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EVALUATING COMPLEX INLET DISTORTION WITH A PARALLEL COMPRESSOR MODEL: PART 1 – CONCEPTS, THEORY, EXTENSIONS, AND LIMITATIONS

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ABSTRACT

Modeling compression systems using parallel compressor theory has been used for the analysis of compression system operability since the 1960s. Parallel compressor models have been traditionally designed and used for the analysis of circumferential distortion effects as a means to evaluate the impact of various inlet flow field disturbances on compressor operation.

This paper (the first of two) provides a review of the parallel compressor concept and discusses extensions to the original theory. These extensions include the incorporation of dynamic response, application to complex distortions, and the application to inlet swirl. Understanding these effects and the application of parallel compressor theory extensions is required to produce analytical models and computer simulations that can be used to enhance the development testing and the understanding of the response of gas turbine compression systems. Once a computer simulation has been constructed for a particular test article, it can be exercised and results compared against test results where distortion-generator devices (such as distortion screens) have been used, generally with favorable accuracy. The usefulness of the extended parallel compressor model is derived from its ease of use, simplicity, and ability for quick turn-around of results. It is often more desirable to have an analysis capability that is easy and quick to use than to have one that is extremely accurate, especially when understanding basic physics is of primary concern during a test operation. Extreme accuracy may require large amounts of computer resources and take days or weeks to compute a single performance point. While this may be acceptable for design, the limitations of high-fidelity simulations make them impractical to use due to the time constraints imposed by the pace of testing. Applying a timely analysis capability, using a parallel compressor simulation can provide a new physical understanding of the effects of complex distortion during the testing process when comparing the analytical and test results. This concept is presented in two companion papers: the first paper, Part 1, concentrates on the parallel compressor concepts, theory and limitations of the methodology while the second paper, Part 2 [1] presents applications of the approximate methods developed and compares results with experimental data.

INTRODUCTION

The effects of distorted inflow to turbomachinery have been a concern since the early days of engine installations. Whether stationary power units or aircraft engines, designs are performed assuming a particular inlet flow characteristic. Variations in the inlet total pressure, total temperature, or flow angle from the design flow characteristic all have an effect on the operation of the turbomachine. The characterization of these effects has been a continuous effort for turbomachinery designers since the early days. Many attempts to characterize the variations and the performance effects of distorted inflow have been made, with a variety of successes. The goal of this paper is to review the known technology and to discuss extensions to the parallel compressor model that can be made to provide a useful tool for the analysis of not only simple distortion but complex distortion as well.

NOMENCLATURE

c	absolute velocity
h	specific enthalpy
P,p	pressure
s	entropy
t	time

T temperature
u internal energy, velocity
U blade velocity
v average relative velocity
w relative velocity

ρ density
θ circumferential extent
$\Delta Pc/P$ circumferential pressure distortion intensity
$\Delta Pr/P$ radial distortion intensity
$\Delta Tc/T$ circumferential temp. distortion intensity

Subscripts
avg average
b blade
i ring counter
n number of measurement probes in the inlet system, or Fourier coefficient counter
o stagnation property
r radial
rel relative
REF reference
s static
ss steady state
T, t total
x coordinate
q tangential

HISTORICAL PERSPECTIVES

With the earliest engines in the 1930s, the prime concerns addressed were of the turbomachine itself, and other than trying to maintain a "smooth" flow into the machine, inlet/engine integration was not a big concern. As the development of the gas turbine engine proceeded into the 1940s and 1950s, the issues associated with inlet flow regimes began to arise. Largely during the 1960s, it was recognized that the total pressure conditions of the inlet flow had an effect on the performance of the turbomachine and designers attempted to characterize the inlet flow conditions with respect to the operation of the engine. The fact that a total temperature variation could be an issue was not recognized until military applications developed to the point where gun gas ingestion and rocket gas ingestion became an issue. Later, inlet swirl effects due to the shape of the inlet itself became recognized as another initiator of compression system stability problems.

The first basic work that started to examine compression system stability and dynamics as a function of inlet total pressure variation was that of Pearson and McKenzie [2], who first proposed the parallel compressor theory. This theory proposed that a compression system under the influence of a total pressure distortion (of a one-per-rev form) could be treated as two compressors operating in parallel. Each of the "parallel" compressors would operate as if each were under undistorted flow conditions, with one inlet corresponding to the original compressor's distorted side and the other corresponding to the original compressor's undistorted side. The two "parallel" compressors are assumed to discharge to the same static pressure. This technique of analyzing inlet distortion assumes that the compression system responds to changes in inlet flow conditions in an instantaneous manner.

Several years later, Reid [3] showed that for small circumferential extent inlet distortion patterns, the parallel compressor model did not hold true. He concluded that there must be some critical angle of circumferential extent for a compressor to respond in an instantaneous manner. Figure 1 is a depiction of the parallel compressor model with and without Reid's concept of a critical angle. It is easy to see the logic of Reid's concept, which depicts the loss in pressure ratio at the compressor stall point at zero circumferential extent distortion as being zero. However, at some angle between 0 and 360 degrees, the loss in stall pressure ratio is a maximum. At 360 degrees, what is distorted doesn't look distorted at all (since the inlet flow is uniform) and the compressor and the loss in compressor pressure ratio at stall is again zero. More on the critical angle concept will be discussed later in this paper.

From the time of Reid's work through the mid-1970s, manufacturers of gas turbines characterized inlet total pressure distortion with correlations developed from test

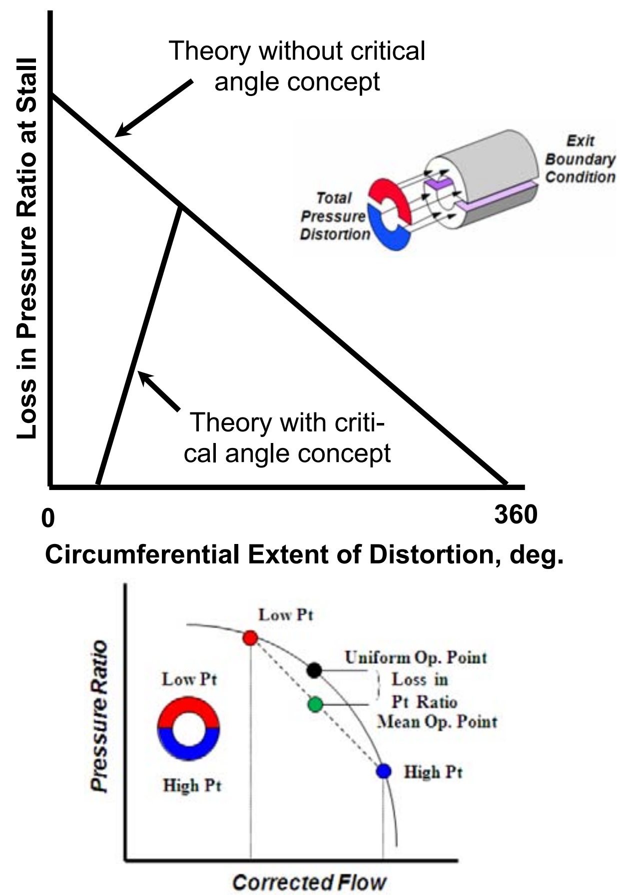

Fig. 1 Parallel compressor theory prediction with and without the critical angle concept [22, 28]

data, mathematical analysis, and simple attempts at modeling. During this time, so many ways of describing inlet total pressure distortion developed that it was not possible to communicate between engine manufacturers, airframers, and customers. Several key works were produced, among them that of Kimzey [4] and Adamczyk and Carta [5]. A committee was formed as a working group under the auspices of the Society of Automotive Engineers (SAE), called the S-16 Committee, made up of representatives of the engine manufacturers, airframers, and customers. As a result of the work of this committee, an aerospace recommended practice (ARP) document was developed and given the annotation of ARP-1420 and titled "Gas Turbine Inlet Flow Distortion Guidelines" [6]. Although the methods proposed in the ARP-1420 document were jointly developed, at the time, no engine manufacturer stopped using their own descriptors and this just added another method to the confusion.

To evaluate and compare all the distortion indices and clear up some of the confusion that existed, the Navy sponsored work at Virginia Tech's Center for Turbomachinery and Propulsion Research to evaluate twelve different distortion indices and the parallel compressor model. This work was performed by Campbell [7] and is a key piece of work in the evaluation of distortion indices. Twelve indices were evaluated, including

1. the common $\Delta P/Pavg$ indices,
2. the Rolls Royce θ critical indices,
3. the NAPC Kθ index,
4. the AVCO Lycoming DI and DIc indices,
5. the Garrett AiResearch CDI index,
6. the Pratt & Whitney KD_2 index,
7. the Pratt & Whitney Ka_2 index,
8. the General Electric Method D system, and,
9. the ARP-1420 method.

Statistical methods were used to determine the accuracy of all the indices. The data used with all the indices included data from the J85-GE-13 turbojet, the TF30-P-3 turbofan, and the T64-GE-6B turboshaft engine. No single index produced the best results for all the data sets, but the ARP-1420 method did provide consistently good correlations. This work helped to set the direction and use of the ARP-1420 methods as a key technique in the industry. Significant references supporting the development of distortion indices are provided in Ref. [7]. In 1983 the Aerospace Information Report number 1419 (AIR-1419) [8] was published. This document discusses many aspects of inlet distortion, summarizes the ARP-1420 procedure, discusses instrumentation, and provides guidelines for the production of engine analytical stability assessments.

While total pressure distortion was the main emphasis for many years, total temperature distortion later became an area of concern as military applications with gun and rocket gas ingestion, vertical take-off, and aircraft carrier operation (steam ingestion) became issues. In commercial applications, ingestion of the exhaust from other aircraft (runway situations) and thrust reverser operation indicated the importance of total temperature distortion. In 1991, the S-16 Committee produced an aerospace resource document (ARD) on temperature distortion [9], which presented an analysis technique similar to that of total pressure distortion.

For over thirty years, airframers and engine manufacturers have defined an aerodynamic interface plane (AIP, often at the engine front flange) as a "connection point" for the airframer's inlet design and the engine manufacturer's engine. Characteristically, the engine distortion tolerance requirements and the inlet distortion generated by the inlet and airframe have been satisfied at this inlet plane. It has become apparent in recent work, that this "piecewise" integration of the inlet and the engine is not the best way to develop the engine system. Better integration of the inlet and the engine (rather than just passing-off information at the AIP) is possible and necessary for future advances in system performance and engine stability. However, the information produced at the AIP is still valuable, as it provides a common interface at which to discuss flow properties, measurements, and analyses. It is at this interface that the parallel compressor model is implemented.

MATHEMATICALLY DESCRIBING AND ANALYTICALLY SIMULATING THE FLOW PHYSICS OF INLET DISTORTION

Total pressure distortion, total temperature distortion, and swirl are all significant contributors to the degradation of stability margin in an engine [10]. There are various methods that can be found to analytically simulate inlet distortion conditions and their effect on compression system stability (such as the CFD work by Gong [11], Chima [12], Hale [13], and others), but often these methods are too complex and take too long to set up and run if looking for guidance related to system interactions during a test program. In cases such as these, the parallel compressor model is still a useful tool, but it requires some enhancements and extensions beyond the basic parallel compressor model previously discussed. These enhancements and extensions are best developed in concert with today's commonly used distortion descriptors (the SAE descriptors) so that a commonality is maintained in describing the true flow physics in both the test data and the simulation results.

THE TOTAL PRESSURE DISTORTION DESCRIPTOR, FOUND IN ARP-1420

Examination of all the well-known distortion descriptors presented in the work by Campbell [7] shows that all the descriptors have one thing in common. That is, they all attempt to describe the total pressure inlet distortion by describing not only the amount of the pressure degradation (the intensity), but also the shape of the distortion pattern. This shape is important because it is directly related to the time response of the inlet/engine integrated system. In ARP-1420, the distortion intensity is divided into two important physical values, the circumferential intensity (ΔPc/P) and the radial intensity (ΔPr/P).

The circumferential distortion intensity is defined as

$$(\Delta Pc/P)_i = (P_{avg\,i} - P_{avg\,low\,i}) / P_{avg\,i} \quad (1)$$

$$P_{avg\,i} = 1/360) \int_0^{360} P(\theta)\,d\theta = \text{ring average pressure} \quad (2)$$

(area under the line between 0 and 360 degrees)

and

$$P_{avg\,low\,i} = (1/\theta^-_i) \int_{\theta^-_i} P(\theta)_i\,d\theta \quad (3)$$

(area under the line between θ_1 and θ_2)

where i denotes the “ring” and the extent of the distortion is θ_i^- as shown in Fig 2.

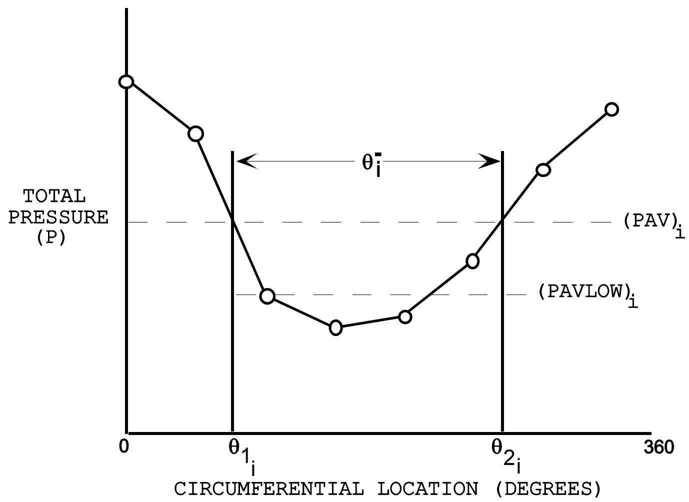

Fig. 2 Distortion description used in ARP-1420 for the calculation of circumferential pressure distortion intensity and defining the circumferential extent of the distortion [6]

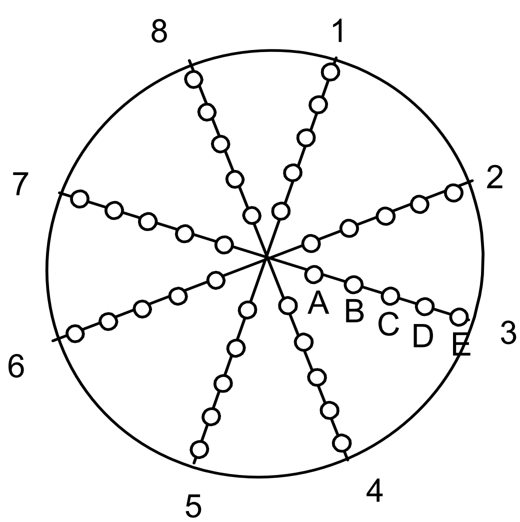

Fig. 3 Typical inlet rake setup defined in ARP-1420 [6]

This is measured on a ring of probes on a set of inlet rakes as shown in Fig. 3. ARP-1420 suggests a rake set made up of 8 rakes with 5 probes on a rake. This is discussed as a “typical” array, and many people read this in error as a requirement. It is not a requirement, but a suggestion of a configuration that works for inlets that have predominately one-per-rev flow features. Most of the early military inlets with high length-to-diameter (L/D) ratios fall in this category. In the work on the T800 engine in Refs. [14, 15], an array of 10 rakes with 4 probes per rake was used, because of the annulus configuration of the inlet. What is important is that the probe configuration is adequate to measure the flow patterns without missing a significant flow detail and without causing too much blockage. In addition, the number of rakes determines the harmonic content that one is able to calculate, as discussed in Ref. [10]. In some cases, there can be more than one low pressure region in the inlet. If there is more than one distorted (low pressure) region, separated by a high pressure region with an extent smaller than a *minimum value* of the positive pressure region (θ_i^+), then the two low pressure regions are considered as one, as shown in Fig. 4.

The minimum value of θ_i^+ is known as the critical angle. This is the same critical angle that Reid [3] discussed

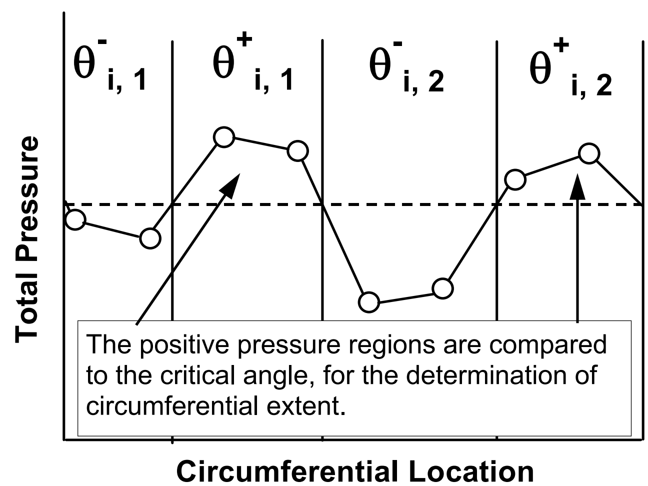

Fig. 4 ARP-1420 description of multiple-per-rev distortion, showing the positive pressure regions, θ^+_i, that relate to the critical angle [6]

in his work with the parallel compressor theory. ARP-1420 says this critical angle must be “determined by experiment,” or in the absence of data, suggests using 25 degrees. Today, the flow physics that lead to a critical angle are better known, as this is related to the time constant of the blade response, which will be discussed in this paper during the discussion of the relationship of rotor dynamics to changes in the inlet flow conditions.

In addition to the circumferential distortion intensity parameter just discussed (ΔPc/P), ARP-1420 calls out the radial distortion intensity (ΔPr/P), defined as

$$(\Delta Pr/P)_i = (P_{face\ avg} - P_{avg\ i}) / P_{face\ avg} \qquad (4)$$

THE TOTAL TEMPERATURE DISTORTION DESCRIPTOR, FOUND IN ARD-50015

The total temperature distortion descriptor, as described in ARD-50015, is similar to the descriptor for pressure distortion, although the concern is the high temperature side of the pattern, rather than the low (Fig. 5). In this case, the temperature distortion intensity is defined as **($\Delta Tc/T$)**, where

$$(\Delta Tc/T)_i = (T_{avghi\ i} - T_{avg\ i}) / T_{avg\ i} \qquad (5)$$

$$T_{avg\ i} = (1/360) \int_0^{360} T(\theta)_i \, d\theta = \text{ring avg temperature} \qquad (6)$$

(area under the line between 0 and 360 degrees)

and

$$T_{avghi\ i} = (1/\theta^+_i) \int_{\theta^+_i} T(\theta)_i \, d\theta \qquad (7)$$

(area under the line between θ_1 and θ_2)

and where as with the θ_i^- extent for pressure distortion, the circumferential extent is described for temperature distortion by θ^+_i. Similar descriptions for temperature distortion (as discussed for pressure distortion) are found in ARD-50015 for the multiple-per-rev content and for radial distortion.

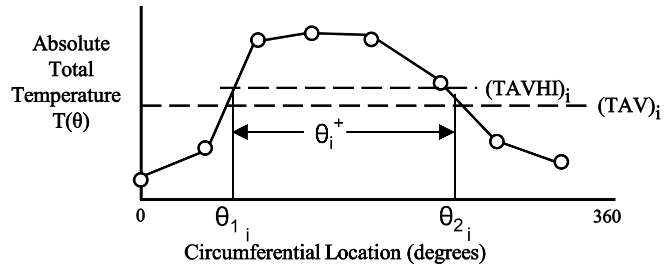

Fig. 5 Circumferential distortion description used in ARD-50015 for the calculation of circumferential temperature distortion intensity and defining the circumferential extent of the distortion [9]

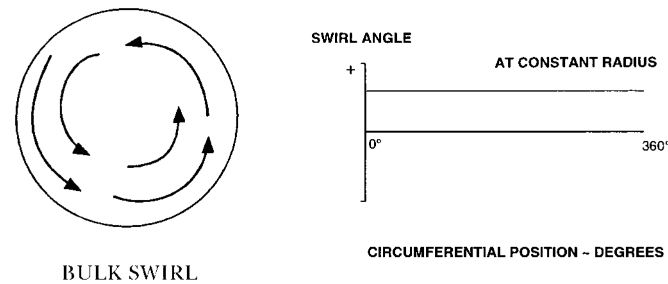

Fig. 6 Projected Velocity Vectors Produced with Bulk Swirl [16]

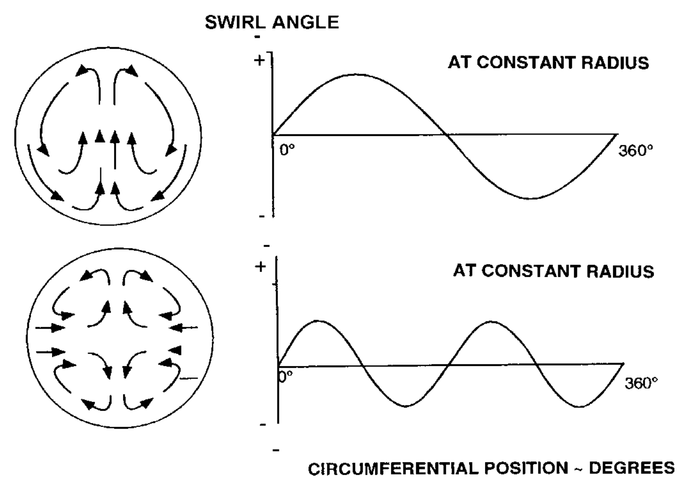

Fig. 7 Projected Velocity Vectors Produced with Paired Swirl [16]

DESCRIPTIONS FOR INLET SWIRL

For many years, people considered the application of the pressure distortion descriptors as an adequate representation of the inlet flow field. Interestingly enough, it is entirely possible to have a total pressure distortion profile that looks rather benign, while having a swirl composition that is in fact, completely unacceptable.

Swirl can be characterized as either bulk swirl or twin swirl (sometimes called peak swirl). It is important to note that the different types of swirl have a different effect on the compression system; therefore, they are characterized separately.

Bulk swirl is characterized by the rotation of the complete flow field coming into the engine (Fig. 6). The swirl is either clockwise or counter-clockwise and is generally described with respect to the direction of rotor rotation as being either pre-swirl or counter-swirl. These two swirl directions have different effects on the compression system component. Counter-swirl, being in the direction opposite rotor rotation, causes an increase in incidence angle to the rotor and as a result, a higher rotor loading and a reduction in stall margin. Pre-swirl, being in the direction of rotor rotation, causes a reduction in incidence angle and therefore a reduction in the flow capacity of the rotor.

Paired swirl, in contrast to bulk swirl, is characterized by having swirl in one direction on one side of the rotor and in the other direction on the opposite side, often with two different magnitudes (Fig. 7). This type of swirl is typically found in installations that have side inlets, such as many helicopters, and also s-duct inlets, commonly found today with embedded engine installations. The physics of swirl, as with inlet distortion, are just a function of basic fluid mechanics, that is, flow in a curved duct generates a static pressure non-uniformity, resulting in the formation of streamwise vorticity, which presents itself as a swirl component. The Society of Automotive Engineers (SAE) S-16 Committee has documented a methodology for considering swirl as part of the inlet-engine compatibility assurance process [16].

COMPRESSION SYSTEM DYNAMIC RESPONSE TO INLET DISTORTION

Since the original work on the critical angle concept by Reid [3], there has been much work done to determine exactly what the physics of compression system response truly are. ARP-1420 indicates that this is determined by test, but says nothing about how this should be accomplished (mainly because not enough work was done at the time to determine this). It suggests using 25 degrees as a critical angle in the absence of any more information, but doesn't say why. Many people make the mistake of thinking that 25 degrees is a *specification* of ARP-1420, but this is not the case. Other work by Oates [17], Plourde and Brimlow [18], and Brimlow, et al. [19] supported the theory that a compressor system would respond to total pressure distortion patterns which persisted on the order of one-per-rev.

Some of the early work performed by Van Deusen and Mardoc [20] showed that the steady state distortion tolerance of a TF30 engine was found to vary inversely as a function of the turbulent energy in the flow field. Therefore, they concluded, the turbulence as well as the steady -state distortion was a prime variable in predicting engine response.

At about the same time, Goethert and Reddy [21] developed the concept of reduced frequency, which is described as the ratio of the residence time of flow in a blade passage to the time the blade is in the disturbance.

In more recent work, Cousins [22, 23] showed that the rotor response time for an axial blade can be calculated as the time for a flow particle to travel the distance from the leading edge of an airfoil to the throat of the blade passage (Fig. 8). This defines the flow physics that determine the sensitivity of the blade to inlet distortion. Cousins presented data that shows the delay in rotor response measured with on-rotor pressure transducers. The key contribution of this work is that it shows the difference in axial and centrifugal compressor rotor response time and explains the critical angle concept in terms of true flow physics.

Pressure rise in all compressors occurs by the conversion of shaft work to fluid pressure. Figure 9 shows the h-s diagram for an axial stage. The enthalpy change across the rotor is represented by

$$h_2 - h_1 = 1/2\,(w_1^2 - w_2^2) \qquad (8)$$

as long as there is no radial shift in the streamlines across the rotor. In reality, there is generally a very small radial shift, due only to the slight convergence of the flow path. This convergence provides insignificant pressure rise when compared to the radial change that occurs in either a mixed-flow rotor or in a centrifugal impeller. (The im-

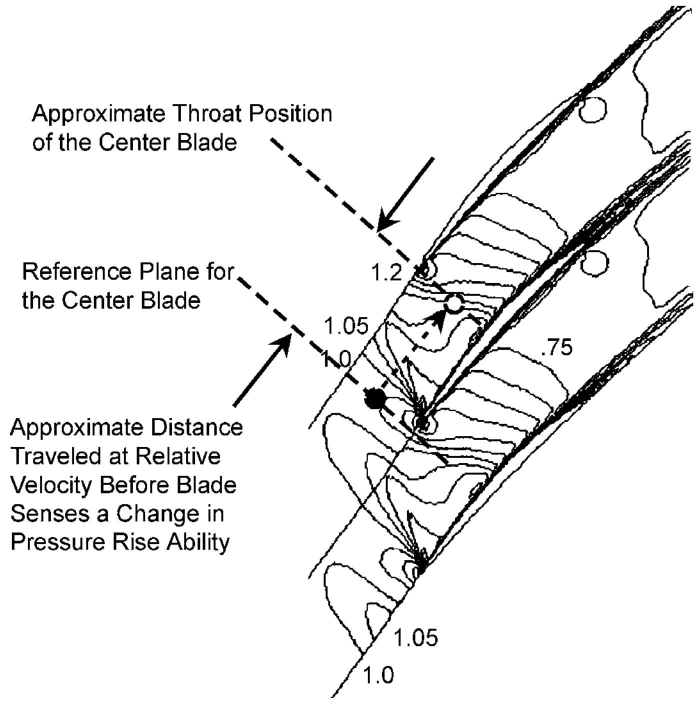

Fig. 8 Distance a particle of fluid must travel to the throat of the blade passage to reach the start of the major diffusion process [22, 23]

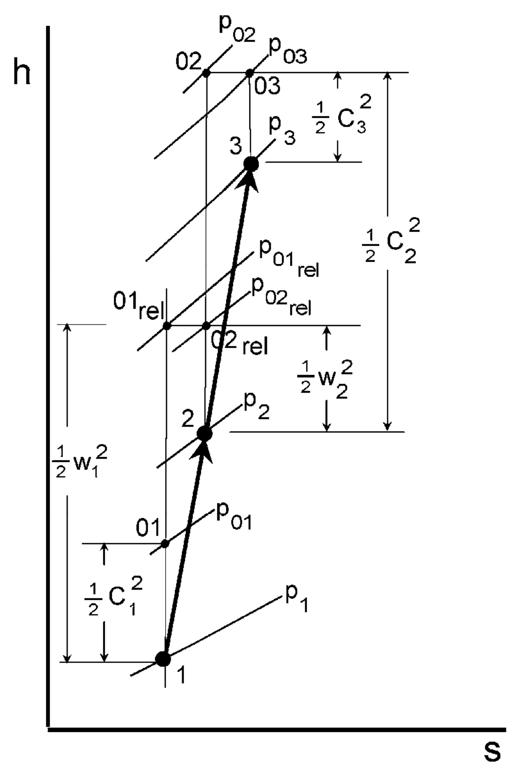

Fig. 9 Enthalpy-entropy diagram for an axial compressor stage [23]

pacts of radial change are discussed in Ref. 23.) For the axial blade, once the blade senses the change in the inlet flow (the particle of fluid reaches the throat), there is a chance that the diffusion process along the remaining portion of the blade can be disrupted, causing flow separation (Fig. 8). Once the flow is separated on an axial blade, the pressure rise capability is severely limited. When a blade passes behind a distorted sector, its response is critical to the impact of that distorted sector on the compression system (Fig. 10). A more complete discussion on this is provided in Cousins [22, 23], where the discussion includes centrifugal impellers.

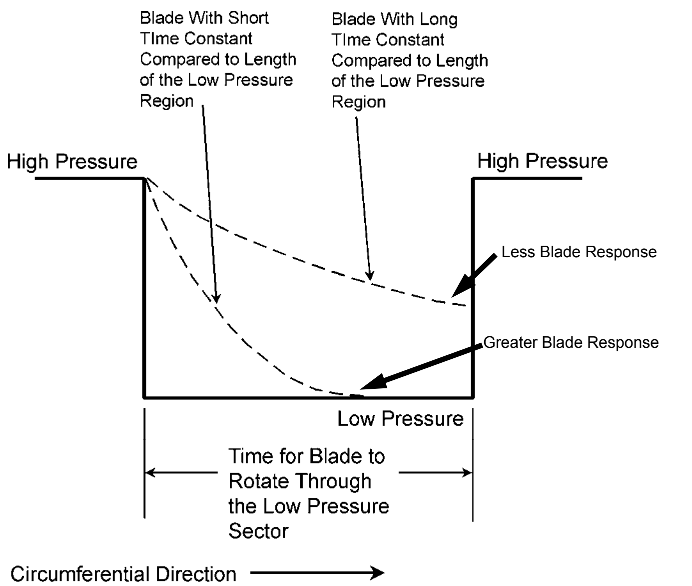

Fig. 10 Blade time constant comparison to a distorted inflow determines the magnitude of the blade response [23]

THE IMPORTANCE OF DYNAMIC DISTORTION

Steady-state distortion measurements do not capture the complete nature of the inlet distortion affecting the stability of the engine compression system. The true nature of inlet flows is to have some dynamic component. In measurements with "standard" measurement probes, this component of the pressure is not obtained. Figure 11 shows a depiction of a pressure trace (which would be from steady-state probes) with the true dynamic component drawn on top. From this pictorial, one can see that the "true" value of pressure distortion is not captured with the steady-state probe. Measurements have shown that this dynamic component can add as much as thirty percent to the steady-state pressure signal. Typically, screens are used to generate a simulated pressure distortion pattern to match that of an inlet wind tunnel test. To account for the dynamic content of the real measurement, the steady-state screen is typically designed either at the RMS value or the peak value of the dynamic measurement. Realizing this is important in the application of extensions to the parallel compressor model.

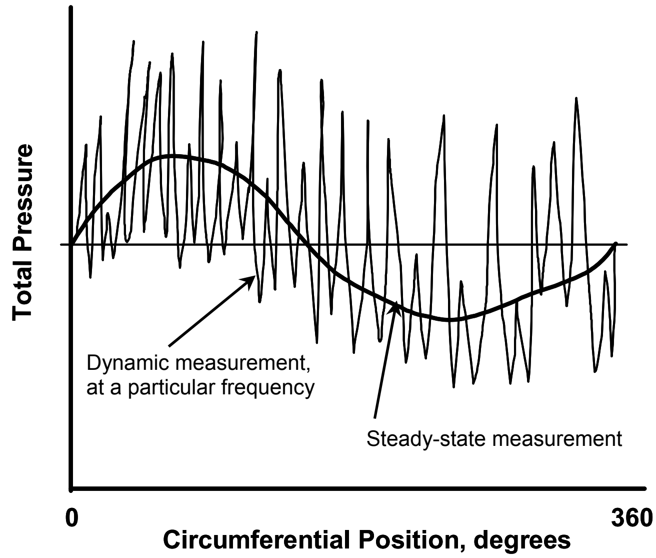

Fig. 11 Graphic pictorial showing steady-state pressure measurement and a high-response measurement [10]

THE DYNAMIC MODEL UPON WHICH PARALLEL COMPRESSOR EXTENSIONS ARE APPLIED

Many one-dimensional models have been developed to aid in the analysis of engine performance and operability. These models provide insight into physical phenomena that may be difficult to understand using test data alone. The present modeling technique (the DYNTECC code [25]) uses a finite difference scheme to simultaneously solve the mass, momentum, and energy equations with or without turbomachinery source terms (mass bleed, blade forces, heat transfer, and shaft work). The compression system source terms are determined from a complete set of stage pressure and temperature characteristics provided by the user. A representative single-spool multistage compression system is shown in Fig. 12, along with the control volume models. The overall control volume is divided into a set of elemental control volumes. Typically, the compressor section is subdivided by stages either as rotor-stator or stator-rotor, depending upon how the experimental stage characteristics are obtained. Acting on the fluid control volume is an axial force distribution, FX, attributable to the effects of the compressor blading and the walls of the system. These effects are lumped together due to the difficulty of separately distilling the effect of each from experimental data. Appropriate inlet and exit boundary conditions are applied at the inflow and outflow boundary locations. Energy supplied to the control volume included the rate of heat added to the fluid, Q, and the shaft work done on the fluid, SW. Mass transfer rates across boundaries other than the inlet or exit (e.g., interstage bleeds) are represented by the distribution W_B.

Time dependent boundary conditions can be specified either at the entrance or the exit of the overall control vol-

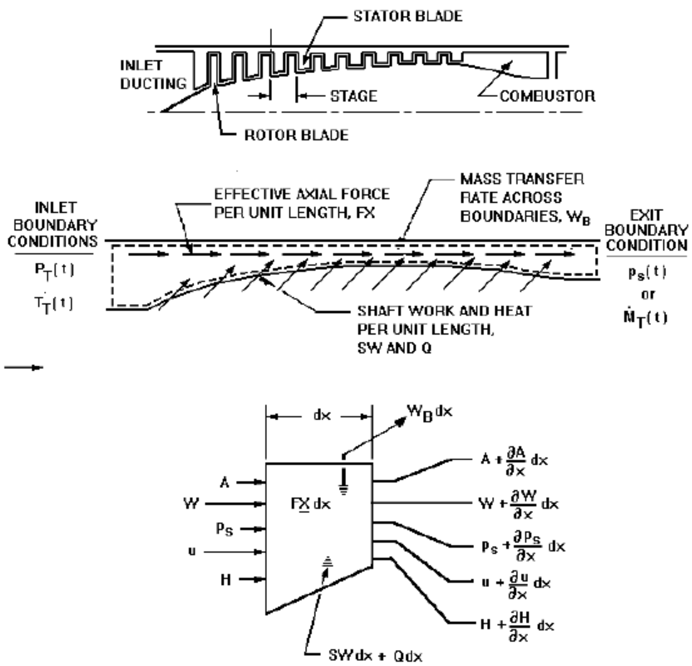

Fig. 12 Representation of the dynamic model to which the parallel compression theory is applied—DYNTECC [25]

ume. Inlet total temperature or pressure time history may be linearly ramped, varied cyclically, or held constant. The same is true for the overall control volume exit pressure, Mach number, and airflow rate. At the entrance, both total pressure and total temperature must be specified. At the exit, however, only one parameter may be specified, usually static pressure or exit Mach number. This model has been used in many investigations, including those in Refs. [23] and [24].

EXTENSIONS TO THE PARALLEL COMPRESSOR MODEL

Applying extensions to the parallel compressor model to enable the simulation of complex distortion (and better simulation of simple patterns) must be performed with careful regard to the flow physics of a compression system. Several items must be considered in the formulation of model extensions. First, the goal is to keep the extensions simple, yet still be physically applicable to the system. Second, it is important to realize that there is a balance between simplicity and solution accuracy. Since the desire is computational speed, along with providing a reasonable interpretation of the trends of total pressure, total temperature, and swirl effects, the accuracy may not be precise enough for design applications. In the testing environment, however, rapid interpretation of trends and effects are critically important and long calculation times (as with many CFD analyses of distortion effects on a compression system) are not acceptable. While this simplified approach may yield less accurate solutions, the method can produce useful guidance and results for testing applications given adequate engineering understanding of the method.

The following extensions to the parallel compressor model are considered:

- increased definition of circumferential distortion effects by mass redistribution,
- inclusion of radial distortion effects by mass redistribution,
- inclusion of swirl effects,
- application of time constants for compression system response,
- dynamic effects near stall, and
- inclusion of a meanline model to eliminate the need for complete compression system maps in the parallel compressor model

Circumferential Mass Redistribution

Circumferential pressure distortion can generate a redistribution of flow in the circumferential direction due to the differences in local static pressure, as shown in Fig. 13.

This cross flow occurs in the rotor-stator gap and can be approximated using a simple orifice flow analogy, developed by Kimzey [4]. The high and low pressure regions can be treated as reservoirs and the rotor-stator gap as an orifice. The flow can be approximated using a simple algebraic expression based on classic orifice flow.

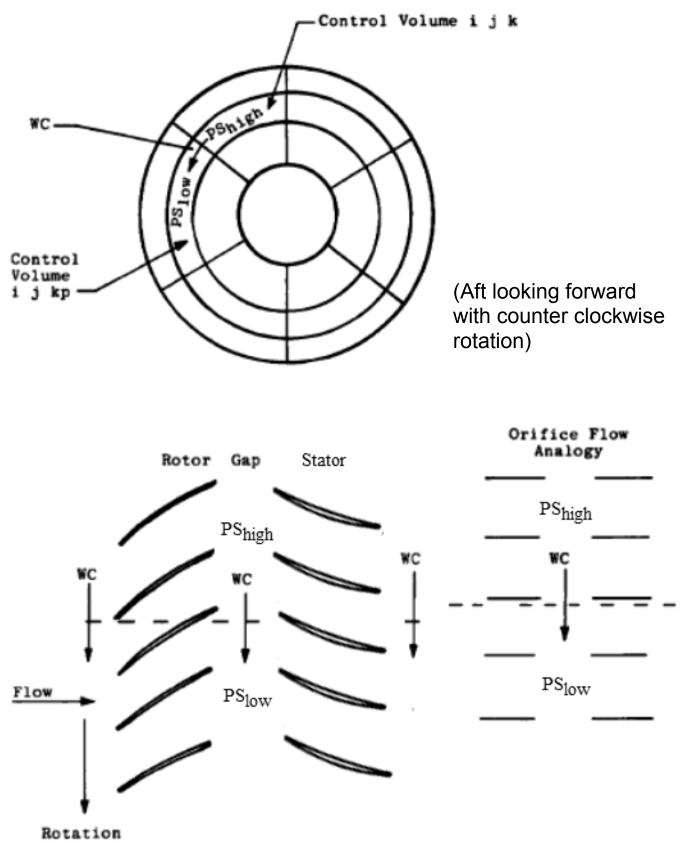

Fig. 13 Circumferential flows are driven by differences in static pressure around the compression system stages [28]

Since flows with small pressure drop can be treated as incompressible, the continuity equation and Bernoulli's equation are used, and the mass flow rate is expressed in terms of a pressure difference and an orifice flow coefficient.

$$W_{cross} = C_{xflow} * A_{gap} [2\rho (Ps_{high} - Ps_{low})]^{1/2} \quad (9)$$

The cross flow coefficient, C_{xflow}, is analogous to the orifice coefficient. For an orifice flow with a pronounced vena contracta and high Reynolds number, the value of the orifice coefficient converges to 0.6, which can be used as a default in the parallel compressor model, to allow a calculation of flow redistribution between the circumferential segments.

Radial Mass Redistribution

Radial mass flow redistribution can occur in a compressor through the passages between the compressor blades. The radial mass flow is similar to the circumferential mass flow redistribution and is driven by differences in static pressure, as shown in Fig. 14.

A relationship similar to that derived for the circumferential flow redistribution can be derived. However, the relationship is not presented here because all attempts to create a radial redistribution model resulted in unsatisfactory results in the parallel compressor simulation. It is believed that the problem with trying to create a radial redistribution model is rooted in the fact that with a simple model such as the parallel compressor model, the radial redistribution actually changes the design radial equilibrium relationship, and therefore the blade loading distribution from hub to tip. With these changes, the relationship between the stage characteristics that are used in the formulation of the parallel compressor model are no longer correct. Attempts at redefining the characteristics were unsuccessful, as the radial distribution does not account for the shift in the radial work distribution that really occurs. This also becomes an issue when examining the stability limit with the parallel compressor model. Typical stalling behavior generally occurs at the tip of the rotor, due to the tip loading. Machines that stall at the hub usually have more influencing the stalling behavior, like a highly loaded hub-flowpath, for example. As a result of this examination, one is strongly cautioned about trying to use a parallel compressor model to predict compression system operation under conditions where the radial distortion is a significant factor.

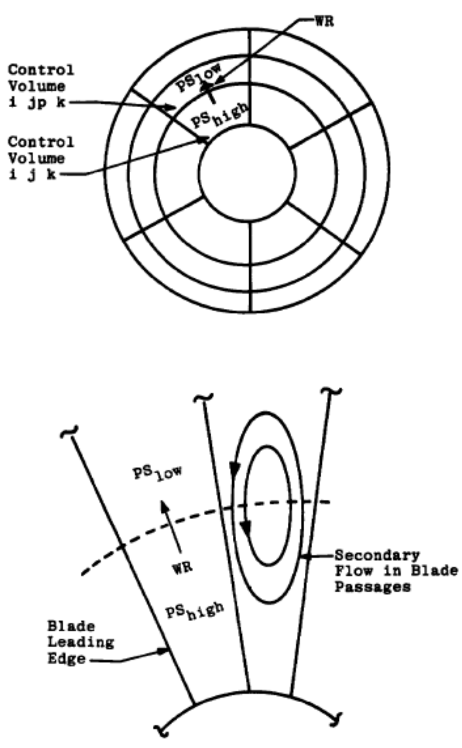

Fig. 14 Radial flows are driven by changes in static pressure (due to the distortion) radially between the blades, actually changing the design radial equilibrium conditions [28]

Inclusion of the Effects of Swirl

To include the effects of swirl in the parallel compressor model, stage, blade-row, or overall system maps must be generated that include the effects of swirl. With these provided, the different sectors of the parallel compressor model can query the appropriate map to obtain the operating point of the given off-design speed line. The off-design maps can be generated from a meanline design code (or even a more complex CFD code) since the goal is to develop a set of maps that contain the effect of varying incidence angle at the front of the compression system. For some configurations, an inlet guide vane (IGV) will control some of the swirl, but at high incidence angles, the IGV will itself stall, causing a high distortion entering the first rotor. In applications where no IGV exists, the incidence angle imposed on the first rotor can cause either movement of the rotor towards stall or towards choke. Providing the stage characteristics to the parallel compressor code under swirl conditions enables the model to obtain the operating point information for any parallel compressor sector under any incidence angle (swirl) conditions. (More on this can be found in Refs. [1 and 10].)

Compression System Response

There are a number of ways the response of the compression system can be included into the parallel compressor model. As discussed previously, a rotor blade passing through a distortion feels the effect of an unsteady flow (due to changing incidence and therefore changing relative velocity). Both the analyses by Goethert and Reddy [21] and the analysis by Cousins [22, 23] (Figure 8) can be expressed as a ratio of the unsteady lift over the airfoils to the steady state lift. This ratio of unsteady lift $|C_l|_{max} / |C_l|_{ss}$ can be shown to be a function of the blade time constant [23], which includes dependence on the reduced frequency of Goethert and Reddy and the airfoil design (passage throat location as discussed by Cousins). Since the dynamic value of the lift coefficient

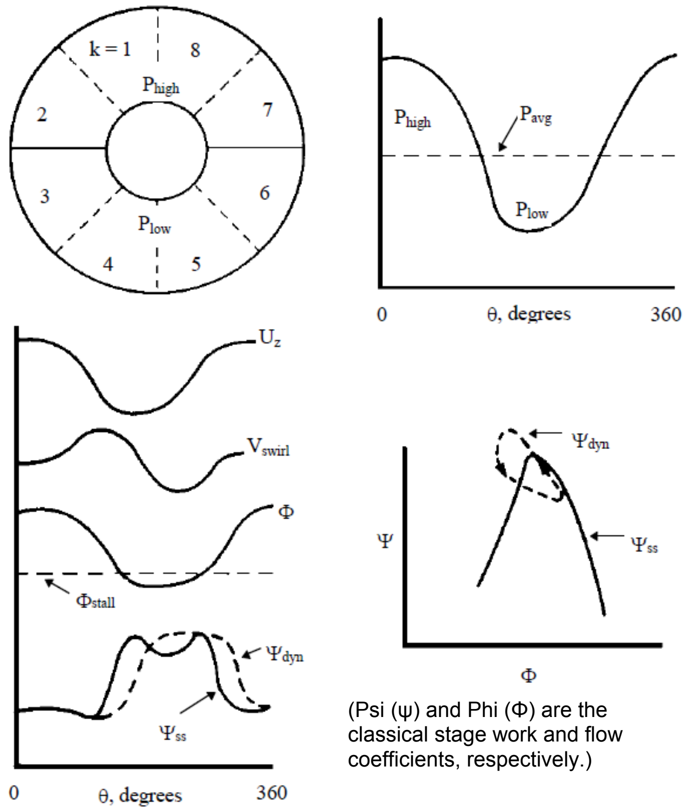

Fig. 15 Model responses to flow disturbances are damped by the lagging of the stage loading [28]

has the same characteristics as the blade stage loading, the ratio of both is equivalent and

$$|C_l|_{max} / |C_l|_{ss} = |\psi|_{max} / |\psi|_{ss} \tag{9}$$

This relationship is called a dynamic lag ratio (DLR) by Kimsey and is included in the dynamic parallel compressor model as a lagging factor on the stage loading as:

$$\psi_{present\ c.v.} = \text{DLR}\ \psi_{present\ ss\ c.v.} + (1 - \text{DLR})\psi_{previous\ c.v.} \tag{10}$$

The effect of the lag ratio is to damp the model response to a flow disturbance (Fig. 15). This is necessary since the characteristics used to determine the stage loading parameter is based upon compressor steady-state operation. Through this method, the dynamic response of the rotor blade is taken into account in the simulation. This is especially important as the compression system nears instability, as the blade dynamics can affect the order in which the stages stall as the loading nears the stability limit.

Inclusion of a Meanline Model

Since the stage characteristics are input to the dynamic model as a family of "look-up-table" characteristics (total pressure ratio, total temperature ratio, corrected speed and corrected flow), these characteristics must be developed prior to model operation (Fig. 16). Families of these characteristics must also be developed to model the

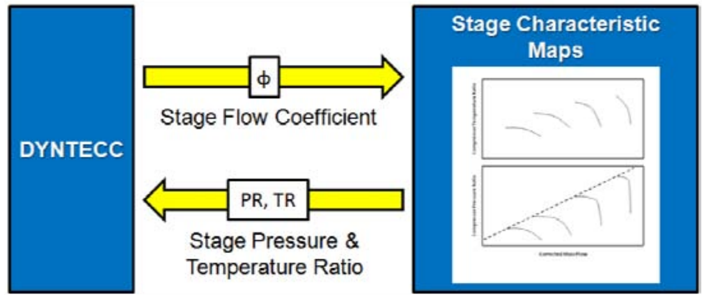

Fig. 16 Typical model operation without a meanline code involves a table look-up operation from characteristics to obtain stage pressure and temperature ratio [27]

impact of swirl on the compression system. Stalling behavior is typically determined by examination of the slope of the pressure stage characteristic. When the slope of the stage characteristic becomes flat, stall for that blade row is indicated. This is a simplifying assumption as some blades can operate on the positive slope of the pressure characteristic.

Integration of a meanline code as a subroutine rather than the tables allows the model to calculate the required pressure ratio and temperature ratio values rather than looking them up from a table (Fig. 17). This also allows the information about incoming swirl angles to be passed to the meanline code so that the proper pressure ratio and temperature ratio can be calculated at the appropriate incidence angle due to swirl.

Another advantage of incorporating a meanline code into the dynamic parallel compressor code is that better predictions of the stability limit can be made, due to more information (such as diffusion factor) being available in the meanline code. Examination of the diffusion factor can provide better prediction of stage stalling since it is a function of the blade design. While typical values of stalling diffusion factor are available for standard circular arc blades (among others) in the NASA SP-36 publication [26], one must be careful as todays airfoils (controlled diffusion airfoils, arbitrary airfoils, etc.) have diffusion factors at stall that can be higher than those in Ref. [26]. That generally does not cause a problem, as the blade design characteristics are generally known as are the diffusion characteristics for the airfoils.

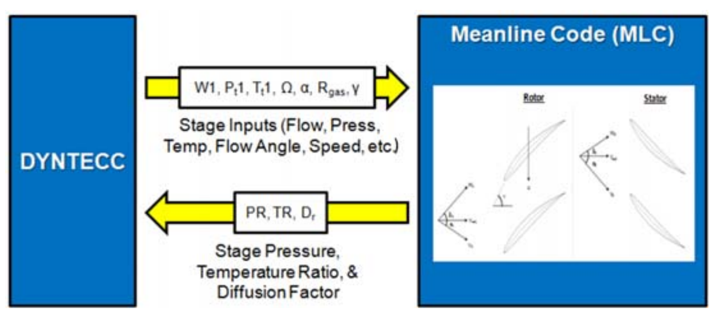

Fig. 17 Model operation with a meanline code eliminated generation of characteristics before operation and provides easy implementation of swirl response and loading parameters [27]

CONCLUSIONS

Parallel compressor theory can be modified to incorporate many basic flow physics of compression systems, and incorporated into a dynamic model. Enhancements provide the capability to simulate more complex distortion patterns, including temperature, pressure, and swirl. Radial distortion effects can not be included easily, due to the changes in radial work distribution that occur. It does not seem that simple models can provide adequate flow physics to make the modeling of radial distortion adequate in a simple parallel compression system model.

The companion paper to this paper, Ref. [1] and other references such as [23] and [28] present many applications to which this modeling system has been applied. The parallel compressor theory, along with extensions, provides an analysis capability that can be used in environments where rapid turn-around of testing trend results is required.

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