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Analysis on Combined Heat and Power, and Combined Heat and Power Hybrid Systems for Unconventional Drilling Operations

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Abstract

Analysis on Combined Heat and Power, and Combined Heat and Power Hybrid Systems for Unconventional Drilling Operations

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The United States (U.S.) has experienced a natural gas "boom" due to the development of unconventional shale plays, but well development is energy intensive. Operations use electric drilling rigs typically powered by either three high-horsepower diesel engines (HHPDE) or three dedicated natural gas engines (DNGE) and associated generators. From a first law analysis, HHPDEs peak at 42% efficiency at full load, while DNGE peak at about 30%. Most of the fuel energy is lost as heat rejected by the exhaust and radiators. Concurrently, during cold seasons rigs utilize boilers to provide steam throughout the rig to prevent freezing and provide comfort. The U.S. Department of Energy (DOE) and the National Energy Technology Laboratory (NETL) recently granted West Virginia University (WVU) funding, under agreement DE-FE0024297, to "develop and validate new knowledge and technology to improve recovery efficiency and minimize environmental implications of unconventional resource development." As part of the funding, WVU was tasked with auditing the energy consumption during the horizontal drilling of an unconventional well, processing the data, and assessing methods to reduce fossil fuel consumption and associated emissions during the development phase.

My research analysis focused on a combined heat power (CHP) approach as a means to improve the utilization factor (UF) of fossil energy consumed during development. Engine activity, boiler fuel consumption, and exhaust gas temperatures were recorded during winter drilling of an entire well in the Marcellus shale. Four characteristic activity cycles were extracted from recorded engine and boiler activity to represent four energy consumption scenarios. Exhaust and jacket water heat exchangers (E-HEX, JW-HEX) were designed and simulated, and results were analyzed in 0-D models for the four case scenarios. A 584-kWh hybrid energy management system (HEMS) was also designed and simulated into the model as another method to reduce fossil fuel energy consumption during well development.

HHPDE UF improved on average from 35.7% to 55.7% if only E-HEXs were used and improved to 72.7% if JW-HEXs were also used. DNGEs were less efficient than HHPDE; therefore, more waste heat was available and at a higher quality (temperature). DNGE average UF increased from 19.0% to 34.9% using E-HEX only. HEMS utilization improved UF up to an average of 76.9% and 39.1% for HHPDE CHP and DNGE CHP systems, respectively

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List of Acronyms

Acronym	Word or Phrase
°C	Degrees Celsius
A	Surface Area
AFR	Air to Fuel Ratio
ASME	American Society of Mechanical Engineers
Bhp	Boiler Horsepower
bhp	Brake Horsepower
BSFC	Brake Specific Fuel Consumption
BTU	British Thermal Unit
CAFEE	Center for Alternative Fuels, Engines, and Emissions
CAN	Communication Area Network
CARB	California Air Resources Board
CAT	Caterpillar®
CH_4	Methane
CHP	Combined Heat Power
CO_2	Carbon Dioxide
DAQ	Data Acquisition
DNGE	Dedicated Natural Gas Engine
DOE	Department of Energy
DPF	Diesel Particulate Filter
ECU	Engine Control Unit
EF	Evaluation Function
EIA	Energy Information Administration
EPA	Environmental Protection Agency
ESG	Exhaust Steam Generator
Ft^3	Cubic Feet
FTIR	Fourier-Transform Infrared Spectroscopy
g	Grams
gal	Gallons
GE	General Electric
genset	Electric Generator Set
GHG	Greenhouse Gas
Gph	Gallons per Hour
h	Specific Enthalpy
H_2O	Water
HEMS	Hybrid Energy Management System
HEX	Heat Exchanger
HHPDE	High Horsepower Diesel Engine
HP	Horsepower
Hr	Hour
HVAC	Heating Ventilating and Air Conditioning
Hz	Hertz
ID	Identification
ISO	International Organization for Standardization
kPa	Kilopascal

kW	KiloWatt
kWe	Electric KiloWatt
Kwh	Kilowatt per Hour
L	Liters
LA	Louisiana
Lbs	Pounds
LHV	Lower Heating Value
LMTD	Logarithmic Mean Temperature Difference
m^3	Cubic Meter
MBH	Million BTU per Hour
Min	Minute
MJ	Mega Joule
MW	Mega Watt
N_2	Dinitrogen
NETL	National Energy Technology Laboratory
NM	New Mexico
NO_x	Oxides of Nitrogen
NTU	Number of Transfer Units
NY	New York
O_2	Oxygen
OH	Ohio
ORC	Organic Rankine Cycle
PA	Pennsylvania
PGN	Parameter Group Number
PM	Particulate Matter
ppm	Parts Per Million
Psi	Pounds per Square Inch
Psig	Pounds per Square Inch Gauge
rpm	Revolutions Per Minute
SAE	International Society of Automotive Engineers
SCF	Standard Cubic Feet
SI	International System
Sm^3	Standard Cubic Meter
SOC	State of Charge
SPN	Suspect Parameter Number
TE	Thermoelectric
TEG	Thermoelectric Generator
TEMA	Tubular Exchanger Manufacturer Association
TX	Texas
U	Heat Transfer Coefficient
UF	Utilization Factor
ULSD	Ultra-Low-Sulfur Diesel
US	United States
V	Volts
VDF	Vector Database File
WHR	Waste Hear Recovery

WJ	Water Jacket
WV	West Virginia
WVU	West Virginia University

1. Introduction

The United States (U.S.) has experienced a natural gas "boom" over the past few decades. Natural gas gross withdrawals have increased 52% since 2010 from 27 million to 41 million cubic feet per year [1]. Projections indicate that natural gas consumption will keep growing as the energy industry phases towards green energy. Higher withdrawals are possible due to the implementation of new techniques such as unconventional drilling and hydraulic fracturing technologies that enable cost-effective extraction from shale gas reserves, but well development is energy intensive. Operations use electric drilling rigs typically powered by either three high horsepower diesel engines (HHPDE) or three dedicated natural gas engines (DNGE) and associated generators. From a first law analysis, HHPDEs peak at about 42% efficient at full load, while DNGEs peak at about 30%. Most fuel energy is lost as heat rejected by the exhaust and radiators. Concurrently, during cold seasons rigs utilize boilers to provide steam throughout the rig to prevent freezing and provide comfort. Strategies are being studied to reduce associated fuel costs and emissions [2].

Government agencies, including the U.S. Environmental Protection Agency (EPA), the California Air Resources Board (CARB), and the U.S. Department of Energy (DOE), are tasked with determining new techniques to reduce greenhouse gas (GHG) engine emissions. The U.S. DOE and the National Energy Technology Laboratory (NETL) recently granted West Virginia University (WVU) funding, under agreement DE-FE0024297, to "develop and validate new knowledge and technology to improve efficiency and minimize environmental implications of unconventional resource development." As part of the funding, WVU was tasked with auditing the energy consumption during the horizontal drilling of an unconventional well, processing the data, and assessing methods to reduce fossil fuel consumption and associated emissions during the development phase.

Combined heat and power (CHP) models allow researchers to quantify and estimate possible waste energy recovery from the same engines as those that operate in the field. The engines are attached to electric generators that power drilling rigs. The engines and generators together are also known as gensets. Evaluating the true impact of the gensets deployed in the field can be accomplished using on-site genset activity data. In-field data were collected from the gensets and boiler during drilling of an unconventional well in the Marcellus shale play, so that representative fuel and energy-saving systems could be modeled. The energy-saving techniques examined with the

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models were CHP and a hybrid energy management system (HEMS). Exhaust heat exchangers (E-HEX) and jacket water heat exchangers (JW-HEX) were designed, modeled, and simulated in the CHEMCAD® software environment, while the HEMS was modeled to meet 584-kW-hr battery technical specifications [2]. Four characteristic cycles were extracted from in-use genset activity to represent possible scenarios during the unconventional drilling. Cycles and energy recovery system models were combined into a SIMULINK software environment for system performance simulations.

Results were calculated in terms of energy utilization factor (UF). The UF was defined as the amount of work and heat produced by the gensets to the fuel's energy presented in Equation 1. Recovering enough heat from the gensets while powering the rig would allow operations to reduce the boiler utilization or eliminate it.

Equation 1. Energy Utilization Factor (UF)

$$UF = \frac{Genset\ Work\ (kJ) + Genset\ Heat(kJ)}{Fuel's\ Energy\ (kJ)} \tag{1}$$

Research Hypothesis: Energy recovery methods deployed in other stationary or mobile applications (CHP and HEMS) could be deployed at unconventional well sites to reduce energy and fossil fuel consumption through possible elimination of boiler systems and overall improvement in energy UF.

To assess this hypothesis, I conducted the following main research tasks:

1) Conducted a literature review on well development and energy-saving technologies.

2) Developed a data acquisition system necessary to characterize the engine and boiler activity during unconventional well development.

      a) Integration of engine ECU data within SCIMITAR Software

      b) Deployed and monitored the system in the field.

      c) Collected and analyzed the activity data.

3) Created multiple activity cycles to assess energy-saving technologies through reduced-order models.

4) Used commercially available modeling software to design and evaluate various heat exchangers.

5) Developed 0-D models to assess potential energy savings – fuel, costs, emissions.

2. Review of Literature

2.1 Unconventional Well Drilling

Estimated natural gas reserves rose to 2074 trillion cubic feet in 2008, from 1532 trillion cubic feet in 2006 [3]. Proved natural gas reserves rose from 213 trillion cubic feet in 2005 to 505 trillion cubic feet in 2018 [4]. Shale gas represented 68% of proved natural gas reserves in the U.S. in 2018 [5]. Estimations based on recoverable resources of dry natural gas in the U.S., assuming the production's annual rate to be similar to the production in 2018, show that the U.S. has enough natural gas to last about 92 years [6]. The U.S. has been importing natural gas to supply the country's demand since 1957; however, due to the growth in known reserves and extraction capabilities, production has overpassed local demand. The U.S. started exporting natural gas in 2017 [7]. Natural gas is found in shale plays formations located in about 30 states: Texas (TX) and Pennsylvania (PA) being the most relevant natural gas producers. Shale plays are areas in which pockets of natural gas are available deep below the surface in tight shale formations. Figure 1 shows a map of the lower 48 major shale plays in the U.S., including Marcellus, Bakken, Barnett, Fayetteville, Eagle Ford, Haynesville, and Utica.
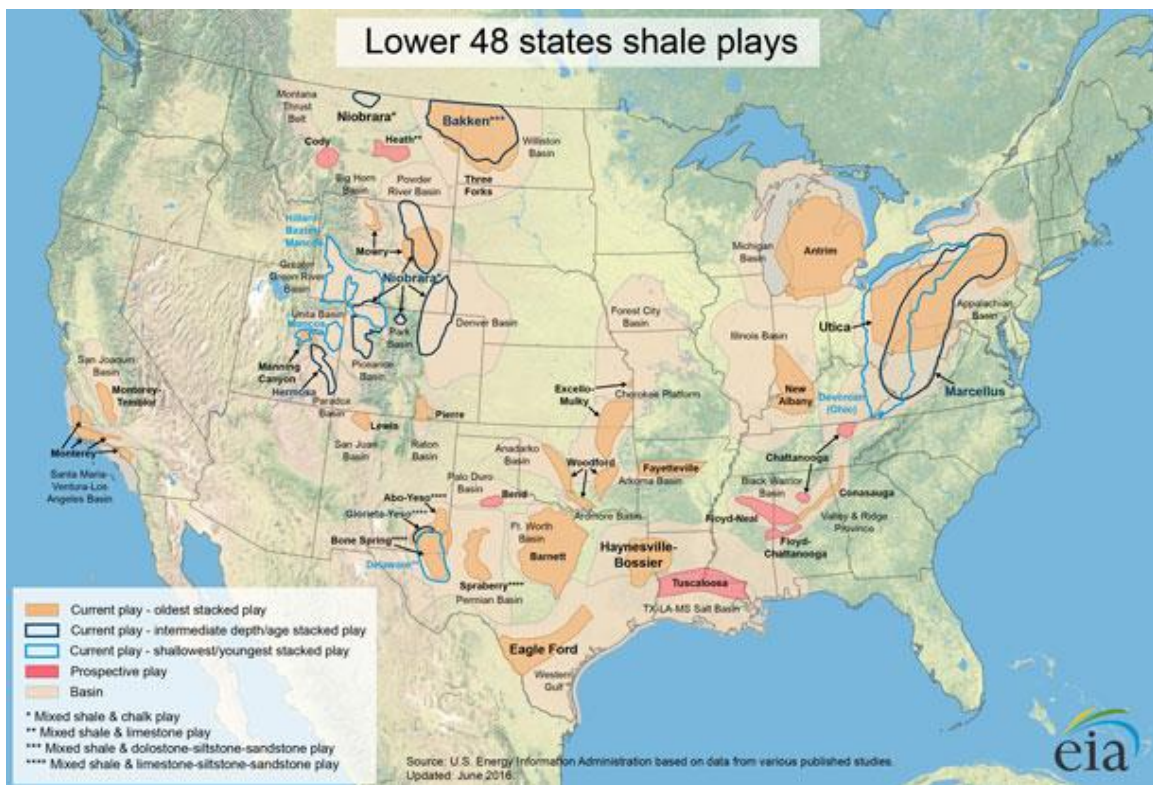
Figure 1. Major U.S. Shale Plays [7]

In 2019, 23.9% of the natural gas production was obtained from TX, while 20% was obtained from PA [7]. The Marcellus shale (PA, WV, OH, and NY) is the most productive play, followed by Permian (TX and NM), Utica (OH, PA, and WV), and Haynesville (LA and TX). Figure 2 presents productions for the major gas shale plays.
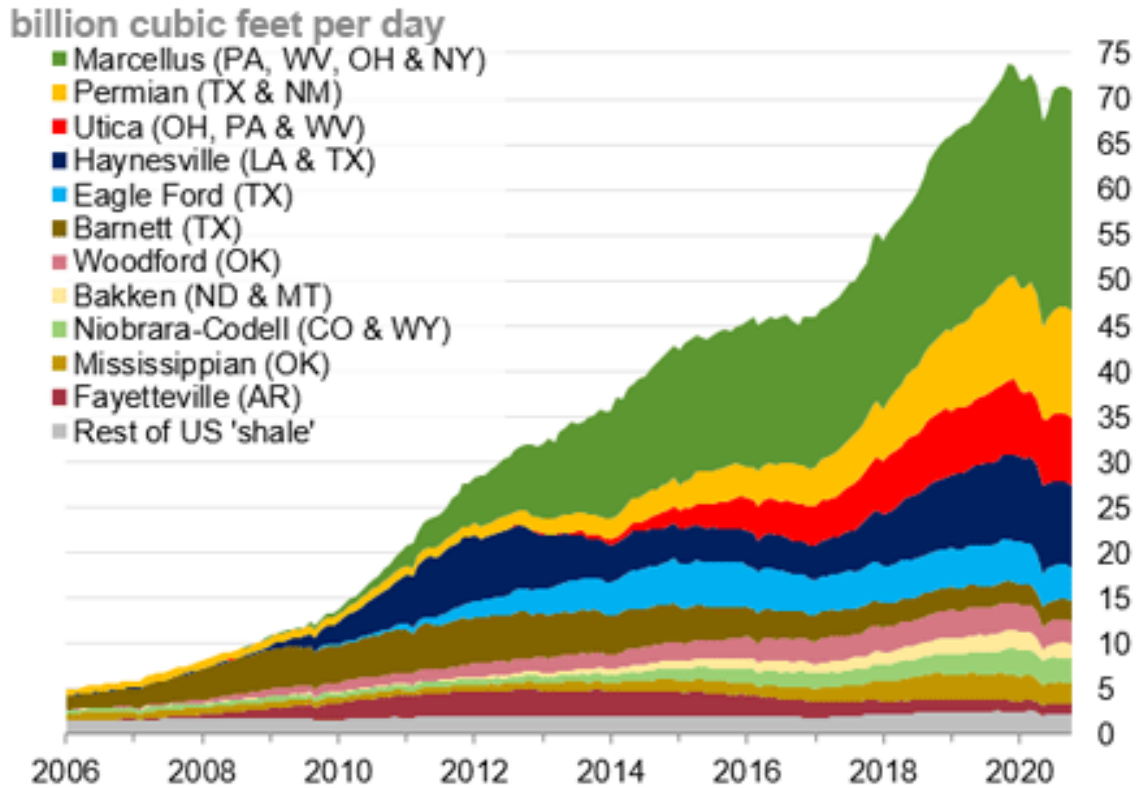
Figure 2. U.S. Dry Shale Gas Production [7]

Conventional (vertical) drilling on shale gas plays proved to have limited effectiveness due to the limited access to the reservoir. Directional drilling provided the opportunity to drill across the play (laterally), enabling access to a larger percentage of the product by drilling longer wells. Natural gas is highly dispersed in rock formations, and because shales have insufficient permeability to allow fluid to flow to a wellbore, hydraulic fracturing must be used. Hydraulic fracturing is a stimulation technique where water, sand, and chemicals are pumped under high pressure into the bedrock formation via the well. This process creates new fractures and extends and connects existing ones, allowing the gas to flow. A typical directional well has around 16 hydraulic fracturing stages [8].

2.2 Energy Consumption During Well Development

Well development is an energy-intensive process. There are multiple stages in unconventional well development that have high energy demands. This energy is provided by HHPDEs or DNGEs depending on natural gas availability and drilling companies' equipment. There are three different common configurations for powering rigs: HHPDE, DNGE, and dual-fuel-kit-adapted diesel gensets. The industry is dominated by either HHPDE or DNGE due to their cost performance. Dual-fuel kits represent a high upfront expense that is not cost-effective given current diesel prices. Diesel gensets are reliable at any site regardless of natural gas availability.

Unconventional wells consist of two stages of drilling: vertical and horizontal. The total length drilled can vary depending on the region in which the well is located and the shale gas's depth. Barnett region wells generally require up to 8000 feet of vertical drilling, but Haynesville region wells can require up to 13,500 feet of vertical drilling [9]. Average lateral lengths have grown from about 2000 feet in 2005 to about 6000 feet in 2015, at a near linear rate, with an average length of about 4000 feet over the entire sample period [10]. Drilling longer horizontals allow the potential for more natural gas to be extracted from a single well. However, horizontal drilling requires more energy and time. This increases the costs of unconventional well development compared to a conventional well. Most current drilling rigs are electric and use three stationary engines coupled to electric generators to produce on-site electricity. The engines and generators together are known as gensets. On average, drilling rigs utilize 2.15 operating gensets, with 1030 kW (1381 HP) per genset. These gensets are estimated to operate 62.6 hours per 1000 feet drilled at an average load of 48.5% [11]. A model developed by the WVU Center of Alternative Fuels, Engines, and Emissions (CAFEE) estimated that, on average, drilling gensets fueled by diesel consumed 6.06×10^6 MJ per well [12]. Using the same model, it was estimated that the amount of energy needed for drilling a well using natural gas was 1.08×10^7 MJ [12].

The three gensets combined working in a closed loop are called the on-site power plant. The power plant operates using its own control system, independent from the control system of the rig. The rig control system uses engine load data from the gensets to dynamically limit the load to prevent overload requests. However, the power plant does not receive a demand signal from the rig control system. Rather, the gensets operate on a closed-loop system that uses voltage and frequency to recognize adequate load to meet the power demand. Since there is no stored energy, the gensets must increase or decrease load in an attempt to match the amount of electrical power that the rig

requires. When the system is balanced at 60 Hertz (Hz) and 600 Volts (V), power production matches rig demand [13]. The total power demand is evenly divided across all online gensets (generally three). Having an excess number of gensets online to be able to handle potential spikes in power demand results in a lower average load on each engine. Therefore, they may operate at lower efficiency and produce higher levels of emissions. Gensets may remain online and running even when fewer gensets could meet power demand because each genset takes several minutes to start up. The gensets need time to warm up and synchronize to the communication area network (CAN) bus. Genset warmup time is a critical consideration in automating engine start and stop decisions while avoiding power limits that interfere with rig operations [13].

2.3 Non-road Land Drilling Engines Emissions

In-use engines vary in age, power, activity, and emissions compliance certification level. Numerous studies have suggested that emissions from unconventional resource development can affect local and regional air quality [14-22]. WVU's model estimated emissions from in-use data. Diesel powered land drilling engines are the primary sources of carbon dioxide (CO_2), carbon monoxide (CO), and methane (CH_4) emissions throughout the unconventional natural gas well development process. If land drilling engines are natural gas-powered, model estimation implies they are the primary CO_2 emitting source compared to the other prime movers (fracking pumps and on-road trucks) [12].

The EPA regulates non-road engine exhaust emissions categorized by rated power and ignition system. In general, compression-ignition drilling engines fall within the highest power rating (kW>900). There have been three tiers of emissions compliance since 2000. Tier 1 (2000-2005), Tier 2 (2006-2010), and Tier 4 (2011-Present) [23]. Large-spark-ignition drilling engines are categorized into two tiers, Tier 1 (2004-2006) and Tier 2 (2007-Present) [24]. Tier 2 compression ignition and Tier 2 spark-ignited engines dominate the unconventional natural gas well industry, and compression ignition engines are slowly shifting towards Tier 4 engines. Tables 1 and 2 present the exhaust emissions standards for compression ignition and spark ignited large bore engine respectively.

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Table 1. Nonroad Compression-Ignition Engines: Exhaust Emissions Standards [23]

Rated Power	Tier	Model Year	NMHC (g/kW-hr)	NMHC + NOx (g/kW-hr)	NOx (g/kW-hr)	PM (g/kW-hr)	CO (g/kW-hr)
kW>900	1	2000-2005	1.3	-	9.2	0.54	11.4
	2	2006-2010	-	6.4	-	0.20	3.5
	4	2011-2014	0.40	-	3.5	0.10	3.5
		2015+	0.19	-	3.5	0.04	3.5

Table 2. Nonroad Large Spark-Ignition Engines: Exhaust Emissions Standards [24]

Tier	Model Year	General Duty-Cycle Standards		Alternative Standards for Severe-Duty Engines		Field Testing Standards	
		HC+NOx (g/kW-hr)	CO (g/kW-hr)	HC+NOx (g/kW-hr)	CO (g/kW-hr)	HC+NOx (g/kW-hr)	CO (g/kW-hr)
1	2004-2006	4.0	50.0	4.0	130.0	-	-
2	2007+	2.7	4.4	2.7	130.0	3.8	6.5

Gensets operate more efficiently when running at higher loads. Exhaust gas emissions are also lower relative to power at higher loads [13]. The industry is attempting to optimize rig and power plant communication, so engines run at the most efficient point, reducing both fuel consumption and regulated emissions.

2.4 Waste Heat Recovery

The growing energy demand has led to increases in GHG production. With the continuous tightening of emissions regulations, GHGs and wasted energy are being curbed but with an economic disadvantage. Multiple industries face losing their competitive edge due to the restrictions currently in place. For the industry to reduce both reliance on fossil fuels and environmental impact, there are two basic options: using renewable energy systems or reducing overall energy consumption. Incorporating renewable energy technologies is an increasingly attractive option as prices fall. However, renewable energy technologies are not suitable for all locations, all industries, and investment costs can still be prohibitive. The alternative, reducing energy demand, can be divided into three further options: reducing total activity, better energy

management, and recovery of otherwise wasted energy [25]. With developments of thermal and physical waste management technologies, multiple companies are utilizing waste heat technologies to recapture previously lost energy to be used in other applications [26].

The EPA has ambitious targets for reducing GHGs and the effects of global warming. Globally, industrial energy use is responsible for 33% of GHGs, and approximately 70% of the industrial sector's energy demand is for heat. All heating processes result in waste heat, up to 50% in some cases, and heat recovery potential is widely acknowledged [26]. Improving system designs will reduce fossil fuel consumption and emissions, allowing industries to comply with regulations and remain competitive financially.

Internal combustion engine waste heat recovery (WHR) has recently garnered interest due to its potential to contribute to the phasing towards green energy and less fossil fuel dependency. There are many WHR systems either being researched or already in production. Some of the most popular are CHP systems, bottoming cycles, or thermoelectric generators (TEG). A TEG is formed by series of thermoelectric (TE) modules. These modules consist of many TE elements connected in series. TEs are made of semiconductor materials. The temperature difference between two dissimilar semiconductors produces a voltage difference [27]. Hawawasam, et al. simulated an internal combustion exhaust TE generator recovery system. It was found that TEGs can recover up to 16.5% of the exhaust heat [27]. Bottoming cycles involve thermodynamic cycles such as the Rankine or Brayton cycle, which involves heat recovery and rejection via a working fluid to drive a turbine and produce mechanical or electrical energy [28]. Shekh and Saiful, at the University of South Australia, designed an experiment on a 40 kilowatt (kW) diesel generator where they found they could recover up to 10% extra power by using an organic Rankine cycle (ORC) with the exhaust heat recovered [29]. Other studies have shown that utilizing ORC on a heavy-duty diesel engine can recover up to 20% of its exhaust heat, improving the system's efficiency by over 50% [30]. CAIN industries is an exhaust heat recovery systems manufacturer which has sold and installed exhaust steam generators (ESG) in multiple locations and industries using natural gas generators. They have installed an ESG recovering exhaust heat from a Jenbacher JMS-320-900 kW natural gas engine, claiming heat recoveries from exhaust up to over 300 kW, using the recovered heat for heating, ventilating, and air conditioning (HVAC) purposes in a hospital, and decreasing wasted heat [31].

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The unconventional natural gas well development industry usually powers an electric rig with a set of Caterpillar (CAT) 3512C diesel engines, or Waukesha L7044GSI dedicated natural gas engines accompanied by electric generators. CAT 3512C engines peak at about 42% efficiency, rejecting 58% of the fuel's energy as waste heat and other minor losses [32]. Waukesha's L7044GSI engine peak at about 33% efficiency, rejecting 67% of the energy consumed as waste heat [33]. Internal combustion engines have room for waste heat recovery improvement, given their large waste heat rate at the exhaust flue and cooling system (radiators). About 34% of the fuel energy consumed by a CAT 3512C is rejected through the exhaust gas, and an additional 16% is lost to the engine coolant [32]. Approximately 30% of the energy consumed on an L7044GSI engine is rejected through the exhaust flue and about the same amount through the radiator [33].

2.5 Heat Exchangers

Heat exchangers (HEX) are devices used to transfer heat between two or more fluids. A solid wall may separate the fluids to prevent mixing, or they may be in direct contact. They are widely used in heating, refrigeration, air conditioning, power stations, engines, chemical plants, petrochemical plants, petroleum refineries, natural-gas processing, and sewage treatment. The heat transfer effectiveness depends on both the HEX's design and the working fluid's properties. Some critical design parameters include the pitch ratio, tube length, tube number, tube layer, and baffle spacing [34].

During engine run time, there are several locations in the engine's configuration where heat is dissipated to the environment. The engine's highest heat rejection areas are exhaust gas and radiator. This heat may be recovered with a heat exchanger's assistance, although it requires upfront equipment costs, installation, and maintenance. In the past, fossil fuel was cheap, and a WHR system was not cost-efficient. Furthermore, high sulfur concentrations in diesel fuel resulted in corrosive exhaust gas, requiring expensive alloys to manufacture E-HEXs.

The EPA has instituted more stringent off-road engine emissions and diesel fuel quality standards to reduce environmental and health impacts. In 1993, the EPA began regulating diesel fuel sulfur levels. Before any EPA diesel fuel sulfur concentration regulations, diesel fuel contained as much as 5000 ppm of sulfur. In 2006, the EPA began to phase in more stringent regulations to lower the amount of sulfur in diesel fuel to 15 ppm. This fuel is known as ultra-low sulfur diesel (ULSD). From 2007 to 2014, low sulfur diesel fuel (500 ppm) and ULSD were phased in for non-road. After

2014, the EPA required ULSD fuel for all non-road operations [35]. Alongside the implementation of oxidation catalysts and diesel particulate filter (DPF) on Tier 2 and 4 engines to comply with the EPA's non-road emissions standards, diesel-fuel sulfur concentration regulations have reduced sulfur oxides and particulate matter (PM). For non-road stationary diesel engine WHR systems, the reduction in sulfur and PM may allow E-HEX to be more practical. The exhaust will be less corrosive and produce less soot, both of which have prevented economic heat recovery in the past.

Another source of waste heat is the engine cooling system. An engine cooling system relies on a HEX where the working fluids absorb heat from the engine via JW to prevent overheating. That heat is later released into the atmosphere. Capturing JW heat from the engine would increase system performance. CAIN Industries offers pre-engineered standardized HEXs for engine exhaust and JW. Their experiments have shown that their systems recover up to 782.2 kW from a 1700 kW natural gas engine in the form of steam vapor at 150 psig [31]. They also offer ebullient steam generators from engine JW, a JW-HEX which produces low-pressure steam. They can be added to the exhaust heat steam generator to improve overall system efficiency.

Current unconventional drilling rigs do not utilize these WHR systems. Their heat sources are industrial boilers fueled by diesel or natural gas. Boilers operate when weather conditions require large amounts of energy for heating - either seasonally or regionally.

2.6 Boilers
A boiler is a steam generator device that combusts fuel to heat and convert water to steam at variable pressures. Boilers are used in multiple industries such as food processing, power generation, and heat production. The boiler's main components are fuel supply, combustion air system, feedwater system, pressurized heat exchanger, and exhaust gas venting system. There are two basic boiler configurations: firetube and water tube. In firetube boilers, exhaust gases pass inside boiler tubes, and heat is transferred to water between the tubes and the shell. In water tube boilers, water passes through the tubes, and exhaust gasses remain on the shell side. Because tubes can typically withstand higher internal pressure than the large chamber shell in a firetube, water tube boilers are used where high steam pressures [over 3,000 pounds per square inch (psi)] are required [36].

Unconventional natural gas wells being developed at cold weather locations often require a boiler to ensure reliable and efficient heat supply for the crew, drilling, and safety equipment. An

industry standard is the HURST series 300 model, rated at 200 boiler horsepower (Bhp) at a maximum heat output of 6695 million British thermal units per hour (MBH) or 1962 kW [37]. These boilers are generally diesel fuel and/or natural gas-powered, depending on natural gas fuel availability on-site. A HURST 300 Series model 200 Bhp is about 81% efficient. At 100% load, the boiler outputs 6900 pounds per hour (lbs./hr.) of saturated vapor at 110 psi (758 kPa) [37]. Maximum boiler fuel consumption is 8400 cubic feet per hour for natural gas and 60 gallons per hour for diesel fuel.

Once saturated steam exits the boiler, it is directed through a system of insulated hoses connected to multiple-sized heaters in offices, drilling fluid tanks, and other accessory locations. Heaters may vary between 320,000 BTU/hr to 1.7 million BTU/hr (94 kW to 498 kW), and they are operated manually by the crew in the rig. Manual operation of heating sources introduces human inefficiencies into the system. Heaters are not automated to maintain a working temperature; thus, rig crew members determine when to power the heaters on or off. On average, a rig operates between 11 to 15 steam heaters, and it is recommended not to use heaters over one million BTU/hr (293 kW) to keep heat demand low [38]. Steam lines are equipped with steam traps. Steam traps are an essential part of any steam heating system. They help keep steam pressure inside the heater cores, allowing the heater to maximize heat usage. They also allow condensate to return to the pre-heated tank, reducing the amount of water consumed. Steam traps are built-in with a thermostat that opens at high temperatures. When the temperature drops below a specified temperature, it closes, capturing condensate water. This water is then redirected towards a preheated tank, also known as a day tank, usually attached to the boiler as a feedwater recovery system.

2.7 Hybridization

The continuous need for emissions reduction and fuel efficiency improvements has led to multiple research approaches to achieve those goals. With improved energy storage technologies and electrically driven motor equipment, hybrid engines are gaining popularity. In the light vehicle industry, hybrid vehicle sales began in 1999 and have become more popular since. In 2019, the U.S. market share reached 2.4% of the total fleet of light-duty vehicles [39]. Hybridization technology is also penetrating the heavy-duty vehicle industry due to the tight emissions regulations, especially those of the CARB. In order to advance towards low emissions engines, Cummins released PowerDrive in 2018, an advanced hybrid system offering both parallel and series capabilities [40].

North America occupies the most extensive electricity consumption market globally. Large-scale energy projects are growing exponentially. Through December 2016, 1227 electricity energy storage projects had been carried out worldwide, and the total installed capacity reached 1930 MW [41]. The electricity storage techniques being used are led by lithium-ion batteries. This type of battery has a year-on-year growth of approximately 55% [41]. Other energy storage technologies are lead-acid batteries, supercapacitors, sodium-sulfur batteries, lithium iron phosphate, and flow batteries.

On July 1st, 2018, a HEMS was introduced to the land drilling industry for commercial operations [13]. The HEMS used battery energy storage and engine automation to reduce the number of gensets operating, increase the efficiency of each genset, and reduce emissions. The power bank (battery system) is connected to the rig's control system in order to maximize engine operation and reduce fuel consumption. Gensets are automatically turned on or off depending on the rig's power demand and the battery state of charge (SOC). In addition to fuel savings, downtime due to issues with generators is avoided. Furthermore, the battery energy storage system can temporarily provide power when an engine fails or requires maintenance. The system was developed over two years and tested at a rig construction facility before being placed into operation on a rig working in the Marcellus shale [13].

Patterson-UTI Drilling Company (Patterson) is starting to use the ECO-CELL. An energy management system that leverages stored energy to optimize fuel efficiency and reduce emissions. This power bank utilizes lithium-ion batteries due to their cycle life, energy and power ratings, and safety characteristics. The power bank acts as a fourth generator and can provide 1.5 times more power at full load. The power bank stores up to 584 kWh of energy and communicates with the rig's operation system for optimal synchronization [42]. Figure 3 presents an experimental HEMS-equipped rig power plant. Three CAT 3512C gensets, the HEMS, and the diesel fuel tank are the major components of the on-site power plant.
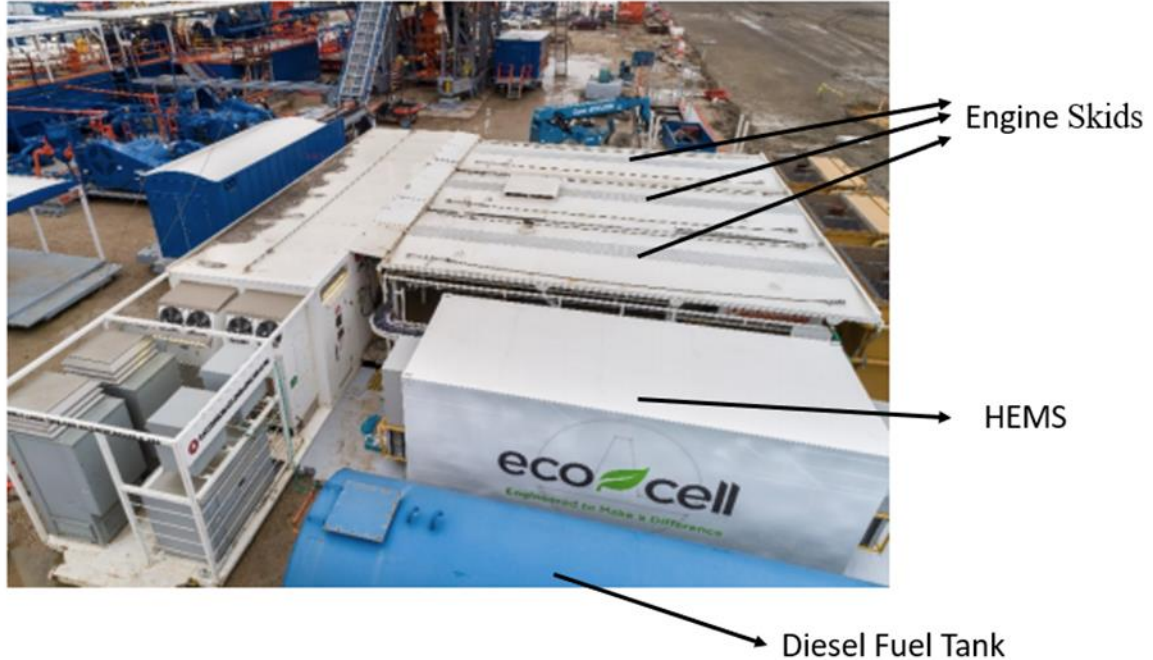
Figure 3. HHPDE-Powered Rig Equipped with a HEMS

For testing purposes, Patterson recorded approximately 600 days of operational data from 7 rigs working in the Marcellus shale over one year. Data analysis established the baseline average number of gensets used in each type of operation. Data recorded from December 2018 was used for normalizing engine activity [13]. During testing, it was found that the HEMS reduced fuel consumption by 13.6% [13]. Total costs savings were $18,284, with an average saving of $659 per day of operation in diesel costs (Based on Standardized $2.17/gal. Fuel Cost [$2.00 plus $0.17 delivery]) [13]. The most savings were detected during tripping and casing operations (transient operations), saving over $11,000 during this operation alone [13]. Similar calculations were performed to estimate emissions reduction. The system resulted in reductions of NO_x by 3%, CO by 31%, and PM by 43% [13]. Also, by reducing active engine time, maintenance intervals were extended, saving expenses and engine availability. Calculations showed that genset usage might be reduced from 1808 hours to 1229 hours with the HEMS in operation versus baseline operation.

Operating a diesel generator set at loads lower than 30% of rated power for extended periods might negatively impact the unit [43]. Diesel gensets are designed to operate between 50 and 85% load [43]. If the engines are operated at low loads for extended periods, the most prevalent consequence

is a phenomenon known as exhaust slobber or wet stack. Running at high idle with little or no load reduces the heat in the cylinder, allowing unburned fuel and oil deposits to leak through the exhaust slip joints [43]. The engine slobber itself will not harm the engine. However, it indicates underload operations that may be accompanied by other underloading effects such as deposit build-up behind piston rings, deposits developing inside the cylinders, and in extreme cases, cylinder liner polishing may occur [43]. During the HEMS testing, the amount of time the gensets operated below a load of 30% was reduced [13]. The total engine operation time under 30% load was 331 hours with the HEMS, compared to 996 hours in the baseline, a 66% reduction [13]. With less time under the 30% load mark, engines will be more efficient, emit fewer pollutants, extend their lifespan, and require less maintenance.

Optimal operation conditions are between 50% to 100% of the rated load for natural gas generator sets. It is recommended not to load the engines below 50% load for any duration in time, and the ideal range is 70% load and above [43]. Natural gas powered engines do not have enough cylinder pressure to maintain oil control in the cylinder at low loads, letting the oil pass the oil rings into the cylinders, leading to ash deposits. These deposits alter the compression ratio, which can diminish the detonation margin. If the detonation margin is reduced sufficiently, unintended detonation may occur. Detonation will decrease the engine's lifespan, harm components, and cause unexpected shutdowns or failures. Like diesel generator sets, the extended operation of gas generator sets at low loads may lead to deposit build-up on the valves, spark plugs, and behind the piston rings [43]. In extreme cases, deposits in the cylinder can develop, causing cylinder liner polishing.

Additionally, many natural gas engines operate at an air to fuel ratio (AFR) below 1 (i.e., rich) at low loads to sustain combustion and guarantee that the engine does not misfire. A rich AFR causes the engine to deviate from the expected emissions levels, potentially leading to non-compliance with required emissions regulations. Also, a rich AFR increases temperatures and can accelerate component wear [43].

3. Data Collection Methods

In order to analyze scenarios that represent the energy utilization of an unconventional drilling rig, it was necessary to collect data from three CAT 3512C engines and a HURST 300 series 200 Bhp in the Marcellus shale region. The gensets and the boilers are the primary fuel consumers during this energy intensive phase. Reducing fuel consumption by recovering waste heat from the engines to assist or eliminate the boiler would reduce operating costs. Other relevant information was obtained from the engine's specification sheets. A dedicated, natural gas powered rig was also modeled and analyzed. Data used for this model was previously obtained by WVU's researchers as part of another DOE funded research program. Data were collected from multiple Waukesha L7044GSI engines during the drilling of wells in two different shale plays. Table 3 presents major engine specifications.Boiler rental expenses were based on a daily fee of $240 and one freight charge fee of $3950 (Appendix F).

Natural gas fuel costs to be recovered with the UFs of 30.3% and 34.9%.

Table 32 shows the return on investment analysis for the DNGE CHP system.

Table 32. DNGE CHP System Return on Investment Analysis

Equipment	DNGE CHP	
	Expenses (+) and Savings (-)	
E-HEX	$408,102	
Boiler Rental	-$108,756	
Natural Gas Costs to Be Saved	-$299,346	
UF	30.3%	34.9%
Time (hrs. of replaced boiler activity)	10,061	8236
Days (replaced boiler activity)	419	343

If normalizing savings per day of replaced boiler activity, the HHPDE CHP system would save approximately $974 per day. If the UF was increased to 34.9%, savings would be estimated at $1190per day.

Table 3. HHPDE And DNGE Engine Parameters [32,33]

Parameter	CAT 3512C	Waukesha L7044GSI
Rated Brake Power	1101 kW	1253 kW
Piston Displacement	52 L	115 L
Exhaust Stack Temperature	397.6 °C	637 °C
Heat Rejected to Exhaust	902 kW	1143 kW
Heat Rejected to Jacket Water	412 kW	1128 kW
Generator Efficiency	95%	95%
Reference LHV	62,780 kJ/kg	35,533 MJ/nm^3

3.1 Data Acquisition System

A data acquisition (DAQ) system containing three computers equipped with WVU CAFEE's DAQ software SCIMITAR was developed. These computers recorded engine activity, exhaust temperatures, and boiler fuel consumption. Figure 4 presents the DAQ system.

Figure 4. Data Acquisition Box

3.2 Engine Activity Data

The engine control unit (ECU) streams a set of public and private channels where multiple engine parameters are broadcasted. Public data generally includes parameters such as engine load, coolant temperature, and fuel consumption. ECU data were obtained via the CAN by connecting the SAEJ1939 engine port from the control panel with a PCAN-USB device to the DAQ box. CAN outputs were indexed using a hexadecimal format. A vector database file (VDF) (.dbc file) was created and adapted to CAT's network IDs to translate data from hexadecimal format to traditional numerical format using CAN-ID-to-PGN public cloud-based files published by CSS Electronics [44]. Figure 5 presents the CDF showing the main engine public channels recorded via CAN communication.

```
 1 BO_ 2901410816 AAAEEC1: 8 Vector__XXX
 2   SG_ SrcAddrssOfCtrllngDvcFrEngCntrl : 40|8@1+ (1,0) [0|255] "" Vector__XXX
 3   SG_ ActlEngPrcntTrqueHighResolution : 4|4@1+ (0.125,0) [0|0.875] "%" Vector__XXX
 4   SG_ EngDemandPercentTorque : 56|8@1+ (1,-125) [-125|125] "%" Vector__XXX
 5   SG_ EngStarterMode : 48|4@1+ (1,0) [0|15] "" Vector__XXX
 6   SG_ EngSpeed : 24|16@1+ (0.125,0) [0|8031.875] "rpm" Vector__XXX        → Engine Speed
 7   SG_ ActualEngPercentTorque : 16|8@1+ (1,-125) [-125|125] "%" Vector__XXX
 8   SG_ DriversDemandEngPercentTorque : 8|8@1+ (1,-125) [-125|125] "%" Vector__XXX
 9   SG_ EngTorqueMode : 0|4@1+ (1,0) [0|15] "" Vector__XXX
10
11 BO_ 2901410560 EEC2: 8 Vector__XXX
12   SG_ SCRThermalManagementActive : 46|2@1+ (1,0) [0|3] "" Vector__XXX
13   SG_ MmntryEngMaxPowerEnableFeedback : 42|2@1+ (1,0) [0|3] "" Vector__XXX
14   SG_ EstPumpingPercentTorque : 56|8@1+ (1,-125) [-125|125] "%" Vector__XXX
15   SG_ DPFThermalManagementActive : 44|2@1+ (1,0) [0|3] "" Vector__XXX     → Engine Load at Current Speed
16   SG_ ActlMxAvailableEngPercentTorque : 48|8@1+ (0.4,0) [0|100] "%" Vector__XXX
17   SG_ AccelPedalPos2 : 32|8@1+ (0.4,0) [0|100] "%" Vector__XXX
18   SG_ VhclAccelerationRateLimitStatus : 40|2@1+ (1,0) [0|3] "" Vector__XXX
19   SG_ EngPercentLoadAtCurrentSpeed : 16|8@1+ (1,0) [0|250] "%" Vector__XXX
20   SG_ AccelPedal2LowIdleSwitch : 6|2@1+ (1,0) [0|3] "" Vector__XXX
21   SG_ RoadSpeedLimitStatus : 4|2@1+ (1,0) [0|3] "" Vector__XXX
22   SG_ AccelPedalkickdownSwitch : 2|2@1+ (1,0) [0|3] "" Vector__XXX
23   SG_ AccelPedal1LowIdleSwitch : 0|2@1+ (1,0) [0|3] "" Vector__XXX
24   SG_ RemoteAccelPedalPos : 24|8@1+ (0.4,0) [0|100] "%" Vector__XXX
25   SG_ AccelPedalPos1 : 8|8@1+ (0.4,0) [0|100] "%" Vector__XXX
26                                                                            → Fuel Consumption
27 BO_ 7398683136 LFE: 8 Vector__XXX
28   SG_ EngAverageFuelEconomy : 32|16@1+ (0.001953125,0) [0|125.5] "km/L" Vector__XXX
29   SG_ EngInstantaneousFuelEconomy : 16|16@1+ (0.001953125,0) [0|125.5] "km/L" Vector__XXX
30   SG_ EngFuelRate : 0|16@1+ (0.05,0) [0|3212.75] "L/h" Vector__XXX
31   SG_ EngThrottleValve2Pos : 56|8@1+ (0.4,0) [0|100] "%" Vector__XXX
32   SG_ EngThrottleValve1Pos : 48|8@1+ (0.4,0) [0|100] "%" Vector__XXX
33                                                                            Coolant Return
34 BO_ 7398682112 ET1: 8 Vector__XXX                                         Temperature
35   SG_ EngFuelTemp1 : 8|8@1+ (1,-40) [-40|210] "deg c" Vector__XXX
36   SG_ EngIntercoolerThermostatOpening : 56|8@1+ (0.4,0) [0|100] "%" Vector__XXX
37   SG_ EngIntercoolerTemp : 48|8@1+ (1,-40) [-40|210] "deg c" Vector__XXX
38   SG_ EngTurboOilTemp : 32|16@1+ (0.03125,-273) [-273|1734.96875] "deg c" Vector__XXX
39   SG_ EngOilTemp1 : 16|16@1+ (0.03125,-273) [-273|1734.96875] "deg c" Vector__XXX
40   SG_ EngCoolantTemp : 0|8@1+ (0.03125,-273) [-273|-265.03125] "deg c" Vector__XXX
41                                                                            → Exhaust Temperature
42 BO_ 7398684160 IC1: 8 Vector__XXX
43   SG_ EngCoolantFilterDiffPress : 56|8@1+ (0.5,0) [0|125] "kPa" Vector__XXX
44   SG_ EngExhaustGasTemp : 40|16@1+ (0.03125,-273) [-273|1734.96875] "deg c" Vector__XXX
45   SG_ EngAirFilter1DiffPress : 32|8@1+ (0.05,0) [0|12.5] "kPa" Vector__XXX
46   SG_ EngIntakeManifold1Temp : 16|8@1+ (1,-40) [-40|210] "deg c" Vector__XXX
47   SG_ EngIntakeManifold1Press : 8|8@1+ (2,0) [0|500] "kPa" Vector__XXX
48   SG_ EngDslPrtclateFilterIntakePress : 0|8@1+ (0.5,0) [0|125] "kPa" Vector__XXX
49   SG_ EngAirIntakePress : 24|8@1+ (2,0) [0|500] "kPa" Vector__XXX
```

Figure 5. J1939 PGN And SPN IDs Used for Engine Data Collection (VDF)

SCIMITAR's built-in, CAN-Interface, was used to record ECU parameters specified on the VDF. Only parameters relevant to energy efficiency were recorded to reduce the risk of data loss and maintain manageable file size for post-processing purposes. Once the DAQ box was installed and connected to the PCANs, SCIMITAR automatically recorded engine parameters when the engines were started. Figure 6 presents the PCAN device utilized for CAN communication.

Figure 6. PeakCAN Connector

3.3 Diesel Fuel Consumption

Although diesel fuel consumption from the three engines was recorded through ECU communication, the HURST boiler was not equipped with a control unit. Two KRAL diesel fuel flow meters were used to record boiler diesel fuel consumption by measuring fuel flow rate and temperature. KRAL OME 20 diesel fuel flow meters were selected due to their measurement range and compact size [45]. Previous research programs conducted by WVU used these flow meters. Researchers determined that KRAL OME 20 accuracy was ±2% of the measured value [46]. Table 4 presents the KRAL flowmeters' technical specifications.

Table 4. KRAL OME 20 Diesel Flow Meters Technical Data

Parameter		OME-20
Nominal Diameter	DN (mm)	20
	DN (in)	3/4
Total Length with DIN flange	(mm)	135
Flow rate L/hr	Q_{max}	2700
	Q_{nom}	1800
	Q_{min}	18
Flow rate L/min	Q_{max}	45
	Q_{nom}	30
	Q_{min}	0.3
Max. Pressure	(bar)	40
Temperature	(°C)	-20 to 125
Viscosity	(mm^2/s)	1 to $1x10^6$
Precision	Of measurement value	±0.1%
Repeatibility		±0.01%
K-factor	K(P/L)	321
Frequency	F at Q_{nom} Hz	161

KRAL flow meters were attached to a KRAL BEM 500 display where the instantaneous reading could be observed. The BEM 500 converted voltage signals into readable data and sent it to the DAQ system via ethernet for SCIMITAR logging. Figure 7 shows the flow meters accompanied by the display used for on-field data collection.

Figure 7. KRAL BEM 500 Display and KRAL OEM 20 Diesel Flow Meters

Before installation, flow meters were calibrated and tested at WVU's engine laboratory. Tests showed that flow meters operated with errors less than 3%, as presented in Appendix A. Each of these flow meters has a turbine meter and a temperature sensor for density correction. One of the flow meters was installed in the boiler diesel fuel supply line, while the other was located on its return. Fuel consumed by the boiler was calculated by measuring fuel flow difference. Boilers are usually fueled by diesel or natural gas, depending on the fuel's availability. Data recorded referred to a diesel fueled boiler; therefore, an energy balance was performed to model a natural gas fueled boiler based on recorded heat demand and assuming constant thermal efficiency of 81% [37]. Boiler heat output was not measured directly. Rather, fuel consumption served as a surrogate along with the assumed constant efficiency as heat output estimations. The lower heating value (LHV) for diesel fuel was assumed to be 42,780 kJ/kg with a density of 838.9 g/L, while LHV for natural

gas was assumed to be 33,533 kJ/m^3 with a density of 0.68 kg/Sm^3 being consistent with values used by CAT and Waukesha in their respective engine specification literature [32,33]. Equation 2 shows the energy balance used for the boiler natural gas fuel consumption model.

Equation 2. Boiler Fuel Consumption Energy Balance

$$\frac{42,780\frac{kJ}{kg}\,x0.8389\frac{kg}{L}\,xDiesel\ Fuel\ (L)}{0.81\%} = \frac{\frac{33,533\frac{kJ}{m^3}}{0.68\frac{kg}{Sm^3}}x(Natural\ Gas\ (kg)}{0.81\%} \tag{2}$$

3.4 Temperatures

Temperatures from the three engine exhausts, boiler water feed, and boiler steam output were measured using K-type thermocouples connected to two 10-channel thermocouple input modules (ICP CONs) located in the DAQ system and connected to the SCIMITAR software. HHPDE exhaust temperature was directly measured with K-type thermocouples installed in the muffler, as shown in Figure *8*8. These temperature sensors have an uncertainty of ±2.2 °C [47]. Temperatures were later used for heat loss detection and addressing efficient heat recovery systems possibilities throughout the rig's energy source configuration.

Figure 8. K=Type Thermocouple Installed in the Engine's Muffler

3.5 Data Collection

Instrumentation occurred on February 17th, 2020, and the engines were fired for powering accessory equipment the following day. Drilling activities were not detected until February 21st, and data collection continued with minimal disruptions until the well was completed. The equipment was removed on March 20th during rig downtime. Therefore, the data collected included all processes from engine start up through the completion of drilling.

The boiler was fired on February 21st and remained in operation until March 5th. Boiler activity was detected on 14 days but was only active continuously with no disruptions for six of those days. The boiler operated intermittently during the remaining eight days, supplying smaller amounts of

heat based on varying weather conditions. Boiler activity depends exclusively on weather temperatures, the crew's tolerance to cold, and liquid temperatures to prevent freezing.

Engine data collection referred to Tier 2 engines for both HHPDE and DNGE. Therefore, the modeling methods include Tier 2 engine performance only. Newer HHPDEs are certified under tighter emissions standards, which involved aftertreatment systems and additional control. These could contribute to different exhaust gas conditions than the engines analyzed in this paper. The unconventional well development industry has not yet shifted towards Tier 4 engines, being still dominated by Tier 2 engines.

4. Modeling Methods

4.1 Fuel Consumption

HHPDE fuel consumption was measured through ECU data streamed by each of the engines' CAN systems. Therefore, diesel fuel consumption for the cycles extracted from engine activity was not modeled. However, when analyzing other possible scenarios, such as the HHPDE CHP hybrid model, fuel consumption was estimated as a function of engine load. Brake specific fuel consumption (BSFC) information was publicly available on the CAT 3512C specification sheet for laboratory conditions at 25%, 50%, 75%, and 100% engine loads [32]. Performance published on this specification sheet complied with the SAEJ1995 standard reference conditions. Diesel fuel density was assumed to be 838.9 g/L as calculated by CAT in their engine specification sheet [32].

The DNGE fuel model was developed using engine activity recorded by previous campaigns in Midland, TX, and Knight Fork, WV in 2016. The BSFC at 100% engine load specified on the Waukesha L7044GSI engine specification sheet was also utilized [33]. Fuel consumption from the specification sheet was based on ISO3046/1-1996 with a tolerance for commercial quality natural gas having a 33,533 kJ/m^3 (900 BTU/ft^3) LHV [33]. Field gas LHV varied due to composition, which impacted the standard fuel curve.

Fuel consumption data recorded from both sites were concatenated and binned in 15 groups. The first group contained any fuel consumption for engine loads between 0-20%, and 5% increments binned the remaining groups. Low engine loads were rare and can induce unnecessary error; therefore, all fuel consumption values indexed to engine loads under 20% were averaged into the first bin. Fuel consumption for the DNGE at 100% load was extracted from the specification sheet and added to the model. Natural gas density was assumed to be 0.68 kg/Sm^3. Figure 9 presents both engine fuel consumption models used for the simulations. The MATLAB® script used to calculate DNGE fuel consumption is available in Appendix B.
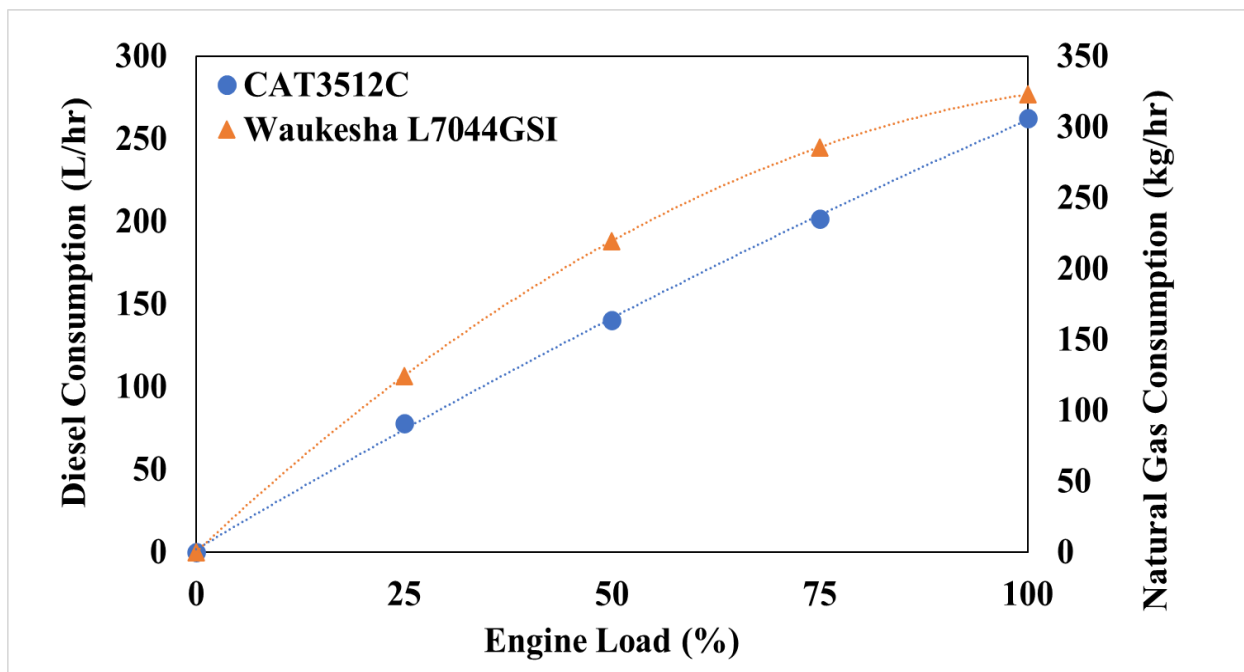
Figure 9. HHPDE And DNGE Fuel Consumption Models

4.2 Exhaust Temperature

An exhaust temperature model for each engine (HHPDE and DNGE) was modeled so simulations of different scenarios could be analyzed. Engine manufacturers generally provide exhaust temperature at full load. CAT also provided exhaust temperatures at 50% and 75% loads. However, no available data addressed low engine loads. To model lower engine loads, exhaust on-site temperature behavior, a quasi-steady engine operation at low load was identified during a transient operation portion for about a minute. All three engines averaged 20% engine load, and exhaust temperatures averaged 306 °C. This data point was added to the exhaust temperatures at 50%, 75%, and 100% engine load specified by the manufacturer, and a polynomial regression was constructed for the HHPDE exhaust temperature model.

DNGE exhaust stack temperature was also modeled based on a regression analysis from data previously recorded from DNGE engines at two other well sites. This data was concatenated and binned into 12 bins with two unique bins (0-10% load and 60%-100% load). The remaining data were binned in 5% increments between 10% and 60% load. Data points in the lowest and highest engine load ranges were averaged in wider bins due to transient operations over these loads. Also, 100% engine load stack temperature was obtained from Waukesha's specification sheet and included in the regression. Table 5 presents exhaust temperatures specified by the manufacturers

at laboratory conditions [32,33]. Figure 10 presents the HHPDE and the DNGE exhaust gas temperature model constructed. The MATLAB® script used for the DNGE exhaust temperature model is included in Appendix B.

Table 5. Exhaust Gas Temperature Data [32,33]

Engine Load	CAT 3512C Exhaust Temperature	Waukesha L7044GSI Exhaust Temperature
%	°C	°C
50	394	
75	391	
100	398	637
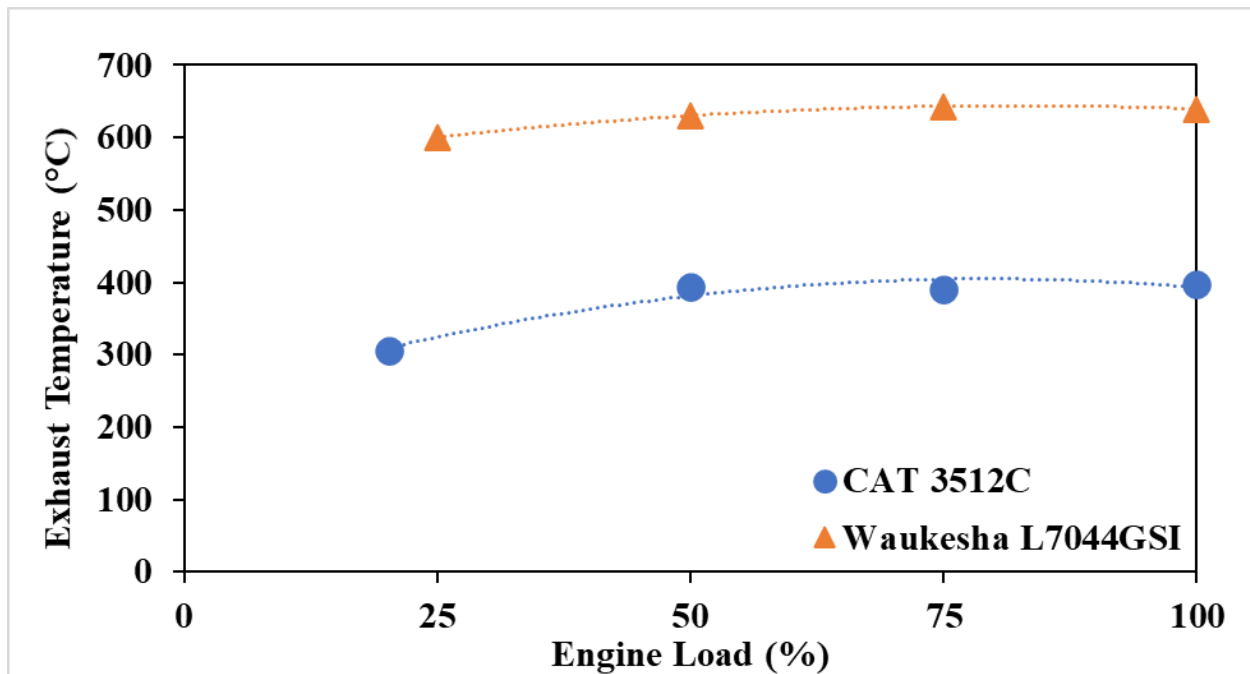
Figure 10. HHPDE And DNGE Exhaust Temperature Models

4.3 Exhaust Mass Flow and Exhaust Heat Rejection Models

Diesel exhaust gas is mainly composed of nitrogen (N_2), water (H_2O), carbon dioxide (CO_2), and oxygen (O_2) [48]. For simplicity, exhaust flow was assumed to be dry air (78% N_2, 21% O_2, 1% other components) and water vapor only. Dry air and water vapor were treated independently for heat rejection calculation purposes. The same assumption was used for the DNGEs where exhaust

composition is mostly N_2, CO_2, and H_2O. Waukesha L44GSI engines run on a slightly rich air-fuel mixture (λ=0.98 on average); therebefore, there was no excess oxygen in the exhaust gas.

CAT provides volumetric exhaust flow rates for 50%, 75%, and 100% engine loads and a mass flow rate for 100% engine load. They also report exhaust heat rejection at these three engine loads. An HHPDE exhaust flow model was constructed using these data points. Table 6 presents engine exhaust information provided by the manufacturer.

Table 6. CAT 3512C Exhaust Specifications [32]

Engine Load	Volumetric Exhaust Flow Rate	Mass Exhaust Flow Rate	Exhaust Heat
%	m³/min	Kg/hr	kW
50	125.9		501
75	173.5		707
100	218.0	6979	902

The exhaust gas composition of a CAT 3512C engine operating on the Marcellus Shale Energy and Environment Laboratory (MSEEL) was measured during a previous WVU campaign [46]. A Fourier-transform infrared spectroscopy (FTIR) device was used to measure exhaust water vapor concentration. The average water vapor concentration was found to be 7.4%. Thus, HHPDE exhaust flow was assumed to be around 7.4% water vapor and 92.6% dry air. Using this ratio of dry air to water vapor in the exhaust, the molar exhaust flow rate was iteratively calculated to estimate exhaust mass flow and heat rejection. Values converged when the minimal error was achieved at 50%, 75%, and 100% engine loads. Iterations were dominated by Equation 3 based on thermal properties for dry air and water vapor which are presented in Table 7. Calculated mass flow rate and calculated heat rejection were compared to data provided on the engine specification sheet. Tables 8 and 9 present estimated exhaust flow rates and heat rejections with their respective errors.

Equation 3. Heat Rejection Equation

$$\dot{Q} = \dot{m_{H_2O}}\left(h_{exhaust,H_2O} - h_{ambient,H_2O}\right) + \dot{m_{Air}}\left(h_{exhaust,Air} - h_{ambient,Air}\right) \qquad (3)$$

Table 7. Specific Enthalpy of Dry Air and Water Vapor Used for HHPDE Exhaust Heat Calculations [32,49]

Engine Load	Temperature	h H_2O	h Air
%	°C	kJ/kg	kJ/kg
Ambient	25	418 (Sat. Liquid)	298
50	394	3265	668
75	391	3260	675
100	398	3273	682

Table 8. CAT 3512C Mass Exhaust Flow Rate Model [32]

Engine Load	Total Mass Exhaust Flow	Total Mass Exhaust Flow Rate by CAT	Error
%	kg/hr	kg/hr	%
50	3632		
75	5157		
100	6637	6797	2.4

Table 9. CAT 3512C Exhaust Heat Model [32]

Engine Load	Heat Release Calculated	Heat Release Specified by CAT	Error
%	kW	kW	%
50	501	501	0.0
75	707	707	0.0
100	924	902	2.3

A DNGE exhaust flow model was built based on the manufacturer's specification sheet information, ECU data, and emissions data recorded during previous projects. Natural gas fuel consumption was converted to SI units by converting standard cubic feet (SCF) to Sm^3 and using a natural gas density of 0.68 kg/Sm^3. Equation 3 was also used for heat rejection calculations. Natural gas LHV was assumed to be 33,533 MJ/m^3 as specified by Waukesha's engine specification sheet [33].

According to data previously recorded, the DNGE AFR is 16.86 [37]. Note that the stoichiometric AFR for natural gas combustion is 17.2 when approximated as methane. With a modeled fuel consumption curve and a fixed AFR assumption, the mass air intake was calculated. It was determined from previous data that DNGE water vapor exhaust concentration was 17.94% on average [37]. Therefore, the exhaust flow of the DNGE was assumed to be 17.94% water vapor and 82.1% dry air. If pure methane was considered for stoichiometric combustion, Equation 4 presents the stoichiometric water vapor concentration in exhaust.

Equation 4. Methane Stoichiometric Combustion

$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 7.52N_2 \tag{4}$$
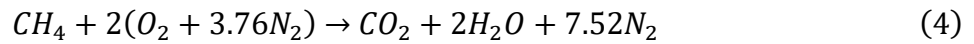
Methane complete combustion products indicate a water vapor exhaust concentration of 19%. On-site data collected referred to field gas composition that differed from pure methane, dropping water vapor concentration to 17.9% of its exhaust composition. Table 10 presents the thermal properties for dry air and water vapor used for the DNGE exhaust heat rejection calculations. Like the HHPDE exhaust heat model, the DNGE model was calculated via iterations until converging to the minimum error governed by Equation 3. Tables 11 and 12 present the DNGE exhaust flow and heat model errors.

Table 10. Specific Enthalpy of Dry Air and Water Vapor Used for DNGE Exhaust Heat Calculations [33,49]

Engine Load	Temperature	h H_2O	h Dry Air
%	°C	kJ/kg	kJ/kg
Ambient	25	419	298
100	637	3787	944

Table 11. Waukesha L7044GSI Mass Exhaust Flow Rate Model [33]

Engine Load	Calculated Mass Exhaust Flow	Total Exhaust Mass Flow Specified by Waukesha	Error
%	Kg/hr	Kg/hr	%
100	4653	5113	9.0

Table 12. Waukesha L7044GSI Exhaust Heat Model [33]

Engine Load	Calculated Heat Rejected	Heat Rejection Specified by Waukesha	Error
%	kW	kW	%
100	1256	1143	9.0

4.4 Exhaust Heat Exchanger Design

HEX design was performed using CC-THERM software, a sub-program to the CHEMCAD® suite developed by Chemstations [50]. CC-THERM software allows the user to design HEXs and performs calculations of hypothetical conditions. This program supports shell-and-tube, air-cooled, plate-and-frame, and double-pipe exchangers. Its integration with CHEMCAD software makes it possible to calculate HEX exit conditions from simulations [50]. This software bundle was used for the design of the E-HEXs and JW-HEXs. These exchangers' physical design was outside of this project's scope, but the focus was to estimate potential heat recovery. HEXs were selected to be automatically sized and designed complying with the American Society of Mechanical Engineers (ASME) and Tubular Exchanger Manufacturers Association (TEMA) standards [51]. Size constraints were implemented so that the HEX could be placed in a standard 20'x8'6''x8' (6.1x2.6x2.4 m) container frame for simple handling and transportation, typical in the industry. Pressure drop was also limited and constituted the primary constraint to avoid exhaust backpressures that would negatively impact engine efficiency [28]. Most diesel generator manufacturers set their maximum allowed backpressure limits ranging from 6.7 to 10.2 kPa (0.97 to 1.48 psi) [28].

Boilers usually provide saturated steam at pressures around 110 psi; therefore, the E-HEX design was intended to match the boiler's pressure and temperature output. Water inlet was selected to be variable while steam output conditions were set to be fixed at saturated steam at 110 psi. Saturated steam distribution lines are equipped with steam traps that redirect saturated and condensed water to a day tank that functions as a recovery feedwater system that feeds the boiler. The goal was not to redesign the heat distribution system but to assist/replace the boiler with a set of HEXs. Therefore, steam distribution system design and specifications had to integrate into the design. Day tank water was selected as the water inlet for the designs. Since the boiler steam output has a fixed pressure, the temperature of the water returned to the day tank was nearly constant, only

affected by ambient temperature. Water inlet temperature (day tank temperature) was assumed to be constant at the average temperature recorded during boiler usage (70 °C).

E-HEX and JW-HEX types were selected to be shell-and-tube due to their popularity in steam generation applications in industries such as combined cycle power plants. These configurations also have the ability to handle high pressures and flows.

CC-THERM used the logarithmic mean temperature difference (LMTD) method instead of the number of transfer units (NTU) method. The NTU method predicted the outlet fluid temperature if the heat transfer coefficient and inlet temperatures were known. The LMTD method was convenient for determining the overall HEX coefficient based on the measured inlet and outlet fluid temperatures. Therefore, the HEX designs were based on the exhaust flow and temperature and water inlet temperature. The software neglects any heat losses to the environment. Exhaust and other higher temperature lines at drilling sites tended to be insulated. HEX designs could be implemented to match well with the externally insulated boundary conditions. Using the LMTD method and software package, an iterative process was performed to find the best HEX design for the required heat transfer coefficient (U) and the surface area (A) while also meeting necessary constraints. Equation 5 was utilized for LMTD calculations.

Equation 5. Heat Exchanger Heat Transfer

$$Q = U \times A \times LMTD \tag{5}$$

Where LMTD is the logarithmic mean temperature difference defined as:

Equation 6. Logarithmic Mean Temperature Difference

$$LMTD = \frac{\Delta T_A - \Delta T_B}{ln \frac{\Delta T_A}{\Delta T_B}} \tag{6}$$

Where ΔT_A is the temperature difference between the two streams at end A and ΔT_B is the temperature difference between the two streams at B. Figure 11 presents a counterflow shell and tube HEX diagram using the LMTD method.
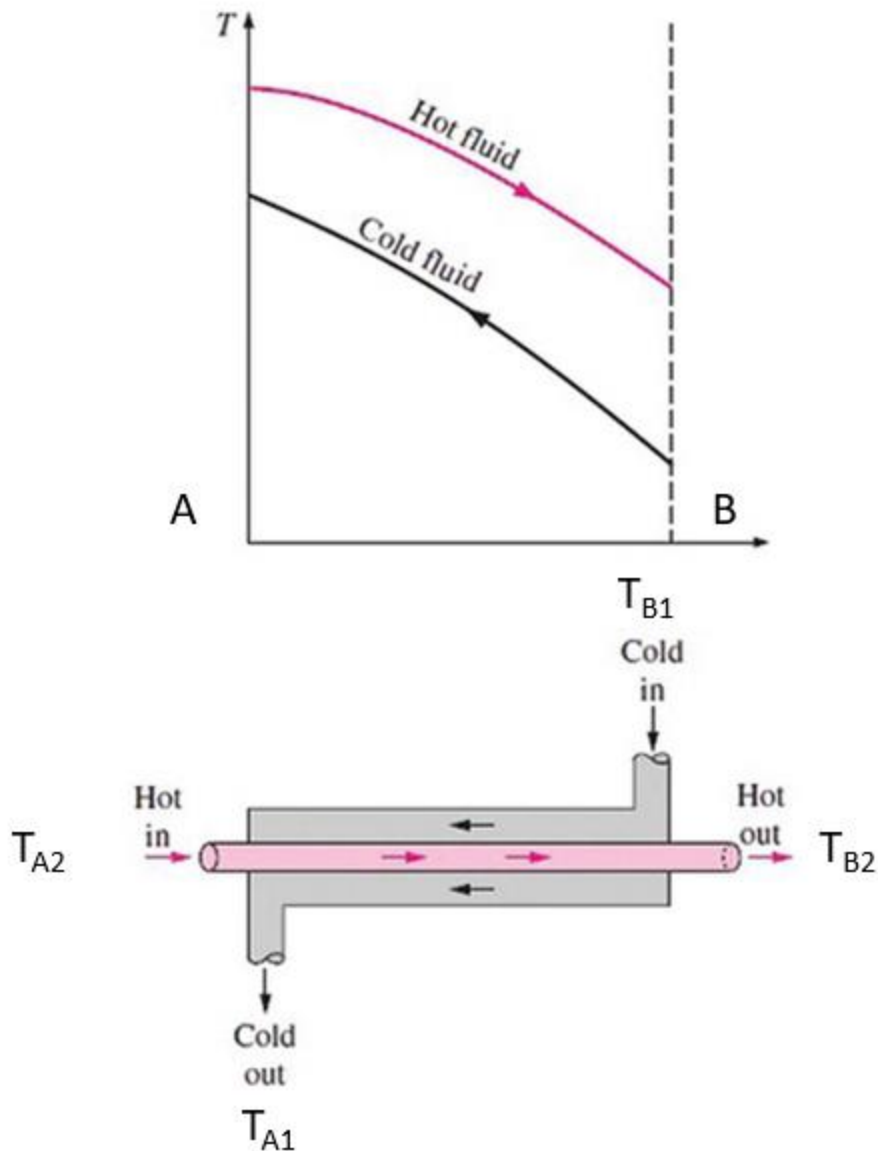
Figure 11. Counterflow Shell and Tube Heat Exchanger Diagram Using the LMTD Method [52]

The HHPDE E-HEX was sized to recover the most heat possible by utilizing engine conditions at 100% load. The water inlet rate was iterated until it converged to the maximum water flow maintaining steam outlet conditions. The system uses a reversible pump to pressurize the water and meet steam pressure requirements to imitate heat output rates. Pump energy consumption was not considered in this first energy analysis for simplicity. In the current rig configuration, water is pumped from the day tank into the boiler. Pump work was small and negligible compared to the other energy demands of the system. The software automatically calculated enthalpy for each fluid

inlet and outlet. The difference in enthalpy at constant pressure represents the energy transfer rate. Therefore, the enthalpy difference between the steam outlet and the water inlet represented the recovered heat in the exchanger. Various combinations of exhaust flow and exhaust temperatures were rated in the software. Their respective heat recovery values were saved to develop a two-entry table. Values that did not fall at the exhaust flow and temperatures simulated were linearly interpolated. Figure 12 presents the HHPDE sizing conditions extracted from the software. The HHPDE E-HEX TEMA sheet is included in Appendix C.
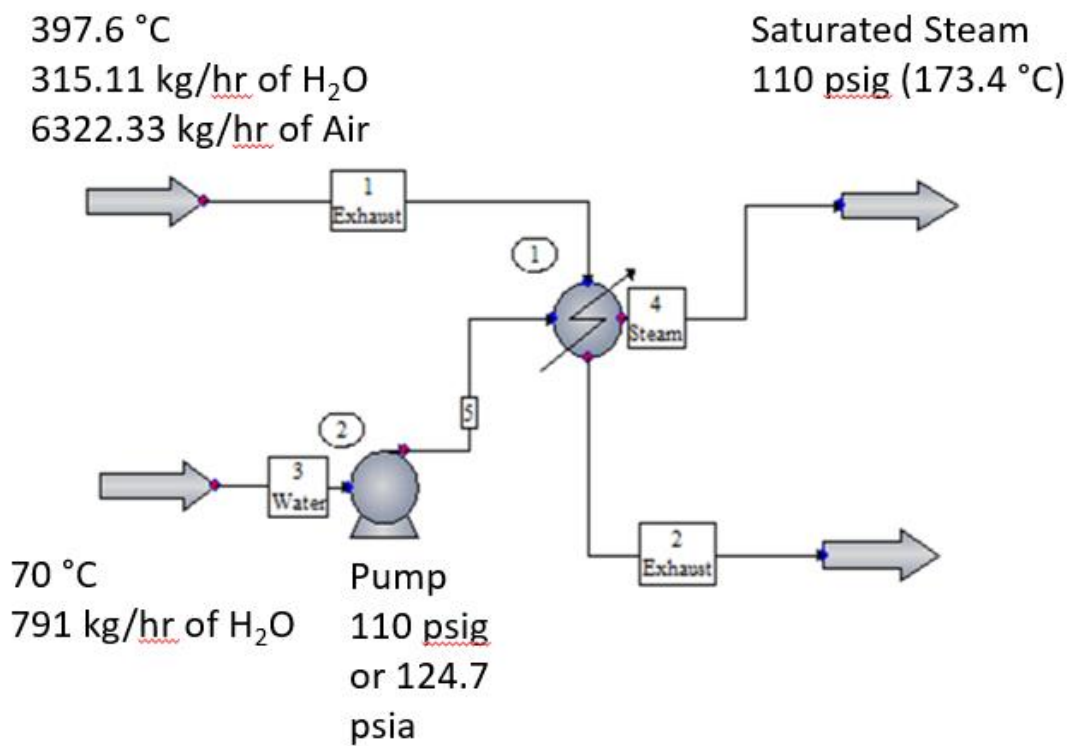
Figure 12. HHPDE E-HEX Sizing and Design Conditions in The CHEMCAD Environment

Similarly, a DNGE E-HEX was designed and sized for the engine's operating conditions at full engine load with the exhaust temperature provided by the specification sheet and exhaust flow calculated by the DNGE exhaust flow model. Allowable pressure drop was assumed to be 10.2 kPa as determined for HHPDE E-HEX. Figure 13 presents the DNGE E-HEX design conditions in the software environment. The DNGE exhaust heat rejection depended only on engine load; thus, DNGE E-HEX heat recovery also depended on engine load. Table 13 presents HHPDE and

DNGE E-HEX main design parameters. The DNGE E-HEX TEMA sheet is included in Appendix C.
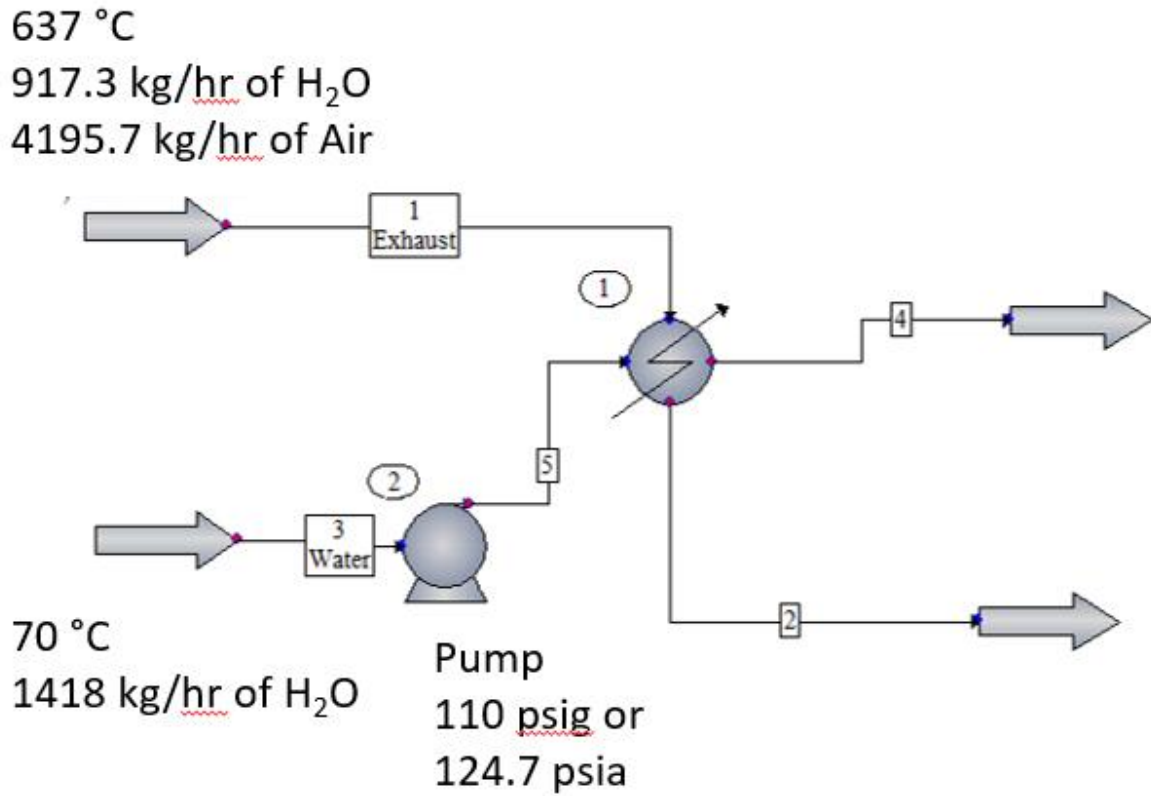
Figure 13. DNGE E-HEX Sizing and Design Conditions in the CHEMCAD Environment

Table 13. HHPDE and DNGE E-HEX Design

E-HEX	HHPDE E-HEX	DNGE E-HEX
Shell Diameter (m)	2.1	2.4
Tube Length (m)	2.4	3.0
Number of Tubes	5557	7508
Effective Transfer Area (m²)	798.3	1352.4
U (W/m2°C)	30.2	21.5
Tube O.D. (cm)	1.91	1.91
Tube I.D. (cm)	1.56	1.56

A commercially available E-HEX was compared with the CHEMCAD designed E-HEX. CAIN industries is an exhaust heat recovery and steam generator system manufacturer. They usually work with large diesel and/or natural gas-powered engines. The best fit unit was selected from their exhaust steam generator (ESG) catalog. The ESG1-B30B14.5CSS unit equipped with an internal economizer was recommended for optimal effectiveness. The ESG performance was simulated by the company's software given exhaust flow and temperatures provided by CAT's engine specification sheet [32]. A regression analysis was performed from the exchanger company's three data points to compare results and validate simulations. Note that ESG heat recovery estimation is only a function of engine load from data recorded in CAT's laboratory at steady loads. The CAIN ESG1-824B16CSS was selected as the best fit for the DNGE engine. Figure 14 presents HHPDE E-HEX, DNGE E-HEX, HHPDE ESG, and DNGE ESG performance as a function of engine load. Both HHPDE ESG and DNGE ESG design and quote can be found in Appendix D.
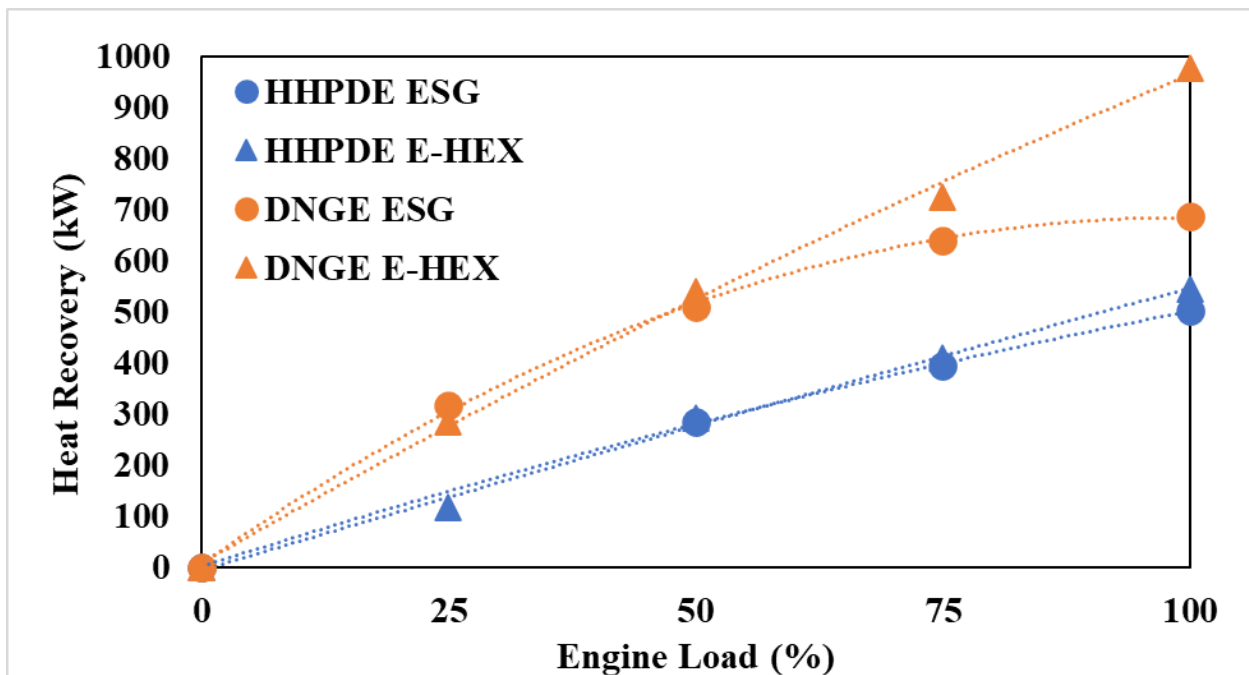
Figure 14. HHPDE ESG, HHPDE E-HEX, DNGE ESG, and DNGE E-HEX Heat Recovery as a Function of Engine Load

4.5 Hybrid System Battery Model

A basic battery design was created based on a hybrid system reported in the literature [13]. The system used a 584-kW-hr battery pack. Only the rate of charge/discharge and usable SOC range

were considered in the design for simplicity. The SOC is the charge of a battery relative to its capacity. The rate of charge/discharge is the amount of energy being supplied or extracted from the battery per unit of time. Coulomb's C-rate is a measure of the rate at which a battery is being charged or discharged. It determined the maximum rate of charge and discharge. The HEMS was designed with a 3C discharge and 1C charging rates, meaning the battery can be fully charged in one hour while it can be discharged in 20 minutes. The allowable range of the SOC was assumed to be between 40% and 100% capacity. Considering the SOC restrictions, the battery could discharge from 100% to 40% in 12 minutes at the maximum discharge rate. Coulombic efficiency is a measure of charging and discharging efficiency that accounts for heat losses through internal impedances. Lithium-ion cells have high coulombic efficiency estimated at around 80% and 90% for charging and discharging, respectively [52]. Coulombic efficiencies were also integrated into the HEMS model. The power output limit of the system was rated at 1752 kW, larger than any of the analyzed individual engines.

The HEMS was designed to enable the operation of fewer engines, requiring those in operation to operate at higher loads near peak efficiency. This reduced transient operations and emissions. The initial battery SOC was assumed to be 70% for scenarios analyzed, and the number of online gensets was calculated given average and maximum power output. The goal was to have as few online gensets as possible at any given time. Initial engine load was assumed to be 75%, and engine load is automatically adjusted based on the last 60 seconds of SOC. Table 14 presents the HEMS ideal engine load controller commands. Also, the HEMS was designed to maintain instant heat recovery at higher rates than the system's instant heat demand when possible

Table 14. HEMS Ideal Engine Load Controller

Previous 60 Seconds of Activity	SOC	Load Adjustment	Ideal Load Output
SOC Continous Increment	SOC≥90	-0.03	Decreased
	SOC≥80	-0.02	
	SOC≥60	-0.01	
SOC Continous Reduction	SOC≤50	+0.03	Increased
	SOC≤60	+0.02	
	SOC>60	+0.01	
Non-continuous SOC Activity Detected	40≥SOC≥100	0.00	Maintained

The HEMS load controller worked on a two-level decision-making unit. The top-level was the limitations due to the HEMS restrictions. These helped the HEMS stay within the allowable SOC working range and its respective charging/discharging limits. The second level included the ideal load controller presented in Table 14 and the heat balance controller.

HEMS top-level load control unit:

- If SOC approached 100%, engine load was decreased to its minimum possible
- If SOC approached 40%, engine load was increased to its maximum possible
- If the rate of charge approached 1C, engine load was decreased to its minimum possible
- If the rate of discharge approached 3C, engine load was increased to its maximum possible

HEMS second-level load control unit:

- Ideal engine load controller as presented in Table 14
- If instant heat demand exceeded instant heat recovery, engine load was increased by 1% load increments, and excess energy was stored in the HEMS battery

In order to compare hybrid to non-hybrid systems, fuel consumption was corrected for battery SOC differences for each cycle. Both HHPDE and DNGE HEMS MATLAB® scripts can be found in Appendix B.

5. Cycles

A set of four activity cycles was extracted from on-site recorded data to study the energy UF during varying timespans that were representative of unconventional well development. These cycles were selected to represent various drilling operations and conditions. These cycles included three short duration cycles (each an hour in length) where drilling operations were powered by either two engines (2E1HR), three engines (3E1HR), and transient tripping pipe/casing operations powered by three engines (TP1HR) (high transient loads). The final cycle was a longer duration (representative of an entire day) that combined all operations (24HR).

Since DNGE activity was not directly recorded, the same cycles recorded on HHPDEs were utilized. The same number of online gensets were used, and the engine load was adjusted to the DNGE power (1253 bkW), assuming a constant generator efficiency of 95%. Generator efficiencies might vary, especially at low loads. However, a constant efficiency reported by manufacturers was utilized for simplicity [32,33]. Figures 15 through 18 present the genset power output for each of the analyzed cycles.

Cycles were characterized by a name, the number of online gensets, average power output, drilling operation, and timespan. Each of these cycle characteristics are specified below:

- 2E1HR Cycle
    - Two online gensets
    - Average Power Output was 1360 kWe
        - The average engine load CAT3 512C was 65.0%
        - The average engine load Waukesha L7044GSI was 57.1%
    - Steady drilling operation
    - One hour long
- 3E1HR Cycle
    - Three online gensets
    - The average power output was 1455 kWe

- The average engine load CAT3 512C was 46.4%
- The average engine load Waukesha L7044GSI was 40.7%
- o Steady drilling operation
- o One hour long
- TP1HR Cycle
  - o Three online gensets
  - o Average Power Output was 801 kWe
    - The average engine load CAT3 512C was 25.5%
    - The average engine load Waukesha L7044GSI was 22.4%
  - o Tripping pipe/casing operation (highly transient)
  - o One hour long
- 24HR Cycle
  - o Two and three online gensets
  - o Average Power Output was 1209 kWe
    - The average engine load CAT3 512C was 38.5%
    - The average engine load Waukesha L7044GSI was 33.9%
  - o Steady drilling and tripping pipe/ casing operations
  - o 24 hours long

Figure 15 presents the 2E1HR cycle, which represents low load steady drilling operations with two engines online for one hour.
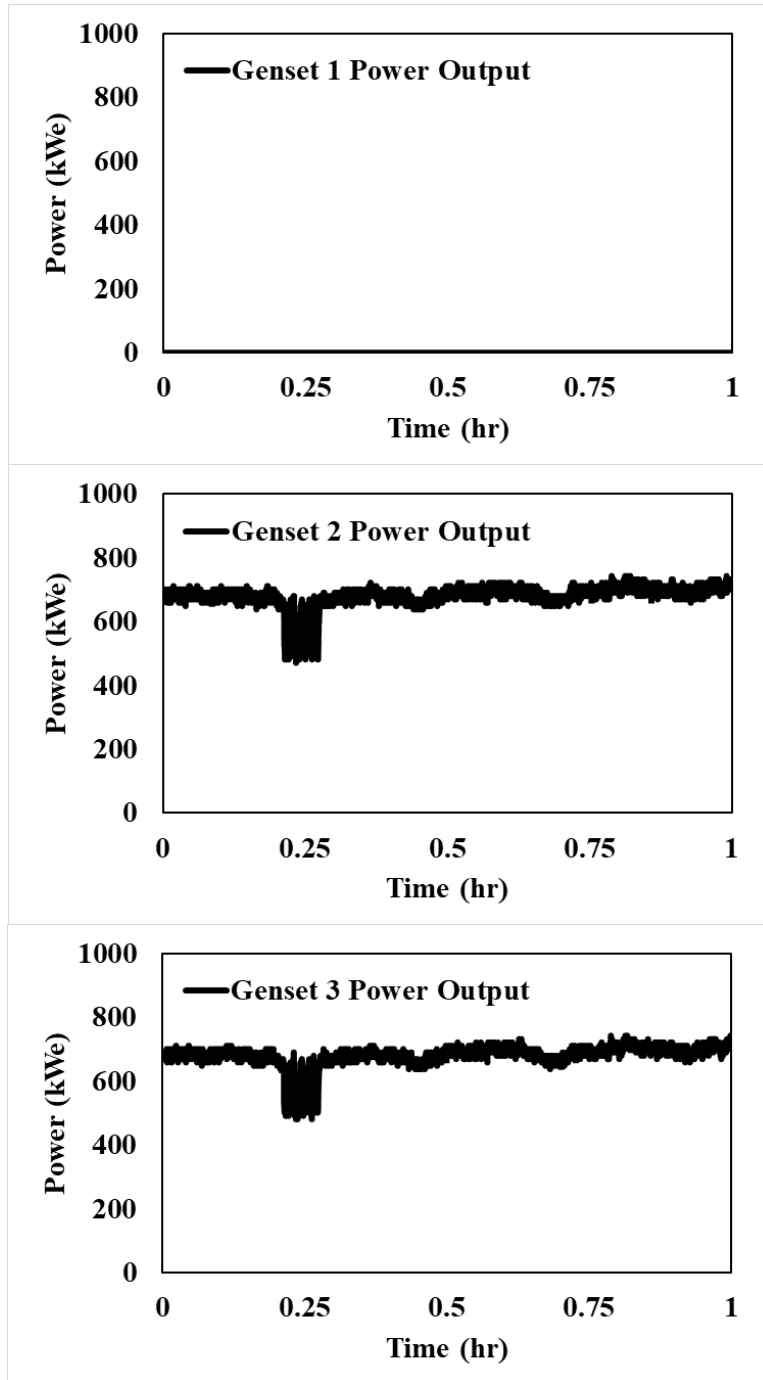
Figure 15. 2E1HR Cycle Genset Power Output

Figure 16 presents the 3E1HR cycle, which is characteristic of a high load drilling operation. The differentiation to the 2E1HR cycle is that it had three engines online and higher electric power outputs.
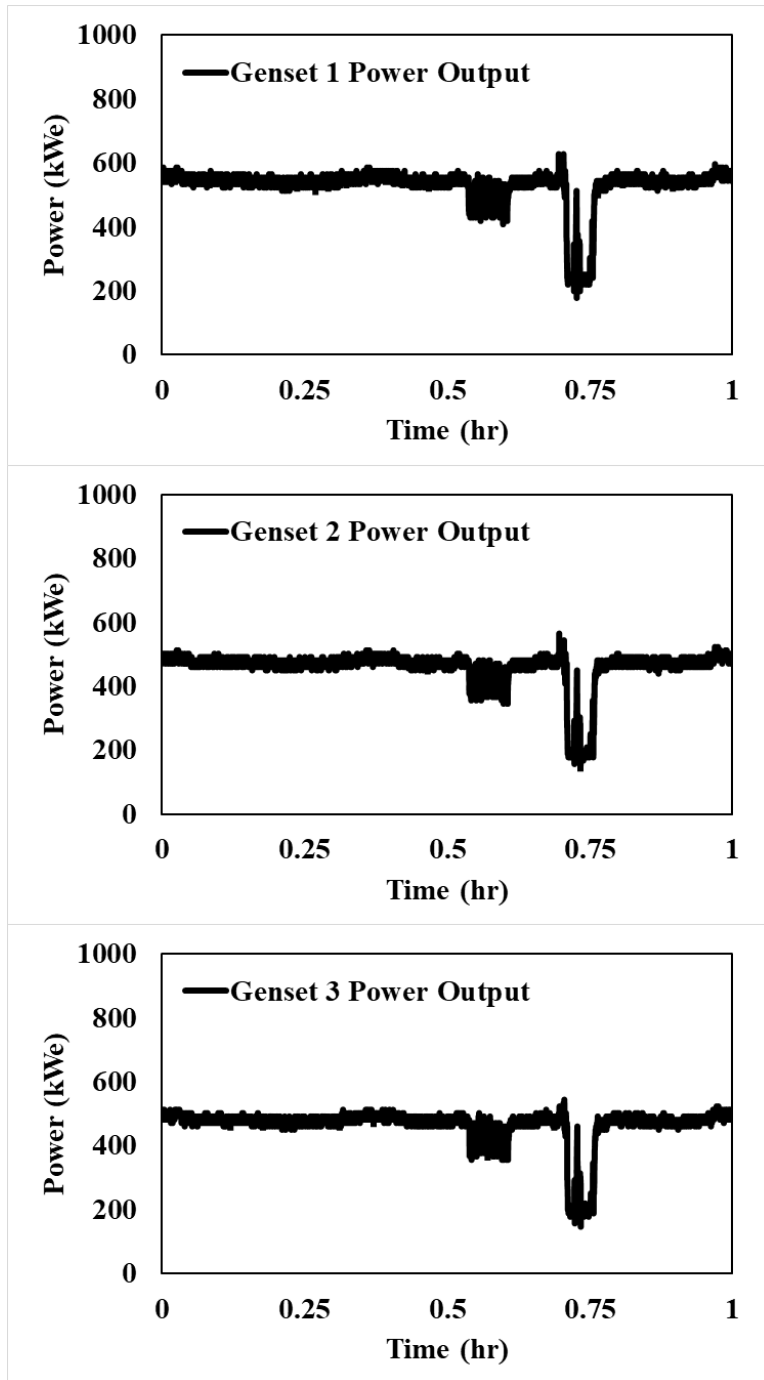
Figure 16. 3E1HR Cycle Genset Power Output

Figure 17 presents the TP1HR cycle, which represents transient activity during tripping pipe/casing operations of the rig where high loads are required after low loads or idling operations causing transient engine operation. This cycle demanded three engines online for one hour.
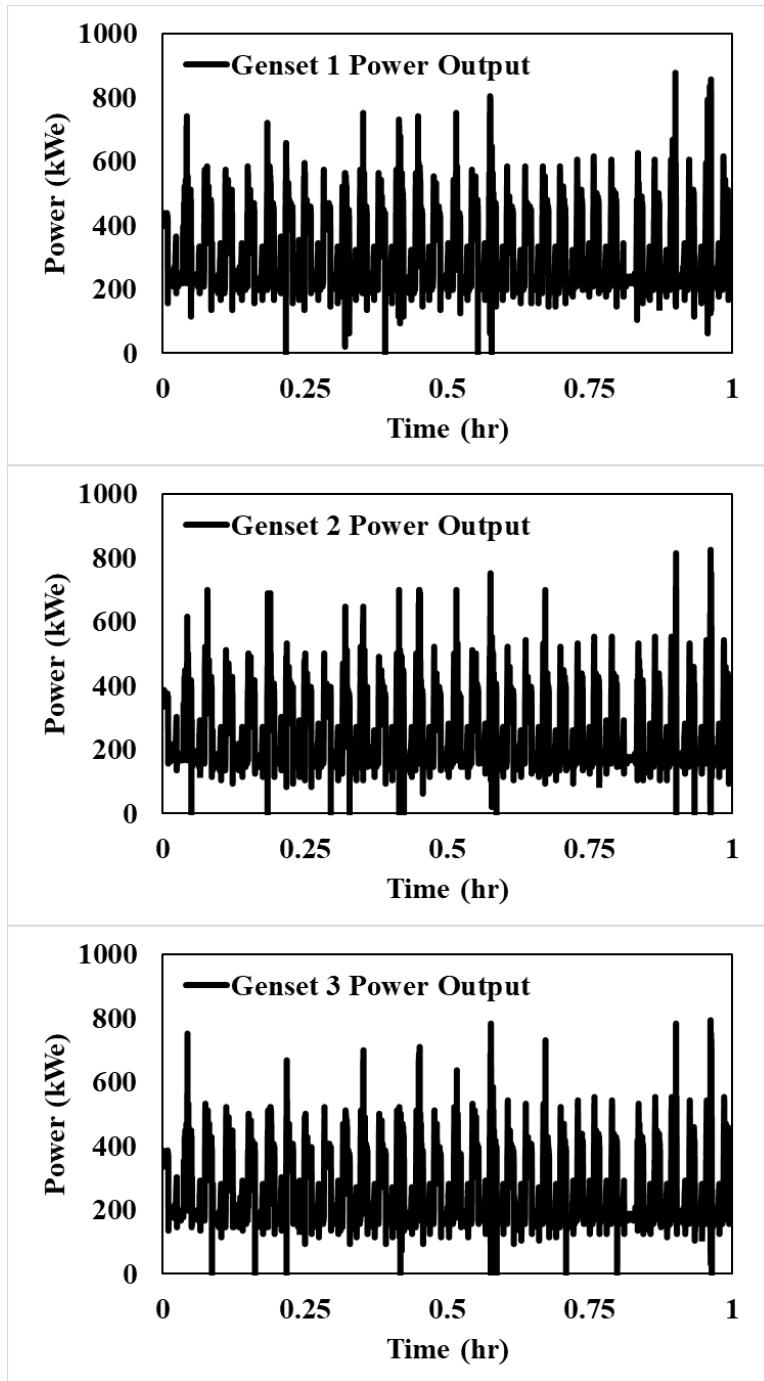
Figure 17. TPHR Cycle Genset Power Output

Figure 18 presents the 24HR cycle, which combined all engine operations on a well pad. This engine operation profile was extracted from 24 hours. It was characterized by having two and three engines online, steady drilling operation, and tripping pipe/casing operations.
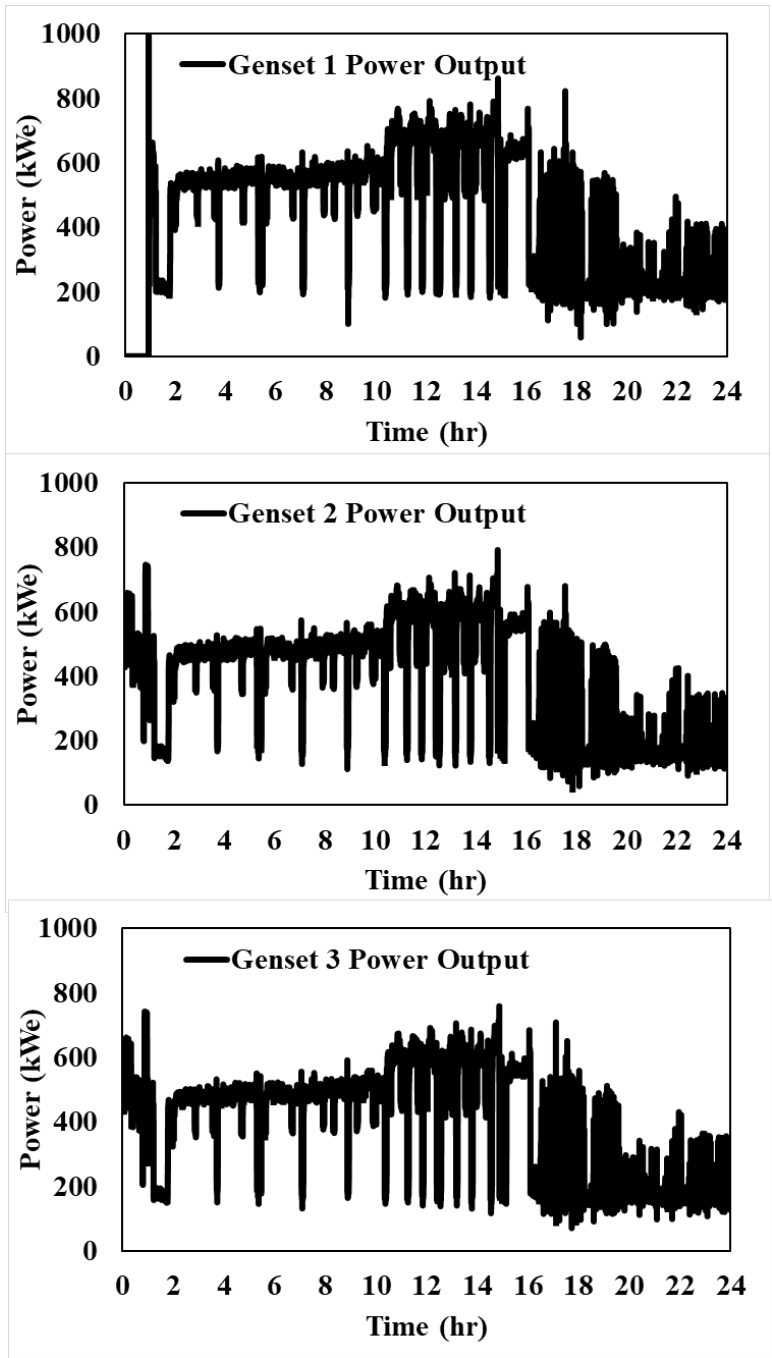
Figure 18. 24HR Cycle Genset Power Output

6. Results and Discussion

6.1 Rig's Energy Demand

Four different scenarios were analyzed over the four characteristic cycles previously defined. These scenarios were:

- High horsepower diesel engine combined heat and power system (HHPDE CHP)
- High horsepower diesel engine combined heat and power hybrid system (HHPDE CHP hybrid)
- Dedicated natural gas engine combined heat and power system (DNGE CHP)
- Dedicated natural gas engine combined heat and power hybrid system (DNGE CHP hybrid)

Table 15 presents the total rig energy demand for the duration of each cycle as well as the average boiler heat demand.

Table 15. Cycle Energy Demand Specifications

Cycle	Rig's Energy Demand	Boiler's Energy Demand	Boiler's Average Instant Demand
Name	MJ	MJ	kW
2E1HR	4895	2619	727.5
3E1HR	5238	2339	649.7
TP1HR	2884	2388	663.3
24HR	104,437	60,760	703.2

6.2 Energy Saving Model: HHPDE CHP

The HHPDE CHP system was simulated in a 0-D model in a SIMULINK environment presented in Figure 19, and results are presented in Table 16. The HHPDE E-HEX is the CHEMCAD designed exhaust heat exchanger and HHPDE ESG is the industrial exhaust heat exchanger being compared in this analysis. UF is the ratio of the energy used (heat or work) to the fuel energy (LHV).

The SIMULINK model for the HHPDE CHP system model specifications is attached in Appendix E and included:

- CAT 3512C engine subsystem
    - Calculated exhaust heat rejection utilizing Equation 3
        - Exhaust mass flow rate modeled as a function of engine load
            - Dry air mass flow rate
            - Water vapor mass flow rate
        - Exhaust temperature recorded by the thermocouples installed in the exhaust muffler
    - Fuel consumption from ECU
    - Power output calculated assuming constant generator efficiency of 95%
- HHPDE E-HEX subsystem
    - Calculated exhaust heat recovery from CHEMCAD developed two-entry table, dry air, and water vapor mass exhaust flow rate from CAT 3512C engine subsystem and exhaust temperature
- HHPDE ESG subsystem
    - Calculated exhaust heat recovery from the exchanger performance provided by the manufacturer and available in Appendix D.
- Boiler subsystem
    - Fuel consumption (diesel) recorded by the KRAL flow meters
        - Calculated heat supplied to the system from fuel consumption assuming constant efficiency of 81%
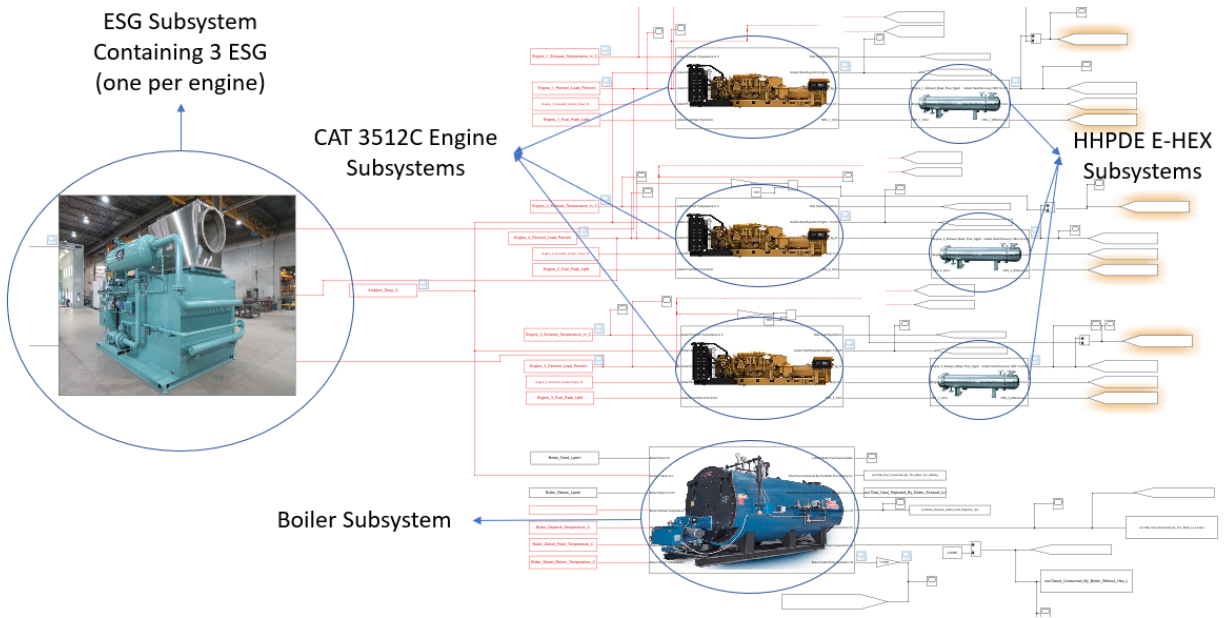
Figure 19. HHPDE CHP 0-D SIMULINK Model

Table 16. HHPDE CHP Heat Recovery

Cycle	HHPDE E-HEX Heat Recovered	HHPDE E-HEX Heat Recovered-Demanded	HHPDE ESG Heat Recovered	HHPDE ESG Heat Recovered-Demanded
Name	MJ	%	MJ	%
2E1HR	3226	123	2517	96
3E1HR	3072	131	2894	124
TP1HR	1220	51	1907	80
24HR	57,070	94	60,326	99

E-HEX heat recovery was sufficient to meet and exceed boiler heat demand for 2E1HR and 3E1HR cycles. However, only 94% and 51% of the required heat was recovered for the 24HR and TP1HR cycles, respectively. For the commercial unit (HHPDE ESG), only the 3E1HR cycle exceeded the boiler heat output required. The other three cycles did not recover enough heat to meet boiler heat demand. The 2E1HR cycle recovered 96% of the boiler heat demanded, the TP1HR cycle recovered 80% of the boiler heat demanded, and the 24HR cycle recovered 99% of the boiler heat demanded.

Although HHPDE E-HEX proved to reduce diesel fuel consumption by supplementing heat, the boiler would still be needed to meet the rig's steam demand. Therefore, the second source of potential heat recovery was assessed. CAT 3512C engines reject up to 412 kW of heat to the environment through their radiator [32]. Table 17 presents JW heat rejection specified in the technical engine sheet.

Table 17. HHPDE Heat Rejection Through the Radiator [32]

Engine Load	Heat Rejection Through Radiator
%	kW
50	263
75	340
100	412

Coolant return temperature was recorded via CAN. Data from the three engines were combined into a single dataset and binned based on engine load. The bins were distributed as follows: 0%-40%, 40%-50%, 50%-60%, 60%-70%, and 70%-100%. Bins were defined this way due to limited data at low and high engine loads. Also, these engine loads were generally only observed during transient operations. Engine load and coolant return temperature were averaged for each bin, and results are presented in Table 18. The MATLAB® script used to model coolant temperature is available in Appendix B.

Table 18. CAT 3512C Coolant Return Temperature Model

Engine Load	Coolant Return Temperature
%	°C
25	74.9
50	78.7
75	82.5
100	86.3

If a heat exchanger replaced the radiator, the working fluid is the coolant that goes through the radiator rather than the coolant that goes through the bypass back into the engine block. Therefore, coolant mass flowrate is variable depending on engine load. The coolant mass flow rate through the heat exchanger was modeled using Equation 7.

Equation 7. Coolant Heat Transfer Calculation

$$\dot{Q} = \dot{m}c_p\Delta T \tag{7}$$

Where c_p is the coolant's heat capacity (approximated as water) at 99°C (4.14 kJ/kgK). Using Equation 7, with the heat rejected presented at Table 17, and a modeled coolant return temperature, a coolant mass flow rate model was developed. Table 19 presents coolant mass flow rates through the radiator as a function of engine load. Figure 20 presents the coolant flow rate through the JW-HEX model.

Table 19. Coolant Mass Flow Model

Engine Load	$\dot{Q}$	JW Outlet Temperature	JW Inlet Temperature	ΔT	$\dot{m}$ Calculated
%	kW	°C	°C	°C	kg/s
50	263	99	78.7	20.3	3.1
75	340	99	82.5	16.5	4.9
100	412	99	86.3	12.7	7.7
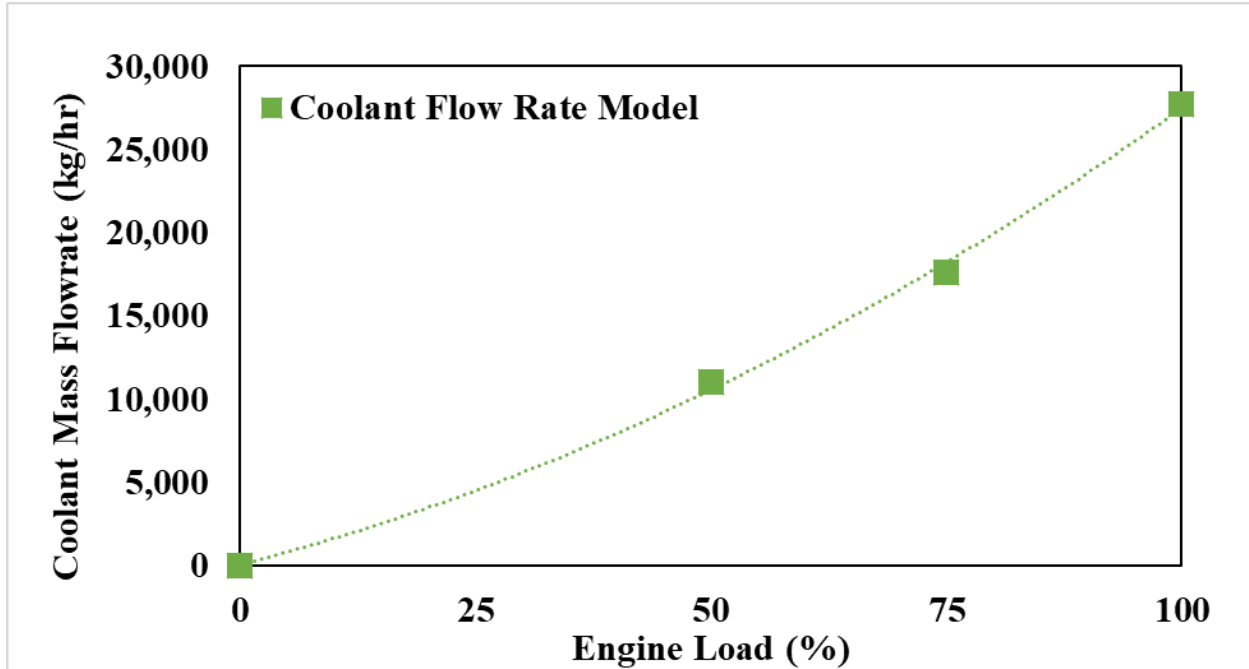
Figure 20. HHPDE JW-HEX Mass Flow Rate Model

Equation 8. Coolant Mass Flow Rate Error

$$Error = \left| \frac{Calculated - Actual}{Actual} \right| x100 \tag{8}$$

Table 20. CAT 3512C Coolant Model Error

Engine Load	$\dot{Q}$ Modeled	$\dot{Q}$ Specified By CAT	Error
%	kW	kW	%
50	251.6	263	4.3
75	352.4	340	3.6
100	408.4	412	0.9

CHEMCAD software was also used for HHPDE JW-HEX design. The HHPDE JW-HEX was sized and designed for full load engine operation to recover the most heat possible. The only restriction on the JW-HEX design was the allowable pressure drop. Allowable pressure drop on the radiator information was not available for the CAT 3512C engine; therefore, information from a CAT G3512 natural gas engine was used for estimating maximum allowable pressure drop.

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Interpolation was used to estimate the maximum modeled coolant mass flow rate into the natural gas G3512 engine. It was estimated that the maximum allowable pressure drop for CAT 3512C engine was 30.3 psi or 208.9 kPa. Figure 21 shows the JW pressure drop on the CAT G3512 engine.
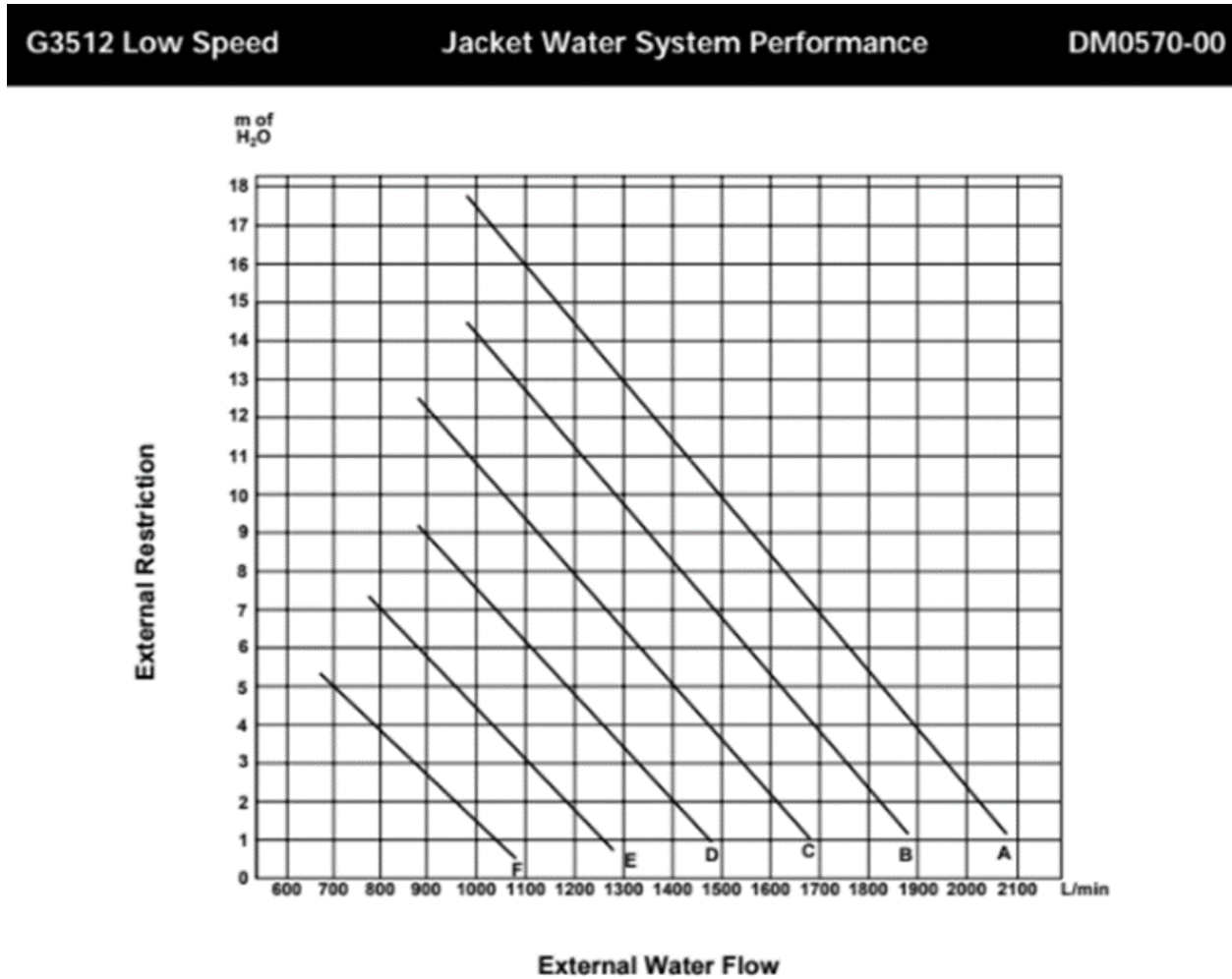
Figure 21. CAT G3512 Low-Speed Pressure Drop (B=1200 RPM)

Size was not a constraint for HHPDE JW-HEX since this exchanger would have to be located by the engine where the radiator is placed in current rig configuration. The JW-HEX could be installed on the genset skid. The JW-HEX was designed in the CHEMCAD software environment for full load engine operating conditions, and the water flow rate was automatically calculated to have an exit temperature matching the conventional coolant return temperature from the in-use operation. Figure 22 presents the HHPDE JW-HEX design diagram. Table 21 summarizes the HHPDE JW-

HEX design main parameters. Figure 23 presents heat recovery as a function of engine load. The HHPDE JW-HEX TEMA sheet is included in Appendix C.

99°C
27,736 kg/hr of H2O

Water Temperature Outlet=
Coolant Temperature Outlet
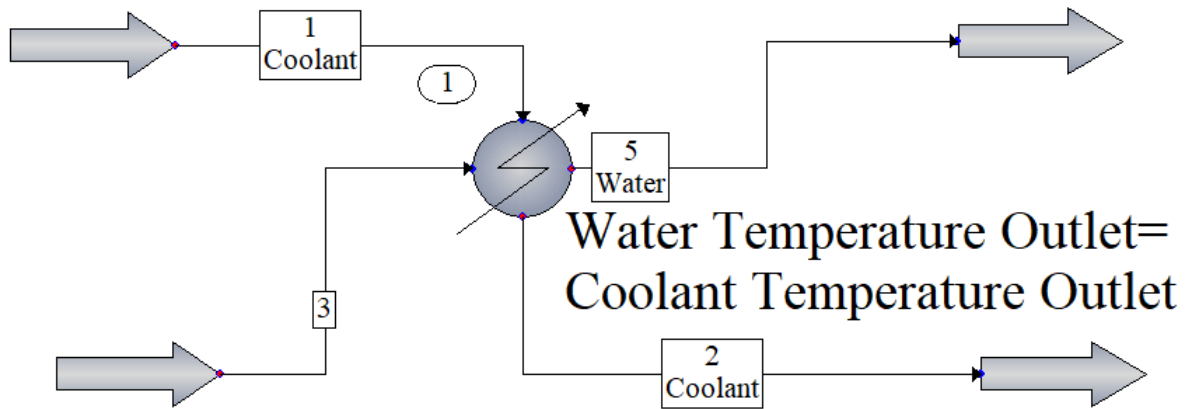
Figure 22. HHPDE JW-HEX Sizing and Design Conditions

Table 21. HHPDE JW-HEX Design

HHPDE JW-HEX	HHPDE JW-HEX
Shell Diameter (m)	0.4
Tube Length (m)	4.9
Number of Tubes	192
Effective Transfer Area (m²)	52.5
U (W/m²°C)	394.3
Tube O.D. (cm)	1.91
Tube I.D. (cm)	1.56
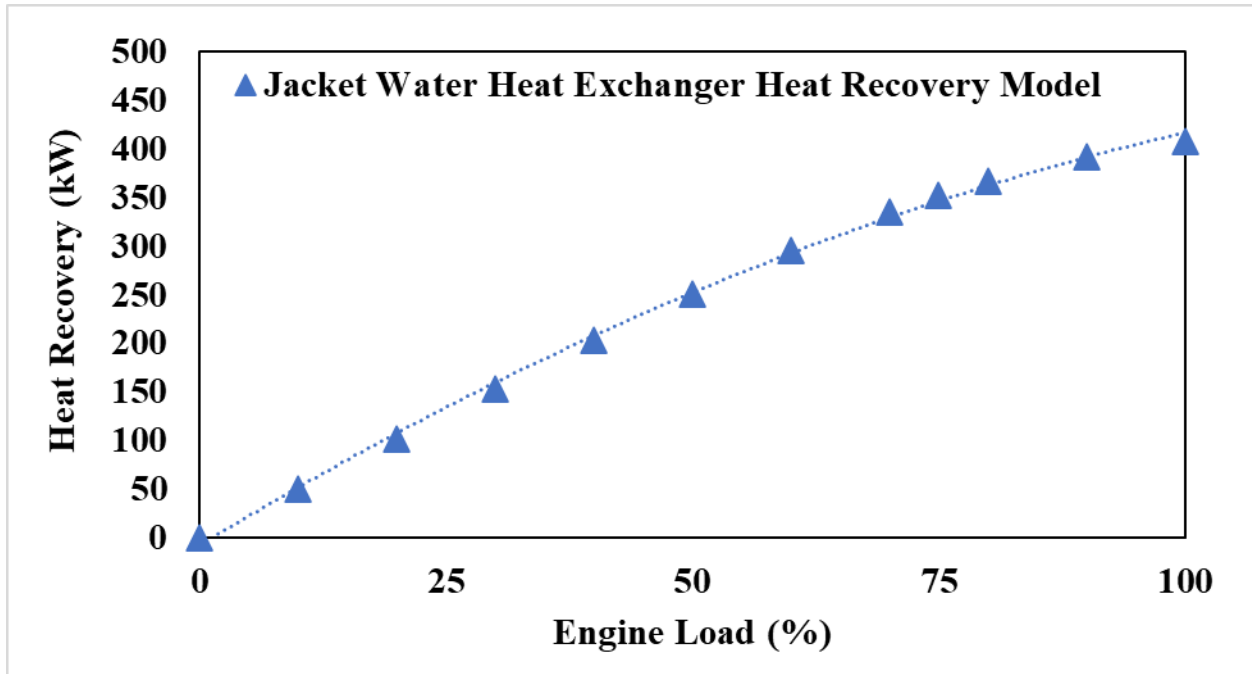
Figure 23. HHPDE JW-HEX Heat Recovery Model

The HHPDE CHP system was simulated again, including both the HHPDE E-HEX and the HHPDE JW-HEX. The HHDE JW-HEX subsystem contains the JW-HEX heat recovery model shown in Figure 23. Figure 24 presents the 0-D HHPDE CHP model with the HHPDE JW-HEX subsystem. Table 22 shows HHPDE CHP system results. Heat recovery over all four analyzed cycles was exceeded by utilizing both E-HEX and JW-HEX designed, where the lowest percentage of heat recovered was 109% of the heat demanded by the rig during the TP1HR cycle using both HHPDE E-HEX and HHPDE JW-HEX. Therefore, enough heat could be captured to eliminate the boiler. Figures 25 through 28 present the instant heat recovery and instant heat demanded by the system throughout the four analyzed cycles.
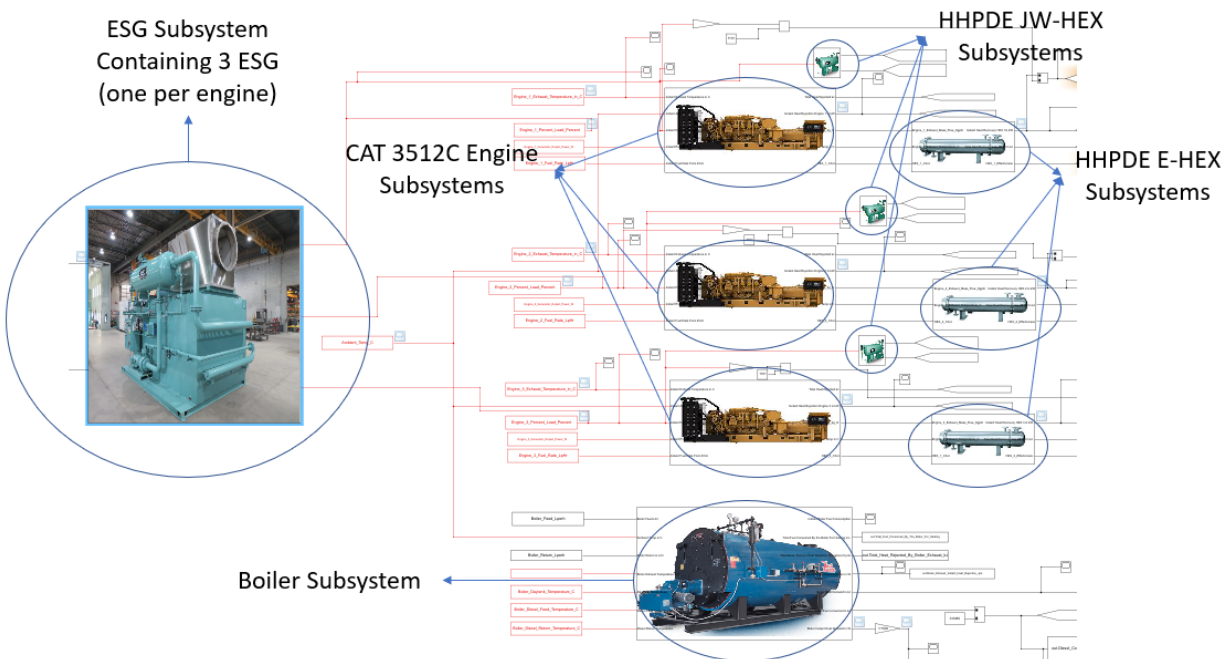
Figure 24. HHPDE CHP System 0-D Model Equipped with HHPDE JW-HEX

Table 22. HHPDE CHP Heat Recovery

Cycle	HHPDE E-HEX+JW-HEX Heat Recovered	HHPDE E-HEX+JW-HEX Heat Recovered-Demanded	HHPDE ESG+JW-HEX Heat Recovered	HHPDE ESG +JW-HEX Heat Recovered Demanded
Name	MJ	%	MJ	%
2E1HR	5494	210	4785	183
3E1HR	5589	239	5411	231
TP1HR	2612	109	3299	138
24HR	106,919	176	110,175	181
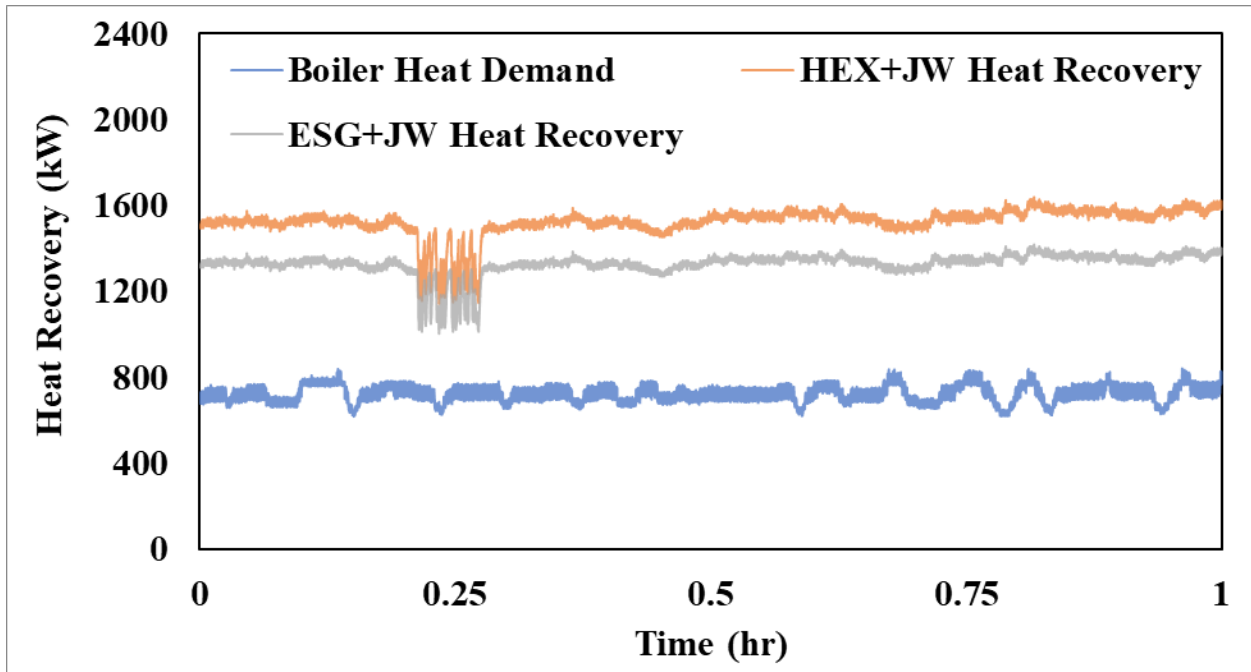
Figure 25. HHPDE CHP System, 2E1HR Cycle Instant Heat Recovery
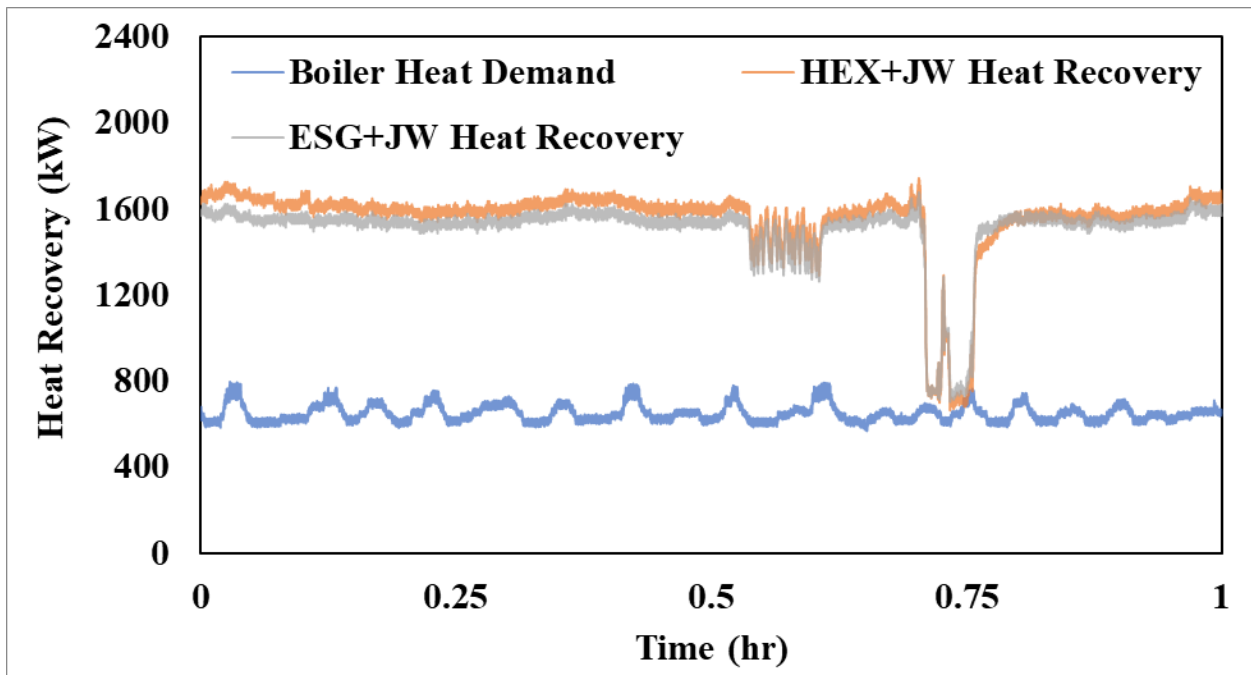
Figure 26. HHPDE CHP System, 31HR Cycle Instant Heat Recovery
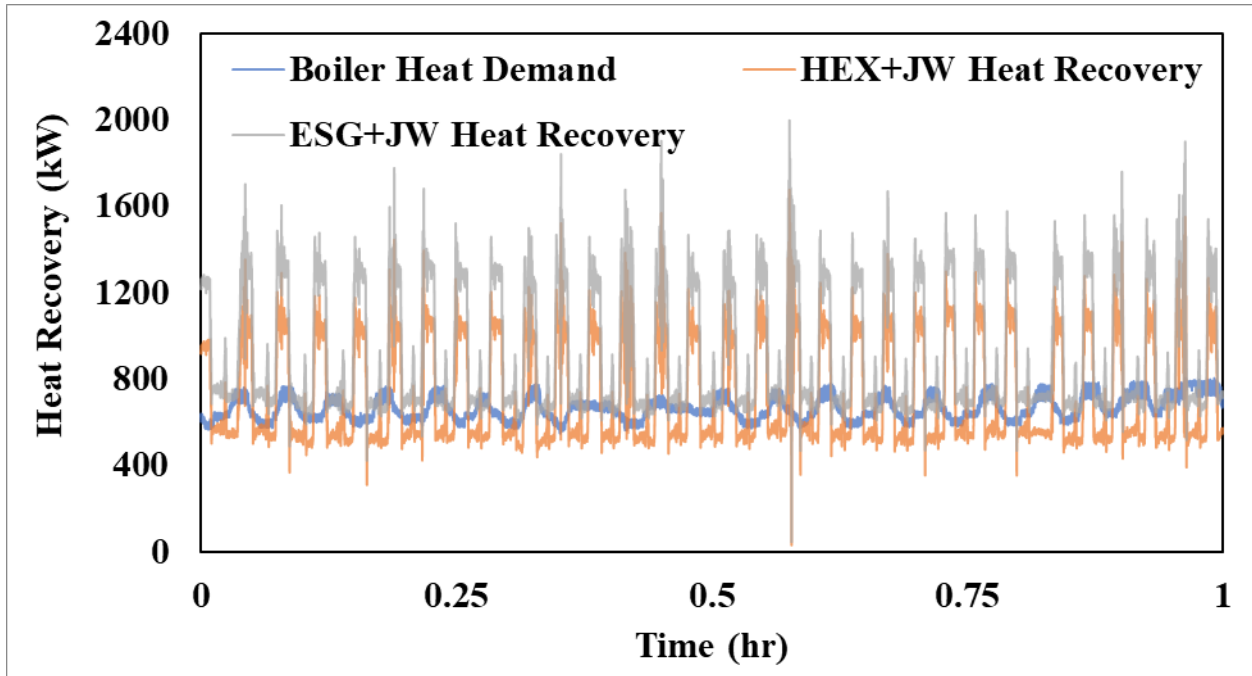
Figure 27. HHPDE CHP System, TP1HR Cycle Instant Heat Recovery
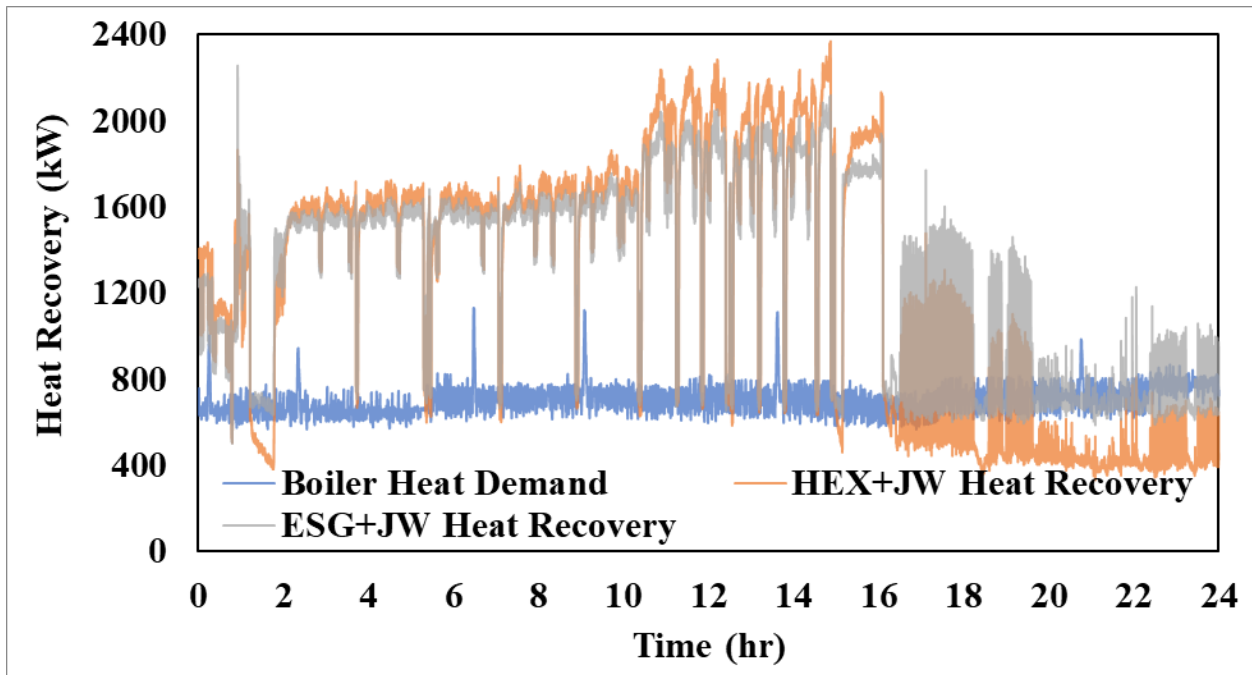
Figure 28. HHPDE CHP System, 24HR Cycle Instant Heat Recovery

Both steady drilling cycles (2E1HR and 3E1HR) recovered enough heat at all times to meet the system's heat demand. However, the TP1HR and the 24HR cycles have periods when instantaneous heat demand was higher than heat recovered. The TP1HR cycle was characterized

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by transient engine operation and thus produced transient heat recovery rates, resulting in insufficient heat demanded for short periods. Average heat demand during the TP1HR cycle was 663 kW, while heat recovered was 726 kW and 916 kW for HHPDE CHP equipped with E-HEX+JW-HEX and ESG+JW-HEX, respectively. The 24HR cycle had a long transient engine operation period at the end of the cycle (from 16 to 24 hrs.). During this section of the cycle, heat recovery failed to recover enough instantaneous heat. However, this portion was highly transient at 1Hz. Therefore, a 30-minute instant heat recovery moving average over the transient operation of the 24HR cycle was created to examine a more appropriate timeframe for the heat analysis. Figure 29 presents that both WHR systems failed to recover enough heat during the last eight hours of engine transient operation of the 24HR cycle. However, this analysis showed that the ESG system would likely yield acceptable operation through hour 20. In this scenario, utilization of the boiler or electric heaters would be necessary to meet heat demands. If electric heaters were used to supply the system with the required heat, engine load would increase, recovering more heat. Another approach to the problem would be to utilize thermal reservoirs to store thermal energy for later usage. An example device could be a steam accumulator. They could help save some of the extra heat recovered from the exchangers to be used when heat demand exceeded heat recovery.
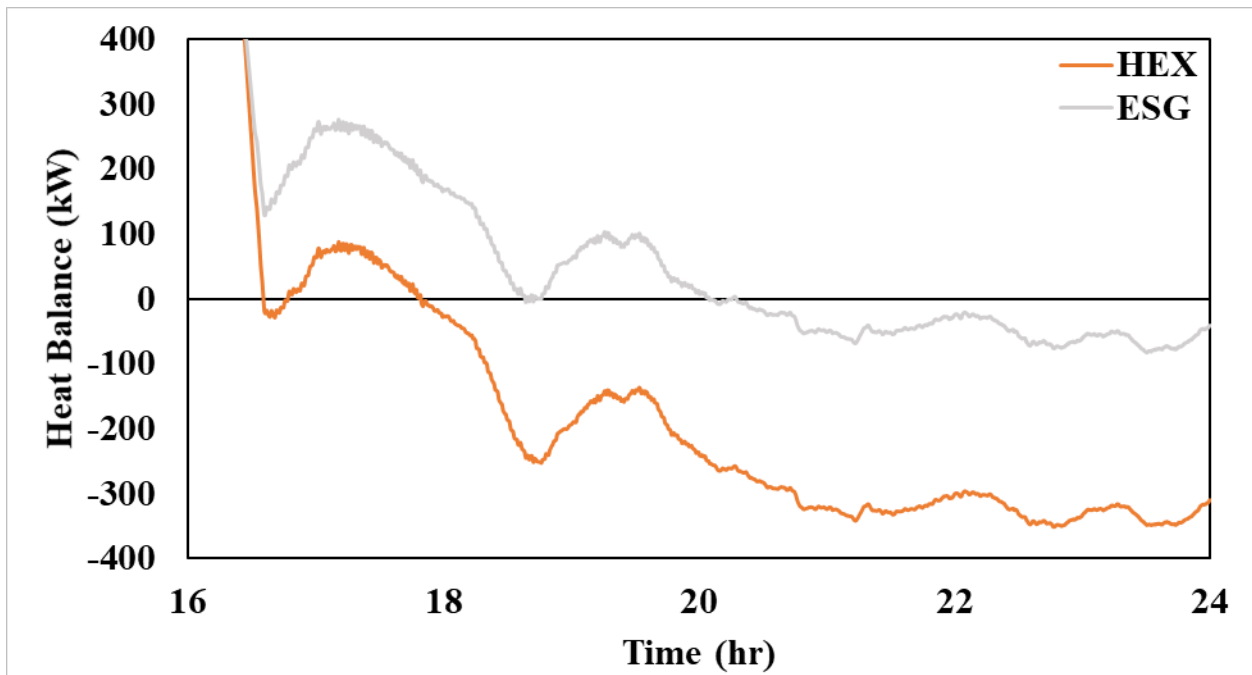
Figure 29. HHPDE CHP System, 24HR Cycle 30-Minute Heat Balance Moving Average

UF was defined as the amount of engine work and recovered heat by the system divided by the fuel's energy throughout the cycle. Table 23 presents a comparison of UFs for the various cycles and HEX combinations. Recovered heat can only be used if the system requires it; otherwise, it is considered lost or potential heat. Note that my data collection did not encompass all cold weather conditions. The highest values presented could be used to estimate performance in colder applications than those that occurred during these data collection efforts.

Table 23. HHPDE CHP UF Analysis

Cycle	HHPDE Without CHP System UF	HHPDE-CHP System UF limited by Heat Required		HHPDE-CHP System Potential UF	
		E-HEX+JW-HEX	ESG+JW-HEX	E-HEX+JW-HEX	ESG+JW-HEX
Name	%	%	%	%	%
2E1HR	38.3	58.8	58.8	81.3	75.7
3E1HR	37.1	53.7	53.7	76.8	75.7
TP1HR	31.8	58.1	58.1	60.6	68.2
24HR	35.7	56.5	56.5	72.3	73.4
Average	35.7	56.8	56.8	72.7	73.2

On average, for the four cycles analyzed, energy's UF could be improved from 35.7% to 56.8%. If potential heat was considered, the UF increased to 72.7% and 73.2% for E-HEX+JW-HEX and ESG+JW-HEX, respectively. Higher UFs translate to less fuel being consumed. Figure 30 shows the HHPDE CHP system diesel fuel savings.
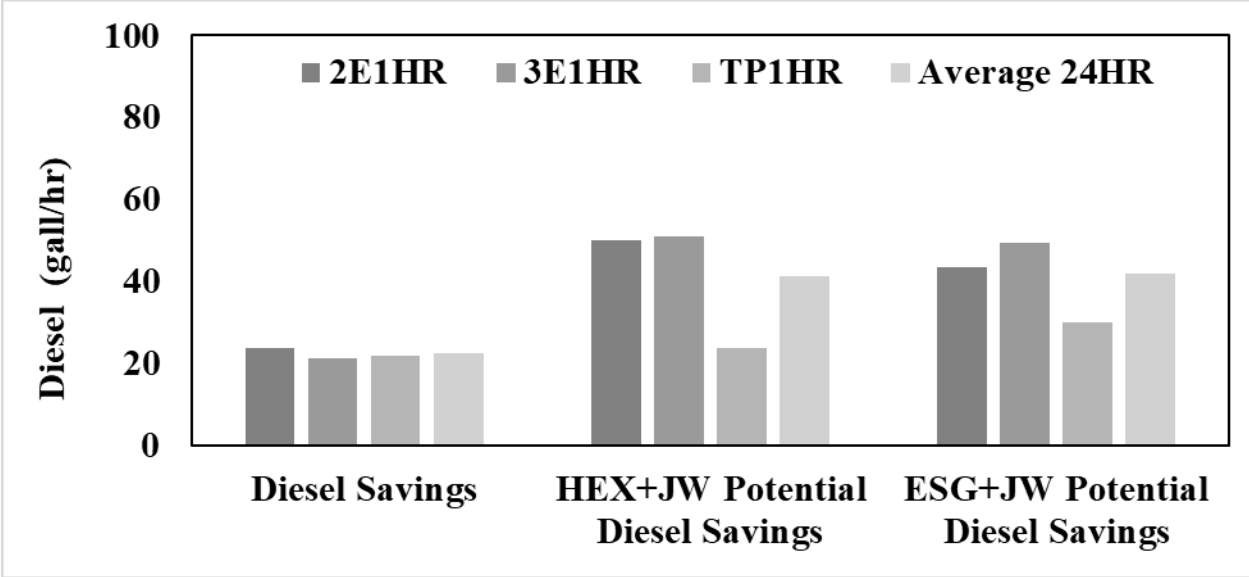
Figure 30. HHPDE CHP System Diesel Fuel Savings

Fuel savings were consistent since the average cycle boiler heat demand ranged between 649.7 kW and 727.5 kW, as presented in Table 15. As shown in Figure 30, potential heat recovery peaked at the two steady drilling cycles (2E1HR and 3E1HR). This is due to their high engine loads, which resulted in large amounts of heat rejection that the E-HEX and JW-HEX recovered. The TP1HR cycle was dominated by low and unsteady engine loads causing a smaller heat recovery potential. Average diesel fuel saved was calculated at 22.5 gph (84.9 L/hr), and potential diesel fuel savings averaged 41.3 gph (156.1 L/hr). Associated CO_2 emissions reductions were estimated at 226.2 kg/hr, and potential emissions savings averaged 418 kg/hr. Assuming a cost of $2.973/gal of diesel fuel, Figure 31 shows the costs savings in U.S. dollars. Average cost savings were estimated at $66.4/hr (20.5% fuel expenses) with potential up to an average of $122.8/hr (36.4% of total diesel fuel expenses).
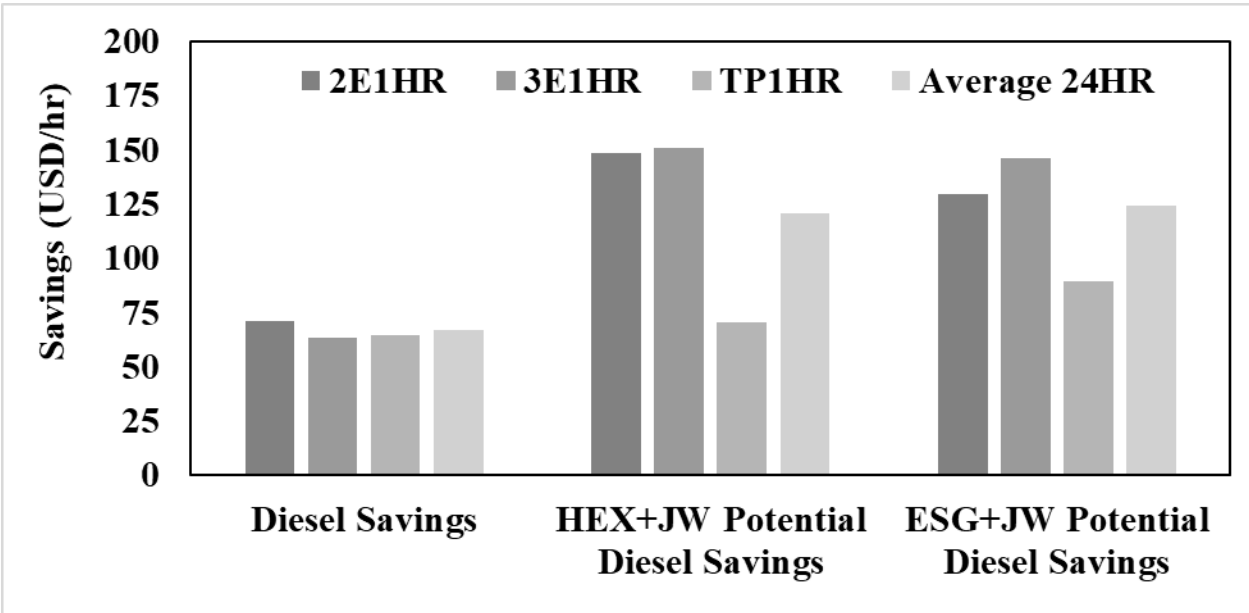
Figure 31. HHPDE CHP Cost Savings

6.3 Energy Saving Model: HHPDE CHP Hybrid

As another method to reduce fossil fuel energy consumption during well development, a HHPDE CHP hybrid model was developed. The HHPDE CHP hybrid system was equipped with the HEMS. This system was intended to reduce the number of required gensets by increasing the average load of those operating, causing them to have a higher thermal efficiency. Figure 32 shows CAT 3512C engine thermal efficiency as specified by the manufacturer [32].
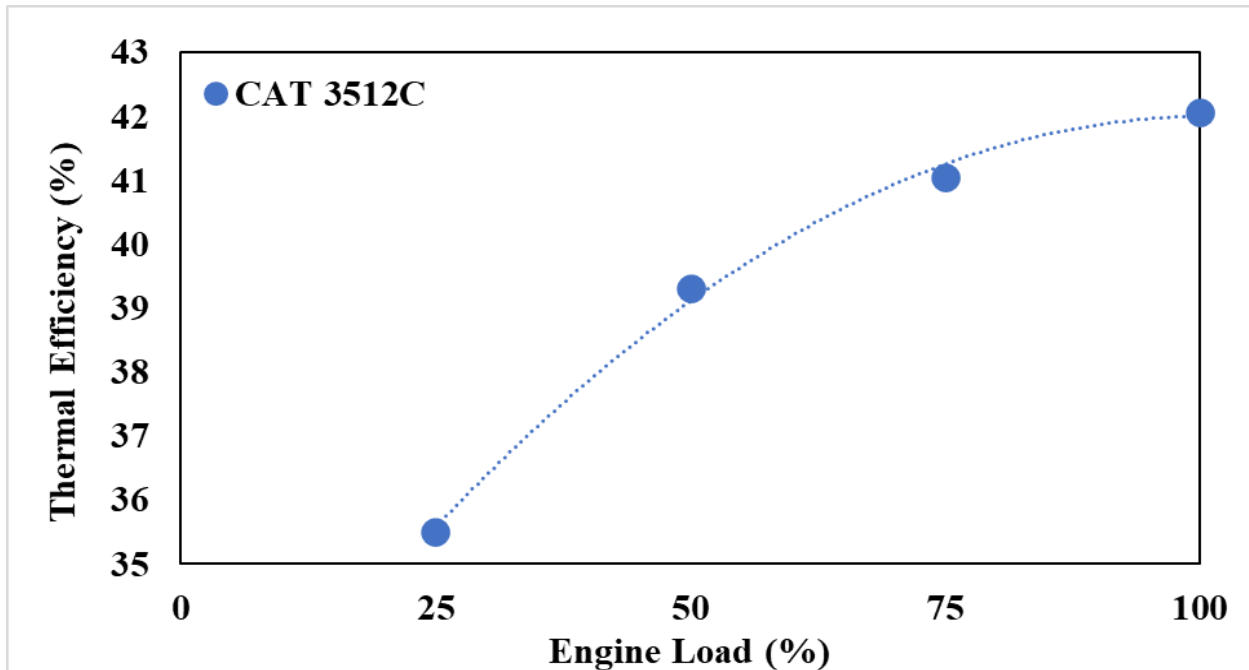
Figure 32. CAT 3512C Thermal Efficiency

A 0-D SIMULINK model was designed for this configuration, and it is presented in Figure 33. The SIMULINK model for the HHPDE CHP hybrid system model specifications is attached in Appendix E and included:

- CAT 3512C engine subsystem
  - Calculated exhaust heat rejection utilizing Equation 3
    - Exhaust mass flow rate model as a function of engine load
      - Dry air mass flow rate
      - Water vapor flow rate
    - Exhaust gas temperature modeled as a function of engine load
  - Fuel consumption modeled as a function of engine load
  - Power output assuming constant generator efficiency of 95%
- HHPDE E-HEX subsystem
  - Calculated exhaust heat recovery from CHEMCAD developed two-entry table, dry air, and water vapor mass exhaust flow rate from CAT 3512C engine subsystem and exhaust temperature models
- HHPDE JW-HEX
  - HHPDE JW-HEX heat recovery model as a function of engine load

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- HHPDE ESG subsystem
  - Calculated exhaust heat recovery from the exchanger performance provided by the manufacturer and available in Appendix D.
- HEMS subsystem
  - 70% initial SOC assumption
  - Minimum number of online gensets possible
  - Instant heat feedback control
  - Genset load control system
  - Charging and discharging coulombic efficiency of 80% and 90%, respectively
- Boiler subsystem
  - Fuel consumption (diesel) recorded by the KRAL flow meters
    - Calculated heat supplied to the system from fuel consumption assuming constant efficiency of 81%
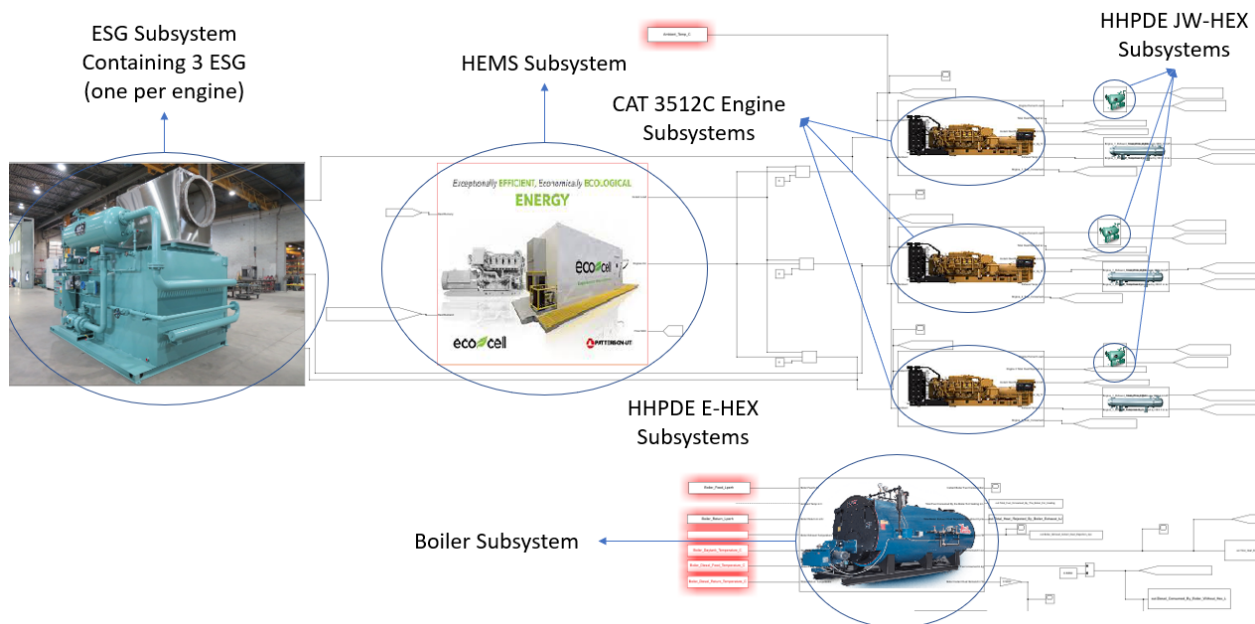
Figure 33. HHPDE CHP Hybrid System 0-D SIMULIK Model Diagram

By running fewer gensets at higher loads, it was expected to obtain higher UF and possibly a tradeoff with potential heat recovery. Table 24 presents the heat recovery results for the HHPDE CHP model equipped with the HEMS. Heat recovery for the HHPDE CHP equipped with the HEMS was sufficient to supply the system's heat demand for all cycles. Figures 34 through 37

show instant heat recovery for the HHPDE CHP hybrid system for the four cycles examined in this paper.

Table 24. HHPDE CHP Hybrid Heat Recovery

Cycle	HHPDE Hybrid E-HEX+JW-HEX Heat Recovered	HHPDE Hybrid E-HEX+JW-HEX Heat Recovered-Demanded	HHPDE Hybrid ESG+JW-HEX Heat Recovered	HHPDE Hybrid ESG +JW-HEX Heat Recovered-Demanded
Name	MJ	%	MJ	%
2E1HR	5175	198	4899	187
3E1HR	5209	223	4931	211
TP1HR	2834	119	2650	111
24HR	105,344	173	103,337	170
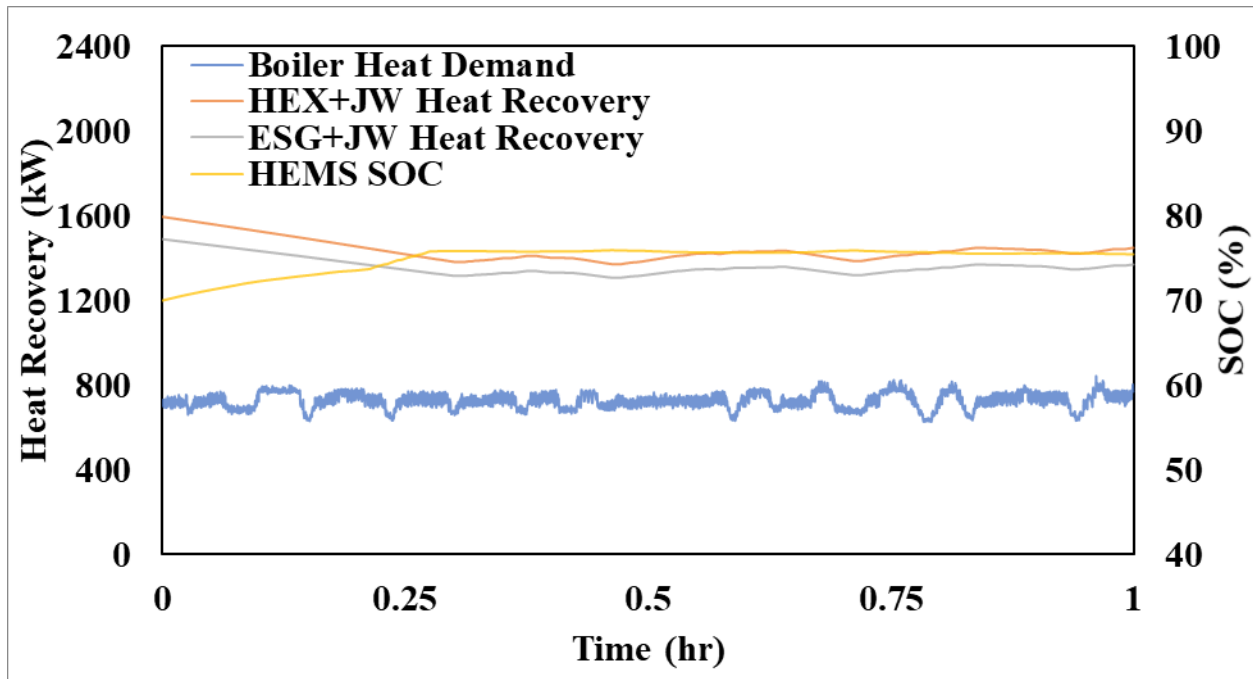
Figure 34. HHPDE CHP Hybrid System, 2E1HR Cycle Instant Heat Recovery
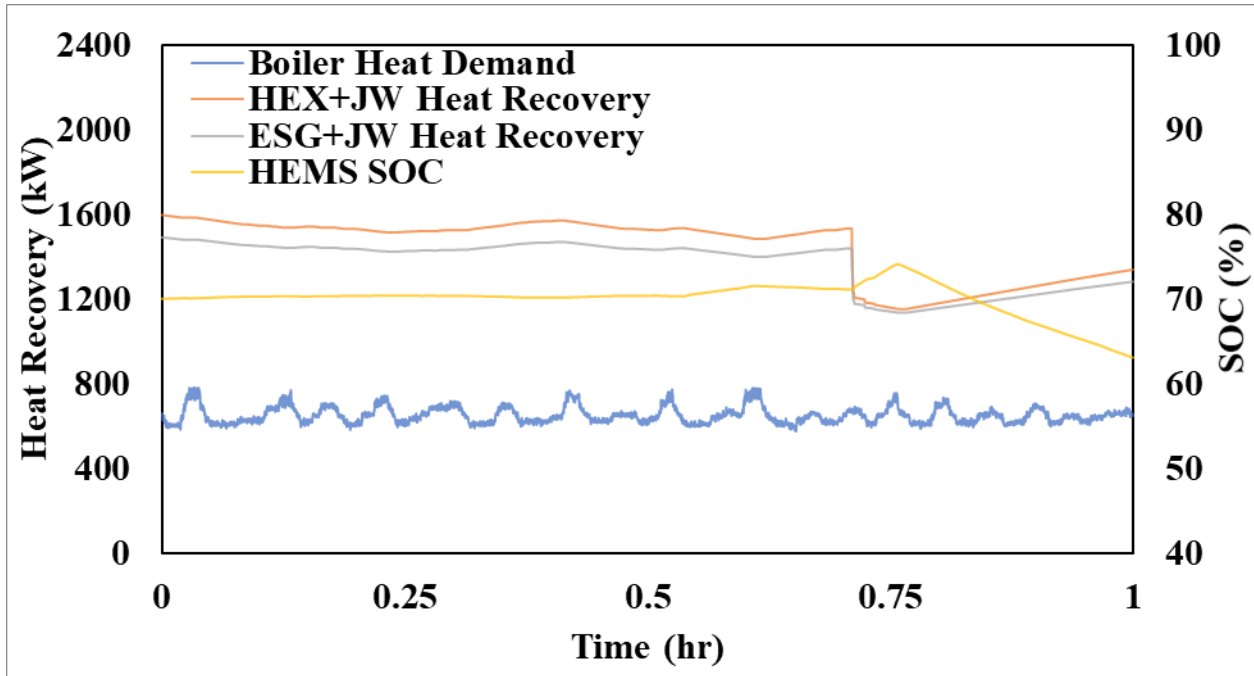
Figure 35. HHPDE CHP Hybrid System, 3E1HR Cycle Instant Heat Recovery
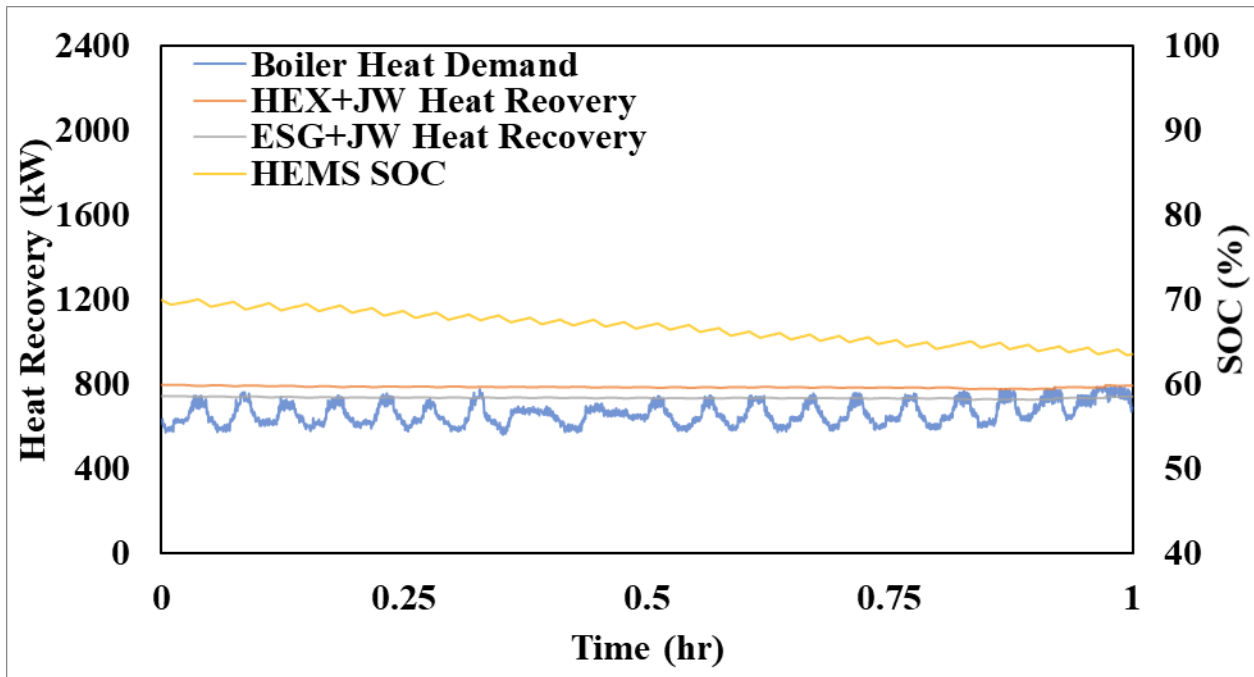
Figure 36. HHPDE CHP Hybrid System, TP1HR Cycle Instant Heat Recovery
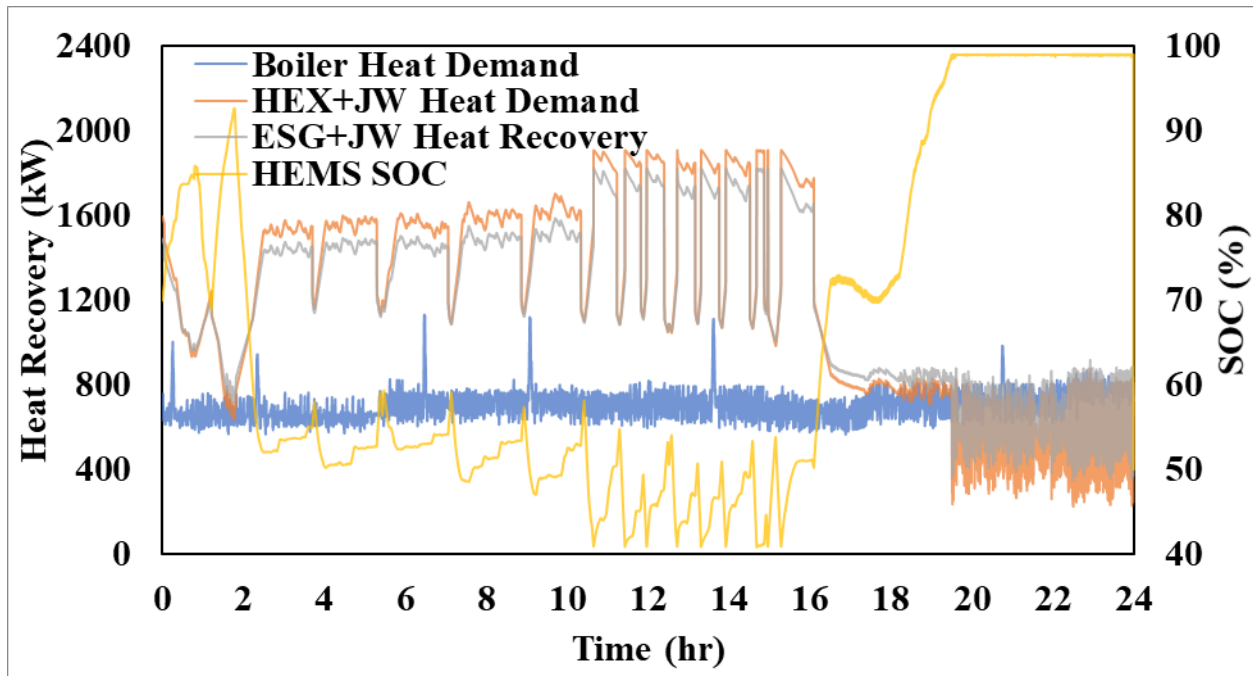
Figure 37. HHPDE CHP Hybrid System, 24HR Cycle Instant Heat Recovery

Instant heat recovered by the HHPDE CHP equipped with the HEMS was sufficient to meet demands through the 2E1HR, 3E1HR, and TP1HR cycles. However, as observed for the non-hybrid system, instant heat recovered for the transient section of the 24HR cycle was not enough to surpass the heat demanded. Figure 38 presents the HHPDE CHP system equipped with the HEMS 30-minute instant heat recovery moving average for the last eight hours of the 24HR cycle. In this case, both systems failed to meet the heat demand during the last ~4 hours.
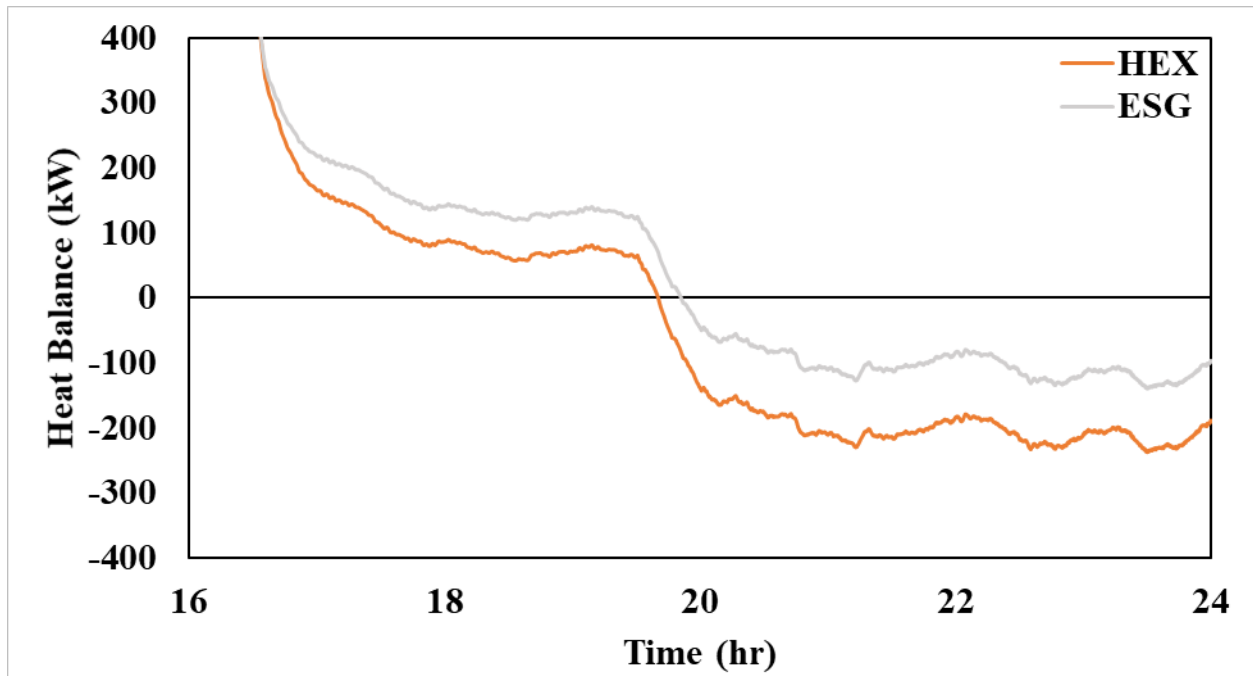
Figure 38. HHPDE CHP Hybrid System, 24HR Cycle 30-Minute Heat Balance Moving Average

During the eight-hour transient section of the 24HR cycle, heat recovery did not meet instant heat demand. However, compared to the non-hybrid system (Figure 29), the HHPDE CHP equipped with a HEMS recovered sufficient heat for approximately two hours longer than the non-hybrid system. Therefore, a more complex heat feedback control could further improve instant heat recovery. The goal of this paper was to assess potential energy utilization in terms of UF and not design a HEMS, but it proves that an energy management system could be a fuel-saving accessory for land drilling engines. Table 25 presents the UF analysis for this system configuration.

Table 25. HHPDE CHP Hybrid UF Analysis

Cycle	HHPDE Hybrid Without CHP System UF	HHPDE-CHP Hybrid System UF limited by Heat Required		HHPDE-CHP Hybrid System Potential UF	
		E-HEX+JW-HEX	ESG+JW-HEX	E-HEX+JW-HEX	ESG+J W-HEX
Name	%	%	%	%	%
2E1HR	37.3	57.2	57.2	76.7	74.6
3E1HR	38.8	56.2	56.2	77.5	75.4
TP1HR	39.2	71.6	71.6	77.7	75.2
24HR	37.6	59.5	59.5	75.6	74.8
Average	38.2	61.1	61.1	76.9	75.0

Average UF on the engines only was increased from 35.7% to 38.2% by using the HEMS, where the cycle that experienced the most significant improvement was the TP1HR cycle. The hybrid system reduced the number of gensets required for three of the four cases analyzed in this research. The number of active engines for the TP1HR cycle, reduced from three online gensets to one. The cycles 3E1HR and 24HR reduced from three online gensets to two. Lastly, the 2E1HR cycle did not reduce the number of engines online. The objective of the HEMS was to have the fewer possible gensets online at a high steady load. The cycles characterized by a high-steady-load did not provide a significant improvement; however, highly transient cycles showed an improvement in UF terms. The TP1HR cycle had the most significant improvement, from 31.8% to 39.2%. The UF of the HHPDE CHP system from 56.8% to 61.1% on average when equipped with the HEMS. The average potential was improved from 72.7% and 73.2% without the HEMS to 76.9% and 75.0% for HHPDE E-HEX+JW-HEX and HHPDE ESG+JW-HEX configurations when equipped with the HEMS. Higher UFs with the same energy demand resulted in higher fuel savings. Average diesel fuel saved was calculated at 27.3 gph (103.1 L/hr), and potential diesel fuel savings averaged 44.9 gph (169.6 L/hr). Associated CO2 emissions reductions were estimated at 276.2 kg/hr, and potential emissions savings averaged 454 kg/hr. Figure 39 presents the HHPDE CHP hybrid system diesel fuel savings. Savings were estimated on an average of $81.1 per hour (27.3% of total fuel expenses) and potential of $133.4 an hour (42.9% of fuel expenses). Figure 40 shows savings per cycle in an hourly rate.
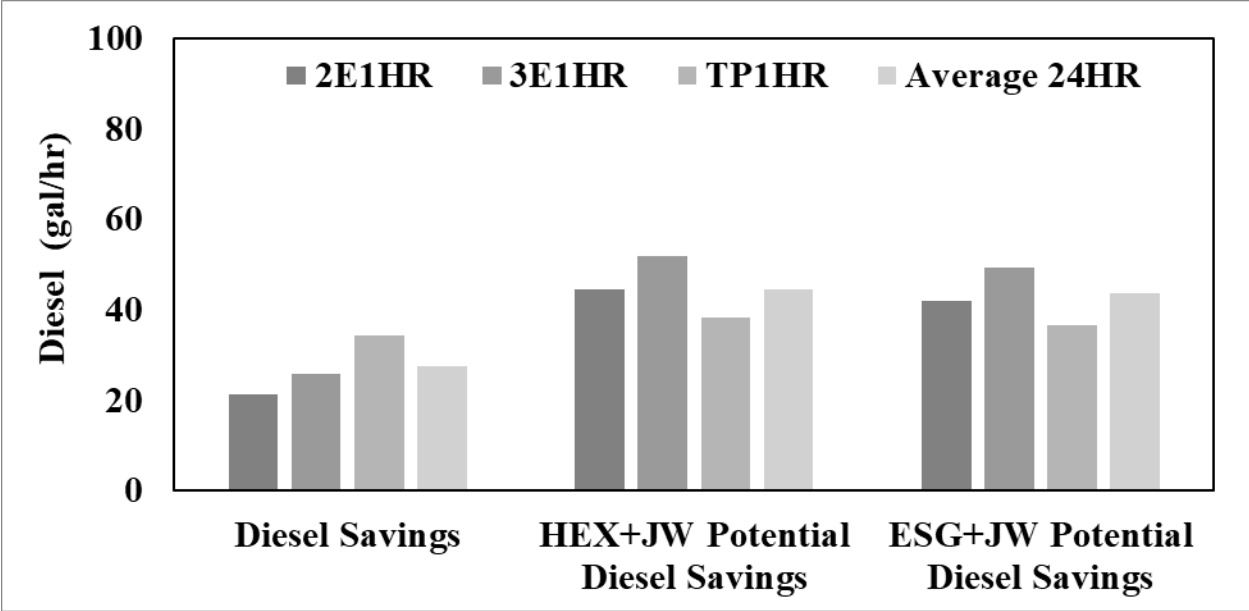
Figure 39. HHPDE CHP Hybrid System Diesel Fuel Savings
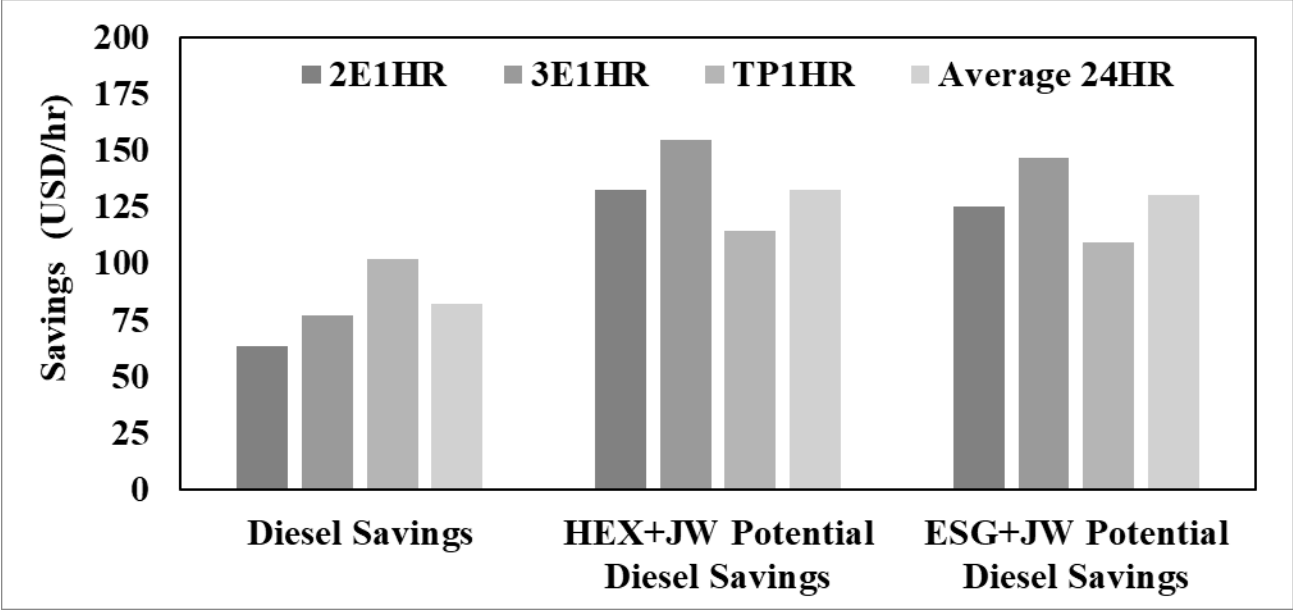
Figure 40. HHPDE CHP Hybrid System Cost Savings

A potential solution to the lack of recovered heat during the last section of the 24HR cycle could be the utilization of electric heaters throughout the rig using stored energy in the HEMS. In order to estimate the feasibility of such a configuration, a simple heater module was developed and implemented in the SIMULINK HHPDE CHP hybrid model. For the simulation, it was assumed that the electric heaters are 100% efficient. The electric heater load control was developed to increase its load when the heat balance (heat demanded - heat recovered) approached zero over the

period of a minute. Figure 41 shows the instant heat recovery over the 24HR cycle for the HHPDE CHP hybrid model using electric heaters powered by the HEMS. Note that the electric heater load control was a simple model with the only objective of testing its feasibility. A more advanced control system could be developed obtaining higher UF simulations. The potential UF decreased from 76.9% using the E-HEX and JW-HEX to 73.7% with the same configuration and utilizing the electric heaters. For the ESG and JW-HEX configuration the potential UF decreased from 75% to 72.7%. While the UF decreased, this approach highlights that HEMs as currently configured had stored more than enough energy to meet the total energy demand (heat + generator power + stored electrical energy). Another possible approach for HEMS stored energy utilization could be the usage of air conditioners when rigs are located in warm weather conditions.
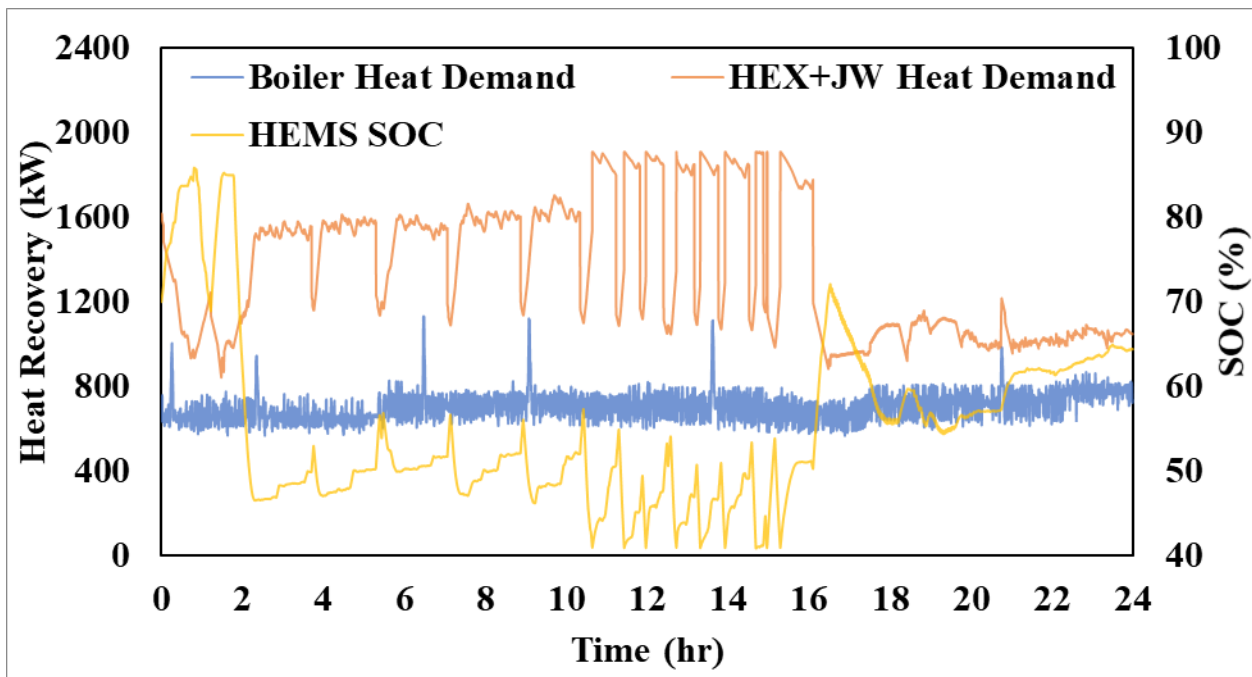
Figure 41. HHPDE CHP Hybrid System, 24HR Cycle Instant Heat Recovery Utilizing Electric Heaters

6.4 Energy Saving Model: DNGE CHP

To assess possible fuel savings on natural gas powered rigs, a DNGE CHP system was also modeled and simulated. The same four cycles were used, and the number of DNGE online gensets was selected to match the HHPDE online gensets amount, although DNGE is rated at higher brake power. A DNGE CHP 0-D SIMULINK model was constructed, and it is presented in Figure 42.

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Table 26 presents the DNGE CHP heat recovery rates. The SIMULINK model for the DNGE CHP system model specifications is attached in Appendix E and included:

- Waukesha L7044GSI engine subsystem
    - Calculated exhaust heat rejection utilizing Equation 3
        - Exhaust mass flow rate from the model as a function of engine load
            - Dry air mass flow rate
            - Water vapor mass flow rate
        - Exhaust temperature model as a function of engine load
    - Fuel consumption as a function of engine load
    - Power output assuming constant generator efficiency of 95%
- DNGE E-HEX subsystem
    - Calculated exhaust heat recovery as a function of engine load
- DNGE ESG subsystem
    - Calculated exhaust heat recovery from ESG manufacturer data provided and available in Appendix D
- Boiler subsystem
    - Fuel consumption (natural gas)
        - Converted from diesel to natural gas utilizing Equation 2
        - Calculated heat supplied to the system from fuel consumption assuming constant efficiency of 81%

Figure 42. DNGE CHP System 0-D SIMULINK Model Diagram

Table 26. DNGE CHP Heat Recovery

Cycle	DNGE E-HEX Heat Recovered	DNGE E-HEX Heat Recovered-Demanded	DNGE ESG Heat Recovered	DNGE ESG Heat Recovered-Demanded
Name	MJ	%	MJ	%
2E1HR	4081	156	4047	155
3E1HR	4368	187	4816	206
TP1HR	2404	101	2905	122
24HR	87,079	143	96,413	159

Heat recovery for the DNGE CHP system exceeded the heat required by the system with exhaust HEX only. Both DNGE E-HEX and DNGE ESG performance were simulated. Enough heat was recovered to replace the boiler; therefore, no DNGE JW-HEX was designed and simulated for the DNGE CHP system. The lowest amount of heat recovered occurred during the TP1HR cycle, where 101 % of the heat demanded by the rig was recovered by the DNGE E-HEX. For the DNGE ESG configuration, the TP1HR cycle was also the cycle that recovered the least heat (122% of the

heat demanded). On the other end, the 3E1HR cycle recovered 187% and 206% of the heat demanded for the DNGE E-HEX and DNGE ESG configurations, respectively. Figures 43 through 46 present DNGE CHP instant heat recovery.

Figure 43. DNGE CHP System, 2E1HR Cycle Instant Heat Recovery

Figure 44. DNGE CHP System, 3E1HR Cycle Instant Heat Recovery

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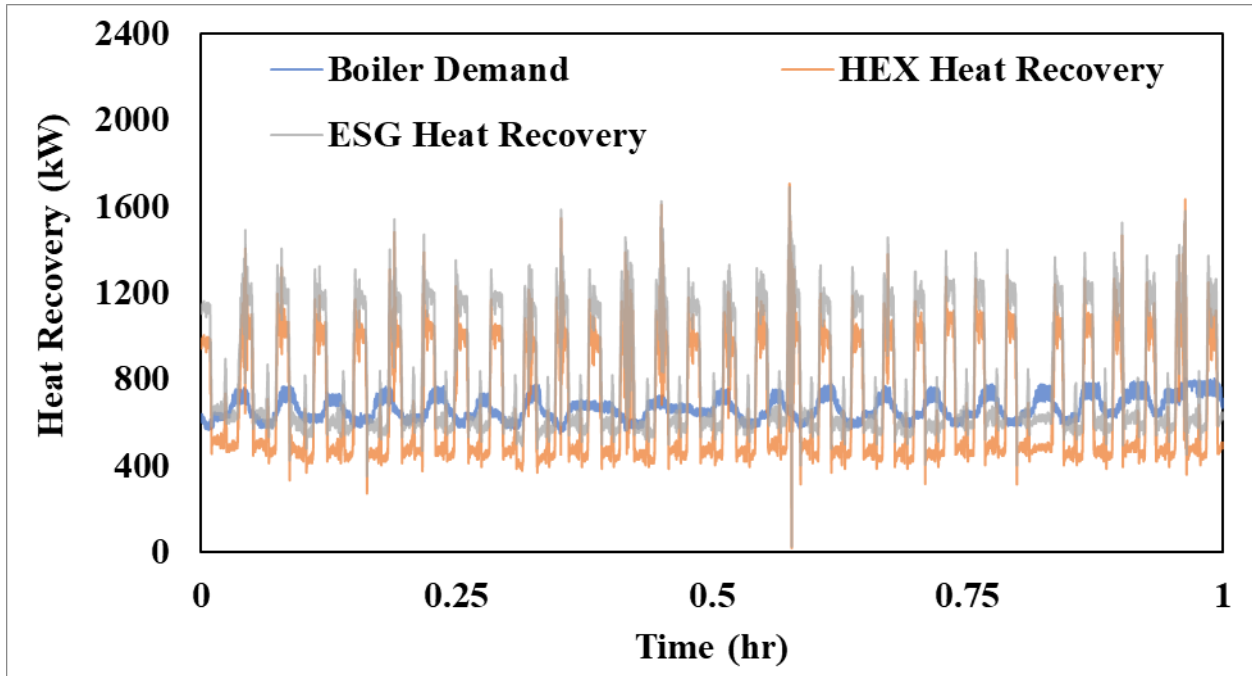
Figure 45. DNGE CHP System, TP1HR Cycle Instant Heat Recovery
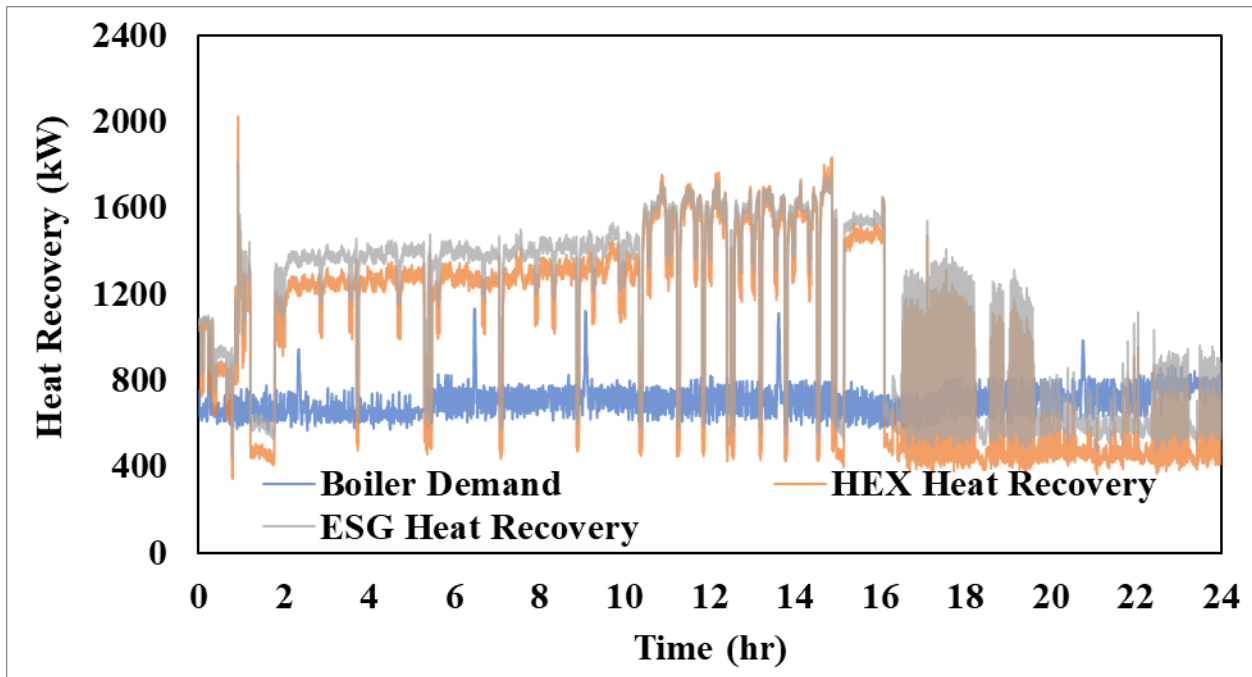
Figure 46. DNGE CHP System, 24HR Cycle Instant Heat Recovery

The DNGE CHP system recovered sufficient heat during the 2E1HR and 3E1HR cycles to surpass the heat demanded throughout the entire cycle. The TP1HR cycle recovered heat at unsteady rates due to transient engine operation, resulting in periods where it did not recover demanded heat. However, the average demanded heat was estimated to be 663 kW, while heat recovery averaged

668 kW and 807 kW for the DNGE E-HEX and DNGE ESG WHR systems. Neither the E-HEX nor ESG systems could recover enough instant heat to meet the heat demanded during the transient section of the 24HR cycle. To better present heat balance over this period in the cycle, a 30-minute instant heat balance moving average was used. Figure 47 presents this curve for the DNGE CHP system. The ESG system behaved similar as in the HHPDE CHP system but was unable to meet heat demand during the last ~6 hours as opposed to the last ~4 hours.
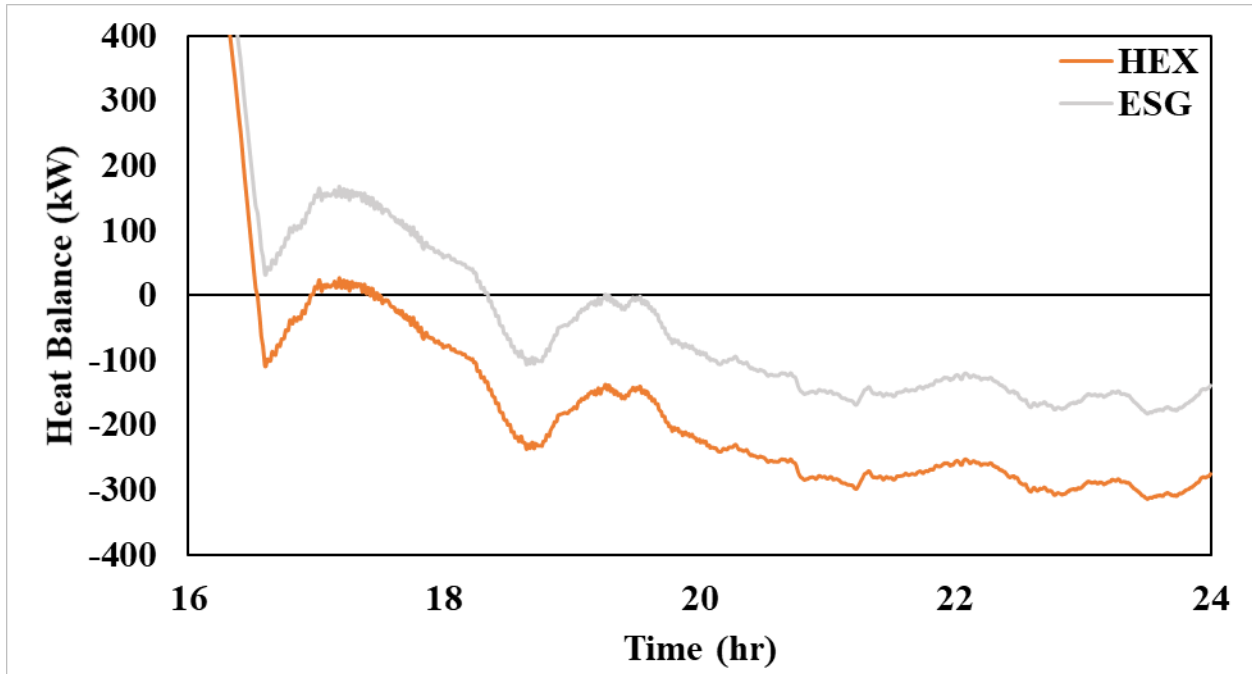
Figure 47. DNGE CHP System, 24HR Cycle 30-Minute Heat Balance Moving Average

As occurred for the HHPDE CHP system, heat recovered over the last eight-hour period of the 24HR cycle was insufficient. Other sources of heat would be necessary to meet the heat demand of the rig. On average, UF was improved from 19.0% to 30.3%. Potential average UF could increase to 34.9% and 36.4% for DNGE E-HEX and DNGE ESG, respectively. Table 27 shows the UF analysis. Figure 48 shows the DNGE CHP system fuel savings. Fuel savings were measured in diesel gallon equivalent (dge) for consistency. Assuming diesel fuel LHV used by CAT in their technical sheet for the 3512C engine, a gallon of diesel gallon equivalent was calculated in 9494 kJ/dge.

Table 27. DNGE CHP UF Analysis

Cycle	DNGE Without CHP System UF	DNGE-CHP System UF Limited by Heat Required		DNGE-CHP System Potential UF	
		E-HEX	ESG	E-HEX	ESG
Name	%	%	%	%	%
2E1HR	20.6	31.6	31.6	37.8	37.6
3E1HR	19.0	27.4	27.4	34.8	36.4
TP1HR	17.6	32.2	32.2	32.3	35.4
24HR	18.9	29.9	29.9	34.6	36.3
Average	19.0	30.3	30.3	34.9	36.4
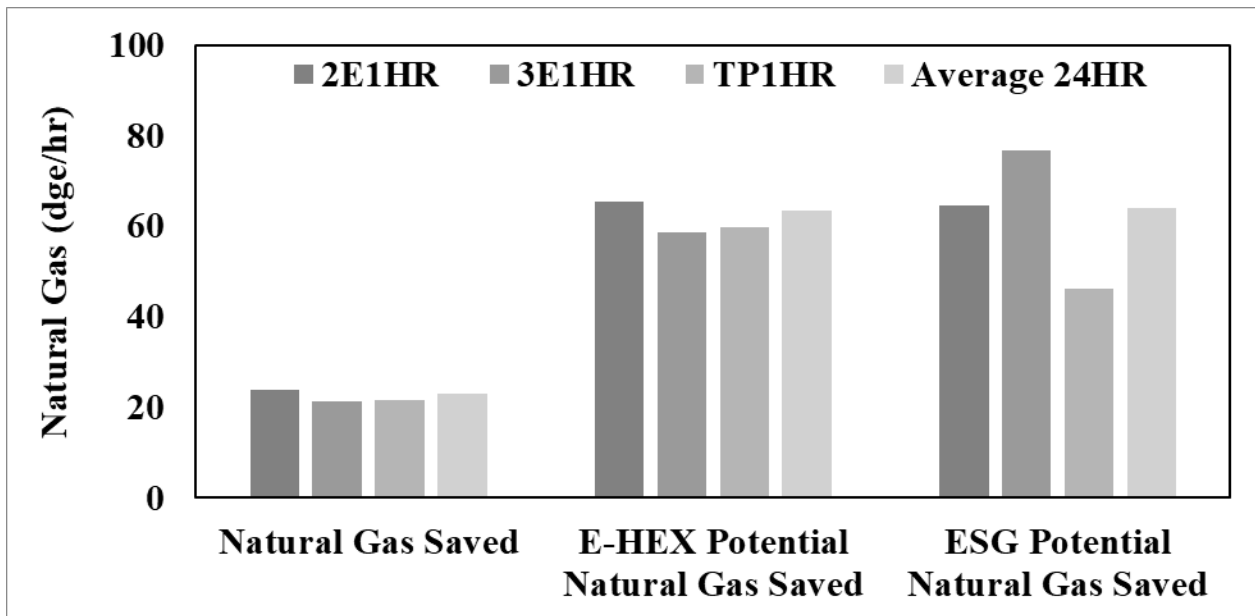
Figure 48. DNGE CHP System Natural Gas Savings

Like the HHPDE CHP system, fuel savings were obtained by eliminating the boiler use. Potential heat recovery peaked during the 2E1HR and 3E1HR cycles, where loads were high and steady throughout the cycle. The lowest heat recovery potential was found on the TP1HR cycle, where low and transient loads dominated engine activities. Average natural gas fuel saved was calculated at 22.5 dge/hr (61.8 kg/hr), and potential diesel fuel savings averaged 61.8 dge/hr (158.9) kg/hr. Associated CO_2 emissions reductions were estimated at 170 kg/hr, and potential emissions savings

averaged 437 kg/hr. Figure 49 shows costs savings assuming a cost of $0.17 per kg of natural gas. Average fuel costs savings were estimated at $10.5/hr (12.2% of fuel costs) with a potential average of $27.0/hr (30.2% of total fuel costs).
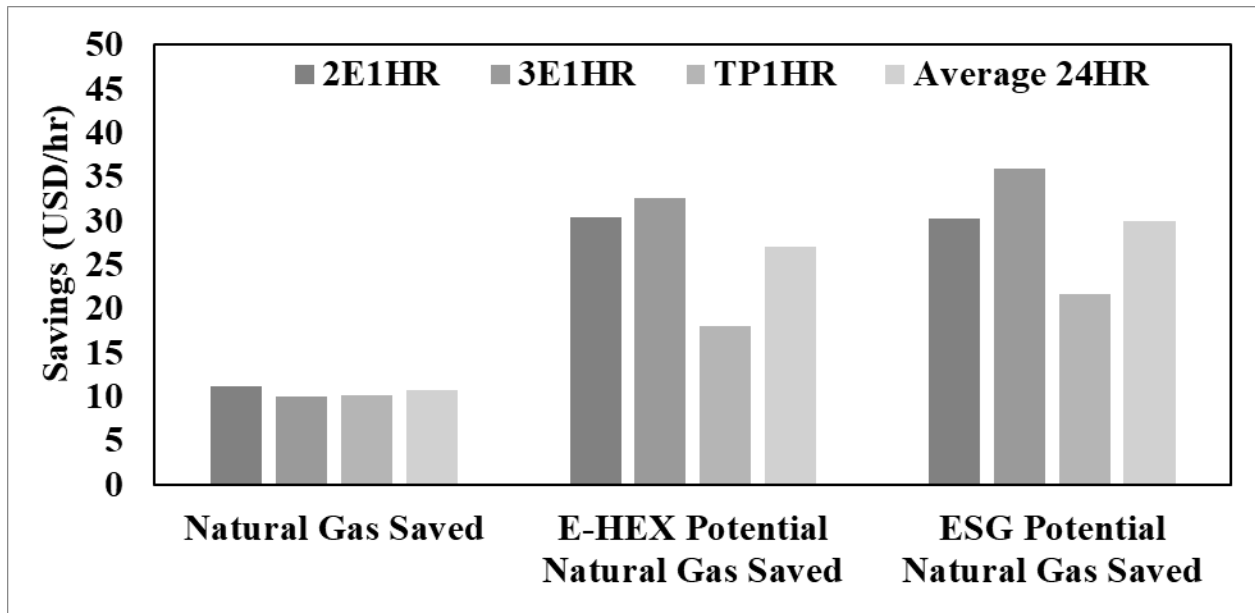
Figure 49. DNGE CHP System Costs Savings

6.5 Energy Saving Model: DNGE CHP Hybrid

The DNGE CHP hybrid system with the specified HEMS was also simulated and analyzed for the four cycles to further improvement in potential fuel savings. The HEMS can output electrical power up to 1752 kWe. Higher loads represent higher thermal efficiency for the DNGEs. Figure 50 shows Waukesha L7044GSI natural gas engine thermal efficiency as a function of engine load. However, previously recorded data showed that on-site thermal efficiency was lower [46]. The same HEMS, parameters, and assumptions as the HHPDE CHP hybrid system were used for the DNGE CHP hybrid system.
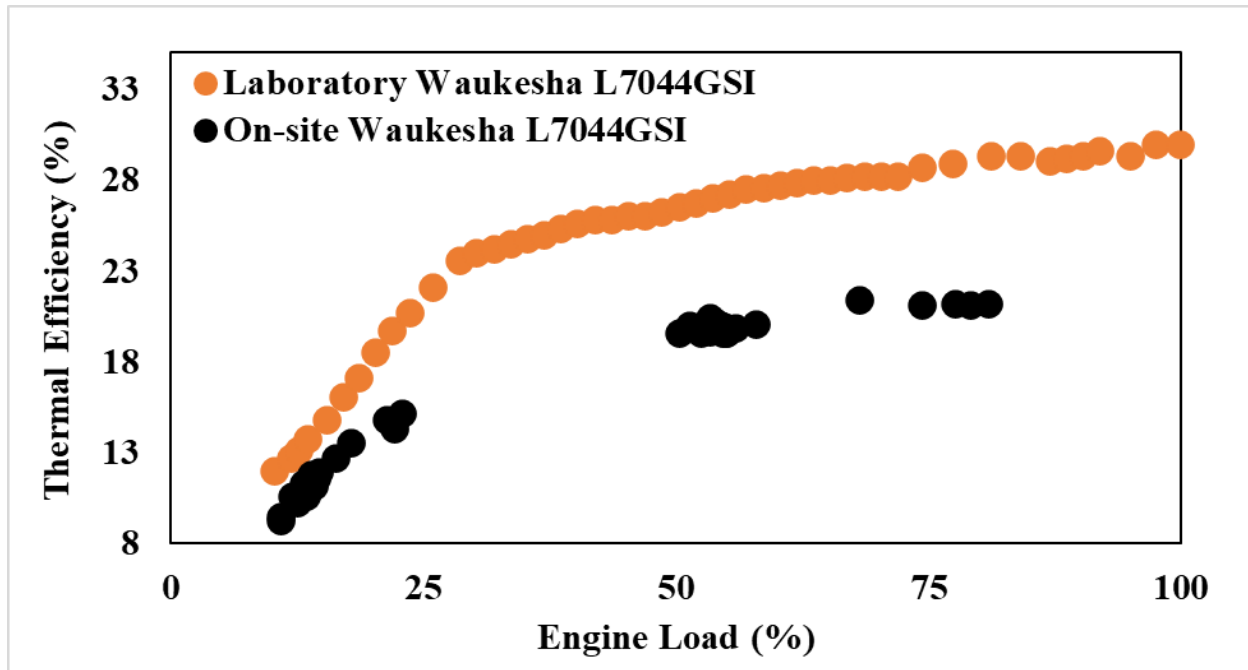
Figure 50. Waukesha L7044gsi Engine Thermal Efficiency

A 0-D SIMULINK model was designed for this configuration, and it is presented in Figure 51. Table 28 shows the heat recovery results for the DNGE CHP system equipped with the HEMS. The SIMULINK model for the DNGE CHP hybrid system model specifications is attached in Appendix E and included:

- Waukesha L7044GSI engine subsystem
  - Calculated exhaust heat rejection utilizing Equation 3
    - Exhaust mass flow rate from the model as a function of engine load
      - Dry air mass flow rate
      - Water vapor mass flow rate
    - Exhaust temperature model as a function of engine load
  - Fuel consumption as a function of engine load
  - Power output assuming constant generator efficiency of 95%
- DNGE E-HEX subsystem
  - Calculated exhaust heat recovery as a function of engine load
- DNGE ESG subsystem
- Calculated exhaust heat recovery from ESG manufacturer data provided and available in Appendix D

- o 70% initial SOC assumption

- o Minimum number of online gensets possible

- o Instant heat feedback control

- o Genset load control system

- o Charging and discharging coulombic efficiency of 80% and 90%, respectively

- Boiler subsystem

  - o Fuel consumption (natural gas)

    - ▪ Converted from diesel to natural gas utilizing Equation 2

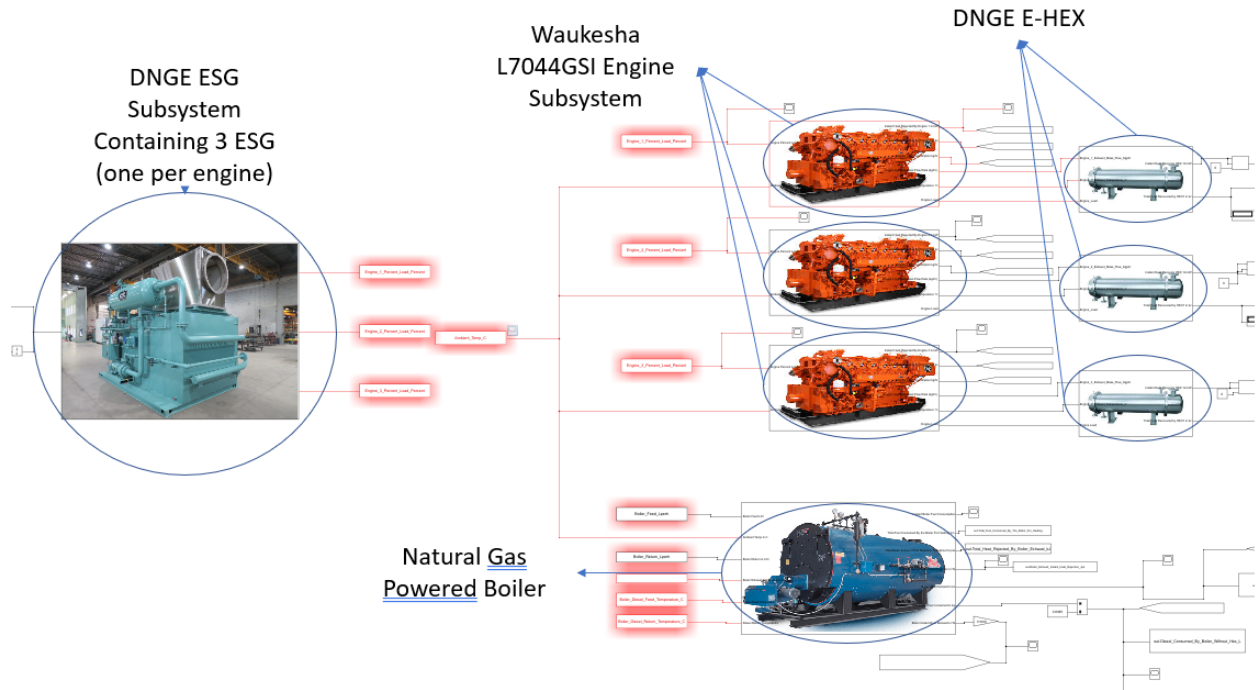   - ▪ Calculated heat supplied to the system from fuel consumption assuming constant efficiency of 81%

Figure 51. DNGE CHP Hybrid System 0-D SIMULINK Model Diagram

Table 28. DNGE CHP Hybrid System Heat Recovery

Cycle	DNGE CHP Hybrid System E-HEX Heat Recovered	DNGE CHP Hybrid System E-HEX Heat Recovered-Demanded	DNGE CHP Hybrid ESG Heat Recovered	DNGE CHP Hybrid ESG Heat Recovered-Demanded
Name	MJ	%	MJ	%
2E1HR	4366	167	4194	160
3E1HR	4378	187	4186	179
TP1HR	2735	115	2338	98
24HR	88,474	146	85,772	141

Reducing the number of gensets online leads to a reduction in fuel consumption; however, it can also reduce heat recovery in some cases. Cycle TP1HR for DNGE CHP hybrid system equipped with the DNGE ESG was not able to recover 100% of the heat required by the system (98%). However, this exchanger was used to compare the designed exchanger (DNGE E-HEX), which exceeded heat demands. In scenarios where the required heat was not able to be recovered, there are two possible solutions. A second engine can be fired to contribute with heat and power cogeneration, or electric heaters can be used in offices. Figures 52 through 55 present the instant heat recovery for each of the four cycles analyzed.
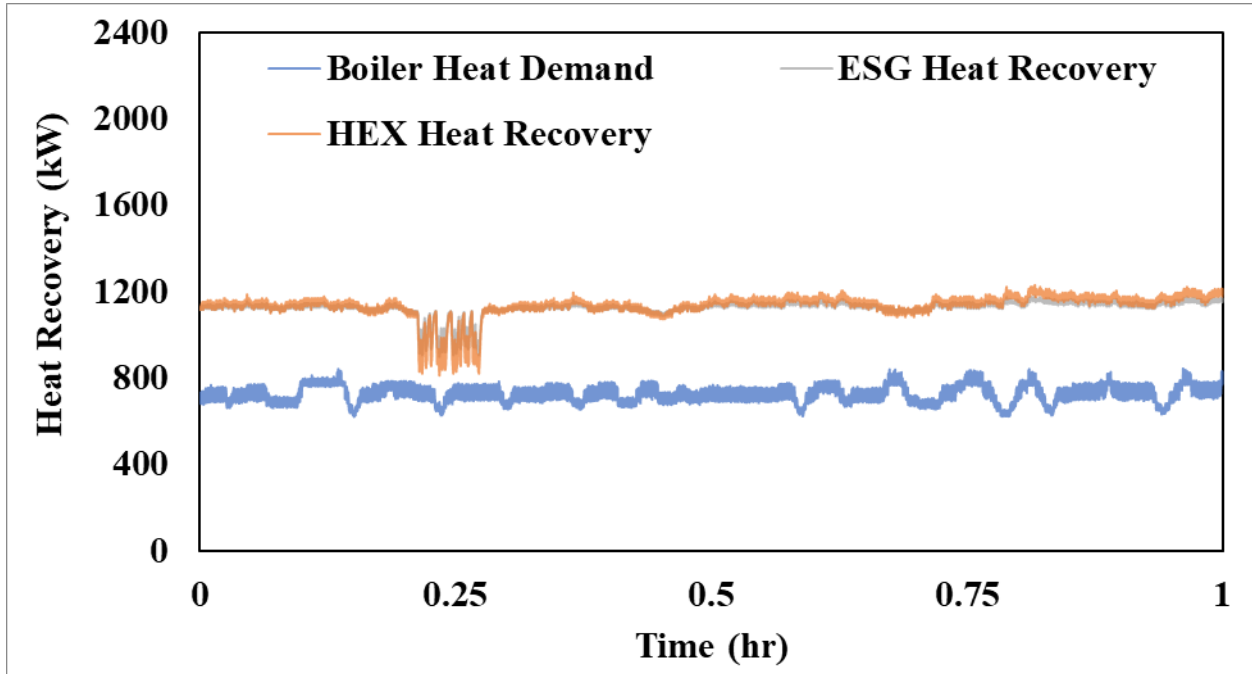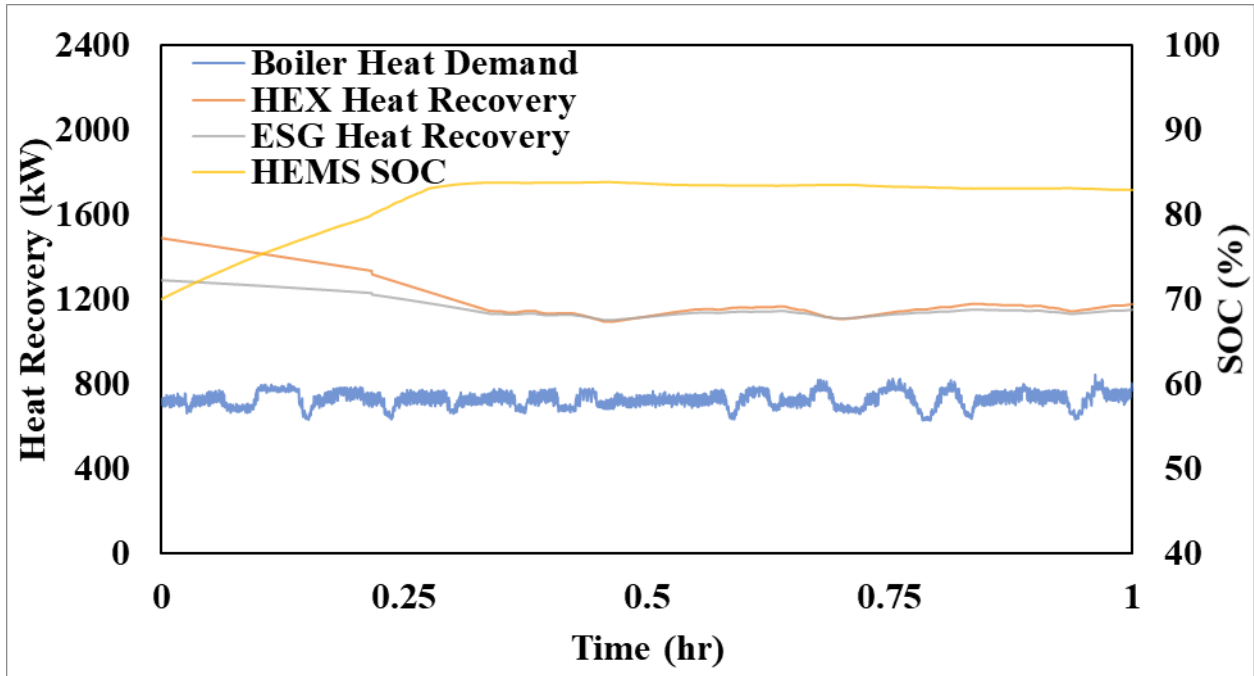
Figure 52. DNGE CHP Hybrid System, 2E1HR Cycle Instant Heat Recovery
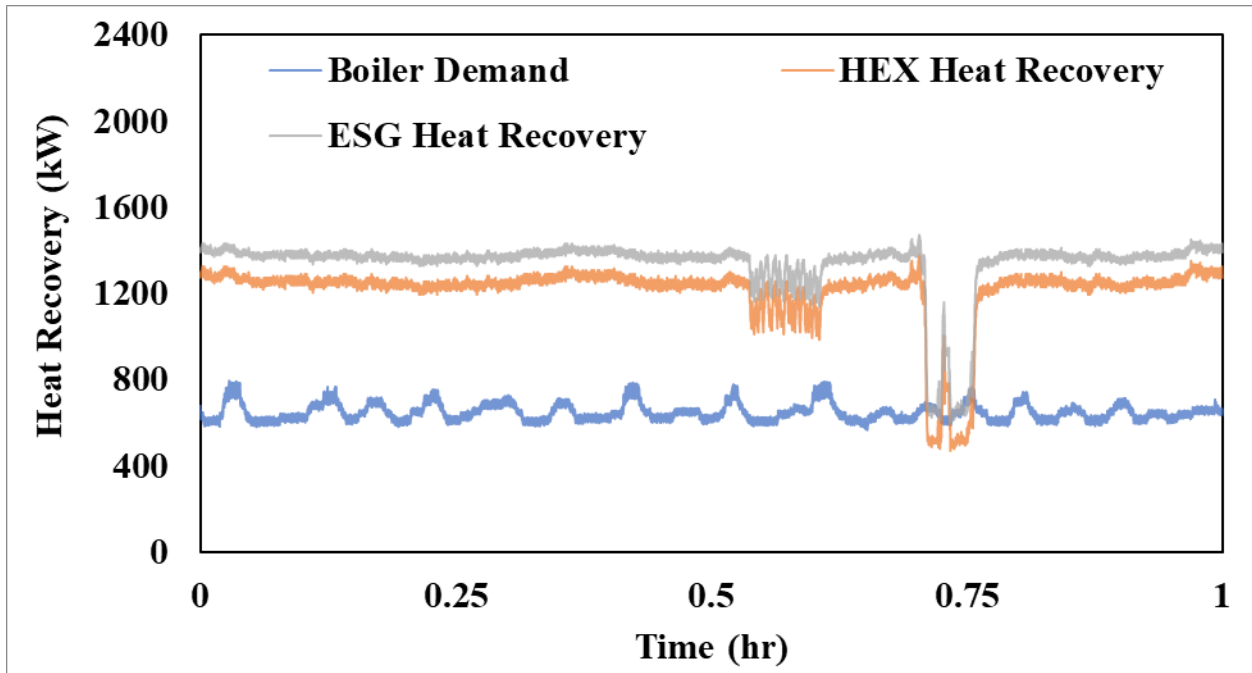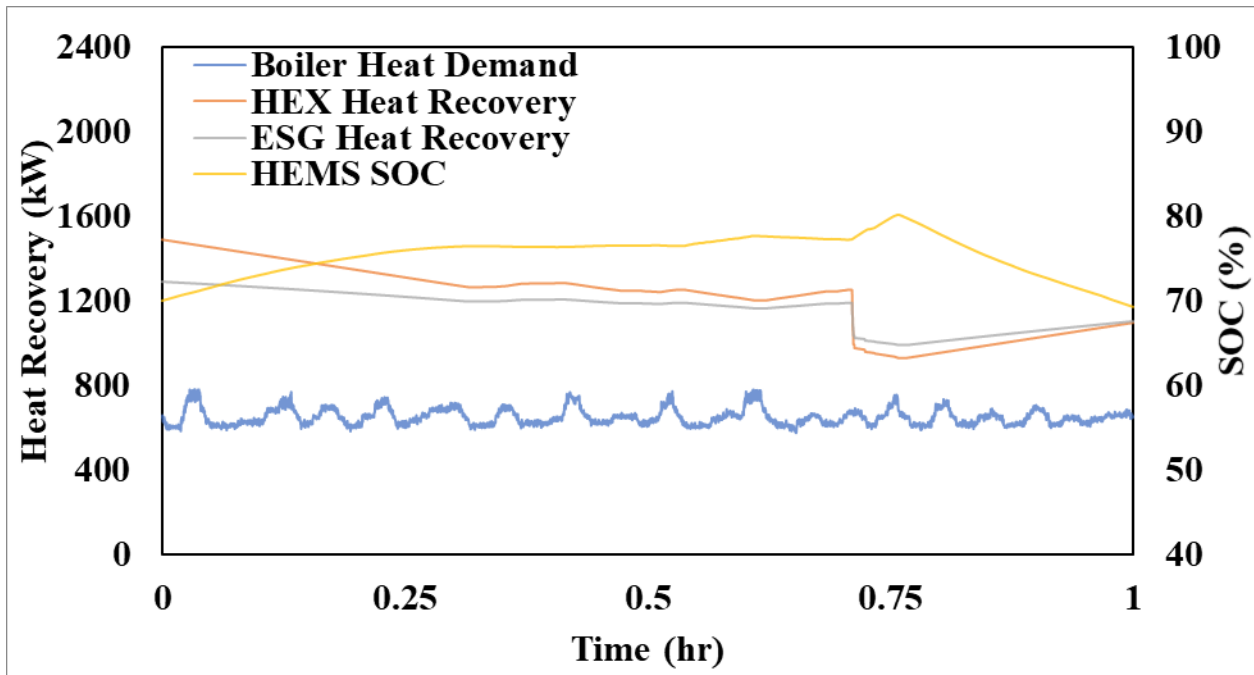

Figure 53. DNGE CHP Hybrid System, 3E1HR Cycle Instant Heat Recovery
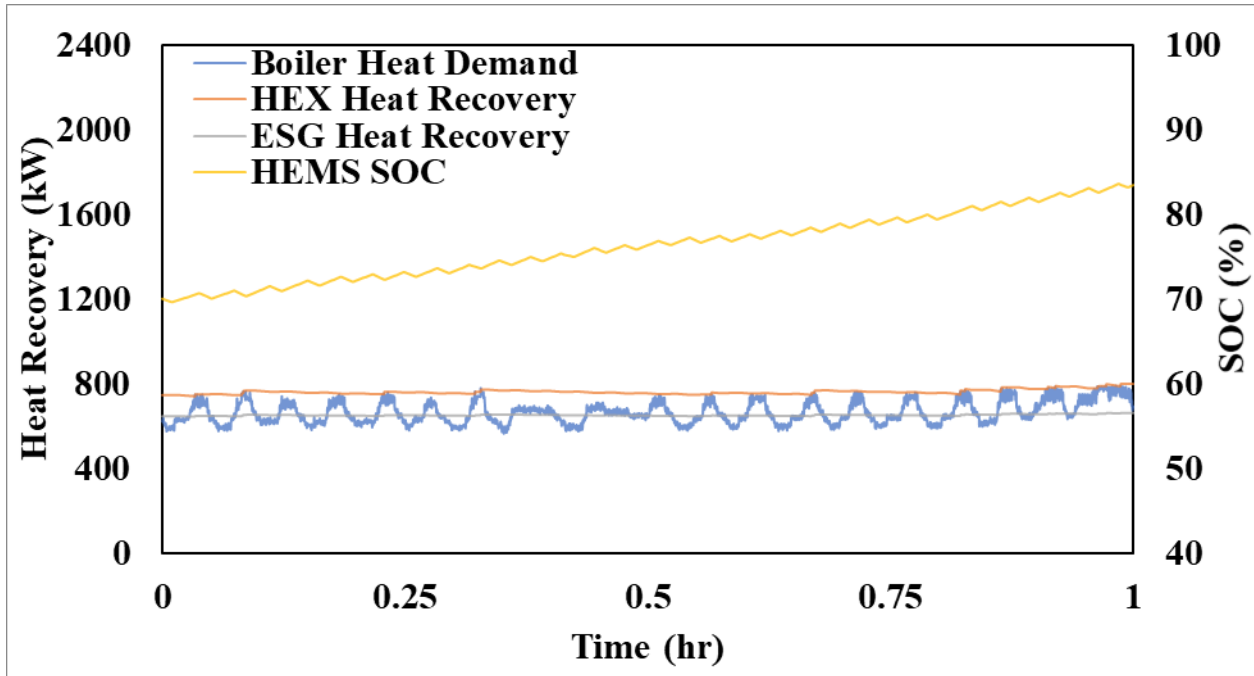
Figure 54. DNGE CHP Hybrid System, TP1HR Cycle Instant Heat Recovery
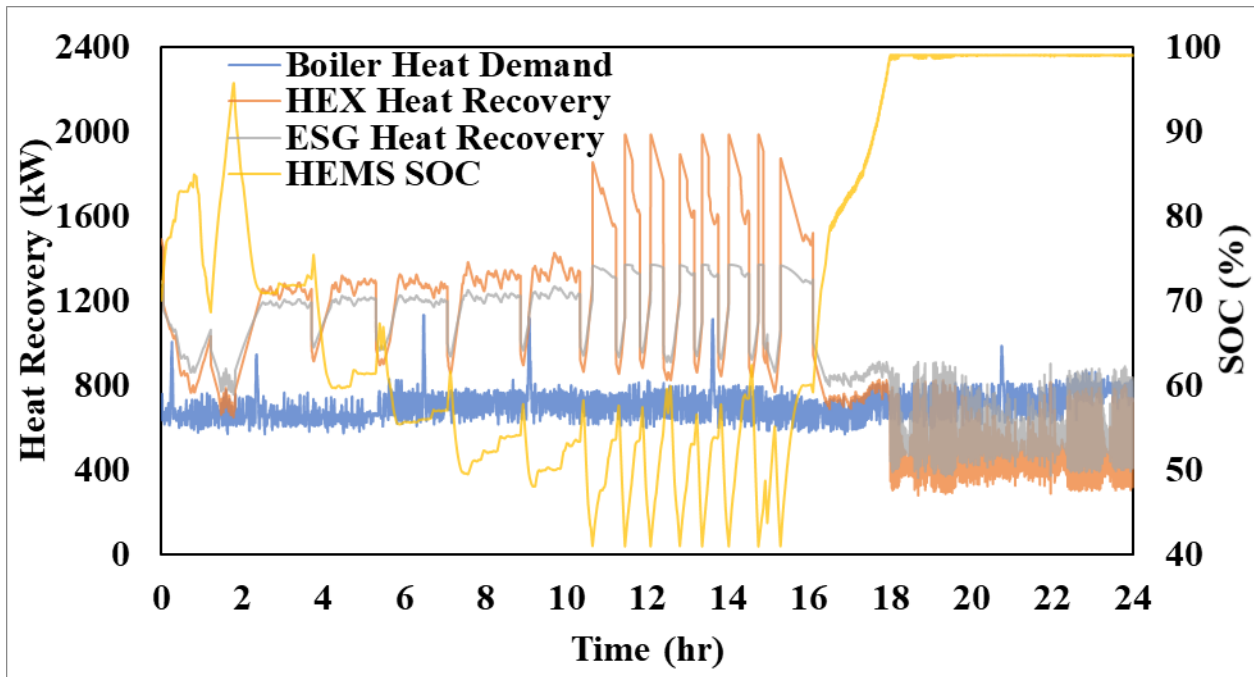
Figure 55. DNGE CHP Hybrid System, 24HR Cycle Instant Heat Recovery

The DNGE CHP system equipped with the HEMS recovered sufficient heat to meet the demands of the rig throughout the entire 2E1HR, 3E1HR, and TP1HR cycles. However, during the transient section of the 24HR cycle, the system did not recover sufficient instant heat. A 30-minute heat balance curve was created to observe heat recovery performance during that period of the 24HR

cycle. Figure 56 presents the DNGE CHP hybrid system instant heat balance 30-minute moving average for the last eight hours of the 24HR cycle. With the addition of the hybrid systems, both the modeled HEX and ESG were able to meet heat demand until just after the 18th hour of the 24 hour cycle.
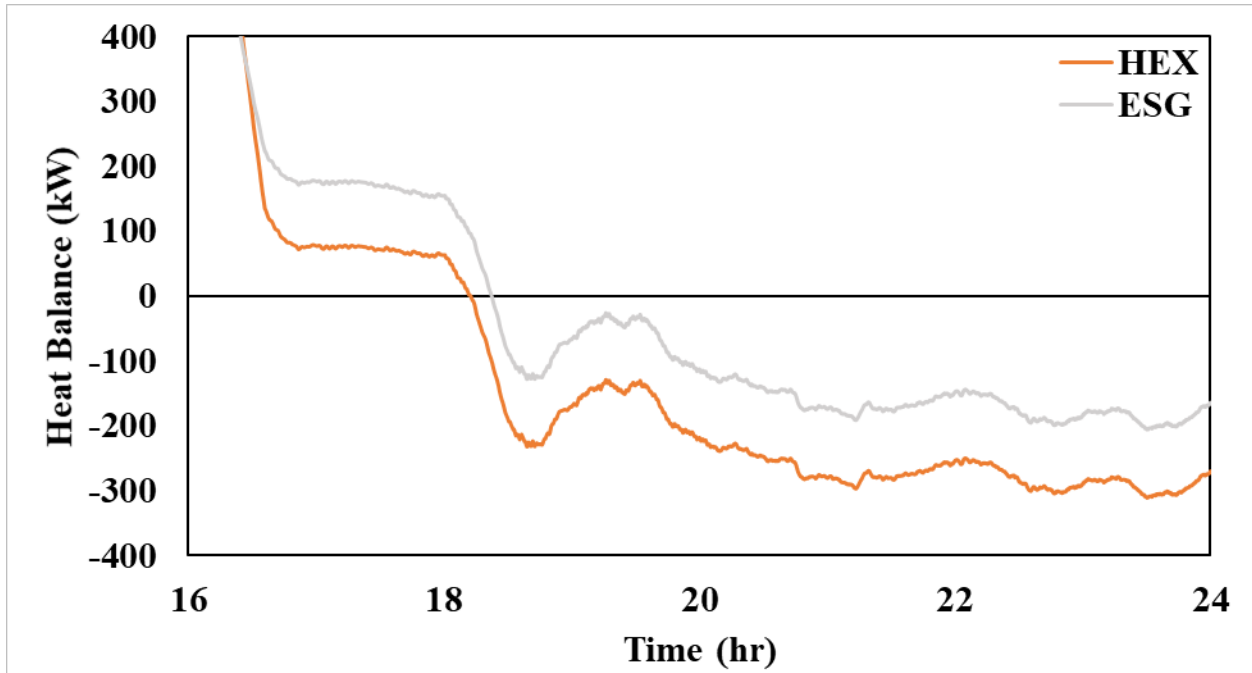
Figure 56. DNGE CHP Hybrid System, 24HR Cycle 30-Minute Heat Balance Moving Average

The DNGE CHP system equipped with the HEMS was able to supply sufficient heat for a more extended period during the 24HR cycle. However, by the conclusion of the cycle, the heat demanded was higher than the heat recovered.

Hybridization provided higher engine efficiency while generating enough heat to keep the rig's equipment functional and provide comfort to its crew. Table 29 shows DNGE CHP hybrid system performance in terms of UF. On average, the UF for the gensets only was improved from 19.0% to 20.8%, and CHP UF increased from 30.3 to 33.2% and 33.1% for the DNGE E-HEX and DNGE ESG units, respectively. Potential UFs reached 39.1% and 37.8% for the E-HEX and ESG configurations, respectively.

Higher UF translates into a more efficient system; therefore, less fuel must be burned for the same amount of energy to be used. Figure 57 shows natural gas fuel savings for the DNGE CHP hybrid system for the four cycles.

Table 29. DNGE CHP Hybrid System UF Analysis

| Cycle | DNGE Hybrid System Without CHP System UF | DNGE-CHP Hybrid System UF Limited by Heat Required | | DNGE-CHP Hybrid System Potential UF | |
| | | E-HEX | ESG | E-HEX | ESG |
Name	%	%	%	%	%
2E1HR	20.2	31.0	31.0	38.2	37.5
3E1HR	21.1	30.5	30.5	38.7	37.9
TP1HR	21.2	38.7	38.3	41.2	38.3
24HR	20.7	32.7	32.7	38.2	37.7
Average	20.8	33.2	33.1	39.1	37.8
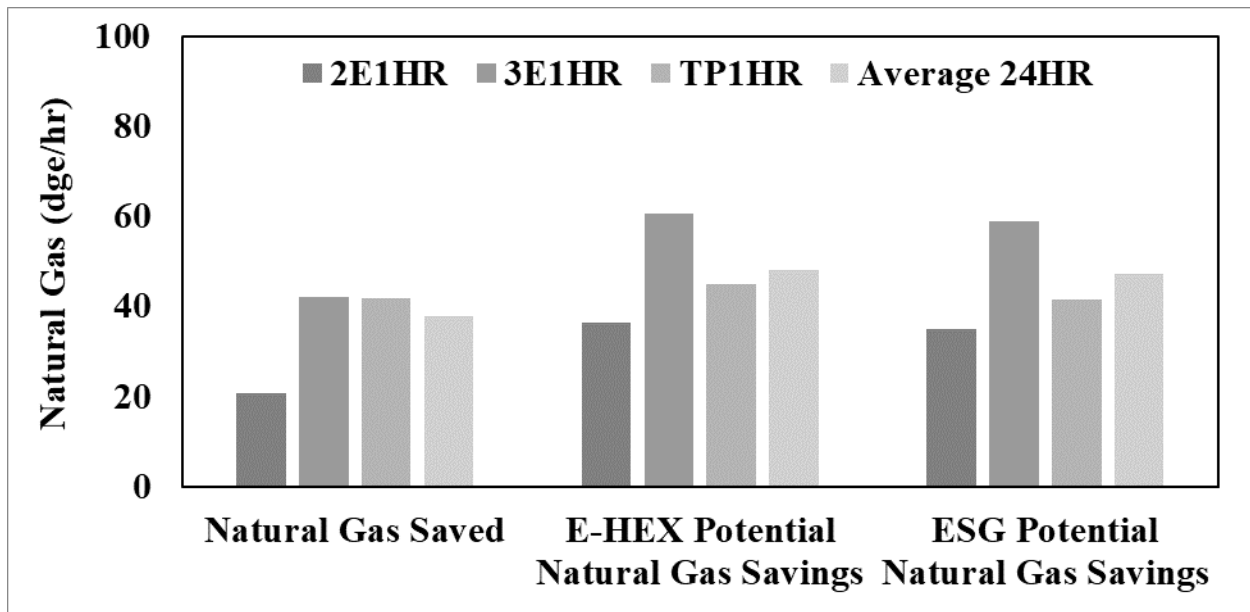
Figure 57. DNGE CHP Hybrid System Natural Gas Savings

Natural gas savings increased compared to the DNGE non-hybrid system due to running fewer engines at higher loads. Note that fuel savings for the 2E1HR cycle were lower than the non-hybrid system by 3.24 dge/hr (8.9 kg/hr). This cycle was characterized by running two engines at high-

steady loads. The objective of the HEMS was to convert any cycle into a high load, steady-state cycle. Therefore, no significant changes were implemented for the hybrid system on this cycle. The TP1HR cycle resulted in the highest fuel savings reducing the online gensets from three to one accompanied by the HEMS. Average natural gas fuel saved was calculated at 35.5 dge/hr (97.7 kg/hr), and potential natural gas fuel savings averaged 47.6 dge/hr (130.8 kg/hr). Associated CO_2 emissions reductions were estimated at 265 kg/hr, and potential emissions savings averaged 360 kg/hr. Figure 58 shows natural gas fuel costs savings for the system which averaged $16.1/hr (21.5% of natural gas fuel expenses) with potential of reaching an average of $22.2/hr (27.9% of natural gas fuel expenses).
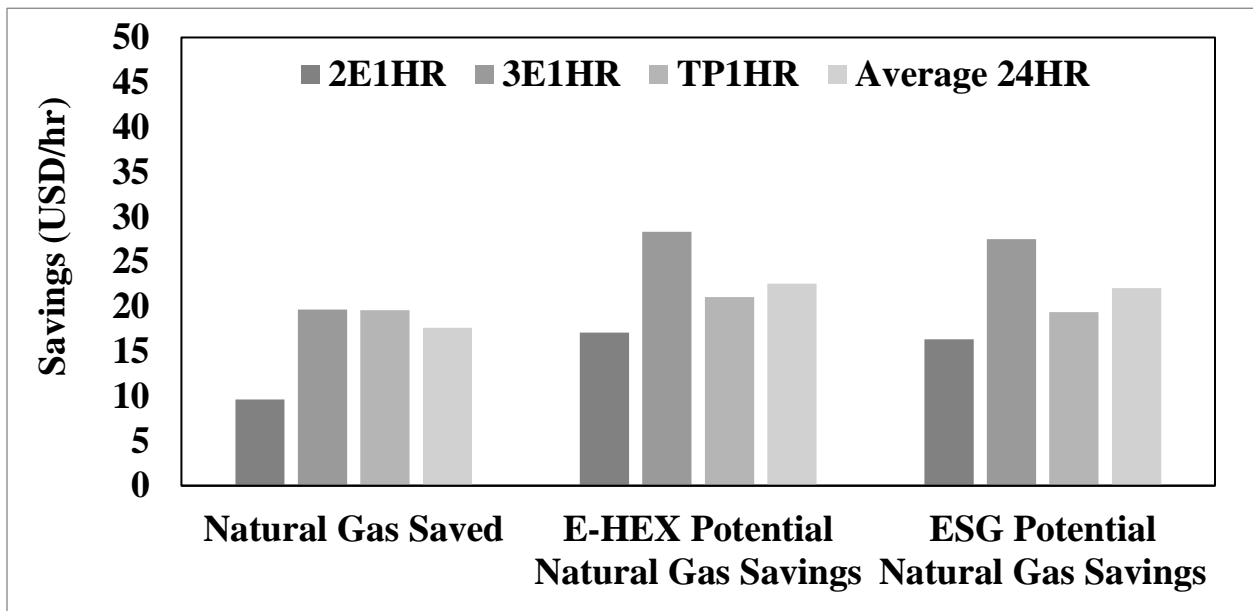
Figure 58. DNGE CHP Hybrid System Savings

In order to assess a solution for the instant heat recovery during the transient section of the 24HR cycle, the same electric heater module was included in the DNGE CHP hybrid model to investigate the feasibility of the system. It was found that using this approach, instant heat recovery exceeded instant heat demand throughout the cycle using electric heaters with a maximum output of 346 kW. Similar to the analysis for HHPDE, this analysis highlights that the total energy demand can be met. The potential UF of the system equipped with the E-HEX and JW-HEX decreased from 38.2% to 36.6% when utilizing the electric heaters. For the system equipped with the ESG and the

JW-HEX, the potential UF decreased from 37.8 to 35.9%. Figure 59 shows the instant heat balance for the DNGE CHP hybrid system equipped with the electric heater system.
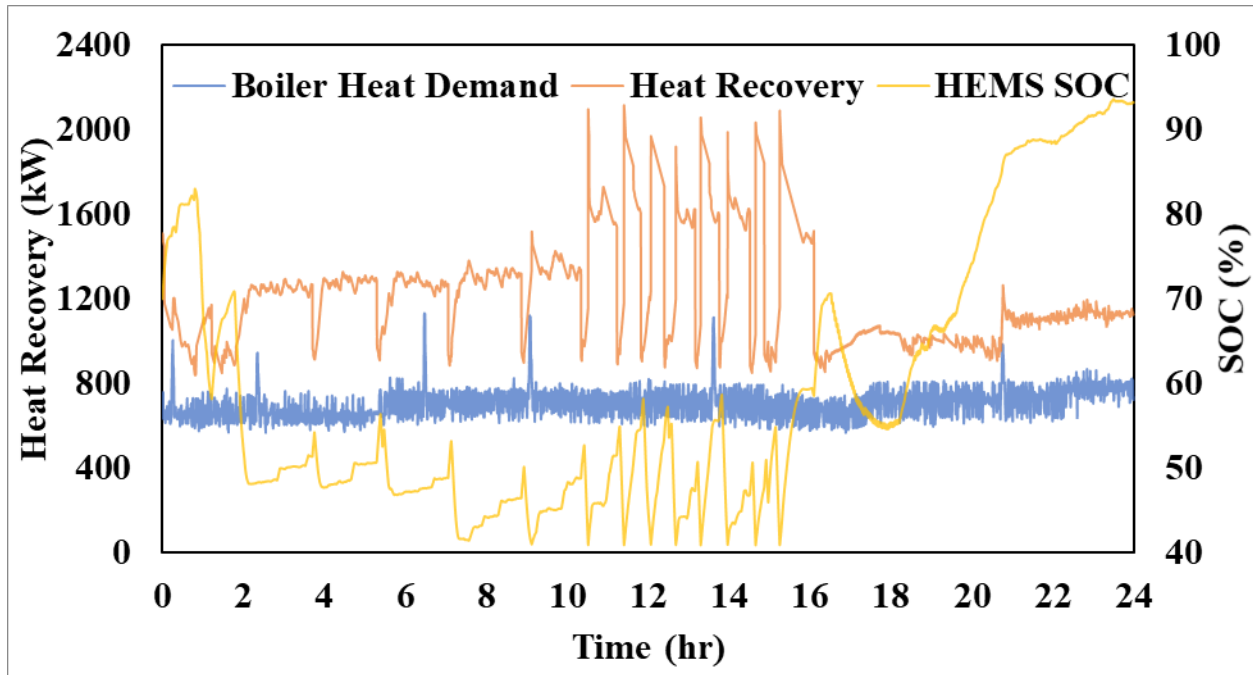
Figure 59. DNGE CHP Hybrid System, 24HR Cycle Instant Heat Recovery Equipped with Electric Heaters

6.5 Cost Analysis

Fuel consumption and emissions reductions may reduce operating costs, improve overall efficiency, and comply with future emissions regulations. However, to achieve these goals, a considerable amount of money must be invested. Rigs equipped with boilers are generally used in cold weather conditions, which vary by location, season, and rig activity frequency. Therefore, as a general cost-efficiency approach, time for investment recovery was calculated based on prices available in the U.S markets and U.S. dollars as a function of replaced boiler activity hours. Equation 9 shows the cost balance equation used.

Equation 9. Return on Investment

$$Expenses\ (hrs\ of\ replanced\ boiler\ activity) = Savings(hrs\ of\ replaced\ boiler\ activity) \quad (9)$$

Where expenses include the equipment needed to be acquired, and savings include equipment no longer required and fuel savings.

Fuel prices vary throughout the U.S., and both diesel and natural gas fluctuate due to several factors that exceed the scope of this research. Therefore, fuel prices were standardized for calculating economic savings according to the models developed. As a reference, the average national diesel price as of 2/22/21, according to the U.S. Energy Information Administration (EIA), was considered ($2.973/gal or $0.787/L) [53]. The last publicly available EIA natural gas prices information established that the industrial price as of November 2020 averaged $3.93 per 1000 ft^3 [54]. The U.S. natural gas heat content in 2019 averaged 1038 BTU/ft^3 or 49.7MJ/kg [55]. The average natural gas price per kg was calculated at $0.17.

For this analysis, the shipping and handling costs of the equipment were not calculated since each company has its own logistics regarding equipment management. The main components that may incur relevant costs were the E-HEXs, the JW-HEXs, the HEMS, the boiler, and the fuel consumed. Both HHPDE E-HEX and DNGE E-HEX were compared in price to the CAIN industries' ESG. Their respective price estimations are included in Appendix D. The HEMS cost was estimated by the price per kWh as reported by the EIA. The last U.S. average installed utility-scale battery storage cost in 2018 was estimated at $625 per kWh [56]. Therefore, the cost for meeting 584 kW-hr capacity could be estimated at $365,000.

For the HHPDE CHP system, the expenses included:

Three E-HEX at $179,354 each with a multiplying factor of 0.93 when purchasing more than one exchanger (Appendix D).

Three JW-HEXs at an estimated price of $27,367 per unit (Appendix D).

The savings for this configuration included:

Boiler rental expenses were based on a daily fee of $240 and one freight charge fee of $3950 (Appendix F).

Diesel fuel costs to be recovered with the UFs of 56.8% and 72.7%.

Table 30 shows the return on investment analysis based on the information provided above for the HHPDE CHP rig configuration.

Table 30. HHPDE CHP Return on Investment Analysis

Equipment	HHPDE CHP	
	Expenses (+) and Savings (-)	
E-HEX	$500,398	
JW-HEX	$82,100	
Boiler Rental	-$58,989	
Diesel Costs to Be Saved	-$523,509	
UF	**56.8%**	**72.7%**
Time (hrs. of replaced boiler activity)	5254	3857
Days (replaced boiler activity)	220	161

If normalizing savings per day of replaced boiler activity, the HHPDE CHP system would save approximately $2648 per day. If the UF was increased to 72.7%, savings would be estimated at $3618 per day.

For the HHPDE CHP system equipped with the HEMS, the expenses included:

Two E-HEX at $179,354 each with a multiplying factor of 0.93 when purchasing more than one exchanger (Appendix D).

Two JW-HEXs with an estimated price of $27,367 per unit (Appendix D).

A HEMS estimated at $365,000.

The savings for this configuration included:

Boiler rental expenses were based on a daily fee of $240 and one freight charge fee of $3950 (Appendix F).

Diesel fuel costs to be recovered with the UFs of 61.1% and 76.9%.

Table 31 shows the return of investment analysis for the HHPDE CHP system equipped with the HEMS.

Table 31. HHPDE CHP Equipped with a HEMS Return on Investment Analysis

Equipment	HHPDE CHP Hybrid	
	Expenses (+) and Savings (-)	
E-HEX	$333,598	
JW-HEX	$54,733	
Boiler Rental	-$68,307	
HEMS	$365,000	
Diesel Costs to Be Saved	-$685,024	
UF	**61.1%**	**76.9%**
Time (hrs. of replaced boiler activity)	6178	4794
Days (replaced boiler activity)	257	200

If normalizing savings per day of replaced boiler activity, the HHPDE CHP hybrid system would save approximately $2931 per day. If the UF was increased to 76.9%, savings would be estimated at $3767 per day.

For the DNGE CHP system, the investments include:

- Three E-HEX, each of their cost was $146,273 with a multiplying factor of 0.93 when purchasing more than one unit (Appendix D).

On the savings side, the DNGE CHP system saved in:

Boiler rental expenses were based on a daily fee of $240 and one freight charge fee of $3950 (Appendix F).

Natural gas fuel costs to be recovered with the UFs of 30.3% and 34.9%.

Table 32 shows the return on investment analysis for the DNGE CHP system.

Table 32. DNGE CHP System Return on Investment Analysis

Equipment	DNGE CHP	
	Expenses (+) and Savings (-)	
E-HEX	$408,102	
Boiler Rental	-$108,756	
Natural Gas Costs to Be Saved	-$299,346	
UF	30.3%	34.9%
Time (hrs. of replaced boiler activity)	10,061	8236
Days (replaced boiler activity)	419	343

If normalizing savings per day of replaced boiler activity, the HHPDE CHP system would save approximately $974 per day. If the UF was increased to 34.9%, savings would be estimated at $1190per day.

For the DNGE CHP system, equipped with the HEMS, expenses included:

- Two E-HEX, each of their cost was $146,273 with a multiplying factor of 0.93 when purchasing more than one unit (Appendix D).
- One HEMS estimated at $365,000.

The savings for this configuration included:

Boiler rental expenses were based on a daily fee of $240 and one freight charge fee of $3950 (Appendix F).

Natural gas fuel costs to be recovered with the UFs of 33.2% and 39.1%.

Table 33 shows the return of investment analysis for the DNGE CHP system equipped with the HEMS.

Table 33. DNGE CHP System Equipped with the HEMS Return on Investment Analysis

Equipment	HHPDE CHP Hybrid	
	Expenses (+) and Savings (-)	
E-HEX	$272,068	
Boiler Rental	-$152,024	
HEMS	$365,000	
Natural Gas Costs to Be Saved	-$485,044	
UF	33.2%	39.1%
Time (hrs. of replaced boiler activity)	14,215	11,827
Days (replaced boiler activity)	592	493

The cost-efficiency analysis relied on the overall system running time, which includes the boiler being fully active. Boilers on well pads are only active at locations where cold weather occurs at cold seasons, varying throughout the country. To make this analysis applicable to all locations, expenses covered by savings were calculated in terms of hours of replaced boiler activity. Average annual boiler activity time at specific locations can be divided by time (hrs) to find the approximate time needed to amortize the potentially acquired assets.

For the weather activity recorded throughout the month of February and March in WV, it was estimated that using the HHPDE CHP system, including three E-HEXs and three JW-HEXs, 220 days of boiler activity would be needed to cover the system investment. For the same case scenario powered by the DNGE CHP system, 419 days would be required to amortize the investment. When optimizing the system with the HEMS, it was proven that two engines and the HEMS could handle any load, saving the third exchanger's costs. Moreover, the cost of the third engine could be categorized as savings. For the purpose of this analysis, the cost of the third engine was not removed from the total costs. It was estimated that approximately 257 days of boiler activity must be replaced to match investments on the system acquisition for the HHPDE CHP system equipped with the HEMS. If DNGE powered the hybrid system, 592 days of replaced boiler activity on average would be needed to pay off initial system investments.

7. Conclusions

The unconventional well development industry has been growing in the U.S. Exploration and production companies are attempting to utilize energy-saving techniques to save on costs and reduce fuel consumption and emissions. In the unconventional well completion industry, the primary consumers, or prime movers, of diesel fuel or natural gas are HHP drilling engines and boilers. Combining both power and heat cogeneration would result in fuel savings and emissions reductions. This has led government agencies such as the U.S. DOE to seek information on these potential solutions. As part of a funding opportunity from the U.S. DOE, WVU was tasked to develop and validate new knowledge and technology to improve efficiency and minimize the environmental implications of unconventional resource development. An energy consumption audit was performed on HHP drilling engines during the horizontal drilling of an unconventional well. Data from the audit were used to assess methods for reducing fossil fuel consumption and associated emissions during the development phase.

In order to complete this task, in-field engine data were collected from the three CAT 3512C diesel gensets. The main data parameters were engine load, coolant temperature, fuel consumption, and exhaust temperature. The engine data were filtered and binned to create engine performance models that included exhaust temperature, fuel consumption, exhaust flow, and coolant temperature, as functions of engine load. Previously recorded emissions data from these engines were used to estimate water concentration in the exhaust flow. Exhaust gas was assumed to be water vapor and dry air only, and heat rejection calculations were based on this exhaust composition. Boiler activity was also recorded to determine fuel consumption and heat output, assuming a constant efficiency of 81%. Four cycles representing different stages in the development of an unconventional well were extracted from recorded on-site data and used for comparisons. These included a steady drilling cycle powered by two engines, a steady drilling cycle powered by three engines, a tripping pipe/casing cycle (transient operation) powered by three engines, and a 24 hour cycle with variable engines online and variable power demand profile. A natural gas powered rig was also modeled based on engine data previously recorded by another research project conducted by WVU.

The heat recovery system was developed using an engine exhaust flow model, engine coolant model, and CHEMCAD software. Exhaust and jacket water heat exchangers were sized, designed,

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and simulated for both HHPDE and DNGE powered rigs. A simple HEMS model was developed on a SIMULINK subsystem based on literature. An overall SIMULINK model was created for each system configuration, including HHPDE and DNGE with no WHR system, HHPDE and DNGE with CHP system, and HHPDE DNGE CHP system equipped with HEMS. SIMULINK models were run using the four cycles extracted from on-site recorded data. Potential heat recovery was calculated and compared to the heat demand provided by the boiler. System efficiency was calculated in terms of the UF of the fuel (diesel or natural gas). The goal was to improve UF by recovering wasted heat rejected from the engines and utilizing the HEMS. This would reduce fuel consumption while providing the same amount of work and heat for the four cycles.

Results based on the cycles analyzed indicated that it would be possible to improve the UF utilization for both CHP and hybrid systems. Each of the cycles analyzed represents a characteristic operation of the rig. It was found that the HHPDE CHP system could increase UF from an average of 35.7 to an average of 56.8 and had the potential to use up to 72.7 of the fuel's energy. This would represent a potential diesel fuel savings of 156.1 L/hr ($122.9 per hour) and an emissions reduction of 418 kg/hr of CO_2. An investment of $582,498 would be required to acquire the equipment. This investment would be recouped after 220 days of replaced boiler operation. When the hybrid system was added to the HHPDE CHP system, the average UF was increased from 35.7 to 38.2 utilizing the HEMS alone. If equipped with the HHPDE E-HEXs and HHPDE JW-HEXs, the UF averaged about 61.1 and had the potential to reach 76.9, representing 169.6 L/hr of diesel fuel savings ($133.5 per hour) and 454 kg/hr of associated CO_2 emissions. Investment for this scenario was estimated at $753,331. This investment would be recouped after 257 days of replaced boiler operation to cover the cost. The DNGE CHP system increased the UF from an average of 19.0 to an average of 30.3 and the potential to use up to 34.9, representing potential fuel savings of 158.9 kg/hr of natural gas and associated emissions reduction of 437 kg/hr of CO_2. An investment of $408,102 would be required for acquiring the equipment. This investment would be recouped after 419 days of replaced boiler operation. When the hybrid system was added to the DNGE CHP system, the average UF was increased from 19.0 to 20.8 using the HEMS only. When utilizing the DNGE E-HEX and the HEMS, the average UF increased to 33.2 and had the potential to reach 39.1, representing 130.8 kg/hr of natural gas and 360 kg/hr of associated CO_2 emissions. Investment for this scenario was estimated at $637,068. This investment would be recouped after 592 days of replaced boiler operation.

Results showed that instant heat recovery was insufficient for extended periods of transient engine operation. Therefore, other sources of heat had to be utilized in those circumstances. If electric heaters were used to supply the system with the required heat, engine load would increase, recovering more heat. Analysis also demonstrated that the stored electrical energy of the HEMS combined with other energy could meet total rig energy demands. Another approach to the problem would be to utilize thermal reservoirs to store thermal energy for later usage. A possible solution would be the installation of a steam accumulator. They could help save some of the extra heat recovered from the exchangers to be used when heat demand exceeded heat recovery. Such an analysis was not performed in this research, but it could be addressed in further investigation.

8. Recommendations for Future Research

8.1 Tier 4 Compression Ignition CHP and CHP hybrid system modeling

The models developed on this analysis were based on Tier 2 engines due to the broad use of these engines within the industry. However, it is expected for the near future, that Tier 2 engines will be replaced with Tier 4 engines as older engines are taken off duty. To assess energy saving potential of CHP and HEMS, a Tier 4 engine data collection campaign should be completed. New models could be developed, and the Tier 4 engine sub models could be implemented into the 0-D models developed in this project and results for future industry prime movers could be obtained.

8.2 Waste Heat Recovery System On-site Validation

Simulations obtained through models developed in this paper suggested the waste heat recovered from unconventional well completion gensets may be able to reduce or eliminate boiler utilization during the drilling phase of wells during cold weather conditions or in cold regions. A cost analysis was performed, showing that such an investment would pay off between 161 and 592 days of boiler activity, depending on the configuration and fuel used. Validation of the system is critical for future industrial applications. It was calculated that the DNGE powered rigs have more room than HHPDE for heat recovery due to lower thermal efficiencies. The next step in the process would be to manufacture or purchase an E-HEX for a DNGE and obtain on-site heat recovery data from the physical design. To be achieved, it would be necessary to develop a control feedwater system for water inlet regulation based on engine activity. It would also be necessary to connect the E-HEX into a working steam line equipped with mass flow meters, k-type thermocouples, and pressure gauges to calculate heat supply. It would also be necessary to add a pump to pressurize the water inlet into the exchanger. Once data is recorded from this experiment, results can be extrapolated into a fully equipped rig power plant (three or four engines).

If data collected during the DNGE CHP system (one genset and one E-HEX) on-site validates the results presented in this paper, the next step would be to do the same with the HHPDE CHP system.

8.3 Emissions Savings Through CHP and CHP Hybrid Systems

Results obtained in this paper suggest that there is potential for fuel savings and reduced GHGs emissions with the use of CHP and CHP hybrid systems. Only CO_2 emissions were estimated through models developed. However, DNGE and HHPDE emissions also include other products regulated pollutants. These include CO, HC, PM, NO_x, and other GHGs such as CH_4.

A model for each engine type can be constructed based on data utilized in this project. Controlled emissions could be estimated to understand the system's impact on regulated emissions. Moreover, previously recorded emissions data from these engines could be used to build an emissions model.

Another critical aspect that has not been approached in this research is the E-HEX exhaust emissions. Temperature drops may affect the chemical composition of exhaust gases when passing through the E-HEX. Suppose the WHR system on-site validation step is taken. In that case, emissions sensors could be instrumented into the exchangers' exhaust for comparison to the engine's emissions with no CHP system to evaluate its impact.

8.4 Heat and Power Hybrid Energy Management System Integration

The power plant (composed of the three HHPDEs or DNGEs) operates using its own control system, independent from the rig's control system. The rig control system uses engine load data to prevent overload requests. However, the power plant does not receive a demand signal from the rig control system. Rather, the gensets operate on a closed-loop system that uses voltage and frequency to recognize adequate load to meet power demand. Since there is no stored energy, the gensets must increase or decrease load in an attempt to match the amount of power that the rig requires. When the system is balanced at 60 Hz and 600 V, power production matches rig demand [13]. The total power demand is evenly divided across all the online gensets. Having an excess number of gensets online to be able to handle potential spikes in power demand results in a lower average load on each engine, causing them to operate at lower efficiency and produce higher brake specific emissions. Engines remain online and running when fewer engines could meet power demand because each genset takes several minutes to start up. The gensets need time to warm up and synchronize to the CAN bus. Genset warmup time is a critical consideration in automating engine start and stop decisions while avoiding power limits that interfere with rig operations [13].

The boiler is not equipped with a control unit system; therefore, the rig's crew is responsible for turning heaters on and off, and the boiler adjusts to the demand accordingly. An automated control system for the boiler could be developed to provide enough heat based on the weather without human intervention. Current technology allows accurate weather prediction in all locations to auto-adjust heat supply and predict future demands. A smart control could be developed, allowing communication with weather report websites for predicted heat demand.

The hybrid system advantage is its control system, allowing the engines to run in a more efficient load, reducing the required number of in-use engines while maintaining the capability to meet unpredictable spikes in power demand. The CHP hybrid integrated control system could consider both heat and power demands to increase the UF by adjusting engine load to produce enough power for the rig and its accessories while outputting enough heat through its exhaust to substitute the boiler's use more effectively. Such an integrated control system could be developed in MATLAB® and run on the already developed SIMULINK models. However, more boiler activity data would be needed to develop a more robust heat demand model as a function of ambient temperature that includes colder scenarios.

8.5 Waste Heat Recovery Analysis on Gas Turbine Powered Rigs

With continuous efforts of the industry to reduce diesel dependency, natural gas-powered drilling rigs could be an alternative for cheaper power generation on-site. In 2017, Siemens developed the SGT-A35 RB gas turbine for the oil and gas industry. The smallest current available gas turbine designed by Siemens for the oil and gas industry is the DGT-A05. Table 34 shows basic performance data published by the manufacturer.

Table 34. Siemens SGT-A05 [61]

Parameter	DGT-A05
Gross Output	4 MW
Functionality	50/60 Hz
Heat Rate	12,137 kJ/kWh
Gross Efficiency	29.7%
Shaft Speed	14,200 rpm
Pressure Ratio	10.3
Exhaust Mass Flow	15.4 kg/s
Exhaust Temperature	560 °C

Using data provided by the manufacturer and using the same fuel composition assumptions used for the previous analysis, a simple SIMULINK model was developed to estimate potential heat rejection. A CHP system minimum effectiveness to recover heat demands for the four cycles analyzed in this paper was analyzed. Figure 60 shows the gas turbine rig powered SIMULINK model developed for estimating feasibility of turbine driven power generation on well development.
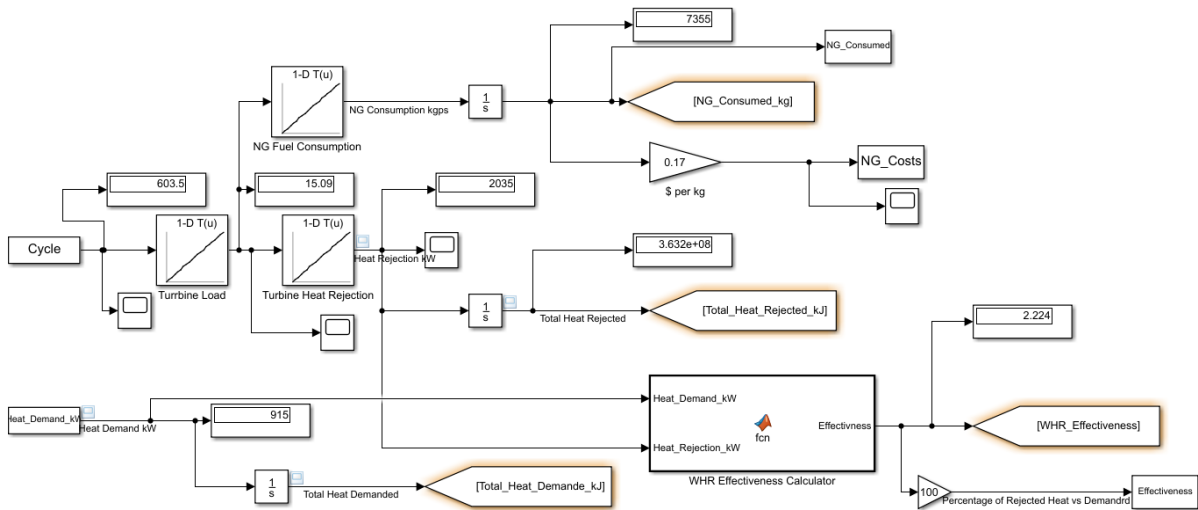
Figure 60. Gas Turbine Powered Drilling Rig SIMULINK Model

After running the four cycles on this model I was found that:

On average the 436.2% of the demanded heat is rejected by the turbine.

Minimum WHR system effectiveness should always be 58.5% to exceed instant heat demand for the four cycles. Considering the potential expansion of the gas turbine power generation on the natural gas development industry, a more advanced model based on recorded turbine activity data could be developed as my CHP findings could be combined with gas turbines for further energy savings.

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9. Appendices

9.1 Appendix A: KRAL OEM 20 calibration test.

KRAL OEM 20 calibration test results obtained at WVU's engine laboratory

	Test 1			Test 2		
	Flowmeter A	Flowmeter B	Scale	Flowmeter A	Flowmeter B	Scale
Average Flow (L/hr)	485.326	484.558		513.468	512.577	
Average Flow (L/s)	0.135	0.135		0.143	0.142	
Time (s)			99.600			99.000
Mass (kg)			10.932			11.612
Volume (L)	13.427	13.406	13.031	14.120	14.096	13.842
Percent Error (%)	3.043	2.880		2.012	1.835	

9.2 Appendix B: MATLAB® Codes

The scripts shown in this section were used for data importation and model development of engine performance.

Natural Gas Fuel Consumption Model Script

```matlab
%Natural Gas Consumption Model
%Waukesha L7044GSI

load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\WV_NG_Post_Files.mat'
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\TX_NG_Post_Files.mat'
TX_NG=[TX_NG_Post_1; TX_NG_Post_2;TX_NG_Post_3;TX_NG_Post_4];%Combine all
testing in Texas together
TX_NG.Properties.VariableNames={'Time' 'AbsoluteTime' 'AmbientTemperature'
'CrankcaseCO2' 'H2O' 'LgrCh4' 'CrankcaseFlow' 'VoltageFlow' 'NO' 'No2Low'
'No2High' 'N2O' 'Nh3Low' 'Nh3High' 'H2O1' 'Formaldehyde' 'CO2' 'CoLow'
'CoHigh' 'Propylene' 'Ethylene' 'Ch4Low' 'Ch4High' 'Ethane' 'Acetylene'
'NdirTemperature' 'NdirCellPressure' 'PDil' 'CNGFlow' 'Total'
'CNGTemperature' 'Load' 'EngineSpeed' 'EngineTorque' 'Throttle'
'ExhaustTemp1' 'ExhaustTemp2' 'RightLambda' 'LeftLambda'};
TX_NG=table(TX_NG.Time, TX_NG.CNGFlow, TX_NG.Load,
TX_NG.ExhaustTemp1);%Obtain Relevant Data only
TX_NG.Properties.VariableNames={'Time' 'CNGFlow' 'Load'
'ExhaustTemp'};%Rename Table Headers
t=1:length(TX_NG.Time);%Adjust Time array
t=t';
TX_NG.Time=t;

%Plot Data from TX
figure(1)
plot(TX_NG.Time,TX_NG.Load)
yyaxis left
ylabel('Percent Load')
hold on
yyaxis right
plot(TX_NG.Time,TX_NG.CNGFlow)
legend('Percent Load','Natural Gas')
```

```matlab
ylabel('Natural Gas Consumption SCFM')
xlabel('Time, sec');
title('Percent Load and CNG Consumption over time, TX')
WV_NG=[WV_NG_Post_1; WV_NG_Post_2;WV_NG_Post_3;WV_NG_Post_4];%Combine all
testing in WV together
WV_NG.Properties.VariableNames={'Time' 'AbsoluteTime' 'AmbientTemperature'
'CrankcaseCO2' 'H2O' 'LgrCh4' 'CrankcaseFlow' 'VoltageFlow' 'NO' 'No2Low'
'No2High' 'N2O' 'Nh3Low' 'Nh3High' 'H2O1' 'Formaldehyde' 'CO2' 'CoLow'
'CoHigh' 'Propylene' 'Ethylene' 'Ch4Low' 'Ch4High' 'Ethane' 'Acetylene'
'NdirTemperature' 'NdirCellPressure' 'PDil' 'CNGFlow' 'Total'
'CNGTemperature' 'Load' 'EngineSpeed' 'EngineTorque' 'Throttle'
'ExhaustTemp1' 'ExhaustTemp2' 'RightLambda' 'LeftLambda'};
WV_NG=table(WV_NG.Time, WV_NG.CNGFlow, WV_NG.Load,
WV_NG.ExhaustTemp1);%Obtain Relevant Data only
WV_NG.Properties.VariableNames={'Time' 'CNGFlow' 'Load'
'ExhaustTemp'};%Rename Table Headers
tt=1:length(WV_NG.Time);%Adjust Time array
tt=tt';
WV_NG.Time=tt;

%Plot Data from WV
figure(2)
plot(WV_NG.Time,WV_NG.Load)
title('Percent Load and CNG Consumption over time, WV')
hold on
yyaxis left
ylabel('Percent Load')
yyaxis right
plot(WV_NG.Time,WV_NG.CNGFlow)
legend('Percent Load','Natural Gas')
ylabel('Natural Gas Consumption SCFM')
xlabel('Time, sec')

%Bin data from TX
TX=sortrows(TX_NG);
edges=[0,20,25,30,35,40,45,50,55,60,80,85,90,95,100];
bins=discretize(TX.Load,edges);
meanPercLoad=splitapply(@mean,TX(:,3),bins);
meanPercLoad=[meanPercLoad;100]; %Add 100% Load value secified by
manufacturer
meanCNG=splitapply(@mean,TX(:,2),bins);
meanCNG=[meanCNG; .42];%Add Max NG Consumption at 100%Load specified by
manufacturer
TX_1=table(meanPercLoad,meanCNG);

%Bin data from WV 214
WV=sortrows(WV_NG);
edges=[0,20,25,30,35,40,45,50,55,60,80,85,90,95,100];
bins=discretize(WV.Load,edges);
meanPercLoad_1=splitapply(@mean,WV(:,3),bins);
meanPercLoad_1=[meanPercLoad_1;100]; %Add 100% Load value secified by
manufacturer
meanCNG_1=splitapply(@mean,WV(:,2),bins);
meanCNG_1=[meanCNG_1; 240.9];%Add Max NG Consumption at 100%Load specified by
manufacturer
WV_1=table(meanPercLoad_1,meanCNG_1);
```

```matlab
%Combine both sites data
All=[WV_NG;TX_NG];
All=sortrows(All);%Bin All data combines
edges=[0,20,25,30,35,40,45,50,55,60,80,85,90,95,100];
bins=discretize(All.Load,edges);
meanPercLoad_2=splitapply(@mean,All(:,3),bins);
meanPercLoad_2=[meanPercLoad_2;100]; %Add 100% Load value secified by
manufacturer
meanCNG_2=splitapply(@mean,All(:,2),bins);
meanCNG_2=[meanCNG_2; 240.9];%Add Max NG Consumption at 100%Load specified by
manufacturer
All_1=table(meanPercLoad_2,meanCNG_2);

%Plot Model
figure(3);
scatter(meanPercLoad,meanCNG);
title('Mean Percent Load vs Natural Gas Consumption');
xlabel('Percent Laod');
ylabel('Natural Gas, SCFM');
hold on
scatter(meanPercLoad_1,meanCNG_1);
hold on
scatter(meanPercLoad_2,meanCNG_2);
legend('TX', 'WV', 'All')

%Obtain best fit equations for model
  p=polyfit(meanPercLoad_2,meanCNG_2,3)
  f = polyval(p,meanPercLoad_2);
  plot(meanPercLoad_2,f,'-b')
  legend('Natural Gas Consumption Best Fit')


 %%Display trendline equation Diesel
 NG_mean_fit=polyfit(meanPercLoad,meanCNG_2,3);
 NG_mean_equation=polyval(NG_mean_fit,meanPercLoad_2);
 trendlineequationcoeffnum=length(NG_mean_fit);
          switch trendlineequationcoeffnum;
                case 2  % means it has slope and intercep 2 numbers
                    NG_Consumption_Model=sprintf('y=%.3f
x+%.3f',NG_mean_fit(1),NG_mean_fit(2));
                case 3
                    NG_Consumption_Model=sprintf('y=%.3f x^2+%.3f
x+%.3f',NG_mean_fit(1),NG_mean_fit(2),NG_mean_fit(3));
               case 4
                  NG_Consumption_Model=sprintf('y=%.3f x^3+%.3f x^2+ %.3f
x+%.3f',NG_mean_fit(1),NG_mean_fit(2),NG_mean_fit(3),NG_mean_fit(4));
               case 5
                  NG_Consumption_Model=sprintf('y=%.3f x^4+%.3f x^3+ %.3f
x^2+%.3f
x+%.3f',NG_mean_fit(1),NG_mean_fit(2),NG_mean_fit(3),NG_mean_fit(4),NG_mean_f
it(5));
          end

htrendlinetext=text((1)+0.5*(3),(2)+0.8*(4),NG_Consumption_Model,'units','nor
malized');
```

```
            display(NG_Consumption_Model)
            htrendlinetext.FontSize=12;
            htrendlinetext.FontWeight='bold';
            htrendlinetext.Color='r';
```

Natural Gas Engine Exhaust Temperature Model

```
%WaukeshaL7044GSIEPA Exhaust Heat Rejection Model
%Diego Dranuta
%11/3/2020

load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\WV_NG_Post_Files.mat'
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\TX_NG_Post_Files.mat'
TX_NG=[TX_NG_Post_1; TX_NG_Post_2;TX_NG_Post_3;TX_NG_Post_4];
t=1:length(TX_NG.Time);
t=t';
TX_NG.Time=t;
figure(1)
plot(TX_NG.Time,TX_NG.Load)
figure(2)
plot(TX_NG.Time,TX_NG.ExhaustTemp1)
WV_NG=[WV_NG_Post_1; WV_NG_Post_2;WV_NG_Post_3;WV_NG_Post_4];
tt=1:length(WV_NG.Time);
tt=tt';
WV_NG.Time=tt;
figure(3)
plot(WV_NG.Time,WV_NG.Load)
figure(4)
plot(WV_NG.Time,WV_NG.ExhaustTemp1)
%Note: Use TX DATA ONLY. WV data is TP (Transient)
TX_NG=table(smooth(TX_NG.Load,1),
smooth(((TX_NG.ExhaustTemp1+TX_NG.ExhaustTemp2)./2),1));
TX_NG.Properties.VariableNames={'Load' 'ExhaustTemp'};
figure(5)
scatter(TX_NG.Load,TX_NG.ExhaustTemp);
 All_1=sortrows(TX_NG);
 edges=[0,10,15,20,25,30,35,40,45,50,55,60,100];
  bins=discretize(All_1.Load,edges);
  meanPercLoad=splitapply(@mean,All_1(:,1),bins);
  meanPercLoad_1=[meanPercLoad;100];             %Add 100% Looad value
from manufacturer specs
  meanExhaustTemp=splitapply(@mean,All_1(:,2),bins);
  meanExhaustTemp_1=[meanExhaustTemp; 637];
  All_2=table(meanPercLoad_1,meanExhaustTemp_1);
  figure(6);
  scatter(meanPercLoad_1,meanExhaustTemp_1,'or');
  title('Mean Percent Load vs Mean Exhaust Temperature in each bin');
  xlabel('Percent Laod');
 ylabel('Exhasut Temperature in C');
 hold on

 %Obtain best fit equations for model
  p=polyfit(meanPercLoad,meanExhaustTemp,3)
  f = polyval(p,meanPercLoad);
  plot(meanPercLoad,f,'-b')
  legend('Dual Fuel Exhasut Temperatre Best Fit')
```

```matlab
%%Display trendline equation Diesel
ExhaustTempMeanFit=polyfit(meanPercLoad,meanExhaustTemp,3);
ExhaustTempMeanFitEquation=polyval(ExhaustTempMeanFit,meanPercLoad);
trendlineequationcoeffnum=length(ExhaustTempMeanFit);
            switch trendlineequationcoeffnum;
                  case 2  % means it has slope and intercep 2 numbers
                      NG_ExhaustModel=sprintf('y=%.3f
x+%.3f',ExhaustTempMeanFit(1),ExhaustTempMeanFit(2));
                  case 3
                      NG_ExhaustModel=sprintf('y=%.3f x^2+%.3f
x+%.3f',ExhaustTempMeanFit(1),ExhaustTempMeanFit(2),ExhaustTempMeanFit(3));
                  case 4
                      NG_ExhaustModel=sprintf('y=%.3f x^3+%.3f x^2+ %.3f
x+%.3f',ExhaustTempMeanFit(1),ExhaustTempMeanFit(2),ExhaustTempMeanFit(3),Exh
austTempMeanFit(4));
                  case 5
                      NG_ExhaustModel=sprintf('y=%.3f x^4+%.3f x^3+ %.3f
x^2+%.3f
x+%.3f',ExhaustTempMeanFit(1),ExhaustTempMeanFit(2),ExhaustTempMeanFit(3),Exh
austTempMeanFit(4),ExhaustTempMeanFit(5));
            end

htrendlinetext=text((1)+0.5*(3),(2)+0.8*(4),NG_ExhaustModel,'units','normaliz
ed');
            display(NG_ExhaustModel)
            htrendlinetext.FontSize=12;
            htrendlinetext.FontWeight='bold';
            htrendlinetext.Color='r';
```

CATERPILLAR 3512C Coolant Temperature Model

```matlab
%Engine Coolant Temperature Model
%Diego Dranuta
%WVU
% Created on 12/16/2020
%Last Updated on 12/16/2020

%%
%Load cycles, extract percent load and coolant temperature data and cat
%together
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 1 hour 3 Engines Boiler
On\Drilling_Cycle_1_hour_3_Engines_on_20200229_From_0300_to_0400_Boiler_on.ma
t'
Engine_1_a_PercentLoad=[emissions1.EnginePercentLoadAtCurrentSpeed];
Coolant_1_a=[emissions1.EngineCoolantTemperature];
Engine_2_a_PercentLoad=[emissions2.EnginePercentLoadAtCurrentSpeed];
Coolant_2_a=[emissions2.EngineCoolantTemperature];
Engine_3_a_PercentLoad=[emissions3.EnginePercentLoadAtCurrentSpeed];
Coolant_3_a=[emissions3.EngineCoolantTemperature];

load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 2 engines Boiler
On\Drilling_Cycle_1_hour_2_Engines_on_20200228_From_1630pm_to_1730pm_Boiler_O
n.mat'
Engine_1_b_PercentLoad=[emissions1.EnginePercentLoadAtCurrentSpeed];
```

```matlab
Coolant_1_b=[emissions1.EngineCoolantTemperature];
Coolant_1_b=zeros(36000,1);
Engine_2_b_PercentLoad=[emissions2.EnginePercentLoadAtCurrentSpeed];
Coolant_2_b=[emissions2.EngineCoolantTemperature];
Engine_3_b_PercentLoad=[emissions3.EnginePercentLoadAtCurrentSpeed];
Coolant_3_b=[emissions3.EngineCoolantTemperature];


load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 24 hours Boiler
On\Drilling_Cycle_24_hours_3_Engines_On_20200229_From_0000am_to_0000am_Boiler
_on.mat'
Engine_1_c_PercentLoad=[emissions1.EnginePercentLoadAtCurrentSpeed];
Coolant_1_c=[emissions1.EngineCoolantTemperature];
Engine_2_c_PercentLoad=[emissions2.EnginePercentLoadAtCurrentSpeed];
Coolant_2_c=[emissions2.EngineCoolantTemperature];
Engine_3_c_PercentLoad=[emissions3.EnginePercentLoadAtCurrentSpeed];
Coolant_3_c=[emissions3.EngineCoolantTemperature];


load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Tripping
Pipe 1
hour\TP_Cycle_1_Hour_3_Engines_on_20200229_From_1631_to_1731_Boiler_on.mat'
Engine_1_d_PercentLoad=[emissions1.EnginePercentLoadAtCurrentSpeed];
Coolant_1_d=[emissions1.EngineCoolantTemperature];
Engine_2_d_PercentLoad=[emissions2.EnginePercentLoadAtCurrentSpeed];
Coolant_2_d=[emissions2.EngineCoolantTemperature];
Engine_3_d_PercentLoad=[emissions3.EnginePercentLoadAtCurrentSpeed];
Coolant_3_d=[emissions3.EngineCoolantTemperature];



Engine_Percent_Load_1=[Engine_1_a_PercentLoad; Engine_1_b_PercentLoad;
Engine_1_c_PercentLoad; Engine_1_d_PercentLoad];
Engine_Percent_Load_2=[Engine_2_a_PercentLoad; Engine_2_b_PercentLoad;
Engine_2_c_PercentLoad; Engine_2_d_PercentLoad];
Engine_Percent_Load_3=[Engine_3_a_PercentLoad; Engine_3_b_PercentLoad;
Engine_3_c_PercentLoad; Engine_3_d_PercentLoad];
Coolant_1=[Coolant_1_a; Coolant_1_b; Coolant_1_c; Coolant_1_d];
Coolant_1(isnan(Coolant_1))=0;
Coolant_2=[Coolant_2_a; Coolant_2_b; Coolant_2_c; Coolant_2_d];
Coolant_2(isnan(Coolant_2))=0;
Coolant_3=[Coolant_3_a; Coolant_3_b; Coolant_3_c; Coolant_3_d];
Coolant_3(isnan(Coolant_3))=0;

Engine_1=table(Engine_Percent_Load_1, Coolant_1);
Engine_2=table(Engine_Percent_Load_2, Coolant_2);
Engine_3=table(Engine_Percent_Load_3, Coolant_3);

figure(1)
scatter(Engine_Percent_Load_1, Coolant_1);
figure(2)
scatter(Engine_Percent_Load_2, Coolant_2);
figure(3)
scatter(Engine_Percent_Load_3, Coolant_3);

%%
All_1=sortrows(Engine_1);
```

```
  edges=[30,40,50,60,70,100];
  bins=discretize(All_1.Engine_Percent_Load_1,edges);
  meanPercLoad=splitapply(@mean,All_1(:,1),bins);
%   meanPercLoad_1=[meanPercLoad;100];                %Add 100% Looad value
from manufacturer specs
  meanCoolantTemp=splitapply(@mean,All_1(:,2),bins);
%   meanCoolantTemp_1=[meanCoolantTemp; 99];
  All_2=table(meanPercLoad,meanCoolantTemp);
  figure(6);
  scatter(meanPercLoad,meanCoolantTemp,'or');
  title('Mean Percent Load vs Mean Coolant Temperature in each bin');
  xlabel('Percent Laod');
 ylabel('Coolant Temperature in C');
```

HHPDE CHP Importation Script

```
%Script for SimulinkWasteHeatModel
%Diego Dranuta
%Last Update 1/14/2021

% %Load TP Cycle
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Tripping
Pipe 1
hour\TP_Cycle_1_Hour_3_Engines_on_20200229_From_1631_to_1731_Boiler_on.mat'

% Load Drilling 2 Engines On
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 2 engines Boiler
On\Drilling_Cycle_1_hour_2_Engines_on_20200228_From_1630pm_to_1730pm_Boiler_O
n.mat'

%Load Drilling 3 Engines On
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 1 hour 3 Engines Boiler
On\Drilling_Cycle_1_hour_3_Engines_on_20200229_From_0300_to_0400_Boiler_on.ma
t'

% Load 24 Hours Cycle
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 24 hours Boiler
On\Drilling_Cycle_24_hours_3_Engines_On_20200229_From_0000am_to_0000am_Boiler
_on.mat'


emissions1 = table2struct(emissions1,'ToScalar',true);
emissions2 = table2struct(emissions2,'ToScalar',true);
emissions3 = table2struct(emissions3,'ToScalar',true);



TimeLength1=(emissions1.Time(end,1)-emissions1.Time(1,1))

%Load Tables and correlations
load 'C:\Users\Derek\Desktop\DataCollection\Needed Files (Not Including
Exhaust Model)\TempKvsCpKjpermolK.mat'
KvsCp=table2struct(TempKvsCpKjpermolK, 'ToScalar', true);
```

```matlab
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Exhaust flow
model\Simulink\Exhaust_Mass_Flow_Rate.mat'
Exhaust_Mass_Flow_Composition_H2O=table2struct(Superheated_Vapor_Mass_Flow_Ra
te_From_Load, 'ToScalar', true);
Exhaust_Mass_Flow_Composition_Air=table2struct(Air_Mass_Flow_Rate_From_Load,
'ToScalar', true);


load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Exhaust flow
model\Simulink\Superheated_Water_Specific_Enthalpy.mat'
Superheated_Water_Properties=table2struct(SuperheatedWaterSpecificEnthalpyTab
le,'ToScalar', true);

load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Exhaust flow
model\Simulink\Cp_Air_K.mat'
CpAirK=table2struct(CpAirK,'ToScalar', true);

load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Exhaust flow
model\Simulink\Ideal_Gas_Properties.mat'
Ideal_Gas_Properties=table2struct(IdealGasPropertiesofgas,'ToScalar', true);

load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Exhaust flow
model\Simulink\Cp_Superheated_Steam.mat'
Cp_Superheated_Steam=table2struct(CpSuperheatedSteam,'ToScalar', true);

load 'C:\Users\Derek\Desktop\DataCollection\Needed Files (Not Including
Exhaust Model)\TempInCelciousVsGramsPerL.mat'
CvsGramsPerLiter=table2struct(TempInCelciousVsGramsPerL, 'ToScalar', true);

load 'C:\Users\Derek\Desktop\DataCollection\Needed Files (Not Including
Exhaust Model)\Temperature_K_vs_Cp_Kg_KgK.mat'
KvsCpKjperKgK=table2struct(TemperatureKvsCpKgKgK, 'ToScalar', true);




load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Heat Exchanger
Design and Files\HEX December 2020 Design\HEX_Performance.mat'
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Heat Exchanger
Design and Files\HEX December 2020 Design\HEX_Water_Mass_Flow_Rate_kgphr.mat'

%Load CAIN HEX Performance
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Heat Exchanger
Design and Files\CAIN\ESG1_1_Performance.mat'
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Heat Exchanger
Design and Files\CAIN\ESG1_2_W_Economizer.mat'
CAIN_Performance=table2struct(ESG1_2_W_Economizer, 'ToScalar', true);

%Load JW HEX
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Coolant HEX\JW
MODEL\JW_HEX.mat'
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Coolant HEX\JW
MODEL\JW_Heat_Rejection_Model.mat'
```

```matlab
JW_Heat_Rejection_Model=table2struct(JW_Heat_Rejection_Model, 'ToScalar',
true);




%
%
%
%  %This is only used for the Boiler Model
% %  load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON
UTI\WasteHeatModel\Boiler Model\Boiler_Model_Lookup_Table.mat'
% %  Boiler_Load_Model=table2struct(BoilerTemperatureConsumptionentryS1,
'ToScalar', true);
%
%
%Emissions 1 Data Output From Excel File (Input into Simulink)
Engine_1_Exhaust_Temperature_in_C=[emissions1.Time
emissions1.Tc14ExhaustEngine1] ;
Engine_1_Percent_Load_Percent=[emissions1.Time
emissions1.EnginePercentLoadAtCurrentSpeed];
Engine_1_Generator_Output_Power_W=[emissions1.Time
emissions1.GeneratorTotalRealPower];
Engine_1_Fuel_Rate_LpHr=[emissions1.Time emissions1.EngineFuelRate];
Engine_1_Instant_Power_From_ECU_in_W=[emissions1.Time
emissions1.GeneratorTotalRealPower];




%Emissions 2 Data Output From Excel File (Input into Simulink)
Engine_2_Exhaust_Temperature_in_C=[emissions1.Time
emissions1.Tc16ExhaustEngine2] ;
Engine_2_Percent_Load_Percent=[emissions2.Time
emissions2.EnginePercentLoadAtCurrentSpeed];
Engine_2_Generator_Output_Power_W=[emissions2.Time
emissions2.GeneratorTotalRealPower];
Engine_2_Fuel_Rate_LpHr=[emissions1.Time emissions2.EngineFuelRate];
Engine_2_Instant_Power_From_ECU_in_W=[emissions2.Time
emissions1.GeneratorTotalRealPower];




%Emissions 3 Data Output From Excel File (Input into Simulink)
Engine_3_Exhaust_Temperature_in_C=[emissions1.Time
emissions1.Tc17ExhaustEngine3] ;
Engine_3_Percent_Load_Percent=[emissions3.Time
emissions3.EnginePercentLoadAtCurrentSpeed];
Engine_3_Generator_Output_Power_W=[emissions3.Time
emissions3.GeneratorTotalRealPower];
Engine_3_Fuel_Rate_LpHr=[emissions1.Time emissions3.EngineFuelRate];
Engine_3_Instant_Power_From_ECU_in_W=[emissions3.Time
emissions3.GeneratorTotalRealPower];




%Ambient Conditions Output From Excel File (Input into Simulink)
Ambient_Temp_C=[emissions1.Time emissions1.Tc7Ambient];
```

```
%Boiler Data Output From Excel File (Input into Simulink)
Boiler_Feed_Lperh=[emissions1.Time emissions1.FlowmeterAFeed];
Boiler_Return_Lperh=[emissions1.Time emissions1.FlowmeterBReturn];
Boiler_Diesel_Feed_Temperature_C=[emissions1.Time
emissions1.FlowmeterATemperature];
Boiler_Diesel_Return_Temperature_C=[emissions1.Time
emissions1.FlowmeterBTemperature];
Boiler_Exhaust_Temperature_C=[emissions1.Time emissions1.Tc8Boiler];
Boiler_Daytank_Temperature_C=[emissions1.Time emissions1.Tc9Daytank];
```

DNGE CHP Importation Script

```
%This script should be run before running
%SimulinkWasteHeatModel_NaturalGas_Dedicated
%Diego Dranuta
%Last Update 11/6/2020


%% Load Tables
%Load Exhaust Temperature Model
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\Simuink\Tables_For_NG_Simulink_Model.mat'

%% Load Cycles

%Load 3 engines 1 Hour Cycle
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\Cycles\Drilling Cycle 1 hour 3 Engines Boiler
on\NG_Drilling_1_Hour_3_Engines.mat'
%Load 24 Hours Cycle
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\Cycles\24 Hours Boiler On\NG_24_Hours_Cycle.mat'
%Load 2 Engines 1 Hour Cycle
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\Cycles\Drilling Cycle 1 hour 2 Engines Boiler
on\Drilling_Cycle_2_Engines_1_Hour.mat'
% %Load Generalized Cycle
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\Cycles\Generalized Cycle\Generalized_Cycle_Natural_Gas.mat'
% %Load TP 1 Hour Cycle
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\Cycles\Tripping Pipe 1 Hour\NG_TP_1_Hour_Cycle.mat'


%% Export Simulink Inputs
 Engine_1_Percent_Load_Percent=[Engine_1.Time Engine_1.PercentLoadWaukesha];
 Boiler_Exhaust_Temperature_C=[Engine_1.Time Engine_1.Tc8Boiler];
 Boiler_Daytank_Temperature_C=[Engine_1.Time Engine_1.Tc9Daytank];
 Boiler_Diesel_Feed_Temperature_C=[Engine_1.Time
Engine_1.FlowmeterATemperature];
 Boiler_Diesel_Return_Temperature_C=[Engine_1.Time
Engine_1.FlowmeterBTemperature];
 Boiler_Feed_Lperh=[Engine_1.Time Engine_1.FlowmeterAFeed];
 Boiler_Return_Lperh=[Engine_1.Time Engine_1.FlowmeterBReturn];
 Engine_2_Percent_Load_Percent=[Engine_2.Time Engine_2.PercentLoadWaukesha];
```

```
 Engine_3_Percent_Load_Percent=[Engine_3.Time Engine_3.PercentLoadWaukesha];
 Ambient_Temp_C=[Engine_1.Time Engine_1.Tc7Ambient];
TimeLength1=Engine_1.Time(end,:)
%% Load Heat NG HEX Chart
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\Heat Exchanger Performance\HEX_NG.mat'


%% Load CAIN ESG1 Nat Gas
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Natural Gas
Only\CAIN\CAIN_ESG1_Performance.mat'
CAIN_ESG1_NG=table2struct(CAIN_ESG1_NG_Performance, 'ToScalar', true)


%% Load Boiler Tables
 load 'C:\Users\Derek\Desktop\DataCollection\Needed Files (Not Including
Exhaust Model)\TempInCelciousVsGramsPerL.mat'
 CvsGramsPerLiter=table2struct(TempInCelciousVsGramsPer, 'ToScalar', true);
 load 'C:\Users\Derek\Desktop\DataCollection\Needed Files (Not Including
Exhaust Model)\Temperature_K_vs_Cp_Kg_KgK.mat'
 KvsCpKjperKgK=table2struct(TemperatureKvsCpKgKgK, 'ToScalar', true);
```

Hybrid Control Script for HHPDE CHP Hybrid System

```
%Hybrid System Control Design CAT3512C Diesel Engine
%This code is meant to be used to program ECOCELL on Simulink for its most
%efficienct performance

%% %Load  Cycle
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Tripping
Pipe 1
hour\TP_Cycle_1_Hour_3_Engines_on_20200229_From_1631_to_1731_Boiler_on.mat'
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 2 engines Boiler
On\Drilling_Cycle_1_hour_2_Engines_on_20200228_From_1630pm_to_1730pm_Boiler_O
n.mat'
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 1 hour 3 Engines Boiler
On\Drilling_Cycle_1_hour_3_Engines_on_20200229_From_0300_to_0400_Boiler_on.ma
t'
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 24 hours Boiler
On\Drilling_Cycle_24_hours_3_Engines_On_20200229_From_0000am_to_0000am_Boiler
_on.mat'
emissions1 = table2struct(emissions1,'ToScalar', true);
emissions2 = table2struct(emissions2,'ToScalar', true);
emissions3 = table2struct(emissions3,'ToScalar', true);

Ambient=[emissions1.Time emissions1.Tc7Ambient];
%% Load Load Adjustment Chart from SOC and ROC
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON
UTI\WasteHeatModel\Hybrid System\Change_in_Load_Chart.mat'
% Change_in_Load_Chart=table2struct( Change_in_Load_Chart, 'ToScalar', true);
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON
UTI\WasteHeatModel\Hybrid System\Adjustment_in_Load_Rate_Of_Charge.mat'
```

```matlab
Adjustment_in_Load_Rate_Of_Charge=table2struct( AdjustmentinLoadRateOfCharge,
'ToScalar', true);
%% Average Power Demand
Average_1=mean(emissions1.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100)
Average_2=mean(emissions2.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100)
Average_3=mean(emissions3.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100)
Average_Total=Average_1+Average_2+Average_3

Demand_1=emissions1.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100 ;
Demand_2=emissions2.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100 ;
Demand_3=emissions3.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100 ;
Total_Demand=Demand_1+Demand_2+Demand_3;
Total_Demand=[emissions1.Time Total_Demand];

Max_Demand=max(Total_Demand(:,2))
Min_Demand=min(Total_Demand(:,2))
Average_Demand=mean(Total_Demand(:,2))
yyaxis left
plot(Total_Demand)

Time_Used=[emissions1.Time emissions1.Time  ];
%% Percent Load And Number of engines for average Power
A=zeros(length(Total_Demand), 1);
L=length(Total_Demand);
first=1;
second=1;
for i=1:L
   next=first+second;
   first=second;
   second=next;
end
% for i=1:L
% %      disp([Total_Demand(i)])
%     if  Total_Demand(i)>2797 && Average_Demand>1045.95
%         Number_of_Engines=2;
%     elseif Total_Demand(i)>3843.9 && Average_Demand>2091.9
%         Number_of_Engines=3;
%     else
%         Number_of_Engines=1;
%     end
%     A(i)=(Number_of_Engines);
%     hold on
% %      i=i+1;
% end
%yyaxis right
%plot(A)
Time=(emissions1.Time(end,1))  ;
Total_Energy_Demand_kJ=sum(Total_Demand./1); %Note: for files in 10Hz Divide
by 10, for files in 1Hz divide by 1
Total_Energy_Demand_kJ=(Total_Energy_Demand_kJ(1,2));
Average_Engine=Total_Energy_Demand_kJ./Time;
Engine_Percent_Load_Average=Average_Engine./0.95;
Energy_Balance=(Average_Engine-Total_Demand);
Energy_Balance=[emissions1.Time Energy_Balance]
figure(2)
plot(Energy_Balance(:,2))
```

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```matlab
yline(584)

yline (-1752)
Ennergy_Balance_Max=max(Energy_Balance)
Energy_Balance_Min=min(Energy_Balance)

%%  %Load Profile Build Up
if Max_Demand <=2797.95 && Average_Total<=(1101*0.95)
    Total_Demand_1_Engine=Total_Demand;
    Number_of_Engines=1
elseif Max_Demand<=3843.9 && Average_Total<=(1101*0.95*2)
        Total_Demand_2_Engines=Total_Demand;
        Number_of_Engines=2
elseif Max_Demand<=4889.85 && Average_Total<=(1101*0.95*3)
            Total_Demand_3_engines=Total_Demand;
            Number_of_Engines=3
end


%% Rate of charge/Discharge & Engine Load Profile

Engine_Average_Load=(Average_Total./Number_of_Engines).*100./(1101.*0.95);
Energy_Balance_Hybrid=((Engine_Average_Load./100.*1101.*0.95.*Number_of_Engin
es)-(Total_Demand));
Energy_Balance_Hybrid=[emissions1.Time Energy_Balance_Hybrid(:,2)]
Ideal_Engine_Percent_Load=Engine_Average_Load;
figure(3)
plot(Energy_Balance_Hybrid(:,1),Energy_Balance_Hybrid(:,2));

%% Energy Balance Load Aadjustment
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON
UTI\WasteHeatModel\Hybrid System\Energy_Balance_Load_Ajustment.mat'
% Energy_Balance_Load_Adjustment=table2struct(EnergyBalanceLoadAjustment,
'ToScalar', true)




%Battery Discharge Upper limit =1752kW for 12 min or 720 s
%Battery Charge upper limit= 584kW for 35 min or 2100 s

%% Hourly Calculations
For_Hourly_Average=Total_Demand(:,2)
Hourly_Demand_Average=movmean(For_Hourly_Average,3600)
Total_Demand_Hourly_Demand=[emissions1.Time Hourly_Demand_Average]

%% Load Models For Simulation
%Load Exhasut Temperature Model
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Exhaust
Temperature Model\Diesel\DE_Exhaust_Temperature_Model.mat'
DE_Exhaust_Temperature_Model=table2struct(ExhaustTempFixedS1, 'ToScalar',
true);
```

```
% Load Fuel Consumption Model
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\CAT3512 Fuel
Consumption\CAT3512_Fuel_Consumption.mat'
CAT3512_Fuel_Consumption=table2struct(CAT3512FuelModelFromSSS1, 'ToScalar',
true);
```

Hybrid Control Script for DNGE CHP Hybrid System

```
%Hybrid System Control for Waukesha L7144GSI Dedicated Natural Gas Engine
%% %Load  Cycle
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Tripping
Pipe 1
hour\TP_Cycle_1_Hour_3_Engines_on_20200229_From_1631_to_1731_Boiler_on.mat'
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 2 engines Boiler
On\Drilling_Cycle_1_hour_2_Engines_on_20200228_From_1630pm_to_1730pm_Boiler_O
n.mat'
%load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 1 hour 3 Engines Boiler
On\Drilling_Cycle_1_hour_3_Engines_on_20200229_From_0300_to_0400_Boiler_on.ma
t'
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON UTI\Cycles\Drilling
Cycle 24 hours Boiler
On\Drilling_Cycle_24_hours_3_Engines_On_20200229_From_0000am_to_0000am_Boiler
_on.mat'
emissions1 = table2struct(emissions1,'ToScalar', true);
emissions2 = table2struct(emissions2,'ToScalar', true);
emissions3 = table2struct(emissions3,'ToScalar', true);

Ambient=[emissions1.Time emissions1.Tc7Ambient];

%% Load Load Adjustment Chart from SOC and ROC
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON
UTI\WasteHeatModel\Hybrid System\Change_in_Load_Chart.mat'
% Change_in_Load_Chart=table2struct( Change_in_Load_Chart, 'ToScalar', true);
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON
UTI\WasteHeatModel\Hybrid System\Adjustment_in_Load_Rate_Of_Charge.mat'
Adjustment_in_Load_Rate_Of_Charge=table2struct( AdjustmentinLoadRateOfCharge,
'ToScalar', true);

%% Average Power Demand
Average_1=mean(emissions1.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100)
Average_2=mean(emissions2.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100)
Average_3=mean(emissions3.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100)
Average_Total=Average_1+Average_2+Average_3

Demand_1=emissions1.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100 ;
Demand_2=emissions2.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100 ;
Demand_3=emissions3.EnginePercentLoadAtCurrentSpeed.*1101.*0.95./100 ;
Total_Demand=Demand_1+Demand_2+Demand_3;
Total_Demand=[emissions1.Time Total_Demand];
Max_Demand=max(Total_Demand(:,2))
Min_Demand=min(Total_Demand(:,2))
yyaxis left
```

```matlab
plot(Total_Demand)

Time_Used=[emissions1.Time emissions1.Time  ];
Time=(emissions1.Time(end,1))  ;
Total_Energy_Demand_kJ=sum(Total_Demand./1); %Note: for files in 10Hz Divide
by 10, for files in 1Hz divide by 1
Total_Energy_Demand_kJ=(Total_Energy_Demand_kJ(1,2));
Average_Engine=Total_Energy_Demand_kJ./Time;
Engine_Percent_Load_Average=Average_Engine./0.95;
Energy_Balance=(Average_Engine-Total_Demand);
Energy_Balance=[emissions1.Time Energy_Balance]
figure(2)
plot(Energy_Balance(:,2))

yline(584)

yline (-1752)
Ennergy_Balance_Max=max(Energy_Balance)
Energy_Balance_Min=min(Energy_Balance)

%%  %Load Profile Build Up
if Max_Demand <=2797.95 && Average_Total<=(1101*0.95)
   Total_Demand_1_Engine=Total_Demand;
   Number_of_Engines=1
elseif Max_Demand<=3843.9 && Average_Total<=(1101*0.95*2)
      Total_Demand_2_Engines=Total_Demand;
      Number_of_Engines=2
elseif Max_Demand<=4889.85 && Average_Total<=(1101*0.95*3)
         Total_Demand_3_engines=Total_Demand;
         Number_of_Engines=3
end
%% Rate of charge/Discharge & Engine Load Profile

Engine_Average_Load=(Average_Total./Number_of_Engines).*100./(1253.*0.95);
Energy_Balance_Hybrid=((Engine_Average_Load./100.*1253.*0.95.*Number_of_Engin
es)-(Total_Demand));
Energy_Balance_Hybrid=[emissions1.Time Energy_Balance_Hybrid(:,2)]
Ideal_Engine_Percent_Load=Engine_Average_Load;
figure(3)
plot(Energy_Balance_Hybrid(:,1),Energy_Balance_Hybrid(:,2));

%% Energy Balance Load Aadjustment
load 'C:\Users\Derek\Google Drive\WVU\Rig339 PATTERSON
UTI\WasteHeatModel\Hybrid System\Energy_Balance_Load_Ajustment.mat'
% Energy_Balance_Load_Adjustment=table2struct(EnergyBalanceLoadAjustment,
'ToScalar', true)




%Battery Discharge Upper limit =1752kW for 12 min or 720 s
%Battery Charge upper limit= 584kW for 35 min or 2100 s
```

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```matlab
%% Hourly Calculations
For_Hourly_Average=Total_Demand(:,2)
Hourly_Demand_Average=movmean(For_Hourly_Average,3600)
Total_Demand_Hourly_Demand=[emissions1.Time Hourly_Demand_Average]

%
```

9.3 Appendix C: CHEMCAD Heat Exchangers TEMA Sheets
HHPDE E-HEX TEMA Sheet:

```
                         Simulation: HHPDE E-HEX


                              TEMA SHEET
 1                            ----------
 2 Customer                                           Ref No.
 3 Address                                            Prop No.
 4 Plant Loc.                                         Date      Rev
 5 Service of Unit                                    Item
 6 Size 7.0ft x 8.0ft Type AXL (Hor/Vert) H Connected in  1 Para  1 Seri
 7 Surf/Unit(G/E) 8728.9/8592.5 ft2; Shell/Unit 1.000000   Surf/Shell
8728.9/8592.5 ft2
 8                      PERFORMANCE OF ONE UNIT
 9 Type of Process              Sensible            Forced Evap
 10 Fluid Allocation            Shell Side          Tube Side
 11 Fluid Name                  Exhaust
 12 Flow                        14633.0              1743.9       lb/h
 13 Liquid                          0.0              1743.9       lb/h
 14 Vapor                       14633.0                 0.0       lb/h
 15 NonCondensable              0.00000             0.00000       lb/h
 16 Steam                         694.7                 0.0       lb/h
 17 Evap/Cond                       0.0              1743.9       lb/h
 18 Density         0.03/0.00  /  0.05/0.00    0.27/61.02 /  0.27/55.79
lb/ft3
 19 Conductivity    0.03/0.00  /  0.02/0.00    0.02/0.38  /  0.02/0.39
Btu/hr-ft-F
 20 Specific Heat   7.47/0.00  /  7.07/0.00   11.32/18.01 / 11.32/18.85
Btu/lbmol-F
 21 Viscosity       0.03/0.00  /  0.02/0.00    0.02/0.41  /  0.02/0.15
cP
 22 Latent Heat                    0.00              875.79       Btu/lb
 23 Temperature(In/Out)     747.680/254.920   158.067/344.164   F
 24 Operating Pressure        14.70              124.70          psia
 25 Fouling Factor               0.001500           0.002000  hr-ft2-
F/Btu
 26 Velocity                     8.83                1.01        ft/sec
 27 Press Drop Allow/Calc   1.450/0.684     1.450/0.162    psi
 28 Heat Exchanged 1.857e+000 MMBtu/h;  MTD(Corrected): 40.64   F
 29 Transfer Rate, Service: 5.3     Calc: 5.3      Clean: 5.4    Btu/hr-
ft2-F
 30                       CONSTRUCTION DATA/SHELL          Sketch
 31                          Shell Side       Tube Side
 32 Design/Test Press psia 0.000000/Code       0.000000/Code
 33 Design Temperature   F      0.000          0.000
 34 No. Passes per Shell        1              16
 35 Corrosion Allowance  ft     0.000          0.000
 36 Connections  IN ID    ft     1.104          0.087
 37 Size &       OUT ID   ft     0.665          0.206
 38 Rating
 39 Tube No. 5557 OD 0.063 ft;Thk. 0.0054 ft;Length. 8.00  ft;Pit. 0.078
ft; Ptn. 60
 40 Tube Type      Bare     Material  TP 304 Stn. Stl.
```

```
41 Shell  A-240-304      7.00 ID   7.08 OD ft         Shell Cover
42 Channel or Bonnet        A-240-304     Channel Cover
43 Tubesheet Stationary     A-240-304     Tubesheet Floating
44 Floating Head Cover                    Impingement Protection: Yes
45 Baffles Cross  A-240-304 Type SSEG  Cut(Diam) 15 Spacing C/C   1.53 ft
46 Baffles Long                          Seal Type
47 Supports Tube C.S.                    U-Bend
48 Bypass Seal Arrangement               Tube-Tubesheet Joint
49 Expansion Joint No.                   Type
50 Rho-V2-Inlet Nozzle 564.68   Bundle Entrance         Bundle Exit
51 Shell Side                            Tube Side
52 Gasket Floating Head
53 Code Requirements                     Tema Class          ASME
54 Weight/Shell
55 Remarks:  Pressure drop shown is total pressure drop.
56
57

CHEMCAD 7.1.4                                        12/14/2020
1:30 PM
```

DNGE E-HEX TEMA Sheet:

```
                    Simulation: DNGE E-HEX


             SUMMARY REPORT
             --------------
 General Data:                    Heat Transfer Data:
  Exch Class/Type      ASME/AXL   Effective Transfer Area   14557.64
  Shell I.D.               8.00   Area Required             14436.63
  Shell in Series/Parallel  1/1   COR LMTD                     60.84
  Number of Tubes          7508   U (Calc/Service)       3.79/3.76
  Tube Length             10.00   Heat Calc                    3.36
  Tube O.D./I.D.   0.0625/0.0517  Heat Spec                    3.33
                                  Excess %                     0.84
  Tube Pattern            TRI60   Foul(S/T)     1.500E-003/2.000E-003
  Tube Pitch               0.08   Del P(S/T)          0.40/0.20
  Number of Tube Passes      16   SS Film Coeff                3.94
  Number of Baffles          11   SS CS Vel                    7.29
  Baffle Spacing           1.63   TW Resist               0.000589
  Baffle Cut, % Diameter     15   TS Film Coeff
212.23
  Baffle Type             SSEG   TS Vel                       1.34
  Baffle space def.      Edge-Edge

 Thermodynamics:
   K: Ideal Vapor Pressure
   H: SRK
   D: Library

 Number of Components: 2

 Calculation Mode: Design
```

```
Engineering Units:
 Temperature         F
 Flow/Hour           (lb/h)/h
 Pressure            psia
 Enthalpy            MMBtu
 Diameter/Area       ft/ft2
 Length/Velocity     ft/(ft/sec)
 Film                Btu/hr-ft2-F
 Fouling             hr-ft2-F/Btu
CHEMCAD 7.1.4                                              3/27/2021
4:02 PM
```

HHPDE JW-HEX TEMA Sheet:

```
                       Simulation: Jacket Water Heat Exchanger


                          TEMA SHEET
 1                        ----------
 2 Customer                                    Ref No.
 3 Address                                     Prop No.
 4 Plant Loc.                                  Date     Rev
 5 Service of Unit                             Item
 6 Size 1.3ft x 16.0ft Type AEL (Hor/Vert) H Connected in  1 Para  1 Seri
 7 Surf/Unit(G/E) 603.2/565.5 ft2; Shell/Unit 1.000000   Surf/Shell
603.2/565.5 ft2
 8                      PERFORMANCE OF ONE UNIT
 9 Type of Process          Sensible          Sensible
10 Fluid Allocation         Shell Side        Tube Side
11 Fluid Name               Coolant           water
12 Flow                     60616.1           47336.4       lb/h
13 Liquid                   60616.1           47336.4       lb/h
14 Vapor                        0.0               0.0       lb/h
15 NonCondensable           0.00000           0.00000       lb/h
16 Steam                        0.0               0.0       lb/h
17 Evap/Cond                    0.0               0.0       lb/h
18 Density     0.00/59.84 /  0.00/60.39   0.00/61.02 /  0.00/60.39
lb/ft3
19 Conductivity  0.00/0.39  /  0.00/0.39     0.00/0.38  /  0.00/0.39
Btu/hr-ft-F
20 Specific Heat  0.00/18.14 /  0.00/18.08    0.00/18.01 /  0.00/18.07
Btu/lbmol-F
21 Viscosity     0.00/0.28  /  0.00/0.33     0.00/0.41  /  0.00/0.33   cP
22 Latent Heat                  0.00              0.00      Btu/lb
23 Temperature(In/Out)     210.200/187.358   158.000/187.358   F
24 Operating Pressure         14.70             124.70        psia
25 Fouling Factor              0.002000          0.002000  hr-ft2-F/Btu
26 Velocity                    1.34              1.08        ft/sec
27 Press Drop Allow/Calc  30.290/7.174    1.450/0.324    psi
28 Heat Exchanged 1.392e+000 MMBtu/h;  MTD(Corrected): 20.84  F
29 Transfer Rate, Service: 118.1   Calc: 123.8    Clean: 273.2   Btu/hr-ft2-
F
30                      CONSTRUCTION DATA/SHELL          Sketch
31                        Shell Side      Tube Side
32 Design/Test Press psia 0.000000/Code       0.000000/Code
33 Design Temperature   F       0.000          0.000
34 No. Passes per Shell         1              2
35 Corrosion Allowance  ft      0.000          0.000
36 Connections  IN ID    ft      0.256          0.256
```

```
37 Size &        OUT ID    ft      0.296               0.256
38 Rating
39 Tube No. 192  OD 0.063 ft;Thk. 0.0054 ft;Length. 16.00 ft;Pit. 0.078 ft;
Ptn. 60
40 Tube Type        Bare     Material   1 Carbon Steel
41 Shell  A-285-C       1.27 ID   1.35 OD ft        Shell Cover
42 Channel or Bonnet       A-285-C       Channel Cover
43 Tubesheet Stationary    A-285-C       Tubesheet Floating
44 Floating Head Cover                   Impingement Protection: Yes
45 Baffles Cross  A-285-C   Type SSEG  Cut(Diam) 15 Spacing C/C   0.27 ft
46 Baffles Long                         Seal Type
47 Supports Tube C.S.                   U-Bend
48 Bypass Seal Arrangement              Tube-Tubesheet Joint
49 Expansion Joint No.                  Type
50 Rho-V2-Inlet Nozzle 1797.66  Bundle Entrance        Bundle Exit
51 Shell Side                           Tube Side
52 Gasket Floating Head
53 Code Requirements                    Tema Class            ASME
54 Weight/Shell
55 Remarks:  Pressure drop shown is total pressure drop.
56
57
```

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Cain Industries 'Savings Analysis Study'

Caterpillar 3512C Diesel Engine
Recovering Exhaust Energy To Generate Steam

Exhaust Heat Recovery Proposal
General Submittal Drawing
Terms of Sale and Warranty
Application Data Information

End User:
West Virginia University
1500 University Ave.
Morgantown, WV 26506

From:
Cain Industries, Inc.
W194 N11826 McCormick Dr.
Germantown, WI 53022

Ryan O'Connor
Sales Engineer
262-415-7875
ryan.oconnor@cainind.com

Ref: 63966   Rev: 1   12/9/2020

"Manufacturing Waste Heat Transfer Products To Save Energy"
Boiler Economizer Systems · Gas & Diesel Cogeneration Systems · Fume Incineration Systems
Exhaust Steam Generators · Finned Tubing

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heat recovery systems

Caterpillar 3512C Diesel Engine
Recovering Exhaust Energy To Generate Steam

Ref: 63966
Rep: 24999015
Rev: 1

Date: 12/9/2020
Page: 1

Engineered For:

West Virginia University
1500 University Ave.
Morgantown, WV 26506

Attn: Diego Dranuta
Ph: 304-389-7302

End User:

West Virginia University
1500 University Ave.
Morgantown, WV 26506

Attn: Diego Dranuta
Ph: 304-389-7302

Exhaust Heat Recovery Proposal:

Cain Industries is pleased to propose the following ESG1 model heat recovery steam generator to recover exhaust heat from a Caterpillar 3512C Diesel Engine and produce 110 psig steam output. The ESG1 package model comes complete with automatic control for either Supplemental (operating with fired steam boilers) or Primary (standalone) steam output.

- The proposed ESG1 product model will reduce the exhaust temperature gas inlet 747ºF to 371ºF, with a 27ºF pinch point @ 344ºF saturated steam temperature.
- The recovered 1,594,668 BTU/hr to steam output shall produce 1,643 #/hr operating at 110 psi based @ 252ºF inlet feedwater temperature.
- See NOTE1 regarding minimum inlet feedwater temperatures and treated water quality requirements.

• The basic ESG1 features a packaged and skid mounted, forced circulation, water tube, heat recovery steam generator.
• The factory tested ESG1 components contain a bypass section, heat exchanger section, and steam flash drum assembly.
• Included as standard is an automatic modulating exhaust gas bypass, controlled by steam pressure, as well as a modulating feedwater water control system.
• The heat transfer tubes are SA178 GrA carbon steel with carbon steel fins.
• Steam output can be produced in 5-10 minutes from a cold start-up.

The heat exchanger section (double seal welded outer shell to 10" W.C.) and the steam flash tank assembly are insulated with protective welded carbon steel exteriors. The exhaust gas bypass shall have 3" thickness insulation and a Stainless Steel protective exterior. After startup/commissoning, the buyer shall be responsible to field insulate the interconnecting piping as deemed necessary.

The Modulating Exhaust Gas Bypass Assembly construction will be designed where all materials in contact with the exhaust gas shall be Stainless Steel.

In the event of loss of electrical power or during an alarm condition, i.e., high water, low water, excess steam pressure, the bypass will be driven to the exhaust bypass position (diverting 99%+ exhaust away from the ESG1).

The steam pressure controller will have Modbus communications for re-transmission of the

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Caterpillar 3512C Diesel Engine
Recovering Exhaust Energy To Generate Steam

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process variable and remote set point capability. The water level controller will also have Modbus communications for re-transmission of the process variable. Contacts will be provided to remotely drive the exhaust gas bypass to the bypass position. The panel is UL approved.

The ESG1 must be installed indoors within a heated/cooled enclosure protected from freezing and/or from excessive heat above 120°F.

Optional Equipment: Add $3639 for auto surface blowdown installed. Add an additional $1190 to add MODBUS communications to the auto surface blowdown controller.

All prices are quoted in US currency. The quoted price does not include freight, export crating, any taxes or duties, or start-up services. See the attached "Terms of Sale" bulletin for start-up service requirements and rates, f.o.b. Cain Industries.

NOTE: The ESG1 product line model is typically selected at a feed water temperature of 212°F as the recommended minimum inlet temperature for softened and chemical treated boiler water. An optional internal economizer, for higher efficiency and closer approach point, can be used when the available feedwater temperature is above the minimum 150°F. An optional external feedwater preheater is also available if the feedwater temperature is below 150°F.

*Upon receipt of written purchase order, Cain Industries shall submit within 7-10 working days, Customer Approval Drawing(s) for review. The product drawing shall be drawn to 'detailed scale' and submitted in the form of AutoCAD .dwg and Adobe .pdf file formats. The submittal drawing shall include various views for plan and elevation customer incorporation into other drawings as required. It is important that these drawing(s) be returned within 60 days from issuance to guarantee quoted price, signed, dated, and marked "Approved for Production" while also forming as a part of the Terms of Sale, and begin the delivery schedule in the proposal.

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Caterpillar 3512C Diesel Engine
Recovering Exhaust Energy To Generate Steam

Ref: 63966
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Waste Heat Exhaust:

Primary Fuel Type:	#2 Fuel Oil
Secondary Fuel Type:	#2 Fuel Oil
Fuel Cost per 100,000 BTU (USD):	
Heat Source:	Caterpillar 3512C Diesel Engine
Exhaust Flow Direction:	Vertical

Model: ESG1-B30B14.5CSS

Overall Length, inches	139	Customer Connections:	18" Dia.
Overall Width, inches	58	Exhaust Connections	8-18"Dia.
Overall Height, inches	110	Steam, RF Flange	3
Dry Weight, lbs.	8000	Boiler Feedwater, NPT	
Wet Weight (6.8 ft³/43 imp gal), lbs.	8430	Blowdown Manifold	
Surface Area, Ft2	2,345	Steam Safety Valve, NPT	
Design Pressure, PSIG	150	Circulator Motor HP	3
Operating Steam Pressure, PSIG	110	Voltage	230/460
Hydrostatic Test Pressure, PSIG	225	Phase 3	60
Fin Design Temperature, °F	550		
Maximum Entering Temperature, °F	900		

Performance:

	Load 1	Load 2	Load 3
Load of Maximum Output, %	100%	75%	50%
Exhaust Entering Temp, °F	747°	736°	741°
Exhaust Flow Rate, SCFM	3,486	2,705	1,955
Exhaust Leaving Temp, °F	371°	363°	355°
Pressure Drop " W.C. Max	1.56	1.01	0.59
*Equivalent Evaporation, lb/hr	1643	1260	943
Boiler Horsepower, BHP	48	37	27
Boiler Feedwater Flow, GPM	3.5	2.7	2.0
+ Sensible Heat Dry Gas	1,339,916	1,028,046	768,848
+ Sensible Heat of H2O Vapor	254,752	195,458	146,178
+ Latent Heat of Condensation	N/A	N/A	N/A
TT BTU/hr Recovered	1,594,668	1,223,504	915,026

(*From 251.6°F feedwater to 110 PSIG steam)

Savings:

	Load 1	Load 2	Load 3
Therms (100,000 BTU) Saved/Hr	37.968	29.842	23.462
Annual Hours of Operation			
Annual Savings Per Load			

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Recovering Exhaust Energy To Generate Steam

Ref: 63966
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Quotation:

Qty	Part #	U/M	Description
1		EACH	ESG1-B30B14.5CSS
			-INCLUDING:
			Skid Mounted Packaged Design
			Full Modulating Exhaust Bypass
			Full Boiler Control Trim
			Factory Insulation
			Water Level Control
			Low Exhaust Friction Loss
			7 Ga. Steel Exterior
			Stainless Steel Inner Lining
			ASME & Nat'l Board Stamped
	914014		55-135 PSIG O.P. (Design:150)
	915015		304 SS Exhaust Gas Bypass
2	974040	EACH	18" Stainless Steel Exhaust Flange
	940060		Alarm Dry Contacts
1	940065		Remote Bypass Contacts
1	940070		Modbus Communications
1	930040		UL Approval - Control Panel
	964025		3" Thks. Factory Insulation
1	964770	EACH	Bypass Insulation ESG1-AXXB 3"T
1	967010	EACH	Corrosion Resistant Coating
	950050		Rotary Electric Sootblower
14203			Internal Economizer

TOTAL PRICE (USD) $179,354

Terms of Sale:
* Estimated Shipping: 14-16 weeks after submittal approval
* Payment Terms: See "Terms of Sale" Bulletin #25500
* See Bulletin 25500 including 'Warranty and Performance Guarantee'.
* Savings projections will vary from fluctuating fuel costs, operating hours, & equipment care

Ryan O'Connor

Ryan O'Connor
Sales Engineer

120920-16.25-CI-RO

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Caterpillar 3512C Diesel Engine

Ref: 63980　　　　　　　　　　　　　　　　Date: 12/9/2020
Rep: 999　　　　　　　　　　　　　　　　　Page: 2
Rev: 0

Waste Heat Exhaust:
Primary Fuel Type:	Natural Gas
Fuel Cost per 100,000 BTU (USD):	
Heat Source:	Caterpillar 3512C Diesel Engine
Exhaust Flow Direction:	Vertical
Heat Sink:	
Liquid Type:	Water

Model: ESGe-220A14.5ALS
Overall Configuration, inches	24.1x76
Overall Height, inches	8.3
Liquid Connection	3/4" NPT
Exhaust Connection	20.9x60.3
Dry Weight, lbs.	230
Wet Weight (0.8 ft³/5 imp gal), lbs.	280
Surface Area, Ft2	272
Design Pressure, PSIG	250
Hydrostatic Test Pressure, PSIG	375
@ Design Temperature, °F	550
Maximum Entering Temperature, °F	900

Performance:

	Load 1	Load 2	Load 3
Load of Maximum Output, %	100%	75%	50%
Exhaust Entering Temp, °F	346°	347°	345°
Exhaust Flow Rate, SCFM	3,317	2,619	1,837
Exhaust Leaving Temp, °F	312°	315°	316°
Pressure Drop " W.C. Max	0.39	0.25	0.12
Liquid Entering Temp, °F	158.0°	158.0°	158.0°
Liquid Flow Rate, GPM	3.3	2.5	1.9
Liquid Leaving Temp, °F	237.5°	237.0°	224.8°
Pressure Drop, PSIG	0.00	0.00	0.00
+ Sensible Heat Dry Gas	110,117	82,935	53,336
+ Sensible Heat of H2O Vapor	20,936	15,768	10,141
+ Latent Heat of Condensation	N/A	N/A	N/A
TT BTU/hr Recovered	131,054	98,703	63,477

Savings:

Therms (100,000 BTU) Saved/Hr	1.311	0.987	0.635
Annual Hours of Operation			
Annual Savings Per Load			

DATE: 12/9/2020 REF#: **63966** FOR: West Virginia University
REV#: **1** c/o: Cain Industries, Inc.

MODEL: ESG1-B30B14.5CSS
HEAT SOURCE: Caterpillar 3512C Diesel Engine

Bul.#10950

PARTS LIST
(SEE DESIGN DATA FOR CONSTRUCTION)
1. LIFTING EYE
2. STEAM PRESSURE CONTROLLER
3. MODULATING FEEDWATER VALVE
4. ASME STEAM SAFETY VALVE
5. STEAM OUTLET FLANGE (STEAM STOP & CHECK VALVE SUPPLIED BY OTHERS)
6. STEAM FLASH DRUM/DRY PIPES (INSULATED)
7. EXCESS STEAM PRESSURE SWITCH
8. WATER LEVEL CONTROL W/LOW & HIGH WATER CUTOUT & MANUAL RESET LOW LOW WATER
9. LOCALIZED PIPING CONNECTIONS: FEEDWATER, BLOWDOWN MANIFOLD, 100 PSIG CONTROL AIR (WHEN REQ'D FOR PNEUMATIC DAMPER ACTUATOR)
10. TUBE REMOVAL ACCESS DOOR
11. STRUCTURAL STEEL BASE
12. MODULATING EXHAUST BYPASS

13. FINNED TUBE HEAT TRANSFER SECTION
14. MAIN BLOW DOWN VALVE ASSY.
15. CIRCULATING PUMP ASSY W/SELF COOLING CIRCULATING PUMP (NO COOLING WATER REQUIRED)
16. NEMA 12 CONTROL PANEL: (SINGLE HIGH VOLTAGE CONN.) FUSE DISCONNECT MAGNETIC STARTER, 120V. STEPDOWN TRANSFORMER, ALARM LIGHTS
17. 10GA. SHELL (3" THKS. INSULATED)
18. CONTINUOUS SURFACE BLOWDOWN
19. MODULATING ACTUATOR W/FAILSAFE TO BYPASS
20. MAIN INSPECTION DOOR, HINGED
21. STAINLESS STEEL INTERIOR

NOTES:
A. OPTIONAL INSULATION AS REQ'D:
—SHELL THICKNESS INSULATION:
4"☐ 6"☐ 8"☐ 12"☐
—BYPASS JACKET INSULATION:
2"☐ 3"☐ 4"☐ 6"☐ 8"☐
(EXTERNAL PIPING INSULATION INSULATED BY OTHERS AS REQ'D)
B. EXHAUST FLANGE CONN.S:
2x2 ANGLE RING (STANDARD)
150LB ANSI HOLE PTN (OPTIONAL) ☐
C. A.S.M.E. & NAT'L BOARD STAMPED — SEC.1, DIV.1

ESG1

PERFORMANCE
AND
DIMENSION DATA

A.	747 °F
B.	371 °F
	110 PSI
	48 BHP
C.	139x58 "
D.	110 "
E.	3 " CONN.
F.	8-18"Dia. " CONN.
	8000 # WGT
	2,345 H.S.
	150 PSIG
	900 TEMP.

(FULL OPERATING POSITION)

(FULL BYPASS POSITION)

EXAMPLE ESG1 CONTROLS AND HEADERS

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The terms of the attached Limited Warranty are included in these Terms of Sale and are incorporated by reference herein. The following "Terms of Sale" forms as a part of the Cain Industries, Inc. equipment proposal as attached herein. All proposed pricing is quoted F.O.B. factory. All pricing is quoted in U.S. currency.

QUOTED DELIVERY TIME:
Delivery times quoted are appropriate for various product lines and based on conditions at the time of quotation. Cain Industries, Inc. will, in good faith, attempt to deliver the equipment within the time quoted. In no case shall Cain Industries, Inc. be liable for incidental or consequential damages resulting from failure to meet requested or quoted delivery schedules. Quoted delivery time is based from the date of receipt of an approved written purchase order including an initial down payment, as required, with either written authorization to proceed with fabrication -or- from date of receipt of submittal drawings (less 10 working days) when required.

OFFER EXPIRATION:
All offers expire 60 days from the quotation date unless otherwise stated and are subject to cancellation by Cain Industries, Inc. at any time prior to the formal acceptance of our offer to furnish equipment quoted.

SUBMITTAL DRAWINGS:
Submittal drawings are issued 5-10 working days from receipt of written purchase order, when required by either Cain Industries, Inc. and/or the Buyer, and must be returned (marked "Approved for Production", signed, and dated) in order to initiate production. Production cannot begin until the approved submittal drawings are returned.

SHIPMENT OF GOODS:
Unless otherwise specifically agreed, all shipments are made F.O.B. Factory via "best way" and shipped prepaid and add. Cain Industries, Inc. responsibility ceases upon acceptance by the carrier. SHOULD GOODS BECOME LOST OR DAMAGED IN SHIPMENT, THE PURCHASER OR RECIPIENT OF THE GOODS MUST IMMEDIATELY NOTIFY AND PLACE CLAIM WITH THE CARRIER, ADVISE CAIN INDUSTRIES, INC. OF ANY DAMAGE OR DISCREPANCY, AND OBTAIN AUTHORIZATION FOR RETURN OR REPLACEMENT. As a courtesy, Cain Industries, Inc. will assist in tracing and recovering lost goods and the collection of just claims, but cannot guarantee safe delivery. Loss or damage in shipment does not release the purchaser from payment of the total invoice.

PAYMENT-ESTABLISHED ACCOUNTS:
Payments for established accounts with a credit limit are due on or before the Net 30 days from date of invoice due date and coinciding with shipment date and/or 'ready for shipment date'.

EXPEDITING:
Expediting charges may be issued in order to improve delivery depending on the shorter delivery time required. Contact Cain Industries, Inc. for pricing for the best possible delivery.

STORAGE:
When the equipment is ready for shipment it will be shipped to the 'ship to' address noted on the Sales Order, unless otherwise indicated. Should there be a request to hold the equipment beyond the 'ready for shipment date', Cain Industries, Inc. will store the equipment for up to 30 days at no cost providing storage space is available. Contact Cain Industries, Inc. for storage costs when equipment is expected to be stored for more than 30 days. If storage space is unavailable the buyer agrees to make provisions to receive the equipment when it becomes ready for shipment.

MINIMUM BILLING:
The minimum order is $100.00, plus shipping costs.

CREDIT LIMIT:
Accounts over credit limit will be on a "Cash with Order" basis until account is brought to below "Credit Limit" status. Special circumstances may occur where credit limits may be adjusted for companies with past credit history satisfactory to Cain Industries, Inc.

TAXES OR SURCHARGES:
Quoted prices do not include sales, use, excise, occupation, processing transportation or other similar taxes which Cain Industries, Inc. may be required to pay or collect with respect to any of the quoted materials. Such taxes which are or may be incurred shall be paid by the purchaser.

PAYMENT-NEW ACCOUNTS:
An initial purchase order received from a new account shall require a 50% down payment with the order, receipt of the completed credit application for immediate processing, and the balance due prior to shipment; or 30% with purchase order and receipt of the completed credit application in conjunction with credit limit and/or progress payment schedules. Orders will be held until credit limit has been established. Allow a 3 week processing period to complete the credit check.

PAYMENT-ORDERS OUTSIDE THE UNITED STATES:
For purchase orders received wherein the final equipment destination and/or the Buyer is located outside the continental United States, payments shall be made according to the guide lines as set forth herein. All payments shall be made in U.S. currency according to the terms of progress payments and shall be paid in full prior to shipment outside the continental United States. Letters of Credit are not acceptable. Any additional costs associated with international payments such as but not limited to: currency adjustments, tariffs, and special taxes, etc. shall be the responsibility of the Buyer.

SERVICE CHARGE:
A 2% per month service charge will be assessed on all past due amounts.

PROGRESS PAYMENT SCHEDULES:
The following are payment schedules for orders exceeding credit limit:

- For purchase orders of $25,000 to $50,000:
  - 30% due with purchase order
  - 30% due at 45 days from receipt of approval drawings
  - Balance due 30 days from shipment.

- Over $50,000 or required for the ESG product orders:
  - 15% due with purchase order
  - 15% due with submittal approval drawings
  - 30% due 45 days from receipt of approved submittal drawings
  - 30% due prior to shipment
  - Balance due 30 days from shipment.

CANCELLATION AND CHANGES:
As many Cain Industries, Inc. products are manufactured and/or adjusted "to order", orders accepted and acknowledged by Cain Industries, Inc. are not subject to change or cancellation without prior consent of Cain Industries, Inc. Down payments are non-cancelable. Order quantity reductions or cancellations, if granted, will be subject to cancellation charges as a percentage of completion based on the period of time from the date released for fabrication through to the estimated ship date.

EQUIPMENT STARTUP & SERVICE:
Pricing for equipment startup or service for all installation locations are quoted per application. Travel, lodging, and subsistence expenses are in addition. Contact Cain Industries, Inc. for a quotation. Startup can only be initiated upon receipt of completed Pre-Startup form. ESG & ESG1 boiler startups must be completed by authorized Cain Industries, Inc. personnel to allow the warranty to become effective, unless otherwise stated in a written agreement issued by Cain Industries, Inc. to the Buyer.

RETURN OF GOODS FOR WARRANTY REPAIR, REPLACEMENT, OR CREDIT:
Authorization to return goods for any reason must be obtained from Cain Industries, Inc. prior to the return of the shipment being made. All items returned for repair, replacement or credit shall be returned freight prepaid. Freight collect shipments will not be accepted. A 30% "minimum" restocking charge will be made on all items returned for credit. Cancellation and/or restocking charges will apply to the balance of the order pending with a maximum of 90% as determined at the point of cancellation dependent on the work in process. Quantities shipped prior to the point of cancellation shall be issued an additional invoice for the difference in price breaks between the original quantity ordered and the total shipped up to the point of cancellation.

PROPRIETARY DATA:
All manufacturing drawings, specifications and technical material submitted by Cain Industries, Inc. are the property of Cain Industries, Inc. and are to be considered as confidential. Except for its original intent the submittal information supplied herein attached cannot be copied, transferred, or used in any way without the express written authorization from Cain Industries, Inc.

LIMITATION OF REMEDIES:
Cain Industries, Inc.'s liability is limited exclusively to its obligations under the attached **Limited Warranty**, the terms of which are incorporated by reference herein. Buyer agrees that in no event will Cain, Industries, Inc. be liable for cost of processing, loss of profits, or any other consequential or incidental damages or cost of any kind resulting from the order and or use of its product, whether arising from breach of warranty, non-conformity to order specifications, delay in delivery or any other loss sustained by buyer.

Cain Industries, Inc. PO Box 189 Germantown, WI 53022 • USA • 262-251-0051 • 800-558-8690 • Fax: 262-251-0118 • sales@cainind.com

021220 Cain Industries, Inc.

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EXHAUST STEAM GENERATORS

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Manufacturing Waste Heat Transfer Products To Save Energy

Boiler Economizer Systems · Gas and Diesel Cogeneration Systems · Fume Incineration Systems · Exhaust Steam Generators · Finned Tubing

GENERAL APPLICATION DATA - Request For Quote:

DATE: 11/30/20

Completed By: Diego Dranuta
Company: WVU
Email Address*: dgdranuta@mix.wvu.edu

Initial Inquiry: *
- ○ 2019 pdf email inquiry ○ phone misc inquiry ○ email misc inquiry
- ○ 2019 tear form inquiry ○ fax misc inquiry ○ mail misc inquiry

☐ Lead

Proposal To: * Fields with an asterisk are required

Company Name:	West Virginia University
Address:	1306 Evansdale Dr
City, State/Province:	Morgantown WV Postal Code: 26505
Country:	United States
Contact Name:	Diego Dranuta
Email Address:	dgdranuta@mix.wvu.edu
Phone:	3043897302 Fax:

End User: ☑ Same As Proposal To

Company Name:	West Virginia University
Address:	1306 Evansdale Dr
City, State/Province:	Morgantown WV Postal Code*: 26505
Country:	United States
Contact Name:	Diego Dranuta
Email Address:	dgdranuta@mix.wvu.edu
Phone:	3043897302 Fax:

1. Exhaust to Liquid Heating Source:
- ○ Oven
- ○ Furnace
- ○ Thermal Oxidizer
- ○ Other
- ○ Fume Incinerator
- ○ Natural Gas Engine
- ◉ Diesel Engine

Burner Type:
- ○ Atmospheric
- ◉ Turbocharged
- ○ Power Burner
- ○ Naturally Aspirated

2. Exhaust Source Description: * ○ New or ◉ Retrofit

Make:	CAT
Model:	3512C
BTU/Hr. input:	3077752.084 Number of Units: 3

For Engine Exhaust Retrofit:

kW@ (engines):	1101
In.³ (engines):	3158
RPM (engines):	1200

3. Design Specifications:
Design Pressure: 110 PSIG
Operating Pressure: 110 PSIG

4. Exhaust Stack Description:
- ◉ Vertical gas flow direction
- ○ Horizontal gas flow direction

Stack Type:
- ○ Rectangular Dimensions: ___ X ___
- ◉ Cylindrical Diameter: 18 in

Economizer Space Limitations:
- ◉ None or ○ ___

5. Type Of Fuel Burned:

Primary
- ○ Natural Gas
- ○ Propane
- ◉ #2 Fuel Oil
- ○ #4, 5, 6 Fuel Oil
- ○ Landfill Gas
- ○ Biogas/Digester Gas

Btu / Ft3: 129305

Standby *(for dual fuel burners)*
- ○ Natural Gas
- ○ Propane
- ◉ #2 Fuel Oil
- ○ #4, 5, 6 Fuel Oil
- ○ Landfill Gas
- ○ Biogas/Digester Gas

Btu/Gallon:

6. Exhaust Gas Flow Entering: (Maximum pressure drop @ 100% load: 40 inches W.C.)

	Load 1	Load 2	Load 3	Load 4
Temperature (°F):	746	735.8	740.66	
○ SCFM, or ◉ ACFM, or ○ #/hr:	7598	5931	4178	
Desired Outlet (°F):				
Content O2%:				
or Content CO2%:	6	6.5	7	
Content Excess Air%:	42	41	39	
Thermal Efficiency %:				
Load % of Input:	**100**	75	50	
Hours of Operation per Load:	1500	1500	1500	

7. Liquid Flow Entering: (Maximum pressure drop @ 100% load: ___ PSIG)

Temperature (°F):				
○ GPM, or ○ #/hr:				
Desired Outlet (°F):				

8. Desired BTU / Hr. Recovery: 6000000 BTU/Hr

9. Savings Analysis Information:
Fuel Cost Per 100,000 Btu: $ 1.706
Total hours per year operation (loads 1 + 2 + 3 + 4): 4500
Hours per day operation: 16
Days per week operation: 7
Weeks per year operation: 40

10. Justification for purchase:
Payback (months): ___   Return on Investment (%): ___

Attachments:
Please attach any files that you would like to submit with this request.

Notes:
Provide additional specifications or requirements (i.e. primary voltage, NEMA, etc.):
3 CAT 3512C Engines run on site; therefore, 3 ESGs are required. 6 million BTU/hr is the current heat supply capability covered by a HURTS SERIES 300, 200 bhp. This investment is to reduce or eliminate the usage of the boiler. Jacket water is also available, up to 412 kW per engine (see LEHW0078-02.pdf).

Attention:

Your combustion source description as listed above including operating conditions, fuel costs, etc. has either been instrument measured and/or noted by you. Because our quotation is a detailed analysis of expected savings for your specific application, please verify that the information is complete and accurate. This will allow us to proceed with compiling a comprehensive proposal for a Cain Industries, Inc. fuel-saving economizer system for your review.

Verified By*: ☑

DESIGNED, ENGINEERED &
HAND BUILT IN THE USA

Cain Industries
'Savings Analysis Study'

Waukesha L7044GSI Natural Gas Engine
Recovering Exhaust Energy To Generate Steam

Exhaust Heat Recovery Proposal
General Submittal Drawing
Terms of Sale and Warranty
Application Data Information

End User:

West Virginia University
1500 University Ave.
Morgantown, WV 26506

From:

Cain Industries, Inc.
W194 N11826 McCormick Dr.
Germantown, WI 53022

Ryan O'Connor
Sales Engineer
262-415-7875
ryan.oconnor@cainind.com

Ref: 64070   Rev: 0   1/20/2021

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Waukesha L7044GSI Natural Gas Engine
Recovering Exhaust Energy To Generate Steam

Ref: 64070
Rep: 24999015
Rev: 0

Date: 1/20/2021
Page: 1

Engineered For:

West Virginia University
1500 University Ave.
Morgantown, WV 26506

Attn: Diego Dranuta
Ph: 304-389-7302

End User:

West Virginia University
1500 University Ave.
Morgantown, WV 26506

Attn: Diego Dranuta
Ph: 304-389-7302

Exhaust Heat Recovery Proposal:

Cain Industries is pleased to propose the following ESG1 model heat recovery steam generator to recover exhaust heat from a Waukesha L7044GSI Natural Gas Engine and produce 110 psig steam output. The ESG1 package model comes complete with automatic control for either Supplemental (operating with fired steam boilers) or Primary (standalone) steam output.

- The proposed ESG1 product model will reduce the exhaust temperature gas inlet 1,098°F to 412°F, with a 68°F pinch point @ 344°F saturated steam temperature.
- The recovered 2,200,310 BTU/hr to steam output shall produce 2,375 #/hr operating at 110 psi based @ 295°F inlet feedwater temperature.
- See NOTE1 regarding minimum inlet feedwater temperatures and treated water quality requirements.

• The basic ESG1 features a packaged and skid mounted, forced circulation, water tube, heat recovery steam generator.
• The factory tested ESG1 components contain a bypass section, heat exchanger section, and steam flash drum assembly.
• Included as standard is an automatic modulating exhaust gas bypass, controlled by steam pressure, as well as a modulating feedwater water control system.
• The heat transfer tubes are SA178 GrA carbon steel with carbon steel fins.
• Steam output can be produced in 5-10 minutes from a cold start-up.

The heat exchanger section (double seal welded outer shell to 10" W.C.) and the steam flash tank assembly are insulated with protective welded carbon steel exteriors. The exhaust gas bypass shall have 6" thickness insulation and a Stainless Steel protective exterior. After startup/commissoning, the buyer shall be responsible to field insulate the interconnecting piping as deemed necessary.

The Modulating Exhaust Gas Bypass Assembly construction will be designed where all materials in contact with the exhaust gas shall be Stainless Steel.

In the event of loss of electrical power or during an alarm condition, i.e., high water, low water, excess steam pressure, the bypass will be driven to the exhaust bypass position (diverting 99%+ exhaust away from the ESG1).

The steam pressure controller will have Modbus communications for re-transmission of the

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Waukesha L7044GSI Natural Gas Engine
Recovering Exhaust Energy To Generate Steam

Ref: 64070
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Date: 1/20/2021
Page: 2

process variable and remote set point capability. The water level controller will also have Modbus communications for re-transmission of the process variable. Contacts will be provided to remotely drive the exhaust gas bypass to the bypass position. The panel is UL approved.

The ESG1 must be installed indoors within a heated/cooled enclosure protected from freezing and/or from excessive heat above 120°F.

Optional Equipment: Add $3639 for auto surface blowdown installed. Add an additional $1190 to add MODBUS communications to the auto surface blowdown controller.

All prices are quoted in US currency. The quoted price does not include freight, export crating, any taxes or duties, or start-up services. See the attached "Terms of Sale" bulletin for start-up service requirements and rates, f.o.b. Cain Industries.

NOTE: The ESG1 product line model is typically selected at a feed water temperature of 212°F as the recommended minimum inlet temperature for softened and chemical treated boiler water. The included internal economizer, for higher efficiency and closer approach point, can be used when the available feedwater temperature is above the minimum 150°F. An optional external feedwater preheater is also available if the feedwater temperature is below 150°F.

*Upon receipt of written purchase order, Cain Industries shall submit within 7-10 working days, Customer Approval Drawing(s) for review. The product drawing shall be drawn to 'detailed scale' and submitted in the form of AutoCAD .dwg and Adobe .pdf file formats. The submittal drawing shall include various views for plan and elevation customer incorporation into other drawings as required. It is important that these drawing(s) be returned within 60 days from issuance to guarantee quoted price, signed, dated, and marked "Approved for Production" while also forming as a part of the Terms of Sale, and begin the delivery schedule in the proposal.

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Waukesha L7044GSI Natural Gas Engine
Recovering Exhaust Energy To Generate Steam

Ref: 64070
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Waste Heat Exhaust:

Primary Fuel Type:	Natural Gas
Fuel Cost per 100,000 BTU (USD):	
Heat Source:	Waukesha L7044GSI Natural Gas Engine
Exhaust Flow Direction:	Vertical

Model: ESG1-824B16CSS

Overall Length, inches	121	Customer Connections:	18" Dia.
Overall Width, inches	58	Exhaust Connections	8-18"Dia.
Overall Height, inches	110	Steam, RF Flange	3
Dry Weight, lbs.	8000	Boiler Feedwater, NPT	3/4
Wet Weight (4.8 ft³/30 imp gal), lbs.	8300	Blowdown Manifold	1
Surface Area, Ft2	1,615	Steam Safety Valve, NPT	1-1/4
Design Pressure, PSIG	150	Circulator Motor HP	3
Operating Steam Pressure, PSIG	110	Voltage	230/460
Hydrostatic Test Pressure, PSIG	225	Phase 3 Hertz	60
@ Design Temperature, °F	550		
Maximum Entering Temperature, °F	1250		

Performance:	Load 1	Load 2	Load 3	Load 4
Load of Maximum Output, %	100%	76%	52%	28%
Exhaust Entering Temp, °F	1098°	1156°	1185°	1179°
Exhaust Flow Rate, SCFM	2,553	2,258	1,775	1,126
Exhaust Leaving Temp, °F	412°	406°	390°	367°
Pressure Drop " W.C. Max	1.43	1.20	0.81	0.38
*Equivalent Evaporation, lb/hr	2375	2305	1922	1243
Boiler Horsepower, BHP	66	64	53	34
Boiler Feedwater Flow, GPM	5.2	5.0	4.2	2.7
+ Sensible Heat Dry Gas	1,848,805	1,794,465	1,496,736	967,716
+ Sensible Heat of H2O Vapor	351,505	341,174	284,568	183,988
+ Latent Heat of Condensation	N/A	N/A	N/A	N/A
TT BTU/hr Recovered	2,200,310	2,135,639	1,781,304	1,151,704
(*From 294.8°F feedwater to 110 PSIG steam)				

Savings:				
Therms (100,000 BTU) Saved/Hr	22.003	21.356	17.813	11.517
Annual Hours of Operation				
Annual Savings Per Load				

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Waukesha L7044GSI Natural Gas Engine
Recovering Exhaust Energy To Generate Steam

Ref: 64070   Date: 1/20/2021
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Rev: 0

Quotation:

Qty	Part #	U/M	Description
1		EACH	ESG1-824B16CSS
			-INCLUDING:
			Skid Mounted Packaged Design
			Full Modulating Exhaust Bypass
			Full Boiler Control Trim
			Factory Insulation
			Water Level Control
			Low Exhaust Friction Loss
			7 Ga. Steel Exterior
			Stainless Steel Inner Lining
			ASME & Nat'l Board Stamped
	914014		55-135 PSIG O.P. (Design:150)
	915015		304 SS Exhaust Gas Bypass
2	974040	EACH	18" Stainless Steel Exhaust Flange
	940060		Alarm Dry Contacts
1	940065		Remote Bypass Contacts
1	940070		Modbus Communications
1	930040		UL Approval - Control Panel
	964035		5" Thks. Factory Insulation
1	964773	EACH	Bypass Insulation ESG1-AXXB 6"T
1	967010	EACH	Corrosion Resistant Coating
1			Internal Economizer

TOTAL PRICE (USD) $139,523

Terms of Sale:
* Estimated Shipping: 14-16 weeks after submittal approval
* Payment Terms: See "Terms of Sale" Bulletin #25500
* See Bulletin 25500 including 'Warranty and Performance Guarantee'.
* Savings projections will vary from fluctuating fuel costs, operating hours, & equipment care

Ryan O'Connor
Sales Engineer

012021-08:44-CI-RO

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Economizer Performance

Ref: 64071		Date: 1/19/2021	
Rep: 999		Page: 2	
Rev: 0			

Waste Heat Exhaust:

Primary Fuel Type:	Natural Gas
Fuel Cost per 100,000 BTU (USD):	
Heat Source:	Waukesha L7044GSI Natural Gas Engine
Exhaust Flow Direction:	Vertical
Heat Sink:	
Liquid Type:	Water

Model: ESGe-224A16ALS

Overall Configuration, inches	24.1x88
Overall Height, inches	8.3
Liquid Connection	3/4" NPT
Exhaust Connection	20.9x72.3
Dry Weight, lbs.	270
Wet Weight (0.9 ft³/5 imp gal), lbs.	320
Surface Area, Ft2	424
Design Pressure, PSIG	250
Hydrostatic Test Pressure, PSIG	375
@ Design Temperature, °F	550
Maximum Entering Temperature, °F	1250

Performance:	Load 1	Load 2	Load 3	Load 4
Load of Maximum Output, %	100%	76%	52%	28%
Exhaust Entering Temp, °F	412°	406°	390°	367°
Exhaust Flow Rate, SCFM	2,553	2,258	1,775	1,126
Exhaust Leaving Temp, °F	341°	334°	320°	304°
Pressure Drop " W.C. Max	0.25	0.19	0.12	0.05
Liquid Entering Temp, °F	212.0°	212.0°	212.0°	212.0°
Liquid Flow Rate, GPM	5.2	5.0	4.2	2.7
Liquid Leaving Temp, °F	294.8°	289.9°	282.8°	274.2°
Pressure Drop, PSIG	0.01	0.01	0.00	0.00
+ Sensible Heat Dry Gas	180,923	163,473	124,952	70,560
+ Sensible Heat of H2O Vapor	34,398	31,080	23,757	13,415
+ Latent Heat of Condensation	N/A	N/A	N/A	N/A
TT BTU/hr Recovered	215,321	194,553	148,709	83,976

Savings:

	Load 1	Load 2	Load 3	Load 4
Therms (100,000 BTU) Saved/Hr	2.153	1.946	1.487	0.840
Annual Hours of Operation				
Annual Savings Per Load				

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DATE: 1/20/2021 REF#: **64070** FOR: West Virginia University
REV#: **0** c/o: Cain Industries, Inc.

MODEL: ESG1-824B16CSS
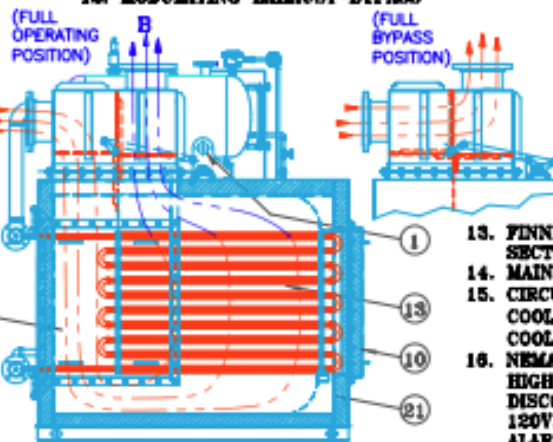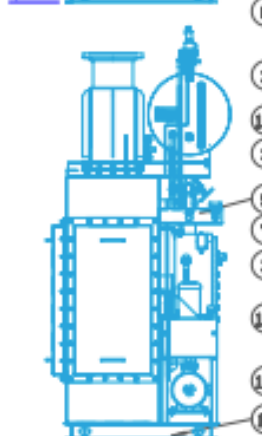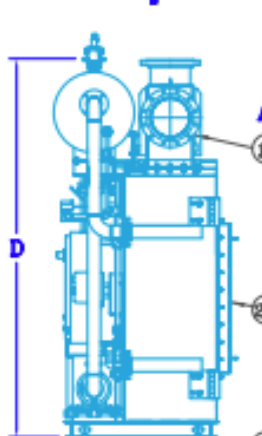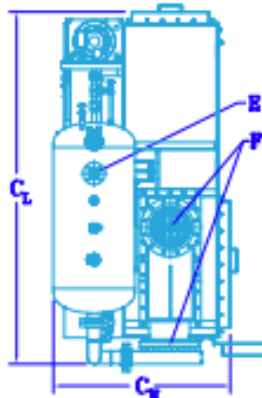HEAT SOURCE: Waukesha L7044GSI Natural Gas Engine

Bul.#10950

PARTS LIST
(SEE DESIGN DATA FOR CONSTRUCTION)
1. LIFTING EYE
2. STEAM PRESSURE CONTROLLER
3. MODULATING FEEDWATER VALVE
4. ASME STEAM SAFETY VALVE
5. STEAM OUTLET FLANGE (STEAM STOP & CHECK VALVE SUPPLIED BY OTHERS)
6. STEAM FLASH DRUM/DRY PIPES (INSULATED)
7. EXCESS STEAM PRESSURE SWITCH
8. WATER LEVEL CONTROL W/LOW & HIGH WATER CUTOUT & MANUAL RESET LOW LOW WATER
9. LOCALIZED PIPING CONNECTIONS: FEEDWATER, BLOWDOWN MANIFOLD, 100 PSIG CONTROL AIR (WHEN REQ'D FOR PNEUMATIC DAMPER ACTUATOR)
10. TUBE REMOVAL ACCESS DOOR
11. STRUCTURAL STEEL BASE
12. MODULATING EXHAUST BYPASS

(FULL OPERATING POSITION)

(FULL BYPASS POSITION)

ESG1

PERFORMANCE AND DIMENSION DATA

A.	1098 °F
B.	412 °F
	110 PSI
	66 BHP
C.	121x58 "
D.	110 "
E.	3 " CONN.
F.	8-18"Dia. " CONN.
	8000 # WGT
	1,615 H.S.
	150 PSIG
	1250 TEMP.

13. FINNED TUBE HEAT TRANSFER SECTION
14. MAIN BLOW DOWN VALVE ASSY.
15. CIRCULATING PUMP ASSY W/SELF COOLING CIRCULATING PUMP (NO COOLING WATER REQUIRED)
16. NEMA 12 CONTROL PANEL: (SINGLE HIGH VOLTAGE CONN.) FUSE DISCONNECT MAGNETIC STARTER, 120V. STEPDOWN TRANSFORMER, ALARM LIGHTS
17. 10GA. SHELL (3" THKS. INSULATED)
18. CONTINUOUS SURFACE BLOWDOWN
19. MODULATING ACTUATOR W/FAILSAFE TO BYPASS
20. MAIN INSPECTION DOOR, HINGED
21. STAINLESS STEEL INTERIOR

NOTES:
A. OPTIONAL INSULATION AS REQ'D:
   −SHELL THICKNESS INSULATION:
   4" ☐ 6" ☐ 8" ☐ 12" ☐
   −BYPASS JACKET INSULATION:
   2" ☐ 3" ☐ 4" ☐ 6" ☐ 8" ☐
   (EXTERNAL PIPING INSULATION INSULATED BY OTHERS AS REQ'D)
B. EXHAUST FLANGE CONN.S:
   2x2 ANGLE RING (STANDARD)
   150LB ANSI HOLE PTN (OPTIONAL) ☐
C. A.S.M.E. & NAT'L BOARD STAMPED − SEC.1, DIV.1

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Phone (262) 251-0051 (800) 558-8690 Fax (262) 251-0118 sales@cainind.com www.cainind.com

Jacket Water Heat Exchanger Quote

Company: West Virginia University
Attn: Diego Dranuta
Email: dgdranuta@mix.wvu.edu

Date: 3/1/2021
Perry Ref: 21-064
Revision: 0

Option 1: Jacket Water Heat Exchanger - Per Customer Data Sheet
Description: AEL-16-192 (16" Diameter Shell / 192" Long Tubes)
Approx Sq Ft: 603 ft2 (each)
Materials: C/S Shellside, C/S Tubeside
Tubes: (192) 3/4" OD X 16 GA (MW) X 192" LG X SA-214 Welded Tubes
Design: TEMA Class C / Type: AEL
Code: Construction is per ASME VIII/D1. Exchanger will be ASME stamped and NB registered.
Notes: Exchanger includes supports for horizontal operation. Tubes to be rolled and expanded into a double grooved tubesheet. External carbon steel to be sandblasted and coated with standard shop primer.

DESIGN	Shell Side (1-Pass)	Tube Side (2-Pass)
PRESSURE:	150 PSI	150 PSI
TEMPERATURE:	250°F	250°F
MATERIAL:	C/S	C/S
CONNECTIONS:	(2) 3" 150# RFSO	(2) 3" 150# RFSO
CORR ALLOW:	0.0625"	0.0625"
RADIOGRAPH:	None	None
EX JT:	None	

MATERIALS OF CONSTRUCTION:
Tubes: SA-214 Carbon Steel
Tubesheets: SA-516-70
Baffles: C/S
Shell / Channel Cylinders: SA-53 E/B
Shell / Channel Nozzles: SA-106B / SA-105
Channel Covers: SA-516-70
Body Flanges: SA-105
Gaskets: Klingersil C4430
Bolting: SA-193-B7 / SA-194-2H (Coated)

Price Qty (3): $82,100
**Shipment:** (6) to (8) weeks after receipt of approval drawings and release for shop fabrication
Drawings: (1) to (2) weeks after receipt of order
All prices are ExWorks, Hainesport, NJ.

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PERRY

PERRY PRODUCTS CORP.
A PERRY EQUIPMENT COMPANY

HEAT EXCHANGER SPECIFICATION SHEET

US Units

Customer	West Virginia University	Job No.
Address		Reference No.
Plant Location		Proposal No. 21-064
Service of Unit		Date 2/26/2021 Rev 0
		Item No.

Size	16 x 192 inch	Type AEL	Horizontal	Connected In 1	Parallel	1 Series
Surf/Unit (Gross/Eff)	603.18 / 596.9 ft2	Shell/Unit 1		Surf/Shell (Gross/Eff)	603.18 /	596.9 ft2

PERFORMANCE OF ONE UNIT

Fluid Allocation		Shell Side		Tube Side	
Fluid Name		Coolant			
Fluid Quantity, Total	lb/hr	60616		47336	
Vapor (In/Out)					
Liquid		60616	60616	47336	47336
Steam					
Water					
Noncondensables					
Temperature (In/Out)	F	210.20	187.36	158.00	187.46
Specific Gravity		0.9599	0.9682	0.9778	0.9683
Viscosity	cP	0.2847	0.3282	0.4041	0.3282
Molecular Weight					
Molecular Weight, Noncondensables					
Specific Heat	Btu/lb-F	1.0275	1.0207	1.0139	1.0204
Thermal Conductivity	Btu/hr-ft-F	0.3914	0.3877	0.3814	0.3879
Latent Heat	Btu/lb				
Inlet Pressure	psia	14.700		124.70	
Velocity	ft/sec	0.78		1.08	
Pressure Drop, Allow/Calc	psi	5.000	1.697	5.000	0.393
Fouling Resistance (min)	ft2-hr-F/Btu	0.00200		0.00200	
Heat Exchanged	1418098 Btu/hr			MTD (Corrected)	20.5 F
Transfer Rate, Service	115.69 Btu/ft2-hr-F	Clean 269.77	Btu/ft2-hr-F	Actual	123.06 Btu/ft2-hr-F

CONSTRUCTION OF ONE SHELL

		Shell Side	Tube Side	Sketch (Bundle/Nozzle Orientation)
Design/Test Pressure	psig	150.00 /	150.00 /	
Design Temperature	F	250.00	250.00	
No Passes per Shell		1	2	
Corrosion Allowance	inch	0.0625	0.0625	
Connections In	inch	(1) 3" 150# RFSO	(1) 3" 150# RFSO	
Size & Out	inch	(1) 3" 150# RFSO	(1) 3" 150# RFSO	
Rating Intermediate				

Tube No. 192	OD 0.7500 inch Thk(Avg) 0.0650 inch Length 16.000 ft	Pitch 0.9375 inch
Tube Type Plain	Material SA-214 Welded Carbon Steel	Tube pattern 30
Shell Carbon Steel	ID 15.375 OD 16.000 inch Shell Cover	
Channel or Bonnet Carbon Steel	Channel Cover Carbon Steel	
Tubesheet-Stationary Carbon Steel	Tubesheet-Floating	
Floating Head Cover	Impingement Plate Rods	
Baffles-Cross Carbon steel	Type Single-Seg. %Cut (Diam) 24.81 Spacing(c/c) 12.000	Inlet 17.000 inch
Baffles-Long	Seal Type None	
Supports-Tube	U-Bend	Type None
Bypass Seal Arrangement pairs seal strips	Tube-Tubesheet Joint Expanded (2 grooves)	
Expansion Joint	Type None	
Rho-V2-Inlet Nozzle 1795.9 lb/ft-sec2	Bundle Entrance 50.37 Bundle Exit 337.91 lb/ft-sec2	
Gaskets-Shell Side Klingersil C4430	Tube Side Klingersil C4430	
- Floating Head		
Code Requirements	TEMA Class C	
Weight/Shell 3159.2 lb	Filled with Water 4390.5 lb Bundle 1604.4 lb	
Remarks:		

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9.5 Appendix E: SIMULINK® 0-D Models.
HHPDE CHP System

HHPDE CHP system engine, E-HEX, and JW-HEX sub model SIMULNK® Environment
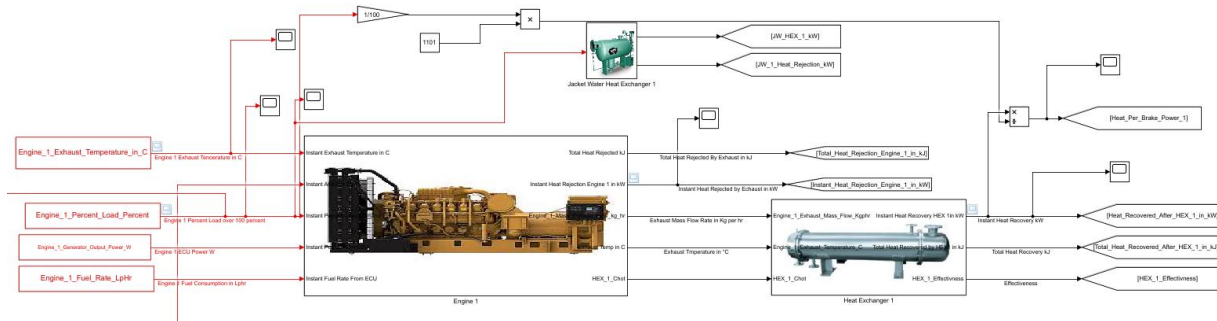
Where engine subsystem inputs were:

- Exhaust temperature
- Engine load
- Genset instant power output
- Engine fuel consumption

Engine Subsystem outputs were:

- Total exhaust heat rejection
- Instant exhaust heat rejection
- Exhaust mass flow rate (dry air and water vapor combined)
- Exhaust temperature

CAT 3512C engine calculations based on Equation 2 are shown in the figures below

HHPDE CHP system engine sub model calculations based on Equation 2.
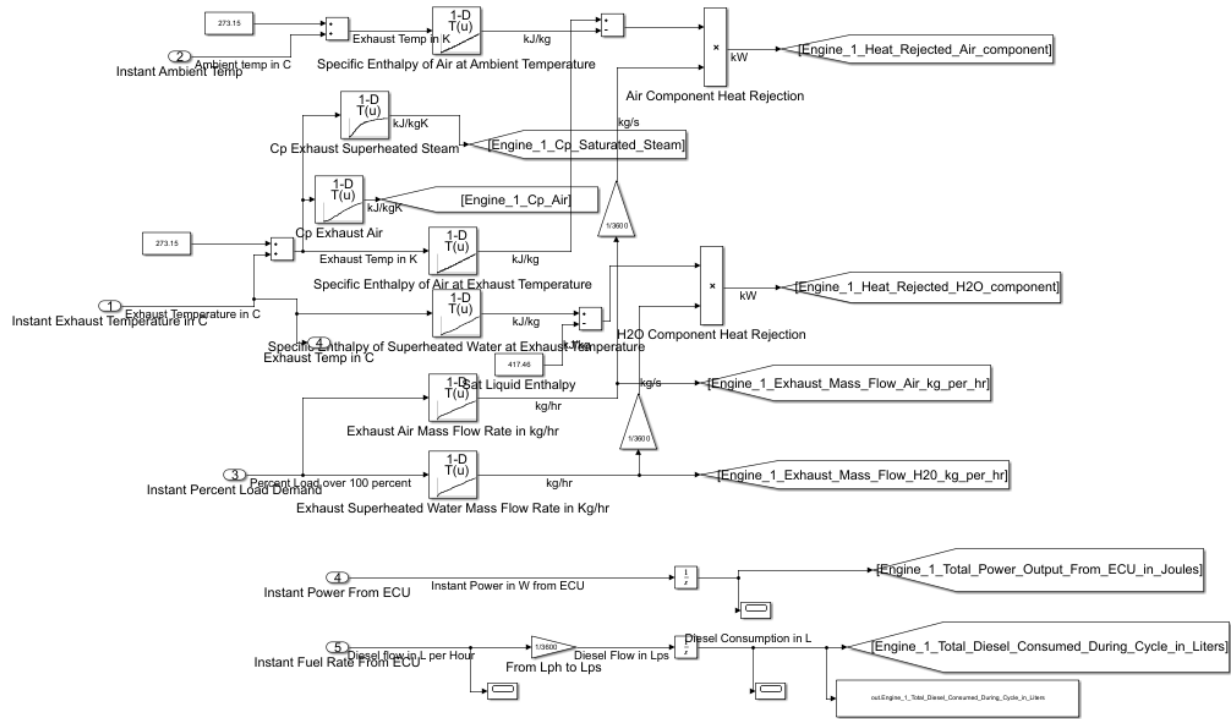
HHPDE exhaust flow dry air and water vapor mass flow rates modeled used on Equation 2 for heat rejection calculations.

Engine Load	Mass Exhaust Flow rate air	H_2O Mass Exhaust Flow rate
%	kg/hr	kg/hr
0	0.0	0.0
5	889.9	44.4
10	1176.2	58.6
15	1462.5	72.9
20	1748.7	87.2
25	2035.0	101.4
30	2321.3	115.7
35	2607.6	130.0
40	2893.9	144.2
45	3180.2	158.5
50	3459.5	172.4
55	3752.7	187.0
60	4039.0	201.3
65	4325.3	215.6
70	4611.6	229.8
75	4911.7	244.8
80	5184.1	258.4
85	5470.4	272.6
90	5756.7	286.9
95	6043.0	301.2
100	6322.3	315.1
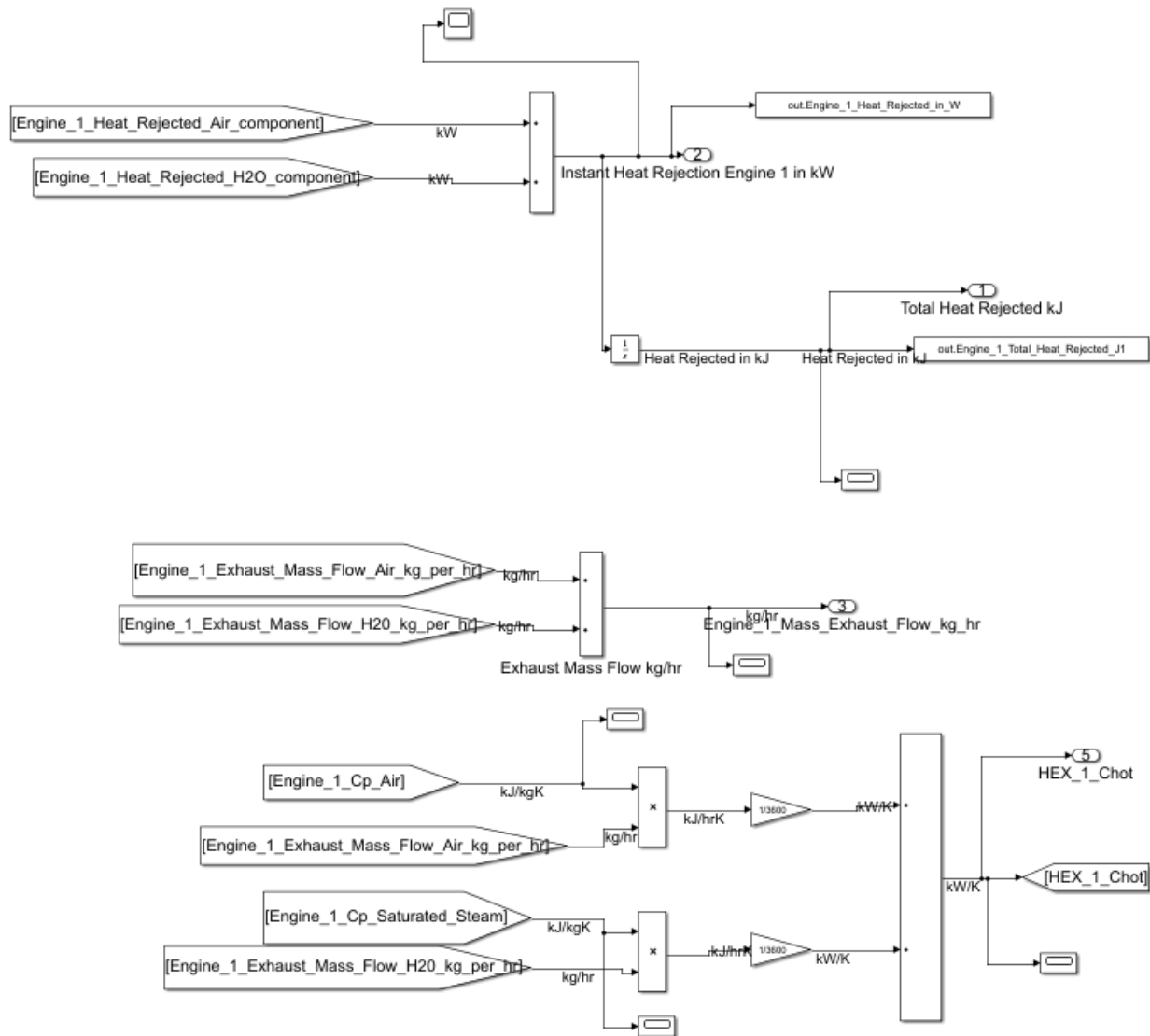
HHPDE E-HEX subsystem inputs were:

- Exhaust mass flow rate (dry air and water vapor combined)
- Exhaust temperature

HHPDE E-HEX subsystem outputs were:

- Instant heat recovery
- Total heat recovery

HHPDE E-HEX calculations are presented in the figures below, along with the two-entry table used for heat recovery calculations and HHPDE E-HEX water flow rate.

HHPDE E-HEX calculations HHPDE exhaust composition model and variable exhaust gas temperatures.
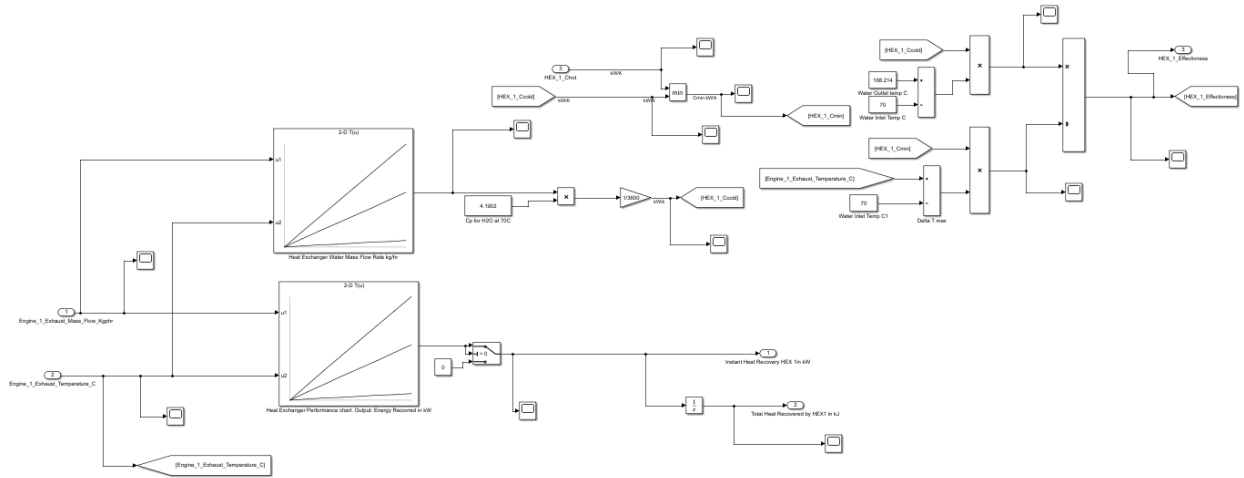
HHPDE E-HEX heat recovery two-entry table based on CHEMCAD® simulated scenarios for modeled exhaust gas composition and variable exhaust gas temperatures.

HEX Energy Performance kW			
Exhaust	Engine Exhaust Temperature (°C)		
Exhaust Flow (kg/hr)	200	400	600
0	0.0	0.0	0.0
1000	9.6	84.7	162.5
2000	19.3	169.3	323.6
3000	28.9	254.7	483.3
4000	38.6	339.4	643.0
5000	48.2	424.1	801.3
6000	58.5	509.4	959.0
7000	68.2	593.4	1115.2

HEX WATER SIDE MASS FLOW RATE KG/HR			
Exhaust	Temperature (°C)		
Exhaust Flow (kg/hr)	200	400	600
0	0	0	0
1000	14	123	236
2000	28	246	470
3000	42	370	702
4000	56	493	934
5000	70	616	1164
6000	85	740	1393
7000	99	862	1620

HHPDE JW-HEX subsystem input was:

- Engine load

HHPDE JW-HEX subsystem outputs were:

- Instant Heat rejection to JW
- Instant heat recovery

The figures below present the models used for the instant heat rejection to JW and instant heat recovery via JW-HEX utilized in the 0-D SIMULINK model.

HHPDE heat rejected to the coolant based modeled JW temperature and mass flow rates as a function of engine load.

HHPDE JW-HEX Heat Rejection Model	
Engine Load (%)	Heat Rejection (kW)
0	0.0
5	30.6
10	60.1
15	88.5
20	115.9
25	142.2
30	167.5
35	191.7
40	214.8
45	236.9
50	257.9
55	277.9
60	296.8
65	314.7
70	331.4
75	347.2
80	361.8
85	375.4
90	388.0
95	399.5
100	409.9

HHPDE JW-HEX performance based on CHEMCAD® simulated scenarios, modeled exhaust gas composition and exhaust gas temperatures.

Engine Load	Heat Recovered
%	kW
0	0
10	49.61
20	100.87
30	152.41
40	202.89
50	250.96
60	295.94
70	334.47
75	351.74
80	367.2
90	392.1
100	407.84

HHPDE ESG subsystem input was:

- Engine load

HHPDE ESG subsystem outputs were:

- Instant heat recovery
- Total heat recovery

The HHPDE ESG subsystem included calculations for the three engines ESGs. Figures below present the HHPDE ESG subsystem and its heat recovery model.

HHPDE ESG SIMULINK® subsystem containing the three ESG (one per genset).
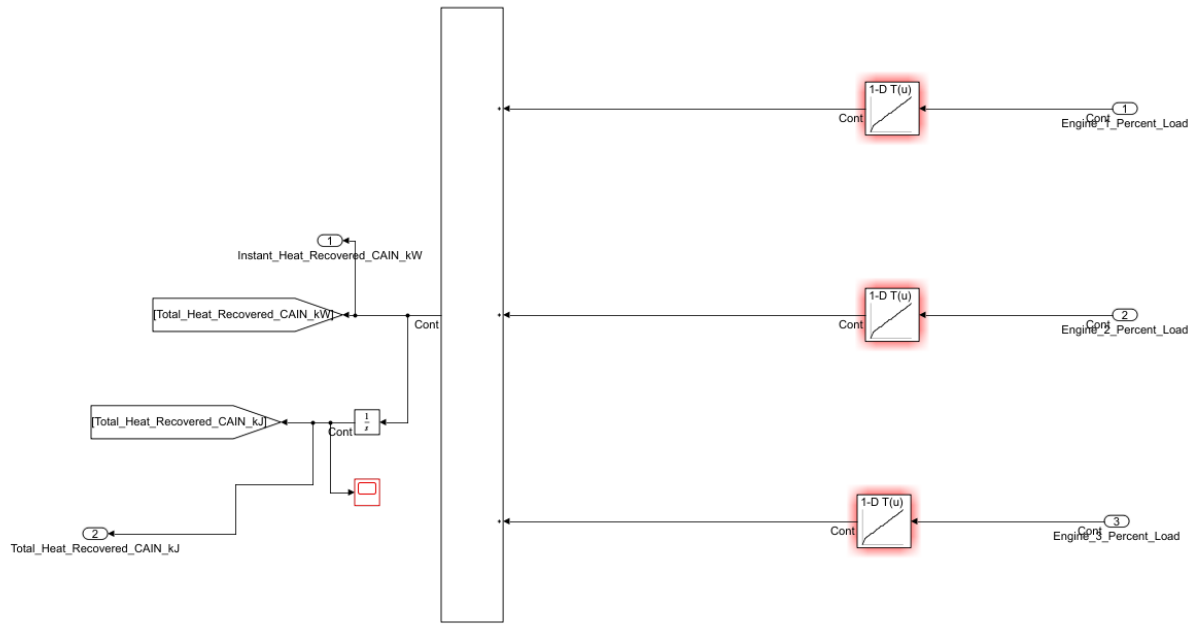
HHPDE ESG heat recovery model based on data provided by CAIN Industries which is attached in Appendix D.

Engine Laod	Heat Recovery
%	kW
0	0.0
5	86.8
10	108.7
15	130.6
20	152.5
25	174.4
30	196.3
35	218.1
40	240.0
45	261.9
50	283.8
55	305.7
60	327.6
65	349.5
70	371.4
75	393.3
80	415.2
85	437.1
90	459.0
95	480.9
100	502.8

HHPDE CHP hybrid System

The HHPDE CHP hybrid system included the same subsystems as the HHPDE CHP system plus the HEMS subsystem. The figures below show the HEMS subsystem and its respective sub models.

HEMS SIMULINK® subsystem used for both HHPDE CHP hybrid and DNGE CHP hybrid models
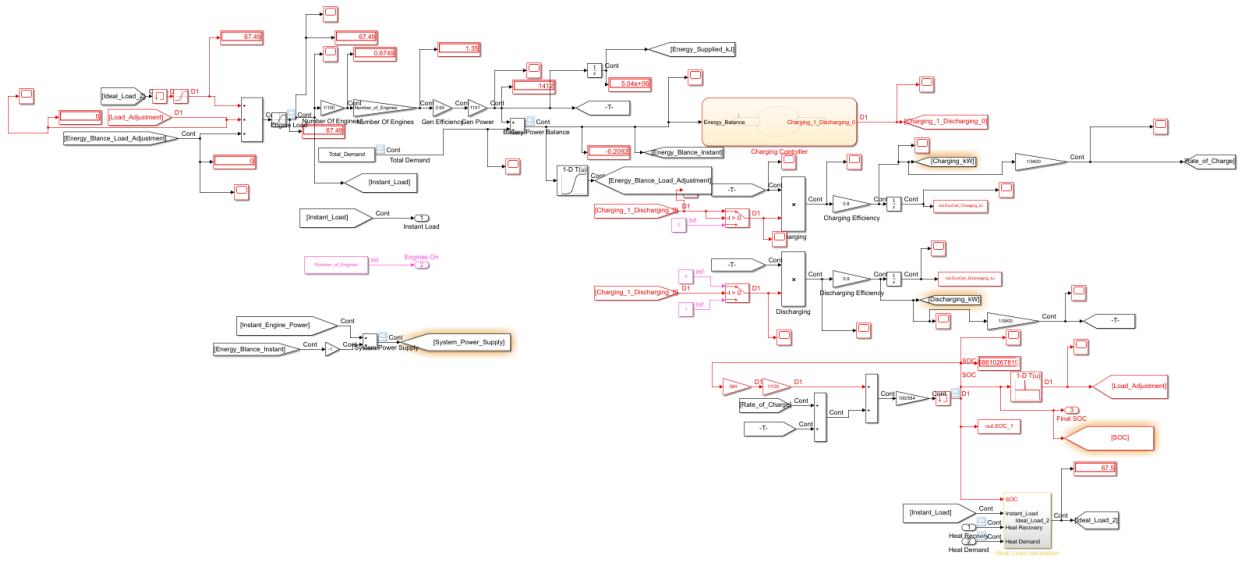
Where the main components of this subsystem were:

- Power demand
    - Imported from recorded cycles
- Energy balance (charging/discharging control)
    - Determined when the HEMS was charging or discharging given power demand and power supply
        - Power demand
        - Power produced by gensets
        - Power supplied by or to the HEMS
- HEMS control system
    - Ideal load control system
        - Utilized the previous 60 seconds of SOC to adjust the load to an ideal steady load to meet cycle power demands
    - Energy balance engine load adjustment control
        - Prevented the HEMS from exceeding charging and discharging rates
    - SOC engine load adjustment control
        - Maintained the HEMS within the SOC operational range (40% to 100%)
    - Heat balance control

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- Adjusted engine load to meet instant power demands if the energy balance engine load adjustment control and the SOC engine load adjustment control allowed it.

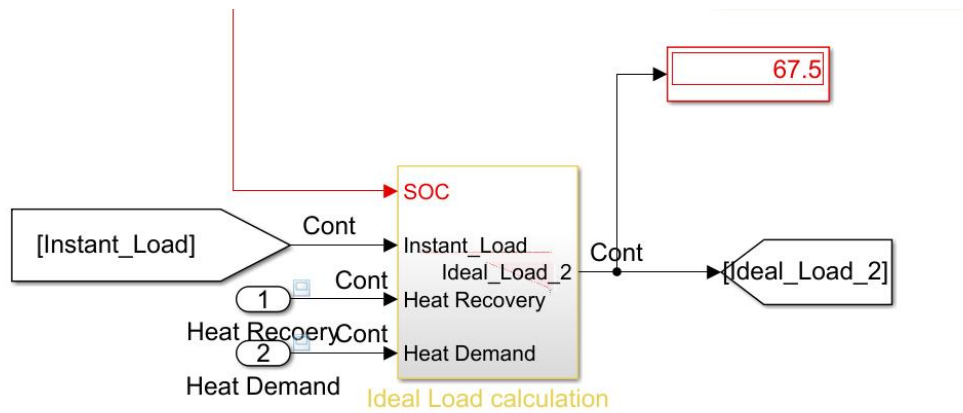The HEMS sub models of the subsystem are presented in the figures below.

HEMS charging/discharging control system modeled.
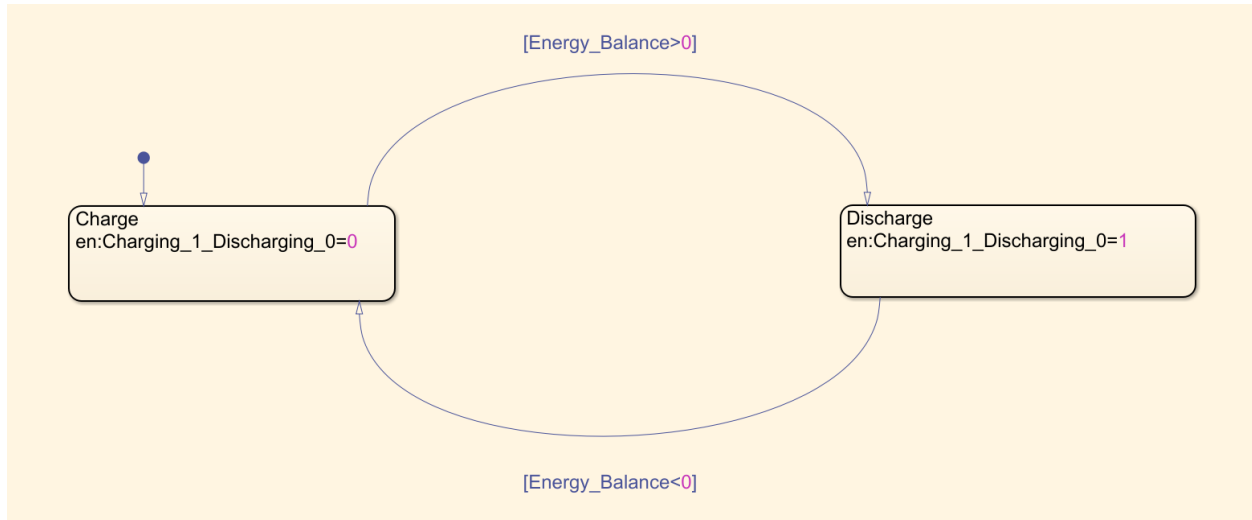
HEMS energy balance engine load adjustment control and SOC engine load adjustment control designed for complying with the HEMS technical specifications.
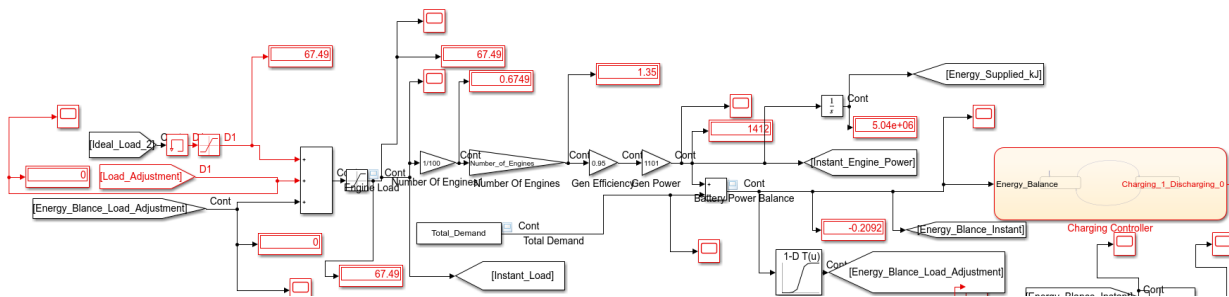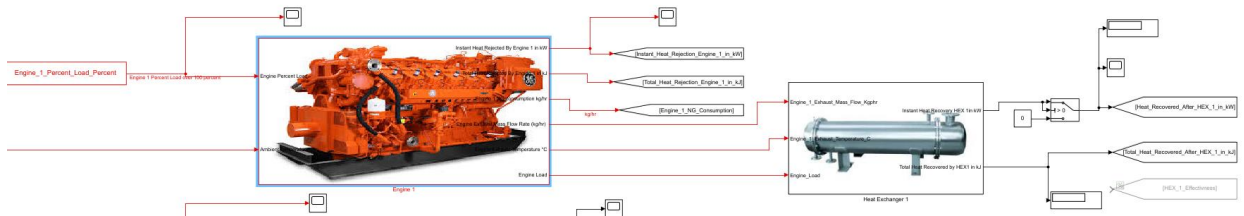
HEMS heat control balance.

DNGE CHP System

DNGE CHP system engine and E-HEX sub models SIMULNK® Environment

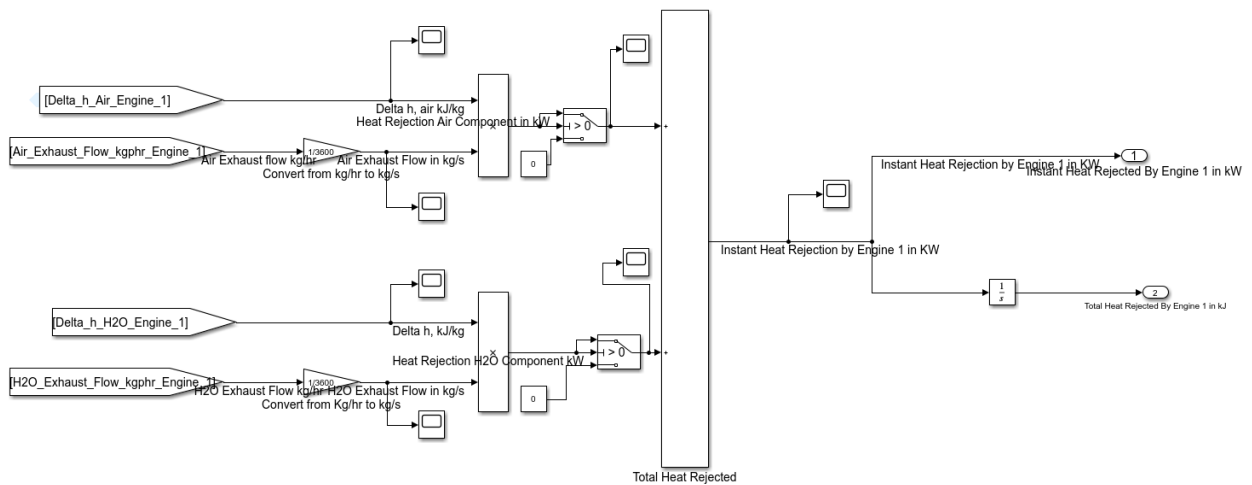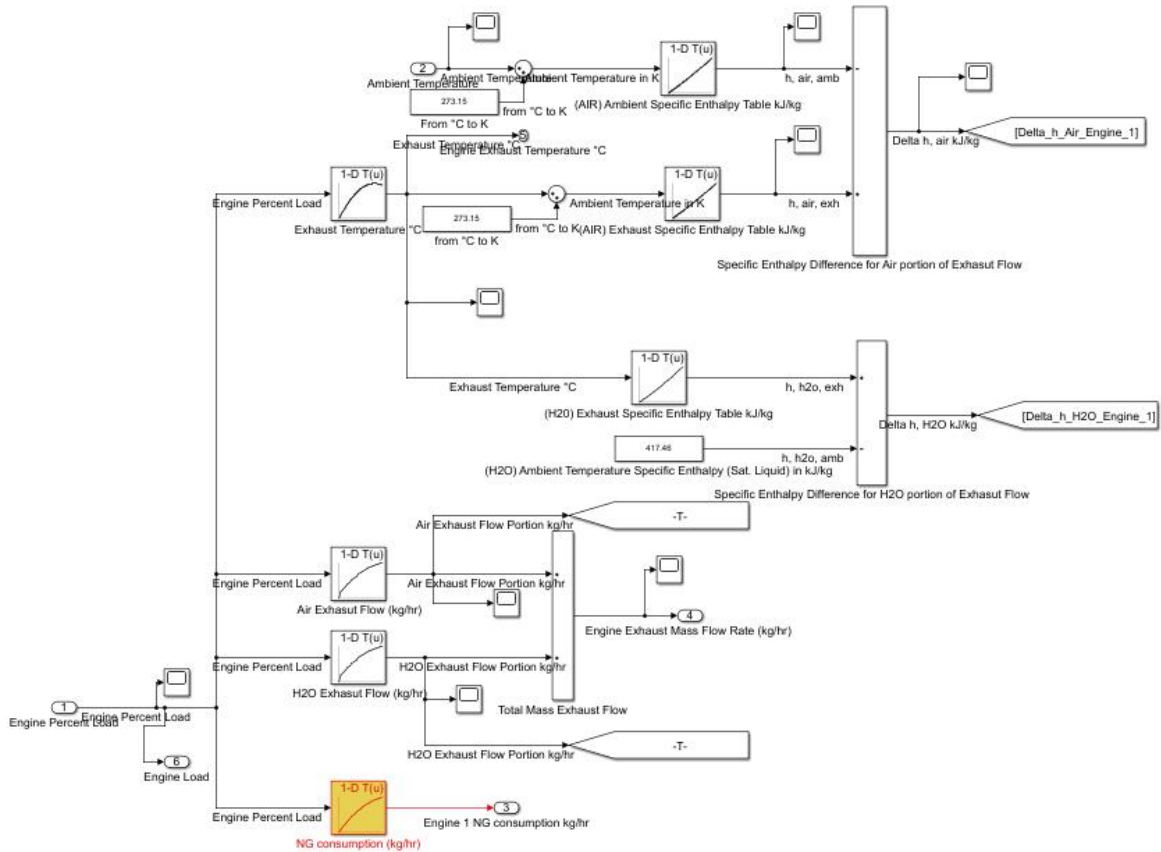Where engine subsystem inputs were:

- Engine load

Engine Subsystem outputs were:

- Total exhaust heat rejection
- Instant exhaust heat rejection
- Exhaust mass flow rate (dry air and water vapor combined)
- Exhaust temperature
- Engine load
- Natural gas consumption

Waukesha L7044GSI engine calculations based on Equation 2 are shown in the figures below.

DNGE CHP system engine sub model calculations based on Equation 2.
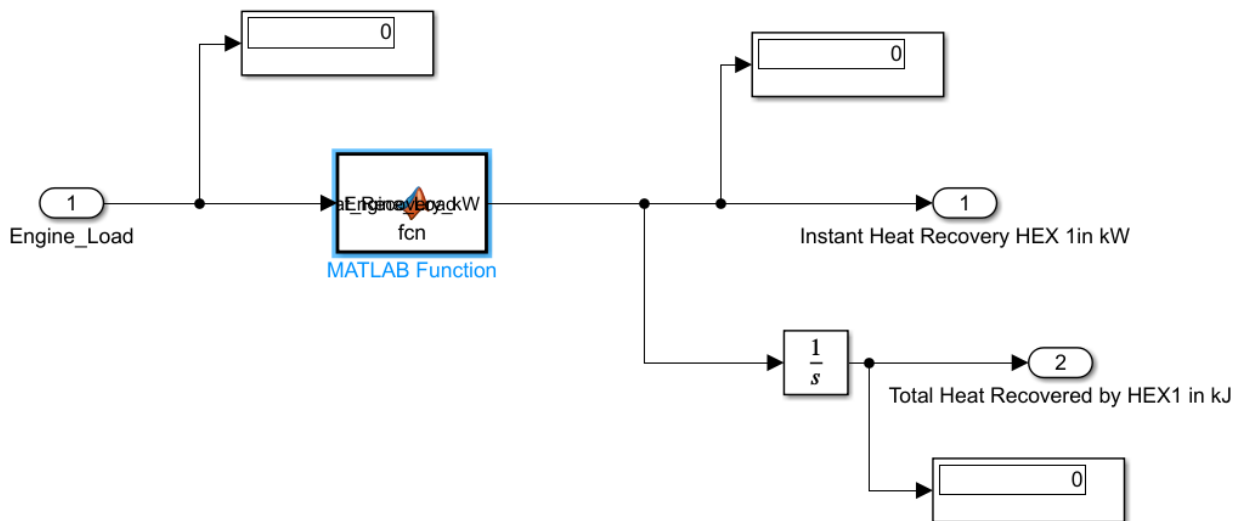
DNGE E-HEX subsystem inputs were:

- Exhaust mass flow rate (dry air and water vapor combined)
- Exhaust temperature
- Engine Load

HHPDE E-HEX subsystem outputs were:

- Instant heat recovery
- Total heat recovery

HHPDE E-HEX calculations are presented in the figures below, along with the two-entry table used for heat recovery calculations and HHPDE E-HEX water flow rate.

DNGE CHP system E-HEX sub model based on engine laod.

Where the DNGE E-HEX model was included in the MATLAB® function block and presented below.

```
function Heat_Recovery_kW = fcn(Engine_Load)
Heat_Recovery_kW = 9.9251.*Engine_Load;
```
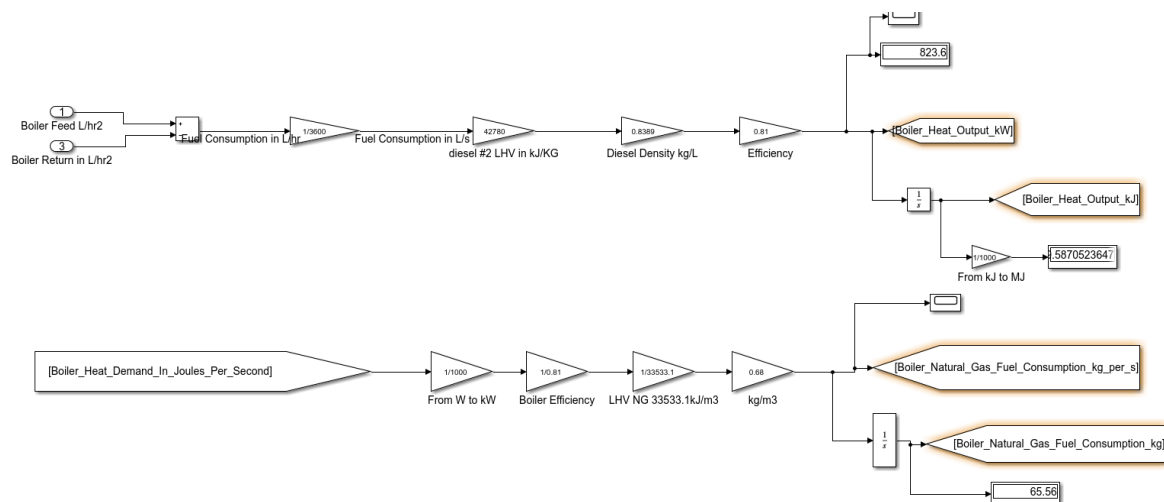
DNGE boiler subsystem inputs were:

- Boiler diesel fuel feed
- Boiler diesel fuel return

DNGE boiler subsystem inputs were:

- o Instant heat output
- o Total heat output
- o Natural gas fuel consumption (converted utilizing Equation 1)
- o Total natural gas fuel consumption

The figures below show the natural gas powered boiler model

DNGE CHP system boiler sub model calculations based on Equation 1.
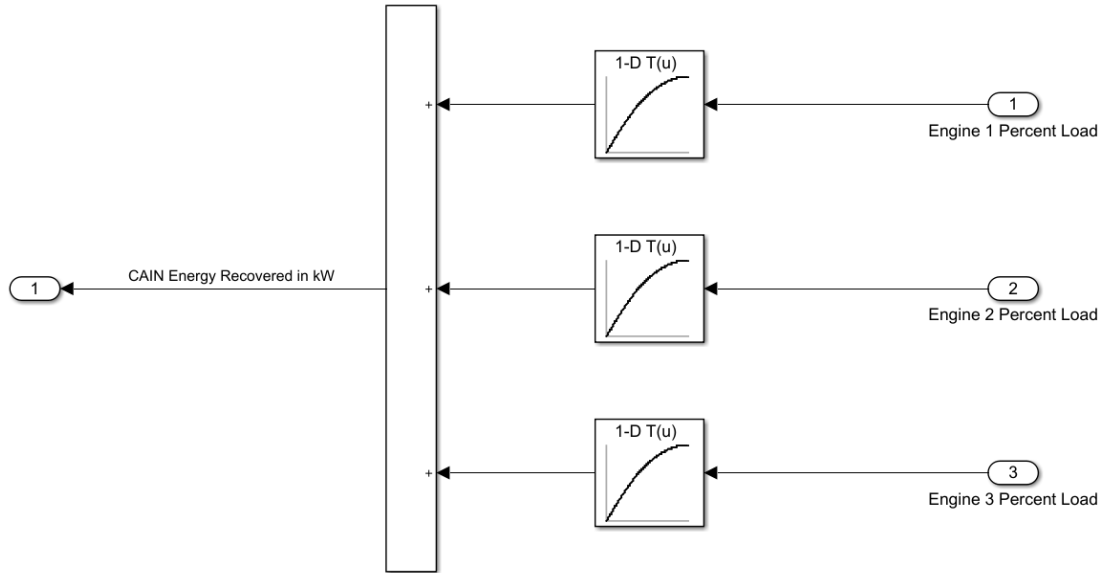

DNGE ESG subsystem input was:

- Engine load

DNGE ESG subsystem outputs were:

- DNGE ESG instant recovered heat
- DNGE ESG total recovered heat

The figures below present the DNGE ESG subsystem, which includes the three ESG (one per genset).

DNGE CHP system SIMULINK® ESG sub model.

DNGE CHP system ESG sub model calculations based on engine load and ESG performance
supplied by CAIN Industries as specified in Appendix D

Engine Load	Heat Recovery
%	kW
0	0
10	131.6
20	249.16
30	352.68
40	442.16
50	517.6
60	579
70	626.36
80	659.68
90	678.96
100	684.2

DNGE CHP Hybrid System

The DNGE CHP hybrid system added the same HEMS system utilized for the HHPDE CHP hybrid system to the DNGE CHP system. Same subsystems and blocks were copied and implemented into a new SIMULINK® environment model.

9.6 Appendix F: Boiler Rental Pricing

Rental Quote

Please Remit To
Power Mechanical, Inc.
4811 Commerce Drive
Newport News, VA 23607

Booking Number BK0007064
Booking Date 4/16/2021
PO Number

Bill To

Diego Dranuta
WVU
Morgantown, WV
Phone: (304) 389-7302

Customer Name WVU

Rental Location

Diego Dranuta
WVU
Morgantown, WV
Phone: (304) 389-7302

Payment Terms

Power Mechanical, Inc. proposes to provide the following rental equipment for your project:

Description 200 HP Steam Boiler, 110 PSI, #2 Oil Or Gas, 480V

Rental Rate $7,500.00 per MONTH

Price is based on a 1 month minimum rental.

Please note that all units are subject to availability at the time of order

Miscellaneous Charges		Quantity	U of M	Unit Price	Line Total
TRUCKING CHARGE	Round Trip Freight Charge	1.00	EACH	3,950.00	3,950.00
START UP T/M	Start up: Billed on a T&M basis. This service is required to be performed by a PMI technician.				

By my signature below, I certify that I have read and agree to the provisions set forthe in this Rental Agreement and in the General Terms and Conditions and Equipment Terms and Conditions posted at www.PMITerms.com. I understand that a copy of the Terms and Conditions can be sent to me at my request.

Customer Signature: _____ Date: _____

PO#: _____ Expected Date of Delivery: _____

Page 1 of 2

Power Mechanical, Inc., 4811 Commerce Drive, Newport News, VA 23607