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Transit IDEA Program

Diesel-Electric Locomotive Energy Recovery and Conversion

Final Report for
Transit IDEA Project 67

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Diesel-Electric Locomotive Energy Recovery and Conversion

Final Report

Contract No: TRANSIT-67

Prepared for the IDEA Program
Transportation Research Board
The National Academies

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1. EXECUTIVE SUMMARY

Internal combustion engines typically convert only one third of the fuel potential into useful power. The remaining fuel potential is lost as low- and high-grade wasted thermal energy. Approximately half of the thermal energy normally vented to atmosphere (wasted) is represented by high-grade (high temperature) exhaust gases. Waste Heat Recovery (WHR) technologies aim at recovering at least a portion of the high grade wasted thermal energy and convert it to useful power, resulting in lower fuel consumption and pollutant emissions.

ThermaDynamics Rail LLC (TDR) is optimizing Waste Heat Recovery (WHR) technologies for minimally invasive retrofitting of large IC engines equipping locomotives. US railroad locomotive fleets are generally represented by a large number of locomotives manufactured more than a decade ago. As of 2012 there were approximately 24,000 locomotives in service, with over 60% of the fleet represented by locomotives older than 12 years (*1*). Retrofitting locomotives with WHR technologies would allow railroad companies to lower fuel consumption and operating expenditures, whilst enabling them to meet increasingly more stringent pollutant reduction requirements. Depending on locomotive type, adopting WHR technologies could also allow railroad companies to avoid or mitigate securing of heavy investments required to replace old locomotives with new locomotives compliant with the most recent pollutant emission standards.

Contract No. TRANSIT-67 was focused on the analysis of WHR technologies performance against averaged locomotive duty cycles with results summarized in a final report containing two stages addressing technical and economic aspects as follows:

Stage 1 “Commuter Rail Locomotive Baseline Data Collection & Computer Modeling”, was dedicated to obtaining experimental data from locomotives equipped with the Electro Motive Division (EMD) 16-645-E3 engine operating at different notch numbers. The experimental data comprised details of the exhaust gas temperature, mass-flow-rate (kg/s or lb/s) and pressure in addition to the locomotive average duty cycle, fuel consumption and particulate matter emission. Based on a set of thermodynamic assumptions, a computer model was developed to predict the potential power recovered by retrofitting the EMD 16-645-E3 with TDR WHR technologies. A parametric analysis was also performed to assess the impact that each of the assumptions could have on the projected recovered power. Stage 1 results estimated that WHR technologies can recover up to 437.6 kW of the EMD 16-645-E3 engine exhaust gas energy thermal power, normally vented to atmosphere (approximately 18% of the EMD engine maximum power). This recovered power, at 2013 fuel cost, averaged locomotive duty cycles, and locomotives with rated power in the 3,000hp to 4,400hp range, represents approximately \$54,000 to \$140,000 annual fuel savings per locomotive, in addition to the environmental benefits induced by lowered thermal pollution and reduced pollutant emissions – through WHR technologies a lower amount of fuel may be utilized to obtain the same locomotive power.

Stage 2 “Installation Test and Analysis” was dedicated to the development of a laboratory scale exhaust gas simulator to validate the results of the computer model. To validate the computer model, a series of custom made High Pressure Heat Exchangers (HiPHEX) were retrofitted with a bank of locomotive exhaust manifold coupled to a 5-gallon/hr diesel-burner. The purpose of this configuration was to generate exhaust gases with thermodynamic conditions similar to those found in a typical locomotive exhaust manifold during operation. Pressurized water was then circulated

inside the thermodynamic loop formed by the HiPHEXs to obtain superheated steam as a result of thermal transfer from the exhaust gases to the water via thermal coupling with the heat exchangers. A data acquisition system was set up to sample the thermodynamic properties of the water/steam at the inlet and outlet of the heat exchangers so as to enable direct comparison with the results projected by the computer model. The experimental set up highlighted some unforeseen problems related to the exhaust gas simulator design. The dimensional constraints imposed by the locomotive exhaust gas manifolds dictated the maximum size of the flange coupling the combustion chamber. As a result, the exhaust gas simulator turned out to be under-dimensioned for complete combustion of the fuel-air mixture, thus inducing poor performance of the exhaust gas simulator and limiting the spectrum of data to be investigated. Re-designing and manufacturing an expanded and optimized combustion chamber was unfortunately outside the scope, timing, and budget dedicated to this project. Consequently, the computer models were only partially verified against experimental data obtained from locomotive testing executed under static conditions through Federal Railroad Administration (FRA) grant FR-RRD-0005-10-01-00, and from experimental data from a GE-Dash 9 locomotive retrofitted with custom made HiPHEX's (described in detail in Stage 2).

The analyses showed that basic HiPHEX designs perform reasonably well at all notch numbers with attractive economic and pollutant reduction benefits (i.e. averaged \$54,000-\$140,000 of fuel savings per year per locomotives with power ratings from 3,000hp to 4,400hp). However, availability of an expanded combustion chamber and additional experimental measurements are still necessary to simulate the full spectrum of locomotive duty cycles, and validate the projected results and increase accuracy of the economic performance figures.

2. STAGE I – WORK PERFORMED

The following chapters, identified by Tasks, describe the work performed in agreement with TRANSIT-67 Subject Work.

2.1. TASK 1 & 2: COMMUTER RAIL LOCOMOTIVE BASELINE DATA COLLECTION

The aim of these initial tasks was to collect baseline data from selected locomotives. The Electro-Motive Division (EMD) 16-645-E3 engine was chosen for this analysis. This engine was selected for practical reasons as a locomotive equipped with this particular EMD engine is operated by the Transportation Technology Center, Inc (TTCI) in Pueblo, Colorado, and was made accessible with the support of the Federal Railroad Administration (FRA). As the California Environmental Protection Agency Air Resources Board (CARB) executed a series of studies aimed at quantifying particulate matter (PM) emissions and possible impact on populations directly exposed to such emissions, exhaust gas data for the execution of this report was extracted from CARB's Roseville Rail Yard Study (2). In this study, a series of exhaust gas measurements sampled at the stack exhaust of locomotives operated at various conditions, were collected and published. Examples of key exhaust gas averaged parameters, including temperature (T), velocity (U) and Particulate Matter (PM) emissions at different locomotive operating conditions, are shown in Table 1.

Another data source utilized for the execution of Tasks 1 and 2 is represented by the United States Environmental Protection Agency (EPA) through regulatory support documents on locomotive emission standards (3). Table 2 summarizes EPA estimate for the time percentage each locomotive operates on average at each notch setting.

Notch	Power (hp)	Fuel cons. (kg/s)	Exhaust U (m/s)	Exhaust T (K)	PM emissions (g/hr)	PM emission (g/hp/hr)
8	3159	0.1514	26.89	661	821.3	0.26
7	2661	0.1285	22.49	660	636.6	0.24
6	1971	0.0900	18.10	616	552.9	0.28
5	1461	0.0690	15.10	584	336.0	0.23
4	1034	0.0490	12.09	545	258.5	0.25
3	686	0.0354	9.70	504	226.4	0.33
2	395	0.0194	6.85	451	134.3	0.34
1	105	0.0105	4.80	385	35.70	0.34
Idle	69	0.00530	3.73	351	47.94	2.82
DB - 1	17	0.0166	9.46	387	80.04	1.16

Note: EMD 16-645-E3 data on PM emissions was extracted from EPA report on locomotive emission standards [REF-4].

TABLE 1: EMD 16-645-E3 Engine Exhaust Gases Stack Flow Conditions

Notch	Line Haul (%)	Passenger (%)
8	16.2	15.6
7	3.0	1.4
6	3.9	2.9
5	3.8	4.0
4	4.4	4.7
3	5.2	5.7
2	6.5	5.1
1	6.5	7.0
Idle	38.0	47.4
DB-1	12.5	6.2

TABLE 2: Duty-Cycles For Current In-use Locomotives (percentage time in notch)

Table 2 reports averaged data obtained from analysis of 63 line-haul trains operating for 2,475 hours, and from 20 Amtrak passenger locomotives operating for 57,500 hours. This data is therefore the results of averaged parameters and is not representative of specific locomotive duty cycles. Additionally, discussions with locomotive operators indicated

that especially for Line Haul operations, locomotives may be operated in excess of 20% of time at Notch 8. On the other hand, Amtrak operations generally involve intercity trains, thus a potentially lower percentage of time spent at high Notch numbers. A more accurate analysis requires case-by-case data, for example from actual duty cycle of operational commuter rail and line-haul locomotive fleets.

2.2. TASK 3: THERMODYNAMIC SOFTWARE MODELING

The aim of Task 3 is to develop a thermodynamic model to represent the thermal-hydraulic behavior of the waste heat recovery and conversion system developed by TDR when configured to operate as a Rankine power cycle. For the execution of this task, Engineering Equation Solver (EES) software has been used as the code-platform for the evaluation of the waste heat recovery and conversion technical performance. EES is an equation-solving program relying on thermodynamic and transport properties database provided for several substances (www.fchart.com/EES). For the purposes of this project the properties of air have been utilized to simulate exhaust gases, while for the working fluid circulating in the closed-loop, water properties have been adopted with R134a refrigerant only as a working fluid for optimization studies and addressing low exhaust gas mass-flow-rate and temperature characteristic of locomotive operations at idle and low Notch numbers.

Figure 1 shows TDR’s waste heat recovery closed-loop Rankine cycle diagram, consisting of a pump, a high-pressure heat exchanger (HiPHEX), and a power conversion unit (PCU) formed by a turbine coupled to an electric generator.

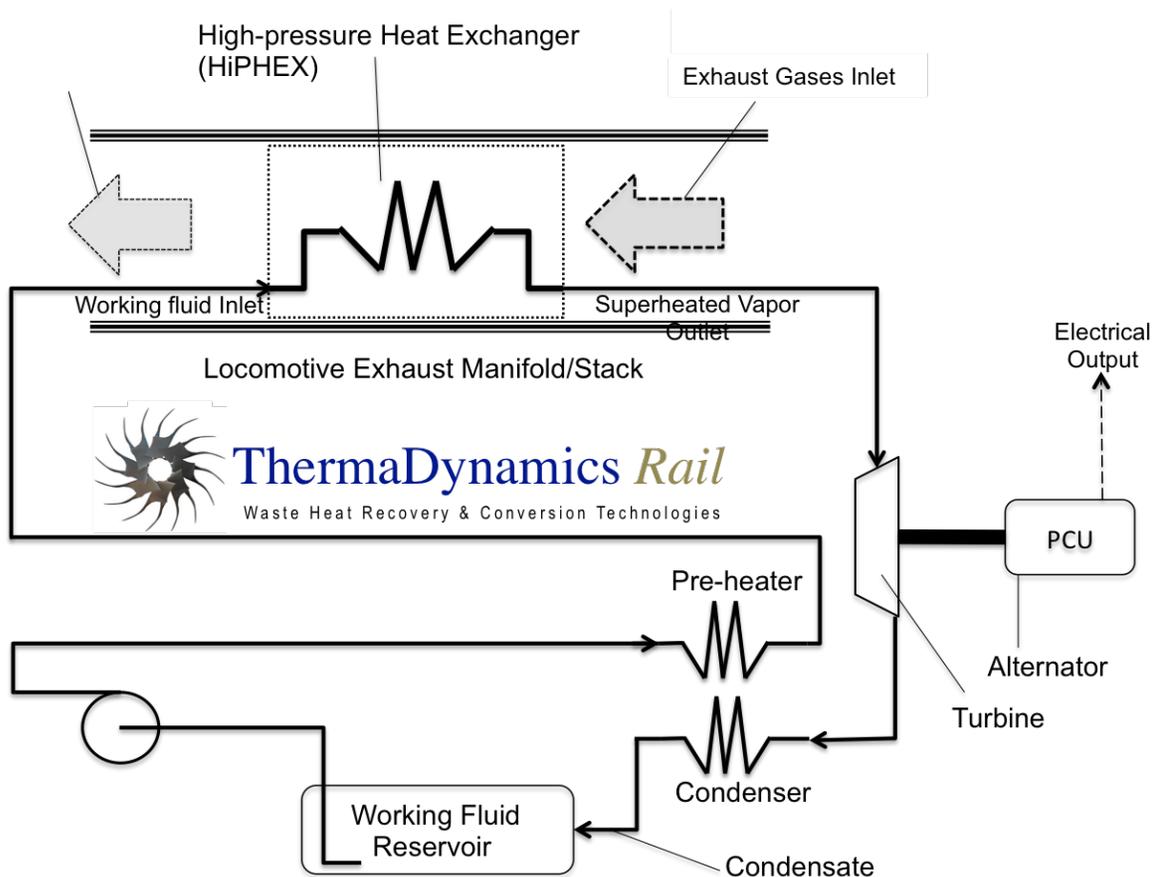


FIGURE 1: Closed-loop Vapor cycle and Thermadynamics Rail components

For the purposes of the representation shown in Figure 1, the working fluid is assumed to be water - however organic fluids could be utilized for higher cycle efficiency and economic performance. With reference to Figure 1, water is pressurized via high-pressure pump from a reservoir, and flows into the High-Pressure Heat Exchangers (HiPHEXs) exposed to the exhaust gas stream. At the HiPHEXs inlet, exhaust gas temperatures and mass flow rates vary according to locomotive operating conditions. At the HiPHEX outlets the exhaust gas temperature decreases by a measure proportional to the amount of thermal energy transferred to the working fluid. The HiPHEX objective is to obtain superheated vapors to drive a turbine or other types of expanders coupled to an electric generator. In this manner, the power conversion unit (PCU) converts the turbine (or expander) mechanical work into electricity. The electricity produced by the generator is then distributed to the locomotive power train (i.e. traction motors), and ultimately converted into pollutant-free propulsion power, thereby reducing fuel consumption and emissions in a manner proportional to the engine duty cycle. To complete the closed-loop Rankine power cycle, working fluid vapor at the turbine discharge is condensed through thermal exchange with pre-heater and condenser heat exchangers (i.e. exchanging thermal energy with the pressurized loop and with the environment). At this point, condensate is pressurized by the pump to reset the thermodynamic cycle. The simplifying assumptions adopted for the execution of the thermodynamic model included the following:

- Working fluid at pump inlet = saturated liquid at $P[1] = 0.75$ bars
- Working fluid at pump outlet pressure = $P[2] = 20$ bars
- Constant pump isentropic efficiency $\eta_{\text{pump}} = 90\%$
- No pressure losses in HiPHEX and pipes such that $P[2] = P[3]$
- Pressure of steam at turbine discharge = 0.75 bars
- Condenser drives the working fluid thermodynamic state to saturation pressure, no pressure losses such that $P[1] = P[4]$
- Constant isentropic turbine and alternator efficiency $\eta_{\text{turbine}} = \eta_{\text{alternator}} = 85\%$

η_{pump}	η_{turbine}	$\eta_{\text{alternator}}$	T[1] (°C)	P[4] = P[1] (bars)	P[2] = P[3] (bars)
0.9	0.85	0.85	80	0.75	20

TABLE 3: Summary of Assumptions

Given the above assumptions, the EES-based Rankine power cycle model estimates the total power recovered from wasted exhaust gas thermal energy given at mass flow rate and temperatures of the exhaust gases for different Notch numbers as reported in Table 1. Figure 2 shows an example of the EES diagram window. The main results, summarized by P_{recovd} and η_{recovd} , are shown in the top right side box. At locomotive operating conditions corresponding to Notch 8, approximately 11.8% of the otherwise wasted thermal energy may be converted into recovered energy, thereby resulting in a net 187.9 kJ/s (factoring the alternator, turbine, and pump inefficiencies) of recovered pollutant-free electric power.

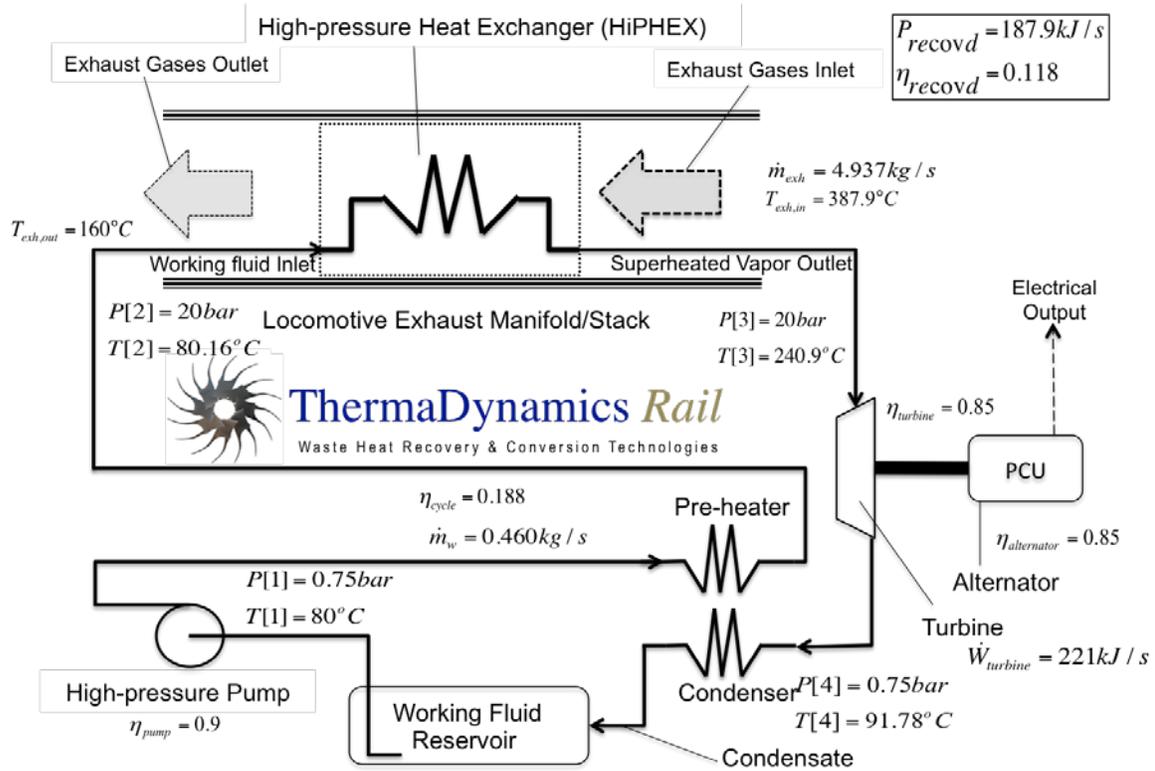


FIGURE 2: EMD 16-645-E3, Notch 8 operating conditions, TDR model estimates

Table 4 shows the estimated recoverable power via Rankine power cycle executed under TDR modeling conditions for the EMD 16-645-E3 engine operated from Notch 3 to Notch 8. As locomotive power decreases, power recovered (P_{recovd}) also decreases. For example, at Notch 8, $P_{recovd} = 187.9 \text{ kJ/s}$ and drops to 9.2 kJ/s at Notch 3.

Notch	$T_{exh,in}$	m_{exh}	$T_{exh,out}$	m_{water}	$T[3]$	W_{turb}	η_{cycle}	η_{recovd}	P_{recovd}
8	387.9	4.937	160	0.460	240.9	221.0	0.188	0.118	187.9
7	386.9	4.136	160	0.392	221.3	183.4	0.187	0.118	155.9
6	342.9	3.566	180	0.240	228.5	113.4	0.187	0.096	96.4
5	310.9	3.138	185	0.163	227.1	76.9	0.187	0.082	64.8
4	271.9	2.692	195	0.085	234.4	40.5	0.187	0.059	34.2
3	230.9	2.336	207	0.023	222.7	10.8	0.187	0.022	9.2

TABLE 4: EMD 16-645-E3 Recoverable Energy at Different Notch Numbers

As previously shown in Table 1, data corresponding to idle and low Notch settings are represented by low exhaust gas temperatures. At these conditions to optimize the Rankine cycle the working fluid pressure may be lowered so as to ensure that expansion in the turbine or expander occurs in the superheated region. Table 5 shows model predictions for EMD 16-645-E3 PM emission reductions. Accordingly, for Notch setting varying from Notch 3 to Notch 8, the

proportion of power recovered, measured as $P_{\text{Engine}}/P_{\text{recovd}}$, varies from 1.8% to 8.0%. Therefore, the absolute amounts of greenhouse gases and particulate matter (PM) produced by this engine when operated at these conditions may proportionally decrease by approximately 1.8% - 8.0% respectively.

Notch	P_{recovd} (kJ/s)	P_{Engine} (kJ/s)	PM emission (g/hr)	PM emission (g/hp/hr)	$P_{\text{Engine}}/P_{\text{recovd}}$ (%)	PM reduction (g/hr)
8	187.9	2356	821.3	0.26	8.00	65.74
7	155.9	1984	636.6	0.24	7.86	50.04
6	96.4	1470	552.9	0.28	6.56	36.27
5	64.8	1089	336.0	0.23	5.95	20.00
4	34.2	771	258.5	0.25	4.43	11.45
3	9.2	512	226.4	0.33	1.80	4.08

Note: EMD 16-645-E3 data on PM emissions was extracted from EPA report on locomotive emission standards [REF-4].

TABLE 5: PM Emission Reduction for EMD 16-645-E3 Engine Retrofitted with TDR Components

Table 6 shows fuel cost and induced savings estimates for the EMD 16-645-E3 engine should it be retrofitted with TDR waste heat recovery and conversion components under the assumptions listed. The results have been obtained assuming averaged diesel fuel cost of 3.545 \$/gallon¹, which is equivalent to approximately 1.042 \$/kg of fuel. Increased average fuel savings can be obtained when retrofitting locomotive engines that produce exhaust gases at higher temperatures than those produced by the EMD 16-645-E3 engine utilized in this example. These aspects are further discussed in Task 6.

Notch	P_{recovd} (kJ/s)	P_{Engine} (kJ/s)	Fuel cons. (kg/s)	Fuel cost (\$/s)	Fuel cost (\$/hr)	$P_{\text{Engine}}/P_{\text{recovd}}$ (%)	\$ Saved (\$/s)	\$ Saved (\$/hr)
8	187.9	2356	0.1514	0.1578	568.1	8.00	0.01262	45.45
7	155.9	1984	0.1285	0.1339	482.1	7.86	0.01010	37.89
6	96.4	1470	0.0900	0.0937	337.5	6.56	0.0062	22.14
5	64.8	1089	0.0690	0.0719	259.0	5.95	0.00428	15.41
4	34.2	771	0.0490	0.0511	183.9	4.43	0.00226	8.15
3	9.2	512	0.0354	0.0369	132.8	1.80	0.00067	2.39

TABLE 6: Cost Savings Estimates EMD 16-645-E3 Engine (under TDR model assumptions)

¹ U.S. Energy Information Administration diesel retail price on Oct/07/2013 in the central Atlantic region.

Table 7 shows the total estimated annual savings from operation of typical line-haul and passenger locomotives, based on the EPA [REF-4] averaged data summarized in Table 2. These results are based on 60% in-service locomotive operation during the year (40% downtime), which is equivalent to 5,256 hours of actual locomotive operation.

Notch	\$ Savings	Line Haul	Total	Passenger	Total
8	45.45	16.2	38,700	15.6	37,266
7	37.89	3.0	5,974	1.4	2,788
6	22.14	3.9	4,538	2.9	3,374
5	15.41	3.8	3,077	4.0	3,239
4	8.15	4.4	1,885	4.7	2,013
3	2.39	5.2	653	5.7	716
2	NA	6.5	NA	5.1	NA
1	NA	6.5	NA	7.0	NA
Idle	NA	38.0	NA	6.2	NA
DB-1	NA	12.5	NA	47.4	NA
Total Saving (\$)	-	-	54,828	-	49,398

NA: not-available as thermodynamic cycle was optimized for high efficiency at Notch settings higher than 2

TABLE 7: Estimated Annual Savings EMD 16-645-E3 Engine Retrofitted with TDR Components

Total savings results reported in Table 7 are based on exhaust gas temperatures sampled at the locomotive exhaust gas stack, thereby post-turbo charger. However, exhaust gases transiting through the exhaust gas piping system, prior to expanding through the turbo-charger, are generally characterized by higher temperatures. As a result, the total savings data reported in Table 7 tend to under-estimate actual savings.

2.3. TASK 4: SENSORS INSTALLATION ON SELECTED LOCOMOTIVE

Figure 3 shows the EMD 16-645-E3 equipped locomotive operated by the Transportation Technology Center (TTCI) and utilized as reference for thermodynamic modeling. The schematic configuration of this engine is represented in Figure 4. As part of Stage I accomplishments, TDR and TTCI representatives inspected the EMD engine to determine minimally invasive methods to equip this locomotive with sensors to support data collection of exhaust gases mass flow rates and temperatures.



FIGURE 3: Views of EMD locomotive operated at the TTCI facility

As shown in Figure 4, the exhaust gas manifolds are equipped with ports to allow installation of probes exposed to the exhaust gas flow. Pressure and temperature sampling may be non-invasively executed by accessing and retrofitting one of two ports provided on the EMD exhaust manifolds and couplers. The location of the ports shown in Figure 4 allows for sampling of exhaust gas parameters prior to the exhaust gases inletting the turbocharger. Port 1 provides access directly onto the coupler connecting the turbo-charger flange to the exhaust manifold, while Port 2, formed by the large flange coupled to the flexible member, is favorable for the insertion of high-temperature thermocouples extending into the exhaust gas free stream. Installation of cabling connecting instrumentation sensors to a power supply and a data acquisition system may be executed by running low-voltage shielded cables across the engine compartment (away from hot surfaces) into the locomotive cab. Intake engine air-mass-flow information is determined by inserting a hot-wire anemometer at the inlet of the coupler hydraulically connecting the locomotive air filter to the turbocharger-driven air-compressor on the compressor suction side.

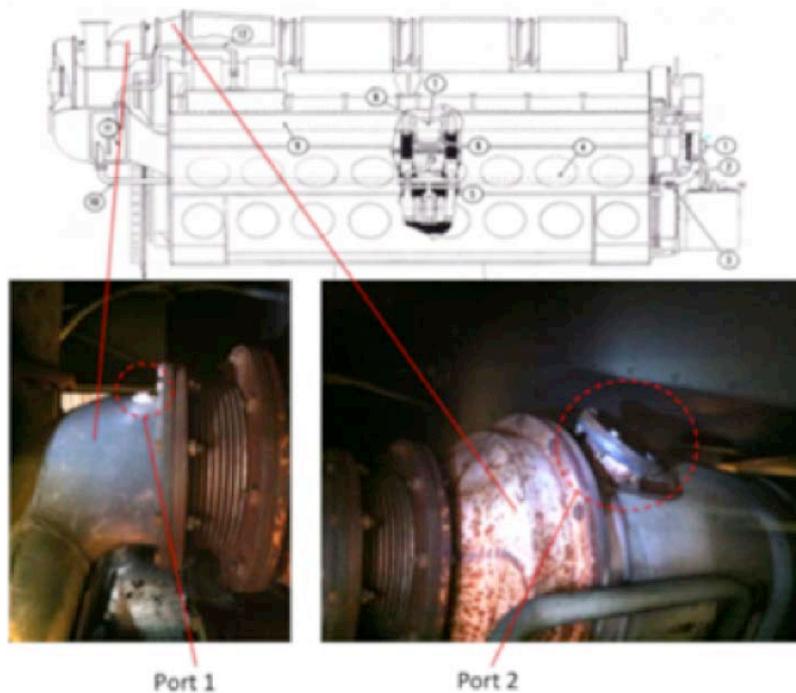


FIGURE 4: Port 1 and Port 2 locations

Figure 5 shows locations for the threaded fitting and removable installation of hot-wire anemometer probe.

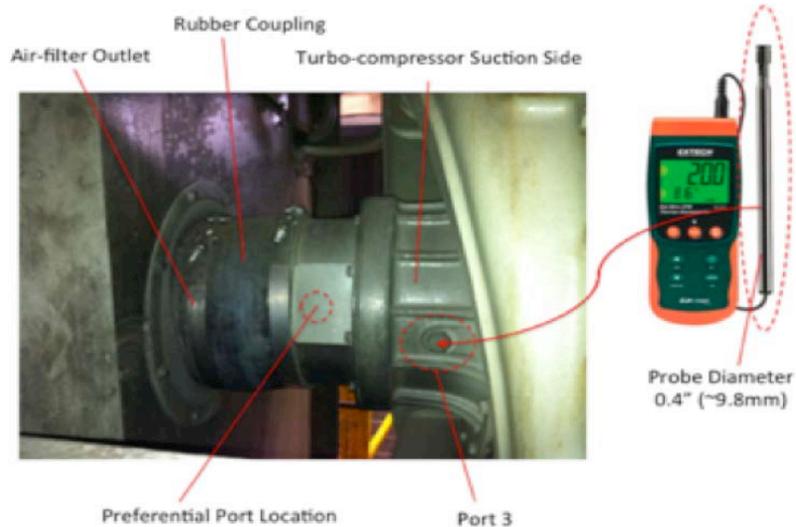


FIGURE 5: Port 3 and hot-wire anemometer probe/sensor location

Figure 6 shows locations for insertion of additional temperature ports so as to record temperature of exhaust gases at the discharge of the turbo-charger and prior to the stack area venting to atmosphere.

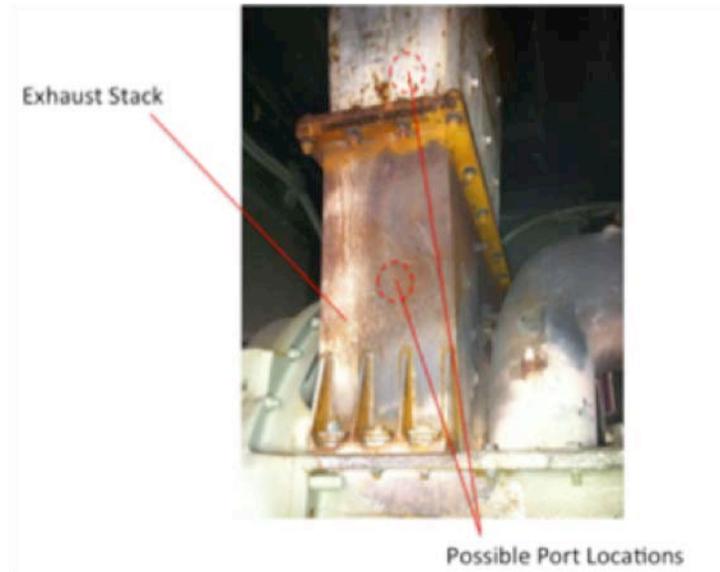


FIGURE 6: Location of thermocouples on stack to sample exhaust gas at turbocharger discharge

2.4. TASK 5: DATA ACQUISITION INSTALLATION ON OPERATING LOCOMOTIVES

EMD engine parameters are logged via onboard computer and provided by TTCL. During inspection of the EMD engine it was determined that the electric bus for “external electric loads” is not rated to provide extra electrical power (i.e. to support TDR high-pressure electric pumps). For this particular locomotive, the electrical power available for computers, servos for automatic radio-locomotive control during testing and logging, is maxed. As a result, a mobile diesel-generator, or access to non-locomotive electric buses (available during static testing) ensures electric power availability for auxiliary components. Alternatively, for the purposes of modeling, exhaust gas data specific to the EMD engine equipping this locomotive can be extracted from publicly accessible authoritative sources (i.e. CARB and EPA reports), and locomotive testing executed under FRA support.

2.5. TASK 6: OPTIMIZATION OF EES-BASED COMPUTER CODES

The EES-based TDR model results were obtained under the assumptions summarized in Table 3. The following discussion addresses assumptions impact in greater detail. The effect of the component efficiencies on the performance of the TDR model are graphically represented in Figure 7, which compares the performance (efficiency) of η (min) when $\eta_{\text{pump}} = \eta_{\text{turbine}} = \eta_{\text{alternator}} = 0.7$ and η (max) when $\eta_{\text{pump}} = \eta_{\text{turbine}} = \eta_{\text{alternator}} = 0.9$. As shown in this Figure, the difference in recovered energy between η (max) and η (min) becomes more significant at high notch numbers. For example, at Notch 8, the power recovered in the η (max) case is 2/3 higher than the η (min) case.

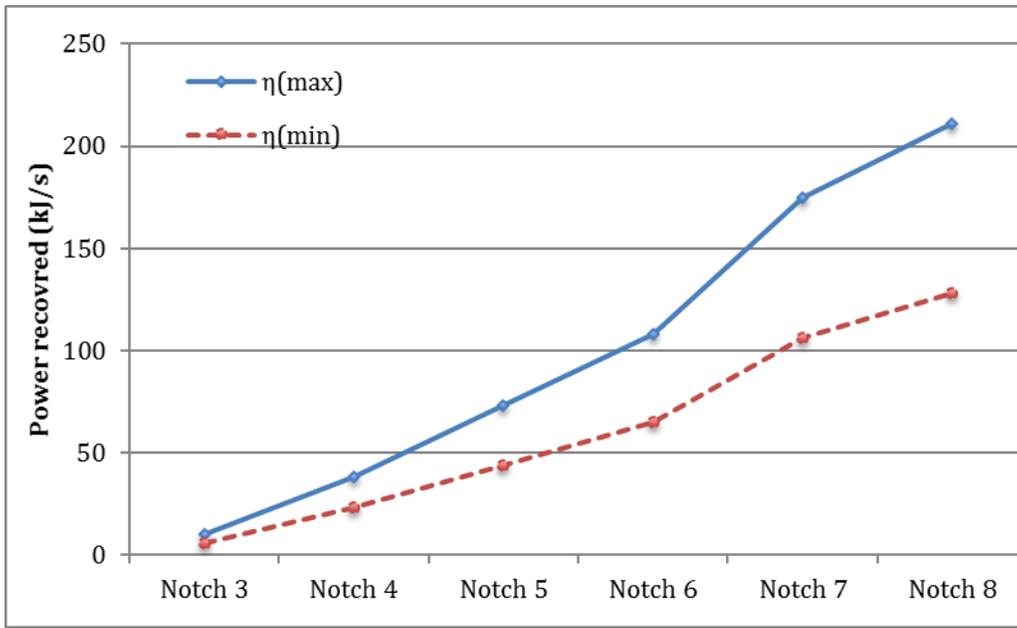


FIGURE 7: Recoverable power EMD 16-645-E3 for η (min) and η (max)

Figure 8 shows the performance of the Rankine power cycle operating under TDR waste heat recovery model assumptions in the case of varying working fluid pressure from P(min) to P(max). The case P(min) models the worst case scenario as the working fluid, prior to expanding in the turbine, is at a pressure of 5 bar and outlets the turbine at a pressure of 1 bar. The case P(max) models the best case scenario, as the working fluid expands in the turbine from a pressure of 20 bars to a pressure of 0.75 bars.

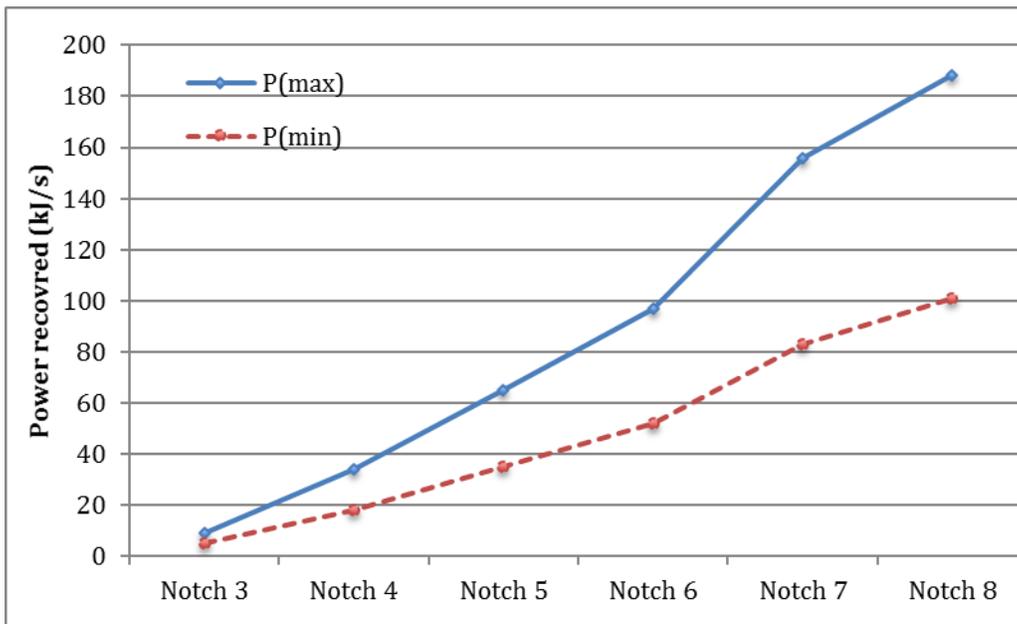


FIGURE 8: Performance of the TDR cycle at different notches for P (min) and P (max)

As mentioned, the above analysis was performed using the exhaust gas temperatures measured at the stack of the EMD 16-645-E3 engine which, as shown in Table 1, reaches a maximum of 388 °C at notch 8. However, typical gas temperatures sampled in the exhaust manifold of generic 4-stroke diesel engines may exceed temperatures of 700 °C (5). TDR waste heat and recovery technologies can be configured to recover and convert thermal energy in the stack as well as in the exhaust manifolds. Table 8 shows the projected performance should $T_{exh,in}$ be increased by 200 °C. This case would generally correspond to a pre-turbo charger retrofitting configuration of the HiPHEXs.

Notch	$T_{exh,in}$	m_{exh}	$T_{exh,out}$	m_{water}	T[3]	W_{turb}	η_{cycle}	η_{recovd}	P_{recovd}
8	587.9	4.937	120	0.820	432.2	514.6	0.210	0.175	437.6
7	586.9	4.136	120	0.700	404.0	422.0	0.206	0.171	358.7
6	542.9	3.566	130	0.520	433.9	327.1	0.210	0.168	278.2
5	510.9	3.138	140	0.420	401.4	252.2	0.205	0.157	214.5
4	471.9	2.692	150	0.310	408.5	188.1	0.206	0.149	159.9
3	430.9	2.336	155	0.250	301.7	130.5	0.193	0.132	110.9

TABLE 8: Rankine Cycle Estimated Performance with $T_{exh,in}$ Increased by 200 °C

Notch Number	\$ Savings	Line Haul	Total	Passenger	Total
8	105.5	16.2	89,850	15.6	86,520
7	87.2	3.0	13,740	1.4	6,413
6	63.9	3.9	13,090	2.9	9,735
5	51.0	3.8	10,190	4.0	10,720
4	38.1	4.4	8,820	4.7	9,420
3	28.8	5.2	7,861	5.7	8,620
2	NA	6.5	NA	5.1	NA
1	NA	6.5	NA	7.0	NA
Idle	NA	38.0	NA	6.2	NA
DB-1	NA	12.5	NA	47.4	NA
Total Saving (\$)	-	-	143,350	-	131,400

NA: not-available as thermodynamic cycle was optimized for high efficiency at Notch settings higher than Notch 3

TABLE 9: Estimated Yearly Savings Under TDR Model Assumptions

Following a similar approach to the one presented in Table 7, Table 9 provides a projection of the total yearly savings of typical line-haul and passenger locomotives operation (>4,000hp rated power), according to EPA (4) locomotive duty

cycle. As mentioned, these results are based on the assumption that locomotives are operated 60% of the time during the year, which is equivalent to 5,256 hours, in agreement with EPA duty cycle shown in Table 2. The Rankine model applied to the EMD 16-645-E3 engine, under the assumptions summarized in Table 3, showed that, for low notch settings, the turbine of the waste heat recovery system operated at relatively high pressure becomes significantly inefficient. One method to configure the turbine to operate at higher efficiency when the locomotive is operated at low notch settings may include the adoption of a throttling valve at the turbine inlet so as to vary the amount of working fluid expanding in the turbine. Other methods involve lowering the working fluid pressure by, for example, controlling pump speed when the locomotive operates at low notch numbers. Alternatively, by changing the physical properties of the working fluid (i.e. utilizing organic fluids), turbines may be designed to operate efficiently at low notch settings. Table 10 shows the performance of the Rankine cycle under TDR model assumptions for the EMD 16-645-E3 engine retrofitted with waste heat recovery components maximizing efficiency at low locomotive notch settings. For example, at Notch 2, optimization occurs by setting the working fluid at the thermodynamic state corresponding to saturated gas prior to inletting the turbine, and pressure set to 4 bars. At Notch 1, optimization is obtained by setting the working fluid pressure at the turbine inlet to 1 bar.

Notch	P[2]=P[3]	T _{exh,in}	m _{exh}	T _{exh,out}	m _{water}	T[3]	W _{turb}	η _{cycle}	η _{recovd}	P _{recovd}
2	4	177.8	1.844	152	0.02	152.2	4.81	0.100	0.016	4.09
1	1	111.8	1.514	99	0.0083	108.9	0.34	0.017	0.0025	0.291

TABLE 10: TDR Cycle Optimization for the EMD 16-645-E3 Engine at Notch 1 and Notch 2

Throughout this analysis the Rankine power cycle under TDR model assumptions utilized water as working fluid. For low exhaust gas temperature applications, organic fluids may be utilized to increase cycle efficiency and recoverable power. At these operating conditions, and utilizing, for example, R134a as working fluid, pressure can be maintained at 20 bars (or higher) even for very low exhaust gas temperatures, and still obtain superheated thermodynamic conditions at the turbine inlet. To obtain a sub-cooled liquid thermodynamic state at the pump inlet at environmental temperatures, the condenser pressure needs to be set to a minimum of 5 bar. Table 11 shows the Rankine cycle performance when TDR model is optimized for operation with R134a at low notch settings of the EMD 16-645-E3 engine. This Table shows that additional analysis focused on waste heat recovery with Rankine cycle operated with refrigerant fluids would provide important insights to increase energy recovery efficiency.

Notch	P[2]=P[3]	T _{exh,in}	m _{exh}	T _{exh,out}	m _{R134a}	T[3]	W _{turb}	η _{cycle}	η _{recovd}	P _{recovd}
1	20	111.9	1.513	66.86	0.314	104.1	9.166	0.11	0.0638	7.791
Idle	20	77.85	1.29	62.81	0.109	72.42	2.628	0.107	0.0342	2.234

TABLE 11: TDR Cycle for the EMD 16-645-E3 Engine at Low Notch Numbers Using Refrigerant R134a

2.6. TASK 7 & 8: EXHAUST GAS SIMULATOR DESIGN, ASSEMBLY & DAQ IMPLEMENTATION

Figure 9 illustrates the functional diagram of a modified oil burner configured to produce variable amounts of exhaust gases as a result of combustion of diesel fuel mixed with air. Oil burners are generally used for domestic or industrial

heating and are equipped with means to change mass flow rate of air and fuel by changing air fan and fuel pump speed respectively. Generally, in these devices, diesel fuel is injected through a nozzle by operating an integral positive displacement gear pump driven by an electric motor that is also mechanically coupled to the intake air fan. For the purposes of simulating exhaust gases at different temperatures and mass flow rates, the air inletting the burner side is mixed with additional environmental air in controlled amounts so as to fine-tune the exhaust gas temperature and mimic exhaust gases when the locomotive is operated at different conditions (i.e. idle to notch 8). As the exhaust gases generated by the oil burners have the same characteristics of those generated by the locomotive, their temperature and mass flow rate may be varied through metering the fuel inletting the burner and by mixing cold air with the combustion gases in a proportion to matches total exhaust gas temperature and mass-flow-rate sampled at the locomotive exhaust gas manifold.

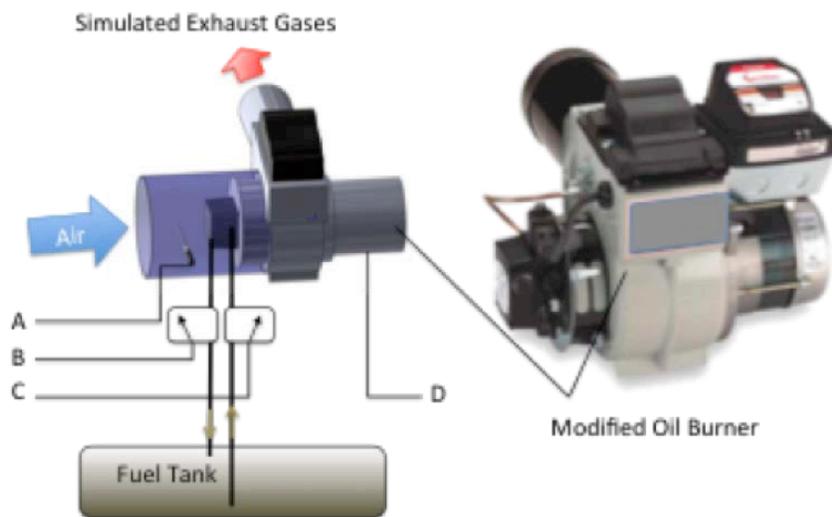


FIGURE 9: Modified oil burner to generate exhaust gases

As a result, modified oil burners may be utilized as “exhaust gas generators” and simulate locomotive exhaust gas parameters at various operational conditions. By regulating the amount of inlet air and fuel inletting and returning from the oil burner pump, the mass flow rate of exhaust gases and their temperature may be fine-tuned so as to impact combustion parameters as desired. In Figure 9, the schematic connection represented by “A” indicates air-flow information/data as measured by means of a hot wire anemometer. Similarly, connections B and C represent information/data to support a totalizer fuel flow-metering computer configured to quantify the amount of fuel inletting the burner and the excess fuel returning to the fuel tank. To lower exhaust gases temperature and accurately mimic locomotive engine operations at idling or low notch settings, a second air fan may be coupled to the burner simulator structure so as to provide an additional controlled cold air mass-flow-rate, thereby diluting the high temperature exhaust gases produced by the oil burner.

2.7. TASK 9: SMALL-SCALE HiPHEXs DESIGN TAILORED TO LOCOMOTIVE APPLICATIONS

The High-Pressure Heat Exchangers (HiPHEXs) are thermal hydraulic systems capable of operating at high pressure while fully engulfed in flames even under total loss of coolant scenarios (i.e. breakage of Rankine loop piping). The

HiPHEXs shown in Figure 10 were utilized to provide data in support of thermodynamic analyses under various operating conditions. A diesel-electric generator was modified to execute locomotive duty cycles by loading the generator with an electric load bank so as to finely mimic line haul and commuter locomotives duty cycles. The HiPHEXs shown in this Figure are coupled directly to a small-scale engine block configured to mimic locomotive operating conditions and exhaust gas temperatures.

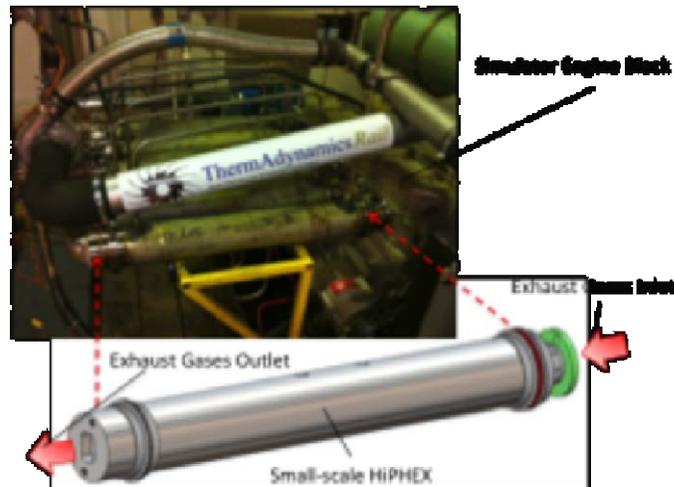


FIGURE 10: HiPHEX retrofitting small-scale diesel-electric generator

2.8. TASK 10: HiPHEX PROTOTYPE ASSEMBLY

Full-scale HiPHEX prototypes dimensioned for minimally invasive retrofitting of the exhaust gas manifolds have been assembled and tested under various operating conditions. For example, in Figure 10, a series of HiPHEXs were non-invasively inserted inside individual exhaust gas manifolds (at pre-turbocharger locations) of a modified diesel-generator. HiPHEXs may be configured for counter-flow, parallel-flow or counter- and parallel-flow combinations so that the working fluid may circulate through different HiPHEXs in a manner that maximizes waste heat recovery. These features allow “fine-tuning” the working fluid temperature, for example, to optimize turbine inlet temperature while addressing exhaust gas minimum temperature requirements with respect to different types of turbochargers (i.e. centrifugal or mechanically-driven).

3. STAGE II – WORK PERFORMED

The following chapters, identified by Tasks, describe the work performed in agreement with TRANSIT-67 Stage II Subject Work.

3.1. TASK 11: SIMULATOR/HIPHEX COUPLING

To satisfy small-scale testing requirements, a series of custom made High-Pressure Heat Exchangers (HiPHEXs) have been non-invasively retrofitted directly inside GE Dash-9 locomotive exhaust manifolds (4,400hp rated power). Such a

configuration minimally impacts back-pressure inside the manifold during locomotive operation and can be used without modifications when coupled to a burner system (see Tasks 7 & 8) to simulate exhaust gases. Coupling the non-invasively retrofitted manifolds with a burner system allows estimating the performance of multiple HiPHEXs when operated on different locomotives, possibly with different exhaust gas manifolds.

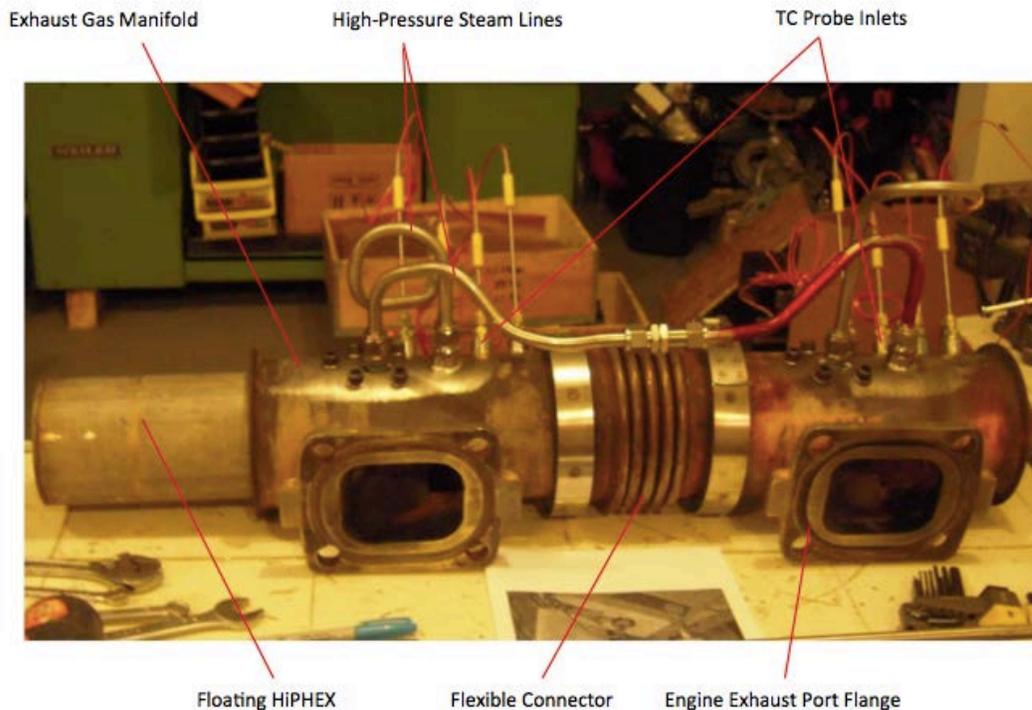


FIGURE 11: HiPHEX’s retrofitted with two-of-eight symmetrical GE Dash-9 exhaust manifold

Figure 11 shows a “floating HiPHEX” on the left (the second HiPHEX is internally coupled to the symmetrical exhaust gas manifold on the right). These HiPHEXs were manufactured according to design optimization obtained in Task 9 and leveraged tests results executed under the Federal Railroad Administration (FRA) grant FR-RRD-0005-10-01-00. As shown in this Figure, each HiPHEX is modular and positioned within an individual exhaust gas manifold separated by a flexible connector. Water, the working fluid considered in this analysis, is pressurized at the inlet of the first HiPHEX (see “high-pressure steam lines” in Figure 11), and flows in the opposite direction relative to the exhaust gas flow, thus both heat exchangers are thermally coupled with the exhaust gases in “counter flow” configuration. The energy content of the water increases as it flows through the first HiPHEX via thermal transfer from the exhaust gases. The working fluid then outlets the first HiPHEX and re-enters the symmetrical high-pressure inlet of the second HiPHEX. Depending on the mass-flow-rate of the water, that of the exhaust gases, and average temperature of the exhaust gases, the water at the outlet of the first HiPHEX may be in a thermodynamic state defined as sub-cooled liquid or superheated vapor.

3.2. TASK 12: SIMULATOR/HIPHEX/PUMPING SYSTEM COUPLING

Figure 12 shows the water supply system utilized to test the HiPHEXs when operated with exhaust gases generated by the burner system. Water is stored in a reservoir (white PVC tank on the left) hydraulically connected to the suction side of an electrically-driven pump, and flows, once pressurized at the pump discharge, through a water flow meter, which displays the flow rate (l/min) and provides data to a dedicated Data Acquisition System (DAQ). The pump is formed by a ½ hp induction motor coupled to a positive displacement rotary pump, which at atmospheric pressure and negligible head (open discharge and zero-elevation) provides a maximum flow rate of approximately 154 GPH (~10 l/min). The water mass-flow-rate is regulated by changing the rotary speed of the positive displacement pump, and by throttling the discharge via the valve. The rotary speed is regulated by means of an electronic controller, while the throttling valve is adjusted manually so as to obtain the desired operational pressure inside the HiPHEXs.

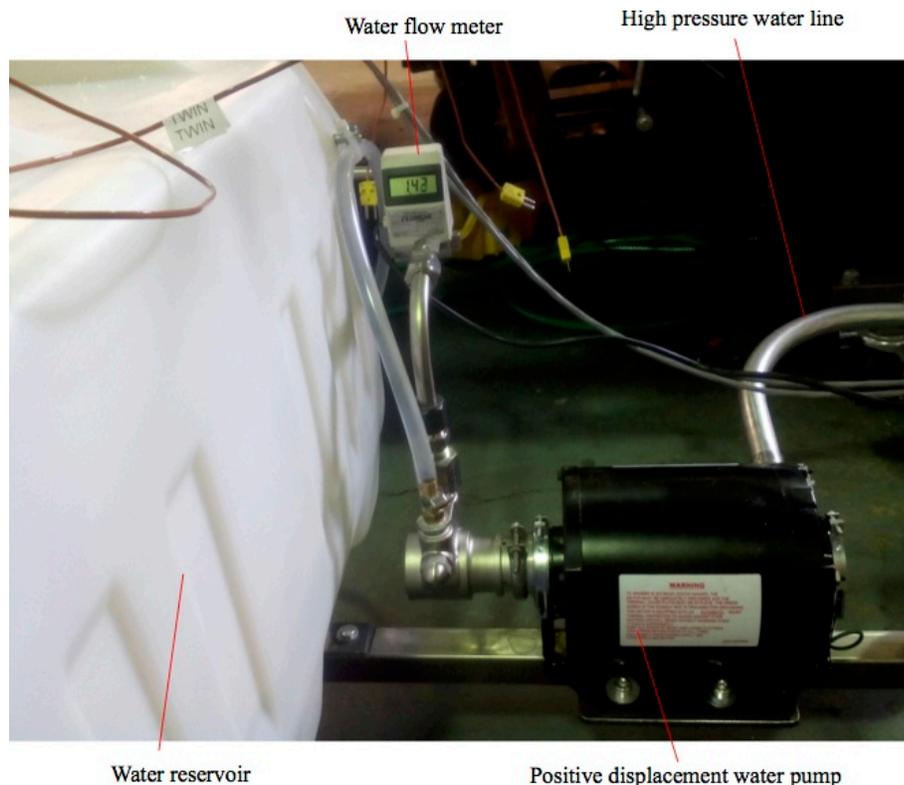


FIGURE 12: Schematics of 1/2 hp positive displacement pump

3.3. TASK 13: DAQ CONNECTION AND TUNING

Figure 13 shows the instrumentation utilized for data collection. Temperature measurements were obtained using K-type thermocouples. The exhaust gas temperatures sampled at the HiPHEX inlet and outlet (corresponding to $T_{exh,in}$ and $T_{exh,out}$ in Figure 2), were measured by positioning thermocouples upstream and downstream of the HiPHEX (Figure 13(d)). Steam temperature at the HiPHEX outlet ($T[3]$ in Figure 2), was measured using a thermocouple positioned inside the high-pressure stainless steel tubing coupling the HiPHEX to the condenser (Figure 13(f)). The temperature of the water in the reservoir ($T[1]$ in Figure 2) was measured by placing a thermocouple directly inside the water reservoir.

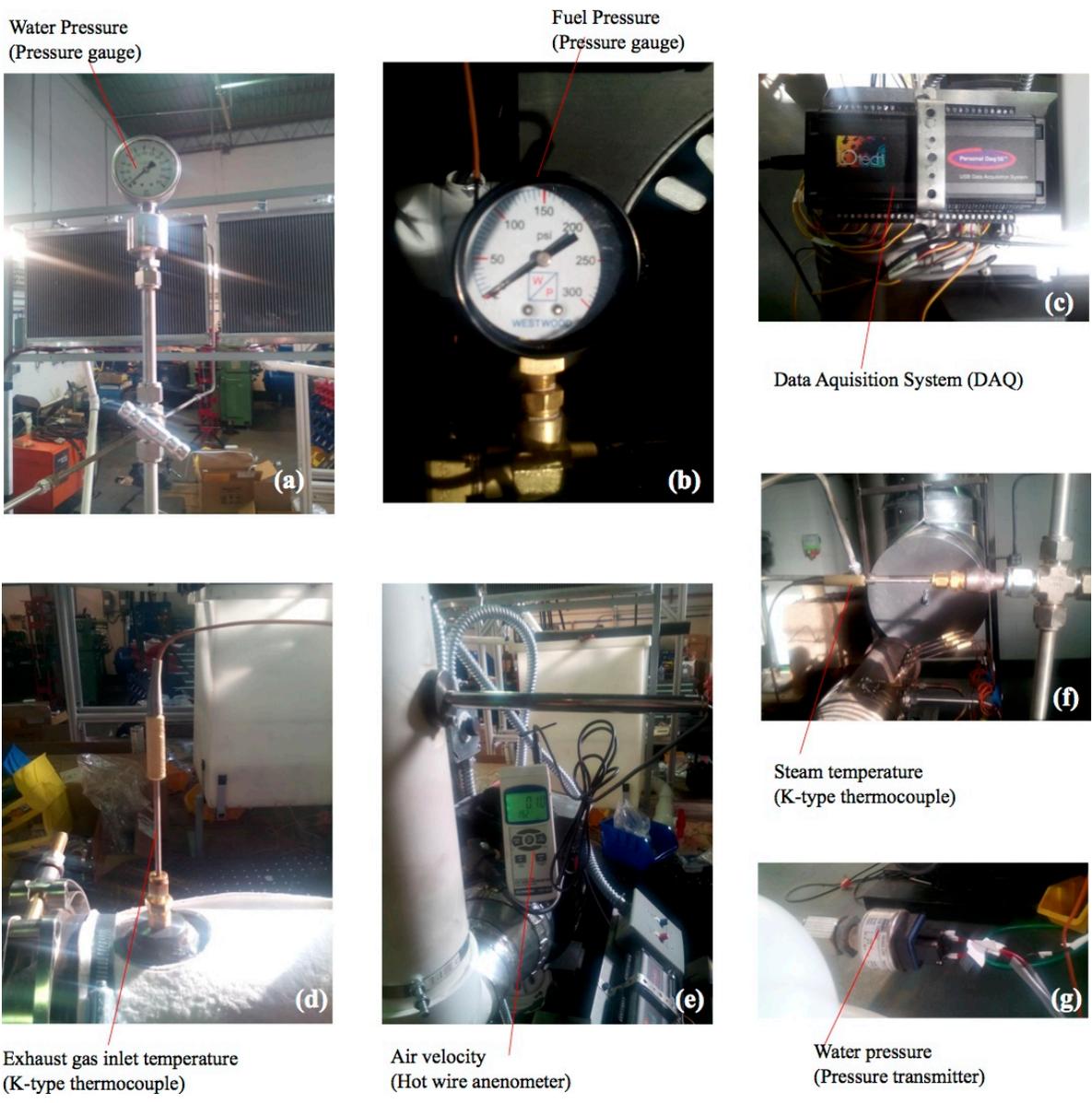


FIGURE 13: Burner instrumentation

The water pressure at the pump discharge (P[2] in Figure 2) was measured using a pressure transmitter (Figure 13(g)), which was also connected to the DAQ. To support testing optimization, an analog pressure gauge (Figure 13(a)) was connected to the high-pressure tube at the HiPHEX outlet (P[3] in Figure 2). As outlined in Task 12 a water flow meter was used to measure the water mass-flow-rate (see Figure 12), which provided data to the DAQ electronic card interfaced to a computer and enabled real-time visual representation of the mass- flow-rate. This allowed fine-tuning of water mass-flow-rate and pressure by adjusting the manual valve.

Finally, the airflow rate utilized to estimate the total mass-flow-rate of the exhaust gases was measured by utilizing a hot wire anemometer. This measurement represented several challenges as the hot wire anemometer cannot operate at high

temperature, thus forcing measurement samplings to be taken at the air-intake conduit feeding the oil-burner forming the exhaust gas simulator. As a result, the air-intake system equipping the oil burner (see Figure 9) was modified by connecting it to a coupler to allow reasonably accurate sampling of the air velocity (see Figure 13(e)), thereby enabling data collection of the total mass of exhaust gases flowing through the HiPHEXs.

3.4. TASK 14: SIMULATOR INITIAL TESTING AND TUNING

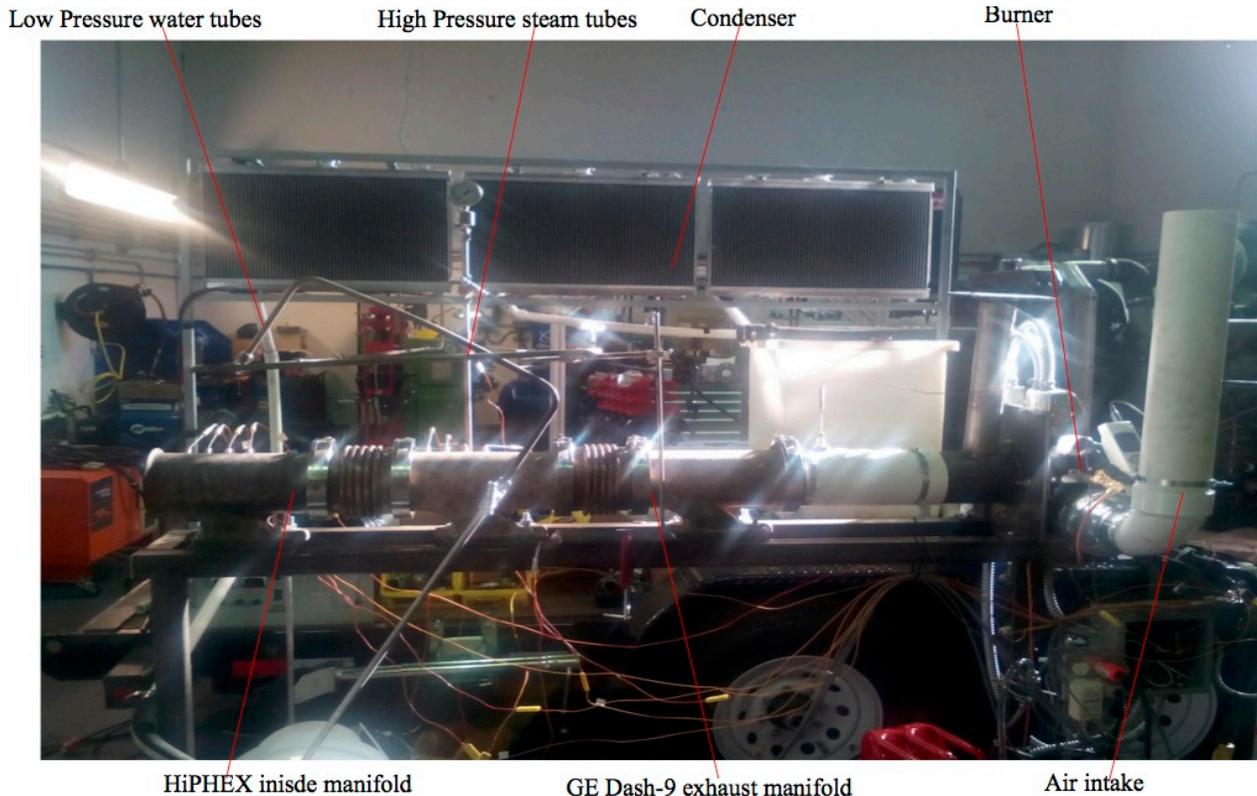


FIGURE 14: HiPHEX Retrofitted GE manifolds coupled to burner/simulator

Figure 14 shows the experimental set up including the HiPHEXs coupled to the exhaust gas generator developed to optimize and validate Stage I computer models and to assess HiPHEX economic performance. As shown in this Figure, the oil-burner (right of Figure 14) is hydraulically coupled to a set of retrofitted GE dash-9 exhaust gas manifolds. High temperature exhaust gases generated via burner are mixed with air supplied by a separate blower so as to obtain similar stoichiometric conditions found in typical locomotive exhaust manifolds (see Task 7&8). As the burner is operated, the exhaust gases flow into the air-mixer and into the exhaust gas manifolds retrofitted with the HiPHEXs.

As mentioned, in this first HiPHEX the water energy content increases while circulating in counter-flow configuration (with respect to the flow direction of the exhaust gases, see Task 11). The resulting sub-cooled liquid water, or steam – depending on mass-flow rates and exhaust gas temperature at the HiPHEX inlet, is then superheated inside the second HiPHEX. The thermodynamic properties of the superheated steam are sampled at the outlet of the second HiPHEX through thermocouple T[3] and pressure gauge P[3], while it is discharged inside a set of radiator heat exchangers with

forced convection to force condensation. Once condensed, water flows by gravity back into the reservoir, thereby resetting the closed-loop cycle. Condensation rates are changed by means of electrical fans regulated via power supply.

3.5. TASK 15: OPTIMIZATION

During testing of the “first-generation” burner system (locomotive exhaust gas simulator) developed as part of the activities contemplated under TRANSIT-67, unforeseen problems emerged. Figure 15 shows a snap-shot of the flame formed by the burner system when firing into the GE exhaust manifold retrofitted with two HiPHEXs. As shown, the flame forms at a relatively long distance from the burner nozzle, thus it manifests downstream of the retrofitted GE manifolds. The flame characteristics, temperature and color indicate that complete combustion is not occurring. This resulted in excessive soot formation and unburned fuel accruing throughout the GE exhaust manifolds and within the exhaust gas extraction system (large cylindrical ducts shown to the left of the experimental set up). After trouble shooting the problem, it was concluded that poor combustion and burnout were mainly caused by the overall small size of the combustion chamber compounded with the fact that the commercial burner’s components (i.e. fuel injection and air-fan tandem driven systems) do not allow pressurization of the combustion chamber. Under-dimensioning of the combustion chamber total volume was “forced” in an effort to maintain the same hydraulic diameter characterizing the GE-Dash-9 exhaust gas manifolds. Under these dimensional constraints, the high-pressure atomized fuel is injected inside a relatively small cylindrical combustion chamber lined with high-temperature ceramic materials. However, at the fuel rates required to mimic scaled locomotive conditions, the flame forming at the nozzle of the burner “wets” the ceramic liner “too early”, or at a distance that is too close to the fuel injector nozzle tip, subsequently, the flame does not have enough volume to properly mix with the oxidizer (air), thus resulting in a delayed and incomplete combustion. As the unburned fuel accumulates on the walls of the HiPHEXs and inner walls of the exhaust gas manifolds, it further cools the flame downstream, thus creating the conditions for potentially dangerous backfires. Figure 15 shows the two HiPHEX retrofitted GE-Dash-9 exhaust gas manifolds, wherein the second floating HiPHEX is flame engulfed.

Under the limitations induced by the combustion chamber assembled for this first generation burner simulator, testing aiming at collecting meaningful data to simulate various notch numbers was not feasible. The flame reached the highest temperatures in the regions where the HiPHEX’s are located, thereby resulting in exhaust gas temperatures in excess of 1,000°C. Such temperature values are too high when compared to those encountered in the actual exhaust gas manifolds of diesel-electric locomotives, even when operating at maximum power rating. The HiPHEXs are designed to operate dry while fully engulfed in high temperature flames so these tests were not challenging the HiPHEXs structural integrity. Tests were executed intermittently at various fuel, air and water mass-flow-rates, and some tests went on for hours. Under full flame engulfment, even under complete loss of coolant (zero mass flow of circulating water), the heat exchangers did not manifest signs of fatigue or damages, thus proving that they can withstand prolonged operations under the harshest operational conditions. These same conditions would normally be catastrophic for conventional heat exchangers.

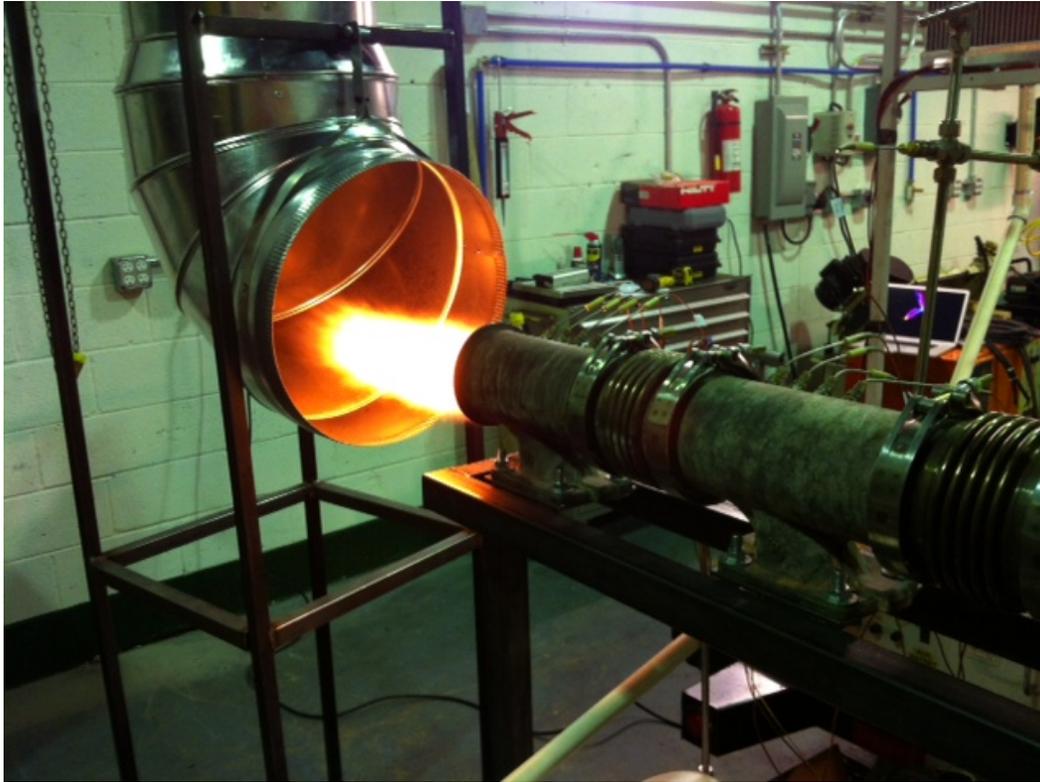


FIGURE 15: Burner firing into the GE Dash-9 exhaust manifold and resulting outlet flame

By expanding the combustion chamber upstream of the exhaust manifold, and by enhancing air mixing to calibrate and more accurately regulate exhaust gas temperatures, the burner simulator can be optimized. More generally, with a larger combustion chamber the atomized fuel and surrounding oxidizer would have enough volume and time to mix to complete combustion. Under these conditions, the exhaust gases could then be mixed with controlled air bled into the system to adjust the exhaust gases final temperature as desired and flow through the exhaust gas manifolds retrofitted with the HiPHEXs. The burner-locomotive simulator objective was to generate controlled exhaust gases with the same properties characterizing exhaust gases generated by diesel-electric locomotives. Modeling and CADs of an expanded combustion chamber based on the shortcoming of the first generation burner-locomotive simulator are underway and will be the subject of a new proposal to be submitted to the IDEA program.

3.6. TASK 16: HIGH PRESSURE TESTING OF SMALL-SCALE HiPHEX's

Each individual high-pressure heat exchanger (HiPHEX), was tested for high-pressure capability. This test consisted of plugging the HiPHEXs' outlets, while hydraulically coupling the inlets to a high-pressure positive displacement pump providing 3,000 PSI (~207 bar). Tests were executed by pressurizing the heat exchangers at maximum pressure for 24 hours. Water was utilized as the working fluid for this test. The pressure pump consisted of a modified 5hp electrically-driven positive displacement piston pump. Pressure was regulated by means of a by-pass valve at the high-pressure side of the positive displacement pump head, and by regulating electrical power to the 5hp induction motor. The induction motor rotary speed and torque was in-turn regulated by means of a "vector-drive" electronic controller. The HiPHEXs

were also tested once connected in series and counter-flow configuration as fully retrofitted with the GE Dash-9 exhaust gas manifolds. This additional test was executed to ensure there would not be leakages developing when the system was coupled to the exhaust gas simulator. No leakages, nor deformations were detected after the high-pressure tests.

3.7. TASK 17: DATA ANALYSIS AND FINE-TUNE TESTING

Task 15 showed that the “first generation” of the exhaust gas simulator generated exhaust gas temperatures with temperatures significantly higher than those characterizing exhaust gases of diesel-electric locomotives, even when locomotives are operated at maximum power (i.e. notch 8). Re-designing and manufacturing an expanded combustion chamber was unfortunately outside the scope, timing, and budget dedicated to this project. However, to increase accuracy of the computer models presented in Stage I section 2, a set of experimental exhaust gas measurements obtained from retrofitting GE-dash 9 exhaust gas manifolds, actually tested on an operational GE Dash-9 locomotive, were utilized as sample data in place of the experimental data to be obtained by operating the exhaust gas burner-locomotive simulator. The exhaust gas data from the GE Dash-9 locomotive were obtained by operating the locomotive statically at different Notch numbers and by dissipating electric energy produced by the engine through resistor banks forming the locomotive dynamic brakes. Locomotive data were then utilized to mimic the closed-loop configuration shown in Figure 14 and described in Task 15. The mass-flow-rate of exhaust gases generated by an actual locomotive is considerably higher than that generated by the small-scale burner simulator as that shown in Figure 14. For example, the total energy contained in the exhaust gases when generated through the diesel-burner simulator is proportional to the energy attainable by burning between 5gal/hr to 10 gal/hr (0.00444 kg/s – 0.00888 kg/s) of diesel fuel. When exhaust gases are locomotive-generated, the energy content may be proportional to burning approximately 60gal/hr-260gal/hr, depending on the locomotive operating conditions (i.e. from idling to Notch 8 respectively).

Notch	T _{exh,in} °C	m _{fuel,L} kg/s	AFR -	m _{exh,L} kg/s	T[2] °C	T[3] °C	m _{w,L} kg/s
8	610	0.104	32.0	3.43	47	177	0.12
7	537	0.0765	35.0	2.75	40	169	0.1
6	511	0.0655	35.4	2.39	33	166	0.085
5	510	0.0535	35.0	1.92	35	155	0.085
4	512	0.0375	32.4	1.25	39	139	0.09
3	508	0.027	30.8	0.86	31	107	0.1
2	360	0.0105	39.1	0.41	28	64	0.1
1	260	0.00485	59.5	0.29	24	45	0.1
IDLE	156	0.00115	171.4	0.40	28	37	0.1

TABLE 12: GE Dash-9 Exhaust Gas Experimental Measurements

Table 12 summarizes key GE Dash-9 locomotive exhaust gas parameters, where $T_{exh,in}$ is the temperature of the exhaust gases prior to entering the HiPHEXs retrofitting two of the exhaust gas manifolds, $m_{fuel,L}$ is the locomotive mass flow rate or fuel consumption, AFR is the air to fuel ratio and $m_{exh,L}$ is the mass flow rate of the exhaust gases generated by the locomotive. Under these exhaust gas conditions, and under Stage I model simplifying assumptions, two HiPHEX's coupled in counter flow configuration increase the temperature of water, flowing from inlet of the first HiPHEX to outlet of the second HiPHEX with a mass flow rate $m_{w,L}$, from T[2] to T[3].

Notice that the same exhaust gas temperatures ($T_{exh,in}$) are obtainable with a 5 gallon/hr (0.00444 kg/s) burner if the AFR is maintained, however at a much lower mass flow rate m_{exh} . Table 13 shows the exhaust mass flow rates m_{exh} , which would result from firing the burner system with a 5 gallon/hr (0.00444 kg/s) nozzle and maintaining the same AFR sampled through the GE Dash-9 test. Although the HiPHEXs configuration for the tests executed with the GE locomotive with retrofitted exhaust gas manifolds is the same as that employed for the small-scale burner system, it is assumed that if m_w is dropped with equal proportions with respect to m_{exh} then the water temperature increases from T[2] to T[3] following the same behavior. This assumption leads to inaccuracies when scaling performance from locomotive conditions to simulator conditions, and vice versa, as the change in m_w also affects the Reynolds number, and therefore impacts heat transfer rates. In other words, the exhaust gas velocity and the velocity of water through the components forming the same retrofitted manifolds are significantly different when scaling from “burner-size” to locomotive size and vice versa. A properly dimensioned combustion chamber coupled to the burner simulator with the ability to pressurize the combustion chamber will allow to better mimic locomotive generated exhaust gas conditions by scaling up and down the gases and water mass-flow-rates and temperatures, therefore enabling the identification of key parameters to increase modeling accuracy.

Table 13 shows the projected exhaust gas parameters obtained through a burner simulator equipped with a 5 gallon/hr nozzle when utilized to mimic locomotive generated exhaust gases. The results at low notch numbers have been ignored in line with the discussion presented in Section 2.

Notch	$T_{exh,in}$	m_{fuel}	AFR	m_{exh}	T[2]	T[3]	m_w
	°C	kg/s	-	kg/s	°C	°C	kg/s
8	610	0.00444	32.0	0.14652	47	177	0.00512
7	537	0.00327	35.0	0.11757	40	169	0.00380
6	511	0.00280	35.4	0.10179	33	166	0.00300
5	510	0.00228	35.0	0.08223	35	155	0.00260
4	512	0.00160	32.4	0.05347	39	139	0.00064

TABLE 13: Assumed Experimental Measurements Obtainable via 5 gallon/hr Burner

Following the same approach as Tasks 2 and 6, the experimental measurements presented in Table 13 were used as data basis for the computer models. By setting the exhaust gases inlet temperature ($T_{exh,in}$) and mass flow rate (m_{exh}), the

code was then utilized to calculate the mass flow rate m_w necessary to increase the water temperature from $T[2]$ to $T[3]$ (as physically sampled from the actual locomotive test).

Notch	Exp. (2 x HiPHEX's)			Exp. (6 x HiPHEX's)			Model predictions			η_{Hex}
	m_w	q_{rec}	W_{turb}	m_w	q_{rec}	W_{turb}	m_w	q_{rec}	W_{turb}	
	kg/s	kJ/s	kJ/s	kg/s	kJ/s	kJ/s	kg/s	kJ/s	kJ.s	%
8	0.00512	13.3	1.546	0.01537	39.9	4.639	0.0307	79.93	9.278	50.0
7	0.00380	9.928	1.136	0.01140	29.784	3.410	0.022	57.58	6.593	51.7
6	0.00300	7.927	0.822	0.00900	23.781	2.467	0.0171	45.2	4.69	52.6
5	0.00260	6.782	0.704	0.00780	20.346	2.112	0.0149	37.3	3.872	54.5
4	0.00064	3.072	0.049	0.00192	9.2166	0.147	0.0064	16.28	0.261	56.6

TABLE 14: Comparison Between Experimental and Computer Model Results

Table 14 compares the experimental measurements of the TDR cycle with the computer model predictions, where q_{rec} is the heat energy recovered by the HiPHEX and W_{turb} is the work performed by a turbine² operated at the given steam conditions. The experiments were performed with only two of six HiPHEXs, which were designed to fit a whole GE-Dash 9 exhaust gas manifold bank – there are a total of two manifold banks for the 16-cylinder engine utilized for this test. Table 14 shows the extrapolated results for a retrofitting configuration involving 6 HiPHEX's; assuming that each HiPHEX transfers the same amount of thermal energy.

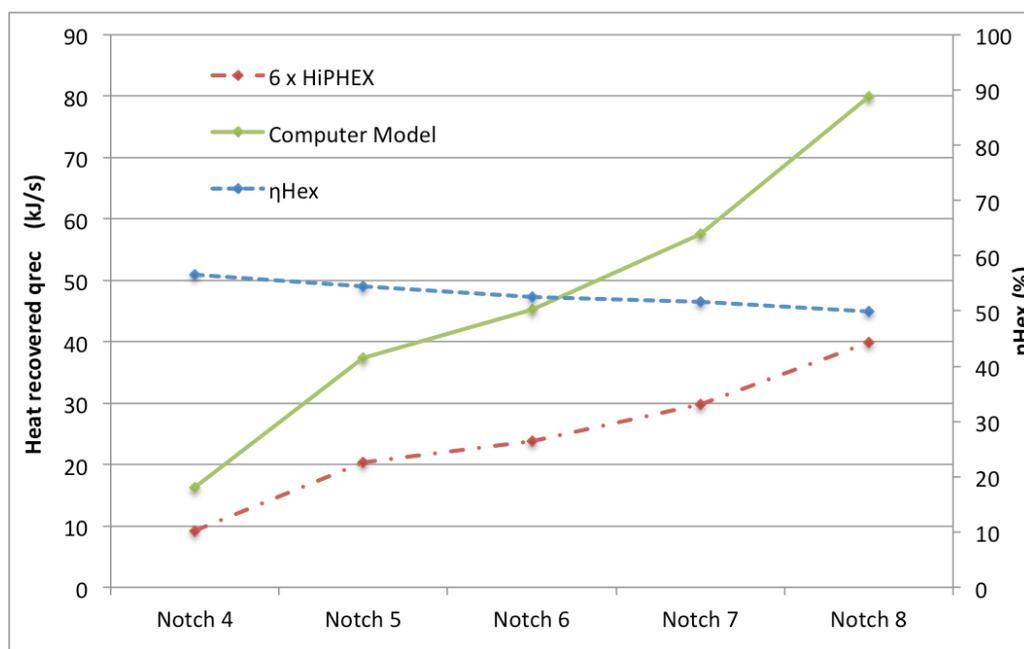


FIGURE 16: Heat recovered and HiPHEX performance at different notch numbers

² The turbine work has been calculated using the same assumptions used in Section 2.

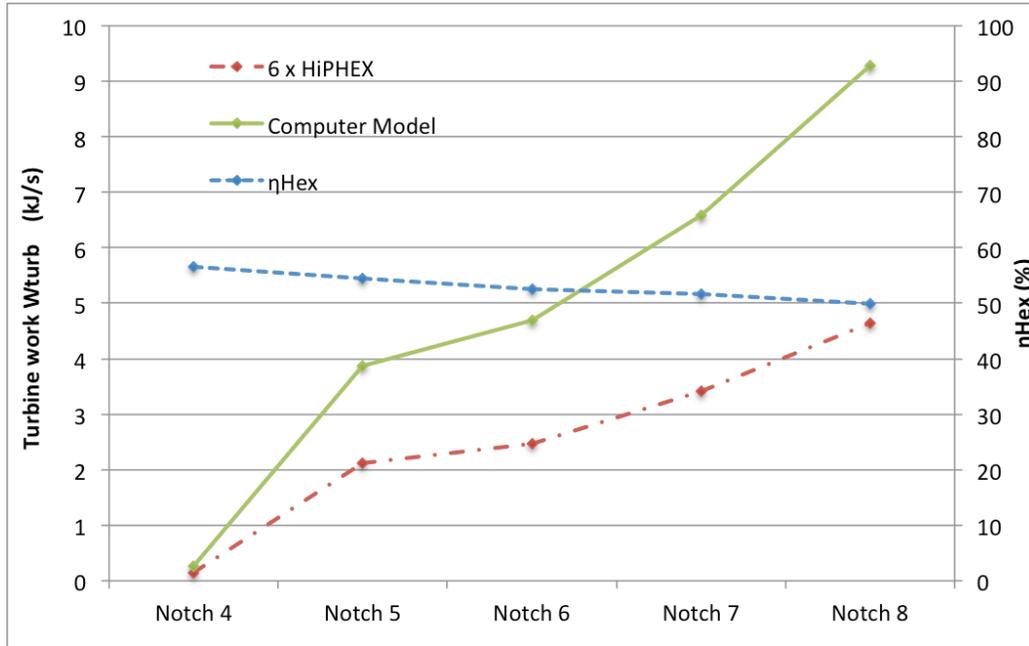


FIGURE 17: Turbine work and HiPHEX performance at different notch numbers

Figures 16 and 17 compare the heat recovered q_{rec} and the potential turbine work W_{turb} obtained experimentally with the computer model. Accordingly, q_{rec} measures the maximum possible energy transferred by the HiPHEX. The values of q_{rec} and W_{turb} obtained with the computer models will thus represent the maximum achievable heat recovered and potential turbine work obtainable with an infinitely extended HiPHEX heat transfer surface.

In Figures 16 and 17, η_{Hex} measures the ratio of q_{rec} obtained experimentally (extrapolated to 6 HiPHEX's) to q_{rec} obtained computationally, and is a measure of HiPHEX performance. However, data from the HiPHEX retrofitted and tested on the GE Dash-9 locomotive once utilized in the computer model also show a decrease of the HiPHEX effectiveness. This is mainly due to the fact that the total mass of exhaust gases flowing through these two heat exchangers was essentially “thermally saturating” the heat exchangers. In other words, above a certain exhaust gas mass flow rate threshold the total surface area of the retrofitted HiPHEX becomes insufficient to transfer additional energy to the working fluid at the given rates. This also shows that extrapolating the behavior of two HiPHEXs to six under a linear assumption leads to inaccuracies. More generally, Figures 16 and 17 show that the most basic HiPHEX design (minimum heat transfer surface area to comply with non-invasive retrofitting requirements) performs reasonably well at all notch numbers (~50%).

Nonetheless additional experimental measurements are necessary to further validate the results and aid the design toward HiPHEXs configurations with higher effectiveness (i.e. extended surface area within the dimensional constraints and to comply with non-invasive retrofitting requirements). Costing analysis of the components forming the waste heat recovery and conversion technologies developed by ThermaDynamics Rail LLC was outside the scope of TRANSIT-67. Generally, retrofitting costs vary proportionally to the efficiency of the Rankine power cycle. The higher the efficiency,

the higher the costs associated with the components forming the thermodynamic cycle. ThermaDynamics Rail is optimizing these components to ensure locomotive owners can achieve ROI not exceeding 2.5 years.

4. SUMMARY

Baseline data collected from literature and from measurements of selected locomotives equipped with widely utilized type of engines were analyzed with respect to technical and economic performance of waste heat recovery technologies developed by Thermadynamics Rail LLC (TDR). The collected data involved identification of key exhaust gas parameters including overall dimensions of exhaust stack, manifolds, temperature and velocity of the exhaust gases sampled at given locations of the hydraulic exhaust gas piping system. Data analysis also included Particulate Matter emissions and fuel consumption rates. Additional information addressing locomotive duty-cycles focused on line-haul and commuter rail operations was also utilized for the purposes of estimating economic performance of TDR waste heat recovery technologies when utilized to retrofit selected locomotive engines.

A thermodynamic model to represent the thermal-hydraulic behavior of TDR waste heat recovery and conversion components was optimized using the commercially available software Engineering Equation Solver (EES). Under a set of simplifying assumptions, the overall performance of the TDR model was investigated for locomotives operating at different notch settings. When analyzing a locomotive engine producing relatively low temperature exhaust gases, a Rankine-based power cycle, executed under TDR waste heat recovery components configuration, can achieve a total energy recovery of approximately 13% at Notch 3 to 17% at Notch 8 of the total waste heat energy otherwise vented to atmosphere. The recovered energy may then be directly converted into pollutant-free propulsion or electric power, thereby reducing fuel consumption and exhaust gas emissions. Accordingly, under industry accepted assumptions of locomotive averaged usage and duty cycles, when the waste heat recovery operated under TDR modeling is retrofitted with engines producing averaged exhaust gas temperatures, the yearly fuel savings can vary between \$54,000 and \$140,000 per locomotive.

The effects of assumptions adopted to simplify the bottom-cycle Rankine model utilized were analyzed by varying the efficiencies of the pump, turbine and alternator. As expected, turbine and alternator efficiencies were found to induce considerable effects on the model performance. Additionally, the effect of pressure ratio between the high- and low-pressure thermodynamic state of the working fluid was also assessed, thus providing results indicating how pressure in the waste heat recovery Rankine cycle may be optimized to increase efficiency at low notch settings.

Two different approaches were considered for achieving waste heat energy recovery when locomotives are operated at idle and low notch settings. One approach involved lowering the Rankine cycle operating pressure. The second approach involved utilization of an organic working fluid such as R134a. In both cases very modest power recovery could be obtained at idle and low Notch numbers.

A “first generation” low-cost locomotive-burner simulator was utilized to validate computer model results. The simulator was developed to generate exhaust gases with characteristics comparable to those generated by locomotive engines. As the locomotive-burner simulator was tested it became apparent that the dimensional constraints imposed by

the locomotive exhaust gas manifolds induced a combustion chamber hydraulic diameter too small for complete combustion of the fuel. As a result, the locomotive-burner simulator did not perform as expected. A redesigned combustion chamber is required. Computer models were then tested against a limited set of data obtained from a GE-Dash-9 locomotive retrofitted with two high-pressure heat exchangers (HiPHEXs). With this limited data set and under model assumptions, the HiPHEX effectiveness tends to decrease for increasing notch numbers. In fact, as the total mass of exhaust gases flowing through the two heat exchangers increases beyond a certain threshold, the total surface area of only two HiPHEXs becomes insufficient to effectively transfer energy from the exhaust gases to the working fluid, thus extrapolation of data becomes increasingly inaccurate with increasing notch numbers.

Overall, the analyses based on limited actual locomotive test data show that the basic HiPHEX design performs reasonably well at all notch numbers (~50%) with attractive economic and pollutant reduction benefits. However, additional experimental measurements are necessary to further validate the results and aid the design toward HiPHEXs configurations with higher effectiveness (i.e. extending heat exchangers surface area within the constraints of executing non-invasive locomotive retrofitting).

All Stage I and Stage II tasks have been executed in agreement with TRANSIT-67 Work Subject.

4.1. FUTURE WORK AND RECOMMENDATIONS

The first generator locomotive exhaust gas simulator under performed mainly due to an under-dimensioned combustion chamber and low controllability of the mass flow rate of the exhaust gases and water circulating in the closed loop thermodynamic cycle.

A first next step, involves the design and manufacture of a more advanced simulator, comprising an expanded combustion chamber, thus allowing enough volume for the air and fuel to mix and fully burn. The expanded combustion chamber would allow parameters such as fuel flow rate and AFR to be adjustable in real-time and over a wide range, thus utilize the simulator to closely tailor data and scale results to a desired duty cycle while simulating locomotive engines of various power ratings. The advanced simulator would also provide a testing tool allowing investigations over full locomotive operational spectrums, thus increase data accuracy and enable validation of the computer models within large variations of thermodynamic conditions.

4.2. INVESTIGATOR PROFILE

Dr. Claudio Filippone, the project investigator, cooperated with TTCI technical staff, railroad consultants, and locomotive owners to access locomotive data required for the purposes of TRANSIT-67. Dr. Filippone is the inventor of several granted patents and patent applications covering waste heat recovery systems, solar-driven displacement of water without electricity or moving parts, spent nuclear fuel passive cooling systems, bio-electric fermenters to support bio-fuel production, high-power lasers, water recovery from exhaust gases, small-scale hybrid engines, and truly transportable compact nuclear generators powered by gas-cooled melt-resistant cores. Dr. Filippone experience with

combustion engines goes back to the 1980s. In 1986, as an electrical engineer, he patented and became involved with the development and testing of hot plasma high-voltage high-frequency ignition systems dedicated to advanced lean-burn stratified engines. The results were published by the Energy Department of the Polytechnic of Turin, Italy and were also tested by Ferrari Race Team in preparation of the 1986 Grand Prix of Brazil. His work demonstrated that the air-fuel ratio can significantly be extended toward ultra-lean condition without inducing misfiring. He also demonstrated that a plasma flame properly controlled within the combustion chamber reduces unburned hydrocarbon emissions by 60%, while operating at extremely low NO_x and CO conditions. From 1999 to 2006 Dr. Filippone worked at the Space Systems Laboratory (SSL) and as a faculty member at the Department of Aerospace Engineering at the University of Maryland where he taught Advanced Space Propulsion Systems II. In 2000 he received the “EB1” Extraordinary Ability green card, and later became a US citizen. He is the developer of CAESAR (Clean And Environmentally Safe Advanced Reactor), a nuclear reactor system with innovative heat transfer mechanisms able to significantly increase nuclear fuel burn up. Since 2010 Dr. Filippone manages ThermaDynamics Rail LLC engineering teams formed by Mechanical, Combustion, Electrical, Turbo-machinists, Manufacturing and Aerospace engineers and cooperates with international engineering firms and universities dedicated to the design, manufacturing and testing of complex thermal-hydraulic and high-speed electrical systems. Over several years as a researcher on thermal-hydraulic systems he developed expertise in conventional and advanced energy producing systems whose power source is represented by renewable energy, fossil fuels, generation III+ nuclear power systems, and advanced small modular reactors. Dr. Filippone is fully dedicated to furthering waste heat recovery technologies.

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