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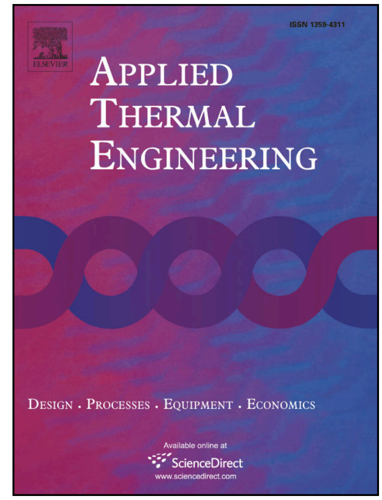

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PII: S1359-4311(19)34381-9
DOI: https://doi.org/10.1016/j.applthermaleng.2020.115087
Reference: ATE 115087

To appear in: *Applied Thermal Engineering*

Received Date: 25 June 2019
Accepted Date: 13 February 2020

Please cite this article as: R. Hackney, T. Nikolaidis, A. Pellegrini, A method for modelling compressor bleed in gas turbine analysis software, *Applied Thermal Engineering* (2020), doi: https://doi.org/10.1016/j.applthermaleng.2020.115087

Applied Thermal Engineering, Volume 172, May 2020, Article number 115087
DOI: 10.1016/j.applthermaleng.2020.115087

A method for modelling compressor bleed in gas turbine analysis software

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Abstract

The modelling of compressor interstage bleed in a gas turbine is required at all phases of the engine design cycle. The effect of the physical geometry of the bleed offtake on compressor flow is an important consideration, but of equal importance is the analysis of the effect of bleed air on compressor work requirements and overall gas turbine cycle efficiency. Knowledge of bleed air properties (temperature, pressure, work) is of paramount importance to carry out a reliable preliminary cycle analysis.

While interstage bleed modelling may be regarded as a small detail in the overall gas turbine cycle, it is of increasing importance in modern engine design due to the requirement to obtain performance optimisation in engines that have already had significant development. Consequently, even small improvements to the modelling process can yield beneficial improvements to the engine cycle.

Current methods to identify these effects vary in terms of complexity and accuracy; in this paper, a novel method is proposed, which offers an innovative - yet simple- way to simulate any number of interstage bleeds within a compressor, without the need to compromise on model accuracy. This is achieved through implementation of two methods - 1) properties of the bleed air are calculated by utilising the polytropic relationship between pressure and temperature, and 2) compressor work requirements are calculated by use of the "superposition principle" by implementing the work requirements of the bleed into an initial 'no bleed' calculation. This method has been implemented in the Cranfield modelling software Turbomatch, and validated against test data from an industrial gas turbine. Analyses so far show that the method is quick, accurate, and compares extremely well to the test data.

This novel, 'integrated' method has been shown to calculate interstage bleed properties in a gas turbine compressor without the need for complex modelling or artificial 'splitting' of components. This method is partially dependent on the assumption that stage pressure ratio can be estimated using constant stage temperature rise and compressor overall polytropic efficiency (although the use of overall isentropic efficiency was also investigated). Case studies performed

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on a number of Siemens industrial gas turbines suggest this is indeed the case, with calculated stage inlet/outlet pressure typically within 3% of the real engine comparison, although older technology compressors tend to be modelled less accurately than newer ones.

For compressor work and overall cycle calculation, comparison of this method against test data shows that accuracies between 0.5-1% can be obtained down to low load conditions. Compared to compressor 'splitting' methods, which demonstrated relatively poor accuracy levels (above 20%), likely due to inaccurate compressor map selection, the alternative method is shown to be a suitable method to model bleed without the need for extensive manual model adjustment. It is also shown how this method can be used to quickly assess the effect of bleed stage and mass flow on the compressor running line, as well as on the surge margin.

It should be noted that, while the engines used in the case study represent different engine designs and technology levels, they are from the same engine manufacturer and represent a gradual evolution of compressor technology using a similar design philosophy.

Keywords: Compressor, Performance, Modelling, Bleeds, Gas Turbine

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Nomenclature

Acronyms

CT Compressor Turbine
PT Power Turbine
DP Design Point
OD Off-Design Point

Symbols

Δh change in specific enthalpy between stations
δ Stage pressure rise
$\dot{m}$ mass flow
$\dot{W}$ Power
η compressor efficiency
γ ratio of specific heats
Π Compressor overall pressure ratio
π Stage pressure ratio
Π_{choked} Pressure ratio when choked
Π_{surge} Pressure ratio at surge
ζ Temperature rise per stage
C_p Specific heat capacity
H Total enthalpy
N Compressor speed
P Total pressure
T Total temperature

Z	Compressor surge characteristic

Subscripts (station numbering is based on ARP 755A)

2	Conditions at station 2 - Compressor inlet
21	Conditions at station 21 - Compressor bleed
3	Conditions at station 3 - Compressor exit
4	Conditions at station 4 - CT inlet
41	Conditions at station 41 - CT exit / PT inlet
5	Conditions at station 5 - PT exit
$fuel$	Conditions at fuel input into combustor
in	Conditions at inlet of component
n	Compressor stage number, where n=specified stage
n_{last}	Total compressor stage count
out	Conditions at exit of component
$poly$	polytropic

1. Introduction

The requirement for constantly increasing thermal efficiency and more stringent emissions legislation demand continuous improvement and development of gas turbine technology. Advances in material technology, computational methods, and general understanding are key enablers for such improvements. However, all stages of the development cycle incur considerable costs. It is therefore important that adequate modelling tools are available so that costs at these phases are minimised.

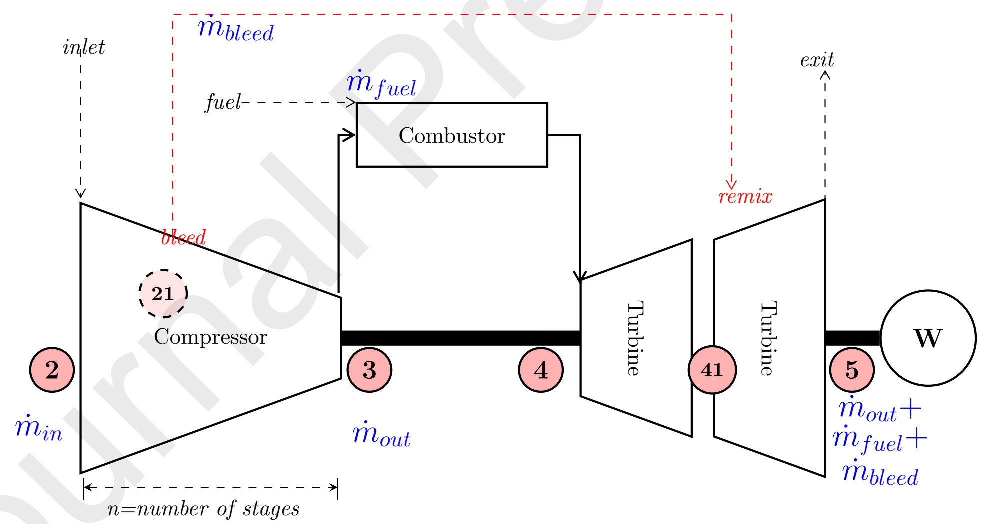
A model can be defined as an abstract, mathematical, representation of a system [1], and varies considerably in terms of detail depending on the tools used. Preliminary design and thermodynamic analysis stages tend to utilise *0-D* or *1D* modelling tools (such as component mapping or flow mean line analysis), where aerodynamic and mechanical design analysis will utilise *2-D* and *3-D* tools (such as CFD design tools). However, with such an increase in detail there is usually an associated increase in time to set up and run the models.

While product costs are typically very low during the initial design phase, the majority of the product cost is committed at these early stages, and hence the ability to reduce costs is also significantly higher than at later stages of the design process [2]. As the development of gas turbine components reaches higher and higher levels, the resulting further development margin is decreasing and has led to the consideration of factors not previously investigated in great detail [3]. One such area is the effect of secondary air systems on cycle efficiency. Secondary air systems extract some of the working fluid from the compression system to service other mechanical needs (such as turbine blade cooling or oil sealing). Generally referred to as 'bleed', the amount of secondary air can be as high as 30% of the total inlet mass flow, therefore can significantly affect the overall gas turbine performance.

1.1. Compressor Bleed

Compressor bleed air can be defined as a quantity of air which has been compressed to a specified stage of the gas turbine compressor for the use in secondary systems (such as cabin air systems, oil sealing or rotor thrust management), or for the purpose of improving start up surge margin. An engine schematic showing this arrangement is shown in figure 1. As shown, this air may be reintroduced into the gas path into components further downstream. As work has already been done on this air, bleed air represents an inefficiency in the overall gas turbine cycle. This inefficiency is manifested by both a reduction of useful air mass flow into the combustor and the work done on the bleed. However, one study [4] suggested that any performance loss can be reduced or even inhibited if the bleed location is selected such that the matching of the compressor and turbine results in a greater pressure ratio or compressor efficiency than in the no bleed case. Clearly, then, being able to model the effect of bleed air on the compressor and overall gas turbine cycle is a necessary requirement for modern design and development projects.

There are two primary aspects to consider when modelling bleed: 1) the effect of bleed on the overall gas turbine cycle, and 2) the thermodynamic properties of the bleed air itself (temperature, pressure, enthalpy, etc.).



Schematic of secondary air systems on a single spool industrial free power turbine engine with interstage bleed remixing between free power turbine and compressor turbine. Numbers represent key stations for thermodynamic calculations. *2=compressor inlet, 21=compressor interstage bleed, 3=compressor exit, 4=compressor turbine inlet, 41=compressor turbine exit, 5=free power turbine exit. W* represents the driven unit

Figure 1: Engine schematic with bleed

1.2. Current bleed modelling methods

A user may choose a range of methods to simulate bleed depending on the level of detail required, information available to the engineer and time available to produce results. At the most detailed level, a full CFD simulation of air flows, compressor geometry, and bleed off-take geometry may be conducted [5]. While such a method may yield detailed results, the time taken to produce the results, and the detailed component knowledge required means that such an exercise may be too resource intense for initial analyses. The use of such methods would also not model the effect on the overall gas turbine cycle, therefore subsequent cycle analyses should be carried out. Less resource intense methods, such as *compressor zooming*, which is defined as using *" . . . a higher order component analysis code . . . and the results from this analysis are used to adjust the zero-dimensional component performance characteristics within the [0-D] system simulation"*[6], may be utilised as they offer a balance between level of detail and resource requirements. However, such a method still requires a considerable level of knowledge of the compressor. Other numerical models may be employed to investigate the effect of bleed [7], and where only the design point compressor characteristics, compressor map, and some basic design features of the compressor (such as the total number of stages) are known, simpler and much less resource intense methods may be considered.

In current *0-D* design tools, a number of methods are utilised. Properties of the bleed air itself can be defined on the assumption that the bleed air shares the same properties (temperature and pressure) as the compressor exit air, or a simple linear extrapolation of temperature and pressure along the compressor stages can be made to define bleed air properties. From this, enthalpy and work may be calculated.

The effect of bleed on the overall gas turbine cycle is also modelled in different ways. Bleed air essentially changes the work requirements of the compressor. For example, the commercially available software *GasTurb* [8] uses a term called *relative enthalpy*, which is a fraction used to determine the enthalpy cost of the bleed based on the location in the compressor. Some methods do not consider the change of work at all (particularly if the bleed amount is small), while some define bleed in fixed locations along the compressor. The assumption that all bleeds take place at the compressor exit is also made by some applications. A potenially more accurate method is the artificial 'splitting' of a compressor into two or more pseudo-compressors, as used by Crafield University software *Turbomatch* [9], for example. The 'splits' are made at the points where bleed is off-taken. However, assumptions and 'educated guesses' must be made of the temperature and pressure at the pseudo compressor boundaries, and the user may even have to develop new compressor maps.

An alternate method is proposed in this study where any number of bleed off-takes may be defined at any stage of the compressor. Properties of the bleed air are estimated by utilising the polytropic relationship between pressure and temperature, and compressor work requirements are calculated by implementing the work requirements of the bleed into a pre 'no bleed' analysis cycle.

2. Theory

Where applicable the station numbering system in figure 1 is used to identify component-specific calculations.

2.1. Identification of bleed properties - flow, pressure and temperature

A calculation of temperature at the bleed offtake(s) is required in order to calculate bleed enthalpy and work, and the effect of remixing the bleed air back into the main gas path. However, as bleed air is often used for downstream secondary systems such as bearing sealing, turbine blade cooling, or cabin pressure air [10], knowledge of the pressure of bleed air is also required. Pressure and temperature at the bleed point may be calculated using a simple assumption that it is equal to the compressor exit temperature and pressure. Such an assumption, however, would introduce unacceptable errors in the cycle calculation, as any remixed air would have an artificially high enthalpy and would overestimate the contribution of the bleed air to useful work (this is discussed in more detail later). A more practical solution is to assume that the compressor stage pressure rise and temperature rise are linear and uniform through the compressor. In other words, temperature rise per stage may be expressed as:

$$\zeta = (T_3 - T_2)/n_{last} \tag{1}$$

and pressure rise per stage may be expressed as a function of inlet pressure and overall compressor pressure ratio:

$$\delta = (P_2 \times (\Pi - 1))/n_{last} \tag{2}$$

In order to establish whether the linear temperature and pressure rise is a reasonable assumption, an investigation of three industrial gas turbines of varying technology levels was undertaken. Shown in table 1 it can be seen that the oldest engine ('C') has a relatively low pressure ratio, lower polytropic efficiency and subsonic rotor tip relative velocities. This early engine was designed with the 'C4' style compressor rotor blades. The more modern Double Circular Arc (DCA) style-blades were used in some stages of engine 'B', which were capable of handling higher relative velocities (around Mach 1)[11], allowing transonic flow without excessive efficiency losses. This effectively resulted in a higher pressure ratio compressor for a lower number of stages, as well as higher efficiency. The more modern engine in the study ('A') benefited from further blade design improvements by using low aspect ratio stages (where aspect ratio is the ratio of blade height to chord length), and implementing Multiple Circular Arc (MCA) and Controlled-Diffusion Arc (CDA) blades in some stages, which were capable of even higher relative velocities and thus a further improvement in pressure ratio and efficiency.

Figure 2 shows normalised temperature and pressure rise for these engines for each stage of the compressor. It can be observed that temperature rise is broadly constant from stage to stage for all engines, and hence it is not unreasonable to assume a constant temperature rise per compressor stage. Pressure rise,

Engine Ref	Engine 'A'	Engine 'B'	Engine 'C'
Approx. Introduction	2000	1998	1982
Technology level assumed	MCA/DCA/CDA/C4	DCA/C4	C4
Inlet Velocity	Transonic	Transonic	Subsonic
Blade stages	11	10	15
Speed (100%) [rpm]	14300	17384	11085
Pressure ratio	1.000	0.836	0.702
Overall polytropic efficiency	1.003	0.997	0.968

Table 1: Engine details. Note that pressure ratio and overall polytropic efficiency are expressed non-dimensionally as a function of the highest respective value of the engines compared

however, is not constant nor linear. Normalised polytropic efficiency (shown as 'Poly' on figure 2) is observed to be broadly constant across compressor stages, although the older technology engines with "C4" technology appear to have a decline in polytropic efficiency towards the later stages.

By assuming constant stage temperature rise and constant polytropic efficiency, stage pressure rise may be calculated. The polytropic relationship between pressure and temperature is a well known thermodynamic principle, and is shown in equation 3 [12]:

$$T_{out} - T_{in} = T_{in} \times \left[\pi^{\left(\dfrac{\gamma - 1}{\gamma \times \eta_{poly}} \right)} - 1 \right] \tag{3}$$

From equation 3 it can be observed that, since the aim is to have constant $(T_{out} - T_{in})$ and constant η_{poly} in each stage (figure 2), T_{in} will increase moving from the first to the last stage, which means that stage pressure ratio (π) has to decrease moving from first to last stage. It is well known [13] that, for a fixed polytopic efficiency, isentropic efficiency increases when pressure ratio reduces. This proves that assuming constant isentropic efficiency across the stages would be inconsistent with the experimental data presented in figure 2 and is, therefore, not acceptable. Polytropic efficiency is therefore shown to be a more appropriate expression to use when assuming constant stage efficiency. Equation 3 may therefore be re-arranged in terms of stage pressure ratio:

$$\pi_n = \left[1 + \left(\frac{\zeta}{T_{in}} \right) \right]^{\eta_{poly} \left(\dfrac{\gamma}{\gamma - 1} \right)} \tag{4}$$

Equation 4 can hence be used to calculate the pressure ratio of each stage for given values of temperature rise per stage, η_{poly}, and stage inlet temperature

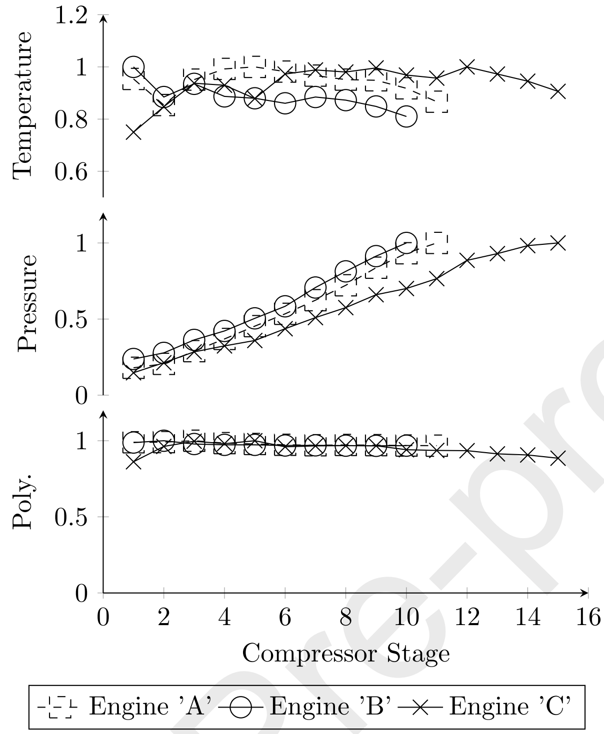


Figure 2: Pressure and Temperature rise per stage. Values are normalised to the peak value for each engine

(T_{in}). Figure 3 shows the comparison between: real engine data (as shown in figure 2), results obtained by applying the methodology just described, and results obtained by assuming a constant pressure rise per stage. The results suggest that the proposed method is accurate (within 3% of real engine data), and can be trusted for the calculation of bleed pressure levels.

2.2. Compressor power and overall cycle change

Considering a case where there is no compressor bleed, the power required to drive the compressor can be defined as the change in specific enthalpy between compressor inlet and exit multiplied by the mass-flow of air flowing in the compressor, as shown in equation 5.

$$\dot{W}_{3-2} = \dot{m}_1 C_p (T_3 - T_2) \tag{5}$$

The power for the compressor is provided by the compressor turbine. Turbine power is calculated using the same principle as compressor power, although in the case of a single spool turbine (i.e. power generation unit) the power produced by the turbine is equal to compressor power plus useful output power.

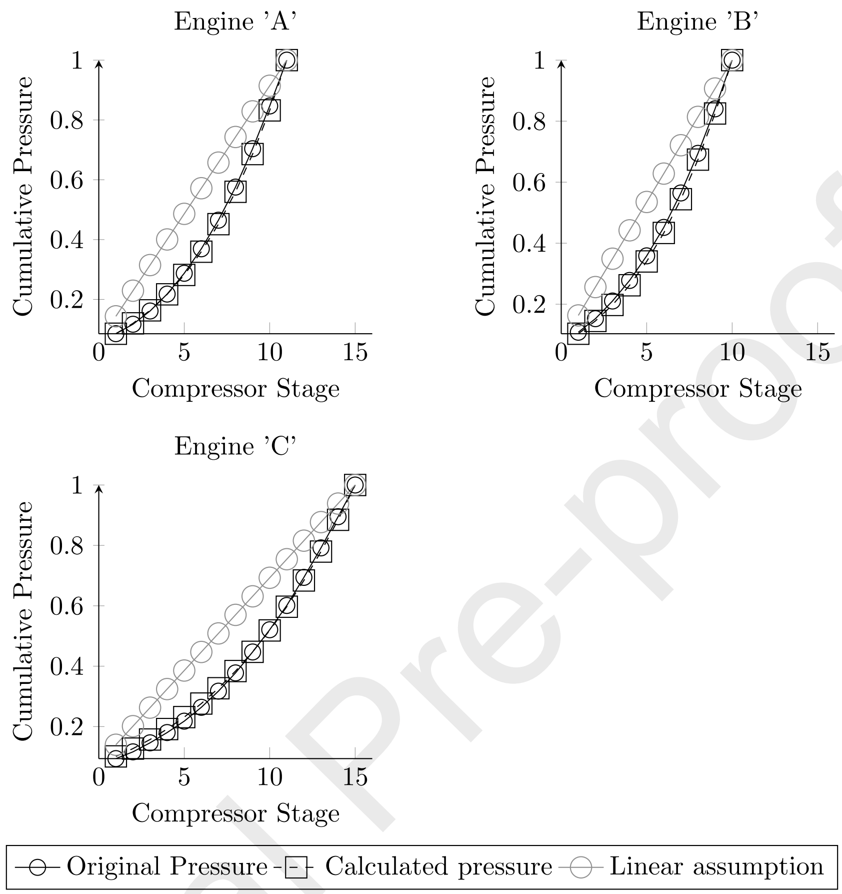


Figure 3: Pressure rise per stage comparison. Values are normalised to maximum pressure for each engine

In reality, there are a number of losses to consider within the compressor (e.g. bearing losses or windage losses), but for simplicity these are ignored

$$\dot{W}_{41-4} = \dot{W}_{3-2} + losses \tag{6}$$

2.3. Compressor work change

The first calculation of compressor power ignores the presence of any bleed. In reality, the reduction in compressor air mass-flow due to a bleed offtake will result in a change of power absorbed by the compressor. As bleed represents a reduction of flow from the compressor, and hence a reduction in the amount of mass flow which has work done on it, the change in compressor work can be calculated by calculating the work which *would have been done* on the bleed air, had it remained part of the main gas path, and subtracting it from the compressor power previously calculated (by ignoring the presence of the bleed). In summary, for bleed quantity $\dot{m}_{21}$ at stage n:

1. Work is still done on $\dot{m}_{21}$ from compressor stages 1 to n
2. Work is no longer done on $\dot{m}_{21}$ from compressor stage $n+1$ to the exit of the compressor
3. The compressor power (8) can be defined as *the original power (5) minus the work per unit massflow which is no longer done on the bleed air (7).*

$$\dot{W}_{3-21} = \dot{m}_{21} C_p (T_3 - T_{21}) \tag{7}$$

$$\dot{W}_{3-2} = [\dot{m}_2 C_p (T_3 - T_2)] - [\dot{m}_{21} C_p (T_3 - T_{21})] \tag{8}$$

This revised value of power and massflow is then passed to the modelling software and the next iteration can occur. This will then result in a change to the matching conditions of the engine and hence different compressor exit temperature and pressure, and overall cycle performance.

Equation 8 shows that bleed air actually *reduces* the power absorbed by the compressor. It therefore stands to reason that the lower the value of n, the greater the impact from not doing work on the bleed air. Note that in terms of *overall performance* of the gas turbine, the reduction in compressor work is clearly offset by the reduction in useful mass flow through the combustor (this is true only if the total inlet mass-flow to the compressor is assumed constant). This leads to the theory that the closer to the outlet section of the compressor the bleed is, the less 'impact" from not having to do work on it (compared to the baseline case with no bleed), and conversely the more bleed at bleed stage n, the greater the reduction in compressor work required.

2.4. Cost of bleed

The compressor cannot be considered an isolated system, and ultimately the effect of bleed is that less air is provided to the combustion system and hence less energy introduced into the flow (assuming constant combustor outlet temperature), even when considering bleed remixing in later stages of the turbine. This must result in a net loss of performance, as can be demonstrated by considering a (simplified) Brayton Cycle. Shown in figure 4, the main gas path air follows the typical real process *1-2-3-4* (*1-2'-3-4'* representing the ideal cycle). The bleed air will follow the same real compression process as the main gas path from *1-2* until it reaches the stage where the bleed air is offtaken (represented by *1B'* in figure 4). There is no combustion of this bleed air (so no equivalent of stages *2-3*), but there will be expansion upon remixing into the turbine (represented by *1B"*). It can be seen that $\Delta(1 - 1B') > \Delta(1B' - 1B'')$, thus showing that bleed offtake will always result in a net loss of overall efficiency compared to a no-bleed case, and hence a net loss of shaft output power.

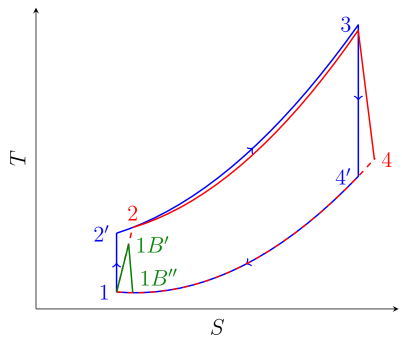


Figure 4: The Brayton cycle, including bleed

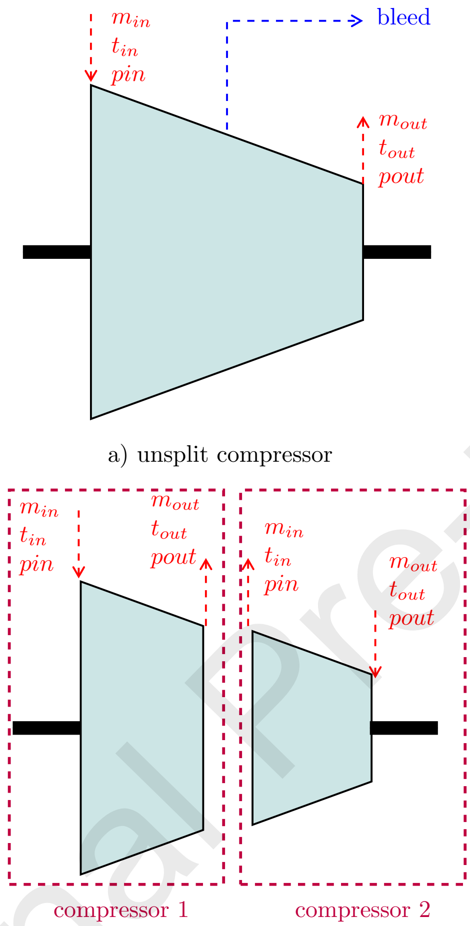
So, in summary, by using the theory presented in section 2.1 to obtain compressor bleed temperature (the calculation of bleed pressure, while a very important consideration for secondary air systems, has no role in the work calculations), and the methods above to calculate the change in compressor work, the effect of bleed on the overall gas turbine cycle may be investigated.

3. Application

Using the modelling software *Turbomatch*, the effect of bleeds on the gas turbine cycle has been investigated. Turbomatch is a software package designed by Cranfield University to simulate steady state (design and off-design point) performance of a gas turbine. Gas turbine performance is calculated by using scaled component characteristic maps for compressors, combustion chambers, turbines (both compressor turbines and free turbines) and a map providing the velocity coefficient for exhaust nozzles. Off-design point performance is determined by using a modified Newton-Raphson method [9]. The engine matching criteria for off-design calculations consist of mass and energy balance.

An initial validation exercise was carried out, modelling bleed by 'splitting' the compressor. 'Splitting' the compressor divides the compressor into a number of sub-compressors according to the location of the interstage bleed(s), as shown in figure 5. The operator must then determine the design point temperature and pressure for each sub-compressor, as well as selecting a relevant non-dimensional map for each sub-compressor. Design point polytropic efficiency must also be used over isentropic efficiency, as isentropic efficiency is highly stage-count dependent. If accurate sub-compressor design point conditions are selected then this method can yield accurate results, but it requires significant manual adjustments, and a very careful selection of the compressor maps used for each "section".

Following validation, the alternative method was implemented in Turbomatch. Test data from an industrial small gas turbine with a single-spool, free power-turbine arrangement was selected for the study (detailed in table 2) . This engine has undergone a number of design modifications over the years to



a) unsplit compressor

b) split into two artificial compressors

Figure 5: Splitting of the compressor

improve efficiency, lifing and emissions. Secondary air system flows are based on engine flow models.

3.1. Effect of varying bleed

Following validation of the new code, two test cases were run at design point conditions:

1. **"Varying Stage"**. Varying bleed stage between 1 and 11, using the same amount of bleed (2%) each time. In this case, compressor exit mass flow and bleed mass flow are constant.

Parameter	**Details**
Power	13.40MW (mechanical power)
Efficiency (thermal)	36.2% (mechanical)
Compresor stages	11
Compressor inlet flow	38.9kg/s
Compressor stator blade modulation	Variable guide vanes
Comp. nominal speed	14100rpm
Comp. pressure ratio	16.8:1
Combustion	Dry-Low Emissions system
Compressor Turbine	2-stage air-cooled turbine
Power Turbine	2-stage high efficiency turbine
Exhaust temperature	555deg. C

Table 2: Engine selected for study

2. **"Varying Flow"**. Varying bleed amount between 0 and 10% at a fixed stage (stage 6). In this case, the temperature and pressure at the fixed bleed stage are constant.

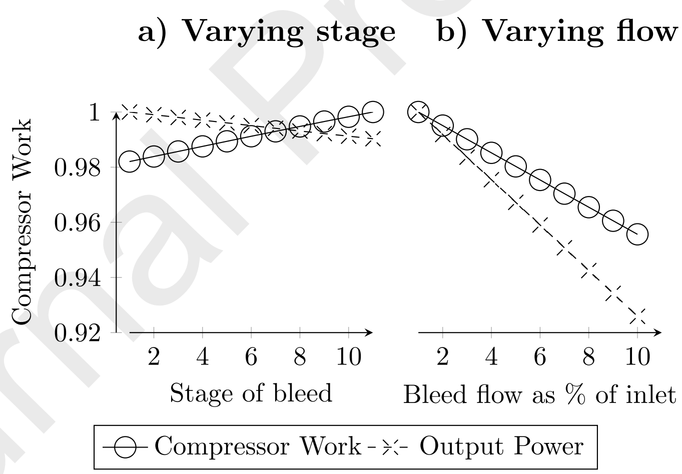


Figure 6: Comparisson of effect of varying bleed amount and location. Values are normalised to max parameter value.

As shown in figure 6, shaft output power is reduced compared to no-bleed for both cases. However, the power loss is clearly greater in the varying amount case, despite the fact that there is a net reduction in compressor work. For the varying stage case it can be seen that, despite the same amount of air being bled for each calculation (and hence no change to net mass flow), shaft output power reduces with increasing stage.

3.2. Comparison to other methods

While section 2.1 shows that the method to derive stage pressure and temperature is sound, and section 3 shows the effect of bleed on gas turbine cycle using the new method, neither prove the accuracy of the method, nor offer a comparison to other methods. Therefore a comparison at design conditions with varying bleed amount has been made against two other methods:

1. **"Method 1. All bleed at compressor exit"**. This method uses the assumption that all bleed air is extracted at compressor exit. No assumptions of bleed temperature and pressure are required, but conversely this means that these values will be highly inaccurate when downstream remixing is introduced (this may or may not be an issue depending on what the bleed air is used for). While this method does ensure that the bleed air is not part of the main gas path into the combustor, it makes no assumption of the change in work requirements for the compressor.
2. **"Method 2. Splitting of the compressor"**. As discussed in section 3, this method 'splits' the compressor into a number of sub compressors using a previous compressor model and dividing it into a number of sections determined by the interstage bleed locations. The biggest limitation of this approach is that, in order to correctly model different stages of a given compressor, realistic maps for the initial, intermediate, and final stages should be used. This is almost never the case, and assuming that the map shapes for these three "sections" of the compressor are the same, is a very rough approximation that may produce inaccurate results in off-design calculations (where the shape of the maps determines the behaviour of the engine)
3. **Method 3. "New integrated method"**. This method consists of the practical application of the theory proposed within this report. It offers the improved accuracy of the 'split' method without the need for the associated manual calculations and assumptions (especially choice of map assumptions) and allows more correct use of the component maps and modelling of the bleeds impact on the compressor and overall engine performance. Furthermore, it uses the polytropic efficiency derived pressure ratio.

Each of the three methods have been compared in *Turbomatch.* Figure 7 shows the effect of varying the amount of interstage bleed at constant combustor outlet temperature. It can be observed that method 1 is significantly different to the other two cases. This suggests that its use as a modelling assumption is limited. Methods 2 and 3, however, show good agreement with each other. Bleed stage temperature and pressure for these methods has been derived using the same method described in section 2.1). In terms of shaft output power *(a)*, both the" new integrated method" and "splitting" method remain within $\pm$ 0.2% of each other. The 'all bleed at compressor exit" method, however, deviates by as much as 5% from the other two methods. This is a trend reflected in the fuel flow plot *(b)*.

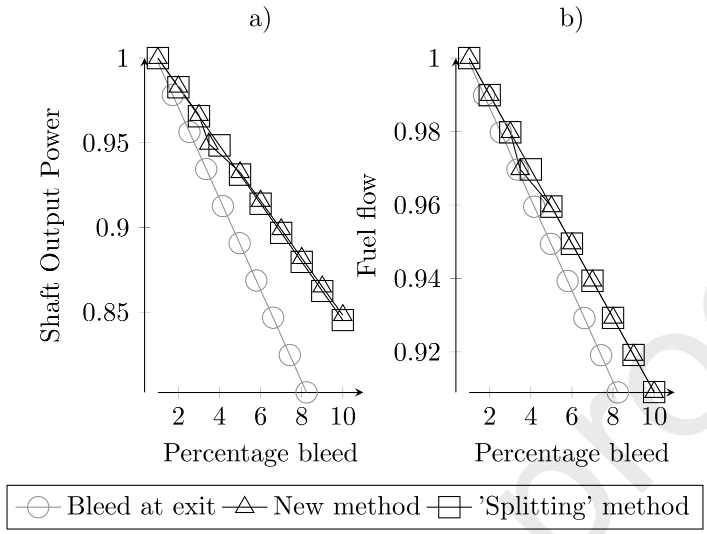
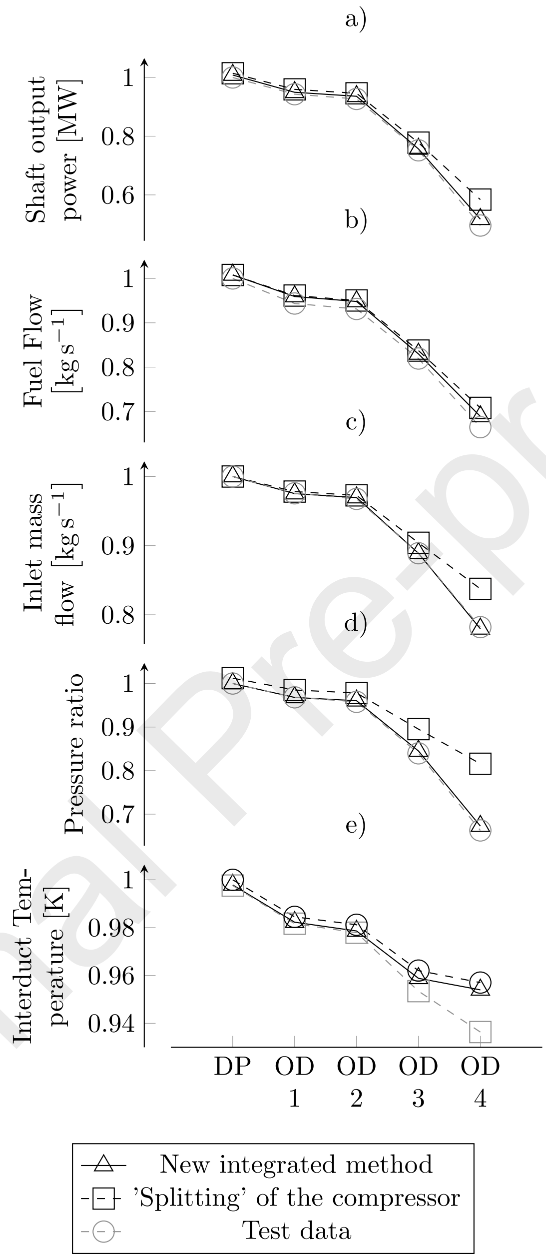


Figure 7: Effect of modifying bleed from 2% to 10%.

Analysis so far has been performed at design conditions. Discarding the method 1 (as this had a relatively poor agreement with the other two methods), the two remaining methods have been run over an engine running line and compared to test data from a small industrial gas turbine. Results taken from an engine factory test over varying load (shown in figure 8) are compared to the two methods above. While the ambient temperature and pressure were approximately 10^oC and $0.996bar$, results have been normalised to standard conditions of 15^oC and $1.013bar$ using ISO correction factors detailed in *BS EN ISO2314* [14].

The results of this comparison, shown in figure 8, show that at the design point the two methods agree well with the test data, but as load is reduced, the 'split' method data deviates significantly from test data. For all parameters aside from fuel flow, the alternative method is within ± 1% of test data for the design point and all but the final off-design point. Fuel flow is typically within 1.5% of test values for all but the final off-design point. For all parameters, the fourth off-design conditions shows a weaker agreement, with 4% difference in power, 4% in fuel flow, and 1.5% difference in pressure ratio. Air mass flow, however, remains within 0.5% for all conditions. The split method shows similar agreement with the alternative method for the design-point and the first off-design condition. However, agreement with test data reduces for subsequent off-design conditions. The difference in output power is over 3% at OD3, and more than 17% at OD4. Similar trends can be seen for the fuel flow and pressure ratio differences



DP= Design Point T_4
OD1 = (design point T_4 - 24K), No bleed (97% load)
OD2 = (design point T_4 - 30K), No bleed (93% load)
OD3 = (design point T_4 - 56K), 3.1% bleed (75% load)
OD4 = (design point T_4 - 48K), 11.5% bleed (48% load)

Figure 8: TurboMatch vs. Test data at OD conditions

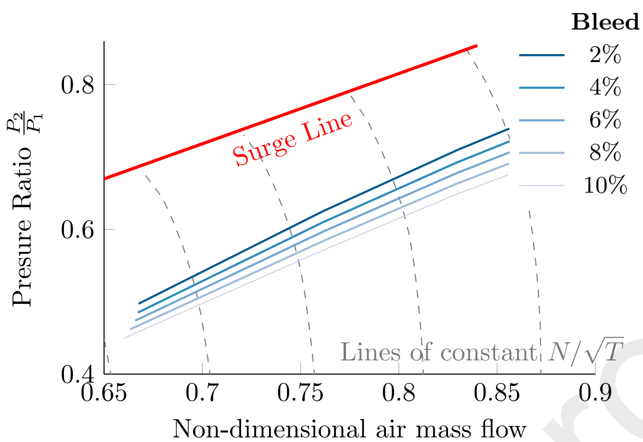


Figure 9: Compressor Map for varying bleed amount at stage 9 (values are normalised)

3.3. Analysis of effect of bleed on compressor running line

Figure 9 shows the compressor running line with varying amount of bleed from 2% to 10%, whilst figure 10 shows the running line with changing bleed stage from stages 2 to 10, but at a constant bleed flow of 5%. Note that all running lines are analysed using the same combustor outlet temperature range. Also shown on these plots are the surge characteristics of the data points. The surge margin parameter, Z, is a measure of how close the data point is to the pressure ratio at which surge will occur. It is expressed as a ratio of current data point to surge line:

$$Z = \frac{\Pi - \Pi_{choke}}{(\Pi_{surge} - \Pi choke)} \tag{9}$$

Thus the higher the value of surge margin parameter, the less margin there is to surge.

For increasing bleed (figure 9), it can be seen that the running line moves *down* relative to to the nominal case. It is shown that for a given inlet air mass flow, pressure ratio is reduced. Conversely, for increasing bleed stage but constant amount of bleed fixed (figure 10), the running line moves to the left. For a given non-dimensional mass flow, increasing the bleed stage has the effect of reducing surge margin at low running conditions, but having less effect near design point conditions.

4. Discussion

4.1. Method of estimating bleed stage pressure, temperature and work

Using an assumption of linear temperature rise and constant polytropic efficiency, and applying the polytropic relationship between pressure and temperature, an estimation of bleed stage compressor characteristics can be made.

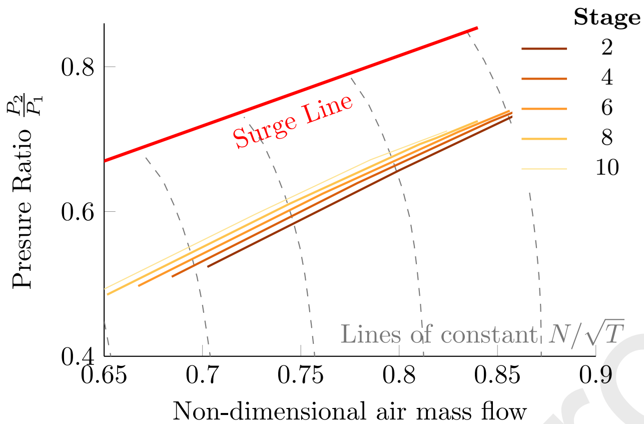


Figure 10: Compressor Map for varying stage using fixed bleed of 5% (values are normalised)

It has been shown in figure 3 that this method yields accurate results when compared to data from a number of engines, each representing different technological development levels. While it is outside the scope of this study to define the maximum acceptable error, it is shown that this method produces much more accurate predictions of pressure than a simple linear pressure rise calculation (within 3% of the baseline value). Ultimately, an assessment of the accuracy will be made depending on the required use of the data and precision requirements.

It should be noted, however, that while a number of different engines with different technology levels were used in this study, they are all from the same manufacturer. It would be beneficial to conduct a similar exercise on an engine from a different manufacturer. Another consideration is that in order to take advantage of the methods used in this study, suitable consideration must be made to the introduction of any bleed flows back into the main gas path downstream of the compressor.

4.2. Effect of bleed on engine cycle at design point conditions

The effect of bleed on engine cycle was investigated in two ways. Firstly, by varying bleed stage between 1 and 11, but using the same amount of bleed each time. Secondly, by varying bleed amount between 0 and 10% at a fixed stage. According to the theory, both should result in net loss of shaft output power. Furthermore, if reasonable ranges of stage variation (mid-stage to near-last stage) and bleed amount (1% to 30%) are assumed, varying the amount of bleed flow should be more detrimental than varying the stage. This is clearly shown in figure 6.

For both cases, shaft output power is reduced compared to the no-bleed case. However, the shaft output power reduction is clearly greater in the varying flow case, despite the fact that there is a net reduction in compressor work (less flow means less work required to achieve the previous pressure ratio). This is because the more flow that is used as bleed, less flow is available in the combustor. Hence for a given temperature, the lower the fuel input and hence the lower the net heat input.

For the varying stage case it can be seen that, despite the same amount of air being bled for each calculation (and hence no change to net mass flow), shaft output power reduces with increasing stage. This is because the later the stage air is bled from, the more work is done to that air, and hence net compressor work requirements increase with increasing stage. Increased net compressor work requirements mean less useful power available to drive the turbine, and hence a net loss in shaft output power.

Overall, the results are consistent with the theories stated in section 2. It has been shown that bleed affects compressor work requirements and the mass flow through the combustor, and due to both of these factors, ultimately affects the gas turbine cycle, output power and efficiency.

4.3. Comparison of new method to current methods and test data at off-design conditions

After discarding the method of assuming all bleed is at the compressor exit due to the relative inaccuracies observed, the new integrated method and the 'split' compressor methods were compared to test data (figure 8). Results show that at design condition both methods were broadly in agreement with test data, but as the engine moves away from design point, the 'splitting' method deviates significantly from the test data, which may be due to incorrect selection of compressor maps (it should also be noted that generic scaled maps were used for all modelling within this paper). However, this does show a weakness with this method - the accuracy is dependent on the user performing extensive manual adjustment to the model to create the 'split'. Design point temperatures and pressures at each sub-compressor must be manually calculated (and if simple linear assumptions are made for pressure then model accuracy will be further reduced), and the user must also ensure that the maps describing each sub compressor are representative of the component to be modelled.

The integrated method, however, demonstrated satisfactory agreement with the test data across the operating range of the engine. This suggests that this method offers a viable alternative to other modelling methods, whilst not increasing the complexity of the modelling tools.

4.4. Effect of bleed on compressor running line

By increasing the amount of bleed (figure 9) it can be seen that the compressor running line moves downwards. As more air is bled from the compressor, there is less air available for combustion, and hence for the same firing temperature there will be less energy available for expansion through the turbine. This means that the turbine does not extract as much energy from the gas path as before, and hence the compressor is not able to deliver such a high pressure ratio (compressor speed will hence reduce). However, the effect of this 'rematching' between the compressor and turbine results in an *increased* surge margin for a given load, which is consistent across the running line.

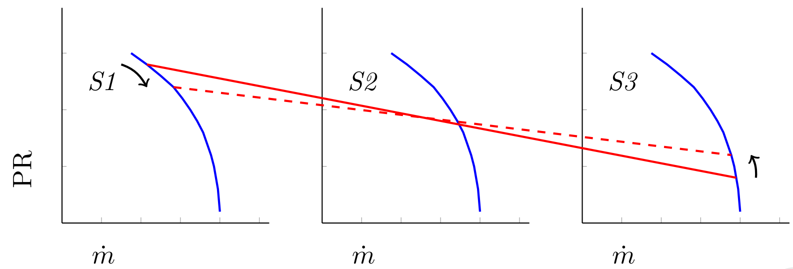


Figure 11: Representation of change in running line

By increasing the bleed stage (figure 10), the running line is observed to move to the left. It is likely that the turbine and compressor match at a lower corrected rotational speed and inlet mass-flow, and hence the running line is moved to the left. Air mass flow at compressor outlet is constant in this case, so there is no penalty though the combustion process. Instead, bleeding at later stages does unnecessary work on the bleed air (unnecessary form the point of view of the engine cycle - not necessarily secondary air requirements) that effectively increases work requirements in the compressor. Increasing the compressor work requirements reduces the useful work available to the cycle, and hence reduces output power.

While changes to the *overall* running line can be identified using this method, care must be taken to ensure that individual blade stages are not pushed into surge regions by such a change in the running line. Reducing the pressure/flow relationship for one stage may result in an increase in pressure/flow relationship for later stages due to the way the compressor matches. This is demonstrated in figure 11, where the non-dimensional flow/pressure relationship for three stages of a compressor are shown. By introducing bleed off-take at stage 1 (*S1*), the stage surge margion is improved (solid red line to dashed red line), but for the same operating condition this effectively pushes later stages towards surge, as seen for stage 3 (*S3*).

5. Conclusion

A method has been proposed which offers an alternative way simulating interstage bleed within a compressor without the need to compromise on model accuracy. This is achieved through two assumptions. 1) constant stage temperature rise; and 2) compressor and gas turbine cycle changes due to bleed can be calculated according to the "superposition principle" (i.e. the assumption that the work done on bleed air m at stage n, plus the work that *would have been done* had the flow remained part of the gas path, is equal to the work done by the same amount of massflow without any bleed offtake). This method has been implemented in the Cranfield modelling software *Turbomatch*, which itself has been validated against test data from an industrial engine. Analysis so far show the method is quick, accurate, and compares well to test data.

The new integrated method was proven to allow for the simulation of interstage bleed in a gas turbine compressor without the need for complex modelling or artificial 'splitting' of components. Furthermore, it was demonstrated that this method offers improved accuracy over the 'splitting' method, which is prone to errors due to inaccurate selection of the compressor map(s). This is the biggest limitation of the 'splitting' approach - in order to correctly model different stages of a given compressor, realistic maps for the initial, intermediate, and final stages should be used. This is almost never the case, and assuming that the map shapes for these three "sections" of the compressor are the same, is a very rough approximation that may produce inaccurate results in off-design calculations (where the shape of the maps determines the behaviour of the engine). The alternative method does not require the selection of new maps, and it has been shown that this produces more accurate results at off-design conditions.

The alternative method is partially dependent on the assumption that stage pressure ratio can be estimated using constant temperature rise per stage and constant polytropic efficiency. Case studies performed on a number of Siemens industrial gas turbines suggest this is indeed the case, although older technology compressors tend to be modelled less accurately than newer ones. Comparison of this method against a compressor 'splitting' methods showed that comparable results can be obtained without the need for extensive manual model adjustment that the splitting method requires. However, it should be noted that while the engines used in this case study represent different engine designs and technology levels, they are from the same manufacturer and represent a gradual evolution of compressor technology using a similar design philosophy.

Further study is recommended, including investigating the use on other engine types, and improvements to the remix modelling following the bleed re-injection into the main gas path. While the method discussed has been based on steady state simulations and engine data, in principal it could be applied to transient events. it is recommended that this is further investigated.

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Declaration of interests

☒ The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

☐The authors declare the following financial interests/personal relationships which may be considered as potential competing interests:

- Compressor bleed influences gas turbine cycle efficiency
- Linear pressure rise assumption in an axial compressor is not accurate
- Compressor work can be calculated by use of the ”superposition principle”
- Accuracy within 3% of real engine data compared to other methods

2020-02-17

A method for modelling compressor bleed in gas turbine analysis software

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Elsevier