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Research Article

Preliminary Design, Simulation and experimental analysis of 1.5 kW Organic Rankine Cycle (ORC) Powered Turbo Expander

Abhijeet Chaudhari[†*], A.K Karthikeyan[‡] and V. V Ganore[†]

[†]CO_2Research and Green Technology Centre, SMBS, VIT University, Vellore- Tamil Nadu, India-632014
[‡]School of Mechanical and Building Science, SMBS, VIT University, Vellore- Tamil Nadu, India-632014
[†]School of Electrical and Electronics Engineering, SENSE, VIT University, Vellore- Tamil Nadu, India-632014

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Abstract

Recovery of waste heat from energy conversion or usage of renewable energy to cut down the fossil fuel consumption which in turn leads to environmental difficulties like Global temperature rise, harmful effects on ozone layer, rapid rise in air pollution and smog is the urgent action required. Hence development of new energy conversion techniques which will efficiently utilize the energy resources for power generation without polluting environment is need of an hour.Typical low-grade heat resources like solar heat, geothermal energy and waste heat such as industrial processes, power systems, ICE, solar collector and OTEC are perfect candidates for new energy resources as their temperature ranges from 60 to 200°C. Energy source generation using low grade heat recovery can be studied using power turbine cycles. Invention of lower boiling point organic fluids has provided Rankine cycle with low temperature heat sources. ORC has been proved to be a very promising technology to utilize the low grade heat. The research is aimed at exploring the opportunity of realizing, with low economic cost, heat recovery system with the following characteristics: Evaporation temperature of 150°C-200°C, suitable for industrial heat recovery., Condenser cooling by ambient temperature water, which can be further used for other applications such as process heating and Development of low cost organic fluid turbine for small scale heat recovery applications.

Keywords: *Organic Rankine Cycle (ORC), Thermal Efficiency, de Laval Turbine, Toluene*

1. Introduction

Energy recovery and development strategies address energy sustentation and emission reduction. Due to the rapid globalization and exhaustion of conventional fuel resources considering their influence on ecosystem, it has become necessary to switch the energy dependency towards renewable resources rather than conventional energy sources. It is also necessary to use the available energy more effectively. Most of the energy is consumed in industrial sector, out of which more than 50% of the fuel energy is lost to the environment in the form of waste heat. Hence, it is important to recover this low temperature heat from industrial exhaust.

There is a huge potential to recover industrial waste heat at low temperatures typically below 200°C. The classification of waste heat sources is done based on the temperature range: High temperature waste heat (>650°C), medium temperature waste heat (230-650°C) and low temperature or low grade heat (<230°C). Most of the statistical analysis shows that out of the total waste heat liberated by industries, low temperature heat lost is beyond 50%. In the current era, the Organic Rankine Cycle (ORC) system, which is the modified version of steam Rankine cycle which operated with different working fluids,has proved to be the effective solution for "waste heat or waste energy recovery". The use of organic fluid instead of steam/water allows ORC to transform low temperature heat into electrical power. Most of the ORC systems in the current era are mostly developed for heat recovery from different heat sources which are available at higher power ratings i.e IC engines, Gas turbine exhaust, Geothermal sources, Furnace industries, chimney exhausts, etc.). The power production capacity of such sources ranges from 50 kW to 500 kW. The ORC systems for small rated capacity, as well as for small IC engines typically Cars or low capacity heat exchangers confined. Also the very limited mechanisms that have been matured for narrow rated systems, on the other hand comes with distinct expanders (scroll, screw, etc.). These types of expanders are either not suitable in small or the confined areas like endogenous parts of a ordinary car or cannot be adopted in such operating conditions.

Selection of a specific type of turbine for a particular operation highly depends on the maximum

*Corresponding author: **Abhijeet Chaudhari**

efficiency under which the turbine can operate at the given operating conditions. Also for the small scale operations, cost effectiveness is the major criteria which cannot be neglected. For micro-CHP systems with capacity of 1-10 kW power, vane and scroll type expanders are better choice (Qju, *et al*, 2011). The efficiency of ORC systems also follows weak function of evaporation temperature and critical temperature of working fluid (Liu, *et al*, 2004)

The aim of this work is to carry out the preliminary design of an expansion device which is able to meet distinct system requirements such as low mass flow rates, low operating pressure, low temperature and small size with less manufacturing complexity and low cost. In this paper, the thermodynamic analysis of ORC system for the given temperature at particular mass flow rate as well as identification of operating parameters are presented. Along with the ORC system analysis, the major focus of the work is on the development of Turbo-Expander for ORC and comparison of analytical results with experimentally obtained values. The main focusis to develop, propose and commercialize a possible design procedure for ORC system and to cover the gap between small scale power range ORC energy recovery system (1 kW- 10 kW) compact enough for recovering heat at small applications.

2. Organic Rankine Cycle analysis

Fig.1 represents the T-s diagram of proposed ORC system with toluene as working fluid.The stage 1-2 represents work done by the pump,2-3' is the heating of organic working fluid in the evaporator, 3'-3 represents the phase change of working fluid, 3-4 shows the expansion in turbine and 3-3a shows the isentropic expansion of working fluid at imposed pressure.

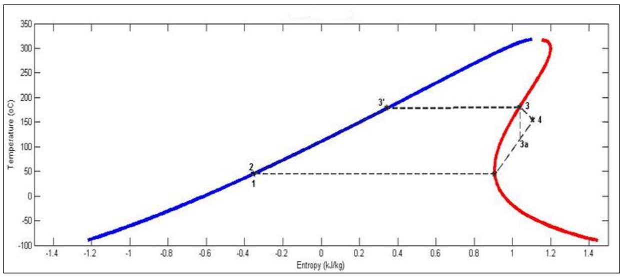

Fig.1 Temperature-entropy diagram for ORC

In order to start the design procedure, it is important to fix the heat source and heat sink temperatures. The aim of the present research is the design, construction and testing of ORC system focusing on the following specifications:

- 1.5 kW electrical power output
- About 180°C evaporation temperature
- About 45°C condensation temperature

In order to avoid the complexity and to reduce the equipment sizing, a direct configuration has been chosen. The heating cycle will consist of a boiler to supply the heat to the thermic oil. ORC working fluid takes the heat directly from the oil through the heat exchanger placed into it.

Heat sink will be a water loop connected directly to the tap to circulate around the condenser coils basically a helical coil heat exchanger.

Before carrying out the thermodynamic analysis of the ORC systems, following assumptions are made:

- Expander efficiency is assumed as 60%
- Pump overall efficiency is assumed to be 65%
- Heat loss through the piping and other equipments is assumed to be negligible.
- The system is to be designed for a heat source at 200°C.
- Mechanical losses in all the rotating equipments are considered to be negligible.
- Cooling water temperature is assumed to be 27°C.

To analyze the ORC system performance the simulation based on steady state condition has been performed using Cycle Tempo software. The design results are validated with the analysis results from the software. The software has been developed by ASIMPTOTE(Advanced Simulation for Power and Total Energy systems). The simulation kit allows rigorous energy and mass balances for a wide range of processes.

The first approximations were very important as they allowed setting and fixing some operating parameters for each component, which are the general and fundamental requirements for the thermodynamic analysis of the cycle.

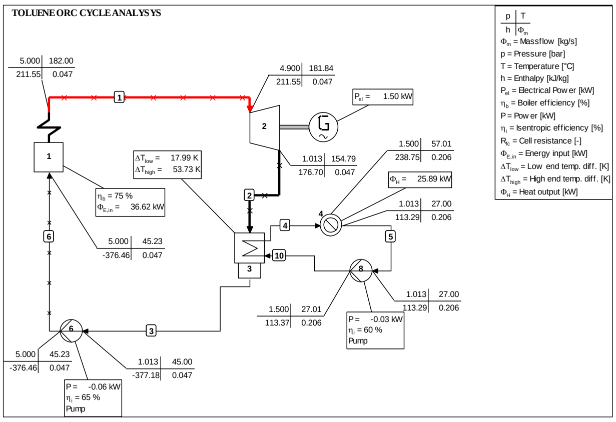

Fig.2 Thermodynamic simulation result from cycle tempo software

In order to verify the analytical results, following correlations considering thermodynamic stateswere applied on the data obtained from experiments. The experimental data and the values obtained from the cycle analysis were closely matched and satisfactory results were also obtained from the same.

Evaporator heat input was calculated using Eq. (1)

$$Q_{in}(kW) = \dot{m}_{wf}(h_3 - h_2) \quad (1)$$

Isentropic efficiency for the turbine is given by Eq. (2)

$$\eta_{is} = \frac{h_3 - h_4}{h_3 - h_{3a}} \times 100 \quad (2)$$

The net power output from the ORC system is calculated by Eq. (3)

$$P_{net} = P_{Generator} - P_{Pump} \quad (3)$$

Where,

$P_{Generator}$ is the power produced by electrical generator, and

P_{Pump} is the power consumed by organic feed pump.

Thermal efficiency of cycle is given by Eq. (4)

$$\eta_{th} = \frac{GeneratorPower\,Output(kW)}{Q_{in}(kW)} \times 100 \quad (4)$$

Net cycle efficiency is obtained from Eq. (5)

$$\eta_{net} = \frac{Net\ Power\ Output(kW)}{Q_{in}(kW)} \times 100 \quad (5)$$

Fig.3 shows the variation of Gross efficiency and the Net efficiency with increase in the evaporation temperature of working fluid. It is clear from the results that the efficiency drops with the increase in evaporation temperature of working fluid. This is because the load on condenser increases as the turbine cannot draw the maximum enthalpy and high temperature fluid is condensed in the condenser. In order to reduce the working fluid temperature, the cooling water pump has to supply water at an increased mass flow rate, thus consuming more power which causes the net and gross efficiency to drop.

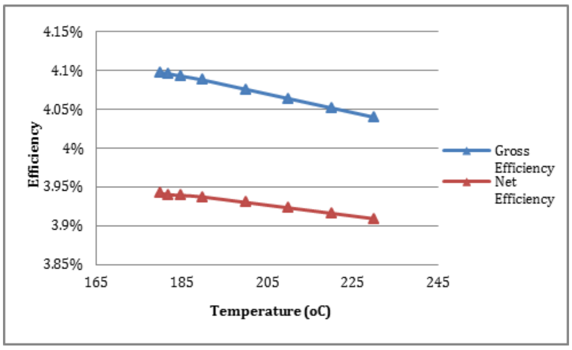

Fig.3 Variation of Gross and Net efficiency (Energy) of ORC system with increase in evaporator temperature

The Fig.4 shows that exergy Gross and Net efficiency also drops with the increase in evaporation temperature.

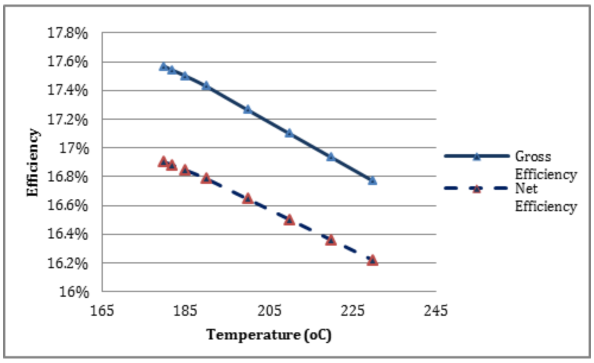

Fig.4 Variation of Gross and Net efficiency (Exergy) of ORC system with increase in evaporator temperature

The drop in efficiency is not much, but in case of small capacity systems this drop in efficiency can hamper the payback period of power plant.

3. Preliminary Design of Organic Turbo-Expander

After the feasibility analysis and thermodynamic simulations of ORC system, the next tread is to move towards the expansion device.Preliminary design of turbine is based on the mean-line flow concept. It is the 1-D approach with the assumptions that flow through blade passages (considering the midspan values) is uniform and unidirectional. The mean-line model consideration allows easy and simplified examination of how to categorize the thermodynamic process and the fluid-dynamic parameters over the entire flow region; because of this fact, the preliminary design is the most vital step of the whole procedure. The most suitable turbine for the ORC application at small scale applications is an Inward Radial flow turbine. The major drawback of this design is the manufacturing complexity and the cost. As the major focus is to develop low cost and high efficiency turbine, the de Laval turbine is selected for this purpose.

The geometrical profile of expander can be determined by using the concept from specific speed. Low area of transition of turbine denotes the low specific speed and vice a versa. The specific speed is the indication of maximum efficiency that can be reached by the turbine. During the initial phase of design, following losses associated with the turbine are considered (Fig.5).

- Losses occurred in stator
- Impeller Losses
- Losses at the tip due to clearance (gap between stationary wall of turbine and impeller)
- Leakage losses between seals
- Kinetic energy losses at the exit of turbine

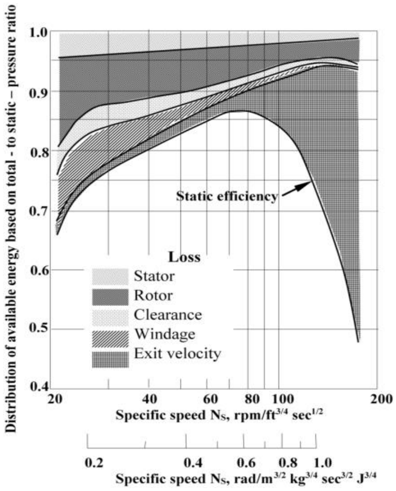

Fig.5 Specific Speed of Turbine Vs. Losses

The design adopted in this case is the straight non-condensing configuration. The turbine output is a function of initial working fluid conditions, mass flow rate of working fluid and pressure at the exhaust of turbine. The output power in such configuration is limited by the process demand, until the external demand is artificially created by the use of vapour vent at the exhaust.

Table 1 Input data for mean-line modeling of turbine stage

Boundary Conditions	Geometrical Constraints	Design Variables
Inlet/Outlet T-P conditions	Nozzle inlet angle	Blade inlet angle
Mass flow rate	Minimum blade thickness	
Stage expansion ratio	Stator-Rotor clearance	Angular speed
Power Output	Minimum blade height	

Table 2 Inlet operating conditions for turbine design

Nozzle inlet temperature T_1, °C	182
Nozzle inlet Pressure P_{01}, bar (gauge)	5 bar
Inlet mass flow rate ṁ, kg/s	0.055
Vapour inlet density ρ, kg/m³	13.819

The flow conditions at inlet are known i.e density and mass flow rate (ṁ). Neglecting the dynamic enthalpy, the peripheral velocity at the inlet (u) can be obtained from

$$u = \sqrt{\frac{h_o - h_2}{\psi_1}} \tag{6}$$

With the value of u, rotor inlet diameter can be calculated as

$$d = \frac{2u}{\omega} \tag{7}$$

The rest of parameters can be computed from initial α_1 and the velocity triangle of the turbine.

Table 3 Design parameters of ORC Turbine

S. No	Parameters	Values
1	Rotation speed N, RPM	0-4865
2	Rotor blade velocity factor	0.85
3	Nozzle velocity factor ϕ	0.89
4	Nozzle outlet angle α_1	22
5	Mean diameter d, mm	138
6	Blade height l_1, mm	15
7	Blade length l_2, mm	27
8	Number of Nozzle	2
9	Number of rotor blade	15

The geometric design shown in Fig.6 also depicts the velocity triangle for analytical design of turbine. The optimal nozzle angle value is adopted from ROHLIK: α_1=22°.

The inlet angle for the rotor is computed as 36°. The blade profile is symmetric about the central axis which facilitates easy manufacturing; though the velocity at the outlet of blade is high there is loss in kinetic energy.

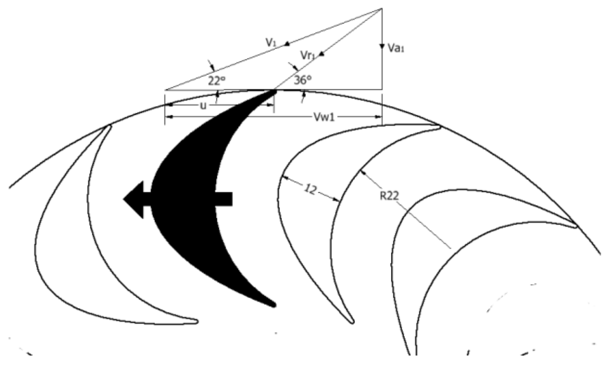

Fig.6 Turbine geometry and velocity triangle

4. Thermo-Structural Analysis (FEM) of Turbine

The next step after completion of the design of turbine rotor, a model both in 2-D and 3-D was made. The 3-Dimensional model of turbine blade was developed using commercially available cad product (Autodesk Inventor®, Autodesk, Inc.). The blade profile was adopted from NACA (National Advisory Committee for Aeronautics) profile. After completing the 2D drawing, the mesh was generated which is composed by triangular elements.

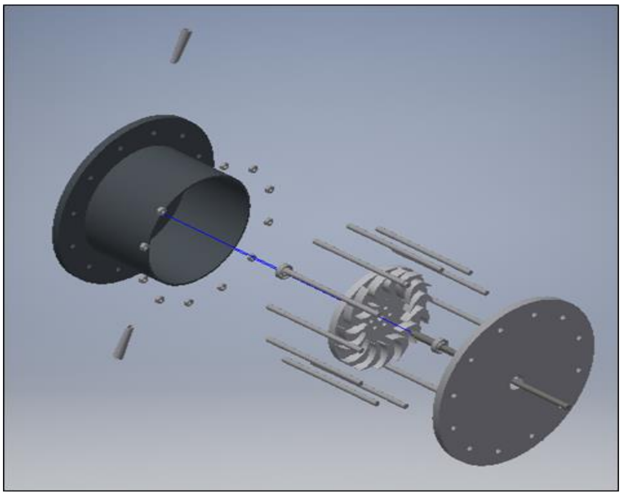

Fig.7 3-D sketch for Turbine assembly

During the preliminary thermo-structural analysis, the simulation was segregated into: simple structural and static stress, heat or the thermal load on turbine (thermal stress) and the last one, a global stress. The global stress represents the summation of the previous ones. The turbine was subjected to a rotational speed of 4865 RPM with rotation axis coinciding with geometric axis of turbine. The only constraint applied is the cylindrical support which restricts the axial movement of turbine and allows the radial movement of turbine.

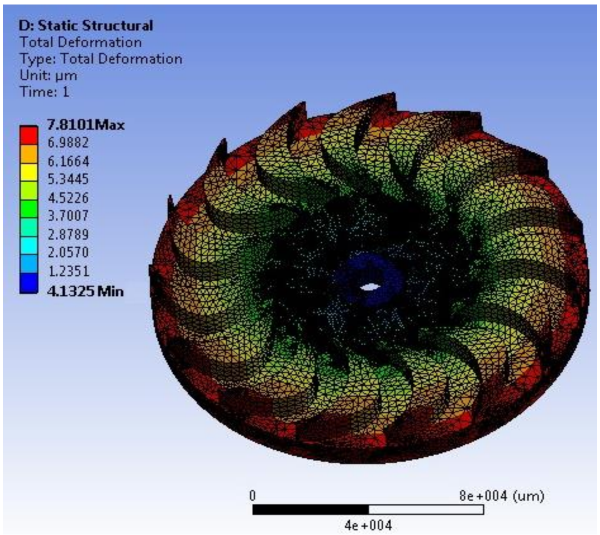

Fig.7 Deformation under applied pressure

The material used under these considerations is Stainless Steel, though in actual manufacturing the material used is SS-304, with slightly superior qualities than standard Stainless steel. The results obtained from the structural simulation clearly indicate that this material is suitable for building the rotor.

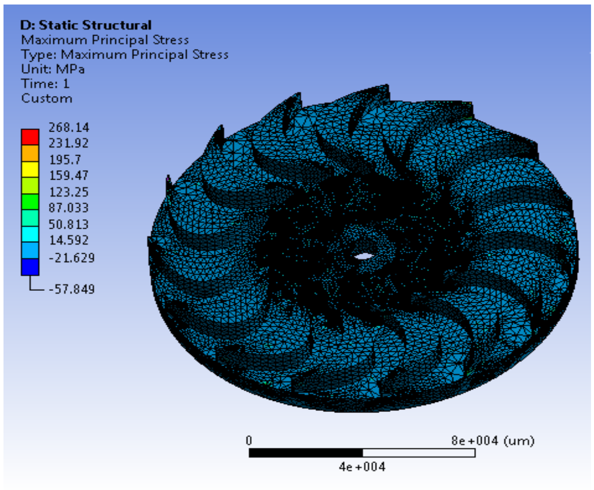

Fig.8 Maximum Principal Stress on turbine

5. Fabrication and experimental analysis of ORC Turbo-Expander

Fig.7 presents the 3-D layout of proposed Turbo-expander. During the manufacturing process it was ensured that tolerance value was under ∓ 0.05 mm. In order to prevent the leakage, VITON seals were used to seal the bearing end of turbine. The nozzles were made of brass material and brazed to the turbine casing. Another important learning was most of the leakage occurred at the sealing ends and also steel tubes and fittings prove to be easy and suitable to use for low rated systems. In order to ensure the leakage free fitting of turbine, the tests were carried out on compressed air.

The pressure was varied from 300 kPa to 800 kPa maintaining the mass flow rate constant. The air was heated to 100°C in order to increase inlet temperature to the turbine. It was identified that increasing the inlet temperature and pressure, the efficiency of turbine is slightly increased. Thus it shows that, through the nozzles, expansion of air causes the net increase in volume flow rate and increase in power output. The maximum power produced by the turbine reached 1.08 kW at 10 bar and 90°C. Almost same power was achieved at 10 bar and 100°C.

Fig.8 Turbine geometry after machining

6. Results and discussion

As expected, the rotor tip of blade experiences more stress, which is mainly due to action of all the forces acting collectively on it, i.e centrifugal force and the thermal stress. In order to determine the safety conditions, the parameter called safety factor (Ratio of Yield strength to maximum equivalent stress) has been determined. The results are achieved satisfactory and the obtained value is within the limit conditions.

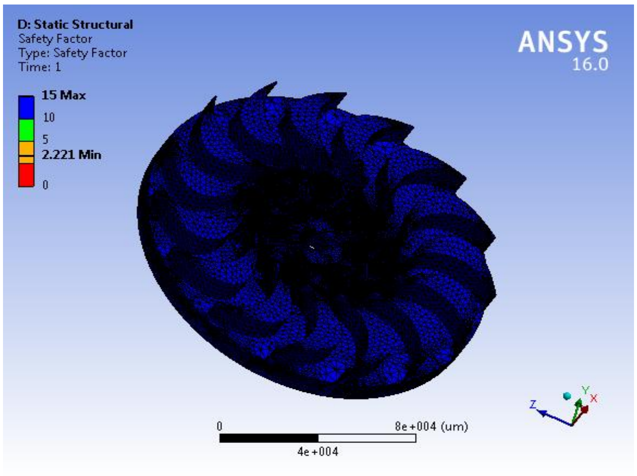

Fig.9 Safety factor for equivalent stress

After the experimental setup was fixed, the system was operated on air at varying pressure. The turbine shaft was connected to 12V DC alternator capable of producing 65 Amps. The DC load bank of 15 bulbs with capacity of 90 W and 100 W each was connected to the alternator. The flow rate was adjusted with the help of flow control valve and a rotameter was connected across the piping. Fig.10 presents the efficiency of turbine with increase in pressure and varying temperature. The maximum efficiency achieved by the turbine was recorded to be 79.82% at 10 bar and 90°C.

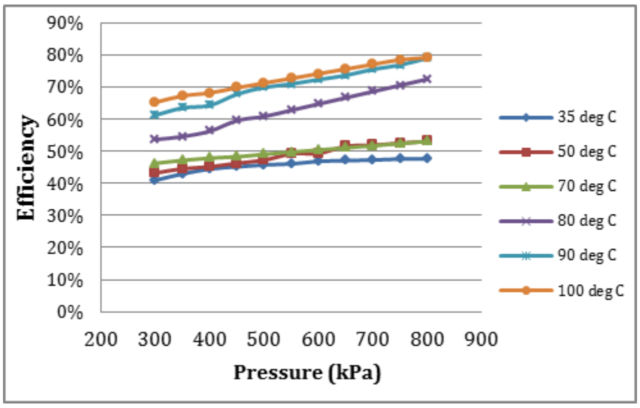

Fig.10 Effect of pressure and temperature on turbine efficiency

The efficiency plot shows that higher efficiency is achieved at elevated pressure ratio. This is because the heat is added at faster rate to the working fluid. We can also state that maximum efficiency is not always correlated with maximum power output, as maximum efficiency operating points and maximum power output locations cannot be same for all the conditions. The minimum efficiency was achieved at 300 kPa and 35°C air temperature.

Conclusions

In the given work, we have successfully developed, simulatedand verified 1.5 kW ORC Turbo-Expander. The experimental analysis was done with the help of working fluid as air instead of ORC fluid. The system was able to generate a net of 1.08 kWe power output with efficiency of 79.82%. Other losses accounted of 5% which covers the transmission loss from turbine shaft to alternator. The following points can be concluded from the above work:

1) A preliminary simulation of the ORC system performance by Cycle Tempo® software, varying the main operative parameters and considering Toluene and air as working fluids has been carried out.
2) Two-Dimensional design of turbine was carried out to find the geometry of the turbine. The material SS-304 was selected for the manufacturing of turbine and the turbine was machined on 3-axex VMC machine.
3) The results revel that it is possible to improve the performance of the turbine by appropriate choice of geometric parameter.

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