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Design of centrifugal compressors for heat pump systems

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HIGHLIGHTS

- A mean-line model for design and analysis of centrifugal compressors is presented.
- The model is validated against five test cases including three different fluids.
- The model is coupled to that of a high-temperature heat pump and optimized.
- The optimization method accounts for trade-offs in cycle and compressor designs.
- A two-stage steam compressor yields the best cycle performance.

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ABSTRACT

This work presents a mean-line model of a centrifugal compressor and a method for a coupled optimization with a heat pump system. The compressor model was validated with five test cases from the open literature including the working fluids: air, refrigerant R-134a and carbon dioxide. Afterwards, the compressor model was coupled and optimized with that of a heat pump cycle supplying steam at 150 °C. Two cycle configurations were considered: an open-loop system using steam, and a closed-loop system with five other working fluid candidates. The compressor was designed using a multi-objective optimization algorithm, which seeks to maximize simultaneously the cycle coefficient of performance and the supplied heat flow rate. The method employed in this work considers the possible trade-offs regarding cycle and compressor design criteria, and can be used to identify cost-effective solutions for the next generation of heat pumps. The obtained results suggest that a two-stage compressor using steam yields the highest values of coefficient of performance and heat supply, and at the same time requires a more challenging compressor design.

1. Introduction

Heat pump systems are regarded as a viable and attractive solution to recover excess heat from low-to-medium temperature heat sources of many industrial processes [1–3].

Chua et al. [4] and Arpagaus et al. [2] reviewed the most important advances in the field. The application of heat pump systems is currently extending to the generation of high-temperature heat (with heat sink temperatures above approximately 100 °C). The International Energy Agency (IEA) [5] produced a report on industrial heat pumps, indicating the markets, level of technological maturity, applications and barriers of heat pumps. The report highlights that the theoretical potential for application of industrial heat pumps increases significantly for heat sink temperatures up to and higher than 100 °C. Nellisen and Wolf [6] investigated the heat demand across different industries in the European market and indicated that about 626 PJ (174 TWh) of heat is reachable by industrial heat pumps, of which about 113 PJ (19%) is in the temperature range 100–150 °C. Recently, Arpagaus et al. [3] presented a comprehensive review on high-temperature heat pump (HTHP) systems, concluding that industrial HTHPs can potentially supply the industrial European demand of 113 PJ.However, the literature highlights that the development of mass-produced units is hindered by many factors, such as low awareness of the heat consumption in companies and the possible technical solutions [5,3]; lack of knowledge of comprehensive heat pump integration methods [3,5,7]; lack of available refrigerants with low global warming potential in the high-temperature range [3,8]; lack of cost-efficient solutions and long payback periods [5,8,9]; and the technical challenges related to the compressor operation at high temperatures [10].

In this respect, the centrifugal compressor (CC) technology is seen as an attractive option compared with positive displacement compressors, since it bears key advantages such as the potential for high efficiency,

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Nomenclature

Symbols

A	cross-sectional area, m^2
AR	vaned diffuser area ratio, –
AS	vaned diffuser aspect ratio, –
b	blade width, m
C	absolute velocity, m/s
C_f	diffuser friction factor, –
COP	coefficient of performance, –
CP	diffuser pressure recovery coefficient, –
d_{hb}	hydraulic diameter, m
D_f	blade diffusion factor, –
DR	degree of reaction, –
$\dot{m}$	mass flow rate, –
h	specific enthalpy, J/kg
k_s	blade surface roughness, m
K	total pressure loss coefficient, –
L_z	rotor axial length, m
L_b	blade hydraulic length, m
LWR	vaned diffuser length over width, –
M	Mach number, –
N	rotational speed, rpm
o	throat, m
p	static pressure, Pa
PR	pressure ratio, –
$\dot{Q}$	heat flow rate, W
r	radius, m
Re	Reynolds number, –
s	specific entropy, J/(kgK)
t	trailing edge thickness, m
T	temperature, K
U	peripheral velocity, m/s
$\dot{V}_{comp,in}$	compressor inlet volume flow rate, m^3/s
W	relative velocity m/s
Z	number of blades –

Abbreviation and acronyms

CC	Centrifugal Compressor
HP	Heat Pump
HTHP	High-Temperature Heat Pump
VD	Vaned Diffuser

Greek letters

α	absolute flow angle, °
β	relative flow angle, °
ρ	density, kg/m^3
ε	clearance gap, m
ε_w	wake fraction at impeller exit, –
η	compressor isentropic efficiency, –
σ	impeller slip factor, –
ϕ	flow coefficient, –
χ	secondary flow mass fraction, –
ψ	stage loading coefficient, –

Subscripts

0	total, impeller inlet
1	impeller throat static conditions
01	impeller throat total conditions
2	impeller exit static conditions
02	impeller exit total conditions
3	diffuser inlet static conditions
03	diffuser inlet total conditions
4	diffuser throat static conditions
04	diffuser throat total conditions
5	diffuser exit static conditions
05	diffuser exit total conditions
a	axial direction
b,bl	blade, back face
ch	choking
cl	clearance
h	hub
vs	interspace
vd	vaned diffuser
tt	total-to-total
ts	total-to-static
th	throat
t	trailing edge, tangential direction
sf	skin friction
s	shroud
rms	root-mean-square
rel	relative
rc	recirculation
r	radial direction, rotor
pp	pinch point
opt	optimal
mix	mixing
m	meridional direction
lim	limit value
is	isentropic
ind	inducer
Inc.	incidence
imp	impeller
id	ideal

the possibility to operate at high pressure ratios, a compact design, and oil-free operation [11,12]. The progress of the technology in the last decades, and, foremost, the introduction of high-speed generators, have made it possible to apply CCs even to smaller units in the refrigeration and heat pump fields. Hastbacka et al. [13] claimed that small centrifugal compressors are expected to replace reciprocating and screw compressors for chilled-water systems in the range 88–281 kW as their high initial cost is offset by improved energy efficiency. Süß [14] developed a centrifugal compressor for refrigeration applications with a nominal power of 7 kW, operating up to 90 krpm and achieving an overall efficiency of 70% (including the motor). Bertsch et al. [15] conducted a study on the use of micro turbo-compressors in HTHPs, concluding that such compressors are a viable alternative to traditional technology. The working fluid can range from natural to synthetic refrigerants, and the compressor can experience considerably different flow phenomena according to the selected medium. The combination of high rotational speed, the variety of possible working fluids, and the compact size often result in highly loaded compressors whose design requires the adoption of suitable numerical tools and modeling strategies.

The first step in the design of a centrifugal compressor is performed using one-dimensional preliminary design models. One-dimensional numerical methods, also called *mean-line models*, provide the design and performance estimation of the centrifugal compressor through the solution of the governing equations at the mean streamline and at the main stations of the compressor components. Compared to more

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