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Theoretical modeling and experimental investigations for the improvement of the mechanical efficiency in sliding vane rotary compressors

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HIGHLIGHTS

- Breakdown of energy flows in sliding vane rotary compressors.
- Comprehensive model to simulate the performances of industrial machines.
- Experimental campaign at different outlet pressure levels and revolution speeds.
- Identification of the friction coefficient.
- Optimization of several design parameters to achieve mechanical efficiency improvements.

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ABSTRACT

Positive displacement compressors lead the market of compressed air production for industrial applications. Among them, sliding vane rotary compressors represent an energetically virtuous alternative to the current compression technologies. In the present work, the effects of compressor design parameters were investigated through a comprehensive approach that aimed at addressing more efficient machines to promote sliding vane compressors as the key enabling technology in compressed air systems. A comprehensive mathematical model was developed to study the main phenomena occurring in this kind of compressors. The model provides the cell volume evolution over a whole rotation during which filling, compression and discharge processes occur. The first and latter phases are described by the quasi-propagatory approach that represents the inertial, capacitive and resistive features of one-dimensional unsteady flows. The dynamics of the compressor blades led to four different arrangements inside the rotor slots while an analysis of the hydrodynamic lubrication established between blade tip and stator wall focused on the oil film thickness evolution to prevent dry contacts. An extensive experimental campaign on a mid-size industrial compressor allowed the model validation at different outlet pressure levels and revolution speeds using a direct measurement of mechanical power and the reconstruction of the indicator diagram from piezoelectric pressure transducers. The friction coefficient at the contact points between blades with stator and rotor was estimated in 0.065 and further improvements of the mechanical efficiency were eventually addressed considering the roles of compressor aspect ratio, revolution speed, and blade tilt. The first two theoretical optimizations might lead to an increase of the compressor efficiency of 2 and 9 percentage points respectively. On the other hand, acting on the blade tilt would not produce relevant improvements.

1. Introduction

Energy saving is today recognized as the primary action to accomplish energetic and environmental commitments of all the Countries in the World. CO_2 concentration in atmosphere is a crucial concern, universally recognized as the tomorrow's main challenge. Therefore, a new energy paradigm based on a sharp reduction of the energy consumptions and a renewable-based energy production is sought by many scientific and institutional contexts.

Compressed air accounts for a mean 10% of the global industrial electricity consumptions (7850 TW h in 2012) which reaches a

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Nomenclature

List of symbols

α	angular coordinate [rad]
γ	specific heat ratio [–]
ξ	contraction ratio [–]
η	efficiency [–]
λ	friction coefficient [–]
μ	oil dynamic viscosity [Pa s]
ζ	force projection angle [rad]
ρ	density [kg/m^3]
τ	QPM parameter [s]
ϕ	blade tilt angle [rad]
χ	force projection angle [rad]
ω	revolution speed [RPM]
Λ	QPM parameter [–]
Ω	QPM parameter [s^{-1}]
a	speed of sound [m/s]
h	film thickness [m]
k	force projection coefficient [–]
m	mass [kg]
p	pressure [Pa]
q	heat flux [J]
t	thickness [m]
u	flow velocity [m/s]
v	slip velocity [m/s]
w	oil flow velocity [m/s]
A	upstream boundary condition slope [N s/m^3]
B	downstream boundary condition slope [N s/m^3]
C	characteristic curve slope [N s/m^3]
D	diameter [m]
E	absolute internal energy [J]
F	force [N]
H	specific enthalpy [J/kg]
K_m	mixture thermal conductivity [W/m/K]
L	length [m]
N	number of cells [–]
Nu^*	corrected Nusselt number [–]
P	power [W]
Q	flow rate [m^3/s]
R	gas constant [J/kg/K]
R_Y	tip radius [m]
S	duct cross section area [m^2]
T	temperature [K]
U	blade tip speed [m/s]
V	cell volume [m^3]

Subscripts and superscripts

$'$	initial state
0	thickness at maximum pressure
∞	steady state
bl	blade
c	centrifugal
cor	Coriolis
d	downstream
drop	oil droplet
evap	evaporation
ext	external
fr	friction
in	inertia
ind	indicated
inj	injection
inl	inlet
mech	mechanical
out	outside the rotor slot
outl	outlet
pn	normal pressure
st	stator
suc	suction conditions
u	upstream
vol	volumetric

Mathematical operators

$\dot{\Upsilon}$	first time derivative of Υ
$\ddot{\Upsilon}$	second time derivative of Υ

share close to 20% if commercial and residential needs are included [1–3]. Facing this issue implies the employment of many saving measures with actions upstream and downstream of the compressed air production: pipeline leakages reduction, adjustable speed drives, optimization of the end use devices, etc. In addition, with reference to electricity consumptions in compressed air systems, the saving potential related to compressor technology has been estimated 25–30% [1]. From an economic point of view, although energy costs for compressed air are predominant with respect to the capital ones (70–75%) [2], an investment increase of 10% in a 10 year operating period would be feasible only if the compressor efficiency increase was 4% greater than the former technology [3].

In industrial applications, rotary volumetric compressors are able to match flow rate and pressure level requirements with an electrical power range from a few to several hundred kilowatts. Among them, Sliding Vane Rotary Compressors (SVRC) revealed a better energetic behavior whether on/off load conditions were taken into account [3]. This operating regime is a common process to match the line pressure needs in terms of flow rate: when the air demand decreases, thanks to the automatic depressurization of the machine and some intrinsic features related to the sealing among vanes (e.g. absence of the so called "blow hole line"), SVRCs accomplish in an efficient way a process that is usually energy wasteful.

In the literature, sliding vane compressors have been investigated both theoretically and experimentally. The machine geometry and vane kinematics were modeled following a trigonometric approach [4,5]. Circular and elliptical stator configurations to achieve higher volumetric ratios or even dual stage compressions were considered [6,7] as well as slanted blade arrangements inside the rotor slots [8,9]. Preliminary studies assumed suction and discharge processes as isobaric while the compression was modeled using semi-empirical formulations [6]. However, zero dimensional models for the cell thermodynamics were also developed based on the energy conservation to predict the pressure evolution over the whole cell rotation [5,9–11]. To detail the friction power, the contribution due to the blade dynamics was widely investigated [12,13]. Comprehensive analyses that involve secondary contributions such as friction at the bearings and on the side covers of the machine [14,15] or the blade tip profile [16,17] were also developed. Leakage paths models were presented assuming clearances as orifices [18]. As concerns the experimental activities, tests at different steady conditions [19–21] and on unconventional configurations like the blade tilting [22] were carried out. Experimental methodologies have been set up to measure the pressure inside the compressor cells [23–25] while theoretical approaches as the Helmotz's resonator one supported the discussion of the results [26,27].