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GAS TURBINE WASTE HEAT RECOVERY USING A 20,000 RPM, SEALLESS, TURBINE GENERATOR/ORC SYSTEM

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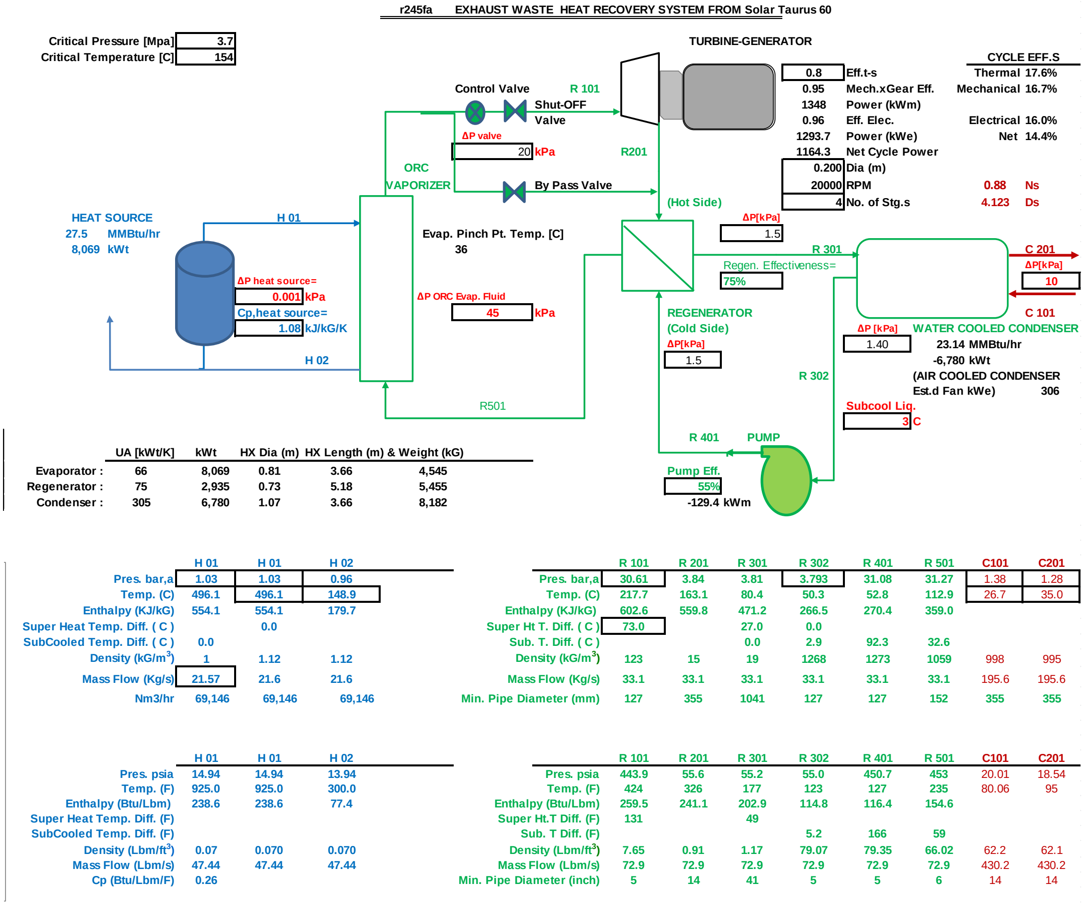
Abstract

The fuel efficiency of small gas turbines can be improved by 25% by using a modular Turbine Generator Unit (TGU) in an Organic Rankine Cycle (ORC) system to recover exhaust gas waste heat. This paper describes a high-speed, 20,000 rpm TGU that can operate completely sealed in the organic working fluid. The TGU is rated for 330 kWe, and consists of a close-coupled turbine and high-speed, permanent magnet generator, with magnetic bearings, in a hermetically sealed package. The TGU is an integral unit, without a shaft seal or lubrication required for its magnetic bearings. The turbine gas path employs a modular cartridge design, allowing for cost-effective optimization and modification for a wide range of operating conditions and working fluids. This modular design results in a compact, robust, and highly-efficient system, with reduced maintenance and operating costs when compared to traditional, open-architecture ORC systems. The instantaneous operating speed of the TGU can be changed to best match any part-load transients required—and the speed control helps to maintain the highest power recovery from the exhaust gas, while providing a constant generator frequency and voltage. A single 300 TGU rated for 330 kWe can recover the exhaust from a 1.25 to 1.75 MWe gas turbine, depending on exhaust temperature and flow rate. This paper describes the TGU's mechanical and electrical design in more detail, and provides results from several case studies of gas turbine applications using the 300 ORC turbine generator.

Introduction and Background

The gas turbine is the prime mover of choice for stationary power generation applications, due to its high fuel efficiency, compact size, and high power-to-weight ratio. It is therefore not surprising that applications for the gas turbine are expanding into many areas of power generation that were previously reserved for reciprocating applications. However, a gas turbine's efficiency is more affected by changes in atmospheric pressure and temperature than internal combustion engines are. The efficiency of a gas turbine can also be reduced by as much as 25% at 50% part-load. Consequently, the thermodynamic reality is that a gas turbine converts 90% of its fuel consumption as either shaft power or as exhaust gas waste heat—which compels the continued exploration of methods for improving the efficiency or power of gas turbines across the operating range. In addition, recent developments in Organic Rankine Cycle (ORC) systems encourage the integration of gas turbines and ORC systems, to accomplish better fuel economy and more power output.

An ORC is a regenerative Rankine cycle that uses an organic–based fluid to power the vapor expander. The organic fluid can be any number of common refrigerants (typically R245fa or R134a), with the choice of a refrigerant based upon the magnitude of the temperature of the waste heat source. Concepts NREC (CN) typically uses R245fa with waste heat temperatures ranging from 500 to 1000°F, taking special care not to exceed the bulk temperature limitations. R134a is used for temperatures below 260°C, and R236fa may be considered for use below 150°C. A thermodynamic cycle analysis is always done to determine the best fluid to use with the ORC system, and the analysis must include the exact specifications of the waste heat source flow rate and temperature. Figure 1 shows a component schematic and state points for a typical ORC system.



	UA [kWt/K]	kWt	HX Dia (m)	HX Length (m)	& Weight (kG)
Evaporator :	66	8,069	0.81	3.66	4,545
Regenerator :	75	2,935	0.73	5.18	5,455
Condenser :	305	6,780	1.07	3.66	8,182

	H 01	H 01	H 02
Pres. bar,a	1.03	1.03	0.96
Temp. (C)	496.1	496.1	148.9
Enthalpy (KJ/kG)	554.1	554.1	179.7
Super Heat Temp. Diff. (C)		0.0	
SubCooled Temp. Diff. (C)	0.0		
Density (kG/m^3)	1	1.12	1.12
Mass Flow (Kg/s)	21.57	21.6	21.6
Nm3/hr	69,146	69,146	69,146

	R 101	R 201	R 301	R 302	R 401	R 501	C101	C201
Pres. bar,a	30.61	3.84	3.81	3.793	31.08	31.27	1.38	1.28
Temp. (C)	217.7	163.1	80.4	50.3	52.8	112.9	26.7	35.0
Enthalpy (KJ/kG)	602.6	559.8	471.2	266.5	270.4	359.0		
Super Ht T. Diff. (C)	73.0		27.0	0.0				
Sub. T. Diff. (C)			0.0	2.9	92.3	32.6		
Density (kG/m^3)	123	15	19	1268	1273	1059	998	995
Mass Flow (Kg/s)	33.1	33.1	33.1	33.1	33.1	33.1	195.6	195.6
Min. Pipe Diameter (mm)	127	355	1041	127	127	152	355	355

	H 01	H 01	H 02
Pres. psia	14.94	14.94	13.94
Temp. (F)	925.0	925.0	300.0
Enthalpy (Btu/Lbm)	238.6	238.6	77.4
Super Heat Temp. Diff. (F)			
SubCooled Temp. Diff. (F)			
Density (Lbm/ft^3)	0.07	0.070	0.070
Mass Flow (Lbm/s)	47.44	47.44	47.44
Cp (Btu/Lbm/F)	0.26		

	R 101	R 201	R 301	R 302	R 401	R 501	C101	C201
Pres. psia	443.9	55.6	55.2	55.0	450.7	453	20.01	18.54
Temp. (F)	424	326	177	123	127	235	80.06	95
Enthalpy (Btu/Lbm)	259.5	241.1	202.9	114.8	116.4	154.6		
Super Ht.T Diff. (F)	131		49					
Sub. T Diff. (F)				5.2	166	59		
Density (Lbm/ft^3)	7.65	0.91	1.17	79.07	79.35	66.02	62.2	62.1
Mass Flow (Lbm/s)	72.9	72.9	72.9	72.9	72.9	72.9	430.2	430.2
Min. Pipe Diameter (inch)	5	14	41	5	5	6	14	14

Figure 1. Typical Organic Rankine Cycle system, with block diagram and state points.

In order to attain maximum efficiency, the operating pressure and temperature of the ORC working fluid may need to be close to the critical pressure and approximately

50-75°F below the recommended maximum bulk fluid temperature of the fluid—as may be observed in the corresponding T-s diagram, shown in Figure 2.

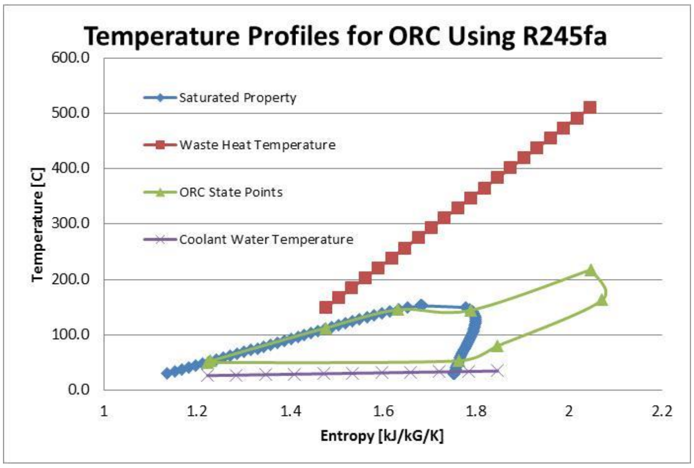


Figure 2. T-s diagram for the ORC system shown in Figure 1.

The efficiency of contemporary gas turbines has been effectively summarized in a plot of efficiency vs. the specific power [kWe/(kg/s)], as shown in Figure 3.[1]. It is clear from Figure 3 that the efficiency of a gas turbine tends to increase with its power rating. For large power prime movers an important subtlety that can be discerned from Figure 3 is that as a consequence of increased thermal efficiency, the power output increases faster than the required mass flow rate through the gas turbine. This implies that the exhaust temperature is hotter and therefore steam Rankine cycle system should be more practical to use for recovering the waste heat from the gas turbine.

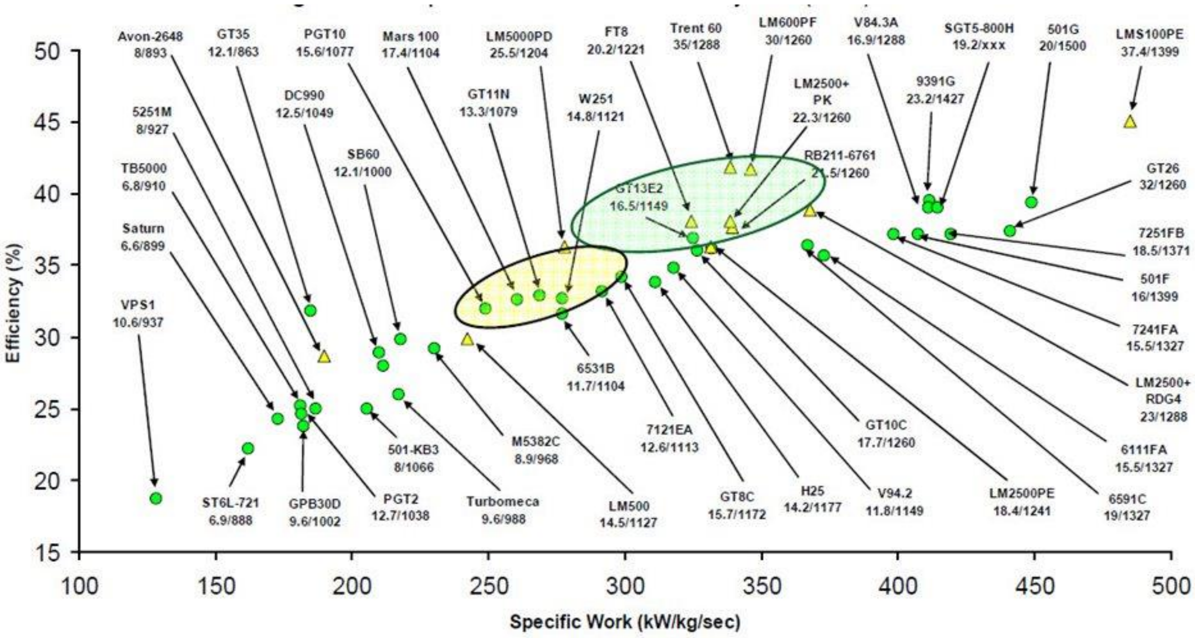
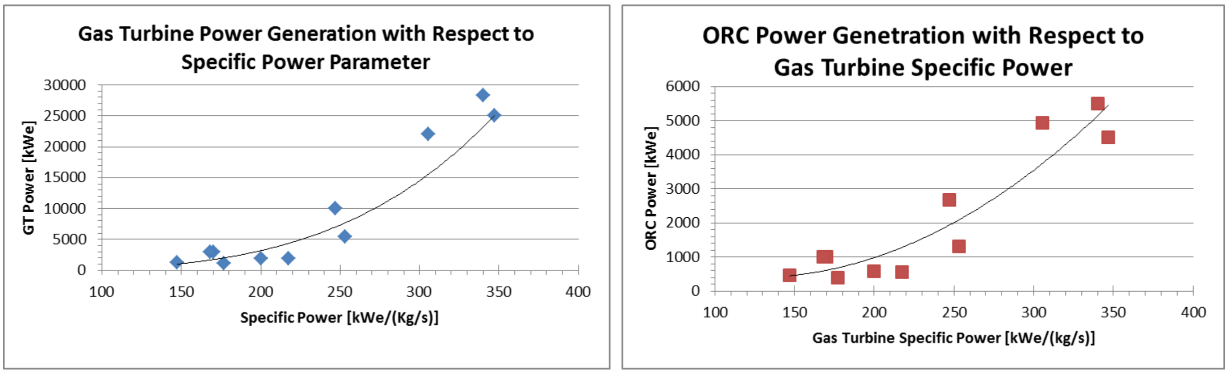


Figure 3. Summary of gas turbine efficiency,
as a function of specific power parameter.

In Figure 4A, which was prepared by this author, the kWe rating of the gas turbine is shown as a function of the specific power. This figure can serve as a modified companion graph to Figure 3. The application of an ORC system to any one of these manufactured gas turbines results in effective power recovery, as shown in Figure 4b.



Figures 4a and 4b. GT power and ORC power recovery, as functions of gas-turbine-specific power parameters.

The data displayed in these two figures can be used to determine the potential power improvement that can be achieved using an ORC system with respect to the specific power parameter. Results of this calculation are shown in Figure 5.

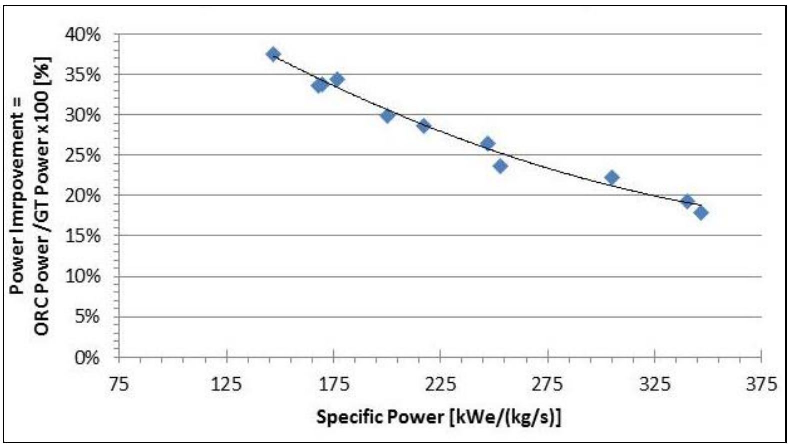


Figure 5. Potential power improvement for a gas turbine using an ORC system for exhaust gas heat recovery.

The interesting, and not unreasonable, conclusion that can be drawn is that there is a power recovery improvement opportunity and benefit if an ORC system is applied to a small gas turbine. This conclusion is understandable, given the lower efficiency of smaller gas turbine systems, as previously indicated by Figure 3. While it is correct that the magnitude of power recovery from an ORC system is in proportion to the power rating of the gas turbine, the simple payback for any prime mover power generator is dependent on the installed cost per kWe ($/kWe) of the prime mover, the availability of the system, and continuous operation and maintenance costs—not upon the absolute magnitude of the rated power.

Typical cost per kWe is shown in Figure 6, illustrating that this cost is inversely proportional to the system's size. (Figure 6 is based on a literature search of published ORC system costs, as presented by independent sources—such as references 2 and 6.) The inverse relationship between $/kWe and rated power is consistent for any developed prime mover system. This is why it is more common for system developers to install large ORC systems with large gas turbine systems.

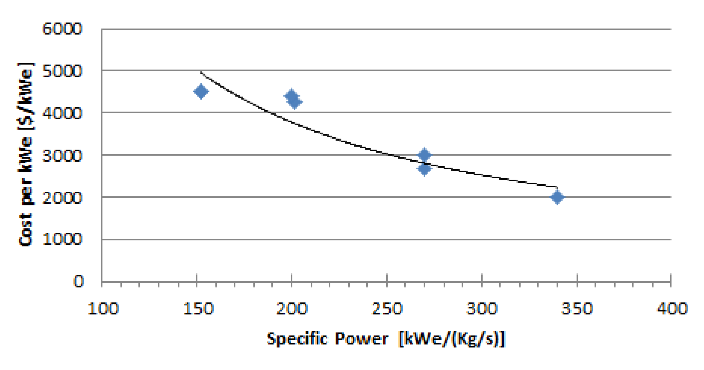
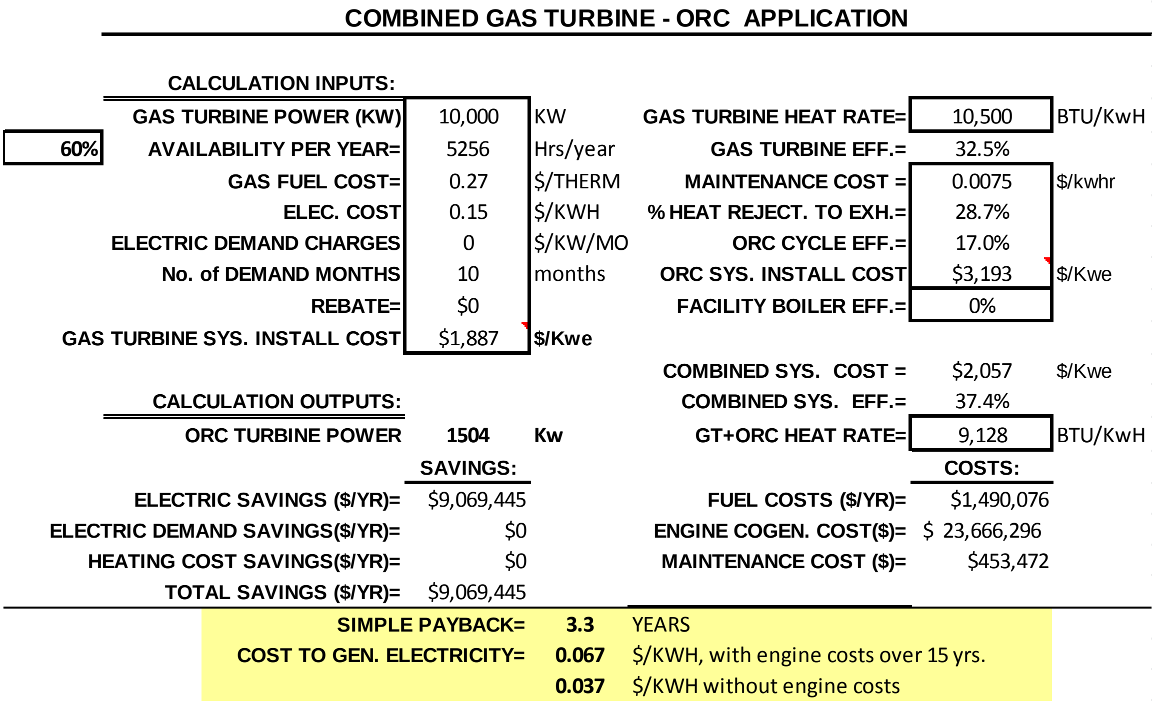


Figure 6. Installed cost per kW of typical ORC systems.

However, the duty cycle, system availability, and operation and maintenance costs for the combined system also affect simple payback—as shown in Table 1 and Tables 2a & 2b. Table 1 illustrates simple payback for a combined gas turbine and ORC installation for a relatively large, power-rated system. The simple payback of 3.3 years is very reasonable at the10 MWe gas turbine rating, and this is only 0.2 years more than if the gas turbine were to operate alone, without heat recovery. The 10 kWe rating was chosen for this example as per a study which found that a combined GT/ORC system can be economical at a gas turbine rating of at least 10.5 kWe.

TABLE 1. Simple payback calculation for a gas/turbine ORC system



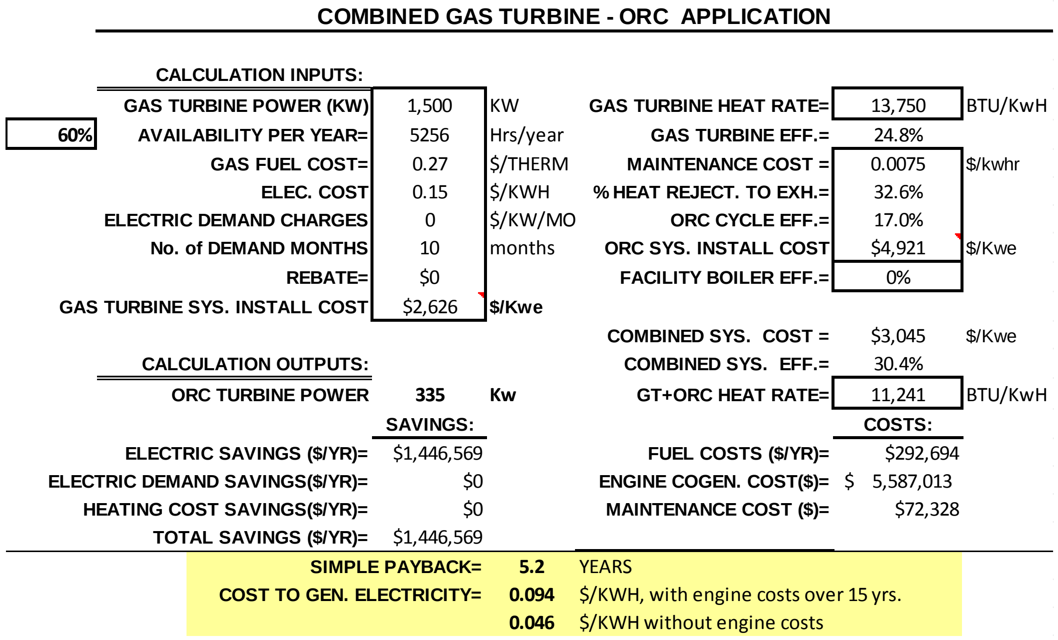
COMBINED GAS TURBINE - ORC APPLICATION

	CALCULATION INPUTS:					
	GAS TURBINE POWER (KW)	10,000	KW	**GAS TURBINE HEAT RATE=**	10,500	BTU/KwH
60%	**AVAILABILITY PER YEAR=**	5256	Hrs/year	**GAS TURBINE EFF.=**	32.5%	
	GAS FUEL COST=	0.27	$/THERM	**MAINTENANCE COST =**	0.0075	$/kwhr
	ELEC. COST	0.15	$/KWH	**% HEAT REJECT. TO EXH.=**	28.7%	
	ELECTRIC DEMAND CHARGES	0	$/KW/MO	**ORC CYCLE EFF.=**	17.0%	
	No. of DEMAND MONTHS	10	months	**ORC SYS. INSTALL COST**	$3,193	$/Kwe
	REBATE=	$0		**FACILITY BOILER EFF.=**	0%	
	GAS TURBINE SYS. INSTALL COST	$1,887	**$/Kwe**			
				COMBINED SYS. COST =	$2,057	$/Kwe
	CALCULATION OUTPUTS:			**COMBINED SYS. EFF.=**	37.4%	
	ORC TURBINE POWER	**1504**	**Kw**	**GT+ORC HEAT RATE=**	9,128	BTU/KwH
		SAVINGS:			**COSTS:**	
	ELECTRIC SAVINGS ($/YR)=	$9,069,445		**FUEL COSTS ($/YR)=**	$1,490,076	
	ELECTRIC DEMAND SAVINGS($/YR)=	$0		**ENGINE COGEN. COST($)=**	$ 23,666,296	
	HEATING COST SAVINGS($/YR)=	$0		**MAINTENANCE COST ($)=**	$453,472	
	TOTAL SAVINGS ($/YR)=	$9,069,445				

SIMPLE PAYBACK=	**3.3**	YEARS
COST TO GEN. ELECTRICITY=	**0.067**	$/KWH, with engine costs over 15 yrs.
	0.037	$/KWH without engine costs

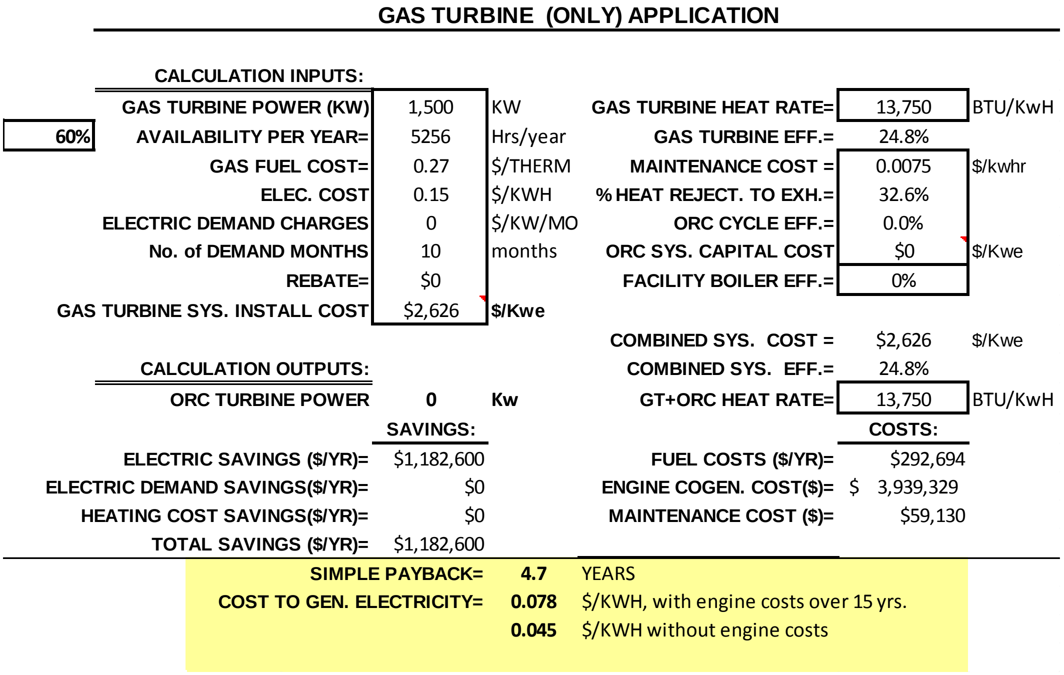
Table 2a provides a similar simple payback analysis, but for a combined gas turbine/ORC system, with the gas turbine rated at 1,500 kWe (the nominal rating required of a gas turbine to produce at least 330 kWe from an ORC system). Table 2b provides the simple payback analysis for just a gas turbine (GT) system without the ORC system. The simple payback analysis used a natural gas cost of $0.27/therm and an electric energy value of $.15/kWh. As the tables show, the change is only 0.5 years between the simple paybacks for the combined gas turbine/ORC system and a system that only uses the gas turbine—and the difference between the simple payback for a large combined (10.5 MWe) gas turbine/ORC system and a smaller 1.5MWe is only two years.

Tables 2a and 2b: Comparing a combined GT/ORC system with a simple GT system



COMBINED GAS TURBINE - ORC APPLICATION

	CALCULATION INPUTS:					
	GAS TURBINE POWER (KW)	1,500	KW	**GAS TURBINE HEAT RATE=**	13,750	BTU/KwH
60%	**AVAILABILITY PER YEAR=**	5256	Hrs/year	**GAS TURBINE EFF.=**	24.8%	
	GAS FUEL COST=	0.27	$/THERM	**MAINTENANCE COST =**	0.0075	$/kwhr
	ELEC. COST	0.15	$/KWH	**% HEAT REJECT. TO EXH.=**	32.6%	
	ELECTRIC DEMAND CHARGES	0	$/KW/MO	**ORC CYCLE EFF.=**	17.0%	
	No. of DEMAND MONTHS	10	months	**ORC SYS. INSTALL COST**	$4,921	$/Kwe
	REBATE=	$0		**FACILITY BOILER EFF.=**	0%	
	GAS TURBINE SYS. INSTALL COST	$2,626	**$/Kwe**			
				COMBINED SYS. COST =	$3,045	$/Kwe
	CALCULATION OUTPUTS:			**COMBINED SYS. EFF.=**	30.4%	
	ORC TURBINE POWER	**335**	**Kw**	**GT+ORC HEAT RATE=**	11,241	BTU/KwH
		SAVINGS:			**COSTS:**	
	ELECTRIC SAVINGS ($/YR)=	$1,446,569		**FUEL COSTS ($/YR)=**	$292,694	
	ELECTRIC DEMAND SAVINGS($/YR)=	$0		**ENGINE COGEN. COST($)=**	$ 5,587,013	
	HEATING COST SAVINGS($/YR)=	$0		**MAINTENANCE COST ($)=**	$72,328	
	TOTAL SAVINGS ($/YR)=	$1,446,569				
	SIMPLE PAYBACK=	**5.2**	YEARS			
	COST TO GEN. ELECTRICITY=	**0.094**	$/KWH, with engine costs over 15 yrs.			
		0.046	$/KWH without engine costs			

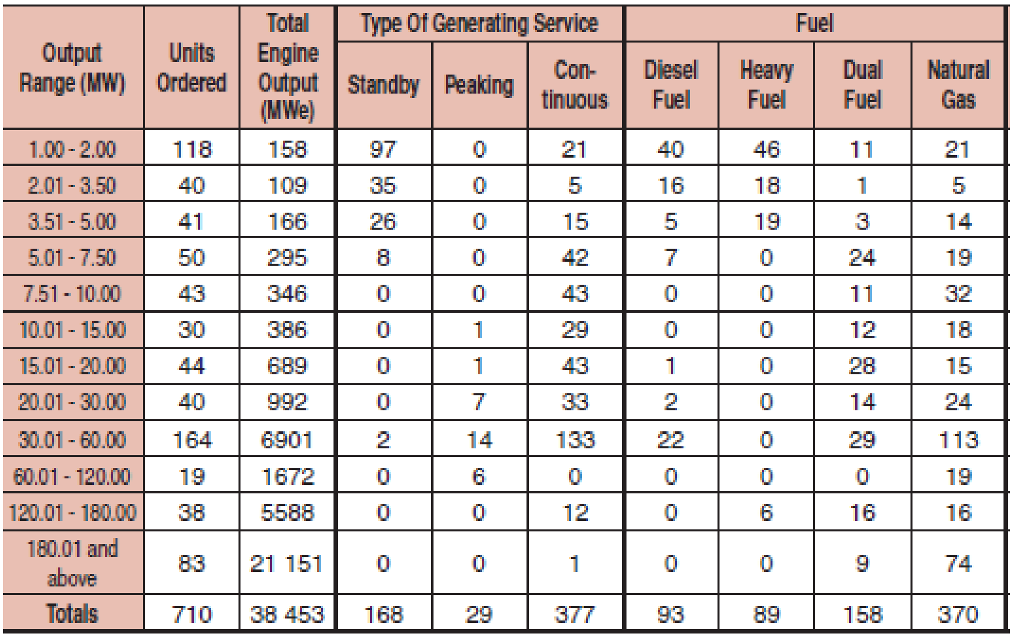


GAS TURBINE (ONLY) APPLICATION

	CALCULATION INPUTS:					
	GAS TURBINE POWER (KW)	1,500	KW	**GAS TURBINE HEAT RATE=**	13,750	BTU/KwH
60%	**AVAILABILITY PER YEAR=**	5256	Hrs/year	**GAS TURBINE EFF.=**	24.8%	
	GAS FUEL COST=	0.27	$/THERM	**MAINTENANCE COST =**	0.0075	$/kwhr
	ELEC. COST	0.15	$/KWH	**% HEAT REJECT. TO EXH.=**	32.6%	
	ELECTRIC DEMAND CHARGES	0	$/KW/MO	**ORC CYCLE EFF.=**	0.0%	
	No. of DEMAND MONTHS	10	months	**ORC SYS. CAPITAL COST**	$0	$/Kwe
	REBATE=	$0		**FACILITY BOILER EFF.=**	0%	
	GAS TURBINE SYS. INSTALL COST	$2,626	**$/Kwe**			
				COMBINED SYS. COST =	$2,626	$/Kwe
	CALCULATION OUTPUTS:			**COMBINED SYS. EFF.=**	24.8%	
	ORC TURBINE POWER	**0**	**Kw**	**GT+ORC HEAT RATE=**	13,750	BTU/KwH
		SAVINGS:			**COSTS:**	
	ELECTRIC SAVINGS ($/YR)=	$1,182,600		**FUEL COSTS ($/YR)=**	$292,694	
	ELECTRIC DEMAND SAVINGS($/YR)=	$0		**ENGINE COGEN. COST($)=**	$ 3,939,329	
	HEATING COST SAVINGS($/YR)=	$0		**MAINTENANCE COST ($)=**	$59,130	
	TOTAL SAVINGS ($/YR)=	$1,182,600				
	SIMPLE PAYBACK=	**4.7**	YEARS			
	COST TO GEN. ELECTRICITY=	**0.078**	$/KWH, with engine costs over 15 yrs.			
		0.045	$/KWH without engine costs			

It is clear from the analysis of simple payback that a smaller system that has a higher availability, lower operation and maintenance costs, and a lower first-time installation cost can also be economical. This is a particularly important conclusion, as many industrial and commercial energy sectors have power demands that can be served by smaller GT/ORC systems that are less than 7.5 MWe. These power demands are thermally equivalent to gas turbines, with specific power levels less than 250 kWe/(Kg/s). For example, the global market for gas turbines [3], itemized by power range, is given in Table 3, where it may be observed that 35% of the turbine sales are less than 7.5 MWe—and of those, 33% are continuous duty.

Table 3. Gas Turbine Sales in 2013



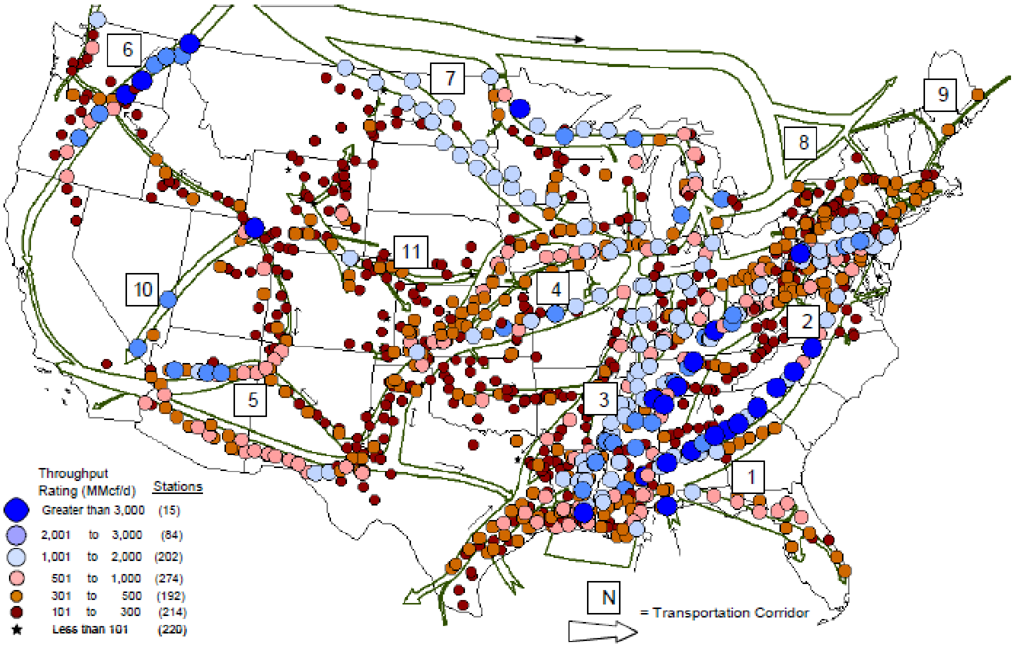
Output Range (MW)	Units Ordered	Total Engine Output (MWe)	Type Of Generating Service			Fuel			
			Standby	Peaking	Con-tinuous	Diesel Fuel	Heavy Fuel	Dual Fuel	Natural Gas
1.00 - 2.00	118	158	97	0	21	40	46	11	21
2.01 - 3.50	40	109	35	0	5	16	18	1	5
3.51 - 5.00	41	166	26	0	15	5	19	3	14
5.01 - 7.50	50	295	8	0	42	7	0	24	19
7.51 - 10.00	43	346	0	0	43	0	0	11	32
10.01 - 15.00	30	386	0	1	29	0	0	12	18
15.01 - 20.00	44	689	0	1	43	1	0	28	15
20.01 - 30.00	40	992	0	7	33	2	0	14	24
30.01 - 60.00	164	6901	2	14	133	22	0	29	113
60.01 - 120.00	19	1672	0	6	0	0	0	0	19
120.01 - 180.00	38	5588	0	0	12	0	6	16	16
180.01 and above	83	21 151	0	0	1	0	0	9	74
Totals	710	38 453	168	29	377	93	89	158	370

In addition, the ORC market in gas compressors/pipeline compressors [4], as shown in Figure 7, indicates that many of the most recent and planned installations are expected to utilize GT drivers, if only for the higher-powered compressor stations. However, as Figure 7 illustrates, 52% of these compressor stations compress less than 500 MMcft/day (million cubic feet per day) of natural gas, and thus have a driver rated for less than 7.5 MWe. It is likely that these stations are now using internal combustion engines rather than gas turbines, but they may consider GT drivers in the future when re-tooling of the existing sites becomes necessary.

With the reduction in the installed cost of an ORC waste heat recovery system, the use of lower power-rated gas turbines in future smaller pipeline compressor stations becomes more economical. For example, according to U.S. Department of Energy researchers, the future needs of the U.S. transportation energy sector are expected to include the use of hydrogen gas as a primary fuel. The hydrogen will need to be transported from the production fields, where renewable energy has been used to produce the hydrogen, to population centers in the U.S. This transportation will involve interstate pipelines that are pressurized by hydrogen pipeline compressors and are each powered by 7,000 kWe gas turbine drivers. Similar markets for gas turbine/ORC applications exist in the industrial energy sectors, in processes that include the liquefaction of natural gas, bio-gas generation and combustion, the combustion of oil field flaring or landfill gas for power generation in GTs and heat recovery, and coal gasification and oxy-combustion technology.

A particularly interesting, and relatively recent, application of gas turbines with waste heat recovery involves the development of small, modular Compressed Air Energy Storage (CAES) systems. Very large CAES systems have been researched since the 1960's, but CAES has only been used by a few electric power utilities to improve the economics of power generation. These improved economics are the result of storing energy during times of less demand, so that it may be used when the demand is higher and the energy can command a higher cost per kWh for the utility.

Recently, there has been research into the deployment of much smaller, almost modular CAES systems that can be paired with power farms of photovoltaic and wind turbines that are generating 3 to 5 MWe of power. The energy stored in the form of compressed air can help the utility grid to stabilize the output from these power farms, despite the inherent intermittent and unpredictable energy fluxes of wind and solar energy sources. The stored, compressed air can be directed into the combustor of a gas turbine, and the instantaneous availability of this pressurized air enables the gas turbine to respond much more quickly to the load changes on the utility grid. During the energy recovery sequence, the exhaust heat from the gas turbine is used to preheat the compressed air from the CAES system—but when the CAES is idle, the gas turbine exhaust waste heat can be recovered in an ORC system, as described in this paper. This approach enables the fuel efficiency of the gas turbine to be maintained as high as possible whenever the turbine is operating.



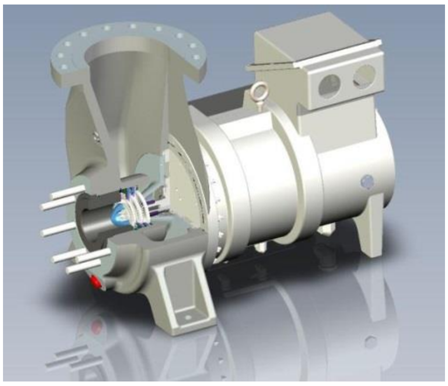
Note: EIA has determined that publication of this figure does not raise security concerns, based on the application of Federal Geographic Data Committee's *Guidelines for Providing Appropriate Access to Geospatial Data in Response to Security Concerns.*
MMcf/d = million cubic feet per day.
Source: Energy Information Administration, Natural Gas Division, Natural Gas Transportation Information System, Compressor Station Database.

Figure 7. Gas pipeline compressor stations, and the number of sites at capacities of less than 500 MMcft/day (equivalent to 7,500 kWe or less).

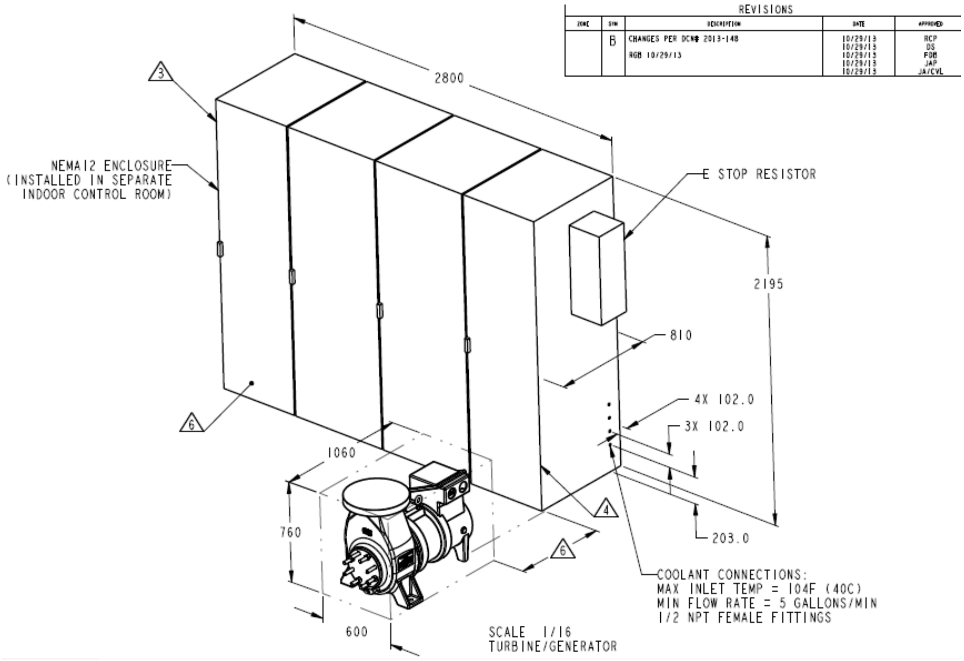
300 kWe Modular, Hermetically Sealed Turbine Generator Unit

Clearly, there is a growing need for an ORC turbine generator that is suitable for smaller applications. Such a system can recover waste heat from internal combustion and gas turbine engines—not just from pipeline compressions systems, but also from bio-gas and flared gas applications that may or may not be using the gas turbine as the prime mover.

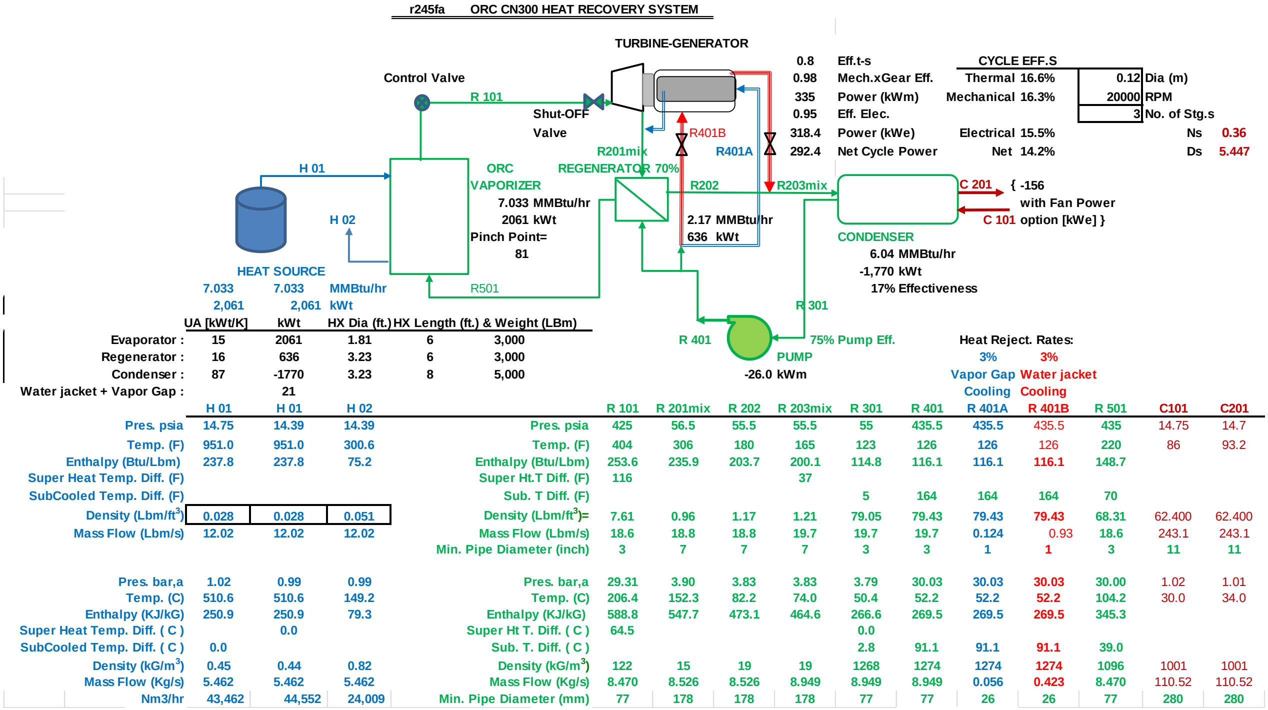
In response to this need, Concepts NREC (*CN*) has developed a TGU with power conditional controls, as shown in Figures 8a and 8b. Typical cycle state points for *CN*'s TGU are shown in Figure 9.



Figures 8a and 8b. *CN*'s Integrated Sealed TGU (1m long x .6m wide x .76m), above, and Power Conditioning System (0.8m x 2.8m x2.2 m), below.
Patent issued July, 2015, number 9083212.



The turbine rotor and high-speed generator are hermetically sealed, so the turbine does not require a shaft seal. Two slip streams of the organic working fluid are used to cool the generator, via evaporative cooling and a separate liquid cooling of the generator housing. The generator shaft and rotor are magnetically levitated, so the TGU does not use a gearbox and lubrication system. The power conditioning electrical controls provide a range of voltages (380 Vac to 480 Vac) and frequencies (50 or 60 hz) that are suitable for any site within the United States.



	UA [kWt/K]	kWt	HX Dia (ft.)	HX Length (ft.) & Weight (LBm)	
Evaporator :	15	2061	1.81	6	3,000
Regenerator :	16	636	3.23	6	3,000
Condenser :	87	-1770	3.23	8	5,000
Water jacket + Vapor Gap :		21			

	H 01	H 01	H 02
Pres. psia	14.75	14.39	14.39
Temp. (F)	951.0	951.0	300.6
Enthalpy (Btu/Lbm)	237.8	237.8	75.2
Super Heat Temp. Diff. (F)			
SubCooled Temp. Diff. (F)			
Density (Lbm/ft^3)	0.028	0.028	0.051
Mass Flow (Lbm/s)	12.02	12.02	12.02
Pres. bar,a	1.02	0.99	0.99
Temp. (C)	510.6	510.6	149.2
Enthalpy (KJ/kG)	250.9	250.9	79.3
Super Heat Temp. Diff. (C)		0.0	
SubCooled Temp. Diff. (C)	0.0		
Density (kG/m^3)	0.45	0.44	0.82
Mass Flow (Kg/s)	5.462	5.462	5.462
Nm3/hr	43,462	44,552	24,009

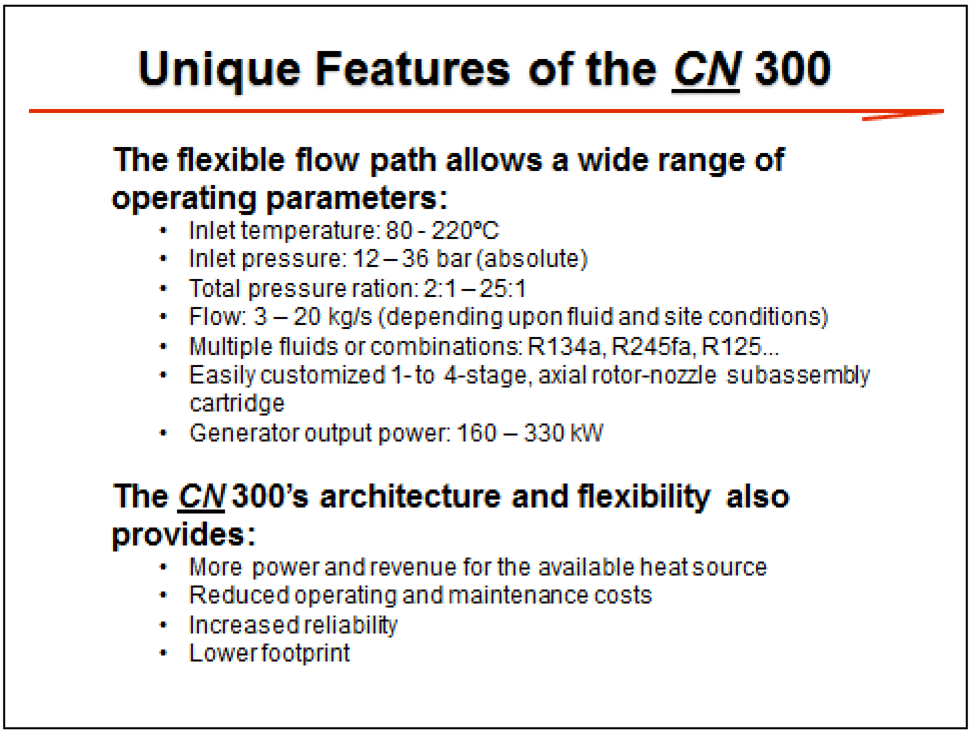
	R 101	R 201mix	R 202	R 203mix	R 301	R 401	R 401A	R 401B	R 501	C101	C201
Pres. psia	425	56.5	55.5	55.5	55	435.5	435.5	435.5	435	14.75	14.7
Temp. (F)	404	306	180	165	123	126	126	126	220	86	93.2
Enthalpy (Btu/Lbm)	253.6	235.9	203.7	200.1	114.8	116.1	116.1	116.1	148.7		
Super Ht.T Diff. (F)	116			37							
Sub. T Diff. (F)					5	164	164	164	70		
Density (Lbm/ft^3)=	7.61	0.96	1.17	1.21	79.05	79.43	79.43	79.43	68.31	62.400	62.400
Mass Flow (Lbm/s)	18.6	18.8	18.8	19.7	19.7	19.7	0.124	0.93	18.6	243.1	243.1
Min. Pipe Diameter (inch)	3	7	7	7	3	3	1	1	3	11	11
Pres. bar,a	29.31	3.90	3.83	3.83	3.79	30.03	30.03	30.03	30.00	1.02	1.01
Temp. (C)	206.4	152.3	82.2	74.0	50.4	52.2	52.2	52.2	104.2	30.0	34.0
Enthalpy (KJ/kG)	588.8	547.7	473.1	464.6	266.6	269.5	269.5	269.5	345.3		
Super Ht T. Diff. (C)	64.5				0.0						
Sub. T. Diff. (C)					2.8	91.1	91.1	91.1	39.0		
Density (kG/m^3)	122	15	19	19	1268	1274	1274	1274	1096	1001	1001
Mass Flow (Kg/s)	8.470	8.526	8.526	8.949	8.949	8.949	0.056	0.423	8.470	110.52	110.52
Min. Pipe Diameter (mm)	77	178	178	178	77	77	26	26	77	280	280

Figure 9. Typical state point diagram for the _CN_ 300.

The _CN_ TGU is rated for 330 kWe. It consists of a close-coupled turbine and high-speed, permanent magnet generator, with magnetic bearings, in a hermetically sealed package. The TGU is an integral unit, operating at 20,000 rpm, without a shaft seal or lubrication required for its magnetic bearings. The turbine gas path employs a modular cartridge design, allowing for cost-effective optimization and modification that can accommodate a wide range of operating conditions and working fluids for each application.

This design makes the system compact, robust, and highly efficient, with reduced maintenance and operating costs when compared to traditional open-architecture ORC systems. Simplicity and remote dispatch capability make the system highly effective at isolated sites where maintenance costs are at a premium. Use of a magnetic bearing and high-speed generator eliminates the lube oil, shaft seal, and shaft seal pressure control system, and reduces operating and maintenance costs, creating a significant reduction over the cost of the contemporary ORC waste heat recovery power systems.

A description of the features of the _CN_ TGU is given in Figure 10.



Unique Features of the *CN* 300

The flexible flow path allows a wide range of operating parameters:

- Inlet temperature: 80 - 220°C
- Inlet pressure: 12 – 36 bar (absolute)
- Total pressure ration: 2:1 – 25:1
- Flow: 3 – 20 kg/s (depending upon fluid and site conditions)
- Multiple fluids or combinations: R134a, R245fa, R125...
- Easily customized 1- to 4-stage, axial rotor-nozzle subassembly cartridge
- Generator output power: 160 – 330 kW

The *CN* 300's architecture and flexibility also provides:

- More power and revenue for the available heat source
- Reduced operating and maintenance costs
- Increased reliability
- Lower footprint

Figure 10. *CN*'s TGU features.

The *CN* 300, as shown in Figure 11, is intended to be mounted directly to the base frame that supports all of the components that make up the Balance of Plant, in a way that is similar to how it is shown Figure 12. Each of the base frame modules contains the necessary vapor generator, regenerator, feed pump, valves, and ancillary subsystems to provide a complete ORC package. A second module (not shown) that contains the power conditioning and control electrical system can be mounted on top of the base frame of the Balance of Plant module.

This module is designed once, eliminating non-recurring engineering expenses for subsequent ORC systems (each with a rating of 330 kWe). Additional modules can be added to the power plant site to increase the total ORC power output when more than 1.5 MWe of power is available from one or more gas turbines or the equivalent waste heat source. In this manner, total system output can be scaled up or down to suit the heat resource, yet the compact package is maintained.

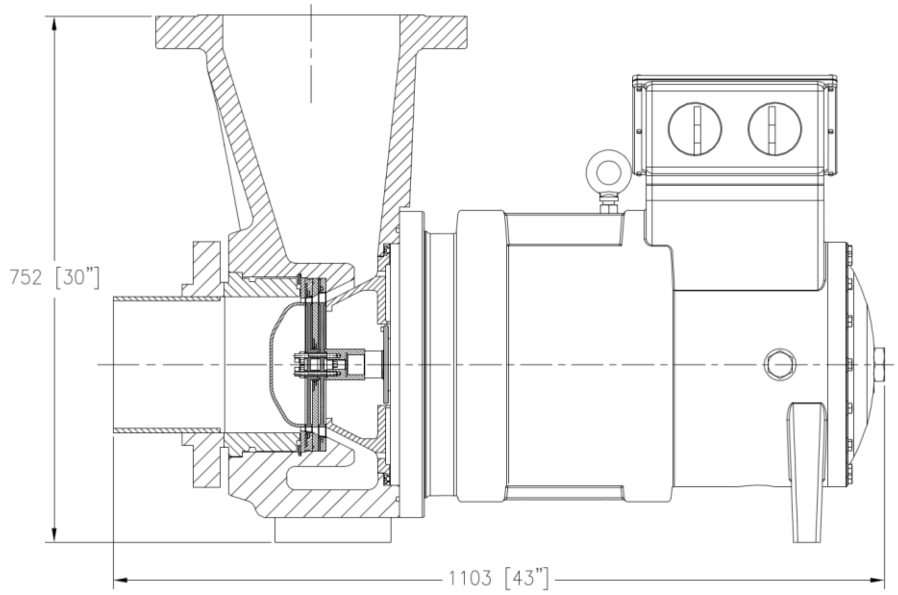


Figure 11. Turbine generator layout, shown with two of four possible impellers. Patent issued July, 2015, number 9083212.

Figure 12. Mounted example of a single 300 kWe ORC module.

A Futuristic Application of Gas Turbines with ORC Waste Heat Recovery

The gas turbine’s high power-to-weight ratio, high thermal efficiency, high ratio of “recoverable” waste heat rejection to fuel consumed at very high temperatures, and high power-to-volume ratio make it possible for gas turbines to find applications in the most power-intensive industries. Several of the more common, industrial applications of large amounts of power were given earlier in this paper, and they include: the liquefaction of natural gas, bio-gas generation and combustion, the combustion of oil field flaring or landfill gas for power generation in GTs and heat recovery, the coal gasification and oxy-combustion technology, and even use with Compressed Air Energy Storage applications, where the stored compressed air is used directly in the combustor of a gas turbine to provide more instantaneous responses to load changes. The introduction of a low-cost TGU that is consistent with all of the beneficial features of a gas turbine, as noted above, enhances the market for smaller 1-3MWe gas turbines.

This author is suggesting an application that is so relatively old that its resurgence may now be considered new: the gas turbine locomotive engine. The first gas turbine prime mover used to drive locomotives in the long distance freight industry was developed in the late 1940s. Application peaked in the early 1960s, after approximately 60 locomotives were manufactured and put into service; at that time, the gas turbine power was evenly split between 3.4 MWe and 6.3 MWe. But few (if any) such locomotives are in use today from this older generation of gas turbine designs—and lack of interest in the use of GTs in locomotives is primarily due to the daily duty cycle, in which the turbine operates at part-load and hence at low thermal efficiency. It is thus useful to consider a combined GT/ORC system that can operate more efficienctly at part-load, to improve power recovery from the ORC system.

An article written in 2013 for RailwayBulletin.com [5] suggested that the use of a more modern gas turbine prime mover for locomotive applications is being considered by General Electric and Caterpillar. The article also mentioned that "…Berkshire Hathaway, Union Pacific and Norfolk Southern have already expressed their interest in the project and actively cooperate with the manufacturers." Although no further announcements have been made, the introduction of a gas turbine for locomotive applications may be an excellent application of smaller gas turbines (3-5 MWe), representing an opportunity to include waste heat recovery using the system technology detailed in this paper. At the very least, the 2013 article encourages the exploration of the data presented in this paper, to determine the specifications for a combined gas turbine/ORC locomotive. The design would use the old 1948-1954 gas turbine locomotive design configuration, depicted in Figure 13, as a starting point to define the constraints that may still be imposed.

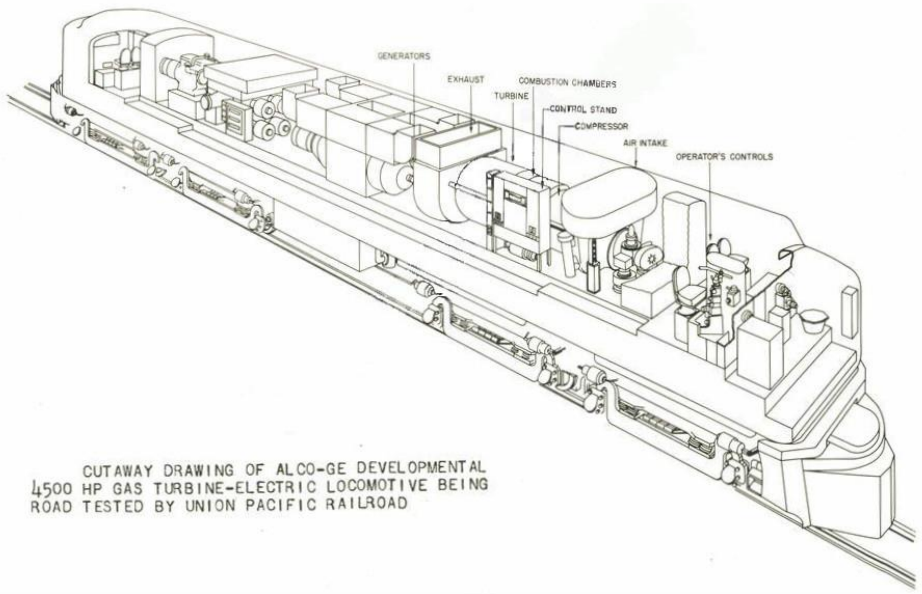
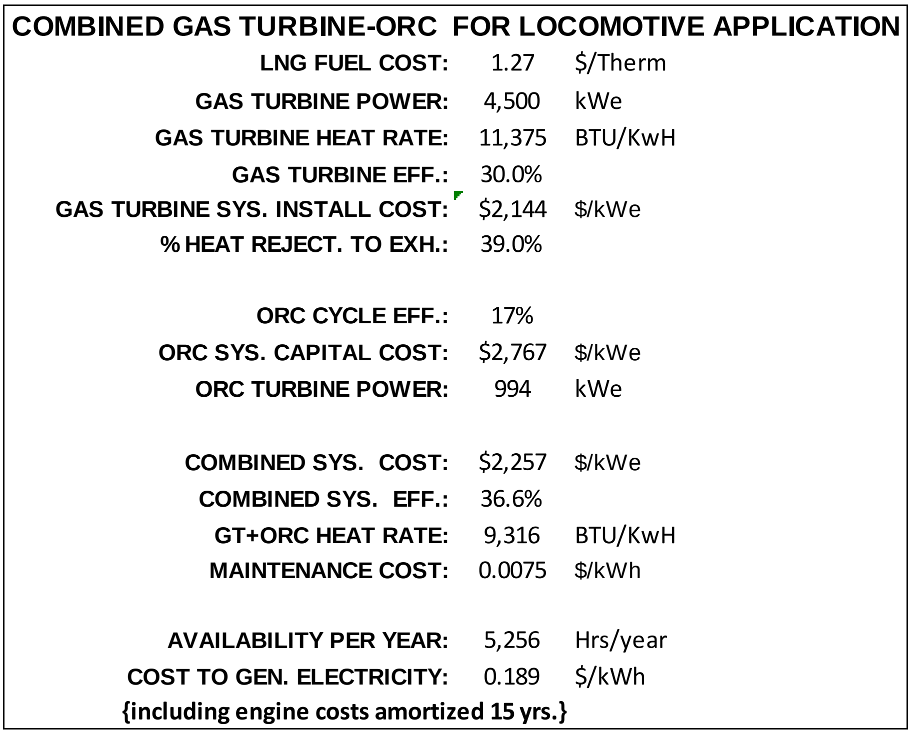


Figure 13. 1948 design of one of the first gas turbine-electric locomotives.

The first specification would include the use of either compressed natural gas or liquified natural gas to improve the economics of the gas turbine, so it would be a dual fuel prime mover. In other words, diesel fuel would provide fuel redundancy. Table 4, on the next page, shows the proposed specification for the gas turbine locomotive/ORC.



COMBINED GAS TURBINE-ORC FOR LOCOMOTIVE APPLICATION		
LNG FUEL COST:	1.27	$/Therm
GAS TURBINE POWER:	4,500	kWe
GAS TURBINE HEAT RATE:	11,375	BTU/KwH
GAS TURBINE EFF.:	30.0%	
GAS TURBINE SYS. INSTALL COST:	$2,144	$/kWe
% HEAT REJECT. TO EXH.:	39.0%	
ORC CYCLE EFF.:	17%	
ORC SYS. CAPITAL COST:	$2,767	$/kWe
ORC TURBINE POWER:	994	kWe
COMBINED SYS. COST:	$2,257	$/kWe
COMBINED SYS. EFF.:	36.6%	
GT+ORC HEAT RATE:	9,316	BTU/KwH
MAINTENANCE COST:	0.0075	$/kWh
AVAILABILITY PER YEAR:	5,256	Hrs/year
COST TO GEN. ELECTRICITY:	0.189	$/kWh
{including engine costs amortized 15 yrs.}		

Table 4. Combined gas turbine/ORC locomotive engine specification.

Conclusion

In this paper, <u>CN</u> has offered a solution to improve the efficiency of small gas turbines, making them more viable in a wider range of industrial energy applications. The integration of an ORC system with a gas turbine can improve the power generation by more than 25% for gas turbines with power ratings below 7.5MWe. Such integrated gas turbine/ORC systems can see increased application in future pipeline compressor stations, particularly when hydrogen gas pipeline compressor stations become necessary. Applications in smaller gas liquefaction and coal gasification plants are also possible, enabling more distributed sites throughout North America and making futuristic gas turbine locomotive applications more economically viable.

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