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Capture of Heat Energy from Diesel Engine Exhaust

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Abstract

Diesel generators produce waste heat as well as electrical power. About one-third of the fuel energy is released from the exhaust manifolds of the diesel engines and normally is not captured for useful applications. This project studied different waste heat applications that may effectively use the heat released from exhaust of Alaskan village diesel generators, selected the most desirable application, designed and fabricated a prototype for performance measurements, and evaluated the feasibility and economic impact of the selected application. Exhaust flow rate, composition, and temperature may affect the heat recovery system design and the amount of heat that is recoverable. In comparison with the other two parameters, the effect of exhaust composition may be less important due to the large air/fuel ratio for diesel engines. This project also compared heat content and qualities (i.e., temperatures) of exhaust for three types of fuel: conventional diesel, a synthetic diesel, and conventional diesel with a small amount of hydrogen. Another task of this project was the development of a computer-aided design tool for the economic analysis of selected exhaust heat recovery applications to any Alaskan village diesel generator set.

The exhaust heat recovery application selected from this study was for heating. An exhaust heat recovery system was fabricated, and 350 hours of testing was conducted. Based on testing data, the exhaust heat recovery heating system showed insignificant effects on engine performance and maintenance requirements. From measurements, it was determined that the amount of heat recovered from the system was about 50% of the heat energy contained in the exhaust (heat contained in exhaust was evaluated based on environment temperature). The estimated payback time for 100% use of recovered heat would be less than 3 years at a fuel price of \$3.50 per gallon, an interest rate of 10%, and an engine operation of 8 hours per day. Based on experimental data, the synthetic fuel contained slightly less heat energy and fewer emissions. Test results obtained from adding different levels of a small amount of hydrogen into the intake manifold of a diesel-operated engine showed no effect on exhaust heat content. In other words, both synthetic fuel and conventional diesel with a small amount of hydrogen may not have a significant enough effect on the amount of recoverable heat and its feasibility.

An economic analysis computer program was developed on Visual Basic for Application in Microsoft Excel. The program was developed to be user friendly, to accept different levels of input data, and to expand for other heat recovery applications (i.e., power, desalination, etc.) by adding into the program the simulation subroutines of the desired applications. The developed program has been validated using experimental data.

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Chapter 1 Introduction

In rural Alaska during 2007, approximately 180 villages consumed about 370,000,000 kWh [1] of electrical energy. During the process of producing electrical power, there is unused heat from the diesel engines. If the waste heat could be used, there would be a significant fuel savings. This project, which studied the selection of the most appropriate engine exhaust heat recovery application, is needed as part of village economic development.

There are many different heat recovery methods available for capturing engine waste heat, including thermal electric conversion; heat-to-power conversion (e.g., Sterling engines, Rankine cycle engines, gas turbines); direct heat application (e.g., space heating, waste water loop heating); heat for refrigeration and air conditioning; and heat for desalination. To optimize the benefit that heat recovery systems can bring to Alaskan villages, the engine performance characteristics and operational conditions that have important effects on the performance of heat recovery systems have to be understood. Important engine properties and operational conditions may include engine-generator load, engine fuel energy, soot produced by exhaust gas, and exhaust gas corrosivity. Engine load and fuel energy will affect the amount of heat that is recovered by the heat recovery system and the soot content in the exhaust. Corrosivity and soot content will affect the maintenance of the heat recovery system.

To understand the load condition of the village diesel generators, engine data sheets of annual generator usage were obtained from some of the Alaskan villages [2], including Ambler, Chevak, Noorvik, and Scammon Bay. These data show the load pattern of village electricity usage in June and December of 2002. Table 1.1 lists the month's load patterns according to the data sheets. The average load, based on this data, was about 65% of the engine rated load year-round. The load varied between 25% and 100% of the rated load.

Table 1.1. Generator Usage

Month	0% to 25% load	25% to 50% load	50% to 75% load	75% to 100% load
June	5%	34%	58%	3%
December	1%	4%	43%	52%

Based on engine specification sheets of similar diesel generators, a rough estimate of the fuel energy distribution is given in Figure 1.1. About half of the village generators have already been equipped with intercooler heat recovery systems, but intercooler heat recovery is not a part of this study. According to Figure 1.1, the heat energy contained in the exhaust was found to be 30% of the total fuel energy. If half of the heat contained in

the exhaust could be recovered for useful applications, the result would be a fuel savings of 15% in Alaskan village power generation.

Figure 1.1. Energy distribution of a diesel generator.

The first task of this project studied the feasibility and economic effects of some of the most appropriate heat recovery applications for Alaskan village diesel generators. According to the cogeneration chapter of the “December 2002 Rural Alaska Energy Plan,” the proposed future cogeneration market segments included five different applications: community water loop temperature maintenance, public space heating using baseboard systems, public space heating using floor radiant systems, residential micro-cogeneration units, and school cogeneration units. The first three proposed applications are applicable to the existing village diesel generators, but the last two are not. Other heat recovery applications which were reviewed but rejected for further consideration included ice-making and refrigeration, desalination, thermal electricity generation, and steam engine electricity generation. Reasons for rejection included limited regions for adoption, limited economic benefit, limited amount of heat recoverable, high capital cost, and high frequency of maintenance requirement. This work was then converged to the feasibility study and economic analysis of an exhaust heat recovery system for heating.

Diesel engine exhaust contains soot and corrosive compounds which may cause performance deterioration, possibly resulting maintenance, and power outage problems with the heat recovery system and the engine exhaust system. No precise data describing the performance deterioration rate, corrosion rate, and maintenance frequency of exhaust heat recovery devices for general diesel engine applications were found in the literature.

According to a simplified experiment [3] of soot accumulation in a small bundle of tubes heated with combustion gas from controlled combustion, the deterioration rate reached the saturation point before the next maintenance was needed. Corrosion might not occur, provided that the exhaust temperature is kept appreciably higher than the dew point of sulfurous acid. Therefore, exhaust heat recovery systems may not cause serious problems that prevent it from being adopted for cogeneration. Soot- and corrosion-related feasibility is further discussed in later chapters of this report.

The objective of this experimental project was to conduct a feasibility and economic analysis of the heat recovery applications mentioned above. The experiment also included the design and installation of a heat recovery system for a given diesel generator for performance data collection, which then was used for analysis. The study results also provided a useful procedure which could be put into usable form for the end user, for designing an exhaust heat recovery system that would economically, reliably, and efficiently capture the unused heat from the village diesel engine generator sets. Considering the recent increase in fuel prices, this study is particularly important.

One recent consideration concerning fuels used in diesel engines has been introduced by the U.S. Environmental Protection Agency (EPA), which mandated a 95% reduction in the sulfur content of fuels used. Synthetic diesel matches the sulfur content requirement and may become an important fuel of the future. Since the fuel composition and combustion performance of synthetic fuels [4, 5] are known to be different from conventional diesel, the heat content and temperatures of the exhaust for these two fuels may be appreciably different, resulting in a difference in the amount of recoverable heat. An experiment to determine exhaust heat content and temperature (i.e., heat quality) was conducted on synthetic fuel and a conventional diesel. The results were then used to predict the advantages and disadvantages of these two fuels in exhaust heat recovery.

Recently, many individuals claimed the observation of significant engine efficiency improvement when using a hydrogen electrolyzer to generate a small amount of hydrogen and inject it into the intake manifold. A considerable change in engine efficiency may also mean a considerable change in exhaust property. This project conducted an experiment to measure the heat content and temperature of exhaust for conventional diesel with different levels of small amounts of hydrogen. The result was then used to estimate the effect of a small amount of hydrogen on exhaust heat recovery.

Since the performance and economic outcomes of installing an exhaust heat recovery system for heating are case-dependent and depend on many existing factors, a computer program which is able to help determine whether or not the installation is beneficial is important. The second task of this project was to develop a computer program for preliminary economic analysis of installing an exhaust heat recovery system in a village diesel generator set. The targeted users were the engineers and technicians of village power plants, who should be able to learn the program with a minimum amount of time and instruction. The program was developed on Virtual Basic for Application in Microsoft Excel, which is available on all personal computers. The program was designed to be user-friendly and able to accept different levels of input data (i.e., data obtained from measurements and manufacturers or data derived from analysis using

standard handbook procedures) of system and component properties, depending on the level of data available to the user. The program was also designed for continuing improvement and expansion, provided more design data and more effective methods become available.

Executive Summary

In rural Alaska, there are nearly 180 villages consuming more than 370,000 MWh of electrical energy annually from individual diesel generators. A similar amount of fuel energy is dissipated into the atmosphere from diesel exhaust. The purpose of this project was to evaluate the feasibility and economic impact of applying exhaust heat recovery on rural Alaska diesel generators. Two major tasks were included in this project. The first task was to conduct feasibility and economic analyses using the data obtained from a designed experiment. This task involved the selection of the best application of recoverable exhaust heat, design and fabricate an experiment setup, testing, and analysis. The second task was to develop a computer program for preliminary economic analysis of applying the selected exhaust heat recovery application to individual village diesel power generators.

Exhaust Heat Recovery System Design and Testing

Among several potential heat recovery applications (e.g. heating, electric power generation, desalination, and ice making), exhaust heat recovery for heating (EHRH) was selected for this project. The selection was based on need, availability, feasibility, and benefit that could be brought to rural Alaska.

An EHRH system was designed for a DD50 diesel engine located on UAF campus. The engine has a rated load of 125 kW and rated speed of 1200 rpm, is instrumented with a variety of sensors for engine performance monitoring, and is connected to a load bank for engine load control. The engine was not equipped with after treatment devices; therefore, the analysis results demonstrated the worst case scenario. The designed heat recovery system has 3 major parts, the exhaust to liquid heat exchanger for exhaust heat absorption, the unit heater for heat dissipation or heating load control, and the piping and control unit for temperatures and flow rate controls. Temperatures and flow rate controls made the system capable of simulating different heating methods at different loads for optimal heat absorption. The recommended working fluid was 40/60 mixture of Propylene glycol and water for cold weather in Alaska.

The designed heat recovery system was installed and instrumented. A National Instrument data acquisition system (DAQ) was used for experimental data collection and reduction and performance monitoring of the system and its components. Engine was running for 350 hours after installation. The system was operated under 25%, 50%, 75%, and 100% engine loads for three different heating operation conditions to simulate three different space heating methods possibly used in Alaskan villages. After the experimentation, the heat exchanger was dismantled for the observation and measurement of corrosion and soot accumulation for maintenance and feasibility study.

Engine performance and exhaust heat contents were also measured for two other fuels: the ultra-clean (synthetic) diesel and conventional diesel with complementary hydrogen (less than 1% by volume or 0.07% by mass). The purpose of the test was to evaluate the potential effects of different fuels on exhaust heat recovery performance.

Results derived from analysis of experimental data and observations on engine and heat recovery system performance led to the following findings. The performance of the exhaust heat exchanger was consistent and reliable. No noticeable difference was observed on engine performance before and after the installation of the heat recovery system. Corrosion and soot accumulation was not observed to be a problem in the laboratory test of 350 hours. For the 125 kW diesel generator used in this experiment, the rate of heat recovered from the exhaust was about 60 kW. Based on this heat recovery rate, the estimated payback time for a 100% use of recovered heat would be less than 3 years for a fuel price of \$3.5 per gallon (Village fuel price is considerably higher than that of cities) at an interest rate of 10% and an engine operation of 8 hours per day. Different types of fuels did not show noticeable effect on engine performance and exhaust heat content

An Economic Analysis Program for Exhaust Heat Recovery for Heating

Since the performance and economic outcomes of installing an EHRH system are case dependent and depend on many existing factors, a computer program, which is able to help determine whether or not the installation for a particular village diesel generator is beneficial, is important. The well known thermal science computation programs, which are not designed for economic analysis of EHRC problems in Rural Alaska, either not appropriate or not applicable, especially when the problems involve specific information about village infrastructure, power usage, heat usages, and local cost information.

The computer program developed for this task was based on the experience obtained from the first task of this project. The computer program is capable of doing preliminary design (if needed) and economic analysis for installing an exhaust heating system to any village diesel generators. This program was developed on Visual Basic for Application in Microsoft Excel, which is available on all personal computers. The targeted users of the program were engineers and/or technicians of the village power plants. The program was designed for user-friendly and being able to accept different levels of input, from measured data, manufactures' data, to data from analytical procedures, depending on the level of data available to the user.

The input of the computer program includes the power plant information, existing village heating system information, interest rate, and all other cost information. The output of the program includes predicted payback period and profits and also detailed information generated along the design and analysis process. The program can also provide a Microsoft Words document of all the significant data generated.

This program has been used to simulate the design and performance of the experiment conducted at UAF and to conduct the economic analysis based on data from simulation results. The results match well to that obtained using the experimental data. The program was also designed for continuation of improvement and expansion provided more design data and more effective analysis methods would become available. For example the program can be easily extended to economic analysis for other exhaust heat recovery applications (e.g. power, desalination) by adding into the computer program new subroutines of the desired applications.

Chapter 2 Experimental

2.1 Background

2.1.1 Introduction

The first task of this project included an experimental study of the feasibility and economic effects of exhaust heat recovery heating for Alaskan village diesel generators. This task involved the experimental procedure of design, procurement, installation, instrumentation, measurements, and data reduction. The first task also included the measurement of exhaust temperatures, flow rates, and emissions of an engine operated with conventional diesel, a synthetic fuel, and conventional diesel with different levels of a small amount of hydrogen. The measured data were used to estimate exhaust heat content and to evaluate the potential effects of different fuels on heat recovery performance. This project's second task was to develop a computer program for economic analysis of applying exhaust heat recovery to individual village diesel power plants. The second task did not involve an experiment.

2.1.2 Literature Review of Exhaust Heat Recovery

This subsection reviews the research that was being conducted in the field of recovery of rejected heat from diesel engines. During engine run time, there are different places in the engine's structure where significant amounts of heat are dissipated to the atmosphere. The rejected heat includes heat from the water jacket, the exhaust, and in more recent engines, the heat from the turbocharger aftercooler. This heat can be used for domestic or commercial purposes by using a recovery process, although this requires the addition of significant hardware which adds to the expense of the installation. In the past, fuel was cheap and it was very difficult to justify the cost of the heat recovery hardware. Furthermore, high sulfur fuel results in corrosive condensates, requiring either expensive alloys for heat recovery from exhaust systems or expensive replacements.

Previously, when the idea of heat recovery was not considered, the engine cooling system was used only for running the engines to prevent overheating. When fuel economy became important due to the energy crisis, especially in the early 1970s, different ways to recover an engine's unused heat for useful purposes became a practical research area. After the 1970s energy crisis eased, heat recovery research activities dropped accordingly, until the recent surge in fuel prices. From previous research, it was found that engine cooling systems and exhaust had abundant amounts of heat. The utilization of this energy would result in an increase in the efficiency of the system.

Experimental studies focused mainly on recovery of exhaust heat, which is one of the heat-carrying sources. The following sections provide the literature review for heat recovery from diesel engines.

Low-sulfur fuels: One recent consideration in fuels used in diesel engines has been the EPA-mandated reduction in sulfur content of fuels used for transportation. This reduction is anticipated to have several significant environmental benefits, including a direct reduction of sulfur oxides, a direct reduction in particulate matter (PM), and an

indirect reduction in other pollutants through the use of catalytic cleanup systems. For stationary diesel engine heat recovery systems, the reduction in sulfur and the reduction in PM mean that exhaust heat exchangers are likely to be more practical, as the exhaust will be less corrosive and form less soot, both of which have prevented economic heat recovery in the past.

Sulfur is an element that is naturally present in crude oil. Conventional diesel sold for transportation in the United States after 2001 contains a maximum of 500 ppm of sulfur [4,5]. This sulfur gets oxidized to sulfur dioxide (SO_2) or sulfur trioxide (SO_3) after combustion. These sulfur oxides play an important role in the corrosion process. The sulfur oxides dissolve in moisture to form sulfurous acid (H_2SO_3) or sulfuric acid vapor (H_2SO_4) [6].

The dew point of exhaust water vapor (37.78°C to 65.56°C) is considerably below typical exhaust temperatures. However, in boiler conditions, the formation of acid occurs at the dew point (115.56°C to 137.78°C), which is more likely to occur in combustion exhaust systems, especially when exhaust heat recovery is applied. The formation of these acids depends on the amount of air, moisture, and sulfur content [6].

The EPA proposed to decrease the sulfur content in the on-road diesel fuels to 15 ppm by 2006. These standards are scheduled to be extended to 2011 for off-road engines. One way of reducing sulfur is to use synthetic diesel (ULSD) created by Fischer–Tropsch, which has near-zero sulfur [4, 7]. This would largely reduce the problem of corrosion due to acids.

Soot formation: The major pollutants of fuel combustion are considered to be nitrogen oxides (NO_x) and PM. Particulate matter is a result of unburned hydrocarbons due to incomplete combustion, and is a major concern because it is a health hazard and impacts visibility. For our case, only PM was considered, as it is the source for soot in the heat exchanger. Soot is defined as a dark powdery deposit of unburned fuel residue mainly consisting of carbon. Significant accumulations of soot have a direct negative impact on the ability of a heat exchanger to extract usable heat from the exhaust.

The composition of PM by mass is as follows [8]:

Metal – 1.2%
 Hydrogen – 2.6%
 Nitrogen – 0.5%
 Oxygen – 4.9%
 Sulfur – 2.5%
 Carbon – 88%

The soluble organic fraction in PM of a diesel engine varies from 5% to 40% by mass. The production of PM depends on the aromatic levels of the fuel. C_2 reacts with C_4 to generate aromatic structure [9], which comprises monocyclic and polycyclic aromatic hydrocarbons (PAH). Nanoparticles, which are formed during the engine cycle of the gas

phase PAH, grow to 2 nm and continue until growth vanishes. These smaller particles coagulate to 10 nm due to van der Waals forces [8].

Most soot ranges from 5 nm to 30 nm in diameter. Laser-induced incandescence (LII) was one of the techniques used to find particulate matter measurements. Primary soot particles (20 nm to 30 nm) absorb light radiation both in ultraviolet and visible regions.

Many experiments have been conducted to illustrate soot growth and coagulation. According to a study by Mathis, the primary soot particles of a diesel engine range from 17.5 nm to 32.5 nm, as determined by transmission electron microscopy (TEM) measurements [10]. This work demonstrated that primary soot particle diameter decreased from 25.9 nm to 17.5 nm at low loads and decreased from 28.4 nm to 21.6 nm at high loads during the start of ignition. Also noted was an effect due to injection pressure, as the primary soot particle diameter was reduced due to increase in injection pressure from 500 to 1100 bar at low loads. There was a small decrease in diameter of soot particle at high load when injection pressure was changed from 800 to 1400 bar.

In another study by Kitsopanidis in 2006, a rapid compression machine was adapted to replicate a diesel engine in which a line-of-sight (LOS) absorption method was used [11] to study soot growth. The volume concentration of soot grew depending on the fuel concentration of the compressed charge over a period of time, but an unexplained initial exponential growth was observed irrespective of the fuel concentrations.

Regarding the soot accumulation inside heat exchangers, a simulation experiment was conducted [3]. The experiment controlled the flow rate and pressure of combustion gas, which crossed a bundle of a small number of cooling pipes. The experiment was operated for an extended time for each of the combinations of selected flow rates and pressures. Soot accumulations were measured along with the experiment. According to the experiment results, the soot that accumulated on finned tubes for each case seemed to approach its plateau within a relatively short period of time.

Heat recovery applications: The idea of using waste heat from an engine to heat a space is not unique to this project, but there are fewer applications of heat recovery from diesel exhaust. During our literature search, we came across various projects that involved heat recovery in different aspects, but could not find any documents that are directly related to our project. Some of the heat recovery projects that were similar to our project were documented as case studies. Some of the most useful information related to diesel exhaust heat recovery in Alaska was obtained through personal communication with engineers working in Alaska.

Heat recovery for Eagle Community School in Eagle, Alaska: In the late 1970s a recovery system was built for the Eagle Community School in Eagle, Alaska, by Summit Logistics, a company owned by Dave Cramer. According to Mr. Cramer, the recovery of heat was done on a 350 kW Cummins diesel engine-generator set. The recovered heat was used to heat a water heating system using a gas-to-gas heat exchanger. "There was an awful effect due to soot on the heat exchanger components and also with the engine," said Mr. Cramer. He recollected that the engine manufacturer did

not recommend such systems. The heat exchanger was installed over the engine exhaust system, which was not appropriate for the given task. The system had an effect on the engine due to the increase in carbon deposits. Though the system was meant to work for 10 years, after a few hundred hours of running, it was observed that the recovery system did not work to Mr. Cramer's expectations, and it had to be removed within the first year of its installation.

"A constant temperature could not be maintained with the variation of loads," said Dave Cramer. The system required constant monitoring and maintenance four times a year to remove soot deposits. Lack of maintenance and observation might be a cause for the improper operation. Mr. Cramer stated that, for an engineer, the loads on the engine cannot be predicted on practical grounds. He estimated the entire system, with copper tubing of 2" to 1½", cost him about \$30K to \$40K.

Industrial waste heat recovery: During the 1970s, the Arab oil shocks convinced many energy users to attempt to reduce their energy consumption. Unfortunately, little effort was made in the recovery of waste heat during the initial stages, but after a few years, this idea was tested. A demonstration project was undertaken in 1976 by American Standard Corporation.

The demonstration project involved industrial waste heat recovery at a plant located in Louisville, Kentucky, that produced thermal energy. The combustion air was preheated electrically to 537.78°C for the coke-fired cupolas. Exhaust gases from iron foundry cupolas was close to 150,000 pounds per hour at a temperature of 760°C. A comprehensive study was performed in close cooperation with the staff of American Standard Corporation and with international experts. Economical and technical studies of heat recovery systems were made, and problems based on structure, space requirements, etc., were given close attention [12].

The new system that was installed included a gas-to-air heat exchanger to preheat the air to 537°C. Flue gas at 760°C was controlled to 676.67°C by a cooling tower to protect equipment and assure stable performance and efficient gas filtering. A wise decision to include air-operated soot blowers with the heat exchanger to remove dirty flue gases was considered. The gas and air tubes were filtered with damper valves to permit flexibility to the entire system.

Multiplying this experience by 10,000 industrial plants leads to an equivalent energy savings of 390,000 MBTU or 67,000 barrels of domestic heating oil for each hour.

Recovery of heat from the exhaust gas of a diesel engine: A large number of projects recover heat from natural gas-fired systems that would reduce carbon dioxide and nitrous oxide emissions, reducing the greenhouse effect. A recovery system described by Verneau was built in a Lucciana, Corsica, power plant on an 11 MW SEMT 18PC 3 diesel engine (much larger than the diesel engines used in Alaskan villages). The thermal power contained in the exhaust was of the same order as the shaft power (38%). The exhaust temperature was 390°C, but could not be cooled below 170°C because of corrosion effects. In this test, the experiments were conducted on a steam cycle and

organic fluid cycle, and the results were compared. Verneau states that better cycle and better expansion led to a net efficiency of 0.21 and 0.15 for organic fluid and steam. The cycle that was selected is shown in Figure 2.1.

Figure 2.1. Schematic diagram.

The organic fluid required a larger heat exchange surface area and higher temperature, which was done by a recuperator. The recuperator increased the temperature of liquid going into the evaporator that avoided corrosion. The working fluid was a mixture of trifluoroethanol and water. The tests showed some corrosion on carbon steel and slightly alloyed steels [13]. However, the major disadvantage of these cycles is the high cost of both the recuperators and the turbines.

Exhaust heat recovery from midsize diesel engines for power generation:

Due to the recent surge in fuel prices, some research activities in exhaust heat recovery for power generation have been conducted with midsize diesel engines that are similar in size to most of the engines used in Alaskan villages. Most of the research activity in this area has focused on truck engines [14, 15, 16], and is still in the early stages. In general, the efficiencies of the prototypes developed are still much lower than exhaust heat recovery for heating applications. It is expected that the efficiency of exhaust for power will be improved and become competitive in the future.

Thermoelectric generators based on heat recovery: Exhaust heat can be used to produce electric power by means of thermoelectric generators (TEG). Several TEG were designed to fit the temperatures at various parts of the exhaust system [17, 18, 19]. TEG is easy to install and have few maintenance problems. There are no moving parts, so

there is less noise. Though the idea is relevant, the efficiency of the TEG is considered to be meager, peaking at about 4% under ideal conditions, and often much lower.

2.1.3 Heat Exchanger Basics

The inlet and outlet temperatures on the hot and cold side of a heat exchanger can be used to design a heat exchanger. The basic equations involved for a heat exchanger design are

$$Q = U_o A F (LMTD) \quad (2.1)$$

where

$\dot{Q}$ = Heat transfer rate

U_o = Overall heat transfer coefficient

A = Heat transfer area

$LMTD$ = Logarithmic mean temperature difference

F = Correction factor

The size of a heat exchanger can be measured by means of number of transfer units (NTU). The equation for NTU is given in Equation 2.2.

$$NTU = \frac{AU}{\dot{C}} \quad (2.2)$$

where

U = Local overall heat transfer coefficient

A = Heat transfer area

$\dot{C}$ = Product of mass flow rate and the coefficient of specific heat

In the absence of phase change, the NTU of a heat exchanger determines the performance in terms of effectiveness (E). Effectiveness is given by the ratio of actual heat transfer rate to the maximum heat transfer rate. The equation for effectiveness is given in Equation 2.3.

$$E = \frac{\dot{Q}}{\dot{Q}_{\max}} \quad (2.3)$$

where

$\dot{Q}$ = Actual heat transfer rate

$\dot{Q}_{\max}$ = Maximum heat transfer rate

There are many types of heat exchangers based upon their differences in temperature range, performance ratios, type of heat exchange carried out, efficiency, and working

condition [21, 22]. Based on ratio of heat transfer area and dimension of heat exchanger, heat exchangers can be categorized into shell and tube type and compact type. The compact type may include plate fin heat exchangers, tube fin heat exchangers, plate type heat exchangers, printed circuit heat exchangers, recuperators, and regenerators.

The main problems commonly faced when using compact heat exchangers are related to fouling due to accumulated solid materials or property changes on the walls of the heat exchanger. In our case, the fouling was high due to exhaust gases. Chemical cleaning, thermal baking and subsequent rising and wise design in maintenance and operation methods may help resolve these problems.

The following are different fouling mechanisms:

1. Crystallization/precipitation
2. Particulates
3. Chemical reactions
4. Corrosion
5. Biological effects
6. Freezing

The following are effects of fouling:

1. Increase in the capital costs
2. Increase in the maintenance costs
3. Loss of production
4. Increase in the energy losses

Compact heat exchangers are mostly the gas-to-gas type. Gas is harder to control than liquid. At the beginning of the project, most of the suppliers that we contacted provided compact heat exchangers, but either the temperature range or the material of construction did not match our project's needs. After further review and a literature search, it seemed that shell and tube type heat exchangers would be a better choice for this project.

Shell and tube heat exchangers are only limited by the materials of construction and have the additional advantage of being designed for special operating conditions, such as vibrations, heavy fouling, highly viscous fluids, erosion, corrosion, toxicity, radioactivity, and multicomponent mixtures. Shell and tube heat exchangers can be made from metal as well as non-metal, and their surface areas range from as little as 0.1 to 100,000 m². They have less surface area per unit volume than compact heat exchangers [21].

2.1.4 Economic Study

Heat recovery systems are not typically used in diesel engine generators due to cost issues. If the cost of purchasing a heat exchanger and installing it is high and the cost of diesel fuel is low, most users will elect to vent the hot exhaust and purchase additional fuel for the required heat loads. Only when the cost of fuel is high compared with the capital cost of heat recovery installation will this option make sense. The economic analysis is directly related to the cost of fuel and the cost of capital as given by the interest rate. These two considerations are discussed below.

Cost of fuel: In Alaska, small rural utilities have cooperated to purchase fuel in bulk quantities from the same vendor in order to reduce fuel costs. The fuel price depends on the crude oil cost, cost at the refinery, and transportation cost. Among these organizations were Alaska Native Industries Cooperative Association (ANICA) and Alaska Village Electric Cooperative (AVEC). ANICA purchases 2 to 3 million gallons of fuel each year to serve 25 communities. AVEC purchases about 6 million gallons of fuel each year to operate electric utilities for 51 rural villages [23].

Discounts of about 15% are available for a purchase of 100,000 gallons of fuel. Further reductions are given for purchases of more than 100,000 gallons of fuel. A reduction of 10 ¢/gal would result in \$2000 to \$4000 in fuel cost savings for a utility that consumes 100,000 gallons of fuel [23].

Interest rates: The interest rate determines the payback amount for money borrowed. Depending on the source of funds, the interest rates may vary from 0% to 12%. The 0% APR is in the case of grants allotted by the organizations that require repayment. The payback method differs from company to company. There are two methods of returning the borrowed money: The first method includes equal amounts to be paid each year. According to this method, the total amount will be calculated for a span of years with the interest rate, which needs to be repaid in equal amounts in equal intervals of time. In the second method, at a constant interest rate, the principle amount differs. The principle amount differs depending on the amount paid each year. The amount repaid by the end of each year will be deducted from the initial borrowed money, and the interest rate accounts for the current principle amount to be paid each year [25].

2.1.5 Effect of Fuel Type on Heat Recovery

Exhaust flow rate, composition, and temperature may affect the heat recovery system design and the amount of heat recoverable. In comparison with the other two properties, the effect of exhaust composition may be less important due to the large air/fuel ratio for diesel engine combustion. This part of the project was to measure the exhaust properties and to estimate the heat contents and heat qualities (i.e., temperatures) of exhausts obtained from diesel engine combustion of three different types of fuel: a synthetic fuel, the conventional diesel, and the conventional diesel with different levels of a small amount of hydrogen. The results were then used to predict the potential difference between the exhausts of the three types of fuel.

2.2 Design of an Exhaust Heat Recovery System

2.2.1 Introduction

In most parts of rural Alaska, diesel generators operate year-round to produce power to supply village residents and remote mining sites with electrical power. The power consumption determines the load pattern on the diesel engines. In most villages, the load in summer is less compared with that in winter due to the decreased demand for lighting and heating during the warm, bright summer months. In nearly all locations, heat was

unused and rejected to the atmosphere. The percentage of fuel heat left unused from the engine is shown in Table 2.1.

Table 2.1. Percentage of waste heat

Location of recoverable heat	Percentage of fuel energy
Exhaust	30
Turbocharged air	7
Jacket water	18

The average consumption of energy in rural Alaskan households is detailed in Figure 2.2.

	Rural Average	Anchorage	Fairbanks	Juneau
Median Household Income (MHI)	\$40,380 ¹⁵	\$55,546	\$40,577	\$62,034
Annual Electric Consumption	5040 kWh	7782 kWh	9048 kWh	10,428 kWh
Average Price ¹⁶ (After PCE)	\$0.20/kWh	\$0.095/kWh	\$0.089/kWh	\$0.102/kWh
Annual Amount	\$1080	\$739	\$805	\$1064
Electricity as Pct of MHI	2.7%	1.3%	2.0%	1.7%
Space Heating Consumption	700 gallons per year	2100 CCF per year	1500 gallons per year	1000 gallons per year
Average Price	\$2.00 per gallon ¹⁷	\$0.40 per CCF	\$0.75 per gallon	\$0.79 per gallon
Annual Amount	\$1400	\$890 ¹⁸	\$1125	\$790
Heating Consumption as Pct of MHI	3.5%	1.6%	2.8%	1.3%
Electric + Heating Consumption as Pct of MHI	6.2%	2.9%	4.8%	3.0%

Figure 2.2. Alaskan household energy consumption [24].

If this heat could be applied to useful applications such as heating, a savings in fuel consumption might result. Since it is difficult and expensive to move heat, one major issue is the need to have a heat sink close to the diesel generator. While most generators are located as far as possible from buildings for noise and emissions reasons, the recent spike in diesel fuel price means that the economics of these systems are changing.

The goal of this project was to design and test a system to utilize the waste heat from a small diesel generator through a recovery process that would be economical and cost efficient for Alaskan villages. A heat recovery system was designed and fabricated to implement the selected heat recovery application and determine the feasibility and

economic effect. To design and test the heat recovery application, a Detroit Diesel Series 50 engine with a 125 kW generator operated at 1200 rpm was used. The details of this approach and the design procedures are discussed in the following sections.

2.2.2 Selection of Waste Heat

The amount of heat present in the exhaust, the turbocharged air, and the jacket water are presented in Table 2.1. As seen in the table, the jacket water and turbocharged air have lower energy than the exhaust, which indicates that the exhaust waste heat is more prominent than the jacket water and the turbocharged air. The following are other reasons behind the selection of exhaust waste heat for this project:

1. Many of the diesel generators were already equipped with jacket water recovery systems, and recovery of this energy is well understood.
2. The temperature of the exhaust was much higher than the turbocharged air, while the mass flow was nearly the same. The heat recovery rate from the exhaust was also expected to be higher.
3. The introduction of new low sulfur fuels, which result in lower corrosivity and lower particulate matter generation, meant a reduction in the major technical barriers to heat recovery from the exhaust.
4. The recent increase in the cost of crude oil resulted in higher diesel and heating oil prices, making heat recovery more economically attractive.
5. A reduction in greenhouse gases would result from a reduction in the consumption of diesel fuel.

2.2.3 Selection of Heat Recovery Application

Several heat recovery applications had the potential to be economical and feasible for Alaskan villages, among which were space and water heating, desalination, ice making, and electric power generation. However, desalination does not appear to be locally useful. According to the Alaska Department of Environmental Conservation in March 2005 [29], the mineral content of groundwater for most Alaskan villages is well within acceptable ranges. Only a few villages in northern Alaska require purification of surface water for drinking, where the permafrost is very deep. Electric power generated from waste heat appears attractive, as it is a convenient form of energy. It was not selected, however, because of its low heat recovery rate—about 6% of fuel savings out of 30% of fuel energy contained in the exhaust—and because it is expensive. Ice making has been demonstrated in Kotzebue, but unless there is a large commercial user for ice in the summer (such as local fishing industry), the costs are not justified. However, heat is almost always needed in Alaska, so heating was considered to serve the purpose in this project. The recovered heat can be used either for space heating, domestic hot water, or warming municipal water supplies to prevent freezing.

2.2.4 Experimental Site

This experiment was conducted at the Arctic Energy Technology Development Laboratory at the University of Alaska Fairbanks, using a Detroit diesel engine-generator set. The rated speed of the generator is 1200 rpm with a rated electrical power of 125 kW. The generator was connected to an external load bank from Load Tech with a power of 250 kW. The engine has a rated speed of 1800 rpm with a power of 235 kW, so the engine was down rated and connected to the generator. The diesel generator set was placed inside an International Organization for Standardization (ISO) container with the load bank on the outside of the ISO container.

This engine and generator set were used for other experiments, including emissions experiments conducted in 2004 and 2005. The setup is discussed in more detail in Telang's master's thesis [4].

For the experiment's heat recovery system design and installation, the experimental site was considered a convenient substitute for a potential heat recovery application that mimics the diesel generators in rural Alaska. Heat was recovered from the exhaust, measured, and then rejected to the atmosphere through a dump load.

2.2.5 Heat Recovery System Design

The design of the heat recovery system involved determining the different functional parameters (e.g., exhaust backpressure, exhaust flow rate, total pressure drop, and temperatures) that needed to be considered. The parameters and the related components are discussed in detail below.

Heat exchanger: The exhaust heat exchanger was the first element to be considered in the design process. The specification sheet of our experimental engine was the primary information considered in coordination with the village data to get a brief idea for the selection of a heat exchanger. There were a few set points for the heat exchanger selection. Those set points include the following:

1. Amount of exhaust heat available
2. Corrosion of the exhaust system
3. Maximum exhaust backpressure permissible by engine
4. Dimensional constraints
5. Weight constraints
6. Maintainability

Given that electrical loads vary, the amount of recoverable heat also varies. In order to select a heat exchanger suitable for fluctuating exhaust heat energy, the heat exchanger needed to have some operational parameters adjustable with the varying engine load conditions and possibly with the heat requirements. Based on this consideration, the heat exchanger was designed for engine exhaust at engine-rated load conditions. At partial engine load conditions, the heat recovery system should be adjustable to the heat requirements. A major design concern for our work was the availability of space. As well, the following parameters needed attention:

1. Flow rate of the exhaust
2. Required heating temperatures
3. Exhaust temperature differential
4. Coolant temperature range

Another important consideration was atmospheric effect. As this project was designed for use in Alaska, the maximum ambient temperature differential in Alaska was considered. Any equipment selected must be capable of functioning in a wide range of ambient conditions.

For this project's design, a gas-to-liquid type heat exchanger was selected. The main reason behind this choice was to permit easy escape of the exhaust gases after passing through the heat exchanger without affecting engine exhaust backpressure. The heat exchanger acts as a muffler also. The design allowed the exhaust temperature to be above the water vapor dew point to reduce exhaust condensation and acid formation. (The issue of acid dew point was not discovered until the project was well underway, but fortunately, we seem to have avoided it.)

A shell and tube type heat exchanger was selected. In this design, exhaust gas passes through the shell, and liquid passes through the core side, that is, through the inner pipes. This design enables better exhaust escape to the atmosphere in a single pass without building much backpressure. This type of heat exchanger provides a relatively large area for heat transfer. For maintenance purposes, the liquid on the tube side and accessibility to remove the core were specified.

The parametric inputs considered for the heat exchanger design were

1. temperatures
2. heat transfer surface area
3. corrosion resistance

Temperature: Inlet and outlet temperatures of the liquid (Table 2.2) were based on a safety margin depending on the load conditions and weather conditions for convective space heating. The inlet temperature was the exhaust temperature of the engine at full load. The outlet temperature was selected based on the trade-off between cost and heat recovery. The possible reason for corrosive exhaust condensate formation was also considered for the outlet temperature selection; that is, the exhaust outlet temperature should be above the water dew point with a safety margin. (The acid dew point was not considered, but the temperatures seem fine for that issue also.) The liquid side temperatures were selected based on the feedwater temperature of 65°C, typical of boilers used for baseboard heating. Thus, a temperature of 77°C would be adequate for preheating the boiler feedwater.

Table 2.2. Inlet/outlet temperatures with respect to the heat exchanger

Gas/liquid	Inlet temperature (°C)	Outlet temperature (°C)
Exhaust	540	177
Liquid	77	87

Heat transfer surface area: Heat transfer surface area directly affects the heat exchange rate. The number of passes, the tube diameter, and the tube pattern affect the heat transfer surface area. The total area is constrained by the overall dimensions allowed for the heat exchanger.

The predesign calculation for the heat exchanger consisted of the overall heat transfer coefficient and the log mean temperature difference (LMTD). The correction factor was considered to be 1. The surface area required was calculated to be 8 m^2 (90.4 ft^2) using the procedure given in [19]. The effectiveness was calculated to be 80%.

Based on the required temperature and surface area from the preliminary design calculations of the heat exchanger, 25 heat exchanger suppliers were contacted. Most of the heat exchanger manufacturers were unable to respond to our heat exchanger requirements for this project. The reasons were as follows:

1. High temperature on the gas side
2. Type of heat exchanger needed (shell and tube type)
3. Material of construction
4. Ease of maintenance

Two of the manufacturers were willing to fabricate the type of heat exchanger similar to the one that we needed in our project and they provided bids, summarized in Table 2.3.

Table 2.3. Quotation comparison

	Quotation 1	Quotation 2
Shell side	Gas	Liquid
Tube side	Liquid	Gas
Gas pressure drop	0.31 PSI	PSI
Maintenance	Removable core, 1 in. gap and straight tubes	U-tube for gas
Heat transfer area	87 ft ²	30 ft ²
Size	51 in. x 28 in. x 28 in. (with detailed drawings)	Shell Φ 6.625 in., Tube: 72 in. (no details)
Weight	725 lb	350 lb
Material of construction	Tube: SS fin tube Shell: SS inner wall	Tube: Plain SS 316 Shell: CS
Insulation	Integrated insulation	No insulation included
Cost (\$)	9,800	7,979

After reviewing both of the quotations and designs, Quotation 1 was considered to be a better option for this project. The following were the main reasons for this choice:

1. Gas on the shell side and liquid on the tube side
2. Stainless steel material of construction
3. Less pressure drop
4. Inbuilt insulation
5. Easy maintenance due to straight tube structure

Even though the cost of Quotation 2 was less than Quotation 1, Quotation 1 was selected due to the insulation. Weight was not an important issue in this case.

Selection of unit heater: In any real system, the heat that is recovered from the heat exchanger needs to be used for a useful purpose. In order to replicate a real application, the recovered heat for space heating was simulated by the dispatched heat through the unit heater with controlled inlet and outlet temperatures and flow rate.

A unit heater is a simple radiator with an associated fan that dissipates the heat to the atmosphere; one can be purchased off the shelf. For this experimental setup, the heat recovered from the exhaust was dissipated to the atmosphere under different load conditions. The designed maximum capacity of heat recovered by the Cain Industries heat exchanger was 290,000 Btu/hr. The unit heater must be able to remove the same amount of heat, thus a unit heater (model S) built by Trane was selected.

Control system: The control system in this experiment used copper tubing that connected the heat exchanger to the unit heater in a closed loop. The components in the system contributed to desired functionality of the entire system. A three-way mixing valve was present in the control system connecting the bypass and the unit heater outlet pipe. The valve was controlled by a nickel immersion sensor and universal controller. The manual set temperatures in the universal controller helped in controlling the valve opening in the mixing valve in coordination with the nickel sensor to attain the required temperatures on the coolant inlet side of the heat exchanger. The major components considered for our heat recovery system were heat exchanger, pump, three-way valve, flow meter, and unit heater. The heat exchanger acted as a heat source to capture heat from the exhaust, while the unit heater acted as a heat sink to simulate the heat flow. The heat exchanger and the unit heater were connected in a closed loop. The purpose of the piping system was to transport the working fluid between the heat source (heat exchanger) and the heat sink (unit heater) to control the heat desired for practical applications in rural Alaska. The piping system needs to be operated in such a way that the inlet of the heat exchanger can be maintained at specific temperatures at variable load conditions. To obtain those temperatures, a three-way mixing valve with a nickel sensor was used.

The pipe design was iterated many times due to the unexpected real sizes of the components and space availability for system installation. The ISO container was preoccupied by the Detroit diesel engine-generator set and the day tanks. There was very limited space for the construction of the pipe system. This caused major changes in the preliminary pipe design. The total pressure drop of 10 psi in the pipe system remained almost the same after the changes in the design. The space that was allocated for building the control system inside the ISO container is shown in Figure 2.3.

Figure 2.3. Space allocated for the control system inside the ISO container.

The final control system design is shown in Figure 2.4. A detailed description of the entire control system is discussed in Section 2.3.

Coolant side pressure drop calculations and pump selection: The pressure drop in the copper tubing decides the pump size. Pressure drop was estimated based on a procedure given in the ASHRAE handbook [20], which involved calculating the preliminary pressure drop. The total pressure drop was calculated to be 10 psi.

The pump was selected based on the calculated total pressure drop in the copper tubing with all components installed. The flow rate was taken from the selected heat exchanger's specifications. Figure 2.5 shows the pump curve used for selecting the pump. A margin of 5 gpm was added to the original flow rate for safety purposes. The calculated pressure drop was 23 ft of water. These two values correspond to a point that was marked in the graph. The graph shows a 1/3 HP pump that was suitable for our system.

Figure 2.5. Owner's manual – Bell and Gossett.

Table 2.4. DAQ channels

Slot	Channel type	Purpose of the channel
1		- Empty -
2		- Empty -
3	SCXI 1120	8-channel isolation amplifier
4	SCXI 1180	Feed-through panel
5	SCXI 1120	8-channel isolation amplifier
6		- Empty -
7	SCXI 1120	8-channel isolation amplifier
8	SCXI 1121	4-channel isolation w/excitation
9		- Empty -
10		- Empty -
11	SCXI 1102	32-channel thermocouple amplifier - heat recovery
12	SCXI 1102	32-channel thermocouple amplifier

Figure 2.6. Block diagram showing the DAQ channels.

Table 2.5. Naming in the DAQ

Slot in SCXI 1102	Name in DAQ
8 to 12	Exhaust outlet temperature (T8)
13	Heat exchanger inlet temperature (T1)
14	Heat exchanger outlet temperature (T6)
15	Exhaust inlet temperature (T7)
17	Before bypass temperature (T2)
18	Bypass temperature (T3)
19	After 3-way valve temperature (T5)
20	Before 3-way valve temperature (T4)
21	Unit heater inlet temperature (T9)
22	Unit heater outlet temperature (T10)
SCXI 1120	Mass flow meter

2.3 Installation and Instrumentation of the Heat Recovery System

2.3.1 Introduction

This section of the report discusses installation, instrumentation, and calibration of different elements in the heat recovery system. The entire system can be divided into three sections: heat exchanger, unit heater, and control system.

The selected heat exchanger has gas on the shell side and liquid on the tube side, and its physical dimensions are 51 in. x 18 in. x 23 in. (LxWxH), with a heat transfer surface area of 87 ft². The heat exchanger weighs about 1000 lb and has 4 in. of inbuilt insulation.

The entire control system was designed to fit the ISO container design in this experimental project. The main aspect that needed to be considered during the heat recovery system design was our space allocation. The ISO container had two regions: (1) the water jacket radiator and (2) the engine, generator, day tanks, control unit, and DAQ. When the engine was running, the temperature in the engine section was maintained well above zero due to the heat radiated by the engine. The instruments and the engine needed to stay warm for normal operation, while the radiator section needed to be at a lower temperature, since the inlet manifold air temperature should be below 40°C.

For the convenience of connecting the heat exchanger to the exhaust pipe and because hot gases prefer to rise, the heat exchanger was installed on top of the ISO container. The size of the unit heater and the improved efficiency of using normal ambient air determined its installation outside the ISO container. For the sake of convenience, the unit heater was also installed on the west wall next to the control system, which was constructed inside the ISO container. Most of the control system needed to be maintained at room temperature for the specified operation of various sensors. For this project we used copper tubing, as it was (relatively) easy to make modifications when needed and is inexpensive.

The initial AutoCAD drawing of the whole system is shown in Figure 2.7. Details of the heat recovery system were split into three sections, based on their location. Figure 2.8 is section 1 showing the heat exchanger and its inlet/outlet pipe connections; Figure 2.9 is section 2, which includes major flow control elements; and Figure 2.10 is section 3, which shows the unit heater and its inlet/outlet pipe connections. Components of the control system are designated by numbers and are described in Table 2.6.

There were significant changes in the design of the control system due to space considerations. The final control system of this project is shown in Figures 2.8, 2.9, and 2.10, with brief descriptions added.

Figure 2.7. AutoCAD heat recovery design drawing.

Parts location and description:

1	Heat exchanger	5	Bronze ball valve
2	Temperature thermocouple	A	Inlet to the heat exchanger
3	Pressure gauge	B	Outlet from the heat exchanger
4	Snubber		

Figure 2.8. Section 1 of the heat recovery system (on top of ISO container).

Parts location and description:

3	Pressure gauge	8	Expansion tank
4	Snubber	9	Circuit setter
5	Bronze ball valve	C	Flow meter line
6	Flow meter	D	Pump line
7	3-way valve		

Figure 2.9. Section 2 of heat recovery system (inside ISO container along west wall).

Parts location and description:

2	Temperature thermocouple	8	Expansion tank
3	Pressure gauge	10	Unit heater
4	Snubber	E	Inlet to the unit heater
5	Bronze ball valve	F	Outlet from the unit heater

Figure 2.10. Section 3 of heat recovery system (sitting outside on the ground next to the west wall of the ISO container).

Table 2.6. Component numbering

1	Heat exchanger
2	Temperature thermocouple
3	Pressure gauge
4	Snubber
5	Bronze ball valve
6	Flow meter
7	3-way valve
8	Expansion tank
9	Circuit setter
10	Unit heater
A	Inlet to the heat exchanger
B	Outlet from the heat exchanger
C	Flow meter line
D	Pump line
E	Inlet to the unit heater
F	Outlet from the unit heater

2.3.2 Heat Exchanger Section

The heat exchanger section of the heat recovery system (see Figure 2.11) includes the heat exchanger, pipe components, and instruments connected to the heat exchanger located outside on top of the ISO container. The inlet and the outlet sides for the heat exchanger are indicated in the figure along with its associated components. For isolating the heat exchanger, bronze ball valves were installed along with the drain. During the experimental process, if there was any clogging due to scales or rust on the core side of the heat exchanger, the pressure would increase. In order to ascertain the presence of any clogging due to these kinds of obstacles, pressure gauges were installed on the inlet and the outlet side. A release valve (from Cain industries) was set on the inlet side of the heat exchanger to free excess pressure and prevent liquid from entering the heat exchanger. Temperature sensors (also from Cain industries) along with temperature thermocouples

were installed on the inlet and the outlet pipes, which help in calculating the heat that can be recovered. As these pipes were considered to be the highest points in the entire piping system, the air vents were installed on them.

Figure 2.11. Heat exchanger inlet/outlet pipe connections.

2.3.3 Control System Section

The control system section of the heat recovery system (Figure 2.12 shows the north half and Figure 2.13 shows the south half) includes the pump and all pipes, fittings, components, and instruments installed inside the ISO container that are between the heat exchanger and the unit heater.

Figure 2.12. North half of the control system inside the ISO container on the west wall.

Pump system: The pump (18 kg) was supported from the roof in the middle of the control system. The support for the pump was fabricated to resist vibrations. A pressure sensor was installed across the pump for intermediate monitoring of its performance. The pump was connected to the copper pipe through dielectric unions which helped in avoiding the effects caused by the contact of two different materials and for the easy removal of the pump when isolated by the bronze ball valves when needed. A strainer was installed in the line to remove any metal residue in the coolant flow. The strainer can be placed anywhere in the control system, but due to space considerations, it was placed after the pump and before the expansion tank. This pipe came from the heat exchanger through the roof and ran along the roof of the ISO container, as that was the only place where it could be accommodated with enough space to walk around it.

Expansion tank in the control system: The expansion tank (Figure 2.14) was installed at a convenient location, and was provided with unions and bronze ball valves across it to isolate that section for maintenance. The expansion tank was fitted with a pressure gauge to note the pressure on the liquid side. This pressure was compared periodically with the initial pressure readings for maintenance purposes.

Figure 2.13. South half of the control system inside the ISO container on the west wall.

Figure 2.14. Expansion tank in the control system

Bypass looping: As noted in the control system discussion above, the unit heater was parallel to the bypass line (Figures 2.15 and 2.16) to maintain the outlet temperature of the parallel lines. The outlet temperature could be controlled to simulate different types of heat recovery applications, such as central floor heating, desalination, and ice making. A mixing valve with an actuator was installed in the end of the bypass, connecting the bypass line and outlet line from the unit heater. A temperature sensor was installed after the mixing valve, which controls the opening of the three-way valve with the help of a signal going to the actuator from the sensor. Thermocouple sensors were installed along and across the bypass for constant monitoring of the temperature difference taking place with the change of the engine load. A circuit setter was installed in the bypass to maintain the pressure and the flow rate.

Flow meter line: The flow meter line (Figure 2.17) was connected to the inlet of the heat exchanger. It was a turbine-type flow meter with bronze ball valves and dielectric unions on both ends. A pressure gauge was installed across the flow meter for intermediate monitoring of flow meter performance. The flow meter is an important component in a control system. The readings were recorded in the DAQ system.

Figure 2.15. Bypass line.

Figure 2.16. Bypass line.

Figure 2.17. Flow meter line.

2.3.4 Unit Heater Section

The unit heater section (Figure 2.18) includes the pipe, fittings, expansion tanks, and thermocouples. The inlet to the unit heater is the pipe coming from the pump. Strainers were attached across the unit heater to trap any possible metal pieces. Thermocouples were installed across the unit heater for monitoring the amount of heat extracted from the heat exchanger.

Figure 2.18. Unit heater inlet/outlet pipe connections.

Operation of control system: Once a control system is running, the pump moves the fluid in the pipe continuously, transferring the heat from the exhaust to the liquid passing through the tubes in the heat exchanger. The heat exchanger was designed for a maximum fluid flow rate of 30 gpm. The flow rate was maintained and monitored. The expected heat recovered was 290,000 Btu/hr with a built-in heat exchanger surface area of 87 ft². When the system was new, all the temperature, pressure, and flow measurements were recorded for reference purposes.

To begin with, liquid passes through the pump and expansion tank, and then enters into the unit heater or the bypass, depending on the three-way valve position. The three-way mixing valve opening was controlled by the temperature sensor attached to it. The temperature at which the mixing valve needs to be operated was set on the universal controller, which sends out a signal to the actuator attached to the mixing valve that

controls valve openings. The temperature on the universal controller was preset manually for the liquid entering the heat exchanger. If the sensor read less than the preset value, the bypass valve would open and distribute more fluid through the bypass instead of the unit heater. If the sensor read more than the preset value, the bypass valve would close, passing more fluid through the unit heater. For simulation of different heat recovery applications, different temperatures can be set accordingly.

Fluid enters the heat exchanger after passing through the flow meter. Pressure gauges were installed across the pump and the flow meter to monitor the inlet and discharge pressure of the pump and the flow meter. These pressure readings also helped in monitoring the performance of the pump and the flow meter.

All the thermocouples were connected to the DAQ system for continuous monitoring. The flow meter gave a 4-20 mA signal that was also connected to the DAQ system for monitoring.

2.3.5 Flow Meter and Load Cell Calibration

The flow meter calibration was calibrated gravimetrically. This involved measuring the weight of the fluid passing through the flow meter during a known time from a source to the sink. The flow meter calibration was set up onsite to carry out the calibration process. In this calibration process, two 15-gallon tanks that served as source and sink were used inside the ISO container. A load cell was fixed under one of the 15-gallon tanks (before the outlet of the heat exchanger) that acted as a sink. Before starting the calibration, the bronze ball valves across the unit heater were closed, allowing the fluid to pass only within the ISO container at room temperature. When the pump was run, the fluid from one tank was pumped to the other tank. The load cell would sense a change in the reading and record this in the DAQ system. The readings of the load cell and the flow meter are compared and graphed in Figure 2.19.

Figure 2.19. Calibration curve for flow meter.

The load cell was also calibrated by known weights. The load cell calibration curve is shown in Figure 2.20.

Figure 2.20. Calibration curve for load cell.

2.3.6 Instrumentation of Exhaust Line Monitoring

The purpose of instrumentation was to monitor the effects of the heat exchanger on engine performance and the exhaust system. The instruments included the thermocouples installed on the inlet and outlet of the heat exchanger on the shell side (Figure 2.21) and on the tube side (Figure 2.22). The thermocouples on the tube side gave the temperature drop across the heat exchanger.

2.3.7 Cost of the System

The total cost of the system includes all the components of the control system, the unit heater, and the heat exchanger. The cost of the experimental system is tabulated in Table 2.7. The total cost for the heat recovery system was \$30,000 (approximately). The estimated cost for the village heat recovery system is tabulated in Table 2.8.

Figure 2.21. Thermocouples on the outlet of the heat exchanger.

Figure 2.22. Thermocouples on the core side.

Table 2.7. Cost of the experimental system

Component	Qty	Unit Price (\$)	Total Price (\$)
Tristand pipe vise	1	319.00	319.00
Box beam 3"X3"X1/8" 20' bar	1	75.57	75.57
Flat stock 3"X3/16" 20' bar	1	24.31	24.31
Angle bar 2-1/2"X2-1/2"X3/16" 20' bar	1	33.22	33.22
Flat stock 1-1/2"X1/8" 20' bar	1	9.50	9.50
Flat stock 1"X1/8" 20' bar	1	6.57	6.57
Box beam 1-1/2"X3"X1/8" 20' bar	2	56.54	113.08
Box beam 1-1/4"X1-1/4"X1/8" 20' bar	2	32.02	64.04
Box beam 2"X2"X1/4" 20' bar	1	87.23	87.23
Flexible, insulated thermocouple probes with exposed junction	15	17.50	262.50
Slip-on flange, 150 psi, 5" pipe dia, 10" OD, 8-1/2" bolt	3	38.03	114.09
Great Stuff insulating foam sealant	6	8.49	50.94
Centrifugal pump 1/3 HP 30 GPM	1	672.00	672.00
Expansion tank	1	34.18	34.18
Heat exchanger	1	10,019.00	10,019.00
Fiberglass insulation - bag	1	61.79	61.79
Signal conditioner 0-5 V	1	524.00	524.00
Turbine flow meter, 4-60 linear range (GPM)	1	1,289.00	1,289.00
Pressure gauge, dc powered, dual alarms	3	375.00	1,125.00
Type K grounded thermocouple probe with dia 0.125"	10	25.90	259.00
Type K ungrounded thermocouple probe with dia 0.062	25	26.80	670.00
Type K ungrounded thermocouple probe with dia 0.062	10	27.60	276.00
Teflon insulated type K thermocouple wire	1	405.00	405.00
Miniature type K thermocouple connector pair	50	4.00	200.00
Type K grounded thermocouple probe with dia 0.062	10	24.00	240.00
Type K grounded thermocouple probe with dia 0.125	10	24.00	240.00
24-gauge galvanized metal (Size 63"X39")	1	78.00	78.00
24-gauge galvanized metal (Size 60"X40")	2	78.00	156.00
24-gauge galvanized metal (Size 66"X42")	1	100.00	100.00
Single-loop controller with two 0-10VDC analog outputs	1	100.45	100.45
1-1/4" 3W BR control valve UFXUF with 0-10VDC	1	292.77	292.77
Nickel immersion temperature sensor	1	35.02	35.02
30VA transformer for RWD controller	1	21.64	21.64
Hydronic unit heater, model S-Unit heater	1	1,655.00	1,655.00
Pipe components			6,303.90
Miscellaneous from warehouse			4,000.00
Total cost			\$29,917.80

Table 2.8. Estimated cost for the village recovery system

Component	Qty	Unit Price (\$)	Total Price (\$)
Box beam 3"X3"X1/8" 20' bar	1	75.57	75.57
Flat stock 3"X3/16" 20' bar	1	24.31	24.31
Angle bar 2-1/2"X2-1/2"X3/16" 20' bar	1	33.22	33.22
Flat stock 1-1/2"X1/8" 20' bar	1	9.50	9.50
Flat stock 1"X1/8" 20' bar	1	6.57	6.57
Box beam 1-1/2"X3"X1/8" 20' bar	2	56.54	113.08
Box beam 1-1/4"X1-1/4"X1/8" 20' bar	2	32.02	64.04
Box beam 2"X2"X1/4" 20' bar	1	87.23	87.23
Slip-on flange, 150 psi, 5" pipe dia, 10" OD, 8-1/2" bolt	2	38.03	76.06
Great Stuff insulating foam sealant	6	8.49	50.94
Centrifugal pump 1/3 HP 30 GPM	1	672.00	672.00
Expansion tank	1	34.18	34.18
Heat exchanger	1	10,019.00	10,019.00
Fiberglass insulation - bag	1	61.79	61.79
Type K ungrounded thermocouple probe	6	26.80	160.80
Teflon insulated type K thermocouple wire	1	405.00	405.00
Miniature type K thermocouple connector pair	50	4.00	200.00
Pressure gauge	3	10.00	30.00
24-gauge galvanized metal (Size 66"X42")	1	100.00	100.00
Single-loop controller with two 0-10VDC analog outputs	1	100.45	100.45
1-1/4" 3W BR control valve UFXUF with 0-10VDC	1	292.77	292.77
Nickel immersion temperature sensor	1	35.02	35.02
30VA transformer for RWD controller	1	21.64	21.64
Pipe components			6,303.90
Labor cost (hours)	80	75.00	6,000.00
Parts	1	400.00	400.00
Total cost			\$25,377.07

2.4 Experiment Setups for Exhaust Properties of Other Fuels

The engine used for this experiment was a 125 kW 4-cylinder DD50 diesel generator without exhaust gas recirculation and after-treatment devices. Engine load was controlled by a 250 kW resistive/reactive load bank. The interface between the engine and the load bank had customized generator load profiles. Exhaust heat content was estimated using exhaust temperatures and exhaust flow rates with specific heat data. The exhaust flow rate was estimated from the measured intake air flow rate and fuel consumption rate. Inlet air mass flow was measured using a laminar flow element manufactured by Mariam Instrument, coupled with a HART differential pressure transducer manufactured by ABB. Fuel flow-rate data were taken from the CANbus of the diesel generator, with appropriate

calibration using a gravimetric method. Exhaust temperature data were taken from the engine CANbus. All the measured data were processed via a National Instruments DAQ system and acquired at 10-second intervals.

2.4.1 Synthetic Fuel

Measurements in intake air flow rate, fuel consumption rate, and exhaust temperature for both synthetic diesel and conventional diesel were performed earlier while the synthetic fuel was available. The tests were conducted for 50% load and 100% load. Exhaust emissions had been measured in previous work [4] for NO_x, CO, and THC.

Testing procedure: The fuel tests performed during this study can be divided into two parts: the synthetic fuel test and the conventional diesel test. These tests were conducted on different days, and the synthetic fuel was tested first. The same procedure was used for both fuel tests. Tests for the synthetic fuel followed the procedure given below:

1. Before a test, the fuel line was emptied, and old filters were replaced with new ones.
2. The engine was then started with the synthetic diesel fuel and operated for 2 hours to purge all the residual conventional diesel fuel from the fuel line.
3. After this cleaning, the synthetic diesel fuel test was started, and data were collected.
4. The measured exhaust temperatures and flow rates were used to estimate the heat content in the exhaust.

2.4.2 Conventional Diesel with a Small Amount of Hydrogen

A small amount of hydrogen was introduced into the intake air stream of the diesel engine in flow rates of 0, 1, 2, 3, 4, 6, 8, 10, 12, 16, 30, 50, 100, and 150 liters per minute (lpm). Figure.2.23 shows the experimental setup for the test. The main components of the experiment were the diesel engine, compressed hydrogen tank, MKS mass flow controller, DAQ system, and exhaust gas analyzer.

The hydrogen tank was a commercially available compressed hydrogen tank used for industrial purposes. The hydrogen pipeline was connected to the inlet air stream of the diesel engine by the MKS mass flow controller. The MKS mass flow controller was connected in turn to the National Instruments DAQ system. The range of the MKS flow controller is 0 to 200 lpm. The DAQ system gives the signal to the mass flow controller for certain flow rates of hydrogen, and then the mass flow controller adjusts the flow of the hydrogen according to the signal received from the DAQ system. Data included exhaust temperature, ambient temperature, and fuel flow rate. The exhaust gases were analyzed by a Testo 350-S exhaust gas analyzer. A sample of exhaust gas was passed through the analyzer, and the data were analyzed for emissions. The analyzed data were stored in the DAQ system. The complete information collected by DAQ was stored in the

form of an Excel spread sheet. After the experiment was completed, the data in the spread sheet were analyzed. Engine load for this test was 56 kW.

Figure 2.23. Line diagram for hydrogen test.

Testing procedure: An experiment was conducted to simulate the performance of a hydrogen electrolyzer (commercially available on the web) and to determine the effect of hydrogen on exhaust heat content and emissions. The difference between this experiment and one actually using an electrolyzer was that we used pure bottled hydrogen, while an electrolyzer decomposes water (H_2O) into hydrogen (H_2) and oxygen (O_2). In general, the mass flow rate of H_2 and O_2 generated by the electrolyzer is much lower than 1% of the intake air mass flow rate.

The following steps explain the test procedure:

Step 1: The diesel engine was started and ran for 3 h at 50 kW load to stabilize the temperatures surrounding the engine. This step ensures that the readings are being recorded at constant ambient temperatures. During this run, no hydrogen was supplied.

Step 2: The DAQ system collected data every 10 sec. The hydrogen was introduced into the intake air stream in regular intervals of 5 min; that is, the experiment was started with 0 lpm of hydrogen for the first 5 min, then hydrogen was introduced at 0.5 lpm to the intake air for the next 5 min, and then again the hydrogen flow was lowered to 0 lpm for the next 5 min and then at 1 lpm for the next 5 min. Likewise the hydrogen flow cycles of 0 lpm and designated flow (0.5, 1, 2, 3, up to 150 lpm) were introduced into the intake air stream. The DAQ collected nearly 30 data points for every hydrogen flow rate.

Step 3: The exhaust gas analyzer (Testo 350-S) is not a continuous measuring device. For every hydrogen flow rate of 5 min, the exhaust gas analyzer was switched “ON” for 3 min for taking the emissions data. Therefore, approximately 15 emissions data were available for averaging for every hydrogen flow rate.

Step 4: Exhaust emissions data for oxygen (O_2), carbon monoxide (CO), nitrous oxide (NO), nitrogen dioxide (NO_2), total nitrogen oxides (NO_x), and sulfur dioxide (SO_2) were collected. The exhaust gas analyzer gave the output of emissions in amperes. The exhaust gas temperatures and air flow rates were used to calculate the average heat content in the exhaust.

Chapter 3 Results and Discussion

3.1 Introduction

This chapter discusses the results obtained from our two tasks. The first task includes two parts. The first part discusses the measured results, performance, and economics of the designed heat recovery system. Discussions focus on verifying the match between the measured performance of the fabricated system and the desired performance of the original design, feasibility, and economic effect. The feasibility study includes the effect of the heat recovery system on engine performance, system efficiency, and reliability issues such as corrosion and soot accumulation. The second part of the first task discusses the heat energy contents in the exhaust from three different fuels. The results are used to compare the heat recovery potential of the exhaust from these three fuels. For the second task, this chapter discusses the structure of the developed economic analysis computer program and its performance. Discussions include the goal and structure of the program, validation of the program for its ability in heat recovery computation, and a demonstration of economic analysis result using an example.

Considering the economic analysis of exhaust heat recovery for heating, the heat recovery system was operated and monitored for nearly 350 hours after the completion of the system installation. The system was operated under generator-rated load conditions (125 kW) for most of the time. The system was also operated under different engine loads (25%, 50%, and 75%) for performance testing. For each load, the inlet or outlet coolant of the heat exchanger was controlled at various temperatures to simulate the requirements for different applications.

At the beginning of testing in the summer, water was used as a coolant for both ease of use and data analysis. However, data collection was not completed before the onset of cooler weather, so the coolant was changed from water to a mix of 40% propylene and 60% water at the 150th hour to avoid freezing. After 250 hours, one of the thermocouples (coolant inlet temperature to the heat exchanger) was recalibrated, and the time constant of the controller for the temperature control valve was adjusted to limit a large fluctuation in the temperature reading. In this chapter, data obtained between 250 and 300 hours were used for most of the analysis because the data are more accurate (Table 3.1).

Based on experimental data obtained, the upcoming sections discuss the verification of the heat recovery system, the consistency of the heat recovery system, the effect of heat recovery on engine performance, the feasibility and maintenance related issues, and the economic analysis.

The test for exhaust properties of the three fuels and their effect on exhaust heat recovery broke down to two parts. The first part involved the comparison of effects of the synthetic fuel and the conventional diesel on heat recovery performance; the second part concluded the effect of a small amount of hydrogen on exhaust heat recovery performance. The test for the synthetic fuel and the conventional diesel was conducted much earlier due to the time availability of the synthetic fuel. Comparison in exhaust emissions between the

synthetic fuel and the conventional diesel had been performed in a previous project; results were given in [4]. Because the tests of the two parts were carried out more than one year apart and the testing environment conditions were different, the results of the studies of the two parts were discussed separately.

Concerning the developed economic analysis computer program for exhaust heat recovery for heating, this chapter discusses the specific goals and desired features of the program, the platform chosen, program structure, input required, and output data and format. This chapter also discusses the links between different routines of the program, the data library, and the principles applied to design and analyses.

Table 3.1. Different cases with engine running on conventional diesel fuel

Heat exchanger outlet temperature	Working fluid	Case
50-hour run 87°C (190°F)	40% propylene glycol	Data analyzed at 100% load once every 10 hours
87°C (190°F)	40% propylene glycol	25%, 50%, 75%, and 100% engine loads
77°C (170°F)	40% propylene glycol	25%, 50%, 75%, and 100% engine loads
65°C (150°F)	40% propylene glycol	25%, 50%, 75%, and 100% engine loads

3.2 Economic Analysis for Exhaust Heat Recovery for Heating

3.2.1 Design Verification

The heat recovery system was tested to verify its design at different engine loads. The flow rate on the coolant side was observed to be constant throughout the experimental process. Figure 3.1 shows the flow rate distribution curve across the bypass and unit heater at different engine loads. There was a change in the flow rate at rated load. This can be explained as the change in the total pressure drop. When the engine is running at rated load, a higher percentage of coolant passes through the unit heater instead of the bypass, which increases the total pressure drop in the pipeline due to the higher flow resistance of the unit heater. This could be adjusted using the circuit setter. However, the flow rate was not adjusted due to very small changes, which might cause a small variation in the heat exchanger efficiency.

The inlet/outlet temperatures across the fluid side of the heat exchanger are shown in Figure 3.2. The temperature differentials at varying engine loads changed according to the heat present in the exhaust and the heat dissipation capacity of the unit heater, which depended on ambient temperature. At higher loads, as the exhaust heat was greater, the inlet temperature to the heat exchanger needed to be much lower in order to maintain the

required outlet temperatures and vice versa. At the same time, the unit heater needed to dissipate the absorbed heat by the coolant. In our case, the unit heater was smaller and could not dissipate enough heat at higher engine load conditions as it reached its maximum heat-dissipating capacity. This limitation resulted in the system being incapable of further lowering the heat exchanger inlet temperature which in turn increased the heat exchanger outlet temperature on the fluid side. Thus, the graph shows an increasing curve on the heat exchanger coolant outlet side and a decreasing curve on the coolant inlet side. However, the inlet temperatures of the heat exchanger on the fluid side matched the temperatures that were set on the temperature control valve, which confirms desired operation of the temperature-controlled three-way valve.

Figure 3.1. Fluid flow distributions across the bypass and unit heater.

Figure 3.2. Temperature on the fluid side across the heat exchanger.

The unit heater was able to serve its purpose by dissipating the maximum amount of heat possible. In a real field situation, the system would be designed based on the heat requirements instead of using a unit heater to simulate the load required. Table 3.2 shows real values at different engine loads.

Table 3.2. Fluid side readings at different engine loads

Parameters	25% Load	50% Load	75% Load	100% Load
Heat exchanger inlet temperature (°C)	81.72	79.66	78.22	75.90
Heat exchanger outlet temperature (°C)	85.43	85.58	86.87	89.90
Total fluid flow rate (Kg/s)	1.45	1.45	1.41	1.22
Flow rate in bypass (Kg/s)	1.389	1.282	1.039	0.415
Flow rate across unit heater (Kg/s)	0.057	0.164	0.368	0.801
Exhaust inlet temperature	223.33	327.42	412.89	463.90
Exhaust outlet temperature	109.54	131.89	159.52	186.86

Figure 3.3. Flow distributions between the bypass and unit heater.

Effect of outside ambient temperature: As expected, the ambient temperature had a great impact on the flow distribution of the system between the bypass and the unit heater. The warmer the outdoor ambient temperature was, the more coolant was needed to pass through the unit heater.

Energy balance: Heat balance was checked between the heat absorbed from the exhaust side and the heat dissipated from the system, including heat loss from the unit heater and pipe. Figure 3.4 shows that the heat absorbed by coolant equals the sum of heat dissipations, as the laws of thermodynamics demand. The difference between the heat absorbed and dissipated by the coolant is less than 3%. Figure 3.4 also shows that the heat dissipated from the unit heater was higher for higher temperatures. This can be explained by the pipeline, at higher ambient temperatures, dissipating less heat which must be compensated for by increasing heat loss through the unit heater to maintain the inlet temperature of the heat exchanger.

Figure 3.4. Energy balance with respect to the ambient temperature.

3.3 Feasibility and Performance

3.3.1 50-Hour Run

The heat recovery system was run for 350 hours to assess the maintenance that might be required for satisfactory system performance. There were two major concerns during this time: soot accumulation resulting in reduced heat flux, and corrosion due to condensation of sulfuric acid. During a 50-hour run, the heat exchanger efficiency was monitored, but no significant change was observed in either the efficiency or the system performance (temperature, pressure, and fluid flow rate) during this time. There was a small variation in the total enthalpy of the exhaust entering and leaving the heat exchanger. The variation followed a change in the conex temperature (graphed in Figure 3.5), and was found to be due to changes in ambient pressure, resulting in a slight change in mass flow. However, the efficiency of the system was seen to be almost constant, where the efficiency is the ratio of the amount of heat gained by the propylene to the amount of heat lost by the exhaust.

Figure 3.5. Enthalpy change in exhaust across the heat exchanger.

The exhaust mass flow rate was calculated using the sum of the engine manifold inlet air mass flow rate and fuel consumption rate. In order to determine the air mass flow rate, a laminar flow element and differential-pressure gage was used. The heat release rate of the exhaust across the heat exchanger is shown in Figure 3.6. The heat flux was calculated by the product of enthalpy change with respect to the standard conditions and flow rate of the exhaust. The enthalpy change was calculated based on the individual mass percentages of the exhaust components after combustion. In Figure 3.6, E_{in} and E_{out} represent, respectively, the total heat present in the exhaust before entering into and after passing through the heat exchanger.

Figure 3.6. Heat flow rates of the exhaust entering and leaving the heat exchanger.

Figure 3.7 shows the heat release rate from the exhaust and the heat absorption rate of the glycol coolant. The heat absorption rate was evaluated using the coolant mass flow rate, the specific heat, and the temperature difference between the heat exchanger inlet and outlet flow. The amount of heat absorbed by the propylene glycol followed the same trend as the heat released from exhaust across the heat exchanger.

3.3.2 Forty Percent Propylene Glycol as Working Fluid

The working fluid was 40% propylene glycol and water mix. Propylene does not affect the environment negatively when spilled. This was the main reason behind its selection for this experiment. The freezing point of 40% propylene is -35°C . Corrosion inhibitors were added to the mix to avoid corrosion in the pipe. There were a few experiments using 40% propylene glycol as the working fluid to check the feasibility and performance of the heat recovery system. The different cases include variations in the heat exchanger outlet temperatures of around 87°C , 77°C , and 65°C , and variations in engine-rated load of 25%, 50%, 75%, and 100%.

This experiment was conducted at lower loads and higher loads. Lower loads were considered to be 25% and 50% loads, while higher loads were considered to be 75% and 100% loads. Table 3.3 shows related engine power to percentage of load. This table represents all cases in this project.

Figure 3.7. Heat release versus heat absorption.

Table 3.3. Load percentage against related kW

Loads	kW
25%	31
50%	62
75%	93
100%	128

The amount of heat that the fluid can recover depends on the heat present in the exhaust. Figure 3.8 shows the heat flow on the gas side of the heat exchanger at different loads.

Figure 3.8. Heat released by exhaust and heat absorbed by glycol fluid as outlet temperature to the heat exchanger on the fluid side is set to 87°C

As predicted, the heat absorption rate varied at different loads. The heat absorption rate followed the same trend as the heat present in the exhaust shown in Figure 3.8. All the related temperature measurements and calculated values for the exhaust and fluid with 87°C as the outlet temperature on the fluid side of the heat exchanger are given in Table 3.4.

Similar system performances were observed for other cases. The related graphs are shown in Figure 3.9 and Figure 3.10.

All the related temperature measurements and calculated values for the cases of 77°C and 65°C as the outlet temperature on the fluid side of the heat exchanger are given in Table 3.5 and Table 3.6.

Table 3.4. Outlet temperature of 87°C on the fluid side of the heat exchanger

Temperature / calculations	25% Load	50% Load	75% Load	100% Load
Exhaust flow (Kg/s)	0.14	0.16	0.19	0.24
Exhaust outlet temperature (°C)	109.54	131.89	159.52	186.86
Heat exchanger inlet temperature (°C)	81.72	79.66	78.22	75.90
Heat exchanger outlet temperature (°C)	85.43	85.58	86.87	89.90
Exhaust inlet temperature (°C)	223.33	327.42	412.89	463.90
Total fluid flow rate (Kg/s)	1.45	1.45	1.41	1.22
Total absorbed heat by propylene (kW)	20.11	32.08	45.60	65.43
E_{in} of exhaust (kW)	28.06	51.43	81.28	114.86
E_{out} of exhaust (kW)	11.75	17.70	27.26	40.84

Figure 3.9. Heat released by exhaust and heat absorbed by glycol fluid as outlet temperature to the heat exchanger on the fluid side is set to 77°C

Figure 3.10. Heat released by exhaust and heat absorbed by glycol fluid as outlet temperature to the heat exchanger on the fluid side is set to 65°C

Table 3.5. Outlet temperature of 77°C on the fluid side of the heat exchanger

Temperature / calculations	25% Load	50% Load	75% Load	100% Load
Exhaust flow (Kg/s)	0.14	0.16	0.19	0.23
Exhaust outlet temperature (°C)	100.97	122.28	150.62	189.88
Heat exchanger inlet temperature (°C)	72.65	69.22	67.10	67.58
Heat exchanger outlet temperature (°C)	76.29	74.93	75.98	84.49
Exhaust inlet temperature (°C)	216.67	319.52	398.70	471.59
Total fluid flow rate (Kg/s)	1.45	1.44	1.38	1.08
Heat absorption rate of propylene (kW)	19.71	30.75	45.75	68.76
E_{in} of exhaust (kW)	26.87	49.65	76.08	115.04
E_{out} of exhaust (kW)	10.45	15.96	24.76	40.93

Table 3.6. Outlet temperature of 65°C on the fluid side of the heat exchanger

Temperature / calculations	25% Load	50% Load	75% Load	100% Load
Exhaust flow (Kg/s)	0.13	0.16	0.19	0.24
Exhaust outlet temperature (°C)	93.02	115.59	154.95	198.13
Heat exchanger inlet temperature (°C)	60.68	58.28	53.76	72.11
Heat exchanger outlet temperature (°C)	64.45	64.26	65.21	89.23
Exhaust inlet temperature (°C)	216.27	320.97	424.02	491.87
Total fluid flow rate (Kg/s)	1.44	1.42	1.08	1.09
Heat absorption rate of propylene (kW)	20.36	31.83	46.32	70.06
E_{in} of exhaust (kW)	26.49	48.92	82.46	122.97
E_{out} of exhaust (kW)	9.23	14.56	25.92	43.88

The total absorbed heat by the propylene with respect to each load and for each case is shown in Figure 3.11. According to Figure 3.11, the amounts of heat absorbed with respect to load for all cases followed the same trend with a difference of less than 4%. The difference may be due to the ambient conditions during the experimental run.

Figure 3.11. Absorbed heat by propylene at 87°C, 77°, and 65°C as heat exchanger outlet temperature.

Heat absorbed by propylene: The total heat present in the exhaust was calculated by assuming no heat exchanger attached to the engine exhaust pipe and allowing all the heat to be released to the atmosphere. For this case, the percentage of heat that can be recovered from exhaust was calculated by the ratio of heat absorbed by the propylene and total exhaust heat. The percentage of heat that can be recovered from the exhaust was observed to be constant for a 50-hour experimental run, which was about 52%. This is graphed in Figure 3.12.

Figure 3.12. Percentage of heat extracted from the total exhaust heat during a 50-hour experimental run time.

At different heat exchanger outlet temperatures on the fluid side, the heat recovery rate followed the same trend. The recovery rate by the heat exchanger was seen to be higher at lower engine loads than at higher engine loads. This can be explained as: at low engine loads the heat content in the exhaust gas was less which the coolant was able to recover most part of it that would show a greater recovery rate by the heat exchanger and vice versa. At higher loads, however, the unit heater needed to be much larger to attain better recovery. For practical application, as we did not deal with the unit heater, the recovery rate remained constant. Figure 3.13 details the recovery rate in different cases. The decrease in the recovery rate in each case can be better explained now as the result of exhaust heat content and the dissipated heat capability of the unit heater. With a larger unit heater, the recovery rate will stabilize.

Figure 3.13. Percentage of heat extracted from the total exhaust heat at different heat exchanger outlet temperatures on the fluid side.

3.3.3 Efficiency

Efficiency is the percentage of the ratio of amount of heat gained by the propylene and the heat loss by the exhaust. The efficiency of the heat exchanger was constant throughout the experimental process. The efficiency of the heat exchanger was calculated to be 85%. The related graph is shown in Figure 3.14.

3.3.4 Soot Accumulation

The accumulation of soot in the heat exchanger was considered a critical performance parameter, as large accumulations of soot are known to have a strong negative effect on the performance of heat recovery systems. Over a span of 350 hours of run time, the total soot produced by the exhaust gas was expected to be 4000 to 6000 grams, as much of the PM passes through the heat exchanger.

After the experiments, the heat exchanger was dismantled. No significant amount of soot was seen deposited on the tube fins or on the shell side that would affect the heat transfer rate. Only a thin layer of soot was seen on the tube fins. The thickness of the soot was considered to be a few hundredths of a millimeter. On cleaning the heat exchanger, about 150 grams of soot was found to be accumulated, which is far less than the 6 Kg maximum expected. With this small amount of accumulated soot, the heat exchanger

would need maintenance not more than two times every year. Figure 3.15, Figure 3.16, and Figure 3.17 show the soot accumulation on the heat exchanger.

Figure 3.14. Efficiency of the heat exchanger.

Figure 3.15. Shell side soot accumulation.

Figure 3.16. Soot accumulation on the fins.

Figure 3.17. Soot accumulation on the core side.

3.4 Corrosion Experiment

The corrosivity of the exhaust was evaluated by collecting condensate and evaluating both the pH and corrosion. In order to collect the condensate, the exhaust gas from the muffler was passed through a finned tube placed in a cold environment. The condensate was collected for different burning fuels. Stainless steel (SS316L) and mild steel (C1010) from Metal Samples were the corrosion coupons that were used in this experiment. The pH values of the condensates are tabulated in Table 3.7.

Table 3.7. pH value of exhaust condensate with the respective fuel burned

Fuel type	pH value
S1	3
S2	4
Conventional diesel	2
Blend*	3
Biodiesel	3

* Blend = 20% biodiesel + 80% conventional diesel

The exhaust condensate was then tested for its corrosive effect on different metals in two different cases:

Case 1: No exposure to air

Case 2: Exposure to air

The setup is shown in Figure 3.18 and Figure 3.19.

In both cases, the coupons were completely immersed in the exhaust condensate. In Case 1, the containers were airtight, while in Case 2, the containers were left to air without airtight caps.

Figure 3.18. No exposure to air.

Figure 3.19. Exposure to air.

The experiment was conducted for nearly four days for each coupon. In both cases, no corrosion was observed on the surface of SS316L coupons, but corrosion was observed on the surface of C1010 coupons. The corroded material was cleaned and checked for its weight. There was a weight loss in C1010 coupons. The changes in the actual weight of the C1010 coupons are tabulated in Table 3.8 for Case 1 and Case 2, and the respective bar graphs are shown in Figure 3.20 and Figure 3.21.

Table 3.8. Weight loss for C1010

	Conventional diesel	S1	S2	Blend
Case 1 (g)	0.0043	0.0013	0.0015	0.0046
Case 2 (g)	0.0067	0.004	0.0049	0.0048

When the heat exchanger was dismantled, the corrosive effect of exhaust gas on the heat exchanger was also investigated by examining the surface for the existence of corrosion spots on the tube and shell side. No trace of any such corrosion was observed. The reason seems to be that the exhaust temperature was always kept above the dew point of the acids and water vapor. This resulted in an absence of condensate in the heat exchanger throughout the run time. The construction material of the heat exchanger is SS316L because it reduces corrosion. This is of concern because even though the exhaust temperature was maintained above the dew point, there might be some acids formed during startup and shutdown that might cause corrosion.

Figure 3.20. No exposure to air.

Figure 3.21. Exposure to air.

3.5 Economic Analysis and Maintenance

Economic analysis was based on the present heat recovery system of the test engine built at UAF, and it assumed 100% use of heat recovery. For further calculations, the heat recovery rate used was 60 kW at rated load 8 hours per day (Table 3.9).

The heating value for conventional fuel = 130,000 BTU/gal

The fuel flow rate at 100% load = 8 gal/hr

Initial cost of the recovery system = \$30,000

Installation cost = \$6,000 (\$75/hr x 8 hours x 10 days)

Airfare, lodging, meals = \$1,950 (600 RTAF + \$90/day x 15 days)

Total capital cost = \$37,950

Simple breakeven in terms of fuel cost savings = $\$37,950/\$11,461 = 3.3$ years

Table 3.9. Estimated annual savings

	Per hour	Per day	Per year
Heat recovery	204,728 BTU	1,637,824 BTU	597 MBTU
Fuel consumption savings	1.57 gal	12.56 gal	4584.4 gal
Fuel cost savings @ \$2.50/gal	\$3.90	\$31.40	\$11,461.00

Payback time: The payback time was calculated based on varying interest rates and increasing fuel prices. The graph in Figure 3.22 shows respective payback times with different interest rates and fuel prices, assuming 100% use of the recovered heat. The value of fuel savings increases with an increase in the fuel prices that would decrease the payback time. Therefore, the graph in Figure 3.22 shows a decreasing curve with increasing fuel prices.

Fuel Sensitivity: The value of fuel savings depends on the present fuel cost. As the fuel prices alter considerably over a span of years, a fuel sensitivity analysis was performed. The curve in Figure 3.23 shows the fuel savings over a period of 5 years against the fuel cost per gallon. If the fuel price is above \$2/gal, the principle amount for the project will be attained in less than 5 years in terms of fuel savings.

The heat recovery system maintenance cost was based on one day of labor (\$75/hr) and a flight ticket (\$600), which comes to \$1200 every visit. The maintenance costs also include additional money (\$300) every year for supplies. Maintenance is required every 6 months.

Figure 3.22. Payback time with respect to fuel price and interest rate.

Figure 3.23. Fuel sensitivity curve.

3.6 Exhaust Emissions, Heat Content, and Temperatures of the Three Fuels

This section discusses the emissions, heat content, and temperatures of the exhausts obtained from an engine running on conventional diesel, synthetic fuel, and conventional diesel with different levels of a small amount of hydrogen. Heat content was estimated using exhaust mass flow rate, specific heat, and the temperature difference between the exhaust manifold and 170°C, which is the dew point of the sulfurous acid of exhaust. Exhaust flow rate was estimated using measured intake air flow rate and fuel consumption rate.

3.6.1 Conventional Diesel and Synthetic Fuel

Emissions: Table 3.10 shows the exhaust emissions for the two different fuels. It has been observed that the synthetic fuel performed better in every emission category.

- In comparison with conventional diesel, CO emission decreased by 40%, NO_x emission decreased by 9%, and THC emission decreased by 37%.

Table 3.10. Emissions for diesel and synthetic fuel

Test fuel	CO (g/kW-h)	NO _x (g/kW-h)	THC (g/kW-h)	Generator fuel efficiency (kW-h/gal)
Diesel	2.657	18.32	0.334	14.64
Synthetic	1.612	16.97	0.21	14.15

Heat Content and Temperatures: The properties of exhausts produced by different fuels at different engine loads are shown in Table 3.11, Table 3.12, and Table 3.13.

- For full load, the difference in specific heat between the exhausts of the two fuels was 0.6%, the difference in the exhaust mass flow rate was 3.6%, the difference in the exhaust temperature was 4% (or 20°C), and the difference in heat content was 9.9%.
- For 50% load, the difference in specific heat between the exhausts of the two fuels was 0.3%, the difference in the exhaust mass flow rate was 1.5%, the difference in the exhaust temperature was 2.4% (or 9°C), and the difference in heat content was 3.3%.

According to Table 3.13, the difference in calculated heat content between the exhausts of the two fuels was 9.9% for full load and 3.3% for 50% load. The estimated accumulated experiment uncertainty for heat content calculation, which involves at least 4 different measurements, is about $\pm 3\%$. In addition, environmental conditions can influence the heat content and temperature of exhaust. For 50% load, a 3.3% difference in heat content is within experiment uncertainty. For 100% load, a 9.9% difference in heat content resulted from a 20°C difference in exhaust temperature and a 3.6% difference in

heat flow rate. The heat content and temperature difference for full load may have some influence on the amount of heat recovered from the exhaust. However, the influence on the total heat recovered by a heat recovery system may be largely reduced when the engine load pattern (or average load about 70%), the design of the heat exchanger, and heat losses from the system (i.e., pipes, heat exchanger, etc.) are all taken into consideration. Table 3.10 shows that the emissions produced by synthetic fuel are less than conventional diesel. In addition, synthetic fuel is expected to have less PM emission [26, 27, 28] due to its zero aromatics and less corrosivity due to its zero sulfur content. Therefore, synthetic fuel is expected to have a slight disadvantage in amount of heat recovered and an advantage in maintenance, with less soot accumulation and corrosion.

Table 3.11. Exhaust property of conventional diesel fuel

% Load	Exhaust flow rate Kg/s	Overall Cp (J/gm-K)	Exhaust temp (T_{in} °C)	Exhaust temp (T_{out} °C)	Q (kW)
50	0.16	1.71	371.05	170.00	54.26
100	0.24	1.79	518.86	170.00	151.25

Table 3.12. Exhaust property of synthetic fuel

% Load	Exhaust flow rate Kg/s	Overall Cp (J/gm-K)	Exhaust temp (T_{in} °C)	Exhaust temp (T_{out} °C)	Q (kW)
50	0.16	1.70	362.09	170.00	52.46
100	0.23	1.78	498.25	170.00	136.31

Table 3.13. Comparison of exhaust property between conventional diesel fuel and synthetic fuel

% Load	% difference in exhaust mass flow rate	% difference in Cp	% difference in exhaust temperatures	% difference in exhaust heat
50	-1.50	0.29	2.42	3.31
100	3.60	0.64	3.97	9.87

3.6.2 Conventional Diesel and Conventional Diesel with a Small Amount of Hydrogen:

Emissions: Emissions data for different flow rates of hydrogen were plotted. Figure 3.24 shows the plot for hydrogen flow rate in lpm versus exhaust gas emissions in amps. The data were plotted for 0, 4, 10, 30, 50, 100, and 150 lpm of hydrogen and were compared with a 0 lpm hydrogen flow rate. The following results were observed:

- When compared with 0 lpm hydrogen, the amount of O_2 , NO , NO_x , and SO_2 in the exhaust gases did not vary significantly with the hydrogen flow rate.

- The emission of nitrogen dioxide (NO₂) increased significantly—by 155% between 0 lpm and 150 lpm hydrogen—with the increase of hydrogen concentration.
- The emission of CO has no regular trend. When compared with 0 lpm hydrogen, the change in CO concentration is not significant.

Figure 3.24. Comparison of exhaust gas emissions for different hydrogen flow rates.

Heat Content and Temperatures: The results for heat content and temperatures are given in Table 3.14, and discussed below.

Table 3.14. Heat in exhaust for different hydrogen flow rates.

H ₂ (lpm)	Exhaust flow rate Kg/s	Overall Cp (J/gm-K)	Exhaust temp (T _{in} °C)	Exhaust temp (T _{out} °C)	Q (kW)
0	0.153	1.72	395.15	170	59.07
10	0.152	1.72	395.02	170	58.86
50	0.152	1.72	402.53	170	61.01
150	0.148	1.72	404.94	170	60.11

- The exhaust gas mass flow rate, specific heat, and temperature did not vary significantly with the hydrogen flow rate.
- The calculated exhaust gas heat energy variation is within the experiment uncertainty range, so the variation in exhaust gas heat content is insignificant.

According to Table 3.14, the insertion of a small amount of hydrogen into the intake air stream has nearly no effect on the quantity and quality of exhaust heat and, therefore, has minimum effect on exhaust heat recovery and corresponding economics, as predicted.

3.7 Economic Analysis Program for Exhaust Heat Recovery System

A software program was developed for estimating the economic feasibility of installing an exhaust heat recovery system at any Alaskan village power plant for space heating and water loop temperature maintenance. This program was developed on Visual Basic for Application (VBA) in Microsoft Excel. The factors involved in the analysis include information about existing village community water loop and heating system infrastructure, existing heat recovery facility of power plant, engine size, power plant configuration, load pattern, and power plant location. The program uses given information to design the recovery system based on the trade-off between cost and amount of heat recovered. The designed system is analyzed for amount of heat recovery, energy requirement for operation, and maintenance requirements. Analyzed results are incorporated with capital cost, operation, and maintenance cost to evaluate the breakeven point and payback time.

In this program, two types of exhaust heat recovery systems can be designed and analyzed for any power plant. The first type uses only one gas-to-liquid (exhaust to coolant)—that is, shell and tube—heat exchanger, as shown in Figure 3.25. In this system the coolant from the community loop is pumped directly into the heat exchanger. The program calculates various physical parameters of the heat exchangers, such as heat transfer area required, overall heat transfer coefficient of heat exchanger, etc.

Figure 3.25. Line diagram of exhaust heat recovery system (one heat exchanger).

The program then goes to pressure-drop calculations in which the pressure drop in the pipe and piping components between the community loop and the heat exchanger (including the heat exchanger) is estimated. From the program library, the user can select various pipe components (such as valves, elbows, and strainers) which account for pressure drop in the system. Some of these selections are made by default, and the user can override them. Based on the pressure drop in the control system, the program suggests a pump size. The user can override the pump size and make another choice.

Then the program goes to cost estimates of the system in which capital investment cost and operation and maintenance costs are calculated. Capital costs include the cost of the heat exchanger, pipe components, pump, installation, etc. Operation and maintenance costs include operation of the system and frequency of maintenance, labor cost for maintenance, etc. Then the program goes to economic analysis, where the heat absorbed by the coolant, the capital costs and operation and maintenance costs (previously calculated), and the fuel costs and interest rates are utilized to do the economic feasibility study. The result of the economic analysis is represented as payback period, breakeven point, and profit gained for the given lifetime of the system. Finally, a copy of the all computations can be saved in the form of a Word document.

The second type of exhaust heat recovery system is one that uses two heat exchangers: a gas-to-liquid (exhaust to coolant)—that is, shell and tube heat exchanger—and a liquid-to-liquid (coolant to water) plate heat exchanger, as shown in Figure 3.26. The high temperature coolant, which is circulated between the gas-to-liquid heat exchanger and the liquid-to-liquid heat exchanger, takes heat from the exhaust in the shell and tube heat exchanger and gives off heat to the water in the plate heat exchanger. In this program, the design and analysis of both the shell and tube heat exchanger and the plate heat exchanger can be done at different loads. The program calculates the physical parameters of both heat exchangers, such as overall heat transfer coefficients, heat transfer area required for both heat exchangers, and amount of heat absorbed by both coolants.

Figure 3.26. Line diagram of exhaust heat recovery system (two heat exchangers).

The program then goes to pressure-drop calculations in the control system, that is, pipe and pipe components (valves, strainers, etc.) between both heat exchangers (the shell and tube HX and the plate heat exchanger), including the heat exchangers. The pressure drop, cost estimations and economic analysis calculations, and the program outputs are similar to the first type of heat recovery system described above. At the end, all computations can be saved in the form of a Word document.

The following example explains in detail the program usage for the second type of heat recovery system in which two heat exchangers are used. In this example, the exhaust conditions of the UAF diesel engine at different loads are used for calculations.

3.7.1 Results

1. The computer program for the exhaust heat recovery system was developed on the Visual Basic for Application (VBA) platform in Microsoft Excel.

2. The program was validated with the experimental results of the exhaust heat recovery system (which is the first task of this project), and the outputs obtained in the program were within comparable range of the experimental results.
3. The program was evaluated for a midsize diesel engine (DD50) which has 125 kW rated power at 1200 rpm. Earlier, an exhaust heat recovery experiment was conducted on the same diesel engine. A comparison of the experimental results and the program outputs are presented below.
 - The heat exhaust heat exchanger surface area used was 8 m^2 , and the overall heat transfer coefficient of the exhaust heat exchanger was $46 \text{ W/m}^2\text{-K}$, which is close to 9 m^2 and $37 \text{ W/m}^2\text{-K}$ calculated by the program.
 - The heat absorbed by coolant in the exhaust heat exchanger at 100%, 75%, 50%, and 25% loads are 68.76 kW, 45.75 kW, 30.75 kW, and 19.71 kW, respectively, and the program calculated values are 76.54 kW, 55.6 kW, 37.8 kW, and 20 kW at respective loads, which are in close acceptance with experimental values.
 - The pressure drop used for selecting the pump in the experiment is 10 psig or 23 feet of water head. The program calculated the total pressure drop in the system to be 9 psig or 21 feet of water head.
 - The calculated payback period for the experimental setup was 2.9 years at a fuel price of \$3.50/gal and 10% interest rate, assuming 100% use of recovered heat which is taken as 60kW. The calculated value of the program for the same amount of heat recovery rate, fuel price of \$3.50/gal, and 10% interest rate was 2.6 years, which is close to the experimental value.
4. The program can do additional tasks, such as analysis of the system for different intermediate loads; can take into account the pressure drop in a liquid-to-liquid heat exchanger; and can accommodate village heating loop modification cost estimates.
5. The program was written in Microsoft Excel, which is available on almost all personal computers. The program instruction manual is given in the Appendix.

Chapter 4 Conclusions

4.1 Experimental Economic Analysis of Exhaust Heat Recovery

Experimental work was conducted using an experimental engine to check the feasibility and economic effect of recovering exhaust heat energy for useful applications. According to the analysis from the experimental data, the following conclusions are made:

1. A procedure of heat recovery system design and analysis was demonstrated. This procedure is applicable to exhaust heat recovery system designs for many Alaskan village diesel generator sets.
2. Performance and economic results will be different from one case to another. Analysis is recommended before the application of an exhaust heat recovery system to a village generator set.
3. The performance of an exhaust heat exchanger is reliable and consistent.
4. In our case, about half the exhaust heat was recovered, which kept the heat exchanger exhaust outlet temperature high enough to avoid major maintenance problems.
5. No effects were observed on engine maintenance frequency due to the heat recovery system.
6. According to the soot analysis, the estimated time for heat exchanger maintenance is about two days every year.
7. Corrosion was not observed to be a problem in the laboratory test of 350 hours.
8. Operation cost is largely case-dependent. Influential parameters would include diesel fuel cost, the electrical power pattern, the use of heat, the existing infrastructure of the community heating system, etc.
9. According to these experimental results, the payback time for 100% use of recovered heat would be less than 3 years at a fuel price of \$3.50 per gallon, an interest rate of 10%, and an engine operation of 8 hours per day. For 80% use of recovered heat, the payback time would be less than 4 years.

4.1.1 Future Work

The following is future work that can be done as an extension to the present work:

1. The present engine-generator set at UAF will allow testing of other fuels. Fuels such as extra low sulfur fuel and biodiesel can be used for testing (e.g., for soot property, soot accumulation, and corrosivity).
2. Field tests need to be done.
3. The control system could be improved.
4. Performance could be investigated by applying exhaust heat recovery for different heating applications such as power and desalination.

5. The current exhaust heating system could be modified to an exhaust Rankine cycle system for exhaust heat recovery for power.

4.2 Evaluation of Exhaust Heat Contents and Temperatures for Different Fuels

Heat content and average temperatures of exhaust from synthetic diesel, conventional diesel, and conventional diesel with different levels of a small amount of hydrogen were evaluated using experimental data. Based on the study results, conclusions about the expected effects of the exhausts of different fuels on heat recovery are derived.

4.2.1 Synthetic Diesel Versus Conventional Diesel

Based on the experimental data, the conclusions are as follows:

1. In comparison with conventional diesel, synthetic fuel has a slight disadvantage in amount of heat recovery.
2. In comparison with conventional diesel, synthetic fuel has an advantage in maintenance for exhaust heat recovery application.

Future work: Since the environmental condition may have a non-negligible effect on exhaust temperature and mass flow rate, more experimental data may be needed to improve the reliability of the conclusion.

4.2.2 Diesel Fuel and Diesel Fuel with a Small Amount of Hydrogen

The self-ignition concentration of hydrogen is 4% by volume in air. In this experiment, the maximum level of 150 lpm of hydrogen amounts to 1.5% by volume of hydrogen in the intake air, which may be an insignificant volume. Based on the experimental data, conclusions are as follows:

1. A significant variation in emissions and exhaust heat content were not observed at this volume.
2. Adding a small amount of hydrogen into the engine intake manifold causes no economic effect on exhaust heat recovery application.

Future work: More tests with a hydrogen concentration higher than 4% by volume is desired in the future. With the hydrogen concentration higher than the self-ignition concentration, significant changes in exhaust heat content, emissions, and engine performance may be observed.

4.3 Economic Analysis Program for Exhaust Heat Recovery

1. Results given in the previous chapter show that the economic analysis program works well for studying the economic feasibility of installing exhaust heat recovery systems on rural diesel generators, as validated by experimental results.
2. The program can be improved further for incorporating more design data, thus providing more accurate design and analysis and more capable design and analysis tools.

3. The program can be easily extended to other exhaust heat recovery applications such as electric power generation by organic Rankine cycle, desalination, etc.
4. The cost estimations and economic analysis part of the program can be used for studying the economic feasibility of other exhaust heat recovery system applications such as organic Rankine cycle, desalinations, etc.

4.3.1 Future Work

In comparison with heating applications, the application of electrical power is more flexible in implementation, more efficient in energy transmission, and applicable year-round in Alaska. In general, the efficiencies of the up-to-date prototypes developed for power applications are still much lower than those for exhaust heating applications. However, it is expected that the efficiency of exhaust for power will be improved and become competitive in the near future. To develop a computer module for heat-to-power conversion (e.g., organic Rankine cycle) and incorporate it with the current computer program would make the current program a more useful tool for both exhaust heat-to-power research and exhaust heat-to-power economic analysis.

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Appendix A: Instruction Manual for Exhaust Heat Recovery System

(Economic Analysis Program for Exhaust Heat Recovery System)

A software program was developed for estimating the economic feasibility of installing an exhaust heat recovery system at any Alaskan village power plant for space heating and water loop temperature maintenance. This program was developed on Visual Basic for Application (VBA) in Microsoft Excel. The factors involved in the analysis include information about existing village community water loop and heating system infrastructure, existing heat recovery facility of power plant, engine size, power plant configuration, load pattern, and power plant location. The program uses the given information to design the recovery system based on the trade-off between the cost and amount of heat recovered. The designed system is analyzed for amount of heat recovery, energy requirement for operation, and maintenance requirements. Analyzed results are incorporated with capital cost and operation and maintenance costs to evaluate the break-even point and payback time.

In this program two types of exhaust heat recovery systems can be designed and analyzed for any power plant. The first type uses only one gas-to-liquid (exhaust to coolant)—that is, shell and tube heat exchanger—as shown in Figure A.1. In this system the coolant from the community loop is pumped directly into the heat exchanger. The program calculates various physical parameters of the heat exchangers, such as heat transfer area required, overall heat transfer coefficient of heat exchanger, etc.

Figure A.1. Line diagram of exhaust heat recovery system (one heat exchanger).

The program then goes to pressure-drop calculations in which the pressure drop in the pipe and piping components between the community loop and the heat exchanger (including the heat exchanger) is estimated. From the program library, the user can select various pipe components (such as valves, elbows, and strainers), which account for pressure drop in the system. Some of these selections are made by default, and the user can override them. Based on the pressure drop in the control system the program suggests a pump size. The user can override the pump size and make another choice. Then the program goes to cost estimates of the system in which capital investment cost and operation and maintenance costs are calculated. Capital costs include the cost of the heat exchanger, pipe components, pump, installation, etc. Operation and maintenance costs include operation of the system and frequency of maintenance, labor cost for maintenance, etc. Then the program goes to economic analysis, where the heat absorbed by the coolant, the capital cost and operation and maintenance (previously calculated),

and the fuel costs and interest rates are utilized to do the economic feasibility study. The result of the economic analysis is represented as payback period, breakeven point, and profit gained for the given lifetime of the system. Finally, a copy of the all computations can be saved in the form of a Word document.

The second type of exhaust heat recovery system is one that uses two heat exchangers: a gas-to-liquid (exhaust to coolant)—that is, shell and tube heat exchanger—and a liquid-to-liquid (coolant to water) plate heat exchanger, as shown in Figure A.2. The high temperature coolant, which is circulated between the gas-to-liquid heat exchanger and the liquid-to-liquid heat exchanger, takes heat from the exhaust in the shell and tube heat exchanger and gives off heat to the water in the plate heat exchanger. In this program, the design and analysis of both the shell and tube heat exchanger and the plate heat exchanger can be done at different loads. The program calculates the physical parameters of both heat exchangers, such as overall heat transfer coefficients, heat transfer area required for both heat exchangers, and amount of heat absorbed by both coolants.

Figure A.2. Line diagram of exhaust heat recovery system (two heat exchangers).

The program then goes to pressure-drop calculations in the control system, that is, pipe and pipe components (valves, strainers, etc.) between both heat exchangers (the shell and tube HX and the plate heat exchanger), including the heat exchangers. The pressure drop, cost estimations and economic analysis calculations, and the program outputs are similar to the first type of heat recovery system described above. At the end, all computations can be saved in the form of a Word document.

The following example explains in detail the program usage for the second type of heat recovery system in which two heat exchangers are used. In this example, the exhaust conditions of the UAF diesel engine at different loads are used for calculations.

Usage of Exhaust Heat Recovery Program

1. An overview of economic analysis

The economic feasibility study for installing an exhaust heat recovery system is done based on expenditure and returns. Figure A.3 shows the flow chart for economic analysis. Expenditures include the total initial capital costs, which include component costs (such as for the heat exchanger, pipe, pipe components, etc.), installation costs (such as for labor, travel, etc.), interest rate on capital and operation and maintenance costs. On the other hand, we have returns in the form of amount of heat recovered, fuel costs, and fuel price escalation charges so that we can calculate the amount saved on fuel and payback period, breakeven point, and profit or loss for the given lifetime of a system.

Figure A.3. Economic analysis flow chart.

2. Exhaust heat exchanger design and analysis

Figure A.4 shows the screen shot for calculating different physical parameters of an exhaust heat exchanger. This example is explained by using the exhaust parameters of a DD50 diesel engine located at the UAF campus. In the design of the exhaust heat exchanger (shell and tube heat exchanger), the exhaust conditions of the UAF diesel engine at 100% load are taken. To add the exhaust conditions of another diesel engine or any power plant, click on “Type of Diesel Engine” button. A window for the addition of a diesel engine appears on the screen as shown in Figure A.5. At this window in the fields provided enter the diesel engine or power plant name (to distinguish from others) and exhaust properties at different loads, and click “Add.” Select the diesel engine and click “OK.” The properties of exhaust at 100% load are automatically taken for the design.

Figure A.4. Exhaust heat exchanger design and analysis.

Next, go to selection of coolant and its properties. To select or to add a coolant from or into the program library click on “Select the Type of Coolant.” A window for addition or selection of coolant appears as shown in Figure A.6. For addition, enter the coolant type and its physical properties in the fields provided and click “Add.” Select a coolant type (high temperature coolant) that circulates between exhaust heat exchanger and liquid-to-liquid heat exchanger. From now on, the high temperature coolant (HT coolant) refers to the coolant circulating between the shell and tube heat exchanger and the liquid-to-liquid heat exchanger.

The design of the exhaust heat exchanger can be done based on the mass flow rate of the coolant or based on the coolant outlet temperature. In this example, it is done based on the coolant outlet temperature. Enter coolant inlet and outlet temperatures in degrees Kelvin in the fields provided

Select the type of Diesel Engine Data

Engine	Medot(kg/s)	Tein(in Kelvin)	Teout(in Kelvin)
☒ UAF Diesel Engine	0.2400	737.0000	460.0000
☐ DD60	0.3000	737.0000	460.0000
☐			
☐			
☐			
☐			
☐			
☐			

OK Delete Engine

Cancel

Add Engine Data

Engine type or Name

Medot (kg/s) Tein(Kelvin) Teout(Kelvin)

100% Load

75% Load

50% Load

25% Load

Add

P.S. Engine Names should be different even if they are of same make (e.g. If you have two engines of DD60, then name them DD60-1 and DD60-2.)

Figure A.5. Adding a diesel engine and exhaust properties.

Select the type of Coolant

Coolant	Specific Heat (Cp in J/kg-K)	Density (kg/m3)	Dynamic Viscosity (Pa-s)	Thermal Conductivity (W/m-K)
☒ Water	4180.000	1000.000	3.5500E-04	0.600
☐ PropyleneGlycol	2481.000	1036.000	3.2200E-03	0.232
☐ PropyleneGlycol/Water 60/40	3800.000	1048.250	1.3500E-03	0.327
☐ PropyleneGlycol/Water 50/50	3532.000	1042.870	1.1200E-03	0.365
☐ EthyleneGlycol	2347.000	1113.000	2.1200E-03	0.276
☐ EthyleneGlycol 50/50	3281.000	1073.350	9.8000E-04	0.380
☐ EthyleneGlycol 60/40	3084.000	1086.270	1.1000E-03	0.349
☐				

OK Delete Coolant Cancel

Add a Coolant

Coolant

Specific Heat (Cp in J/kg-K)

Density (Kg/m3)

Dynamic Viscosity (Pa-S)

Thermal Conductivity (W/m-K)

Add

P.S. Coolant Names should be different even if they have same composition (e.g. If you have two different property values for Propylene Glycol 50/50 then give Propylene Glycol 50/50-1 and Propylene Glycol 50/50-2.)

Figure A.6. Coolant selection and addition window.

Next, go to the specifications of the shell and tube heat exchanger. The heat exchanger specifications, such as the tube outer diameter, tube thickness, thermal conductivity of tube material, and fouling-factor specifications, are to be entered in respective fields. The film heat transfer coefficients of exhaust on the shell side and the coolant on the tube side can be input by the user, or the program can calculate the values using heat transfer principles.

The outputs of the exhaust heat exchanger design window are overall heat transfer coefficient of heat exchanger, heat transfer area requirement for heat exchanger, heat absorbed by HT coolant, and effectiveness and efficiency of heat exchanger. If the design is based on the coolant outlet temperature, then the coolant mass flow rate is output and vice versa.

Then, if wanted, the heat exchanger can be analyzed for different intermediate loads by going to analysis mode (click on “Analysis” tab) as shown in Figure A.7.

Diesel Engine data at different loads		
	Medot (kg/s)	Tein(Kelvin)
100% Load	0.24	737
75% Load	0.19	686
50% Load	0.16	601
25% Load	0.14	497

Diesel Engine data used for calculations	
Percentage Load (in Percent)	80 %
Mass Flow of Exhaust (Medot)	0.2 kg/s
Exhaust inlet Temperature (Tein)	696.2 Kelvin

Interpolate

Coolant Data	
Coolant Inlet Temperature (Tcin)	333 Kelvin
Mass Flowrate of Coolant (Mcdot)	1 kg/s 15.1987 GPM
Coolant Outlet Temperature (Tcout)	Kelvin

Calculations	
Overall Heat Transfer Coefficient of HX (Uo)	36.555 W/m2-K
Area of Exhaust HX (A)	8.907 m2
Mass flowrate of coolant(Mcdot)	kg/s GPM
Coolant outlet temperature (Tcout)	350.41 Kelvin
Exhaust Outlet Temperature (Teout)	421.63 Kelvin
Heat absorbed by Coolant (Qc)	61492.12 W
Effectiveness of Heat Exchanger (Qactual/Qmax)	0.7558
Efficiency of Heat Exchanger	69.29 %

Figure A.7. Exhaust heat exchanger analysis window.

In the example shown in Figure A.7, the heat exchanger is analyzed at 80% load of engine by interpolating between 100% and 75%. The analysis is done based on mass flow rate of the coolant. In analysis the overall heat transfer coefficient of the heat exchanger and the area of the heat exchanger remain constant from the design mode. The results of the analysis mode are exhaust outlet temperature, heat absorbed by coolant, and effectiveness and efficiency of the heat exchanger. If the analysis is based on the coolant mass flow rate, then the coolant outlet temperature is output and vice versa.

3. Liquid-to-liquid heat exchanger design and analysis

Next, go to Liquid-to-Liquid Heat Exchanger (plate heat exchanger) Design and Analysis. The HT coolant temperature inlet-to-plate heat exchanger is taken to be equal to the HT coolant outlet temperature of the exhaust heat exchanger. If one clicked on “Go to Liquid-to-Liquid HX Design and Analysis” when in design mode, the mass flow rate of high temperature coolant and high temperature coolant inlet temperature for plate heat exchanger (i.e., coolant outlet temperature of shell and tube heat exchanger) are taken from the design calculations, and when in analysis mode, the values are taken from analysis calculations. Figure A.8 shows the design and analysis window for liquid-to-liquid heat exchanger (i.e., plate heat exchanger).

Liquid-to-Liquid Heat Exchanger Design and Analysis

Design | **Analysis**

Coolant from Exhaust Heat Exchanger
 High Temperature Coolant used in Exhaust HX: PropyleneGlycol/Water 50/50
 Specific Heat of the High Temperature Coolant (Cpht): 3532 J/kg-K
 Mass Flowrate of High Temperature Coolant (Mhtc): 0.7 kg/s
 High Temperature Coolant inlet Temperature (Thitin): 363 Kelvin
 High Temperature Coolant outlet Temperature (Thitout): 340 Kelvin

Type of Low Temperature Coolant and Application
 Select the type of Low Temperature Coolant: Water
 Specific Heat of Coolant (Cp): 4180 J/kg-K
 Density of Coolant: 1000 kg/m3
 Thermal conductivity of Coolant: 0.6 W/m-K
 Dynamic Viscosity (μ): 0.000355 Pa-s (or) Kg/m-s
 1) Space Heating 2) ☒ Community water Loop Temperature maintenance
 a) ☐ High Temperature base board SH
 b) ☐ Low Temperature base board SH
 c) ☐ Central floor Radiant SH
 Low Temperature Coolant Inlet Temperature (Tlctin): 288 Kelvin
☐ Mass Flowrate of Low Temperature Coolant (Mltc): Kg/s GPM
☒ Low Temperature Coolant Outlet Temperature (Tlctout): 303 Kelvin

Plate Heat Exchanger Specifications
 Thickness of the Plate (t): 0.02 inches
 Gap between Plates (Usually 0.2 to 0.4 inches): 0.3 inches
 Thermal Conductivity of Plate Material (K): 35 W/m-K
 Convective Heat Transfer Coefficient of Low temperature Coolant (hl): ☒ User Input: 1200 W/m2-K ☐ Calculate
 Fouling Resistance of Low Temperature Coolant (Rlt): 0 m2-K/W
 Convective Heat Transfer Coefficient of High temperature Coolant (ho): ☒ User Input: 1200 W/m2-K ☐ Calculate
 Fouling Resistance of High Temperature Coolant (Rht): 0 m2-K/W

Calculations
 Overall Heat Transfer Coefficient of Plate HX (Uo): 594.82 W/m2-K
 Area of Plate Heat Exchanger (A): 1.71 m2
 Mass Flowrate of Low Temperature Coolant (Mltc): 0.91 kg/s 14.4238 GPM
 Low Temperature Coolant Outlet Temperature (Tlctout): Kelvin
 Heat Absorbed by Low Temperature Coolant (Qltc): 57057 W
 Effectiveness of Plate heat exchanger (Qactual/Qmax): 0.3077
 Efficiency of Plate Heat Exchanger (Thitout=300K): 36.63 %
 Efficiency of Plate Heat Exchanger (Thitout=Tlctin): 30.77 %

Buttons: Calculate, Go to Pipe Pressure Drop Calculations, Go back to Exhaust HX Design, Reset, Cancel

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Figure A.8. Plate heat exchanger design and analysis window.

The design and analysis of a liquid-to-liquid heat exchanger is very similar to that of an exhaust heat exchanger. The low temperature coolant can be selected from the library or can be entered manually. The type of heat application, such as space heating or community water loop temperature maintenance, can be selected from the options provided. The physical parameters (overall heat transfer coefficient, heat transfer area required, etc.) of a plate heat exchanger can be calculated based on low temperature coolant mass flow rate or low temperature coolant outlet temperature. In this example we selected based on coolant outlet temperature. The plate heat exchanger specifications can be entered in the next column. Specifications such as plate thickness, gap between plates, thermal conductivity of plate material, and fouling factor of high temperature coolant and low temperature coolant can be entered in respective fields. The convective heat transfer coefficients of high temperature coolant and low temperature coolant can be input by the user or the program can make the calculation. The results of the liquid-to-liquid heat

exchanger (i.e., plate heat exchanger) design are expressed as the overall heat transfer coefficient of the plate heat exchanger, heat transfer area required, heat absorbed by low temperature coolant, and effectiveness and efficiency of the heat exchanger. If the design is based on the low temperature coolant outlet temperature, then the coolant mass flow rate is output and vice versa.

After designing the plate heat exchanger (i.e., after calculating the overall heat transfer coefficient of the heat exchanger and heat transfer area required), an analysis of the plate heat exchanger can be done; that is, we can analyze the heat exchanger for different low temperature coolant outlet temperatures and mass flow rates by clicking on the “Analysis” tab. Figure A.9 shows the analysis window of the plate heat exchanger. In analysis, the overall heat transfer coefficient and the area of the plate heat exchanger remain constant from the design mode.

Liquid-to-Liquid Heat Exchanger Design and Analysis

Design Analysis

Coolant from Exhaust Heat Exchanger
 Mass flowrate of High Temperature Coolant (Mh_{tc})
 0.7 kg/s
 High temperature Coolant inlet Temperature (Th_{tcin})
 363 Kelvin

Low Temperature Coolant Inlet Temperature (Tl_{tcin})
 288 Kelvin
☒ Mass flowrate of Low temperature Coolant (Ml_{tc})
 1 kg/s 15.8503 GPM
☐ Low Temperature Coolant outlet temperature (Tl_{tcout})
 Kelvin

Calculate

Calculations

Overall Heat Transfer Coefficient of Plate HX (U_o)
 594.82 W/m²-K
 Area of Plate Heat Exchanger (A)
 1.71 m²
 Mass Flowrate of Low Temperature Coolant (Ml_{tc})
 kg/s GPM
 Low Temperature Coolant outlet temperature (Tl_{tcout})
 301.72 Kelvin
 High Temperature Coolant outlet Temperature (Th_{tcout})
 339.8 Kelvin
 Heat Absorbed by Low Temperature Coolant (Ql_{tc})
 57349.6 W
 Effectiveness of Plate heat exchanger (Q_{actual}/Q_{max})
 0.3093
 Efficiency of Plate Heat Exchanger (Th_{tcin}=300K)
 36.82 %
 Efficiency of Plate Heat Exchanger (Th_{tcin}=Tl_{tcin})
 30.93 %

Go to Pipe Pressure Drop Calculations Go back to Exhaust HX Design Reset Cancel

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Figure A.9. Liquid-to-liquid heat exchanger analysis window.

4. Pressure-Drop Calculations

After the design of both the shell and tube heat exchanger (exhaust heat exchanger) and the plate heat exchanger (liquid-to-liquid heat exchanger), the program goes to pressure-drop calculations in pipe and piping components (valves, strainers, elbows, etc.) forming the control system between the two heat exchangers (including the heat exchangers).

Figure A.10 shows the pressure-drop calculations window. The high temperature coolant, which is circulating between the exhaust heat exchanger and the liquid-to-liquid heat exchanger, and its mass flow rate and physical properties (i.e., density and viscosity) are automatically taken for the pressure-drop calculations, as shown in the Figure A.10. The pipe material and the pipe diameters can be selected from the program library. Length of pipe and elevation change (i.e., the elevation difference between the lowest and highest point in the piping system) are user inputs. The pressure drop in both heat exchangers can be entered in respective fields.

Pressure Drop Calculations in Pipes

Pipe Specifications

Mass Flow rate of Coolant: 0.7 Kg/s

Density of Coolant: 1042.87 kg/m3

Volume Flowrate of Coolant: 10.6391 GPM

Dynamic Viscosity(μ): 0.00112 Pa-s

Elevation Change (Z): 10 Feet(ft)

Pipe Material: Copper

Absolute Roughness: 0.000005 Feet(ft)

Pipe Type and Nominal Diameter: copper Type L tubing 1.50"

Pipe Internal Diameter: 1.505 Inches

Length of the Pipe: 46 Feet(ft)

Pressure Drop in Exhaust Heat Exchanger (on Liquid Side): 1 PSIG

Pressure Drop in Liquid-to-Liquid Heat Exchanger: 2 PSIG

Select the type of Valves and Calculate their total cost

Total K-Factor for valves: 36.43

Total Cost of Piping Components: \$6000

Pressure Drop Calculations

Velocity of Fluid flow: 1.9187 fps

Reynolds Number: 20817.0852

Friction Factor: 0.02568

Total Pressure Drop: 8.706 PSI

Buttons: Calculate, Reset, Cancel, Go back to HX Design, Go to Cost Estimates

Figure A.10. Pressure-drop calculations window.

The piping components (i.e., valves, elbows, strainers, etc.) can be selected by clicking on "Select the Type of Valves and Calculate Their Total Cost." Figure A.11 shows the screen shot for selection of piping components. For selecting a fitting, enter a value greater than zero in the quantity field. The K-factor for fittings is taken from the

manufacturer's catalog. The user can override the default values provided by the program. The total cost of the components can be calculated by entering the unit cost of each component in its respective field, or a total approximate cost can be entered in the "Total Cost of Piping Components" field provided in the pressure-drop calculations screen.

The results of the pressure-drop calculations are shown as fluid velocity in the piping system, Reynolds number, friction factor in the pipe, and total pressure drop in the system between the two heat exchangers (including the heat exchangers). The pressure drop can be calculated in psig (or) head of water in feet.

Select the type of Valves

Valves | Elbows and Tees | Bends | Mitre Bends | Check Valves and Strainers | Pipe Exit and Entrance | Other Components

Fitting	L/D	Quantity	K-Factor	Unit Cost (\$)
Angle Valve	55	4	1.155	0
Angle Valve	150	0	3.15	0
Ball Valve	3	0	0.063	0
Butterfly Valve		0	0	0
Gate Valve	8	0	0.168	0
Globe Valve	340	0	7.14	0
Plug Valve Branch Flow	90	0	1.89	0
Plug Valve Straightway	18	0	0.378	0
Plug Valve 3-Way Thru-Flow	30	0	0.63	0

Other K-Factor: 0

Calculate Total K-Factor and Component Cost

Total K-Factor: 36.43

Total Cost of Pipe Components: \$ 0

Go to Pressure Drop Calculations | Reset | Cancel

Figure A.11. Piping components (i.e., valves, elbows, strainers, etc.) window which amount for pressure drop in control system.

5. Cost Estimations

After pressure-drop calculations in the piping system, the program goes to cost estimations of the system, where the total capital cost and operation and maintenance

costs can be calculated. The total capital cost includes the cost of heat exchangers, total cost of piping components, total cost of structural components which are used to build the system and do not account for pressure drop, and initial installation costs which include cost of labor, number of days for installation, freight charges, etc. Operation and maintenance costs include the cost for operating the system (electricity charges for running the pump, etc.) and maintenance costs such as labor, frequency of maintenance per year, number of days for maintenance per visit, freight charges, etc. Figure A.12 shows the screen shot for cost estimations.

Minimum components required for building the system are given in the first two tabs, labeled “Initial Capital Costs-1” and “Initial Capital Costs-2”. If the user has more components, there is an additional tab labeled “Other Initial Costs,” which is left blank so that the user can enter the component name, quantity, and unit cost, as shown in Figure A.13. Installation costs, which are added to total initial capital costs, can be calculated in the tab labeled “Initial Capital Costs-2,” as shown in Figure A.14. There, all costs add up to Total Initial Capital.

Cost Estimations

Initial Capital Costs-1 | Initial Capital Costs-2 | Operation and Maintenance Costs | Other Initial Costs

Component	Unit Cost(\$)	Quantity	Total Cost(\$)
Exhaust Heat Exchanger			10019
Liquid-to-Liquid Heat Exchanger			5000
Pump Cost	672	1	672
Pipe Cost	3 per foot	46 feet	138
Pipe Components Total Cost (Valves, Strainers, etc.,)			6000
Box Beam	62.84	6	377.04
Flat Stock	24.37	3	73.11
Angle Bar	33.22	1	33.22
Slip on Flange	38.3	2	76.6
Insulating Foam Sealant	8.49	6	50.94
Expansion Tank	34.18	1	34.18
Insulation	61.79	1	61.79

Continued in Next Tab.....

Total initial Capital (Component Costs + Installation Costs) 31388.34 \$

Total Operation and Maintenance Costs 4317.11 \$

Go back to Pipe Pressure Drop Calculations Calculate Reset Go to Economic Analysis Cancel

Figure A.12. Cost estimations screen shot.

The operation and maintenance screen shot is shown in Figure A.15. If the user has specific values for total installation costs and total operation maintenance costs, then the calculated values can be overridden. The total operation and maintenance costs are calculated for a year.

Cost Estimations

Initial Capital Costs-1 | Initial Capital Costs-2 | Operation and Maintenance Costs | Other Initial Costs

Component	Unit Cost(\$)	Quantity	Total Cost(\$)
Parts	400	1	400
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0
	0	0	0

Total initial Capital (Component Costs + Installation Costs) 31388.34 \$

Total Operation and Maintenance Costs 4317.11 \$

Figure A.13. Additional structural components window.

Cost Estimations

Initial Capital Costs-1 | Initial Capital Costs-2 | Operation and Maintenance Costs | Other Initial Costs

Component	Unit Cost(\$)	Quantity	Total Cost(\$)
Un-grounded Thermocouple Probes	26.8	6	160.8
Teflon insulated Thermocouple wire	405	1	405
Miniture Thermocouple connector pair	4	50	200
Pressure Gauge	10	3	30
24 gauge galvanized metal	100	1	100
Nickel immersion temperature sensor	35.02	1	35.02
30VA transformer for RWD controller	21.64	1	21.64

Installations Costs

Total number of days for installation: 10 Days

Working hours per day: 8 per Day

Labor required for installation: 1

Labor cost per hour: 75 \$/hr

Round trip Total Airfare for all persons: 600 \$

Lodging and meals cost per day per person: 90 \$/Day

Total other installation costs: 0 \$

Total Installation costs: 7500 \$

Total initial Capital (Component Costs + Installation Costs): 31388.34 \$

Total Operation and Maintenance Costs: 4317.11 \$

Go back to Pipe Pressure Drop Calculations | Calculate | Reset | Go to Economic Analysis | Cancel

Figure A.14. Installation costs window.

Cost Estimations

Initial Capital Costs-1 | Initial Capital Costs-2 | Operation and Maintenance Costs | Other Initial Costs

Operation Costs

Number of hours engine is run per year: 2920 hours

Electric power required to run the pump: 0.33 hp

Cost of Electric power: 2 per KWH

Other operation costs per year: 0 \$

Total Operation Costs per year: 1437.11 \$

Maintenance Costs

Number of maintenance visits required per year: 2

Total number of days for maintenance per visit: 1 Days

Labor required for maintenance: 1

Cost of Labor per hour: 75 \$/hr

Round trip Total Airfare for all persons per visit: 600 \$

Lodging and meals cost per day per person: 90 \$/Day

Other maintenance costs (supplies etc): 300

Total Maintenance costs per: 2880 \$

Total initial Capital (Component Costs + Installation Costs): 31388.34 \$

Total Operation and Maintenance Costs: 4317.11 \$

Go back to Pipe Pressure Drop Calculations | Calculate | Reset | Go to Economic Analysis | Cancel

Figure A.15. Operation and maintenance costs screen shot.

6. Economic Analysis

After cost estimations of the system, the user goes to economic analysis of the system, where the total initial capital and operation and maintenance costs are taken from the previous cost estimations window. Figure A.16 shows the screen shot for economic analysis. In economic analysis, the heat recovered to the low temperature coolant is taken from the liquid-to-liquid heat exchanger window. The heating value of fuel, engine hours run per year, fuel price, and fuel price escalation rate are user inputs (program gives default values which the user can override). By default, the program gives 2920 hours (8 hr per day * 365 days) of engine run time per year.

The output of the economic analysis window has two types. Based on simple interest, the heat recovered per year and total savings on fuel, simple breakeven point and payback period are calculated. Based on compound interest and the lifetime of the system, the cost of capital per year (i.e., net present value), the total cost of heat recovery system per year (i.e., cost of capital per year + operation and maintenance costs per year), fuel savings for

the given lifetime of the system, and profit or loss made during the lifetime of the system are calculated.

Economic Analysis

Capital Costs and Operation & Maintenance Costs

Total Initial Capital Cost: 31388.34 \$

Total Operation and Maintenance Costs per year(B): 4317.11 \$

Interest rate on capital (Simple Interest for Payback Period): 10 %

Lifetime of System: 10 years

Interest rate on capital (Compound Interest for Capital cost per year): 6 %

Heat Recovered and Fuel Price

Heat Recovered from Exhaust: 57057 W

Engine hours per year: 2920 Hours

Heating value of Conventional Diesel Fuel per gallon: 130000 BTU/gal

Fuel price per gallon: 4.5 \$/gal

Percentage increase of fuel price per year: 2 %

Calculations

Heat Recovered per year: 568514866.351 BTU/year

Total savings on fuel per year: 19679.36 \$/year

Simple Breakeven (Capital/Savings on Fuel): 1.59 years

Payback period for given interest rate on capital: 2.37 years

Cost of Capital per year(A): 4264.67 \$

Total cost of Heat Recovery system per year (A+B): 8581.78 \$

Total investment for given Lifetime of system(C): 85817.8 \$

Fuel Savings for given lifetime of system(D): 215483.51 \$

Profit or Loss made during life time of system (D-C): 129665.71 \$

Go back to Cost Estimates Calculate Reset Cancel

Save as Word

Figure A.16. Economic analysis screen shot.

All the computations made—from the design of the exhaust heat exchanger to the economic analysis—can be saved as a Word document by clicking on “Save as Word.” Figure A.17 shows the screen shot for a Word document saved for this example.

At any point in the program computations, the user can go back to a previous window to check computations, make required changes in the inputs, and recalculate.

Figure A.17. Saved as Word document.

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