The Selection and Design of a Heat Recovery Application for Diesel Generators Used in Alaskan Villages

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***Abstract*— In rural Alaska, there are nearly 180 villages consuming about 374,000 MWh of electric energy annually from individual diesel generator facilities. For modern diesel engines, about 40% of fuel energy is released as waste heat from engine exhaust and turbocharger compressed air. If half of the released heat was put to appropriate use, there would be a great fuel savings. The purpose of this study was to select an appropriate exhaust heat recovery application for Alaskan village diesel generators and to study its feasibility and economic effect.**

Recovery heat from the exhaust of Alaskan village diesel generators has long been avoided due to the probable problems of corrosion caused by sulfuric acid and sulfurous acid and performance deterioration caused by corrosion and exhaust soot accumulation. Since the new regulation that requires diesel engines to use ultra low sulfur diesel, the intensity of corrosion and performance deterioration problems may be reduced mostly and the exhaust heat recovery may become feasible and economical. This report presents the selection process of an appropriate heat recovery application for Alaskan village diesel generators, the design of an experimental heat recovery system for testing, and the result of the effect of exhaust on the fouling factor and corrosion of the heat recovery system. Also included in this report is the result obtained from feasibility and economic analysis.

***Index Terms* – diesel, exhaust, heat recovery,**

I. INTRODUCTION

In rural Alaska, there are nearly 180 villages consuming about 374,000 MWh of electric energy from individual diesel generators [1]. During the process of producing electric power, there is unused heat from the engine. The single largest amount of unused heat from the engine is the exhaust heat, which contains about 30% of the fuel energy. If the waste heat were put to appropriate use, there would be great energy or fuel savings. Work, which studies the selection of the most appropriate engine heat recovery method, is needed as part of the village economic development.

There are many different heat recovery methods available for capturing engine waste heat, such as thermal electric conversion [2], heat to power conversion (e.g. Sterling engines, steam engines) [3], direct heat application (e.g. space heating, community water loop heating) [4], heat for refrigeration and air conditioning [5], heat for desalination, etc [6, 7, 8]. To optimize the benefit that heat recovery systems can bring to the Alaskan villages, the engine performance characteristics and operational conditions that have important effects on the performance of heat recovery systems have to be understood. The important engine performance characteristics and operational conditions may include diesel-generator load, fuel composition, soot produced, and exhaust gas corrosivity. Engine load and fuel composition will affect the amount of heat to be recovered by the heat recovery system from the exhaust. Exhaust gas corrosivity and soot content will affect the maintenance of the heat recovery system [9].

This project studies feasibility and economic effect of applying exhaust and charge air heat recovery to Alaskan village diesel generators. This paper focuses on studying the application of exhaust heat recovery. Recovery of heat from the exhaust of Alaskan village diesel generators has long been avoided due to the probable problems of corrosion caused by sulfurous acid and sulfuric acid and performance deterioration caused by corrosion and exhaust soot accumulation. Decades ago, attempts of recovering exhaust heat were tried and discarded due to serious drawbacks caused by the systems [10]. Since then heat exchanger design [11, 12] and control systems have been improved and a heat recovery system using new industrial components may work better than systems of decades ago. In addition, the new regulation requires diesel engines to use ultra low sulfur diesel fuel, the intensity of corrosion [13] and performance deterioration problems may be reduced mostly and the exhaust heat recovery system may become feasible and economical.

In this study, space heating was the selected application for exhaust heat recovery. The selection of space heating was due to its potential in optimizing the economic benefit to Alaskan villages. Space heating, which included public space heating using baseboard system and floor radiant systems, was also proposed by Alaska energy authority as candidates for future rural Alaska cogeneration development [4]. The following sections describe details of the diesel generator and the heat recovery system used for this study along with system instrumentation, experimental results, and the final conclusion.

II. TEST BED

This section discusses the test bed which included a diesel generator facilitated with digital load control capability and details of the designed heat recovery system.

A. *The Diesel Generator System*

This experiment was conducted at Energy Center, University of Alaska Fairbanks using a DD50 diesel-generator set. The generator was connected to a 250 kW external resistive/reactive load bank from Load Tech for engine load control. Load bank was interfaced with LABVIEW IV system for load control signals. Property of the diesel engine is listed in Table I. The diesel generator set and data acquisition system were placed inside an ISO container while the load bank sitting outside of the ISO container (Figure 1).

TABLE I
DIESEL ENGINE PROPERTY

Number of cylinder: 4	Number of Stroke: 4
Bore x Stroke: 130mm x 160mm	Fuel Injection: Electronic unit injector
Aspiration: Turbocharged	Displacement: 8.5 L
Rated Power: 125 kW	Rated Speed: 1200 rpm
Exhaust Gas Recirculation: None	Exhaust After treatment: None

Figure 1 Diesel Generator Location

B. *Exhaust Heat Recovery System*

The exhaust heat recovery system, which included both heat recovery component as heat source and heat dissipation component as heating load, was designed according to the following requirements:

Apply existing technology,
Optimize overall efficiency for varying engine load (In general, village generator load ranging from 25% to 100% of rated load).
Minimize corrosion to exhaust system.
Meet back pressure requirement (13.8 kPa or 2 psi).
Meet dimension constraints (very limited space).
Be able to emulate different heating applications of Alaskan villages.
Be able to measure system and component performance.
Be easy to maintain and do not increase maintenance frequency.
Work under a wide range of ambient temperatures (-40^{o}C to 33^{o}C)
Be reliable

Based on the requirements, a heat recovery system was designed and installed. The system had three major sections: the heat source section (Figure 2A), the heat sink or load section (Figure 2B), and the pipe and control section (Figure 2C). The source section had a heat exchanger and auxiliary components to capture heat from engine exhaust. The heat sink section had a unit heater and auxiliary components to dissipate heat and was design to work in such a way that the source or load had to be controlled at specific temperatures at different engine loads. The pipe and control section was used to transport working fluid between the heat source and the heat sink and able to control the flow rate and pressure distributions of the system.

Based on the computation results [13] and the availability of commercial products for high temperature exhaust, a shell and tube type of heat exchanger was selected. The exhaust gas passed through the shell side and the liquid coolant through the core side. For maintenance purposes, the core was removable for cleaning when necessary. In order to have the heat exchanger be able to absorb the most amount of heat, the designed size of the heat exchanger was based on the exhaust condition at full engine load. For partial engine load conditions, the heat recovery system was designed with parameters adjustable to make the coolant inlet and outlet temperatures match the requirements of different space heating applications. The coolant side temperatures used for design (Table II) were selected based on the feed water temperature of 65 ^{o}C typical of boilers used for baseboard heating. So a temperature of 77 ^{o}C would be adequate for pre-heating the boiler feed water. The inlet gas temperature used for design was the full load exhaust temperature and the outlet gas temperature was based on the condensation formation temperatures [14] of the corrosive components of the exhaust and also based on the trade off between the cost and the amount of heat to be recovered. Some of the details of the selected heat exchanger are shown in Table III.

Figure 2A Heat Exchanger Section (On top of container)

Figure 2B Unit Heater Section (Outside of west sidewall of ISO Container)

The heat dissipating capacity of the selected unit heater was above the expected maximum amount of heat absorption rate of the heat exchanger (i.e. 290,000 Btu/hr. for this system) with a safety margin. A bypass loop and a 3 way temperature control valve were selected for the control of the outlet temperature of the load section by appropriately distributing the total flow between the bypass loop and the unit heater.

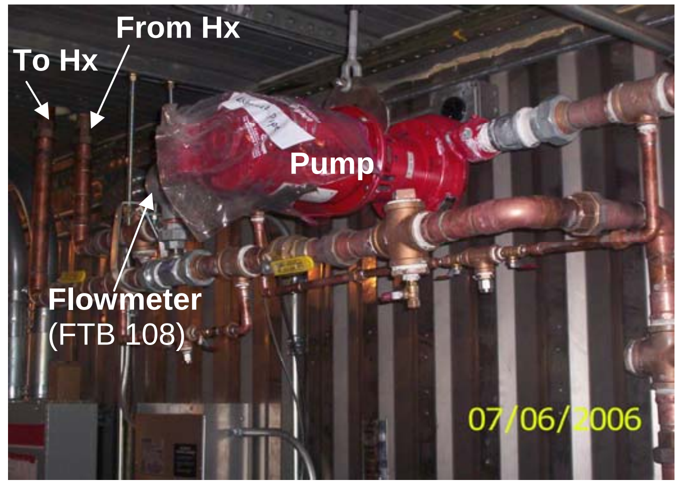

Figure 2C-1 Pipe and Control Elements (South half – inside of west sidewall)

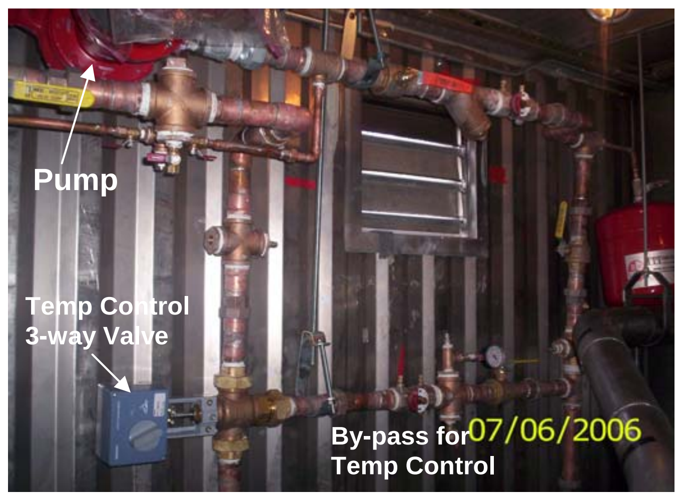

Figure 2C-2 Pipe and Control Elements (North half – inside of west sidewall)

The pipe and control section included the pipes between the heat exchanger section and the load section, a pump and other miscellaneous components (i.e. ball valves, expansion tank, circuit setters, snubbers, dielectric unions, fittings, insulation material, etc.) were designed to control the system operation and to meet the needs for maintenance, instrumentation, corrosion prevention, etc. The pump was selected based on the design flow rate and calculated total pressure drop [15] of the system with the effects of all the expected system components.

TABLE II
INLET/OUTLET TEMPERATURES OF HEAT EXCHANGER

	Inlet temperature	**Outlet temperature**
Exhaust	540 ^{0}**C**	177 ^{0}**C**
Coolant	77 ^{0}**C**	87 ^{0}**C**

TABLE III
HEAT EXCHANGER PARAMETERS

	Quotation 1
Shell side	Gas
Tube side	Liquid
Gas pressure drop	0.31 PSI
Maintenance	Removable core
Heat transfer area	87 ft^2
Size	51”x28”x28”
Weight	725 lbs
Material of construction	Tube- SS fin tube Shell- SS inner wall
Insulation	Integrated insulation

III. INSTRUMENTATION AND TEST PROCEDURE

A. Instrumentation

The purpose of instrumentation was for system operation monitoring and system data collection. Parameters to be monitored and collected included temperatures, pressures, flow rate, and humidity. Sensors installed to the heat recovery system included temperature sensors, pressure gauges, and a flow meter. A National Instrument data acquisition system (DAQ) was used to document the experimental readings. Temperature sensor signals were sent to a SCXI 1102 board and the flow meter signal was sent to a SCXI1120 board of the DAQ system. Temperature data were used to estimate the system and components efficiencies, possibility of condensation formation in exhaust pipe and heat exchanger, and fouling factors. Pressure data were used to monitor and evaluate the condition of the flow path and the functions of the components. The pressure data were occasionally checked and recorded manually, but not recorded automatically. The locations of the temperature sensors and the coolant flow meter along the pipe system are shown in Figure 3. There were also 5 thermal couples installed at the heat exchanger exhaust outlet to measure the instantaneous average temperatures of exhaust and 10 thermal couples installed on the outside walls of the heat exchanger tubes to monitor the temperature drop across the heat exchanger. The exhaust temperature at the inlet of the heat exchanger was obtained from a sensor came with the original engine generator set.

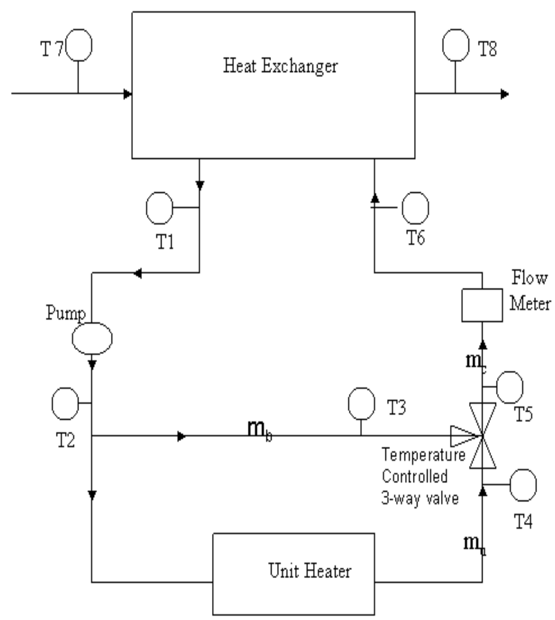

Figure 3 Locations of Temperature Sensors and Flow Meter

In addition to the measurements mentioned above, engine inlet air flow rate and property (i.e. temperature, pressure, humidity) and engine fuel consumption rate were also measured and used for exhaust flow rate calculations. Details can be found in [16, 17]. Sensors for mass flow rates of fuel consumption and coolant flow were calibrated gravimetrically.

B. Test Procedure

The heat recovery system was operated and monitored for nearly 350 hours after 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 heat recovery system was tested for three different types of space heating applications. For each type of application, the inlet or outlet coolant temperature of the heat exchanger was controlled to the practicing temperature corresponding to the application.

During the first 150 hours of test, water was used as coolant and the purpose was to investigate the performance of the system and components and the effect of the exhaust heat recovery system on engine performance. Several defects were found (i.e. such as leakages and inappropriate calibrations of some of the sensors) and fixed. At the 150th hour, the coolant was changed from water to 40% propylene and 60% water mix to avoid freezing and the flow meter was recalibrated.

After the 250th hour, one of the thermocouples was recalibrated and the time constant of the temperature control valve was adjusted to limit the fluctuation in the temperature readings. During the period between 250th hour and 300th hour, the engine was operated under full load. The data obtained for this period were used to investigate the system performance. During the period between 300th hour and 350th hour, the engine was operated under 4 different engine loads (i.e. 25%, 50%, 75%, and 100% of rated load). The data obtained during this period were used to investigate the effect of the load on the performance of the heat recovery system. This paper reports the results of system performance and feasibility and economic analyses based on the data for conventional based board heating application with engine at full load.

IV. RESULTS AND DISCUSSIONS

This section discusses the measured results, system performance, and feasibility and economical analysis results. Feasibility analysis includes the effect of the heat recovery system on engine performance, system efficiency and reliability issues such as corrosion and soot accumulation.

A. System Performance

Based on the first 100 hours test data, no engine performance was found significantly altered. Some of the data is listed in Table IV. Performance data of the exhaust heat recovery system obtained during the period between 250-hour and 300-hour is given in Table V.

The performance data showed that the ambient temperature has a noticeable effect on the flow distribution of the system between the bypass and the unit heater (Figure 4) as expected. The warmer the outdoor ambient temperature, the more coolant needed to pass through the unit heater to keep the heat load (i.e. the sum of heat dissipated from the bypass pipe and the unit heater) nearly constant.

TABLE IV
ENGINE PERFORMANCE DATA AT 50th HOUR AND 100th HOUR

Engine hours	Exhaust T (Degree C)	Turbocharger T (Degree C)	Fuel consum-ption (L/hr.)
Before the installation	542	122	34
50th hour	540	142	34
100th hour	533	140	34

TABLE V
PERFORMANCE DATA OF THE EXHAUST HEAT RECOVERY SYSTY DURING THE PERION BETWEEN 250-HR AND 300-HR

Measured Data	10Hr	20Hr	30Hr	40Hr	50Hr
Exhaust Flow (Kg/s)	0.25	0.24	0.24	0.24	0.25
Exchanger exh inlet T (C)	511.5	503.0	487.0	495.9	517.1
Exchanger exh outlet T C)	215.6	220.6	207.9	214.5	226.6
Coolant flow rate (Kg/s)	1.45	1.48	1.45	1.45	1.43
Exchanger coolant inlet T (C)	76.89	76.53	77.07	77.03	75.82
Exchanger coolant outlet T (C)	88.45	87.05	87.74	87.53	87.34
Before bypass (load) T (C)	88.02	86.79	87.36	87.00	86.83
After 3way valve (load) T (C)	76.91	76.94	77.08	76.75	75.52
Flow rate in bypass (Kg/s)	1.00	1.13	1.02	1.00	0.83
Flow across unit heater (Kg/s)	0.45	0.35	0.43	0.45	0.60
Outside ambient T (C)	-23.0	-30.0	-18.0	-15.0	-7.0

The energy balance was checked between the heat absorbed from the exhaust side and heat dissipated from the load side. Figure 5 shows that the heat absorbed by coolant equals the sum of heat dissipated with an error of less than 3%. It also shows that the heat dissipated from the unit heater was higher for higher temperatures. This can be explained as, at higher ambient temperatures, the pipeline dissipates less heat which needed to be compensated by increasing heat loss through the unit heater to maintain the inlet temperature of the heat exchanger.

Figure 6 shows the heat release rate from the exhaust and the heat absorption rate of the coolant. The heat release rate of the exhaust across the heat exchanger was based on enthalpy change. The enthalpy was calculated based on the individual mass percentages of the exhaust components after combustion. The heat absorption rate was evaluated using the coolant mass flow rate, specific heat and the temperature difference between the heat exchanger inlet and outlet flow.

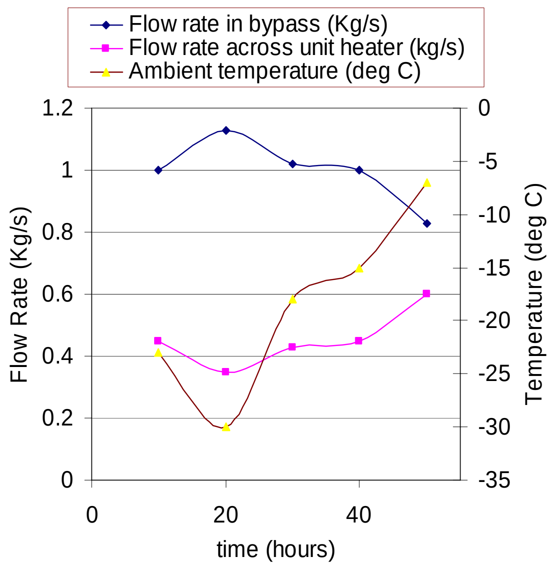

Figure 4 Flow Distribution between Bypass and Unit Heater

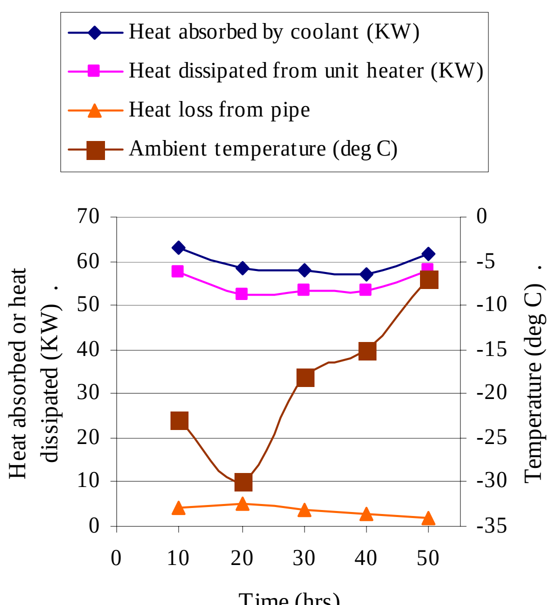

Figure 5 Energy Balance with respect to the Ambient Temperature

The amount of heat absorbed by the coolant followed the same trend as the heat released from exhaust across the heat exchanger.

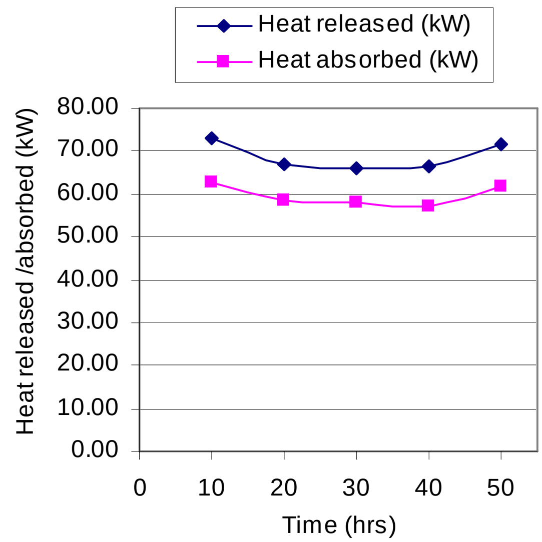

Figure 6 Heat Release vs Heat Absorption with respect to Time

B. Observation of Soot Accumulation and Corrosion

Both soot accumulation and corrosion were considered critical parameters, as they were known to have negative effect on the performance and maintenance of heat recovery systems. This section presents the results obtained from the observation of soot accumulation and corrosion inside the heat exchanger after 350 hours of operation of the exhaust heat recovery system.

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 and cleaned and about 150 grams of soot was found to be accumulated, which is far less than the maximum of 6000 grams. Figure 7 shows the soot accumulated on a arbitrarily selected finned tube. This result matches a general believe that soot accumulated on a finned tubes may approach to its asymptote within a relatively short period of time [18]. With this small amount of soot accumulated, the heat exchanger needs maintenance not more than twice a year, which is considered no impact to the maintenance frequency of the diesel generator set.

When the heat exchanger was dismantled, the corrosion effect of exhaust gas on the heat exchanger was also investigated by examining the surface for the existence of corrosion spots on both tube and shell sides. No trace of any such corrosion was observed. The reason seems to be that the exhaust temperature was always kept above the dew point. This resulted in an absence of condensate in the heat exchanger throughout the run time. The material, SS 316L, used to construct the inside of the heat exchanger may be the other reason.

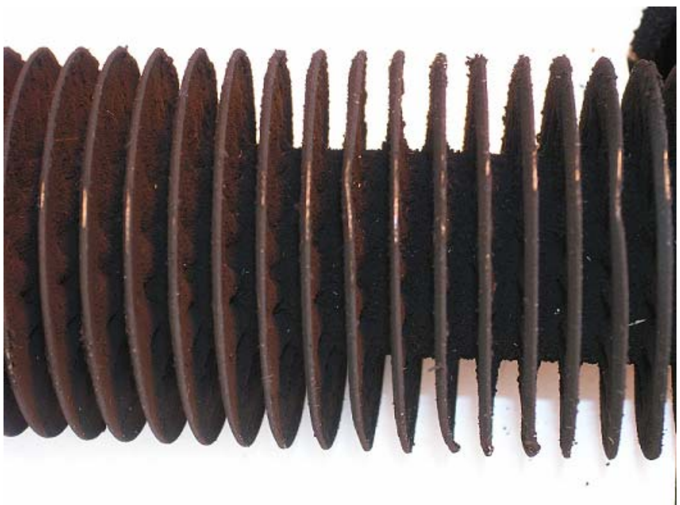

Figure 7 Soot Accumulations on the Fins

C. *Economic Analysis and Feasibility*

The economic analysis was done based on the present heat recovery system and diesel generator used for this experiment and an assumption of 100% use of the recovered heat. The heat recovery rate was taken to be 60 kW at rated load with 8 hours per day. Other parameters used in this calculation include:

Initial cost of the recovery system = $30,000.
Installation cost = $ 5,000 ($75/hr x 8 hrs /day x 10 days).
Airfare, lodging, meals = $1,950 ($600/round trip + $90/day x 15 days).
Total capital cost = $37,950.

Heat recovery per hour = 204,728 BTU.
Heating value for conventional fuel = 130,000 BTU/gal.

Pay back time [19] was calculated based on two different interest rates and two different fuel prices. Figure 8 shows the calculated payback times with respective interest rates and fuel prices. It is noticed that a reasonable payback time is attainable with an interest rate of 10%and a fuel price of above $2 per gallon.

The heat recovery system maintenance cost was based on one day of labor ($75/hr) and a round trip flight ticket ($600) which comes to $1,200 for 6 months of engine operation. The maintenance costs also include additional money ($300) every year for supplies.

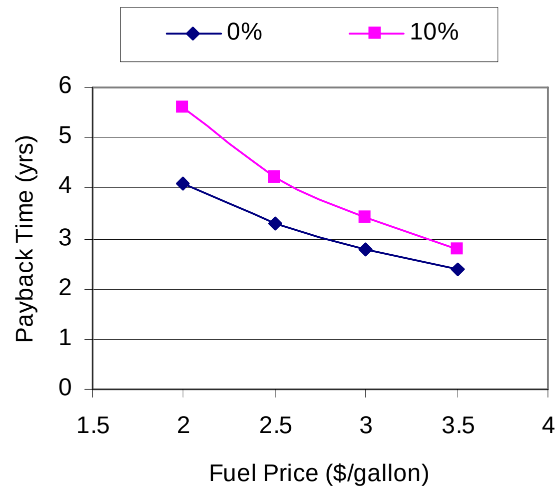

Figure 8 Payback Time with Respect to Fuel Price and Interest Rate

Based on results described in the previous sections, applying exhaust heat recovery system to a diesel generator is expected to cause no significant impact on performance and maintenance frequency of the engine generator system and the payback time for a 100% use of recovered heat would be about 3 years based on the current fuel price.

V. Conclusions

An experimental work was conducted to check the feasibility and economic effect of applying exhaust heat recovery to a diesel engine generator system. The collected data and analysis results lead to the following conclusions:

1. The performance of our exhaust heat exchanger was reliable and consistent.
2. For the 125 kW diesel generator used in this experiment, the rate of heat recovered from the exhaust was about 60 kW.
3. No effects were observed on the engine performance and maintenance frequency due to the heat recovery system.
4. According to the soot accumulation data obtained

from this experiment, the estimated time for heat exchanger maintenance is less than two days per year.

5. Corrosion was not observed to be a problem in the laboratory test of 350 hours.
6. Based on experimental data obtained from this experiment, the estimated payback time for a 100% use of recovered heat would be about 3 years for a fuel price of $2.5 per gallon. For 80% use of the recovered heat, the payback time would be 4 years.
7. Operation cost is largely case dependent. Influential parameters would include diesel fuel cost, the application of the recovered heat, location of the power plant, etc.
8. Performance and economic outcomes will be different from one case to another. However, analysis is recommended before the installation of an exhaust heat recovery system to a village generator set.

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