Energy 35 (2010) 436–450

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Design, experimental investigation and multi-objective optimization of a small-scale radial compressor for heat pump applications

J. Schiffmann [a,*], D. Favrat [b]

[a] Fischer Engineering Solutions AG, Birkenweg 3, CH-3360 Herzogenbuchsee, Switzerland
[b] Ecole Polytechnique Fédérale de Lausanne, EPFL STI IGM LENI, Station 9, CH-1015 Lausanne, Switzerland

ARTICLE INFO

Article history:
Received 24 May 2009
Received in revised form
24 September 2009
Accepted 8 October 2009
Available online 4 November 2009

Keywords:
Domestic heat pump
Oilfree compressor
Centrifugal compressor
Design optimization

ABSTRACT

The main driver for small scale turbomachinery in domestic heat pumps is the potential for reaching higher efficiencies than volumetric compressors currently used and the potential for making the compressor oil-free, bearing a considerable advantage in the design of advanced multi-stage heat pump cycles. An appropriate turbocompressor for driving domestic heat pumps with a high temperature lift requires the ability to operate on a wide range of pressure ratios and mass flows, confronting the designer with the necessity of a compromise between range and efficiency. The present publication shows a possible way to deal with that difficulty, by coupling an appropriate modeling tool to a multi-objective optimizer. The optimizer manages to fit the compressor design into the possible specifications field while keeping the high efficiency on a wide operational range. The 1D-tool used for the compressor stage modeling has been validated by experimentally testing an initial impeller design. The excellent experimental results, the agreement with the model and the linking of the model to a multi-objective optimizer will allow to design radial compressor stages managing to fit the wide operational range of domestic heat pumps while keeping the high efficiency level.

1. Introduction

According to the International Energy Agency [1] the worldwide energy fraction consumed by the residential sector accounts for 35% of the total energy consumption, out of which 75% are used for space and tap water heating. As domestic heating requires relatively low temperature levels (30 °C for floor space heating and 60 °C for tap water), renewable energy sources (environmental heat) may offer an interesting alternative to fossil fuels. Recently Favrat et al. [2]. have introduced an exergy efficiency indicator, compared different heating technologies and reported on the significant advantage of heat pumps in this respect. Other studies also show that the future combination of efficient trigeneration techniques with advanced hybrid fuel cell–gas turbine systems and heat pumps could further improve the prospects for a more rational use of energy (Burer et al. [3]).

The targeted topic in this particular study is the design optimization of a radial compressor for driving a heat pump for domestic heating applications. The main driver for small scale turbomachinery in low power heat pumps is the potential for reaching higher efficiencies than volumetric compressors currently used in these machines and the potential for making the compressor oil-free, which bears a considerable advantage in the design of advanced multi-stage heat pump cycles. In an earlier paper Schiffmann et al. [4] investigate on the feasibility of a twin-stage radial flow compressor for driving a two stage heat pump. They come to the conclusion that a compressor unit with two impellers of 20 and 18 mm tip diameter, rotating at 250 krpm is technically feasible. Recently Schiffmann and Favrat [5] have published experimental data demonstrating the technical feasibility of a small scale, oil-free and direct driven turbocompressor for domestic heat pump applications. Their proof of concept consisted of an oil-free gas bearing supported radial turbocompressor with a tip diameter of 20 mm that was tested to speeds up to 210 krpm, reaching pressure ratios in excess of 3.3 and internal isentropic compressor efficiencies of up to 79%.

Centrifugal compressors are extensively used in a wide spectrum of industrial applications, with nearly any combination of pressures, powers, applications and gases. Turbochargers for internal combustion engines alone embrace automotive as well as marine applications, with impeller diameters varying from approximately 30 mm to more than 1 m. Other typical applications are micro-turbine generators, industrial refrigeration cycles and helicopter gas turbines where pressure ratios of up to 1:10 are

* Corresponding author. Tel.: +41 (0)62 956 88 50; fax: +41 (0)62 956 22 00.
E-mail address: juerg.schiffmann@fischerprecise.ch (J. Schiffmann).

0360-5442/$ – see front matter
doi:10.1016/j.energy.2009.10.010

Nomenclature

Roman symbols

$\dot{m}$	mass flow [kg s^{-1}]
Re	Reynolds number
A	section [m^2]
b	channel width [m]
c	absolute velocity [ms^{-1}]
C_f	friction coefficient [–]
d	diameter [m]
D_f	diffusion coefficient [–]
D_h	Hydraulic diameter [m]
e_{back}	impeller backplate clearance [m]
e_{blade}	impeller blade thickness [m]
E_{el}	electric energy [J]
e_{is}	specific isentropic work [J kg^{-1}]
e_{tip}	impeller tip clearance [m]
h	specific enthalpy [J kg^{-1}]
K_f	disc friction coefficient [–]
K_L	pressure loss coefficient [–]
l	axial length [m]
L_h	hydraulic length [m]
N_{Bl}	number of main blades [–]
n_d	number of days [–]
N_{rot}	rotational speed [rpm]
N_{Sp}	number of splitter blades [–]
p	pressure [Pa]
r	impeller radius [m]
Ra	surface roughness [m]
T	temperature [$^\circ$C]
u_t	tip velocity [ms^{-1}]
w	relative velocity [ms^{-1}]

Greek symbols

α	absolute flow angle [deg]
β	relative flow angle [deg]
η	efficiency [–]
η_{is-k}	isentropic compressor efficiency [–]
η_{y-k}	seasonal isentropic compressor efficiency [–]
μ	viscosity [Pas]
Π	pressure ratio [–]
ρ	density [kg m^{-3}]

Subscripts

1	inducer inlet
2	impeller inlet
3	impeller inlet throat
4	impeller discharge
5	diffuser discharge
7	volute discharge
θ	tangential component
air	Air
bl	blade loading & diffusion
cl	clearance
df	disc friction
ext	external
h	hub
in	inlet condition
inc	incidence
ind	inducer
is	isentropic
m	Meridional component
meas	measured value
opt	optimum
out	exhaust condition
rc	recirculation
rms	rms-value
s	shroud
sf	skin friction
*St*1	stage 1
*St*2	stage 2
tt	total–total
water	water

required, ideally on a single compressor stage. In aerospace applications saving weight is the main driver, leading to a run for highest possible pressure ratios on one stage. Recently the experimental investigation of a single stage 1 MW impeller with a pressure ratio 1:11 processing air was presented by Higashimori et al. [6]. The diversity in terms of applications, pressure ratios and powers is further increased by different operational requirements. In some stationary industrial applications the aim of the compressor is to deliver a nominal mass flow and pressure ratio with the least possible energy consumption, whereas in other applications the emphasis is rather on a wide range of possible pressure ratios and mass flows, necessarily leading to a compromise on efficiency. The present publication proposes a way to facilitate the design of small scale turbocompressors requiring operation on a wide range in terms of pressure ratio and mass flows.

The focus on the design procedures, performance prediction and especially on the efficiency gain of radial compressors is still today mainly driven by large industrial compressor stages. Today's small scale turbomachinery applications, however, are essentially driven by micro-gas turbine applications. Already in 1980 Strong [7] presented a directly fired heat pump, where an Organic Rankine Cycle turbine drove the compressor of a reversed Organic Rankine heat pump cycle. The radial compressor was 26.2 mm in diameter and rotated at 160 krpm. The measured efficiency was 60% at a pressure ratio of 6.65 and a mass flow of 0.027 kg s^{-1}. In 1996 Yun and Smith [8] presented a study of a two stage compressor for automotive air-conditioners. The fluid was an R123 and the impeller diameters 48 and 38 mm delivering pressure ratios of 3.2 and 1.94. The pressure ratios were unbalanced in order to achieve the best possible efficiency, taking into account the tip clearance losses. The overall peak efficiency including the intercooler between the two stages was predicted to be around 77%. In 1998 Mehra et al. [9] presented a theoretical study on a small gas turbine motor with an electrical output power of 10–40 W contained in a volume smaller than 1 cm^3. Due to size and manufacturing issues the proposed impellers were composed of 2D vanes. Calculations based on CFD predicted a pressure ratio of 4.4 and an efficiency of 63%. Two years later Frechette et al. [10] showed first results of the turbine of the small 2D turbine reaching 1.4 Mrpm and an output of 5 W. However, neither efficiencies nor turbine maps are published. In 2004 Epstein [11] presents the state of the art of the MIT micro-gas turbine project as a whole, giving, among other, insights in the manufacturing based on semiconductor industry derived processes allowing to etch silicon and silicon carbides to submicron tolerances. In 2001 Isomura et al. [12] present the feasibility study of a slightly larger gas turbine generator with a power output of 100 W, using 3D radial impellers with a compressor pressure ratio of 3, a rotational speed of 870 krpm and a predicted efficiency of 65%. The drivers for the micro-turbine generators are mainly military applications; the power density of these machines is