Similarity Numbers for the Scaling of Hydroelectric Generator Ventilation Flow

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***Abstract*—Recently, the design of cooling air circuits for hydro power generators has transitioned towards utilization of computational fluid dynamics (CFD) simulations, increasingly supplementing or even replacing the previously favored network calculations. To validate the calculations for individual generators, common practice involves direct measurements of the pressure drop and volume flow characteristics using scaled models of the corresponding generators. This paper delves into the analysis of generator model ventilation flow and derives a set of similarity numbers to dimensionlessly represent this flow problem. The set of similarity numbers specified herein aids in determining the key considerations when scaling a generator for a ventilation model test. The focus of these similarity numbers lies on pressure loss characteristics, with no exploration of heat transfer aspects due to their usual exclusion from ventilation model tests. Two of these similarity numbers, although not appearing in existing literature, are predicted to significantly impact generator cooling flow. To evaluate and discuss this impact, CFD simulations of a generic generator cooling flow section are conducted. These CFD simulations primarily illustrate the importance of the newly introduced similarity numbers while maintaining adherence to the state of the art.**

***Index Terms*—Rotating Machines, Similarity, Model Test, Hydro Generator Ventilation, CFD**

I. Introduction

This paper addresses the scaling of hydro generator ventilation flow within the context of pressure loss investigations that are conducted through model tests. It describes a top-down approach of nondimensionalization. The main contribution herein is the derivation of a unique set of similarity numbers based on the application's specific characteristic parameters. More precisely, these similarity numbers result from nondimensionalizing the volume forces of the fluid mechanics laws applied to generator cooling flow. The derivation of similarity numbers presented here is entirely analytical. Additionally, computational fluid dynamics (CFD) calculations of a simplified benchmark rotor segment are employed to illustrate and discuss the effects of the newly proposed similarity numbers.

A typical hydro power plant consists of two primary sections, the turbine and the hydro power generator (HPG). These large electrical machines with rotor diameters up to 15 meters convert the mechanical power into electrical power. Despite their high efficiency, HPGs necessitate cooling to maintain operational temperature limits. A closed air flow circuit across the entire machine commonly enables cooling of all heat-generating components. This airflow is generated either by external fans, rotor-attached fans, or radial ventilation of the rotor itself. The choice of cooling mechanism depends on the machine and operation parameters [1].

There are two categories of cooling air circuit design methods. The classical approach involves so-called network calculations, which model the flow system with thermal and flow networks. Significantly more expensive yet potentially more accurate is the direct application of CFD simulations. Given the crucial role of adequate cooling, both network calculations and CFD simulations must be validated through experimental studies. One common validation strategy for the ventilation design includes measurements on a down-scaled model of the generator. These model tests are advantageous over measurements on the original machine, since they can be conducted before the machine's construction and thereby potentially aid the design process. Furthermore, measurements inside an HPG pose challenges due to strong magnetic fields and demanding geometric accessibility for probes. The authors performed such industrial tests for many years. Laws of similarity are typically employed to ensure that model measurements are transferable to the original generator. The challenge for model design lies in the selection and definition of similarity numbers and the handling of partial similarity, implying not all similarity numbers of the model test flow match the original. Depending on similarity number requirements, different scaling approaches for the flow problem can be considered. Despite ongoing research over several decades,

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openly available literature regarding reliable guidelines for the selection of similarity numbers and scaling approaches for the generator cooling flow is sparse. This knowledge is primarily confined to company internal know-how.

Numerical-based investigations on the generator heat transfer problem are frequently researched [2], [3], [4]. Boglietti [5] offers a comprehensive and historical overview of numerical simulations for the generator flow problem. However, only in some instances scaled models are used to gather experimental data in addition to CFD results. Moradnia et al. [6] compare scaled model experiment results with a CFD simulation, but with limited success in terms of quantitative results. The experimental setup in [6] mirrors the boundaries in the present paper, specifically cold conditions without heat transfer. Jamshidi et al. [1] place higher emphasis on model scaling in their work, where they combine a scaled generator model test with CFD simulations. The research institute of Hydro-Québec (IREQ) has been a leading research group in the field of scaled generator models in the past decade, operating a 1:4 scaled model of a hydroelectric generator used for providing validation data for CFD calculations [7], [8]. This model is also used for measurements such as particle image velocimetry (PIV) [9]. Recent investigations regarding the experimental validation of generator flow CFD simulations are performed by Diebel et al. [10], although scaling is not taken into consideration in this context.

II. Fluid Mechanical Description of Generator Ventilation Flow

In this paper, two cylindrical coordinate systems are used: one each for stationary and rotating flow sections. The designations draw from the theory of turbomachinery. The origin of coordinates is situated at the rotor's center on the rotation axis, which is the z-axis. Fig. 1 shows the coordinate systems on a generic generator geometry. The stationary coordinate system is referred to as the inertial system. The directions of the unit vectors in this system are:

$$\begin{pmatrix} r \\ \phi \\ z \end{pmatrix} = \begin{pmatrix} \text{radial direction} \\ \text{angular tangential direction} \\ \text{axial direction} \end{pmatrix} \tag{1}$$

For the flow inside the rotor the inertial coordinate system is transformed into a rotating coordinate system with an angular velocity ω around the z axis. It is called relative system.

$$\vec{\omega} = \begin{pmatrix} 0 \\ 0 \\ \omega \end{pmatrix} \tag{2}$$

The generator problem assumes a steady rotation. Hence the time derivative of the rotational speed is: $\frac{\partial \omega}{\partial t} = 0$. The variables in relative coordinates are denoted with primes. The directions of the unit vectors in the relative coordinate system are analogous to (1):

$$\begin{pmatrix} r' \\ \phi' \\ z' = z \end{pmatrix} = \begin{pmatrix} \text{rotating radial direction} \\ \text{rotating angular tangential direction} \\ \text{axial direction} \end{pmatrix} \tag{3}$$

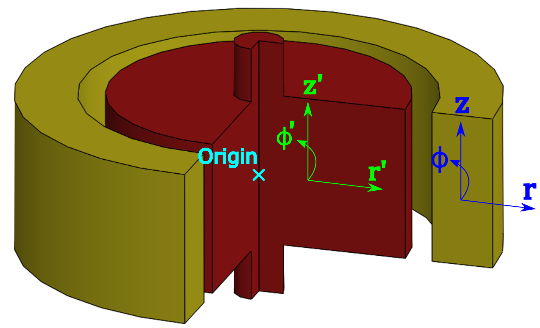

Fig. 1: Directions of coordinates in inertial system and relative system (primed variables).

The foundation for the description of fluid dynamics problems lies in the conservation equations [11], which postulate the conservation of mass flow, momentum and energy. The Navier-Stokes-Equation, a common model for momentum conservation in a continuum, is presented in its conservation form in (4) [12]. As heat transfer is not considered in the paper, and the continuity equation does not introduce additional similarity numbers, only (4) is pertinent to this study.

$$\rho \frac{\mathrm{D}\vec{v}}{\mathrm{D}t} = -\nabla p + \eta \nabla^2 \vec{v} + \left(\xi + \frac{\eta}{3}\right) \nabla \left(\nabla \cdot \vec{v}\right) + \vec{F} \tag{4}$$

The volume force $\vec{F}$ for the inertial coordinate system, corresponds to the gravitational acceleration $g \cdot \rho$ in negative z-direction. A transformation of the flow equations into the relative coordinate system must be taken into account to describe the flow within the rotating sections. The main distinction arising from this transformation is the alteration in the volume force vector, although the momentum conservation equation retains its original form:

$$\begin{aligned} \rho \frac{\mathrm{D}\vec{v}'}{\mathrm{D}t} &= \rho \left[\frac{\partial \vec{v}'}{\partial t} + \left(\vec{v}' \cdot \nabla\right) \vec{v}' \right] \\ &= -\nabla p + \eta \nabla^2 \vec{v}' + \left(\xi + \frac{\eta}{3}\right) \nabla \left(\nabla \cdot \vec{v}'\right) + \vec{F}' \end{aligned} \tag{5}$$

The volume force in the relative coordinate system with a constant angular velocity still includes the gravitational volume force $\vec{F}'_g$. Additionally, it introduces terms for the centrifugal volume force $\vec{F}'_{cf}$ in the radial direction and terms for the Coriolis volume force $\vec{F}'_{Co}$ in both the radial and circumferential directions.

$$\vec{F}' = \vec{F}'_g + \vec{F}'_{cf} + \vec{F}'_{Co} \tag{6}$$

$$\vec{F}'_g = \vec{F} = \begin{pmatrix} 0 \\ 0 \\ -g\,\rho \end{pmatrix} \tag{7}$$

$$\vec{F}'_{cf} = -\rho\,\vec{\omega} \times \left(\vec{\omega} \times \vec{x}'\right) = \begin{pmatrix} \rho\,\omega^2\,r' \\ 0 \\ 0 \end{pmatrix} \tag{8}$$

$$\vec{F}'_{Co} = -2\,\rho\,\vec{\omega} \times \vec{v}' = \begin{pmatrix} 2\,\rho\,\omega\,v_{\phi'} \\ -2\,\rho\,\omega\,v_{r'} \\ 0 \end{pmatrix} \tag{9}$$

III. Nondimensionalization of the Generator Ventilation Flow

The nondimensionalization of the conservation equations results in similarity numbers that describe the flow problem. Through this process, the fluid behavior is detached from its dimensional properties. Nondimensionalization is accomplished by replacing all dimensional variables with the ratio of a dimensionless equivalent and a dimensional characteristic variable, e.g. $\vec{v} = v_0 \vec{v}^*$ for velocity. This approach is widely accepted and will not be described in detail in this paper. The reader is referred to [13] for a more detailed explanation of the process of nondimensionalization. The components of the volume forces are expressed with the summation $\sum_i \left(F_{0,i}\,\vec{F}^{\,*}_i\right)$.

$$\rho v_0 f_0 \frac{\partial \vec{v}^*}{\partial t^*} + \rho \frac{v_0^2}{l_0} \left(\vec{v}^* \cdot \nabla^*\right) \vec{v}^* = -\frac{p_0 - p_\infty}{l_0} \nabla^* p^* + \eta \frac{v_0}{l_0^2} \nabla^{*2} \vec{v}^* + \left(\xi + \frac{\eta}{3}\right) \frac{v_0}{l_0^2} \nabla^* \left(\nabla^* \cdot \vec{v}^*\right) + \sum_i \left(F_{0,i}\,\vec{F}^{\,*}_i\right) \tag{10}$$

Beyond the already introduced characteristic velocity v_0, the characteristic variables in (10) include the characteristic frequency f_0, the characteristic length l_0 and the characteristic reference pressure difference $p_0 - p_\infty$, where p_∞ is a reference pressure, such as the ambient pressure. The volume force term of the generator model problem leads to similarity numbers that are not typically found in fluid mechanics literature.

This section entails the derivation and explanation of these similarity numbers. To yield all similarity numbers from (10), the equation is multiplied with the factor $l_0/\rho v_0^2$. The term corresponding to the volume forces acting on the fluid becomes:

$$\frac{l_0}{\rho v_0^2} \sum_i \left(F_{0,i}\,\vec{F}^{\,*}_i\right) = \frac{l_0}{\rho v_0^2} \Big(F_{g,0}\,\vec{F}^{\,*}_g + F_{cf,0}\,\vec{F}^{\,*}_{cf} + F_{Co,r',0}\,\vec{F}^{\,*}_{Co,r'} F_{Co,\phi',0}\,\vec{F}^{\,*}_{Co,\phi'}\Big) \tag{11}$$

The characteristic volume forces $F_{0,i}$ are provided through the definitions of the volume force components in (7) to (9):

$$F_{g,0} = -g\,\rho \tag{12}$$
$$F_{cf,0} = \rho\,\omega^2\,r' \tag{13}$$
$$F_{Co,r',0} = 2\,\rho\,\omega\,v_{\phi'} \tag{14}$$
$$F_{Co,\phi',0} = -2\,\rho\,\omega\,v_{r'} \tag{15}$$

Subsequently, the incorporation of the characteristic volume force terms into (11), will enable the derivation of the corresponding similarity numbers in the following subsections. Each characteristic length l_0 is measured in the direction along the corresponding coordinate of the force term component.

A. Centrifuge Number: Centrifugal Force

The component of the centrifugal volume force in the radial direction in (11) is:

$$\frac{F_{cf,0}\,l_0}{\rho\,v_0^2} \vec{F}^{\,*}_{cf} = \frac{\rho\,\omega^2\,r'\,l_0}{\rho\,v_0^2} \vec{F}^{\,*}_{cf} \tag{16}$$

The characteristic length in radial direction is the characteristic radius r_0, which significantly impacts centrifugal compression. The dimensionless factor of the radial component of the volume force becomes the sought similarity number. It is assigned the name Centrifuge number Cf.

$$\mathrm{Cf} = \frac{\omega^2\,r_0^2}{v_0^2} \tag{17}$$

The Centrifuge number equals the pressure rise of the rotor, normalized with the dynamic pressure of the flow. It is a special form to express the general kinematic similarity, where the flow field's directions and relative velocity across the entire flow region must be preserved to maintain similarity.

B. Radial Rossby Number: Coriolis Force in Radial Direction

The radial component of the Coriolis volume force in (11) is:

$$\frac{F_{Co,r',0}\,l_0}{\rho\,v_0^2} \vec{F}^{\,*}_{Co,r'} = \frac{2\,\rho\,\omega\,v_{\phi'}\,l_0}{\rho\,v_0^2} \vec{F}^{\,*}_{Co,r'} \tag{18}$$

For this term, a similarity number named radial Rossby number Ro_r is introduced. This naming is based on the atmospheric Rossby number, which describes the effect of the Coriolis force to inertia ratio in the atmosphere. The radial Rossby number is also caused by the Coriolis acceleration. The characteristic length l_0 is set to r_0, equivalently to the Centrifuge number. The radial and tangential components of the characteristic velocity v_0 are $v_{r',0}$ and $v_{\phi',0}$.

$$\mathrm{Ro}_r = \frac{2\,\omega\,v_{\phi',0}\,r_0}{v_0^2} \tag{19}$$

With a characteristic velocity defined in the direction of the flow and α as the angular deviation of the flow from the radial direction on a plane normal to the z-direction: $\sin(\alpha) = v_{\phi',0}/v_0$, Ro_r can be written as:

$$\mathrm{Ro}_r = \frac{2\,\omega\,\sin(\alpha)\,r_0}{v_0} \tag{20}$$

The definition of the radial Rossby number has an inverse character compared to the atmospheric Rossby number, where small values indicate a high influence of the Coriolis force. Here a low radial Rossby number means a low Coriolis force impact and a radial Rossby number of zero is equal to no Coriolis force effect. The reason for this difference in definition is that (20) remains a mathematically valid expression for stationary parts. Note that Yeranee et al. [14] uses similar Coriolis force terms when studying the similarity in turbine blade cooling investigations.

C. Tangential Rossby Number: Coriolis Force in Tangential Direction

Analogous to (18), the tangential force term is:

$$\frac{F_{Co,\phi',0}\,l_0}{\rho\,v_0^2} \vec{F}^{\,*}_{Co,\phi'} = \frac{-2\,\rho\,\omega\,v_{r'}\,l_0}{\rho\,v_0^2} \vec{F}^{\,*}_{Co,\phi'} \tag{21}$$

The reciprocal absolute value of the dimensionless factor is the definition of the tangential Rossby number Ro_ϕ. The naming is again inspired by the atmospheric counterpart, where a "second Rossby number" also exists [15]. The characteristic length is the tangential flow channel length $l_{\phi,0}$. The characteristic radial velocity is the radial component of the characteristic velocity $v_{r',0}$.

$$\mathrm{Ro}_\phi = \frac{2\,\omega\, v_{r',0}\, l_{\phi,0}}{v_0^2} \tag{22}$$

With a characteristic velocity defined as the radial velocity component $v_{r',0} = \cos(\alpha)\, v_0$, (22) yields:

$$\mathrm{Ro}_\phi = \frac{2\,\omega\, \cos(\alpha)\, l_{\phi,0}}{v_0} \tag{23}$$

The tangential Rossby number expresses the ratio of the inertia of the tangentially aligned fluid column in the rotating channel and the radial flow inertia. In the special case of a radially aligned flow, the tangential Rossby number becomes similar to the Rotation number known from turbomachinery, except that the characteristic length of the Rotation number is the hydraulic diameter. This definition seems less applicable, since the described flow behavior (see section IV-B) depends on the ratio of channel height and width [16]. The designation Rotation number is not adopted for this paper due to this different definition, the presence of the radial Rossby number as a second Coriolis force effect, and the incapability of the Rotation number to treat angled flows correctely.

D. Equation of Momentum Conservation including Similarity Numbers

By incoporating the definitions of the similarity numbers from (17), (20), (23), as well as the definitions for the Strouhal number Sr, the Euler number Eu, the Reynolds numbers Re and Re_b as well as the Froude number Fr, the equation of nondimensionalized momentum conservation (10) becomes:

$$\begin{aligned}&\mathrm{Sr}\frac{\partial \vec{v}^*}{\partial t^*} + (\vec{v}^* \cdot \nabla^*)\,\vec{v}^* = -\mathrm{Eu}\nabla^* p^* + \frac{1}{\mathrm{Re}}\nabla^{*2}\vec{v}^*\\ &+ \left(\frac{1}{3\,\mathrm{Re}} + \frac{1}{\mathrm{Re}_b}\right)\nabla^*\left(\nabla^* \cdot \vec{v}^*\right) + \mathrm{Cf}\vec{F}^{\,*}_{cf}\\ &+ \mathrm{Ro}_r\,\vec{F}^{\,*}_{Co,r'} + \mathrm{Ro}_\phi\,\vec{F}^{\,*}_{Co,\phi'} - \frac{1}{\mathrm{Fr}}\,\vec{F}^{\,*}_{g}\end{aligned} \tag{24}$$

While these similarity numbers have a general applicability to all rotating flows, their characteristic parameters have been specifically tailored to suit the ventilation problem in generators, much in the same way as similarity numbers and dimensionless metrics have been chosen for various other problems, for instance, turbine blade internal cooling problems as discussed in [17], [18], and [14].

The distinctions between the newly introduced similarity numbers and the well-established ones need to be further elaborated. The first one is the Rotation number, denoted as Rot. It pertains to a rotating 2-D channel and is expressed according to [19] by:

$$\mathrm{Rot} = \frac{2\,\omega\, l_{\phi,0}}{v_0} \tag{25}$$

For the specific scenario of a radially oriented pole gap flow, Rot is equivalent to the tangential Rossby number. However, after (20) and (23) in case of an angled flow, both the tangential Rossby number and radial Rossby number deviate from the Rotation number. This discrepancy can arise in the rotor hub flow as well as between the poles. Consequently, the Coriolis force will not be accurately represented by (25). The distinct characteristic length variables lead to the fact that the Pythagorean Addition of Ro_r and Ro_ϕ is not equal to Rot.

Furthermore, another well-established dimensionless parameter used to analyze rotating flows is the buoyancy parameter Bo [14]. It describes the combination of rotational force and temperature difference driven buoyancy. The corresponding expression, as delineated by [18], is:

$$\mathrm{Bo} = \left(\frac{\Delta\,\rho}{\rho}\right)_x \mathrm{Rot}^2\,\frac{r_x}{d_h} \tag{26}$$

Even in the special case without static buoyancy, the centrifuge number differs from the buoyancy parameter, thus isolating the contribution of centrifugal force instead of combining the overall rotational effects. Since these distinct contributions lead to different phenomena in the generator flow (see Section IV), the separate consideration proves more advantageous for this heat transfer neglecting scaling application.

The main focus of this paper is on the new similarity numbers Cf and Ro_ϕ from (24) as well as their effect on the generator flow problem. Compared to the other two, the radial Rossby Number's importance is relatively low for the mainly radially oriented flow in the generator model.

IV. Effect of the new Similarity Numbers

To investigate the influence of Cf and Ro_ϕ, CFD studies were conducted using the commercial solver STAR-CCM+. The analyzed model is a section of a generic geometry of a generator model, see Fig. 2. The radius of the rotor is $r_{rotor} = 0.7\,\mathrm{m}$. The rotor section includes two radial rim channels and one pole gap. Two axial stages are simulated to verify the periodic boundaries at the top and the bottom wall of the pole gap. The poles diverge with an angle of 7.5° downstream of the pole gap inlet, where the reference width is $b_{gap} = 0.15\mathrm{m}$. This gap is used to calculate the characteristic length of the Reynolds number with the hydraulic diameter $l_0 = d_{hydr} = 2\,b_{gap}$ and the tangential Rossby number $l_{\phi,0} = b_{gap}$. This dimension is chosen for the Reynolds number due to the primary focus being on the gap section between the poles. The characteristic velocity v_0 is defined with a realistic reference volume flow for two rim channels of $\dot{V} = 0.3\,\mathrm{m}^3\,\mathrm{s}^{-1}$ and the cross section area at the pole gap inlet. The 3-D-model of the fluid volume for one geometry is shown in Fig. 3. The geometric parameters represent the model of a typical generator design. The bulk Reynolds number Re_b is not considered separately, since the fluid properties as well as the thermodynamic state are constant for all simulations, meaning a constant Re implies a constant Re_b.

The air is treated as an ideal gas for simulation purposes. The polyhedral mesh of the finite-volume-discretization contains boundary prism layers that resolve the flow with $y^+ \leq 1$. The mesh size reduces for the pole gap, as the investigations

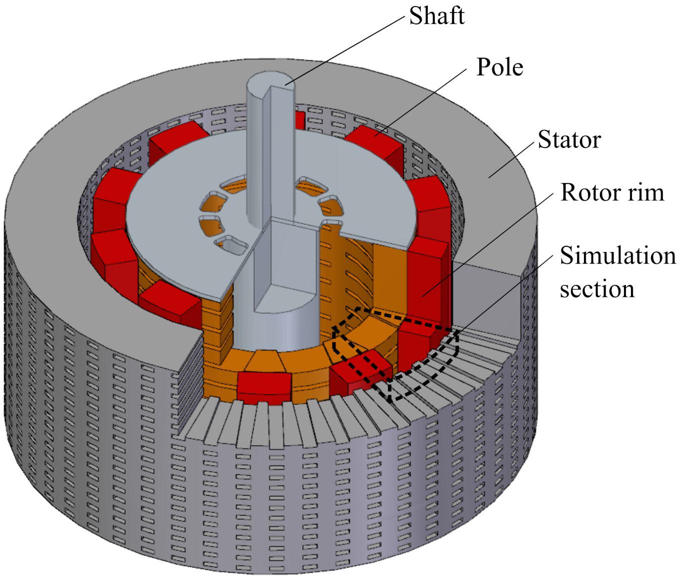

Fig. 2: Sketch of the generic generator model with the simulation section of the rotor.

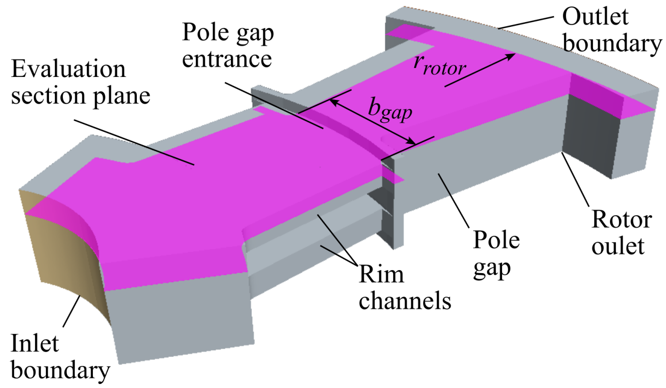

Fig. 3: One simulation model geometry with the highlighted evaluation section plane (purple).

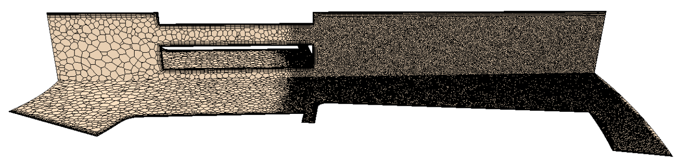

Fig. 4: Mesh on two section planes of the simulation model in Fig. 3.

focus on this section. The transport equations are solved using a steady Reynolds-Averaged-Navier-Stokes (RANS) approach and the k-ω shear-stress transport (SST) turbulence model. Steady models are chosen because the investigated similarity numbers also exist in a steady form of (24). The k-ω SST model is a two-equation model that combines the advantages of the k-ϵ-model and the k-ω-model by effectively switching the turbulence model inside the boundary layer [20]. An exemplary mesh is shown in Fig. 4. A comprehensive mesh convergence study over the mesh base size and boundary layer parameters was used to determine the optimal cell size. The pressure difference from inlet to outlet, the residuals, and the averaged volume integrals of all three velocity components over the whole simulation continuum were used to assess convergence. Note that the importance was not obtaining realistic data for the generator but reasonable and stable conditions to study the impact of the similarity numbers. Therefore, a radial mass flow inlet is placed upstream of the rim channel. The outlet is modeled as a pressure outlet within the rotating reference frame. Thus, the deceleration influence of the stator is not modeled. The reason for not implementing a rotor-stator interface is to isolate the effects caused by the characteristic dimensions of the rotor section itself.

The variable characteristic parameters of the simulation are the characteristic velocity v_0, which is the mean velocity in the pole gap inlet, the pole gap inlet width $l_0/2$, and the rotational speed ω. The system of characteristic variables is therefore fully constraint by the three similarity numbers.

A. Effect of the Centrifuge Number

In order to understand the influence of the Centrifuge number on the rotor channel flow, a series of three simulations with increasing Centrifuge numbers was conducted, while keeping Reynolds and tangential Rossby numbers constant. To vary the Centrifuge number, the channel width, characteristic velocity and the rotational speed ω are adjusted.

With the three definitions of Re , Cf in (17) and Ro_ϕ in (23), the system is fully constrained. To illustrate the parameter selection of a simulation series, this process is explained on the example of one variation of the Centrifuge number:

Beginning with the reference case, where b_{gap}, l_0 and v_0 are predetermined as specified above, the Centrifuge number is incremented by increasing the ratio of rotational speed to characteristic velocity. To preserve the Reynolds number, the rim channel gap width is reduced by the same factor by which the characteristic velocity is amplified. The product of the rotational speed's enlarging factor and the pole gap's decreasing factor equals the enlarging factor of the characteristic velocity, which ensures the maintenance of the tangential Rossby number. The corresponding values are detailed in Tab. I.

The ventilation characteristics are determined by the pressure rise and the pressure drop of the generator model. Fig. 5 depicts the increase in total pressure with rising Centrifuge number while Reynolds number and tangential Rossby number are kept constant. Consequently, the rotor's pressure rise corresponds to the total pressure and amplifies with the Centrifuge number. Hence, the Centrifuge number is crucial for a successful generator ventilation model design. Moreover, the compliance with the Centrifuge number to scale the pressure rise of the rotor is advantageous to simply maintaining the pressure rise. Maintaining the pressure rise of the rotor would be the trivial approach for a generator ventilation model design, in the case of only using well-known similarity numbers. The transition from pressure rise to

TABLE I: Characteristic flow dimensions of Centrifuge number study.

b_{gap}[m]	v_0[m s^{-1}]	ω[1/s]	Cf[−]	Ro_ϕ[−]	Re[−]
0.15	16.61	20	0.710	0.722	3.49E5
0.125	19.93	28.8	1.023	0.722	3.49E5
0.1	24.92	45	1.598	0.722	3.49E5

pressure rise coefficients provides this advantage. As a result, the Reynolds number can be adjusted by altering the flow velocity, offering greater flexibility in the design process.

The streamlines from all three simulations reveal a flow detachment behaviour inside the pole gap. However, the flow separation from the pole's suction side surpasses the flow separation from the pressure side due to a flow deflection. In the case of Cf $= 1.023$, there is a noticable variation in this deflection, even with the Reynolds and tangential Rossby numbers remaining constant.

B. Effect of the Tangential Rossby Number

The effect of the tangential Rossby number on the flow is examined through three simulations with increasing tangential Rossby numbers. Similarly to the preceding section, both the Reynolds number and the Centrifuge number are kept constant at their respective reference values. The tangential Rossby number is varied by adjusting the channel width, characteristic velocity, and the rotational speed in a manner akin to the Centrifuge number variation. Starting from the reference value, one configuration decreases the tangential Rossby number while another increases it. A higher tangential Rossby number corresponds to an elevated ratio of the Coriolis volume force to the flow's inertia.

In Tab. II and Fig. 6 the three configurations are compared. Unlike the previous section, Fig. 6 depicts the velocity magnitude in the rotating frame. The velocity is more relevant for the flow inertia than the total pressure and thus better suitable to study the Coriolis volume force. The streamlines in Fig. 6 highlight the effects covered by the tangential Rossby number: flow's attachment to the pressure side and separation from the suction side when the flow channel broadens in the pole gap. This behavior suggests inadequate cooling of one pole flank by the airflow. These pole flanks house the pole windings, one of the generator's heat sources due to electrical losses. Incomplete cooling of half the pole windings would lead to unacceptable high temperatures especially of the adjacent insulation material and therefore would result in a complete cooling system failure. Hence, modeling this flow phenomenon is highly relevant for the generator model. As this effect prompts a transition from one behavior to another, regime boundaries could be estimated for tangential Rossby numbers with or without the separation effect.

V. Conclusions

The dimensional analysis of the generator cooling flow ventilation problem results in a set of similarity numbers that are, or could potentially be, relevant for correct scaling. Notably, the two new similarity numbers Cf and Ro_ϕ, may impact the validity of the scaled model as significantly as established similarity numbers. However, these are not represented in their current form in the literature. By considering these numbers, the process of scaling a generator model for cooling air flow investigations can be improved. The conducted CFD calculations demonstrate the suitability of both the Centrifuge number and the tangential Rossby number in describing the

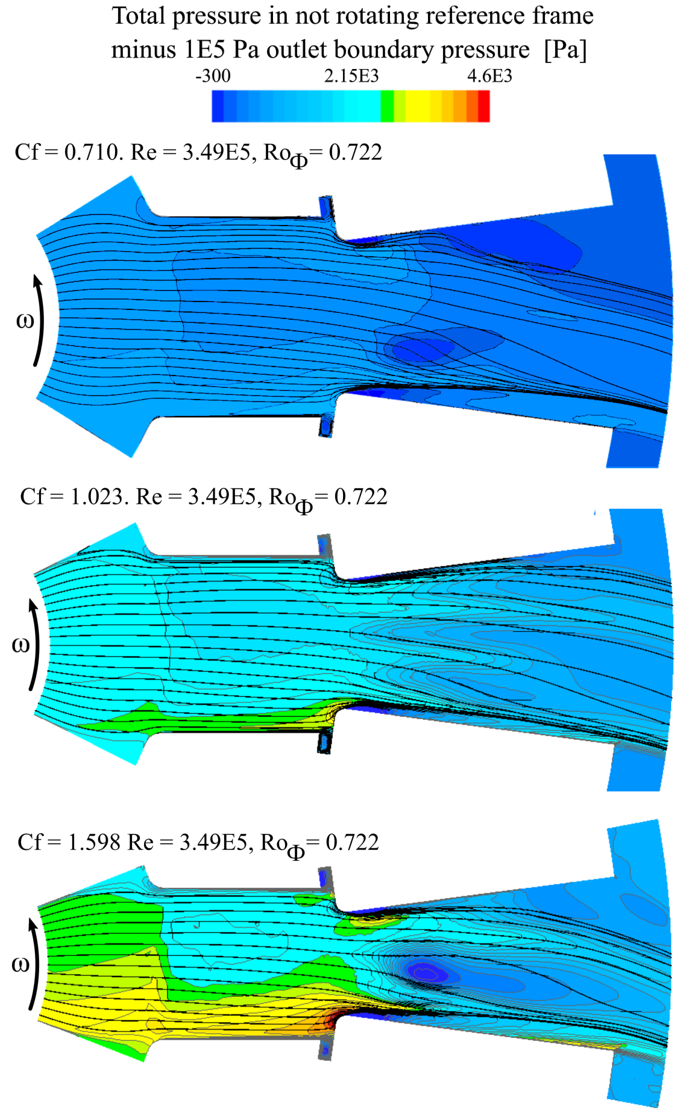

Fig. 5: Total pressure nonrotating reference frame and streamlines in rotating reference frame of generic rotor flow for variable Centrifuge number.

TABLE II: Characteristic flow dimensions of tangential Rossby number study.

b_{gap}[m]	v_0[m s^{-1}]	ω[1/s]	Cf[−]	Ro_ϕ[−]	Re[−]
0.1	24.92	30	0.710	0.482	3.49E5
0.15	16.60	20	0.710	0.722	3.49E5
0.2	12.46	15	0.710	0.963	3.49E5

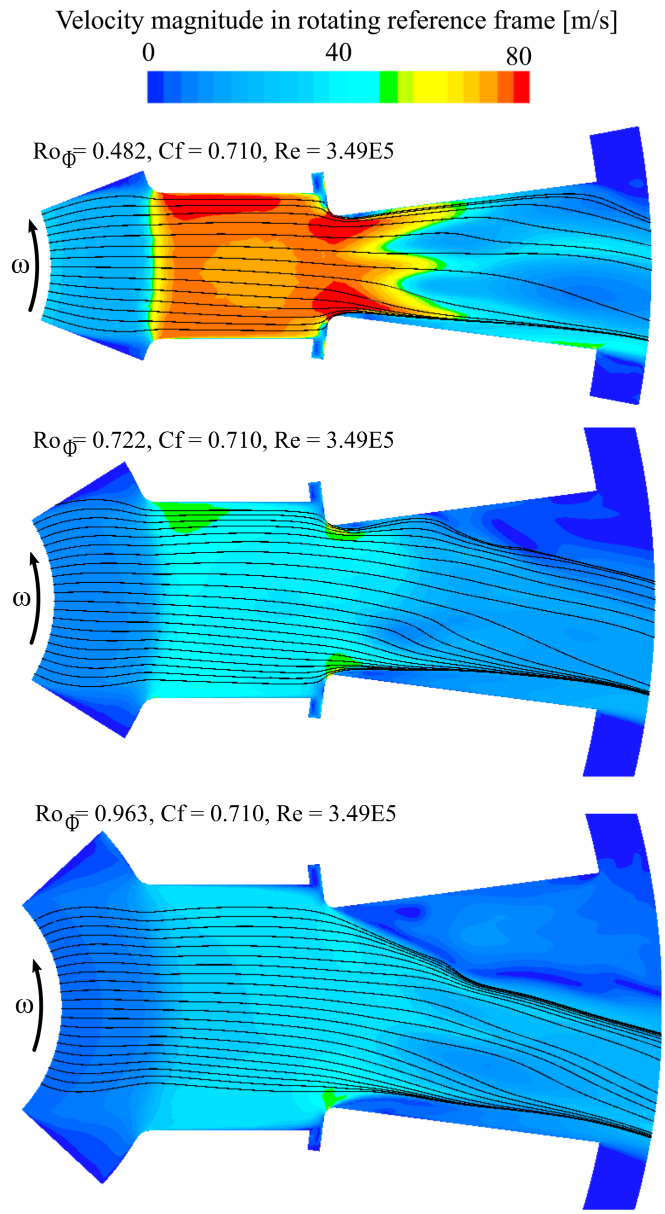

Fig. 6: Velocity magnitude and streamlines in rotating reference frame of generic rotor flow for variable tangential Rossby number.

effects caused by the rotational acceleration of the rotor flow. In line with this simulation example, the system of characteristic variables is fully constrained for the combination of the well-established Reynolds number and the new similarity numbers Cf and Ro_ϕ. This provides an approach for scaling a generator ventilation model. However, in the event of further requirements for the model, e.g. manufacturing precision or model power drive limitations, additional trade-offs may be necessary to satisfy the observed similarity numbers in a fully constrained system.

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VI. Biographies

David Noelle has been a researcher in the field of experimental aerodynamics at the Chair of Flight Dynamics and Control at Technische Universität Dresden, Germany, since 2021. He received his Diploma in Engineering from Technische Universität Dresden in 2017. In 2018, he joined the Experimental Aerodynamics team at Technische Universität Dresden, where he has been involved in designing scaled aerodynamic ventilation models of hydrogenerators for industrial projects.

Dr. Veit Hildebrand has been a research associate and head of the wind tunnel facility at Technische Universität Dresden, Germany, since 2002. He received his Ph.D. from Technische Universität Dresden in 1993. Prior to that, he was a scientific researcher in the fields of thermofluid dynamics, power plant technology, and turbo machinery at Technische Universität Dresden.

Bastian Diebel received the master's degree (M.Sc.) in aerospace engineering at University of Stuttgart, Stuttgart, Germany. In 2016 he joined Voith Hydro Holding GmbH & Co. KG in Heidenheim, Germany. In the department "Fluid Mechanics" he is responsible for research and development on cooling and ventilation of hydro generators. This includes the development and validation of 1D and 3D thermal design tools.

Dr. Thilo Dauch completed his master's degree (M.Sc.) in mechanical engineering with a major in turbomachinery and jet engines at RWTH Aachen University, Aachen, Germany. Afterwards he worked as research associate and PhD candidate at the Institute of Thermal Turbomachinery (ITS) at the Karlsruhe Institute of Technology (KIT) in Karlsruhe, Germany. In 2020 he joined Voith as development engineer in the field of hydro generator ventilation. In 2024 he became "Head of Fluid Mechanics" and is now responsible for R&D activities related to fluid mechanics in hydro generators and turbines.

Prof. Harald Pfifer is the Chair in Flight Dynamics and Control at the Technische Universitaet at Dresden, Germany. He received his Ph.D. from Technical University Munich, Germany, in 2013. Before joining Technische Universitaet at Dresden in 2021, he was an assistant professor at University of Nottingham, and a post-doctoral associate at University of Minnesota. His main research interests include aeroservoelastic control, uncertain dynamical systems, and robust and linear parameter-varying control.