Aerodynamic analysis of the small-scaled centrifugal compressor for micro-turbojet engine applications

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Abstract. This paper presents the results of an aerodynamic analysis of a small-scaled transonic centrifugal compressor for micro turbojet engine (TJE) applications. The analysis was conducted using a CFD model validated by experimental data collected for the gaged JetCat P200-RX micro-TJE. The loss coefficients of the impeller, vaned diffuser and deswirler were estimated for 4 design points corresponding to 70%, 80%, 90% and 100% rotation speeds to perform the loss balance diagram. The flow angle spanwise variation demonstrated an intensive flow separation zone at the top 25% of the vaned diffuser span due to the pressure shock appearance. It was shown that the main source of the losses in the investigated compressor is the deswirler due to non-optimal flow turning conditions. The diffuser loss coefficient was estimated as 0.18 at the compressor full load.

1. Introduction
Centrifugal compressors are widely used for numerous applications such as aircraft engines, gas compressor units, turbochargers, and distributed power generation systems. Their efficiency varies from 70% up to 90% depending on the compressor dimensions and power output. A higher efficiency level corresponds to high capacities. Contrariwise, the efficiency of high-speed small-scaled machines typically does not exceed 76 – 80% due to high relative leakages, secondary and friction losses (see, e.g., Casey (1985) [1], Pampreen (1973) [2], Senoo and Ishida (1987) [3], Cumpsty (2004) [4]). Compressors of this type are commonly used for micro turbojet engines (TJE). Performance of state-of-the-art micro-TJE varies from 26% to 32% (see, e.g., Fathy et al. (2018) [5], Shukla (2013) [6], [7], [8]). One of the factors for such a situation is the comparably low efficiency of the turbine and compressor. In many cases, micro-TJE compressors are highly loaded because of the strict size-weight limitations which lead to the occurrence of local supersonic zones and additional losses. Micro-TJEs are generally used for different drones and unmanned aerial vehicles (UAV). Since the UAV international market is expected to see a 1.5 growth rate (see J’son & Partners [9]) micro-TJE performance improvement becomes extremely important. Therefore, understanding of the main loss mechanisms in the small-scaled compressors flow paths is of high interest.

The traditional experimental approach is not fully applicable for examining the flow phenomena in small-scaled turbomachinery since the pressure probes and thermocouples are similar in size to the blade height. Scaling of the flow path using similitude rules leads to the elimination of the strong curvature and Reynolds number effects (see, e.g., Barsi et al. (2019) [10]). Integral methods, when the rotor torque is measured, are not able to provide an understanding of the loss breakdown. A 3D CFD approach provides a huge amount of flow parameters but in many cases thorough validation of the model is needed. Xiang et al. (2016) [11] provided detailed validation of the numerical model of the KJ-66 micro-TJE compressor. Reasonable agreement between CFD and experimental data was
achieved. Bogdanets et al. (2018) [12] validated the numerical model of the Eckardt compressor test case with 400 mm impeller tip diameter and vaneless diffuser. The CFX Frozen rotor model was used for the coupling of the stationary and rotating domains. Han et al. (2017) [13] studied different deswirler configurations of the centrifugal compressor with an impeller tip diameter of 78 mm and rotating speed 63000 rev/min. The numerical model was validated based on the surrogate experimental stage without deswirler. A good agreement between the calculation and experiment was obtained. Surprisingly, CFD calculations demonstrated a slightly higher stall margin combined with the fact of the deswirler absence in the experimental stage. Authors observed that in some cases the so-called fishtail diffuser could provide stage efficiency growth and decrease the pressure non-uniformity. Danish et al. (2014) [14], Mosdzien et al. (2018) [15] showed the serious potential of impeller modification and employing the tandem configuration to increase the efficiency of the centrifugal compressors with a vaneless diffuser. Li et al. (2020) [16] demonstrated prospects of a tandem impeller configuration for the compressors of interest both numerically and experimentally. Since the efficiency of small-scaled centrifugal compressors for micro-TJE still has high potential to be improved the scope of the present paper is to provide an aerodynamic analysis of the flow phenomena and outline the main loss sources for this type of machines.

2. Methods

2.1. Experimental compressor

The centrifugal compressor of the JetCat P200-RX micro-TJE was chosen as the object of the study. Its main geometric parameters and operation conditions are presented in table 1. A 3D model of the compressor is presented in figure 1a.

| Parameter | Dimension | Value | Parameter | Dimension | Value |
|-----------|-----------|-------|-----------|-----------|-------|
| $D_1$     | mm        | 62    | $Z_{VD}$  | -         | 17    |
| $D_2$     | mm        | 84    | $Z_{DS}$  | -         | 34    |
| $D_3$     | mm        | 114   | $u_{2\text{ max}}$ | m/s | 492.6 |
| $D_4$     | mm        | 130   | $\pi_{s_{1}}$ [17] | - | 4.0 |
| $Z_{\text{IMP}}$ | - | 6+6 | $\eta_{1_{s_{1}}}$ [17] | - | 0.72 |

2.2. Numerical Set-up

The Ansys CFX commercial solver was used to perform 3D steady-state Reynolds Averaged Navier Stokes (RANS) simulations. A summary of the solver settings is presented in table 2, a representation of the compressor “fluid” model – in figure 1b.
Table 2. A summary of the solver settings

| Parameter         | Setting                                      | Parameter         | Setting          |
|-------------------|----------------------------------------------|-------------------|------------------|
| Approach          | Steady-state RANS                            | Heat transfer     | Adiabatic walls  |
| Advection scheme  | Adaptive 2nd order (High resolution)         | Energy option     | Total energy     |
| Timescale         | Physical, 2e-5 s                             | Turbulence model  | various          |
| Working fluid     | $C_p$ – zero-pressure polynomial function    |                   | $\lambda$ – constant |
|                   | $\mu$ – Sutherland equation                 |                   |                  |
|                   | $\lambda$ – constant                         |                   |                  |

The numerical approach presented is similar to other related publications (see, e.g., Barsi et al. (2018) [10], Xiang et al. (2016) [11], Han et al. (2017) [13], Ling et al. (2007) [18]). The authors provided additional rotor-stator interface type investigation, which is presented in figure 2a. It was shown that using the Frozen rotor interface type leads to an unexpected impeller pressure ratio decrease. Therefore, it should be avoided in the case of the impeller – vaned diffuser coupling. Nevertheless, it is suitable for the impeller – vaneless diffuser coupling as was shown by Bogdanets et al. (2018).

Figure 2. Compressor pressure ratio for the Stage and Frozen rotor interface models (a) and results of the grid convergence study (b)

Tetra mesh with prism layers was used since the deswirler has a non-axisymmetric hub endwall. A grid convergence study was conducted for 3 sequentially refined meshes − 2.1M nodes (coarse grid (CG), 1M / 1.1M for the impeller / diffuser), 4.14M nodes (medium grid (MG), 1.36M / 2.78M for the impeller / diffuser), 6.31M nodes (fine grid (FG), 3.29M / 3.02M for the impeller / diffuser). Simulations for the coarse and medium grids were performed using wall functions ($30 < y^+ < 300$), for the fine grid – without wall functions (average $y^+$ is less than 3). The results of the grid convergence study are presented in figure 2b for 3 compressor speedlines (80%, 90% and 100% of nominal rotation velocity). The obtained results demonstrate that a medium grid with wall functions provides sufficient accuracy even in the stall region. For the investigated compressor this result is related to the high volume of the flow separation zones in the diffuser caused by the flow turning from a radial to axial direction.

2.3. Numerical model validation

Validation of the numerical model presented was conducted based on experimental data collected for the gaged JetCat P200RX micro-TJE in terms of the outlet total temperature and compressor total-to-total pressure ratio. The scheme of the engine gaging is provided in figure 3, the results of the model validation – in figure 4a.
Analysis of the outlet total temperature demonstrates a good agreement between numerical and experimental data. Meanwhile, numerically obtained total-to-total pressure ratio values are significantly higher than experimental ones and closer to the manufacturing data (17). A possible reason for such a deviation is the effect of the engine gaging which is also observed at the comparison thrust plot for the gaged and ungagged engine (see fig.4b). Since the numerically obtained outlet total temperature is close to the experimental values for the gaged engine, for the ungagged engine the numerical model is expected to provide slight (in range of 0.3…0.7%) efficiency underprediction.

3. Results and discussion

Key factors which lead to efficiency deterioration of small-scaled compressors are pressure shocks appearance and a high volume of the flow separation zones compared with the flow path dimensions (see Pampreen (1973) [2], Senoo and Ishida (1987) [3]). These flow phenomena are illustrated for the examined compressor in figure 5 for 4 design points corresponding to the 70%, 80%, 90% and 100% rotation speeds. Strong pressure shock appearance at the top half of the impeller leading edge is particular to compressor rotation speeds higher than 90% of nominal speed. However, flow separation at the impeller blade suction side appears at approximately the top 25% of the impeller span and does not demonstrate a direct correlation with the pressure shock since it is common with all investigated rotation velocities. This phenomenon has a complex nature. Mainly it is related to the impeller tip leakage which enhances the effect of the flow separation due to its turning from an axial to radial
direction as illustrated in figure 5. On the other hand, the energy equation in the relative frame demonstrates the condition while the relative velocity approaches zero (reversed flow occurrence):

\[
(w_M^2)_{Sp} = 2(h_{i,Sp} - h_{r,Sp}) + (u_M^2 - u_i^2)_{Sp} + (w_i^2)_{Sp},
\]

where “M” and “Sp” subscripts denote the current streamwise and spanwise coordinates respectively. Analysis of equation (1) shows that this condition fulfillment is possible when the absolute value of enthalpies difference exceeds the difference of circumferential velocity squares. Physically this could mean that the shape of the impeller meridional section is non-optimal relative to the impeller pressure ratio.

![Figure 5. Relative frame Mach number plots and streamwise velocity plots in the impeller (right column)](image)

Analysis of the loss balance diagram (fig.6a) demonstrates that total-to-total impeller efficiency decreases from 0.864 at 70% rotation speed to 0.818 at 100% rotation speed. These are sufficiently high efficiency values for the transonic impeller. The vaned diffuser total pressure loss coefficient in figure 6a includes the rotor-stator interaction losses. At 70% rotation speed the total pressure loss coefficient of the vaned diffuser is approximately 1.6 times lower than its value for the deswirler. The loss balance diagram demonstrates that total pressure losses in the vaned diffuser and deswirler become substantially identical as the rotation speed exceeds 90%. The total pressure distribution along the flow path (fig.6b) shows a dramatic increase of the total pressure losses in the vaned diffuser at 90% and 100% rotation speeds.
Figure 6. The loss balance diagram of the compressor (a) and stationary frame total pressure distribution along the compressor flow path (b).

Analysis of the flow angle span variation at the vaned diffuser inlet (fig.7a) shows the occurrence of the secondary flow zones. The hub separation zone occupies approximately 10% of the blade span and does not change its volume for all the rotation speeds. It tends to minimize towards the vaned diffuser outlet as illustrated in figure 7b. The shroud separation zone increases as the rotation speed growths: from 14% of span at 70% rotation speed to 26% of span at 100% rotation speed. This is related to the appearance of the pressure shock above 75% of the blade span which leads to the flow separation at the blade suction side. This is a reason for increasing the vaned diffuser loss coefficient with the growth of the rotation speed. The shroud separation zone also tends to decrease towards the vaned diffuser outlet (see fig.7b).

Figure 7. The flow angle span variation at the inlet of the vaned diffuser (a) and Mach number plots along its flow path at full load (b) (zero flow angle values correspond to the reversed flow).

Analysis of the flow angle span variation at the deswirler inlet (fig.8a) demonstrates a significant hub reversed flow zone which occupies approximately 20% of the blade span. At the top 80% of span inlet flow the angle varies from 30° to 40°. Nevertheless, the flow has a significant centripetal velocity component which leads to the formation of the shroud secondary vortex as illustrated in figure 8c. This vortex occupies up to a half of the deswirler cross-section at the outlet. As a result, the outlet flow angle (fig.8b) varies from 35° to 70° at 75% of span. This causes additional losses due to the secondary vortex dissipation and flow mixing.
Figure 8. The flow angle span variation at the deswirler inlet (a) and outlet (b) (zero values correspond to the reversed flow zones), Mach number plots along its flow path at full load (c)

4. Conclusion

It was shown that the main source of losses for the compressors of interest is the diffuser which combines the vaned diffuser and deswirler. Its loss coefficient may reach up to 0.18 at the compressor full load. Pressure shock appearance at the top 25% of the vaned diffuser span causes flow separation at the suction side. This leads to the increase of the vaned diffuser loss coefficient from 0.053 to 0.086 as the rotation speed growths from 70% to 100%. The deswirler loss coefficient varies from 0.082 to 0.095 due to non-optimal conditions of the flow turning from a radial to axial direction. The outlined flow phenomena cause relatively low efficiency of small-scaled high-speed centrifugal compressors compared to full-scale machines.

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Nomenclature

- \( C_p \) specific heat capacity at constant pressure (kJ/(kg·K))
- \( l_{\text{norm}} \) span normalized coordinate
- \( n \) rotation speed (rev/min)
- \( u \) circumferential velocity (m/s)
- \( \zeta \) total pressure loss coefficient
- \( \lambda \) thermal conductivity (W/(m·K))
- \( \eta_{\text{t-t}} \) total-to-total efficiency
- \( p \) pressure (Pa)
- \( T \) temperature (K)
- \( w \) relative velocity (m/s)
- \( Z \) number of blades
- \( \pi_{\text{t-t}} \) total pressure ratio
- \( \mu \) dynamic viscosity (Pa·s)

Abbreviations

- DS deswirler
- IMP impeller
- VD vaned diffuser
- TJE turbojet engine

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