ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

PERFORMANCE ANALYSIS OF AN ORGANIC RANKINE CYCLE

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Abstract. *The Organic Rankine Cycle (ORC) is a promising alternative for power generation through the use of sources of low and medium temperature. The result is a better use of energy resources. This paper evaluated the behavior of an ORC cycle operating with different organic fluids. It is performed the computational, unidimensional and tridimensional simulation of a 1000 kW radial turbine evaluating its performance and flow characteristics. The study was conducted from the First and Second Laws of Thermodynamics, using previously established conditions for the cycle in question. It is presented an adequate set of the working fluid to evaluate the cycle's efficiency and turbine's geometry in order to extract more energy from the heat source. The data was obtained with an evaluation of the parameters of significant influence on the efficiency of ORC cycles. Through these results, it is possible to consider about the best configuration to be used, seeking to optimize the working cycle and the utilization of the heat source. The R-600 and R-600a fluids stood out for application in this study.*

Keywords: *ORC Cycle; Organic Fluids; Energy Resources; Radial Turbine.*

1. INTRODUCTION

Currently, energy and development are strongly associated concepts, reason why one could ensure that the progress of society depends on a broad, efficient and economic energy supply.

The increased electricity consumption has become a feature of our modern society, and can be taken as an important social indicator. In this way, energy and energy self-sufficiency are matters that must be related to a country organization, planning and development.

The increasing energy demand must be associated with the search for new energy sources as well as an environmental impact control, creating alternatives to energy matrices. The pursuit for more acceptable environmental standards will lead to a progressive use of potential energy efficiency, in both energy use and production, and will also prompt a more thrifty use of industrial energy-intensive basic supplies.

Thereby an increasing incentive in renewable energy research and innovation must be provided, aiming at new sources of clean energy introduction and consolidation. The development of systems that promote greater energy efficiency provides several benefits, such as CO_2 and other pollutants emission reduction, consumer economy, and greater use of local energy sources.

This article aimed to perform an Organic Rankine Cycle analysis, evaluating different types of working fluids, followed by a 1000 kW radial turbine analysis, appraising its characteristics with a Thermodynamic Cycle.

2. CHARACTERISTICS OF ORGANIC RANKINE CYCLE (ORC)

The ORC cycle is at present the most commonly thermal system used to recover heat that is commercially available for systems between 200-4000 KW. The spread of these systems has been rapidly developed - in 1994 there were approximately 11 units installed in Austria, Switzerland, Italy and Germany, and 13 more units were being built in those same countries. Currently, there are between 300 and 400 ORC plants in Europe. The growth in number of projects and installed power of ORC in the last 20 years has had an exponential character, and the main energy source being used in these systems is biomass, followed by geothermal energy and residual heat recovery, respectively (QUOLIN and LEMORT, 2009).

The Organic Rankine cycle has emerged as an efficient technology for conversion of low and medium temperature heat into electricity. According to Shengjun, *et al.*, (2010) the possibility of utilizing efficient energy resources, with smaller systems and great economic performance, enhance this technology attractiveness.

The Organic Rankine cycle is composed of its main components: evaporator, turbine, pump and condenser; and its operating principle is similar to a Conventional Rankine Cycle. The main difference between the two cycles is the working fluid used – the Conventional Rankine Cycle uses water while the Organic Rankine Cycle can use hydrocarbons or coolants (organic fluids), that provides its special characteristics.

ISSN 2176-5480

The working fluids in an ORC cycle present boiling points and latent heat of vaporization inferior to that of water, allowing evaporation to lower temperatures, which provides better yield of the heat supplied by a hot source, and highlight its use to low and medium temperature sources (SILVA, 2010).

Being a cycle which utilizes low and medium temperature heat sources, the Organic Rankine cycle operates between 60 to 200° C for low-temperature sources, reaching 350° C in the case of medium-temperature sources (YAMAMOTO *et al.*, 2001).

Thus this level of maximum temperatures allows the work in different and varied types of regimes in a range of considerably high power, fluctuating between 100 kW to 4000 kW (SILVA, 2010).

The ORC has different applications, and among the most usual are: heat recovery from biomass combustion, electric power production from geothermal and solar sources, heat recovery of biogas combustion obtained from bio-digestion, and saltwater desalination from solar source and residual heat recovery from industrial processes (SILVA, 2010).

A series of industrial processes and power plants emit waste heat at low temperatures (between 80 and 200° C). The waste heat recovery from industrial processes and the possibility of converting it into electricity have several benefits. However, the technology is not well established yet and this recovery process has some challenges if compared to the recovery of high temperature combustion gases, which is best used in the present scenario. The biggest challenge is precisely the low level of heat extraction from the source, which reduces the power produced in the cycle. Organic Rankine cycles must be selected for each application, as appropriate (ABU, 2011).

2.1 Working Fluid

For ORC cycles it is very important to select the appropriate working fluid and delineate the work cycle, optimizing it accordingly to its expected purpose. More than 50 working fluids have been considered in the literature; however, some of them have already been rejected due to environmental concerns, as required by international protocols. The organic fluids may have distinct characteristics, therefore their analysis and proper selection is very important to each operation. Important criteria in the working fluids selection is that they are evaluated according to thermodynamic and physical properties, stability, environmental impacts, safety and compatibility, availability and cost – the working fluid selection has being a great motivation to studies concerning power generation process optimization in Organic Rankine Cycles (CHEN , *et al.,* 2010).

The literature concerning working fluids selection is very diverse, for their characteristics are totally linked to the cycle operating conditions. The literature holds extensive analyses and comparisons between different thermodynamic cycles and organic working fluids, nevertheless most comparisons are conducted under certain fixed aspects, predefined temperature condition, and using only a few working fluids.

It appears then that the need to select the best working fluid and the highest efficiency cycle is not always taken into account in operations analysis with distinct conditions. It is important to point out that for each operation characteristic, and every property variation in the ORC cycle, one should consider the best cycle and working fluid options in order to get the desired features (MANOLAKOS, *et al.*, 2007).

Hung (2010) describes a proper working fluid selection as a critical factor to achieve safe and efficient operation. According to this author, each fluid has its own range of applicability, giving its thermo-physical properties under consideration of high-efficiency and safe operation. For Hung (2009), the three main characteristics of an organic fluid are: specific heat, latent heat and slope of the saturation curve.

For the cycle analysis, it was chosen not to determine a specific heat source, thus taking into consideration only the heat flow required. However an important parameter for the evaporator project is the Minimum Temperature Difference (MTD) between the fluids in the heat exchanger, which can be observed in Figure 1. The point where MTD occurs (the Pinch point) depends on two factors:

1) Pressure and temperature of the organic fluid leaving the evaporator.
2) Temperature of the heat source fluid at the entrance of the evaporator.

The temperature difference in the evaporator and the source fluid has no influence on the cycle thermal efficiency. However the heat flow necessary to the cycle will be influenced by the MTD, which is influenced by the working fluid mass flow.

Vetter and Wiemer *et al* (2013) describe the MTD influence in an ORC cycle; analyzing an ORC cycle using propane; it is observed a relationship between MTD and network generated in the turbine. A difference of 20 K Pinch reduces network in 29%, compared to a difference of 5 K. MTD variation between 20 K and 5 K, with a constant turbine inlet temperature, provided an increase of 18% on net power generated. By varying MTD and optimizing heat source fluid parameters it is possible to reach a net power up to 22% higher for lower MTD.

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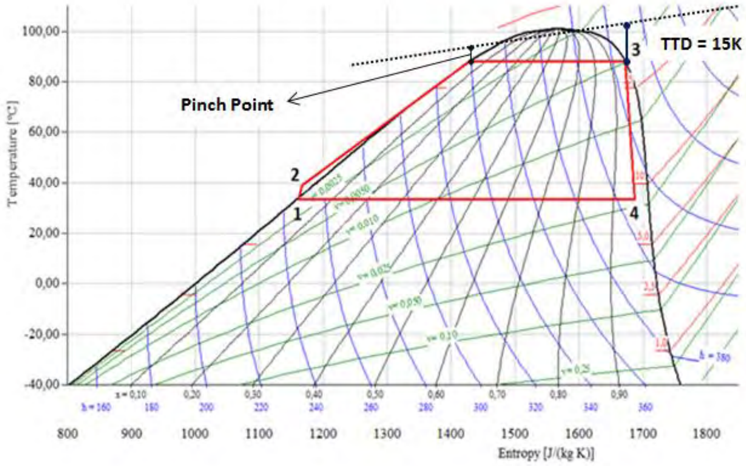

Figure 1 –Pinch Point and TTD Representation on a Txs Diagram.

2.2 Saturated Vapor Curve:

An important consideration about organic fluids cycles is regarded to their saturation curve in a T-s (ds/dT) diagram. According to Chen *et al.* (2010), the slope of the saturation curve in a T-s diagram Figure 2 can be positive (*e.g.* isopentane), negative (*e.g.* R22) or vertical (*e.g.* R11), and the fluids are accordingly called "wet", "dry" or "isentropic", respectively.

The value of (dT/ds) tends to infinite to isentropic fluids, and the inverse of this slope (ds/dT) is used to express how "dry" or "wet" a fluid can be. For Chen *et al.* (2010), if (ξ = ds/dT), the type of fluid can be sorted by its ξ value, in other words, $\xi > 0$: dry fluid (*e.g.* pentane), $\xi \cong 0$: isentropic fluid and $\xi < 0$: wet fluid (*e.g.* ammonia).

Dry and isentropic fluids have better efficiencies when compared to wet fluids. These fluids do not change phase during the turbine expansion, reason why they dispense superheated vapor. Thus, this translates into a more efficient cycle. Wet fluids will need higher temperature at the turbine entrance to ensure title greater than 90%, but there is less concern with the cooling load in the condenser.

When a fluid is "too dry" the expanded vapor leaves the turbine with a substantial overheating, what is a waste and increases the cooling load in the condenser. In this case a heat recovery system is necessary to preheat the working fluid after the feed pump, increasing considerably the cycle efficiency.

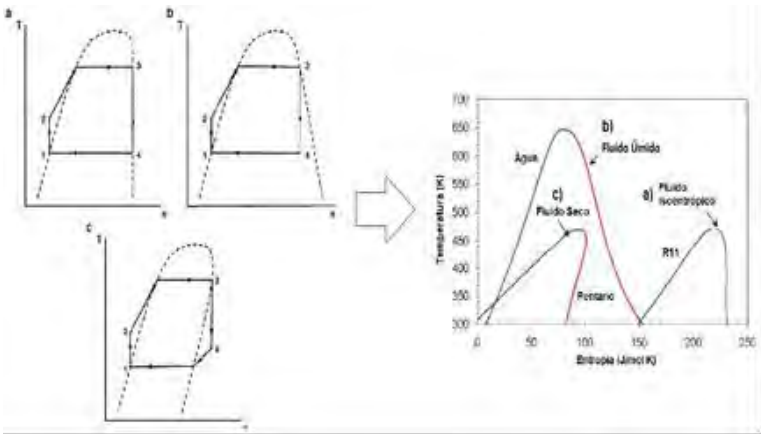

Figure 2- Comparison of Working Fluids: a) Isentropic, b) Wet, c) Dry
(CHEN, *et al.*, 2010).

3. THERMODYNAMIC ANALYSIS

The ORC cycle operating conditions and different organic working fluids performance was defined, in this study, using a modeling from the First and Second Laws of Thermodynamics.

Equation (1) presents the First Law of Thermodynamics for a system at steady-state:

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$$0 = \dot{Q}_{VC} - \dot{W}_{VC} + \sum_e \dot{m}_e \left(h_e + \frac{V_e^2}{2} + g z_e \right) - \sum_s \dot{m}_s \left(h_s + \frac{V_s^2}{2} + g z_s \right) \tag{1}$$

The Terminal Temperature Differential (TTD) in the evaporator was set at 15 K, between heat source and working fluid. To reduce security measures and materials costs, the maximum operating pressure was fixed at 80% of the critical pressure. Mass balance solution and energy determine working fluid flow ($\dot{m}$), energy efficiency (η) and operating temperatures at each point of the system. In the thermodynamic analysis the following points were assumed:

1) Kinetic energy, potential energy and heat losses are not considered;
2) Operation in steady-state;
3) Operation at full load;
4) Exhaust steam title in the turbine output above 90%.

The analysis aimed a selection of different organic fluids and different operating conditions and performance for an ORC cycle, using a radial turbine with a power established in 1000 kW.

As a research tool, Aspen HYSYS®, version 7.0 the process simulator software, was used; and allowed the characterization and analysis of the process for different predetermined fluids, as shown in Figure 3, and its modeling is based on the First and Second Laws of Thermodynamics.

The pump and turbine efficiencies were set at 85%. For the cycle analysis, the working fluid was considered to be, after the condenser, a saturated liquid. The condition of saturated steam at turbine was also established in order to create a standard of comparison between the chosen fluids.

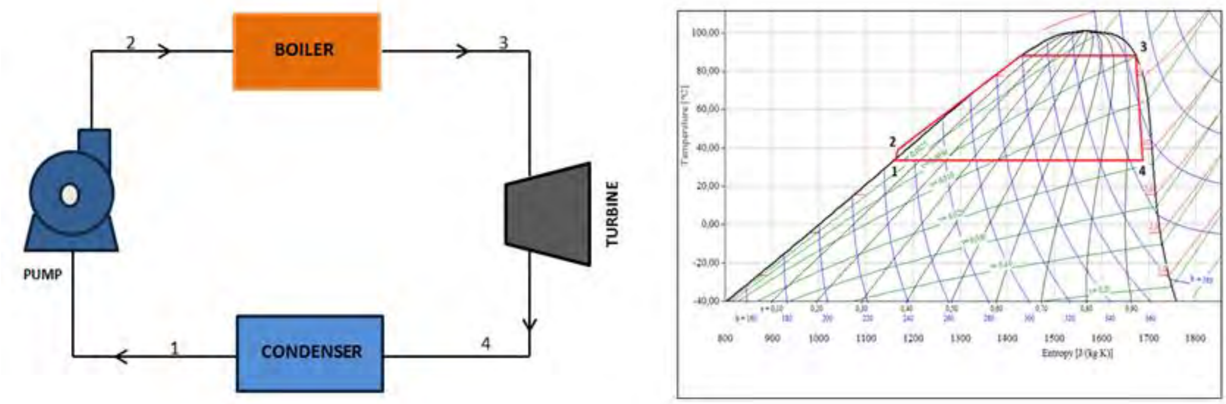

Figure 3 – Cycle Schematic Drawing (left) and T-s Diagram Representation (right).

The definition of thermal efficiency can be regarded as efficiency based on the First Law of Thermodynamics, and is presented in Eq. (2) (VAN WYLEN, *et al.*; 2003).

$$\eta_{th} = \frac{\dot{W}_{Net}}{\dot{Q}_H} \tag{2}$$

The device performance, whose purpose is to perform a task, such as steam power plants, can be evaluated as a ratio of actual work performed by theoretical maximum work. This ratio is a type of exergetic efficiency formulation (Second Law), presented in Eq. (3) (MORAN and SHAPIRO, 2002).

$$\eta_{II} = \frac{\dot{W}_{Net}}{\dot{Q}_H\left(1 - \frac{T_L}{T_H}\right)} \tag{3}$$

In a real thermodynamic system, it is impossible to convert all available energy into work, so the concept of irreversibility to calculations must be introduced. For simplification, it was considered that the internal irreversibilities in the equipment that constitute the cycle (pump, evaporator, condenser and turbine) are negligible. The internal pressure differential between the components (pump, condenser and evaporator) is also considered irrelevant (HUNG, 2010).

The Equation (4) shows the rate of irreversibility expressed for uniform flow conditions operating at steady-state (HUNG, 2010):

$$\dot{I} = \dot{m} T_0 \left[(s_{out} - s_{in}) - \frac{q}{T_0} \right] \tag{4}$$

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Based on Eq. (4), one can calculate irreversibility for all equipment existent in the cycle, equipment shown in Figure 2.

Pump: For this equipment, the loss of heat to external environment is disregarded; and the irreversibility is determined by Eq. (5):

$$\dot{I}_{1-2} = \dot{m}T_0(s_2 - s_1) \tag{5}$$

Evaporator: The evaporation is an isobaric heating process. In this equipment the external heat source steadily transfers heat to the working fluid, at a T_H temperature. The irreversibility in the equipment is given by Eq. (6):

$$\dot{I}_{2-3} = \dot{m}T_0\left[(s_3 - s_2) + \frac{h_2 - h_3}{T_H}\right] \tag{6}$$

Turbine: As well as in the pump, the loss of heat to external environment in the turbine can be disregarded; and the irreversibility can be calculated by Eq. (7):

$$\dot{I}_{3-4} = \dot{m}T_0[(s_4 - s_3)] \tag{7}$$

Condenser: Condensation is a cooling isobaric process. This equipment transfers heat from the working fluid to a cold source at temperature T_L. The irreversibility is given by Eq. (8):

$$\dot{I}_{4-1} = \dot{m}T_0\left[(s_1 - s_4) + \frac{h_4 - h_1}{T_L}\right] \tag{8}$$

Combining Eqs. (5), (6), (7) and (8), the total irreversibility can be determined by Eq. (9):

$$\dot{I}_{tot} = \sum_i \dot{I}_i \simeq \dot{m}T_0\left[-\frac{h_3 - h_2}{T_H} + \frac{h_4 - h_1}{T_L}\right] \tag{9}$$

4. CYCLE ANALYSIS

Two major components and a great part of the interrelated cycle are the working fluid and the turbine. The selection of an appropriate working fluid and operating conditions, as well as an optimized turbine design for these conditions, determine the amount of energy that can be extracted from a resource (SAURET and ROWLANDS, 2011).

Different fluids, due to differences in their thermodynamic properties, under certain temperature and pressure, are able to extract a greater or lesser amount of energy from a given source. The fluids that provide the highest specific work (from the turbine expansion) have as consequence lower mass flow in the cycle, thus requiring lesser radial turbine size.

The configuration established for this study does not express an optimum operation for the present cycle. This approach is a parameter for comparison of different working fluids and their influence on a radial turbine, not taking into account the operating conditions optimization. To enable more efficient configurations, the ORC cycle analysis must verify in depth the working fluids. Different parameters, such as thermo-physical properties, environmental impact, health and safety, availability and cost, as well as improving the cycle configuration, should be evaluated. Various configurations can be employed to an ORC cycle in order to increase substantially its overall efficiency.

4.1 Working Fluid:

The choice of a suitable working fluid is of great importance for a better cycle performance, considering the different characteristics of each fluid. The slope of the saturation curve (T-s diagram) that defines: $\xi > 0$ dry fluid, $\xi \cong 0$ isentropic fluid, $\xi < 0$ wet fluid, has a fundamental importance for cycle analysis. This study selected seven fluids in order to address different characteristics between dry, isentropic and wet.

In general, for the working fluid selection, the focus was kept on the utilization of low and medium temperatures sources. The hot sources origin was not considered (solar, biomass, waste heat, heat of biogas combustion among others), but only their characteristics, according to the fluids properties. The selected working fluids and some of their characteristics are listed in Table 1.

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Table1 - Working Fluids Properties.

Working Fluids	R-600$_a$	R-600	R-134$_a$	R-290	R-152$_a$	R-1270	R-143$_a$
Chemical Formula[1]	C_4H_{10}	C_4H_{10}	$C_2F_4H_2$	C_3H_8	$C_2H_4F_2$	C_3H_6	$C_2H_3F_3$
Critical Temperature[1] [°C]	134,83	152,00	101,00	96,67	113,89	92,44	72,73
Critical Pressure[1] [bar]	36,55	37,96	40,55	42,42	44,44	46,64	37,64
Volume [1] [m^3/kg Mol]	0,2592	0,2563	0,2004	0,1087	0,1835	0,1884	0,1950

[1] CHEN *et al.*, 2010.

- **R-143a**: Wet fluid with low critical temperature and reasonable critical pressure; its more convenient and promising usage is in a Supercritical Organic Rankine cycle. Other fluids with similar features are: R-170, R-744, R-41, R-116, R-125, R32.

- **R-152a, R-1270:** Also classified as wet fluids, but unlike R-143a, these fluids have relatively high critical temperature and, because of this condition, they allow overheating use, and usually this setting is necessary when applied. R-152a can be used in supercritical cycles, if the heat source is suitable for this condition. On the other hand the R-1270 is not normally used for supercritical configuration, because of its low molecular weight. The use of R-1270 under these conditions would involve a greater structural system size compared to other fluids with higher molecular weight, which would increase substantially the installation price, making this type application impracticable.

- **R-134a, R-290**: Fluids with the same characteristics as R-134a and R-290 are classified as isentropic fluids. Both fluids can be used in conventional organic Rankine cycles and in supercritical settings, after all they present low critical temperatures and can be used efficiently for low heat sources. One of the main advantages of these fluids is to ensure a superheated steam condition after the turbine expansion, without overheating.

Other fluids with similar characteristics are: R-21, R-R-141b, 142b, R-123, R-245ca, R-245fa, R-236ea, R-124, R-227ea and R-218.

R-600, R-600a: Such fluids are considered dry fluids and in a similar manner to the isentropic, they guarantee a superheated steam condition after the turbine expansion, without overheating. These fluids can be used both in conventional and supercritical ORC cycles. It can be observed that these fluids have a high critical temperature; therefore they can be used in conventional ORC cycles with high temperature sources.

Other fluids with similar characteristics are: R-601, FC-4-1-12, R-C318, R-3-1-10.

4.2 Turbine Analysis

A turbine is utilized to extract energy from a working fluid and convert it into mechanical work, which subsequently is converted into electricity. The two main configurations used in turbines are axial and radial. Single-stage axial machines are commonly used in systems with high flow rate and low pressure ratio; while radial flow machines are suitable for systems with lower flow rate but higher pressure, which makes them attractive for ORC systems (SAURET and ROWLANDS, 2011).

In the past decade, inflow radial turbines or centripetal turbines have been used in many applications, for they operate with lower flows, being more efficient than axial turbines operating in low power conditions. They also present a number of attractive features, such as simplicity, reliability, different types of fuels usage, low manufacturing cost, relatively high performance, and easy installation and maintenance (DÓREA, *et al.*, 2006).

According to Marcuccilli (2008), radial turbines can be optimized if small changes are made, allowing better yield for different heat sources, while maintaining a high efficiency, through the utilization of nozzles with variable vane at the entrance. Furthermore, they are more robust and support higher loads on the blades, caused by greater density working fluids operation, in both configurations supercritical and subcritical (UTAMURA, 2010 ; SÁNCHEZ, *et al.,* 2008).

4.2.1 Dimensionless Parameters:

Specific speed is a parameter defined by Eq. (10):

$$N_s = \frac{N\sqrt{\dot{V}}}{(\Delta h)^{3/4}} \tag{10}$$

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Where $\dot{V}$ is a volumetric flow rate through a turbine, at a rotor output, N is the rotation of the rotor and Δh enthalpy variation of the turbine.

Specific speed is used to relate geometrically similar turbines performance, but with different dimensions. Generally, the efficiency of two turbines with similar specific speed is the same, except for Reynolds number major differences. The practice of preserving specific speed is a common methodology for those turbines operating at different flow rates (MATHIS, 1996).

Specific diameter is defined by Eq. (11):

$$D_s = \frac{d(\Delta h)^{3/4}}{\sqrt{\dot{V}}} \tag{11}$$

Where *d* is a turbine rotor diameter, when at axial and radial flow.

4.2.2 Turbine Simulation

In this step, a radial turbine numerical simulation is presented, which was performed using "TurbinaDP.exe" - a computer program used for one-dimensional calculation. This program was developed and discussed by (MIRANDA, 2010), and is used in a radial turbine aerothermodynamics preliminary project, where FORTRAN language is used. It is also presented in a three-dimensional flow analysis, for which Computational Fluid Dynamics (CFD) is used. This equipment analysis was carried out by studying its behavior, when operating with an R134a working fluid.

The analysis through "TurbinaDP.exe" uses a one-dimensional solution to determine flow characteristics through a turbine along the medium current line, for project point conditions and out, at a steady-state of a single cycle radial flow turbine with one axis. Table 2 describes aerothermodynamics parameters and input values used for conditions of simulation calculation. This analysis follows the development proposed by (MIRANDA, 2010).

These basic geometry parameters as well as working fluid operating conditions in turbine inlet should be provided prior to the analysis start. This approach aimed to establish a correct relationship between one-dimensional and three-dimensional analyses. A three-dimensional analysis is performed through a Computational Fluid Dynamics analysis.

Table 2 – Inlet Parameters of Turbine's Simulation.

Inlet Parameters of Turbine	Symbol	Unit	Value
Relative angle of the blade at inlet rotor	β_2	°	-25
Relative angle of the blade at outlet rotor	β_{3s}	°	-60
Diameter's Ratio at output rotor	υ	--	0,2395
Total efficiency nozzle	η_b	%	90
Total-static efficiency rotor	η_{ts}	%	85
Clearance in the shroud rotor	f_{br}	mm	1
Molar Mass of gas	M_{mg}	Kg/kmol	102,03
Relation of specific heat	k	--	2,00
Total temperature inlet turbine (Nozzle)	T_{00}	K	363,5
Total Pressure inlet turbine (Nozzle)	P_{00}	kPa	3247,19
Mass rate flow	$\dot{m}$	Kg/s	4,5
Relatives speed ratio of rotor	w_r	--	3,28
Total- static expansion ratio of turbine	RP_{ts}	--	3,30
Thickness of blade rotor	t_r	mm	1
Thickness of blade nozzle	t_b	mm	1
Ideal clearance between rotor and nozzle	Δr	mm	2

The radial turbine simulation proposed here, following Miranda (2010) indications, consists in disassembling the equipment in sectors, whose operating characteristics are modeled in their operational characteristics and the output data of each component is converted into the input data for the next component Figure 4. The radial turbine project was divided into two analysis stages: a one-dimensional and a three-dimensional. Initially, the one-dimensional analysis was

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performed to determine the flow conditions through the turbine along the medium current line, in order to evaluate the components.

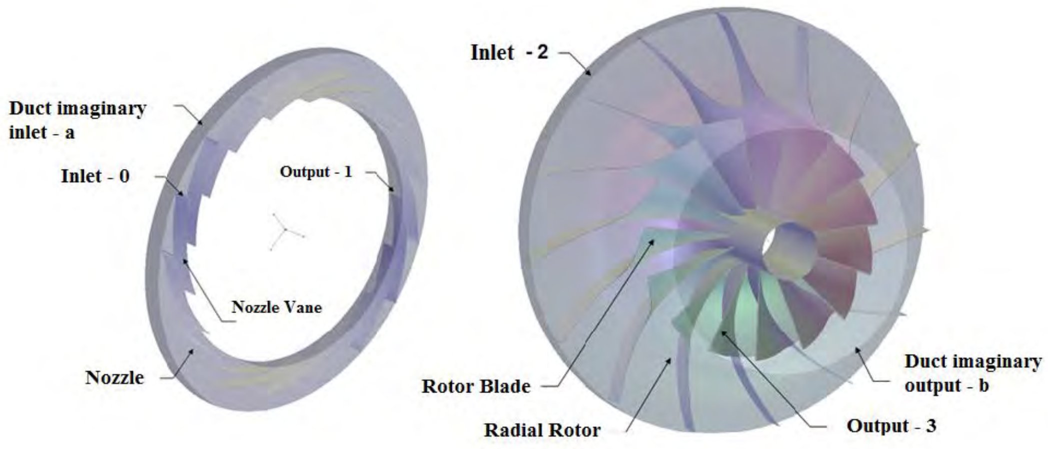


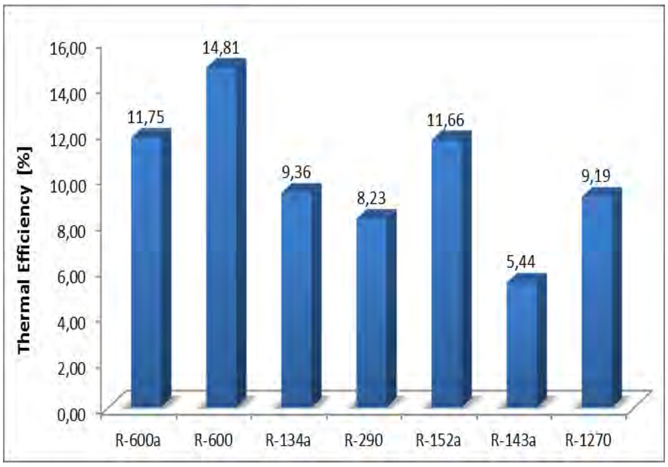
Figure 4 – Turbine's Nozzle and Rotor Three-Dimensional View (MIRANDA, 2010), ANSYS CFX 14.0®

5. RESULTS AND CONCLUSIONS

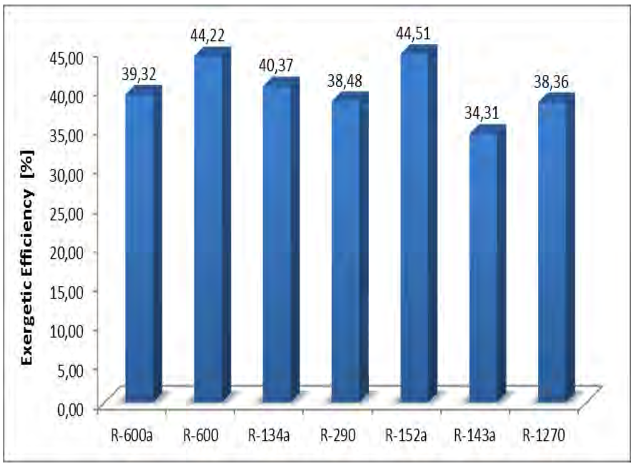
5.1 Thermodynamic Analysis

An appropriate choice of working fluids and conditions of operation is essential to the best performance of a cycle. This study was based on a comparison of different working fluids on conditions presented and discussed throughout the work, where the turbine inlet pressure was set at saturation point for all fluids. The intent of this cycle configuration step is to evaluate different working fluids to operate in a 1000 [kW] radial turbine, performing a cycle study based on the First and Second Laws of Thermodynamics, where the cycle operates with different fluids.

The working fluid that showed the highest thermal efficiency was R-600, followed by R-600a - both considered "dry" fluids, which require no overheating. In regards to the conditions set up for the cycle in this study, they admitted the highest turbine inlet temperatures, as shown in Figure 5a. ORC cycles operate in heat recovery from low quality energy sources. The temperature in these sources is not high, which gives these cycles greater exergetic efficiency than thermal. Figure 5b shows that R-152 has greater exergetic efficiency, followed by R-600.



(a)



(b)

Figure 5 - Result of Thermal Efficiency (a) and Exergetic Efficiency (b) for each fluid.

ISSN 2176-5480

In Table 3, it is observed that the greatest losses occur in the evaporator, for all fluids analyzed, contributing about 60% of cycle irreversibility. This analysis indicates that an evaporator optimization requires a later thermodynamic cycle improvement, seeking economically viable solutions to allow more net power, allying with smaller equipment irreversibilities.

Table 3 – Results of Irreversibility Calculation for each component.

	$I_{turbine}$		I_{pump}		I_{boiler}		I_{cond}	
	[Kw]	**[%]**	**[Kw]**	**[%]**	**[Kw]**	**[%]**	**[Kw]**	**[%]**
R-600a	149,5628	14,41	17,07745	1,65	705,84	68,01	165,3203	15,93
R-600	148,2364	17,74	13,90565	1,66	541,7204	64,84	131,6017	15,75
R-134a	159,1405	16,64	14,13067	1,48	603,1185	63,06	179,9889	18,82
R-290	160,5369	15,96	25,50258	2,54	618,6893	61,51	201,159	20
R-152a	156,5353	19,46	7,260675	0,9	495,4897	61,6	145,1076	18,04
R-143a	163,9942	12,98	19,4213	1,54	767,8606	60,76	312,4682	24,73
R-1270	149,0817	15,53	30,2123	3,15	601,5672	62,68	178,879	18,64

5.2 Turbine

Figure 6 shows a geometric configuration and a nozzle meridional plane representation of the simulated radial turbine. The analysis was evaluated at 50% of the blade height in the meridional plane of the nozzle.

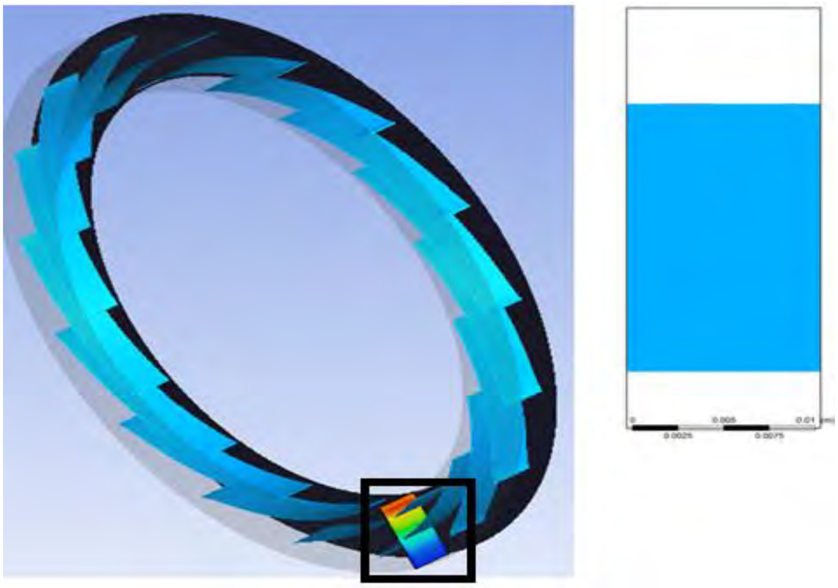


Figure 6 – Radial Turbine's Nozzle. Geometry and Meridional Plane ANSYS CFX 14.0®.

Figure 7a demonstrates the velocity vectors distribution in the trailing edge and on the surface at 50% of the nozzle blade height. There is a uniform behavior, without disturbance along the passage, in which the flow enters with low speed and then increases it in the nozzle blade output. Similarly occurs to Mach-number distribution, being low at the entrance and increasing towards the nozzle exit. It is observed that the flow in the nozzle is entirely subsonic, thus preventing clogging of the flow Figure 7b.

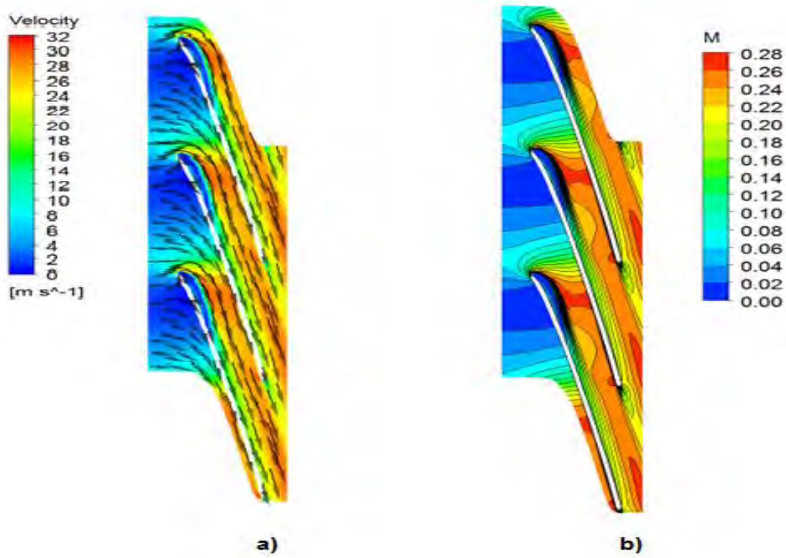


Figure 7 - Nozzle's Velocity Vector Distribution (a) and Mach Number (b). ANSYS CFX 14.0®.

The three-dimensional analysis is of vital importance to evaluate flow dynamics, allowing an in-depth turbine study, enabling geometry changes and also improving flow aerothermodynamics distribution.

The calculation program is a performance tool of great utility, especially in the project preliminary phase, as it can be used to quickly calculate and test different settings, even before any nozzle or rotor blade detail is manufactured. The use of this calculation program along with computational analysis can significantly reduce the number of laboratory tests and prototypes, thus reducing project costs and time. Miranda (2010). CFX proved to be a useful tool, enabling geometry changes and tests of the designed radial turbine, and therefore enabling adjustments to losses and improvements in the flow aerothermodynamics distribution. Based on the results observed, the one-dimensional analysis ratifies its Computational Fluid Dynamics three-dimensional analyses.

Specific diameter and specific speed are used to correlate turbines performances. Balje (1981) presents extensive analytical studies, which result in turbine efficiency charts, correlating specific diameter and speed for various turbines types. These charts are of utmost importance for turbines initial sizing and performance estimation Figure 8 (BALJE, 1981). The range of better use for radial turbine utilization is the specific speed in the range of $30 \leq S \leq 100$.

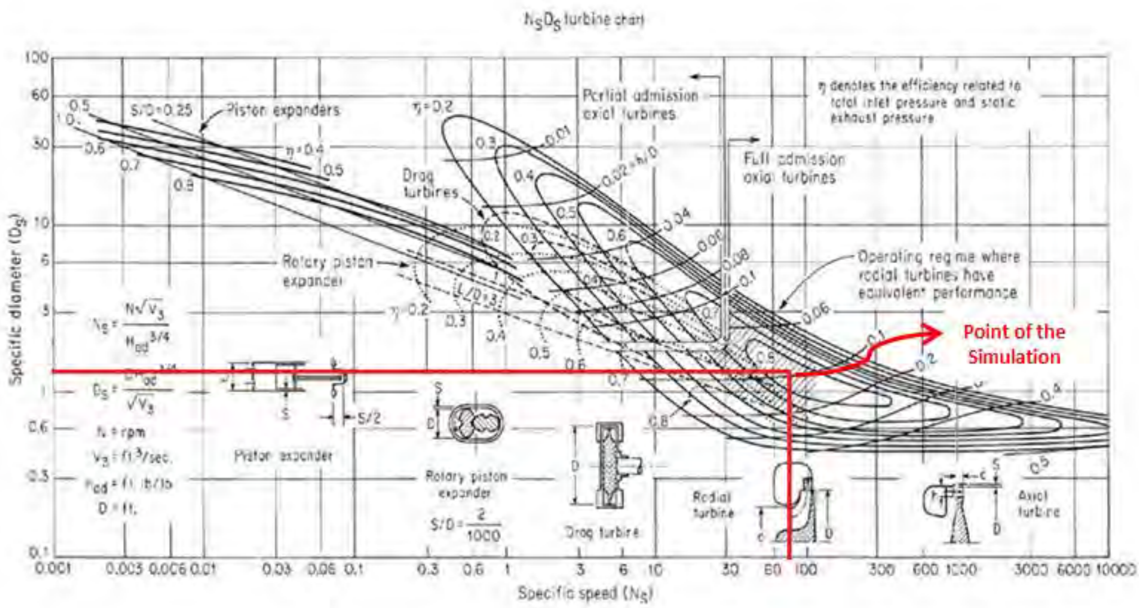


Figure 8 – Specific Diameter and Speed (BALJE, 1981).

ISSN 2176-5480

Through one-dimensional simulation, the diameter and specific rotation of the turbine were determined, parsed with 85% isentropic efficiency, as previously determined, obtaining Ns = 74, and Ds = 1.33. This simulation is shown in Figure 8. From this analysis and the characteristics of each working fluid studied in this cycle, the rotor diameter of a 1000 [kW] radial turbine was determined for each fluid - data shown in Table. 4.

Many studies evaluate thermodynamic cycles without a comprehensive equipment analyses. In determining initially the efficiency of the studied turbine, the fluids that provided greater thermal and exergetic performance, as R-600, R-600a, and R-152a also presented smaller's turbine's rotor.

Due to the capacity of removing more energy from the source, the R-600 fluid provided more thermic and exergetic efficiency to the cycle. According to the specified conditions it also provides great specific work from the turbine's expansion and less mass flow rate. That implies in a smaller turbine's rotor, according to the preliminary one-dimensional analyses.

Table 4 - Radial Turbine's Parameter at Simulation.

	R-600a	**R-600**	**R-134a**	**R-290**	**R-152a**	**R-143a**	**R-1270**
Δh	21952,27	29546,24	7544,09	12575,45	13997,49	3616,867	12522,71
V [ft^3/s]	6,2340	5,0961	8,8365	11,7127	6,9589	21,8068	10,8102
N [rpm]	53436	73853	20145	25670	36089	7388	26636
D [m]	0,09	0,07	0,13	0,13	0,10	0,25	0,13

The radial turbine analysis operating in the proposed cycle demands high rotation, by means of a gear box to reduce the speed or using a high rotation generator. The high rotation generators are prefered for being compact and also connecting directly to the turbine without the gear box. The advantages are the lower weight, higher efficiency and cheaper maintenance (AKIRA*, et al*; 2004).

The caracteristics of a saturation curve slope (T-s diagram), which defines dry, wet and isentropic fluids, is one of the principal factors for ORC cycle analysis. Nevertheless, latent heat is also regarded as an essential feature for working fluids evaluation. According to Chen, *et al;* (2010), fluids with higher latent heat provide greater power generation in the turbine, when temperature and other parameters are stable.

Other important properties are the specific heat and specific mass. A lower latent heat and lower specific heat provide more energy absorption from the source, consequently lower mass flow rate. That implies in smaller equipment's and less work performed by the pump. Higher specific mass results in more compact equipment's.

The analysed working fluids that present higher latent heat have also a higher temperature at the boiler exit. The turbine work was set at 1000 [kW], then these fluids will have less mass flow rate, enabling the usage of smaller equipment in the turbine. However, this examination must be thorough, both in relation to thermodynamic cycle and equipment. Tab.6 displays the studied ORC cycle thermodynamic analysis results, evaluating together the radial turbine rotor dimension for different working fluids in the new operation point.

Table 6 – Results Performance Analysis.

	ηth [%]	**ηII [%]**	**Latent Heat [kJ/kg]**	**T3 [°C]**	**Flow Rate [kg/s]**	**D [mm]**
R-600	14,80	44,22	336,82	135	13,46	0,07
R-600a	11,75	39,31	303,44	117	17,94	0,08
R-152a	11,66	44,50	249,67	101	28,13	0,10
R-1270	9,18	38,36	284,34	87	29,24	0,13
R-290	8,22	38,48	292,13	86	31,31	0,13
R-134a	9,35	40,37	155,42	82	52,29	0,13
R-143a	5,43	34,30	124,81	61	108,86	0,25

It can be concluded that the Organic Rankine cycle optimization process is indeed interactive and a necessary challenge for the application of this technology. Selecting the best operating conditions for distinct working fluids, through a thermodynamic analysis, not necessarily will result in more efficient turbine projects. According to Sauret and Rowlands (2011), this type of analysis can result in an overall lower efficiency and result in a lower net power produced, rather than the expected based on the previously presumed turbine yields.

Similarly, it is observed that different organic fluids, operating under the same fixed efficiency for the turbine, result in large physical dimensions differences for the radial turbine; what could limit a certain fluid selection for a given operating condition.

A turbine three-dimensional analysis is of vital importance to evaluate flow dynamics, allowing a turbine in-depth study, enabling geometry changes, and also improving the flow aerothermodynamics distribution.

6. ACKNOWLEDGEMENTS

The authors wish to thank to CAPES - Brazil, CNPq - Brazil (Proc. N° 310069/2012-2) and FAPEMIG - Brazil for their collaboration and support in the development of this work.

7. REFERENCES

Abu, 2011, "*The Organic Rankine Cycle Power Production From Low Temperature Heat*".
Akira Chiba, Rahman, *et al*. 2004. "*Super High Speed Electrical Machines*" – IEEE-PES, Meeting at Denver-Panel Session on Super High Speed Drive.
Balje, O. E. , 1981, "*A guide to Design, Selection, and Theory*".
Chen, H.; Goswami, D. Y.; Stefanakos, E. , 2010, "*A review of Thermodynamic Cycles and Working Fluids for the Conversion of Low-Grade Heat*". Renewable and SustainableEnergy Reviews.
Dórea, F.; Mendoza, O.; Martins, G.; Pitillo, J., 2006, "*Modelagem Semi-Empírica de Turbinas Radiais*". Simpósio de Pós-Graduação em Engenharia Mecânica, Uberlândia, n. 16.
Hung, T. , 2009, "*Waste Heat Recovery of Organic Rankine Cycle Using Dry Fluids*". Energy Conversion and Management.
Hung, T. C. , 2010 , "*A Study of Organic Working Fluids on System Efficiency of an ORC Using Low-Grade Energy Sources.*". Energy Conversion And Management, V. 42.
Manolakos, D.; Papadakis, S.; S., Kyristsis; Bouzianas, K. , 2007, "*Experimental Evaluation of an Autonomous Low-Temperature Solar Rankine Cycle System for Reverse Osmosis Desalination*".
Marcuccilli, F. , 2008, "*Advantages of using radial inflow turbines for kalina and organic*". *GRC Transactions*.
Mathis, d. M. , 1996, "*Fundamentals of Turbine Design*". Tempe, Arizona, USA.
Miranda, R., 2010, "*Projeto e Análise de Desempenho de Turbinas Radiais*", Brasil.
Moran, M.; Shapiro, H. , 2002, "*Princípios de Termodinâmica para Engenharia*" 4ª. ed. Editor Santuário.
Quolin, S.; Lemort, V. , 2009, "*Technological And Economical Survey Of Organic Rankine Cycle Systems*".
Sánchez, D.; Chacartegui, R.; Muñoz, A; Sánchez, T. , 2008, "*Parametric Analysis and Optimization of a High Temperature Fuel Cell e Supercritical CO2 Turbine Hybrid System*. " Proceedings of ASME turbo expo 2008: power for land, sea and air, GT2008-ASME, Berlin, German.
Sauret, E.; Rowlands, A. S. , 2011, "*Candidate Radial-Inflow Turbines And High-Density Working fluids For Geothermal*". *Energy*.
Shengjun, Z.; Wang, H.; Guo, T., 2010, "*Performance Comparison and Parametric Optimization of Subcritical Organic Rankine Cycle (ORC) and Transcritical Power Cycle System for Low-Temperature Geothermal Power Generation*", Energy, n. 88.
Silva, J. , 2010, "*Projecto de um Ciclo Rankine Orgânico para Produção de 1MW de Eletricidade*". The 9th Latin-American Cogress on Eletricity Generation and - CLAGTEE.
Utamura, M. , 2010, "*Thermodynamics analysis of part-flow cycle supercritical CO_2 gas*". Journal of Engineering for Gas Turbines and Power.
Van Wylen, J. G.; Sonntag, R. E.; Borgnakke, C. , 2003, "*Fundamentals of Thermodynamics*". 6ª. ed.
Vetter, Christian; Wiemer, Hans-Joachim; Kuhn, Dietmar; J. , 2013, *"Comparison of sub- and supercritical Organic Rankine Cycles for Power Generation from Low-Temperature/low-enthalpy Geothermal Wells, Considering Specific net Power Output and Efficiency*"
Yamamoto, T.; Furuhata, T.; Arai, N.; Mori, K. , 2001, "*Design and Testing of the Organic Rankine Cycle*". Energy, n. 26.

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