Design guidelines for axial turbines operating with non-ideal compressible flows

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1 Introduction

Design guidelines based on scaling analysis and synthesized in the form of efficiency maps as function of work and flow coefficient are essential to obtain a preliminary estimate of the size, the shape of the velocity triangles, and the fluid-dynamic performance of turbomachinery stages [1]. While maps for the preliminary design of steam and gas turbines are readily available and their accuracy has been improved over time [2], design criteria for unconventional turbomachinery, i.e., turbomachines operating with flows departing from ideal gas state, are yet to be established. Examples of unconventional turbomachines are turbines for organic Rankine cycle (ORC) power systems [3], turbines for the oil and gas industry operating with heavy fluid molecules, supercritical CO₂ (sCO₂) compressors [4], and high-speed compressors for refrigeration and air-conditioning systems.

In these machines, the occurrence of strong thermo-physical fluid property gradients [5], compressibility, and non-ideal flow effects [6] along the expansion or the compression process arguably alters the share of dissipation induced by the various loss mechanisms, i.e., viscous friction, shocks, and mixing, as compared to standard gas turbines. This may eventually result in optimal designs significantly differing from those that would be attained through the application of existing best practices [1].

Attempts to devise design charts for unconventional turbines can be found in Refs. [7–9]. However, the existing body of work solely relies on semi-empirical loss correlations to predict the stage performance and lack of a thorough validation of results by means of high-fidelity models or experimental data. The adoption of such loss models is debatable in the context of non-ideal compressible fluid dynamics (NICFD), where the flow characteristics in the blade passages can considerably depart from those considered when deriving and calibrating the existing semi-empirical loss correlations. Recent studies [10] pointed out that an adequate physical interpretation of loss mechanisms in ORC turbines and an accurate loss accounting is arguably possible only by resorting to computational fluid-dynamic (CFD) calculations. These, however, are machine-specific and their use would be excessively demanding for comprehensive investigations targeting the conception of design guidelines.

This work aims at addressing this knowledge gap by means of a theoretical and numerical framework suited to unconventional turbomachinery. The framework encompasses a reduced-order turbine stage model based on classical similarity parameters, e.g., the duty coefficients, and on dimensionless quantities suited to unconventional turbomachinery, namely, the volumetric flow ratio [7] and the isentropic pressure–volume exponent [11], and a validation procedure based on high-fidelity CFD. The fluid-dynamic losses are calculated through a loss model [12,13] derived from first-principles and extended to arbitrary thermo-physical fluid models [14]. The model is used to compute efficiency maps for axial turbines operating with siloxane MM and CO₂ at different volumetric flow ratios in the ideal and non-ideal thermodynamic regime. The selection of these working fluids is dictated by their increasing molecular complexity as compared to air, see Table 1, and by their relevance for industrial applications, i.e., organic Rankine cycle engines [3] and sCO₂ power systems [15]. Herein, the molecular complexity of a fluid is evaluated as the number of active degrees-of-freedom at its critical temperature, as suggested in Ref. [6].

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Μ/Μₘₐₓ</th>
<th>N</th>
<th>γₛₑ</th>
<th>γₚₑ,ₘᵢₙ</th>
</tr>
</thead>
<tbody>
<tr>
<td>H₂</td>
<td>0.07</td>
<td>3.0</td>
<td>1.67</td>
<td>1.63</td>
</tr>
<tr>
<td>Air</td>
<td>1.00</td>
<td>5.0</td>
<td>1.40</td>
<td>1.44</td>
</tr>
<tr>
<td>CO₂</td>
<td>1.52</td>
<td>7.0</td>
<td>1.29</td>
<td>0.86</td>
</tr>
<tr>
<td>R134a</td>
<td>3.52</td>
<td>21.6</td>
<td>1.09</td>
<td>0.81</td>
</tr>
<tr>
<td>MM</td>
<td>5.61</td>
<td>77.4</td>
<td>1.03</td>
<td>0.39</td>
</tr>
<tr>
<td>D₆</td>
<td>15.36</td>
<td>211.7</td>
<td>1.01</td>
<td>0.23</td>
</tr>
</tbody>
</table>

The heat capacity ratio is computed in dilute gas state; the minimum value assumed by γₚₑ is evaluated in the thermodynamic region defined by s > 1.01x and by the thermal stability limits of each molecule.
\[ N = \frac{2M_{C, \text{ad}}(T_c)}{R} \] (1)

The results are eventually validated by means of three-dimensional (3D) RANS and URANS computations on stage geometries reconstructed to ensure full similarity with the correspondent dimensionless parameters used in the reduced-order model. The specific objectives of the work are (i) to draw design guidelines and fluid-dynamic design maps valid for unconventional axial turbines, (ii) to investigate potential design limits, (iii) to gain insight of the relative share of the various loss mechanisms in such turbine stages, and (iv) to provide physical understanding on the impact of fluid molecule and associated non-ideal flow effects on the efficiency of axial turbine stages. The structure of the paper is as follows. The technical approach is presented first. Next, the results are analyzed and discussed. Finally, concluding remarks summarize the lessons learnt and give a perspective on future work.

2 Methodology

2.1 Scaling Analysis for Unconventional Axial Turbines. In classical similarity analysis, the efficiency of an axial turbomachinery stage is expressed as

\[ \eta = f(\lambda, \phi, \chi, \text{Re}, \text{Ma}, \sigma) \] (2)

where the vector \( \sigma \) represents the stage dimensionless geometrical characteristics (e.g., blade solidity, aspect ratio, etc.) and the fluid properties are usually assumed to be modeled with the perfect gas law. Equation (2) hereinafter will be referred to as classical similarity equation (CSE). In the typical design process of gas turbine stages, the cascade Mach numbers are evaluated once the velocity triangles and stage velocities are computed [16]. The design is then iteratively adjusted to attain stage layouts, which can comply with admissible mechanical and operational constraints. The same procedure can be adopted to design unconventional turbine stages, but at the expense of overlooking the characteristic Mach numbers. As an illustrative example, consider a turbine stage operating with a complex fluid molecule. These stages are characterized by low values of specific work, but may exhibit volumetric flow ratios largely exceeding those of gas turbine stages. This is essentially due to the low heat capacity ratio and high molecular weight of complex working fluids. The normalized isentropic specific work of a turbine stage, which can be written as

\[ \psi_v = \frac{\Delta h_{\text{in}}}{R_{\text{ref}}^* \cdot T_{\text{ref}}^*} = \frac{R^*}{R_{\text{ref}}^*} \frac{T_{\text{ref}}^*}{T^*} \left( 1 - \frac{a_v^*}{\gamma} \right) \] (3)

where \( R_{\text{ref}}^* = 287.06 \) is depicted in Fig. 1. The trends show that the normalized work reduces while increasing the fluid molecular complexity and is mildly affected by a change in \( \gamma \).

In the dilute gas state, whereby \( \gamma \approx \gamma \) for highly complex fluid molecules. In other words, stage compressibility effects, ascribed here to variations of the stage volumetric flow ratio, can be drastically enhanced—or mitigated—by slightly varying its prescribed specific work. Such effect is not only related to the type of fluid molecule, but also to the thermodynamic regime of the fluid. This influence can be conveniently predicted through the evaluation of the isentropic pressure–volume exponent, i.e., the thermodynamic property allowing one to cast the isentropic pressure–volume relation in the general form \( P_v = K \), as documented in Ref. [11].

\[ \gamma_{P_v} = \frac{-v \frac{\partial P}{\partial v}}{P} \mid_T = \frac{-v c_v \frac{\partial P}{\partial v}}{P c_v} \mid_T \] (4)

This state variable can be related to the fundamental derivative of gas dynamics [17] as follows:

\[ \Gamma = \frac{1}{2} \left( \gamma_{P_v} + 1 - \frac{v}{\gamma_{P_v}} \frac{\partial \rho_{P_v}}{\partial v} \right) \] (5)

and its value asymptotically approaches the specific heat capacity ratio of the fluid in the dilute gas state

\[ \frac{P_v}{R} = \frac{\partial P}{\partial v} \mid_T = \frac{RT}{\nu^2} \] (6)

In turn, an explicit form of Eq. (3) valid for either ideal and non-ideal flow conditions can be obtained by substituting the heat capacity ratio with the isentropic pressure–volume exponent, under the assumption of constant \( \gamma_{P_v} \). The impact of NICFD on the normalized work is displayed in Fig. 1 for siloxane MM in two non-ideal conditions, characterized by \( \gamma_{P_v} > \gamma \) and \( \gamma_{P_v} < \gamma \). It can be seen that non-ideal effects lead in both cases to a variation of the normalized isentropic work, ultimately affecting the stage Mach numbers and performance. Since \( \gamma_{P_v} \) is a function of the local thermodynamic state, its average value over the expansion process can be therefore deemed a convenient parameter to quantify the impact of NICFD effects in turbomachinery. This concept is further illustrated in Fig. 2 by means of the application of a quasi-1D framework to compute isentropic expansions of siloxane MM at \( x = 4 \) in three different thermodynamic conditions, identified by the values of the reduced inlet temperature \( T_i = T_{\text{in}}/T_c \) and pressure \( P_i = P_{\text{in}}/P_c \), as reported in Table 2.

In the dilute gas state, whereby \( \gamma_{P_v} \approx \gamma \), the evolution of flow quantities can be accurately predicted with the perfect gas law. Conversely, in the proximity of the critical point, \( \gamma_{P_v} \) can be either higher or lower than \( \gamma \), leading to non-ideal fluid-dynamic effects, which entails significant variations of the resulting Mach number. In particular, for the same \( x \), if \( \gamma_{P_v} > \gamma \), the average Mach number along the expansion largely exceeds the value obtained in ideal gas conditions, and vice versa.

Based on these considerations, it can be argued that the best choice of the key conceptual stage design parameters may depend on the considered fluid molecule and the associated thermodynamic flow conditions. In order to investigate whether this is the case, it is more convenient to turn the CSE into a form, which explicitly includes the volumetric flow ratio \( a \) and the isentropic pressure–volume exponent \( \gamma_{P_v} \). The volumetric flow ratio \( a = \beta \gamma \) is a more informative scaling parameter than the expansion ratio \( \beta = P_{\text{in}}/P_3 \), as it allows one to account for compressibility effects while mitigating the influence of the considered working fluid.
when operating in dilute gas state. In the light of the above, the generalized form of the CSE can be rewritten as

$$g = f(K_{is}, \phi, \chi, \alpha, \gamma_{pe}, Re, \sigma)$$  (7)

In the following, Eq. (7) will be referred to as non-ideal similarity equation (NISE). Note that the loading coefficient and the degree of reaction are now normalized with the total-to-static enthalpy drop of the stage to be consistent with the definition of $a$. The calculation of axial turbine stage efficiency by means of the NISE is carried out according to the method described in the following.

### 2.2 Stage Layout

The representative turbine stage analyzed in this work is depicted in Fig. 3. The three-dimensional blade geometry is assumed to be constituted by a series of two-dimensional airfoils, characterized by negligible thickness and parabolic camber line, radially stacked according to free-vortex design. The blade spacing and stagger are given in terms of solidity $\sigma = c/s$ and axial solidity $\sigma_{ax} = c_{ax}/s$, while the annular geometry is characterized by the blade aspect ratio $AR = (H_{in} + H_{out})/2c_{ax}$ and hub-to-tip radius ratio. The gap between rotor tip and turbine casing is expressed in terms of percentage of blade height $g/(H_{in} + H_{out})$, whereas the stator has no clearance. All turbine stages are designed at constant mean radius and constant meridional speed. The velocity triangles are univocally determined by the choice of the duty coefficients $K_{is}, \phi, \chi$, the midspan radius variation in the rotor and the stator inlet flow angle, see Eq. (8). The latter is set according to repeated stage assumption, i.e., $\alpha_0 = \alpha_3$

$$\alpha = 2 \left( \phi_2 \tan \alpha_2 - \frac{R_{m3}}{R_{m2}} \phi_3 \tan \alpha_3 - \frac{R_{m3}}{R_{m2}} \right)^2$$

$$K_{is} = \lambda + \phi_3^2 + \left( \phi_3 \phi_2 \phi_3 - \frac{R_{m3}}{R_{m2}} \right)^2$$

$$\chi' = \frac{1}{K_{is}} \left( 1 - \phi_2^2 \right)^2 + \phi_3^2 \left( 1 + \tan^2 \alpha_2 \right) +$$

$$- \phi_2^2 \phi_3 \left( 1 + \tan^2 \alpha_2 \right)$$

$$\beta_2 = \arctan \left( \frac{\tan \alpha_2 - 1}{\phi_2} \right)$$

$$\beta_3 = \arctan \left( \frac{\tan \alpha_3 + \frac{R_{m3}}{R_{m2}} \phi_3}{1} \right)$$  (8)

### 2.3 Numerical Framework for Axial Turbine Stage Design

The numerical framework consists of a reduced-order turbine stage model schematically described in Fig. 4.
The streamwise flow distribution within the blade passages is modeled assuming the simplified blade surface velocity profile depicted in Fig. 5 [13] and the spanwise evolution of flow quantities is obtained imposing free-vortex design. The two parameters $D_V$ and $k$ featuring the velocity profile on the pressure and suction side of each blade section are computed by resorting to the circulation theorem and the tangential momentum balance, respectively. The latter is solved iteratively to account for compressibility effects and non-linearity introduced by the use of arbitrary thermodynamic models [14]

\[ \int V ds = (V_{ss} - V_{ps}) c_s = \Delta V_s s \]  

With the purpose of providing quantitative assessment of the influence of each parameter of Eq. (7), a set of 900 stage designs is generated at varying flow and loading coefficients and the results shown in the form of efficiency maps. More specifically, upon specification of the volumetric flow ratio, reduced inlet conditions and an initial guess of the dimension of the blade channel relative to the mean radius, the model is applied to perform a stage design for each couple of $\phi_2$ and $K_{ss}$, while optimizing $\chi$. The effect of Reynolds number is disregarded in this study, since the turbine stages are assumed to operate in fully turbulent regime. The geometrical characteristics of the stage are either computed

\[ m \Delta V_i = (P_{ps} - P_{ss}) c_{ax} H \]  

\[ \frac{H_i}{R_m} = \frac{\eta_i}{\rho_{ss} V_{ax, i}} \]

\[ \Delta H_i \frac{H_i}{R_m} \frac{H_i}{R_m} < \text{toll} \]  

Yes

No

\[ \eta_i \]  

\[ D_{mid} = \text{const.} \]

\[ V_{ax} = \text{const.} \]

\[ a_0 = a_3 \]
3 Loss Modeling

3.1 Boundary Layer Loss. The entropy production rate due to viscous dissipation in a two-dimensional boundary layer can be estimated as [12]

$$\dot{S} = \int C_d \frac{\rho V^3}{T_e} d\alpha$$

(11)

where $C_d$ is the dimensionless dissipation coefficient

$$C_d = \frac{T_e}{\rho_0 V^3} \int_0^\infty \rho V_s(s-s_c)dy$$

(12)

Once the velocity distribution on the blade pressure and suction side is known, the local thermodynamic state at the edge of the boundary layer can be computed by resorting to the conservation of energy and by assuming that the core flow is isentropic within the blade passage. The use of the dissipation coefficient is advantageous in presence of turbulent boundary layers, because, differently from the friction factor, its value is weakly dependent on the state of the boundary layer. Denton [12] suggests that for conventional turbomachinery applications, where the average $Re$ is of the order of 1000 and the shape factor is in the range $1.2 < H < 2$, a reasonable approximation is to take $C_d = 0.002$. The same value is adopted in this work. Note, however, that the research documented in Ref. [18] revealed that the value of the dissipation coefficient is function of the flow regime for fluids made by simple molecules, ultimately affecting the calculated boundary layer loss.

3.2 Mixing Loss. The overall entropy production due to mixing downstream of a cascade can be estimated with a control volume analysis, applied between the throat section and a far downstream boundary, where the flow is assumed to be uniform. The conservation laws in the control volume read

$$\dot{m} = \rho_0 V_a (a - \delta^*) = \rho_{mix} V_{cos} \cos \varphi_{mix}$$

$$\dot{m} V_a - \rho_0 V_a^2 \theta + P_{at} + P_{tg} \cos \varphi_a = \dot{m} V_{mix} \cos \delta + P_{mix} \cos \varphi_a$$

$$h_a + \frac{V_a^2}{2} = h_{mix} + \frac{V_{mix}^2}{2}$$

(13)

If the flow at the outlet section of the cascade is supersonic, the average flow conditions at the choked throat section can be computed resorting only to the conservation of energy; otherwise, the tangential momentum balance should be included in Eq. (13) to provide closure. As this work focuses on transonic turbine stages, the tangential momentum balance is disregarded from the system of equations. The base pressure $P_b$ can be roughly correlated with the total-to-static expansion ratio of the cascade, see Ref. [19]. Due to their minor influence on the final loss estimation, in this study the boundary layer parameters are set to reference values measured in transonic cascades, i.e., $\theta/\tau = 0.075$, $\delta^*/\tau = 2$, as suggested in Ref. [20]. With this information, Eq. (13) can be solved to obtain the entropy rise due to mixing $\Delta s_{mix} - s_a$, and the deviation angle $\delta$.

3.3 Shock Loss. The entropy generation across an oblique shock can be estimated with the Rankine–Hugoniot relations for two-dimensional flow [21], yielding

$$h_B - h_A = \frac{1}{2} (P_B - P_A) (v_A + v_B)$$

$$\frac{P_B - P_A}{v_B - v_A} = (\rho_A V_A \sin \epsilon)^2$$

$$\rho_A \tan \epsilon = \rho_B \tan (\epsilon - \delta)$$

$$V_A \cos \epsilon = V_B \cos (\epsilon - \delta)$$

(14)

where $\epsilon$ stands for the shock angle, $\delta$ is the deviation angle and $A$ and $B$ correspond to the pre-shock and the post-shock states, respectively. If the pre-shock state and the shock angle are known, Eq. (14) can be solved for the post-shock state, the deviation angle, and the resulting entropy rise $\Delta s = s(P_B, \rho_B) - s(P_A, \rho_A)$. In this study, the shock angle and the pre-shock state are correlated to the cascade outlet state according to the results of a set of two-dimensional RANS simulations of representative stages operating in ideal and non-ideal conditions, designed according to the method described in Sec. 5.1.

3.4 Endwall Loss. Endwall loss here includes viscous dissipation on the hub and casing boundary layers and viscous mixing of secondary flows. Due to the inherent complexity of secondary flows, a simple endwall loss model based on first principles is not available. Therefore, the improved semi-empirical correlation proposed in Ref. [22] is employed, see Appendix.

3.5 Tip-Leakage Loss. Tip-leakage loss is considered for the rotor only by assuming unshrouded blades. The entropy
production is calculated by means of the first-principles model proposed in Ref. [12], formulated as

$$T\Delta x = \frac{1}{m} \int V^2 \left(1 - \frac{V_{ps}}{V_{ss}}\right) d\eta_j$$

(15)

where the infinitesimal leakage mass flow rate can be calculated as

$$d\eta_j = C_{cu} \sqrt{2\rho_{ps}(P_{ps} - P_{ss})} dz$$

(16)

and the discharge coefficient $C_{cu}$ is equal to 0.3–0.4 for rotating cascades [12].

4 Case Studies

Best practices for design are investigated for axial turbine stages operating with siloxane MM and CO$_2$. The test cases are summarized in Table 2. For siloxane MM, three operating conditions are examined. For the test case labeled as $i$MM, the inlet conditions are selected such that the stage operates in the ideal gas thermodynamic region. In contrast, the conditions for the test cases denoted as $ni$MM are chosen to assess the impact of strong non-ideal flow effects on loss mechanisms and optimal stage layout. In particular, for case $ni$MM$_1$, the averaged value of $\gamma_{ps}$ is higher than $\gamma$, and vice versa for case $ni$MM$_2$. For what regards CO$_2$, two operating conditions are analyzed. In the same fashion as before, case $i$CO$_2$ entails an expansion in the ideal gas regime, whereas sCO$_2$ refers to a supercritical expansion, where $\gamma_{ps}$ is slightly higher than $\gamma$. For all case studies, the stages feature a total-to-static volumetric flow ratio equal to 4 and a design space delimited by $K_{is}$ in the range 3–10. The flow coefficient varies in the range 0.4–1.4 and 0.3–1.5 for the ideal and non-ideal test cases, respectively.

4.1 Influence of Fluid Molecule. The impact of the fluid molecule on turbine efficiency is evaluated by computing the design map of representative stages operating with MM and CO$_2$ in ideal gas conditions at $z = 4$, in the form $\eta_{opt} = f(K_{is}, \phi, x_{tip})$. The resulting efficiency contours are shown in Fig. 6. The difference in predicted $\eta_{opt}$ in the two test cases is within 1%. This efficiency difference substantially increases when computing the same design maps by fixing the total-to-static expansion ratio, thus confirming the superiority of $z$ as scaling parameter for transonic turbine stages.

The loss breakdown of two stage designs featuring largely different values of loading coefficient, i.e., $K_{is}$ = 3 and $K_{is}$ = 10, and the correspondent optimal values of flow coefficient, is depicted in Fig. 7 for both CO$_2$ and MM. At $K_{is}$ = 3, the largest discrepancies are observed for mixing and secondary losses, which are lower for the simpler fluid molecule. Boundary layer and tip leakage losses practically coincide, whereas shock loss is higher for CO$_2$ due to the comparatively higher Mach numbers and the higher heat capacity ratio. Overall, the effect of mixing and secondary losses is predominant and leads to the small efficiency deficit of the stage operating with MM. At $K_{is}$ = 10, the difference in Mach numbers between the two stages is close the previous case, but the average stage Mach numbers are significantly higher. In turn, compressibility effects assume primary importance, promoting an inversion of trend in mixing loss and a larger difference in shock and tip leakage losses, while boundary layer and secondary losses are virtually coincident. Altogether, at high loading coefficient, the stage operating with MM is expected to be slightly more efficient than the one operating with CO$_2$. As opposed to stage efficiency, the fluid molecule has large implications in terms of peripheral speed, therefore centrifugal loads. As discussed above, fluid made by heavy molecules like MM are characterized by comparatively lower enthalpy drops, therefore by lower tip speed for the same volumetric flow ratio over the entire design space, as displayed in Fig. 6. The values of tip peripheral speed reported for the two opposite values of $K_{is}$ reveal that the volumetric flow ratio of axial stages operating with

![Fig. 6 Design maps at $z = 4$ in dilute gas region. The dots correspond to the designs at optimal $\phi$ for each level of $K_{is}$, while the solid line represents their spline interpolation of order three. The labels show the peripheral speed at rotor tip corresponding to the lower and upper bounds of loading coefficient. (a) iCO$_2$ test case and (b) iMM test case.](image)

![Fig. 7 Lost efficiency breakdown for test cases $i$MM and $i$CO$_2$ at $K_{is}$ = 3–10 and optimum flow coefficient](image)
increasing stage loading and decreasing \( x \). In other words, for each \( K_a \), the best efficiency point and the corresponding optimal stage layout become function of the volumetric flow ratio and the selected working fluid. The physical reason thereof is intimately related to the trend of shock and mixing losses. The share of both these loss sources is directly proportional to the characteristic Mach numbers of the cascades, which also directly scale with \( \phi \), \( \gamma \), and \( z \). Therefore, at high \( z \), shock and mixing losses become predominant, and an attenuation of total entropy production can be only obtained by designing the stage at lower values of flow coefficient. This effect is more pronounced as the value of \( \gamma \) increases, namely, the simpler is the fluid molecule.

4.3 Influence of Non-Ideal Flow Effects. The impact of NICFD on axial turbine design and performance is analyzed for stages operating with siloxane MM representative of supercritical organic Rankine cycle applications, namely, the test cases labeled niMM1 and niMM2. The result of an additional calculation performed with CO2 in supercritical conditions, i.e., test case sCO2, is included in Fig. 10, but the correspondent design map and loss breakdown for the best efficiency point are omitted for brevity. The reason of that is twofold. First, \( \gamma_P \) shows only limited deviation relative to \( \gamma \) along the prescribed supercritical expansion, thus leading to small variations of performance trends and optimal duty coefficients with respect to case iCO2. Second, due to the different shape of the isentropic lines, a CO2 expansion in the proximity of the critical point would entail excursion in the two-phase region, and the study of expansion with phase change is beyond the scope of this work.

The resulting performance maps are shown in Fig. 11. For case niMM2 the absolute value of efficiency is significantly higher over the entire design space and the region of optimal design is shifted toward larger flow coefficients as compared to case niMM1. The efficiency values and the Smith line for case iMM lie in between, see Fig. 6(b).

As anticipated, these trends can be predicted by analyzing the variation of \( \gamma_P \), along the correspondent isentrope. Compared to the thermodynamic process in dilute gas conditions, where \( \gamma_P \approx \gamma \), for case niMM1, the isentropic pressure–volume exponent assumes, on average, larger values, and the opposite occurs for case niMM2. This leads to remarkable deviations in the Mach number trend along the expansion, see Fig. 2(b), ultimately affecting the share of the various loss sources, as depicted in Fig. 12. The predicted drop or gain in total-to-total efficiency is substantial. As expected, the main causes of such variation can be attributed to the significant deviations in mixing and shock-wave losses, which are intimately related to the characteristic stage Mach numbers.
In general terms, the effect of \( C_{PV} \) on stage design and performance is equivalent to a change of volumetric flow ratio. In other words, an average value of \( C_{PV} \) higher than \( C_r \) resembles an increase of \( a \), inducing larger shock and mixing losses, thus lowering the stage efficiency and the optimal flow coefficient. Note, however that, differently from \( a \), the departure of \( C_{PV} \) from \( C_r \) is function of the fluid molecular complexity and thermodynamic state. In absolute terms, the significance of NICFD effects is therefore strictly fluid-dependent, with molecularly complex compounds expected to provide the highest efficiency disparities.

5 Computational Fluid Dynamics Validation

5.1 Three-dimensional Stage Geometry Reconstruction.

Three-dimensional RANS and URANS simulations of the flow past turbine stages reconstructed from the output of the dimensionless turbine stage model are carried out with ANSYS-CFX [23]. The objective is to assess the capability of the first-principles loss model to provide accurate efficiency trends in both ideal and non-ideal flow conditions. Three turbine stages designed at \( a = 4 \) and increasing blade loading, correspondent to test cases iCO2 and iMM are selected to verify the effect of the molecular complexity. The impact of NICFD is quantified by means of two additional simulations performed on stages designed at \( a = 4 \) and minimum blade loading, for the non-ideal operating conditions corresponding to cases niMM1 and niMM2. The duty coefficients characterizing the selected stage layouts are summarized in Table 4.

In order to ensure full similarity between the reconstructed stage geometry and the correspondent dimensionless design, an automated design chain coupling the in-house turbine stage model, the open-source turbomachinery design system Multall [24] and the grid generator [23] is developed. The design workflow is summarized in Fig. 13. The procedure is as follows: for a given point on the performance map, the actual stage design is computed by first assigning a dimensional quantity, e.g., the mass flow rate, and then performing a mean-line design by means of the MEANGEN software, ensuring to comply with the set of nondimensional parameters provided by the in-house turbine stage model. During this process, the mass flow rate or the design point radius is iteratively adjusted to guarantee a minimum inter-row Reynolds number of the order of \( 3 \times 10^6 \), thus ensuring fully turbulent flow. The three-dimensional blade profiles are then generated using the STAGEN software by projecting a set of two-dimensional airfoils on the computed stream surfaces. The resulting curved profiles are then stacked along their centroids according to free vortex to be consistent with the assumptions made at preliminary design phase. The three-dimensional stage geometry is finally exported and meshed by means of a structured grid generator [23].

5.2 Numerical Setup. The boundary conditions of the CFD simulations are imposed in terms of flow direction, total pressure, and total temperature at the inlet, whereas an average static pressure is assigned at the outlet, enabling maximum local deviations within 5%. In order to reduce the computational cost, the minimum allowable number of blade passages is simulated by exploiting phase-lagged boundary conditions. \( k-\omega \) SST turbulence

Table 4 Dimensionless characteristics of the turbine stages selected for CFD validation

<table>
<thead>
<tr>
<th>Case name</th>
<th>( K_w )</th>
<th>( \phi )</th>
<th>( \eta^* )</th>
</tr>
</thead>
<tbody>
<tr>
<td>iCO2 3.0</td>
<td>0.55</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>iCO2 4.0</td>
<td>0.65</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>iCO2 5.0</td>
<td>0.8</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>iMM 3.0</td>
<td>0.55</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>iMM 4.0</td>
<td>0.75</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>iMM 5.0</td>
<td>0.85</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>niMM1 3.0</td>
<td>0.55</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>niMM2 3.0</td>
<td>0.55</td>
<td>0.3</td>
<td></td>
</tr>
</tbody>
</table>
model is employed together with adequate cell clustering near walls to guarantee $y^+ < 1$. Turbulence boundary conditions are set in terms of inlet turbulence intensity ($k = 5\%$) and eddy viscosity ratio ($\mu / \mu_t = 10$), while the turbulent Prandtl number is set to $Pr_t = 1$, in accordance with Ref. [25]. The advective and turbulent fluxes are discretized with total variation diminishing schemes [26], while a central difference scheme is adopted for discretizing the viscous fluxes. The time derivative term is discretized with a second-order backward Euler scheme. In order to speed up the evaluation of thermo-physical fluid properties, a look-up table method is employed by resorting to the reference thermodynamic library [14]. After performing a sensitivity analysis, a grid size of $4 \times 10^6$ cells and a thermodynamic mesh of $1 \times 10^6$ elements are set as optimal tradeoff between accuracy and computational cost. All the unsteady simulations are initialized with the results of steady-state RANS computations performed on the same stage geometries by using the mixing plane stator–rotor interface.

### 5.3 Results.

The time-averaged total-to-total efficiency obtained by steady and unsteady CFD for the test cases iCO$_2$ and iMM characterized by increasing loading coefficient is displayed in Fig. 14, along with the efficiency trend calculated with the turbine stage model coupled to the physical loss model as well as to the semi-empirical Traupel loss model [27]. The color bars in the charts are indicative of $\pm 1\%$ efficiency deviation with respect to the time-averaged values computed by 3D URANS.

Three main conclusions can be derived from these results. First, the efficiency values obtained by means of the first-principles loss model are within $\pm 1\%$ deviation bands for all considered test cases in ideal flow conditions, whereas the Traupel loss model.

### Table 5  Comparison of $\eta_{gt}$ predictions for stages designed at $K_a=3$, $\phi=0.55$ and $\chi'=0.3$ operating with MM in ideal and non-ideal thermodynamic conditions

<table>
<thead>
<tr>
<th>Case name</th>
<th>URANS</th>
<th>RANS</th>
<th>Physical</th>
<th>Traupel</th>
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</thead>
<tbody>
<tr>
<td>iMM</td>
<td>92.48%</td>
<td>92.03%</td>
<td>91.41%</td>
<td>87.43%</td>
</tr>
<tr>
<td>niMM$_1$</td>
<td>92.77%</td>
<td>89.28%</td>
<td>86.97%</td>
<td>84.76%</td>
</tr>
<tr>
<td>niMM$_2$</td>
<td>93.98%</td>
<td>93.75%</td>
<td>91.57%</td>
<td>87.59%</td>
</tr>
</tbody>
</table>

in Fig. 14, computed with the physical loss model, Traupel loss model [27], 3D RANS and URANS. (a) iCO$_2$ test case and (b) iMM test case.

![Fig. 13](https://example.com/fig13.png) Design workflow for the reconstruction of the stage geometries selected for CFD validation. (a) Target design point, (b) stage geometry reconstruction, and (c) 3D mesh.

![Fig. 14](https://example.com/fig14.png) Total-to-total efficiency trends of selected test cases, see Table 4, computed with the physical loss model, Traupel loss model [27], 3D RANS and URANS. (a) iCO$_2$ test case and (b) iMM test case.

![Fig. 15](https://example.com/fig15.png) Trend of the mass flow-averaged values of isentropic pressure–volume exponent and entropy generation along the normalized streamwise coordinate for test case niMM$_1$. The $\Delta s$ is normalized with respect to its maximum value. The URANS data are time-averaged.

---

**Table 5** Comparison of $\eta_{gt}$ predictions for stages designed at $K_a=3$, $\phi=0.55$ and $\chi'=0.3$ operating with MM in ideal and non-ideal thermodynamic conditions

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leads to a systematic underestimation of the stage performance. Both the physical and the semi-empirical loss models allow one to correctly capture the efficiency trend, with the former providing a slightly lower rate of decay as function of the blade loading. Second, the results obtained by CFD confirm that the influence of the working fluid is strongly attenuated when designing the stages at constant volumetric flow ratio in ideal gas conditions. In other words, the generalized isentropic exponent can be correctly predicted only approximately, of the stage performance. In turn, one can infer that the effects produce an underestimation of the stator efficiency and, ultimately, the spurious shock waves reflection at the interface. Overall, these effects resemble the efficiency drop with loading for stages operating with fluids made by simple molecules. The net consequence is that, for complex fluids, it is possible to design efficient turbine stages even at high load coefficient and volumetric flow ratio.

The results computed by RANS, URANS, and the turbine stage model coupled to the physical and the semi-empirical loss model for test cases iMM, niMM1, and niMM2 at minimum blade loading are reported in Table 5. The comparison shows that both the results of steady-state CFD and of the reduced-order model follow the same trend, namely, the total-to-total efficiency of the stage is inversely proportional to the average value of $\gamma_P$, along the expansion process. While the results of 3D URANS are consistent with this tendency for test cases iMM, niMM1, and niMM2 at minimum blade loading, i.e., increasing the stage Mach numbers, until an inversion of trend is observed around $K_{in} = 5$ at $x = 4$.

The results computed by RANS, URANS, and the turbine stage model are validated through high-fidelity CFD. Based on the outcomes, the following conclusive remarks can be drawn.

6 Conclusions

Design guidelines for axial turbine stages operating with non-ideal compressible flows have been derived using a first-principles loss model extended to arbitrary thermodynamic fluid models and the results validated through high-fidelity CFD. Based on the outcomes, the following conclusive remarks can be drawn.

(1) A generalized similarity equation valid for turbomachinery operating with non-ideal compressible flows can be formulated by introducing the volumetric flow ratio $x$ and the isentropic pressure-volume exponent $\gamma_P$, in place of the cascade characteristic Mach number and the perfect gas specific heat capacity ratio $\gamma$.

(2) Both $x$ and $\gamma_P$ have a strong impact on the compressibility effects of the stage. The larger the $x$, the higher the mixing and shock losses; thus, the lower the stage fluid-dynamic performance and the optimal flow coefficient. An average value of $\gamma_P$ larger than $\gamma$ along the expansion process resembles the effects of an increase of $x$, and vice versa.

(3) An increase of the volumetric flow ratio provides higher efficiency drops with loading for stages operating with fluids made by simple molecules. The net consequence is that, for complex fluids, it is possible to design efficient turbine stages even at high load coefficient and volumetric flow ratio.

(4) The first-principles loss model is accurate within 1–2 percentage points as compared to 3D RANS for all the selected test cases. In contrast, the semi-empirical Traupel loss model [27] significantly underestimates the stage efficiency, but it provides an accurate trend prediction with loading. The results obtained with 3D URANS are in accordance with those obtained by steady-state simulations and the physical loss model, except for the case niMM1, in which the trend of $\gamma_P$, is not monotone along the expansion.

Summarizing, the optimal flow coefficient, thus the preliminary stage layout, and the stage efficiency are significantly affected by the thermodynamic regime and by the operating conditions. The reduced-order stage model based on the NISE documented in this work is suited to devise design guidelines for axial turbines operating with arbitrary working fluids and thermodynamic conditions. Future research will target the in-depth study of turbine expansion processes characterized by non-monotonic variations of $\gamma_P$ by means of unsteady CFD.

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- Aard- en Levenswetenschappen, Nederlandse Organisatie voor Wetenschappelijk Onderzoek (Grant No. 17091; Funder ID: 10.13039/501100003246).

Nomenclature

Symbols

- $a$ = sound speed
- $c$ = chord length
- $c_{ax}$ = axial chord length
- $c_p$ = specific heat capacity at constant pressure
- $c_t$ = blade camberline length
- $c_v$ = specific heat capacity at constant volume
- $C_{du}$ = discharge coefficient
- $C_d$ = dissipation coefficient
- $g$ = blade tip gap
- $h$ = specific enthalpy
- $H$ = blade height
- $K_{in}$ = total-to-static loading coefficient

Downloaded from http://asmedigitalcollection.asme.org/gasturbinespower/article-pdf/143/1/011004/6608775/gtp_143_01_011004.pdf by Delft University of Technology, Library user on 15 June 2021
\[ Y = \frac{0.038 + 0.41 \tanh(1.26' / H)}{\sqrt{\cos \gamma \cos \zeta_{\text{in}}}} \left( \frac{H}{C_s} \right)^{0.55} \left( \frac{C_s \cos \zeta_{\text{out}}}{c_h} \right)^{0.55} \frac{H}{c} \leq 2 \]

\[ Y = \frac{0.052 + 0.56 \tanh(1.26' / H)}{\sqrt{\cos \gamma \cos \zeta_{\text{in}}}} \left( \frac{H}{C_s} \right)^{0.55} \left( \frac{C_s \cos \zeta_{\text{out}}}{c_h} \right)^{0.55} \frac{H}{c} > 2 \] (A1)

where

\[ Y = (P_{\text{in}} - P_{\text{out}}) / P_{\text{t, out}} \quad \Delta s = s(P_{\text{t, out}}, h_{\text{in}}) - s(P_{\text{t, in}}, h_{\text{in}}) \]

Appendix B

The turbine stage geometries used for the CFD validation can be freely downloaded at the website link.1

References


1https://bitbucket.org/pp-team/nicfdaxialturbines.git