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CAE Approach to the Development of Rotary Compressors

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CAE APPROACH TO THE DEVELOPMENT OF ROLLING PISTON TYPE ROTARY COMPRESSORS

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ABSTRACT

This paper describes the computer aided engineering (CAE) approach to investigate the vibration of rolling piston type rotary compressors in the design and analysis stages. At first, the optimum design procedure for the compressor and piping system is presented. Then ,a compressor data base used for a dynamic simulation of the royary compressors are carried out. Finally, a practical application of the CAE approach to the optimum design of the rotary compressor is considered.

INTRODUCTION

Rolling piston type hermetic compressors have been used widely for room air-conditioners and unitary air-conditioners, because of high efficiency and reliability ,small size ,and light weight. However ,exciting forces generated by the rotary compressors are sometimes comparable to the conventional reciprocating compressors. Because,the motor and the components are directly fixed to the hermetic housing in order to reduce the size and improve the productivity. This means that an unbalanced inertia force around the crankshaft directly transmits to the hermetic housing which is suspended with three coiled springs. Therefore,it is important task to design the piping system connected to the rotary compressor in view of low vibration.

A general concept of the CAE approach to the piping system in outdoor units has been established at DAIKIN and has already been reported.[1] At our rotary compressor division we utilize the CAE approach to design the rotary compressors with reduced vibration. In this paper we introduce the CAE approach to the reduction of the excitation forces generated by the rotary compressors in operation.

OPTIMUM DESIGN PROCEDURE FOR THE COMPRESSOR AND PIPING SYSTEM

The optimum design procedure for the compressor and piping system is shown in Fig.1. Following are the key tasks in this procedure.

(1) Solid modeling technique is employed to obtain precise geometries and mass properties of compressors, because it is a difficult task to measure the mass properties experimentally.

(2) Kinematic simulations of compressors are carried out, in which excitation forces generated by the compressors are calculated based upon the unbalanced mass and the fluctuation of compression load.

The results of previous tasks (geometries,mass properties,and excitation forces) are stored in a compressor data base.

(3) Three dimensional piping design is performed using a 3-D CAD system.

(4) Piping geometries are converted into Finite Element model.

(5) Eigenvalue analysis is carried out with the piping system model which comprises the beam elements and the rigid elements representing compressor and accumulator by Building Block Approach.

(6) Calculated results are compared with modal testing data in the form of transfer function.

(7) Dynamic responses on the piping system are calculated and local stress distribution is derived from the displacement boundary condition.

The alternative design cycles will be repeated until the optimum design is obtained which satisfies the design criterion.

Fig.2 shows a cross section of a rolling piston type rotary compressor in which the motor and the compressor components are each directly fixed to the upper or lower portion of the hermetic housing.

Design process for a rotary compressor starts with performance evaluation of the compressor,which is followed by dynamic simulation of rolling piston and discharge valve,and also lubrication analysis. In the view of vibration,the prediction of the the excitation forces is also a key factor in designing rotary compressors,because dynamic responses of piping system depend upon the excitation forces. Therefore,before the detailed design of rotary compressors,the prediction of the excitation forces must be carried out.

THE PREDICTION OF EXCITATION FORCES

Solid Modeling Of Compressors

The gravity center and the moment of inertia (mass properties) are necessary parameters in calculating the excitation forces of the rotary compressors. However,it is a difficult task to measure the mass properties experimentally,because the geometric shapes of the compressor components are very complicated.

Therefore,solid modeling technique is employed to obtain precise geometries and mass properties of compressors. In Fig.3 the computer model of 2 HP rotary compressor is shown. Fig.4 shows the computer models of several components which are comprised in the compressor. And Table 1 shows the mass properties.

Excitation Force Analysis

(1) Momentum Equation

The torque balance around the center of rotational axis can be written as the following equation of motion.[2]

$$I_r \cdot \ddot{\theta} = T_m - T_g - T_v \tag{1}$$

where, I_r = moment of inertia of rotating components
θ = crank angle
T_m = motor torque

T_g = loading torque due to the cylinder pressure
T_V = loading torque due to the blade force

In the momentum equation mentioned above, the loading torque due to the mechanical friction is neglected for the simplicity.

Using the mass properties mentioned previously, the moment of inertia around the rotational axis, I_r, is obtained by the following equation.

$$I_r = I_{MR} + I_{CR} + m_{RO} \cdot e^2 \tag{2}$$

where, I_{MR} = moment of motor-rotor inertia
I_{CR} = moment of crankshaft inertia
m_{RO} = mass of roller
e = eccentricity of crankshaft

(2) Calculation Of Excitation Forces

On the coordinate system shown in Fig.5, each component of the force and the moment acting at the cylinder center can be written as the combination of unbalanced inertial forces.

$$F_X = m_{BU} \cdot \ddot{x}_{BU} + m_{RO} \cdot \ddot{x}_{RO} + m_{CR} \cdot \ddot{x}_{CR} + m_{BL} \cdot \ddot{x}_{BL} \tag{3}$$
$$F_Y = -m_V \cdot \ddot{y}_V + m_{BU} \cdot \ddot{y}_{BU} + m_{RO} \cdot \ddot{y}_{RO} + m_{CR} \cdot \ddot{y}_{CR} + m_{BL} \cdot \ddot{y}_{BL} \tag{4}$$
$$F_Z = 0 \tag{5}$$
$$M_X = m_{BU} \cdot \ddot{y}_{BU} \cdot l_{BU} + m_{BL} \cdot \ddot{y}_{BL} \cdot l_{BL} \tag{6}$$
$$M_Y = m_{BU} \cdot \ddot{x}_{BU} \cdot l_{BU} + m_{BL} \cdot \ddot{x}_{BL} \cdot l_{BL} \tag{7}$$
$$M_Z = -I_r \cdot \ddot{\theta} \tag{8}$$

where, m_{BU} = mass of upper balance weight
m_{BL} = mass of lower balance weight
m_{CR} = eccentric mass of crankshaft
l_{BU} = gravity center of upper balance weight
l_{BL} = gravity center of lower balance weight

Therefore, excitation forces around the gravity center of the compressor, the location of which is (X_G, Y_G, Z_G) in the cylinder coordinate system, can be written as follows.

$$F_{GX} = F_X \tag{9}$$
$$F_{GY} = F_Y \tag{10}$$
$$M_{GX} = M_X + Z_G \cdot F_{GY} \tag{11}$$
$$M_{GY} = M_Y - Z_G \cdot F_{GX} \tag{12}$$
$$M_{GZ} = M_Z - X_G \cdot F_{GY} + Y_G \cdot F_{GX} \tag{13}$$

(3) Results Of Analysis

Fig.6 shows the cylinder pressure fluctuation used for the analysis. Fig.7 shows the loading torque fluctuations in steady state, in which the loading torque due to the cylinder pressure is dominant. Fig.8 shows the angular velocity and acceleration around the rotational axis which calculated from equation (1).

The excitation forces are calculated from the set of equations (3)～(13) using the mass properties as well as angular velocity and acceleration obtained above. The excitation forces in both X and Y axes are shown in Fig.9,and the excitation moments around X,Y,and Z axes are shown in Fig.10. Fig.11 and Fig.12 show their Fourier transformed data. It can be seen from Fig.10 that the excitation moment around Z axis is the largest component. Table 2 shows the maximum values of the excitation force in each direction.

APPLICATION TO THE OPTIMUM DESIGN OF THE ROTARY COMPRESSOR

Optimum Design Of Balance Weight

The unbalanced forces other than the inertia force of the blade can be canceled by the optimum design of upper and lower balance weights. The comparison between current design and the optimum design is shown in Table 3 ,in which the balance weight specifications and the resultant excitation forces are tabulated.

Table 3 Comparison between Current Design and Optimum Design of Balance Weights

		Current Design	Optimum Design
Upper Balance Weight Thickness	(mm)	5.5	4.5
Upper Balance Weight Mass	(Kg)	0.0527	0.0431
Lower Balance Weight Thickness	(mm)	16.0	11.717
Lower Balance Weight Mass	(Kg)	0.1730	0.1267
FGX (max)	(N)	100.079	8.167
FGY (max)	(N)	71.823	-16.089
MGX (max)	(N·m)	4.344	-1.286
MGY (max)	(N·m)	5.897	0.580
MGZ (max)	(N·m)	8.208	7.678

In Fig.13 the fluctuations of the excitation forces are shown. It can be seen that all the excitation forces are reduced by the optimum design of the balance weights.

Influence Of Gravity Center Change

It is difficult to conform the gravity center of the rotary compressor to the cylinder center,because the motor is the largest mass in all the compressor components and is located in the upper portion of the compressor. Therefore,it is interesting problem to investigate how the excitation forces are influenced by changing the gravity center.

Fig.14 shows the comparison between current design and modified ones,in which the gravity center is changed. It can be seen that the excitation moments around the both X and Y axes are reduced by conforming the gravity center to the cylinder center. Therefore, it is desirable to locate the gravity center as nearly as possible to the cylinder center.

Influence Of Blade Inertia Force

The blade is the only reciprocating element in the rotary compressor components. It can be seen in equation (4) that the excitation force in Y axis direction is influenced by the inertia force of the blade. We investigated the influence by changing the blade mass.

In Fig.15 the comparison between current design and modified ones, in which the blade mass is changed,is shown. The excitation force in Y axis direction can be reduced by increasing the blade mass.

CONCLUSIONS

The CAE approach to the development of rolling piston type rotary compressors was introduced focusing on the investigation of the vibration characteristics,which is summarized as follows.

(1) The loading torque due to the cylinder pressure is most dominant within the fluctuation of torques which generates the excitation forces of rotary compressor,and the excitation moment around the rotational axis is the largest force component.

(2) It was confirmed that the excitation forces could be reduced by the optimum design of each component.

(3) The results of this analysis are effectively used for the optimum design of the piping system in air conditioning units as a compressor data base.

The verification of the accuracy of the results remain as a future work.

ACKNOWLEDGEMENT

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[1] T.Hirano,"CAE Approach to the Research and Development of Air Conditioning Units,"10th International Seminar on Modal Analysis,K.U. Leuven,Belgium,1985.

[2] T.Yanagisawa et al.,"Study on Vibration of Rolling Piston Type Rotary Compressor,"Transaction of the Japan Society of Mechanical Engineers (Series C),Vol.49.NO.444,1984.

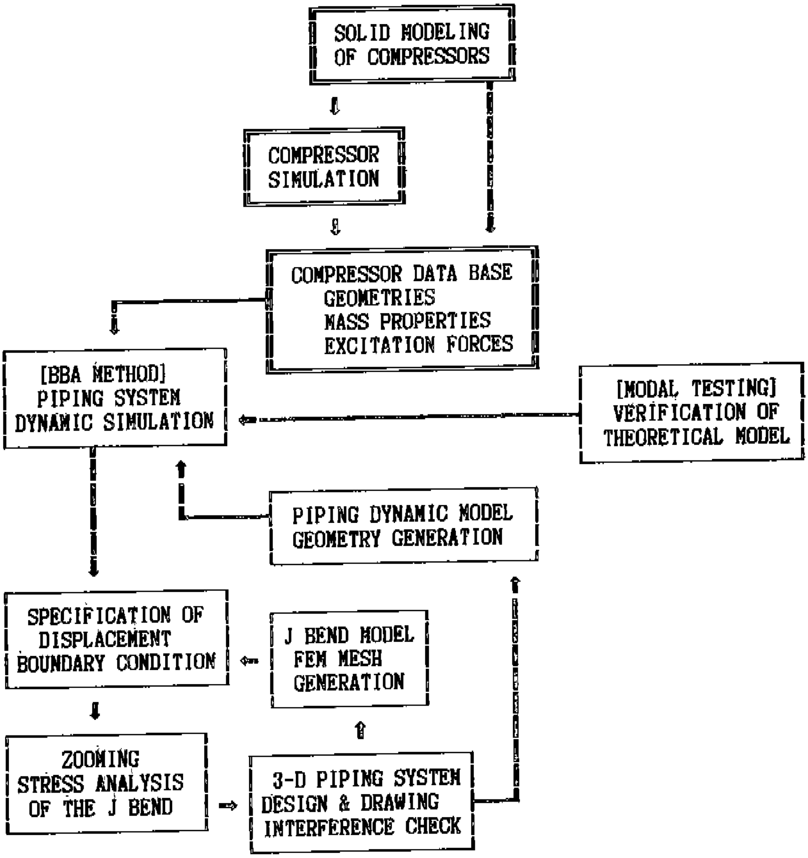


Fig.1 Optimum Design Procedure for the Compressor and Piping System

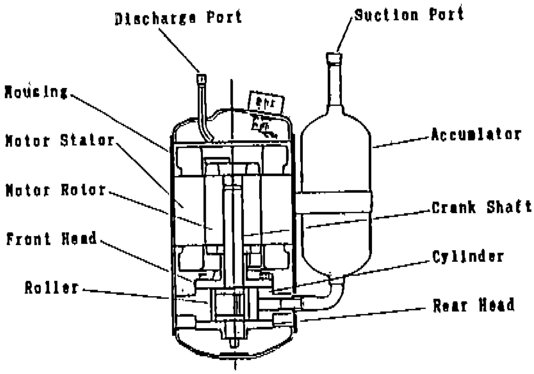


Fig.2 Cross Section of Rolling Piston Type Rotary Compressor

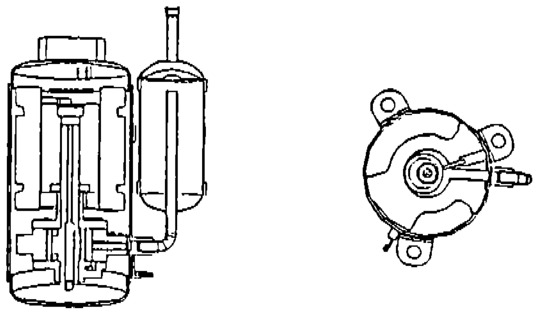


Fig.3 Computer Model of 2 HP Rotary Compressor

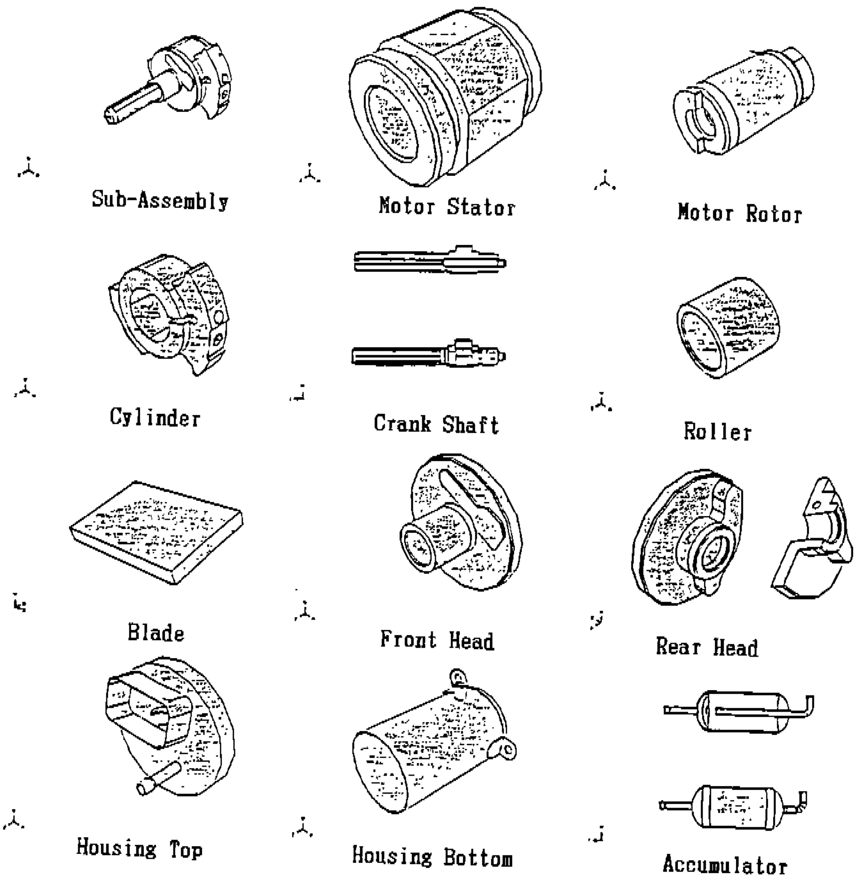


Fig.4 Computer Models of Main Components

Table 1 Mass Properties of Compressor Components

COMPONENT	GRAVITY CENTER (mm)			MOMENT OF INERTIA (N·m)		
	X	Y	Z	Ixx	Iyy	Izz
HOUSING TOP	1.306	11.719	281.462	14.634	13.080	24.441
HOUSING BOTTOM	1.257	0.449	112.107	268.983	268.498	147.134
MOTOR STATOR	0.000	0.000	183.067	200.709	198.273	187.004
MOTOR ROTOR	0.420	0.000	182.624	25.790	25.788	12.050
FRONT HEAD	0.639	0.836	99.821	2.249	2.247	3.709
REAR HEAD	0.593	-1.978	45.312	2.351	2.398	4.391
CRANK SHAFT	-1.122	0.010	113.273	14.558	14.398	0.494
CYLINDER	2.388	-3.824	71.984	14.484	22.040	33.481
ROLLER	-4.000	0.000	71.975	0.550	0.550	0.672
BLADE	30.126	10.916	72.000	0.039	0.052	0.018
ACCUMULATOR	118.288	0.000	204.356	31.969	32.029	9.354
2HP ROTARY	6.754	0.267	151.494	1195.189	1317.142	569.344

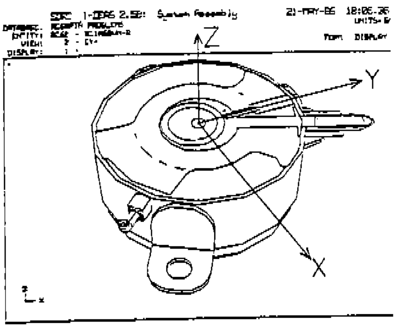


Fig.5 Coordinate System used for Analysis

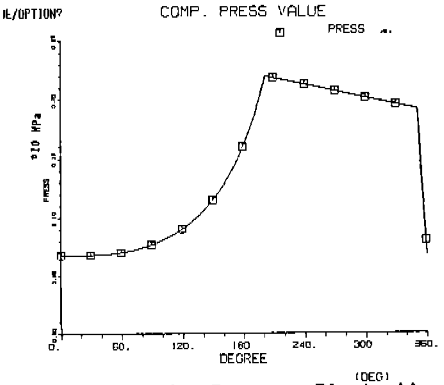


Fig.6 Cylinder Pressure Fluctuation used for Analysis

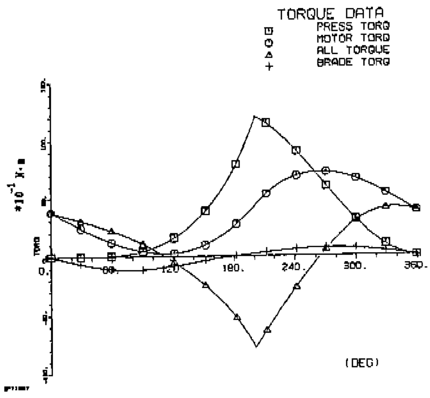


Fig.7 Torque Fluctuation in Steady State

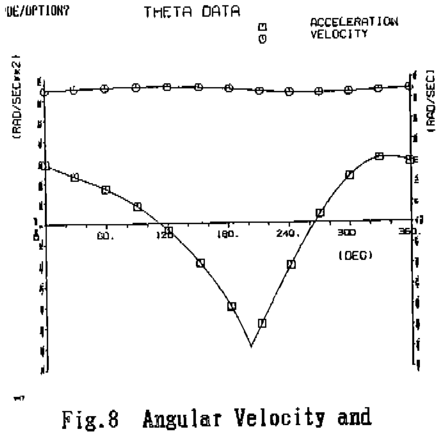


Fig.8 Angular Velocity and Acceleration

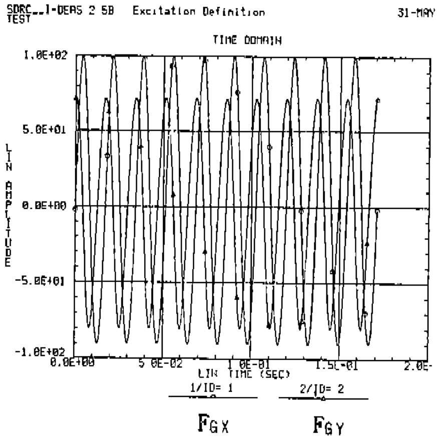


Fig.9 Excitation Forces in Both X and Y Axes

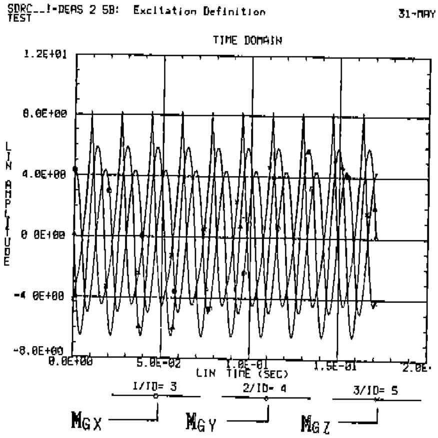


Fig.10 Excitation Moments around X,Y,and Z Axes

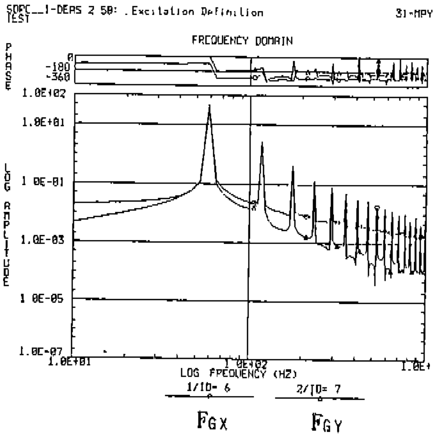


Fig.11 Fourier Transformed Data in Both X and Y Axes

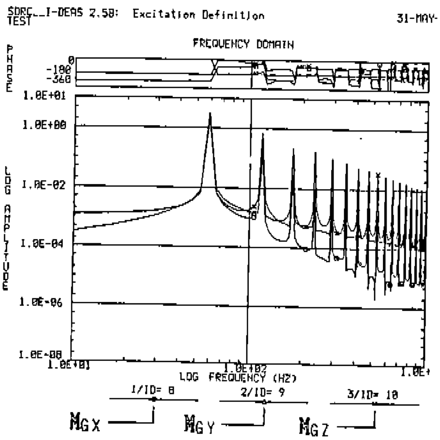
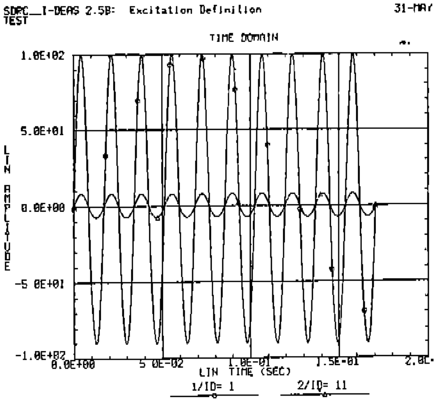


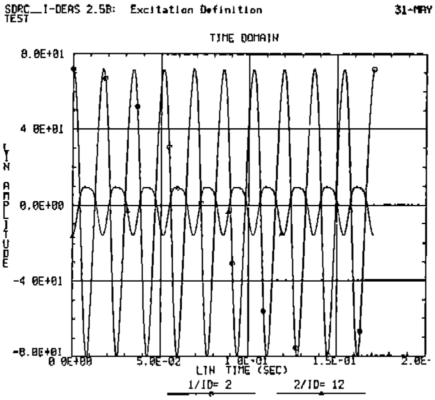
Fig.12 Fourier Transformed Data around X,Y,and Z Axes

Table 2 Maximum Values of Excitation Forces (Current Design)

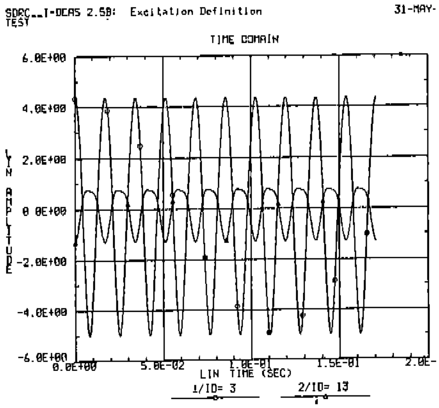
	Maximum Value
FGX (N)	100.079
FGY (N)	71.823
MGX (N·m)	4.344
MGY (N·m)	5.897
MGZ (N·m)	8.208



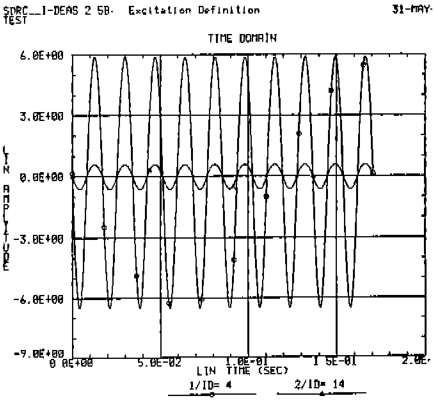
(a) Excitation Force in X Axis



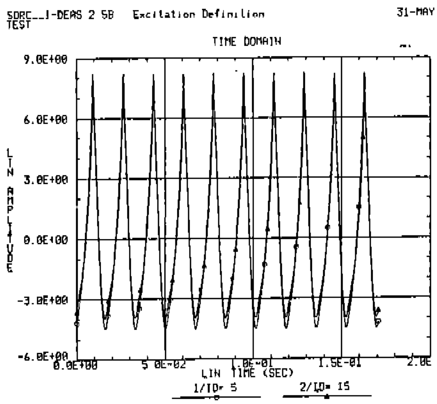
(b) Excitation Force in Y Axis



(c) Excitation Moment around X Axis



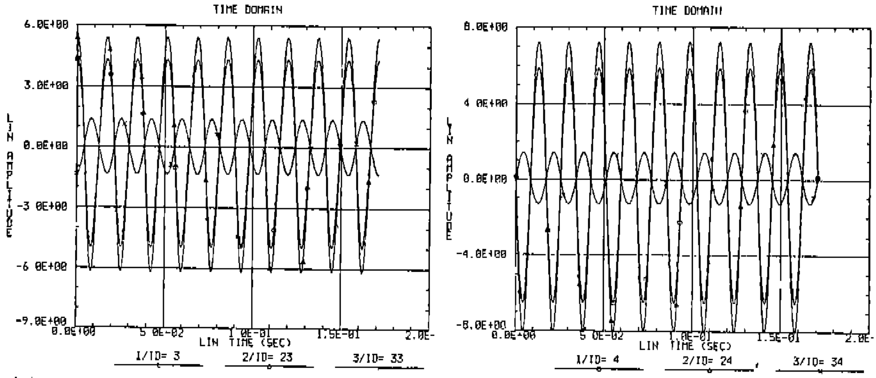
(d) Excitation Moment around Y Axis



(e) Excitation Moment around Z Axis

Fig.13 Comparison between Current Design and Optimum Design of Balance Weights

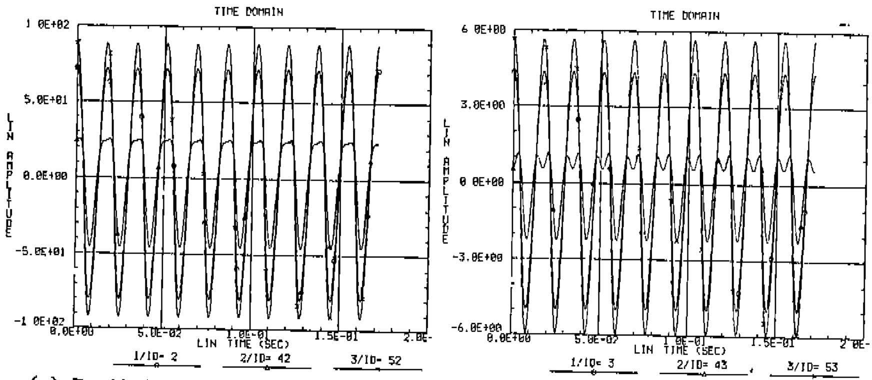
○: Current Design
△: Optimum Design



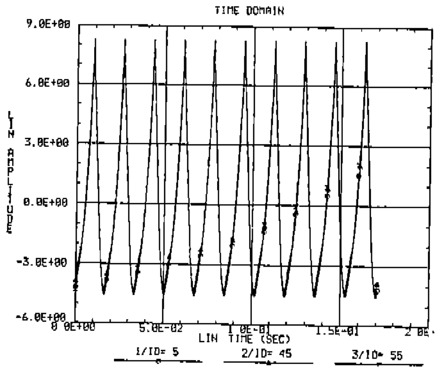
(a) Excitation Moment around X Axis (b) Excitation Moment around Y Axis

○ :Current Design, △:15 mm Upper than Current Design, ×:Cylinder Center

Fig.14 Comparison of Gravity Center Change



(a) Excitation Force in Y Axis (b) Excitation Moment around X Axis



(c) Excitation Moment around Z Axis

Fig.15 Comparison of Blade Mass Chage

○:Current Design
△:Heavy Blade Mass
×:Light Blade Mass