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Geometry Optimization of Power Production Turbine For A Low Enthalpy (≤100˚C) ORC System

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Abstract

The present paper is examining the geometry optimization of a power production turbine, in the range of 100kW$_{el}$, for a low enthalpy Organic Rankine cycle system (≤100˚C). In the last years, accelerated consumption of fossil fuels has caused many serious environmental problems such as global warming, ozone layer destruction and atmospheric pollution. It is this reason that a growing trend towards exploiting low-enthalpy content energy sources has commenced and led to a renewed interest in small-scale turbines for Organic Rankine Cycle applications. The design concept for such turbines can be quite different from either standard gas or steam turbine designs. The limited enthalpic content of many energy sources imposes the use of organic working media, with unusual properties for the turbine. A versatile cycle design and optimization requires the parameterization of the main turbine design. There are many potential applications of this power-generating turbine, including geothermal and concentrate solar thermal fields or waste heat of steam turbine exhausts. An integrated model of equations has been developed, thus creating a model to assess the performance of an organic cycle for various working fluids such as R134a and isobutane-isopentane mixture. The most appropriate working fluid has been chosen, taking its influence on both cycle efficiency and the specific volume ratio into consideration. This choice is of particular importance at turbine extreme operating conditions, which are strongly related to the turbine size. In order to assess the influence of various design parameters, a turbine design tool has been developed and applied to define the geometry of blades in a preliminary stage. Finally, as far as the working fluid is concerned, the mixture of 85% isopentane-15% isobutane has been chosen as the most suitable fluid for the low enthalpy ORC system, since its output net power is 10% higher compared to the output net power of R134a.

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Keywords: Organic Rankine Cycle; turbine design; heat recovery; low enthalpy energy sources.

1. Introduction

In recent years, the exploitation of low-temperature heat sources has become more and more attractive, owing to increasing concern about energy shortage and global warming, [1]. Out of a variety of technical solutions to this problem, the Organic Rankine Cycle (ORC) has been proven to be a very promising option, already in use for industrial applications [2][3]. The most common approach is the employment of renewable sources, such as geothermal [3], biomass [4] or solar [5][6] units, as low-temperature heat sources. The electrical power produced by ORC is thereby, considered *"clean",* since it is derived from recovery of waste heat from renewable sources, without any fossil fuel consumption.
The ORC process works similarly to a normal Rankine steam cycle, with the particularity that the temperatures and pressures are significantly lower[7][8]. In low-temperature ORC the turbine entry temperature (TET) is about 100 °C, a very low value indeed, compared to that of a modern steam turbine; pressures in ORC are usually 20-25 bar, while in regular Rankine cycle the values of pressure may typically reach multiples of that figure, even when a subcritical case is considered. Therefore, the

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Nomenclature		Greek symbols	
Symbols		η	efficiency [%]
w	specific work [kJ/kg]	Ψ	loading coefficient [-]
q	specific heat [kJ/kg]	Φ	flow coefficient [-]
P	pressure [bar]	α	absolute angle [º]
T	temperature [K]	ρ	density [kg/m^3]
Rn	degree of reaction [%]	Subscript	
U	rotational speed [m/sec]	th	thermal
$\dot{m}$	mass flow rate [kg/sec]	el	electrical
$\dot{W}$	power [kW]	m	mechanical
V	velocity [m/sec]	x	axial component
h	enthalpy [kJ/kg]	P	pump
A	cross sectional area [m^2]	T	turbine

installation cost of ORC plants is usually lower than regular steam plants. Moreover, since they operate using recovery waste heat rather than fossil fuel, ORC plants do not need all the components related to fuel pre-treatment and reduction of pollutant emissions (like DeNOx, DeSOx, Scrubber, etc.), which represents a non-negligible cost for the plant.

Due to the low values of temperatures and pressures, ORC needs a working fluid different than water. The choice of the working fluid is among the most important aspects in the optimization. Thus it is very important for the researcher to select the working fluid with utmost care [9]. Previous studies [10][11] have shown that the working fluid should have the following features: (i) low specific volume ratio between turbine outlet and turbine inlet, allowing the use of cheaper and simpler turbines; (ii) high heat transfer capacity, allowing a better recovery of the heat source; (iii) critical temperature higher than the TET of the cycle, in order to avoid chemical decomposition of the fluid; (iv) critical pressure not higher than 25 bar, since higher operating pressures would lead to increasing costs; (v) it should not be either toxic, flammable or corrosive. The importance of the specific volume of the working fluid and in particular of the volume ratio between turbine outlet and inlet cannot be overemphasized. Thus, the choice of the working fluid strongly influences the design of the turbine, which represents the most important as well as expensive plant component.

In this paper, a 100 kWel ORC plant is examined, considering three different working fluids and their mixtures. The scope of this work is to find the most appropriate solution to increase efficiency, considering at the same time the characteristics of the working fluid, with particular emphasis on the specific volume. The aim of this investigation is therefore, to define the optimal turbine design, considering different geometries at choked flow conditions in the turbine. The transonic conditions allow for a volume reduction and thereby for a simpler and cost-effective turbine design. The ultimate goal of this work is to parameterize and generalize turbine design aiming to integrate it into a larger cycle design framework.

2. Cycle Analysis

Figure 1 shows the typical ORC plant layout: the main components of the system are a pump, an evaporator, a turbine and a condenser. The pump (process 1-2) pressurizes the working fluid and leads it to the evaporator; At this station, the working fluid vaporized (process 2-3), employing the waste heat from a geothermal, a solar-thermal source or biomass incineration; the high pressure vapor expands in the turbine (process 3-4), which represents the most important and expensive plant component, producing electrical power; finally, the working fluid enters the condenser, bringing it back to the same conditions mentioned in point 1.

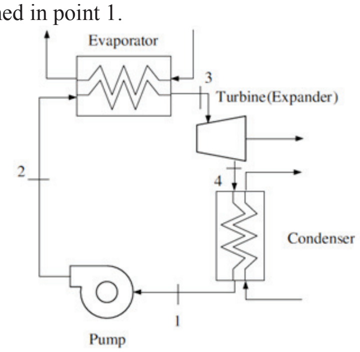

Figure 1: ORC plant lay-out

2.1. Thermodynamic Analysis

Three different working fluids have been considered in the present analysis: tetrafluoroethane (R134a), isopentane (methylbutane) and isobutane (R600a), whose basic properties are reported in Table 1.

Working fluid	R134a	Isopentane	Isobutane
Formula	CH2FCF3	C5H12	C4H10
Critical temperature (K)	374.25	460.35	408.05
Critical pressure (bar)	40.6	33.8	36.5
Boiling point (K)	247.25	245.15	261.45
Molecular weight (g/mol)	102.03	72.15	58.12
Toxicity	-	-	-

Table 1: main characteristics of working media

Referring to the properties of the ideal working fluid, as reported in the introduction, it is worth noting that it is very difficult to identify a fluid presenting all the desired properties simultaneously. Three different working fluids have been considered to underline the influence of the working fluid choice on thermal efficiency, volume ratio and fluid mass flow, which are parameters of primary importance in ORC optimization [13][14]. It is also worth observing how evaporation temperature T_3 is strongly influenced by the working fluid: considering a low-temperature heating source (i.e. geothermal, solar), having a low T_3 is a parameter of primary importance. Figure 4 shows volume ratio V_4/V_3 versus thermal efficiency, which are the two most important parameters for the working fluid choice: volume ratio between turbine outlet and inlet should be as low as possible, in order to have a simpler and cheaper turbine.

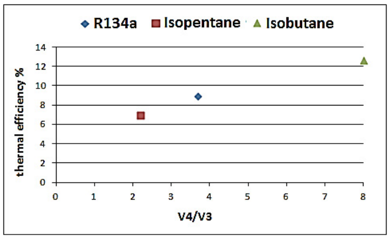

Figure 2: Thermal efficiency vs volume ratio

R134a represents a good solution, since the thermal efficiency is acceptable, the volume ratio and the evaporator temperature are low and it does not present toxic or dangerous features. Isobutane presents the best efficiency, but the volume ratio is very high too; isopentane presents a good value of volume ratio, however its thermal efficiency is low. In order to produce a new highly efficient working fluid; a new mixture of isopentane and isobutane has been investigated. The following diagrams represent the thermal efficiency, speed of sound, turbine power output and gear ratio for different chemical compositions of the new mixture.

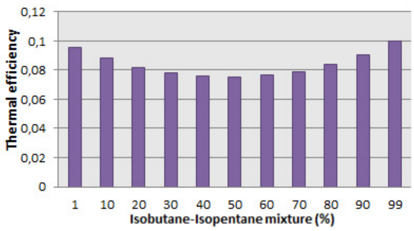

Figure 3: Thermal efficiency for different mix compositions

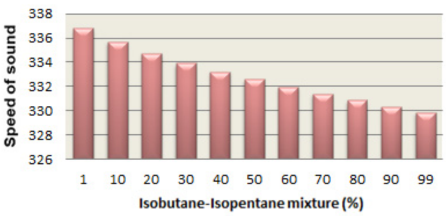

Figure 4: Speed of sound for different mix compositions

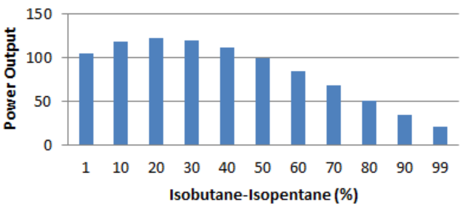

Figure 5: Turbine power output for different compositions

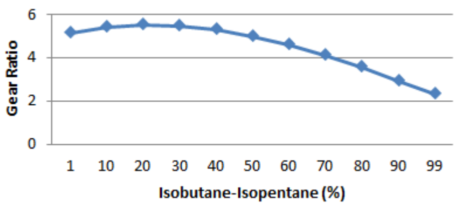

Figure 6: Gear ration of the required gear box for different compositions

Taking into consideration the results above it has been decided the usage of the 15% isobutane – 85% isopentane mixture as the most optimal working fluid because of the increased thermal efficiency as well as the power output of the turbine.

2.2. Main assumptions

To evaluate thermal efficiency, some thermodynamic parameters have been fixed, according to literature values [12], pressure losses in evaporator have been assumed equal to 5%; since low-temperature heat sources are assumed in this analysis, super-heating is not considered, thus the quality of the working fluid at turbine inlet is equal to 1.

Some additional assumptions have been considered:

- Pump efficiency η_P equal to 0.70
- Turbine efficiency η_T equal to 0.85
- Mechanical efficiency equal to 0.97
- Alternator efficiency equal to 0.97.

2.3. Thermal efficiency calculation

The specific work of the pump can be calculated as follows:

$$w_P = \frac{P_2 - P_1}{\rho_1 \eta_P} \quad (1)$$

The heat transferred to the working fluid in the evaporator is:

$$q_{Hot} = h_3 - h_2 \quad (2)$$

The specific work of the turbine is:

$$w_T = (h_3 - h_{4IS})\eta_T \quad (3)$$

The heat transfer in the condenser is given by:

$$q_C = h_4 - h_1 \quad (4)$$

Considering mechanical and electrical efficiencies, the net work is:

$$w_{NET} = (w_T - w_P)\eta_m \eta_{el} \quad (5)$$

The thermal efficiency is the net work and thermal power inlet ratio, as follows:

$$\eta_{Th} = \frac{w_{NET}}{q_H} \quad (6)$$

Knowing the working fluid and its physical properties, the calculation is straightforward to perform:

- Since T_1 is known, the pressure in the condenser and the density of the liquid can be found;
- Knowing the pressure in the evaporator, thermodynamic properties at the turbine inlet can be calculated;
- Considering the isentropic expansion in the turbine and knowing turbine efficiency, thermodynamics properties at the turbine outlet can be evaluated.

The main thermodynamic cycle results for R134a and isobutane-isopentane mixture are reported in fig.7 and fig.8 respectively. In particular, enthalpy and density values h_3, h_4, ρ_3, ρ_4, which have been used for the turbine design optimization.

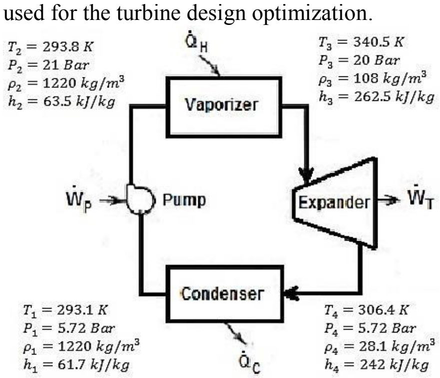

Figure 7: ORC thermodynamic results for R134

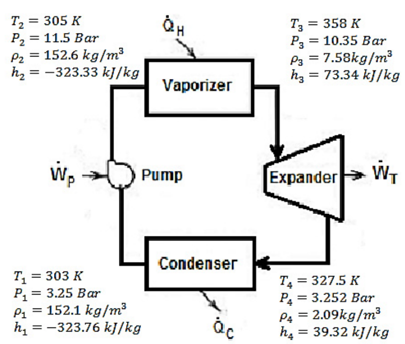

Figure 8: ORC thermodynamic results for 15% Isobutane- 85% Isopentane mixture

Both results of working media are presented in the table below. Even though the thermal efficiency of R134a is slightly higher it was prefered to use the new mixture in our project because of the high value in net power output and the smaller gear box that would be needed.

Working medium	R134a	Isobutane-Isopentane mix
Turbine output power (kW)	100	100
Mass flow rate (kg/sec)	4.35	2.94
Thermal efficiency (%)	8.9	8.4
Gear box ratio (-)	14.6 (44,000RPM)	6.4 (19,300RPM)
Pump power (kW)	10.6	1.24
Net power output (kW)	89.4	98.7

Table 2.Thermodynamic data output

3. Turbine Design

Some additional data are required for the turbine design, namely, the speed of sound of the specific working medium, the loading coefficient Ψ, the flow coefficient Φ, the degree of reaction *Rn*, the absolute inlet angle α_1 and the electrical frequency *f* (for direct driven generators, it determines the turbine rotational speed). For electrical power production the rotational speed of rotor must be 3000 rpm, corresponding to a frequency of 50 Hz. In this project the value of turbine frequency is multiple compared to 3000rpm, thus the use of a gearbox is necessary. Main data are shown in Table 4.

Ψ	2	-	**Rn**	0	%
Φ	0.65	-	**Speed of sound**	322.75	m/sec^2
α_1	0°	deg	**f**	19,300	RPM

Table 3: main data for turbine design

Blade design with zero degree of reaction has the following advantages: a) zero (minimal) pressure drop across the rotor tip b) zero (minimal) axial load on the shaft c) mature blade technology which is easier to build in a rather inexpensive manner. For the absolute inlet angle α_1 which is equal to α_3 according to Turbine Euler Equation (TEE):

$$\Delta h = \dot{W}/\dot{m} = U(V\theta_2 - V\theta_3) \tag{7}$$

to maximize the work per unit mass: $V\theta_3 = 0$, i.e V_3 is equal to V_x, which means a_3=0°. The loading coefficient Ψ can be easily calculated from the following equation:

$$\Psi = 2(1 - Rn - \Phi \cdot \tan a_1) \tag{8}$$

The main flow and geometrical parameters at the turbine inlet have been calculated. Thereafter, the velocity is fixed where the Mach number becomes equal to one, which means that the flow is choked. Finally, a single-stage turbine has been assumed.

Turbine rotor

Since the net power of the turbine is assumed equal to 100 kW, the mass flow of the working fluid can be easily calculated:

$$\dot{m} = \dot{\mathrm{W}}/(h_3 - h_4) \tag{9}$$

A single stage turbine has been considered, turbine outlet conditions have been calculated starting from mass flow continuity equation:

$$\dot{m} = \rho \cdot A \cdot V_x = const \tag{10}$$

Since outlet density ρ_4 and axial velocity are known, the cross sectional area of the rotor can be calculated; then, the other main geometrical parameters can be found (blade height, blade length, shaft diameter, turbine's size).

A well-designed turbine should follow the following criteria:

1. Axial velocity less than 0.3 Mach is used in order to ensure controllability of the machine under all operating conditions.
2. Hub to tip ratio close to 0.8 is used for standard turbine design.
3. Reduced values of flare angle have been used in order to decrease the possibility of occurrence recirculation flow at the hub of the blade.

A 2-D design of the turbine stage and the velocity triangles of turbine's blades can be seen at the next figures.

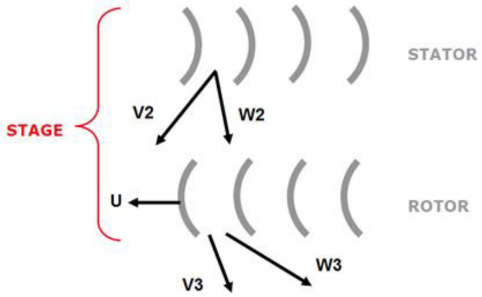

Figure 9: One stage turbine for ORC system

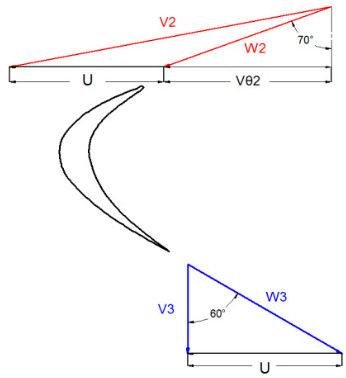

Figure 10: Rotor blade morphology through the velocity triangles

4. Results Analysis and Discussion

For the present ORC, the isobutane-isopentane mixture has been chosen as the best option for working fluid; starting from the results of the thermodynamic cycle. Different chemical compositions of the mixture have been studied for the system, in order to find out the best solution, with particular attention to turbine's volume reduction. From this point of view and considering also the thermal efficiency (fig.3) and output power of the turbine (fig.5), the best result have been obtained with the usage of 15% isobutane – 85% isopentane as working medium.

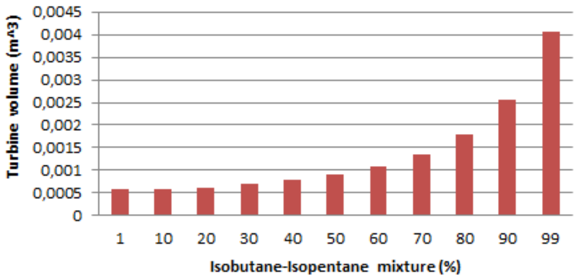

Figure 11: Volume reduction of the turbine for the optimal fluid

For the blading design a 2-D model has been considered and the calculations were estimated at the "base line" of the blade which morphology is shown in fig.10 through the velocity triangles. The geometrical values of the turbine are presented in the table 4 below.

Rotational speed U (m/sec^2)	153.47
Blade height (m)	0.0253
Blade length (m)	0.0202
Shaft diameter (m)	0.152
Inlet cross-sectional area (m^2)	0.0116
Outlet cross-sectional area (m^2)	0.0221
Flare angle (˚)	28.7
Turbine volume (m^3)	$2.04 \cdot 10^{-3}$

Table 4: Geometrical datas of the turbine

Finally, it should be noted that in the present study only the turbine design has been considered for the various working fluids, since it represents an important component regarding both the overall efficiency as well as cost. However, in the future, it would be interesting to analyze the influence of the working fluid also on the volume of other components, with particular emphasis on the evaporator and the condenser.

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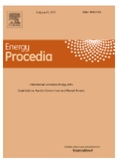

Biography

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