# Rotordynamics on the PC: Further Capabilities of ARDS

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## ABSTRACT

Rotordynamics codes for personal computers are now becoming available. One of the most capable codes is ARDS (**A**nalysis of **R**otor**D**ynamic **S**ystems) which uses the component mode synthesis method to analyze a system of up to 5 rotating shafts. ARDS was originally written for a mainframe computer but has been successfully ported to a PC; its basic capabilities for steady-state and transient analysis were reported in an earlier paper.

Additional functions have now been added to the PC version of ARDS. These functions include: 1) Estimation of the peak response following blade loss without resorting to a full transient analysis; 2) Calculation of response sensitivity to input parameters; 3) Formulation of optimum rotor and damper designs to place critical speeds in desirable ranges or minimize bearing loads; 4) Production of Poincaré plots so the presence of chaotic motion can be ascertained.

ARDS produces printed and plotted output. The executable code uses the full array sizes of the mainframe version and fits on a high density floppy disc.

Examples of all program capabilities are presented and discussed.

**Key Words**: Rotordynamic Analysis; Machinery Vibration Analysis; Vibration Control; Transient Analysis; Personal Computers

## INTRODUCTION

Personal computers are now used extensively for engineering analysis; their capability exceeds that of mainframe computers of only a few years ago. Programs originally written for mainframes have been ported to PCs to make their use easier. One of these programs is ARDS (Analysis of RotorDynamic Systems) which was developed at Arizona State University to quickly and accurately analyze rotor steady state and transient response using the method of component mode synthesis [1]. The original ARDS program was ported to the PC in 1995 and reported in [2].

The mainframe ARDS was augmented by Nelson, et al, at Arizona State to increase its capability [3-6]. These enhancements include: 1) Estimation of the peak response following blade loss without resorting to a full transient analysis; 2) Calculation of response sensitivity to input parameters; 3) Formulation of optimum rotor and damper designs to place critical speeds in desirable ranges or minimize bearing loads. These enhancements have now been added to PC ARDS. In addition, squeeze film damper calculations now allow the damper to be placed in series with the shaft bearing (the normal industry practice). Furthermore, transient analysis output can now be viewed on Poincaré plots to better enable the detection of chaotic or other nonharmonic motion.

Program results were verified by comparison with the ARDS mainframe output reported in [3-6]. Additionally, the series squeeze film damper analysis was verified by comparing results with those in [7].

## PROGRAM DESCRIPTION

ARDS was originally written in Fortran 77; this language has been retained for the present PC version.

Peak blade loss response.-Transient vibration amplitudes following a suddenly applied rotor imbalance may be larger than subsequent steady-state vibration. Design of rotating machinery must account for this. Time transient analyses, such as performed by ARDS, can predict this vibration with good accuracy, but may be too time-consuming for the preliminary design phase. However, a quick approximation of the peak rotor displacement may be obtained via an analog of the shock spectrum analysis used in structural engineering. In concept, the response of each rotor mode to the sudden imbalance is obtained, and the responses then summed in some manner. This method has been implemented in ARDS, where four summation schemes are used: sum of the absolute values of the modes, a root-mean-square sum, and two combinations of these. The procedure is described in detail in [3].

Sensitivity of response to parameter changes.-When designing a rotor system, it is often desirable to have a set of sensitivity coefficients which quantitatively predict a change in specific system characteristics as design parameters change. The dynamic characteristics of usual interest are the system whirl frequencies, the critical speeds, and steady unbalance response. With the finite element analysis used by ARDS, the sensitivity coefficients (partial derivatives of system characteristics with respect to parameter changes) can be calculated simply. The parameters considered are bearing coefficients, inertial properties of concentrated masses (discs), and the distributed mass and stiffness of the rotating shafts.

Having sensitivity coefficients makes calculation of critical speeds easy; ARDS could not do this prior to this enhancement. Full details of the procedures for determining sensitivity coefficients and critical speeds are in [4].

Determination of optimum rotor system design.-Successful design of rotor systems requires the satisfaction of several operational constraints. Two of the more significant are placement of critical speeds (usually required to be outside the normal operating speed range) and minimization of bearing forces. Satisfying the first of these constraints is greatly facilitated by having sensitivity coefficients, described above. In theory, an optimum design could be formulated with the sensitivity coefficients and reanalysis using a cut-and-try process; however, the large number of design parameters makes this a daunting task. Use of available optimization codes makes the automation of the optimization process feasible.

The enhanced ARDS casts the optimum design process as a nonlinear programming problem that minimizes an objective function subject to performance constraints and bounds on the design variables. Two types of problem have been programmed: 1) Placement of undamped critical speeds by optimization of bearing stiffness, shaft stiffness and mass, and bearing location. 2) Minimization of bearing load for steady unbalance response by optimization of squeeze film damper design. This can be done over a range of rotating speed. In this analysis, ARDS can now handle a series arrangement of damper and shaft bearing, with a bearing mass interposed between the damper and bearing. Squeeze film properties are calculated using the short bearing approximation. ARDS optimization techniques are described more fully in [5] and [6].

Production of Poincaré plots.-A Poincaré plot produces a picture similar to what one would see if the rotor were illuminated by a strobe light. That is, a Poincaré plot records the position of the rotor at periodic intervals, usually once per revolution. The original ARDS transient analysis included the option of x-y amplitude plots. Therefore it was a simple matter to modify the plotting program to place a point on the plot once per revolution.

## EXAMPLES OF ARDS ANALYSES

The analyses described above were implemented on a 166 Hz Pentium PC; plots were produced by ARDS on a laser printer. Any consistent units may be used for ARDS input. The example problems below use in-lb-sec system input; displacement is therefore in inches and forces in pounds.

Peak blade loss response.-The NASA Lewis "workhorse" rotordynamics test rig was modeled for this exercise. The model of the shaft drawn by ARDS appears as Figure 1. This rig has a slender shaft with three discs at stations 1, 4, and 7. The discs indicated in the figure at stations 2 and 4 represent the masses of the bearing housings, and are much smaller than the others. A sudden imbalance representing a blade loss was simulated at station 1 for various shaft speeds. Figure 2 shows the peak amplitude predicted by ARDS at station 4, the rotor midpoint, for different calculation methods. The curves plotted are not smooth because results were obtained only for discrete speeds at 100 rad/sec intervals. Curves labeled "s" and "t" represent results of steady state unbalance response and a full transient analysis, respectively, and curves 1-4 represent various combinations of response spectra results. These curves illustrate how the peak transient response can be much higher than the steady state response. The true response is most closely simulated by curve 2, which is the rms value of the modal contributions; the other methods, in particular the absolute value sum represented by curve 1, considerably overestimate the true peak. This was also the case for the results reported in [3]. Note that, at the critical speed occurring near 1200 rad/sec, curve 2 predicts a peak response lower than the steady state response, although the percentage difference is small. Despite the disparity between the approximate methods and the true peak response, the saving in calculation time (45 sec vs 9 sec for the calculations of fig. 2) makes the response spectra method useful in the early design phases. Of course, a full transient analysis should be used to verify the final design.

Sensitivity of response to parameter changes.-A sensitivity analysis was run for the same rotor system as above for the blade loss data. Samples of printed output are shown as Figures 3 and 4.

Figure 3 gives the sensitivity of the first critical speed (764 rad/sec for the base conditions) to changes in several system parameters. The rotor system analyzed is symmetric end-to-end; therefore sensitivity coefficients for, e. g., station 2 also apply to station 6.

Program output is interpreted as follows. The number of damped critical speeds requested (in this case 4) are presented. Following that is a table showing the size of parameter changes used in calculating the sensitivity coefficients. ARDS allows the size of the parameter changes to be specified to make interpretation of output easier. For the example presented, large changes were specified, in most cases 100 percent of the base values. Since the sensitivity analysis calculates derivatives, the size of the change used is arbitrary, although the actual response to such changes is often not linear.

Next the sensitivity coefficients for changes in disc properties are shown. The data show that for an increase of the disc mass at station 4 of the amount shown in the table (which equals the original mass), the first critical speed will drop by 126 rad/sec. Changes in diametral and polar inertia of this disc have virtually no effect, because the first mode is symmetric, and station 4 is the midpoint of the shaft.

Sensitivity coefficients for bearing changes are then shown. Figure 3 shows that an increase of 10 000 (lb/in.) of direct stiffness in one bearing will increase the critical speed by 55 rad/sec, while an increase of 2 (lb sec/in.) direct damping will increase the critical speed by only 1.4 rad/sec. Sensitivity coefficients for cross-coupled bearing coefficients are also calculated.

This section of sensitivity output concludes by showing changes in critical speed due to changes in shaft element length, unit mass (mass per unit length), and stiffness.

It is appropriate to state again that the sensitivity analysis calculates derivatives; results are thus strictly applicable for only small changes in parameters. The large changes of the example are for ease of interpreting results.

Figure 4 shows sensitivity of unbalance response to changes in various parameters. Center of gravity (cg) eccentricities of 0.05 mm were assumed for each of the three discs. Response to the original imbalance at the shaft speed chosen (8000 rpm in this case) is shown first. This is followed by change in response due to imbalance changes at the discs (these changes are shown on the change unit panel of fig. 3). The data show that imbalance at station 4 has a greater effect on response than that at station 1 (and also, because of the symmetric system, at station 7).

The next section deals with the effect of bearing properties. Again because of the symmetric system, the two bearings affect the response equally; only that for the bearing at station 2 is shown.

The results show that increasing the stiffness of either bearing produces a large increase in the unbalance response. Changes in bearing damping have a much smaller effect.

Finally, changes in response due to changes in shaft element stiffness are shown. Shaft stiffness has a significant effect on response, as would be expected from the large amount of shaft bending indicated in the response to initial imbalance.

Determination of optimum rotor system design.-This will be illustrated by designing a squeeze film damper for a five-disc rotor system similar to the three-disc rotor used above. In [7], a damper was designed to enable the rotor to pass through the first bending critical speed with minimum vibration amplitude. The system is illustrated in Figure 5. The shaft is supported by ball bearings which in turn are supported by a squeeze film damper and centering spring. The damper design of [7] produced low bearing force at the first critical speed, but force increased substantially above the critical speed. ARDS offers the opportunity to design a damper that will optimize response over a range of speeds. The speed range chosen was 6000-16 000 rpm, which includes the first rigid support critical speed of 8280 rpm. Maximum bearing force occurs at the maximum speed of 16 000 rpm. ARDS was instructed to optimize damper length and centering spring stiffness, while damper radius and clearance were held constant. Figure 6 shows that the ARDS-optimized design reduces the bearing force at this speed, but not substantially. It is likely that the bearing housing mass would need to be reduced to obtain a further reduction in bearing load. Table 1 lists the original and optimized damper characteristics.

**Table 1. Squeeze film damper parameters.**

| | Original design | Optimum design |
|---|---|---|
| Radius | 39.6 mm | 39.6 mm |
| Length | 14.4 mm | 10.8 mm |
| Clearance | 0.127 mm | 0.127 mm |
| Centering spring stiffness | 228 kN/m | 290 kN/m |
| Fluid viscosity | 12 mPa s | 12 mPa s |

Production of Poincaré plots.-Nonharmonic motion can be difficult to discern in the usual time transient plot, as shown in part (a) of Figure 7. This is for the magnetically suspended system of [2] where the rotor drops onto a backup bearing after failure of the magnetic bearing. The backup bearing is assumed to have a friction coefficient of 0.5. The figure shows a station near the rotor midpoint. From this figure one cannot discern the period of motion or even if the motion is periodic. A Poincaré plot, which shows the position of the rotor once per revolution, enables more information to be obtained. Figure 7 (b) illustrates this for the same simulation as Figure 7 (a). One can now see that the motion is periodic, with a period equal to four revolutions.

## CONCLUDING REMARKS

The versatility of ARDS for personal computers has been increased by the enhancements reported herein. In addition to the basic rotordynamic calculations of natural frequencies, unbalance response, and transient motion due to blade loss, rubs, etc., for an interconnected system of up to five rotors, ARDS now has the capability to 1) estimate the peak response following blade loss without resorting to a full transient analysis; 2) calculate response sensitivity to input parameters; 3) formulate optimum rotor and damper designs to place critical speeds in desirable ranges or minimize bearing loads; 4) produce Poincaré plots so the presence of chaotic or other nonharmonic motion can be ascertained.

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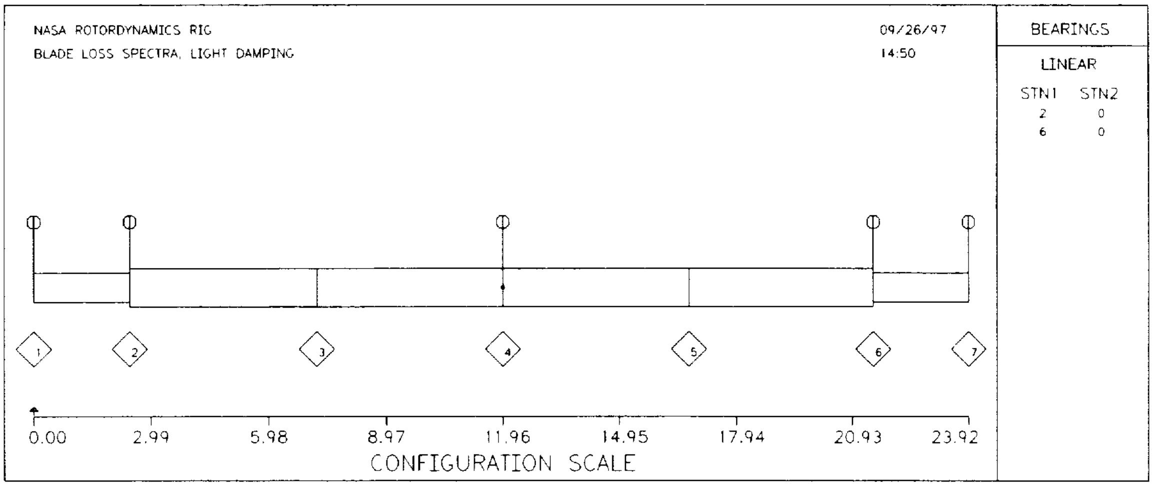



**Figure 1.—ARDS plot of NASA rotordynamics rig.**

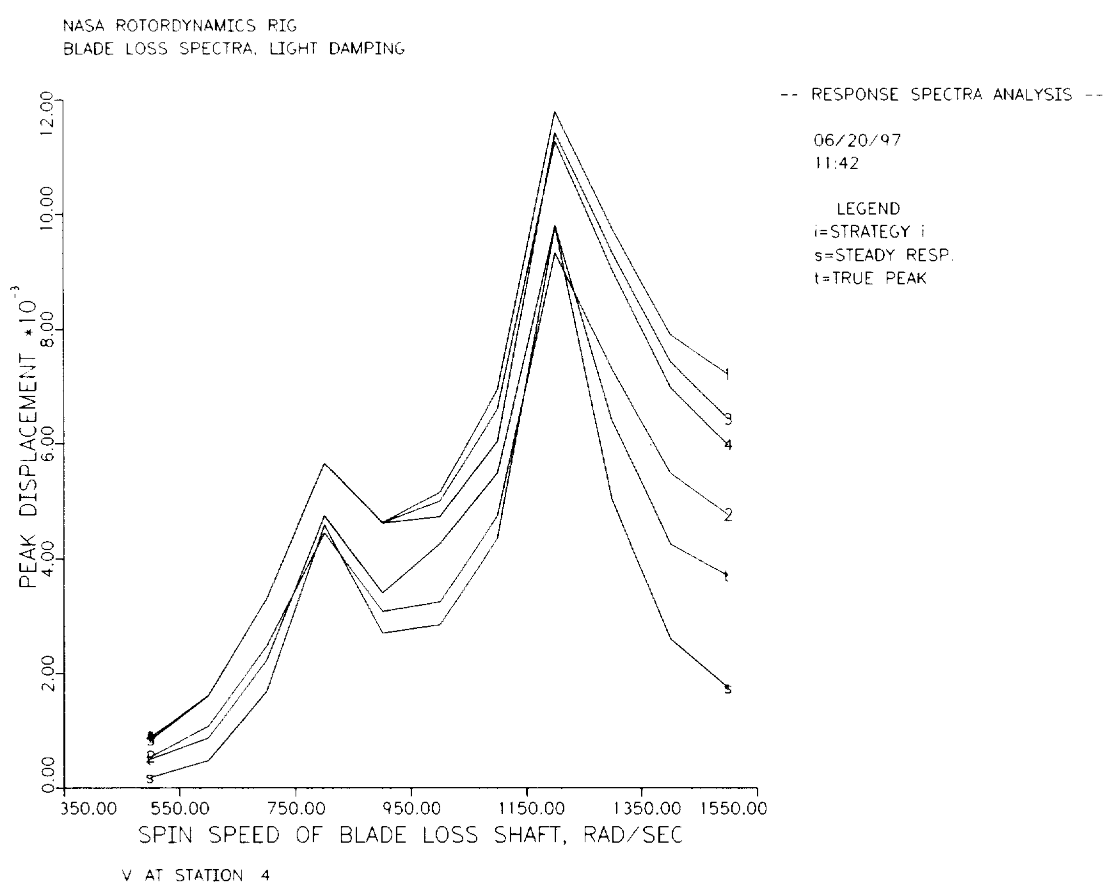



**Figure 2.—Peak rotor amplitude following blade loss.**

```
*************** DAMPED CRITICAL SPEEDS, SYNCHRONOUS WITH SHAFT  1   **************

 NO   ***************  SHAFT(1) SPIN SPEED  ***************        DAMPING COEFFICIENT
  1     7296.74    RPM  =   764.11    R/S  =     121.6     HZ               -33.9000
  2     7352.10    RPM  =   769.91    R/S  =     122.5     HZ               -36.4016
  3     9258.14    RPM  =   969.51    R/S  =     154.3     HZ               -88.1485
  4     9465.02    RPM  =   991.17    R/S  =     157.8     HZ               -92.4001

************************ DAMPED CRITICAL SPEED SENSITIVITY ANALYSIS **********************

                         * * * * * * * * * * * * * * * * * * * * * * *
                         *       Output values in the following        *
                         *     sensitivity analyses are associated     *
                         *     with changes of the magnitude shown     *
                         *          for each listed parameter          *
                         *                                             *
                         *  **  PARAMETER  **    **  CHANGE UNIT  **   *
                         *                                             *
                         *   RIGID DISC                                *
                         *      Mass                   0.40700E-02     *
                         *      Diametral inertia      0.35600E-02     *
                         *      Polar inertia          0.71200E-02     *
                         *      CG eccen               0.20000E-02     *
                         *      CG angle                10.000         *
                         *                                             *
                         *   BEARING                                   *
                         *      Stiffness(trans)        10000.         *
                         *      Stiffness(rot)          1.0000         *
                         *      Damping(trans)          2.0000         *
                         *      Damping(rot)            1.0000         *
                         *                                             *
                         *   SUBELEMENT                                *
                         *      Length                  1.0000         *
                         *      Unit mass              0.58180E-03     *
                         *      Bending stiffness      0.15000E+07     *
                         * * * * * * * * * * * * * * * * * * * * * * *

**** CRITICAL SPEED NO  1

SHAFT (1)  SPIN SPEED=   764.11    R/S

                  **********  RIGID DISC SENSITIVITY COEFFICIENTS **********

                  STN          MASS              DIAM.              POLAR
                   NO                            INERTIA            INERTIA
                    1        -7.4135          -0.75017            -1.5295
                    4        -125.84          -0.75851E-30        -0.13174E-29

              *****************  BEARING SENSITIVITY COEFFICIENTS *****************

              **********  STIFFNESS  **********      ***********  DAMPING  ***********

STN1 STN2       KVV      KVW      KWV      KWW        CVV      CVW       CWV      CWW
 NO   NO
  2    0       55.23    11.92   -11.92    55.23      1.447   -8.521     8.521    1.447
  6    0       55.23    11.92   -11.92    55.23      1.447   -8.521     8.521    1.447

               ***********  SHAFT ELEMENT SENSITIVITY COEFFICIENTS    ***********

              ELE  SUBEL        LENGTH             UNIT MASS              EI
               NO    NO
                1     1         5.1401             -5.6636            -0.33616E-01
                2     1        -38.217            -38.165              11.811
                3     1        -38.155            -76.394              68.243
```

Figure 3.—Critical speed sensitivity coefficients.

******************* STEADY SYSTEM RESPONSE DUE TO SHAFT 1 UNBALANCE *******************

SHAFT 1 SPIN SPEED = 6000.00 RPM = 628.319 R/S = 100.000 HZ

| STATION NO | SEMI-MAJOR AXIS | SEMI-MINOR AXIS | ATTITUDE ANGLE |
|---|---|---|---|
| 1 | 0.990014E-03 | 0.990014E-03 | -16.2020 |
| 4 | 0.181737E-02 | 0.181737E-02 | -15.6806 |
| 7 | 0.990014E-03 | 0.990014E-03 | -16.2020 |

******************** STEADY UNBALANCE RESPONSE SENSITIVITY ANALYSIS ********************

***** UNBALANCE RESPONSE SENSITIVITY COEFFICIENTS DUE TO RIGID DISC STATION NO 1 *****

*** C.G. ECCENTRICITY ***

| STATION NO | SEMI-MAJOR AXIS | SEMI-MINOR AXIS | ATTITUDE ANGLE |
|---|---|---|---|
| 1 | 0.841221E-03 | 0.841221E-03 | 4.49288 |
| 4 | 0.249459E-03 | 0.249459E-03 | -2.04940 |
| 7 | -0.101247E-03 | -0.101247E-03 | -0.862307 |

***** UNBALANCE RESPONSE SENSITIVITY COEFFICIENTS DUE TO RIGID DISC STATION NO 4 *****

*** C.G. ECCENTRICITY ***

| STATION NO | SEMI-MAJOR AXIS | SEMI-MINOR AXIS | ATTITUDE ANGLE |
|---|---|---|---|
| 1 | 0.250040E-03 | 0.250040E-03 | -3.63057 |
| 4 | 0.131846E-02 | 0.131846E-02 | 4.09880 |
| 7 | 0.250040E-03 | 0.250040E-03 | -3.63057 |

********** UNBALANCE RESPONSE SENSITIVITY COEFFICIENTS DUE TO BEARING SUPPORT **********

******* BEARING COEFFICIENTS BETWEEN STATIONS 2 AND 0 *******

| STATION | ******** STIFFNESS ******** | | ******* DAMPING ******* | |
|---|---|---|---|---|
| NO | AMPLITUDE | ATTITUDE ANGLE | AMPLITUDE | ATTITUDE ANGLE |
| 1 | -0.222391E-02 | 33.0850 | -0.718388E-04 | -16.1736 |
| 4 | -0.186304E-02 | 18.6166 | -0.742048E-04 | -7.38093 |
| 7 | 0.122457E-03 | 4.24847 | -0.922489E-05 | 0.890587 |

***** UNBALANCE RESPONSE SENSITIVITY DUE TO SHAFT SUBELEMENT BENDING STIFFNESS (EI) *****

ELE NO 1 SUBEL NO 1

| STATION NO | SEMI-MAJOR AXIS | SEMI-MINOR AXIS | ATTITUDE ANGLE |
|---|---|---|---|
| 1 | -0.242712E-03 | -0.242712E-03 | -1.72391 |
| 4 | -0.286551E-04 | -0.286551E-04 | 0.246839 |
| 7 | 0.791026E-05 | 0.791026E-05 | 0.527836E-01 |

ELE NO 2 SUBEL NO 1

| STATION NO | SEMI-MAJOR AXIS | SEMI-MINOR AXIS | ATTITUDE ANGLE |
|---|---|---|---|
| 1 | 0.381831E-05 | 0.381831E-05 | -0.310422 |
| 4 | -0.747739E-04 | -0.747739E-04 | 0.474326 |
| 7 | 0.152281E-05 | 0.152281E-05 | 0.268108 |

ELE NO 3 SUBEL NO 1

| STATION NO | SEMI-MAJOR AXIS | SEMI-MINOR AXIS | ATTITUDE ANGLE |
|---|---|---|---|
| 1 | 0.747387E-04 | 0.747387E-04 | 2.55804 |
| 4 | -0.771583E-03 | -0.771583E-03 | 1.38706 |
| 7 | 0.179561E-04 | 0.179561E-04 | 2.55872 |

**Figure 4.—Unbalance response sensitivity coefficients.**

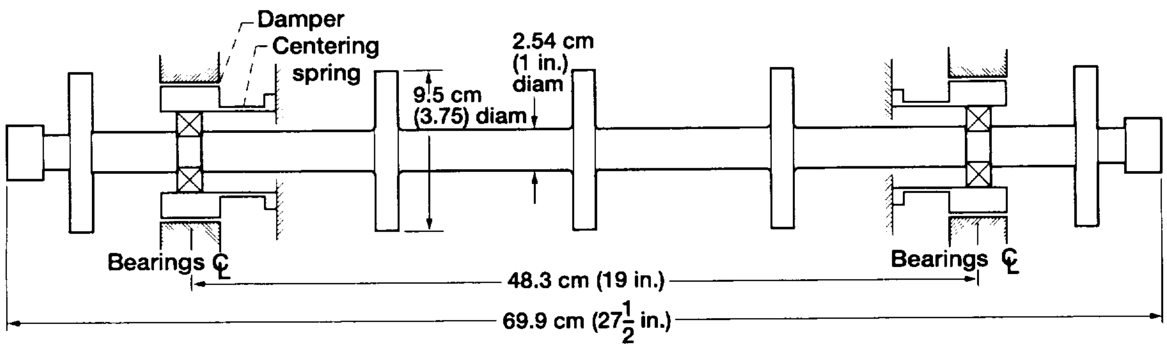



Figure 5.—NASA five-disc flexible rotor system.

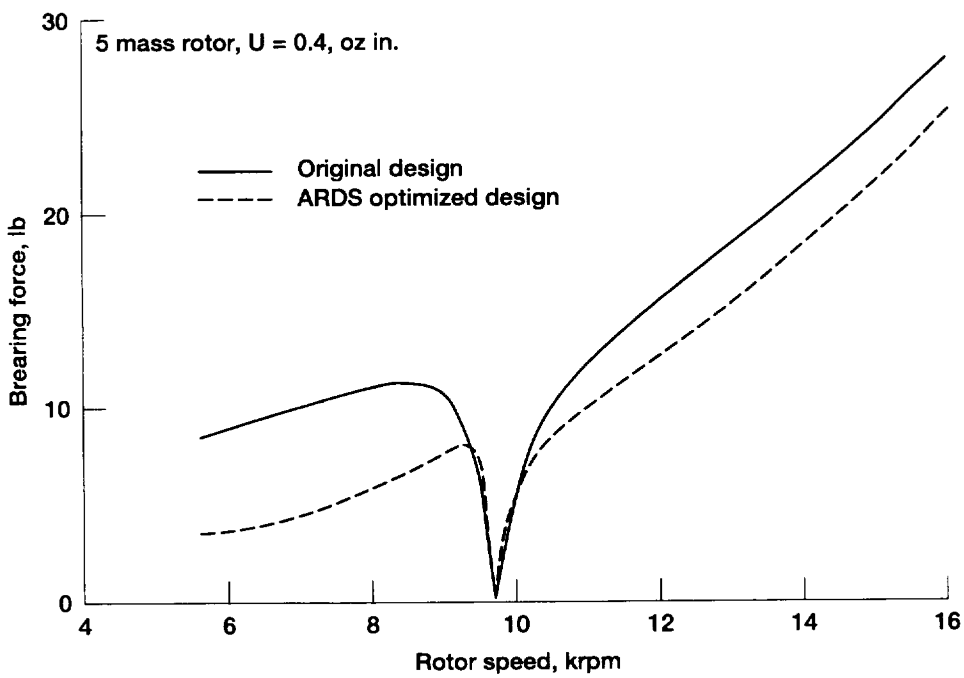
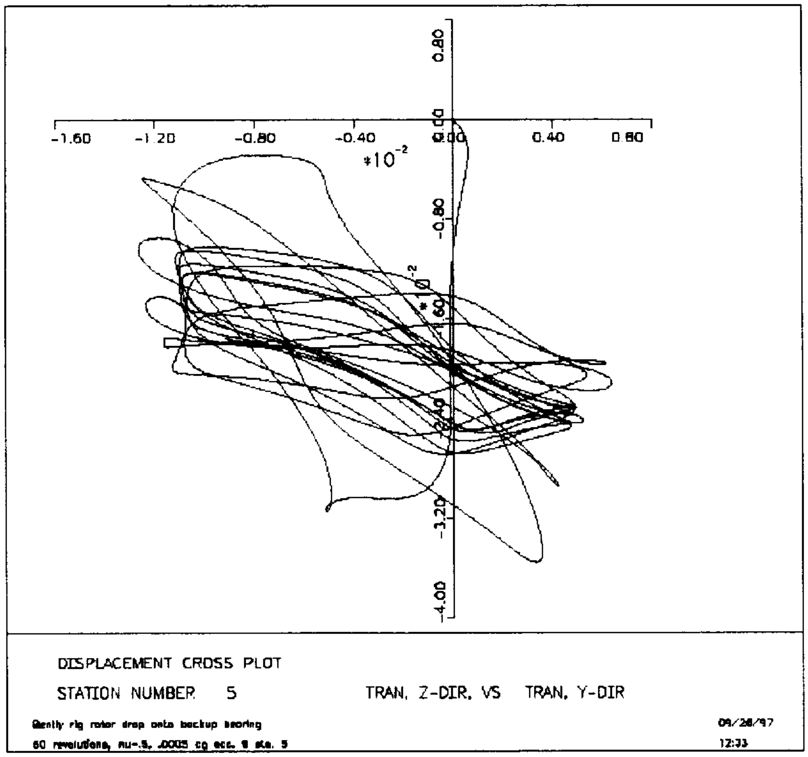
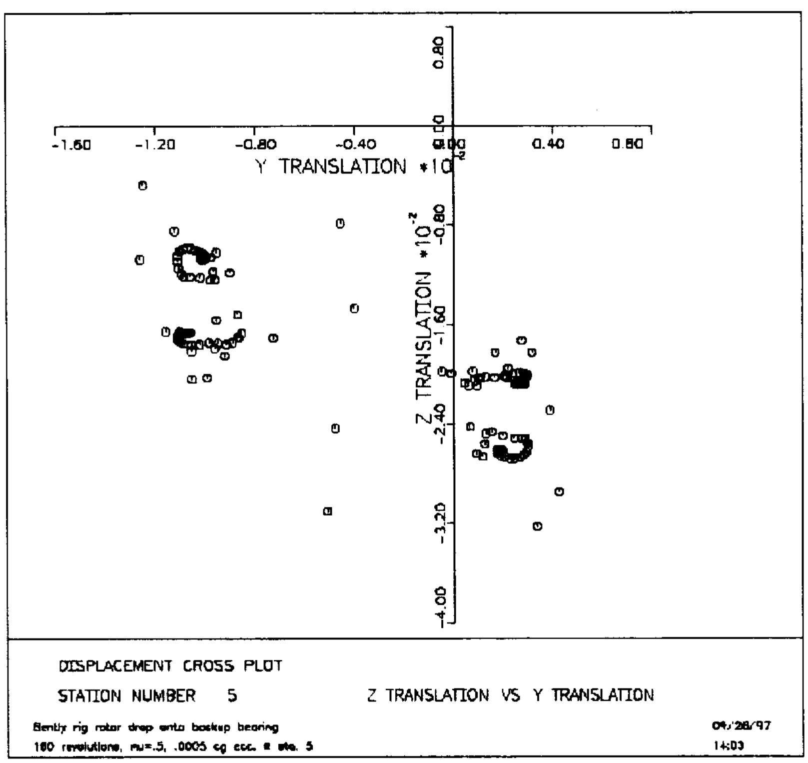



Figure 6.—Bearing force for squeeze film damper supported rotor.




(a) Orbit plot.




(b) Poincaré plot.

**Figure 7.—Rotor drop onto backup bearing.**

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