INFLUENCE OF REYNOLDS NUMBER VARIATION METHOD ON CENTRIFUGAL COMPRESSOR LOSS GENERATION

J. Tiainen - A. Jaatinen-Värri - A. Grönman - J. Backman

LUT School of Energy Systems, Lappeenranta University of Technology, Lappeenranta, Finland, jonna.tiainen@lut.fi

ABSTRACT
Decreasing Reynolds number increases frictional losses in kinetic compressors, resulting in deteriorated performance. Reynolds number can be varied by changing either compressor size or compressor inlet conditions. In this study, the effect of decreasing Reynolds number on compressor performance is numerically investigated in centrifugal compressors. First, the Reynolds number is varied by scaling the compressor while keeping the inlet conditions constant and after this by keeping the dimensions constant and varying the inlet conditions. The fractions of different loss generation mechanisms at different Reynolds numbers are estimated and loss generation mechanisms are compared. The results can be utilized to improve performance of low Reynolds number compressors, e.g. in micro-scale gas turbines or unmanned aerial vehicles.

KEYWORDS
AMBIENT CONDITIONS, SCALING, LOSS COEFFICIENTS

NOMENCLATURE

Latin alphabet

b	blade height	[m]
c	chord length	[m]
C_d	dissipation/discharge coefficient	[-]
c_m	meridional velocity	[m/s]
C_pb	base pressure coefficient	[-]
c_s	blade surface length	[m]
p	blade pitch, static pressure	[m],[Pa]
Re_b2	Reynolds number, $Re_\mathrm{b2} = \frac{U_2 b_2}{\nu_1}$	[-]
Re_c	Reynolds number, $Re_\mathrm{c} = \frac{w_1 c}{\nu_1}$	[-]
t	tip clearance	[m]
t_b	trailing edge thickness	[m]
U_2	tip speed	[m/s]
w	relative velocity	[m/s]
w_th	throat width	[m]
y^+	dimensionless wall distance	[-]

Greek alphabet

α	flow angle from the axial direction	[°]
δ^*	displacement thickness	[m]
ν	kinematic viscosity	[m^2/s]
ρ	density	[kg/m^3]
Θ	momentum thickness	[m]

Abbreviations

FB	full blade
PS	pressure side
SB	splitter blade
SS	suction side

Subscripts

1	impeller inlet
2	impeller outlet

INTRODUCTION
Varying ambient conditions or scaling the compressor size results in changes in Reynolds number, which affects performance of kinetic compressors. For example, the unmanned aerial

vehicles flying at high altitudes and micro-scale gas turbines for distributed energy production suffer from additional frictional losses due to low Reynolds numbers. The critical chord Reynolds number $Re_{\mathrm{c,crit}}$ of 200 000 is achieved at the altitude of around 20 000 m above the sea level due to decreased pressure.

Previous studies on the Reynolds number effect have been focused on axial compressors (Smith et al. 2015, Choi et al. 2008) and transonic centrifugal compressors (Zheng et al. 2013). However, less attention has been paid to subsonic centrifugal compressors and previously only deterioration in performance is investigated while the importance of different loss generation mechanisms is not discussed.

Experimental results of Smith et al. (2015) indicated that, even though the Reynolds number is highly dependent on the inlet temperature, it does not affect the performance of an axial compressor. However, the studied range of 680 000 to 840 000 was above the critical Reynolds number.

Numerical results of Choi et al. (2008) indicated that a stronger hub-corner flow separation at a decreased Reynolds number from 244 000 to 24 400 increased losses of an axial compressor operating at high altitude. Choi et al. estimated the amount of losses by Denton's (1993) loss coefficients which showed that the main loss generation mechanism was the trailing edge loss while the tip leakage loss decreased with an increasing Reynolds number.

Schleer and Abhari (2005) varied the inlet pressure of a subsonic centrifugal compressor in the range of $30 - 100$ kPa. As a result, the total-to-static pressure ratio decreased 0.5% and no effect on a stall margin was distinguished. Schleer and Abhari (2005) calculated Reynolds number based on tip speed and blade height Re_{b2} which results in lower values than chord Reynolds number Re_{c} definition. Consequently, the chord Reynolds numbers investigated by Schleer and Abhari could have been several times higher than the presented blade height Reynolds numbers. Therefore, it is reasonable to assume that the blade height Reynolds numbers of 160 000 and 530 000 studied by Schleer and Abhari correspond to chord Reynolds number above the critical chord Reynolds number (200 000). Above the critical chord Reynolds number the change in performance due to varying Reynolds number is insignificant (Casey & Robinson 2011).

Zheng et al. (2013) studied the effect of Reynolds number on the performance of a transonic centrifugal compressor. In their study, they reduced the Reynolds number from 986 000 to 296 000 and were able to measure a 7.9% decrease in total-to-total pressure ratio and 6.9% decrease in total-to-total isentropic efficiency. Zheng et al. defined the Reynolds number by impeller inlet tip speed and impeller inlet tip diameter. As the reported performance decrease is considerably higher than what Schleer & Abhari (2005) reported, it is plausible that the chord Reynolds numbers of Zheng et al. were closer to the critical chord Reynolds number. The experimental results of Zheng et al. indicated that efficiency and pressure ratio decreased almost linearly with a decreasing Reynolds number due to a thicker boundary layer especially in the inducer and more severe boundary layer separation. However, their numerical results overestimated performance and predicted a different trend.

Analytical and numerical investigations of the Reynolds number effect on compressor performance are combined in this paper. The effect of a decreasing Reynolds number is studied by varying the Reynolds number using two methods. Firstly, the Reynolds number is varied by scaling the compressor and secondly, by keeping the dimensions constant and varying the inlet conditions.

The previous results of the authors (Tiainen et al. 2016) indicated that the increased boundary layer and tip clearance losses were the most significant loss generation mechanisms in the compressor with splitter blades. As the previous paper (Tiainen et al. 2016) investigated the Reynolds number effect by decreasing the compressor size, the authors speculated that the losses associated with tip clearance might not take as significant fraction of the total losses in the compressor where low Reynolds numbers are achieved by varying the ambient conditions. This speculation is investigated in this paper and the hypothesis is that the tip clearance losses are less prominent if Reynolds number is decreased by varying the ambient conditions instead of decreasing the compressor size.

This paper contributes to the field by presenting the possible differences in the loss generation mechanisms between different Reynolds number variation methods. The paper presents the results of the most significant loss generation mechanisms in a centrifugal compressor when Reynolds number is decreased by varying ambient conditions. These results are compared with the previous results about Reynolds number losses in downscaled compressors (Tiainen et al. 2016).

NUMERICAL METHODS

Two centrifugal compressors presented in the previous paper by the authors (Tiainen et al. 2016) are studied numerically. The main difference between the compressors is that one has splitter blades and the other one only full blades. For the reader's convenience, details of compressor geometries and important dimensionless numbers at the design/peak efficiency point at high Reynolds number are shown in Table 1. The compressor with splitter blades is modeled at the design operating point at $n/n_{\mathrm{DES}} = 1.0$ (Jaatinen-Värri et al. 2013). The compressor without splitter blades is modeled at its peak efficiency point at $n/n_{\mathrm{DES}} = 0.8$ (Ziegler et al. 2003). The Reynolds number effect is analytically estimated in the compressors with an efficiency correction equation based on experimental data (Dietmann & Casey 2013).

The commercial software ANSYS CFX 17.0 is used for numerical calculations. Total pressure and total temperature are specified at the inlet boundary and mass flow rate at the outlet boundary. Turbulence is modeled using SST $k - \omega$ model. Numerical model was validated against experimental data in the previous paper by the authors (Tiainen et al. 2016).

When decreasing Reynolds number by varying ambient conditions, tip speed Mach number and flow coefficient are kept constant. To model the altitude effect, compressor inlet pressure and temperature are decreased based on the properties of the standard atmosphere. To remain the compressor operating point same at low Reynolds number (25 km above sea level) as at high Reynolds number (sea level), rotational speed and volume flow rate are decreased with respect to the change in the ambient temperature.

ANALYTICAL METHODS

To compare loss generation mechanisms in compressors where the Reynolds number is varied by changing either ambient conditions or compressor size, a quantitative prediction needs to be established. The sources of entropy in the impeller blade passages are estimated by two methods.

Method of loss coefficients

Several researchers, including Denton (1993) and Prust (1973), have presented loss coefficients to estimate the amount of losses in the blade passages. Due to the complexity of the centrifugal compressor flow field and the difficulty in defining the free stream velocity, the kinetic energy loss coefficients (Prust 1973) cannot be calculated inside centrifugal compressor

Table 1: Technical data for design point of compressor with splitter blades and peak efficiency point of compressor without splitter blades at high Reynolds number.

	With splitter blades	Without splitter blades
Number of blades	7 + 7	15
Relative blade height (b_2/D_2)	0.058	0.041
Relative tip clearance (t/b_2)	0.052	0.045
Flow coefficient ($\phi = \frac{q_\mathrm{v}}{U_2 D_2^2}$)	0.065	0.051
Pressure coefficient ($\psi = \frac{\Delta h_\mathrm{s}}{U_2^2}$)	0.520	0.450
Specific speed ($N_\mathrm{s} = \frac{\omega\sqrt{q_\mathrm{v}}}{\Delta h_\mathrm{s}^{0.75}}$)	0.830	0.830
Tip speed Mach number ($Ma_\mathrm{U} = \frac{U_2}{a_1}$)	0.920	1.170

blade passages.

Consequently, first method utilizes Denton's (1993) loss coefficients for which the free stream velocity does not need to be defined. The loss generation mechanisms are divided into boundary layer, trailing edge, and tip leakage losses. The boundary layer loss coefficient is defined as

$$\zeta_{\mathrm{B.L.}} = 2\sum \frac{c_\mathrm{s}}{p\cos\alpha_1}\int_0^1 C_\mathrm{d}\left(\frac{w}{w_1}\right)^3 d\left(\frac{x}{c_\mathrm{s}}\right). \tag{1}$$

The equation (1) integrates the entropy increase over the blade surface and sums the loss generation on both blade surfaces. The dissipation coefficient C_d is 0.002 as suggested by Denton (1993).

The trailing edge loss coefficient is calculated by

$$\zeta_{\mathrm{T.E.}} = -\frac{C_\mathrm{pb} t_\mathrm{b}}{w_\mathrm{th}} + \frac{2\Theta}{w_\mathrm{th}} + \left(\frac{\delta^* + t_\mathrm{b}}{w_\mathrm{th}}\right)^2. \tag{2}$$

The base pressure coefficient C_pb is defined as

$$C_\mathrm{pb} = \frac{p_\mathrm{b} - p_\mathrm{ref}}{0.5\rho_\mathrm{ref} w_\mathrm{ref}^2}, \tag{3}$$

where the term p_b refers to the average pressure acting on the base of the trailing edge and the subscript ref refers to the value on the blade suction surface immediately before the trailing edge.

The tip leakage loss coefficient is defined as

$$\zeta_{\mathrm{T.L.}} = \frac{2C_\mathrm{d} tc}{bp\cos\alpha_1}\int_0^1 \left(\frac{w_\mathrm{SS}}{w_1}\right)^3 \left(1 - \frac{w_\mathrm{PS}}{w_\mathrm{SS}}\right)\sqrt{\left[1 - \left(\frac{w_\mathrm{PS}}{w_\mathrm{SS}}\right)^2\right]} d\left(\frac{x}{c}\right), \tag{4}$$

where a typical value of 0.8 (Denton 1993) for a discharge coefficient C_d is used.

In the loss coefficient equations, the relative velocity is defined at the edge of the boundary layer. Exact calculation of boundary layer thickness is difficult in radial turbomachines due to complex flow: the flow field is three dimensional, blades are curved, and free stream velocity is difficult to specify. Thus, the boundary layer thickness is estimated by studying when the change in values between two adjacent observation points is less than 1% corresponding to the definition of $U_\delta = 0.99U_\infty$ for the boundary layer. Similar method for estimating the boundary layer thickness is used also by Choi et al. (2008), who stated that the free stream velocity cannot be defined in the blade passage. To estimate the boundary layer momentum and displacement thicknesses, a velocity profile for turbulent, incompressible flow past a flat plate is assumed.

Method of specific entropy

In a second method, the impeller outlet is divided into several regions corresponding to different causes of loss. This method was also used in the previous paper by the authors (Tiainen et al. 2016), but it is represented here for the readers' convenience. The subregions at the impeller outlet are referred to as tip clearance, passage wake on the full blade suction side, passage wake on the splitter blade suction side, and boundary layers.

Tip clearance region is defined as a rectangular plane in the tip clearance. The wakes on the full and splitter blade suction sides are defined as low meridional velocity regions (Eckardt 1976) for which the definition $c_\mathrm{m}/U_2 \leq 0.2$ holds. As discussed above, the definition of the boundary layer thickness is difficult inside centrifugal compressor blade passages. Therefore, to estimate the boundary layer losses at the blade and hub surfaces at the impeller outlet, the areas of tip clearance and passage wakes are subtracted from the total impeller outlet area. Finally, the increase in specific entropy is calculated between the compressor inlet and these above-mentioned subregions.

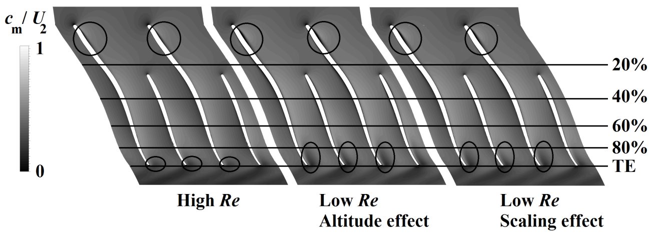
RESULTS

Before the quantitative analysis of the loss generation mechanisms, the flow fields inside the blade passages are qualitatively studied from the meridional velocity contour plots. After that, the loss generation mechanisms in centrifugal compressors with and without splitter blades are analyzed using two above-mentioned methods.

Flow field

Contours of normalized meridional velocity inside impeller at mid-span are studied to find out if the Reynolds number variation method has any influence on the flow fields. In the compressor with splitter blades (Fig. 1), the boundary layer separation near the leading edge on the full blade suction side (highlighted with circles) strengthens when the chord Reynolds number is decreased from high ($17 \cdot 10^5$) to low (altitude effect: $0.6 \cdot 10^5$, scaling effect: $0.9 \cdot 10^5$). Circles at the trailing edge highlight wakes on the blade suction sides. At low Reynolds number, wake stretches from the trailing edge 20% of the chord length upstream. There is not as clear strengthening of boundary layer separation in the compressor without splitter blades (not shown). However, similar stretching of wake at the trailing edge is observed.

The influence of the decreasing Reynolds number on the wake at the trailing edge is further studied in Fig. 2, where contours of normalized meridional velocity are shown in spanwise direction for the compressor with splitter blades. The results indicate that the influence of a decreasing Reynolds number is similar in both compressors and thus the contours for the compressor without splitter blades are not shown. At high Reynolds number, wake is located at the shroud suction side corner. At low Reynolds number, wake stretches towards the hub and shifts closer to the blade suction side. With increasing and strengthening wake the velocity differences



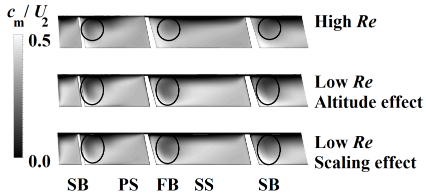


Figure 1: **Highlighted areas of boundary layer separation and wake.**

Figure 2: **Highlighted areas of wake.**

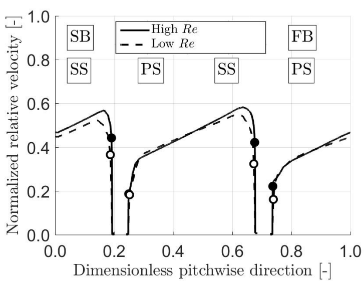
become larger at the trailing edge, resulting in increased mixing losses. The important finding when studying the flow fields at low Reynolds number is that there is no meaningful difference between the Reynolds number variation method.

Boundary layer thickness

The increase in boundary layer thickness with a decreasing Reynolds number is investigated next because it affects Denton's loss coefficients. Figures 3 and 4 show normalized relative velocity distributions in the compressor with splitter blades 50% of the chord length downstream from the full blade leading edge. The effects of altitude and scaling are shown by dashed lines in Figs. 3 and 4, respectively. The baseline high Reynolds number velocity distribution is shown by solid line in both figures. The edges of the boundary layers on blade surfaces are marked by circles (filled and non-filled circles refer to high and low Reynolds numbers, respectively).

Figure 3 presenting the altitude effect shows that relative velocity decreases slightly with a decreasing Reynolds number, which is totally opposite result compared to Fig. 4 presenting the scaling effect. The decreased velocity with low Reynolds number in Fig. 3 results from the numerical method. In order to remain all the other dimensionless numbers constant, except the Reynolds number, rotational speed and volume flow rate needs to be decreased with a decreasing Reynolds number. Therefore, the velocity components are decreased with a decreasing Reynolds number, but the velocity triangle and incidence angle remain equal to the high Reynolds number case as shown in Fig. 5.

On the other hand, in Fig. 4 the relative velocity increases with a decreasing Reynolds number due to the velocity components equaling to the high Reynolds number case. The increased velocity is due to the blockage. The blockage results in increased velocity also when



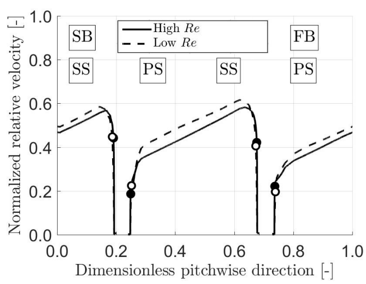


Figure 3: **Velocity profile. Altitude effect.** Figure 4: **Velocity profile. Scaling effect.**

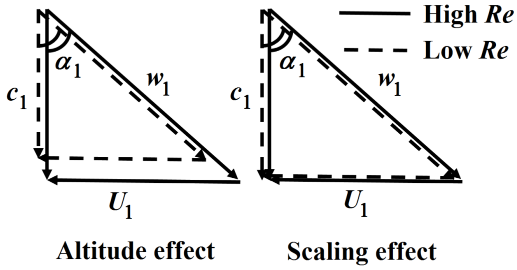


Figure 5: **Velocity triangles at the impeller inlet.**

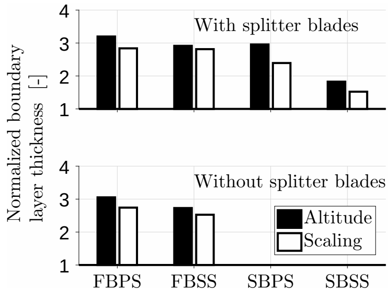


Figure 6: **Increase in normalized boundary layer thickness.**

the Reynolds number is decreased due to the effect of altitude, which is seen in Fig. 3 as an almost equal velocity distribution with low and high Reynolds numbers.

The boundary layer thickness increases in both cases. The weakness of the boundary layer definition is that it depends on the location of the data points, which are arbitrarily selected. In this study, the number of evenly distributed data points in the pitchwise direction from the full blade to the full blade is 10 000 corresponding to approximately 140 data points per millimeter in the compressor with splitter blades and 180 data points per millimeter in the compressor without splitter blades. Because the results for the compressors with and without splitter blades are similar, the corresponding figures of the velocity distributions are not shown for another compressor.

Figure 6 shows how the relative boundary layer thickness increases from high to low Reynolds number. Filled and non-filled bars present the increase in boundary layer thickness due to altitude and scaling effect, respectively. Only the mean values are presented on blade pressure and suction sides. The largest difference between the altitude and scaling effect is visible on splitter blade pressure side (24%), but mainly the difference is insignificant between the Reynolds number variation methods. Therefore, it can be concluded that the boundary layer thickness is increased approximately as much with both Reynolds number variation methods and the boundary layers are approximately three times thicker at low Reynolds number than at high Reynolds number.

Boundary layer loss coefficient

The challenge in using Denton's (1993) loss coefficients is that they are based on the velocity at the boundary layer edge. As shown above, it is difficult to estimate the boundary layer thickness in the blade passage of a centrifugal compressor. On the other hand, even though the boundary layer thickness increases in all the studied cases, relative velocity at the boundary layer edge decreases when the Reynolds number is decreased by varying inlet conditions. Therefore, the increased boundary layer thickness would indicate increased boundary layer losses, but decreased relative velocity indicates decreased boundary layer loss coefficient calculated by Eq. (1).

The normalized boundary layer loss coefficient, which is a sum of coefficients for full and splitter blades, is shown for the compressor with splitter blades in Figs. 7 (altitude effect) and 8 (scaling effect). Similar results are obtained for the compressor without splitter blades (not shown). These results indicate that the boundary layer loss coefficient is very sensitive

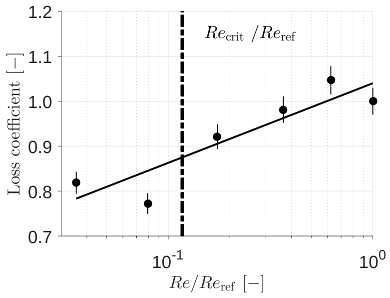


Figure 7: **Boundary layer loss coefficient. Altitude effect.**

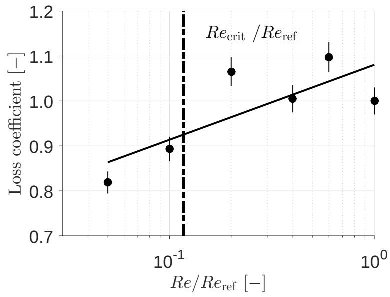


Figure 8: **Boundary layer loss coefficient. Scaling effect.**

to relative velocity. According to the sensitivity analysis, the boundary layer loss coefficient changes by 3% when relative velocity at the edge of the boundary layer is changed by 1%. Errorbars in Figs. 7 and 8 indicate this change of 3%.

In addition to the boundary layers at the blade surfaces, boundary layers at the shroud and hub cause losses. However, these losses are not estimated here because of the complexity of the flow field in a centrifugal compressor due to the tip leakage flow, flow separation, and secondary flows, which make the endwall losses difficult to be separated from other loss generation mechanisms. Instead of the endwall losses, the mixing out loss of the boundary layers and the entropy production due to the trailing edge are accounted for in the trailing edge loss coefficient, which is calculated in the following section.

Trailing edge loss coefficient

The terms of the trailing edge loss coefficient in Eq. (2) refer to the loss due to the low base pressure acting on the trailing edge, mixed out loss, and combined blockage of the trailing edge and the boundary layers (Denton 1993). Denton stated that in the compressors the third term might be more dominant than the first term due to boundary layer blockage being thicker than the trailing edge blockage. However, the results of this study indicate that the first term is more important than the third one, meaning that even though the boundary layer thickness increases indicating increased mixing losses at the trailing edge, the trailing edge loss coefficient does not change in the compressor with splitter blades as shown in Fig. 9 or even decreases as in Fig. 10 depending on the Reynolds number variation method.

These results might result from the difference between axial and radial machines. The base pressure term might be more dominant in centrifugal compressors than in axial ones due to the centrifugal effect, which causes higher pressure rise compared to that in axial machines. Since Denton's trailing edge loss coefficient is derived for axial blade row for which typical values of base pressure coefficient C_{pb} in Eq. (3) are in the range -0.1 to -0.2, it might be that the centrifugal effect causes lower values of base pressure coefficient, resulting in more dominant base pressure term in a trailing edge loss coefficient of centrifugal compressors.

Due to the shape of the blade trailing edge in the compressor without splitter blades, the average pressure acting on the base of the trailing edge is higher than the pressure on the blade suction side immediately before the trailing edge, resulting in negative base pressure term in Eq. (3). Dominant base pressure term in Eq. (2) results in negative trailing edge loss coefficient in the compressor without splitter blades (results not shown).

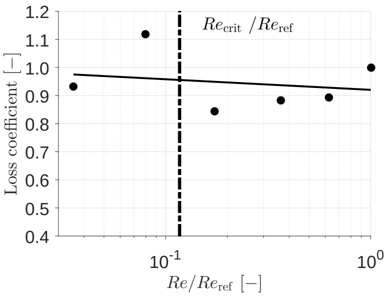


Figure 9: **Trailing edge loss coefficient. Altitude effect.**

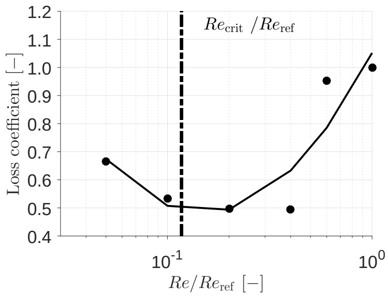


Figure 10: **Trailing edge loss coefficient. Scaling effect.**

The results about thickening boundary layers with a decreasing Reynolds number indicate that mixing losses behind the trailing edge are increased with a decreasing Reynolds number. Also the meridional velocity contours in Figs. 1 and 2 indicate increasing mixing losses due to the increased and strengthened wake. However, the dominance of the base pressure term in Eq. (2) causes decreasing of the trailing edge coefficient with a decreasing Reynolds number in this study.

Tip leakage loss coefficient

The last studied loss coefficient estimates the magnitude of the tip leakage loss. Similarly as the boundary layer loss coefficient, the tip leakage loss coefficient is sensitive to relative velocity; it changes by 3% when relative velocity at the edge of the boundary layer is changed by 1%, resulting in a significant uncertainty in the loss coefficient when even the boundary layer thickness is difficult to estimate. Especially, if the velocity on the blade suction side is lower than that on the blade pressure side, Eq. (4) results in complex value. The velocity on the blade suction side at the edge of the boundary layer could be lower than that on the pressure side near the blade leading edge when the flow separates.

This sensitivity of the tip leakage loss coefficient is even more severe in the tip clearance region close to the blade tip, where the jet-wake flow structure is dominant and more flow separation occurs than closer to the hub. Near the blade tip it is more difficult to get data where the suction side velocity is larger than pressure side velocity. Because it is not reasonable to estimate the tip leakage loss coefficient far away from the tip clearance region or delete undesirable data points $(w_{\mathrm{SS}} < w_{\mathrm{PS}})$, the tip leakage loss coefficient is not presented here.

Increase of specific entropy in impeller outlet subregions

The second method for investigating the loss generation mechanisms divides the impeller outlet into the subregions corresponding to different sources of loss and specific entropy change is calculated between compressor inlet and these regions. Figure 11 presents the increase in specific entropy when the Reynolds number is changed from high to low value. The upper subplot shows the values for the compressor with splitter blades and the lower one for the compressor without splitter blades. In both compressors, the altitude effect (filled bars) gives larger increase in losses than the scaling effect (non-filled bars). The maximum difference between the altitude and scaling effects occurs in the tip clearance, the difference being 22 percentage points for the compressor with splitter blades and 17 percentage points for the compressor without splitter blades.

The results given by the altitude and scaling effects are of the same order (in average, the differences are 15 and 13 percentage points for the compressors with and without splitter blades, respectively). Therefore, the hypothesis can be refuted. It was assumed that the tip leakage losses might not take as significant fraction of the total losses in the case of the altitude effect as in the case of the scaling effect, because the rotational speed is not changed when the inlet conditions are varied, unlike when the compressor is downscaled. However, the results indicate that the increase in tip leakage losses is not dependent on the rotational speed but on the Reynolds number if other dimensionless numbers are kept constant.

The results of the increased boundary layer thickness with a decreasing Reynolds number in Fig. 6 indicated that the boundary layer thickness increases approximately by 200% when the Reynolds number is decreased by 95%. However, Fig. 11 indicates that the boundary layer losses increase around 50% in the compressor with splitter blades and 35% in the compressor without splitter blades even though a linear correlation between the changes in boundary layer thickness and boundary layer losses could be assumed.

The weakness of the two-equation turbulence models might be the reason for the under-estimated increase in specific entropy in the boundary layers. It is generally known that the two-equation models under-predict the viscous losses and at low Reynolds numbers where the boundary layer thickness is relatively thicker, the two-equation models under-predict the viscous losses relatively more.

Effect of losses on performance

As the results presented above indicate, the performance of the compressors is deteriorated with a decreasing Reynolds number. Figure 12 shows normalized total-to-total isentropic efficiency for the studied compressors with and without splitter blades (SBs) in the case of the altitude (A) and scaling (S) effect. The numerical results in Fig. 12 are validated against the efficiency correction equation based on experimental data (Dietmann & Casey 2013).

According to the results, the efficiency is not decreased as much as the correction equation predicts below the critical Reynolds number. In the previous paper by the authors (Tiainen et al. 2016), it was speculated that the difference could be caused by the manufacturing tolerances in a small-scale compressors. The manufacturing tolerances were neglected in the numerical study and all the geometric dimensions of the compressors were downscaled by the same scaling factor, as described more in detail in the previous paper (Tiainen et al. 2016). However, the

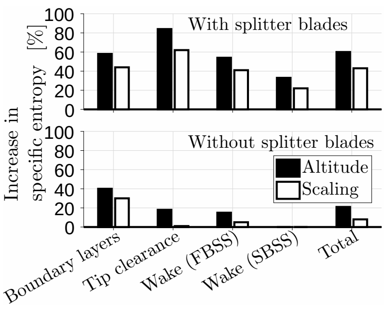


Figure 11: **Change in specific entropy with a decreasing Reynolds number.**

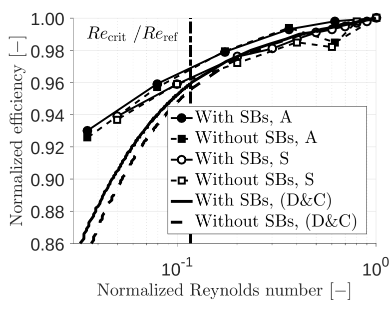


Figure 12: **Change in efficiency with a decreasing Reynolds number.**

reason for the discrepancy between the numerical results and the correction equation should not be caused by the manufacturing limitations in the case of the scaling effect, because the altitude effect gives identical results.

The effect of relatively larger surface roughness height is not accounted for when the compressor is downscaled since smooth surface approximation is used in the numerical study. However, the estimation of the viscous sublayer height ($y^+ \leq 5$) from the numerical results indicates that the roughness elements do not affect the performance since they are of the same order of magnitude or smaller than the viscous sublayer in the whole studied Reynolds number range. Therefore, the smooth surface approximation is valid in the numerical study.

The results indicate that the viscous sublayer increases as much in the case of the altitude effect as in the case of the scaling effect, that is from approximately 0.01% of the blade pitch at high Reynolds number to 0.20% at low Reynolds number. Surface roughness height is approximately 0.01% of the blade pitch.

Since both the altitude and scaling effect give similar results and the difference between them and the correction equation is as much with both studied compressors, it seems that the numerical model over-predicts the compressor performance below the critical Reynolds number. As discussed in the previous section, the two-equation models underestimate the viscous losses below the critical Reynolds number relatively more due to relatively increased boundary layer thickness. The difference between the numerical results and the correction equation could result from the use of two-equation model on the one hand, or the volute excluded from the numerical study on the other. With a decreasing Reynolds number the frictional losses increase also in the volute and since the numerical study is performed in the impeller and diffuser only, the losses associated with the volute are neglected in the numerical study but accounted for in the correction equation.

CONCLUSIONS

This paper aimed to answer to the following research question: Does the most dominant loss generation mechanism change if the Reynolds number is varied by changing the ambient conditions instead of changing the compressor size?

It can be concluded that the Reynolds number variation method does not markedly affect the loss generation since all the results indicate that there is no difference between the Reynolds number variation methods. Between two Reynolds number variation methods: 1) Flow fields are identical at low Reynolds numbers, 2) Boundary layer and viscous sublayer thicknesses increase as much, 3) Loss distributions are identical, and 4) The increase in specific entropy from high to low Reynolds number is of the same order. Therefore, the hypothesis about the differences in the loss generation mechanisms between the Reynolds number variation methods is refuted.

Additional conclusions are the following: 1) The results of the estimated loss coefficients cannot be utilized in the centrifugal compressors, 2) The challenge of using Denton's (1993) loss coefficients in the centrifugal compressors is the sensitivity of the loss coefficients to the value of velocity at the boundary layer edge, 3) When even the estimation of the boundary layer thickness is difficult, the value of velocity at the boundary layer edge results in uncertainty in the loss coefficients, 4) The less sensitive method for analyzing the loss generation mechanisms inside the blade passages is to divide the impeller outlet roughly into subregions, which are based on different causes of loss, and 5) The method based on impeller outlet subregions could be made more accurate in the future, if a general method for defining the boundary layer thickness in the blade passage was defined.

In the future, the Reynolds number effects can be investigated either by downscaling the baseline compressor or by varying the compressor inlet conditions. However, the researcher has to ensure that all the other dimensionless numbers except the Reynolds number remain constant between the baseline and low Reynolds number case so that the same operating point is compared between the high and low Reynolds number cases.

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