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A numerical model for the prediction of the fluid dynamic and mechanical losses of a Wankel-type expansion device

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HIGHLIGHTS

- A model is proposed to predict the real performances of a Wankel expander.
- The model is based on a lumped parameter approach developed in Matlab®.
- Different losses were considered in the modelling.
- A close matching was found between numerical and experimental results.
- A maximum value of 0.87 at 1250 rpm was found for the isoentropic efficiency.

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ABSTRACT

This paper presents a numerical model of the fluid dynamic processes and mechanical losses of a Wankel-type expansion device for microgeneration purposes. The prototype was built at the University of Pisa using the rotating parts of a gasoline-fueled Wankel engine. Considering the geometrical features of the prototype, the model simulates the fluid-dynamic losses across the valves, the thermal exchange with the outer ambient and the mechanical losses of the whole mechanism. The numerical results were then validated by the comparison with the pressure cycle obtained at the dynamic test bench using compressed air.

1. Introduction

The environmental advantages and the possibility of substituting fossil fuels push to increase the renewable sources exploiting in energy applications. In this field, the micro generation represents an effective solution for some energy sources, such as biomasses, waste flue gases and thermal solar, since it allows to reduce the distance between the productive site and the users.

For most of the above-mentioned cases, Organic Rankine Cycle (ORC) technology is a suitable method to produce electrical power with a high efficiency due to the relatively low temperature at which heat is available. ORCs have been thoroughly analyzed during the past years, both from the point of view of the working fluid selection, the operating parameters and the technology of the expander. Although a generally applicable technology has not found yet, most of the authors agreed on the convenience of employing volumetric expanders when the size of the plant is below 100 kW [1–7]. Scroll expanders cover low values of power (1–10 kW), while screw expanders are more suited between 20 and 100 kW [8]. Moreover, volumetric expanders do not show problems due to the moisture at the end of expansion and they may be employed with almost all the operating fluids.

Many published scientific papers deal with the numerical modelling of a volumetric expander for efficiency and power prediction sake. Guangbin et al. in [9] developed a numerical model of a scroll expander considering suction and discharge losses, thermal exchanges and fluid leakages while in [10] a scroll expander was experimentally investigated with air and ammonia under different operating conditions. Moreover, Quoilin et al. in [4] proposed a semi-empirical model of an ORC plant employing a scroll expander and Ma et al. in [11] presented a dynamic modelling of a scroll expander. The research was carried out considering also single screw expander [12–16] and twin screw expander [17–19] in which experimental and numerical aspects were focused. Other papers [20–23] investigated the performances of rotating vane expander and a comparison between experimental and theoretical models was shown in [24]. Other authors, such as Baek et al. in [25], modelled a piston expander considering the lubrication theory and Fukuta et al. in [26] developed a reciprocating expander with four cylinder arranged radially; however, a list of the experimental studies about the previous expander technologies can be found in [27].

The employment of the Wankel mechanism for the realization of an

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Nomenclature

Dh	hydraulic diameter [m]
e	eccentricity [m]
f	friction
F	force N [N]
h	enthalpy [J/kg]
H	enthalpy of the chamber [J]
hint	internal heat exchange coefficient $\left[\frac{W}{m^2K}\right]$
hext	external heat exchange coefficient $\left[\frac{W}{m^2K}\right]$
n	rotating speed [n]
y	general function
m	mass [kg]
$\dot{m}$	mass flow rate $\left[\frac{kg}{s}\right]$
k fluid	thermal conductivity of the fluid $\left[\frac{W}{m\,K}\right]$
ks	thermal conductivity of the stator $\left[\frac{W}{m\,K}\right]$
kl	thermal conductivity of the insulating $\left[\frac{W}{m\,K}\right]$
Nu	Nusselt number
Pr	Prandtl number
p	operating pressure [Pa]
P	general component of the force
$\dot{Q}$	total flow rate $\left[\frac{W}{m^2}\right]$
r average	average radius of the chamber [m]
R	length side of the rotor [m]
R bearing	radius of the rotor bearing [m]
Re	Reynolds number
t	time [s]
U	total heat exchange coefficient $\left[\frac{W}{m^2K}\right]$
V	volume chamber [m^3]
v	specific volume $\left[\frac{m^3}{kg}\right]$

Greek symbols

α	angular coordinate of the rotor [°]
Δ	variation
ϒ	angular coordinate of the shaft [°]

Subscripts

a	chamber a
Atm	atmospheric
b	chamber b
bearing	bearing
c	chamber c
eff	effective
ext	external
in	inlet
inertia	inertia
out	outlet
iso	isoentropic
i	insulating
s	stator
l	lateral surface
p	covering plates surface
x	x direction
y	y direction
z	z direction

expander was proposed by Badr et al. in [28,29].

The authors of the present paper also published some papers [1,30–32] in which the Wankel expander was analyzed; the results indicated the possibility of realizing machines in the power range 10–50 kW depending on the employed working fluid and rotating speed. Although the internal combustion engines based on the Wankel scheme demonstrated their limits due to the high fuel consumption and pollutant emissions, the conversion into an expander can eliminates or greatly reduce these disadvantages [1,30–32], while retaining its advantages in terms of compactness, economy and low vibration magnitude level. Respect to many piston expanders, in which intake and exhaust ports are located in the cylinder head and therefore they are quite close each other, in the Wankel expander the exhaust ports are located relatively far away from the intake ones, thereby reducing the heat losses.

The work here presented is the prosecution of the previous published studies in which the authors studied the correct matching with the working fluid [1] and the influence of the valves timing and geometry. In this work a numerical model of this expander was developed in order to combine the prediction of the indicated cycle with the calculation of mechanical, thermal and fluid-dynamic losses.

2. Materials and method

2.1. The Wankel expander prototype

The prototype used as a reference for this work (see Fig. 1) was already described in previous published papers [1,30–32] and its main features are summarized in the following lines. This prototype was built using the shaft and bearings of an internal combustion Wankel engine (AIXRO 50). Conversely, the stator case was newly built to accommodate the intake and exhaust valves. The original rotor was substituted with a specifically built one in which the typical cavities were eliminated to increase the geometrical compression ratio with the aim to improve the isentropic efficiency of the device.

The original apex seal elements were replaced with others built in PTFE and glass fiber, because the working conditions of the expander are less severe than those of combustion engine (see Fig. 1).

2.2. Model geometry

In this paragraph the fundamentals of the geometry definition are recalled for the sake of clarity. The coordinate α is the angular

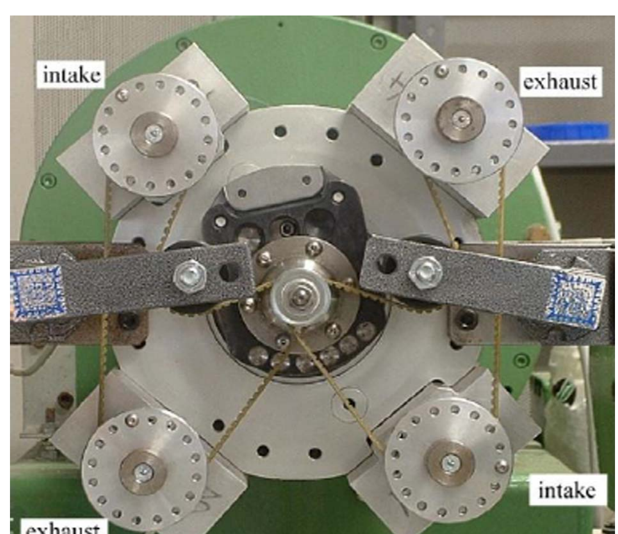

Fig. 1. Wankel expander prototype.