THE STATE OF THE ART OF HIGH SPEED OVERHUNG CENTRIFUGAL COMPRESSORS FOR THE PROCESS INDUSTRY

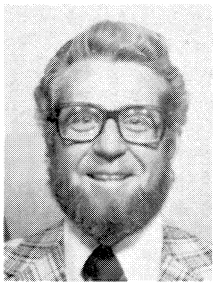
by

Hans Pennink

Chief Engineer

Turbonetics Incorporated

Latham, New York



Hans Pennink is chief engineer at the Turbonetics Division of M.T.I. (Mechanical Technology Inc.) where he is responsible for the development of high speed centrifugal expander/compressors as well as high speed gear driven centrifugal compressors for process applications. Mr. Pennink has some 18 years of experience in high speed turbomachinery, in particular aerodynamics, controls and design. He joined M.T.I. in 1966 and has been involved in turbomachinery projects and was responsible for the development of several product lines. Prior to this, Mr. Pennink has worked in various fields of turbomachinery at the Boeing Co. (Turbine Division), Alco Products and G.E. gas turbines. Mr. Pennink holds a B.S. degree in Mechanical Engineering from the University of Delft, Holland, and M.S. degree in Mechanical Engineering from Rensselaer Polytechnic Institute (R.P.I.) of Troy, New York.

ABSTRACT

Advances in 3-D finite element stress analysis, as well as powerful analytical technique in rotor dynamics and bearing analysis using high speed digital computers, have created the environment to build process centrifugal compressors with steel impellers running at tip speed in excess of 1500 fps. At these tip speeds the centrifugal compressor is capable of producing adiabatic heads of up to 45,000 ft. in a single stage at good efficiency and range.

The intent of this paper is to enumerate the advantages of the high speed gear driven single stage overhung centrifugal compressor as well as to indicate its limitations for application in the process industry.

This paper discusses the various aspects of the aerodynamic components and performance as well as the mechanical features which are required to make this type of machine competitive with the multi-stage centrifugal compressor from the standpoint of reliability with excellent efficiency at lower cost.

INTRODUCTION

The advances of late in 3-D finite element stress analysis as well as powerful analytical techniques in rotor dynamics, gear and bearing design have contributed a great deal to the fact that pinion mounted overhung impeller design compressors are built to run above the first lateral critical speed at tip speeds in excess of 1500 fps, using precision cast steel impellers, with good efficiencies.

Why then the frequent reluctance of the Process Industry to accept these advances in design? There appear to be several reasons.

- Most engineers and maintenance personnel have grown up with reciprocating equipment or multi-stage centrifugals and are comfortable with the modest speeds.
- The multi-stage centrifugal compressor has served the industry well — so why change?
- Many engineers who select equipment are not aware that high speed single stage centrifugal compressors are available for their applications.
- Most manufacturers of traditional multi-stage centrifugal compressors do not offer high speed single stage compressors.
- Last but not least, the Process Industry is a conservative industry and rightly so. Breakdown of equipment could cause losses in production of many times the cost of the equipment so that only equipment is selected which has a known record of reliability.

It is, therefore, up to the high speed equipment manufacturer to show that his equipment is as reliable, or more so, than slow speed equipment, with advantages in cost, efficiency, maintenance, size and weight.

RELIABILITY

Reliability of a typical machine design can only be demonstrated statistically by actual field operation of many typical units for the design life of the machine. Practically this is not always possible.

The following definition is suggested to determine the mechanical reliability on a yearly basis:

$$\text{Reliability} = \left(1 - \frac{\text{Total Number of Failures of a Given Type of Machine}}{\text{Accumulated Years of Operation of the Given Type of Machine}}\right)$$

As an example, one manufacturer of high speed single stage overhung compressors can point to more than 150 units of one type with an average accumulated running time of 3 years. This statistic result would indicate a reliability of:

$$\text{Reliability} = \left(1 - \frac{3}{3 \times 150}\right) = .9933$$

Other indirect criteria for reliability are the correlation of simplicity of design, ease of assembly and ease of maintenance. Figure 1 shows a cross-section of a typical horizontal split gearbox of a single stage overhung centrifugal compressor for high speed application. The advantage of this design is that the machine can be inspected for clearances, tooth contact, runout, end play and slow roll with all the rotating components in operating position. There is no substitute for this kind of final inspection before the machine is buttoned up, and it virtually eliminates problems at production testing.

Figure 2 is a side view of a high speed single stage compressor, which shows that the machine can be completely self-

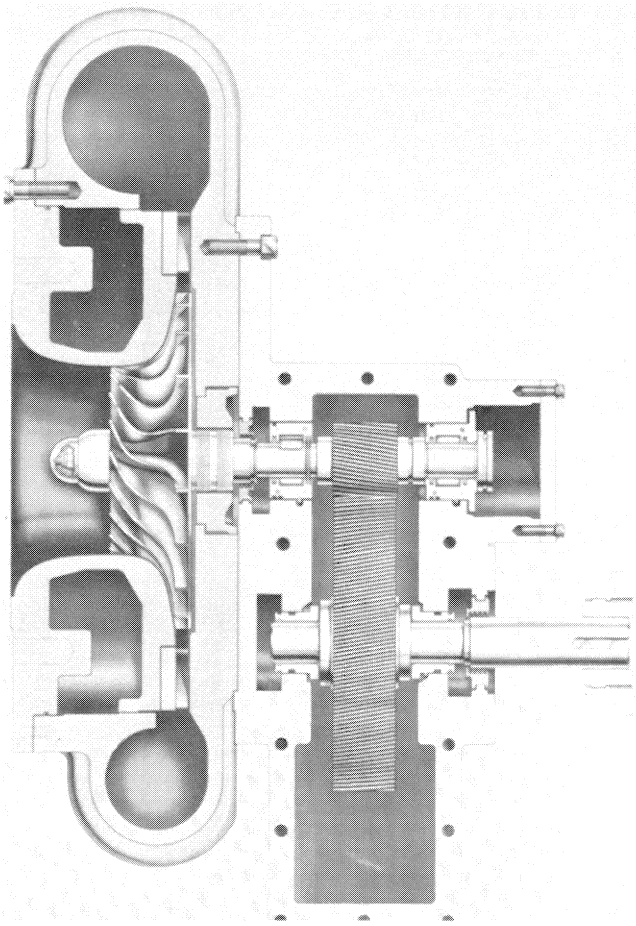


Figure 1. Compressor-Gearbox Assembly.

contained with the auxiliary equipment and controls integrally mounted. Because of the integral design, the machine requires only light hold-down bolting, flange connections and electrical power and control connections. All components operate exactly as on the test stand and a minimum of field installation effort is required. The indirect criteria for reliability have been found to contribute a great deal to overall machine reliability.

PERFORMANCE

The performance characteristic is the first item to be examined in order to determine if a compressor can be applied in a process. This should be preferrably a characteristic based on actual rather than predicted performance.

Figure 3 shows the test performance of a typical high speed single-stage overhung centrifugal compressor, operating on air at a pressure ratio of 3.3:1.

The design conditions required to obtain a performance as shown may be summarized as follows:

Tip Speed	1450. fps
Speed	29156 rpm
Impeller Dia.	11.4 Inches
Discharge Blade Angle	20°
Adiabatic Efficiency	$\eta_{ad} = .79$
Polytropic Efficiency	$\eta_{pol} = .823$

Where:

$$\eta_{ad} = \frac{T_1}{(T_2 - T_1)} \left\{ (P_2/P_1)^{\frac{k-1}{k}} - 1 \right\}$$

$$\eta_{pol} = \frac{\left(\frac{k-1}{k}\right) \ell n\,(P_2/P_1)}{\ell n\ \ T_2/T_1}$$

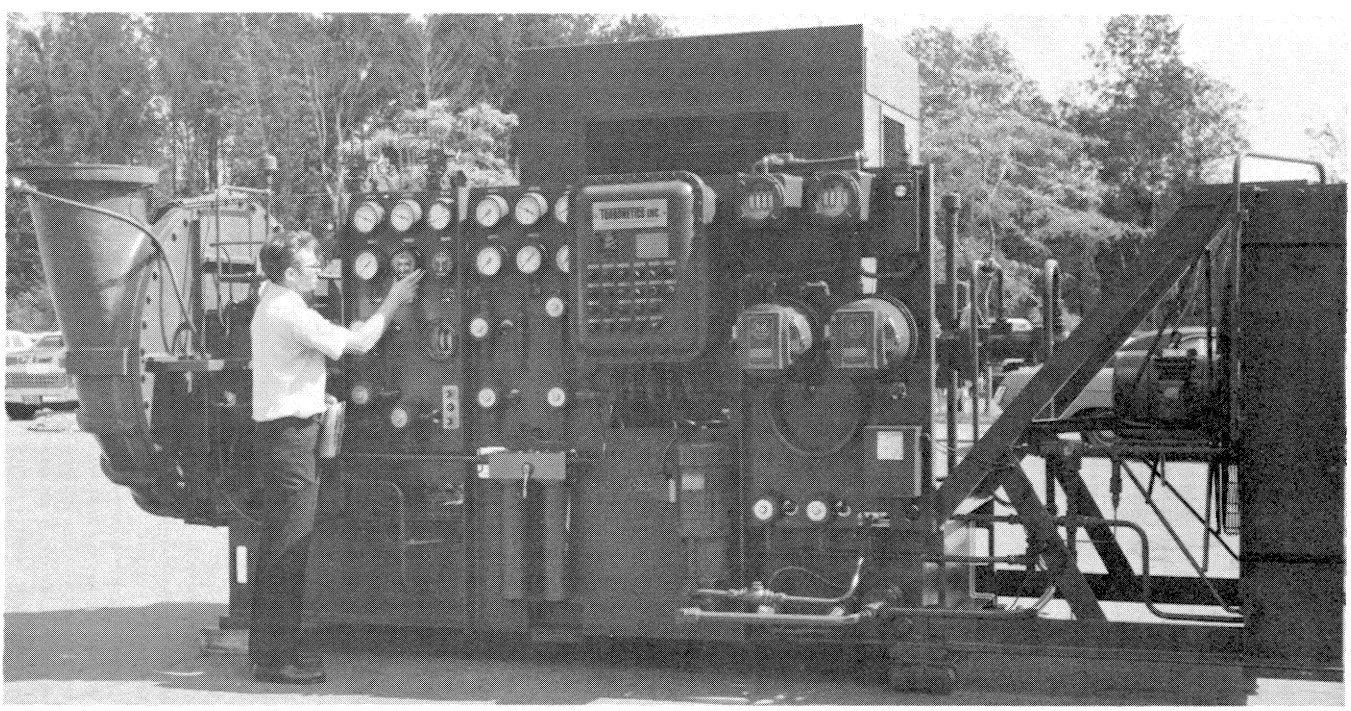


Figure 2. Compressor Package.

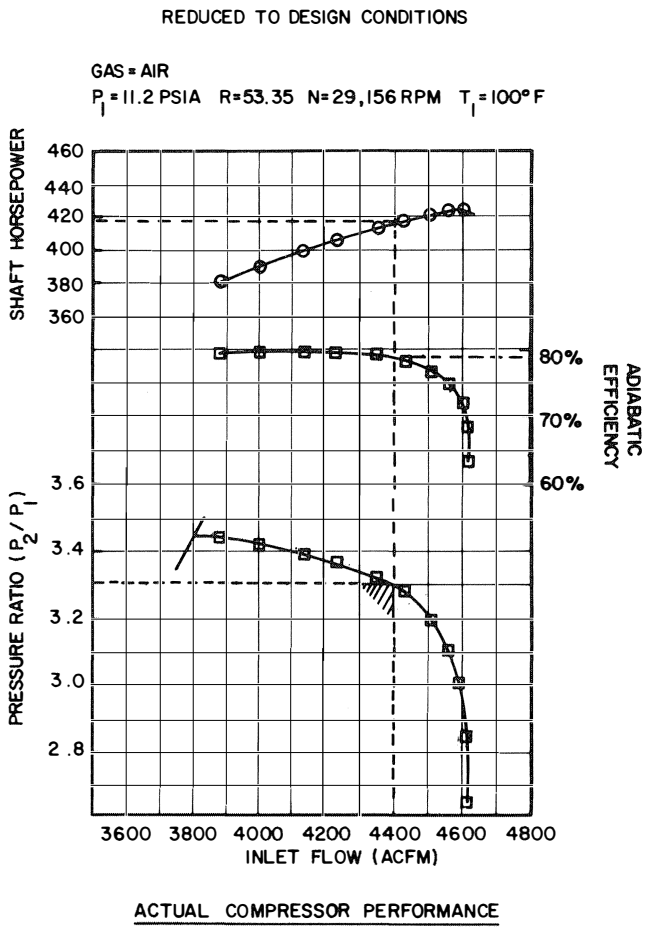


Figure 3. Actual Compressor Performance.

k = specific heat ratio = 1.395
P_1 = Inlet Total Pressure
T_1 = Inlet Total Temperature
P_2 = Discharge Total Pressure
T_2 = Discharge Total Temperature (Actual)

Since the user is generally more interested in power consumption, Figure 3 shows the input HP to the gearbox as measured with a torque meter at operating speed in addition to adiabatic efficiency and pressure ratio. The gearbox power loss can be determined by the difference of input HP and aerodynamic HP.

Another parameter of interest is the head rise to surge value, which is defined as the ratio of the increase in head from design point to the surge point along a speed line, to the head at the design point. For the characteristic shown on Figure 3, the head rise to surge value is 5 percent.

Surge margin is another characteristic of considerable importance and may be defined as follows:

$$\text{Surge Margin} = \frac{Q_{\text{Design}} - Q_{\text{Surge}}}{Q_{\text{Design}}} \times 100\%$$

Where: Q = Inlet Actual CFM

The surge margin for the characteristic of Figure 3 is 16 percent.

Efficiency and surge margin are a function of the pressure ratio (Mach number) and decrease as the pressure ratio increases. This is shown very clearly in Figure 4, which is based on actual test data. A curve is drawn through the highest points of surge margin and efficiency since these points represent readily attainable values of efficiency and surge margin with least compromised aerodynamic design. The band reflects the effects of specific speed and impeller casting size which are generally not optimum because of the common use of standardized gearsets and impeller castings.

Better efficiencies and surge margins are possible using machined impellers (thinner inducer blading) at optimum speed, with careful matching of impeller, diffuser and volute.

The operating point on the compressor characteristic is determined by the intersection of the curve describing system resistance after the compressor and the head-flow curve of the compressor. As long as the intersection of the two is clearly defined by two lines of sufficiently different slope (generally compressor curve negative and system resistance positive), the operation of the system is stable regardless of surge margin or head rise to surge. Most system characteristics consist of parallel and series restrictions which tend to give a steep parabolic curve and therefore a very well defined intersection point. The exception is a liquid level resistance which tend to give a very flat characteristic.

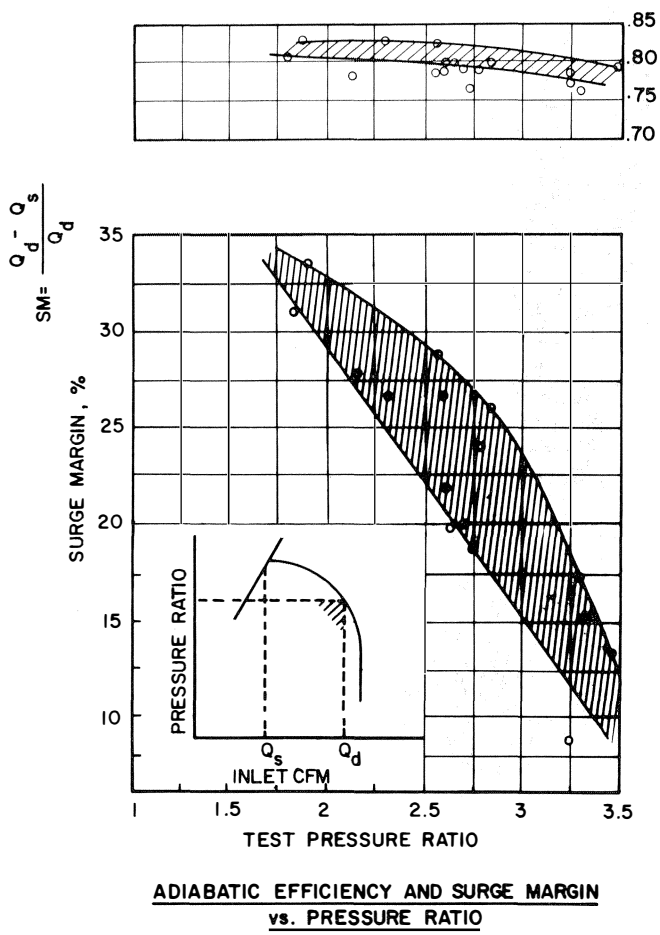


Figure 4. Adiabatic Efficiency and Surge Margin vs. Pressure Ratio.

AERODYNAMIC COMPONENTS

General

Figure 1 shows the aerodynamic components, which consist of a conventional bell mouth inlet followed by a three dimensional inducer type unshrouded impeller with backward leaning vanes, which in turn is followed by a cascade diffuser and a volute. The cascade diffuser is preceeded by a vaneless space.

Impeller

Impellers are typically unshrouded precision investment castings of three dimensional design, having backward leaning vanes and a full inducer. Although 17-4 PH stainless steel is a common standard, alternate materials such as Aluminum (C-355) or Titanium (Ti 6 Al – 4V) are also used. For tip speeds much in excess of 1500 FPS, only Titanium alloys are used.

Figure 5 shows a typical impeller in finished machined form, mounted on a high speed pinion. The blading is designed by "radial ruling," so that at every point in the vane the centrifugal load vector is in the metal. In this manner the bending moments in the vane roots are minimized. Sufficient taper is provided to obtain uniform loading. Generous radii are required to avoid stress concentrations.

After completion of a new aerodynamic design, which includes an analysis of the velocity distribution on the pressure and suction surface along the hub, mean, and tip stream line, the design is subjected to a 3-D finite element stress analysis of the blade-hub elements, including the bore geometry. Corrections are made to reduce stresses if necessary, usually to the blade thickness and taper as dictated by stress or blade natural frequencies. Only then is the impeller released for manufacturing to the casting vendor or the machine shop. Casting vendors are required to provide material certifications, oven charts, X-rays, dye check result and dimensional tolerance reports. If the casting is found to be acceptable, it is released for machining.

After completion of final machining, the blades are rung individually to verify calculated frequencies. Vibration testing is done to determine the mode shapes and frequencies of the disc. Quality and critical dimensions are carefully checked.

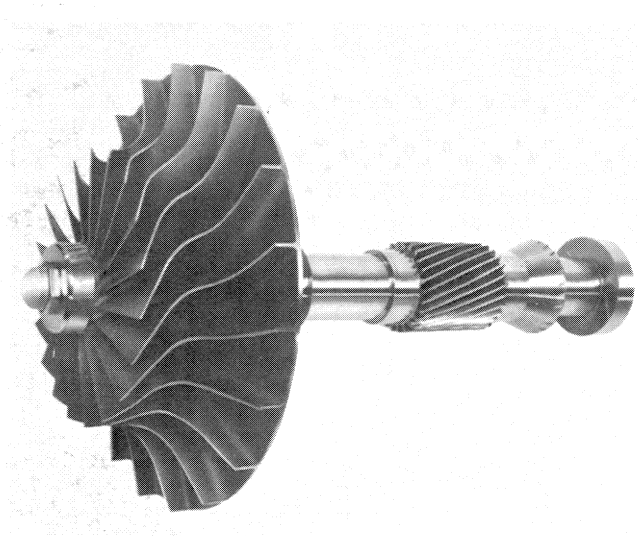


Figure 5. Impeller Mounted on Pinion Shaft.

Large impellers receive a shot peening treatment along the blade root.

Finally the impellers are balanced to maximum balance machine accuracy which exceeds Navy Specifications (Unbalance $< \frac{4 \times \text{Weight}}{\text{RPM}}$ Oz. Inch) by a factor of about 2, and the impeller is spin tested to 15 percent overspeed (25 percent for turbine drive applications). It should be noted here that the API specification 617.2.18.3 for residual unbalance is not quite applicable.

For most applications standardized impeller castings can be used to fit the design requirements. A profile trim of the impeller vanes and/or impeller outside diameter will produce the desired characteristic. This is normally done using a computer program which calculates the new profile based on a specified area schedule to assure acceptable decelerations.

If the required trim is too severe so that efficiency or characteristic is impaired to an unacceptable degree or if higher efficiencies are required, it will be necessary to make a new scaled impeller. The reason for scaling rather than an all new design is the assurance that the performance will be predictable, based on test data of similar units which have proven performance.

Diffuser

The diffuser of a high speed centrifugal compressor can be one of three different types:

a. Vaneless

b. Cascade (single or two stage)

c. Channel (vane island)

For large flow range requirements the vaneless diffuser is normally selected. However, above a pressure ratio of 1.8 it does not produce the best efficiency. Below a pressure ratio of 1.8 on air the vaneless diffuser is preferred and has good efficiency. Stiefel [1] shows that a short vaneless diffuser followed by a matched diffusing volute can also be efficient at pressure ratios over 3.0:1 on air.

The cascade and channel diffusers are very close in terms of surge margin and performance. The cascade diffuser, however, can be designed within a smaller diameter and therefore is preferred, particularly in high pressure applications. Figure 6 shows a typical cascade diffuser.

The vaneless space preceeding the vaned diffuser is normally designed with axial contraction to account for the blade blockage reduction. The diameter ratio is required to assure smooth entrance of the fluid into the vaned diffuser, as well as to allow for dissipation of the impeller blade wakes, particularly for the high mach number, high pressure ratio configuration. The vaned diffuser is designed to criteria for optimum efficiency as well as surge margin. The various parameters affecting performance are discussed in detail in such works as Rundstadler [2] et al.

The diffuser is designed to match the impeller at the design point to produce the desired compressor characteristic. The key parameters are throat area, boundary layer blockage, Mach number, inlet angle, inlet radius and number of vanes. A computer program is used to scale basic vane geometry from stored data to the desired parameters. A computer plot of two vanes is produced in the "spiral (radial) flow" field as well as a conformal transformation of the vanes into the "straight flow" field as suggested by Smith [3]. If the diffuser is found to be acceptable, a tape is produced by the computer program describing the vanes in Cartesian coordinates. This tape can be

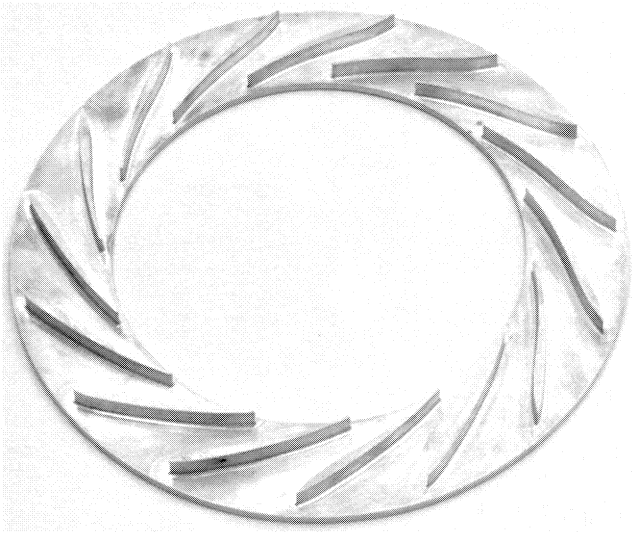


Figure 6. Cascade Diffuser.

used directly on the N.C. milling machine to machine the diffuser.

Volute

Although velocities in the volute are generally low, the volute is designed carefully in order to recover as much as possible of the entrance velocity head.

The volute and exit diffuser operate at peak efficiency at only one value of flow. Because of the unacceptable cost of providing a new volute housing with each application, volutes are therefore generally mismatched except for design flow conditions of the base line compressor. The throat of the volute is sized to match the diffuser exit velocity at 100 percent of the maximum flow of the impeller at full operating speed. As a result, at reduced speed and reduced diameter, the volute is undersized. At reduced impeller profiles, the volute is oversized.

MECHANICAL DESIGN

General

Figure 1 shows the integral design of a compressor-gearbox assembly. The impeller is overhung mounted on the pinion shaft and the pinion gear is straddled by tilting pad journal bearings. The bearing farthest from the impeller has tapered land thrust bearing surfaces on both sides of the bearing cage matching the thrust collars on the shaft. The impeller is mounted either by a tapered fit and tiebolt for small units or by polygon and tiebolt for larger units. Sufficient length is provided between the outboard bearing and the impeller to provide adequate room for various seal designs.

The low speed bullgear is of conventional design and shrunk fit onto a shaft. Combination sleeve-thrust bearings support this shaft.

A strong gearbox encloses the gears and is mounted on a common combination base and oil reservoir, which also support the driver, instrument panel, and all accessories.

The most critical part of the machine is the high speed rotor-bearing system. A detailed and complete rotor dynamics analysis must be made of the high speed rotor prior to final design or difficulties are certain to occur which can be very costly to correct.

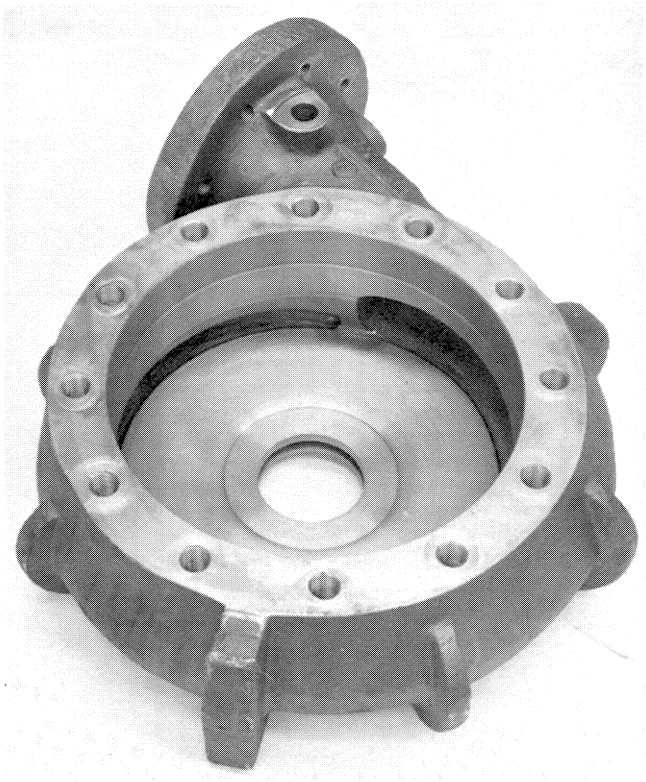


Figure 7. Volute.

Journal Bearings — High Speed

Before a rotor dynamics analysis can be made the journal bearings for the high speed shaft must be defined and completely analyzed. In order to avoid potential instability problems, it is advisable to use tilting pad journal bearings which are inherently stable. Experience has shown that five pad centrally pivoted tilting pad bearings with a geometric pre-load factor of 0.2 to 0.3 have served very well in the past. By designing the bearing pads to have axial pivoting in addition to tangential pivoting, the possibility of edge loading of the pads is eliminated. Generally a diametral clearance of .0015 inch/inch results in quite satisfactory bearing stiffness and power loss. Although thick babbitt may sound old fashioned to some engineers, it does pay off in terms of dirt tolerances which cannot be ignored even with the most careful cleaning procedures. The thick babbitt is also the reason for limiting the bearing loads to 350 psi. This value insures that with the preload and the residual dynamic unbalance the babbitt fatigue life is respected. Loading on the pad is the preferred orientation. Figure 8 shows a typical 5 pad tilting shoe journal bearing. The bearings are fed by holes drilled through the pad stops, so that fresh oil is provided at the leading edge of each pad. The holes serve also as the restrictors. The bearings are completely flooded and the outflow is restricted by either garter spring-loaded bushings or tight fitting two step seals on either side.

Thrust Bearings — High Speed

The high speed thrust bearing can be either a self-equalizing tilting pad bearing or a tapered land bearing. For most applications, the tapered land thrust bearing is more than adequate and has the advantage of lower power losses. A third alternative is the rolling cone thrust bearing, with even a further

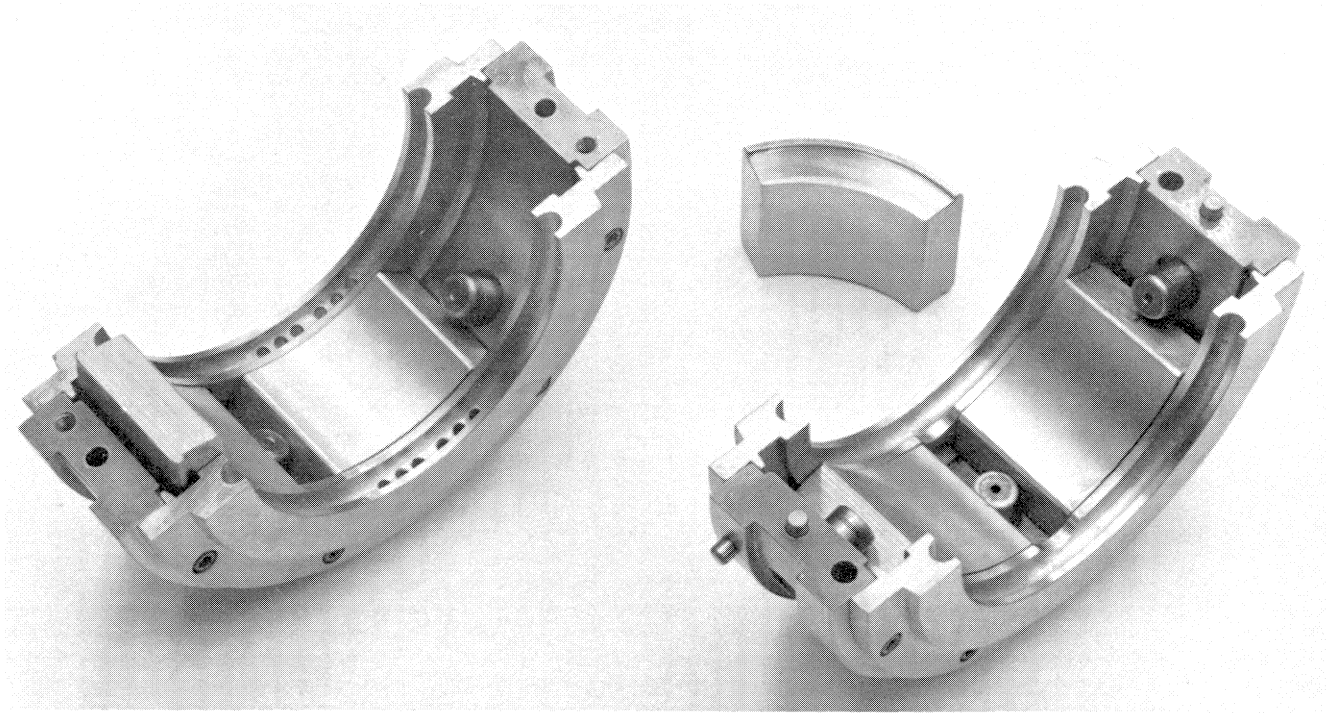


Figure 8. High-Speed Journal Bearing.

reduction in power losses. The final selection of thrust bearings is more a question of preference and power loss, and does not enter into the dynamics of the system.

The tapered land thrust bearing is fairly foolproof and can be combined with the journal bearing into a compact bearing by providing tapered land thrust surfaces on either side of the journal bearing cage which match the thrust collars on the pinion shaft as shown in Figure 9.

In case of a combination thrust bearing/journal bearing the thrust faces are fed by a ring channel supplied by fresh oil from between the pads as well as side leakage oil from the journal pads. For very high thrust loads, the thrust feed grooves can be fed directly.

Thermocouples in the loaded pads and thrust faces to monitor babbitt temperature particularly for large units is very inexpensive insurance since it tends to give advance warning of potential trouble.

Low Speed — Journal/Thrust Bearing

The low speed journal bearing requirements can be easily handled by conventional two-groove sleeve (split cylindrical) bearings. The bearing should be so oriented that during startup as well as running the load vector does not end up near the feed groove. Normally the feed grooves are in horizontal position and it only takes about 10 to 15 degrees indexing to assure that the load vector stays well between the feed grooves.

The low speed thrust bearing is normally combined with the journal bearing by providing slider thrust faces with multiple feed grooves on the flanges of the journal bearings. The multiple feed grooves receive oil directly from the two journal feed grooves as well as from side leakage of the journal.

Gears

In order to operate helical gears with high reliability at pitchline velocities in the order of 30,000 FPM, very high quality tooth accuracy is required. AGMA Q13 is typical of the accuracy specified. There are several gear manufacturers who can supply this quality on a consistent basis even for fairly large sizes. Excellent performance and reliability have been obtained with case carburized (931OH) or nitrided (4140) ground gears with the added advantage of minimum size.

Figure 10 shows a summary of an optimized computer based gear analysis using all the latest criteria and calculation procedures of AGMA (211., 215., 221., 225.). Note that the design is controlled by the maximum surface endurance fatigue stress rather than the tooth bending fatigue stress, for a given life (10 years normally). This is done so that a gear set will give indication of distress through increased vibration rather than outright loss of teeth. With the proper (involute) gear tooth design, conservative loading limits, and proper tolerances on lead and profile errors as well as adequate cooling and lubrication, there is no difficulty in attaining a gear life of 10 years or better at these pitch line velocities. Careful attention must be paid to crowning and profile relief so that with a tooth overlap of three smooth engagement of the teeth is insured.

It is extremely important that a gear tooth contact check is made at final assembly on at least three locations on the bull gear. The contact strip should show at least 85 percent contact.

The hand of the helix angle is normally selected such that the axial thrust of the helical pinion is directed opposite to the impeller thrust, so that the thrust bearing load is decreased to a minimum.

Preferred design is where the pinion is pushed down into the bearings by the bullgear. It must be verified, however, that the bullgear does not lift itself out of the bearings, otherwise the reverse arrangement is required.

From the helix angle, pressure angle, and torque loads it is now possible to determine the journal bearing loads and load vector orientations.

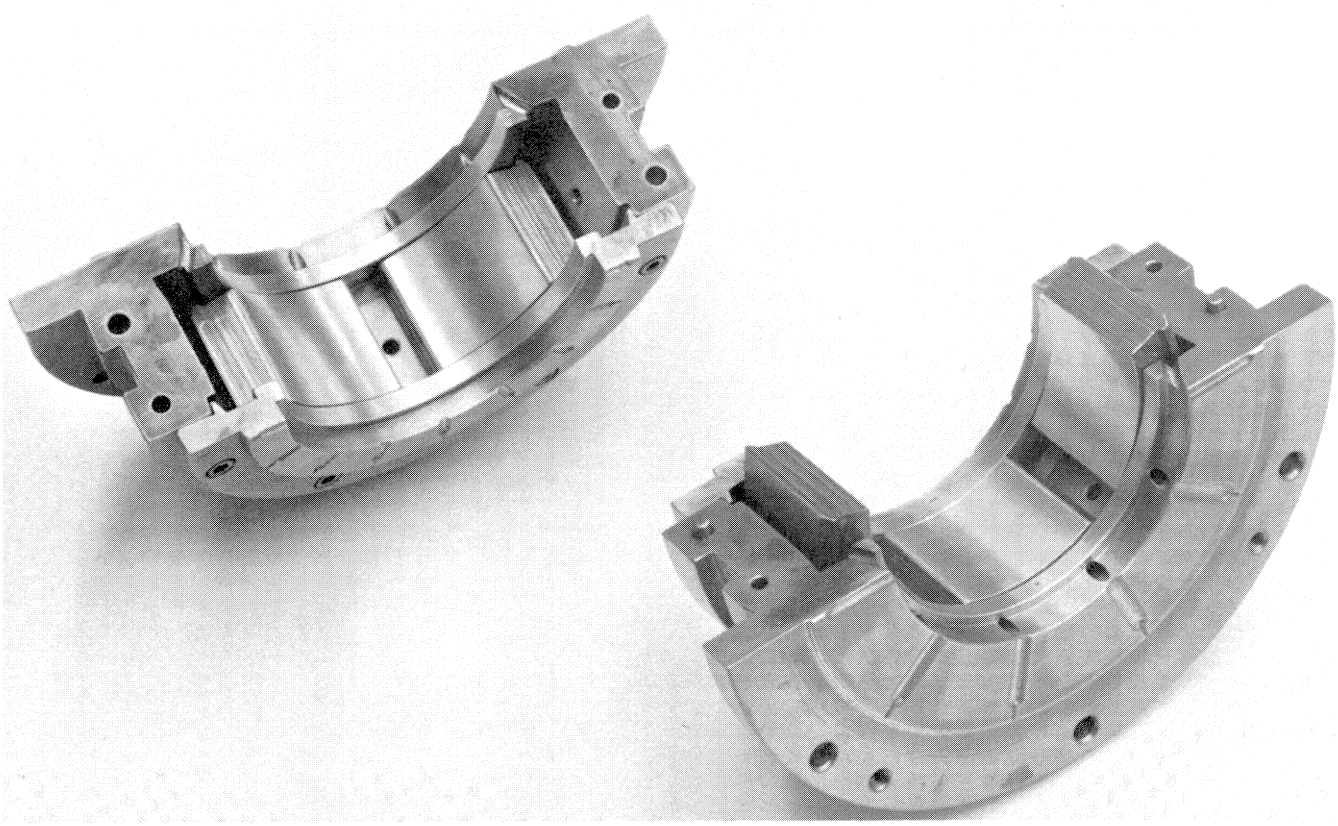


Figure 9. High-Speed Journal/Thrust Bearing.

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(AGMA STANDARDS 211.02.  211.02A.  221.02.  221.02A.  &  421.06)
(CUSTOMER)  ******   TURBONETICS INC.   ******
(JOB IDENTIFICATION)   ***   SCIENTIFIC DESIGN SC-6   ***

    ITEM                              PINION              GEAR
TOOTH FORM                                SINGLE HELICAL
NUMBEROF TEETH                            32                441
PITCH DIAMETER (OPERATING)            2.341210            32.264804
               (STANDARD)             2.341210            32.264804
TOOTH THICKNESS (TRANSVERSE)          0.1119              0.0934
OUTSIDE DIAMETER, EFFECTIVE           2.4840              32.4100
FACE WIDTH, TOTAL                     4.3750              4.3750
BASE CIRCLE DIAMETER                  2.11260501          29.11433744
NUMBER OF MESHING MEMBERS                1                  1

     GEAR MESH RATIO            13.7812500    CIRCULAR PITCH, BASE  0.20740
    HELIX ANGLE                 12.500000     HELIX OVERLAP          4.220
     CONTACT RATIO              1.5216        FACE WIDTH TO DIAMETER  1.869
                                              RATIO
     AGMA QUALITY NO.           014           LENGTH OF LINE OF ACTION  0.3156
     CENTER DISTANCE: (STANDARD).   17.30301  ; (OPERATING).  17.30301]
                                   NORMAL                TRANSVERSE
     PRESSURE ANGLE (CUTTING)      25.000000             25.530503
                    (OPERATING)    24.999996             25.530503
     DIAMETRAL PITCH               14.000000             13.668144
HOB DTA (PIN=1.): TIP RADIUS..2800: ADDENDUM.L1.3000  PRVTUBERANCE..
   L                               PINION                GEAR
T  O   D  REVOLUTIONS PER MINUTE    49268.00             3575.00
O  A   A  CYCLES IN  87600.0 HOURS  2.590E+11            1.879E+10
O  D   T  TORQUE (POUND INCHES)     1599.                22037.
T  I   A  HORSEPOWER                          1259.0
H  N      PITCH LINE VELOCITY (FT/MIN)        30198.
   G      TOOTH LOAD )LBS): TANG. = 1366.0   AXIAL = 302.8  SEP. = 652.

          SURFACE ENDURANCE AND BENDING STRENGTH
                                          CALCULATED   ALLOWABLE   BASIC
  CONTACT STRESS NUMBER (SAC )  PINION    98536.       100614.    180000.
                                GEAR      98536.       116916.     180000.
BENDING STRESS NUMBER (SAT)     PINION    22470.        38885.     55000.
                                GEAR      20755.        42526.     55000.
MATERIAL:  PINION, "CASE CARBURIZED STEEL"   58 ROCKWELL C (MIN. )
           GEAR.  "CASE CARBURIZED STEEL"   58 ROCKWELL C (MIN)

HOTSEPOWER: SURFACE DURABILITY.  1303.8     BENDING FATIGURE 2164.0
AGMA SCORING INDEX  8574.  C1 = 1.1929   C2 = 3.205   C3= 1637.084
K1 = 0.5095   PAC2 = 6259.04    PATF2 = 4221.31   PATG2 = 4570.13
"K " FACTOR                 143.        UNIT LOAD                 4371.
GEOMETRY FACTOR  (J) PINION  0.642      GEOMETRY FACTOR(I)    0.267
GEOMETRY FACTOR (J) GEAR     0.695      HARDNESS RATIO FACTOR  1.000
CURVATURE FACTOR             0.181      SIZER FACTOR          1.000
SURFACE FACTOR               1.000      TEMPERATURE FACTOR     1.000
LOAD SHARING RATIO           0.678      OVERLOAD (SERVICE@)   1.5000
                                           FACTOR
C SUB M = 1.365  K SUB M = 1.255         LIFE FACTOR, DURAB.   0.559
                                           PINION
DYNAMIC FACTOR: C = 0.557  B = 0.557                GEAR    0.650
SAFETY FACTOR                1.000               BEND. PINION 0.707
CSUBU CURVE NUMBER           2                      GEAR    0.773
ELASTIC COEFFICIENT          2300.        RATING ADJ. FACT.: C=3.679 B= 3.382
MEMBERS GIVEN HEAT TREATMENT FOLLOWING FINISH MACHINING: NEITHER
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Figure 10. Gear Data Sheet.

Rotor Dynamics

One of the key requirements for a successful high speed rotor system design is an in-depth rotor dynamics analysis. With the preliminary information on the bearings and bearing loads as well as the design of the rotor, it is possible to mathematically model the rotor, calculate the bearing coefficients and subject the design to a rigorous computer based rotor dynamics analysis as described by Malanoski [4]. Both a lateral dynamic analysis of the high speed pinion and a torsional analysis of the entire train are carried out.

Lateral Analysis

The results of the lateral analysis can be represented by three basic diagrams which predict the rotor behavior within an accuracy of 5 percent. These are:

a. The undamped lateral critical speed map

b. The unbalance response

c. The damped first natural frequency mode diagram with associated log decrement.

Figure 11 shows a lateral critical speed map of frequency vs. bearing stiffness of the high speed pinion. Tilting pad journal bearing stiffnesses are normally in the order of 0.5 to 2 × 10^6 lb/in. at operating speed and loads as shown. The map shows the various operating modes of the rotor as a function of frequency and bearing stiffness. The operating point is normally selected between the 1st and 2nd critical speed. It is recommended that the operating point be at least 15 to 20 percent away from a critical speed.

Figure 12 shows a diagram of an Unbalance Response characteristic. By assuming discreet unbalances in particular at

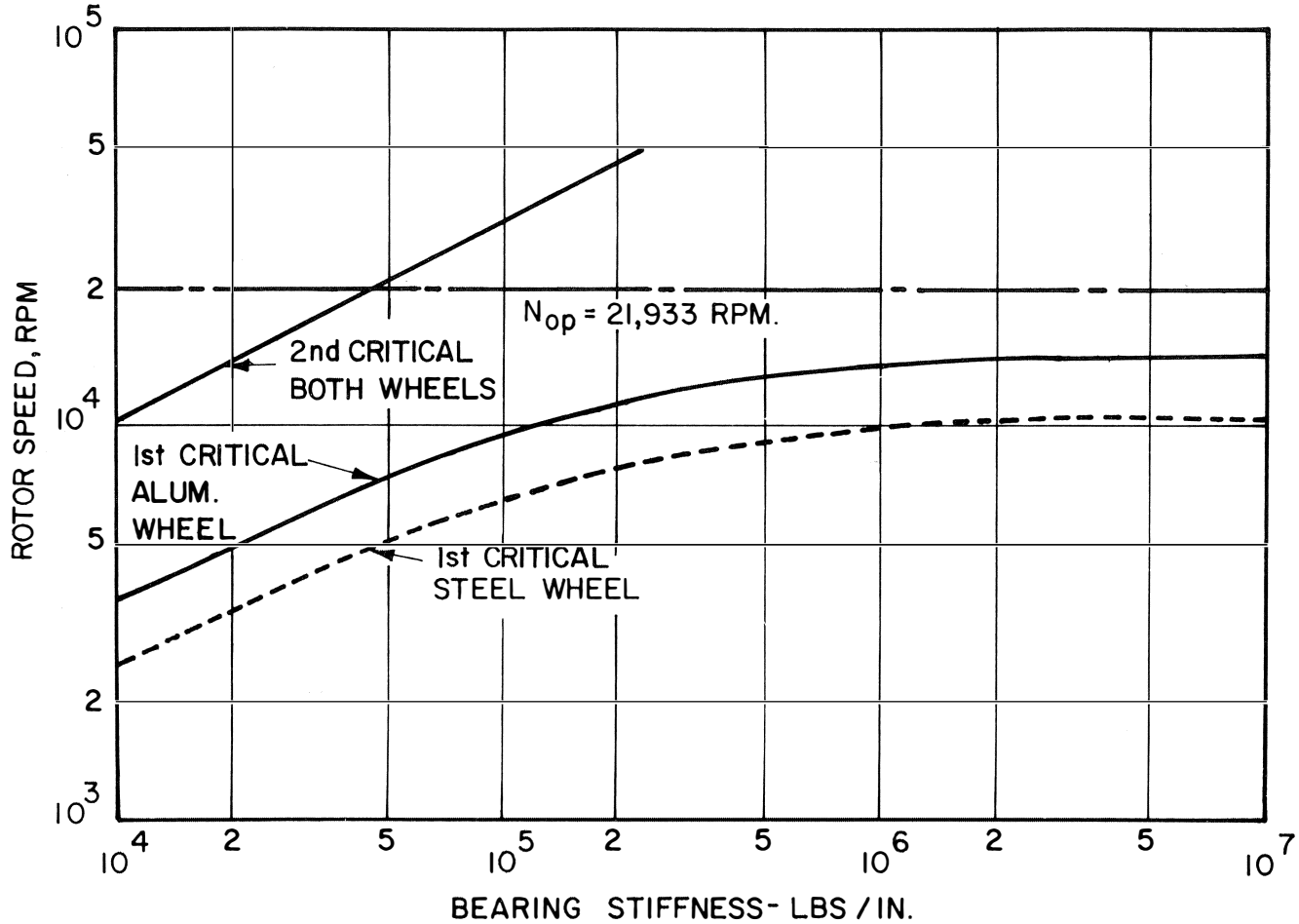


Figure 11. Undamped Lateral Critical Speed Map.

the overhung impeller at specific locations, the behavior of the rotor can be predicted. The amplitudes of synchronous vibrations are predicted. As long as the amplitude ratios are mild and the rotor is well balanced, there should be no problems.

Figure 13 shows the mode shape of the rotor in a subsynchronous mode while operating at the design speed. The log decrement ($\ell n \frac{X_1}{X_2}$) defined as the natural logarithm of the ratio of two consecutive amplitudes of subsynchronous vibration of the shaft is an important criteria of the stability of the rotor system. Experience has shown that the following holds true:

$\ell n \frac{X_1}{X_2}$	=	.50 ~ 1.0	Very Stable
	=	.3 ~ .50	Stable
	=	.2 ~ .3	Marginally Stable
	=	0 ~ .2	Unstable

Torsional Analysis

Finally a torsional critical speed calculation is made. Naturally this cannot be done until the driver inertia is known, however, Figure 14 shows a typical Campbell diagram of the torsional natural frequencies. If a torsional critical excitation is indicated it can generally be corrected by modifying the coupling stiffness.

It is particularly important for turbine drive applications, where it is possible to operate on a critical at reduced speeds, or with synchronous motor driven units where the motor slip frequency is a form of torsional excitation. Machines designed along these principles of rotor dynamics will in general run at vibration levels of 0.3 to 0.5 mils or less at the bearings, where approximately 0.2 mils can be contributed to electrical and mechanical run out.

Seals

From an operational standpoint, the high-speed shaft seal is one of the most critical items of a machine. The most common and most reliable is the buffered labyrinth seal, as shown in Figure 15, which has been used very successfully in the process industry. However, buffer gas consumption can be excessive if working pressures are much above atmospheric.

Some other seals that have been successfully applied on this type of machine at speeds up to 52,000 RPM are the:

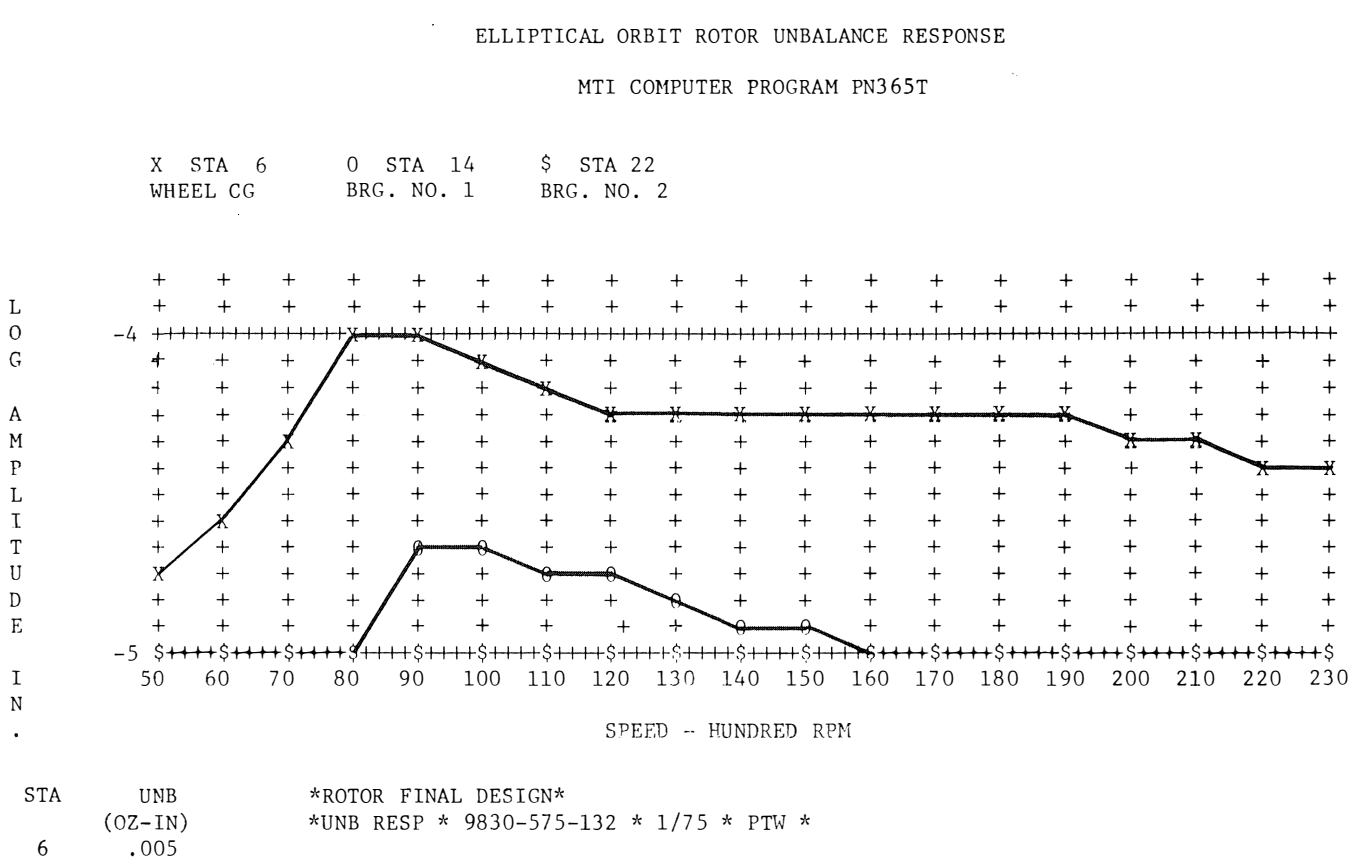


Figure 12. Unbalance Response.

Oil Bushing Seals (Figure 16)
Dry Kinetic Wedge Seal (Figure 17)
(Spiral Groove Face Seal)
Combination Seals

Since the shaft is running at very high speed, seal cooling is very critical. Bushing and face seals require detailed analysis in heat transfer, temperature and pressure distortion as well as seal dynamics. If this is done properly, the seal will operate successfully.

The dry kinetic wedge seal has been tested at temperatures of up to 450°F and pressures to 500 psi at 52,000 RPM with excellent results. Several are operating successfully in the field, and appear to have great promise.

Gearbox

The gearbox forms the basic structural component of the machine and must be designed solidly. It carries the compressor housing and must also be able to take the flange forces.

As pointed out earlier, the horizontal split of the gearbox allows assembly procedures which are straightforward and simple, and contribute towards the reliability.

The larger gearboxes of over 800 HP are normally fabricated. Design is such that different gear ratios and center distances can be accommodated without major redesign.

ACCESSORY SYSTEMS

Lube System

The lubrication system of a high speed centrifugal compressor should in essence comply to API -614. Some exceptions are taken. Whether or not API is followed, the objective is to provide a support system which does not detract from the reliability of the mechanical train.

The lube oil system can be conveniently integrated on the compressor package by using the base as the reservoir and panel mounting of all the lube oil system components.

Controls and Instrumentation

Controls and instrumentation are very much influenced by customer standards and can vary from basic switches and gauges to a full blown and complex electric, electronic, pneumatic system, with failsafe monitoring, process controls and seal support system controls. Controls and instrumentation can be panel mounted on the compressor base to obtain the reliability of the integrated system approach.

APPLICATIONS

The most logical fields of application of the single stage, high speed centrifugal compressor are those where high stage head and/or special materials for the gas passage are required, since the number of parts exposed to the gas stream are few.

Typical application where this type of machine has been used successfully are:

Air and Gas Compression
Vapor Compression
Recycle Compression
Oxygen Compression
Corrosive Gas Compression

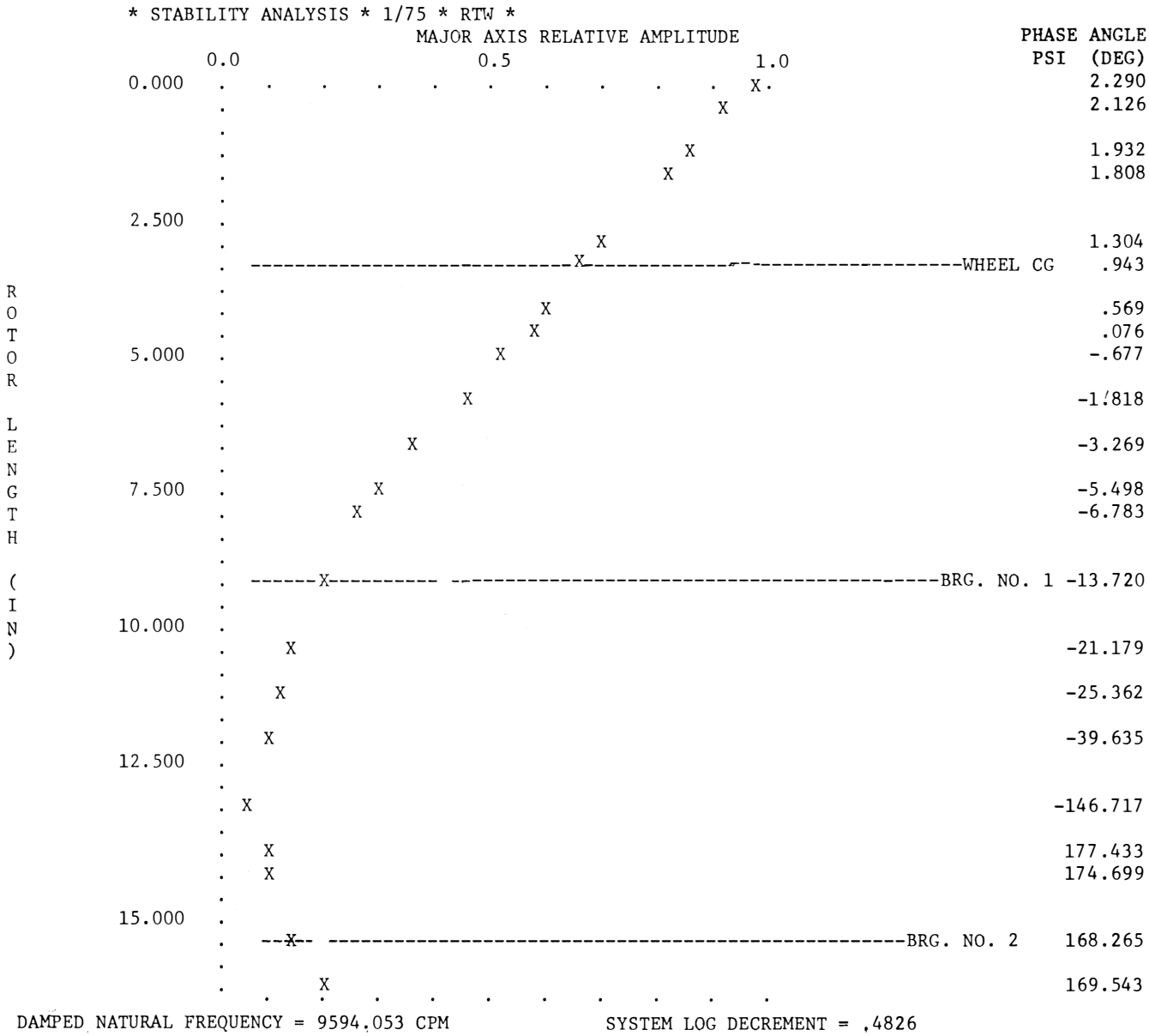


Figure 13. Sampled First Natural Frequency Mode with Associated Log Decrement.

Toxic Gas Compression

Because of the single stage, few parts are required compared to a multistage. This enhances the reliability, with less chance of rubs since only one seal and one impeller are involved.

FUTURE DEVELOPMENTS

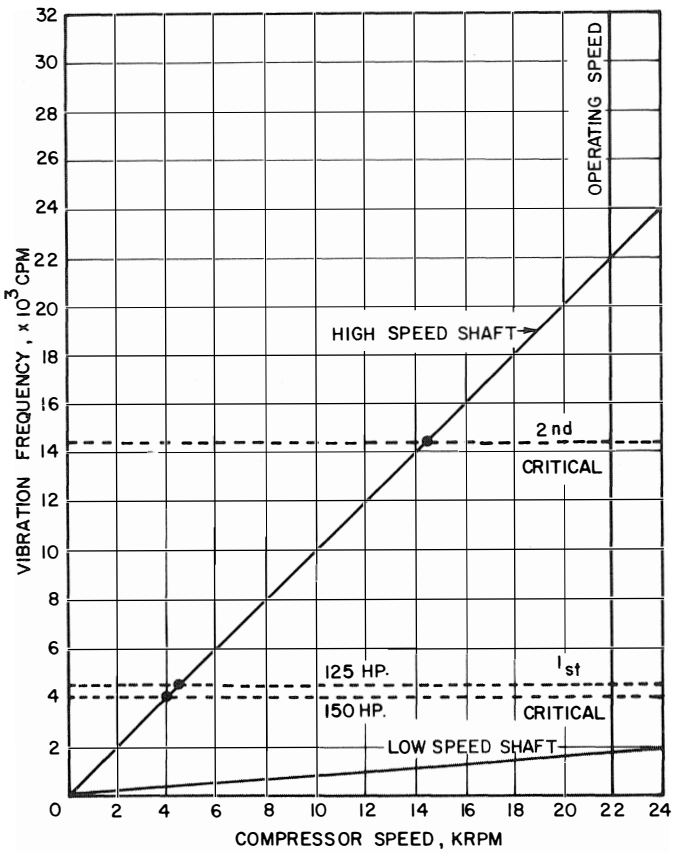
Performance improvement in terms of efficiency and surge margin are to be expected. The quoted efficiencies are not as high as some of the advanced developments as quoted by C. Rogers [5] et al., which reflect the utmost in sophisticated aerodynamic design, machined impellers and very high surface finish and perfectly matched components. Investment castings still suffer from nonuniformities, material impurities, (requiring thicker blading) and surface finish, all affecting efficiency and surge margin.

Surge margin can be increased by variable inlet guide vanes, but only in combination with a variable diffuser or vaneless diffuser. Variable inlet guide vanes have been supplied on many industrial compressors. Variable diffusers are still in the development stage as far as the process industry is concerned, but basically should not present problems.

Costs of machined impellers have declined steadily because of N.C. machining and with the increased pressures of energy shortage the efficiency penalty will eventually offset the cost difference between machined and cast impellers. This trend will result in improved impeller efficiencies because of thinner blading, uniform blade spacing as well as smooth surface finish. It also would remove the restriction of selecting impellers from a given set of castings.

CONCLUSIONS

Reliable high speed single stage overhung centrifugal compressors are now available for application in the process industry



UNDAMPED TORSIONAL CRITICAL SPEED

Figure 14. Undamped Torsional Critical Speed.

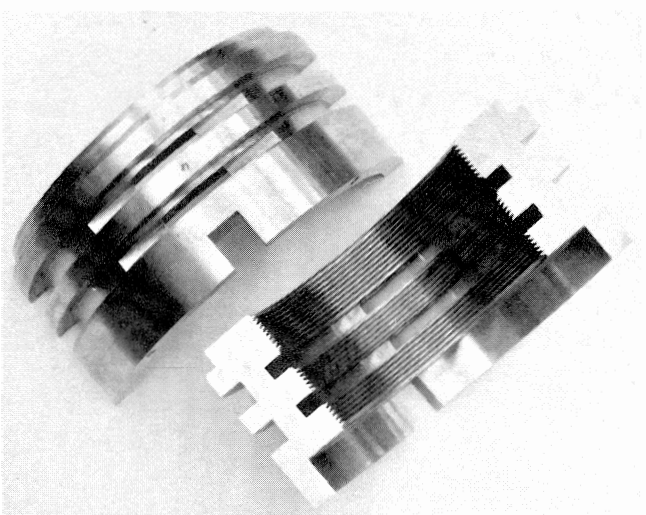


Figure 15. Buffered Labyrinth Seal.

as an alternative to the multi-stage centrifugal compressors. They are competitive from a reliability and efficiency standpoint at generally lower cost.

Present restrictions and limitations may be summarized as

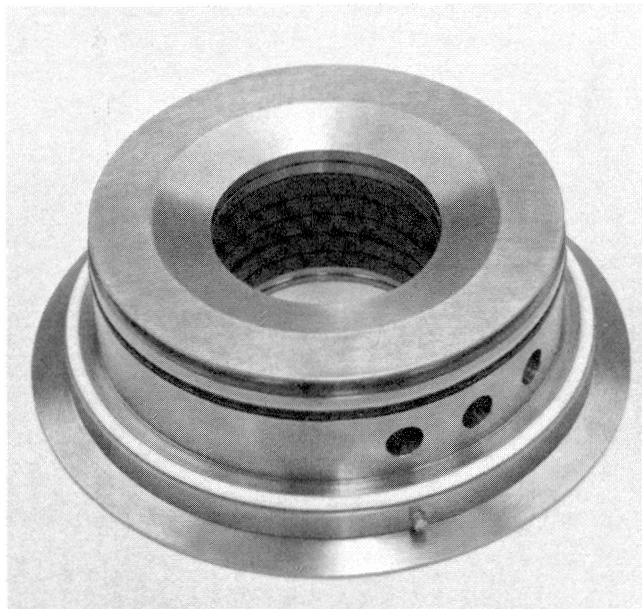


Figure 16. Oil Bushing Seal.

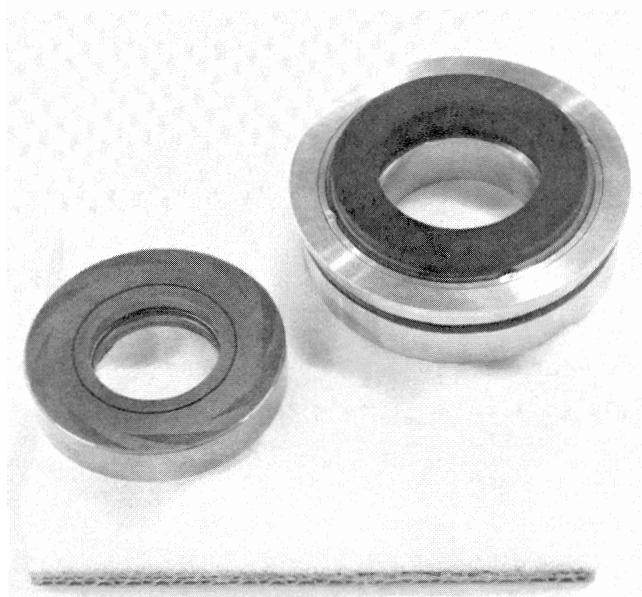


Figure 17. Dry Kinetic Wedge Seal.

follows:

Head	45000 feet
Tip Speed	1500 FPS (17-4 PH Impellers) (Higher Tip speeds and heads are offered in Titanium alloy)
Efficiency	82% Polytropic, 79% Adiabatic
Range	16% at max. heads (see Figure 4)
Head rise to surge	5%

Stable operation regardless of range or head rise to surge exist as long as the intersection of system resistance and compressor characteristics is well defined.

Reliability and minimum field start up problems are enhanced by the integrated design approach, complete system testing and horizontal split gearbox design.

Buffered labyrinth seals, oil bushing seals and dry kinetic wedge face seals have been employed successfully for a large range of operating conditions.

Variable inlet guide vanes and diffusers will improve the range and application of the high speed single stage centrifugal compressor in the future.

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