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SMALL-SCALE ORC ENERGY RECOVERY SYSTEM FOR WASTED HEAT: THERMODYNAMIC FEASIBILITY ANALYSIS AND PRELIMINARY EXPANDER DESIGN

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Summary

- Objectives of the Work
- Organic fluids
- Thermodynamic simulations of ORC cycle
- ORC expander design
  - Blade design
  - Structural analysis
- Results and Conclusions
- Future developments

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Objectives of the Work

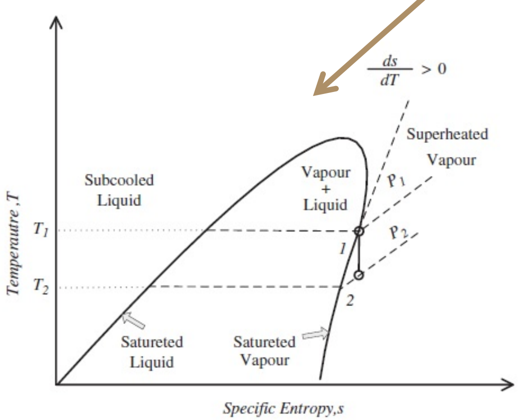
- Thermodynamic feasibility of an innovative Organic Rankine Cycle (ORC) recovery system for automotive applications;
  - a small-scale ORC energy recovery system fed by the exhaust gases of a diesel engine or GT device;

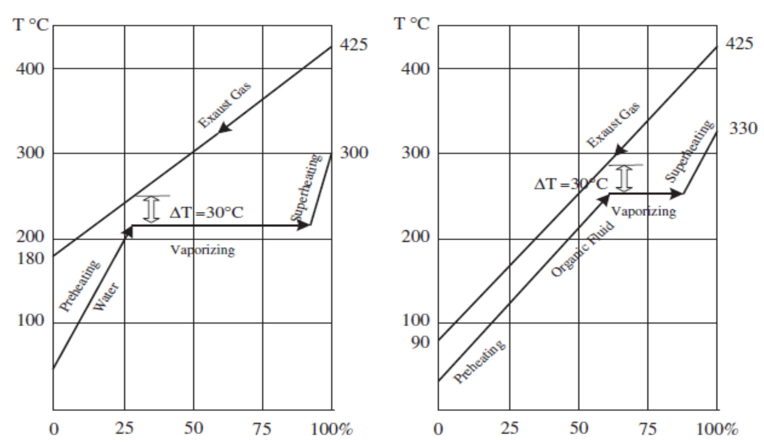
- Preliminary design of the expander→ dynamic turbine
  - single-stage radial turbine

Organic fluids - 1

Organic fluids present interesting thermodynamic properties for heat recovery from low temperature sources compared to water they have:

1. a lower vaporization heat;
2. a lower vaporization temperature at the same pressure than water;
3. an high heat capacity due to their molecular length.
4. some of them have a positive slope of the saturation vapor curve;





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Organic fluids - 2

- First three properties allow to use low temperature sources, the fourth consents to have an expansion from a non-superheated vapor point without enter in the vapor dome.
- This aspect is fundamental, because is not necessary to superheat the fluid because at the end of the expansion the fluid is still in the vapor phase avoiding any problems in the expander, especially if it is a turbine.
- In an ORC design is the choice of the fluid based on the source; in practice, most used fluids are three:
  1. R134a (1,1,1,2-Tetrafluoroethane)
  2. R123 (2,2-Dichloro-1,1,1-trifluoroethane)
  3. R245fa (1,1,1,3,3-Pentafluoropropane)

Organic fluids - 3

- In addition to the thermodynamically properties, the organic fluids for ORC have to be secure for people and environment:

1. No flammable
2. No explosive
3. Non-toxic
4. Low Ozone Depletion Potential (ODP)
5. Low Global Warming Potential (GWP)

Organic fluids - 4

After various tests, which were not reported, but that were made in previous papers has been chosen the fluid→ R245fa (1,1,1,3,3-Pentafluoropropane)

Properties of HFC-245fa	
Chemical Name	1,1,1,3,3-pentafluoropropane
Molecular Formula	CF3CH2CHF2
Molecular Weight	134
Flammability Limits in Air @ 1atm** (vol.%)	None
Flash Point *	None
Water Solubility in HFC-245fa	1600 ppm
ASHRAE Safety Group Classification	B1

**Flashpoint by ASTM D 3828-87; ASTM D1310-86*
***Flame Limits measured at ambient temperature and pressure using ASTM E681-85 with electrically heated match ignition, spark ignition and fused wire ignition; ambient air.*

Standard International Units*		English Units*	
Boiling Point °C @ 1.01 bar	15.3	Boiling Point (°F) @ 1atm	59.5
Freezing Point °C @ 1.01 bar	<-107	Freezing Point (°F)	<-160
Critical Temperature** (°C)	154.05	Critical Temperature** (°F)	309.29
Critical Pressure** (bar)	36.4	Critical Pressure** (psia)	527.9
Critical Density** (m3/kg)	517	Critical Density** (lb/ft3)	32.28
Vapor Density @ Boiling Point (lb/ft3)	5.921	Vapor Density @ Boiling Point (lb/ft3)	0.3697
Liquid Density (kg/m3)	1339	Liquid Density (lb/ft3)	83.58
Liquid Heat Capacity (kJ/kg K)	1.36	Liquid Heat Capacity (Btu/lb °F)	0.33
Vapor Heat Capacity @ constant pressure, 1.01 bar (kJ/kg K)	0.8931	Vapor Heat Capacity @ constant pressure, 1atm (Btu/lb °F)	0.218
Heat of Vaporization at Boiling Point (kJ/kg)	196.7	Heat of Vaporization at Boiling Point (Btu/lb)	84.62
Liquid Thermal Conductivity (W/m K)	0.081	Liquid Thermal Conductivity (Btu/hr ft °F)	0.0468
Vapor Thermal Conductivity (W/m K)	0.0125	Vapor Thermal Conductivity (Btu/hr ft °F)	0.0072
Liquid Viscosity (mPa s)	402.7	Liquid Viscosity (lb/ft hr)	0.9744
Vapor Viscosity (mPa s)	10.3	Vapor Viscosity (lb/ft hr)	0.025

**Properties at 77 °F / 25 °C unless noted otherwise*
***NIST Refprop v 7.0*

R245fa

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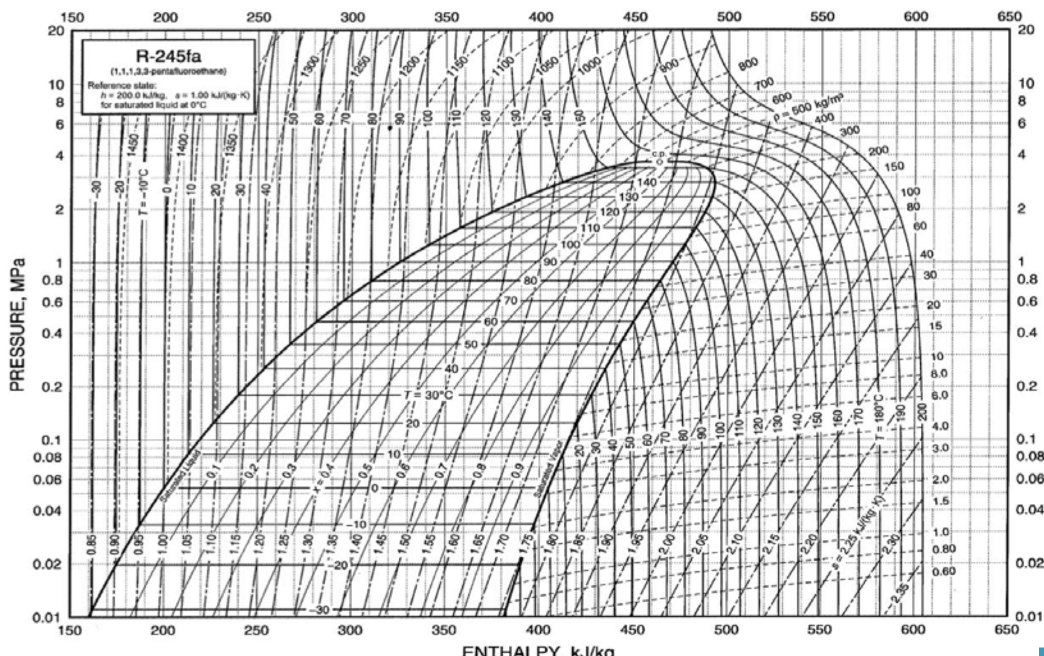
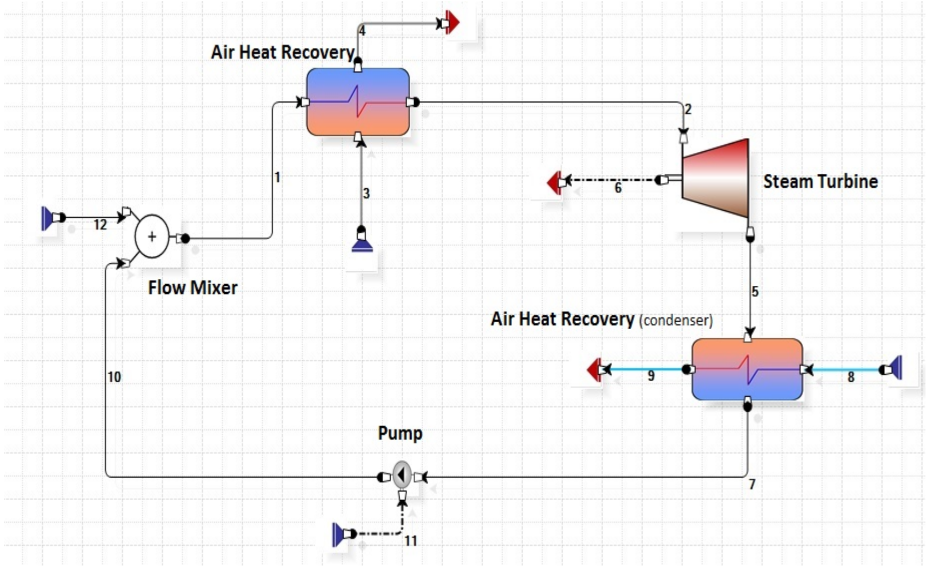
- A positive slope of the saturated vapour curve
- No superheating needed
- It doesn't damage the ozone layer
- It's nearly non-toxic
- No degradation at the temperature of the cycle



Pressure-Enthalpy Diagram for Refrigerant 245fa

Thermodynamic simulation -1

- The thermodynamic simulation of the ORC cycle was performed by CAMEL-Pro™ process simulator, a software developed at the Mechanical Engineering Department of the University of Rome "*Sapienza*".



Input constrains :

- The output power from the turbine (flow 6): 5kW
- The available mass flow rate of the exhaust gas (flows 3 and 4): 0,16 m^3/s
- Temperature of the exhaust gas (flows 3 and 4): 848K

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Thermodynamic simulation -2

- From the thermodynamics tables for R245fa this fluid condenses at 301K with a pressure of 170kPa. Since the expander is a single-stage turbine, it has a limit in the pressure ratio, for this reason the upstream pressure is fixed too.
- Analyzing the very few models available a β = 3,2 has been considered which means a $p_{upstream} = 550kPa$.
- The turbine adiabatic efficiency has been underrated at $\eta_{ad} = 0.75$ to be sure that the outlet power was at least 5 kW and the efficiency of the pump is fixed to 0,9.

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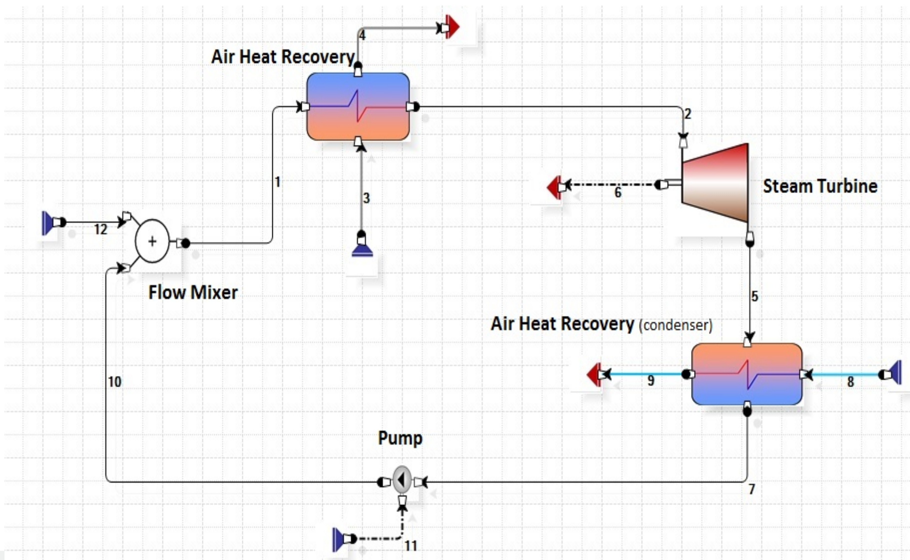
Thermodynamic simulation -3

The state properties of the computed ORC cycle



Working Fluid			R245fa		
State	m [kg/s]	p [kPa]	T [K]	h [kJ/kg]	s [kJ/kgK]
1	0.293	550	301	-187.2	-0.656
2	0.293	550	358	49.1	0.04
5	0.293	170	334	31.6	0.06
7	0.293	170	301	-187.5	-0.656
10	0.293	550	301	-187.2	-0.65

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ORC Expander
Briefly considerations

- The expander is the fundamental component of the ORC and several types of expander are used:

  - **Turbines**
  - **Scroll expanders**
  - **Screw expanders**

Turbines are not very suitable for ORC especially for small-scale plants, but they have the advantage of small dimensions

Volumetric machines remain the best choice.

Scroll expanders are developed from scroll compressors operating in an expander mode.

Screw expanders have the advantage of a simple architecture and they can achieve an outlet power above 20kW

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ORC Expander Design

Objective of the Work →feasibility of using a dynamic expander in ORC cycles

The main challenge of the analysis are the imposed size and weight limitations that require a particular design.

Proposed expander design method -1

- The procedure adopted to design the radial turbine is based on Rohlik study
- The three independent variable from which started Rohlik optimization procedure are:

$$52° < \alpha_1 < 83°$$

$$0.04 < h_1/D_{2,m} < 0.68$$

$$0.2 < D_{2,m}/D1 < 0.6$$

- 0 stator inlet, 1 rotor inlet and 2 rotor outlet. The angle α_1 is the complementary angle formed by the direction of U and V in the velocity triangle. h_1 is the height of the rotor blade, D_{2m} diameter of the midspan section at the exit of the rotor and, finally, D_1 is the diameter at the rotor inlet.

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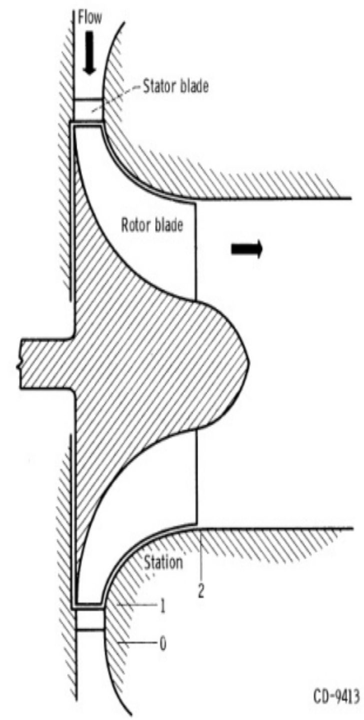
Proposed expander design method -2

- The losses considered are stator loss, rotor loss, tip-clearance loss, windage and exit kinetic energy

- Rohlik developed some charts that show optimal geometric and kinematic parameters for every specific speed n_s.

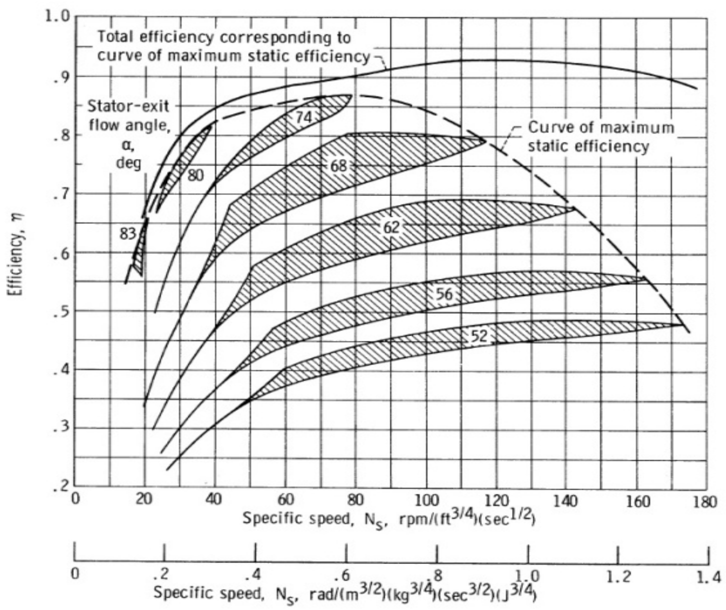
- n_s. is expressed through the three independent variables :

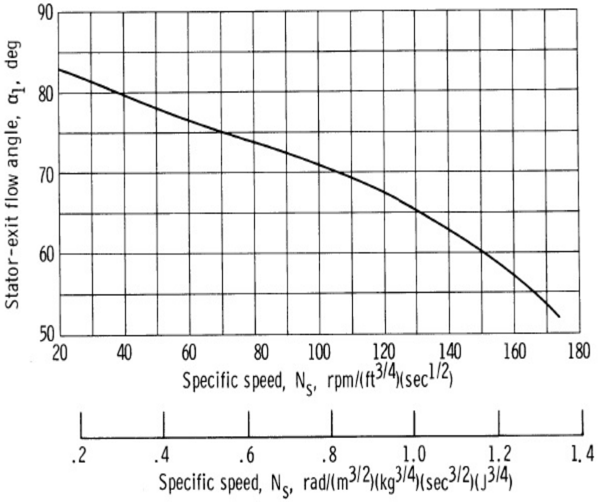
$$n_s = \frac{60*(2g)^{3/4}}{\sqrt{\pi}} * \left(\frac{\Delta h}{\Delta h'}\right)^{3/4} * \left(\frac{u_1}{V_j}\right)^{3/2} * \left(\frac{V_2}{u_2}\right)^{1/2} * \left(\frac{D_{2,m}}{D_1}\right)^{3/2} * \left(\frac{b_1}{D_{2,m}}\right)^{\frac{1}{2}}$$



Proposed expander design method -3

Rohlik chart for blade design





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Proposed expander design method -4

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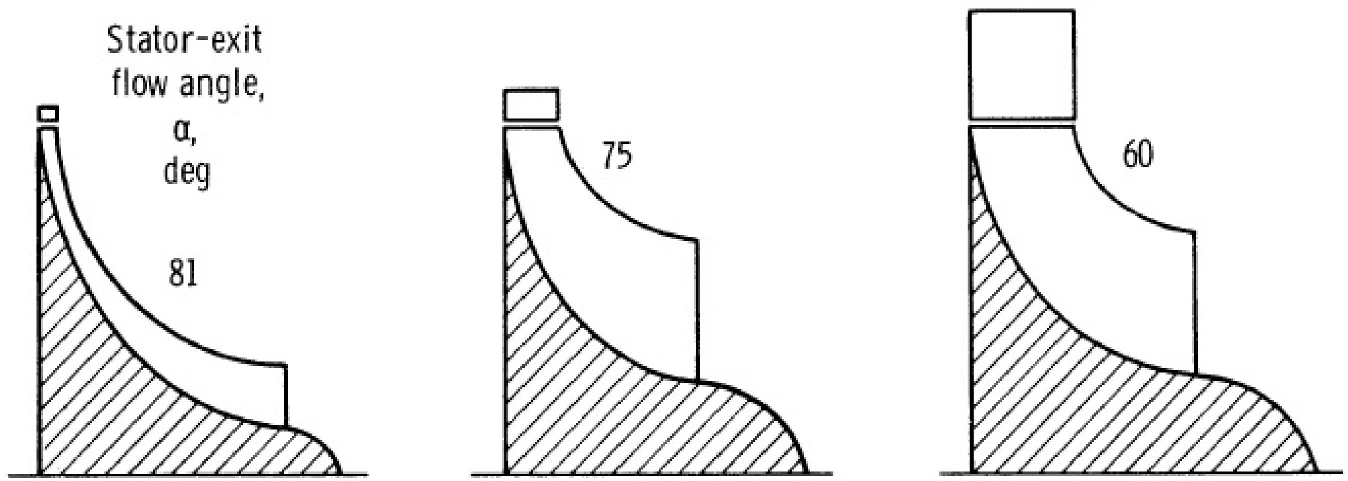
- The charts show the optimal trend, as a function of specific speed, of the ratio of the stator blade height to rotor inlet diameter b_1/D_1 and the ratio of rotor exit tip diameter to rotor inlet diameter D_{2e}/D_1.

- There is an upper limit on D_{2e}/D_1 which must not exceed 0,7. Finally, Rohlik also presented three optimum turbines sections geometry, that correspond to the curve of maximum static efficiency at three values of specific speed.

Proposed expander design method -5

Optimum turbine geometry by Rohlik



(a) Specific speed, 30 rpm per foot$^{3/4}$ per second$^{1/2}$ (0.23 rad/(m$^{3/2}$)(kg$^{3/4}$)(sec$^{3/2}$)(J$^{3/4}$)); blade-jet speed ratio, 0.68.

(b) Specific speed, 70 rpm per foot$^{3/4}$ per second$^{1/2}$ (0.54 rad/(m$^{3/2}$)(kg$^{3/4}$)(sec$^{3/2}$)(J$^{3/4}$)); blade-jet speed ratio, 0.70.

(c) Specific speed, 150 rpm per foot$^{3/4}$ per second$^{1/2}$ (1.16 rad/(m$^{3/2}$)(kg$^{3/4}$)(sec$^{3/2}$)(J$^{3/4}$)); blade-jet speed ratio, 0.62.

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Preliminary design of radial turbine for ORC-1

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- Setting the degree of reaction R_ρ at 0.5, pressure, enthalpy, temperature and, consequently, density at rotor inlet section have been calculated.

Rotor Inlet β = 3.2		
h_1	40298	J/kg
p_1	3,5	bar
T_1	344	K
ρ_1	17,8	kg/m^3

- There are five degrees of freedom in the design procedure. Setting $\psi_2 = 0$, in order to optimize Euler work, and R_ρ= 0.5, that means $\psi_1 = 1$ and radial blade at the rotor inlet because of structural reason and in order to optimize efficiency leading to Chen and Baines chart, has been calculated the blade speed.

Preliminary design of radial turbine for ORC-2

- Inlet section

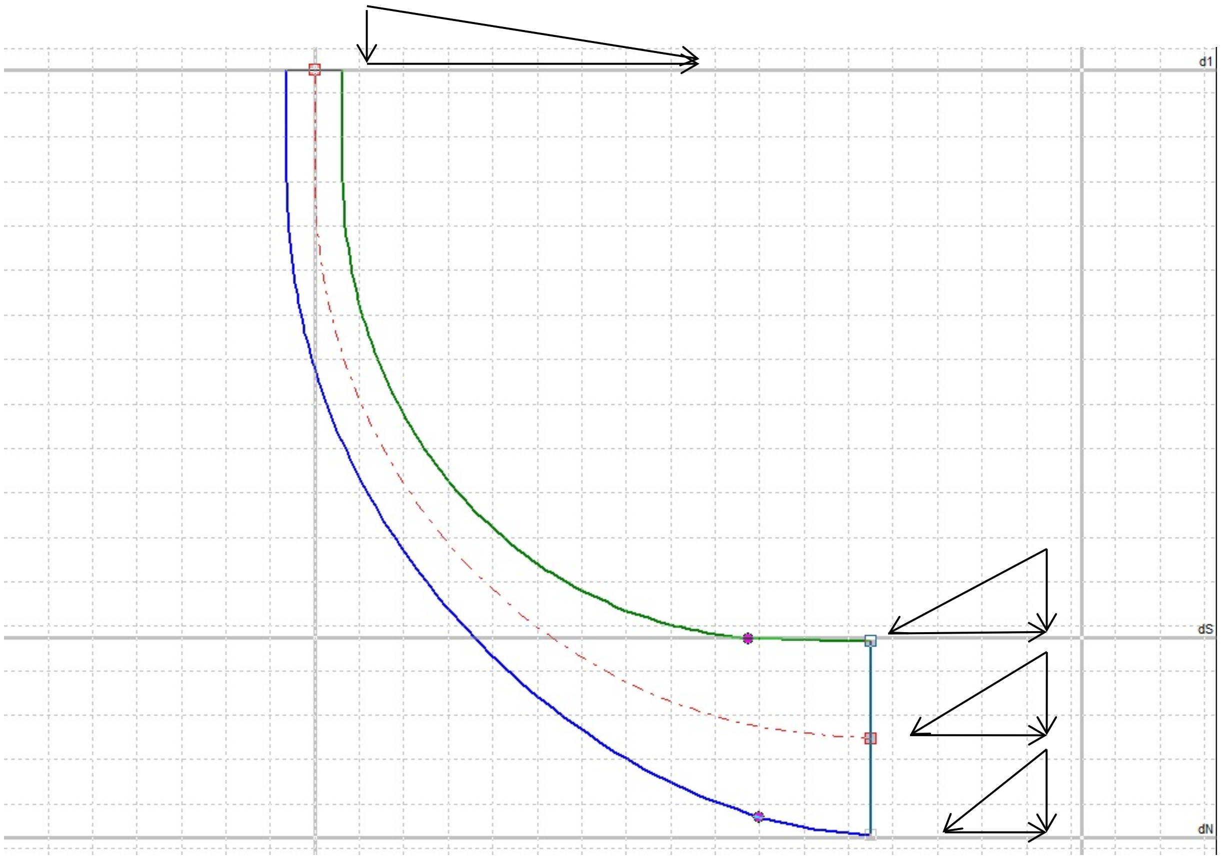
D_1 [m]	0.1	U_1 [m]	131
(90- Rohlik) [°]	10	V_{1t} [m]	131
φ_1	0.18	W_1 [m]	23
ψ_1	1	V_1 [m]	133
b_1 [m]	0.0025	V_{1m} [m]	23
b_1/D_1	0.025	Ma_{1v}	0.9
		β_1 [°]	90

- Outlet section

φ_c	0.25	V_{2m} [m/s]	32
D_{2m}/D_1	0.4	U_{2m} [m]	52
D_{2m} [m]	0.04	W_{2m} [m]	62
D_{2i} [m]	0.031	β_2 [°]	32
D_{2e} [m]	0.049	φ_2	0.63
D_{2e}/D_1	0.5	ψ_2	0
b_2 [m]	0.0091	χ_2	0.6

Preliminary design of radial turbine for ORC-3

- Blade shape and velocity triangles



Blade profile - 1

- First, the mean line has to be defined: we chose to use a portion of straight line and an arc linked. 20 points as equidistant as possible are considered.

- having the y coordinates of each point, it is possible to calculate the rotational speed at each point.

- Knowing the initial value W_1 and the final value W_2 of the relative velocity, a linearly increasing trend of W along the mean line of the blade has been assumed.

- The value of the angle β at any point of the mean line has imposed

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Blade profile - 2

- For temperature a linear progression along the mean line (the rotor inlet temperature, pressure and density have been calculated from the degree of reaction, entering in the *Mollier* chart of R245fa) are adopted, while pressures were determined through the relationship between pressure and temperature in the case of the polytrophic expansion.

- The points defining the hub and shroud lines were obtained respectively by subtracting and adding to the coordinates of the mean line, previously defined, the half-height of the blade profile in a direction perpendicular to the tangent to the curve.

- Finally, to maintain a certain uniformity of the meridian velocity and a constant Euler work on the section, the blades necessary must be twisted.

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Blade profile - 3

Design procedure results

			HUB						TIP			
p.	V_t [m/s]	U [m/s]	W [m/s]	V [m/s]	β [°]	α [°]	V_t [m/s]	V [m/s]	W [m/s]	V [m/s]	β [°]	α [°]
0	131,3	131,3	23,1	133,3	90,0	10,0	131,3	131,3	23,1	133,3	90,0	10,0
1	118,1	118,2	27,5	121,3	90,0	13,1	118,1	118,2	27,5	121,3	90,0	13,1
2	112,7	112,2	29,4	116,5	91,1	14,6	112,1	112,7	29,4	115,9	88,8	14,7
3	106,1	106,2	31,3	110,6	89,8	16,4	104,9	107,4	31,4	109,5	85,5	16,6
4	98,8	100,3	33,1	104,2	87,5	18,5	97,0	102,1	33,4	102,5	81,3	18,8
5	91,2	94,5	34,8	97,6	84,7	20,8	88,8	97,0	35,6	95,4	76,8	21,3
6	83,3	88,8	36,4	90,8	81,4	23,4	80,4	92,1	37,9	88,1	72,0	24,1
7	75,4	83,3	38,0	84,0	78,0	26,2	71,8	87,4	40,3	80,8	67,2	27,3
8	67,3	78,0	39,3	77,2	74,2	29,4	63,1	83,1	42,8	73,6	62,2	31,0
9	59,2	72,9	40,6	70,5	70,4	32,8	54,6	79,1	45,4	66,6	57,3	35,0
10	51,4	68,0	41,7	64,1	66,6	36,7	46,3	75,5	48,2	60,1	52,7	39,6
11	43,7	63,4	42,7	57,9	62,5	41,0	38,3	72,4	51,0	53,9	48,0	44,7
12	36,4	59,1	43,7	52,1	58,7	45,7	30,8	69,7	53,9	48,4	43,8	50,4
13	29,5	55,1	44,5	46,9	54,9	51,0	24,0	67,6	56,8	43,6	39,9	56,6
14	23,0	51,4	45,4	42,2	51,3	57,0	17,9	66,0	59,7	39,7	36,4	63,2
15	17,0	48,1	46,3	38,3	47,9	63,6	12,6	64,9	62,5	36,6	33,3	69,8
16	11,7	45,3	47,4	35,4	44,9	70,8	8,2	64,2	65,2	34,5	30,9	76,2
17	6,9	43,1	48,8	33,5	42,2	78,0	4,7	63,9	67,7	33,1	29,0	81,9
18	3,2	41,7	50,6	33,0	40,5	84,4	2,1	63,8	69,9	32,9	28,0	86,3
19	0,0	41,0	52,5	32,8	38,7	89,9	0,0	64,1	72,0	32,8	27,1	89,9

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Number of Blades

- To minimize losses but at the same time a well guided flow, the analytical equation for the number of blades imposes:

$$Z_{min} = 2\pi \, cotg\,(\alpha_1)$$

- Glassman suggests to calculate the minimum number of blades using the following empirical relationship :

$$Z_{min} = \frac{\pi}{30}(\alpha_1 + 20)\, cotg\,(\alpha_1) \qquad = 19$$

- Actually, the number of blades should be corrected through the slip factor but in our case its influence can be neglected and then the number of blades does not

$$\sigma = 1 - \theta \frac{\pi \sin \beta_2}{Z} \quad \theta\text{: (blockage factor = 0.5-0.6)}$$

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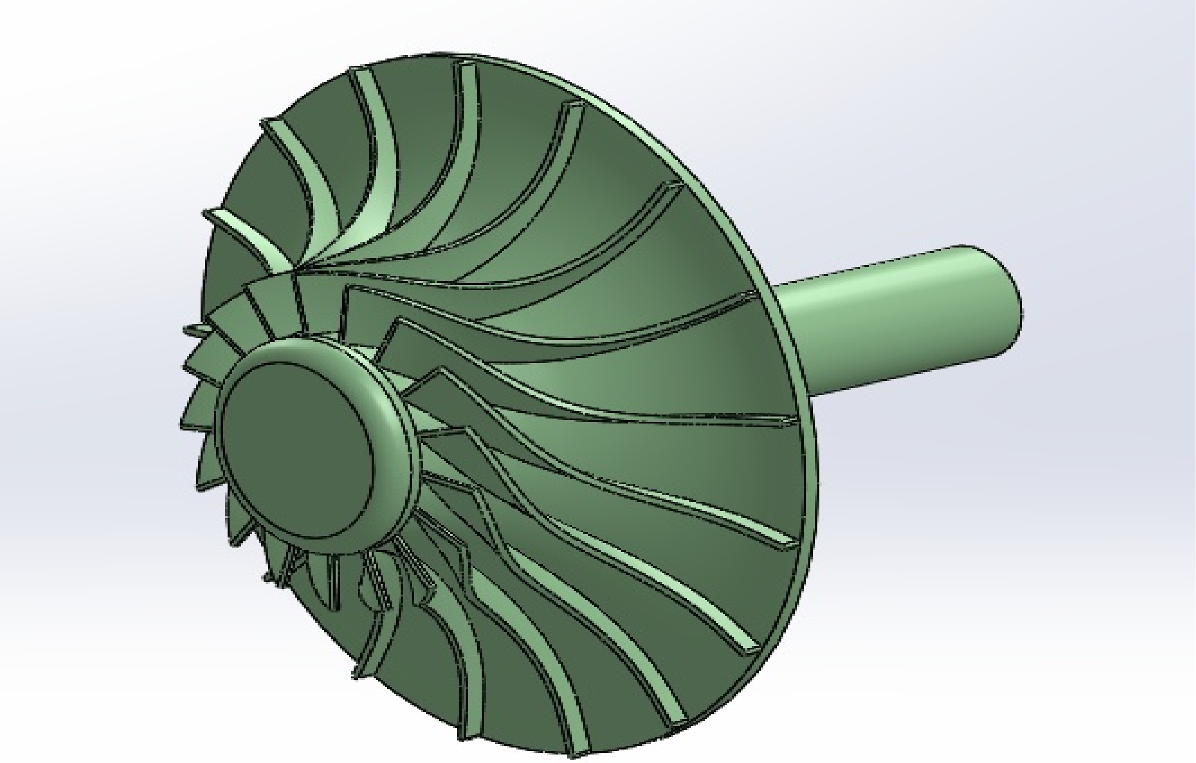
Impeller 3D sketch

Sketch of the obtained turbine blade



Structural analysis and simulation: choice of material - 1

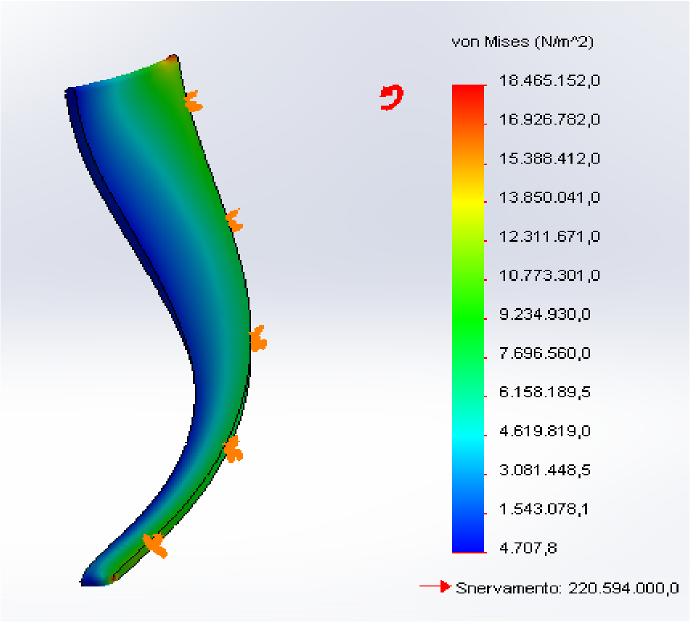
Material	Elastic modulus [Mpa]	Yield [MPa]	Displacement [mm]	Strain	Von Mises [MPa]
Molten carbon steel	200000	248	$3.363\ 10^{-4}$	$7.536\ 10^{-5}$	18.1
Carbon steel	210000	221	$3.195\ 10^{-4}$	$6.822\ 10^{-6}$	18.4
Chromium stainless steel	200000	172	$3.354\ 10^{-4}$	$7.163\ 10^{-5}$	18.5
Steel alloy	210000	620	$3.154\ 10^{-4}$	$6.735\ 10^{-5}$	18.2

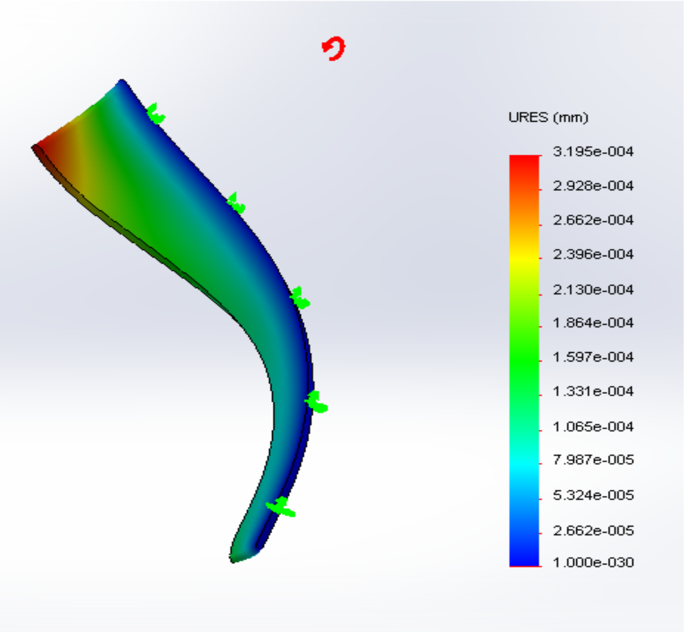
Structural analysis and simulation: choice of material - 2

The material chosen after the simulations is carbon steel The same material will be used for shaft, stator and volute. On the software, centrifugal force has been set as main stress, setting the value of the rotational speed and the axis of rotation. In this first phase the contribution of the pressure drop has been neglected, and for “computational constrains” only single blade has been simulated. The analysis supply structural resistance to mechanical stress of the blade and the main characteristic of the blade behavior by:

- Von Mises stress
- Displacement
- Strain

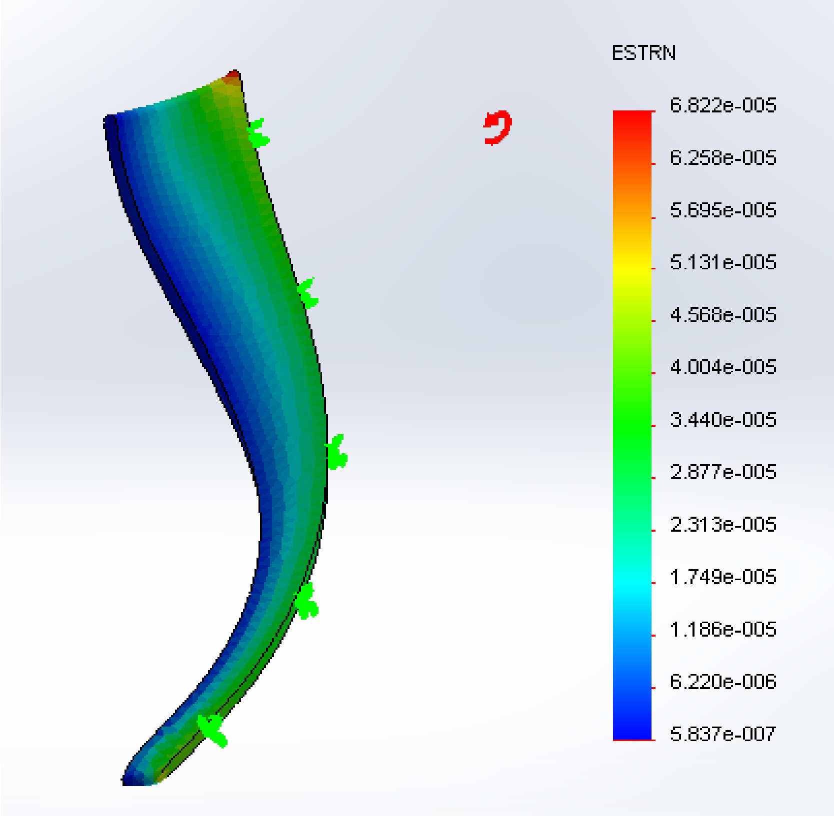
Structural analysis and simulation: choice of material - 3





(Von Mises & Displacement)

Structural analysis and simulation: choice of material – 4 (strain)



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Conclusions - 1

- This paper presents the thermodynamic feasibility of an innovative Organic Rankine Cycle and a preliminary design of its main component, the expander.
- The novelty lies in the choice of this component. The intention is to see if you can make this device, "breaking" the usual practice, that today uses volumetric expanders
- The main challenge of the analysis are the imposed size and weight limitations that require a particular design.

Conclusions - 2

- Possible system layout has been analyzed and the requirements for a prototypal application investigated.

- The simulation has been important both because it has provided feedback on the values of the expander and supply information on the nominal power of the system and about the behavior of the R245fa, as working fluid.

Future developments

Further development of the study will include :

- to complete the design procedure, including stator and volute;
- to supply a complete layout of the device;
- to perform other thermodynamics simulation with other organic fluids such as R134 to investigate by a thermodynamic study the super-heater and/or a pre-heater components and to study the interaction between components and the organic fluid;
- to perform a market and cost analysis.

All this data can be a valid support for the final step of the research, that is the realization of the prototypal ORC system.

Small-Scale ORC Energy Recovery System for Wasted Heat: Thermodynamic Feasibility Analysis and Preliminary Expander Design

Thank you for your attention!

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