Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45519

CORRELATIONS HIDDEN IN COMPRESSOR MAPS

Joachim Kurzke
Gas Turbine Performance Simulation Software
85221 Dachau, Max Feldbauer Weg 5, Germany
kurzke@gasturb.de
www.gasturb.de

ABSTRACT

Realistic compressor maps are the key to high quality gas turbine performance calculations. When modeling the performance of an existing engine then these maps are usually not known and must be approximated by adapting maps from literature to either measured data or to other available information. There are many publications describing map adaptation processes, simple ones and more sophisticated physically based scaling rules. There are also reports about using statistics, genetic algorithms, neural networks and even morphing techniques for re-engineering compressor maps. This type of methods does not consider the laws of physics and consequently the generated maps are valid at best in the region in which they have been calibrated. This region is frequently very narrow, especially in case of gas generator compressors which run in steady state always on a single operating line.

This paper describes which physical phenomena influence the shape of speed and efficiency lines in compressor maps. For machines operating at comparatively low speeds (so that the flow into each stage is subsonic), there is usually considerable range between choke and stall corrected flow. As the speed of the machine is increased the range narrows. For high-speed stages with supersonic relative flow into the rotor the efficiency maximum is where the speed line turns over from vertical to lower than maximum corrected flow. At this operating condition the shock is about to detach from the leading edge of the blades.

The flow at a certain speed can also be limited by choking in the compressor exit guide vanes. For high pressure ratio single stage centrifugal compressors this is a normal case, but it can also happen with low pressure ratio multistage boosters of turbofan engines, for example. If the compressor chokes at the exit, then the specific work remains constant along the speed line while the overall pressure ratio varies and that generates a very specific shape of the efficiency contour lines in the map.

Also in other parts of the map, the efficiency varies along speed lines in a systematic manner. Peculiar shapes of specific work and corrected torque lines can reveal physically impossibilities that are difficult to see in the standard compressor map pictures.

Compressor maps generated without considering the inherent physical phenomena can easily result in misleading performance calculations if used at operating conditions outside of the region where they have been calibrated. Whatever map adaptation method is used: the maps created in such a way should be checked thoroughly for violations of the underlying laws of compressor physics.

NOMENCLATURE

A	m²	flow annulus area
c_p	J/kg*K	specific heat at constant pressure
H	m²/s²	specific work
k	-	a constant value
M	-	Mach number
N	rpm	spool speed
P	kPa	total pressure
R	J/kg*K	gas constant

T	K	total temperature
T_s	K	static temperature
Trq	Nm	torque
u	m/s	circumferential velocity
v	m/s	velocity
VGV	-	variable guide vane
W	kg/s	mass flow
α	°	stator exit angle
β	°	rotor exit angle
Φ	-	flow coefficient
ρ	kg/m³	density
γ	-	isentropic exponent
π	-	pressure ratio
Ψ	-	work coefficient

Indices:

1	compressor inlet
2	compressor exit
ax	axial
eff	effective
is	isentropic
u	circumferential direction

INTRODUCTION

Realistic compressor maps are the key to accurate gas turbine performance calculations. One percent error in fan or high pressure compressor efficiency translates up to one percent error in specific fuel consumption of a typical turbofan, for example.

This paper is written for those who use compressor maps within an overall system simulation; compressor specialists will barely find new information. Most of the considerations presented are based on one-dimensional flow correlations. The importance of these correlations was emphasized by Cumpsty in his Leroy Smith Keynote (ref. 18) two years ago stating that if a multistage compressor has a significant shortfall in performance it is likely to be the result of errors in one-dimensional matching. Cumpsty regrets that one-dimensional flow in turbomachinery tends to be forgotten by people working in turbomachinery design.

Since more than 15 years, the topic of compressor maps is on the agenda of the yearly IGTI conference as can be seen from the list of references. The references 1 to 17 discuss methodologies for deriving compressor maps from a limited amount of measured or calculated gas turbine data.

Many different approaches to the task of creating a complete compressor map have been proposed. Calculating a map requires knowledge of the detailed geometry; the results from even the most sophisticated codes available today are not accurate enough for predicting the performance and operability of a newly designed engine. Only if a code has been calibrated with rig data one can get for slightly different compressor designs sufficiently accurate results.

Measuring the map on a dedicated compressor rig using engine parts yields the best result. Such a map can be scaled to represent the maps of slightly modified versions of the same compressor.

Calculating or measuring a compressor map is only feasible for the engine manufacturer and research organizations, for the gas turbine user this is not an option. For him the only way getting a map is adapting a map from literature in such a way that it fits to the known data.

Most compressor map adaptation methodologies begin with a map from open literature. The map is scaled and tweaked to make it line up with the given data. For this process various methodologies have been proposed; many of them employ sophisticated mathematics like neural networks, genetic algorithms, fuzzy logic and numerical optimization. Even the use of morphing (morphing is a graphical procedure which changes one image into another through a seamless transition) has been proposed (ref. 14) for map creation.

The problem with purely mathematically based methodologies is that they pay no heed to the laws of physics while modifying the shape of the speed and efficiency lines of the map.

In the following, the correlations between pressure ratio, flow, speed, specific work, torque and efficiency that are inherent in any compressor map are discussed in the context of various compressor design features.

COMPRESSOR MAP BASICS

A compressor map is usually a graph with lines of pressure ratio over corrected flow for various corrected spool speeds. Additionally there might be contour lines for constant efficiency in the same graph or efficiency is shown for each corrected spool speed in a separate plot over corrected flow or pressure ratio.

Let us first clarify what is meant with the terms "corrected flow" and "corrected spool speed". Corrected flow usually stands for the term $W\sqrt{T}/P$ that can be expanded introducing the Mach number M_{ax}:

$$W\frac{\sqrt{T}}{P} = \frac{A * M_{ax} * \sqrt{\gamma/R}}{\left(1+\frac{\gamma-1}{2}M_{ax}^2\right)^{\frac{\gamma+1}{2*(\gamma-1)}}}$$

If we consider dry air at Standard Day temperature only, then the isentropic exponent γ and the gas constant R are invariable values and it becomes obvious that $W\sqrt{T}/P$ represents the axial flow Mach number in the compressor face annulus area A.

Note that the relation between $W\sqrt{T}/P$ and M_{ax} is not a linear one. As Mach number approaches the sonic condition, the flow parameter becomes very flat and reaches a maximum at the exactly sonic condition. Therefore, at high airflows the speed lines become closely bunched together.

Corrected spool speed stands for N/√T, which is proportional to corrected circumferential speed u/√T. Introducing the static temperature T_s leads to

$$\frac{u}{\sqrt{\gamma * R * T_s}} = \frac{u}{\sqrt{T}} * \frac{\sqrt{1+\frac{\gamma-1}{2} M_{ax}^2}}{\sqrt{\gamma * R}}$$

where $u/\sqrt{(\gamma * R * T_s)}$ is the circumferential speed u divided by the local velocity of sound - it is the circumferential velocity expressed as the Mach number M_u.

$$\frac{u}{\sqrt{T}} = M_u * \frac{\sqrt{\gamma * R}}{\sqrt{1+\frac{\gamma-1}{2} M_{ax}^2}}$$

Thus at a given corrected flow (i.e. a given compressor face Mach number M_{ax}) u/√T stands for the circumferential Mach number M_u.

The circumferential Mach number M_u can reach any value. The corrected flow through a compressor, however, is limited since the annulus Mach number M_{ax} cannot exceed one. This knowledge can be used for map extrapolation to extremely high corrected spool speeds, provided that the flow annulus area at the compressor face is known. Thus in the top speed region of the map a hidden correlation exists between corrected speed and flow. This correlation, however, is not of a big practical value because in reality compressor design face Mach numbers are not higher than approximately 0.7.

Finally, pressure ratio can also be expressed as a Mach number because $H_{is}/(\gamma * R * T_1)$ is formally a Mach number squared.

$$\frac{P_2}{P_1} = \left(1+\frac{H_{is}}{c_p * T_1}\right)^{\frac{\gamma}{\gamma-1}} = (1+(\gamma-1) * M^2)^{\frac{\gamma}{\gamma-1}}$$

THE SHAPE OF SPEED LINES

Much of the off-design performance behavior of a multi-stage compressor is determined by matching of the stages from front to rear. In particular, very far from design at very low or very high airflows the local flow parameters vary considerably in magnitude, causing the front, middle and rear stages to simultaneously operate over the range from stall to choke at the same RPM. Overall compressor performance reflects the totality of performance for each stage, front to rear.

Nevertheless, it is useful considering how the basic shape of a speed line (i.e. a line with constant corrected speed) of a single stage compressor operating at comparatively low Mach numbers looks like: it can be derived from a velocity diagram analysis.

Let us introduce two useful quantities first, the flow coefficient $\Phi = V_{ax}/u$ and the work coefficient $\Psi = H/u^2$. In the ideal case, there is a linear relationship between these two quantities when the flow angle at the exit of each blade and vane row is independent from the incidence angle and equal to the blade metal angle:

$$\psi = 1 - \phi * (\tan\alpha + tan\ \beta)$$

In a real compressor map the correlation between effective Ψ and Φ is essentially linear in the low flow region only as can be seen from fig. 1 which has been created from the measured data published in ref. 20 with the program Smooth C developed by the author. Effective Ψ describes the work input while Ψ_{is} describes implicitly the pressure ratio achieved. The difference between Ψ and Ψ_{is} is caused by the losses which are lowest in the middle (near to the design conditions) of the flow coefficient range and increase towards both ends, i.e. surge and choke.

Since the spool speed does not appear in the Ψ and Φ correlation above one would expect that all speed lines in terms of Ψ and Φ agree. This, however, is usually not the case as can be seen from fig. 1, for example.

Note that the ideal phi-psi slope is flat when the blade exit angles are close to the axial direction.

There are two regions in a compressor map where the Ψ - Φ correlation is not very helpful: the very low speed region and in regions with supersonic flow.

Speed Line Zero

At very low spool speeds the Ψ - Φ correlation gets very sensitive due to numerical reasons. If spool speed is zero, Ψ is undefined (because work input is zero) and Φ is infinite - the correlation cannot be used. However, one can find the basic shape of the speed line zero from simple considerations.

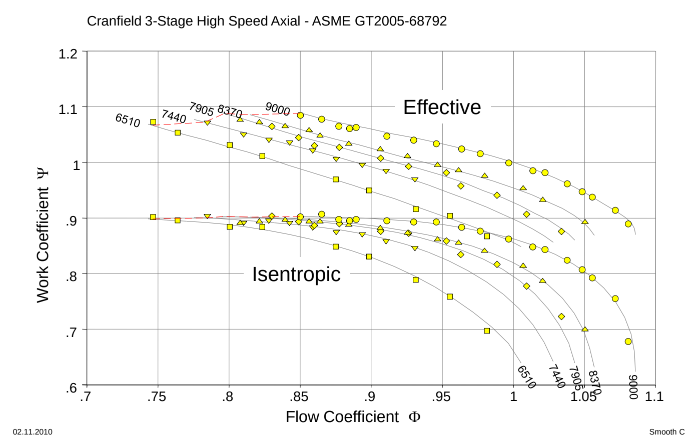

Fig. 1 Work and flow coefficient in a measured map

A non-turning compressor is nothing else than a pipe with a complex internal geometry that creates pressure losses. Imagine a test with constant inlet pressure P_1 and steadily decreasing downstream pressure P_2. While the resulting flow velocity through the compressor is not too big, the pressure loss is proportional to compressor inlet flow velocity squared:

$$P_1 - P_2 = k\frac{\rho}{2} V_{ax}^2$$

Conversion to compressor map coordinates yields:

$$\frac{P_2}{P_1} = 1 - k \frac{\gamma * M_{ax}^2}{2 * \left(1 + \frac{\gamma - 1}{2} M_{ax}^2\right)^{\frac{\gamma}{\gamma - 1}}}$$

This formula describes a line passing through the point {pressure ratio 1; mass flow zero}. With moderate Mach numbers, while k is truly constant, the shape of speed line zero resembles that of a parabola. One obtains the maximum inlet Mach number when in the exit vanes, where the flow area is small and the pressure is low due to the losses created by the upstream stages, sonic velocity is reached. Lowering P_2 further has no effect on the flow field within the compressor, the inlet Mach number remains constant and that means that speed line zero becomes vertical.

In reality, the precise shape of the speed line zero is rarely known. Nevertheless even knowing it only approximately can be of help when extrapolating and adapting a compressor map to low speed data. A smooth transition of the speed line shape towards that of the speed line zero should be ensured in any case.

Supersonic Flow

Supersonic velocities can exist in the flow relative to the rotor of the first compressor stage. In such a case, the lines for higher corrected speed are vertical over a wide range of pressure ratios because the flow field upstream of the first rotor does not change between the points A and C, see fig. 2.

The efficiency maximum is near to point C where the speed line turns over from vertical to lower than maximum corrected flow. At this operating condition the shock is about to detach from the leading edge of the blades.

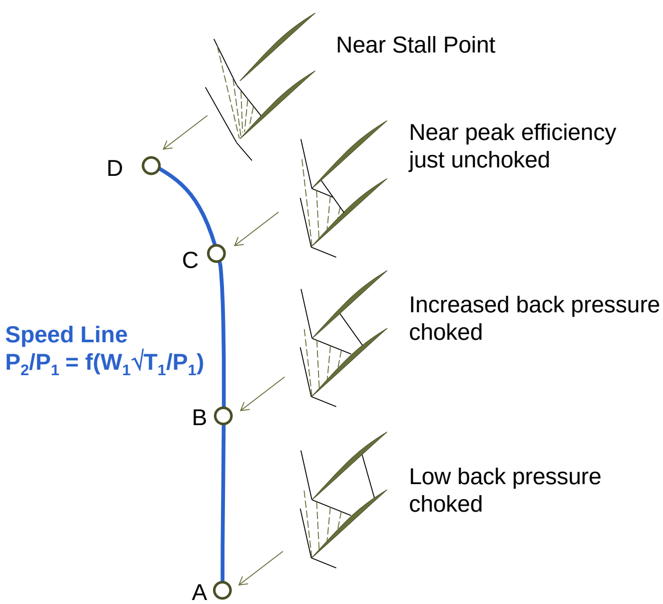

Fig. 2: Supersonic flow makes speed lines vertical

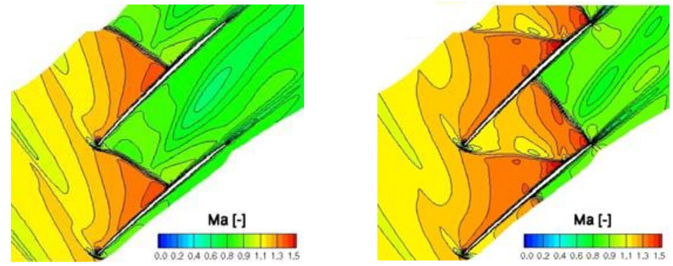

Fig. 3: CFD results for cases B and D (ref. 26)

The map of a single stage fan shown in fig. 4 (ref. 22 and 23) is an example in which supersonic flow in the rotor is the reason for the speed lines being vertical in the high-speed region.

Supersonic flow fields upstream and within the rotor passages are not the only reason for vertical speed lines in a compressor map. High pressure ratio single stage centrifugal compressors have inherently high circumferential tip velocities and this results in high inlet Mach numbers to the diffusor. When due to low backpressure sonic flow at the inlet to the exit vanes is reached, then the same flow structure appears in the vane passages as described above. Changing the backpressure affects only the position of the terminal shock in the vanes but not the flow field in the rotor. The speed lines are vertical in a big part of the map as can be seen in fig.5 (ref. 24).

Boosters of un-geared turbofan engines, for example, operate inevitably with very low circumferential speed and will never see supersonic flow at their entry. Fig. 6 shows the predicted map of a 5-stage low pressure compressor (ref. 25) - similarly as in case of fig. 4 in the high-speed region, the speed lines are vertical.

The reason for the speed lines having vertical sections is in this case that sonic velocity is reached in the exit guide vanes.

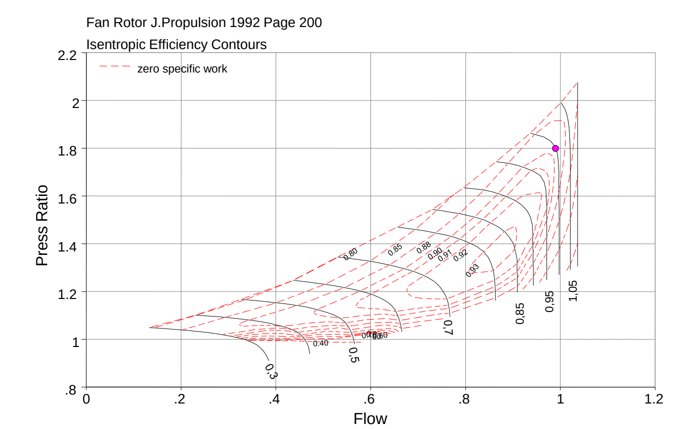

Fig. 4: Fan map with vertical speed lines

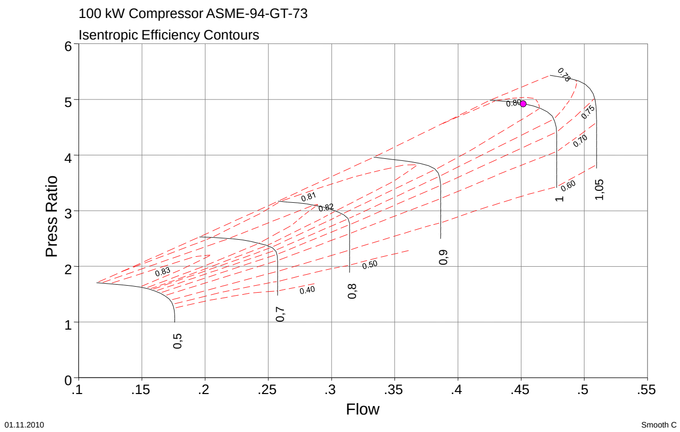

Fig. 5: Map of a high-speed centrifugal compressor

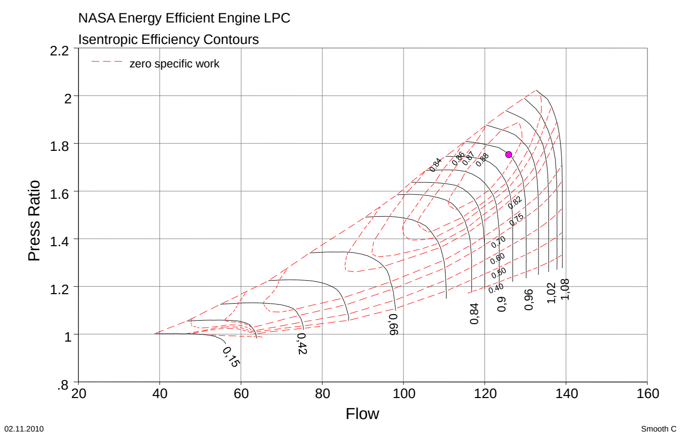

Fig. 6: Map of a 5-stage booster

Specific Work

If a compressor map contains speed lines with vertical sections: is it possible to find out if this is due to supersonic flow in the first rotor or due to a choked blade row at the compressor exit? A plot of specific work over pressure ratio tells a clear story.

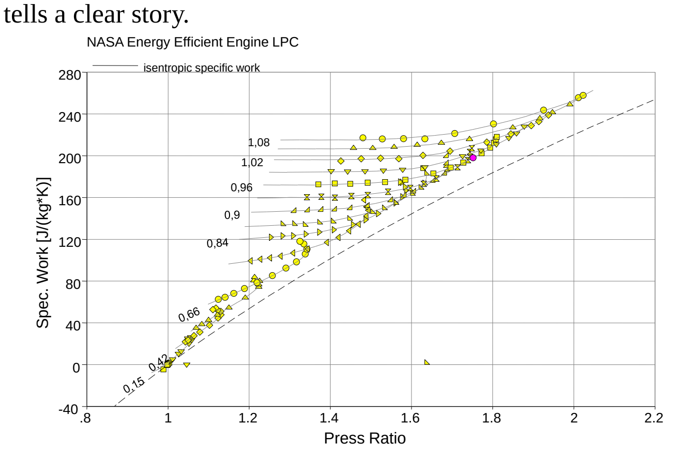

Fig. 7: Specific work for the 5 stage booster map (fig. 6)

If specific work is a horizontal line in this plot, then the compressor exit chokes. This is because reducing the backpressure below the pressure ratio at which the compressor exit chokes has no effect on the flow field in the rotor(s). Since work is done only in the rotor(s) (where the flow field does not change) specific work must remain constant. Fig. 7 shows this effect for the booster map presented as fig. 6.

Plotting specific work over pressure ratio for the centrifugal compressor map presented as fig. 5 tells the same story: as soon as the vaned diffusor chokes the work done in the rotor remains constant, see fig. 8.

If specific work is not a horizontal line in the plot of specific work over pressure ratio (fig.9) while the speed line is vertical, then a supersonic flow field must exist in the first rotor. Different terminal shock positions in the rotor (fig. 2 and 3) cause a change in relative rotor exit velocity and this yields (because both u and V_{ax} are constant) a change in the circumferential component of the absolute velocity at rotor exit. Consequently, specific work depends on pressure ratio.

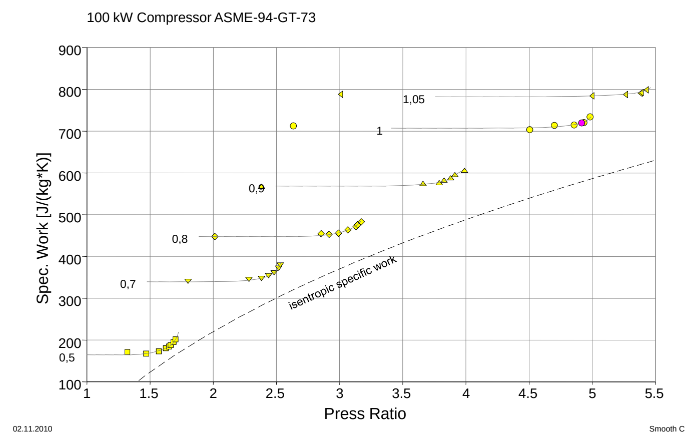

Fig. 8: Specific work for the centrifugal compressor (fig. 5)

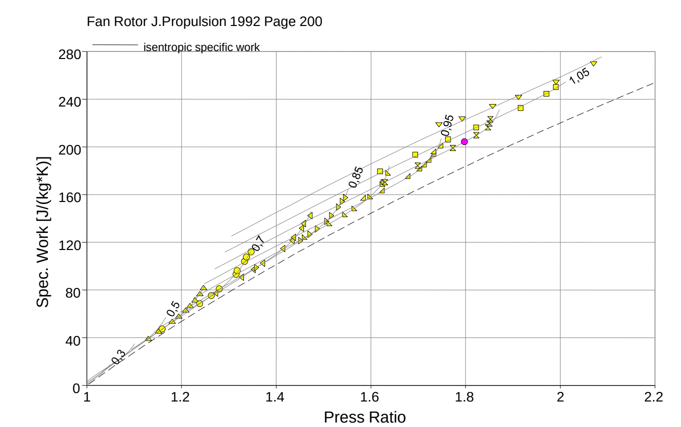

Fig. 9 Specific work for the single stage fan (fig. 4)

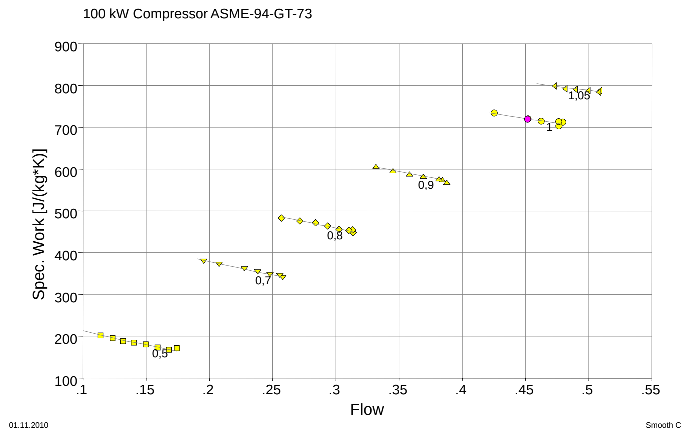

Fig. 10 Specific work for the centrifugal compressor (fig.5)

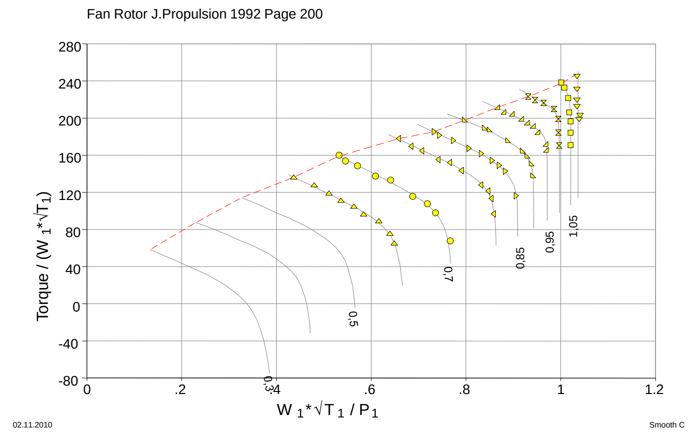

Fig. 11: Corrected torque for the single stage fan (fig. 3)

Torque and Efficiency

Let us have a second look at fig. 1: in the low to medium flow range of a speed line, where the flow is subsonic, the Ψ_{eff} - Φ correlation is essentially a straight line. At a given spool speed Ψ_{eff} is proportional to specific work. For low Mach numbers (incompressible flow) the axial velocity V_{ax} is within a certain Mach number range proportional to corrected flow. Therefore Φ is proportional to corrected flow and specific work plotted over flow is in that region of the map a straight line.

In case of high pressure ratio centrifugal compressors the lines of specific work over flow are straight over the complete flow range. See this in fig. 10, which is another view on the data of ref. 24 (fig. 5).

Instead of corrected specific work H/T_1 one can also consider corrected torque $Trq/(W_1*\sqrt{T_1})$ because these two properties are interconnected. Compressor shaft power can be expressed in two ways:

$$PW = k*N*Trq = W_1*H$$

Rewriting the formula yields

$$\frac{Trq}{W_1*\sqrt{T_1}} = \frac{\frac{H}{T_1}}{k*\frac{N}{\sqrt{T_1}}}$$

Fig. 11 shows for the single stage fan example that the torque lines are linear over a significant part of the map.

How efficiency changes along a speed line depends with subsonic inlet Mach number to the blade and vanes on the incidence angle and cascade inlet Mach number. The useful incidence range decreases quickly at higher inlet Mach numbers, see fig. 12.

Consequently, if the general Mach number level in a compressor is high, the efficiency islands in the map are narrower than in the map of a compressor with a moderate Mach number level. This can be seen, for example, from the comparison of figures 13 and 14.

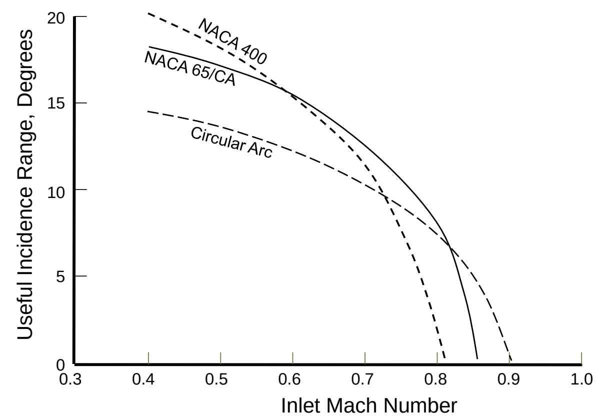

Fig. 12: Incidence range of standard series airfoils (ref. 27)

Both maps are from compressors with approximately the same design pressure ratio. The difference is in the general Mach number level, which must be higher in the single stage fan map (fig. 13) than in the map of the moderately loaded three-stage compressor (fig. 14).

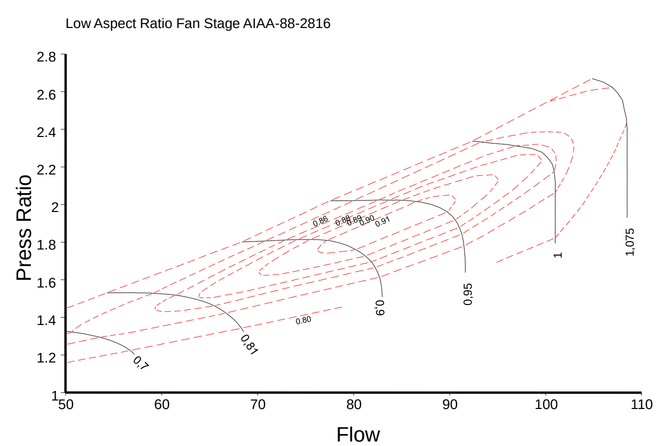

Fig. 13: Single stage fan map (ref. 28)

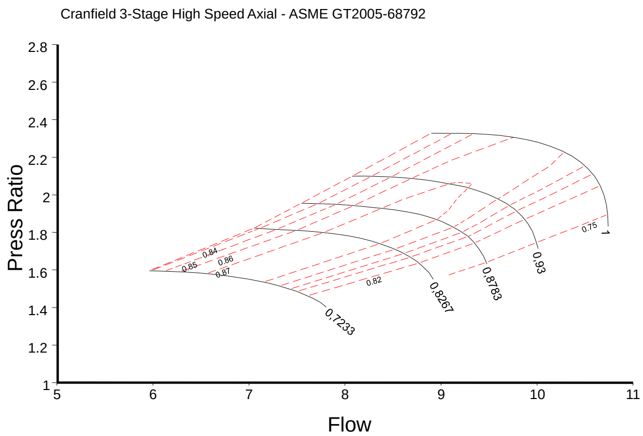

Fig. 14: 3-stage compressor map (ref. 20)

CORRELATIONS WITH SPEED

In the previous sections, we have discussed the shape of pressure ratio, flow, specific work, torque and efficiency lines with constant corrected speed. Next, we will have a look at correlations between various corrected spool speeds. For doing that, it is necessary selecting points on each speed line for which the effect of speed changes are compared. It would be desirable to compare points with equal Ψ and Φ or equal corrected exit flow $W\sqrt{T_2}/P_2$ or any other physically meaningful quantity. Ψ, however, is not a universally applicable parameter because it is invariable in map regions with compressor exit choke. Lines of constant $W\sqrt{T_2}/P_2$ are also not universally applicable because they are very steep in maps of low pressure ratio compressors; they do not connect all the speed lines.

In off-design cycle calculations, the compressor maps need to be evaluated numerically. This is not a trivial task because one cannot read a map with given corrected speed and corrected flow in the region where the speed lines are vertical. One cannot read the map with given corrected speed and pressure ratio either because in the low flow region the speed lines sometimes bend over and then two solutions for the corrected mass flow exist.

One can solve the map-reading problem by introducing auxiliary coordinates - frequently called m-lines or ß-lines - that have unique crossings with each of the speed lines. Thus, the map can be read with given corrected speed and an estimated value for the auxiliary coordinate. The final value of the auxiliary coordinate is found by iteration in such a way that all the flow and energy balances in the cycle calculation are fulfilled.

From principle, the shape of the auxiliary coordinates is arbitrary; linear or parabolic shapes are in use as well as lines in parallel to a line connecting all the points with the highest efficiency on each of the speed lines. The cycle code GasTurb developed by the author uses parabolic ß-lines, for example.

Parabolic ß-lines do not stand for a specific physical value but allow covering completely any type of compressor map. In case of subsonic compressors without variable geometry, parabolic ß-lines can be selected in such a way that they have the shape of constant Ψ lines in the map region with un-choked exit guide vanes. In the map region with choked exit guide vanes the ß-lines are equally spaced auxiliary coordinates and do not represent any specific physical value.

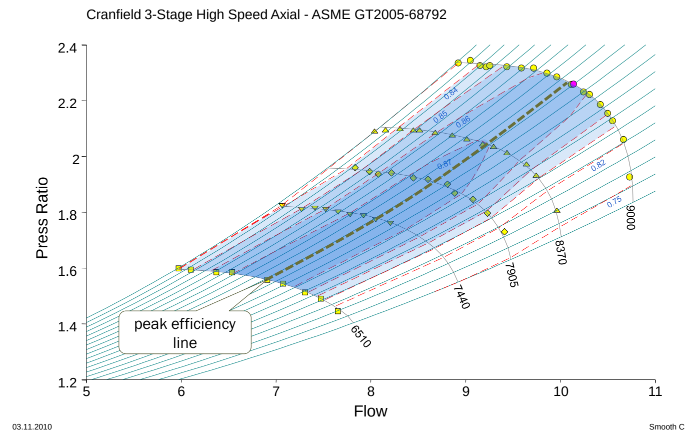

Fig. 15: ß-lines in a compressor map

The ß-line grid as plotted for example in fig. 15 allows showing how the various map parameters change with speed. Along the peak efficiency line the effective work coefficient Ψ is nearly constant; the velocity triangles are very similar to each other, which indicates that the incidences to the blade and vanes are low.

From fig. 16, one can see that specific work is a linear function of speed squared for all ß-lines, which indicates that $\Psi = H/N^2$ is constant along all ß-lines. The top dashed line passes through the origin; the lower crosses the zero specific work line at some positive speed. In a small region of the map, at very low speed, the compressor can operate as a turbine and deliver work instead of consuming it.

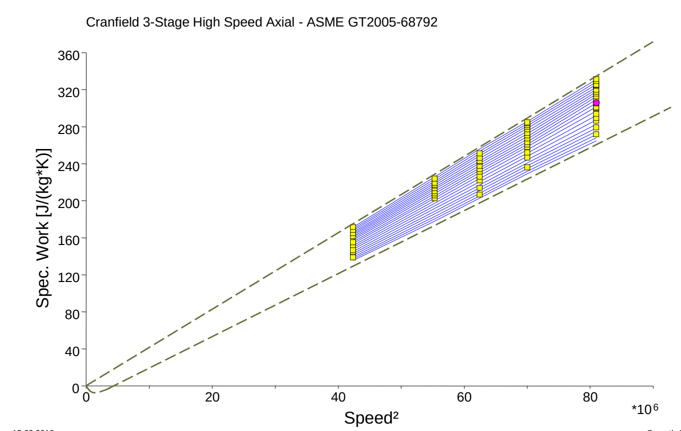

Fig. 16: Specific work is a linear function of speed²

Fig. 17 shows flow as a function of speed for the single stage fan from ref. 22 and 23. For low to medium speed values, flow increases linearly with speed along any ß-line. At the top right end the flow increase with speed becomes less and less which indicates that the axial velocity approaches Mach 1. Since the rightmost speed line in fig. 4 is vertical, all ß-lines collapse for this speed in fig. 17.

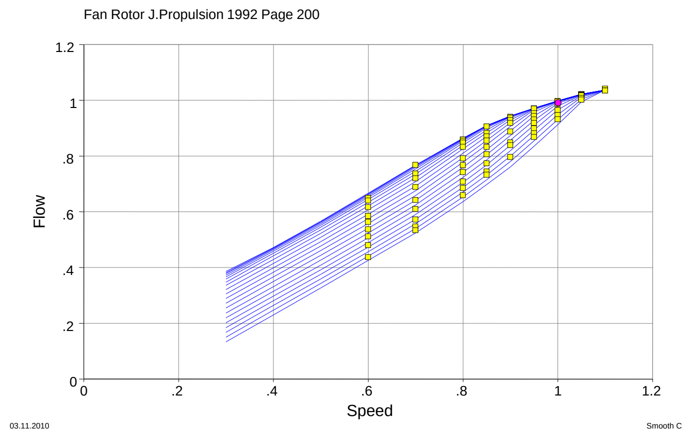

Fig. 17: Flow - speed correlation for ß=constant

Variable guide vanes

All compressors without variable guide vanes show linear H - N² correlations similar to those of fig. 16. The W-N correlation shown in fig. 17 is typical; with subsonic compressors, the top right part - where the ß-lines level out - is missing. In any case, the ß-lines are smooth.

Moving the variable guide vanes to a new position changes the shape of the map. Closing the VGV's makes the map shrink: the speed lines as well as the efficiency islands are shifted towards lower pressure ratio and mass flow. As long as the guide vane movements are moderate, the peak efficiency in the map is not affected very much.

A complete and accurate performance description of a compressor with variable guide vanes requires a whole series of maps. Measuring multiple maps on a compressor rig is very expensive, calculating them might not be accurate enough. Therefore, a different approach is frequently employed: for each value of corrected speed, the VGV setting is optimized separately. All the speed lines with their respective optimal VGV settings are combined to form a single map.

Such a map is valid for a specific VGV schedule with VGV position as a function of corrected speed. Stator exit angles are no longer constant values and therefore the Ψ - Φ correlation is a different one for each $N/\sqrt{T}$:

$$\psi = 1 - \phi * (\tan\alpha + tan\ \beta)$$

One can see this in plots of a map with embedded VGV schedule in figures equivalent in to fig. 16 and 17 as a bend in case the variable guide vanes schedule is not a smooth function of corrected spool speed.

Mechanical stops justify a discontinuity in the VGV schedule; mechanical failures can lead to an unexpected deviation from an otherwise smooth schedule. Thus a discontinuity of H=f(ß,N²) and W=f(ß,N) in the medium speed range can be a hint to a malfunction of the VGV mechanism.

SUMMARY

When looking at the compressor maps plotted with efficiency contour lines (fig. 4, 5 and 6), one cannot see that there is a hidden correlation between efficiency and pressure ratio in those parts of the maps where the speed lines are vertical, for example. Therefore, it is very probable that any modification of efficiency using purely mathematical methods like neural networks will result in physically impossible correlations of specific work and pressure ratio.

Creating a compressor map from scratch using morphing or other methodologies not based on physics will almost certainly lead to a physically impossible result. Adapting a physically sound map to given data by scaling is much less dangerous.

Whatever methodology one uses for adapting a map to given data, one should check the final map by plotting various parameters with the aim detecting tendencies that are not in line with the laws of physics.

ACKNOWLEDGMENTS

The author thanks Uwe Schmidt-Eisenlohr, former head of the MTU Aero Engines compressor department, for his critical review of the paper.

REFERENCES

[1] Joachim Kurzke
How to get component maps for a aircraft gas turbine performance calculations.
ASME Turbo Expo 1996, Paper No 96-GT-164

[2] Sieros G, Stamatis A, Mathioudakis K.
Jet engine component maps for performance modeling and diagnosis.
J Propul Power 1997;13(5): 665–74.

[3] Claus Riegler, Michael Bauer, Joachim Kurzke
Some Aspects of Modeling Compressor Behavior in Gas Turbine Performance Calculations
ASME Turbo Expo 2000-GT-0574
Journal of Turbomachinery, April 2001.

[4] Joachim Kurzke, Claus Riegler
A New Compressor Map Scaling Procedure for Preliminary Conceptional Design of Gas Turbines
ASME 2000-GT-0006
Aircraft Engine Committee Best Paper Award

[5] Barry Curnock and Pericles Pilidis
Compressor Characteristics in Gas Turbine Performance Modeling,
ASME Turbo Expo 2001, 2001-GT-0384

[6] C. Kong, J. Ki and M. Kang,
A New Scaling Method for Component Maps of Gas Turbine Using System Identification,
J. of Engineering for Gas Turbines and Power,125(4), pp. 979-985, 2003

[7] C. Kong, S. Kho and J. Ki,
Component Map Generation of a Gas Turbine Using Genetic Algorithms
ASME Paper No. GT2004-53736

[8] Kong CD, Ki JY.
Components map generation of gas turbine engine using genetic algorithms and engine performance deck data.
Trans ASME J Eng Gas Turbines Power 2007;129(2):312–7.

[9] Changduk Kong, Seonghee Kho, Jayoung Ki, Yongmin Jun
A Novel Method for Component Map Identification of a Gas Turbine Using Intelligent Method and Engine Performance Deck Data
ASME Turbo Expo 2007, GT2007-27569

[10] Yu Y, Chen L, Sun F, Wu C.
Neural network based analysis and prediction of a compressor's characteristic performance map.
J Appl Energy 2007; 84(1): 48–55.

[11] Kaveh Ghorbanian, Mohammad Gholamrezaei
Axial Compressor Performance Map Prediction Using Artificial Neural Network
ASME Turbo Expo 2007 GT2007-27165
ASME Turbo Expo 2008 GT2008-51068

[12] M. Zwingenberg, F.-K. Benra, K. Werner
Improvement of Performance Prediction by Automated Assimilation of Gas Turbine Component Maps
ASME Turbo Expo 2008, GT2008-51068

[14] Chris Drummond, Craig R. Davison
Capturing the Shape Variance in Gas Turbine Compressor Maps
ASME Turbo Expo 2009, GT2009-60141

[15] Chris Drummond, Craig R. Davison
Improved Compressor Maps Using Approximate Solutions to the Moore Greitzer Model
ASME Turbo Expo 2009, GT2009-60148

[16] Y.G. Li, M.F. Abdul Ghafir, L. Wang, R. Singh, K. Huang, X. Feng
Non–Linear Multiple Points Gas Turbine Off-design Performance Adaptation Using a Genetic Algorithm
ASME Turbo Expo 2010, GT2010-22285

[17] Changduk Kong, Semyeong Lim , Seonghwan Oh, Jihyun Kim
Inverse Generation of Gas Turbine Component Performance Maps From Experimental Test Data
ASME Turbo Expo 2010, GT2010-22102

[18] N.A. Cumpsty
Some Lessons Learned
ASME Turbo Expo 2009, GT2009-60368

[19] H Cohen, GFC Rogers, HIH Saravanamuttoo
Gas Turbine Theory
4th edition, 1996

[20] D. Lippett, et alii
The Design, Development and Evaluation of 3D Aerofoils for High Speed Axial Compressors. Part 1: Test Facility, Instrumentation and Probe Traverse Mechanism.
ASME Turbo Expo 2005, GT2005-68792

[21] J D Denton, L Xu
The Effects of Lean and Sweep on Transonic Fan Performance
ASME Turbo Expo 2002, GT2002-30327

[22] C Freeman
Method for the Prediction of Supersonic Compressor Blade Performance
Journal of Propulsion, Vol 8, No 1, 1992

[23] N. Cumpsty
Jet Propulsion - A simple guide to the aerodynamic and thermodynamic design and performance of jet engines
Cambridge University Press, 1997

[24] Hiroshi Uchida et alii
Development of Centrifugal Compressor for 100kW Automotive Ceramic Gas Turbine
ASME 94-GT-73, 1994

[25] C. J. Michael, J.E.- Halle
Energy Efficient Engine
Low-Pressure Compressor Component Test Hardware Detailed Design Report
NASA CR-165354, 1981

[26] R. Schnell, A.Giebmanns, E. Nicke, T.Dabrock
Aerodynamic analysis of a fan for future ultra-high bypass ratio aero engines
AIAA Paper 2009 - 1149, ISABE 2009, Montreal, Canada

[27] D. E. Hobbs, H. D. Weingold
Development of Controlled Diffusion Airfoils for Multistage Compressor Application
Journal of Engineering for Gas Turbines and Power
April 1984, Vol. 106/271, ASME Paper 83-GT-211

[28] T. L. Creason, S. Baghdadi
Design and Test of a Low Aspect Ratio Fan Stage
AIAA-88-2816, 1988

