DESIGN AUTOMATION OF CENTRIFUGAL COMPRESSORS

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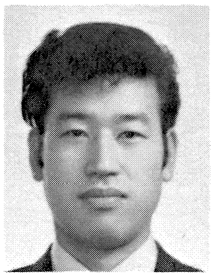

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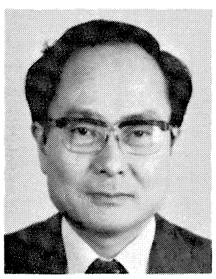

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ABSTRACT

Centrifugal compressors used in various plants come in diverse specifications with regard to capacity, pressure and gas handled. In designing such compressors, important considerations are to shorten design time and save labor as well as to meet user's requirements. In view of these needs, the authors developed a design automation system and has successfully applied the system to the design of all centrifugal compressors for various chemical plants. Design automation of such centrifugal compressors are described.

INTRODUCTION

With recent construction of huge plants and normalization of their operation, the application of centrifugal compressors, which are among the essential components of such plants, has expanded considerably, resulting in the diversification of flow capacity, pressure and gases that they must handle.

Any serious accident with such compressors may endanger the lives of inhabitants in the neighborhood or jeopardize the environment. As such accidents may threaten the existence of plants, requirements for equipment reliability have become increasingly severe.

As a result of recent technological advances in process engineering, it has become necessary to speed up the design of compressors for new services and coordinate it with the overall planning of the plant in a short span of time.

In short, reliability and rapid design have become essential requirements in recent years. The use of computers has become indispensable to meet these advanced requirements in compressor design.

In consideration of these requirements, this paper describes the necessity of design automation for multistage centrifugal compressors to be custom manufactured, citing several examples.

SEQUENCE OF CUSTOM DESIGNING

The basic sequence of custom designing is as shown in Figure 1. Compressor designing consists of fluid dynamic study, stress analysis, material selection with consideration of corrosion resistance, design of mechanisms such as bearings and seals, and vibration study, including critical speed and response due to mass unbalance distribution. Needless to say, each design requires rich experience and good judgment of design engineers.

What is most important is that these individual designs are well balanced on the whole. If the consideration is inclined too much to one field of technology; for instance, if more attention is paid on the fluid dynamic performance than on stress or vibration analysis, the diameter of the shaft might become too small or spacing between the impeller too large, and eventually it will give an unfavorable result such as low critical speed or a shaft which is too sensitive to mass unbalance.

Formerly, only a small part of design was worked out with the aid of computers, leaving the greater part to the engineer's judgment. In such designs, however, differences among design engineers were carried into finished products, and also case study took a lot of time when applying compressors to a new process.

STANDARDIZATION AND AUTOMATION

Standardization will be the first step in optimizing the engineering process.

Standardization may be interpreted that standard series of geometrical dimensions and materials are prepared in advance for certain products or parts and the user's specification is brought into conformity with such a series.

In the case of parts, such as impellers, standardization proves significant if a plurality of machines of the same design can be sold to different customers as the gas to be handled is nearly the same and the pressure and capacity are limited to certain ranges.

In the case of centrifugal compressors to be made to order, for which the gas to be handled, capacity, pressure and temperature are so diverse, the fluid dynamic design probably cannot be the same as that of compressors previously manufactured.

Also, judging from recent technological advances in this field of industry, it can be readily imagined that entirely new types of centrifugal compressors for new gas or new services will be required in the future.

If the impellers standardized in the conventional sense are to be applied under the above-mentioned condition, therefore, dimensions of the standardized impellers, such as the outside diameter or outlet width, will have to be modified.

If, for example, the flow rate and the velocity triangle at the exit of the impeller are the same but the molecular weight of the gas to be handled is different, the shroud angle α, as shown in Figure 2, increases with increasing molecular weight because of the greater density ratio across the stage. In this case, the relation between the flow rate at the inlet and the head shifts to the right as the molecular weight increases as shown in Figure 3.

Figure 4 shows how the deviation of design operating point at the second stage of a four-stage centrifugal compressor affects the overall performance. *A* shows a correct example, *B*

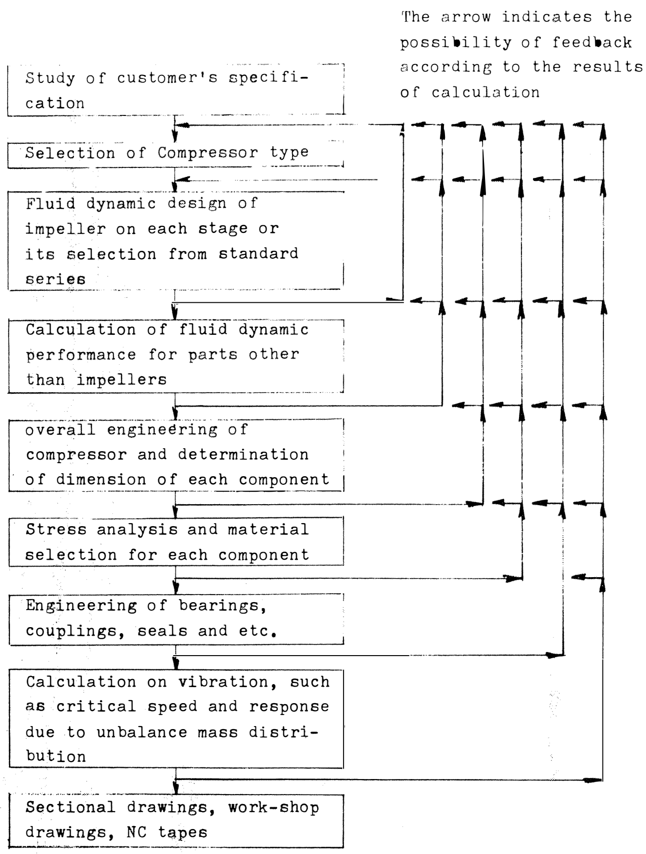
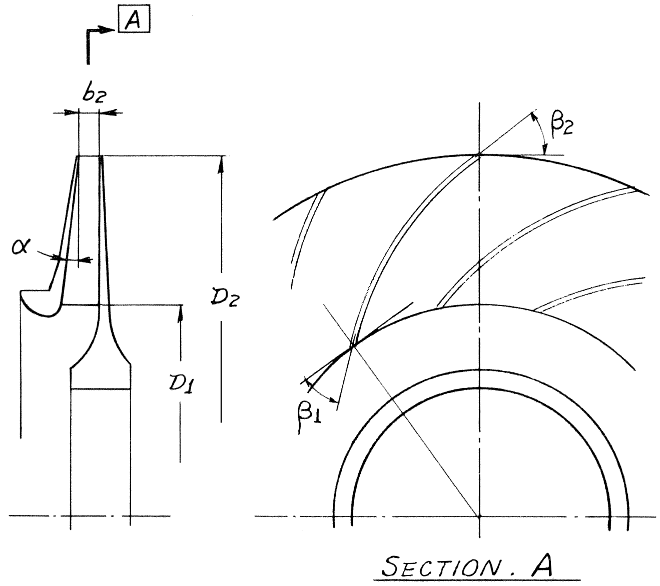
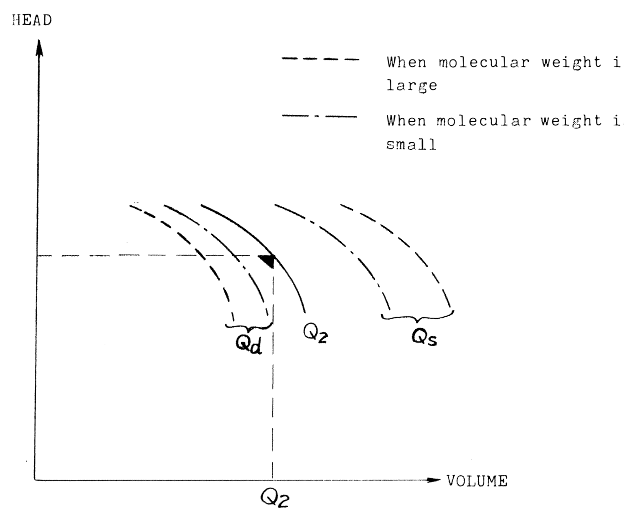


Figure 1. Typical Engineering Sequence for Multistage Centrifugal Compressors (the Arrow Shows a Loop for Correction or Optimization on the Basis of the Results of Calculation of Fluid Dynamic Performance, Stress and Vibration Analysis and Etc.).



Note: D_2 = outside diameter; D_1 = inside diameter; β_2 = outlet angle of blades; β_1 = inlet angle of blades; α = shroud andle; b_2 = outlet width

Figure 2. Major Geometrical Dimensions of Impeller in Reference to the Performance.



Q_s ; flow rate at impeller inlet

Q_2 ; flow rate at impeller outlet

Q_d ; flow rate at inlet of the next stage

Figure 3. Relation Between Flow Rates at Inlet and Outlet and Molecular Weight in Case Where the Velocity Triangle at the Impeller Outlet is the Same.

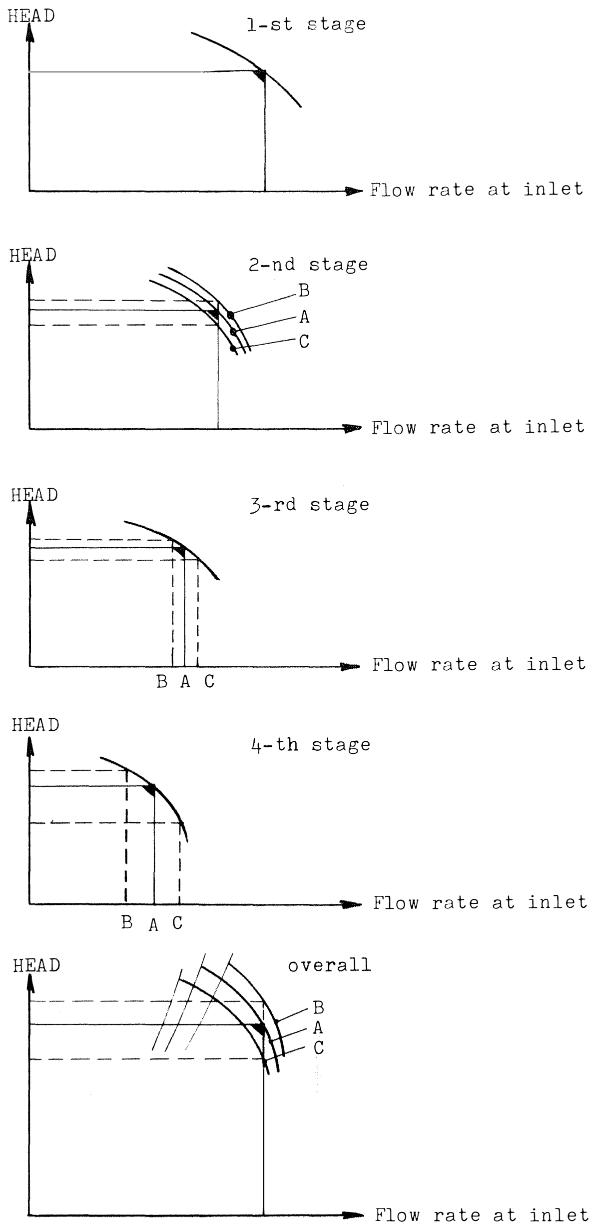


Figure 4. Effect of the Deviation of Designed Operating Point on the Overall Performance (Note: ◥ Indicates the designed Operating Point).

an example where the compressor is operated at a smaller flow rate, and C an example where the compressor is operated at a larger flow rate. In the case of B and C, the deviation of operating point increases gradually. Concerning the overall performance, the allowance for the surge point is decreased in the case of B, but the desired head cannot be obtained in the case of C. In multistage compressors, even a small error in designing at a stage increases the deviation of the operating point at the succeeding stages, resulting in a large difference in expected overall performance. If there is no impeller in the standardized series to satisfy the requirement for the second stage, then some modification must be done to an impeller which is the nearest to the required one. However, the modification of dimensions of a standard impeller means deviations from the geometrical standardization.

Assuming that, for example, the outside diameter D_2 of the impeller and the outlet angle β_2 of the blade as shown in Figure 2 are fixed, let us change in 10 ways each of three major geometrical dimensions, a, D_1 and β_1, which are related to the performance of a two-dimensional impeller. These changes would result in 1,000 kinds of impellers. To select one out of these 1,000 kinds, the velocity triangles at the inlet and outlet of the impeller and the rate of deceleration in relative velocity in the impeller passage must agree with the past results.

Therefore, the standardization of impellers is not the standardization of geometrical dimensions. It is fluid dynamic standardization. The geometrical dimensions are no more than the results of fluid dynamic standardization.

The validity of such fluid dynamic standardization can be proven by the close analysis of the results of factory tests or field tests. Generally, it is impossible to obtain the performance at each stage of a multistage centrifugal compressor from its overall performance. However, if all stages are similar fluid-dynamically, the performance at each stage can be determined by the iteration method from overall performance. The author et al. have developed an optimum design system. This design system will be described in detail in the succeeding chapter. This system is largely based on fluid dynamic standardization which is always proven by the analysis of test results.

Fluid dynamic standardization makes it necessary to compare the performance of all impellers in excess of 1,000 kinds in the design stage, as mentioned before, and to manufacture a plurality of impellers, each having different geometrical dimensions. To eliminate this complexity, an automatic design system employing electronic computers and a numerically-controlled machining system connected to the design system are required.

AN EXAMPLE OF AUTOMATIC DESIGN SYSTEM

Based on the above-mentioned, the automatic design system for multistage centrifugal compressors which has been developed by the author et al. will be introduced hereinafter.

This system is designed to accomplish all of designing steps shown in Figure 1 through optimization process. The system is epoch-making in that sectional drawings, work shop drawings and data for NC machining can be automatically worked out.

Type of System

This system employs the so-called batch system but not the interactive system. That is, the customer's specification is used as input data and all tasks associated with judgment and selection are incorporated into the system so that processing is continued until the optimum results are obtained. In the interactive system, judgment is made by design engineers. Accordingly, differences in the quality, sense or taste of engineers may result in the lack of unity of engineering concept.

The outline of this system is given below.

Calculation on Gas Properties

The first problem to be encountered in the design of centrifugal compressors is to know accurately the properties of the mixture of gas to be handled, such as enthalpy, entropy and compressibility, at a certain pressure and temperature. This system has computing programs of gas properties, such as (1) NGPSA data [1] [2], (2) computing programs based on the B.W.R. method [3] and Redrich-Kwong method [4], and (3) designation according to input data. Therefore, this system is capable of computing the physical properties of almost all gases to be handled by centrifugal compressors, such as air, nitrogen, methane, ethane, propane and other hydrocarbon gases, carbon-dioxide, hydrogen sulfide, and mixture of these gases. The results calculated by the B.W.R's equation or Redrich-Kwong's equation would not be accurate enough at all times, because these depend largely on the coefficients which must be given to the computer, and therefore this system employs a program which was appropriately modified by comparing the results of these equations with other data.

As an example, the results of calculation for a centrifugal compressor, which is 1 ATA in suction pressure and 260 ATA in discharge pressure, for a urea synthesis process is shown in Figure 5. The gas to be handled by this compressor is mainly carbon-dioxide, with a small percentage of nitrogen and hydrogen. Three intercoolers are provided between four compression groups. As the gas composition is varied slightly from group to group, four sets of Mollier diagram are calculated and drawn by an X-Y plotter, as shown in Figure 5.

Design of Fluid Dynamic Performance

This design concerns the calculation of the fluid dynamic performance of impeller, diffuser, return vane, scroll, etc., as well as the determination of dimensions according to the results of calculations.

The basic concepts underlying the fluid dynamic design of impellers are that the flow distributions at the inlet, interior and outlet of the impellers are always made similar for each stage. The compressor type, rotating speed, shaft power and the dimensions of parts relative to the fluid, such as impellers at all stages are obtained as the result of calculations.

Needless to say, the above-mentioned calculation includes the calculation of the so-called internal leakage through the impeller eye and interstage labyrinths and through balance labyrinth and external leakage in the shaft end labyrinth seals. These leaks are taken into consideration in designing the impeller.

The return vanes are of the modified NACA airfoil section to guide the flow to the impeller inlet at the next stage correctly with minimum loss.

The results of performance tests of a multistage centrifugal compressor designed under the above-mentioned concepts are fed into the feedback program so that the performance of each part, such as the impeller and the diffuser at each stage, is estimated from the overall performance of the compressor by the iteration method. Thus, the validity of design can be checked at all times.

Stress Analysis

The finite element method is employed for the calculation of centrifugal stress and deformation of rotation parts such as impellers, inter-stage sleeves and balance drums, and for the calculation of complex stresses for the diaphragms subjected to differential pressure, and the casing containing internal pressure.

As examples, the results of calculation for the impeller and diaphragm are shown in Figures 6 and 7. This finite element method is incorporated into the system after comparing the results of calculation with the stresses actually measured using a large number of strain gauges [5] [6], and therefore this is considered to be a most reliable calculation method.

Composition of Mechanical Elements

Component parts such as journal bearings, thrust bearings, housings for them, vertically split casings and end covers have been standardized for each series and stored in the system.

The diameter of the middle of the shaft to which the impeller of each stage is shrink-fitted is determined mainly according to the shape of the sleeve forming the fluid path, stress and the rigidity of the shaft in consideration of vibration to be described in the next paragraph. As the diameter of the shaft end outside of the end labyrinth must be smaller than that of the middle part, shaft ends of two or three standard sizes are provided for one frame. These standardized parts are interchangeable among the frames, thus reducing the number of standard parts to be made available.

At the shaft end, either a labyrinth seal or an oil-film seal is used according to the specification of the input data. In the case of labyrinth seals, the amount of external leakage is calculated. In the case of oil-film seal, the clearance between the seal ring and the shaft is determined with due consideration of all special conditions, including the starting conditions. The amount of oil required for sealing, oil pressure and oil temperature rise are calculated for all operating conditions, and the rating of the seal oil pump is decided.

The coupling is selected from the standard gear coupling series. The connection of the coupling to the shaft is either taper fit with key or hydraulic shrink fit.

The diameter of the balance drum is determined so that the residual thrust is less than approximately ¼ of the allowable specific load of the thrust bearing under the normal operating conditions and is less than half of the allowable load when the gear coupling slips with a coefficient of friction of 0.25 [7].

For the rotating parts such as the impeller, sleeve, balance drum and shaft, whose dimensions have been determined through the fluid dynamic and stress calculations, their weight, moment of inertia and center of gravity are calculated as a step preparatory to the vibration calculations to be done next.

Calculation of Vibratory Characteristics

The dimensions of all component parts of a centrifugal compressor have been determined as the result of the calculations described above. Typical vibration calculations include (1) the calculation of lateral critical speed, (2) the response due to unbalanced mass distribution, and (3) the calculation of torsional vibration.

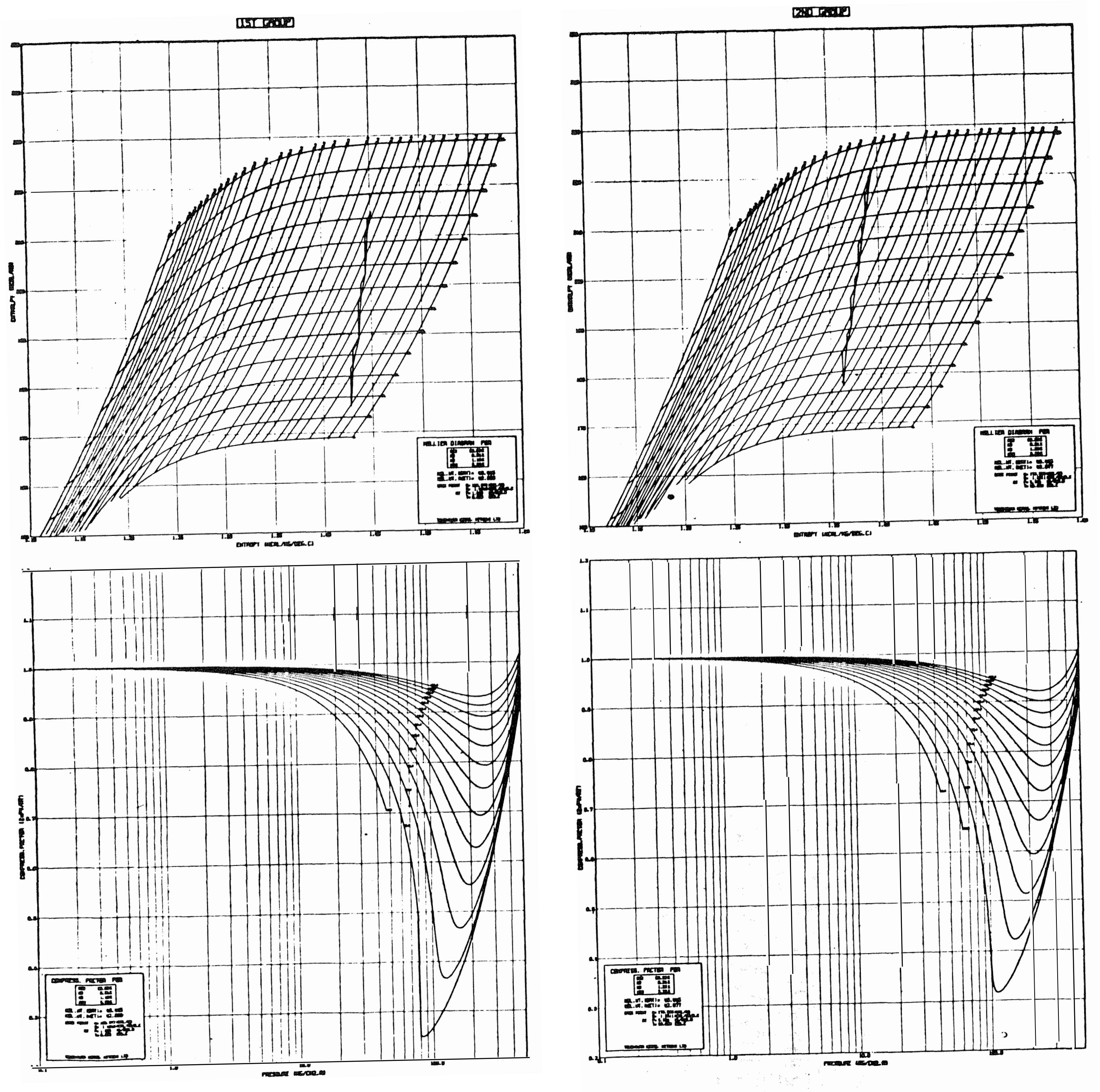
For the calculation of lateral critical speed and response due to unbalance mass distribution, the stiffness and damping coefficient of the oil film in the bearing are taken into consideration. For example [8], the transfer matrix method is employed for the calculation of critical speed, thus obtaining not only the critical speed but also the relative amplitude in each cross section of the shaft, phase angle and moment. An example of the output map thus obtained is shown in Figure 8.

For the calculation of response due to unbalanced mass distribution, the modified Myklestadt-Holzer method is employed [9]. As this type of centrifugal compressor is operated at a speed between the first critical speed and the second critical speed, the calculated unbalances are added so as to cause vibrations at these critical speeds.

An example of calculation results is given in Figure 9.

Process of Optimization

The explanation of designing multistage centrifugal compressors has been given in the first five subsections. For smooth operation of this system, it is essential to provide a control center for controlling these calculation processes. If any abnormality is caused in the course of these complicated calcu-



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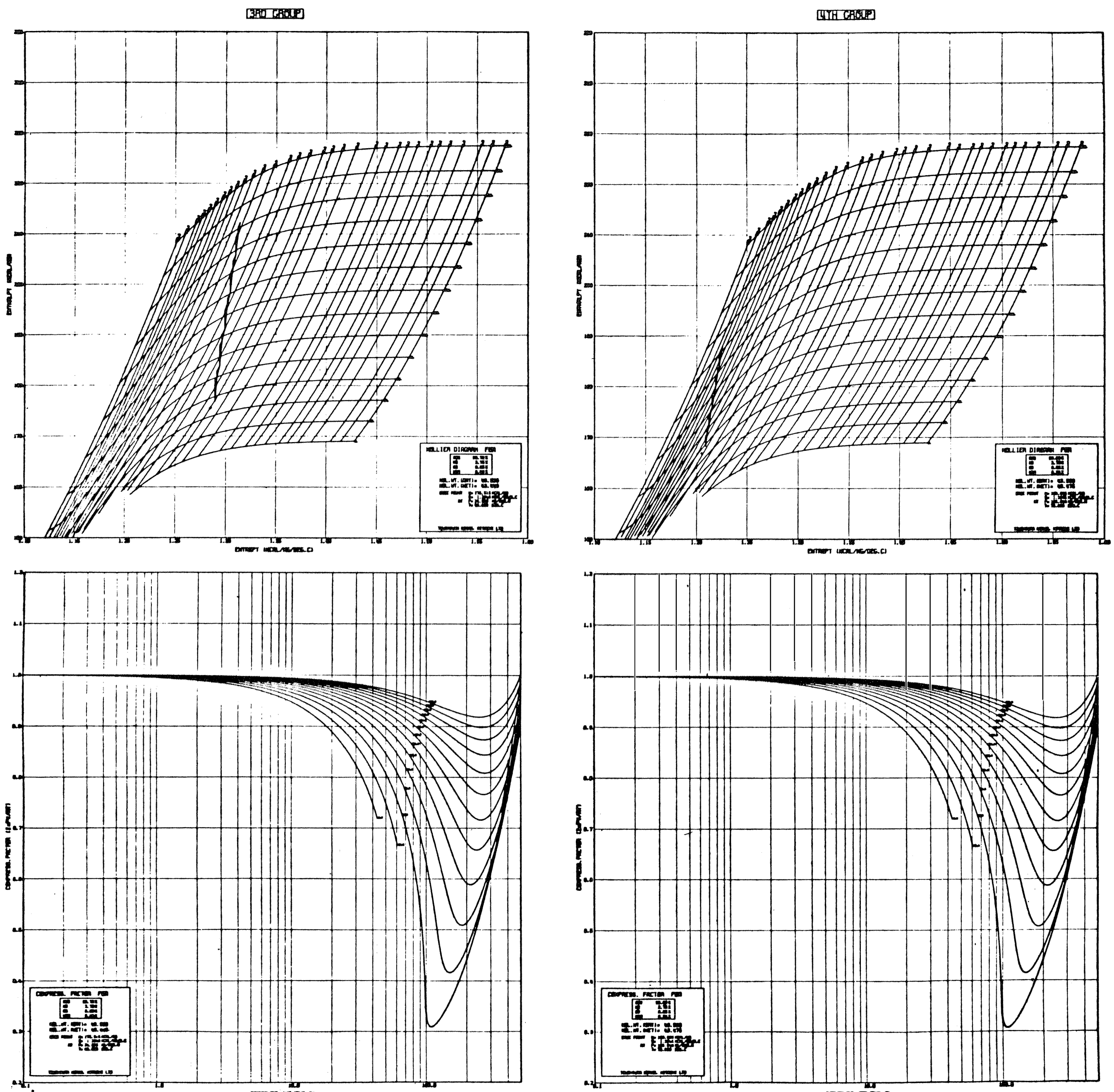

Figure 5. Example of Calculation Results of Gas Properties (Carbon Dioxide Gas Compressor for Urea Synthesis Plant).

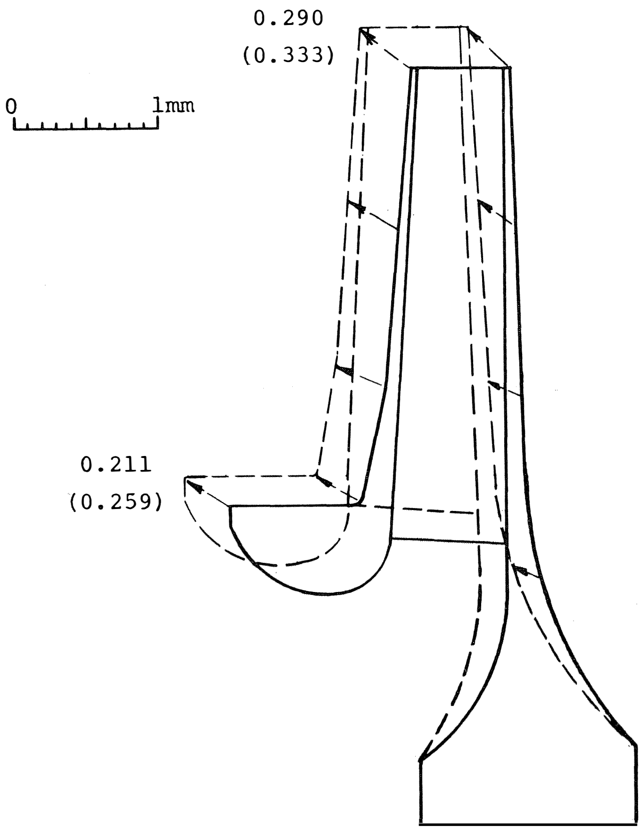

Figure 6. Results of Calculation on the Deformation of Impeller Due to Centrifugal Force (Finite Element Method)

Figures in the Upper Row Indicate the Amount of Displacement (mm) in the Radial Direction, but the Figures in Parentheses Indicate the Amount of Displacement (mm) in the Axial Direction.

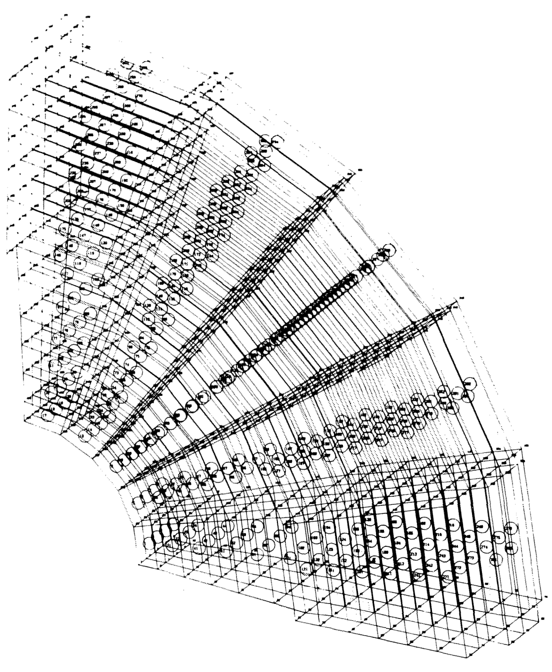

Figure 7. An Example of Calculation of the Deformation of a Half-Split Diaphragm Due to Pressure Difference Between Stages (Finite Element Method).

lations, judgment must be exercised as to the effect on the dimensions and selection of parts and materials.

For this purpose, this system incorporates a program for judgment relating to the fluid dynamic performance, stress and vibration, as well as a program for comprehensive judgment. Thus, the calculation is continued in a comprehensive manner until results, satisfactory in all respects, are obtained while changing the dimensions, shape, material and standardized parts to be selected through the process of optimization by the above-mentioned closed loops.

Information on Automatic Preparation of Sectional Drawings and on Manufacture

As the automatic design of a centrifugal compressor has been completed through the above-mentioned steps, the results of calculations are printed out by a line printer, as shown in Figure 10. At the same time, a sectional drawing is automatically drawn by an X-Y plotter. An example of the automatically drawn sectional drawing of an 8-stage centrifugal compressor is shown in Figure 11.

Information for manufacturing is given in the form of a work-shop drawing or NC tape. An example of a work-shop drawing automatically drawn by an X-Y plotter is shown in Figure 12. An example of NC tape check is shown in Figure 13. Figure 12 shows a part of the work-shop drawing for impeller eye labyrinth. Figure 13 shows a tool path for NC milling of a return vane.

PRACTICAL APPLICATION AND FUTURE DEVELOPMENT

Since this system was developed, almost all multistage centrifugal compressors have been designed by this system. At present, several dozens of compressors designed by this system are in satisfactory commercial operation at various plants.

This system is capable of designing the compressors of the same structure with the highest possible degree of judgment and reliability without relying upon the opinion or experience of design engineers. As the system is connected directly with the workshop drawings or NC tapes, errors attributable to engineering are entirely eliminated. Furthermore, as the system takes into consideration the manufacturing techniques and assembly jig to be employed, errors in manufacture are also eliminated. As the design of one centrifugal compressor is completed in about half a day, the system can be applied for technical quotations that must be completed in a short span of time.

New developments to meet progress have been continued. Efforts must be made continually to update the system by incorporating the new developments into the system.

CONCLUDING REMARKS

1. The necessity and examples of design automation for centrifugal compressors have been outlined which employs the batch system so that all design works from the customer's specification to make sectional drawings, workshop drawings of all parts and NC tapes.

2. It is emphasized that the fluid dynamic similarity in design based on the results of analysis of actual test data is the most important from the performance point of view.

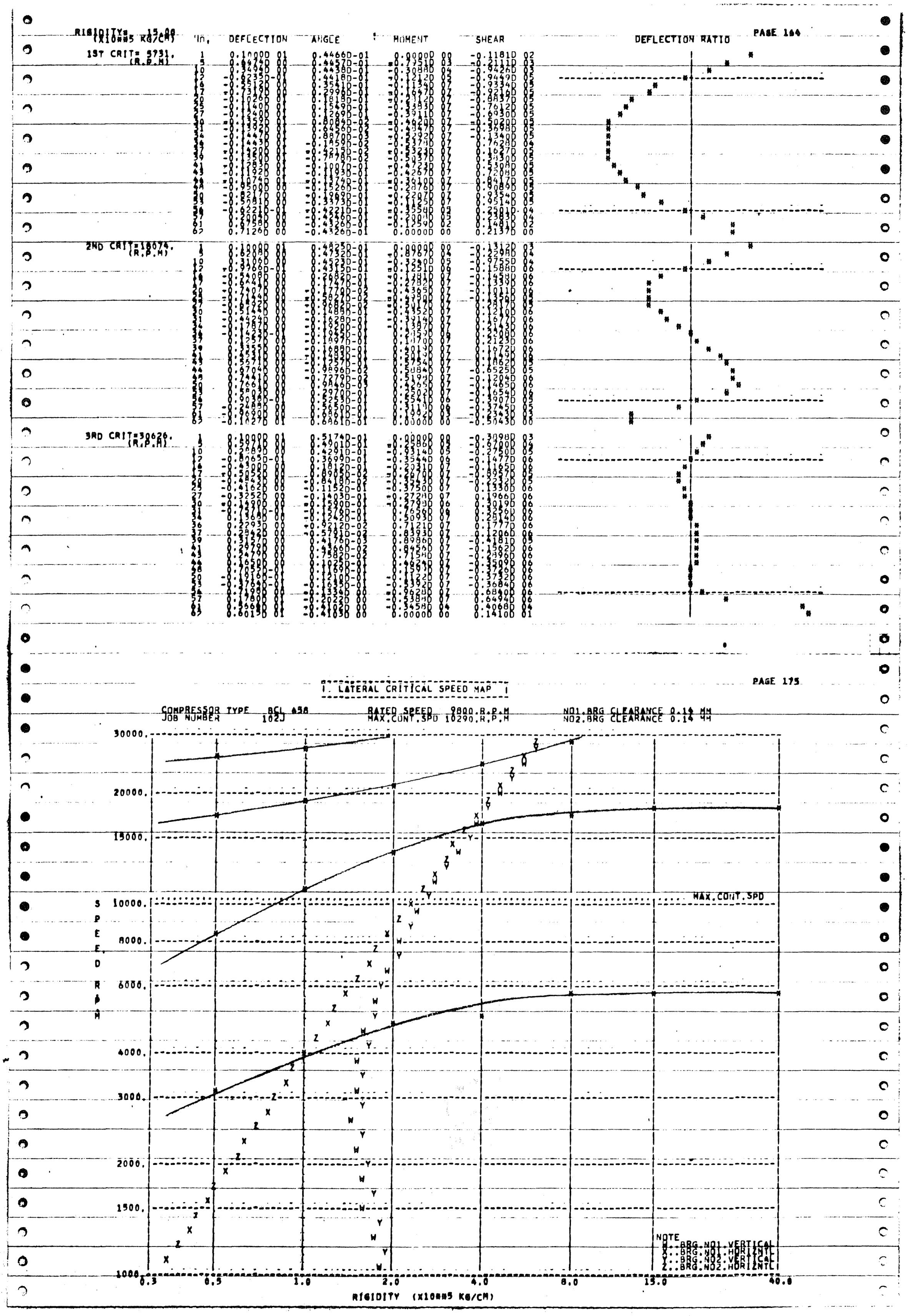

Figure 8. An Example of Calculation Results of Critical Speed (Line Printer).

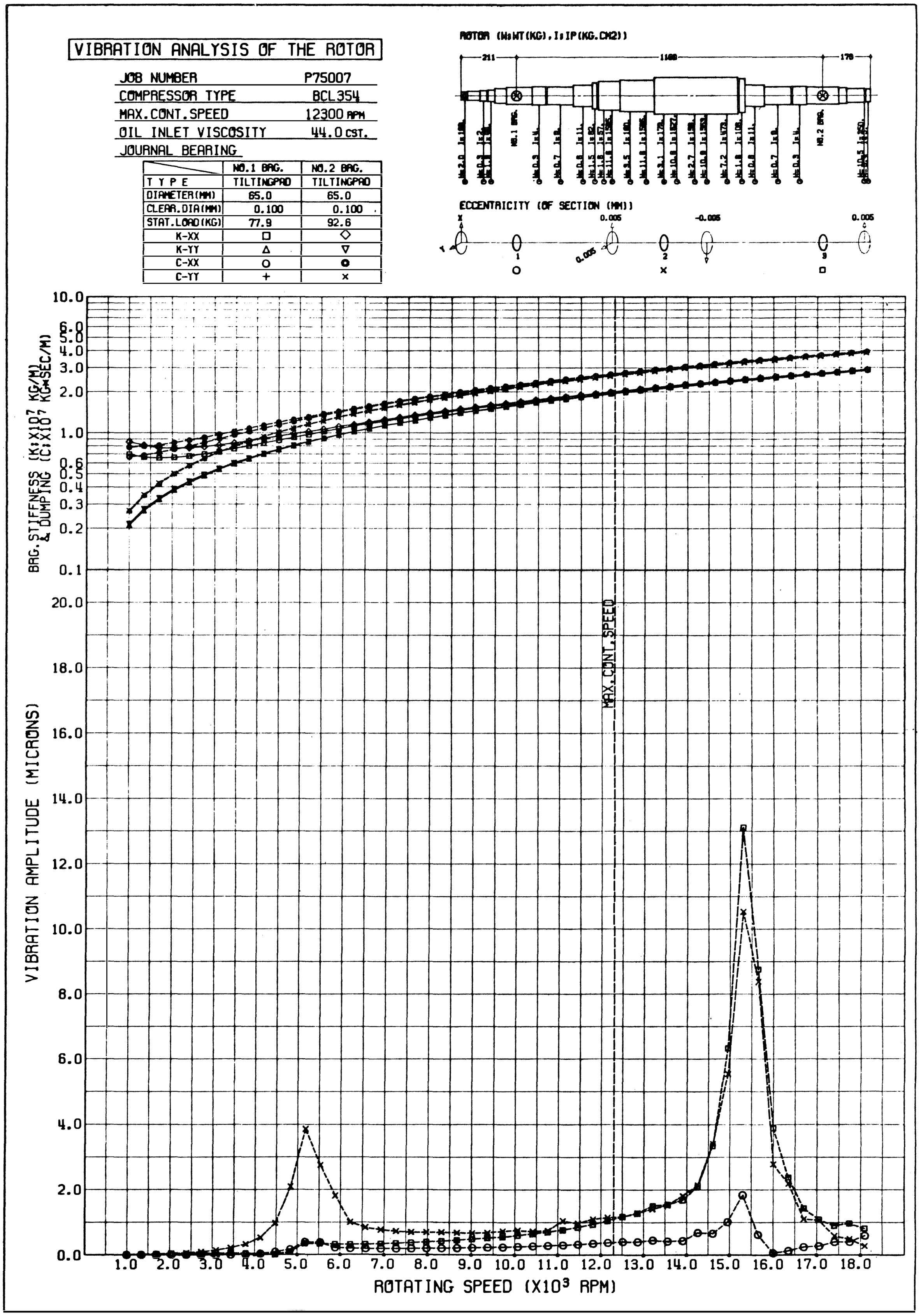

Figure 9. An Example of Calculation Results of Response Due to Unbalance Mass Distribution (X-Y Plotter).

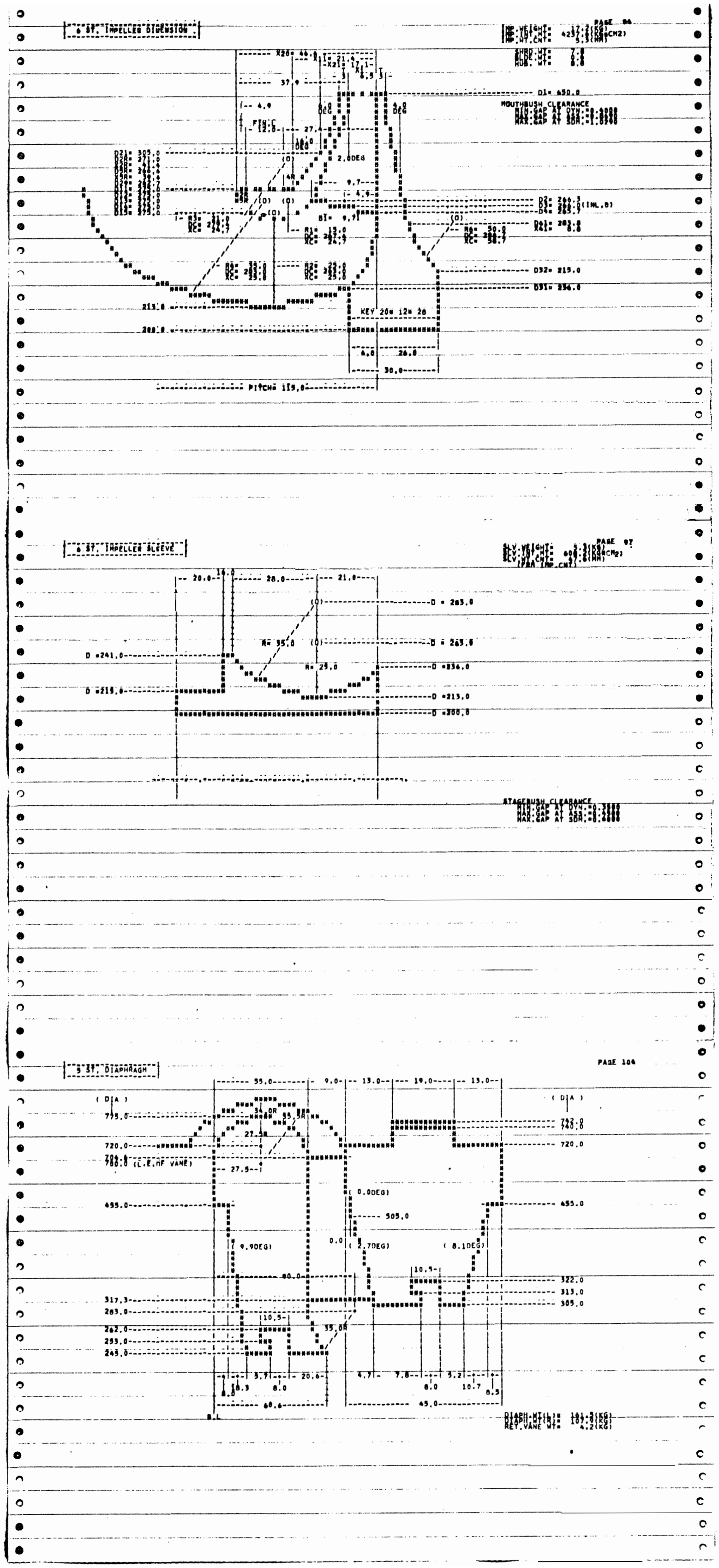

Figure 10. An Example of Output List by a Line Printer.

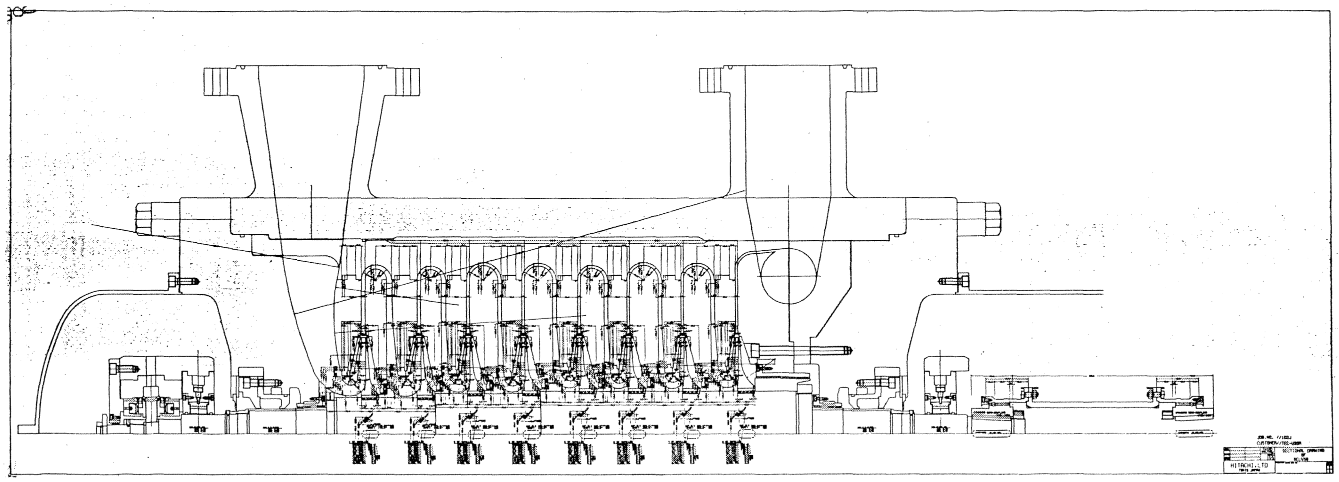

Figure 11. Sectional Drawing of an Automatically Designed 8-Stage Centrifugal Compressor Which has been Automatically Drawn by an X-Y Plotter.

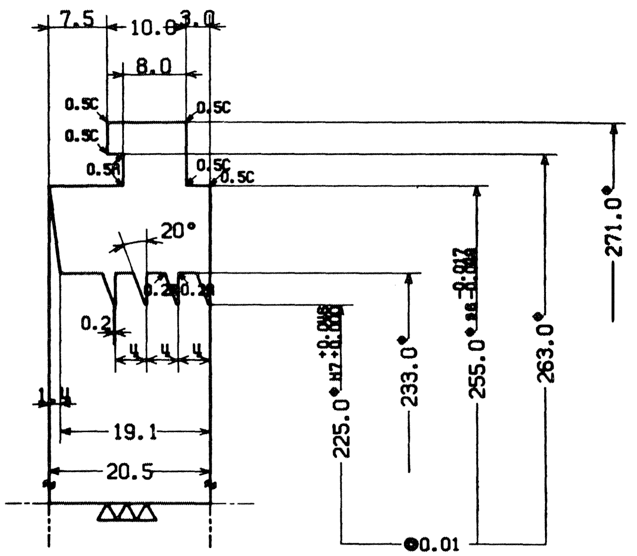

Figure 12. An Example of Automatically Drawn Work-Shop Drawing (Impeller Eye Labyrinth).

3. Finite element method is included in this system for the stress analysis of rotating parts such as impellers, sleeves as well as the stationary parts such as diaphragms and casings. The reliability of this method was proven by comparing the results of calculation with the stresses actually measured.
4. It is essential to provide the process of optimization for judgment and control of calculations for fluid dynamic performance, stress analysis and vibration analysis.
5. Almost all multistage centrifugal compressors have been designed by this system, and at present, several dozens of compressors designed by this system are in satisfactory commercial operation at various plants.

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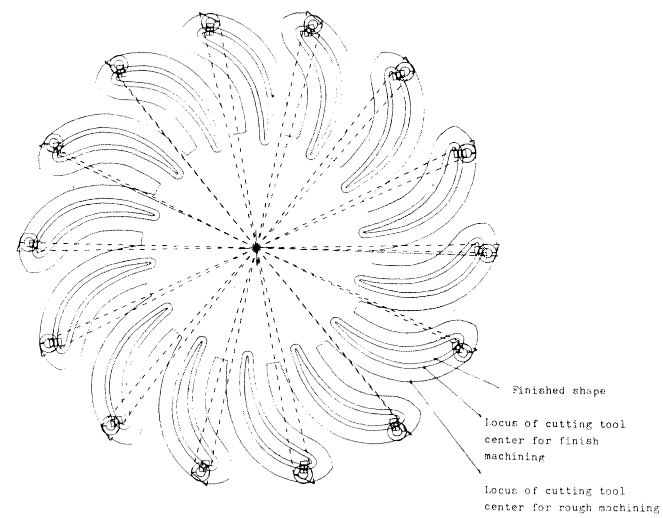

Figure 13. Tool Path for NC Milling of Return Vanes.

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