Def Sci J, Vol 38, No. 4, October 1988, pp 397-412

Gas Turbine Engine Component Development : An Integrated Approach

Willem Jansen

Northern Research and Engineering Corporation
Woburn, Massachusetts 01801, USA

ABSTRACT

Computer-aided engineering methods have made a significant impact in the design technologies of advanced machinery. These methods have been applied in several areas such as aerodynamic and fluid dynamic theory for high efficiency, stress and vibration theory for reliability, and manufacturing strategies to produce machined components at low cost and with short time schedules. The integration of these various design technologies offer the opportunity for even greater productivity in the engineering design and manufacturing process.

This paper addresses the application of various engineering disciplines to the demand of producing a reliable, efficient design and the subsequent manufacture of components with short lead times through the interaction of these computer-aided engineering technologies. The concept is further illustrated by sample cases for a centrifugal compressor and a gas turbine.

1. INTRODUCTION

Computer-Aided Engineering (CAE) methods integrate design and manufacturing technologies. The design process must yield a balanced design that satisfies the often opposing goals of good efficiency, high reliability and reasonable manufacturing costs and time. Thus, the designer must sequentially address a variety of technological disciplines; the most common being : (a) Fluid dynamics theory, for high efficiency; (b) Stress and vibration theory, for reliability; and (c) Manufacturing strategies for low cost and compressed schedules.

Received 25 May 1988

In the following sections each of these technologies and their interactions will be discussed. Sample cases for a centrifugal compressor and a gas turbine will be used to illustrate the concepts.

2. FLUID DYNAMIC DESIGN CONSIDERATIONS

The performance of a turbomachine is often characterised by its pressure ratio (or head), flow, work and efficiency. These quantities are derived from fluid dynamic/hydraulic analyses and involve theory and test results. The pressure ratio follows directly from the rate of change of angular momentum, (ΔrC_θ), of the fluid times the angular velocity, ω, of the rotating part, and is decreased by the internal losses the flow experiences on its way through the machine. Thus for compressors

$$\frac{P_{out}}{P_{in}} = \left[\frac{T_{in} + \dfrac{\cdot\omega\, x(\Delta\, rC_\theta)}{J g_o c_p}}{T_{in}} \right]^{\frac{\gamma}{\gamma-1}} \quad \frac{\Sigma\Delta P_{loss}}{P_{in}} \tag{1}$$

where γ is the ratio of specific heats and c_p the specific heat of the medium; J and g_o are conversion factors. Equivalent equations are valid for turbines.

The efficiency, η, is given by the ratio of the useful work, and the actual work input and for pumps can be written as :

$$\eta = \frac{P_{out} - P_{in}}{P_{out} - P_{in} + \Sigma\Delta P_{loss}} \tag{2}$$

Thus to maintain a high efficiency, pressure losses must be kept to a minimum

2.1 The Origin of Pressure Losses

2.1.1 Identification

Pressure losses in compressors, and turbines arise from a variety of sources such as surface friction, leakages, flow incidences to the blades at off-design, boundary layer separation/diffusion, shock waves (high Mach number), and cavitation (low water pressure). The magnitude of these losses depends on the geometry that is used, and selecting the optimal geometry is the major objective of the design task.

2.1.2 Surface Friction

Surface friction plays a significant role when the surface is rough, in a relative sense. (Roughness for a 30 mm diameter machine is not the same as that for a machine with a one meter diameter). Smooth surfaces are preferred which lead to the use of machined surfaces or investment castings.

2. Leakage

Leakages occur across clearances and seals. Therefore, small clearances should be maintained to keep these losses low.

2.1.4 Boundary Layer Separation and Diffusion

The majority of the losses occur as a result of boundary layer separation due to rapid local deceleration of the flow. The flow separates from the surface when the velocity decreases by a factor of roughly 1.8. This factor is dependent on many parameters such as the rate of deceleration, the curvature of the surface (for convex surfaces it can be as low as 1.2 and for concave surfaces as high as 3.0), the transverse pressure gradients as in rotating passages. At off-design conditions when the flow does not enter the blading parallel to the leading edge, the flow tends to break away from the surface, and the subsequent losses are another manifestation of boundary layer separation. To limit these losses, we must have a detailed knowledge of the behaviour of the flow inside the entire field.

2.1.5 Shock Waves

Shock waves give rise to pressure losses across the shocks, while strong shocks also give rise to boundary layer separation. Therefore, high Mach numbers along the surfaces should be avoided.

2.1.6 Summary of Pressure Losses

In summary, if high efficiency is required to reduce excessive power cost, we must have means to calculate the velocity distribution throughout the region of interest. In addition, we must be able to interpret the velocities in terms of the generation of losses within the region.

2.2 Flow Calculation Methods

The most popular methods for analysing flows in turbo-machines have been pseudo three-dimensional methods. Novak[1] provides a critical overview of the methods. In these methods, the flow is first analysed in the hub-to-shroud plane and then in the blade-to-blade direction. In other words, two-dimensional methods are applied twice. Truly three-dimensional methods are becoming available, but they are not in the form that may be used in day-to-day design routines.

2.2. Velocity Distributions

Typical cross sections of a centrifugal compressor and an axial turbine are shown in Fig. 1. The fluid dynamic design programs will generate the velocity and pressure distribution along the blade surfaces (pressure and suction side) from hub-to-shroud. These velocity distributions are found for a number of streamtubes in the hub-shroud plane.

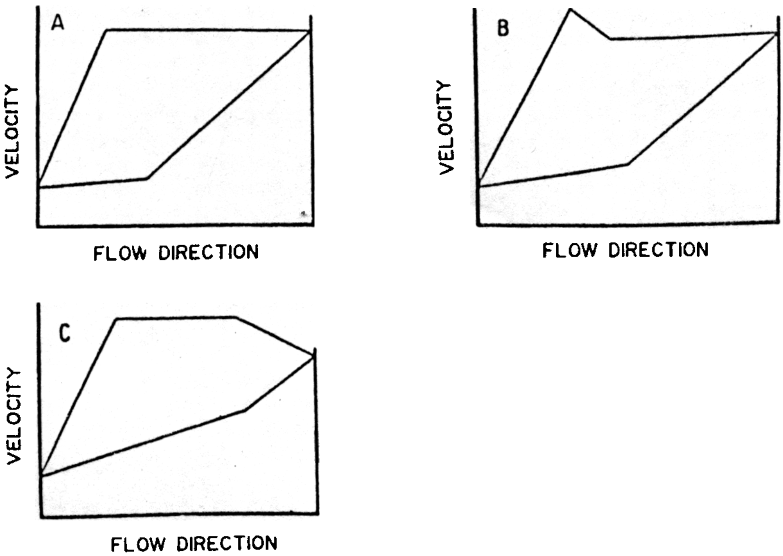

Figure 3. Three types of velocity diagrams in axial turbii

(iii) Specifying velocity distributions : Since the velocity distribution is so critical, it would be more convenient if this distribution could be specified at the outset, and the geometry that would yield such a distribution could be calculated. Stanitz[2] described such a method, and many designers follow this or a variant of it to arrive at suitable geometries. Most gas turbine blades are designed in this manner, and there is a great deal of competitive effort expended currently to design the so-called 'diffusion-controlled' axial compressor blading as discussed by Sanger[3] and Rechter[4] *et al.*

2.3 The Application of Sound Fluid Dynamic Principles

The foregoing sections have shown that a high performance turbomachine can only be obtained through the application of sound aerodynamic methods. These methods lead to certain geometries that are optimal in a performance sense. However, they may not be optimal in other regards, and the following sections identify constraints dictated by reliability and manufacturing considerations.

3. STRESS AND VIBRATION CONSIDERATIONS

3.1 Reliability – Why do Blades Fail?

Failure in turbomachinery blading is almost always caused by a combination of high steady and vibratory stresses. The steady stresses can be caused by centrifugal forces and bending moments. Vibratory stresses are generally low, except when a blade resonance condition is encountered. At resonance, an oscillating force exerts its influence on the blade at a frequency that is the same or is a multiple of the blade's

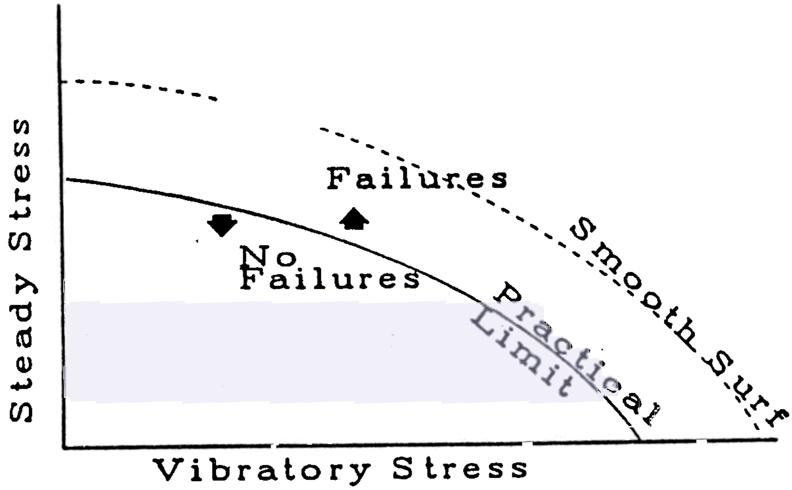

Figure 4. Goodman diagram.

natural frequency. In resonance, the vibratory stress level can be very high and when sustained over a period of time can cause fatigue failures. The period of time need not be great. For example, a 500 Hz vibration accumulates 10 E8 cycles in two days, a sufficient time to cause a fatigue failure. The vibratory stress level to produce a fatigue failure is shown in a so-called Goodman diagram. Each material has its own Goodman diagram. A typical diagram is shown in Fig. 4. It shows that for an existing steady stress only a limited amount of vibratory stress can be allowed. Furthermore, smooth polished surfaces have a higher limit than do eroded and pock-marked surfaces.

In summary, it is important to calculate both the steady and vibratory stresses in turbomachinery blading. If the allowable bounds are exceeded then the design or the material must be changed. However, if the design and geometry is changed, a fluid dynamic analysis of the modified geometry is necessary to ensure that the expected loss level is maintained.

3.2 Steady Stresses

3.2.1 Centrifugal Compressors

Areas of high steady stress (Fig. 5) are usually associated with the blade roots and with the shaft bore. For shrouded impellers high stresses may occur where the blades attach to the shroud near the inlet.

There are two distinct types of methods for finding steady stresses in centrifugal impellers. First, there are pseudo two-dimensional methods described by Loeffler[5] and secondly, there are the Finite Element Methods (FEMs).

The first method is fast but approximate. It does not allow for blade bending (i.e., the blade consists of radially elemented segments), and it assumes that radial cross-sections remain plane (they may tilt). The method gives good results for bore stresses and blade root stresses at the inlet, particularly when the impeller axial length

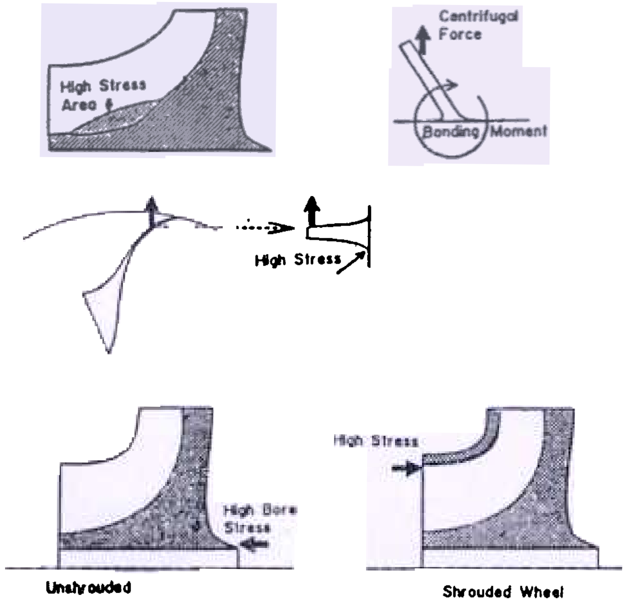

Figure 5. Areas of high stress

is less than the radius of the wheel. The method does not account for the blade root stresses near the discharge due to centrifugal bending of backward-swept blades.

Several FEMs are commercially available (NASTRAN, ANSYS, SUPERSAP, etc.), but they all require the generation of a disk/blade mesh at which the calculations are to be made. Generating a mesh is a very time-consuming exercise, and as a result, not many FEM analyses are conducted for centrifugal impellers. It should be stressed that a general mesh generator when applied to impellers may lead to erroneous results. Only when automated mesh generators for impellers become available will FEMs have an established position in turbomachinery design. Some practical examples of how FEMs are applied to impellers are described by Smith[6] *et al.*

3.2.2 Axial Compressors and Turbines

For this category of turbomachinery, there are also pseudo two-dimensional and FEMs that are the major design and analysis tools. In the pseudo 2-D method, each radial blade section is mass integrated with the total force acting at the centre of gravity. Steady stresses are found from the centrifugal forces acting on the surface areas. Additional steady stresses are found from moments that occur when the blade is not stacked on the axis that connects the centre of gravity.

FEMs are easier to use in axial geometries since the interface with the disk is well defined, but again the generation of the mesh is no easy task. When axial blades are designed with damping devices such as lacing wires, pins, Z-shrouds, and packets, then the FEMs are much more difficult to implement, and the pseudo 2-D methods are preferable, particularly for long narrow blades.

3.3 Vibratory Stresses

Judging from the Goodman diagram, there are two ways to eliminate fatigue failures, i.e., either not allowing any resonances to occur in the operating range, or when resonance cannot be avoided, maintaining low vibratory stresses through damping. It is evident that to decide which action should be taken, the natural frequency of the blades must be found. These frequencies cover the bending, lateral, and torsional motions of the blade, and they occur in basic as well as in higher modes. Thus, it is not uncommon that up to 15 frequencies must be calculated to cover the first five modes of the three basic blade motions. When damping devices connect the blades, these natural frequencies and their mode shapes become rather complicated, and their resolution becomes difficult. Nevertheless, the same methods that were applicable for steady stresses apply here too. Note that the pseudo 2-D method does not yield the plate modes that vibrate in axial direction while, FEMs will identify these. The blade natural frequencies are represented against speed on a Campbell diagram in Fig. 6.

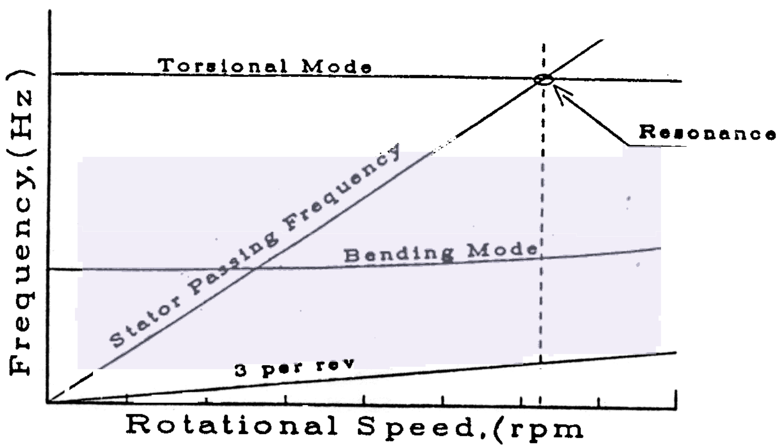

Figure 6. Campbell diagram.

The Campbell diagram is convenient for identifying resonances since frequencies of possible excitation mechanisms are also plotted on the graph. Excitations originate usually from wakes discharging from upstream stationary blades. A rotating blade experiences a fluctuating force field from the wakes of the stationary blades. Stationary blades become excited by wakes from rotating blades.

For example, if a rotor follows a stator with 53 blades, then the rotor will experience 53 intermittent forces for each revolution, thus, there is a 53 'per rev' excitation. If the rotor rotational speed is 3600 rpm, then the excitation frequency is $53 \times 3600/60 = 3180$ Hz (see line on Campbell diagram). Other excitations are the 2, 3 and 4 per rev caused by inlet distortions, shaft movements, and shaft unbalances.

3.4 Summary on Stress and Vibration Considerations

The designer must design the turbomachine within the steady stress limits of the material. Furthermore, operation in resonance at the operating point must be avoided. If resonance cannot be avoided, the vibratory stresses must be calculated and plotted together with the steady stresses on the Goodman diagram. If the combined stresses exceed the limits for the chosen material, then the material must be changed or more likely, the geometry should be changed to bring the combined stresses within the allowable limits.

4. MANUFACTURING CONSIDERATIONS

4.1 Casting and Machining

Rotating turbomachinery parts are either cast or machined. Casting has been preferred for large quantities since the piece price becomes much smaller than for machined pieces. However, casting has some disadvantages which have led many manufacturers to re-examine their preferences. These disadvantages are :

i) Casting material properties are not as uniform and consistent as those for forged materials; undetectable inclusions are sources for failures.

ii) Cast geometries differ from one casting to the next. Since performance is finely tuned to an exact shape, no similar performance between two machines exists. Handwork may be necessary to obtain consistent performance, but this is a very time-consuming and inconsistent practice.

iii) Casting shape deviations may lead to premature surge in compressors, and lower than anticipated power output in turbines.

iv) Casting shape deviations also affect spare parts and their use in overhauls. Inserting a new part of slightly different shape often requires a retuning of the control system.

v) Preparing casting patterns is a time-consuming task that is expensive. Changes cannot readily be made. Casting cost advantages only apply if the same patterns can be continuously used.

vi) Casting is the preferred method for parts that are subjected to high temperatures. The hardness of such materials requires substantial machining time. However, recent work has been directed towards rapid metal removal for such components.

In the light of these considerations, many manufacturers have turned to machining parts for both prototyping, small and even large scale production. Machining is accurate and precise, consistent, fast and allows for easy geometry modifications.

4.2 Five-Axis Milling Machines

The complex shapes of turbomachines require the application of five-axis milling machines to be able to reach all sections of the geometry. In modern design, the geometry to be machined is available from a CAD system, and this is converted into tool movements in a CAM system. This output is then converted via a post-processor to the movements of the five individual axes of the milling machine. For turbomachinery parts, CAM systems currently available on CAD/CAM design computers are too general, and specialised CAM methods must be applied. There are four distinct parts of the operations : (a) Roughing out passages between the blades; (b) Milling the floor; (c) Milling the leading and trailing edges; and (d) Finishing the blade surfaces with slight cuts.

Special care must be taken to reach inaccessible locations on the blade that may cause interference of the cutter tool with adjacent blades.

4.2.1 Point Cutting versus Flank Cutting

The most general method for cutting turbomachinery blading is by the so-called point cutting method (Fig. 7). Here each point on the surface is cut by the end-ball of the cutter as a point. Thus, when the cutter moves along the surface, a groove or cusp is left behind whose depth depends on the fineness of the cut. This method is acceptable, but it creates some inherent difficulties :

i) The blade surface is not smooth, and slight grooves will remain. These grooves should always be parallel to the direction of flow.

ii) For highly curved blades that are close together, interference with adjacent blades is difficult to avoid.

iii) Machining time is long if the effect of the grooves is to be minimised. In other words, many passes around a surface must be made.

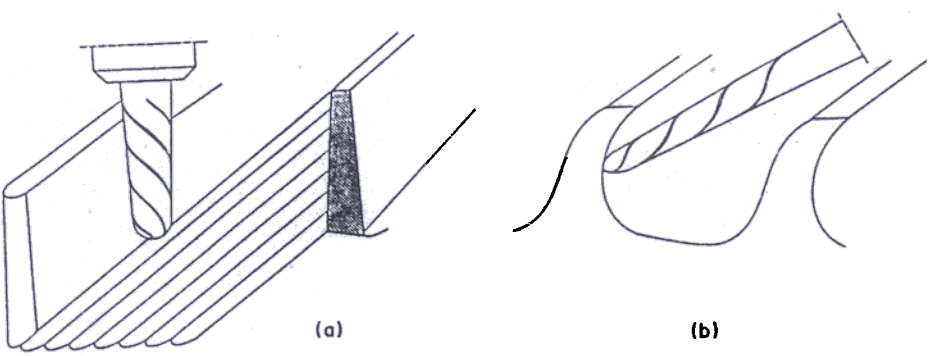

Figure 7. (a) Flank cutting, and (b) point cutting.

4.2.2 Flank Cutting and the Fluid Flow Problem

For these reasons, the machining operation with flank cutting (Fig. 7) is preferred. It eliminates the above stated difficulties and results in smooth blading and fast final machining. With flank cutting, the entire side of the conical tool is in contact with the material, and the implication is that the surface must consist of straight line elements. This restriction is important to the fluid dynamicist who originally generated the blade design from fluid flow principles and had complete freedom in the choice of geometry. In order to obey these restrictions. the blade can only be aerodynamically shaped along three streamtubes. The blade shape is formed by straight line element surfaces that connect the three streamtubes. However, in practice some geometry manipulation is required which further restricts the specification of the blade shape to straight line elemented surfaces that connect only two streamtubes; usually the hub and the tip.

Test experience indicates that specifying the velocities along two streamtubes does not degrade the performance for low to medium specific speed machines, but when large blade heights are used, surfaces consisting of straight line elements are no longer effective.

4.2.3 Flank Cutting and the Stress Problem

The application of straight line elements for ease of manufacturing has an effect on the stress analysis as well. Highly stressed blades are often tapered so that the high root stress is taken up by a larger area. In fact, the best taper is an exponential variation in thickness, but the limitation of a straight line flank only allows a straight taper. Nevertheless, this limitation gains importance only when the blade height becomes large compared to its chord length, such as in the last stages of axial turbines or for high-flow coefficient centrifugal compressors.

5. IMPLEMENTATION OF INTEGRATED DESIGN/MANUFACTURING

The integrated design methods, based on the three aforementioned technological disciplines operate on a database that contains the geometry. Thus, schematically, we can place the database at the centre with the technologies interacting with it. In a practical sense, the designer must satisfy the client who requires a compressor, or turbine with certain specifications.

These specifications for a centrifugal compressor may be a flow of 5 kg per second, a pressure ratio of four-to-one, and a speed of 20,000 rpm. The results would be an impeller wheel milled with a 5-axis machine that satisfies performance, stress and vibration (i.e., reliability) and can be sold in time because of the ease of manufacturing.

5.1 Application to a Centrifugal Compressor

Figure 8* shows the representation of the integrated design method. It represents a system that is commercially available and is used at many major turbomachinery manufacturers. The process starts with the PREDIG program where flow, rpm, and pressure ratio are entered via a pre-processor.

5.1.1 Fluid Flow Considerations

PREDIG generates the main dimensions such as diameters, widths, and blade angles from the given conditions. There are three parts to the PREDIG program; the ANALYZER, the OPTIMIZER, and the CALIBRATOR. The CALIBRATOR is used to enter test data to improve the loss correlations; the OPTIMIZER calculates

* Similar systems may be available at various organisations. However, a specific system developed by the author's organisation and used worldwide, is discussed here for purpose of illustrating this approach.

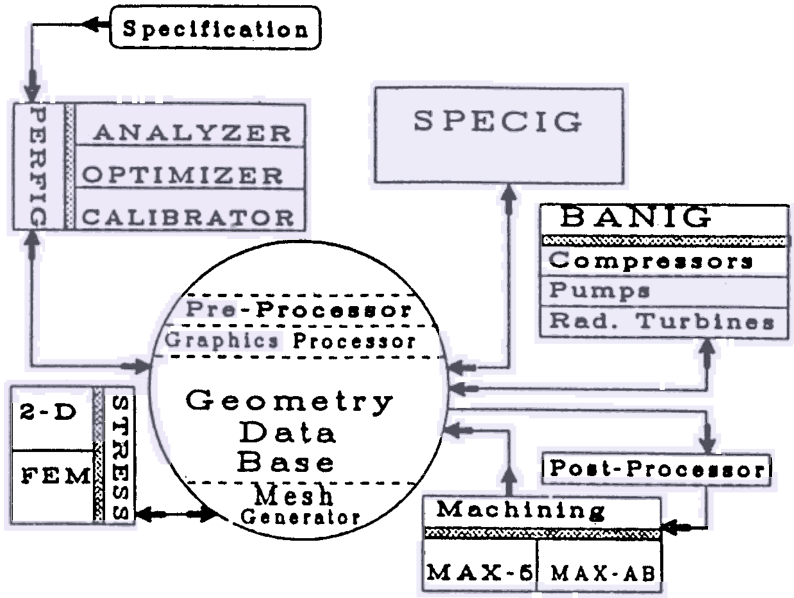

Figure 8. Schematic for integrated design and manufacturing of radial turbomachinery.

the dimensions for best performance taking into account either the tested loss correlations or the default values imbedded in the program. The ANALYZER calculates the performance map to indicate the compressor performance at other speeds and flows with the associated surge and choke flow limitations.

Given the main dimensions, the geometry program SPECIG is entered. The hub and tip blade shapes are found from given velocity distributions and given blade thickness along the hub and tip lines; straight lines are then drawn from hub to tip In the BANIG section, the blading is analysed over its entirety and velocity distributions along all streamlines is found. Thus, here a check is made to see if the velocity distributions along streamtubes other than those at the hub and tip are satisfactory. Often the calculations are repeated in the BANIG section when SPECIG does not provide satisfactory blading from stress or manufacturing considerations and modifications in the blading must be made.

5.2.2 Mechanical Integrity

When a satisfactory blade geometry is found, the database program prepares a file readable by the stress program (either pseudo 2-D or FEM; for the latter an optimal mesh is also transferred). Stresses and vibrational characteristics are found and compared with allowable values. When stresses are above allowable values for the specified material, two approaches are possible. The material may be upgraded by heat treatment or by selecting a different material, for example, titanium over aluminum. Second, the stresses may be improved by decreasing the speed or changing the geometry such as the blade exit angle, the blade taper, or the backface contour to improve the bore stresses, If blade modifications are made to decrease the stresses, another iteration involving BANIG is made to assure that the velocity distributions still satisfy their specifications.

5.2.3 Machining

Following the check on the mechanical integrity of the unit, the database prepares files for the machining program, MAX-5. This specialised program tries, with the available cutters in stock, to generate tool cutting paths. Various problems may arise here. The selected and available cutter may not be able to reach the narrow hub sections and a narrower cutter must be selected. In many cases a new cutter requires a delivery time in the order of weeks, and this may be unacceptable, to maintain the given time schedule for the project. At the same time, a smaller tool may increase cutting time since smaller cuts must be taken to maintain the same tool bending deviations. It is, therefore, often more convenient to redesign the blade passages by decreasing the number of blades, introducing splitter blades, or decreasing the fillet radii at the root of the blades. In any case another iteration involving the BANIG and STRESS programs may be required.

When this cycle of iterations is completed, the design will satisfy the fluid dynamic, the material strength, and the manufacturing criteria of the design, while also satisfying the initial performance specifications.

5.3 Axial Turbine

The implementation of the integrated design and manufacturing method for gas turbines follows a similar path as that for centrifugal compressors. However, in the preliminary design phase, the number of stages must be selected which is the main difference from the previous discussions. This selection involves a number of constraints such as speed (rpm), meanline diameter, Mach number, and axial length. When the number of stages has been found that satisfies the given constraints, then each stage is designed as indicated before.

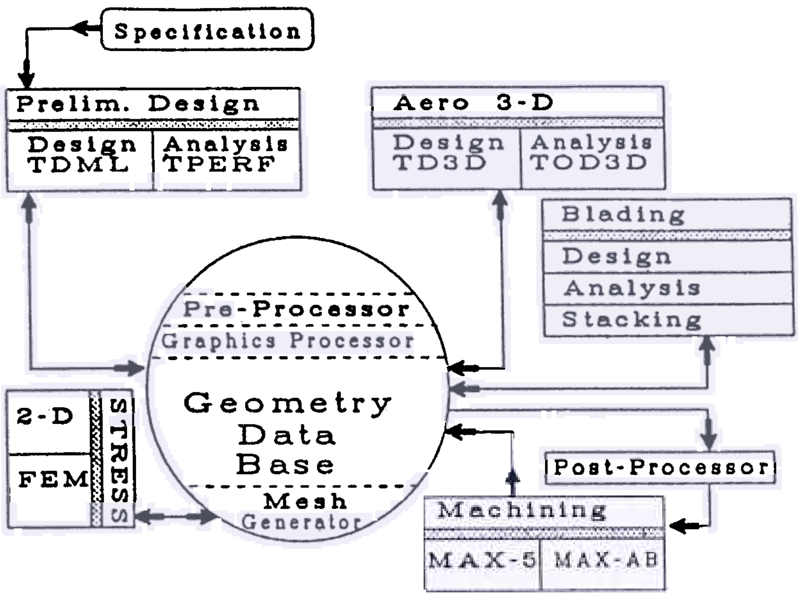

Figure 9. Schematic for integrated design and manufacturing of axial turbines.

Figure 9 shows the procedure for axial turbines. The sequence is as follows :

i) The operational specifications in terms of power, inlet and discharge pressure, rpm, and available flow rates are given.

ii) The preliminary design program TDML is used to find the number of stages, the hub and tip diameters, and the work done in each stage.

iii) The preliminary analysis program TPERF is used to find the off-design performance.

iv) The quasi-3-D design program TD3D is used to find the inlet and exit velocities for each stage at all blade heights.

v) The BLADE program selects the blade shapes at each blade height, stacks the blade, and generates a mesh for use in a FEM or pseudo 2-D stress analysis. Short blades used in the initial stages of a turbine are shaped with straight line element surfaces. Intermediate and last stages usually must be of arbitrary surface geometry.

vi) Stress and vibrational characteristics are found and examined. When modifications are required an iteration between items (iii), (iv) and (v) is necessary until a satisfactory blade is generated. Resonance with upstream blading may require a change in the number of upstream blades. Resonances with low multiples of the rpm requires a re-definition of the blade.

vii) Turbine blading is usually machined as single blades. In these cases there is no iteration necessary with the stress and fluid flow programs since the cutting paths will show no interferences. MAX-5 is used for machining blades that have straight line element blade surfaces; MAX-AB is used when the blades have an arbitrary shape. Recently, turbine blades have been made part of the disks and constructed as a BLaded dISKS (so-called BLISKS). For these units some iterations between MAX-5 (or MAX-AB) and the fluid flow programs may be necessary since interferences may occur here.

6. CONCLUSIONS AND RECOMMENDATIONS

Recent advances in computer hardware and software technologies have made it possible to use an integrated design and manufacturing approach. Constraints imposed by the various design disciplines can be satisfied such that an optimally performing and reliable machine for minimal cost results in the shortest time. Future computer software advances, such as multiple windows, will allow these disciplines to be executed interactively, where a change in one geometry variable shown in one window will yield the resulting change in the fluid dynamic, material strength, and machining aspect of the design in adjacent windows.

On the other hand, it cannot be strongly enough emphasised that there are grave dangers in the continuing computerisation of the design effort. Smith[7] offers warnings with respect to accepting computer results at face value. There must also be continuing efforts in laboratory testing of fluid dynamic effects, material strength aspects in fatigue, and manufacturing methods using better cutter designs and other material removal techniques such as ECM and EDM procedures.

REFERENCES

1. Hawthorne, W.R. & Novak, R.A., The Aerodynamics of Turbomachinery, *In* Annual Reviews, Vol. 1, (Annual Reviews, Inc., Palo Alto, California), 1969, pp. 341–366.
2. Stanitz, John D., Design of Two-Dimensional Channels with Prescribed Velocity Distributions Along the Channel Walls, NACA Report 1115,1953.
3. Sanger, N.L., *Journal of Engineering for Power,* **105** (1983).
4. Rechter, H., Steinert, W. & Lehmann, K., *Journal of Engineering for Gas Turbines and Power,* **107** (1985).
5. Loeffler, K., Die Berechnung von rotierenden Scheiben und Schalen, (Springer-Verlag, Berlin, Germany), 1961.
6. Smith, G.E., Holland, J.E. & Jansen, W., The design of two radial inflow turbines, ASME Paper No. 84-GT-236.
7. Smith, G.E., The dangers of CAD, *In* Mechanical Engineering, (ASME, New York), 1986, pp. 58–64.