INTGRATED DESIGN METHOD OF A ROCKET ENGINE TURBOPUMP SUB-SYSTEM FOR SUPPRESSING ROTOR LATERAL VIBRATION

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Abstract

A rocket turbopump is the high-speed and high-power turbomachinery that raise the pressure of cryogenic fluid to higher than the combustion chamber pressure. Various rotor vibration problems had occurred caused by rotation in the development phase of the turbopumps for the Japanese rocket engines. In order to suppress the rotor instability vibrations, rotordynamic fluid forces induced by the turbomachinery components should be taken account in the rotordynamics analysis. The design method is a kind of multi-disciplinary optimum design and is formed in two steps.

This paper presents a new the integrated design method concept developed to provide good stability for rotor lateral vibration while rotating at all operation range.

1 Introduction

A rocket turbopump is often described the heart of an engine as an indication of its importance. As a human heart pumps blood through the body, a turbopump feeds high-pressure propellant to the combustion chamber. The rocket turbopump is high-speed, high-power turbomachinery. In the development phase of the turbopumps for Japanese rocket engines, various rotor vibration problems caused by the increase in energy density have occurred [1-3]. Such problems are the result of the interaction between excitation and damping, and remain unresolved without reference to dynamics, e.g., the frequency characteristic, the axial mode shape and so on. It is generally considered that the rotor vibration characteristic depends on the bearing characteristics sustaining the rotor, which is especially dominant in the case of stiffness of ball bearing. However, in order to meet the demands of the high efficiency and performance for the rocket turbopump, the clearance between the rotating part and stationary part is very narrow. Therefore, it is required consideration of how the small clearance area will affect the rotor.

Rotordynamic fluid forces (hereafter termed “RD fluid forces”) with a frequency dependence, which are generated in wetted or gas surface areas (i.e., such as an impeller, an inducer, a shaft seal, a balance piston), have a large effect on rotor vibration. Especially, tangential fluid force which foster a destabilizing action, may lead to self-excited vibration (destabilizing vibration).

Establishment of an integrated design method consisting of three turbopump sub-systems, i.e., a pump system, a turbine system and a rotor system, is expected to suppress rotor vibration. The design method consists of “multidisciplinary optimum design” and “design of morphology” considering RD fluid forces and rotordynamics. This paper presents a concept of the new integrated design method which is developed to provide stability for rotor lateral vibration in the whole operational range.

2 Dynamics Design for Turbopumps

2.1 Rotordynamic Fluid Forces and Rotordynamics

A cross section of a hydrogen turbopump for a LE-7 engine is shown in Fig. 1. The sources of RD fluid force acting on the rotor exist at locations where the inducer, the impeller, the seals, the bearings and the turbine have small clearance area filled with fluid. Typical fluid forces which should be considered in rotordynamics analysis are as follows.

1. Mechanical excitation force caused by residual mass unbalance or misalignment.
2. Fluid-dynamic unbalance force.
3. Bearing reaction force.
4. Internal damping into rotors (negative damping at over-critical speed).
5. Destabilizing force caused by shaft seals.
6. RD fluid force due to impellers.
7. Fluid force caused by cavitation and rotating stall.
8. Unbalance torque caused by axial turbines.

Typical RD fluid forces acting on the rotor for a rocket turbopump are shown in Fig. 2. M is additional mass matrix, K is stiffness matrix, C is damping matrix and f represents RD fluid force. Each matrix is used for the linearly expressed RD fluid forces. The matrix of each element consists of direct action and cross-coupled action by a diagonal term and a non-diagonal term respectively. Most of the rotorvibrations induced by the action of the fluid forces is self-excited vibration. So the state of the vibration is decided by the stability of the rotor. This vibration occurs at each natural frequency of fluid phenomena [4] and at a wider range of rotational speed than the resonance frequency. RD fluid forces are decomposed in

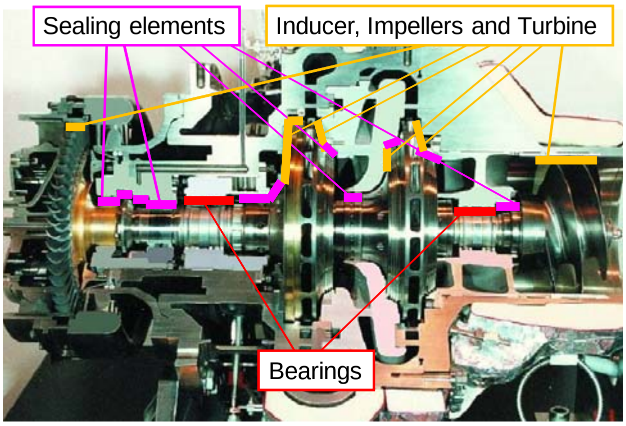


Fig. 1 LE-7 Fuel turbopump

the tangential direction (F_t) and the radial direction (F_n) as shown in Fig. 3. The destabilizing action is strongly affected by the tangential fluid force with the same whirling direction as the rotor spin direction, and is generated at each small clearance area in the turbopump. It has a negative effect on the damping ratio of the rotor system. The damping ratio is the real part of the complex eigenvalue given by the rotordynamics analysis. Occurrence of the self-excited vibration is decided by the damping ratio of the rotor system (i.e. in case of negative for the real part, the system is stable). Therefore, it is necessary to individually obtain the RD fluid forces of major components and the elements which are sensitive for swirl. When the rotordynamics analysis is carried out

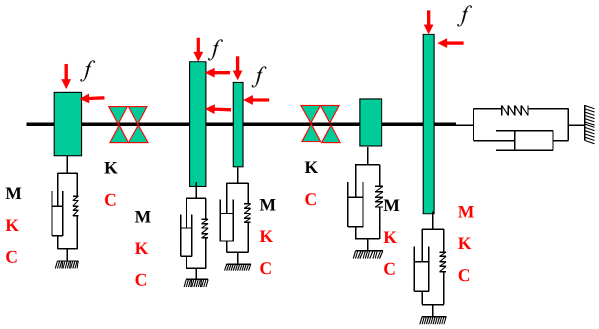
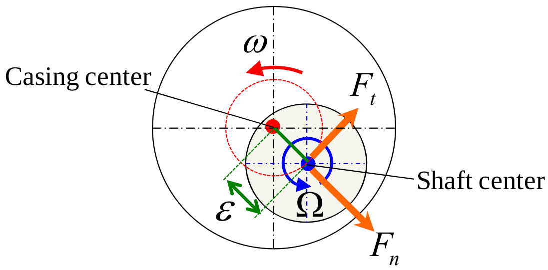


Fig. 2 Rotordynamic fluid forces acting on a rotor.



Rotordynamic force effect at $\omega/\Omega > 0$

$F_n > 0$: inertia effect
$F_n < 0$: restoring effect

$F_t > 0$: destabilizing effect
$F_t < 0$: damping effect

Fig. 3 Schematic of the RD fluid forces acting on an impeller whirling in a circular orbit.

by using the finite element modeling, the rotor behavior can be expressed by the equation of motion as follows.

$$\sum_{j=1}^{n}\left[m_{ij}\ddot{x}_j(t) + c_{ij}\dot{x}_j(t) + k_{ij}x_j(t)\right] = f_i(t) \quad (i = 1, 2, \cdots, n) \tag{1}$$

where m_{ij} is the mass coefficient, c_{ij} is the damping coefficient, k_{ij} is the stiffness coefficient between the nodes i and j, $f_i(t)$ is the excitation force. And $\ddot{x}_i(t)$, $\dot{x}_i(t)$, $x_i(t)$, n are the displacement, velocity, acceleration for "i" and the number of node respectively. These are expressed by using matrices as follows:

$$[m_{ij}] = M, \quad [c_{ij}] = C, \quad [k_{ij}] = K \tag{2}$$

If the displacement depends on linear vibration system, the following equalities are true.

$$[M] = [M]^T, \ [C] = [C]^T, \ [K] = [K]^T \tag{3}$$

where exponent T means transposed matrix. By using the relational expressions (4), the equation of motion (1) is easily stated as equation (5):

$$\sum_{j=1}^{n} x_j(t) = \{x(t)\}, \ \sum_{j=1}^{n} f_j(t) = \{F(t)\} \tag{4}$$

$$[M]\{x(t)\}+[C]\{x(t)\}+[K]\{x(t)\} = \{F(t)\} \tag{5}$$

On the other hand, RD fluid forces which act on the mechanical element of the turbopump are expressed by the linearized rotordynamics coefficient (i.e., m_{xx} and m_{xy} are additional mass coefficients, c_{xx} and c_{xy} are the damping coefficients, and k_{xx} and k_{xy} are stiffness coefficients). Also, RD fluid forces in the rotordynamic analysis are entered as matrices into the node corresponding to the locations acting on the RD fluid forces. In case of the rotor is whirling at the condition of a circular whirl motion of eccentricity ε and whirl angular velocity ω, so that displacement $x(t) = \varepsilon\cos\omega t$ and $y(t) = \varepsilon\sin\omega t$, it is conventional to decompose the RD fluid forces into additional mass, damping and stiffness matrices according to:

$$\begin{bmatrix} F_n & -F_t \\ F_t & F_n \end{bmatrix}\begin{Bmatrix} x/R \\ y/R \end{Bmatrix} = -\begin{bmatrix} m_{xx} & m_{xy} \\ -m_{xy} & m_{xx} \end{bmatrix}\begin{Bmatrix} \ddot{x}/R\Omega^2 \\ \ddot{y}/R\Omega^2 \end{Bmatrix} - \begin{bmatrix} c_{xx} & c_{xy} \\ -c_{xy} & c_{xx} \end{bmatrix}\begin{Bmatrix} \dot{x}/R\Omega \\ \dot{y}/R\Omega \end{Bmatrix} - \begin{bmatrix} k_{xx} & k_{xy} \\ -k_{xy} & k_{xx} \end{bmatrix}\begin{Bmatrix} x/R \\ y/R \end{Bmatrix} \tag{6}$$

Equation (6) means the RD fluid forces can be expressed as quadratic function of whirl ratio "ω/Ω". Thus the non-dimensional RD fluid forces are written by:

$$f_n = m_{xx}(\omega/\Omega)^2 - c_{xy}(\omega/\Omega) - k_{xx} \tag{7}$$

$$f_t = -m_{yy}(\omega/\Omega)^2 - c_{xx}(\omega/\Omega) + k_{xy} \tag{8}$$

According to the expressions (7) and (8), the RD fluid forces are dealt with as six rotordynamics coefficients for the rotordynamics analysis, therefore the effect of the RD fluid forces on the rotor vibration can be estimated. As an example, well-known rotordynamic expressions of unbalanced torque force (Thomas force) acting on a rocket turbine is described below.

Thomas force is explained by the unbalanced torque which results from the different leakage at the clearance between the tip of the turbine and the casing. This force acts as the destabilizing action which decreases the damping ratio of the rotor system. Thomas model is used for turbine-rotordynamic-coefficient calculation as follows.

$$\begin{aligned} F_t &= \sum \frac{T_0}{R}\left(1+\frac{T_1-T_0}{T_0}\cos\theta\right)\cos\theta \\ &= \int_0^{2\pi}\frac{1}{2\pi}\frac{T_0}{R}\left(1+\frac{T_1-T_0}{T_0}\cos\theta\right)\cos\theta d\theta \\ &= \frac{T_{total}}{2R}\frac{T_1-T_0}{T_0} \end{aligned} \tag{9}$$

where R is the turbine radius, T_0 is torque at the mean clearance, T_1 is torque at the maximum clearance, T_{total} is the entire torque of the turbine, and $\theta = \omega t$. Furthermore the cross-coupled stiffness coefficient definition is:

$$K_{xy} = T\beta \diagup D_m L \tag{10}$$

Where β is defined as "the change in thermodynamic efficiency per unit of rotor displacement", D_m is the mean blade diameter, L is the turbine blade height, T is the torque. As explained above, Thomas force is expressed as the cross-coupled stiffness, therefore only the destabilizing action is generated.

2.2 Multidisciplinary Optimal Design and Design of Morphology [5]

For realizing a next generation flagship launch vehicle, a manned launch vehicle and a reusable space transportation system, higher reliability, higher robustness, higher energy density, and longer operating life for the rocket engines will be more and more required. In order to figure out appropriate solutions for a wide variety of requirements from the rocket engine system, largeness and depth of the design flexibility and expandability of the turbopumps is very important. Although the shortage of performance and efficiency of the turbopump decreases a rocket engine power and performance, it doesn't result in the catastrophic failure and functional damage. However, rotor vibration of the turbopump may cause the functional failures of the rocket engine and eventually mission failure. Therefore, it is not too much to say that one factor which threatens the reliability of rocket engines and launchers is rotor vibration in the turbopump. Thus, the design flexibility of the rotordynamics in the turbopump has a possibility of becoming a primary constraint condition during the development of a future rocket engine. Now the inverse design method and the multi-objective optimization are applicable as a useful tool for designing the blade profiles, establishing "design of morphology" optimization technique of the rotor system is eagerly expected in the purpose of suppressing rotor vibration.

We JAXA are now studying the above-mentioned optimization method. It is the design method of optimal rotor system by using multidisciplinary optimization due to integration of the subsystems of the turbopump (e.g. pump, turbine and rotor). Herein, the each subsystem is discussed as a discipline. Because the rotor vibration is the result of the interaction between excitation and damping, knowledge and insight obtained by the "Dynamics" point of view are essential. Therefore, this design method is approached by "Dynamics", and its key words are "Dynamics design", "Multidisciplinary optimal design" and "Design of morphology".

(1) The dynamics design
This design takes the motion and the response (e.g., the unsteadiness, the disturbance, the mutual interference and the stability) into consideration, that is, the phenomena of time and frequency domains are focused. In contrast, "static design" is the method of focusing performance, efficiency, constant and non-interactive phenomena.

(2) Multidisciplinary optimal design
The optimal solution for the rotor system is pursued to stabilize the rotor at the upper level hierarchy. The dynamics characteristics, e.g., RD coefficient, from each rotational element at the lower level are input into the upper level hierarchy. The subsystems are designed and also optimized to increase stability at a discrete area.

(3) Design of morphology
"Morphology design" method does not pursue each element shape such as the detailed blade profile, but determines the location and/or the layout and/or permutation of the subsystems as the system form first.

The selection of the rotor form (layout, permutation of the rotating elements) is located as the upper level hierarchy in the design optimization problem. The optimization of each the pump subsystem, the turbine subsystem and the rotor subsystem which includes seals and bearings are located as the lower level hierarchy. While each subsystem is optimized according to some independent objective functions in the lower level hierarchy, matching between the upper and the lower level hierarchies must be adjusted. In parallel, the rotor form is optimized in the upper level hierarchy. So, changing from the conventional design method, in which the fluid performance is the first priority, to the improved design method, in which the rotor stability is the first priority, makes the reliability and robustness of turbopump higher.

The design flow concept of the improved design method is shown in Fig. 4. This process for achieving to the optimized design solution is divided into two steps.

The first step is the important process for figuring out some initial optimal forms as shown at the left side in Fig. 4. After the turbopump is given the required design specifications from the engine system, rough physical size of the mechanical elements with the discrete rotational speed parameter are calculated for each subsystem. The obtained each rough size information and data of the RD fluid force estimated by the rough size are taken in the platform to control the optimization system and programs. Also, the platform administers data from each subsystem and manages the interfaces between the subsystems. The optimized rotor forms given by integration of the elements are finally obtained from the complex eigenvalue analysis of the rotordynamics program [6]. In the first step, the unique technique is adopted to get the appropriate solution within a suitable time. Table.1 shows the samples of the selection rules used in the morphology design process. In this task, improbable forms as the turbopumps are screened by the rules of morphology design.

The second step is the optimization process for determining the distance from the axial front edge to each elements and shaft diameter at each element with regard to some excellent solutions

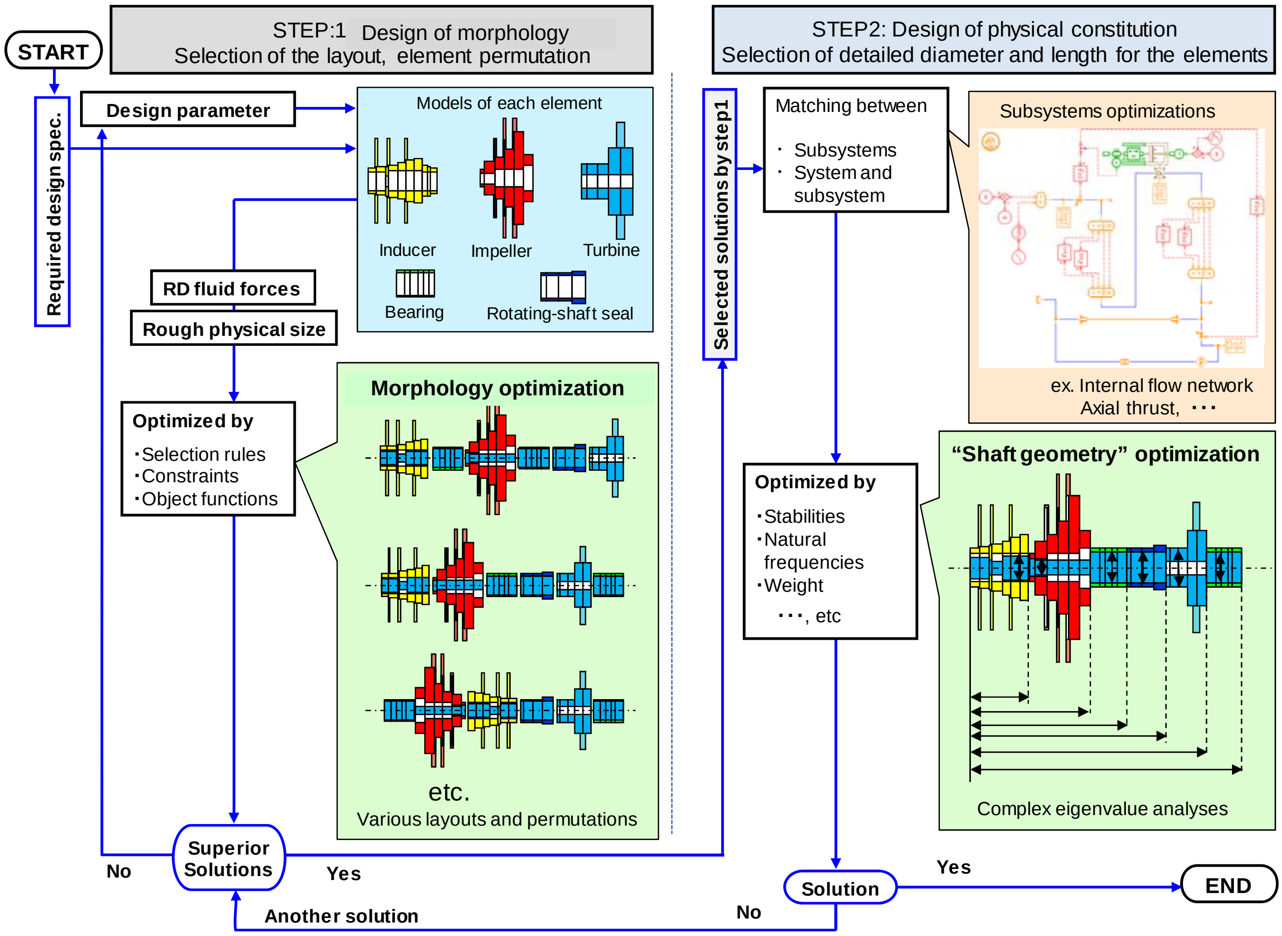


Fig. 4 Concept of the dynamic design.

Table.1 Selection rules for morphology design.

No.	The selection rules for the form design	Reasons
1	In case of bilateral symmetry for two permutations, one of two is calculated.	Avoid of geometric multiple.
2	In case of two pairs of bearings adjoins, a spacer is located at between two pairs.	Meaningless alignments.
3	There is a pair of bearings or nothing at between a inducer and first impeller. Another element cannot locate at there.	Avoid of a complicated flow path.
4	A seal is not located at a end of a shaft.	Meaningless alignments.
5-1	A turbine is located on right hand of a seal.	Avoid of a complicated structure. Constraint of shaft length.
5-2	An inducer, first impeller and second impeller are located on left hand of a seal.	
5-3	A seal orientation is remained unchanged by roles (5-1) and (5-2).	Meaningless alignments.
6	No.2 bearings (a pair of bearings) is located on right hand of No.1 bearings (a pair of bearings).	Avoid of geometric multiple.
7	Flow direction of an inducer, No.1 impeller and No.2 impeller are	Avoid of a complicated flow path.
8	No.1 impellor is located at downstream of an inducer. No.2 impeller is located downstream of No.1 impeller.	Avoid of a complicated flow path.
9	In case of a seal is located at downstream of No.2 impeller,	
9-1	⇒a spacer is located at between a No.2 impeller and seal.	Keep a depressurization area.
9-2	⇒when No.2 impeller, No.1 bearings and No.2 bearings form a line, a spacer is located between No.1 and No.2 bearings.	Meaningless alignments.
9-3	⇒when No.2 impeller, No.1 bearing and a seal form a line, a spacer is located downstream of No.1 bearings.	Keep a depressurization area.
9-4	⇒when No.2 impeller, No.2 bearings and a seal form a line, a spacer is located upstream of No.2 bearings.	Keep a depressurization area.
10	In case of a top of inducer face in a turbine, a spacer is located ahead of a top of inducer.	Avoid of a complicated flow path and structure.
11	In case of No.1 and No2. impeller adjoin, a spacer is located between impellers.	Avoid of a complicated flow path and structure.

obtained by the first step. In this step, an example of the performance indexes are the real part (stability) and the imaginary part (natural frequencies) of the complex eigenvalue analysis in the rotordynamics program.

The optimization in the first step is given a variety of equality constraints and inequality constraints from the subsystems. For example, these are the upper limit of the DN value for the bearings, the upper limit of the differential pressure of the shaft seals, the shaft power of the turbines (equality constraint), the upper limit of the tip-circumferential velocity of the impellers and the turbines, the criterion apart from the critical speeds, the upper limit of the whirl amplitude at each node corresponding to the mode shape, and so on. Therefore, the most important thing for the development of the morphology design method will be how to set the constraints, the performance indexes and the design parameters. While solving the optimization problem for the layout and permutation of the elements, the sizes of the rotating elements are fixed. Also in contrast, while solving the optimum problem for the rotor geometry and shape (shaft diameter and distance from the front edge to the elements), the layout and permutation of the elements are fixed. The optimization of the shapes such as the blade profile etc., in the subsystems will be carried out by using an inverse design method or the multi-objective optimization under the given constraints after the completion of the optimization as the rotor system by the morphology design method.

3 Conclusion

New concept regarding the morphology design method for suppressing the rotor lateral vibration for the rocket engine turbopumps is presented in this paper. This design method takes into account the RD fluid force generated from the rotating elements. Although it is not easy to obtain data of the RD forces by CFD or EFD, but need of the fluid forces will be higher with the increase in the energy density of the turbomachinery. In addition, in order to suppress rotor vibration, deep knowledge and insight about the RD fluid forces and rotordynamics is needed. The rotor vibration problem for the rocket engine turbopumps is directly related to the reliability of rocket engines, therefore the development of this design method is expected strongly and widely.

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