SELECTION AND CONVERSION OF TURBOCHARGER AS TURBO-EXPANDER FOR ORGANIC RANKINE CYCLE (ORC)

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ABSTRACT

Turbo-expander is the most crucial and expensive component in Organic Rankine Cycle (ORC) power generation systems. The turbine engineering, design and research development process for ORC system is costly, time-consuming, and difficult for new-entrants in the ORC field. This paper investigates the re-design and retrofit of an off-the-shelf turbocharger for use in a 1 kW ORC system. The expander requirements were determined for the 1 kW Capstone Gas Turbine bottoming cycles from the ORC system design. A radial turbine was selected and an off-the-shelf turbocharger from a small petrol-driven vehicle was acquired. The turbocharger was simulated to study the off-design performance in an ORC system. The turbocharger was evaluated by expansion of compressed air at room temperature to assess the leakage of working fluid and the lubrication requirements. A preliminary design was developed to deal with leakage and lubrication issues. The conceptual design of an ORC turbine was presented including the casing design and selection of auxiliary components such as bearings, seals and lubrication systems.

1. INTRODUCTION

The potential for utilization of low grade heat sources for power generation is contributing to increasing research and development of ORC systems. The ORC uses refrigerants as the working fluid to provide higher thermal cycle efficiency compared to the conventional steam Rankine cycle at resource temperatures below 300^{o}C (Vankeirsbilck, Vanslambrouck, Gusev, & De Paepe, 2011). The design and development of a turbo-expander for an ORC is the most expensive component in the whole system. Stosic found that the turbo-expander in a 1MW ORC system typically costs over half of the total capital cost (Leibowitz, Smith, & Stosic, 2006). The high cost of the turbo-expander is attributed to the intensive research and engineering development required in the turbine design and manufacturing process.

The type of turbine has to be selected before proceeding to turbine design and analysis. A wide variety of turbo-expanders can be chosen for ORC applications. Positive displacement machines are popular for small scale ORC systems due to their simplicity and low cost, while turbo-expanders are widely applied for large scale power generation systems and have achieved high turbine efficiency and high reliability. The type of turbo-expander is determined by shaft speed, mass flow rate and the nominal power range. Recent developments in small scale ORC systems for research applications have produced an expander selection methodology, which covers the scroll expander, screw expander, and radial turbine (Quoilin, Declaye, & Lemort, 2010). Application of scroll expanders is limited to power generation below 30 kW due to the excessive leakage at increasing scroll diameter and scroll height (Quoilin et al., 2010). A screw expander was adopted by ElectraTherm for a 100 kW ORC system (Qiu, Liu, & Riffat, 2011). However, the screw expander is not as widely validated as the scroll expander and radial turbine. Turbo-expanders are preferable for a wide range of power generation systems ranging from 10 kW to a few MW. A 20-115 kW turbo-expander was developed by Verdicorp, a 175 kW radial impeller was commercialized by Tri-O-Gen and a 250 kW turbo-expander was designed by UTC for the ORC systems. Large power units above 500 kW are mainly monopolized by axial turbine. The current market available ORC units are shown in Figure 1. There are some gaps between 20 kW and 500 kW in which a flexible design approach of the turbo-expander would contribute to fill in the market gap.

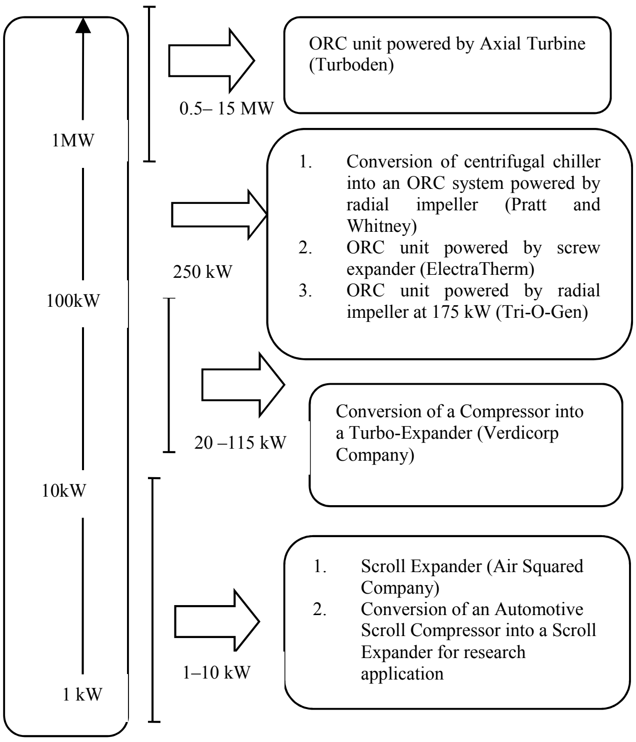

Figure 1: Current Market Available ORC Unit

Figure 2 shows the capital cost of installed ORC units based on different turbine development schemes. The cost per kWh in Figure 2 is only valid for the ORC system over 100 kW. At small scale power generation system, the prototype of an expander would incur an excessively high cost (Qiu, Liu, & Riffat, 2011). Even though conversion of an automotive scroll compressor into a scroll expander costs less than 2000 USD (Orosz et al., 2009) as opposed to a turbo-generator which is over half a million USD (I.K. Smith et al., 2005), the cost per unit kWh for the scroll expander is higher than the turbo-generator due to the low power output from the scroll machine. Figure 2 shows that the cost of an ORC unit based on turbine design and development scheme is in the range of 2000 $/kWh and 2500 $/kWh whereas the cost of an ORC system by converting a centrifugal compressor into an ORC unit is in the range of 1300 $/kWh and 1350 $/kWh (Holdmann & List, 2007). The cost of the ORC unit developed by converting an off-the-shelf turbine is not included due to the lack of published information regarding the economic analysis of the ABB ORC system in Switzerland. In order to develop a flexible design approach to fill in the market gap in Figure 1 with low capital cost per unit kW, the retrofit of the existing turbine into an ORC turbine is proposed by the authors. Radial turbines are the best candidates due to the mass-manufacture and low cost as automotive turbochargers. A turbine selection chart developed by the authors in Appendix I based on the literature reviews shows that the radial turbine is suitable for a wide range of operating conditions for the ORC system. An ORC system for the application of waste heat recovery was developed by converting an automotive turbocharger by ABB Schweiz AG (Buerki, Bremer, Paice, & Troendle, 2010).

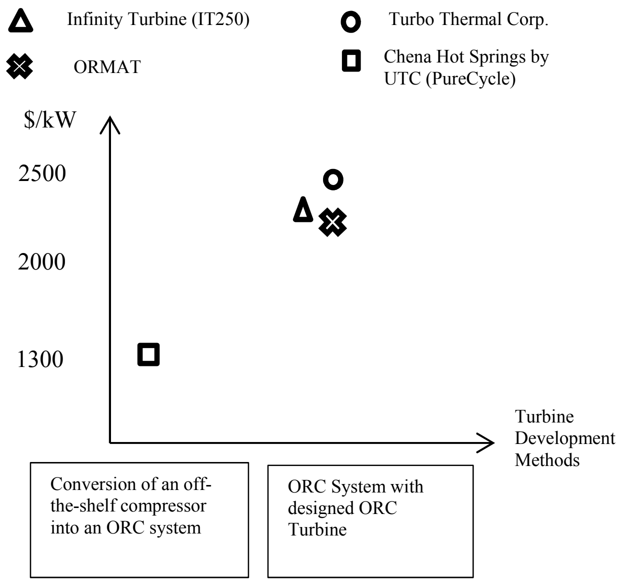

Figure 2: Capital Cost of Installed ORC Units based on different Turbine Development Schemes

Radial turbine design for an ORC system has two approaches, turbine design or turbine selection. The turbine design process involves preliminary design of the radial turbine, followed by a computational fluid dynamic (CFD) analysis of the blade design. This approach is typical for turbomachinery developers such as GE Energy and Turboden. However, turbine selection is a more preferable approach for new ORC developers if an existing turbo-machine can be retrofitted for ORC systems. The radial turbine could be developed by retrofitting automotive turbochargers which are easily accessible, relatively low cost, and manufactured in a wide range of sizes. Once a suitable turbine wheel is chosen, auxiliary components have to be selected and the casing has to be re-designed. The turbine selection and retrofit process is shown below.

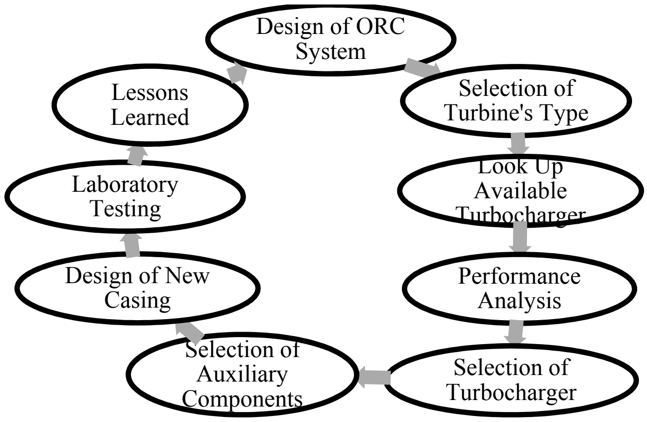

Figure 3: Selection and Retrofit of a Turbocharger as ORC Turbine

This paper demonstrates the selection of turbine type, evaluation of performance, and conversion of a turbocharger into an ORC turbine. One-dimensional off-design performance analysis models are coupled with established empirical loss models to estimate the pressure ratio and isentropic efficiency of the turbine from the turbocharger given inlet thermodynamic conditions. This paper then discusses the required modification in retrofitting the turbocharger as ORC turbine.

2. SELECTION OF TURBINE'S TYPE

A number of turbine selection methodologies have been proposed for different types of turbines and expanders for different applications. The selection methodologies were developed by Balje (Balje, 1981), Dixon (Dixon & Hall, 2010), Baines (Japikse & Baines, 1995), and Lemort (Quoilin et al., 2010). Lemort has proposed a selection methodology covering scroll expanders, screw expanders and radial turbines based on nominal power (Quoilin et al., 2010). Balje, Dixon and Baines have developed a turbine classification and selection methodology based on specific speed.

The 1kW ORC system has the system specification as below.

1. Working fluid is R245fa
2. Inlet total temperature, T_{01} is 370 K
3. Inlet total pressure, P_{01} is 17 bar
4. Mass flow rate, $\dot{m}$ is in the range of 0.1 and 0.3 kg/s.

A radial turbine was chosen from the turbine performance chart in Figure 4 (Balje, 1981) by assuming specific speed as 0.6. Optimal rotational speed was determined as 59,450 rev/min, 42,040 rev/min, and 34,320 rev/min for 0.1 kg/s, 0.2 kg/s and 0.3 kg/s respectively. An optimal range of specific diameter was determined as 0.28 and 0.38 from Figure 2. By using equation (2), the optimal range of turbine diameter was determined in the range of 9 and 21cm.

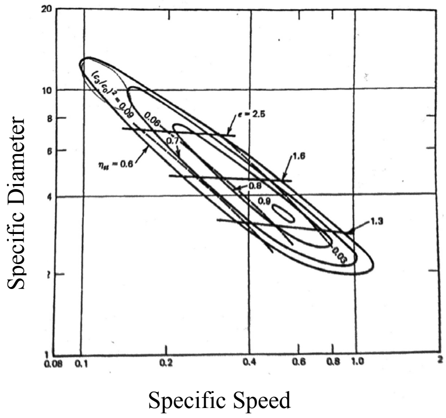

Figure 4: Performance Chart of Radial Turbine by Balje (Balje, 1981)

$$n_s = \frac{\omega\sqrt{\dot{m}/\rho}}{\left(\Delta h_{is}\right)^{0.75}} \tag{1}$$

$$d_s = \frac{D\left(\Delta h_{is}\right)^{0.25}}{\sqrt{\dot{m}/\rho}} \tag{2}$$

The radial turbine was also chosen for several reasons. Radial turbines are cheaper than axial turbines. The radial turbine is more robust under high blade loading by high density working fluids such as R245fa and stable from the standpoint of rotor-dynamic due to high stiffness (Ventura, Jacobs, Rowlands, Petrie-Repar, & Sauret, 2012). Table 1 shows geometry and overall dimensions of a selected radial turbine (known as TC-1) from a turbocharger for petrol-driven vehicle.

Table 1: Basic Dimensions Value of Turbocharger TC-1

	Unit	Value
Inlet radius	r_1 **(mm)**	**22.3**
Outlet tip radius	$r_{2,tip}$ **(mm)**	**19.3**
Outlet hub radius	$r_{2,hub}$ **(mm)**	**6.5**
Inlet blade height	b_{in} **(mm)**	**6**
Total blade height	b_{total} **(mm)**	**18.5**
Blade number	**Z**	**9**
Inlet blade angle	$\beta_{1,b}$	**0**
Outlet blade angle at tip	$\beta_{2t,b}$	**55**
Outlet blade angle at hub	$\beta_{2h,b}$	**45**

3. NUMERICAL MODEL

Off-design performance of radial turbine was first developed in FORTRAN by Glassman as part of the NASA gas turbine program in 1970 (Wasserbauer & Glassman, 1975). The performance analysis process was widely applied in performance estimation of turbochargers for both steady-state (Serrano, Arnau, Dolz, Tiseira, & Cervelló, 2008) and unsteady flow conditions (Rajoo, Romagnoli, & Martinez-Botas, 2012). The performance model was developed under the assumption of perfect gas, in which the ideal gas equation is applied and specific heat capacity is assumed to be constant at all temperatures (Meitner & Glassman, 1980, 1983; Wasserbauer & Glassman, 1975). In this study, as the performance analysis was conducted in Engineering Equation Solver (EES), the in-built real gas properties of EES is applied to determine the thermodynamic properties of real fluids instead of using the ideal gas law. The fluid behaviour can be predicted at higher accuracy compared to the application of the ideal gas equation. The loss models will not be discussed in detail as the information can be obtained in the relevant sources.

3.1 Passage Loss

Passage loss is defined as total fluid flow loss inside the blade passage, including secondary flow loss, mixing flow loss, the blockage loss, and the kinetic energy loss due to growth of a boundary layer along the blade surface (Moustapha, Zelesky, Baines, & Japiske, 2003). Baines's textbook reported that the initial passage loss model by NASA cannot accurately predict the passage loss for all types of radial turbines under different operating conditions. An improved version of passage loss is presented in his work, which is shown below.

$$\xi_{passage} = K_p \left\{ \frac{L_h}{D_h} + 0.68\left[1-\left(\frac{r_2}{r_1}\right)^2\right]\frac{\cos\left(\beta_2\right)}{b_2/C} \right\} * \left(\frac{W_2^2 + W_1^2}{W_2^2}\right) \tag{3}$$

3.2 Incidence Loss

The incidence loss model is developed as a function of change in relative tangential velocity energy of the working fluid (Ghosh, Sahoo, & Sarangi, 2011). The fluid approaches the turbine wheel at an angle different from the optimal angle. The optimal relative flow angle is determined based on the empirical formulation below. The incidence loss is then determined as fluid flow loss at any incidence angle other than the optimal flow angle.

$$\tan\left(\beta_{1,opt}\right) = \frac{-1.98 tan\left(\alpha_1\right)}{Z_r\left(1-\frac{1.98}{Z_r}\right)} \tag{4}$$

$$\xi_{incidence} = \left[\frac{W_1 \sin\left(\beta_1 - \beta_{1,opt}\right)}{W_2}\right]^2 \tag{5}$$

3.3 Trailing Edge Loss

Trailing edge loss is calculated from the relative total pressure loss (Ventura et al., 2012). The relative total pressure loss is proportional to the relative kinetic energy at the rotor exit (Ghosh et al., 2011). The relative total pressure loss is then converted to enthalpy loss coefficient as below (Ventura et al., 2012).

$$\xi_{te} = \frac{2}{\gamma M_{2,rel}^2}\frac{\Delta P_{0,rel}}{P_{02,rel}} \tag{6}$$

3.4 Tip Clearance Loss

The tip clearance is modelled as an orifice with shear flow inside the passage (Ghosh et al., 2011). A study by Baines shows that radial clearance has strong influence on the exit flow condition compared to axial clearance (Moustapha et al., 2003). An in-depth examination has showed that the variations of efficiency with axial and radial clearance are not independent as some cross-couplings exist. The cross coupling coefficient was represented with a coefficient, $K_{a,r}$ and the tip clearance loss is represented by the equation below.

$$\xi_{tip} = \frac{U_1 Z_r}{8\pi}\left(K_a \varepsilon_a C_a + K_r \varepsilon_r C_r + K_{a,r}\sqrt{(\varepsilon_a \varepsilon_r C_a C_r)}\right) \quad (7)$$

3.5 Windage Loss

Windage loss is frictional loss due to leakage of fluid between the back face of the rotor and the back plate. Daily and Nece (Daily & Nece, 1960) considered a simple case of a disk rotating in an enclosed casing. The empirical equations were modified to account for the frictional torque on the back face of turbine rotor. The loss model was then modified and expressed as power loss (Moustapha et al., 2003) and enthalpy loss (Ghosh et al., 2011). Equation (8) shows the windage loss as enthalpy loss.

$$\xi_{windage} = k_f \frac{\bar{\rho} U_1^3 r_1^2}{2\dot{m} W_2^2} \quad \textbf{(8)}$$

3.6. Performance Analysis Flow

The performance analysis is conducted by first guess-estimating a value for relative flow velocity at turbine outlet and entropy drop across the turbine wheel. Based on the given turbine dimensions, the blade speed is determined. The velocity diagram is then constructed. The thermodynamic properties at both turbine inlet and outlet are then calculated. The new value of relative flow velocity is then calculated from continuity equation. If the error of the relative flow velocity is less than the tolerance value, the calculation process will be iterated to achieve convergence on the relative flow velocity. The individual loss of radial turbine is then calculated from the loss models discussed in previous sections. A new value of entropy drop across the turbine is then calculated. The calculation process will be iterated until convergence is achieved on the entropy drop. Finally, the stage efficiency is calculated based on the calculated loss coefficient. The design process overview is presented in Figure 5.

For rotor with radial blade and no guided nozzle, the relative flow angle β_1 is zero and the tangential component of flow velocity $C_{\theta 1}$ is equal to blade tip speed (Serrano et al., 2008).

4. PERFORMANCE ANALYSIS OF TC-1

The simulation result shows that the power output of the radial turbine is increasing with increasing rotational speed and mass flow rate. From Figure 6, at low rotational speed, the power difference is small for different mass flow rate. However, the power difference is significant for different mass flow rate at high rotational speed. The pressure ratio across the turbine wheel is increasing with rotational speed. At high rotational speed and a mass flow rate of 0.1 kg/s. there is a significant increase in pressure ratio when compared to high flow rates of 0.5 kg/s and 1.0 kg/s.

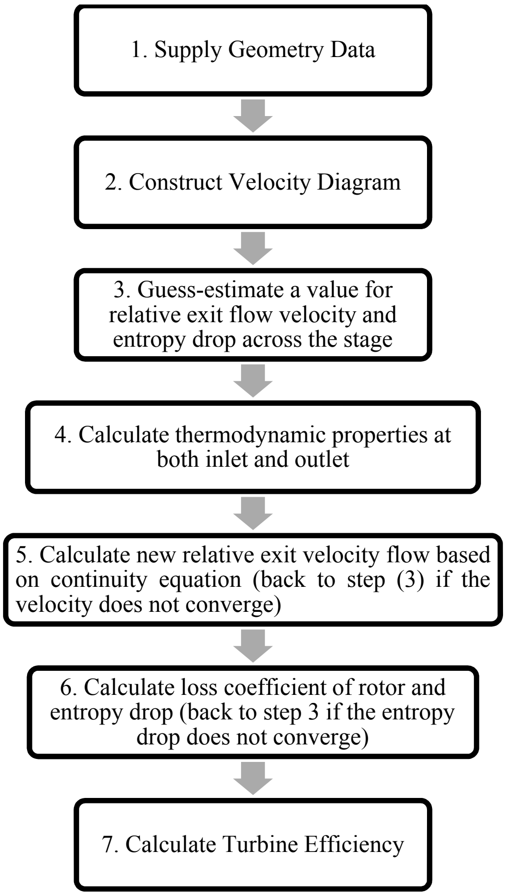

Figure 5: Performance Analysis of Radial Turbine by Implementing Real Fluid Database

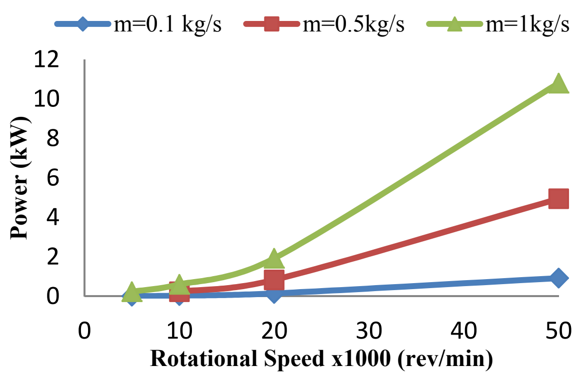

Figure 6: Variation of Power with Shaft Speed and Mass Flow Rate

Figure 7 shows the performance curve at different mass flow rates and rotational speeds. A clear correlation between the investigated parameters is not found. The figure shows an increase in efficiency at 1 kg/s up to 20,000 rpm before a constant efficiency between 20,000 and 50,000 rpm. The turbine shows a near-to-constant efficiency at 0.5 kg/s and 0.1 kg/s. The smaller mass flow rate of 0.1 kg/s shows a lower efficiency (about 55%) compared to the efficiency at 0.5 kg/s (about 77%).

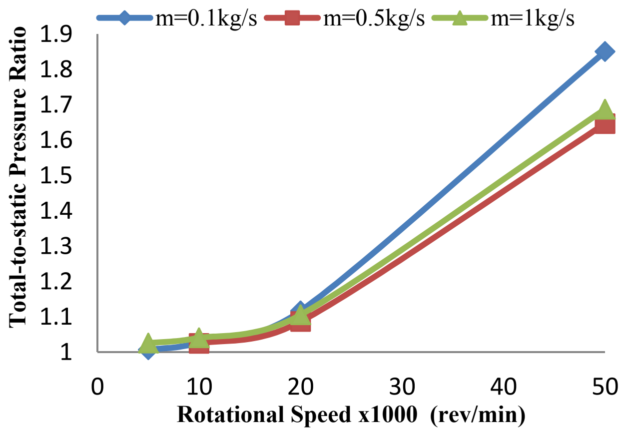

Figure 7: Pressure Ratio across the Turbine Wheel at Different Rotational Speed and Mass Flow Rate

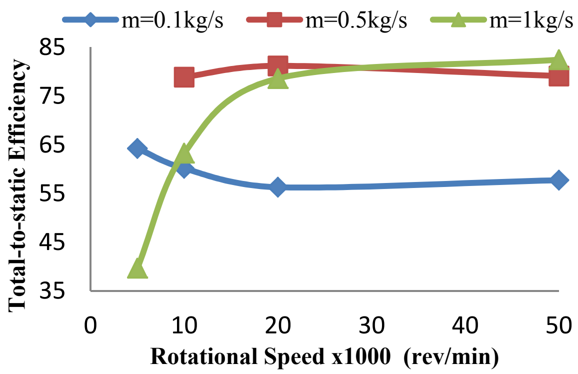

Figure 8: Turbine Total-to-Static Efficiency at Different Rotational Speed and Mass Flow Rate

5. EXPERIMENTAL TESTING OF TURBOCHARGER

Turbochargers cannot be directly applied into ORC system due to excessive leakage, poor sealing, unsuitable housing geometry and lubrication issues. The chosen turbocharger was tested in research laboratory in University of Canterbury with the following findings.

5.1 Leakages

Excessive leakage was found through the turbo gasket which is sandwiched between the turbine casings. The second major leak site was the bypass valve which is originally used to limit the pressure ratio across the turbine section and prevent over-speed of the shaft. The third leak site was along the shaft from the turbine side to the compressor side. Proper seal is not required in turbocharger as the exhaust gas could be safely released to the atmosphere.

5.2 Housing Geometry

Turbochargers are usually designed for large exhaust flow rate. Thus, a low pressure ratio was found by testing the turbocharger at a small 0.05 kg/s mass flow rate of compressed air. The chosen turbocharger does not have a nozzle to direct and accelerate the flow into the turbine wheel, thus reducing the overall turbine efficiency. Lastly, the cross-sectional area of the volute is over-sized for the desired mass flow rate. The ORC turbine requires a new design of turbine housing with small inlet flow and inlet nozzle to guide the flow into the turbine wheel.

4.1 Lubrication System

Oil-based lubricant is applied to lubricate the bushing and shaft in the turbocharger. This lubrication system is not suitable in this 1 kW ORC turbine due to high differential pressure between evaporator outlet and condenser inlet. A separate oil pump with an external oil circuit is required for the ORC turbine. However, the mixture of oil and refrigerant will reduce the lubricating film thickness on the bearing and shaft which will be discussed in the following section. The proposed design would use working R245fa as direct lubrication which is described in the following section.

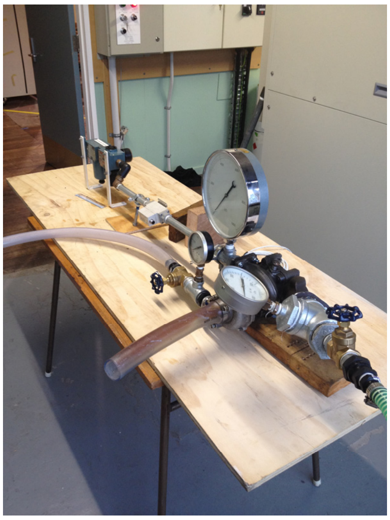

Figure 9: Laboratory Test Rig of Turbocharger TC-1

6. MODIFICATION OF TURBOCHARGER INTO ORC TURBINE

This section discusses the modification required to convert a turbocharger into ORC turbine. From the laboratory testing of the turbocharger, it was concluded that the turbine wheel would be adapted but the casing and bearing would be re-designed and re-selected. The turbine casing would be re-designed to a fully enclosed model to avoid leakage through open drive shaft. The turbine shaft would be coupled with a magnetic coupling which facilitates the coupling with an external high speed generator. O-rings will be installed between the turbine casings to prevent leakage of high pressure working fluid through the casing's gap. Figure 10 and Figure 11 show the proposed design of ORC turbine by retrofitting the turbocharger. Modular block design is proposed in the design of prototype in order to simplify the manufacturing process.

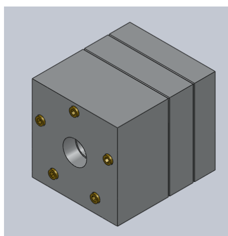

Figure 10: Isometric View of Proposed ORC Turbine

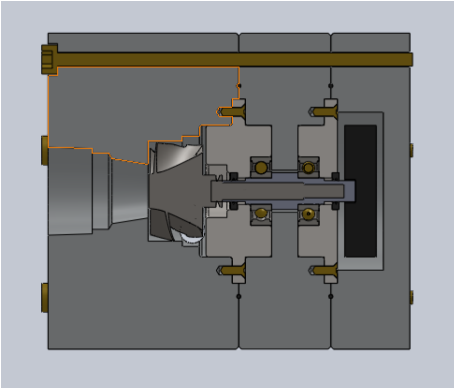

Figure 11: Proposed Design of ORC Turbine by Retrofitting a Turbocharger

6.1 Selection of Bearing

Bearings are required to support a rotor shaft in the turbine assembly. The turbine assembly commonly requires radial bearings and thrust bearings to support both the radial load and axial thrust. Literature reviews on steel bearing and hybrid bearing show that hybrid bearing is the most suitable bearing for ORC application.

In a study by Moleneaux, a hybrid bearing was found to be more suitable than steel bearings when used in an ORC turbine (Molyneaux & Zanelli, 1996). The application of hybrid bearings for refrigerant has also been suggested by Jacobson as the hybrid bearings have an optimal of raceway topography to ensure the maximum elastic deformation of the asperities in Hertzian contact when compared to a steel bearing (Jacobson & Espejel, 2006). The optimization of the raceway allows an effective separation of the mating surfaces, further allowing the bearing to withstand pure refrigerant lubrication for long periods (Jacobson & Espejel, 2006). The proposed design of ORC turbine would be equipped with a pair of hybrid bearings to provide a longer life under lubrication from pure refrigerants.

6.2 Selection of Lubricants

In an ORC application, bearings are usually lubricated by the working fluid for small size turbo-generators, while bearings for medium size turbo-generator are lubricated by proper oil lubricants with a separate oil circuit and oil pump. Two journal bearings and a thrust bearing are lubricated with R134a in an 18 kW oil-free compressors investigated by Molyneaux (Molyneaux & Zanelli, 1996) and journal bearing and thrust bearing are lubricated with toluene in a 25 kW solar-powered ORC system investigated by Nesmith (Nesmith, 1985). Common issues in bearing failures by using refrigerants include the attack of bearing surfaces by some refrigerants such as ammonia (Jacobson & Espejel, 2006). Many refrigerants tend to dissolve into lubricants, such as R134a in ester oil. The mixture reduces the initial lubricant viscosity and pressure viscosity coefficient thus reducing the film thickness during operation. Furthermore, the boiling point of refrigerant-oil mixture is reduced at the local pressure (Yamamoto, Gondo, & Kim, 2001). The reduction in boiling point allows vaporization of lubricant on the bearing surface when the bearing surface temperature is higher than the boiling point of the mixture, thus reducing the lubricant film thickness (Jacobson & Espejel, 2006). Due to the high cost and complexity of the external lubrication system, the ORC turbine would use the working fluid as lubricant for the bearings.

The rheology of three working fluids, R245fa, R134a and R123 was studied to choose an optimal lubricant for the bearing. Among the three working fluids, R245fa is found to be a better lubricant for rolling bearings followed by R134a and lastly R123 (Jacobson & Espejel, 2006). Though R134a is a relatively poor lubricant compared to R245fa, R134a is employed as the lubricant in oil-free compressors (Molyneaux & Zanelli, 1996) as the pressure viscosity coefficient of R134a is very similar to the pressure viscosity coefficient of poly alpha olefin oil (Jacobson & Espejel, 2006). The proposed design of turbo-expander in Figure 9 is designed to be lubricated by R134a and R245fa.

6.3 Casing Design and Seal Selection

The turbine casing can be categorized into enclosed and open-drive designs. In an open drive design, the turbine shaft is coupled to an external generator for power generation. The working fluid will leak to atmosphere due to high differential pressure between the fluid inside the turbine and the atmosphere. The most common sealing system for an open drive design is a dry gas seal and a mechanical seal which are not suitable for small size turbo-generators. The need for an extra gas buffer system and lubrication system to the seal increases the cost of the turbo-generator.

In a fully enclosed design, the turbine shaft is coupled to a high speed generator within the enclosed casing. Due to the scarcity of off-the-shelf high speed generators, the proposed design of ORC turbine is connected to magnetic coupling which facilitates the coupling to an external generator. The casing surrounding the magnetic coupling is made from aluminium to allow the operation of magnetic coupling.

7. CONCLUSION

This paper presents the selection of turbine type, performance evaluation of the automotive turbocharger and modification of the turbocharger into an ORC turbine. A suitable type of turbo-expander was chosen from Balje's turbine performance chart. Performance analysis was then conducted to study the performance behaviour of the chosen turbocharger. If two or more turbochargers are chosen during the selection stage, simulation of turbine off-design performance can assist the selection of the turbine wheel. Lastly, this paper demonstrates the conceptual design stage of ORC turbines, making consideration for the casing design, the selection of bearings, lubricants and seals.

8. FUTURE WORK

The future work is categorized into a number of different sections, which are the investigation of the utilization factor of the turbo-expanders during prospecting phase and feasibility phase, construction of the ORC turbine and development of Expander Selection and Implementation (ESI) method).

The first section aims to investigate the performance of the turbo-expanders. The utilization factors are then applied in the ORC plant prospecting phase and feasibility phase to facilitate the quantification of the economic merits of the ORC plants before proceeding into detailed design and construction phase.

The second section aims to construct a prototype of the ORC turbine. Laboratory performance testing and rotor-dynamic balancing will then be included. Finally, an Expander Selection and Implementation (ESI) method will then be proposed and refined based on the experience in selecting, evaluating, and retrofitting an existing turbine into the ORC turbine.

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NOMENCLATURE

C: chord length (m); *D*: diameter (m); *L*: length (m); *M*: Mach number; *m*: mass flow rate (kg/s); *K*: coefficient; *P*: pressure (kPa); *r*: radius (m); *T*: temperature (K); *U*: turbine blade speed (m/s); *W*: relative flow velocity (m/s); *Z*: blade number

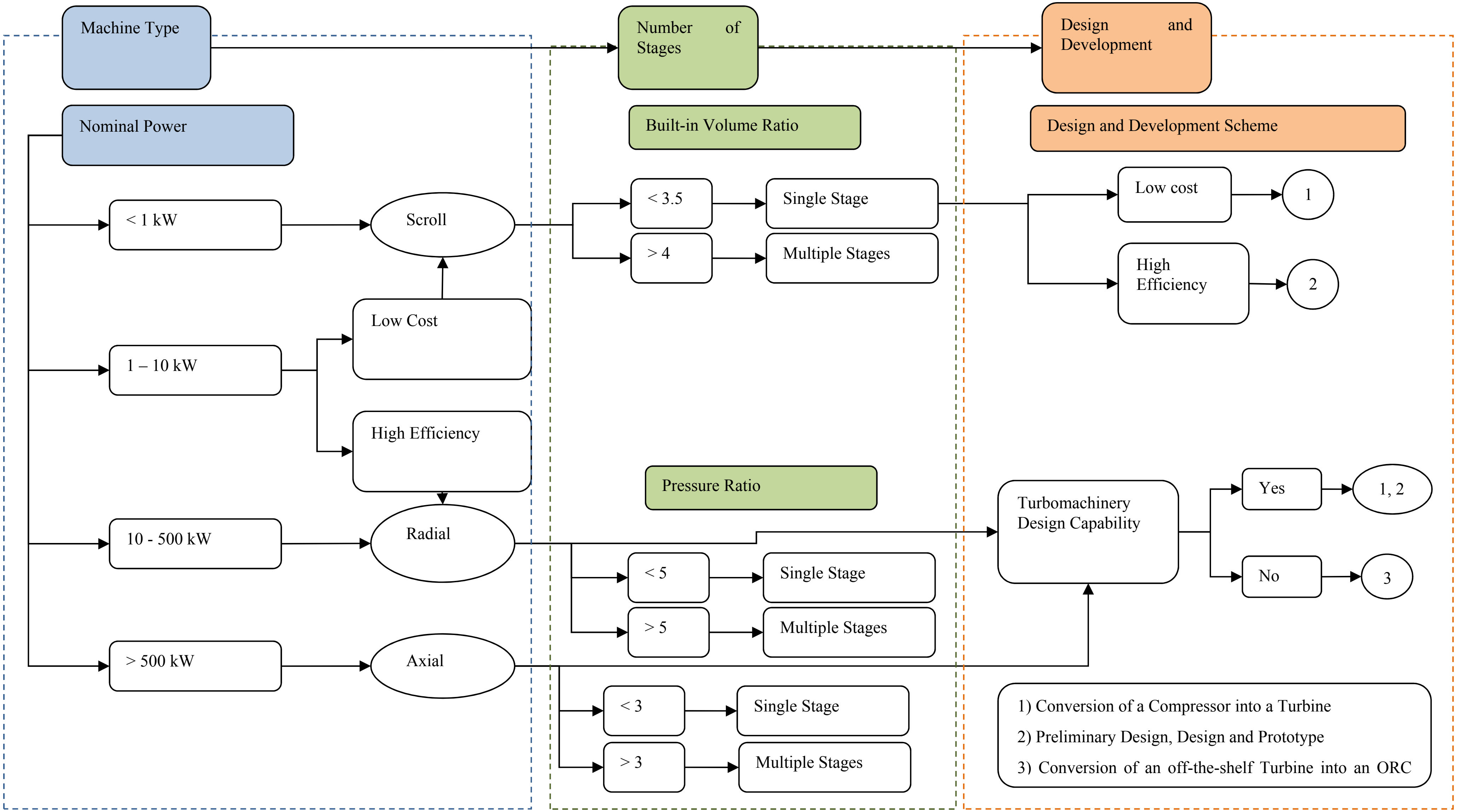
Greek symbols

α: absolute flow angle; β: relative flow angle; ε: clearance; ξ: turbine loss coefficient; ρ: density; γ: specific heat ratio

Subscripts

0: Stagnation condition; *1*: turbine inlet; *2*: turbine outlet; *a:* axial; *b:* blade; *h:* hydraulic; *opt:* optimal; *p:* passage; *r:* rotor or radial; *rel:* relative; *te:* trailing edge

APPENDIX I



Machine Type
Number of Stages
Design and Development
Nominal Power
Built-in Volume Ratio
Design and Development Scheme
< 1 kW
Scroll
< 3.5
Single Stage
> 4
Multiple Stages
Low cost
1
High Efficiency
2
Low Cost
1 – 10 kW
High Efficiency
Pressure Ratio
Turbomachinery Design Capability
Yes
1, 2
No
3
10 - 500 kW
Radial
< 5
Single Stage
> 5
Multiple Stages
> 500 kW
Axial
< 3
Single Stage
> 3
Multiple Stages
1) Conversion of a Compressor into a Turbine
2) Preliminary Design, Design and Prototype
3) Conversion of an off-the-shelf Turbine into an ORC