DEVELOPMENT OF A TURBO-GENERATOR FOR ORC SYSTEM WITH TWIN RADIAL TURBINES AND GAS FOIL BEARINGS

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ABSTRACT

The interest in ORC plant is increasing over recent years in terms of the energy and the environment costs. But the capital cost and maintenance cost of an ORC plant are the main obstacles of the wide spread in the global market. To overcome them, it is necessary to decrease the manufacturing and maintenance cost, and to increase the turbine efficiency and the system availability.

Korea Institute of Energy Research (KIER) and Jinsol Turbomachinery have jointly developed a novel turbo-generator applicable to the low temperature heat sources to meet those needs.

The turbo-generator developed is almost maintenance-free, highly efficient and completely hermetic. A high speed permanent magnet synchronous generator (PMSG) was applied to get the high efficiency. Radial turbines were directly coupled with the PMSG without a gear box to reduce the power transfer loss and cost. The rotor shaft was supported by gas foil bearings to increase the system availability through the non-contacting bearings. Twin radial turbines were assembled with the rotor shaft in the way of face-to-face to cancel out the axial load caused by the pressure difference. This configuration made it possible to apply a gas foil bearing as a thrust bearing despite of its low load capacity. A gas foil bearings is the simple and cheap solution for the rotor support system and the completely hermetic turbo-generator. The high efficiency of the radial turbine was acquired by the real gas modelled turbine design with optimum specific speed.

The developed turbo-generator was integrated with the 100kWe ORC plant installed at KIER and it showed that the turbo-generator efficiency was about 80% as the result of the performance test with the temperature difference of 70℃ between the heat source inlet and the heat sink inlet.

1. INTRODUCTION

In terms of energy issues and environmental aspects, the interest in ORC (Organic Rankine Cycle) plant is increasing over recent years because of its applicability and availability for low temperature heat sources. An ORC has nearly the same components with a conventional steam Rankine cycle except using organic compounds instead of water. The organic compounds applicable to the low temperature heat source (<100~150℃) usually have lower boiling point than the water because ORC plants are applied to generate power from low-temperature heat sources (Quoilin *et al.*, 2013).

Although ORC power plant is considered as one of the promising technologies for generating power form low-temperature heat sources, its high capital cost and maintenance cost are the main obstacles in the wide spread on the global market (Quoilin *et al.*, 2013 and Wang *et al.*, 2011). Mainly the high capital cost and low cycle efficiency cause the low IRR for the ORC power plant. It is necessary to improve the capital cost, maintenance cost, cycle efficiency and system availability to obtain high IRR. In terms of cost, the capital cost of heat exchanges should be reduced at first because it is more than half of the total power plant capital cost and the costs of other components should be reduced secondly. The maintenance cost also should be reduced more. For revenue from the power generation, it is more important to obtain high cycle efficiency (by the components with high efficiency and system optimization) and high system availability (low maintenance shut-down by the robust components) to get more power from the given conditions.

In spite of its drawbacks, ORMAT, Turboden, BNI, UTC, Electratherm, Access Energy and so on have supplied their ORC power plant in the global market. And new manufacturers appear in the market continuously. Most of them select the turbines as their expanders while some others such as Electratherm and Kobelco use the screw type expanders especially in small power capacity plant (Kang, 2012, Takahashi *et al*., 2013, Yuksek and Mrimobin, 2013). Many research studies were conducted with various types of expanders. According to the summary of Fu *et al*. (2015), a scroll expander is dominant in the ORC system with less than 50kW power output. But the reason is that the power capacities of the investigated studies are less than 10kW except some cases.

The expander and the generator are the most important components in the ORC power plant because they convert the electric power from the temperature difference between the heat source and heat sink. To overcome ORC's disadvantages, the expander and the generator must be highly efficient and robust at design point and off-design points. In addition, the expander-generator set must have the advantages such as high system availability, low manufacturing cost, light weight, small volume, easy maintenance, hermetic configuration, and excellent endurance performance.

This study presents the design and development of a novel turbo-generator applicable for low temperature heat sources in 100kW class ORC power plant to meet the needs suggested above. In addition, this paper presents the test results of the 100kW-class turbo-generator under some operation points.

2. DESIGN OF TURBO-GENERATOR

This 100kW power class turbo-generator was integrated to the 100kW class ORC power plant in the Korea Institute of Energy Research (KIER), Korea. R245fa was selected as the working fluid in the ORC system on the basis of the available heat source and heat sink temperature. Although various refrigerants were adopted in the numerous studies (Fu *et al*., 2015, Bao and Zhao, 2013), there exists the proper refrigerant which might be used in the ORC system considering the various aspects such as cycle efficiency, fluid density, safety, and environmental effects.

2.1 Mechanical Layout

The turbo-generator was designed to meet the requirements listed above: high efficiency, high system availability, low manufacturing cost, light weight, small volume, easy maintenance, hermetic configuration, and excellent endurance performance. The turbo-generator was suggested to adopt the radial in-flow turbine, permanent magnet synchronous generator, gas foil bearing, and direct coupling between turbine and generator. The mechanical arrangement was that the twin radial in-flow turbines are located in the front-to front way and the generator is located between them as shown in Figure 1.

By selecting the radial in-flow turbine, the manufacturing cost can be reduced with the high efficiency. And compact size and light weight can be obtainable. The high rotational speed of turbine is inevitable to meet the optimum specific speed but this high speed makes the turbo-generator compact and small.

Due to the high speed, the high speed permanent magnet synchronous generator (PMSG) and the low speed generator with a gear box can be the candidates. For this turbo-generator, the PMSG was selected as a generator. It has higher efficiency than other type generators, light weight, and small volume due to the high rotational speed. It can rotate at various speeds regardless the grid power frequency. That makes the radial turbine rotate at its optimum speed with the maximum efficiency. But its small volume has small surface area for its cooling. It makes hard to design the cooling system in spite of its low heat generation due to its high efficiency. Also it needs power converting system (PCS) to convert the generator side power-generating-frequency that depends on the rotational speed of generator to grid side one. But the power converting efficiency is higher than that of a gear box. PCS does not need the lubrication oil, an oil cooling system, and its maintenance such as oil change. It is more attractive to adopt the PMSG and PCS in the turbo-generator for ORC power plant.

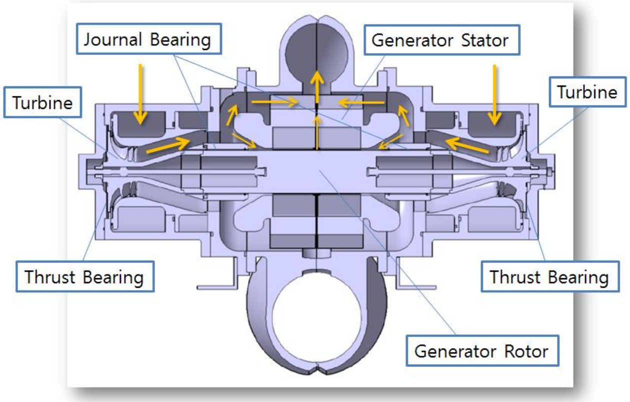


Figure 1: Mechanical Layout of the 100kW turbo-generator for ORC plant

A gas foil bearing was applied in the rotor support for the journal bearing and the thrust bearing. It is the gas foil bearing that is one of the non-contact bearing and oil free bearing. It makes the maintenance-free turbo-generator possible due to its special characteristics mentioned above. A magnetic bearing also has these characteristics but it has many complicate components (its own exclusive controller, position sensors and so on) and more power consumption. It may increase the cost of the ORC plant.
But the gas foil bearing has small load capacity. While it is not the problem in journal bearing, it can be a severe problem in thrust bearing because it should have the load capacity for the axial load caused by the pressure difference between the front side and back side of the radial turbine.

Table 1: Design requirements of radial turbine

Parameters	**Unit**	**Values**
Inlet Temperature	℃	59.8
Inlet Pressure	bar	4.56
Expansion Ratio	-	4.67
Mass Flow Rate	kg/s	2.5 (for each turbine) (total: 5 for twin turbines)
Output Power	kW	58.5 (for each turbine) (total: 117 for twin turbines)
Efficiency	%	85
Working Fluid	-	R245fa

As shown Figure 1, the turbine and the generator's rotor were coupled directly to skip the gear box and its oil handling system. And the twin turbines were arranged in the front-to front formation. It makes possible to cancel out the axial load caused by the pressure difference in each single turbine.

Theoretically the axial load caused by the turbine is zero, which allowed us to use the gas foil bearing as the thrust bearing. The thrust bearing was located at the back face of each twin turbines as shown in Figure 1. In addition, the generator was located between the twin turbines' exits. The relatively cool working fluid can flow into the generator from the turbine exit after expansion. It can cool down the generator by the cooling passage shown in Figure 1. It makes possible to omit the other cooling passage and to save the loss of working fluid for the cooling passage. Table 1 shows the design requirements of radial turbine on the basis of ORC plant cycle design.

2.2 Radial Turbine and PMSG

The radial turbine was designed on the basis of the thermodynamic properties of working fluid, R245fa to meet the specification shown in Table 1. The stage efficiency of the radial turbine was expected as high as 90% with Reynolds Number effect corrected. The rotational speed of the turbine was selected as an optimal value which corresponds to the optimal specific speed. The main flow sections of the turbine stage, i.e. rotor exit, rotor inlet were designed on the basis of optimum velocity triangles which makes the rotor exit relative flow velocity at the tip radius and the rotor inlet absolute flow velocity as minimum respectively and no swirl at the rotor exit. The minimum velocity design minimizes the aerodynamic losses and no swirl at the outlet of turbine rotor makes the radial component of the exit velocity negligible (Aungier, 2006). The dimension of the main flow path is shown in Table 2 for the designed radial turbine.

Table 2: Dimension of the main flow path from the design results of radial turbine

Items	Unit	Values
Stator Inlet Radius	mm	105
Stator Outlet Radius	mm	82
Stator Blade Height	mm	10.6
No. of Stator Blade	-	20
Rotor Inlet Radius	mm	76
Rotor Inlet Blade Height	mm	10.6
Rotor Outlet Tip Radius	mm	54.6
Rotor Outlet Tip Radius	mm	20
No. of Rotor Blade	-	14

The wall contour of the turbine rotor and the blade profiles were generated with the proprietary in-house design code on the basis of the main flow path shown in Table 2 and the real gas model of R245fa using REFPROP developed by NIST. The final radial turbine is shown in Figure 2. Figure 2 (a) shows one of the twin turbine rotors and Figure 2 (b) shows one of the twin turbine stators.

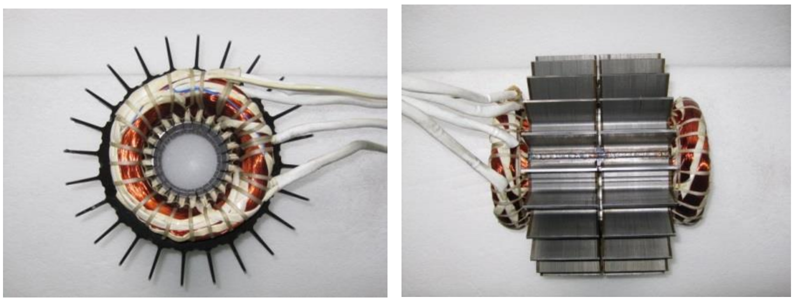
(a) (b)

Figure 2: Radial Turbine Rotor (a) and Stator (b)

The generator was designed in the type of PMSG to obtain higher efficiency. A type of the rare earth magnet was used to get the strong permanent magnet. A samarium-cobalt magnet (Sm_2Co_{17}) is preferred as the rotor magnet for PMSG because of its higher demagnetized temperature than others. Due to this, it can operate at higher temperature. The design specification and design results are shown in Table 3. The stator core was designed and constructed to have several cooling fins along the outside of it extended in the radial direction as shown in Figure 3 (a) to dissipate the heat generated by the iron loss and copper loss of the stator. It has also cooling passage in the middle of the stator core along the axial direction shown in the Figure 1 and Figure 3 (b). The cooling passage in the middle of the stator core is mainly the one for rotor shaft cooling, which can dissipate the heat from the can loss and the permanent magnet loss of the rotor shaft.

Table 3: Design specification and its result of PMSG

Items	Unit	Values
No. of Phase	-	3
No. of Poles	-	2
Rated Voltage(line-to-line)	V	480
Rated Current	A	137
Rated Power	kW	110
Efficiency	%	95
No. of Slots in Stator	-	24
Stator Outer Radius	mm	220
Stator Inlet Radius	mm	89
Rotor Radius	mm	85
Length of the Rotor Magnet	mm	150



(a) (b)

Figure 3: Stator for the PMSG Type Generator

2.3 Gas Foil Bearings

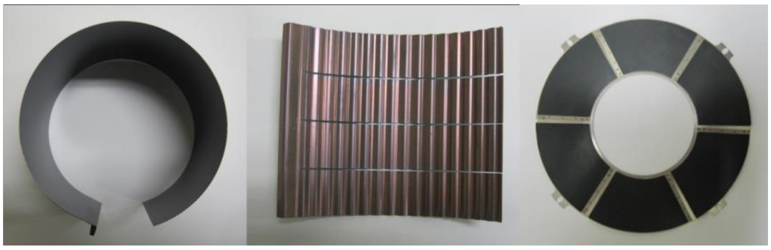
Twin turbine rotors were assembled at each end of the generator rotor shaft which was the part of the turbo-generator rotor to obtain the face-to-face configuration. The whole rotor was supported by the journal bearings and the thrust bearings as shown in Figure 1. The journal bearings support the radial load caused by the rotor weight and the thrust bearings support the axial load caused by the pressure difference between turbines, respectively. According to the literatures, the load capacity of a journal foil bearing is given by Equation (1) (Kus and Neksa, 2013).

$$C_{JB}=f_{JB}L_{JB}D_{JB}^{2}N \tag{1}$$

For the thrust bearing, the axial load on the rotor shaft could be ideally cancelled out to be zero because the rotor shaft is bilaterally symmetric. Therefore the axial load on the rotor is not significant. The thrust load capacity that can be supported by a thrust foil bearing, also is given by Equation (2) (Kus and Neksa, 2013).

$$C_{TB}=f_{TB}\pi w D_{TB}^{2}N \tag{2}$$

According to the in-house design practice, the journal and thrust bearings were designed and made as shown in Figure 4. Figure 4 (a) is the top foil of the journal bearing and Figure 4 (b) shows the bump foil of the journal bearing placed between the top foil and the bearing housing. Figure 4 (c) is the thrust bearing located at the backside of each twin turbine.

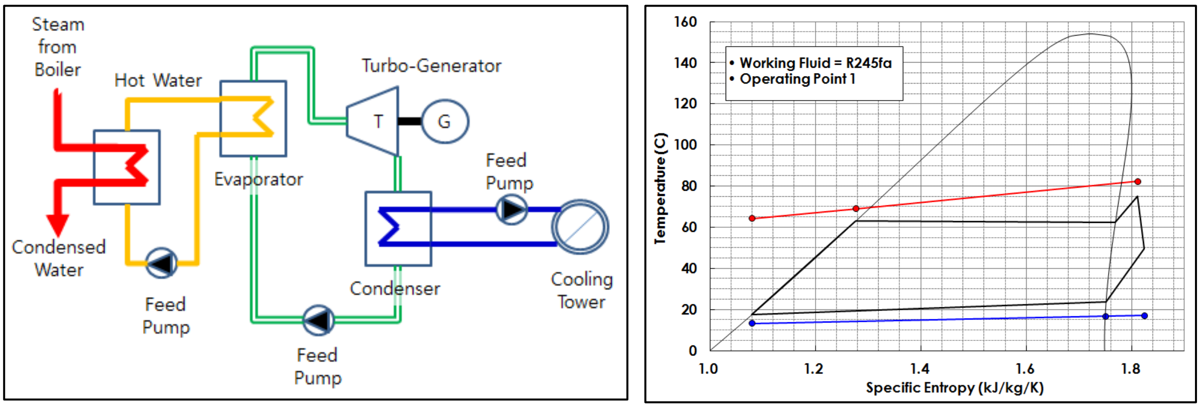


(a) (b) (c)

Figure 4: Gas Foil Bearing: Journal Bearing (Top Foil: a, Bump Foil: b) and Thrust Bearing (c)

3. TEST RESULTS AND DISCUSSION

Figure 5(a) shows the experimental apparatus for the turbine performance test. Actually the whole ORC power plant system was required to run the turbo-generator and to test its performance by the electrical power output from the generator. Then, the performance of the turbo-generator could be evaluated.



(a) (b)

Figure 5: ORC power plant as a turbo-generator performance test rig

The orange line loop in Figure 5(a) is the hot water loop heated by the steam from the boiler as a heat source. The green line loop is the working fluid (R245fa) loop. The blue line loop is the cooling water line to absorb the heat rejection from the condensation of the working fluid. Steam line (red) and hot water loop (orange) were installed to simulate the hot-water type heat source. Figure 5(b) shows a plot of thermodynamic cycle in the temperature entropy diagram (T-S diagram) for an operating point
Table 4 together with Figure 6 (a) show the operating conditions of the ORC power plant and the electrical power output at each operating point. The performance of the turbo-generator was evaluated on the basis of this power output. The mass flowrate of the heat source (hot water) was set to a constant value, 17kg/s. And the temperature difference between the heat source inlet and the heat sink inlet was also controlled to a constant value, about 70℃. Since the temperature difference was nearly

constant, the rotational speed of turbine was also kept nearly constant within 6% deviation. Figure 6 (a) also shows that the electrical power output was proportional to the mass flow rate of the working fluid.
The electrical power output was measured at the power output terminal of the generator. The (gross) cycle efficiency which is defined in Equation (3) increased from 5.2% to 7.3% while the mass flowrate increased. That means 40.4% increase along the flowrate increase. The efficiency of the turbo-generator means that of the total equipment composed of the turbine and generator. It is shown in Equation (4) that the efficiency is the ratio of the electrical power output to the ideal power output from the turbine.

Table 4: Electrical power output and control inputs at each measured operation point

Operating Point	Electric Power Output (kW)	Control Inputs			
		Working Fluid Flowrate (kg/s)	Heat Source Inlet Temperature (℃)	Heat Sink Inlet Temperature (℃)	ΔT (℃)
1	91.20	5.17	82.32	13.17	69.14
2	81.50	4.71	82.78	12.48	70.30
3	70.43	4.16	82.85	11.62	71.23
4	60.34	3.69	81.77	10.94	70.84
5	48.93	3.15	82.52	10.27	72.26
6	35.17	2.60	81.93	10.15	71.78

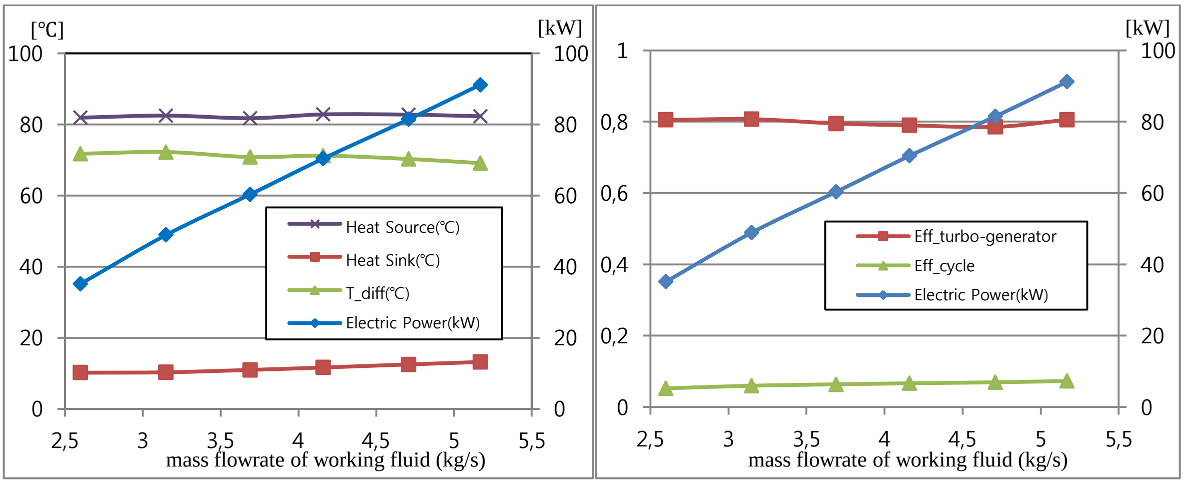


Figure 6: Operation conditions and performance results

It was difficult to measure the temperature and pressure right after turbine exit because of the unique configuration of this turbo-generator. The temperature and the pressure were measure merely near the end-turn of the generator. Assuming that the pressure drop between them is negligible, ideal enthalpy drop can be calculated from the turbine inlet and outlet conditions. But the temperature difference was not negligible because of the temperature rise due to the generator loss and windage loss. Therefore only the turbo-generator efficiency was evaluated instead of the turbine efficiency. The calculated efficiencies of the turbo-generator ranged from 78.5% to 80.7% as shown in Figure 6. It matched well with the product of the design efficiency of the turbine and generator (0.85*0.95) without considering the windage loss.

$$\eta_{cycle} = \frac{P_e}{\dot{Q}_{in}} \tag{3}$$

$$\eta_{t-g} = \frac{P_e}{\Delta h_i \cdot \dot{m}_{ref}} \tag{4}$$

$$\eta_{t-g} = \eta_t \cdot \eta_g \cdot (1 - \xi_{wl}) \tag{5}$$

As shown in Equation (5), the efficiency of the turbo-generator is the product of the turbine efficiency, the generator efficiency, and the efficiency decrease due to the windage loss of the rest part of the rotor except the generator shaft. If the generator is designed with the specified efficiency (95%) and the windage loss is ignored, the turbine efficiency was 85% in the case of the maximum turbo-generator efficiency (80.7%).

4. CONCLUSION

The present study was the design and construction of a novel turbo-generator applicable for the low temperature heat sources to meet the market needs. It is almost maintenance free, highly efficient and completely hermetic. A unique configuration was developed including high speed permanent magnet synchronous generator (PMSG), twin radial turbines, and gas foil bearings. Twin radial turbines were assembled to the rotor shaft in the face-to-face configuration.

The performance test results of the turbo-generator showed that the maximum power output was 91.2kW. The maximal efficiency of the turbo-generator and cycle efficiency were 80.7% and 7.3%, respectively under the condition of 70℃ temperature difference between heat source and heat sink.

And the future study will focus on the clarification of the turbine and generator efficiencies, cost estimation compared with the other types, turbine efficiency change with respect to the operation points, and dynamic response of turbo-generator.

NOMENCLATURE

C	load capacity	(N)
D	diameter	(m)
f	bearing performance coefficient	(N/m^3/krpm)
L	axial length	(m)
$\dot{m}$	mass flow rate	(kg/sec)
N	rotor speed	(krpm)
P	power	(kW)
$\dot{Q}$	heat flow rate	(kW)
w	radial extent of the top foil	(m)
ξ	energy loss coefficient	(-)
η	efficiency	(-)
Δh	enthalpy difference	(kJ/kg)

Subscript

JB	journal bearing
TB	thrust bearing
t-g	turbo-generator

t	turbine
g	generator
e	electric
i	ideal
ref	refrigerant
wl	windage loss

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