

Water Distillation Using Waste Engine Heat from an Internal Combustion Engine

by

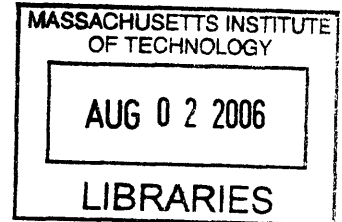
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# Water Distillation Using Waste Engine Heat from an Internal Combustion Engine

by

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## **ABSTRACT**

To meet the needs of forward deployed soldiers and disaster relief personnel, a mobile water distillation system was designed and tested. This system uses waste engine heat from the exhaust flow of an internal combustion engine to vaporize water for the purpose of removing impurities. The vapor is condensed back down to water in a finned condenser that experiences forced convection.

The system pumps heat transfer oil through a 0.61 meter long, cross flow, annulus-type heat exchanger installed over a section of exhaust pipe where the oil experiences a  $\Delta T$  of  $7^{\circ}\text{C}$ . The hot heat transfer oil is then piped to a boiler where it releases its heat to the water and returns to the exhaust heat exchanger to be reheated. Testing demonstrated that the system has a heat up time of 30 minutes, and a steady state distillation rate of 2 gallons per hour. In steady state, the system removes and transfers heat from the exhaust at a rate of 4600 Watts.

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## **1.0 Introduction**

Commercial water purifiers on the market today either require their own source of power, or any number of expensive filters, chemicals, and membranes. For small scale water purification, hand-pump units can be used, but these devices are typically fragile and need frequent filter replacement. For large scale water purification, reverse osmosis is the preferred method. Reverse osmosis is very expensive and requires large pieces of equipment to be shipped into the area where water is to be purified. There are circumstances, however, where neither of these extremes properly addresses the needs of the particular situation. Disaster relief groups and forward deployed military units frequently require more water than hand-pumps can realistically provide, but do not have the time or the supply chain necessary to make use of reverse osmosis water purification.

Some of the very first disaster relief units on the scene of a catastrophe and military units on the scene of a battle are wheeled or armored units. These units travel in wheeled vehicles (usually humvees) that not only provide transportation for relief workers and soldiers, but provide power for communication and medical equipment. These vehicles are used nearly around the clock in many regions of the world. The heat from these vehicles' engines is currently dumped into the atmosphere where it does no constructive work. This project involved designing, building and testing a system that uses the waste engine heat from an internal combustion engine to distill dirty water. This system would not decrease vehicle performance, but rather make use of an energy source currently not being utilized. This purification system would be of great use in areas of the world where other methods of water purification are simply infeasible or inaccessible.

## **2.0 Theoretical Modeling**

The distillation system is comprised of three major parts: a heat exchanger that uses heat transfer oil to remove heat energy from the exhaust of an internal combustion engine, a boiler where this hot oil releases its heat to vaporize dirty water before cycling back to the heat exchanger, and a condenser to condense the vapor back down to water. A pump provides the pressure gradient to drive the heat transfer oil throughout the system.

Theoretical modeling is essential to determine the necessary length of the exhaust system heat exchanger to ensure adequate heat transfer given the flow rate determined by the system pump. Modeling is also necessary to determine the length of tubing required for the oil to fully release its heat to the water in the boiler. Finally, the condenser must be modeled to ensure that the vapor fully condenses instead of being lost to the environment.

## ***2.1 Exhaust Pipe Heat Exchanger***

The major design work on this project went into developing the exhaust system heat exchanger. As the core of the distillation system, it is essential that the heat exchanger removes enough heat from the exhaust stream to distill water at the desired rate of 1 gallon of water per hour.

The total energy input required to vaporize a certain amount of liquid is a function of the latent heat of vaporization for that liquid,  $h_{fg}$ , and the mass,  $m$ , of liquid present.

$$Q = h_{fg} m_{liquid} \quad (1)$$

To convert this total energy input to a rate of energy addition, one needs only to divide by the total time,  $t$ , in seconds.

$$\dot{Q} = \frac{h_{fg} m_{liquid}}{t} \quad (2)$$

In this project, the desired rate of water distillation is one gallon per hour. To find the heating rate necessary to achieve this goal, the mass of one gallon of water is used for  $m$ , and 3600 seconds is used for  $t$ .

Once the desired rate of heating has been determined, it is necessary to determine what temperature drop in the exhaust stream will transfer this much heat to the working fluid. The model assumes no losses, which is not the case, but for the purpose of approximating basic parameters of the system, this assumption will suffice. Based on this no losses assumption, the internal energy increase of the working fluid is equivalent to the internal energy loss of the exhaust stream.

$$Q_{exhaust} = m_{exhaust} c_p (T_{in} - T_{out}) \quad (3)$$

In this equation,  $m_{exhaust}$  is the mass of the exhaust,  $c_p$  is the specific heat capacity of exhaust (approximated as the specific heat capacity of air), and  $T_{in}-T_{out}$  is the temperature drop of the exhaust stream due to interaction with the working fluid. It is immediately apparent that the mass of exhaust is nearly impossible to quantify, so it is more useful to make **Equation 3** dependent on time by converting it to a rate. This is done by replacing  $m$  with  $\dot{m}$ , the mass flow rate. As the specific heat capacity and the temperature difference are constant in time, this small change yields the rate at which the exhaust stream transfers heat to the working fluid.

$$\dot{Q}_{exhaust} = \dot{m}_{exhaust} c_p (T_{in} - T_{out}) \quad (4)$$

The mass flow rate of exhaust from an engine under light load is between 0.015 and 0.03 kg/second (Hendricks, 2006). For the purposes of this project the semi-conservative figure of 0.02 kg/second was used.

The rate of energy transfer required for the pre-determined rate of vaporization found in **Equation 2** is then substituted in for  $\dot{Q}_{exhaust}$  in **Equation 4**. This again assumes that no energy is lost from the working fluid as it is transferred to the boiler and into the water itself. After substituting the result from **Equation 2** into **Equation 4**, one solves for  $\Delta T$ :

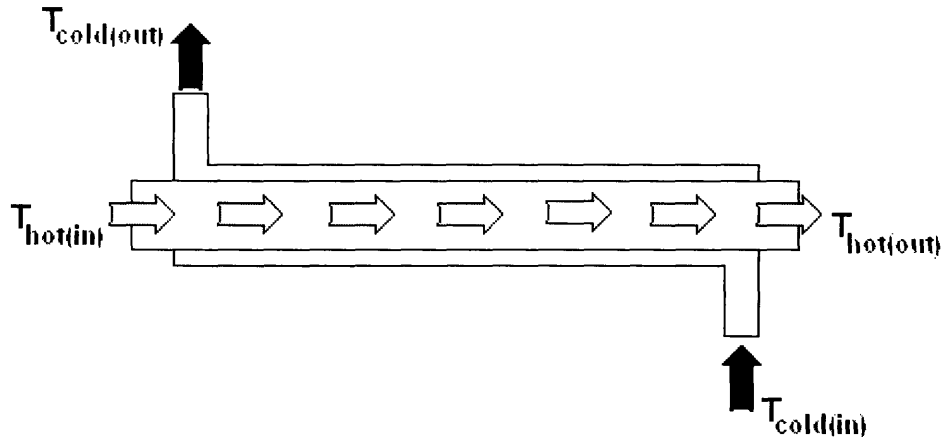
$$\Delta T_{exhaust} = \frac{\dot{Q}}{c_p \dot{m}_{exhaust}} \quad (5)$$

In order to vaporize one gallon of water per hour, this is the change in exhaust stream temperature required to transfer enough internal energy from the exhaust to the internal energy of the water. Solving this equation yields a temperature drop of 104C.

To find the resulting rise of the temperature of the heat transfer oil as it travels through the heat exchanger and gains heat from the exhaust stream, **Equation 5** is again used but with the  $c_p$  and  $\dot{m}_{oil}$  of the heat transfer oil instead of the exhaust.  $\dot{m}_{oil}$  for the transfer oil is determined from the flow rate of the pump. The gear pump is capable of driving water at 3 gpm. To determine what mass flow rate of oil this corresponds to, simple conversions are necessary.

$$\dot{m}_{oil} = \frac{3 \text{ gal}_{water}}{1 \text{ min}} * \frac{8.4 \text{ lbs}}{\text{gal}_{water}} * \frac{1 \text{ kg}}{2.2 \text{ lbs}} * \frac{1 \text{ m}^3}{1000 \text{ kg}_{water}} * \frac{950 \text{ kg}_{oil}}{1 \text{ m}^3} * \frac{1 \text{ min}}{60 \text{ sec}} = 0.1814 \frac{\text{kg}}{\text{sec}} \quad (6)$$

$\dot{m}_{oil}$  is substituted into **Equation 5**, along with the  $c_p$  of the heat transfer oil.  $\dot{Q}$  remains the same, as the total energy drop from the exhaust stream is equivalent to the total energy gain of the heat transfer oil, under the assumption that there are no losses in the system. Solving **Equation 5** for the oil flow yields a temperature increase of 7°C from the inlet to the outlet of the heat exchanger.



**Figure 4:** The temperature of each fluid at the inlet and outlet is shown. The increase in the temperature of the cold stream (the working fluid) is 7C, while the corresponding decrease in the temperature of the exhaust is 104C.

Now that the differences in the inlet and outlet temperatures for both fluids are known, one can calculate the actual temperatures at these points for both flows. Measurements show that the temperature of the exhaust going into this section of exhaust piping can be as high as 750°C. With an exhaust stream temperature drop of 104°C, this equates to an exhaust outlet temperature of 646°C. For the oil, the working fluid has just come from the boiler, so one can assume that it has released all of its heat to the water and its heat to the water so the exchanger inlet temperature is that of the temperature of the boiler (in the case of boiling, 100°C). With a temperature rise of 7°C, the outlet temperature of the oil is 107°C.

At this point a governing equation is needed that brings together the chosen flow rate of the system and the resulting temperatures with the heat transfer coefficients and the geometry of the heat exchanger. From the definition of cross flow heat exchanger effectiveness, it is determined that (Lienhard, 2005):

$$\frac{T_{exhaust(out)} - T_{oil(out)}}{T_{exhaust(in)} - T_{oil(in)}} = \exp\left(\frac{-PLU}{\dot{m}_{exhaust} c_p}\right) \quad (7)$$



The left side of **Equation 7** is constant, as it is determined by the inlet and outlet temperatures of each fluid found in the previous section. The only variables on the right hand side are the length,  $L$ , of the heat exchanger, and the wetted perimeter,  $P$ .  $U$  is the overall heat transfer coefficient of the system which accounts for both the convective resistance from the exhaust stream to the exhaust pipe wall and from the exhaust pipe wall to the heat transfer oil. The conductive resistance through the exhaust pipe wall is disregarded as it is extremely small compared to the other resistances. In order to find the necessary minimum length of the heat exchanger, and to ensure that the heat exchanger will fit over the available exhaust section without significant vehicle modification, this equation must be solved for  $L$ .

To determine  $U$ , the thermal resistances on the inside and the outside of the exhaust pipe need to be found. First, looking at the exhaust flow through the exhaust pipe, the Reynolds number is calculated for this flow. The Reynolds number is used to determine whether the flow is laminar or turbulent, to ensure that the correct correlation is used.

$$\text{Re} = \frac{4\dot{m}}{\pi D_h \mu} \quad (8)$$

Where  $\dot{m}$  is the mass flow rate of the exhaust,  $D_h$  is the hydraulic diameter of the duct (in the case of a circular duct, the hydraulic diameter is the same as the internal diameter of the exhaust pipe), and  $\mu$  is the dynamic viscosity of the exhaust. For this particular case, the Reynolds number is 9900, so the flow is turbulent. This turbulence is to be expected given the vibrations in the exhaust as well as the relatively high mass flow rate for such a low density substance. Using the correlation for turbulent flow in a smooth duct (assuming fully developed hydrodynamics), the Nusselt number is given by (Lienhard, 2005):

$$Nu = \frac{\left(\frac{f}{8}\right)(Re-1000) Pr}{1 + 12.7 \sqrt{\frac{f}{8}} (Pr^{\frac{2}{3}} - 1)} \quad (9)$$

where  $f$  is the friction factor for the pipe,

$$f = (0.79 \ln(Re) - 1.64)^{-2} \quad (10)$$

and  $Pr$  is the Prandtl number for exhaust, approximated as the Prandtl number for air at the same temperature. The resulting heat transfer coefficient for the internal exhaust flow,  $h_{insidepipe}$ , is given by:

$$h = Nu \frac{k}{D_h} \quad (11)$$

where  $k$  is the conductivity of the exhaust in this case. In turn, the thermal resistance for the exhaust side of the pipe is provided by the equation for convective thermal resistance.

$$R_{convection} = \frac{1}{hA} \quad (12)$$

In this equation,  $h$  is the convective coefficient, and  $A$  is the surface area experiencing convection. For the exhaust flow,  $h$  is 31.68 W/m<sup>2</sup>K

To find the convective coefficient off the outside of the exhaust pipe, the math is nearly the same. The Reynolds number for the flow is again found by using **Equation 8**, but in this case  $\dot{m}$  is now the mass flow rate of the oil,  $\mu$  is the dynamic viscosity of the heat transfer oil, and  $D_h$  is the hydraulic diameter of the annulus. For an annulus,  $D_h$  is defined as the difference between the IDs of the outer and inner tubes (Cravhalo, 2006).

$$D_h = D_o - D_i \quad (13)$$

For the oil flow, the Reynolds number is determined to be 13500, so the flow is turbulent. The Nusselt number for turbulent flow in an annulus (with a perfectly insulated outer tube) is given by a similar correlation used for turbulent flow in a smooth duct (Cravhalo, 2006).

$$Nu = \frac{\left(\frac{f}{8}\right)(Re-1000) Pr}{1 + 12.7 \sqrt{\frac{f}{8}} (Pr^{\frac{2}{3}} - 1)} \left[ 0.86 \left(\frac{D_i}{D_o}\right)^{-0.16} \right] \quad (14)$$

Using this correlation, the Nusselt number is calculated, and the corresponding convective coefficient and thermal convective resistance are determined.  $h$  for this situation is found to be 129 W/m<sup>2</sup>K.

The overall thermal heat transfer coefficient,  $U$ , is found by using Newton's Law of Cooling. Newton's law of Cooling states that:

$$\dot{Q} = hA(T_s - T_\infty) \quad (15)$$

where  $T_s$  is the temperature of the surface over which the fluid is flowing, and  $T_\infty$  is the temperature of the fluid. In the case of multiple heat transfers, the single heat transfer coefficient,  $h$ , is replaced by the overall heat transfer coefficient,  $U$ . **Equation 15** becomes:

$$\dot{Q} = UA(T_s - T_\infty) \quad (16)$$

Just as the convective resistance,  $R_{convection}$ , for a flow is  $\frac{1}{hA}$  (see **Equation 12**) the total

thermal resistance for a flow is  $\frac{1}{UA}$ . In other words:

$$\dot{Q} = UA(T_s - T_\infty) = \frac{T_s - T_\infty}{\sum R_{convection}} \quad (17)$$

Manipulating **Equation 17**, one finds that:

$$\frac{1}{UA} = \sum R_{convection} \quad (18)$$

As the two resistances in this situation are in series,  $\sum R$  is equal to the sum of the two resistances.

$$\frac{1}{UA} = \sum R = \frac{1}{h_{insidepipe} A} + \frac{1}{h_{outsidepipe} A} \quad (19)$$

As the wall of the exhaust pipe is very thin, the heat transfer surface area,  $A$ , is assumed to be the same for both the exhaust and the oil flow. As it is present on both sides of the equation,  $A$  can be removed from **Equation 19**. From this, it follows that:

$$U = \frac{1}{\frac{1}{h_{insidepipe}} + \frac{1}{h_{outsidepipe}}} \quad (20)$$

Having already solved for both heat transfer coefficients, the coefficients can be put in **Equation 20**. The total heat transfer coefficient,  $U$  is determined to be 25.5. **Equation 20** is then put into **Equation 7**. At this point, there are only two variables remaining in the equation, and if a value for one is chosen, the equation can be solved for the other. In the case of this project, the whetted perimeter,  $P$ , was determined based on the size of the existing exhaust system, and the amount of space available under the vehicle. Changing the perimeter also has little effect as it changes several other values that have the effect of canceling each other out.  $P$  was set based on a 3.5 inch pipe (the maximum size that

would fit) fitted over the 2.5 inch exhaust pipe. This left just under a 0.5 inch gap between the two pipes for the oil to flow. Such a gap size represented a good balance of low pressure flow, and outlet temperature of the oil given the 0.1814 kg/s mass flow rate of the oil.

$L$  on the other hand greatly affected the rate at which water would be vaporized. As this system was intended to be compact and unobtrusive, the minimal length of the heat exchanger necessary to achieve the desired distillation rate was needed. It was determined from all the above calculations, that a 0.61 meter long annulus heat exchanger was needed to obtain the desired distillation rate of 1 gallon per hour.

## **2.2 Boiler**

It is necessary to develop a model for the copper tube in the boiler to determine the proper length of copper tubing to be used. As the heat exchanger provides a  $\Delta T$  of  $7^{\circ}\text{C}$  (2333 Watts), the oil is assumed to enter the boiler  $7^{\circ}\text{C}$  warmer than the ambient water. The copper tubing is long enough to allow the oil to release all this heat to the water. The oil releases its stored heat, and returns to the heat exchanger at the temperature that it raised the water bath to at its last pass (the oil can never have a lower temperature than the water, except during start-up). After multiple cycles, the temperature of the water has risen to near  $100^{\circ}\text{C}$ . At this point, the temperature of the water no longer rises, and released heat is in the form of latent heat of vaporization. However it is critical to determine the length of the copper tubing necessary to release all of the heat from the oil to the water during boiling. If the copper tubing is too short, the oil temperature will continue to slowly rise with each cycle until the pump is damaged or destroyed.

To determine the length of tubing required to release all exhaust heat to the water during boiling, the boiler is modeled as natural convection over a horizontal cylinder. The Nusselt number correlation for natural convection over a horizontal cylinder is (Cravahlo):

$$Nu = 0.36 + \frac{0.518 Ra^{\frac{1}{4}}}{\left[ 1 + \left( \frac{0.559}{Pr} \right)^{\frac{9}{16}} \right]^{\frac{1}{9}}} \quad (21)$$

Where Pr is the Prandlt number for air, and Ra is the Rayleigh number, defined as:

$$Ra = \frac{g\beta\Delta TD^3}{\alpha\nu} \quad (22)$$

In this equation  $g$  is the gravitational constant,  $\beta$  is the expansion coefficient of the liquid surrounding the cylinder (in this case, water),  $\Delta T$  is the difference in temperature between the bulk temperature of the water bath (100°C) and the log mean temperature ( $T_{LM}$ ) of the oil in the copper tubing,  $\alpha$  is thermal diffusivity of water, and  $\nu$  is the kinematic viscosity of water. The log mean temperature is defined as:

$$T_{LM} = \frac{T_{out} - T_{in}}{\ln\left(\frac{T_{out}}{T_{in}}\right)} \quad (23)$$

With an inlet temperature of 107°C and an outlet temperature of 100°C, the log mean temperature is found to be 103.46°C, yielding a  $\Delta T$  for the system of 3.46°C.

Substituting the proper values for the Rayleigh and Prandlt numbers into **Equation 21** yielded a Nusselt number 11.3. Using **Equation 11**, the corresponding heat transfer coefficient was determined to be 810 W/m<sup>2</sup>K. Replacing  $\dot{Q}$  with 2333 Watts (the energy removed from the exhaust stream),  $h$  with 810 W/m<sup>2</sup>K, and  $(T_s - T_\infty)$  with the boiler's  $\Delta T$  of 3.46°C, and solving for A, yields the tubing surface area necessary to fully release the exhaust energy.

$$A = \pi DL = \frac{2333}{801(3.46)} \quad (24)$$

Dividing this area by the circumference,  $\pi D$ , of the 0.375 inch tube gives the necessary length of tubing required. The minimum length of copper tubing to ensure that the system will run indefinitely under boiling conditions without further heating up and subsequent pump damage is 9.25 meters (30 feet).

### **2.3 Condenser**

To get a baseline idea for the size and complexity of the condenser, an initial model was based on a straight section of 0.375 inch copper pipe experiencing natural convection with 30°C air. The Nusselt number correlation for natural convection over a horizontal cylinder is again given by **Equations 21** and **22**. In the case of the condenser,  $\beta$  is the expansion coefficient of the air surrounding the condenser,  $\Delta T$  is the difference in temperature between the ambient air temperature and the temperature of the steam in the copper tubing (100°C),  $\alpha$  is thermal diffusivity of air, and  $\nu$  is the kinematic viscosity of air. Substituting in the appropriate values for air at a temperature of 30°C yields a Nusselt number of 3.04. Using **Equation 11**, the heat transfer coefficient is found to be 8.44 W/m<sup>2</sup>K.

To find the length of copper tubing necessary to condense the steam, the same process used to determine the length of the boiler tube is used, but with the proper values for the condenser. Assuming an air temperature, of 30°C and a steam temperature of 100°C (a  $\Delta T$  of 70°C), the total area,  $A$ , necessary to remove this much heat with only natural convection is 4.14 meters. Provided that the 0.375 inch copper tube has a circumference of 0.0299 meters, nearly 140 meters of pipe would be required.

To reduce the amount of pipe required, it was necessary to raise the heat transfer coefficient. In fact, for a reasonable length of 2 meters of pipe to condense the steam, it was necessary that the heat transfer coefficient be increased to 556 W/m<sup>2</sup>K, a number nearly impossible to achieve with even forced convection over a straight pipe. A finned array was chosen, with large 4 cm square fins, 0.5mm wide, spaced 0.5 cm apart. This array was tested to ensure effectiveness, yet it was not directly modeled.

## 2.4 Pump Pressure Head

To provide a rough model of the pressure head in the system in order to choose the correct pump, the Darcy-Weisbach Law was used (Cravalho, 2006).

$$\Delta P = \rho_{oil} f \left( \frac{L}{D} \right) \frac{v_{avg}^2}{2} \quad (25)$$

In this equation  $f$  is the pipe wall friction factor defined in **Equation 10**,  $\rho_{oil}$  is the density of the heat transfer oil,  $L$  is the total length of tubing in the system,  $D$  is again the internal diameter of the copper tube, and  $v_{avg}$  is the average velocity of the oil, given by:

$$v_{avg} = \frac{\dot{m}_{oil}}{\rho_{oil} A} \quad (26)$$

where  $A$  is the cross sectional area of the tubing in which the oil flows. Minor head losses also contribute to the pressure head of the system, but as there were only a few major bends in the system, and valves with low minor head loss coefficients were used, this contribution is ignored as quick calculations showed that it would only contribute another 5-10% to the pressure drop in the system. With a total of 15 meters of copper tubing in the system, **Equation 25** yielded a pressure drop of 25 psi. It was decided that a pump capable of driving at least a 30 psi head should be used.



## **3.0 Experimental Apparatus and Procedure**

This section describes the vehicle for which the system was built, as well as traces the evolution of the design of the system from its inception to its creation and testing.

### ***3.1 Experimental Apparatus***

The test vehicle for this project was a 1998 Dodge Ram 1500, with a 5.2L V-8 engine and a 2.5 inch exhaust system. As the water distillation system is intended for use on large military and disaster relief vehicles, the Dodge Ram was a reasonable test vehicle to use for this project. Its engine is slightly smaller and less powerful than these vehicles so it would provide a conservative figure for expected system performance. All design work, system construction, and testing was performed by Kevin Mears and Stephen Samouhos. Steve Haberek, a technical instructor with the Pappalardo Lab at MIT, was responsible for the welding on the exhaust system heat exchanger.

### ***3.2 Experimental Procedure***

The overarching objective of this project was to develop and test a mobile water distillation system that operated off an automobile internal combustion engine. The goal was to design a system made of easily produced or off-the-shelf components that drew minimal parasitic power from the engine, while producing clean water at a rate of at least 1 gallon per hour. Looking at a few numbers, it was determined that internal combustion engines are on the order of 20% efficient, meaning that almost 80% of the energy released by combustion is wasted and leaves the engine as heat. This low efficiency made it clear that the optimal system would scavenge some of this waste heat from the engine, and use that heat to boil the water in the distillation process. A parasitic heater that drew additional power from the engine to operate would be incredibly wasteful and would significantly affect the vehicles fuel efficiency in a negative manner. It made much more sense to use the heat that currently going to waste.

The initial stages of the project focused on finding the ideal source of heat for the system. The second major portion of the project involved designing a system around the chosen heat source using easily obtainable resources and supplies. The final stage of the project included building several successively more robust systems and evaluating them for performance, integrity, and reliability.

### **3.2.1 Finding an appropriate heat source**

Heat from a vehicle's internal combustion engine is removed in a few ways. Over 1/3 of the energy released in the combustion process conducts through cylinder walls and the engine block, where it is removed by way of the engine's liquid coolant system. This extracted heat is then dumped into the atmosphere by way of the finned heat exchanger known as the radiator. Heat also is carried away in the engine oil, which flows over and around all major internal engine components. The coolant system is rather efficient, and the operating oil temperature typically is below 100-150°C. The other main source of heat from an internal combustion engine is the exhaust resulting from combustion. These exhaust gasses can be upwards of 700°C.

As the coolant system removes the majority of the combustion and friction heat from the engine, the initial focus of the project was to use this system as the heat source for the distillation process. In addition, the proximity of this heat source to the radiator and to air flow into the engine bay would aid in integrating and installing the condensing portion of the system, the part of the system responsible for removing the sensible and latent heat necessary to change the steam back down to water.

The fundamental problem with the cooling system as a heat source was the fact that the coolant system is pressurized to keep the coolant water from vaporizing. The higher pressure keeps down the temperature of the coolant water. The working temperature was right around 100°C. Using a heat source at nearly the same temperature

as the boiling point of pure water was just not practical, and even if it worked, it would not boil water at nearly the desired rate. The only way to use the source would be to boil water in a reduced pressure vessel. Such a vessel would require specialty pumps and complex, low tolerance manufacturing. As the project called for simple, robust design, it became clear that the coolant system was not to be the heat source for the system.

The search for a heat source then turned to the engine oil. This was the natural next choice, as many inexpensive off-the-shelf products exist for tapping into an engine's oil supply for the purpose of extracting and returning oil to the engine block. Automotive oil coolers use a device known as a sandwich plate that fits between the bottom of the engine block and the oil filter. Driven by pressure gradients created by the sump, the sandwich plate redirects the oil that would normally travel directly to the oil filter. This redirected oil is sent instead through a high temperature hose that is connected to a small radiator-like heat exchanger. This exchanger removes much sensible heat from the oil, preserving the thermal integrity of the lubricant. The oil then passes through the filter, and returns to the engine block. The existence of such a simple and time-tested method for removing engine oil and passing it through a heat exchanger lent itself to modification for the purposes of heating and boiling water.

After installing an oil cooler on the test vehicle as well as a number of thermocouples, it was found that the device was so effective (in air) that oil temperatures hovered below 100°C. Submerging the heat exchanger in water would raise the heat transfer, further lowering the operating temperature of the oil. While it could be possible to insulate the boiler in such a way to capture the heat released by the oil and to eventually achieve the onset of boiling, this process would take more time than necessary to achieve the desired rate of one gallon of distilled water per hour. In addition, when the engine was under a light load (when the vehicle was idling or driving at low speeds), oil temperatures, even without the oil cooler, would languish around 80-90°C. It quickly became clear that the engine oil was not to be the heat source of choice for the distillation system.

The final significant source of heat from an internal combustion engine is the exhaust created by combustion. As the exhaust gas has a very low density, hence a low mass flow rate, the maximum amount of energy it can transfer (especially to a much more dense liquid) is limited. For this reason, it was the last source of heat to be considered. In addition, the difficulty of harnessing the energy in exhaust presented a significant number of obstacles that needed to be worked around. Could the water be flash boiled off the exhaust system itself? Could the exhaust gas travel through a piping network built in a boiler? Could the exhaust system be wrapped in copper coil, and the water circulated through the copper? These were all questions that needed to be addressed in order to use the exhaust. However, as the only heat source that consistently sees temperatures in great excess of 100°C it was the only reasonable source to use.

Once the heat source was chosen by process of elimination, the design focused on how best to use the exhaust to heat the water. Wrapping sections of the exhaust system proved extremely difficult, and the contact area for conduction between the aluminized steel exhaust piping and the copper was extremely small, no matter how tightly the copper tubing was wound. Making a heat exchanger through which the exhaust traveled involved complex manufacturing, and would affect the back pressure in the exhaust system. Tampering with back pressure negatively affects many aspects on engine performance, and as the distillation system was intended to be as unobtrusive as possible, it quickly became evident that significantly modifying the geometry of a vehicle's exhaust system was just out of the question. Whatever device was designed would have to work with the exhaust setup of the vehicle and not require complex manufacturing nor significant modification of existing exhaust system.

In addition, another problem arose. Water could not be pumped directly through meters of tightly wound tubing as the pressure head would be do great for most pumps. Also, in the event of a cold start up, the exhaust piping would be cold, and water would not boil, but simply enter the condenser as dirty water. Designing a control system that either constantly monitored the temperature of the exhaust and used this information to determine pumping rates through the coil, or one that measured moisture content of the

steam at a particular point in the coil and adjusted flow rates would be beyond the scope of this project.

Water could not be exposed directly to a section of the exhaust. Even if the water exposed directly to the exhaust did boil, collecting and transporting the stream uphill to a storage vessel or condenser would be very difficult considering that the exhaust system is typically the low point of a vehicle and pumping a steam water mix is a very complicated and maintenance intensive operation. Furthermore, if water were to be boiled directly on a section of the exhaust network, impurities from the dirty water would quickly collect on the exhaust pipe and decrease system performance and heating rate. While such collection of impurities is to be expected, this particular system would be very difficult to clean unless the vessel that was fitted around the exhaust pipe to hold the water was easily removable. In addition to being removable, the vessel would have to have a watertight seal, be able to withstand temperatures up to 700°C, and be robust enough to survive under a vehicle. Such a vessel would require much more advanced manufacturing than was appropriate for this project. The whole objective was to create a simple, easy to use, cost effect, and robust system, and the direct exposure method of boiling seemed to violate all of these requirements. It became clear that the only viable answer was that a working fluid was needed to transfer the heat from the exhaust to the water.

As the wound copper method was already discarded after tests demonstrated the difficulty in obtaining a reasonable area of surface contact, an extremely large pressure head, and the fragility of exposed copper tubing on a part of a vehicle that could experience terrain impacts, this method was not considered for heating whatever working fluid that would be used in the system. The only reasonable and simple way to heat the working fluid was using a direct exposure method. While this method would not work with direct boiling, it would work well for another working fluid. As the working fluid would not interact directly with the dirty water, the heat vessel installed on the exhaust system to hold the working fluid would not have to be detachable for cleaning. This fact greatly simplified the manufacturing process. In addition, after being heated, the working

fluid itself could be pumped to another location, alleviating the need to worry about transporting and pumping steam. The working fluid would never vaporize, so it could easily be pumped throughout the system.

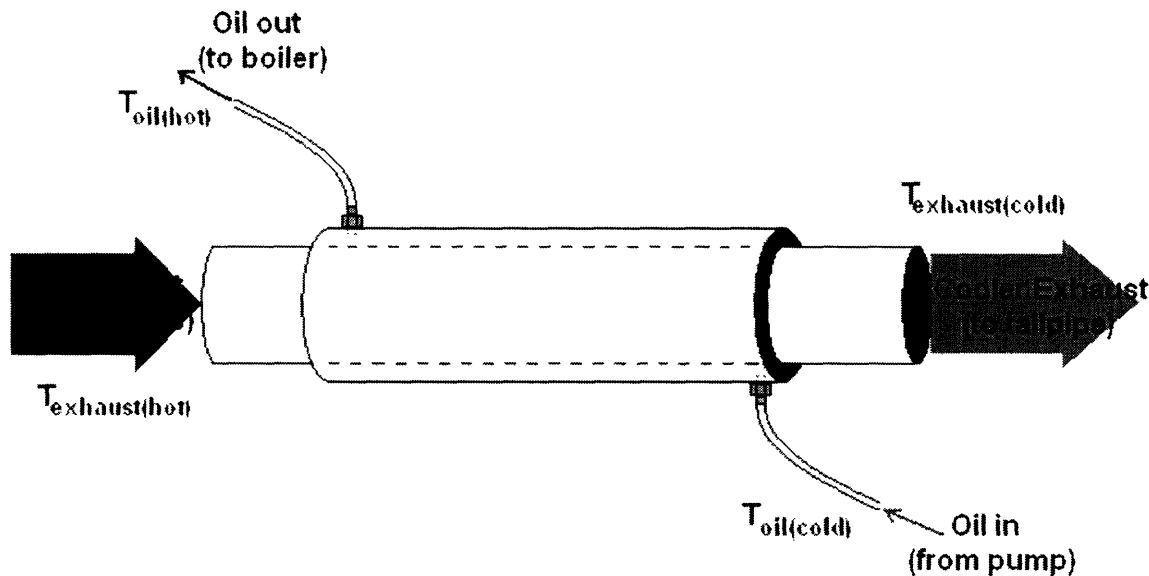
### **3.2.2 Choosing a Working Fluid**

The question now became: “what working fluid can withstand the extremely high temperatures of the exhaust system without breaking down or vaporizing, is easy to pump, and in the very unlikely event that some of the fluid enters the distilled water, is non hazardous?” Only one fluid met these requirements: Duratherm 630. Duratherm 630 is a non-toxic heat transfer oil that can withstand bulk temperatures of over 630 degrees Fahrenheit in an open system without breaking down or vaporizing. While 630°F is significantly less than the 500-700°C that the exhaust pipe walls can see under high load conditions, it is much higher than the bulk temperature of the working fluid to be used in this system. Conversations with Duratherm engineers and private tests showed that if the flow rate of the Duratherm over the pipe could be kept high enough, that the heat transfer oil would neither break down nor boil. Another desirable quality of the Duratherm is that its viscosity is relatively constant at high temperatures. This constant viscosity is essential for proper pumping. If the viscosity of the working fluid were to drop significantly, the pump would have difficulty maintaining the desired flow rate.

### **3.2.3 Exhaust System Heat Exchanger**

After the heat source was chosen, the issue of designing and sizing the proper heat exchanger became the focus of the project. The only section of exhaust that was straight enough and long enough to install any sort of reasonable size heat exchanger was downstream of the first catalytic converter, just in front of the second catalytic converter on the exhaust section connecting to the Y Pipe. This section was roughly three feet long, and as it contained no catalytic converters or oxygen sensors, it could be modified without significantly affecting the performance or monitoring of the exhaust and

emissions system. It was decided that a larger pipe would be fit over the section of exhaust pipe, and end caps would be welded on. Ports would be drilled and tapped into the top and the bottom, and the male ends of high temperature heat transfer oil hoses would be attached to the ports.



**Figure 1:** A large diameter pipe is fitted over a section of exhaust, and functions as a cross flow heat exchanger to remove heat from the exhaust.

The original intention of the project was to have the larger pipe be removable in case of fouling or the need for cleaning, but it was eventually decided that as long a non corrosive working fluid was used, and as long as the working fluid was not exposed to excessive dirt or impurities, the need for removal was not essential.

The diameter of the exhaust pipe on the test vehicle is 2.5 inches. It was decided that a 3.5 inch OD pipe be fitted over the exhaust pipe. This would leave a gap of just under 0.5 inch between the outside of the exhaust pipe and the inside of the heat exchanger pipe. Such a gap would not greatly restrict flow by creating a huge pressure

head. In addition, this was the largest pipe size that would clear the underbody of the test vehicle. A 3.5 inch Schedule 40 black steel pipe was used.

Another fear was that if the gap was too small, the working fluid would have to be pumped through quickly so as not to become too hot and breakdown. Pumping a fluid quickly through a small gap requires a large pressure head; a head larger than many small DC pumps could provide. In addition, the increased flow rate could mean that the hot working fluid would not be able to completely release its heat energy to the water. During steady state operation, the oil would return to the heat exchanger from the boiler with a slightly higher temperature than it had the previous cycle. The only way to compensate for this would be increasing the length of copper tubing in the boiler, a parameter limited by available space. In lieu of additional tubing, the increasingly hot working fluid would pass through the pump as it is sent back into the exhaust system heat exchanger. Few pumps, especially small DC pumps capable of driving the fluid near the desired flow rates, are capable of pumping such a high temperature liquid. The seals will begin to melt, and the electric motor may overheat. This was a very real concern, as if the system was to be robust and reliable; an overheating pump would simply not be acceptable. It was desirable that the oil temperature into the pump be kept below 120°C to preserve its longevity.

The length of the exhaust system heat exchanger was the next area of attention. If the heat exchanger was too long, the problems with an excessive pressure head, and excessive temperatures mention in the last section would again be concerns. If the heat exchanger was too short, not enough heat would be transferred to the working fluid to heat and boil the water at the desired rate. The theoretical model in **section 2.1** demonstrated that the necessary energy transfer rate was to be obtained by using a heat exchanger 0.61 meters in length.

Determining how to install the outer pipe over the exhaust system was again a process of elimination. As the distillation system was intended to be as unobtrusive as possible, the first attempts to attach the heat exchanger involved cutting the outer pipe in



half, and welding the two halves back together once the pipe was fitted over the exhaust. This was to allow installation without altering the existing exhaust system. It quickly became apparent however, that not only was cutting the pipe in such a manner difficult, but four feet of welding was much too labor intensive for this method to be feasible.

Other methods of attaching were debated before it was decided that the section of exhaust pipe over which the heat exchanger was to be fitted would have to be removed. Once the exhaust pipe section was removed from the vehicle, the black steel heat exchanger could be installed, and the whole unit could be put back up into the vehicle. The only issue with this method is that when installing a section of exhaust into a car, the new section must be longer than the missing section. This is so that the ends of the section to be installed can be expanded to fit over the ends of the existing exhaust system, and then welded in place (exhaust pipe is too thin to be effectively butt-welded).

This problem of how to install the unit into the vehicle's exhaust system can be dealt with in two ways. The first way involves using the original piece of exhaust piping that was cut out of the vehicle. Slightly larger diameter sections of exhaust pipe can be used as sleeves to put the original piece of exhaust (with the heat exchanger installed) back into the vehicle. This presents the problem of installing the heat exchanger over a dirty, rusted, and pitted section of exhaust. While the pipe can be cleaned, if it is not sanded down thoroughly, pieces of rust and corrosion could break off and clog the system and the pump. The best solution for optimal system performance is to install the heat exchanger over a new, longer section of exhaust piping, expand the ends of the new unit, and slide it over the existing exhaust system. The clean pipe ensures optimal system performance, and eliminates the time intensive cleaning process. In addition, new sections of exhaust pipe are inexpensive and easy to come by.

After it was decided to cut into the exhaust system to install the heat exchanger section, end caps needed to be designed to affix the black steel pipe to the exhaust pipe. The easiest solution to the end cap issue was to simply use threaded steel end caps, with holes cut in them. The black pipe was threaded, and a hole the size of the exhaust pipe

was cut into each end cap on the lathe. The caps were screwed onto the ends of the black pipe, and sealant was used to obtain a good seal. The black pipe (with the end caps installed) was then slid over the new section of 2.5 inch exhaust pipe. The intention was to weld the end caps to the exhaust pipe, but a problem quickly became evident. While the end caps appeared to be of the same material as the black pipe, it became clear that they were not the same material. During welding, the end cap was quickly eaten away by the heat before the exhaust pipe was even hot enough to obtain a good weld. It was concluded that the end caps were made of ductile iron, a conclusion supported by the supplier. As the 3.5 inch pipe size is being phased out, no other threaded end caps could be obtained for the purpose of this project.

The alternative to the end caps was a steel disc, cut to fill the gap between the exhaust pipe and the black heat exchanger pipe. While requiring fabrication, these discs were welded nicely to the heat exchanger and the exhaust pipe, providing a water and pressure tight seal. The heat exchanger and exhaust pipe unit was then able to be installed into the vehicle.

To ensure that there were no air gaps between the heat transfer oil and the pipe wall in the heat exchanger, and to force the exchanger to be full of heat transfer oil at all times, the in port was tapped into the bottom, and the out port was tapped into the top of the black pipe. While this port arrangement placed a bit more load on the pump, it was essential in order to maintain even heating of the oil. Air pockets would make the gap in which the oil traveled smaller. A smaller oil gap could mean higher temperatures which could damage the oil. In addition, these air bubbles could enter the pump causing cavitation and decreased performance.

### **3.2.4 Choosing a Pump and Piping**

The pump was the next major component of the system to consider. The pump had to be 12v DC (to run off the vehicle's electrical system), pump at a 2-3 gallon per minute flow rate, be able to handle heat transfer oil, be able to operate continuously for a

few hours, and be able to handle relatively high temperatures. The pump would be placed downstream of the boiler, so the fluid it would see would be at the lowest temperature of any point in the system. Having just come from the boiler, the Duratherm had already released its heat to the water. For this reason, the highest temperature the pump would routinely see would be 100°C. Even finding a pump capable of handling a working fluid at 100°C was difficult. After considerable searching, a small DC gear pump capable of driving a 30 psi pressure head was acquired. This pump met every project need, and could handle the pressure head of the system assuming that 0.375 inch tubing was used.

0.375 inch tubing was used for the entire system as it offered an acceptable working pressure given the flow rate of the system determined by the pump. In addition, the pump was threaded for 0.375 inch hose, so using the same size eliminated the need for reducers or other significant minor head loss contributing pieces of plumbing. To preserve pump life by preventing cavitation, an expansion tank was placed just up-stream of the pump. Oil, coming from the boiler, would flow into the expansion tank, which was open to the atmosphere. Any air bubbles trapped in the tubing would escape to the environment. The pump would draw from the expansion tank, and send heat transfer oil to the heat exchanger to be heated. The expansion tank was insulated to prevent excess heat loss, and covered to prevent the introduction of impurities into the system.

It was debated whether or not a filter should be placed on the end of the suction line going to the pump, but this idea was voted down. If the filter was not constructed well, it could break down and be sucked into the pump where it could cause far more damage than small impurities. In addition, the filter could create so much head, that the pump may overheat, or otherwise function improperly. Finally, as there was no place in the system where impurities could realistically be introduced to the oil, it was decided that the oil filter just was not necessary.

### ***3.3 Boiler Design***

After the cooler oil from the expansion tank is pumped to the exhaust pipe heat exchanger to collect heat from the vehicle's exhaust, the oil is piped to the boiler. The boiler was a 6 gallon stainless steel pot containing 30 feet of coiled 0.375 inch copper tubing. The copper tubing was connected to the lid of the pot by way of a pressure fitting, a union, and a male threaded end. The flexible heat transfer hose from the pump attached to the male end of the copper coil connector. The hot oil was pumped into the boiler, traveled through the 30 feet of copper coil, and exited the pot through a similar connection as described before, but on the opposite side of the lid. As detailed in the theory section, 30 feet of tubing ensured that all the heat that was transferred from the exhaust to the oil was transferred from the oil to the water. This is essential to avoid temperature creep which could destroy the pump. The cooler oil coming from the boiler was then piped back to the expansion tank to repeat the cycle.



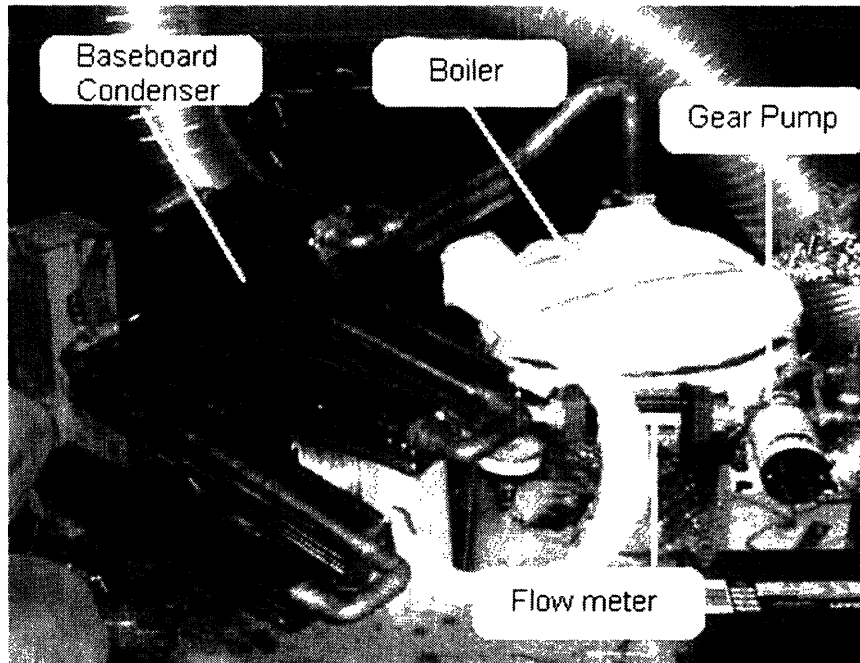
**Figure 2:** The 30 feet of wound copper tubing can clearly be seen. The wound copper has a larger radius at its base so that when much of the water has vaporized, the remaining water in the boiler still sees significant energy transfer. The funnel secured to the lid of the boiler pot is also visible.

After the cycle had been repeated numerous times, the temperature of the water in the boiler would begin to boil. The resulting steam had to be directed to a condenser, so its excess energy could be removed, and it could change phase back down to water. The first idea was to drill a hole in the lid, and attach a copper tube to direct the steam to the condenser. Tests showed however that much of the steam would condense on the lid as the hole was too small to allow much steam out of the boiler at once. It became apparent that a larger hole was needed. The next idea was to cut a large hole in the lid and to fabricate a steel cone that would fit over the hole and direct the steam to a section of copper tubing. With a thermal conductivity of 46 W/mK steel conducted heat so well that, even when insulated, steam would condense on the steel cone. The solution to the early condensing problem came in the way of a plastic funnel. Polyurethane has 1/100<sup>th</sup> the thermal conductivity of steel. In addition, the funnel already had the shape desired to direct the steam into a section of copper tubing. The plastic funnel was attached to the boiler lid with high temperature plumber's putty, and the entire lid/funnel unit was insulated.

### ***3.4 Condenser Design***

The funnel directs the steam into a 0.75 inch copper tube that is immediately angled down to prevent any steam from condensing on the walls of the copper and dripping back into the boiler. After the turn, the steam travels through a 6 foot length of 0.75 inch finned baseboard. The baseboard is cut into three two foot sections, and the sections are connected using small copper pipe sections and 90° copper elbows. The baseboard sections are all angled downward, so if steam condenses anywhere in the condenser, gravity will drive the water out of the baseboard and into the collection pot. To facilitate condensation, ducts were mounted to the vehicle and used in conjunction with single-ply helically corrugated aluminum duct hose to direct air over the finned array. As the system was built in the bed of the test vehicle for the purposes of this project, the duct work was to simulate the airflow the condenser would see if it was mounted in the front of the vehicle as it would in a production system. The project

vehicle was to be tested when in motion, but the forced convection off the finned array could also be replicated using an electric fan.



**Figure 3:** Shown is the boiler unit, finned condenser, and the gear pump used to drive the system. Not pictured is the exhaust pipe heat exchanger. The line carrying oil from the heat exchanger is visible in the far right of the picture, insulated with the aluminum tape insulation.

### ***3.5 System Troubleshooting***

After the entire system was designed, manufactured, and installed in the test vehicle, the system was filled with oil, and preliminary tests were run. One factor that quickly became clear was that the system took a considerable length of time to reach boiling and steady state operation. It did not help that the air temperature (as well as the starting temperature of the exhaust system, heat exchanger, and heat transfer oil) was  $-5^{\circ}\text{C}$  during these tests, but it took nearly one hour to bring 5 gallons of  $30^{\circ}\text{C}$  tap water to a steady boil. It became clear that much insulation was needed on all exposed parts of the

system. The heat exchanger was insulated well as were the heat transfer hoses, and the lines to and from the pump. In addition, the exhaust pipe of the test vehicle was wrapped with high temperature performance exhaust wrap from the headers down to the heat exchanger to prevent excess convection off the exhaust pipe to the atmosphere before the exhaust even reached the heat exchanger. Insulation proved to be extremely important, and the second test saw the water temperature rise in about half the time of the first test.

During the second test, another problem arose. The fuse between the vehicle's battery and the pump blew out. The fuse was replaced, but it quickly blew again. The fuses used were slow burn fuses, so it was clear that it was not the initial current spike of starting the pump that was the problem. The test vehicle had a faulty electrical system. Instead of spending hundreds of dollars troubleshooting this problem, a second battery was obtained to power the pump. This new standalone battery, albeit 12v, was much less powerful than the original battery used in the vehicle's electrical system. The reduction in power decreased the flow rate of the oil through the system by 0.5 gpm down to 1.6 gpm. Quick calculations showed that this flow rate would not significantly affect the system in a negative manner. The initial concern was that the oil could now become too hot in the heat exchanger, but the theory showed that the temperature increase was not too great to cause breakdown of the oil. In addition, the boiler was over-designed, so the length of copper coil in the boiler pot was long enough to remove the extra heat from the oil so as not to overheat the pump. The negative heat transfer effects of the slower flow rate were offset by the higher temperature of the heat transfer oil, and system performance remained relatively unchanged.

While running a few last shakedown tests before collecting data, it was discovered that the pump motor was becoming quiet warm. Some disassembly showed that the shaft powering the drive gears was the same shaft that went into the electric motor; heat from the working fluid was conducting up the shaft into the motor. To aid in cooling the motor, a second duct and duct hose was installed on the vehicle. This one directed air flow over the pump, to cool it by forced convection. Once in place, the motor

temperature, monitored by a thermocouple, stayed well below where it had been previously.

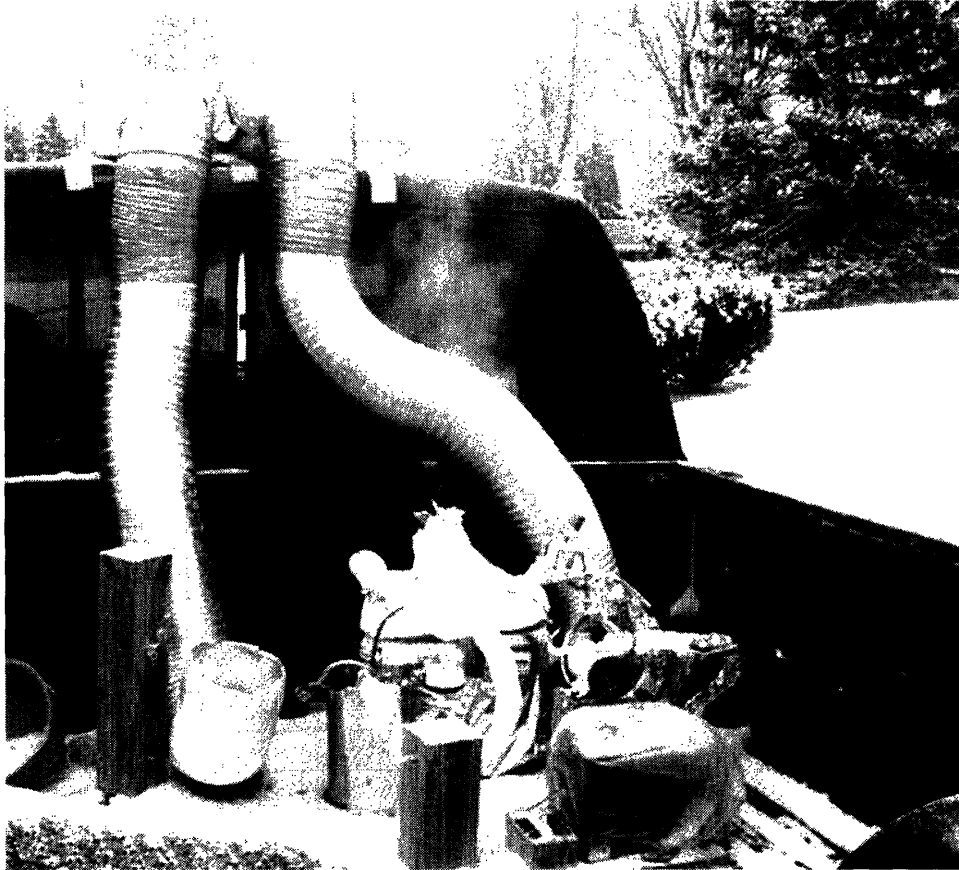
## 4.0 Results of Final System Testing

The final test of the system was conducted in Cambridge, MA in February. The day of the test saw snow, 25 knot winds, and  $-7^{\circ}\text{C}$  to  $-4^{\circ}\text{C}$  temperatures (without wind chill). While this made testing difficult, it would prove, indisputably, if the mobile distillation system worked. Thermocouples in were placed in the boiler to monitor the temperature of the water. The boiler was filled with 5 gallons of  $35^{\circ}\text{C}$  water (the temperature out of the tap) and the test vehicle was started and the pump was turned on. The vehicle was driven out of the parking lot, and onto the city roads. Within 15 minutes of slow stop and go city driving (being extra slow and careful because of the winter conditions), we got onto the highway. Once on the highway, the cruise control was set at just below 55 mph. The point of the cruise control was to eliminate the impulse to rev the engine to help our system heat up faster. The more load the engine is under, the more gasoline is combusted, and the higher the mass flow rate of the exhaust becomes. A higher exhaust mass flow rate equates to a higher heat transfer to the oil and subsequently to the water in the boiler. To best simulate light load, real world conditions so as to obtain very conservative results, engine RPM was kept as low and as constant as possible.

Initially, the temperature of the water began to decrease. The oil in the system has sat overnight, so it started at the same  $-7^{\circ}\text{C}$  as the exhaust pipe and the outside air. Once the engine was started, it took time before the exhaust pipe upstream of our system was warm enough to begin transferring significant amounts of heat to the heat exchanger. Once the section of exhaust over which the heat exchanger was installed began to rise in temperature, it took a while to heat up the oil to the initial temperature of the water. Due to the exceptionally low starting temperature of the system, this process took nearly 10 minutes. Once the oil and water temperatures had equalized, the temperature began to rise much faster. Once on the highway, temperatures began to rise by over  $10^{\circ}\text{C}$  per



minute, and within 30 minutes of total operation (including the 10 minute equalization time), the 5 gallons of water in the boiler was at a steady boil.



**Figure 5:** Shown here, with the condenser removed for demonstration purposes, is the system making steam after 30 minutes from a cold start-up in  $-7^{\circ}\text{C}$  winter weather.

On the return trip to Cambridge, the system continued to run, and by the time we arrived back at MIT, 30 minutes later, the system had distilled 1 gallon of water. In a total run time of 1 hour, from a cold startup, the system distilled 1 gallon. The energy input rate required to heat 5 gallons of water from  $35^{\circ}\text{C}$  to  $100^{\circ}\text{C}$  in 30 minutes is just over 2900 Watts, 600 Watts more than the energy input around which the system was designed. In addition, the energy input required to vaporize water at the rate of 2 gallons an hour (1 gallon in 30 minutes) is 4600 Watts. This rate far exceeded our goal, and can

be attributed to the numerous conservative estimates and approximations used throughout our theoretical prediction of system.

## 5.0 Discussion

The measured rate of energy transfer during the warm up period was 25% higher than the steady state energy transfer predicted by the model of the system. The steady state heat transfer rate of 4600 Watts is twice that predicted by the model. There are three significant points to discuss here: why the initial heat transfer rate was so much lower than the steady state, why the measured rate was so much higher than the modeled rate, and why the system worked well in testing without overheating.

There is a relatively simple explanation for the difference in energy input rates. When the vehicle was first started for the test, and the system was turned on, the engine and exhaust pipe were cold. It took a significant length of time before the exhaust pipe was warm enough to properly heat the oil. During testing it was noted that the system required 10 minutes before the temperature of the oil had matched the temperature of the water. Once equalized, the system heated the water to boil in 20 minutes. This time corresponds to an energy input rate of 4400 Watts, much closer to the steady state value. The model did not take into account this initial difference in temperatures.

The experimental distillation rate of 2 gallons per hour (in steady state) is twice as high as the rate around which the system was designed. This can be explained by conservative estimates made across the board, and on two major factors: the inlet exhaust pipe temperature, and the mass flow rate of the exhaust. As temperature measurements of the exhaust temperature could not be made at the point where the heat exchanger was installed, the wall temperature was used. If the actual exhaust inlet temperature was significantly higher, much more thermal energy would be transferred to the oil in the heat exchanger that was designed around the lower temperature. The mass flow rate was chosen from a range of typical automobile exhaust flow rates, as measuring a value

proved to be exceptionally difficult. However for a moderate sized V-8 engine, this number could have been a significant underestimate. If a mass flow rate of 0.03 kg/s (the high end of the light duty range) was chosen instead of the 0.02 kg/s value used for the theoretical calculations (the lower end of the range), the predicted energy transfer rate would increase by almost 1200 Watts. The more this flow rate was increased, the more heat could be extracted by the exhaust assuming the temperature drop remained unchanged.

During testing, the oil temperature did not greatly increase over time during the boiling process. A thermocouple in the expansion tank recorded a nearly constant oil temperature. Reasons the system temperature not rising include heat losses throughout the system. These losses countered the heat creep effect of the boiler coil length not being long enough to properly transfer all of the oil's heat energy to the water.

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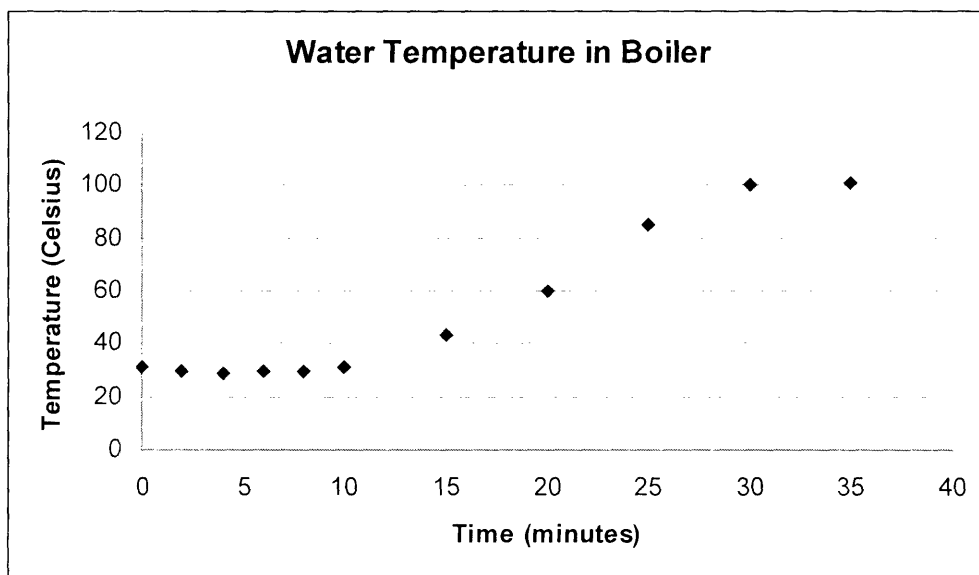
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## Appendix A- System test results

Time (minutes)	Temperature of Water (Celsius)
<i>Engine start</i>	31
0	31
2	30
4	29
6	30
8	30
10	31
15	43
20	60
25	85
35	101



## Appendix B- Duratherm properties

Property Vs. Temperature  
Duratherm 630 - Metric Units  
Temperature (Celsius)  
Minimum: -5 Maximum: 335

Temperature (Celsius)	Density (kg/m <sup>3</sup> )	Kinematic Viscosity (Centistoke)	Dynamic Viscosity (Centipoise)	Thermal Conductivity (W/m.K)	Heat Capacity (kJ/kg.K)
-5	873.46	710.87	620.91	0.144	1.832
5	866.75	298.21	258.47	0.144	1.862
15	860.04	143.64	123.54	0.143	1.892
25	853.33	77.36	66.01	0.143	1.933
35	846.62	45.62	38.62	0.142	1.963
45	839.81	29.31	24.62	0.142	1.993
55	833.11	20.17	16.80	0.141	2.023
65	826.40	14.53	12.01	0.141	2.063
75	819.69	10.87	8.91	0.140	2.093
85	812.98	8.40	6.83	0.139	2.123
95	806.17	6.67	5.37	0.139	2.153
105	799.47	5.42	4.33	0.138	2.193
115	792.76	4.49	3.56	0.138	2.223
125	786.05	3.78	2.97	0.137	2.253
135	779.24	3.24	2.52	0.137	2.283
145	772.53	2.81	2.17	0.136	2.323
155	765.83	2.46	1.89	0.136	2.353
165	759.12	2.18	1.66	0.135	2.383
175	752.41	1.95	1.47	0.135	2.423
185	745.60	1.76	1.31	0.134	2.453
195	738.89	1.60	1.18	0.134	2.483
205	732.18	1.46	1.07	0.133	2.513
215	725.48	1.34	0.97	0.133	2.553
225	718.77	1.24	0.89	0.132	2.583
235	711.96	1.15	0.82	0.132	2.613
245	705.25	1.08	0.76	0.131	2.643
255	698.54	1.01	0.70	0.131	2.684
265	691.84	0.95	0.66	0.130	2.714
275	685.13	0.90	0.61	0.130	2.744
285	678.32	0.85	0.58	0.129	2.784
295	671.61	0.81	0.54	0.129	2.814
305	664.90	0.77	0.51	0.128	2.844
315	658.20	0.74	0.48	0.128	2.874
325	651.49	0.70	0.46	0.127	2.914
335	644.68	0.68	0.44	0.126	2.944