Final Report

Modifications and Optimization of the Organic Rankine Cycle to Improve the Recovery of Waste Heat

Donna Post Guillen
Jalal Zia

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INL
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Jalal Zia, General Electric Co.

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Idaho National Laboratory
Idaho Falls, Idaho 83415
http://www.inl.gov

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EXECUTIVE SUMMARY

This research and development project exemplifies a shared public/private commitment to advance the development of energy efficient industrial technologies that will reduce the U.S. dependence upon foreign oil, provide energy savings, and reduce greenhouse gas emissions. This project developed and demonstrated a Direct Evaporator for the Organic Rankine Cycle (ORC) for the conversion of waste heat from gas turbine exhaust to electricity. In conventional ORCs, the heat from the exhaust stream is transferred indirectly to a hydrocarbon-based working fluid by means of an intermediate thermal oil loop. The Direct Evaporator accomplishes preheating, evaporation and superheating of the working fluid by a heat exchanger placed within the exhaust gas stream. Direct Evaporation is simpler and up to 15% less expensive than conventional ORCs, since the secondary oil loop and associated equipment can be eliminated. However, in the past, Direct Evaporation has been avoided due to technical challenges imposed by decomposition and flammability of the working fluid. The purpose of this project was to retire key risks and overcome the technical barriers to implementing an ORC with Direct Evaporation.

R&D was conducted through a partnership between Idaho National Laboratory (INL) and General Electric (GE) Global Research Center (GRC). The project consisted of four research tasks: (1) Detailed Design & Modeling of the ORC Direct Evaporator, (2) Design and Construction of Partial Prototype Direct Evaporator Test Facility, (3) Working Fluid Decomposition Chemical Analyses, and (4) Prototype Evaluation. Issues pertinent to the selection of an ORC working fluid, along with thermodynamic and design considerations of the direct evaporator, were identified. The failure modes and effects analysis (FMEA) and hazards and operability analysis (HazOp) safety studies performed to mitigate risks are described, followed by a discussion of the flammability analysis of the direct evaporator. A testbed was constructed and the prototype demonstrated at the GE GRC-Niskayuna facility.

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ACRONYMS

CFD	Computational Fluid Dynamics
DOE	Department of Energy
EES	Engineering Equation Solver
ESD	Emergency Shut-Down
FMEA	Failure Modes and Effects Analysis
GE	General Electric
GRC	Global Research Center
GT	Gas Turbine
HazOp	Hazards and Operability Analysis
HX	Heat Exchanger
IDT	Ignition Delay Time
INL	Idaho National Laboratory
LES	Large Eddy Simulation
LMTD	Log Mean Temperature Difference
ORC	Organic Rankine Cycle
R&D	Research and Development
TEG	Turbine Exhaust Gas

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1. INTRODUCTION

1.1 Project Need

In the manufacturing sector, waste heat is one of the most abundant sources of energy, as it accounts for more than 2000 T Btu/year of wasted energy. Refineries, chemical processing plants, concrete plants, iron smelters, and a vast array of other industrial processes produce copious amounts of waste heat. If even a fraction of this waste heat could be economically converted to useful electricity, it would have a tangible and very positive impact on energy consumption and carbon emissions in the United States manufacturing sector.

The Department of Energy (DOE) has set the goal of reducing the energy intensity of the domestic manufacturing industry by 25% over the next 10 years. The combination of several technologies will be required to meet this aggressive goal, but one of the single largest opportunities lies in the waste heat rejected from energy-intensive manufacturing processes, which exceeds 2 quadrillion Btu per year (69 GW). Current waste heat recovery technologies, including Organic Rankine Cycles (ORCs) and thermoelectrics, are technically feasible but economically unattractive; thus, limiting their current use to a small number of niche applications. Idaho National Laboratory (INL) and General Electric (GE) Global Research Center (GRC) have partnered to leverage previous research in advanced ORCs and develop a new Direct Evaporator technological solution to reduce the ORC cost up to 15%, enabling the rapid adoption of ORCs in the manufacturing sector. GE has the exceptional ability to develop and commercialize a high-quality, efficient ORC product. If within 10 years the manufacturing sector were to convert just 10% of the available waste heat into useful electricity with this technology, the manufacturing sector would save over 1 GW of electricity per year and reduce CO_2 emissions by over 6,000,000 tons annually. Figure 1 provides an overview of the project.
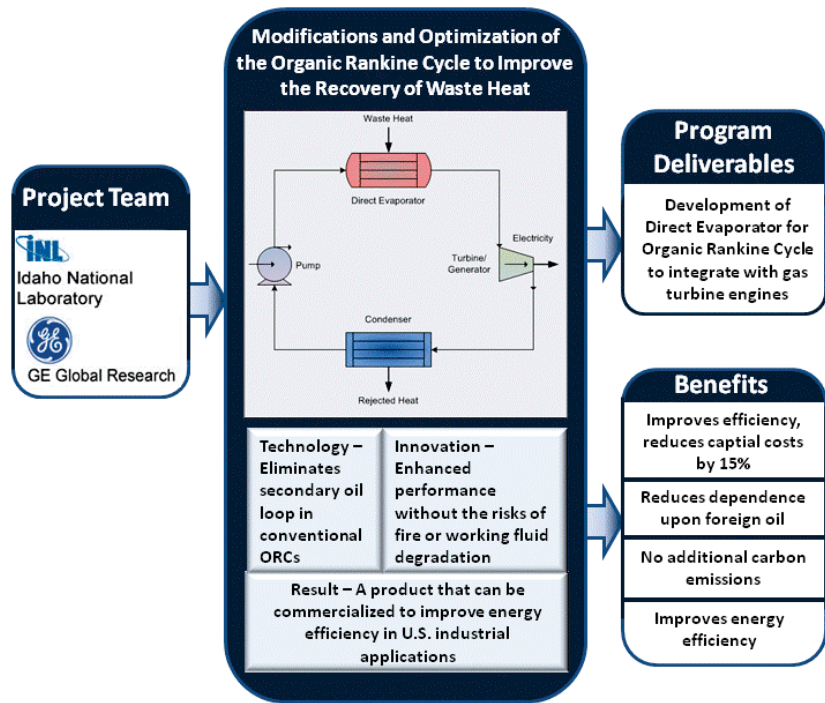
Figure 1. Overview of program team, concept, and benefits.

Waste heat is nearly always born in a low-pressure gaseous stream, such as the exhaust gas from a turbine
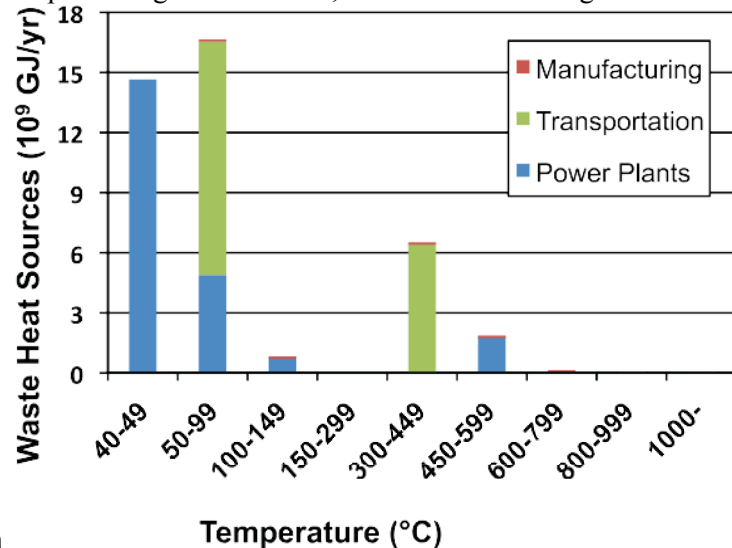
or a reciprocating engine. As seen in

Figure 2, the vast majority of this waste heat occurs in heat sources that are thermodynamically very "cold," typically in the 100 to 500°C range.

GE manufactures a wide range of power-producing engines that are used in various industries, particularly the petroleum industry. GE's gas turbines and reciprocating engines have state-of-the-art efficiencies, yet they still typically produce more waste heat than electricity, see Figure 3. Both make more waste heat than electricity, and reject all of the waste heat at temperatures well below 500°C. A technology that is viable for GE's gas turbines and reciprocating engines can also be used for other sources of manufacturing waste heat.

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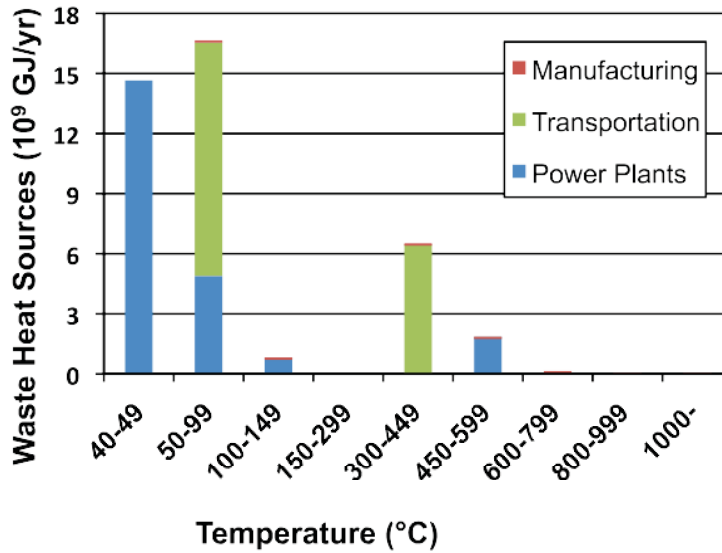
Figure 2. The distribution of waste heat as a function of temperature.[a]
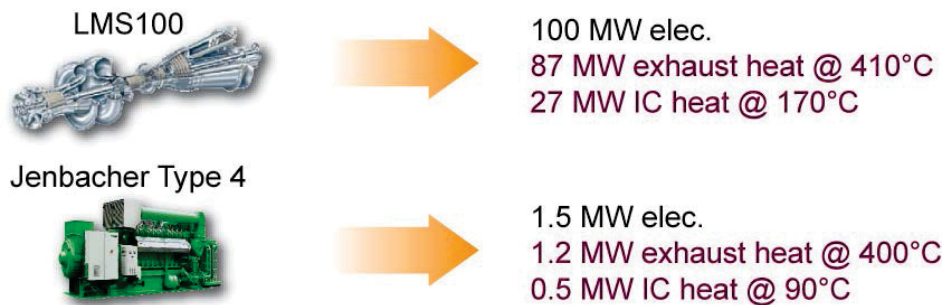
Figure 3. Examples of a GE Oil and Gas–Gas Turbine, and a GE Jenbacher reciprocating engine.

The best use of waste heat streams is when a manufacturing plant requires a source of relatively low-temperature process heat. However, most plants have already integrated this form of waste heat utilization when low-temperature process heat is required. Therefore, the next most useful and flexible outcome is to convert the waste heat into electricity. There are several technologies available for converting waste heat to electricity at these low temperatures, as shown in Table 1. Steam Rankine cycles are by far the most common form of waste heat recovery, as massive units are installed in combined cycle power plants to capture the waste heat from large frame gas turbines. But in these plants, the gas turbine exhaust temperature exceeds 600°C. At temperatures below 500°C, and especially in sizes much less than 100 MW, steam Rankine cycles become uneconomical. Technologies such as thermoelectrics and thermo-tunneling devices promise the elegance of the direct conversion of heat to electricity through a solid-state material. But these technologies currently have low-conversion efficiencies and are presently too expensive for serious waste heat recovery. The most viable technology for converting waste heat to electricity in the next decade is the ORC.

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[a] Rattner, A.S., and Garimella, S., Energy harvesting, reuse and upgrade to reduce primary energy usage in the USA, Energy 36 (10), 2011.

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Table 1. Waste heat recovery technologies.

	Rankine Cycles		Thermoelectrics
	Organic	Steam	
Turbo-expander Size	Small turbine	Large turbine at temperatures <500°C	No turbine.
Performance	Efficient at source temperature less than 500°C	Efficient at source temperatures above 500°C.	Current technology has lower efficiencies than Rankine cycles.
Maturity	Field-proven in waste heat recovery and geothermal applications.	Field proven in large combined cycle applications.	Field proven in remote (e.g., pipelines) and harsh (e.g., space) applications.
Cost	Barely cost effective at low source temperature less than 500°C.	Cost effective at temperature above 500°C.	Current technology is expensive on a per kilowatt basis.
Temperature Limitations	Organic fluids must be kept below 320°C for thermal stability.	Maximum temperature dictated by metallic components, not fluid.	Off-the-shelf materials work at <250°C. More exotic compounds work at 1000°C.
Technical Challenge	Organic fluid stability and flammability require secondary heat transfer loop	A compact and efficient small-scale steam turbine.	New material breakthrough.

1.2   Project Goals and Objectives

The purpose of this project is to develop and demonstrate a Direct Evaporator for an ORC system. The focus of this effort is land-based gas turbines, which are used in a broad range of applications to produce both shaft and electrical power. Most commonly known for generating electricity either as peaking units or as base load units, they are also used to directly drive pumps, compressors, or other machinery requiring shaft power. Simple cycle gas turbines have the advantage of a short startup time relative to coal-fired and nuclear units. However, they incur a significant penalty on their efficiency. Large-frame gas turbines usually are combined with bottoming steam-based Rankine cycles to increase the overall efficiency of the system and thereby improve their cost performance. Small-frame gas turbines with exhaust temperatures around 500°C could, in principle, benefit from steam bottoming cycles, but rarely use them in practice because of the high capital cost of the steam system. Particularly for base-load small-frame gas turbines, akin to those used in pipeline applications, an increase in efficiency is highly desirable.

This technology improves cycle efficiency and cost by eliminating the usual secondary heat transfer loop. The project's technical objective is to eliminate the pumps, heat exchangers, and all other added cost and complexity of the secondary loop by developing an evaporator that resides in the waste heat stream, yet virtually eliminates the risk of a working fluid leakage into the gaseous exhaust stream. The research team comprising INL and GE engineers has leveraged previous research in advanced ORC technology to develop a new direct evaporator design that will reduce the ORC system cost by up to 15%, enabling the rapid adoption of ORCs for waste heat recovery.

Currently available technology employs an intermediate heat transfer loop between the hot exhaust stream and the ORC evaporator. This precaution regulates the maximum temperature that the working fluid can reach and prevents accidental leaks of fluid into the exhaust stream. This over-conservative

design prevents excessive degradation of the working fluid and eliminates the risk of fire by auto-ignition of the working fluid. Nevertheless, a direct exchange of heat, with no intermediate loop, can be implemented if adequate precautions are taken to avoid fires and working fluid degradation due to overheating. GE GRC in Munich, Germany has developed a concept that enables the direct heating of a working fluid without increased risk of degradation or fire.

1.3   Technical Description

The ORC is a vapor power cycle that operates using the same principles as the steam Rankine cycle, except that a fluid with a lower boiling point (and higher molecular mass) is used. An organic working fluid is evaporated, instead of boiling water to create steam that runs through a turbine to generate electricity. Since the cycle operates at much lower temperatures than a steam Rankine cycle, the ORC can utilize the energy from low temperature waste heat sources to produce electricity. ORCs are a viable option to recover the exhaust waste heat using the ambient air as a heat sink.

The Rankine cycle is comprised of four main components: evaporator or boiler, turbine or expander, condenser, and pump. Depending on the working fluid, a recuperator may be advantageous, depending on the residual enthalpy of the fluid as it exits the expander. Typically, the heat of the exhaust stream is transferred indirectly to the ORC by means of an intermediate thermal oil loop. The Direct Evaporator eliminates the need for an oil loop by transferring heat to the working fluid with just one heat exchanger unit placed directly in the exhaust gas stream. Figure 4 compares the indirect versus direct evaporation arrangement. Figure 5 shows a layout of the Direct Evaporator ORC system mated to a PGT25 gas turbine engine.

(a)   (b)
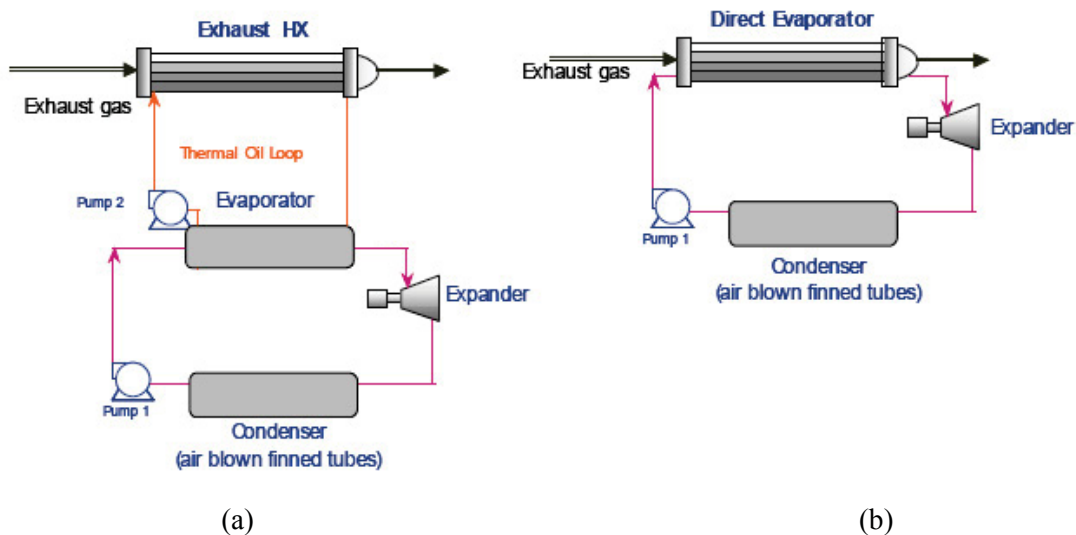
Figure 4. ORC with (a) indirect evaporation versus (b) direct evaporation.
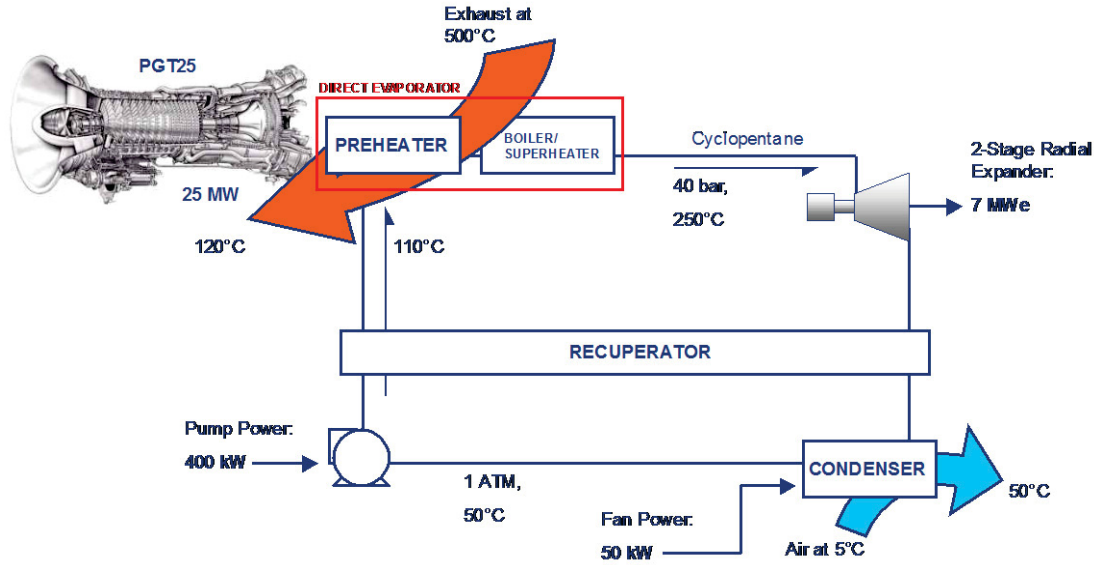
Figure 5. Typical layout for an ORC in the role of a bottoming cycle for a gas turbine.

The efficiency of an ORC depends both on the initial temperature of the waste heat and the level of irreversibility introduced as that heat is transferred to the cycle fluid, then from the cycle fluid to the sink. Figure 6 compares the maximum attainable Carnot efficiency with that of an endoreversible process. The endoreversible process is a much more accurate measure of heat engine efficiency in that the two processes of heat transfer are not treated as reversible [1]. An ambient air temperature of 18°C is used in the calculation. The Turbine Exhaust Gas (TEG) is available at temperatures between 400 and 550°C. However, the temperature to which the working fluid may be heated is limited by the chemical stability of the fluid. Although, in practice typical ORC efficiencies are around 10 to 20%, by integrating an ORC with a gas turbine (GT) engine, total system efficiency can be increased by roughly 20% to 30%. Real-world GT efficiencies in the 25 MW power range of interest are between 35% and 40%, with 60 to 65% of fuel energy wasted as heat. If the ORC can harness 10% to 20% of the wasted heat energy (i.e., of the 65%), the total system efficiency increases to ~45%.
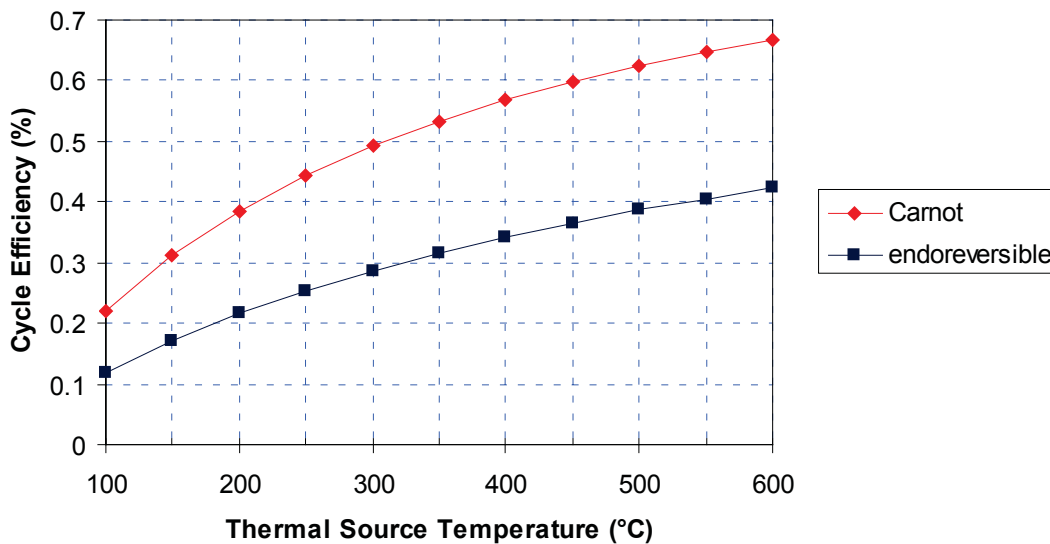
Figure 6. Comparison of Carnot and endoreversible cycle efficiencies.

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The losses and productive output of the cycle can be represented graphically in an exergy diagram, in which the useful power output may be compared against the theoretical entitlement (shown in Figure 7 for an ORC heated by the exhaust stream from a GE PGT-25 gas turbine). A cycle that minimizes exergy destruction is sought.
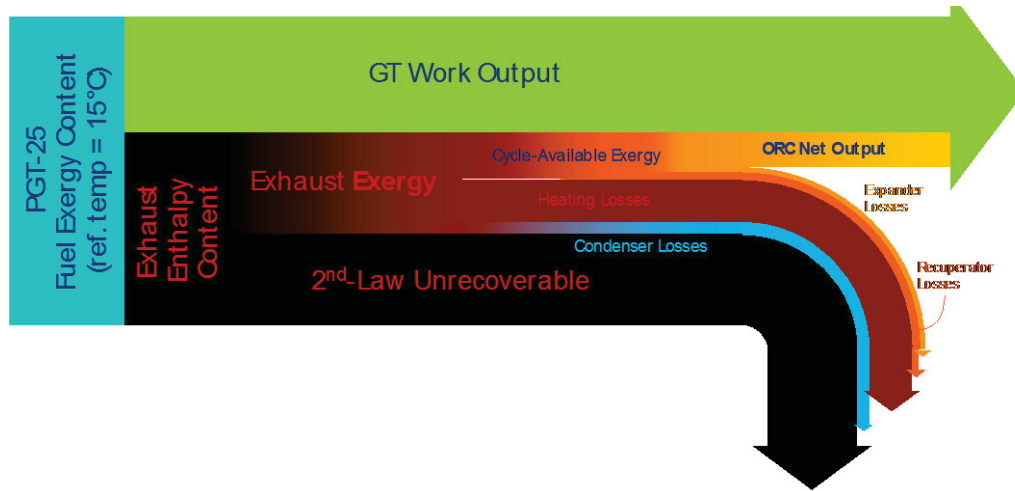
Figure 7. Exergy diagram for a PGT-25 gas turbine with ORC bottoming cycle.

1.4   Research Tasks

The project consisted of four research tasks: (1) Detailed Design and Modeling of the ORC Direct Evaporator, (2) Design and Construction of Partial Prototype Direct Evaporator Test Facility, (3) Working Fluid Decomposition Chemical Analyses, and (4) Prototype Evaluation.

Task 1: Detailed Design and Modeling of Direct Evaporator

Task 1 involved the detailed design and modeling of the Direct Evaporator. GE performed the detailed hardware design and INL completed analyses and modeling. The GE GRC-Munich research team oversaw the detailed hardware design and were responsible for the overall design of the evaporator, architecture, sizing, materials, etc. Analyses and testing were performed to assess the flammability of the working fluid in the hot exhaust gas stream stemming from a potential leak in the evaporator. There were three milestone reports delivered as part of this task:

Milestone Report 1 − Design and Modeling of Direct Evaporator

Milestone Report 2 − Direct Evaporator Leak and Flammability Concerns

Milestone Report 3 – Direct Evaporator Failure Modes Effects Analysis (FMEA) and Hazards and Operability Analysis (HazOp) Study

Task 2: Design and Construction of Partial Prototype Evaporator Test Facility

To ensure evaporator performance, GE conducted full-temperature evaporation tests on a prototype section of the evaporator. This task consisted of conceptual design, design and build, and shakedown testing of the test facility.

Task 3: Working Fluid Decomposition Chemical Analyses

To ensure optimal performance and long-term reliability of the ORC, decomposition of the working fluid was quantified. This task, performed by INL, identified reaction pathways and decomposition products. A milestone report, Milestone Report 4, "Working Fluid Decomposition," was delivered as part of this task.

Task 4: First Prototype Direct Evaporator Evaluation

GE GRC-Niskayuna commissioned a heat exchanger vendor to manufacture a Direct Evaporator prototype. This task culminated in a test of the prototype Direct Evaporator. The results of the test were documented in Milestone Report 5, "Test Bed and Fouling Tests."

2.   RESULTS AND DISCUSSION

2.1   Working Fluid Selection

2.1.1   Performance

ORC systems offer a wide range of parameters for optimization, with the most obvious being the selection of working fluid. The working fluid selection dictates the operating pressures on the condenser and evaporator side, expander design, need for a recuperator, etc. The operating pressures are strongly dependent upon the available heat source and sink temperatures. For different combinations of heat source temperature range and heat sink temperature range, there would likely be a different optimal fluid. To simplify the optimization process, the initial down-selection process focuses on cycle performance, with other considerations introduced later in the process.

The selection of the optimal working fluid is the result of a systematic comparison of over 40 different fluids on the basis of their suitability for use in an ORC cycle. Fluids were compared on the basis of chemical stability, flammability, toxicity, performance under the boundary conditions of the gas turbine exhaust application, and environmental risk in the event of a leak. Other considerations that factor into working fluid selection include corrosiveness and tendency to foul. When selecting a fluid, cost was not considered a factor since the pressure level, component selection, operating temperature, and other attributes—independent from fluid choice—most influence cost. The candidate fluids fall under five broad chemical groups:

1. Simple aliphatic hydrocarbons, such as butane, pentane, and hexane, are attractive because their near-ambient boiling points enable condensation near atmospheric pressure

2. Fluorinated (or otherwise halogenated) hydrocarbons (including most refrigerants), are attractive because of their efficient expansion behavior and lack of need for a recuperator

3. Aldehydes and ketones are variations on simple hydrocarbons that can be chosen so as to combine the benefits of hydrocarbons and refrigerants

4. Silicones with extremely high chemical stability at elevated temperatures to guarantee continued performance over the lifetime of the machinery

5. Aromatic hydrocarbons that combine high stability with good expansion properties, but generally boiling well above ambient temperatures.

Perfluorocarbons, chlorofluorocarbons, and hydrochlorofluorocarbons have very attractive properties for the ORC, but unfortunately have an extremely high greenhouse warming potential, and therefore were not considered. The down selection of working fluids was based primarily on performance in an ORC subject to the constraints identified above. Further selection was guided by consideration of the stability of the chemical at high temperature, health hazards, and potential to cause environmental harm. Important characteristics that govern working fluid performance are as follows:

- A high stability and critical point such that the fluid may be boiled at a relatively high temperature, allowing the recovery as work of a relatively high fraction of the embodied heat energy (enthalpy) of the fluid. The use of fluid blends versus a single fluid presents concerns over unmixing.

- Vertical to positive slope of the vapor curve on T-s diagram to eliminate need for superheating, increase efficiency, and lower condenser cost. A tendency of the expanding vapor is to remain close to saturation, without need for superheating. If the vapor is close to saturation as it is discharged from the turbine, its temperature will not differ greatly from the condensation temperature, and irreversible transfers of heat (within a recuperator) from the vapor to the cooler liquid returning from the pump will not be required. Any such irreversibility decreases cycle efficiency.

- Sufficiently high volatility to boil at or above ambient temperature, meaning that the condenser can be operated at or above atmospheric pressure. A lower-than-atmospheric (i.e., vacuum) condenser is undesirable, since such systems incur additional cost and complexity to prevent in-leakage of ambient air. Fluids with low vapor pressures at ambient temperatures require the use of subatmospheric condensers or costly, cascaded cycles. Condensers that operate at pressures below atmospheric are unacceptable in ORCs because the ingress of air and moisture through unavoidable minute leaks catalyzes degradation reactions in the working fluid [2].

Other desirable characteristics of the working fluid include:

- High thermal conductivity in the vapor phase to maximize heat transfer

- High autoignition temperature, preferably above TEG temperature

- High specific heat ratio

- Low environmental impact and toxicity

- Low overall system pressure to reduce component cost

- Minimal reactivity with air or materials of construction

- Low flammability rating and transport hazard class

- Low freezing point, as this affects operability in cold climates.

On the basis of these criteria, cyclopentane was chosen as the best match to system requirements. Table 2 shows the comparative scores of the leading fluid selections from each category from 1 to 5, with 1 being best. Some selection criteria may, in practice, have a stronger weight than others, so the additive scoring used in the table must be regarded as approximate. Overall, cyclopentane shows the best alignment with the selection priorities, with thiophene and acetone close in second place. Cyclopentane (C_5H_{10}) is an alicyclic hydrocarbon derived from petroleum. Its saturation temperature is slightly above typical ambient air temperatures at atmospheric pressure, and it has a relatively high critical temperature and pressure compared with linear (n-) and iso-pentane. Among the working fluids tested, the halogenous refrigerants best displayed the desired tendency to approach the saturated condition during expansion. Unfortunately, the chemical fragility of these compounds, along with their tendency to form corrosive hydrofluoric acid in the process of breaking down, limits the range of their operating temperatures. Among the refrigerants, methyl iodide had the best thermodynamic performance, but was excluded due to its high toxicity. Among the aldehydes, acetaldehyde performed best thermodynamically, although low stability and extremely high flammability make it impractical for an ORC product. The last group, silicones, tended to perform poorly as a category due to the relatively large amount of heat that the recuperator must remove from the expander discharge vapor of these fluids. The recuperator is a significant source of irreversibility in a silicone-based ORC.

Among simple aliphatic hydrocarbons, cyclopentane produced the best performance, as well as a relatively low flammability compared with the non-cyclic pentane isomers commonly used in ORC cycles. Also, like other simple hydrocarbons, cyclopentane displays a tendency to superheat during the expansion of the vapor, but along with the other stable cyclic aliphatic hydrocarbon, cyclohexane, the tendency is much less than in the case of standard, straight-chain hydrocarbons with comparable

molecular weight such as n-pentane and n-hexane. Cyclopentane also has relatively high stability at temperatures between 250 and 300°C.

If, for any reason, cyclopentane cannot be used, acetone or thiophene would be adequate replacements with marginally lower performance. Use of the ketone acetone would result in a performance drop of nearly 10%, but otherwise shares many of the virtues of cyclopentane, in addition to a substantially higher auto ignition temperature that may relieve the necessity for certain fire-prevention strategies in the design. By contrast, aromatic hydrocarbons performed exceptionally well as a group, although among them only thiophene and toluene had sufficiently low toxicity to be practical as cycle fluids, with toluene's low volatility giving it suboptimal high-condensation temperature at 1 bar. Thiophene, like cyclopentane, exhibits high-temperature stability and a weak tendency to superheat during vapor expansion.

Table 2. Ranking of leading fluid selections for use in the Direct Evaporator ORC system.

| | Methyl Iodide | Acetaldehyde | Thiophene | Cyclopentane | Acetone |
	Halogenous Hydrocarbon	Aldehyde	Aromatic Hydrocarbon	Aliphatic Hydrocarbon	Ketone
Performance	1	1	3	2	4
MSDS Flammability	2	4	3	3	2
Stability	2	4	1	2	2
MSDS Toxicity	5	2	2	1	1
TOTAL	10	11	9	8	9

2.1.2   Thermal Stability

The hydrocarbon working fluid is susceptible to decomposition at high temperatures. For increased efficiency, it is desired to operate the cycle with the working fluid at the highest permissible temperature. However, excessive chemical decomposition of the working fluid can decrease performance and lifetime of the fluid, which increases maintenance and operating costs. Therefore, laboratory experiments were performed to quantify the decomposition rate of the cyclopentane at various temperatures.

The decomposition of research grade cyclopentane was measured in a recirculation loop (Figure 8) at temperatures of 240, 300, and 350°C and a pressure of 43 bars in a heated glass-lined tube. Complete details of the experiment are given by Ginosar et al.'s 2011 journal article. In the absence of air at the lower-two temperatures, decomposition was minor after 10 days of continuous operation. The total cyclopentane decomposition products were approximately 68 and 220 parts per million by weight (ppm) at 240 and 300°C at the end of 10 days. At 350°C, decomposition products were significantly higher and reached 1,500 ppm. Figures 9 through 11 show the total weight of the decomposition products as a function of time on stream. Tables 3 through 5 list the end of run decomposition products.

When the feed was saturated with air, the decomposition rate increased dramatically, as seen in Figure 12. Residues found in the reactor after the decomposition experiments (Figure 13) were examined by a number of different techniques. The mass of the residues increased with experimental temperature, but was lower at the same temperature when the feed was saturated with air. Analysis of the residues suggested that the residues were primarily heavy saturated hydrocarbons.
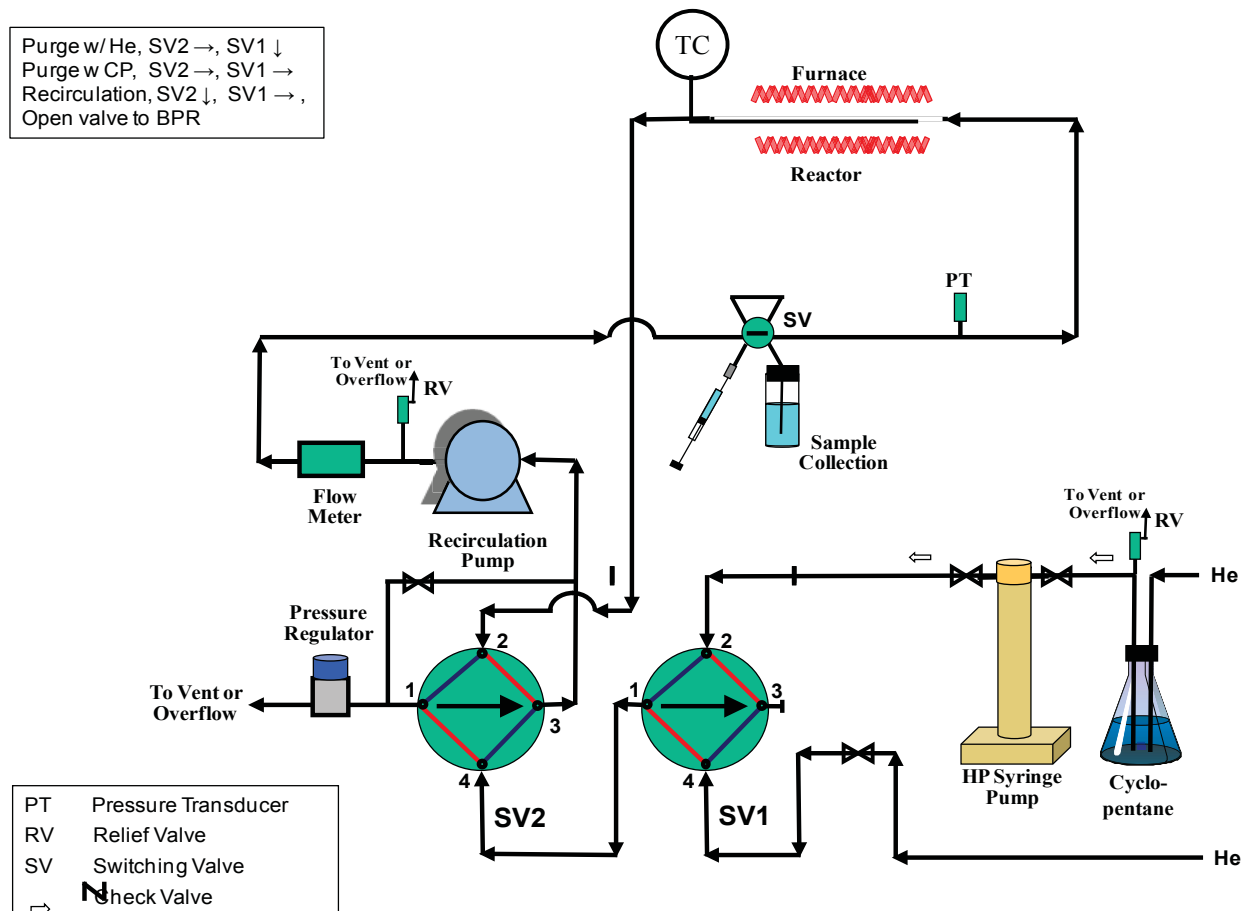
Figure 8. Schematic of the organic fluid decomposition test system.
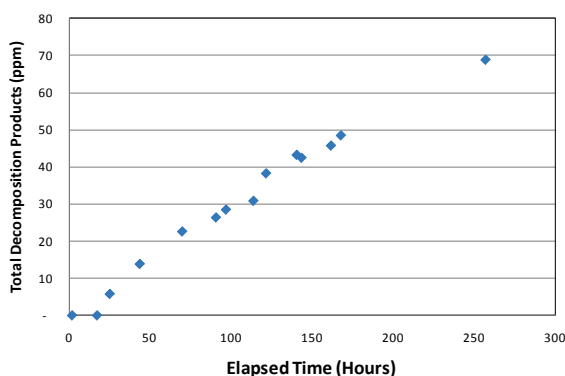
Figure 9. Total decomposition products as a function of time on stream (240°C, 43 bars).

Compound	Concentration (ppm)
Propylene	2
Butanes	14
Pentene	16
Cyclopentene	20
Hexanes/Hexenes	9
Heptane/Heptene	4
Octane/Octene	5

Table 3. End of run compounds and concentrations (240°C, 43 bars).
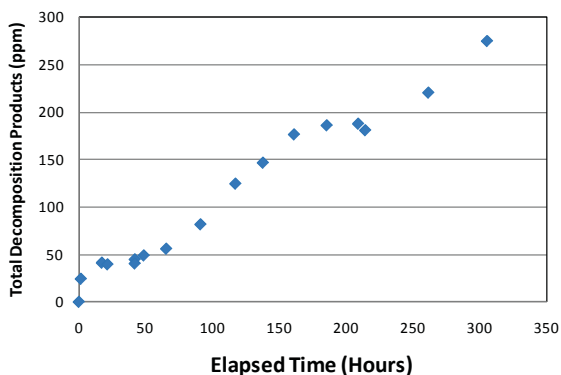
Figure 10. Total decomposition products as a function of time on stream (300°C, 43 bars).

Compound	Concentration (ppm)
Ethylene	1.3
Ethane	2.5
Propylene	21
Butene	5.2
Butanes	57
Pentene	8.8
Cyclopentene	41
Hexanes/Hexenes	27
Heptane/Heptene	53
Octane/Octene	11
Decanes/Decanes	45

Table 4. End of run compounds and concentrations (300°C, 43 bars).
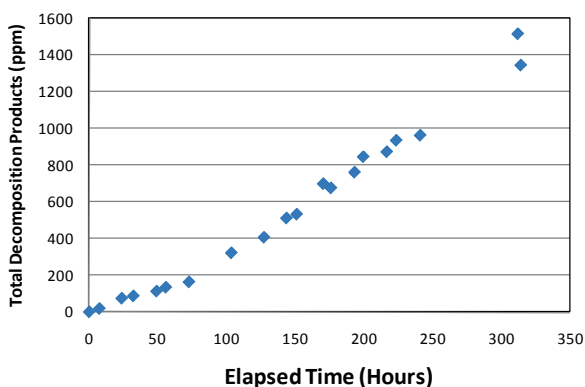
Figure 11. Total decomposition products as a function of time on stream (350°C, 43 bars).

Compound	Concentration (ppm)
Methane	4
Ethylene	39
Ethane	20
Propylene	202
Butenes	32
Butanes	67
Pentene	64
Pentanes	51
Cyclopentene	220
Hexanes/Hexenes	24
Heptanes/Heptenes	330
Octanes/Octenes	232
Decane/Decane	159
Dodecane/Dodecene	72

Table 5. End of run compounds and concentrations (350°C, 43 bars).
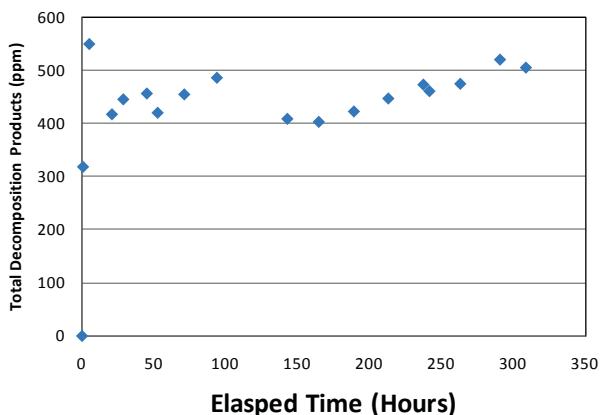
Figure 12. Total decomposition products as a function of time on stream for air-saturated feed. Experimental set points: 300°C, 43 bars.

Figure 13. Decomposition residues.

Cyclopentane should be a relatively stable compound to use as a working fluid in a direct evaporator as long as bulk fluid temperatures are kept below 300°C and air is excluded from the system. These experiments were performed in a laboratory setting using a glass-lined reactor and high-purity cyclopentane. The data help determine an upper limit for the ORC working fluid temperature under idealized conditions. Impurities, the presence of oxygen, and materials of construction can affect the fluid decomposition rates. Longer duration testing using industrial grade cyclopentane in a prototypic ORC loop is required to obtain a realistic estimate of the working fluid degradation in an operational environment.

2.2   Thermodynamic Analysis

The performance of a particular working fluid, even once the source and sink temperatures for the ORC have been specified, is not uniquely determined. A principal variable strongly affecting performance is the pressure at which the working fluid boils. For each fluid, given a particular initial heat source flow and temperature, the electrical output of the ORC will be maximized for a particular pressure level. Here, a single source and sink temperature are specified.

To perform a comparison of fluids, the following five criteria were imposed on the computer simulations of ORC systems:

1. Fixed initial heat source temperature

2. Fixed log-mean temperature difference (LMTD), rather than fixed minimum temperature difference (the distinction is explained below), in evaporators and condensers

3. Fixed expander technology and expander adiabatic efficiency

4. Fixed pump efficiency

5. Use of an additional fixed-LMTD heat exchanger (recuperator), if its inclusion would be beneficial in the particular case, to transfer heat from the fluid vapor as it is discharged from the expander to the fluid in the liquid phase as it returns from the pump.

By using the criterion of a fixed LMTD, rather than a fixed minimum temperature difference between the two flows in each heat exchanger, one possible source of variability between fluids is eliminated. Under given conditions of flow rate and flow inlet temperatures, the effectiveness of a heat exchanger is limited by the requirement that the temperature of the heated fluid may at no point exceed that of the cooled TEG. For this reason, strategies that guarantee more nearly parallel temperature profiles for the warmed and cooled fluid within the exchanger can permit a lower overall LMTD than would be possible if the temperature profiles of either flow were strongly "kinked," as when, at certain low pressures, the process of boiling at constant temperature absorbs roughly the same amount of heat as it took to steadily increase the temperature of the liquid phase from ambient level to the point of boiling. A lower overall LMTD implies lower irreversibility in the transfer of heat, and consequently a more efficient cycle. If the point of minimum approach (in a boiler, this generally occurs at the onset of boiling in the liquid) is the limiting factor in the design of the heat exchanger, strategies such as supercritical heating, or mixing two working fluids together to produce a binary fluid that boils at progressively increasing temperature, can alleviate the limitation and increase the cycle efficiency. But, if source temperatures are sufficiently high, and sink temperatures sufficiently low in relation to the cycle fluid temperature (conditions which hold for our own application), the point of minimum temperature approach will <u>not</u> limit the cycle performance, regardless of the strategy used (i.e., supercritical boiling, binary fluid mixtures). In this case, it is only the heat exchanger's size that controls its effect on the cycle performance, and the implied size changes roughly in proportion to the LMTD of the exchanger. To compare the different fluids on the basis of similar equipment size and cost, LMTD is constrained to be constant across all fluid simulations in order to eliminate it as a source of performance variation. Therefore, the difference in temperature between the two fluids at the point of closest temperature approach varies slightly between different fluid

trials. In practice, this "minimum ΔT" will not measure less than a certain value, so if in any case the chosen LMTD would have forced a minimum ΔT of less than 10°C, the LMTD was increased until a 10°C minimum ΔT was reached. A ΔT of 10°C at the pinch point is fairly typical for large industrial heat exchangers.

2.3   Direct Evaporator Design

A successful design of the direct evaporator must satisfy the required duty (i.e., the amount of heat to be transferred per unit time given the inlet temperatures and mass flows), and meet certain constrains specific to the working fluid and application. For extraction of heat from a low-pressure gas by a high-pressure fluid, finned-tube heat exchangers are employed because of their suitable characteristics of low-pressure loss on the gas side along with high-surface-area ratio between the fins, where the heat transfer coefficient is low, and the tube inside, where the heat transfer coefficients of the fluid are typically about two orders of magnitude larger, leading to a high overall heat transfer coefficient with a relatively compact volume.

Optimizing the heat exchanger design demands a compromise between size (i.e., capital-intensive heat exchange area), and tolerable pressure losses in each of the fluids streams. However, in this case specific constraints require a distinctive approach in regard to dimensions, geometry, and layout. The primary constraints imposed to the heat exchangers of the direct evaporator by the working fluid are:

- Limiting working fluid maximum temperature to avoid excessive working fluid degradation

- Ensuring safety in the event of working fluid leak

- Observing fin surface temperature lower limit

- Maintaining TEG temperature above dew point temperature for nitric acid formation (otherwise, cannot use carbon steel tubes)

- Limiting backpressure from the ORC to within allowable limits to avoid choking the GT.

The most severe design constraint is the upper limit imposed upon fluid temperature above which decomposition is accelerated. As the highest fluid temperature is found in the boundary layer of the fluid close to the wall of an externally heated duct, the inside wall temperature of all heat exchanger pipes must remain below this temperature limit at all times. The thermal stability of the fluid determines the lifetime of the working fluid, affecting life-cycle cost, and has safety implications if undesirable chemical decomposition products are generated.

The dehydrogenation reaction results in hydrogen evolution that, since hydrogen in non-condensable, dramatically reduces expansion pressure ratio, maximum output power and efficiency. Longer-chain hydrocarbons may form, which can leave a gummy or coke type of residue that is deleterious to system components (especially the pump and heat exchanger). Undesired reaction products, including non-condensables, should be periodically or continuously removed from the heat transfer loop. Avoiding oxygen ingress into some working fluid is critical, since experiments conducted at INL show that decomposition products (measured in solution) increase five-fold upon a bulk temperature increase from 300°C to 350°C, whereas solid product deposition is three times higher. Also, avoiding materials of construction or contaminants that contain catalysts that can promote working fluid degradation is recommended.

2.3.1   Strategies for Tube Wall Temperature Control

The inside tube wall temperature depends on the fluid bulk temperature, the heat transfer coefficient, and the heat flux. All three of these parameters are designed such that the inside wall temperature remains below 300°C.

To enhance the internal heat transfer coefficient, the velocity of the fluid inside the tube needs to be high, putting a limit on the number of parallel tube passes, at the same time necessitating a certain pressure loss to be tolerated even when the tube length is kept as short as the aspect ratio and manufacturing costs allow. Internal heat transfer coefficients are highest in the evaporator section, where a nucleate and convective boiling regime prevails. The lowest heat transfer coefficients are found in the economizer section. The working fluid bulk temperature design point at the heat exchanger outlet was set to 250°C to allow ample margin for the wall temperature to stay below 300°C. The highest fluid temperature, 230°C–250°C, is encountered in the superheater section of the boiler. Within the GE proprietary concept of "protective staging," the superheater section is placed in the exhaust gas stream behind the evaporator section (see Figure 14), where the exhaust gas is already cooled although still hot enough to ensure reasonable heat transfer performance in a counterflow layout. The evaporator section is placed first in the exhaust stream in a parallel flow arrangement, such that the highest exhaust gas temperature is met by a fluid bulk temperature at the boiling point, but at low vapor quality. In this regime the heat transfer coefficients are highest and can only increase with the heat flux. The danger of local wall dryout (that leads to sharp wall temperature spikes as the heat transfer coefficient collapses) is minimal because of the low vapor quality. As the vapor quality increases along the evaporator, the exhaust gas temperature and the heat flux decrease, by the time dryout is expected, the high-volume flow rate has increased the gas phase heat transfer coefficient while the exhaust gas temperature is low enough so that the wall temperature limit is not exceeded. The implementation of such a concept is schematically shown in Figure 15. A further measure to limit the heat flux and decrease the inner wall temperature where it can become critical is to reduce the number and area of the fins on the tube outside. This concept of "variable finning" of the tubes (see Figure 16) throughout the length of the heat exchanger includes smaller and less densely spaced fins in the superheater and evaporator section where the temperatures are high and the internal heat flux needs to be limited. Figures 16 (a) through (d) show four different types of fins (internal and external) that can be used in the different sections of the Direct Evaporator. However, in the economizer section where the temperatures are lower and the wall temperature is not critical, larger diameter and serrated fins with tighter fin spacing are used to improve heat transfer performance and reduce the number of rows necessary to achieve the required duty.
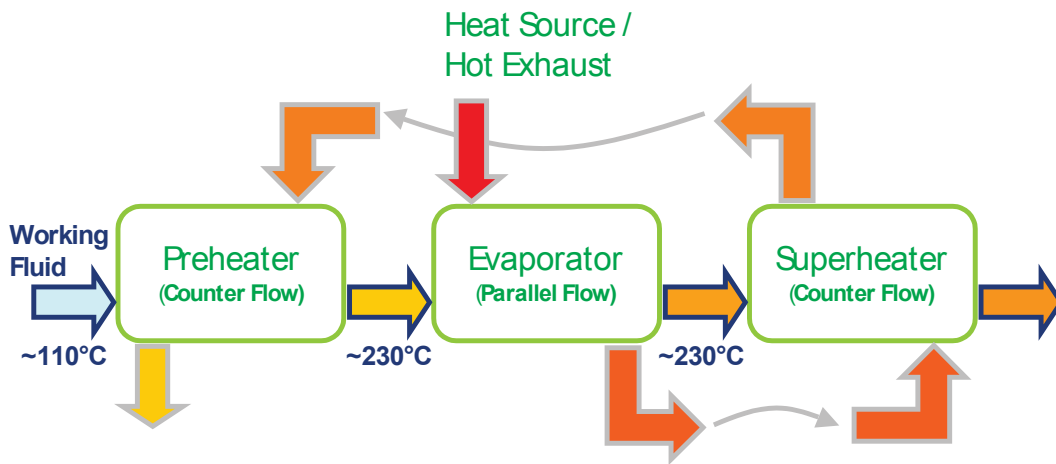

Figure 14. Concept of protective staging applied to a gas turbine exhaust stream.
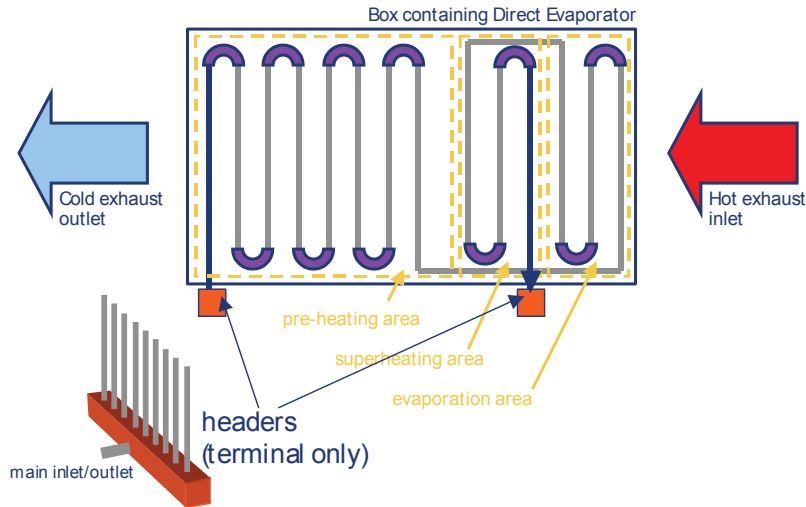
Figure 15. Implementation of protective staging (evaporator in parallel setup).
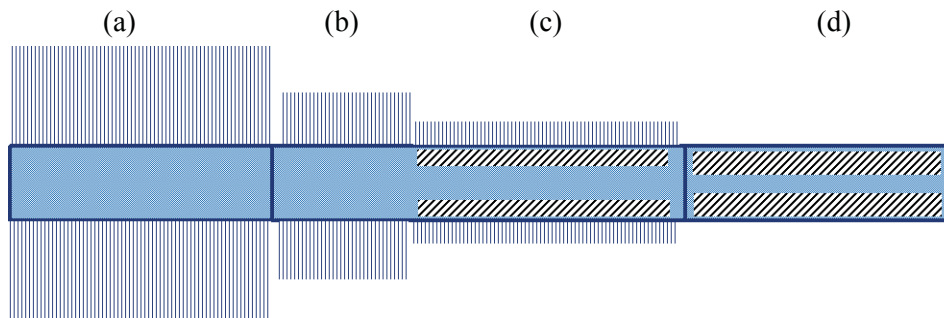
Figure 16. Variable finning of the tube and enhancements of the internal heat transfer surface to control heat flux and tube side film temperature.

2.3.2   Heat Exchanger Modeling Results

A detailed thermodynamic model was constructed using the Engineering Equation Solver (EES) software (F-Chart Software Inc.) to design and simulate the heat exchanger performance for a direct evaporator using the Protective Staging concept. It was concluded that the Zudkevitch-Joffe equation of state for cyclopentane provides the best match to the NIST property data for cyclopentane. The output of this model includes all heat flows, heat fluxes, and temperatures at the inlet and outlets of each section on the working fluid and the exhaust gas side. It contains detailed correlations for the pressure loss and heat transfer coefficient of the fin tubes measured in-house. The current preliminary design is based on these data. The outputs of this tool were compared to the outputs of the commercial heat exchanger design program, ASPEN ACOL Exchanger Design and Rating, and agreed very well (see Figure 17). The wall temperatures are predicted to stay well below the critical limit and pressure losses of the exhaust gas and the working fluid are less than the design limits as well. An off-design mass flow/pressure loss calculation carried out for each section of the boiler indicated stable flow behavior without the danger of oscillations characteristic for once-though steam boilers that are caused by maldistribution in parallel flow passes. Flow stability is attained when the gradient of pressure over mass flow is always positive throughout the range of operation, which was shown in the calculations.
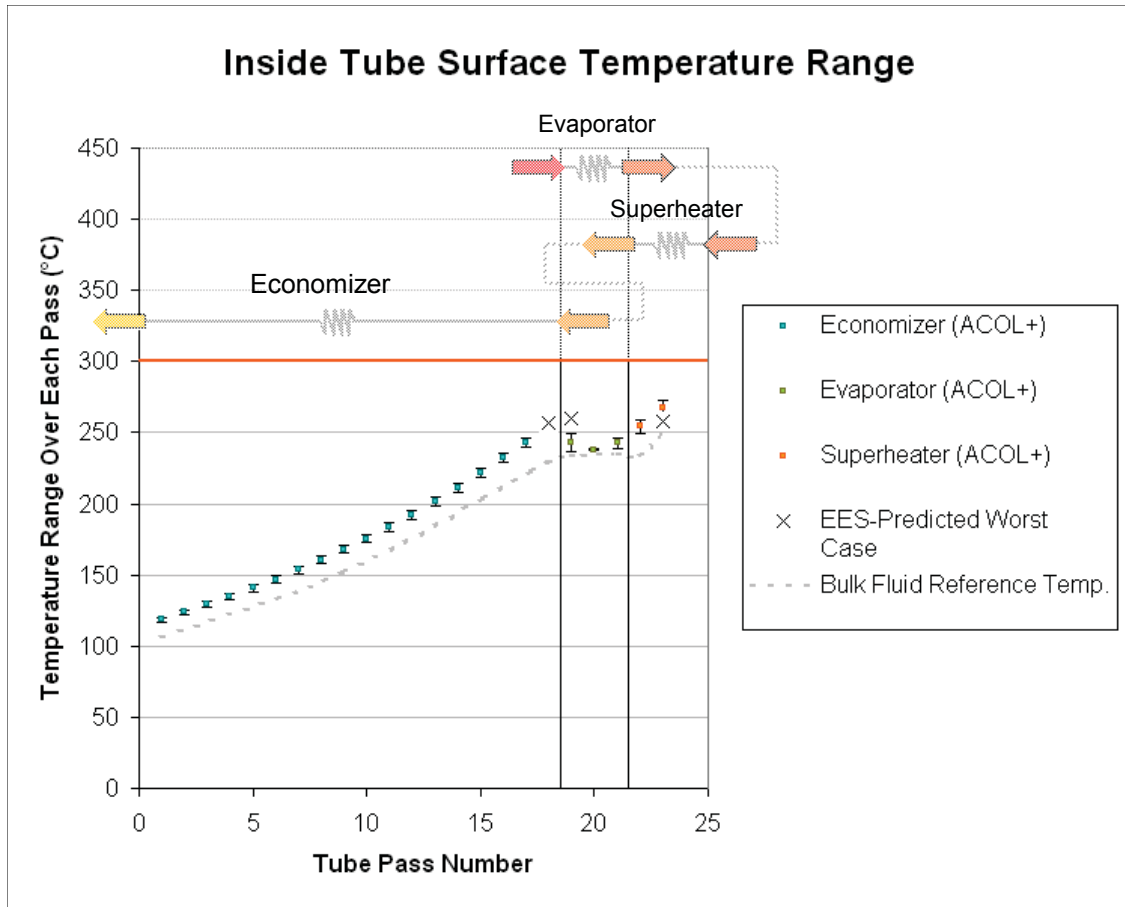
Figure 17. Working fluid side surface temperature in direct evaporators.

2.3.3　Heat Exchanger Design Summary

The preliminary design of the boiler includes three harps of fin tube bundles, economizer, superheater, and evaporator, suspended in a rectangular duct. The number of rows is 24, 2, and 2, respectively, to give a total of 28 with 60 tubes per row. The economizer section only has two rows per pass for lower pressure loss; economizer and superheater are in counterflow to the exhaust gas and the two evaporator rows are arranged in parallel flow. The headers are laid out as manifolds on top of the tube bundles, with each bundle having a once-through flow path. The headers are tentatively placed outside the duct to prevent having zones of hot, slow moving fluid near walls and headers with weld seams, in which in case of a leak the risk of autoignition could increase. A simple schematic of the direct evaporator design is shown in Figure 18.
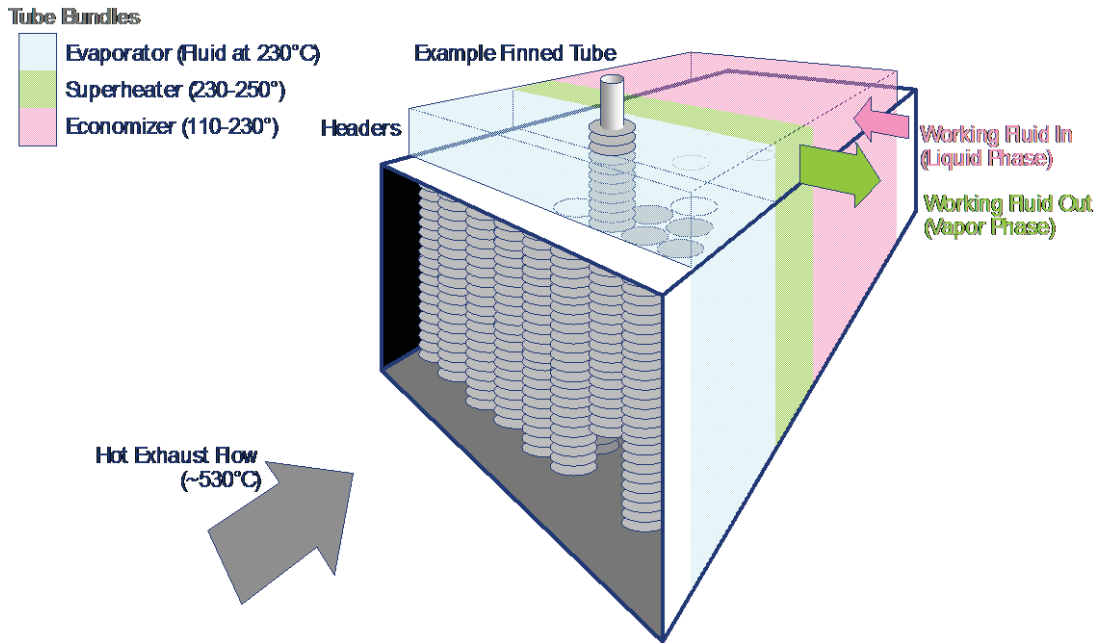
Figure 18. Direct evaporator tube bundle arrangement.

2.4 FMEA/HazOp Study

Placing a heat exchanger operating with a flammable hydrocarbon working fluid directly in the hot exhaust gas stream presents potential safety risks. To mitigate risks, FMEA and HazOp safety studies were performed. The most serious risks anticipated from heating the working fluid in direct proximity to a hot gas turbine exhaust were examined. Potential causes of tube breaches that would initiate a leak include thermal fatigue, mechanical vibration, corrosion, and manufacturing defects.

2.4.1 Thermal Fatigue

Following a cold start of the direct evaporator, metal temperatures increase over a range of hundreds of degrees Celsius. Differential thermal expansion in the various materials used in the construction of the evaporator can put large stresses on the material interfaces, especially on welded joints between the working fluid tubes and the frame. Over the life of the evaporator unit, repeated cycles of startup and shutdown can eventually aggravate small imperfections in the weld to open cracks through which working fluid under high-pressure escapes from the tube into the hot TEG. A large leak would be noticed immediately from the measurable loss of working fluid, while the smallest leak could persist for weeks or months before it is recognized and repaired.

2.4.2 Mechanical Vibration

The boiling process within the tubes, as well as the aerodynamic buffeting experienced by the tube banks during steady-state exhaust flow, contribute to vibration that can eventually fatigue and weaken the tube joints. Excessive strain of fatigued members could potentially open cracks in the tube material or welded joints, allowing a release of working fluid into the hot exhaust flow. These phenomena must be addressed during the design of the direct evaporator.

2.4.3 Corrosion

Although substantially depleted of oxygen, the residual oxygen content, as well as the water content, of the exhaust flow from the gas turbine have a non-negligible potential to corrode the carbon steel of the direct evaporator fluid tubes over time. The risk of corrosion is already significantly reduced by observing a minimum exhaust temperature to prevent so-called "acid gas" exhaust components from precipitating

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out of the gaseous phase and corroding metal surfaces. The concern arising from corrosion of the evaporator tubes is that it may ultimately open pinhole leaks in the tube wall, or simply weaken the wall sufficiently such that thermal or mechanical stresses could induce rupture. Routine inspection of all system components is recommended.

2.4.4 Manufacturing Defects

Defects in the piping material or welded seams, if not discovered through pressure tests during commissioning, remain as potential nucleation points for cracks throughout the lifetime of the evaporator.

Various more serious effects could result from leaks in any of the above scenarios, including ignition of the fluid causing hot spots and gradual weakening of the structure, as well as the possibility that leaks would feed larger cells of combustible gas, which could explode suddenly causing catastrophic failure. The estimated magnitude of these risks was necessarily quite provisional, as no research had yet been performed on the detailed mechanism for each of the failure mechanisms. However, reports on steam boiler technology provide examples of rupture mechanisms originating from corrosive interactions with the tube steel. For many boiler applications, leaks are described in the boiler literature as an inevitable symptom of aging.

2.4.5 Operational Safety

By identifying a safe means of detecting and handling leaks in general, all the particular leak scenarios, including corrosion, thermal, or mechanical strain failure, can be simultaneously addressed. A safety mechanism that could anticipate and "disarm" leaks, allowing no opportunity for ignition or explosion, would conclusively mitigate all conceivable leak scenarios at once. In line with this approach, subsequent analyses and experiments served to set safe limits on the range of velocities and temperature of hot that no amount of leaked working fluid could ignite. Specifically, even at exhaust temperatures as high as 600°C and flow rates as low as one-third of the normal operating level, any leaked fluid would be expelled from the system before it had the chance to ignite. A prototype is being constructed for testing at GE GRC's testbed located in Niskayuna, New York. During the prototype testing, leak tests are planned to confirm the retirement of fire risks, test detectors, and safety chain under real-life conditions.

The safety system (Figure 19) addresses possibility of a fluid leak into the hot exhaust flow for several different operating modes:

1. GT and ORC in normal, steady-state operation

2. GT operating normally, ORC in emergency shut-down (ESD), diverter switching from "open" to "bypass" condition

3. GT in ESD, ORC in ESD, diverter moving from "open" to "bypass"

4. GT operating normally, ORC in ESD, diverter fully in "bypass"

5. GT in ESD, ORC in ESD, diverter fully in "bypass"

6. GT operating normally, ORC cold (shutdown), diverter fully in "bypass"

7. GT operating normally, ORC cold beginning startup, diverter initially in "bypass"

8. GT operating normally, ORC hot beginning startup, diverter initially in "bypass"

9. GT operating normally, ORC cold starting up, diverter switching from "bypass" to "open"

10. GT operating normally, ORC hot starting up, diverter switching from "bypass" to "open"

11. GT shut down, preparing to restart, ORC boiler cleared by purge with GT crank and diverter "open"

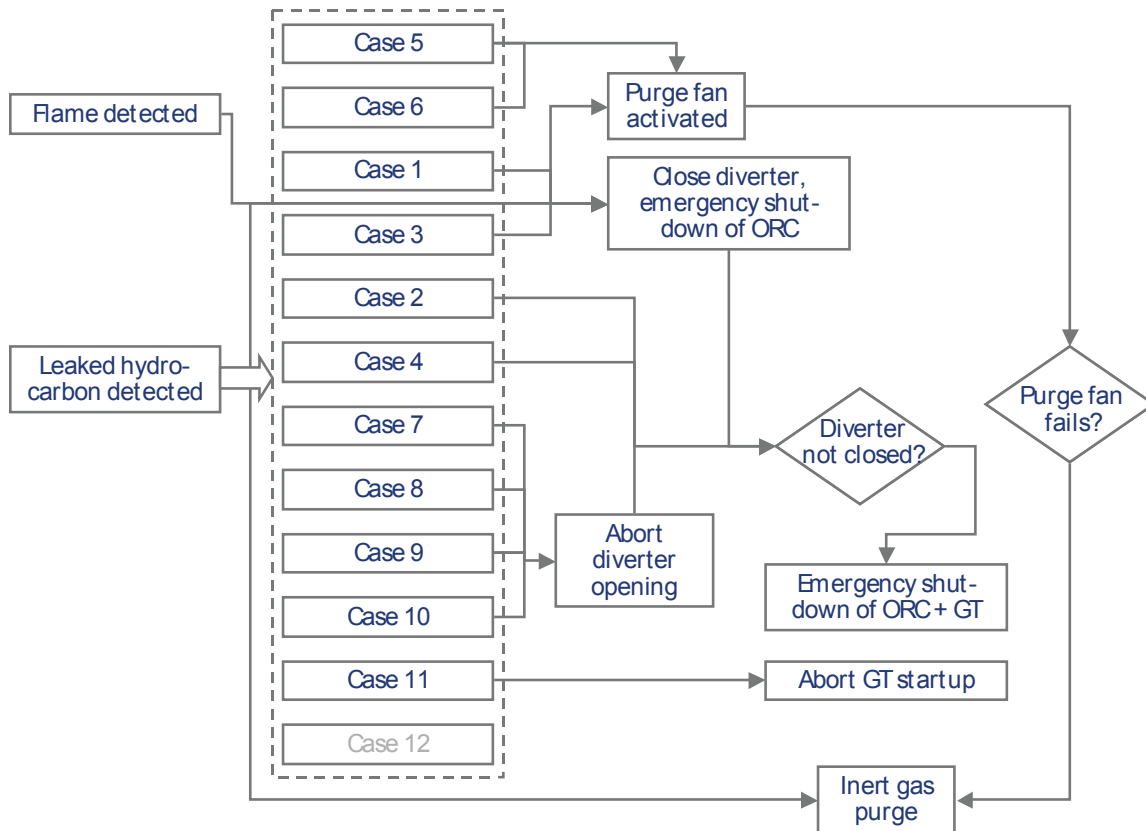12. ORC shut down for maintenance, diverter fully in "bypass."

Figure 19. Safety system logic diagram.

Specific recommendations to minimize the potential for a deflagration in the direct evaporator unit include:

1. Do not allow flammable/explosive concentrations of working fluid-TEG mixtures to stagnate. It is advisable to sweep such mixtures through the system. A minimum TEG velocity should be observed.

2. Purge the oxygen out of the ORC with an inert gas upon system startup.

3. Incorporate hydrocarbon sensors in appropriate locations to detect leaks. If a leak is detected, a system to divert the hot gas to bypass stack should be activated and the working fluid system depressurized. Since the working fluid is heavier than air, any escaped liquid or vapor will tend to settle in low areas or travel some distance along the ground or surface. Potential ignition sources should be identified and safeguarded.

2.5   Flammability Analysis

As shown in Figure 20, oxygen, heat, and fuel are the three elements necessary for a fire to occur. Autoignition occurs when sufficient self-heating by chemical reactions takes place to accelerate the rates of reactions to produce full-scale combustion. Combustion is the sequence of exothermic chemical reactions that occur between a fuel and an oxidant accompanied by the production of heat and conversion of chemical species. Combustion feeds a fire with heat, enabling the process to continue. In the proposed ORC design, oxygen, heat (from the TEG), and fuel (i.e., the working fluid) are present creating the potential for a fire in the direct evaporator.

Figure 20. Fire triangle (courtesy of Wikimedia Commons).

The reaction rate depends on the mean species concentration of the mixture and the local mean temperature. The concentration field and the progress of chemical reaction are affected by the topology of the turbulent flow field [3]. Ignition processes are usually very complex and involve many intricate physical and chemical steps [4]. These steps take a finite amount of time and the period between the start of injection and the start of combustion is referred to as *ignition delay time* (IDT). The IDT is a latent period in the combustion process, during which the temperature remains nearly constant [5]. The delay time is comprised of a physical delay and a chemical delay component. The physical delay is due to the finite rate of mixing of injected working fluid with hot exhaust gas and is the time needed for the flammable gas mixture to reach the autoignition temperature. The chemical delay is due to pre-combustion reactions of the combustible gas mixture that lead to autoignition. In reality, both the physical and chemical processes are occurring simultaneously and cannot be decoupled. Therefore, the actual autoignition delay time in a flowing system is difficult to determine, as it will be affected by the [6]:

- Time taken for fuel and TEG to mix

- Time for the fuel temperature to rise to that of the TEG

- Chemical kinetic time for the autoignition reactions to initiate.

2.5.1   Autoignition Tests

An IQT™ apparatus was used to determine the IDT of cyclopentane at atmospheric pressure for vitiated air (13.3% oxygen). The test matrix encompassed equivalence ratios of 0.5, 1, 3, and 5 and chamber temperatures ranging from 673 to 823 K to establish a set of ignition delay curves. As seen in Figure 21, IDT was observed to decrease with increasing temperature and equivalence ratio. For the cases tested, no ignition was observed at temperatures at or below 723 K or at an equivalence ratio of 0.5.

For these conditions, cyclopentane was observed to ignite in the range of 0.62 s ($\Phi = 5$, T = 823 K) to 3.75 s ($\Phi = 1$, T = 803 K) from the time of injection at 13.3% oxygen concentration and 1 bar pressure. At lean conditions ($\Phi = 0.5$), no ignition was observed, even though the conditions were held for up to 15 seconds. For $\Phi = 1$ and 3, ignition was not observed for temperatures at or below 773 K. All of the cases performed at $\Phi = 5$ ignited.

Because of the nature of non-premixed combustion, the conditions tested by the current set of experiments may not have captured the minimum IDT for an ORC leak scenario. The fuel-rich mixtures exhibit the smallest ignition delays. Therefore, in a leak situation, it is expected that fuel-rich regions would ignite first.
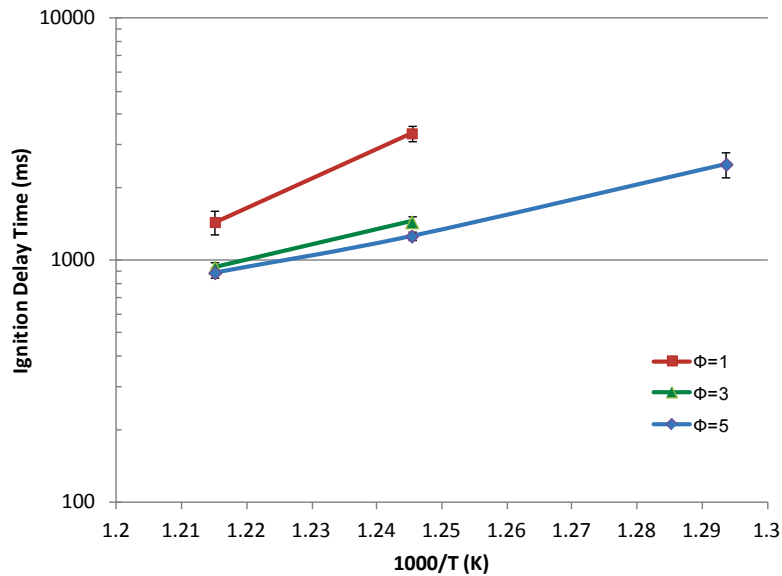
Figure 21. Average measured ignition delay time for cyclopentane.

2.5.2 Computational Fluid Dynamic (CFD) Analyses

Computational fluid dynamic (CFD) analyses were performed to assess the flammability of the selected working fluid in the hot exhaust gas stream stemming from a potential pinhole leak in the evaporator. The primary concern here is the potential for leaked working fluid to become trapped in the recirculation regions behind of the finned tubes. The stabilization of a flame in the eddy region behind a bluff body in a high velocity gas stream is a well-known phenomenon used to anchor the flame in the combustors of jet engines[b]. A flame stabilized in this manner can spread throughout the entire flammable mixture. The residence time of gases in the recirculation zone behind a bluff body dictate whether the flame will propagate or extinguish. The scenario of concern is that fluid released a small leak in a finned tube could ignite, burn undetected for a long time, and potentially degrade surrounding materials or ignite secondary fires.

CFD analyses were performed to assess flammability of a direct evaporator due to a potential pinhole leak in a finned tube. In this non-premixed scenario, there are locally fuel-rich and fuel-lean regions within the heat exchanger. Thus, the leaked fluid concentration falls between the upper and lower flammability limits at some locations.

A three-dimensional large eddy simulation (LES) of a direct evaporator was developed that predicts: (1) residence time of particles representing leaked working fluid in a finned-tube heat exchanger, and (2) temperatures within the heat exchanger after a postulated combustion event. The predicted residence time was compared with the experimentally obtained IDT to determine if the temporal characteristics of the flow recirculation in the wake region of the tubes might lead to ignition. For a baseline operating condition, a combustion simulation is employed to investigate whether temperatures are sufficiently high to affect the structural integrity of the shell-and-tube heat exchanger.

Since well-defined chemical reaction mechanisms were not identified or used for the candidate working fluid, an eddy breakup model was employed. The model considers hot TEG flowing across a finned heat exchanger tube bank section and interacting with vaporized working fluid released from a

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[b] Longwell, J. P., et al., 1953, "Flame Stability in Bluff Body Recirculation Zones." *Industrial and Engineering Chemistry*, Vol. 45, Issue 8, pp. 1629–1633.

volumetric source emanating from the 9 o'clock, 6 o'clock, or 3 o'clock position on a finned tube. At the flow conditions examined, fluid released from a pinhole leak at 6 and 9 o'clock positions on the tube would not be caught up in the wake, but rather be swept downstream before autoignition could occur. Combustion caused by a leak emanating from the 3 o'clock position is simulated, which is the worst case from the standpoint of cyclopentane getting trapped in the wake of the breached tube. This scenario is based upon combustion occurring immediately after the release of cyclopentane into the hot TEG. Figure 22 shows the temperature distribution at 3 s from the start of combustion. The corresponding temperature distribution on the surface of the fins is shown in Figure 23. The maximum fin temperature (1234 K) is observed to occur on the upstream side of the tube directly downstream of the one with the leak. The fin temperatures are below the melting temperature for the carbon steel fins. Therefore, if combustion were initiated as the result of a leak, the simulation shows that finned tubes would not be damaged.
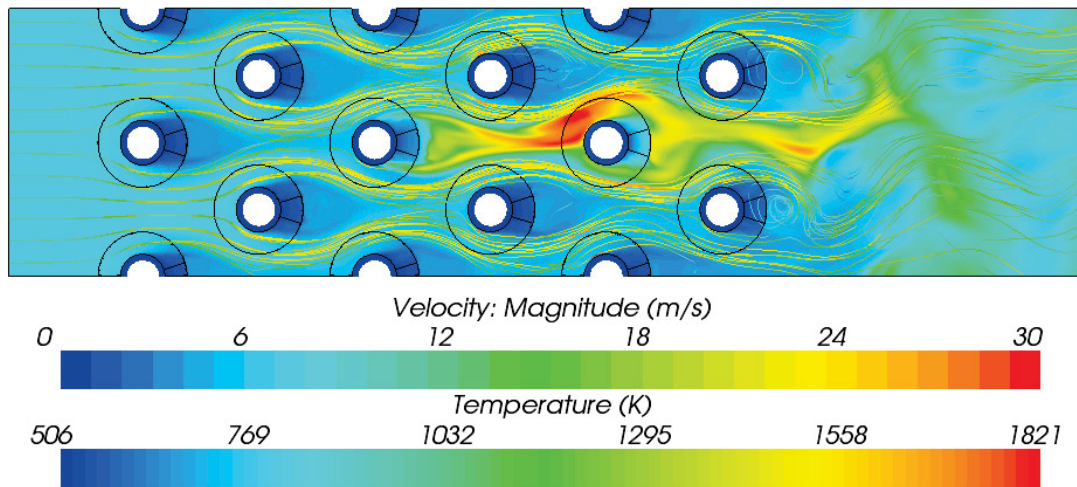

Figure 22. Streamlines overlaid on temperature distribution resulting from combustion.
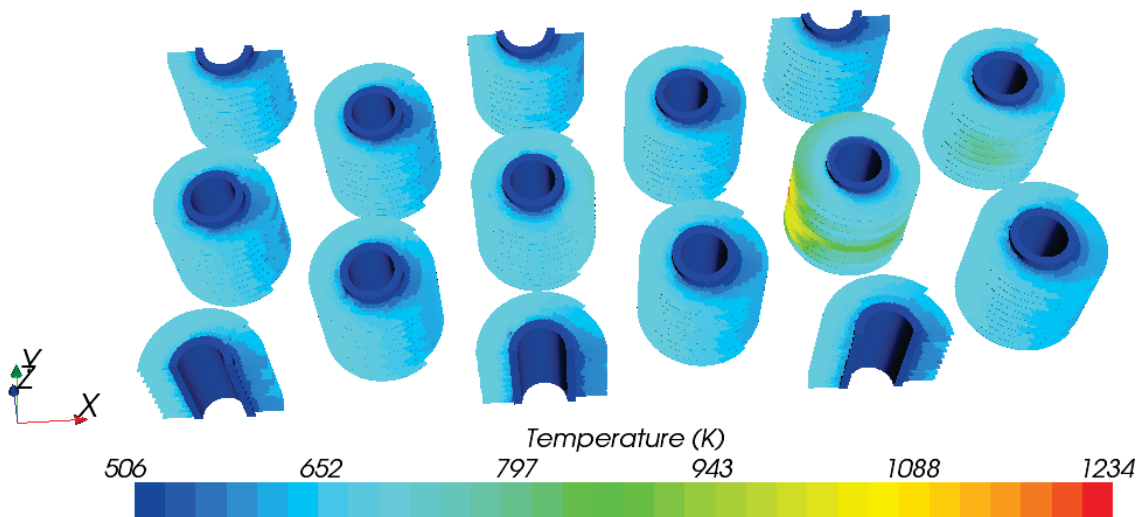

Figure 23. Temperature distribution of fin surfaces exposed to flame.

2.6 Prototype Test

2.6.1 Testbed and Equipment

Figure 24. Direct evaporator installed at GE test facility.

Figure 24 shows the direct evaporator and supporting equipment installed at the GE test facility. The testbed consisted of a cooling water loop, direct evaporator prototype, hot exhaust gas supply, working fluid loop, compressor, container/trailer, and leak containment. The test and setup were designed to characterize an experimental heat exchanger with the working fluid cyclopentane in the direct path of an exhaust stream to simulate the heat exchanger behind a GE PGT25 gas turbine. The working fluid cycled in a loop consisting of the test heat exchanger, a heat recovery vapor generator (HRVG) (Deltak, Inc.), a condenser (Thermal Products Inc.), and a pump (ChemPump). The pressure in the system was maintained by a rack of hydraulic piston accumulators (Pearse Bertram). The maximum temperature of the vapor exiting the Heat Exchanger (HX) superheater (I-16) is 250°C, and the system pressure was controlled at 508 psia (35 bar) as measured at the inlet to the HX. Prior to entering the pump, fluid is condensed and subcooled to approximately 165°C.

This test was run via a remote station equipped with remote monitoring cameras on the heat exchanger and on the inside of the container. After a short performance characterization test, the system was run for >300 hours with periods for ramping and deramping the system with predetermined sampling intervals to measure the degradation products.

Prior to starting full-scale tests with flammable cyclopentane, a detailed shakedown plan was drawn up to test various subsystems and procedures with a benign chemical (10% alcohol/water mixture). Performing a full subsystems and procedures shakedown test with the above fluid allowed a further benefit of precleaning the test rig before finally charging with cyclopentane.

One of the main features of the protective staging concept was that it mitigated the risk of high wall temperatures in the working fluid loop. Furthermore, variable fin density in each of the three sections of the heat exchanger (economizer, evaporator, and superheater) on the outside of the working fluid tubes was used to control the heat flux into each section.

It was generally observed during the experiment that even where the hot exhaust temperature was >400°C, the wall temperature of the finned tubes remained below 300°C. Therefore, the fluid was not exposed to high-risk temperatures, validating the effectiveness of the protective staging concept and the design of the heat exchanger.

It has been shown by means of experiment that:

- The temperature of a boiling (and superheating) ORC working fluid can be controlled by using protective staging when the hot resource temperature is significantly higher than the working fluid decomposition temperature.

- The heat transfer rate into the working fluid can be controlled be using variable fin densities in the evaporator, superheater, and economizer.

- The working fluid degradation/decomposition after 320 hours of hot run time was 30 ppm—an insignificant amount—suggesting the protective staging and variable finning approach was very effective.

- The safety hazard of direct exposure of a flammable fluid to a hot combustion gas was effectively mitigated in this experiment. There were no safety incidents involving the flammable fluid throughout this experiment.

2.6.2   Test Results

The test was run for over 300 hours at conditions that produced a nominal working fluid temperature of 185°C. Figure 25 shows the variation of nominal working fluid temperature through the heat exchanger. The working fluid temperature was increased to a higher average temperature of 213°C during the last two runs. The test rig was operated for a cumulative time of over 333 hours.

Data collected from the test consisted of the following:

- HX inlet flow profiles to determine flow uniformity at the HX inlet
- Thermocouple data at various locations in the heat exchanger
- Mass flow rates and pressure drop for the exhaust gas and working fluid
- Calorimetric data to measure heat leak from the HX shell to the environment
- Calorimetric data to measure net heat transfer from hot exhaust gas into the working fluid
- Wall temperature of the tubes to monitor excursions into high temperatures that risk fluid decomposition
- Chemical tests of the samples to determine the level of decomposition products
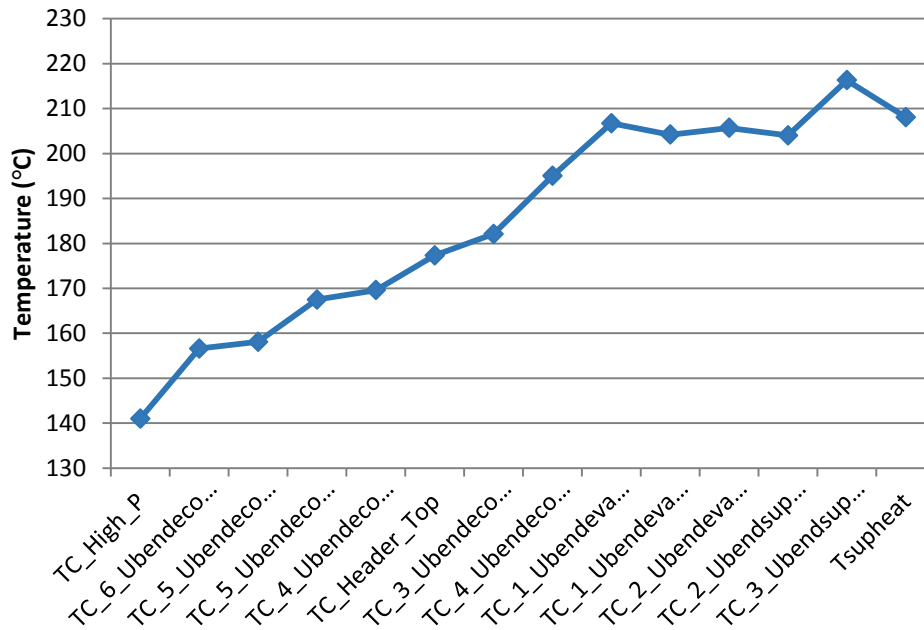- Analysis of residuals in the HX tubes to assess fouling.

Figure 25. Variation of nominal working fluid temperature through heat exchanger.

3. BENEFITS ASSESSMENT

For an engine that is 35% efficient, 65% of fuel energy is turned into heat. The ORC can harness 10 to 20% of the wasted heat energy. Therefore, the efficiency gains are around 20 to 40%.

The advantages of an ORC with Direct Evaporation over a conventional ORC system with a secondary oil loop are:

- Overall compactness of plant – Removing the conventional oil heating loop from the ORC system reduces plant footprint by 30%

- Lower cost/kW – Anticipated 15% reduction in overall plant cost, same kW

- Simplicity/reliability of operation – Significant reduction in number of cycle components (oil system pumps, valves, storage tanks) leads to easier maintenance, more predictable plant behavior

- Lower lead time for startup/shutdown – With gas turbine running, heating/vaporization of working fluid can begin immediately, without waiting for the oil loop to heat up.

There are potential economic benefits from the adoption of this technology. Electricity from waste heat is the cheapest form of new energy generation. Tax incentives would boost sales if waste heat recovery was recognized appropriately as a renewable or renewable equivalent in federal legislation. Potentially 720,000 new American manufacturing jobs could result from the production and sales of this equipment. Similar technology can be used to generate zero carbon emission electricity from geothermal and solar energy sources.

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4. COMMERCIALIZATION

GE has performed a market assessment of competing technologies and has determined the niche for the ORC Direct Evaporator technology. A major advantage of the GE ORC system is that it will integrate with existing GE products without invalidating the engine's warranty. The team has been contacted by numerous businesses interested in this project. Industrial gas turbines are widely used for generating onsite heat and power in a diverse range of applications. The market segments for these units are very diverse and include:

- Process industries (e.g., food and drink, pulp and paper, ceramics, chemicals and pharmaceuticals, textiles, and petrochemicals)

- Utilities and municipalities (including district heating)

- Buildings and institutions (hospitals, airports)

- Independent power producers and contract energy managers.

Among the various potential applications of Direct Evaporator technology in the field of waste heat recovery, the GE ORegen™ ORC system, powered by a gas turbine exhaust stream, was chosen as the testbed for the research work conducted under this contract with DOE. The ORegen™ is the first GE product designed for waste heat recovery, and will provide the most suitable point of entry for the Direct Evaporator into the mainstream market. Accordingly, the working fluid, the dimensions of the prototype evaporator, and the properties of the exhaust stream have all been chosen in accordance with a typical gas turbine-driven ORC bottoming cycle.

Commercialization of the Direct Evaporation ORC will be tackled in two phases. The first phase consists of offering a product that couples to GE gas turbines. The product will be offered as retrofit for the installed base of gas turbine engines and pre-packaged with the sale of new units. According to GE's Oil and Gas Division, GE has an installed base of 1800 simple cycle gas turbines, generating 60 GW of power. If 20% were retrofitted with ORCs, they would produce 3 GW of additional electricity, which translates into ~$3B/year of savings. The second phase consists of offering a modular product with generic capability to couple to a wide variety of gas turbines from other manufacturers.

5. PROJECT ACCOMPLISHMENTS

This section includes research harvest from this DOE award, including patents, publications, presentations, and reports.

5.1 Patents

Patents applied for or resulting from this award are listed in Table 6.

Table 6. Patents filed by the research team relating to this work.

	Title	Number	Date
1	An improved Organic Rankine Cycle system and method	EP2423473A2	2012-02-29
2	Compound closed-loop heat cycle system for recovering waste heat and method thereof	US20110113780A1	2011-05-19
3	Direct evaporator apparatus and energy recovery system	US20110120129A1	2011-5-26
4	Direct evaporator apparatus as well as system and method for energy recovery	EP2348200A2	2011-07-27
5	Direct evaporator system and method for Organic Rankine Cycle systems	WO2011066089A1	2011-06-03
6	Dual reheat Rankine Cycle system and method thereof	EP2345793A2	2011-07-20
7	Fire extinguishing system for an Organic Rankine Cycle hydrocarbon evaporator	US20120023943A1	2012-02-02
8	Heat exchanger for direct evaporation in Organic Rankine Cycle systems and method	US20110100009A1	2011-05-05
9	Heat exchanger with surface-treated substrate	US20100263842A1	2011-10-21
10	Inert gas purging system for an ORC heat recovery boiler	US20120000200A1	2012-01-05
11	Organic Rankine Cycle system and method	US20100281865A1	2011-11-11
12	System and method for controlling an expansion system	US20110094230A1	2011-04-28
13	System and method for power generation in Rankine cycle	7,594,399	
14	System and method for recovering waste heat	US20090000299A1	2009-01-01
15	System for recovering waste heat	EP2203630A2	2010-09-29

5.2 Publications

The following articles were presented at conferences or published in the open literature:

1. Guillen, D. P., Klockow, H., Lehar, M., Freund, S., and Jackson, J., "Development of a Direct Evaporator for the Organic Rankine Cycle," *TMS 2011 Annual Meeting, San Diego, California, February 27–March 3, 2011.*

2. Guillen, D. P., "Effect of Materials on the Autoignition of Cyclopentane," *TMS 2012 Annual Meeting, Orlando, Florida, March 11–15, 2012.*

3. Ginosar, D. G., Petkovic, L., and Guillen, D.P., "Thermal Stability of Cyclopentane as an Organic Rankine Cycle Working Fluid," *Energy & Fuels*, No. 25, September 2011, p. 4138–4144.

4. Guillen, D. P., "The Autoignition of Cyclopentane in an Ignition Quality Tester," *Journal of Materials*, Vol. 64, No. 8, August 2012.

5.3 Presentations

The following presentations were made to DOE or in public forums:

1. DOE Industrial Technologies Program, 2008 Corporate Peer Review, "Energy Intensive Processes Portfolio Review," Washington, DC, October 29, 2008.

2. Project Kickoff Meeting, Idaho Falls, Idaho, May 13–14, 2009.

3. DOE Industrial Technologies Program, "Portfolio Review," Washington, DC, November 4, 2009.

4. GO/NO-GO Decision Meeting, Niskayuna, New York, June 3, 2010.

5. Development of a Direct Evaporator for the Organic Rankine Cycle, TMS 2011 Annual Meeting, San Diego, California, March 3, 2011.

6. DOE/EERE Industrial Technologies Program, "Energy Intensive Processes '08 Projects," *Portfolio Review Meeting, Golden, Colorado, June 14–15, 2011.*

7. "Waste Heat Recovery Research at the Idaho National Laboratory," Texas Industries of the Future Technology Forum on Low Temperature Waste Heat Recovery in Chemical Plants and Refineries, Houston, Texas, May 16, 2012.

5.4 Milestone Reports

Five milestone reports were prepared and submitted to DOE:

1. Milestone Report 1, "Design and Modeling of Direct Evaporator," Idaho National Laboratory, December 2009.

2. Milestone Report 2, "Direct Evaporator Leak and Flammability Concerns," INL/EXT-11-23768, Idaho National Laboratory, September 2013.

3. Milestone Report 3, "Direct Evaporator Failure Modes Effects Analysis (FMEA) and HazOp Study," Idaho National Laboratory, August 2010.

4. Milestone Report 4, "Working Fluid Decomposition," INL/LTD-10-20378, Idaho National Laboratory, November 2013.

5. Milestone Report 5, "Prototype Test," INL/EXT-12-27526, Idaho National Laboratory, September 2013.

6. CONCLUSIONS AND RECOMMENDATIONS

The direct evaporator shows promise for future ORC systems due to its simplicity and lower cost. Overall efficiency can be increased by eliminating the efficiency losses and costs associated with the oil loop. A prototype ORC system using direct evaporation technology has been fabricated and tested. The selection of an optimum ORC working fluid has been outlined. The conditions and constraints imposed on the thermodynamic analysis and heat exchanger design have been discussed. Safety risks recognized during the FMEA/HAZOP are given, along with mitigation strategies. Leak ignition processes are identified and the methodology employed in the flammability analysis is provided.

Successful demonstration of the Direct Evaporator will facilitate the next stage of the ORegen ORC product line from GE. It will provide a more compact and less expensive follow-on product to the Gen-1 system currently available to customers. A market success for the current product will open a pathway for sales of the more advanced Gen-2 system employing the Direct Evaporator.

7. ACKNOWLEDGMENTS