrm{kg}$ *and* $\dot{m} = 0.05\,kg/s$

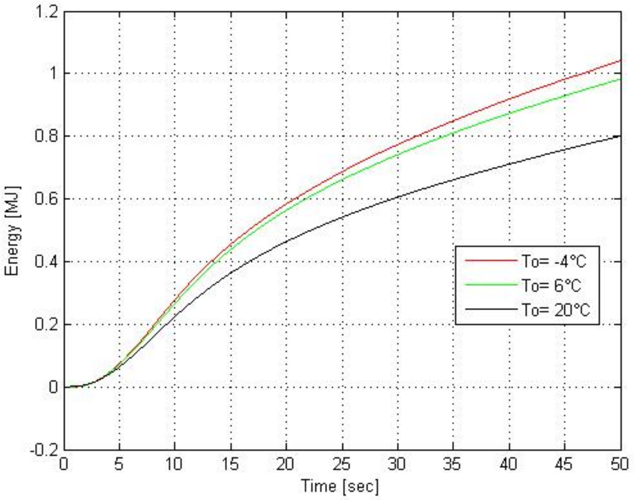

Figure 4.3: Energy in measurements with $M = 10\,\mathrm{kg}$ *and* $\dot{m} = 0.2\,kg/s$

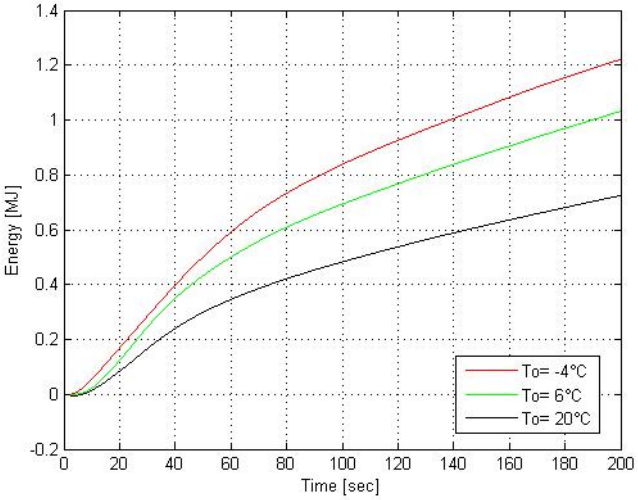

Figure 4.4: Energy in measurements with $M = 10\,\mathrm{kg}$ *and* $\dot{m} = 0.05\,kg/s$

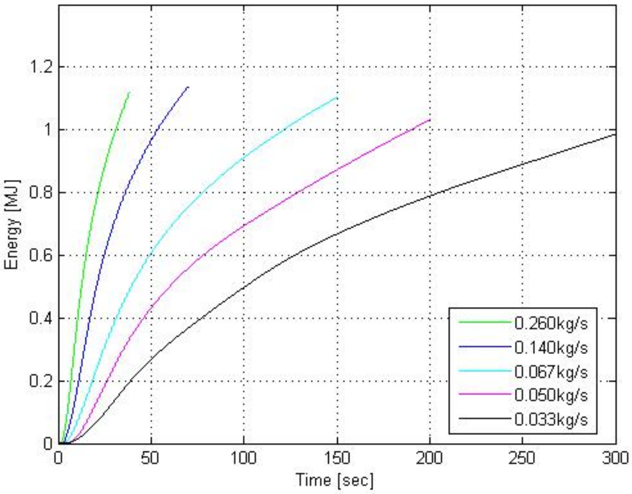

Figure 4.5: Energy in measurements with $M = 10\,\mathrm{kg}$ *and* $T_o = 6\,^\circ\mathrm{C}$

4.2 Theoretical results

The theoretical model introduced in chapter 2 was simulated to find out how high the temperature of the engine block and the cylinder head would become during pumping time of the hot water. The model had unknown parameters that had to be fitted with the data. These parameters were the overall heat transfer coefficient multiplied by the surface area through which the heat transfer took place (U_sA_s and U_aA_a) and the mass being heated (m_s and m_a), for both the engine block and the cylinder head. The parameters were fitted to data from experiments with all three initial engine temperatures. To get the best fit of the parameters the outflow temperature of the hot water (T_e) and the mean value of the cylinder head temperatures (T_{v2} and T_{v4}) were used. The data used was from experiments with $M = 5\,\text{kg}$ and $M = 10\,\text{kg}$, both with mass flow of $\dot{m} = 0.2\,\text{kg/s}$. The parameters gave better fit with the outflow temperature than with the cylinder head temperature. The results from the fits are in figure 4.6 were the measured data are plotted with the solid line while the calculated values are plotted with the broken line. The measured value for $T_o = -4\,°\text{C}$ did differ 4-5 °C from the expected initial temperature used in the model. The parameters were found with least square method for non linear equations in Matlab and the mean value from these two fits gave the results shown in table 4.6.

Table 4.6: Fitted parameters for UA and m

	$UA[W/°C]$	$m[kg]$
Engine block	1.5526	11.5278
Cylinder head	0.1750	7.6794

The results shown in figure 4.7 were obtained by using the Runge-Kutta method to solve the first order differential equations 2.12 and 2.13. The result from these equations gave the temperature change of the engine block ($\dot{T_s}$) and the cylinder head ($\dot{T_a}$) over the pumping time using the fitted parameters. The figure shows the temperature change over the pumping time with $M = 10\,\text{kg}$, $\dot{m} = 0.2\,\text{kg/s}$ and for T_o = -4 °C, 6 °C and 20 °C.

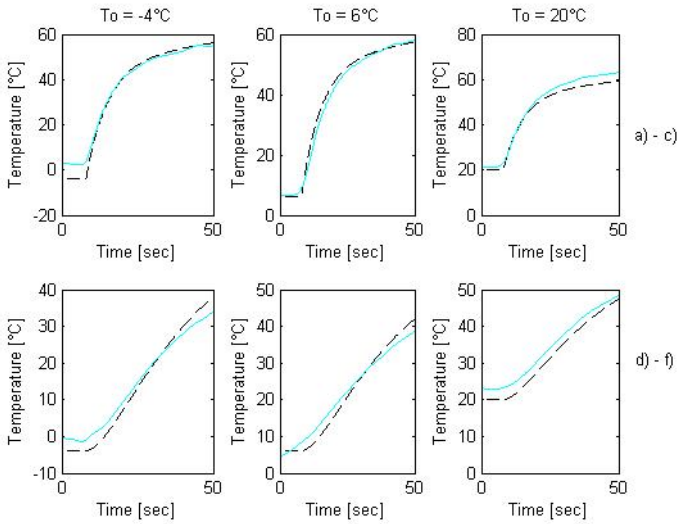

Figure 4.6: The fit of T_e (a - c) and T_a (d - f)

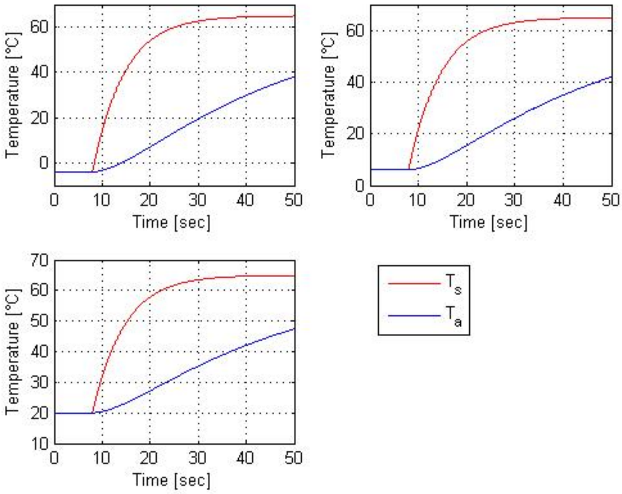

Figure 4.7: Calculated temperature change of the engine block (T_s) and the cylinder head (T_a) over pumping time.

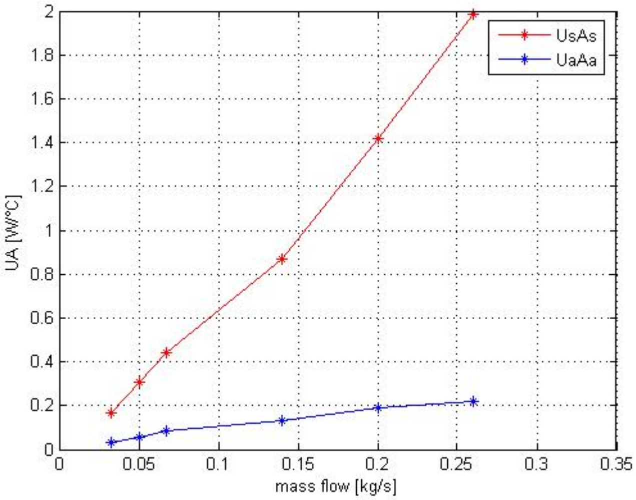

Figure 4.8: Change in UA with different $\dot{m}$

Different mass flows were used to calculate their influence on the heat transfer coefficient. The fitted values for m$_s$ and m$_a$ were used with $M = 10\,\text{kg}$ and $T_o = 6\,°\text{C}$ for mass flow spanning 0.033-0.260 kg/s. Changing the mass flow did have more affect on the heat transfer coefficient of the engine block than of the cylinder head, see figure 4.8. For the cylinder head the slowest mass flow only gave $U_aA_a = 0.0341\,\text{W}/°\text{C}$ while the fastest gave $U_aA_a = 0.2220\,\text{W}/°\text{C}$. For the engine block the coefficient obtained were higher. The slowest mass flow gave $U_sA_s = 0.1674\,\text{W}/°\text{C}$ and the fastest mass flow gave $U_sA_s = 1.9843\,\text{W}/°\text{C}$. The engine mass gains more heat with higher UA. This change in UA with different mass flow was only carried out for initial temperature $T_o = 6\,°\text{C}$ (5) as that represents the mean temperature in Reykjavik last 4 years at 8:00 in the morning and at 17:00 in the afternoon, which are the times when most people are driving to and from work.

The mass of hot water pumped through the engine has an effect on the heat amount transferred to the engine block and the cylinder head. The model was used to evaluate this effect under constant mass flow rate. The more mass of hot water pumped through the engine the longer time it takes which results in the engine gaining higher temperatures until it reaches steady state at 65°C which is close to the temperature of the hot water flowing through the engine, as can be seen in figure 4.9. Similar results were obtained for the temperature gain at engine initial temperatures $T_o = -4\,°\text{C}$ and $T_o = 20\,°\text{C}$.

The experimental results showed that more energy was gained from 10 kg then 5 kg

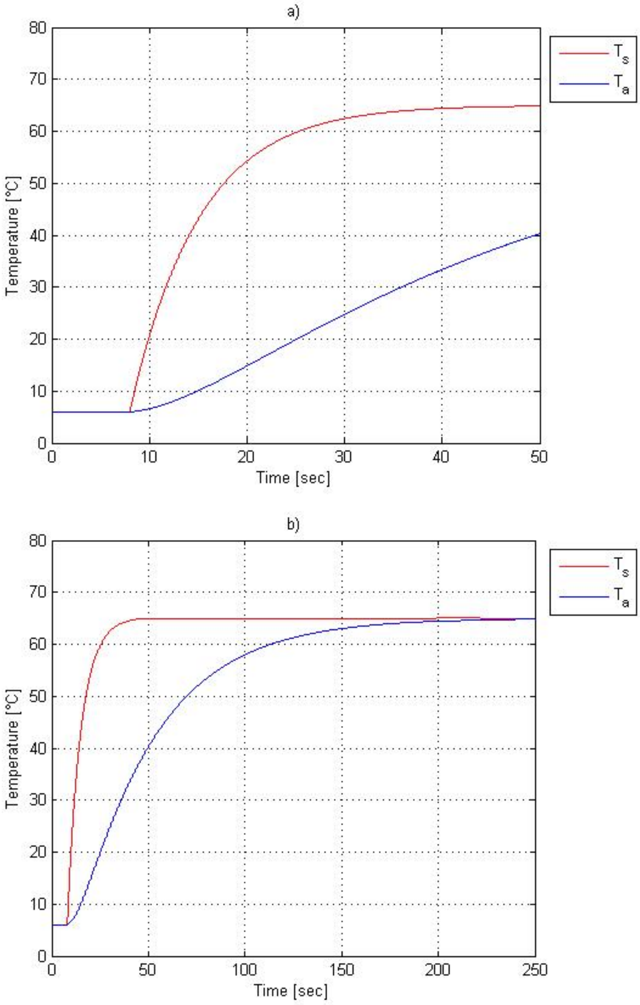

Figure 4.9: Temperature change in the engine block and cylinder head with a) $M = 10\,\mathrm{kg}$ *and b)* $M = 50\,\mathrm{kg}$

of hot water. The results from the model show that the engine reaches higher temperatures when 10 kg are pumped through it. When comparing the temperatures the engine reaches when 50 kg of water are pumped through it to the results from pumping 10 kg, it can be seen that the engine does reach higher temperatures. But this happens after longer time then with 10 kg and the difference is temperature gained is not proportional with the time. That is the cylinder head reaches 65 °C when 50 kg of hot water are pumped through the engine, which takes 250 seconds while it only takes 50 seconds to pump 10 kg and the cylinder head reaches 40 °C. With a gain of only 25 °C it takes 5 times as long, which makes it user unfriendly.

5 Conclusions

The aim of this project was to investigate the feasibility of heating an engine before ignition by pumping hot water through its coolant flow path. This results confirm that it is possible to use hot water and eventually hot cooling liquid to heat the engine in a relatively short time. The results clearly show that there is considerable energy gain from pumping hot water through the engine before starting.

The experimental results showed that the engine did reach quite high temperatures after short time of pumping the hot water. There are differences between the experimental results and the theoretical results regarding which mass flow rate seems to be optimum for $T_o = 6\,^\circ\mathrm{C}$. The experimental results show most energy gained with mass flow $\dot{m} = 0.140\,\mathrm{kg/s}$ while the theoretical results indicate highest heat transfer coefficient for $\dot{m} = 0.260\,\mathrm{kg/s}$. With higher heat transfer coefficient the more heat is transferred to the engine, as the contact area is not changing only the mass flow. For optimum results the mass flow rate should not be the same for different initial engine temperatures. It gives best results to pump the hot water slowly when the initial engine temperature is low but as it increases so should also the mass flow rate.

By igniting the engine after it has been heated up to 30-40 °C the emissions from it will decrease, as a cold or cool start involves most engine emissions of polluting chemicals. When the engine is warm or hot at ignition it takes the catalytic converter much shorter time to reach its reaction temperature and gain full functionality. A great advantage is for the driver and the passengers of the automobile as the heating system will start immediately to blow warm air into the passenger compartment. This will help to demist the car windows and give the driver a better view of the surroundings when driving. This will only take up to 1 minute to work, according to the results presented. Short pumping time makes the method more user friendly, but the pumping time is dependent on the mass of hot water pumped and mass flow rate. It has also to be taken into consideration that with higher mass of water used to heat the engine the heavier the whole system will be and that it should not exceed 1-2 % of the total weight of the car, in order not to increase the fuel consumption.

The cylinder head did reach temperature, as high as 55 °C, after hot water had been pumped through the engine. The maximum temperature was not reached until shortly after the pumping had stopped, it took 10-50 seconds, depending on engine initial temperature and mass flow rate. This and how high temperature was reached indicates that it would be best to start the engine a few seconds after the pumping has stopped, so the heat gained from the hot water will not be lost to the environment.

It is clear that a preheating system has many positive effects in the Icelandic climate, especially during the winter months. With widespread usage, the pollution from cars could be reduced significantly, especially in cases where cars are mainly used for short trips.

5.1 Future work

The results clearly show that pumping hot coolant through an engine can be used as a method to preheat it before ignition. A logical next step is to design and build a prototype of the heat storage for the hot coolant. The prototype will have to be tested with respect to pollution and fuel consumption. The heat transfer model designed in this project assumes that the environment is unchanged except the engine initial temperature. When testing the prototype in use the model presumably will need to be adapted to the outdoor conditions. For example, parameters such as wind and precipitation need to be added to the model. Besides, the size and weight of the prototype need to be optimized. First to mention there is generally little free space under the bonnet of a car for extra equipment and therefore it is a limiting factor how large the heat storage can be. The weight of the heat storage affects the fuel consumption because if it exceeds more than 1 to 2 % of the total weight of the car there will be no saving in fuel consumption. As expected the optimal size and weight of the heat storage is dependent of the car. It would be interesting and also most effective to decrease pollution to develop the heat storage in collaboration with a car manufactory. The manufacturer could give available advice for example concerning where the possible positions are for a heat storage. If the development of the equipment will succeed it will take shorter time to have it as standard equipment in personal cars.

Bibliography

[1] Bilmiljögruppen (1995). Electric engine heater, ScanTech Development AB.

[Böðvarsdóttir] Böðvarsdóttir, A. R. Mælingar á loftmengandi efnum í reykjavík 2005.

[2] Cengel, Y. (2002). *Heat Transfer: A Practical Approach*. McGraw-Hill.

[3] Darkwa, K. and O'Callaghan, P. (1997a). Green transport technology (GTT): Analytical studies of a thermochemical store for minimising energy consumption and air pollution from automobile engines. *Applied Thermal Engineering*, 17:603–614.

[4] Darkwa, K. and O'Callaghan, P. (1997b). Mathematical modelling of a thermochemical energy store: automobile application. *Proseedings of the Institution of Mechanical Engineers*, 211, Part D:337–346.

[5] Gísladóttir, G. . (2006). Veðurgögn frá reykjavík kl.8 og 17 árin 2003-2006.

[6] Guðmundsson, H. J. (2005). Umhverfisvísar reykjavíkur 2005, samgöngur á landi og orkumál.

[7] Gunnarsson, O. A. (2000). Hreyfilhitari.

[8] Hagstofan (2007). Hagtölur, mannfjöldi. *http://www.hagstofa.is/?PageID=67*. Date retrieved: March 26. 2007. Date last modified: February 20. 2007.

[9] Heywood, J. B. (1988). *Internal Combustion Engine Fundamentals*. McGraw-Hill.

[10] Höglund, P. G. and Ydstedt, A. (1998). Reduced air pollution and fuel consumption with preheated car engines.

[11] Hönnun, V. (2006). Samgönguskipulag í reykjavík, fyrsti hluti, greining á stöðu og stefnu.

[12] Holman, J. (2002). *Heat Transfer, Ninth Edition*. McGraw-Hill.

[13] Hunyadi-Kiss, I. (1998). Engine pre-warming heat storage systems as solutions to cold-starting problems of diesel engines. *Periodica Polytechnica Ser. Transp. Eng.*, 26, NO. 1-2:115–130.

[14] Müllerstöm, J., Theorin, L., Nielsen, P., Greatorex, J., Möller, J., Guðmundsson, H., Sigurðsson, ., Nerpin, L., and Silfverberg, K. (2006). Environmental indicators of major nordic cities.

[15] Þorbjörnsson, J. B. (1999). Notkun rafknúinna hreyfilhitara í bifreiðum landsvirkjunar, hagkvæmniathugun.

[16] Orkustofnun (2004). Orkutölur, eldsneyti, eldsneytisnotkun. *http://www.orkutolur.is/mm/eldsneyti/eldsneytisnotkun.html.* Date retrieved: March 27. 2007.

[17] Piotrowski, G. and Schaefer, R. (1992). Evaluation of a schatz heat battery on a fleixible-fueled vehicle, phase ii. Technical report.

[18] Silfverberg, K., Möller, J., Nyhuus, S., Müllerström, J., Christensson, K., Nerpin, L., Guðmundsson, H., Sigurðsson, ., Nielsen, J., and P.Nielsen (2003). Nordic larger cities environment indicators.

[19] Stone, R. (1999). *Introduction to Internal Combustion Engines, Third Edition.* Palgrave.

[20] Umferðarstofa (2006). Upplýsingatorg, bifreiðatölur. *http://www.us.is/id/4174.* Date retrieved: March 26. 2007. Date last modified: December 31. 2006.

[21] Vasiliev, L., Burak, V., Kualkov, A., Mishkinis, D., and Bohan, P. (2000). Latent heat storage modules for preheating internal combustion engines: application to a bus petrol engine. *Applied Thermal Engineering*, 20:913–923.

[22] Wingbrant, H., Svenningstorp, H., Salomonsson, P., Tengström, P., Lundström, I., and Spetz, A. L. (2003). Using a misicfet device as a cold start sensor. *Sensors and Actuators B*, 93:295–303.

[23] Ydstedt, A., Björklund, S., and Höglund, P. G. (1996). Reduced cold start emissions through use of electric engine heaters.

Appendix A

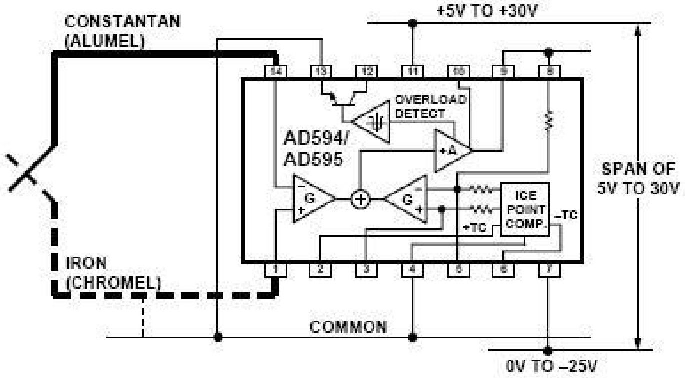

Figure 1: Dual supply operation

Figure 1 is gotten from a data sheet that was enclosed with the thermocouple amplifier AD 595 from Analog Devices, see further at:
$http : //www.analog.com/UploadedFiles/Data_Sheets/AD594_595.pdf$.

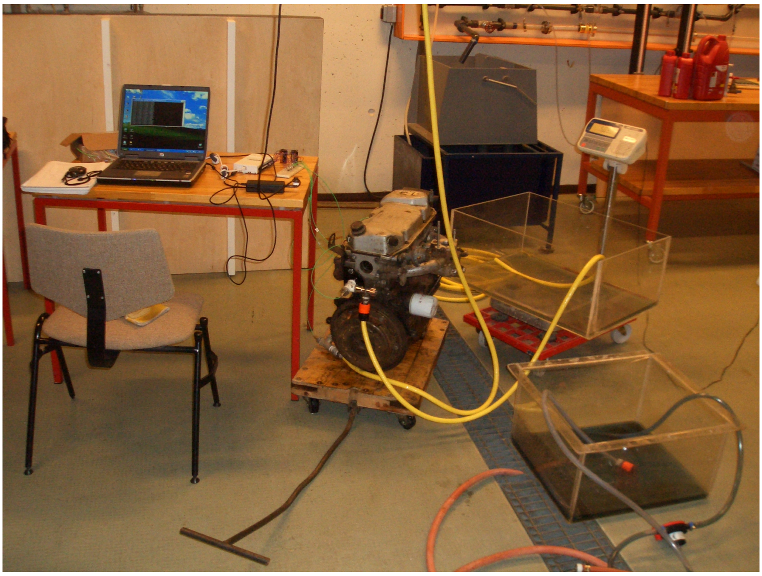

Figure 2: The setup of the engine and the measurement equipment

Figure 2 shows the setup of the engine and the measurement equipments at the measurements location at the University of Iceland.