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Loss development analysis of a micro-scale centrifugal compressor

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ABSTRACT

The ever-increasing demand for more efficient energy conversion has placed designers under increasing pressure to develop processing equipment that can meet contemporary needs. It has long been known that a decreasing Reynolds number has a negative effect on centrifugal compressor efficiency. The drop in efficiency can be accounted for relatively easily in the design process using various empirical correlations. However, the correlations only account for a reduction in performance; they do not offer any consideration of the extent to how the drop in efficiency can be countered in the design process. To identify potential methods by which it is possible to improve the performance of centrifugal compressors operating at low Reynolds numbers, the loss development in centrifugal compressors with a reducing Reynolds number must be studied. Recent works on loss development, in general, have focused on the overall performance deterioration, and the differentiation of the losses originating from different causes with the reducing Reynolds number has been studied only in an axial compressor. The present paper examines loss development in a centrifugal compressor with a vaneless diffuser with respect to the Reynolds number and differentiates between the losses that originate from different causes. A new hybrid method is used to calculate the boundary layer thickness inside a complex flow field. The results show that the diffuser plays a significant role in the performance deterioration of centrifugal compressors with a low Reynolds number and should be included in the loss development analysis. A study of the boundary layers, flow fields and loss development indicates that growth in the impeller hub and diffuser boundary layers should be reduced to improve the performance of the compressor.

1. Introduction

The sustainable development goals of the United Nations aims at reducing greenhouse gas emissions, improving energy efficiency and increasing the share of renewable energy sources [1]. Additionally, the European Union has similar goals [2]. Finland has committed to the EU targets and aims at increasing self-sufficiency in energy [3]. The industrial sector accounts for, on average, 50% of the overall electricity consumption [4]. A cost-effective way to achieve the international and national targets involves improving energy efficiency [5]. The improvement of compressor performance, in particular, plays an important role in improving energy efficiency and reducing the end-use electricity demand, as compressors alone account for 15% of the overall electricity consumption within industry [4].

Micro-scale centrifugal compressors (impeller outlet diameter less than 30 mm [6]) have great potential for efficiency improvement due to their clearly low performance. The performance of micro-scale centrifugal compressors is worse than that of the larger compressors due to the losses caused by low Reynolds numbers, the larger relative blade thickness, surface roughness and tip clearance [7]. The effect of Reynolds number on the compressor performance was discovered e.g. by Yang et al. [8].

The improvement in the efficiency of the micro-scale centrifugal compressors could result in e.g. the increased technological feasibility of micro-scale gas turbines [9]. Micro-scale gas turbines (less than 100–1000 kW [10]) could represent a potential solution for combined heat and power applications to cut greenhouse gas emissions [11]. These machines are both flexible and scalable [12]. Therefore, they could also increase the share of renewable energy sources and self-sufficiency in energy [9]. In addition to distributed energy generation, micro-scale gas turbines also hold potential in applications that require a compact, portable power source due to high power density; e.g., unmanned aerial vehicles [13]. A micro-scale centrifugal compressor could also replace a displacement compressor in small refrigeration systems to achieve lower power consumption and weight [14].

The effect of the Reynolds number on the compressor efficiency can be accounted for relatively easily in the design process with empirical correction equations; however, these equations do not consider whether

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Nomenclature

Latin alphabet

A	area [m^2]
a	fraction of Reynolds-number-independent losses in Eq. (2) [–]
a	speed of sound [m/s]
b	blade height [m]
b	fraction of Reynolds-number-dependent losses in Eq. (4) [–]
B_{ref}	coefficient in Eqs. (5) and (6) [–]
c	absolute velocity [m/s]
c	chord length [m]
c	coefficient in Eq. (3) [–]
c_f	friction coefficient [–]
C_{pr}	pressure recovery coefficient [–]
c_p	specific heat capacity at constant pressure [J/kgK]
D	diameter [m]
f	friction factor [–]
h	specific enthalpy [J/kg]
K_p	total pressure loss coefficient [–]
Ma_U	tip speed Mach number [–]
n	Reynolds-number-ratio exponent in Eqs. (2) and (4) [–]
n	rotational speed [rpm]
N_s	specific speed [–]
p	pressure [Pa]
q_m	mass flow rate [kg/s]
q_v	volume flow rate [m^3/s]
R	specific gas constant [J/kgK]
r	radius [m]
Re_c	chord Reynolds number [–]
T	temperature [K]
t	tip clearance [m]
U	tip speed [m/s]
U_δ	velocity at the boundary layer edge [m/s]
U_∞	free-stream velocity [m/s]
w	relative velocity [m/s]

Greek alphabet

α	flow angle [°]
δ	boundary layer thickness [m]
η	efficiency [–]
μ_0	work input coefficient [–]
ν	kinematic viscosity [m^2/s]
ω	angular velocity [rad/s]
ϕ	flow coefficient [–]
π	pressure ratio [–]
ψ	pressure coefficient [–]
ρ	density [kg/m^3]

Abbreviations

DES	design point
FB	full blade
LE	leading edge
NC	near choke
NS	near stall
PE	peak efficiency point
PS	pressure side
SB	splitter blade
SF	scaling factor
SS	suction side
TE	trailing edge

Subscripts

1	impeller inlet
2	impeller outlet
3	diffuser outlet
ave	average
crit	critical
r	radial
ref	baseline case
s	isentropic, static
t	total

the efficiency drop can be countered somehow. Thus, in order to find potential ways to improve the performance of low-Reynolds-number compressors, loss development in centrifugal compressors with reducing Reynolds number is studied in this paper.

Recent works on loss development in low-Reynolds-number compressors have, in general, focused on the overall performance deterioration in the compressor stage. In a centrifugal compressor, the results of Schleer and Abhari [15] showed a 0.5% decrease in the total-to-static pressure ratio. In addition, the results of Zheng et al. [16] showed a 6.9% decrease in the total-to-total isentropic efficiency of a centrifugal compressor. In an axial compressor, the study of Choi et al. [17] indicated approximately a 69% increase in the total pressure loss coefficient. In addition to the total pressure loss, Choi et al. [17] investigated the differentiation of losses originating from different causes with the reducing Reynolds number in the axial compressor. To the author's best knowledge, the differentiation of losses with the reducing Reynolds number has not previously been investigated in centrifugal compressors apart from the previous work by the authors, where the loss development was studied in the downscaled centrifugal compressors [18]. And later in the centrifugal compressors with varying inlet conditions [19].

The above-mentioned recent works on the differentiation of losses in low-Reynolds-number centrifugal compressors have focused on the impeller, while considerably less attention has been placed on the diffuser. This is because, according to Dietmann and Casey [20], more losses occur in the impeller than in the diffuser due to higher velocities. The hypothesis of this work is that the diffuser plays a marked role in the performance deterioration of the compressor. Thus, the first novel aspect of this study is that the role of a vaneless diffuser in the loss development is analysed. The second novel aspect of the study is that it demonstrates how the hybrid method [21] for calculating the boundary layer thickness inside the complex flow field of a centrifugal compressor enables a more sophisticated analysis of the losses associated with the blade and endwall boundary layers from the impeller inlet to the diffuser outlet than in previous works by the authors. Additionally, the question of whether the transition model should be used when modelling the low-Reynolds-number centrifugal compressors is addressed in this paper.

2. Methods

The effect of the Reynolds number on centrifugal compressor performance and losses were assessed in two centrifugal compressors: one with splitter blades and the other without. The compressor geometries and computational domains are shown in Fig. 1. Both compressors included a vaneless diffuser. The compressor with splitter blades was studied experimentally and numerically at Lappeenranta University of Technology, Finland [22]. The compressor without splitter blades is the test case Radiver, for which the measurements were carried out at the Institute of Jet Propulsion and Turbomachinery at RWTH Aachen,