Influence of impeller’s elastic deformation on the stability of balance piston mechanism of rocket engine turbopump

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Abstract. Turbopump of rocket engine is one of the most important components and has a role to rise pressure of propellant and send to combustion chamber. Turbopump is required to generate high power compared to its small size, so vibration problems often occurred. Axial vibration is one of these vibration problems. Rocket engine turbopump generates large axial thrust due to its high discharged pressure. However, bearings cannot support the axial thrust because turbopump operates in cryogenic condition. Balance piston (BP) mechanism, which is a self-balancing mechanism of axial thrust, is hired for rocket turbopump to deal with the large axial thrust. BP has two orifices at impeller back shroud, and turbopump rotor is designed to be movable in axial direction. Pressure of impeller back-shroud changes in accordance with the change of two orifices clearances when rotor moves in axial direction, and axial thrust is automatically balanced. Although BP mechanism is statically stable, it has a potential to be dynamically unstable due to compressibility of fluid. If BP mechanism becomes unstable, self-excited vibration in axial direction occurs and turbopump can be destroyed due to contact of impeller and casings. In recent researches, stability of BP mechanism is investigated by one-dimensional model of BP. In this approach, rotor including impeller is modeled in lumped mass. However, in the case of open impeller with large diameter, rigidity of impeller decreases compared to closed impeller, and the elastic deformation of impeller affects the stability and vibration mode in axial direction. In such cases, conventional lumped mass model cannot be used. In this paper, influence of elastic deformation of impeller on the stability of axial motion is investigated by the model of BP mechanism including impeller’s elastic deformation.

1. Introduction

One of the major problems of rocket engine turbopumps is a vibration problem. Turbopump is one of the most important component in rocket engine and plays a role to pressurize the propellant such as liquid oxygen and liquid hydrogen. Because of the severe constraints for weight, turbopump is designed to operate in high speed rotation and high discharged pressure to respond to the requirement of engine system. Turbopump generates high power compared with its size, so if its damping decreases, severe vibration occurs and results in the destruction of hardware. In the development of rocket turbopump not only radial vibration problems, but also axial direction-vibration problem occurred.

High discharged pressure of turbopump generates large axial load, which becomes several tons. Because operation fluid of turbopump is cryogenic, special bearings, which are designed for cryogenic
operation, cannot support the large axial load. Balance piston mechanism (BP) [1], which is a self-thrust balancing system which using the pressure of internal flow of the pump, is hired to balance the axial thrust. Figure 1. shows the cut-model of rocket turbopump and figure 2. shows the operating principle of BP. BP is installed in the back side of impeller, and consists of two orifices, which is called No1 orifice and No2 orifice, BP chamber surrounded by these two orifices. Fluid goes from impeller outlet through No1 orifice, BP chamber, and No2 orifice with pressure drop. Turbopump rotor with BP is movable for axial direction, so No1 and No2 orifice clearances changes in response to rotor axial motion. The change of two-orifice clearance result in the change of BP chamber pressure, which acts as self-balancing system of axial load. BP is a kind of an axial direction spring using fluid pressure, and it is stable statistically. However, it can be unstable dynamically.

**Figure 1.** Rocket engine turbopump (hydrogen turbopump).

**Figure 2.** Balance piston mechanism. Impeller is moveable in axial direction, No1 and No2 orifice clearance changes according to impeller movement. Pressure of BP chamber, backside of impeller, is adjusted to balance axial load with variable orifice clearance.
Conventionally research for axial thrust of pumps is conducted. Kurokawa studied the characteristics of axial thrust in centrifugal impellers [2] and the gap flow between impeller shroud and pump casing [3]. Horiguchi investigated dynamic characteristic of gap flow between rotating disk and stationary disk [4]. These studies are mainly focused on gap flow. On the other hand, pressure drop at No1 and No2 orifices has a big influence on BP characteristic. Hayashi constructed one-dimensional model of BP with lumped mass, volume and two orifices, and investigated the mechanism of destabilization in axial direction [5]. Kimura conducted dynamic CFD analysis of BP and reported dynamic characteristic of BP [6]. However, in these research deformation of impeller is not included. In case of large size impeller or open impeller with no front shroud, impeller deformation by fluid pressure is not negligible, so it is possible that deformation affect the stability in axial direction. Existing research regard impeller and rotor as a rigid body, so deformation has not been taken into account.

In this paper, one-dimensional modeling including structure deformation is conducted, and results of investigation about stability by linear analysis are showed. In addition, time-domain simulation using FEM model is conducted to observe the vibration mode of BP including structure deformation.

2. One-dimensional model of balance piston and linear analysis

2.1. BP modeling

In this paper simulated device of BP is studied. Existing research [6][7] make clear that the change of No1 and No2 orifice clearance and the volume change of BP chamber caused by rotor motion is important for axial stability. These components are equipped in this device, showed in figure 3. BP is placed at the backside of impeller, so No1 and No2 orifice is annular clearance and BP chamber is disk-shaped volume. However, simulated device studied in this paper beam structure is hired to facilitate studying the effect of structure deformation. No1 orifice clearance is straight gap in this device.

![Balance piston model](image)

**Figure 3.** Balance piston model. (a) is usual balance piston of turbopump. (b) is simulated device studied in this paper.
One-dimensional modeling of BP is conducted here. In order to check the effect of deformation of beam part, three cases are studied.  
- **Case1:** rigid body only  
- **Case2:** beam part only  
- **Case3:** assembly of rigid body and beam part

Existing research is conducted only for Case1 model. Case2 is useful to check the beam part characteristic without rigid body. Case3 is able to know characteristic including beam part and rigid body. The way of modeling is showed in figure 4. In modeling of Case3, beam part which simulate impeller disk is modeled into equivalent stiffness and mass. Upstream pressure of No1 orifice $P_1$ and downstream pressure of No2 orifice $P_2$ is boundary condition. BP chamber pressure $P$ is calculated from pressure losses in No1 and No2 orifice. In axial load calculation BP chamber pressure $P$ is assumed to be uniform in BP chamber and axial load can be determined from multiplication of $P$ and BP chamber load area $A_{BP}$. Because of deformation of beam part, displacement of rigid body and beam part can be different. Positive direction of displacement is defined to the same direction in which No1 orifice clearance becomes narrower, No2 orifice clearance becomes wider and BP chamber volume becomes larger.

**Figure 4.** One-dimensional model of balance piston. (a) is the definition of pressure and displacement. (b) is Case1, rigid model. (c) is Case2, beam model. (d) is the main target of the study, assembly of beam part and rigid body model.

For these three cases theoretical analysis is conducted. Basic equation is showed below. For each case the first equation is equation of motion, and the second is equation of fluid compressibility which represents the relation between volume change and pressure change in BP chamber.
Case1: rigid body only

\[ M_b \frac{d^2 x_b}{dt^2} + K_b x_b = P A_{BP} \]  

\[ \frac{dP}{dt} = \frac{K_f}{V_0} \left( C_{d1} b D_1 (x_{10} - x_r) \sqrt{\frac{2(P_1 - P)}{\rho}} - C_{d2} \pi D_2 (x_{20} + x_r) \sqrt{\frac{2(P - P_2)}{\rho}} - A_{BP} \frac{dx_r}{dt} \right) \]  

Case2: beam part only

\[ M_b \frac{d^2 x_b}{dt^2} + K_b x_b = P A_{BP} \]  

\[ \frac{dP}{dt} = \frac{K_f}{V_0} \left( C_{d1} D_1 (x_{10} - x_b) \sqrt{\frac{2(P_1 - P)}{\rho}} - C_{d2} \pi D_2 (x_{20} + x_r) \sqrt{\frac{2(P - P_2)}{\rho}} - A_{BP} \frac{dx_b}{dt} \right) \]  

Case3: assembly of rigid body and beam part

\[ M_b \frac{d^2 x_b}{dt^2} + K_b (x_b - x_r) = P A_{BP}, \quad M_r \frac{d^2 x_r}{dt^2} + K_b (x_r - x_b) = 0 \]  

\[ \frac{dP}{dt} = \frac{K_f}{V_0} \left( C_{d1} \pi D_1 (x_{10} - x_b) \sqrt{\frac{2(P_1 - P)}{\rho}} - C_{d2} \pi D_2 (x_{20} + x_r) \sqrt{\frac{2(P - P_2)}{\rho}} - A_{BP} \frac{dx_r}{dt} - A_{BP} \left( \frac{dx_b}{dt} - \frac{dx_r}{dt} \right) \right) \]  

Parameter values are showed in Table 1 (including parameters used for FEM simulation showed in next section).

| Item          | Description          | Value | Unit |
|---------------|----------------------|-------|------|
| \( m \)       | mass of piston*      | 1     | kg   |
| \( b \)       | beam width           | 0.05  | m    |
| \( h \)       | beam thickness       |       | m    |
| \( L \)       | beam length          | 0.2   | m    |
| \( P_1 \)     | No1 upstream pressure| 2000  | Pa   |
| \( P_2 \)     | No2 outlet pressure  | 0     | Pa   |
| \( \rho_b \)  | beam density         | 2700  | kg/m3|
| \( E_b \)     | beam Young's modulus | 200   | GPa  |
| \( \rho \)    | fluid density        | 1.2   | kg/m3|
| \( K_f \)     | fluid bulk modulus   | 0.14  | MPa  |
| \( V_0 \)     | BP chamber volume    |       | cc   |
| \( \chi_{all} \) | total orifice clearance | 0.003 | m    |

* Including mass of beam part.

** Beam material is Aluminum. Working fluid is air.
2.2. Linear analysis

Root locus calculated by linear analysis is shown in figure 5. BP chamber volume is changed from 150cc to 1500cc with step value 150cc. All results show that value of real part increases (stability decreases) as BP chamber volume increases. This trend is the same as existing research (in case of rigid body). In the right side of figure 5, calculation results in case of 15mm of beam thickness. Results of case3 is showed in two graphs, lower side and upper side. Lower side graph shows the root locus of mode1 (lower frequency), and upper side the rood locus of mode2 (higher frequency). Mode1 is consistent to the result of case1 (rigid only). Otherwise, mode2 shows slightly higher frequency than case2 (beam only). The results of 4mm of beam thickness are shown in right side of figure 5. Frequency of mode1 becomes smaller than results of 15mm, and the difference between results of case2 (beam only) and mode2 becomes smaller. Frequency of mode2 reduces due to reduction of beam thickness, and the change of damping ratio becomes larger.

The next result, which is calculated by changing thickness of beam thickness, is shown in figure 6. Thickness is changed from 3mm to 15mm. Results in the left side of figure 6. shows that real part of mode2 increases (stability decreases) and frequency becomes higher. In the right-side graph result of mode1 is shown, change of real part of mode1 is small.

![Figure 5](image)

**Figure 5.** Root locus calculated by linear analysis. (a) is the case of 15mm of beam thickness, (b) is the case of 4mm of beam thickness. “rigid” is results of case1, “beam” is results of case2 and “mode1” “mode2” is results of case3.
3. Time-domain simulation with FEM model

3.1. FEM modeling

Self-excited vibration occurs when damping ratio of BP decreases and becomes negative value. The results of the previous section show that two kinds of mode (mode1 and mode2) becomes unstable in the case of large BP chamber volume. In order to investigate the behavior of self-excited vibration, time-domain simulation with FEM model is conducted.

FEM model is shown in figure 7. Beam part is divided into 5 elements. Load by P1 and P is added to each node, and outer force and mass of rigid part is added to right end node (node1). Displacement change in up-and-down direction and beam deformation results in change of orifice clearance and BP chamber volume, which causes change of BP chamber pressure.

In constructing FEM model stiffness matrix, which express the relation between the two nodes next to each other, is used [8]. Mass matrix is also defined, and the equation is shown in (7). x represents displacement w and angle β in each node, shown in (8). First line of (7) is equation of motion for x. K is stiffness matrix and M is mass matrix. A_{bp} represents load area of each node. F_{0} is load of each node by P_{1}. A_{v} is a matrix which represents the area of beam deformation that causes to volume changed of BP chamber. Q_{1}, Q_{2} is flow rate passing through No1, No2 orifice. Simulation is conducted by solving this equation using New Mark β method. Parameter values used for simulation is shown in table 1.

$$\begin{pmatrix} M & 0 \\ 0 & 0 \end{pmatrix} \dot{\begin{pmatrix} x \\ \dot{x} \end{pmatrix}} + \begin{pmatrix} 0 & V_{0}/K_{f} \\ A_{v} & 0 \end{pmatrix} \begin{pmatrix} x \\ \dot{p} \end{pmatrix} + \begin{pmatrix} K & -A_{bp} \\ 0 & 0 \end{pmatrix} \begin{pmatrix} x \\ p \end{pmatrix} + \begin{pmatrix} 0 \\ Q_{1} - Q_{2} \end{pmatrix} + \begin{pmatrix} F_{0} \\ 0 \end{pmatrix} = 0$$

(7)

$$\begin{pmatrix} x \\ \dot{x} \end{pmatrix} = \begin{pmatrix} w^{1}, \beta^{1}, w^{2}, \beta^{2}, ..., w^{n}, \beta^{n} \end{pmatrix}^{T}$$

(8)

$$A_{v} = \begin{pmatrix} h_{L}/2, 0, h_{L}, 0, h_{L}, 0, h_{L}, 0, h_{L}, 0 \end{pmatrix}$$

(9)
3.2. Simulation results

In showing simulation results, displacement of node 1 and node 6 are displayed (figure 9.). Results are shown in figure 10. BP chamber volume is set to 1350cc, and thickness of beam is changed. (a) and (b) is a result of beam thickness 4mm. The difference between (a) and (b) is the scaling of x-axis. In case of 4mm, vibration of $x_b$ is growing with time (mode 2, discussed in section 2). Although $x_r$ is also oscillating, vibration amplitude is smaller than $x_b$. (c), (d) is a result of beam thickness 6mm. In this case, vibration of $x_b$ is also growing with time. (e), (f) is a result of 8mm. Amplitude of $x_b$ becomes smaller than previous case, and the vibration of $x_r$ is clearly observed. In this case $x_b$ and $x_r$ shows almost the same motion (mode 1, discussed in section 2). (g), (h) is case of 15mm. This result denotes the same tendency of results of 6mm.

Linear analysis results, discussed in section 2, is shown in figure 11., which is the relation of damping ratio between mode 1 and mode 2 when beam thickness is changed. Both damping ratios are negative, so mode 1 and mode 2 are both unstable. However, simulation results show that mode 2 appears only in the case of small beam thickness.
Figure 10. Simulation results. The difference between left side and right side is x-axis scaling. (a), (b): beam thickness is 4mm. (c), (d): 6mm. (e), (f): 8mm. (g), (h): 15mm.
Figure 11. Linear analysis results (discussed in section 2) with changing thickness of beam part. Smaller thickness results in smaller damping ratio of mode 2. On the other hand, damping ratio of mode 1 becomes large as thickness decreases.

4. Conclusion
The effect of structural deformation on the axial stability of balance piston mechanism, which is an axial thrust self-balancing system, is investigated. Simulated device which consists of elastic beam and rigid boy is studied by linear analysis of one-dimensional model and time-domain simulation with FEM model. Results of linear analysis shows that the stability of vibration mode in which beam deformation is dominant decreases as the beam stiffness decreases. Time-domain simulation results shows that vibration mode in which beam deformation appears with small beam thickness, and vibration mode in which beam and rigid body act in the same motion. appears with large beam thickness. From these results structural deformation has an influence on the stability of balance piston mechanism, it is necessary to be careful in designing smaller stiffness impeller, such as large-diameter open impeller.

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