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Generating Physically Accurate Compressor Maps

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ABSTRACT

The accuracy of gas turbine performance predictions depends on the quality of the compressor maps used. These maps can originate from measurements, calculations, or from literature. The most accurate data comes from measurements taken on a compressor rig. This data must undergo preprocessing before it can be used in a performance program. In this process, the unavoidable scatter in the given data, the uneven distribution of the measured points, and the occasional scarcity of points for a given speed line are difficult to cope with.

Ideally, one would automatically and unambiguously generate the map from the measured data using specialized software. However, creating such software is not a trivial task. Simply drawing a smooth line through the given data will not produce an acceptable result. Therefore, a semi-automatic approach is often used, in which an experienced engineer manually corrects a draft created by the computer. The disadvantage of this approach is that the result depends on the person doing the job.

This paper describes the challenges of creating a compressor table generator that adheres to compressor physics. The software must account for the type of compressor (e.g., axial or radial; single-stage or multi-stage; with or without variable guide vanes), as each type has unique characteristics. The Mach number level must also be considered because it affects the shape of correlations between pressure ratio, speed, mass flow, efficiency, torque, specific work, and loss descriptors, for example. Many of these correlations have a shape that can be derived from compressor theory.

This paper does not propose a specific methodology for automatically generating compressor map tables. Instead, it presents the requirements that such software would need to meet. One of these requirements involves checking the shape and slope of a specific torque-mass flow correlation. As demonstrated by comparisons with measured data from seven very different compressors, this correlation is applicable to every compressor map.

INTRODUCTION

The quality of off-design gas turbine performance simulations using programs such as NPSS, PROOSIS, GSP, and GasTurb, depends on the quality of the compressor maps. Many researchers have recognized this, resulting in numerous publications about compressor map generation.

Publications of compressor maps often do not show measured data; instead, they show one what has been derived by post-processing rig data. Such maps consist of smooth speed lines in the pressure ratio – mass flow plane complemented with contour lines for efficiency. Some publications aim to represent the map using purely mathematical algorithms, such as polynomials. Zagorowska (2017) concludes that Chebyshev polynomials allow the map to be reconstructed accurately from fewer points compared to third order polynomials. Shuang et al. (2011) propagate the use of Bezier lines and Tsoutsanis et al. (2012) model the speed lines with segments of elliptic curves in the pressure ratio – mass flow plane which go down to zero pressure ratio. The efficiency contours are expressed by the cutaway of a 3D ellipsoid in the pressure ratio – mass flow plane. It is assumed that each efficiency line belongs to an elliptic curve.

Tsoutsanis et al. (2014) have developed this modelling method further to simultaneously determine the best elliptical curves to a set of compressor map data. The coefficients that determine the shape of compressor maps' curves are analysed and tuned through a multi-objective optimization algorithm to meet the targeted set of measurements.

Li et al. (2018) propagate the partial least squares (PLS) regression method. They claim as advantage that it is strong and robust, requires a low sample number and has physical meaning. In their model the basic function of the PLS, a power function polynomial, is replaced by a trigonometric function.

The use of neural networks for representing compressor maps is popular in the scientific community. The paper of Palmé et al. (2012) presents a study where artificial neural networks are used as a curve fitting method applying measured data from an axial compressor test rig to predict the compressor map. Another example is the paper of Gholamrezaei and Ghorbanian (2010).

All neural networks have one thing in common: they require a large amount of data to be sufficiently trained to achieve highly predictive accuracy and stable calculations. Where do these data come from? In the example presented by Palmé et al., approximately 50 measurements were made for each speed line. This is more than five times the number of measurements usually taken on a compressor rig. Li et al. (2018) state that neural networks are not suitable for compressor map modelling when the number of experimental data is very limited.

In the introduction to their paper, Misté and Benini (2013) give an extensive overview of various methods for adapting maps to given data. They acknowledge that differences arise between the maps related to centrifugal and axial turbomachines, as well as between low- and high-pressure ratio compressors of the same type, and these make it difficult to find a general regression model that is valid for each existing compressor. Their map adaptation methodology uses mathematical optimization with constraints which consider physics-based limits for the shape of speed lines and efficiency contours.

All the compressor map generation methodologies mentioned above focus solely on the common graphical representation of compressor performance as a plot of pressure ratio versus mass flow rate at different speeds, with contours of efficiency levels. The quality checks are usually based on only a single map which is taken as a perfect representation of the compressor performance. The possibility of outliers in the original data is not discussed.

All of the compressor map generation methodologies use complex mathematical correlations with many input parameters. These parameters must be determined before a map can be used in a performance calculation program. Using look-up tables is much faster and simpler than any of these methods. These tables use an auxiliary coordinate called β when performing gas turbine performance calculations, as described by Kurzke, Halliwell, and Hill (2025).

I know of no fully automatic methodology which can generate accurate lookup tables from a limited amount of measured data (with outliers), works for single and multistage, axial and radial, with or without variable guide vanes and can consider interstage bleed without human interference.

The program Smooth C developed by the author (Kurzke 1998, 2011, 2019) can handle this task but relies on the expertise of the user to create a high-quality smooth map fully in line with the laws of physics. Parabolic β lines are used as auxiliary coordinates because they can be shaped to represent lines of constant work coefficient. This is feasible, at least in the low Mach number region of a compressor map.

This paper presents maps of various compressor types that were processed using an experimental version of the Smooth C program. It focuses on physics-based rules which can be applied to adapt the smooth lines so that they optimally pass through the scattered data. These examples illustrate the challenge of creating software that can automatically generate accurate look-up tables describing compressor performance without human intervention.

METHODOLOGY

Our starting point is that we have measured data on pressure ratio, mass flow, and efficiency from a compressor test rig. All the data is affected by measurement noise, and points near the surge line often have greater uncertainty. The distance between measured points on the same speed line is random. Surge points are available for some or all speed lines.

The task is to create look-up tables that describe the compressor's performance accurately. In GasTurb, for example, a compressor map consists of tables for mass flow, pressure ratio and efficiency as function of speed and the auxiliary coordinate β .

According to the laws of compressor physics, any change in properties along a speed line must be smooth. The user is guided through the process of smoothing the map with the aid of numerous graphs that are rooted in those fundamental principles.

Mathematical smoothing methods

Drawing a smooth line through a bunch of scattered data is a standard mathematical task. We could use simple polynomials for that or so-called Chebyshev polynomials which have some advantages.

Figure 1 shows what happens when in Smooth C Chebyshev polynomials are applied to the data for pressure ratio, mass flow and efficiency published by Suder (2011). On the left side is the generated map shown in the conventional format. One can see immediately the polynomials match to the measured data quite well in the lower speed range. However, in the upper speed range, where the speed lines are vertical, the match between the smooth lines and the measured data is unsatisfactory.

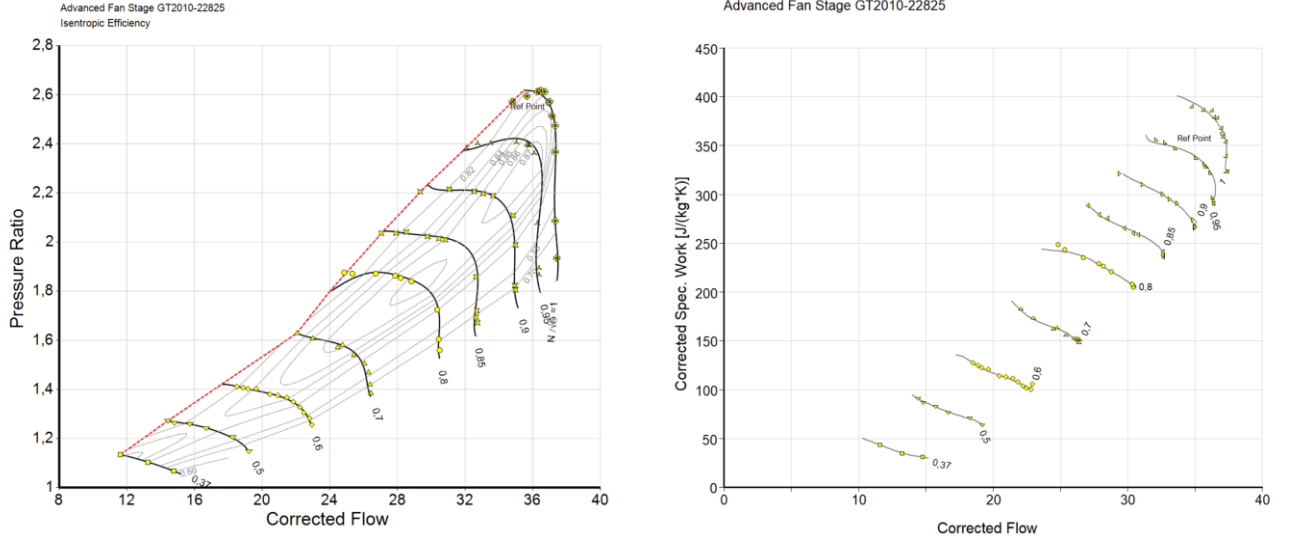


Figure 1: A compressor map that was smoothed using 5th-order Chebyshev polynomials.

Other mathematical procedures could be applied to these data with a seemingly more satisfactory result. At the first glance the efficiency contour lines look reasonable, but, that is not important. Efficiency is not the driving property in the gas turbine performance program; rather, it is the power needed to drive the compressor, which is the product of mass flow ($\dot{W}$) and specific work (H). In turn, H is a function of pressure ratio and efficiency. While we cannot make statements about how efficiency changes along a speed line, we can derive how specific work changes with mass flow and pressure ratio from theory.

Physics based smoothing

A compressor map has two distinctly different regions. At low spool speeds and mass flows, the compressibility effects in the flow field are minimal, so incompressible compressor flow theory can be applied. At high spool speeds and mass flows, transonic or supersonic flow exists, which leads to choked flow and the speed lines become vertical.

Incompressible flow region

The right side of Figure 1 shows how the smooth lines, which are purely mathematical in nature, align with the measured specific work data. The shape of the $H=f(W)$ lines looks peculiar not only in the upper speed range, but also nearly everywhere else on the map. However, we can derive from first principles how a line $H=f(W)$ must look like.

Let us consider incompressible compressor theory and look at the correlation between work coefficient $\psi=H/U^2$ and flow coefficient $\Phi=\dot{V}/U$. In theory, the correlation between ψ and Φ is independent from circumferential speed U .

If we consider the form of the ψ - Φ relationship for a single stage compressor with symmetrical velocity triangles in simple terms, we can conclude from Figure 2 that ψ is a linear function of Φ . This follows from the fact that the flow leaves a blade or vane row in the direction given by the trailing edge geometry.

$$\psi = 1 - \Phi * (\tan \alpha_1 + \tan \beta_2) = 1 - c_1 * \Phi \quad (1)$$

Work input H done on the air by the compressor is represented by a straight line when plotted as Ψ against Φ . The only prerequisite for the validity of Equation 1 is that the flow direction downstream of the blades and vanes is determined by their geometry. In other words, the rotor exit flow direction is determined by the blade exit metal angle.

It is nice to know that $H=f(W)$ must be a straight line in at least a part of the map. However, we can derive more useful information from incompressible theory.

Compressor power can be expressed either as the product of flow $\dot{W}$ and specific work H , or as the product of angular speed ω and torque Trq :

$$PW = \dot{W} \times H = \omega \times Trq \quad (2)$$

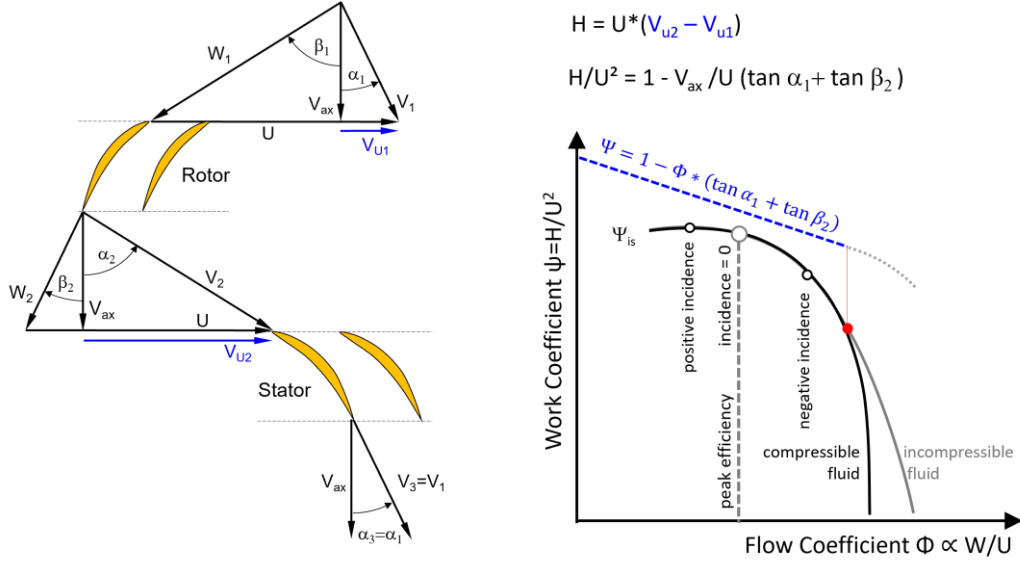


Figure 2: Incompressible compressor theory

Rearrangement and insertion of equation 1 yields

$$\frac{Trq}{W^2} = c_2 \frac{H}{U^2} \frac{U}{W} = c_2 \frac{1 - c_1 \Phi}{\Phi} = c_2 \frac{1}{\Phi} - c_1 c_2 \quad (3)$$

This equation is valid where flow velocity V_{ax} is proportional to mass flow W - in other words, for incompressible flow. Under this condition, Trq/W^2 is a linear function of $1/\Phi$ and, for a given circumferential speed U , Trq/W is a linear function of W :

$$\frac{Trq}{W} = c_2 U - c_1 c_2 W \quad (4)$$

The partial derivative $\partial(Trq/W)/\partial U$ - the slope of the line $Trq/W=f(W)$ - is independent of the circumferential speed U . All lines of $Trq/W=f(W)$ are therefore straight and parallel. In the following we show, with four examples of maps from axial compressors, that in a significant part of the respective maps straight, parallel lines in the plot of Trq/W over W represent the measured data.

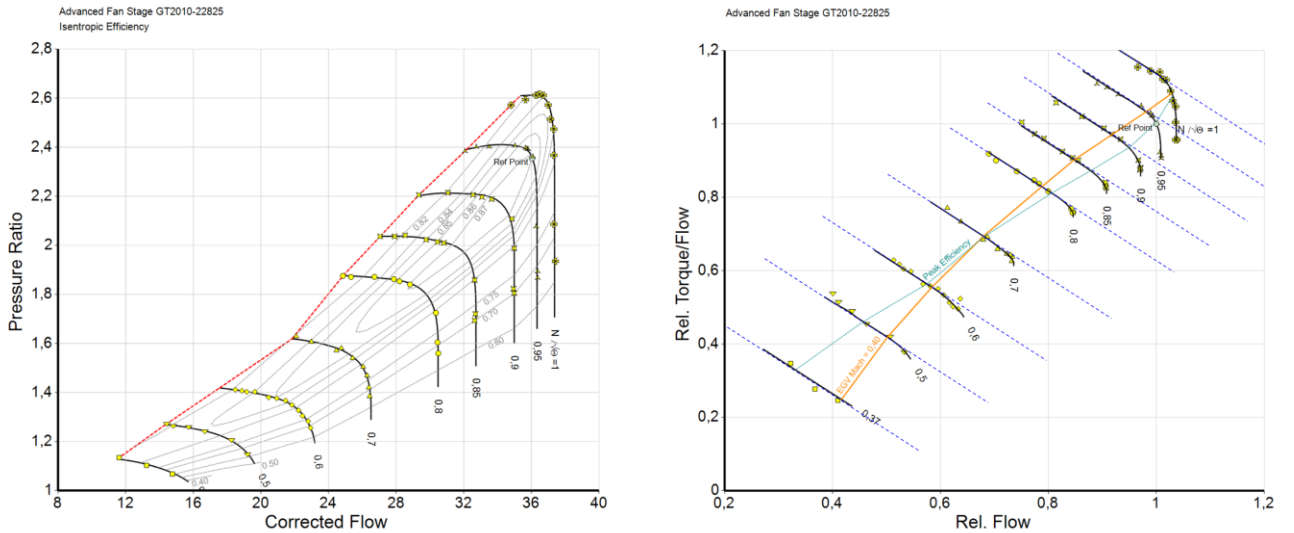


Figure 3: Smoothed map from a single stage fan (Suder K. L., Praht P. S. and Thorp S. A., 2011)

Figure 3 shows what the map in Figure 1 looks like when the above-described theory is applied. The blue dashed lines are straight and parallel. Some iteration is required to shape the pressure ratio, flow, and efficiency lines so that the $Trq/W = f(W)$ lines are straight and have a slope independent of speed. We observe that the theory applies well above the peak efficiency line and above the orange line which represents an exit guide vane inlet Mach number of 0.4.

Knowing that the $\text{Trq}/W = f(W)$ lines are straight and parallel is helpful for identifying outliers in the data. Outliers are to be expected especially near the surge line because measurements there are difficult to obtain accurately due to fluctuations.

Vertical speed lines

The theory provides insight into how torque changes with mass flow. However, at high spool speeds, mass flow changes little or not at all along the speed line and we cannot apply this theory. This raises the question: How do efficiency and specific work vary along a vertical speed line? The key to the answer to this question is the consideration of the conditions at the exit of the compressor.

From inlet mass flow, pressure ratio, and efficiency, we can calculate the corrected flow at the compressor exit. The exit guide vane (EGV) only causes total pressure losses, not a change in total temperature. We can make assumptions about the pressure losses and use them to calculate the corrected flow at the inlet to the EGV, see Kurzke (2019). The pressure losses in the EGV are not very significant, so inaccuracies in estimating them affect the outcome of the subsequent considerations only slightly.

Since choking can restrict the flow at the EGV inlet, where the flow area is smaller than at the compressor exit, it is of interest to know the Mach number at the EGV inlet. We can determine this Mach number for any point in the map if we estimate M_{EGV} at a suitable reference point.

The assumed reference Mach number $M_{\text{EGV,ref}}$ yields a value for the corrected flow per area $W\sqrt{T_{\text{EGV}}/(P_{\text{EGV}} \times A_{\text{EGV}})}$. From mass flow, pressure ratio, efficiency and an assumption about the EGV pressure loss we can calculate the corrected flow $W\sqrt{T_{\text{EGV}}/P_{\text{EGV}}}$ at the map reference point. With this knowledge we find the EGV inlet area A_{EGV} . Once this area is known, the Mach number can be determined at any point on the map.

The question now arises of how to choose a suitable reference point and related EGV inlet Mach number. Since no strict rules exist, this requires some engineering judgment. One reasonable approach is to place the reference point on the peak efficiency line, which connects the points with the highest efficiency on each speed line. To select the Mach number at the reference point, we can look at the Mach number 0.4 line in Figure 3. This line approximately limits the map region in which incompressible flow theory is applicable, and the $\text{Trq}/W = f(W)$ lines are straight and parallel.

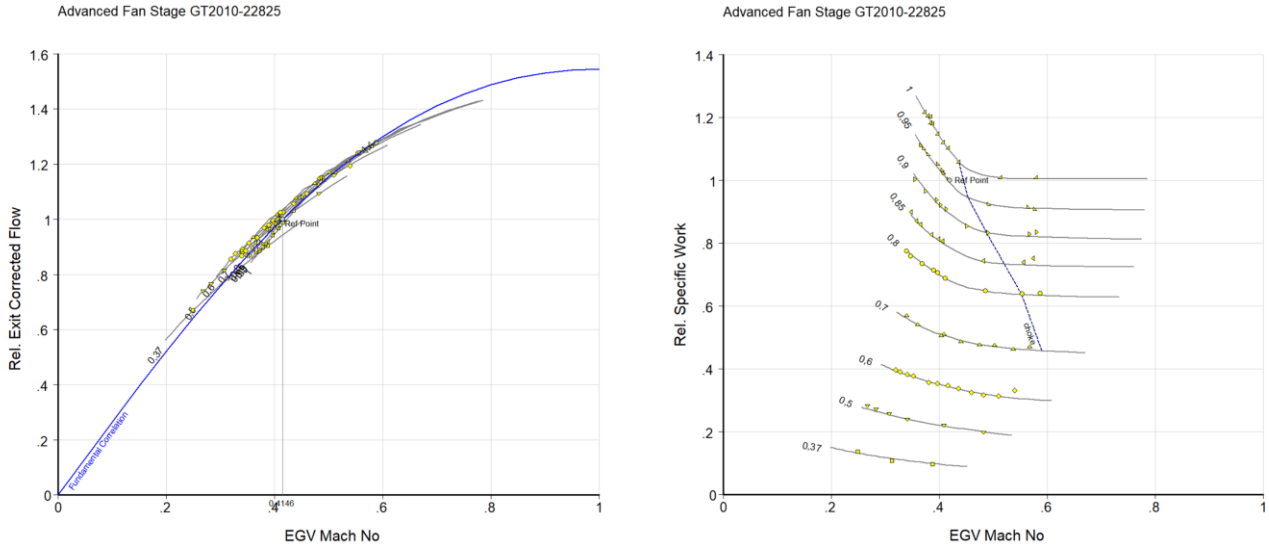


Figure 4: Correlations with exit guide vane inlet Mach number

Examining the left side of Figure 4 is also worthwhile, as it allows one to assess the alignment of the measured corrected exit flow data with the theory (the blue line). Using a Mach number that is too high for the reference point will result in values that exceed the theoretical maximum of the corrected flow-per-area function for low pressure ratios.

Once the Mach number at the reference point has been set, one can examine the right side of Figure 4, which shows specific work over EGV Mach number. All lines must end in the subsonic region. For the upper speed range, the specific work lines tend to be horizontal. The change in efficiency along the speed line can be derived unambiguously from the pressure ratio and the specific work. This yields the efficiency contour lines shown in the left part of Figure 3. They are smooth and nice looking, but more importantly, they align with the laws of compressor physics.

Checking the applicability

Using only one example to demonstrate that a compressor map smoothing methodology is superior is not necessarily convincing. The following shows that the outlined methodology produces reasonable results for all types of compressors. All the input data used in this section originate from compressor rig tests, not from calculations.

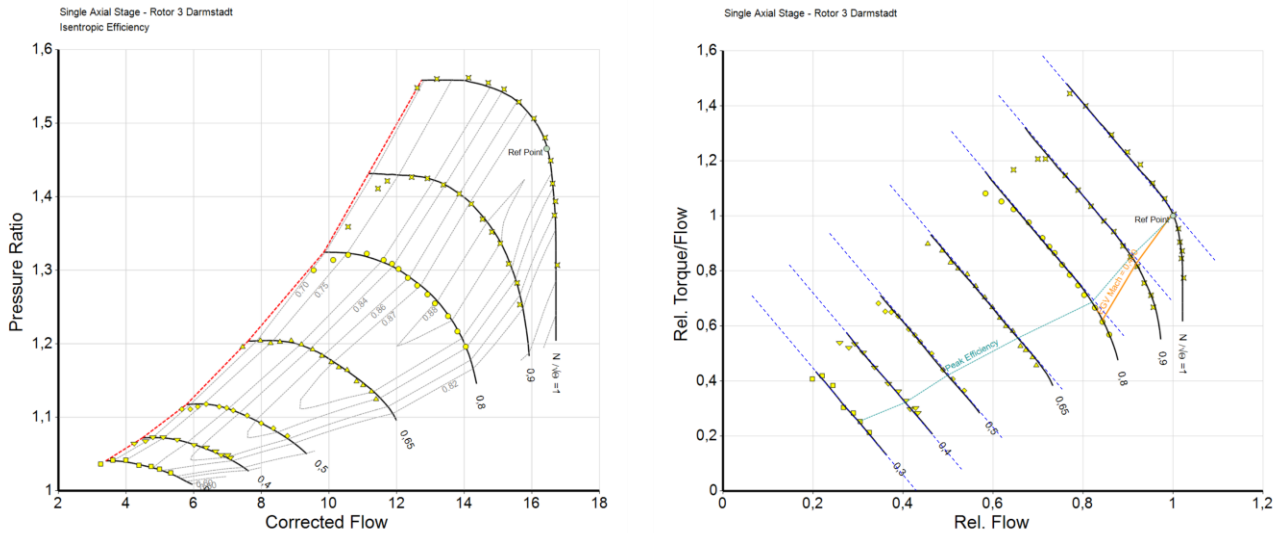


Figure 5: Map from a single-stage transonic compressor

Single-stage axial compressors

The map used to introduce the methodology was from an advanced fan with a design pressure ratio of 2.5 and a tip circumferential Mach number of 1.5. Kablitz's (2003) doctoral dissertation contains data from a 1.5 pressure ratio stage with a much lower tip Mach number of 1.17. This map is shown in Fig. 5 and contains an unusual large number of test points per speed line.

In most of the operating range, straight and parallel $\text{Trq}/W=f(W)$ lines approximate the measured data very well. Outliers near the surge line, which are probably caused by measurement problems due to fluctuations, can be clearly identified. It can be concluded that the map smoothing method is applicable to single-stage compressor maps regardless of the Mach number level.

Multistage axial compressors

The methodology for generating smooth efficiency lines is based on the consideration of single-stage compressors. Can it also be applied to multistage compressors? Let's look at the maps of three- and eight-stage compressors.

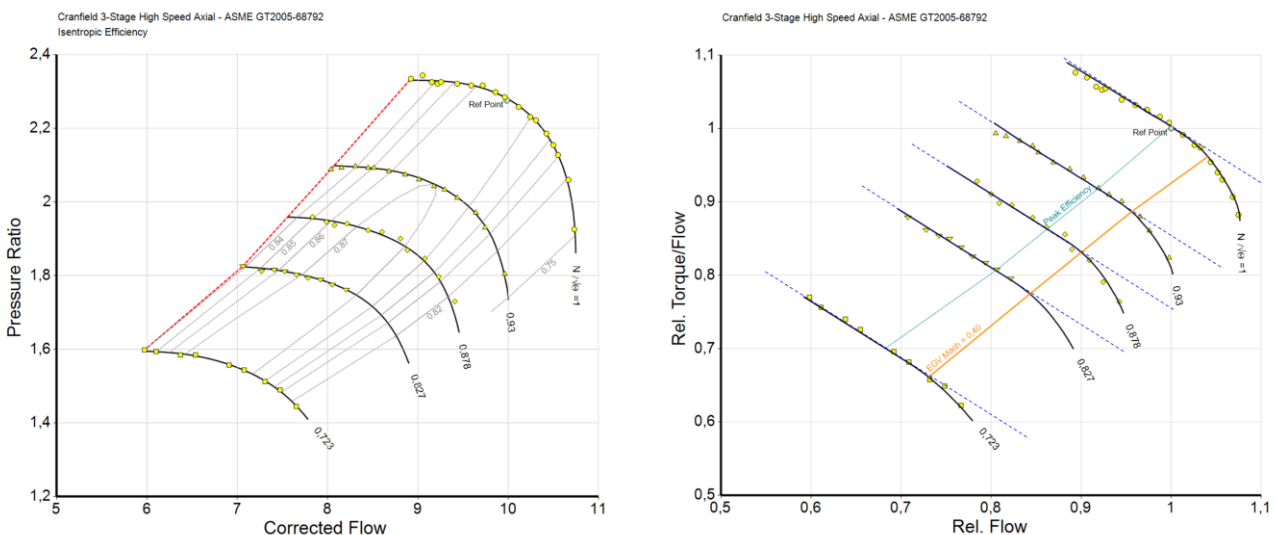


Figure 6: Map from a three-stage compressor

Lippert et al. (2005) published measured data from a three-stage compressor designed with a tip Mach number of 0.75 and a pressure ratio of 2.4. This data was used to generate Figure 6. The EGV Mach number line of 0.4 clearly distinguishes

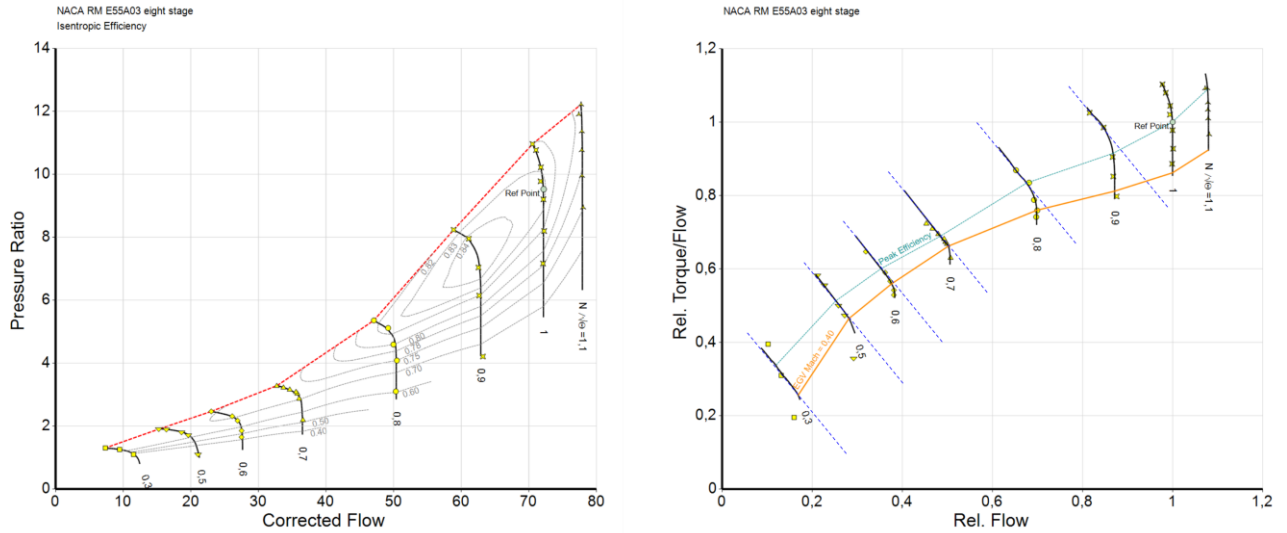


Figure 7: Map from an eight-stage compressor

the incompressible flow region from the region where compressibility effects appear. The $\text{Trq}/W=f(W)$ lines are straight and parallel in this map of a three-stage compressor as well.

Standahar and Geye (1955) published measured data from an eight-stage research compressor. The design pressure ratio was 10.3, and the circumferential tip Mach number was 1.09. Figure 7 shows that, even in this compressor map, the $\text{Trq}/W = f(W)$ lines are straight and parallel across much of the map. The change in efficiency along the vertical part of the speed lines can be quite well derived from the specific work correlation with the EGV Mach number. Not only do the efficiency contour lines in the map look good, but they are also based on the laws of physics.

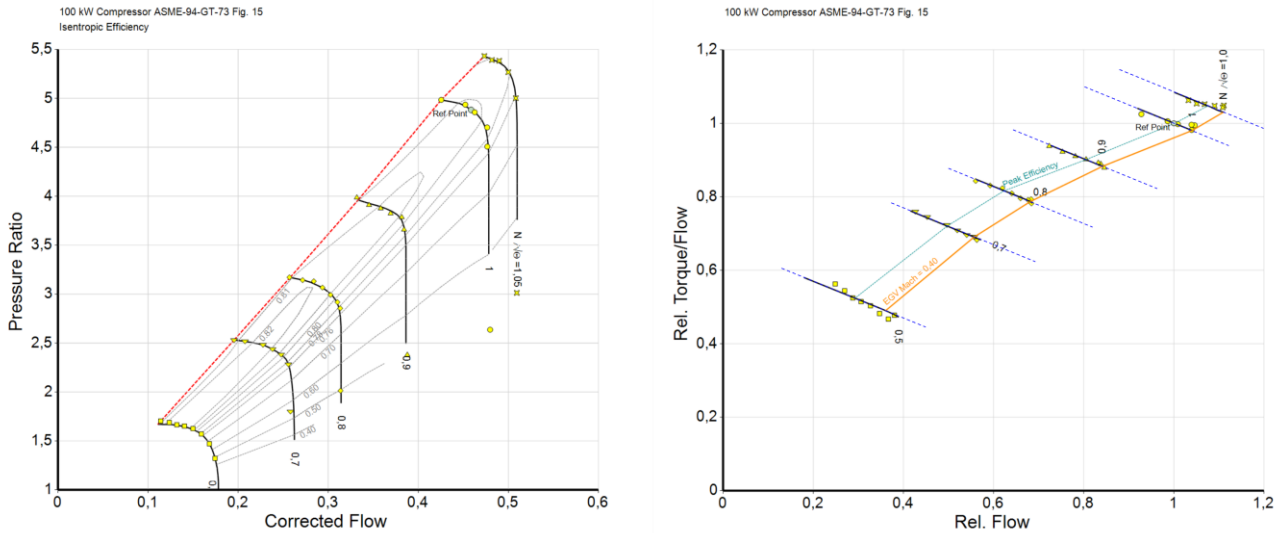


Figure 8: Map from a single-stage radial compressor

Single stage radial compressor

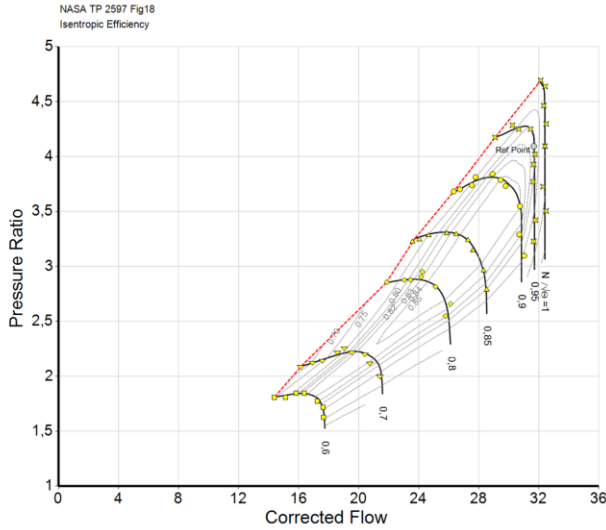
It is somewhat bold to apply a theory derived from axial compressors to the data from a radial compressor published by Uchida et al. (1994). Nevertheless, this has been done, and the results are shown in Figure 8. As in previous cases, assuming the $\text{Trq}/W = f(W)$ lines are straight and parallel helps interpret the data and detect outliers.

Unlike axial compressor maps, the torque lines do not bend down at high mass flow rates. This is because the compressor's work remains constant when the pressure ratio is reduced below the point at which the exit guide vanes choke. A further reduction in exit pressure does not affect the rotor's flow field where the work is done.

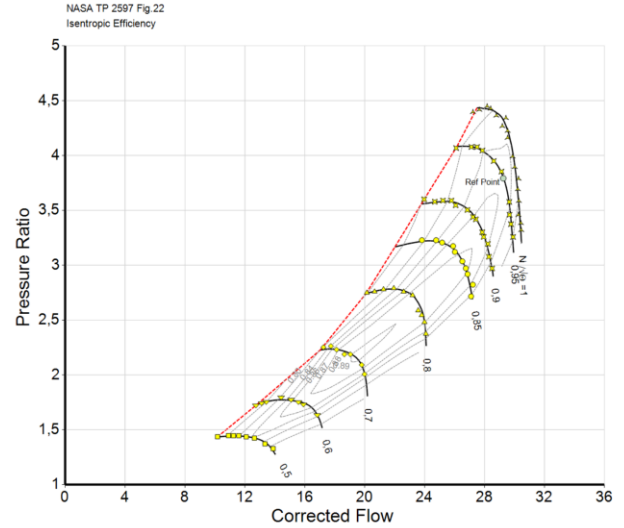
Axial compressors with variable guide vanes

Normally, any map should be smooth. However, this is not the case for maps from compressors with variable guide vanes, since kinks in the VGV schedule create kinks in the speed-flow correlation.

Figure 9 shows two maps of the same three-stage compressor which has variable guide vanes (VGV). The map on the left was measured with the VGV in the same position at all speeds, while the map on the right was created with an optimized VGV position for each speed. The speed lines in both maps have been smoothed using the same slope of the $\text{Trq}/W=f(W)$ lines.

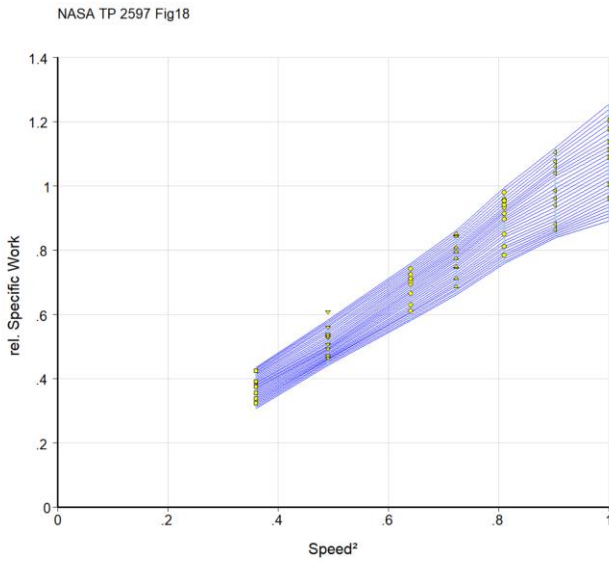


NASA TP 2597 Fig. 18: Nominal VGV settings

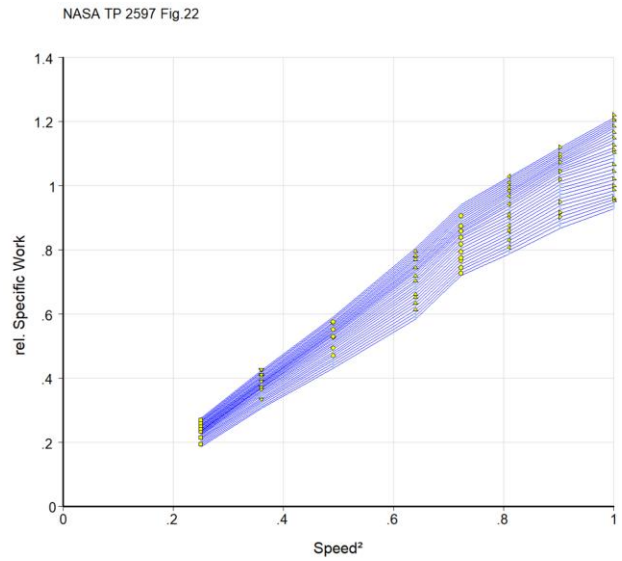


NASA TP 2597 Fig. 22: VGV setting for maximum efficiency at each speed

Figure 9: Maps from a three-stage compressor with variable guide vanes



NASA TP 2597 Fig. 18: Nominal VGV settings



NASA TP 2597 Fig. 22: VGV setting for maximum efficiency at each speed

Figure 10: Kinks in the VGV schedule are visible in the work – speed² correlation

The auxiliary coordinates (the β -lines) used in the Smooth C program are parabolas that can approximate lines of constant work coefficient, $\psi = H/U^2$. When specific work is plotted against the square of speed, the β lines are essentially straight, provided the compressor geometry is fixed. This is evident in the left map of Figure 9 and the left part of Figure 10. The right part of the same figure shows that the β lines are no longer linear functions of speed squared; kinks appear at speeds of $N = 0.85$ ($N^2 = 0.72$) and $N = 0.8$ ($N^2 = 0.64$). Eliminating these kinks during the map smoothing process would be a mistake. All fully automatic map smoothing methodologies applied to a compressor with variable guide vanes that neglect the kinks in the variable guide vane schedule yield questionable results. In fact, it's quite simple to determine whether a map originates from a compressor with variable geometry as can be seen in Figure 10.

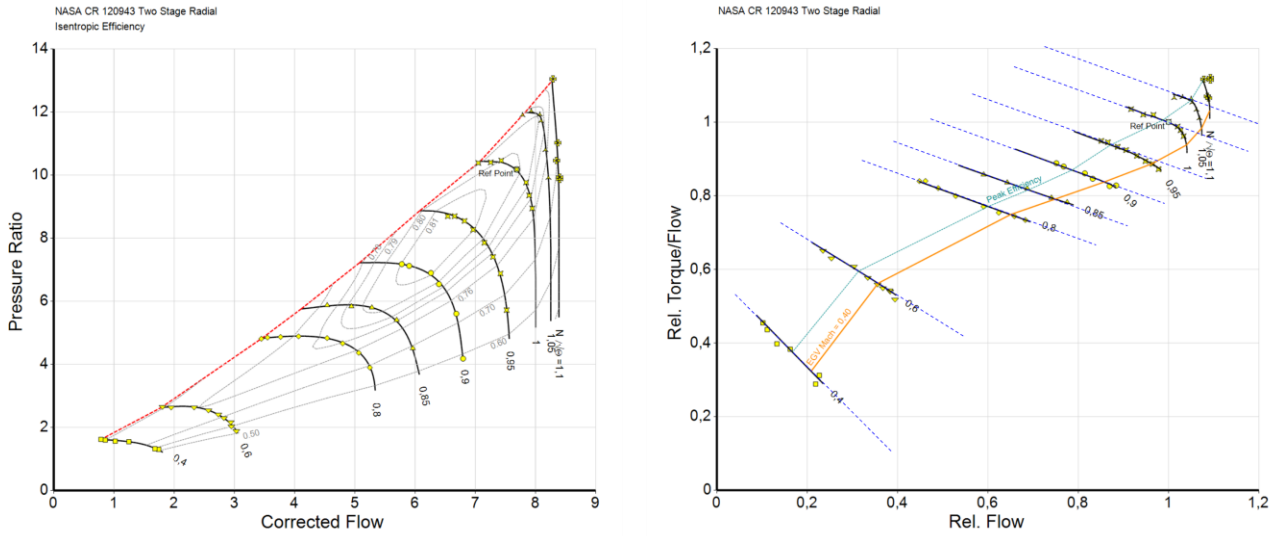


Figure 11: Map from a two-stage radial compressor with interstage bleed

Interstage Bleed

The topic of the report by Bryce et al. (1973) is the design of a two-stage centrifugal compressor. A measured map of such a compressor is shown as an example in that report. In the common map format, the map looks like many others with pressure ratio over mass flow and efficiency contour lines, as seen on the left in Figure 11. However, the $\text{Trq}/W = f(W)$ lines on the right look peculiar. In the upper pressure ratio region, the lines are straight and parallel, as seen in all the other compressor maps. The $\text{Trq}/W = f(W)$ lines for speeds of 0.4 and 0.6, however, are much steeper. This behaviour can be explained by a handling bleed between the two stages. The more interstage bleed air that is taken off, the less power (or torque) is needed to drive the second stage of the compressor, which is reflected by steeper $\text{Trq}/W = f(W)$ lines.

RESULTS AND DISCUSSION

The data measured on a compressor rig test have inevitable uncertainties. The aim of postprocessing these somewhat scattered data is to generate smooth correlations that accurately describe the compressor's performance.

Smooth C, a specialized drawing program in which the user defines smooth lines passing through given data with their mouse, was used to process the example maps. The software provides various views of the data that are meaningful in terms of compressor physics. The experimental version of Smooth C employed for preparing this paper makes it easy to create straight, parallel $\text{Trq}/W = f(W)$ lines.

Working with Smooth C is an iterative process. First, the pressure ratio-mass flow lines are shaped with the mouse to be smooth and pass through the measured data. In the second step, the efficiency lines $\eta = f(W)$ and $\eta = f(\pi)$ are shaped so that the smooth efficiency line represents the measured efficiency values.

This process yields lines of work $H = f(W)$ and $\text{Trq}/W = f(W)$ that are not straight. However, they should be straight! One can try to adjust the lines for $\eta = f(W)$ and $\eta = f(\pi)$ to create straight $\text{Trq}/W = f(W)$ lines. However, that is often not feasible because the pressure ratio-mass flow correlation created in the first step does not allow it. So, one must go back to the first step and reshape the pressure ratio-mass flow correlation. A further iteration may be necessary to optimally choose the slope of the $\text{Trq}/W = f(W)$ lines, which should be the same for all speeds. Depending on the user's expertise and knowledge, smoothing a map with Smooth C can take some hours.

Data from several types of compressors, as presented in the previous section, show that straight, parallel $\text{Trq}/W = f(W)$ lines exist in all examined maps. Clearly, the assumption that the flow follows the rotor exit metal angle regardless of the flow angle at the rotor inlet is reasonable.

Considering the $\text{Trq}(W) = f(W)$ lines helps to identify outliers in the data, which are sometimes seen near the surge line. This is related to the challenge of accurately detecting the surge point because, in its vicinity the flow becomes unstable, and the concomitant fluctuations render measurement arduous.

In regions where the speed lines are vertical, examining correlations with the EGV Mach number helps analyse how efficiency changes along such speed lines.

CONCLUSIONS

Smooth C was introduced around 1990 at MTU Aero Engines by the author. The outcome of the data interpretation process with that software depends on the person conducting it, their knowledge of compressor physics, and how many hours they can dedicate to the task. The program is still in use, but there's a long-standing desire to replace it with a fully automatic map smoothing program.

Writing such a program is a challenge. An automatic map smoothing program must be able to differentiate between axial and radial compressors, distinguish between those with and without variable guide vanes, and also between those with and without interstage bleed air off-take. Purely mathematical algorithms for smoothing pressure ratio, mass flow, and efficiency lines will not produce satisfactory results.

The most important thing for engine performance calculations is shaft power, which is calculated after reading the compressor map look-up tables. The shapes of the specific work and torque correlations must be correct; the shapes of the efficiency lines are not an important criterion.

Checking the result remains indispensable. This is true regardless of the software used to generate or smooth a map. The generated map should be checked, especially with respect to the shape of the $\text{Trq}/W = f(W)$ lines. If these lines are not straight or if their slope changes with speed, the result may not align with first principles.

NOMENCLATURE

A	area	m ²
C	a constant value	
EGV	exit guide vane	
H	specific work	kW/(kg/s)
P	total pressure	kPa
PW	shaft power	kW
T	total temperature	K
Trq	torque	Nm
U	circumferential speed	m/s
V	absolute velocity	m/s
VGv	variable guide vane	
W	mass flow	kg/s
W	relative velocity	m/s
α	absolute flow angle	deg
β	relative flow angle	deg
β	auxiliary map coordinate	
Ψ	work coefficient H/U^2	
Φ	flow coefficient V_{ax}/U	
π	pressure ratio	
ω	angular velocity	deg/s
Indices		
1	rotor inlet	
2	rotor exit	
ax	in axial direction	
u	in circumferential direction	

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