AN ALTERNATIVE COMPRESSOR MODELING METHOD WITHIN GAS TURBINE PERFORMANCE SIMULATIONS

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Abstract

The precise prediction of compressor operation within gas turbine performance simulation is key to an accurate performance simulation model. This paper describes the modeling of multistage axial-flow compressors in gas turbine performance simulations. An alternative modeling technique to the conventional characteristic based approach is presented. The purpose is an improved prediction of the compressor's overall performance as well as of its interstage pressures and temperatures when changing its geometry by bleed extraction or displacement of the variable stator vanes.

Usually, within performance programs the compressor is modeled using a characteristic which describes the relation of its non-dimensional thermodynamic parameters in order to predict the off-design behavior. The performance effect caused by changes in variable geometry has so far been modeled differently within performance programs. Therefore, this paper starts with an overview about the various modeling methods within performance simulations. As an alternative method the coupling of a compressor meanline calculation with an aero engine performance simulation as a substitute to the compressor map is presented. The required changes to the performance program as the compressor module, the adjacent components including the secondary air system and the matching scheme are described in detail. When running the coupled system, the compressor overall and stage-wise off-design behavior is predicted by the meanline program whereby the performance program defines the actual working point by taking the operation of the surrounding components into account.

The successful operation of the whole simulation package has been demonstrated. The resulting working lines for a certain stage are shown. Furthermore the ability of forecasting the performance effect of geometry changes is presented. Finally an outlook on the range of applications is given which could benefit from employing the presented method.

NOMENCLATURE

Abbreviations

ISA	International Standard Atmosphere
VSV	Variable Stator Vanes

Indices

in	Compressor Entry
out	Compressor Exit
red	Reduced
ref	Reference Condition
is	Isentropic
t	Total condition

Symbols

c_p	Specific Heat Capacity at Constant Pressure
h	Specific Enthalpy
$\dot{m}$	Mass Flow
M	Torque
N	Rotational Shaft Speed
P	Pressure
T	Temperature
R	Specific Gas Constant
rh	Relative Humidity
Φ	Entropy function
β	Auxiliary Compressor Map Coordinate
η	Efficiency
κ	Isentropic exponent
π	Compressor Pressure Ratio
φ	Flow Coefficient
ψ	Work Coefficient

1. INTRODUCTION

Each *model* is an abstract image of the system which it shall represent. Thereby the model is similar to the appropriate system but simpler. The model depth as well as its degree of detail are left to the model creator, who is aware of its purpose. During the modeling process the creator has to balance the model accuracy (in order to create a model that is as close as possible to the real system) and its complexity (in order to be capable to understand and work with the model). Not every detail is worth consideration. Some model features increase complexity, but do not impact the model accuracy in a positive way. A good model is always a compromise between realism and simplicity.

Having once finished and validated a model, it allows the prediction of changes to the system without doing experimental tests on the system itself. Those tests might be too expensive, impractical or even impossible. The process of model operation is called a *simulation* [1].

Gas turbine performance models simulate gas turbine behavior throughout the operating envelope at various power levels in a 0-D manner. The range of performance model applications is broad. They are primarily employed to evaluate the potential of new gas turbine cycles, to predict engine operating behavior, to support engine tests and analyze test data. Finally they are increasingly applied for In-Service support [2].

During aero engine development and certification an important issue is the verification of compressor stability throughout the flight envelope. Consequently, for performance programs the accuracy for modeling the compressor during engine development is of high relevance.

This paper provides a detailed overview about compressor modeling in performance simulation programs. Usually, a compressor characteristic is employed to describe the correlation of the compressor's non-dimensional parameters. Additionally, for a rigorous engine analysis second-order effects and geometry changes are incorporated as modifications to the characteristic parameters.

In order to increase compressor simulation accuracy an alternative modeling technique for multistage axial-flow compressors is presented. It employs a compressor meanline calculation as a substitute for the compressor characteristic. This kind of *component zooming* results in a more precise prediction of compressor stage and overall performance. Especially, performance effects caused by a variation of compressor geometry can be considered more accurately.

2. PERFORMANCE MODEL SETUP

Throughout the years, the application range of gas turbine performance simulation has expanded widely. The field of applications includes gas turbine system design, performance prediction of existing gas turbines as well as engine test support. Moreover performance simulations are increasingly employed to improve maintenance by an enhanced diagnosis and prognosis [2].

Gas turbine performance models are intended to simulate the physical process of a gas turbine cycle through a mathematical representation of the engine. Thereby the prediction of the overall engine as well as the aerothermal component performance is one of the major tasks. Parameters as thrust, respectively power output, fuel flow and shaft speed(s) are of primary interest.

The basis for a precise performance prediction is an accurate computer simulation model. In the past model complexity was often limited by available computational power to relatively simple equations. Due to continuously progressing computer power an increase in numerical model details has emerged.

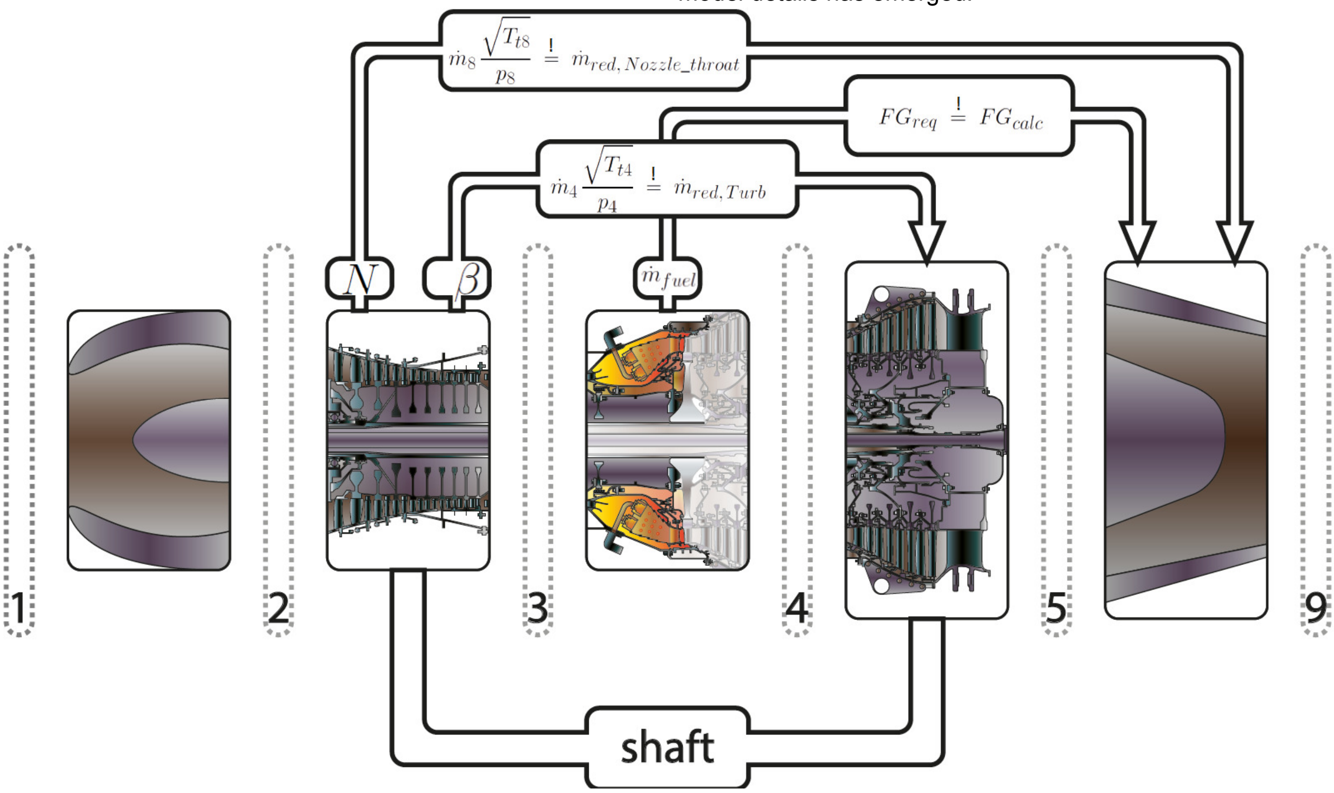

Figure 1: Schematic performance model structure of a one shaft turbojet engine

A modern *modular* gas turbine performance program features a number of modules with each module representing a specific gas turbine component. Component operation is defined by appropriate change of state equations that describe its thermodynamic behavior. In this context the choice of the employed fluid model is of high relevance for the model accuracy. Nowadays performance simulation programs comprise complex gas property equations as summarised in [3]. In document [4] various aspects of thermo-fluid modeling for gas turbines performance calculations are described. Due to the fact that performance simulation is a 0-D approach, the state variables used for calculation represent averaged flow path condition.

The modules are set up generically to guarantee a high reusability. Specific component off-design behavior is provided in form of a *characteristic* or *map*. This map usually characterises the relationship of non-dimensional or quasi non-dimensional component parameters. Additional phenomena which have a second-order effect as Reynolds number or humidity effects have to be considered for a thorough engine analysis [5].

Gas turbine model design evolves from a natural composition of the modules. Usually, the modules are arranged in the order as they are passed by the fluid. The modules are interconnected with each other by stations, which contain and pass information of the fluid condition before and after each module. International standards define the nomenclature for stations and modules [6], [7].

For a steady state simulation the basic requirements on a gas turbine performance program are:

- Power balance for each shaft
- Continuity of mass
- Equality of rotational speed for turbo-components placed on one shaft
- Static pressure balance at mixing surface (if present)

To satisfy the above mentioned requirements, a series of non-linear equations needs to be solved [5]. This can be achieved with nested loops. However, with rising model complexity and the associated increasing number of constraints this method becomes inefficient. Therefore today, the preferably used algorithm is a Newton-Raphson method, which utilises partial derivates to solve the mathematical problem. An employment of the Newton-Raphson algorithm allows the separation of physics modeling and numeric. This enables solving arbitrary engine concepts [8]. Figure 1 shows an exemplary model structure of a one spool turbojet engine including the required matching scheme.

For validation purpose a performance model is best compared against test data. In the process performance simulations are employed to analyze engine test data. The means for test data analysis is preferably a model based approach called *ANSYN – ANalysis by SYNthesis*. In this approach the discrepancy between performance model and test data is minimised by using appropriate model modifiers. Hence, ANSYN allows model validation and enhancement within one step [9].

A description of several modern performance simulation programs is provided in [10], [11], [12], [13].

3. COMPRESSOR MODELLING WITHIN PERFORMANCE SIMULATIONS

Due to its high impact on engine performance and safety a precise prediction of the compressor operation within performance simulation is important. As any other component in modular performance programs the compressor is modeled by a specific compressor module. Within this module the changes to the entering fluid are accounted appropriately. Specific compressor data is provided in form of the compressor characteristic in order to predict its off-design behavior. Besides these *first-order effects, second-order effects* (as Reynolds or humidity effects) are considered to account for deviations from the non-dimensional behavior. For the pure analysis of compressor performance any information about compressor geometry such as number of stages is of no relevance.

Figure 2 illustrates the typical entry and exit parameters of a compressor module.

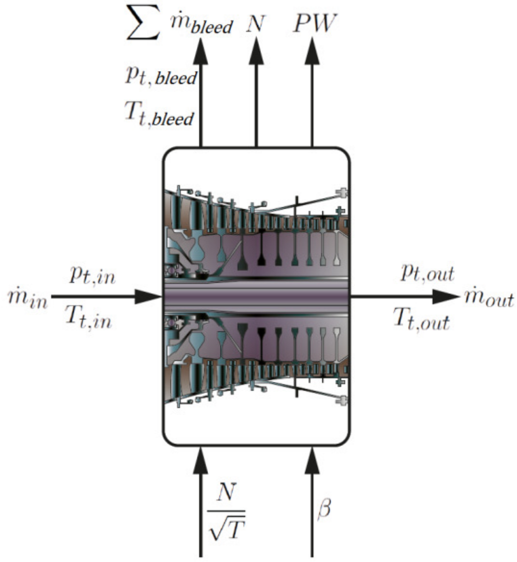

Figure 2: Compressor module entries and exits

3.1. Compressor Module Calculations

The compressor module calculates the thermodynamic change of state of the fluid (1)-(5) by a given set of the compressor characterising non-dimensional input parameters. These are commonly π, η_{is}, $\dot{m}_{red,in}$.

(1) $$P_{t,out} = \pi * P_{t,in}$$

(2) $$T_{t,out} = \frac{\pi^{(\kappa-1)/\kappa} - 1}{\eta_{is}} \cdot T_{t,in} + T_{t,in}$$

(3) $$\dot{m}_{in} = \dot{m}_{red,in} \cdot \frac{P_{t,in}}{\sqrt{T_{t,in}}}$$

(4) $$\Delta h_t = c_p \cdot \Delta T_t$$

(5) $$PW = \dot{m}_{in} \cdot \Delta h_t$$

Instead of pressure ratio similarly reduced work can be employed. For the internal compressor module calculation it is recommended to employ reduced work. 3.2.5 gives an overview about possible input parameter combinations.

For a design point calculation the input parameters are specified directly, whereas they are read from a characteristic for off-design analysis.

Figure 3 illustrates the proceeding in an entropy-enthalpy diagram.

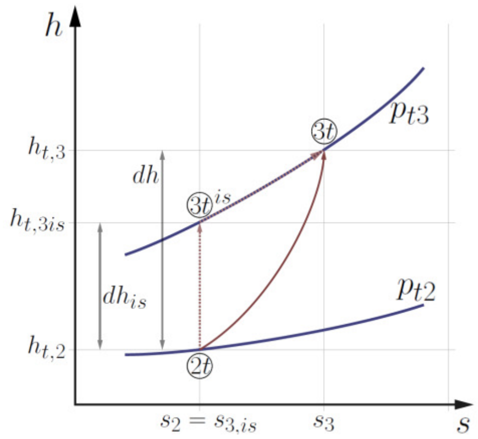

Figure 3: Schematic presentation of thermodynamic compressor calculation in an h-s diagram

According to [5] the use of constant c_p and κ can lead to calculation errors up to 5% in engine performance parameters. In order to increase modelling accuracy, rigorous enthalpy and entropy calculation have to be considered [8]:

- The caloric properties are a function of temperature for an ideal gas. E.g. for isobaric specific heat

(6) $$\left(\frac{\delta h}{\delta T}\right)_p = c_p(T)$$

- For an isentropic compression process the entropy function shall be employed:

(7) $$lnP_2 - lnP_1 = \Phi_1 - \Phi_2$$

Where the entropy function Φ is defined as:

(8) $$\Phi(T) = \int_{T_{ref}}^{T} \frac{c_p(T)}{RT} dt$$

Considering real gas properties increases model accuracy further. The caloric fluid properties within a real gas model are a function of temperature and pressure. [3] shows the deviation between heat capacities calculated from an ideal and real gas model and recommends to apply real gas calculations in regions where the error will exceed 0.5%.

3.2. Compressor Map

The compressor off-design behavior is usually modeled by a 0-D representation of the relationship of its non-dimensional parameters which is called the *compressor map*. These non-dimensional or quasi non-dimensional parameters represent Mach number similarities. Their employment accounts for varying inlet conditions and hence reduces the number of variables used to describe compressor off-design operation.

The general representation of a compressor map is demonstrated in Figure 4. Pressure ratio and isentropic efficiency are plotted against non-dimensional mass flow for a series of referred speed lines. The operational boundary is the surge line which limits the region of stable compressor operation.

A compressor characteristic is created at a fixed environmental reference condition $T_{ref,in}, P_{ref,in}, rh_{ref,in}$, which is usually ISA sea level assuming dry air. A variation of the entering fluid compared to the reference condition can impose second-order effects causing significant deviations from ideal non-dimensional behavior [5].

Considering only ideal non-dimensional relationship (*first-order effects)*, the following assumptions are valid [5]:

- For a fixed compressor geometry (stator position, bleed offtake, tip clearance) the compressor map is unique.
- Due to Mach number similarity each operating point in the map has a unique velocity triangle.
- The operating point in the map is dictated by the surrounding components.

At early design stages the characteristic is a product of numerical compressor simulation programs of higher order. When the appropriate hardware is in place, it is created from compressor rig test measurements.

Particular compressor maps are owned by the developing company. They are rarely published in the open literature. Kurzke [11] provides a collection of published data for a range of axial-flow and radial-flow compressors.

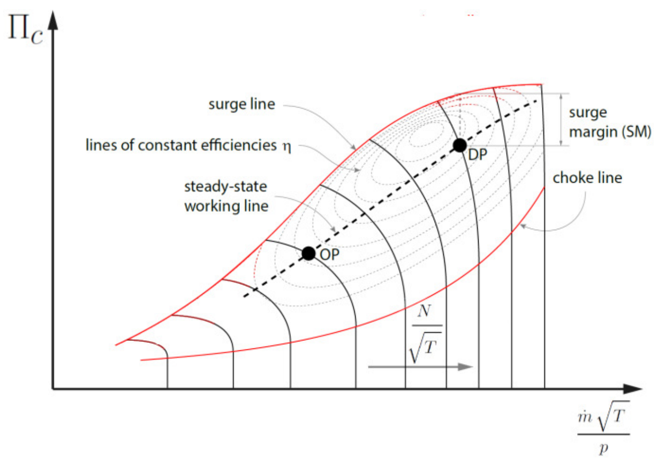

Figure 4: General compressor map representation

3.2.1. Additional Map Preparation for Use in Performance Programs

Before the compressor characteristic can be applied within a performance program, it is commonly *smoothed* in order to account for the gradient based numerical solution algorithm. A useful program to conduct smoothing is SmoothC [14].

A further issue that is related to numerical stability is the choice of lookup variables in order to clearly read out the compressor map. Looking at the general description of a compressor map in Figure 4 it seems practical to read out pressure ratio and efficiency from non-dimensional speed and mass flow. However, at high power region where speed lines tend to get vertical, a bijective read out is not possible. For a particular combination of non-dimensional mass flow and speed a number of map points can be identified. Vice versa the problem appears at low power regions for horizontal speed lines when using π and $N/\sqrt{T_{t,in}}$ alternatively. These examples show that the variables need to be collinear in order to verify a unique readout. Jones investigated the use of various physical and non-physical parameters [15]. He concludes that the most convenient approach is the usage of non-dimensional speed and the auxiliary coordinate β. Nevertheless, the incorporation of β as second lookup parameter necessitates the use of three lookup tables instead of two.

(9) $$\pi, \eta_{is}, \frac{\dot{m}_{in}\sqrt{T_{t,in}}}{P_{t,in}} = f\left(\frac{N}{\sqrt{T_{t,in}}}, \beta\right)$$

There are a few issues which need to be considered when setting up beta lines for a compressor map. In addition to the avoidance of collinearity, Kurzke proposes to define them parallel to the peak efficiency line to reduce interpolation errors [14]. The peak efficiency line connects the efficiency maxima of each speed line. Figure 5 shows an exemplary incorporation of beta lines into a compressor map.

The provision of the compressor map as a series of tables implies the necessity for interpolation between the map nodes. The choice of the number of nodes as well as the employed interpolation method is eminent for precise performance prediction. Akima [19] proposes a good method for interpolation that avoids kinks at map nodes as linear interpolation and non-intended bumps as polynomials. Interpolation errors can easily culminate in regions of second-order effects. Of course the interpolation method as well as the node count enlarges the required calculation time.

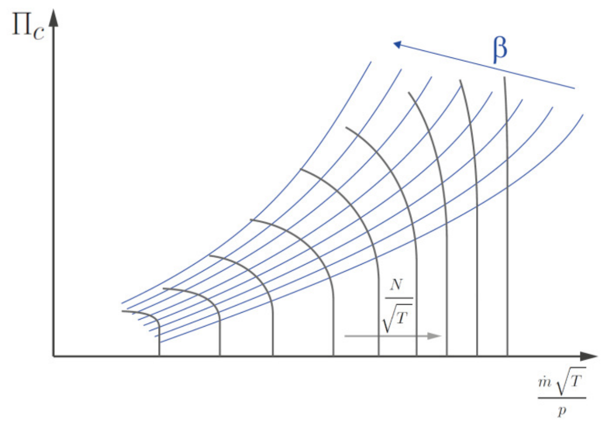

Figure 5: Compressor map including beta lines

3.2.2. Surge Line Data

For the sake of stability margin assessment some information about the compressor surge line needs to be provided. This is usually done by defining the surge pressure ratio as a function of non-dimensional inlet mass flow [16].

(10) $$\pi_{surge} = f\left(\frac{\dot{m}_{in}\sqrt{T_{t,in}}}{P_{t,in}}\right)$$

A definition of the surge margin SM is given with

(11) $$SM = \left(\frac{\pi_{surge} - \pi_{operating}}{\pi_{operating}}\right)_{\dot{m}_{red,in}}$$

3.2.3. Provision of Interstage Values

In the absence of a stage wise compressor modeling some information about the temperatures and pressures needs to be provided to model the performance impact of the secondary air flows. Equation (12) proposes a definition of referred interstage parameters.

(12) $$X_{t,stage,ref} = \frac{X_{t,stage}}{X_{t,in}}$$

Where X is either pressure or temperature. Figure 6 shows an example of referred interstage parameters as defined by equation (12).

Constant stage loading is a good assumption in case of missing details regarding interstage data.

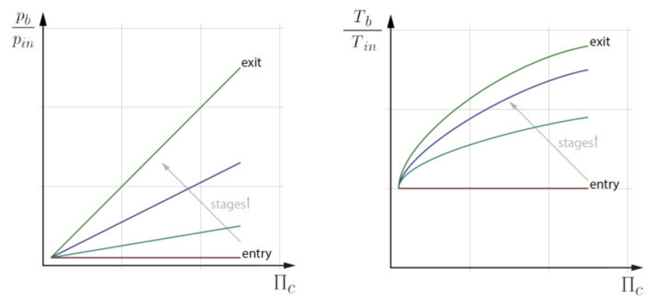

Figure 6: Examplary progress of interstage bleed pressure and temperature

3.2.4. Determination of the Compressor Operating Point

During gas turbine operation the compressor operating point is dictated by the surrounding components. The same applies to performance simulation. In order to satisfy the physical equations of state, several constraints have to be set up. Figure 1 shows the constraints for a turbojet engine. When those constraints are matched properly the working point for the compressor has been determined.

When using a compressor map as function of non-dimensional speed and β, the operating point of the compressor is found when:

- The non-dimensional mass flow predicted by the turbine characteristic matches to non-dimensional combustor exit mass flow.
- The mass flow entering the nozzle is under current nozzle entry total temperature and pressure capable to pass nozzle throat area.

Besides that for a prediction of a steady state operating point the shaft power balance and the equality of rotational speed for turbo components on one shaft need to be fulfilled.

3.2.5. Alternative Map Presentations

The presentation of the characteristic is not fixed to the parameters $\pi, \eta_{is}, \dot{m}_{red,in}$. Walsh and Fletcher [5] state that any combination of dimensionless input parameters is suitable, as long as they define mass flow, pressure ratio and temperature rise. Of course, the order of the calculation changes with different sets of non-dimensional inputs.

Possible alternative combinations are:

(13) $$\frac{M}{\dot{m}\sqrt{T_t}}, \pi, \frac{\dot{m}_{in}\sqrt{T_{t,in}}}{P_{t,in}} = f\left(\frac{N}{\sqrt{T_{t,in}}}, \beta\right)$$

(14) $$\frac{dh_t}{T_{t,in}}, \pi, \frac{\dot{m}_{in}\sqrt{T_{t,in}}}{P_{t,in}} = f\left(\frac{N}{\sqrt{T_{t,in}}}, \beta\right)$$

(15) $$\psi, \psi_s = f\left(\varphi_{out}, \frac{N}{\sqrt{T_{t,in}}}\right)$$

The map presentation depends on the model purpose. For extreme part load operating conditions as starting or windmilling, where speed lines are going to be 0, the efficiency reaches a discontinuity. Therefore the representation including efficiency is not very useful. For zero speed simulation, which occurs during engine start, a representation including corrected torque (13) is suitable [16].

Switching from one map representation to another can imply several problems:

- When reading the map at ambient temperatures deviating from reference condition, discrepancies within the non-dimensional parameter correlation (e.g. $dh_t/T_{t,in}$ vs π) will occur due to the temperature dependency of the caloric fluid properties. Reading the map at reference condition and proceed internally with the parameters $dh_t/T_{t,in}, \eta_{is}, \dot{m}_{red,in}$ will solve that problem.
- Errors resulting from interpolation differences can easily exceed second-order effects.

3.3. Consideration of Second-Order Effects

Phenomena that lead to a deviation from ideal non-dimensional component or engine behavior are called *second-order effects*. For a rigorous performance analysis it is necessary to account for considerable second-order effects. According to [5] and [16] the second-order effects appearing within the compressor are:

- Inlet distortion effects
- Unintended geometry changes
- Reynolds number effects
- Working fluid effects
- Variable geometry
- Transient effects

For the modeling of the impact of second-order effects, detailed component knowledge is required. A simple means for considering these effects is the employment of modifiers on component performance parameters, which are set up to a reference condition. The modifiers are mostly defined as a function of engine power and the difference to the predefined reference condition.

(16) $$MOD = f(PowerParam, \Delta refParam)$$

For a complete and accurate performance description of a compressor including the consideration of second-order effects, a whole series of maps is required. Thereby the deviation of the respective effect compared to map reference condition needs to be defined as additional lookup parameters.

The effect on compressor stability is for all of the below described effects usually accounted through respective threads on stability margin.

3.3.1. Inlet Distortion

Inlet flow distortions are non-uniformities in either temperature, pressure or swirl which primarily influences compression system stability. Moreover, a performance impact in form of an efficiency degradation can be observed as well. In the open literature the effect ranges between 1% and 5% [24].

To account for temperature and pressure distortion effects within performance simulations Kurzke [24] proposes to utilise a parallel compressor model supplied with different entry pressures or temperatures. The operating points of both parallel working compressors are found by a static pressure balance behind the compressor.

By employing mass flow and pressure ratio modifiers the effect of bulk inlet swirl can be modeled. The modifiers can assumed to be proportional to the swirl angle.

3.3.2. Tip Clearance Variation

Tip clearance variations are minor geometry changes that can have a significant impact on compressor performance - mainly efficiency and surge margin. For steady state condition a variation in ambient temperature can have an effect on the tip clearance. During transients the clearance may vary, due to thermal expansion and centrifugal forces. The effect on the characteristic is by trend comparable to the Reynolds number effect.

As several other second-order effects, tip clearance effects are accounted compared to a reference condition. Due to differing internal operating conditions for transient operation, tip clearance deviates from steady state. Therefore [20] and [21] propose to employ alternatively state space models in combination with a stage wise compressor modeling for the accounting of *tip clearance* within multi-stage axial compressors.

3.3.3. Blade Untwist

In order to account for the increasing circumferential speed from hub to tip the compressor rotor blades are twisted. Within engine operation untwist occurs due to centrifugal forces. Those are related to mechanical speed and hence change the relationship of the non-dimensionals.

The untwist effect is comparable to a rotor blade restaggering. The effects can be included in performance modeling by adding modifiers on mass flow, work and efficiency. For a more accurate modeling multiple maps shall be employed.

3.3.4. Reynolds Number Influences

The Reynolds number decreases with rising altitude and thus falling ambient pressure. Below a critical Reynolds number the viscosity has a significant effect on engine component performance. Growing boundary layers reduce efficiencies and mass flow and effect the ideal non-dimensional parameter relationship which is defined by the map.

A compressor map is set up at a reference Reynolds number. At differing Reynolds number modifiers on efficiency and mass flow are conventionally defined as shown with equation (16).

According to [17] the efficiency change caused by Reynolds number changes can roughly be described exponential.

3.3.5. Humid Air

The presence of water in the air is called humidity. Humidity changes the caloric fluid properties since the gas composition differs at varying levels of water in the air. The conventional employment of a quasi non-dimensional parameter representation of compressor maps does not consider a variation in fluid properties. Ignoring this property change can lead to errors of 2-3% in establishing component efficiencies [22]. Though, according to [5] it is acceptable to assume the humid gas mixture to be a perfect gas up to a specific humidity of 10%.

Correction factors as a ratio from dry to moist air can be derived from the full non-dimensional parameter representation.

$$\left(\frac{\dot{m}_{in}\sqrt{T_{t,in}}}{P_{t,in}}\right)_{ref} = \frac{\dot{m}_{in}\sqrt{T_{t,in}}}{P_{t,in}} \cdot \sqrt{\frac{\kappa_{ref}R}{\kappa R_{ref}}} \tag{17}$$

$$\left(\frac{N}{\sqrt{T_{t,in}}}\right)_{ref} = \left(\frac{N}{\sqrt{T_{t,in}}}\right) \cdot \sqrt{\frac{\kappa_{ref}R_{ref}}{\kappa R}} \tag{18}$$

3.3.6. Water Ingestion

Liquid water entering a compressor does affect compressor stage performance. Obermüller et al. [23] describe the major impacts of liquid water ingestion:

- Compared to dry air operation compressor exit temperature reduces due to the required enthalpy of evaporation.
- Non-dimensional inlet mass flow at a non-dimensional speed rises.
- Compressor aerodynamic performance may be negatively affected depending on the size of the droplets.

Due to the impact on thermodynamic fluid properties as well as on compressor aerodynamic, a requirement for the presences of a compressor meanline calculation within performance simulation is evident. There are multiple different modelling methods which aim for the correction of water ingestion in the compressor. They are summarised in [23].

3.3.7. Transient Effects

During transient maneuvers second-order effects resulting from shaft inertia, heat soakage and volume packing can occur. Shaft inertias effects impose a working line excursion and do not have a map effect. They are therefore not further considered in this paper.

A detailed overview about transient effects and their impact on gas turbine component performance is provided by Fiola [25]. Peitsch describes the importance of accurate modeling of transient effects for the design and verification of the control system [26].

For steady state conditions equal fluid and metal temperatures are assumed. For transient calculations this assumption is no longer valid. As a result there is a heat transfer between the fluid and the covered surface which is called *heat soakage*. The relevance of this second-order effect is depending on the temperature difference between fluid and component metal, metal mass, heat transfer coefficients, component geometry and the maneuver speed.

For a short period of time the mass flow entering the compressor may not be equal to the flow out of it due to the effect of density changes. This phenomenon is called *volume packing*. Assuming to start and end at steady conditions, the mass flow integral for the entire simulation time of compressor inlet and exit flow should be balanced. The relevant parameter for the significance of volume packing is the component volume.

A mathematical description for modeling transient effects within performance simulations is provided by Thomson [27].

3.4. Consideration of Variable Geometry

When employing variable geometry such as bleed air and variable stator vanes, the map representation with a fixed relationship between the non-dimensional map parameters becomes inadequate.

3.4.1. Bleed Offtake

In addition to its primary purpose which is increasing total pressure, the compressor needs to ensure the provision of mass flow with sufficient pressure to the secondary air system for several internal and external applications.

In order to account for the law of conservation of mass, the mass flow at compressor exit needs to be reduced by the amount of bleed mass flows.

$$\dot{m}_{out} = \dot{m}_{in} - \sum \dot{m}_{bleed} \tag{19}$$

The fluid bled off at compressor interstage doesn't receive full compressor work. Thus the compressor power calculated by equation (5) needs to be decreased by the amount of energy required to compress the bleed mass flow from offtake position to compressor exit. Equation (20) demonstrates the satisfaction of the energy balance.

$$PW = PW_{map} - \sum \dot{m}_{bleed}\,(h_{t,out} - h_{t,bleed}) \tag{20}$$

Bleed extraction can be regarded as a compressor annulus change resulting in a variation of the velocity triangles and hence compressor stage performance. The higher the extracted mass flow, the bigger is the impact on the compressor map. Therefore the baseline compressor map should include at least the bleed mass flows used for internal purposes which are usually not modeled variable.

A possible option to account for the shift of stage matching is the split of the compressor map into several parts. In [28] a distribution separated by the bleed offtake positions is proposed.

However, besides the loss of mass flow, bleed offtakes induce changes in the flow field due to swirl and temperature distribution changes and distortion. These effects change the compressor map too and need to be considered as well. Thorough modeling of variable bleed mass flows as customer or anti icing bleeds results in multiple maps per stage, whereby the variables to read out the compressor stage maps will be extended by the amount of bleed mass flows at the respective stages.

As Figure 7 exemplarily shows, the effect of bleed offtake usually lowers compressor surge line due to resulting flow distortion.

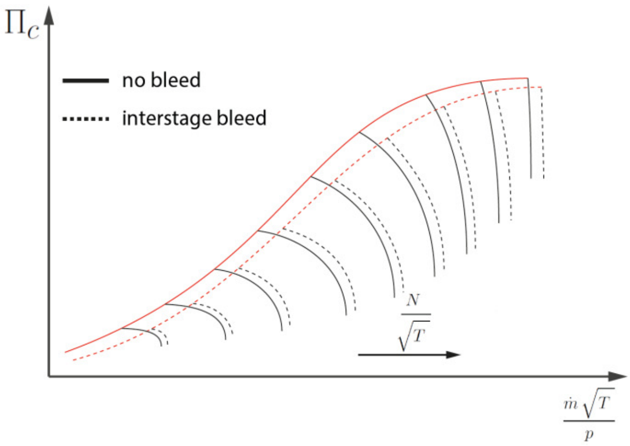

Figure 7: Bleed effect on compressor characteristic

3.4.2. Variable Stator Vanes (VSV)

Within multistage axial-flow compressors the variable inlet guide vane and the variable stator vanes are employed to enlarge compressor operating range. They are set up in order to provide an ideal angle of attack to the following rotor. An optimised setting results in a surge line increase and efficiency improvement at off-design condition.

The change of the variable inlet guide vane angle provokes primarily a shift in compressors non-dimensional flow-speed correlation. Figure 8 presents the effect of more closed variable guide vanes on the characteristic. During engine operation the vanes are closed only at low power to secure part load operation. For steady state operation at design speed the nominal angle is optimal per definition and does not need to be adopted.

When multiple variable stator vane rows are in place they are conventionally coupled by a ganged system. This approach simplifies scheduling due to the fact that only one row needs to be set. This is usually the variable inlet guide vane. The other variables operate corresponding to their gear ratio. To retrieve the gear ratio, for each value of corrected speed, the VSV setting of the various stages needs to be optimised separately.

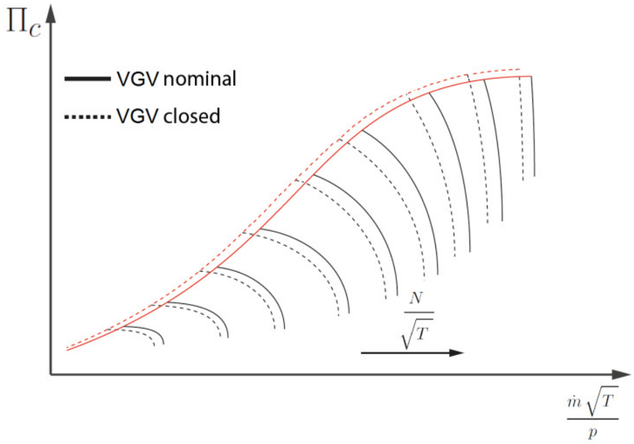

Figure 8: VGV effect on compressor characteristic

Since the ganged variable stator vanes are generally scheduled versus compressor non-dimensional speed, which is also used as a variable to read the compressor characteristic, the compressor map is commonly a composite map. Within such a map each speed line represents a unique compressor geometry including a specific variable stator vane setting [5].

When modeling a deviation from the baseline VSV schedule a few different procedures have been chosen which are summarised below.

An approach is the consideration of exchange rates on η, π and $\dot{m}_{in}\sqrt{T_{t,in}}/P_{t,in}$ as defined by equation (16). The magnitude of the performance parameter change needs to be assessed beforehand.

Kurzke proposes in [29] a simple empirical method where the performance parameters are a linear or quadratic function of Δ VSV angle α as shown in equations (21)-(23).

(21) $$\Delta\dot{m} = k_1\alpha$$

(22) $$\Delta(\pi - 1) = k_2\alpha$$

(23) $$\eta = \eta_{map}(1 - k_3\alpha^2)$$

Therkorn presents an analytical approach based on the assumption of constant compressor exit volume flow for a VSV displacement [18]. The requirement yields further a constant pressure ratio function to non-dimensional mass flow constraint.

(24) $$\frac{\dot{m}_{in}\sqrt{T_{t,in}}}{P_{t,in}} = k_1\pi^{1-\frac{R}{c_p}}$$

The isentropic work coefficient is adopted to that constraint. The efficiency is corrected by a quadratic equation.

(25) $$\eta = \eta_{map}(1 + k_2|1 - k_1|)^2$$

The knowledge about the modifiers k_1 and k_2 is retrieved from test results or numerical simulations of higher order.

In [30] and [31] for the purpose of precise VSV displacement modeling a multi-map approach has been chosen. Herein the VSV offset angle is applied as additional lookup parameter for the compressor map. The information about stability line shift has been included as well.

In order to account for an un-ganged VSV system [2] proposes to use multi-maps with the angle of each variable stator row as map lookup parameter. By employing such a map, an optimisation of the VSV schedule can be conducted through a performance simulation.

3.5. Compressor Map Scaling

When conducting an engine test the compressor performance measurements are compared to the simulation results of the performance model. Differences may due to wrong assumptions concerning the baseline maps, the disregard of second-order effects or an incorrect assumption about the amount of second-order effects (of course measurement uncertainties can be the reason as well). The deviation to the measurements requires an adaptive compressor map scaling beyond the accounting for second-order effects. Thus an ANSYN task is conducted to minimise the measured to model predicted parameter differences by defining appropriate component modifiers. Regarding the compressor the question is how to define the slope of the resulting component modifiers (Figure 9). Kurzke propose in [32] to scale along:

- constant $\Delta h_t/N^2$,
- a parallel to the peak efficiency line or
- non-dimensional exit flow.

The decision, which approach is taken, needs to take the intended application into account.

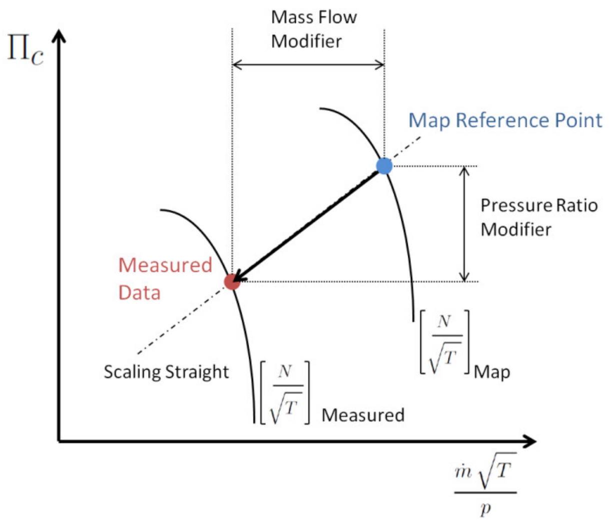

Figure 9: Definition of compressor map scaling line

4. EMPLOYMENT OF A COMPRESSOR MEANLINE CALCULATION

As summarised above a rigorous conventional modeling of the effects of variable compressor geometry changes requires an extension of the compressor map dimensions. According to that, a compressor with conventional controlled variable stator vanes and two variable interstage bleed offtake positions will have 5 independent map parameters ($N/\sqrt{T_{t,in}}$, β, ΔVSV, $\dot{m}_{bleed,lower\ stage}$, $\dot{m}_{bleed,higher\ stage}$). If an un-ganged VSV system is considered, the number of variables can rise even more. This will lead to a huge amount of data, requiring lots of time for interpolation.

4.1. Program Coupling

Alternatively, in order to increase the degree of modeling detail as well as model accuracy in terms of compressor geometry changes, a compressor zooming can be implemented by integrating a compressor meanline calculation into the performance simulation. This physics-based 1-D calculation allows for a precise accounting of the effect of VSV and bleed offtake for each compressor row. Depending on the degree of modeling detail of the meanline program, it is even possible to increase the precision of accounting for other secondary order effects on compressor performance and stability. A further advantage of an integration of the compressor calculation instead of a map that consists of a number of discrete nodes is the avoidance of interpolation.

The coupling has been performed with the Rolls-Royce performance tool MARS and a compressor meanline program. MARS is a C++ written modern modular performance simulation program that employs maps for modeling compressor off-design behavior and is able to account for all of the above mentioned second-order effects. The compressor meanline program is FORTAN based and allows for the computation of the flow parameters along the center streamline in a 1-D manner. The calculation is performed at every rotor and stator entry and exit on the basis of the Euler equation. The accounting of profile losses, slip factor, various profile geometries and boundary layer losses is considered by the use of correlations. Similarly hub and tip temperatures and pressures are determined defining the bleed air properties. Furthermore, the surge line can be predicted by determination of the surge mass flow for a particular speed line.

The meanline program is provided in form of a dynamic link library (dll). The compressor module of the performance program has been equipped with an interface to load the dll, pass the required inputs into it and finally run it.

The definition of the respective compressor design is declared to the meanline program. Moreover, for each meanline computation the inputs provided by the performance simulation are map reference conditions, non-dimensional speed non-dimensional mass flow, amount and stage of variable bleed flow and VSV offset angle. The non-dimensional speed should be corrected to dry reference condition beforehand by employing equations (17) and (18).

The performance simulation from engine point of view does not change. The compressor operating point is determined by the operation of the surrounding components. The meanline calculation provides precise information about compressor stage performance and the effect of geometry changes.

Two different options of a meanline integration have been implemented. Within the first variant the map is substituted completely. The second option is a hybrid method that employs map and meanline calculation.

4.2. Map Substitution

For the substitution of the map, the global performance matching scheme (see Figure 1) is adopted to the inputs required by the meanline calculation program. Instead of the parameters speed and β that are used to read out the map, the parameters speed and compressor exit flow are employed. They define uniquely the working point. The meanline simulation outputs are parameters π and η_{is}. The remaining compressor performance parameters are calculated as described by equations (1)-(5). All second-order effects are assessed as for a conventional map read out. The secondary air flow conditions are directly defined by the respective hub and tip interstage values. The surge line predicted by the meanline program is used as is for the performance calculation.

The calculation time for one point was maximum 3s per steady state point, which is about 5 times higher compared to a compressor map based model.

4.3. Hybrid Meanline Modeling

The full substitution of a map is useful at the beginning of an engine development program, in case the meanline program represents best design understanding. This may be valid for preliminary design phase or latest up to engine development stage 2 exit. Beyond, when a measured compressor map is in place, and hence the actual compressor performance is known to be different what the meanline calculation predicts, a different method needs to be employed.

An option is the adaption of the meanline program to the measurements and proceed with the method presented above. However, to tweak the meanline program in order to meet the measurements is not a trivial but time intensive task and is therefore usually not done.

Thus, the program coupling has been developed further. A compressor map describes the first-order effects. This implies that the global performance iteration does not need to be changed. For an assessment of the effects of geometry changes a meanline calculation is incorporated. The meanline calculation is conducted twice, first at nominal engine setting and second at changed geometry. The delta in performance parameters, interstage parameters and surge line pressure ratio between both simulations is added to the baseline characteristic.

To assess the delta, a representative point for each of the two meanline calculations has to be identified. The first meanline calculation needs to be conducted at a condition that is comparable to the actual working point of the compressor map. This task equals the problem of map scaling in an ANSYN. In order to create physically comparable points, the first meanline calculation is carried out at non-dimensional speed and compressor exit mass flow of the operating point read from the map. The second meanline calculation operates at changed compressor geometry. The delta between both calculations is besides the consideration for other second-order effects added to the map parameters in order to retrieve the compressor output values. Figure 10 illustrates the process.

The calculation time was at maximum 6s per steady state point.

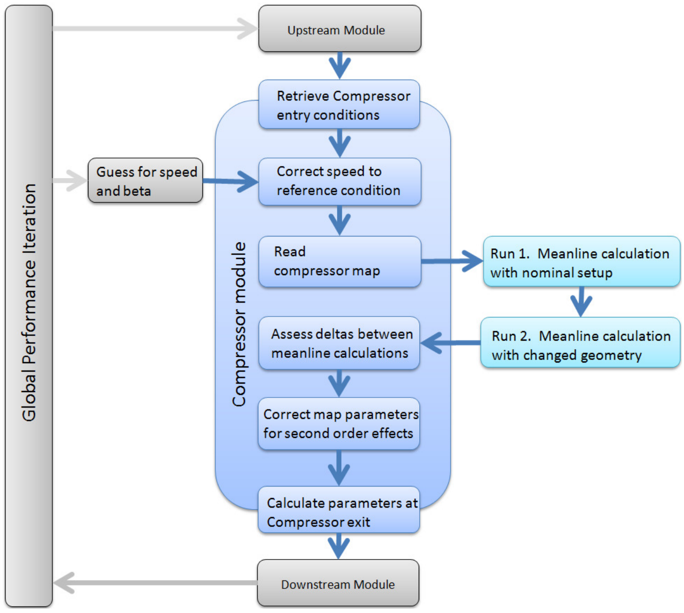

Figure 11: Flow chart displaying hybrid compressor modelling

4.4. Results

For validation purpose a performance model of a modern two shaft turbo fan engine with a 10 stage axial compressor has been employed. A steady state working line at ISA sea level static condition has been calculated.

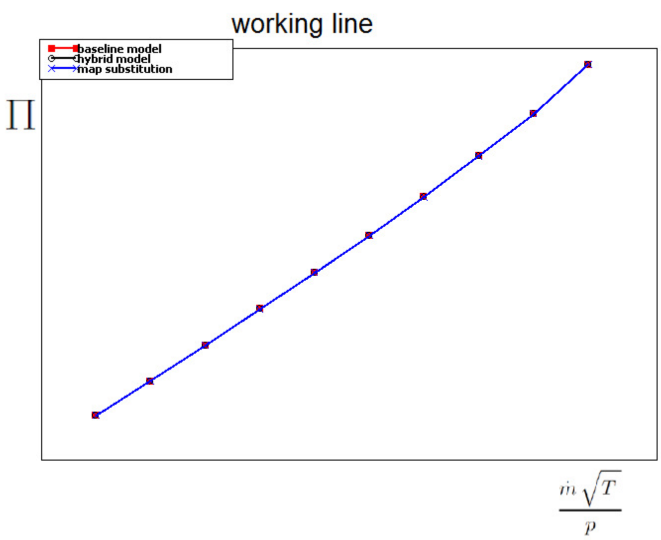

Figure 10: Working line comparison of map based against meanline integrated calculation

Thereby, the results of a conventional computation based on a map, (which was beforehand produced by the meanline program) has been compared against the full substitution and the hybrid modeling. Figure 11 shows the results of the working line as total pressure ratio against reduced compressor inlet mass flow. The results of the different calculations calculations match well. The deviation in major performance parameters is within iteration tolerance.

Another working line has been computed with the hybrid model with a VGV of 5° closed. The outputs have been compared against results of a map based model. The map based model has conducted a calculation with nominal VGV setting and VGV 5° closed. For the latter a correction in form of modifiers according to equation (16) have been employed. The hybrid model matches the conventional approach as can be seen in Figure 12. Small differences can be traced back to assumptions and simplifications for the creation of the modifiers for the map based approach. Figure 12 shows further that closing the VGVs induce a shift in flow-speed correlation to lower non-dimensional mass flows at a specific non-dimensional speed.

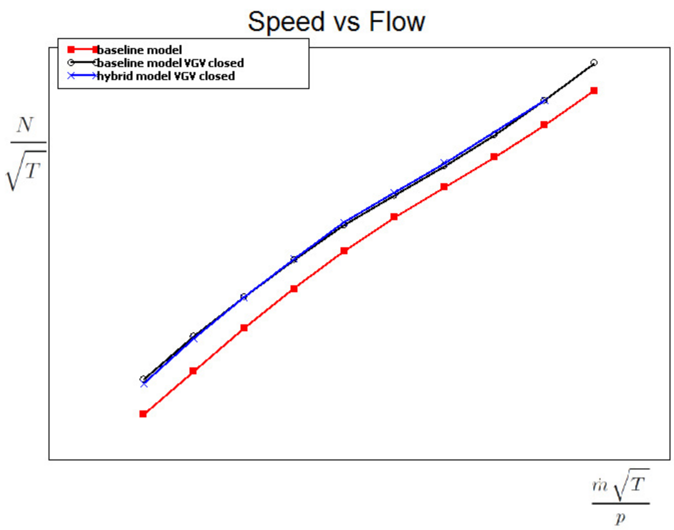

Figure 12: Hybrid modeling showing the effect of VSV angle 5° closed on speed-flow correlation

Finally, a working line calculation with bleed offtake at stage 4 has been conducted employing hybrid modeling. The stage performances of stage 3 and 6 have been compared against a nominal engine configuration.

The results shown in Figure 13 indicate the benefit of the integrated meanline calculation. A detailed analysis of compressor stage matching is possible. Figure 13 shows that stage 3, which is located in front of the bleed offtake, is de-throttled. This stage operates therefore at a lower pressure ratio for a non-dimensional mass flow. In contrast the loading of stage 6 rises due to the upstream located bleed offtake.

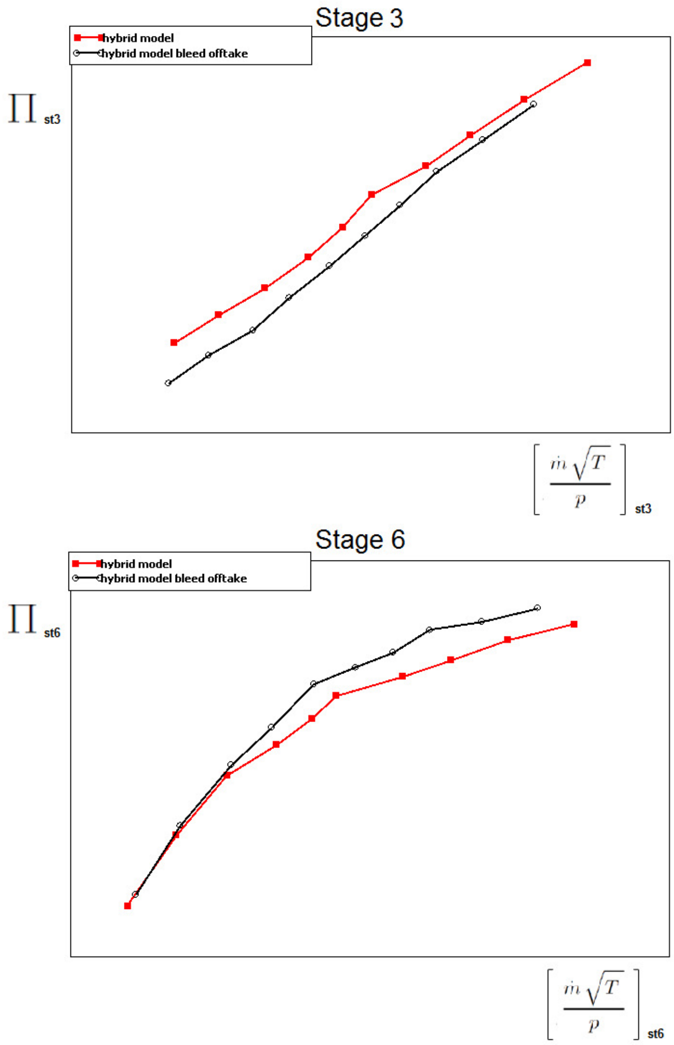

Figure 13: Effect of 5% bleed offtake at stage 4 on stage matching of stage 3 and 6

5. SUMMARY

This paper provides an overview about compressor modeling in gas turbine performance simulations. An alternative modeling technique to the conventional map based approach has been presented. It incorporates a compressor meanline calculation into the performance simulation as a substitute to the compressor map. The physics-based 1-D meanline simulation allows an improved prediction of variable compressor geometry changes. The methodology has been developed further to account for a deviation between compressor map and the prediction of the meanline program. The model accuracy could be increased with small rise of model complexity. The calculation speed is an issue at the moment. It will be improved in the future.

The successful coupling of the engine performance and compressor meanline program has been demonstrated. Some first results indicate the benefit of the resulting increase in model accuracy.

A number of applications benefit from a detailed compressor modeling within performance calculation.

- For liquid water entering the compressor a precise consideration of evaporation of the water across the compressor stages is imaginable. A change of the velocity triangles due to profile geometry changes caused by water film can be considered as well.
- The prediction of heat soakage and tip clearance effects during transient maneuvers can be improved knowing detailed interstage data. In addition, the knowledge about individual stage performance can help to identify the surge trigger.

The author intends to employ the presented methodology to optimise the VSV setting regarding engine efficiency at off-design conditions.

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