FLUID-DYNAMICS OF THE ORC RADIAL OUTFLOW TURBINE

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ABSTRACT

It is well documented that axial turbines and radial inflow turbines have traditionally been the selected solutions for ORC, both with an overhung configuration. In the last years a different turbine technology for ORC has been developed, engineered, manufactured and tested by Exergy: the radial outflow turbine.

In order to better understand its potential and limits, the present study has the purpose of conducting a fluid-dynamic study of the ORC radial outflow turbine. To pursue this aim, here firstly a summary description of the radial outflow turbine and of its features is given, by means of mechanical and thermodynamic fundamentals.

Secondly, moving from the hypothesis of direct coupling with generator, boundary conditions for a 2 MW case are chosen and a radial outflow turbine is studied, focusing on fluid-dynamic design: after a preliminary mean line study a CFD simulation of the machine is performed.

The described analysis includes also a comparison with an axial ORC turbine with the overhung configuration directed coupled with a generator: this approach could allow to valuate fluid-dynamic losses in both technologies and can explain the reason why the radial outflow turbine shows a higher efficiency than the axial overhung one in many ORC applications.

1. INTRODUCTION

In these days the ORC market is in expansion and as a matter of fact a rising interest about it can be observed in the scientific community. In terms of system performance, turbine is the most important and critical component of ORC systems (Macchi, 2013) and for this reason it is subject to many researches and studies.

An interesting innovation in this panorama is the ORC radial outflow turbine developed by Exergy, which has several unique characteristics qualifying this unconventional configuration as advantageous for many ORC applications, as it ideally matches the process conditions typical for these kinds of uses. In fact it has been demonstrated (Macchi, 2013 – Frassinetti et al., 2013) that this machine is competitive with both axial and radial inflow turbines, solutions usually adopted in ORC applications with the overhung configuration as it allows to have compact machines and to reduce sealing problems (Salucci et. al, 1983).

2. THE ORC RADIAL OUTFLOW TURBINE AT A GLANCE

To better understand the particular features of the radial outflow turbine (see Figure 1) some considerations about turbomachinery fluid-dynamics and about thermodynamics are necessary.

The well-known general Euler equation for turbomachinery, ignoring minor negligible losses, provides a formulation of the specific work for a single stage as (for instance Stodola, 1927):

$$\boldsymbol{l = u_1\ c_{t1} - u_2\ c_{t2}} \tag{1}$$

where:
- u is the Peripheral velocity,
- c_t is the Tangential component of the absolute velocity,
- 1 is the Inlet section of the turbine,
- 2 is the Outlet section of the turbine.

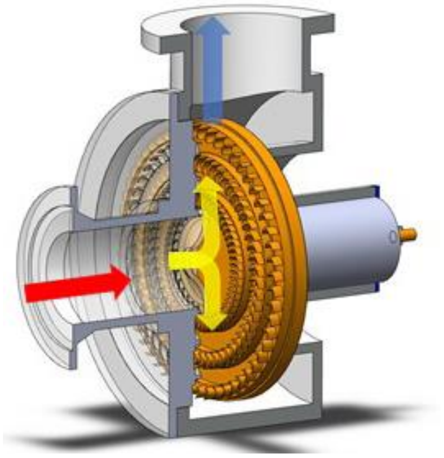

Figure 1: Radial outflow turbine

As a result of Equation (1) in order to maximize the specific work of a single stage, the first term should be significantly higher than the second: assuming that in a turbine C_{t1} must be much bigger than C_{t2}, the highest specific work of a single stage is thus achieved by the radial inflow configuration, which has intrinsically a higher peripheral velocity at the inlet and a lower one at the outlet.

The radial outflow configuration has instead a low specific work per stage due to the increase of the peripheral velocity while expanding the vapor ($U_1 < U_2$). Furthermore, from thermodynamics it is known that the expansion of fluids with low molecular weight, like water, at operative conditions which are typical for power production, is characterized by high enthalpy drops, high volumetric flows and high volumetric ratios (Poling et al., 2000).

Thus, the choice of the radial outflow turbine with water steam faces a serious limit: a significant number of stages is mandatory to convert the enthalpy drop of the fluid into mechanical energy. Owing to this reason, Ljungstrom developed his counter rotating radial outflow turbine configuration, in order to reduce the number of the turbine stages by increasing their specific work.

Moreover, due to the remarkable volumetric flow and its ratio between the inlet and outlet section (considerable for steam), the turbine blades would necessarily have a large height even for small power output turbines. Thus, for the very large diameter disk necessary to accommodate all the required stages and for the too long blades, the radial outflow turbine configuration demonstrated severe limitations while processing steam and was therefore deemed not suitable.

These issues led to no significant development of such type of turbine, which was phased out for steam applications by axial turbines.

At the operative conditions typical for ORC, fluids with high molecular weight lead to significantly lower enthalpy drops, volumetric flows and volumetric ratios than steam (Poling et al., 2000): this made possible for Exergy to reconsider the radial outflow turbine configuration for this application in binary power plants, as the intrinsic limits of this type of technology are no longer relevant.

If compared to traditional organic fluid turbines, meaning overhung axial turbines, Exergy overhung radial outflow turbine demonstrates several mechanical and fluid-dynamic differences hereby summarized.

2.1. Mechanical analysis

Axial turbines are characterized by having only one stage mounted on each single disk (in this paper called single-disk / single stage configuration). This arrangement in overhung axial turbines limits the number of stages, for rotordynamics reasons, to up to 3 stages. The radial outflow turbine allows instead to have several stages arranged on the same disk (see Figure 1).

The single-disk / multi stage configuration has thus the advantage of minimizing vibrations and static and dynamic loads on the bearings, due to the reduced distance between bearings and the turbine center of gravity. This makes possible to decrease the maintenance and to extend the useful life of the rotating components (see Figure 2).

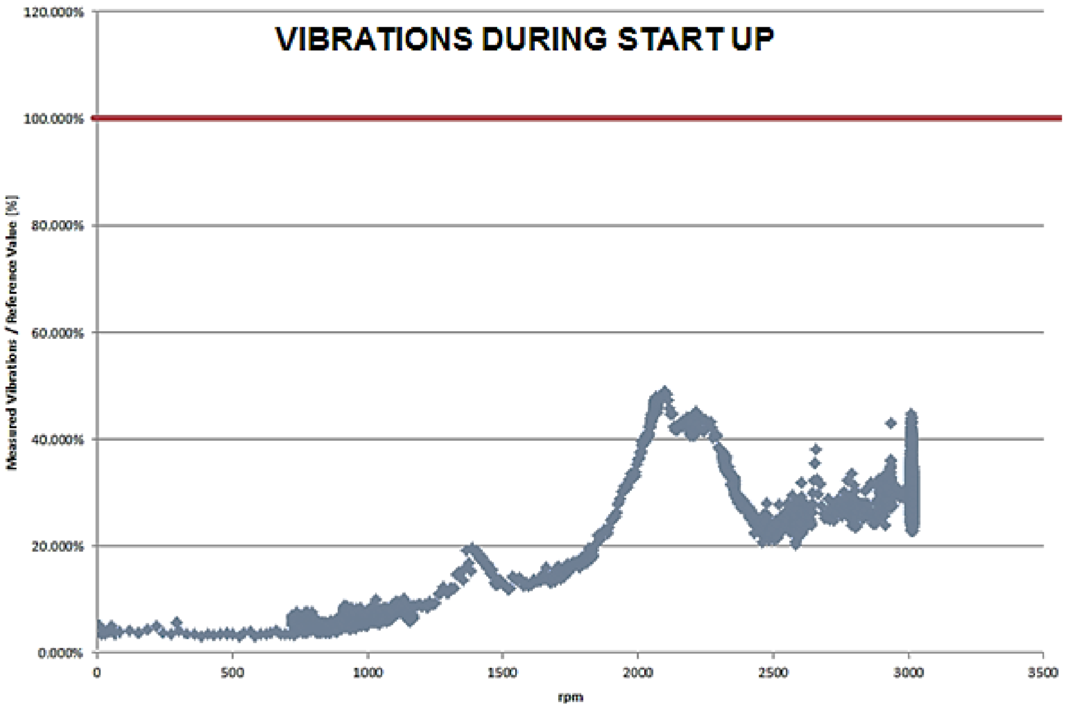

Figure 2: Vibrations during start up. Reference Value from ISO 10816-3 (From Frassinetti et al., 2013)

Finally, being the peripheral velocity constant along the blade span, velocity triangles at hub and tip do not change and the blades are prismatic instead of twisted.

2.2. Fluid-dynamic analysis

Having a cross section increasing proportionally to the radius, during the expansion the radial outflow turbine matches the volumetric flow behavior better than the axial turbines, which usually require high flaring angles. This means that it is possible to have lower blades at the last stage, leading to evident mechanical advantages, and higher blades at the first stages, thus reducing the endwall and leakage losses (for instance Sieverding, 1985 - Sharma and Butler, 1987 or Duden et al, 1999).
For these reasons, as initial stages have a better aspect ratio, they do not need partial admission, avoiding additional losses related to this aspect (Suter and Traupel, 1959 – Horlock, 1966). As a consequence, the possibility to manage higher volumetric flow ratio allows to have a higher pressure at turbine inlet while keeping the same condensing pressure, therefore giving the opportunity to increase the thermodynamic cycle efficiency.
Finally, as the enthalpy drop of the fluid is divided on several stages for the single-disk / multi stage configuration, the radial outflow turbine is characterized by a better recovery factor (Horlock, 1966 - Dixon, 1998 - Lakshminarayana, 1986 or Moustapha et al., 2003) and by lower stage work coefficients. This results in a subsonic or at most transonic expansion (in spite of the low speed of sound of organic fluids), instead of supersonic one typical of the other configuration and in a higher fluid-dynamic efficiency both in nominal and off-design conditions.

3. THE FLUID-DYNAMIC STUDY

In the last years a rising interest in the radial outflow turbine has been noticed and some studies about its fluid-dynamics have been proposed (Spadacini et al., 2011 - Pini et al., 2013 - Persico et al., 2013 - Spadacini et al., 2013).
Until today this configuration has been studied alone. In this work we would like to focus our attention on a comparison between the radial outflow and the axial configurations, considered as the reference technology.

3.1. Case study description

To perform the above mentioned comparison a common case study has been used for both configurations. The boundary conditions are listed in Table 1.

Table 1: Boundary conditions for common case study

Fluid	[-]	Pentane
Turbine Inlet Pressure	[bar]	10.3
Turbine Inlet Temperature	[°C]	130
Turbine Outlet Pressure	[bar]	1
Turbine Mass Flow Rate	[kg s^-1]	25

In order to simplify the mechanical configuration of the system and to reduce costs of investment, operation and maintenance due to the presence of gear, a direct coupling with the generator is chosen: so both turbines must rotate at 3000 rpm.

An in-house made 1D code has been used to design the two machines: the code simulates the expansion of the working fluid in the turbine and it uses the losses model by AMDCKO (Ainley and Mathieson, 1951 – Dunham and Came, 1970 – Kacker and Okapuu, 1981); thermodynamic properties are calculated with Refprop 9.1 (NIST, 2013).

As output the radial outflow turbine results in having 5 stages and producing an estimated power of 1924 kW, with an isentropic efficiency of 85.50%; the axial turbine, with 3 stages, has instead the power of 1852 kW and an efficiency of 82.30%.

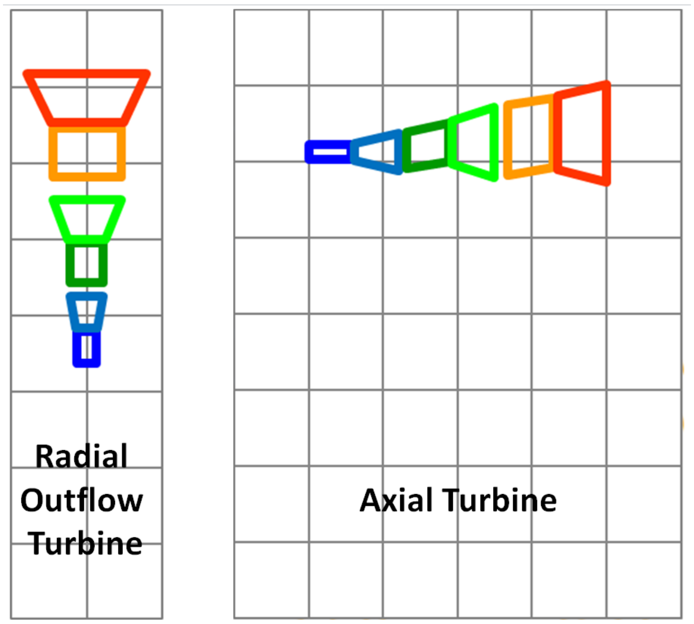

Figure 3: Example of modeled geometry for the radial outflow turbine (left) and axial turbine (right) – From Spadacini et al., 2011

In Table 2 the main results of the 1D analysis are provided respectively for the radial outflow and axial turbine.

Table 2: Main results of 1D mean line analysis

		Radial Outflow Turbine	Axial Turbine
Profile and T.E. losses	[%]	3.572%	5.353%
Endwall losses	[%]	6.635%	6.925%
Leakage losses	[%]	2.974%	4.110%
Disk friction	[%]	1.332%	1.332%

In Table 2 four categories of fluid-dynamic losses are listed: the profile and trailing edge losses and the endwall ones are defined as in AMDCKO model (Ainley and Mathieson, 1951 – Dunham and Came, 1970 – Kacker and Okapuu, 1981).
Leakage losses are calculated as indicated by Egli (Egli, 1935) and disk friction as Daily and Nece (Daily and Nece, 1960): to compute these losses, clearances are supposed to be the same for both configurations.

Important conclusions can be deduced by analyzing results in Table 2.
In fact profile losses in the axial configuration are higher than in the radial outflow, as the first results to have its stages more loaded than the other one with a highly supersonic flow; on the other side the expansion in the radial outflow turbine is at most transonic.
The axial turbine has also higher endwall losses, as a consequence of the lower h/c and h/d ratios (Traupel, 1966 – Horlock, 1966 - Dixon, 1998 - Lakshminarayana, 1986 or Moustapha et al., 2003).
Finally, as axial blades are smaller than the radial ones and because of the lower h/d ratio, leakage losses are minor in the radial outflow turbine.
As a consequence of this considerations, for the present case study the radial outflow turbine results to have a higher efficiency than the axial one.

3.2. CFD simulation

After the conclusions reported above, the comparison must be continued; for this reason a viscous 3D CFD analysis of both turbines is performed. The code employed in this step is ANSYS CFX.
In both of the domains periodic boundary conditions are used and for interface between the two cascades the stage method is utilized for modeling frame change (ANSYS, 2013).

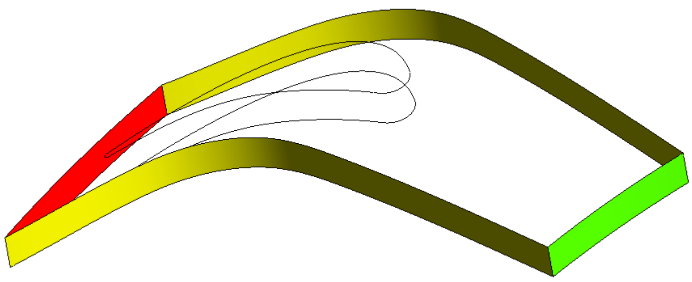

Figure 4: Interfaces for each cascade studied (periodic boundary conditions on yellow surfaces)

To generate blades of both the turbines an in-house database has been used. For thermodynamic properties look-up tables (Pini et al., 2015) are compiled with data from Refprop 9.1 (NIST, 2013).
The used turbulence model is the k-omega SST with the high resolution advection scheme (ANSYS, 2013).
In a first step for both the turbines a computational grid of about 2 million nodes and 7.5 million of tetra and hexa elements has been used; mesh has been then refined to about 8.5 million nodes and 30 million elements with differences in calculated results below 1%. Skewness has been always below 0.9 and maximum Aspect Ratio results to be 62.

3.3. CFD results

The hypotheses just described lead to the following results: the radial outflow turbine with a mass flow rate of 25 kg s^-1 produces 1938 kW, while the axial produces 1857 kW; the calculated isentropic efficiencies are respectively 86.10 % for the first and 82.5 % for the second one.

A piece of information about the expansion in both of the turbines can be extracted from Figure 5 and Figure 6, which confirm that stages in the radial outflow turbine are at most transonic, while in axial turbine they are highly supersonic.

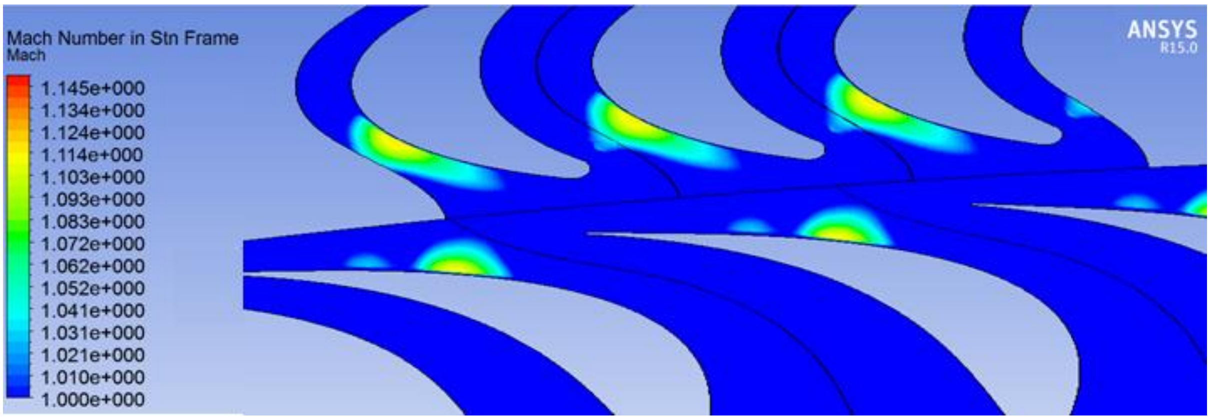

Figure 5: Maximum absolute Mach number at the outlet of stator vanes in the radial turbine

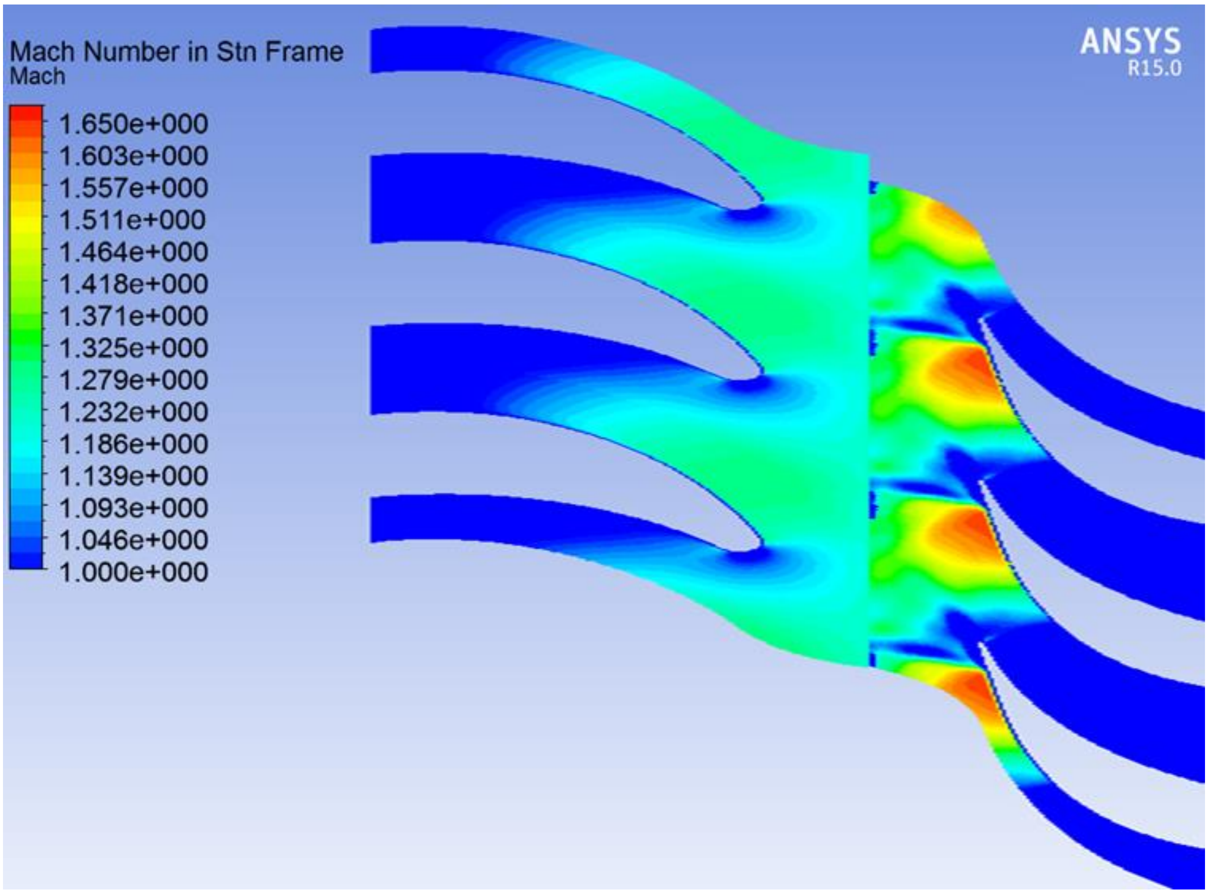

Figure 6: Maximum absolute Mach number at the outlet of stator vanes in the axial turbine

To complete information given by Figure 5 and Figure 6, in the radial turbine the maximum expansion ratio is about 1.6 while in the axial turbine the average value is about 2.8.

To analyze dissipations and losses during the expansion process in both turbines two parameters are chosen: entropy (Figure 7 and Figure 8) and turbulent kinetic energy (Figure 9 and Figure 10).

In fact from thermodynamics it is known that the entropy generated in a process is a measure of the internal irreversibilities (Moran and Shapiro 2004 or Gyftopoulus and Beretta, 1991, for instance). In turbomachinery the sources of entropy are in general viscous effects due to boundary layer and mixing processes and shock waves (Denton, 1993): all these phenomena lead to the losses discussed before in Paragraph 3.1.

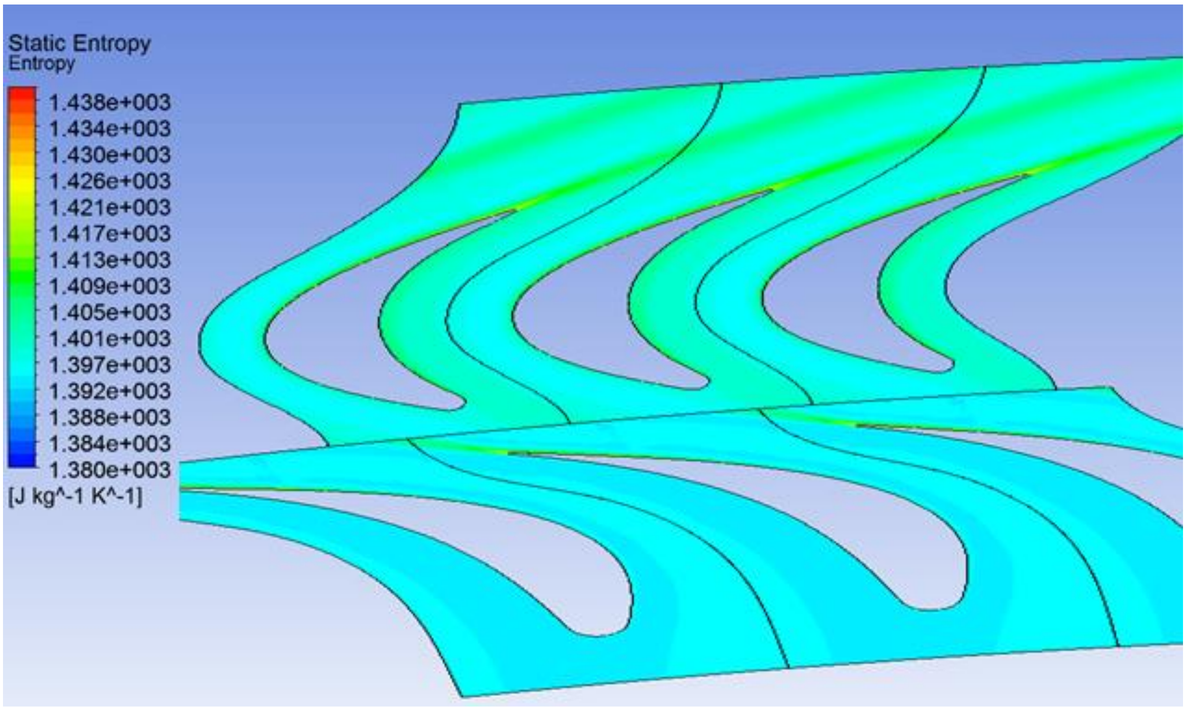

Figure 7: Generated entropy at the last stage of the radial turbine

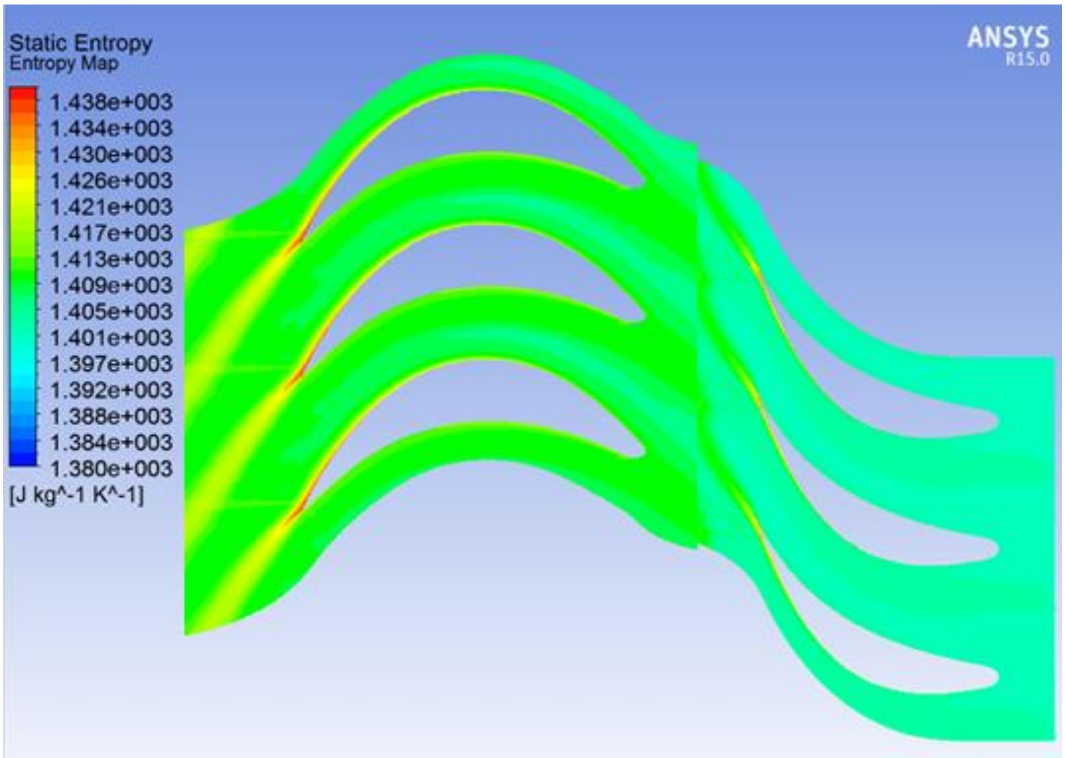

Figure 8: Generated entropy at the last stage of the axial turbine

In addition another important piece of information about losses and dissipation can be deduced from the turbulent kinetic energy distribution. In fact, turbulent kinetic energy (Hinze, 1975) is considered to be an indicator of viscous actions on total pressure losses (Gregory-Smith, 1988 - Moore, 1987).

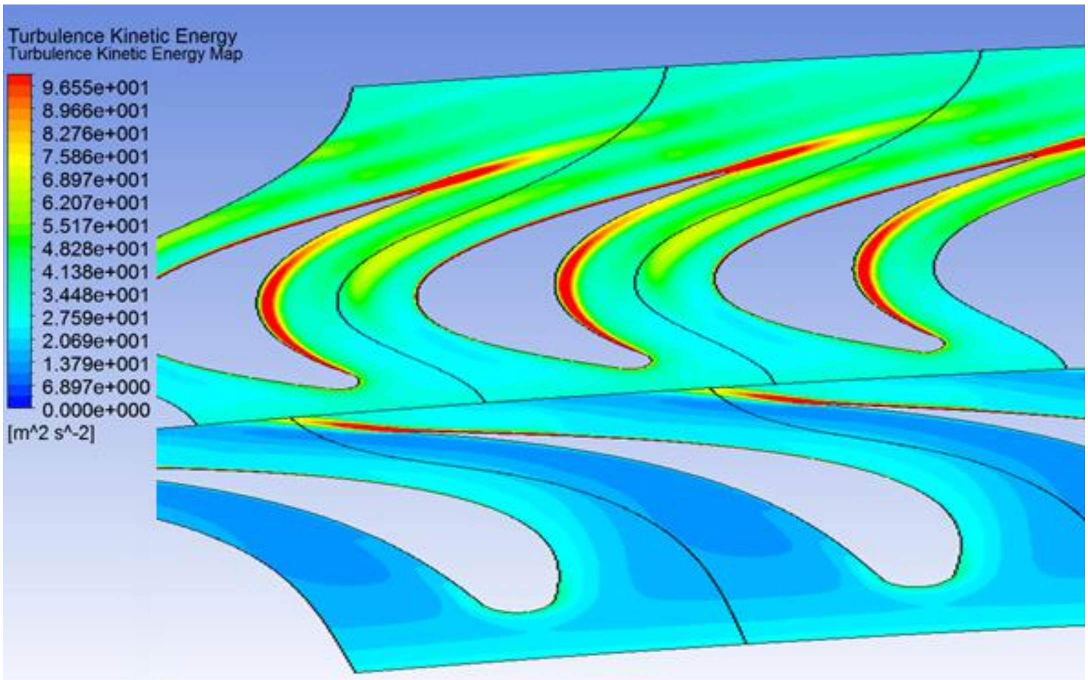

Figure 9: Turbulent kinetic energy at the last stage of the radial turbine

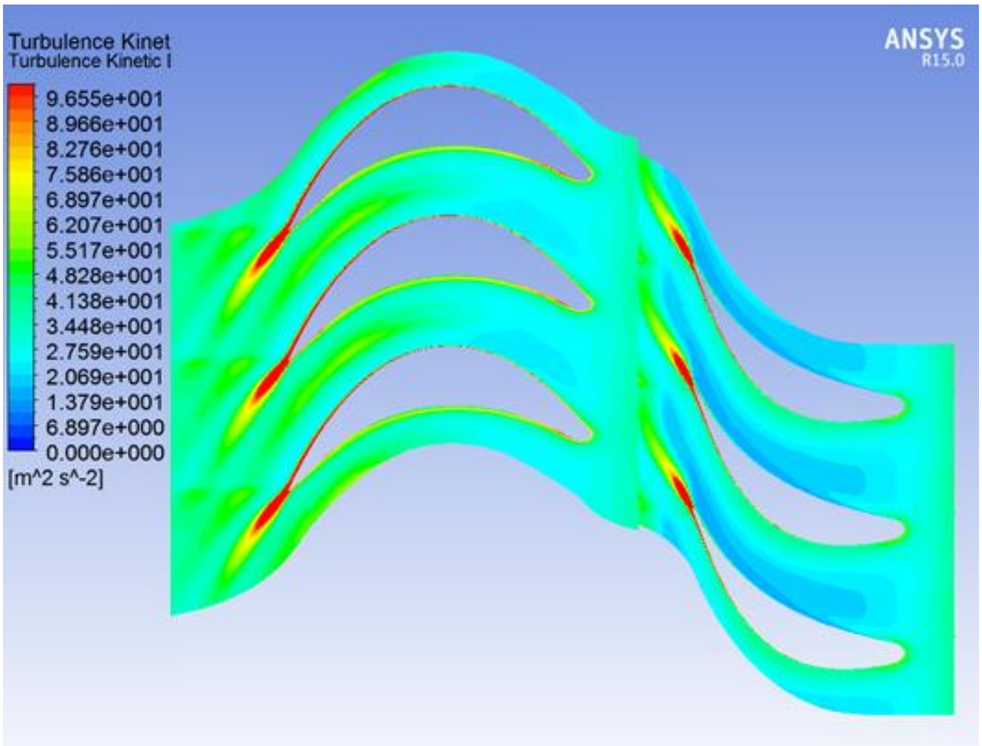

Figure 10: Turbulent kinetic energy at the last stage of the axial turbine

4. CONCLUSIONS

It has been highlighted that the radial outflow turbine is a solution well matching with the typical ORC process characteristics, i.e. high molecular weight, low speed of sound and limited volumetric expansion ratio. Furthermore, for a 2 MW case with the rotational speed of 3000 rpm, a study of the fluid-dynamics of this configuration has been carried out, firstly with mean line 1D methods and then with a CFD 3D simulation. As a result of this analysis, because of the feature of having more stages on a single disk, fluid expansion is characterized by lower losses and dissipations compared to the axial turbine. Consequently, the radial outflow turbine in some ORC applications can be more efficient than the axial one.
Future development of this work should focus on investigating the behavior of the ORC radial outflow turbine in off-design conditions, better if by a comparison with the axial overhung one.

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