Numerical Investigation of Trailing Edge Flow in Centrifugal Pump Impellers

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ABSTRACT
In the fields of aerodynamics and turbomachinery trailing edge flows are of great interest as they can substantially affect performance, vibration and sound emission. The present paper discusses the trailing edge and wake flow in centrifugal pump impellers. In the first part of the study the effect of the trailing edge flow on the performance of equiangular impeller blades was investigated by means of 2D steady RANS simulations for a wide range of blade angles and flow coefficients. The effect was quantified by the slip coefficient as a measure for the overall trailing edge flow deflection from the theoretical, inviscid flow through an impeller with an infinite number of blades. For the typical cut-off or knuckle-type trailing edge configuration an attached and a separated flow regime could be identified, mainly depending on the blade angle. Consequently, in the second part of the study, detached-eddy simulations were carried out, revealing the flow structures of the different trailing edge flow regimes in greater detail.

KEYWORDS
CENTRIFUGAL PUMP, IMPELLER, TRAILING EDGE, SLIP FACTOR, DES

NOMENCLATURE

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| \( \beta \) | Flow angle (measured towards \( \theta \)) |
| \( \gamma \) | Meridional inclination angle |
| \( \rho \) | Fluid density |
| \( \sigma_{CFD} \) | Slip coefficient \( \sigma_{CFD} = 1 - \Delta C_{m2}/U_2 \) |
| \( \varphi \) | Local flow coefficient \( \varphi = C_{m2}/U_2 \) |

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INTRODUCTION

Trailing edge flows are of great importance in aerodynamics and in turbomachinery applications. The flow around the trailing edge of an airfoil or an impeller blade can strongly affect the aerodynamic or hydraulic performance in terms of lift or work. Also vibrations can arise from unsteady phenomena as well as sound emission. Therefore these types of flows have been studied in the past both experimentally and also numerically. For example, an important experimental work in this field was done by Blake (1975), who carried out wind tunnel experiments of a flat parallel strut with different exchangeable trailing edges by employing surface pressure measurements and hot-wire anemometry. Paterson and Peltier (2005) presented unsteady RANS and detached-eddy simulations of this experimental setup. More recently, van der Velden et al. (2016) performed wind tunnel experiments of a very similar setup as well as a large-eddy simulation of the setup including far-field acoustics prediction. Detailed flow measurements within a centrifugal impeller were performed by Ubaldi et al. (1998) by means of laser laser Doppler velocimetry (LDV). Pedersen et al. (2003) presented results from particle image velocimetry (PIV) and LDV measurements within a centrifugal pump impeller at design and off-design conditions. A comprehensive series of investigations of the flow through a radial diffuser pump was published by Feng et al. (e.g. 2007, 2009a, 2009b, 2011) presenting detailed flow features obtained from PIV and LDV measurements as well as from numerical simulations. For centrifugal and mixed-flow pump impellers recent literature indicates that the trailing edge shape can strongly influence the pumps head and also efficiency. Wu et al. (2015) reported up to 25 percent head increase at nominal flow rate for a mixed-flow pump by modifying the trailing edge suction side with different radii. Also the maximum efficiency was increased by around 5.5 percent while shifting the best efficiency point by around 12 percent to a higher flow-rate. The effect of different trailing edge profiles on the performance of a low-speed centrifugal pump was investigated by Gao et al. (2016). They could achieve an increase in maximum efficiency of 2.5 percent and also a significant decrease of the unsteady pressure pulsations for some trailing edge profiles. A significant effect of trailing edge modifications on head and efficiency of centrifugal waste-water pump impellers was reported by Litfin et al. (2017).

The present paper focuses on the trailing edge flow of equiangular impeller blades. When designing an impeller an accurate estimation of the deflection of the impeller exit flow from the theoretical, inviscid flow through an impeller with an infinite number of blades is of great importance. This flow deflection directly affects the work input and is a direct consequence of the angular momentum balance, which can be expressed with the well known Euler turbomachinery equation

$$g \cdot H = U_2 \cdot C_{\theta 2} - U_1 \cdot C_{\theta 1}$$

for an ideal frictionless flow. The flow deflection is usually taken into account by employing a slip factor model. Such a model gives an estimation for the slip velocity $C_{\text{slip}}$ by which the theoretical circumferential velocity $C_{\theta 2,\infty}$ is reduced. With this slip velocity a slip factor can be defined in the form

$$\sigma = 1 - \frac{C_{\text{slip}}}{U_2}.$$
slip velocity approximation of

\[ C_{\text{slip}} = K \cdot U^2 \cdot \frac{\pi \cdot \sin \beta b}{Z}, \]  

(3)

where \( K \) is an additional coefficient to improve the approximation and is usually taken as unity. A series of works dealing with the exact solution of the two-dimensional frictionless flow through a radial impeller was published around the 1920s amongst the work of Busemann (1928) is the most prominent. Busemann published mathematical exact solutions for the two-dimensional flow through radial equiangular vane systems with varying radius ratio and blade numbers. He presented his results in the form of curves for basically two correction factors to the Euler turbomachinery equation (Eqn. 1). For the case of zero pre-swirl, this equation can be rewritten to

\[ g \cdot H_{\text{th,\infty}} = U^2 - \frac{U \cdot C_m^2}{\tan \beta b}, \]  

(4)

which gives the Euler head for the frictionless flow through an impeller with an infinite number of infinitesimally thin blades. With the correction factors of Busemann, Eqn. 4 reads

\[ g \cdot H_{\text{th}} = h_0 \cdot U^2 - h_\nu \cdot \frac{U \cdot C_m^2}{\tan \beta b}. \]  

(5)

The correction factor \( h_0 \) arises from the "displacement flow" produced by the impeller rotation. The other factor \( h_\nu \) originates from the "through flow" through the impeller at rest. Here it has to be noted that, while this form of Eqn. 5 is convenient for backward curved impellers, the corresponding equation looks slightly different in Busemann’s original publication due to a different definition of the blade angle. Busemann’s curves show that the \( h_\nu \) correction factor is close to unity if the blade length is large compared to the vane spacing. This is usually the case for backward curved pump impellers. Subsequently, for typical centrifugal pump impellers the \( h_0 \) correction factor is dominant. Also including the slip velocity (Eqn. 2) into the Euler turbomachinery equation (Eqn. 1) yields

\[ g \cdot H_{\text{th}} = U^2 \cdot (C_{\theta,\infty} - C_{\text{slip}}), \]  

(6)

which can be rewritten with Eqn. 2 as

\[ g \cdot H_{\text{th}} = \sigma \cdot U^2 - \frac{U \cdot C_m^2}{\tan \beta b}. \]  

(7)

Comparison with Eqn. 5 shows that for \( h_\nu = 1 \) Busemann’s \( h_0 \) correction factor equals the slip factor \( \sigma \) in Eqn. 2. Wiesner (1967) gave a review of various methods for slip factor estimation as well as a simple empirical expression fitted to Busemann’s results:

\[ \sigma_{\text{Wiesner}} = 1 - \sqrt{\sin \beta b} \]  

(8)

More recently, von Backstroem (2006) proposed a slip factor model based on the assumption of a single relative eddy for the whole rotor. His derivations revealed the impeller solidity as the main parameter, yielding to a slip factor estimation of

\[ \sigma_{\text{SRE}} = 1 - \frac{1}{1 + 5(l/s^2)\sqrt{\sin (\beta b)}}. \]  

(9)
with the impeller solidity

\[ l/s_2 = \frac{(1 - RR)Z}{2\pi \cdot \sin(\beta_2b)} \]  

(10)

A different approach based on a blade loading estimation was presented by Qiu et al. (2011), yielding to additional flow dependent terms in the slip factor calculation.

\[ \sigma_{Qiu} = 1 - \frac{F_\pi \sin(\beta_2b)\sin(\gamma)}{Z} - \frac{F_2\phi_2}{4\sin(\beta_2b)} \left( \frac{d\beta}{dm} \right)_2 + \frac{F_2\phi_2\sin(\beta_2b)}{4\rho_2} \left( \frac{d\rho_b}{dm} \right)_2 \]  

(11)

Here the two flow dependent terms include the blade turning rate \(\frac{d\beta}{dm}\) and the passage width variation \(\frac{d\rho_b}{dm}\), while \(F\) is a shape factor, that can be derived from the impeller geometry.

In the present work the slip factor is evaluated from viscous flow simulations based on the Navier-Stokes equations. Like described above the theory of impeller slip is based on the assumption of frictionless flow. Strictly, it is not possible to evaluate the slip factor from a viscous CFD solution. This is due to the fact that the overall reduction in circumferential velocity, that can be deduced from viscous CFD simulations, includes not only the theoretical slip velocity but also a velocity change due to viscous effects. In this context also it has to be kept in mind that Busemann’s calculations assume infinitesimally thin blades. Hence, effects of the trailing edge shape of blades with finite thickness on the flow at the impeller exit were beyond the scope of his work. Moreover, like Busemann’s calculations show, the theoretical slip factor \(h_0\) or \(\sigma\) is a constant for a given impeller and is therefore independent of the flow conditions. This is usually not true if a slip factor is calculated from viscous flow data (Qiu et al., 2011, Litfin et al., 2017). Keeping this in mind and in order to avoid confusion with the theoretical slip, a CFD slip coefficient \(\sigma_{CFD}\) is used. This coefficient incorporates all phenomena that modify the absolute circumferential velocity at the impeller exit and therefore is calculated according to the following formula:

\[ \sigma_{CFD} = 1 - \frac{\Delta C_{\theta_2}}{U_2} \]  

(12)

with

\[ \Delta C_{\theta_2} = C_{\theta_2,\infty} - C_{\theta_2} \]  

(13)

where \(C_{\theta_2}\) is evaluated by mass-flow averaging at the impeller exit radius.

**NUMERICAL METHOD AND PROCEDURE**

**Blade Geometry**

In the present study equiangular blades were considered with blade angles \(\beta_b\) ranging from 15 to 60 degrees. The diameter ratio \(D_1/D_2\) was chosen to be 0.5 while the blade number was 8 for all cases. The relative blade normal thickness \(t/D_2\) was 0.02. The blades all had an elliptic leading edge shape with an ellipse ratio of two. The trailing edge shape was either the typical cut-off (knuckle-type) trailing edge or an elliptic shape with an ellipse ratio of 20. The impeller Reynolds number based on the impeller exit diameter \(D_2\) and the rotational velocity at the exit \(U_2\) was around \(7.5 \times 10^6\). No stator equipment was considered in the present study because the focus of the study is on the isolated effect of the trailing edge shape on the impeller flow and the apparent impeller slip. This is of course a major simplification since centrifugal pumps usually have a volute or diffuser or a combination of both, which can significantly affect the impeller flow due to rotor-stator interaction and it’s induced unsteadyness.
2D Steady RANS

Two-dimensional numerical simulations were performed by solving the steady incompressible Reynolds-averaged Navier-Stokes equations with the CFD-solver ANSYS® CFX® 18.1. The computational domain comprised of one impeller blade by making use of the rotational periodicity of the impeller blading. The domain was extended upstream and downstream of the impeller blade in order to assure that the boundary conditions are specified sufficiently far away from the impeller blade. The necessary length of the extensions was obtained in a preliminary study. Since the employed numerical solver is purely three-dimensional the computational meshes were generated as quasi two-dimensional meshes consisting of only one cell layer in direction of the third dimension. The meshes for the different blade angles, all generated with the same settings, had an average of 240,000 elements, varying between 160,000 and 430,000 depending on the blade angle. Inflation layers were used to maintain a non-dimensional wall distance $y^+$ below one. A velocity boundary condition was specified at the inlet, while at the outlet the average static pressure was given. Rotational periodic boundary condition was used, since only one blade was modeled. Translational periodicity was specified in the third dimension. The blades were modeled as smooth walls with a no slip condition specified. For spatial discretization of the advection term a second order upwind scheme was used. Turbulence was modeled by using the shear–stress–transport model (Menter, 1994).

Detached-Eddy Simulation

Detached eddy simulation is a hybrid RANS-LES numerical method in which a turbulence model functions as a LES subgrid-scale model in regions where the grid is fine enough for LES, while it acts as a standard RANS model in the other regions. The basic idea behind hybrid RANS-LES methods like DES is to apply a RANS turbulence model in the boundary layer while using LES for the outside flow. This approach results in an enormous reduction in grid cells as the RANS model requires much less grid resolution in the boundary layer than LES. The method was originally proposed by Spalart et al. (1997). To overcome some deficiencies of the standard DES method regarding the switch from RANS to LES, modifications to the method were made by adding a shielding function that prohibits undesired switching from RANS to LES inside the boundary layer (Menter et al., 2003, Spalart et al., 2006). This modification was termed delayed detached eddy simulation (DDES). In the present work the improved delayed detached eddy simulation model (IDDES) as proposed by Shur et al. (2008) with an underlying SST RANS model was used. This DES variant is an improvement to the standard DDES.
method including wall-modeled LES (WMLES) capabilities, which enables the use of RANS in a much thinner region close to the wall. The simulations were performed with the CFD solver Simcenter™ STAR-CCM+™ 11.06.010. The computational domain was similar to the 2D simulations, except the spanwise direction, which was chosen to be 10 percent of the impeller exit diameter $D_2$. A coarse and a fine mesh variant were generated for every case both consisting of hexahedral elements. The fine meshes were generated with half the grid spacing as the coarse meshes in all directions with exception of the wall normal spacing in the near-wall region. For all meshes a $y+$ value below one was maintained. The coarse meshes had a cell count between 7.5 and 10.3 million, while the fine meshes varied between 45.3 and 75.7 million cells, depending on the blade angle. At the inlet the velocity was specified. Also synthetic turbulence was applied at the inlet. At the outlet a pressure boundary condition was used. The blade was modeled as a smooth no slip wall. Rotational and translational periodicity was used at the other boundaries. A sample computational domain and a detailed view of the mesh is depicted in Fig. 1. An incompressible implicit unsteady segregated solver was used. A high accuracy second order temporal scheme was employed. The nondimensional time step was set to $\Delta t \cdot U_2/D_2 = 1.53 \times 10^{-3}$. Five inner iterations per time step were used to ensure time step convergence. For the convective terms a hybrid scheme was used, which switches between a bounded central differencing scheme in LES regions and a second order upwind scheme in the RANS regions. Each DES simulation was initialized with the converged corresponding 3D-RANS solution. After a time corresponding to five complete impeller rotations statistical sampling was performed for ten impeller rotations.

RESULTS

2D RANS Results

Simulations of blades with blades angles from 15 to 60 degrees in steps of 5 degrees were performed for both trailing edge configurations. For the first set of simulations the flow coefficients for the different blade angles were specified to maintain blade aligned flow at the leading edge according to the equation

$$\phi = \frac{C_{m2}}{U_2} = \tan(\beta_b) \cdot \left(\frac{D_1}{D_2}\right)^2,$$

which is valid for radial impellers with constant width and equiangular blades. The CFD slip coefficient was calculated according to Eqns. 12 and 13, where $C_{\theta2}$ was evaluated by mass-flow averaging at $r = R_2$. In Fig. 2 the slip coefficient evaluated from CFD is presented for the two different trailing edge configurations over the whole range of simulated blade angles. The two trailing edge configurations show a very different behavior. For the elliptic trailing edge the slip coefficient drops with increasing blade angle almost linearly from 0.82 at a blade angle of 15 degrees to 0.71 at 60 degrees. In case of the cut-off trailing edge the slip coefficient initially drops from 0.82 to 0.77 between 15 and 25 degrees. From a blade angle of 25 to 30 degrees the slip coefficient increases to 0.8. After that the slip coefficient decreases with increasing blade angle to a value of 0.77 at a blade angle of 60 degrees. This behavior with the jump in slip coefficient between 25 and 30 degrees indicates a significant change in the trailing edge flow. Here it has to be pointed out that the significant differences between both trailing edge shapes are even more remarkable when keeping in mind that both variants share the exact same camberline. For comparison slip factor values from Busemann taken from his original publication (Busemann, 1928) are shown. While at lower blade angles the CFD slip
Figure 2: Comparison of CFD slip coefficients for cut-off and elliptic trailing edge

coefficient for the cut-off trailing edge is much lower than the values calculated by Busemann, the curve from CFD fits Busemann’s results remarkably well at blade angles above 30 degrees. The CFD slip coefficient values for the elliptic trailing edge are well below Busemann’s results over the whole range of blade angles. Also the corresponding slip factor values for the models from Stodola, Backstroem and Qiu et al. are depicted. The values calculated with Backstroem’s model are very close to Busemann’s results, like already shown in his original publication. The model proposed by Qiu et al. falls back to a Stodola model with a correction factor, since both flow dependent terms in Eqn. 11 are zero for the investigated case with equiangular blades and no passage width variation. In the present case the correction factor, calculated according to the original publication, leads to a strong underprediction of the impeller slip. Contour plots of the non-dimensional relative velocity $W/U_2$ at three different blade angles are depicted in Fig. 3 for both trailing edge shapes. The velocity fields on the left side of the figure show that for the elliptic trailing edge the overall flow pattern does not change substantially with the blade angle. For all three blade angles there is a small separation zone at the last portion of the trailing edge’s suction side. Furthermore, it can be clearly observed that the wake departs the trailing edge at a lower angle than the camberline angle of the blade. On the right side of Fig. 3 the flow field around the cut-off trailing edge is depicted. For this trailing edge shape two different flow regimes can be distinguished. At a blade angle of 25 degrees there is no significant separation zone. The flow on the pressure side is accelerated around the obtuse corner and remains attached for most of the cut-off surface, resulting in additional flow deflection towards the circumferential direction. At both higher blade angles the flow behavior changes substantially. The flow cannot follow the obtuse corner anymore and detaches completely, resulting in a large separation zone. Due to the flow separation the flow deflection is strongly decreased resulting in less slip and therefore a higher slip coefficient. This phenomenon explains the jump in slip coefficient shown in Fig. 2.

In a second step the effect of the flow conditions at the impeller exit on slip coefficient and trailing edge separation was studied. The flow conditions are characterized by the local
Figure 3: Trailing edge flow fields for elliptic (left) and cut-off (right) trailing edge for blade angles of 25, 35 and 45 degrees (from top to bottom)

flow coefficient $\varphi = C_{m2}/U_2$. Therefore, additional simulations at different flow coefficients were performed. Since the trailing edge flow was of interest, possible effects of misaligned leading edge flow and resulting leading edge separation was prohibited by specifying also a circumferential component of the inlet velocity in order to maintain aligned leading edge flow. The results are shown in Figs. 4 and 5. In the figures shown, the flow rate is presented in terms of the angle $\beta_{1,\varphi} = \text{atan}(\varphi \cdot (D_2/D_1)^2)$ according to Eq. 14. This makes it easier to see if the flow through the blade channel is in nominal, part load or overload conditions. From hereon this angle will be referred to as the inlet flow coefficient angle $\beta_{1,\varphi}$. It has to be noted that regardless of $\beta_{1,\varphi}$ the inlet flow angle $\beta_1$ in the simulations is always kept equal to the blade angle $\beta_b$ by the specified circumferential inlet velocity component. Fig. 4 shows a 3D surface plot as well a color map of the slip coefficient as a function of the blade angle $\beta_b$ and the inlet flow coefficient angle $\beta_{1,\varphi}$. The graph shows a plateau with high slip coefficients for blade angles above 30 degrees. Between a blade angle of 25 and 30 degrees there is a jump in slip coefficient for most of the inlet flow coefficient angles. Interestingly, the jump or change in flow regimes does
not shift significantly to other blade angles for the most part of the simulated flow rate range. Only at very low inlet flow coefficient angles the jump almost vanishes, while at very high flow coefficients the jump is shifted slightly towards smaller angles. At low blade angles and high inlet flow coefficient angles the slip coefficient decreases strongly. The results shown in Fig. 2 are also included in this graph and are located on an imaginary diagonal section from the "north-east" to the "south-west" corner through the figure, where $\beta_b = \beta_{1,\varphi}$. In Fig. 5 the slip coefficient map for the elliptic trailing edge shape is depicted. In contradistinction to Fig. 4 the plotted surface is rather smooth without a clearly noticeable jump in slip coefficient. For low blade angles and high inlet flow coefficient angles the slip coefficient also drops dramatically. A plateau area like in the case of the cut-off trailing edge is not existing. For all blade angles there is a monotonic decrease of slip coefficient with increasing flow.
Detached Eddy Simulations

Detached eddy simulations of the cut-off trailing edge shape were performed for different blade angles ranging from 20 to 45 degrees, which includes the blade angle region, where the flow regimes changes. Again, the flow rate was set according to the blade angle for aligned leading edge flow. Fig. 6 depicts the time-averaged non-dimensional relative velocity and isosurfaces of the Q-criterion colored with the spanwise velocity for the blade angles 25, 35 and 45 degrees and the fine mesh. Like in case of the 2D-RANS simulations, two trailing edge flow regimes can be clearly distinguished. At a blade angle of 25 degrees, the flow remains attached, while at the higher blade angles the flow is separated after the obtuse corner of the trailing edge. The isosurfaces of the Q-criterion show the turbulent structures of the flow. While at 25 degrees the small vortical structures are shed from the sharp corner at the end of the blade,
there is massive separation and vortex shedding from the whole trailing edge at both higher blade angles. In case of the higher blade angles also a large-scale shedding can be observed through the sinusoidal wake structure.

The slip coefficients evaluated from DES are presented in Fig. 7 together with the values from the 2D-RANS simulations. The DES results of the fine mesh are close to the 2D-RANS results. The jump in flow coefficient, which indicates the beginning of the separated flow regime, also occurs between 25 and 30 degrees. The transition between flow regimes seems to be a bit smoother and at a blade angle of 35 degrees the fine DES predicts a slightly higher slip factor. In case of the coarse DES the flow regime change is delayed as at 30 degrees a low slip coefficient is predicted.

Finally, the turbulent kinetic energy (TKE) was evaluated from the fine DES. In Fig. 8 contours of the TKE are shown for the 2D-RANS simulations on the left and for the fine DES on the right. In case of the DES, the total TKE is shown, which is the sum of the modeled TKE from the underlying RANS turbulence model (SST) and the resolved TKE evaluated from the variance of the velocity field. Since the RANS model was active only in a small portion of the boundary layer, most of the TKE shown here is contributed by the resolved turbulence field. The comparison reveals some differences especially concerning the magnitude of the TKE. The DES shows higher levels of TKE within the separation region than predicted by the RANS simulations, while the locations of the TKE maxima are similar.

![Figure 7: CFD slip coefficients for cut-off trailing edge from 2D-RANS and DES](image)

**CONCLUSIONS**

2D RANS and detached eddy simulations of impellers with equiangular blades for a wide range of blade angles and with fixed blade number and diameter ratio were performed. Two different trailing edge shapes were considered. The effect of varying blade angle and trailing edge shape on the slip factor was studied. The following conclusions can be drawn from the obtained results:
(a) Both trailing edge shapes show very different flow behavior and therefore lead to a huge difference in slip factor.

(b) None of the slip factor models used for comparison can predict the total impeller slip properly over the whole range of blade angles.

(c) For impeller blades with a cut-off trailing edge two different flow regimes exist. At low blade angles up to 25 degrees the flow around the trailing edge remains attached. Between a blade angle of 25 and 30 degrees the flow regime changes and the flow becomes massively separated at the trailing edge, which results in a sudden increase in slip coefficient. To the knowledge of the authors this behavior is reported here for the first time.

(d) Simulations for a wide range of flow coefficients were performed, resulting in maps for the slip coefficient. This investigation revealed that the separation behavior is not affected significantly by the flow rate.

(e) The differences in slip coefficient and flow behavior for the two trailing edge shapes show that the shape of the trailing edge is extremely important for the impeller’s performance. Therefore successful slip factor modeling has to account for the trailing edge shape and the flow separation behavior.

The presented study shows the effect of the trailing edge shape on the slip factor for two-dimensional impeller blades at fixed diameter ratio, blade number and blade thickness. Further
studies have to investigate the effect of these parameters in order to incorporate the results in a slip factor model. Moreover, the presented results were obtained under the assumption of a quasi two-dimensional impeller and smooth blade surfaces. Three-dimensional effects could possibly hide or smear the transition between the two flow regimes observed for the cut-off trailing edge since both regimes could exist along the trailing edge at the same time depending on the local flow conditions. Moreover, blade surface roughness could possibly have a similar effect, because significant surface roughness will modify the boundary layer and therefore affect the separation behavior. Also the viscous part of the impeller slip will be increased by surface roughness. Therefore, three-dimensional effects including wall boundary layers on hub and shroud surfaces as well as surface roughness effects have to be taken into account for future works.

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