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J. Kurokawa
Yokohama National University
Yokohama, Japan

K. Kamijo and T. Shimura
National Aerospace Laboratory
Kakuta, Japan

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AXIAL THRUST ANALYSIS IN LOX-PUMP

Junichi Kurokawa*
Yokohama National University
Yokohama, Japan

Kenjiro Kamijo** and Takashi Shimura***
National Aerospace Laboratory
Kakuta, Japan

Abstract

The LOX pump in the first stage of the next large launch vehicle, H-II rocket, in Japan was developed by National Aerospace Laboratory and showed fairly good axial thrust performance. However, the behavior of the axial thrust is not well known because of the complicated mechanism of the thrust-balancing device. In order to reveal the flow characteristics in the thrust balancing device and to improve the device, the internal flow in the device is fully analyzed. The analysis developed here is confirmed to give satisfactory results by comparing with the measurements. Using the present analysis, the axial thrust behavior of the LOX pump is revealed for various combinations of balancing piston, balancing hole, swirl breaker and so on. The present analysis is applicable for any types of thrust balancing device.

Nomenclature

AM : angular momentum of leakage flow
b : radial gap width of annular flow
C_D : drag coefficient of fastening bolt
C_q = $(q/2\pi r_2^3 \omega) Re^{1/5}$: leakage parameter
K = $v_\theta/r\omega$: tangential velocity ratio of core fluid
M : friction moment of the wall
n : number of balancing holes, fastening bolts, or grooves of a swirl breaker
p : pressure
P = $2p/\rho u_2^2$: non-dimensional pressure
q, Q : leakage flow-rate and pump discharge
r, r₂: radius and main impeller radius
Re = $r_2 u_2/\nu$: Reynolds number
s : axial gap width of disk flow
u₂ : main impeller tip speed
v : absolute velocity
z : axial distance from the wall
 ψ : pressure head coefficient of pump
 λ : friction coefficient of annular gap
 ν, ρ : kinematic viscosity and density of fluid
 ω : angular velocity of shaft
 τ : wall shearing stress
 ϕ : discharge coefficient of main impeller

Subscripts
O : at the design point of main pump
B, P : at balancing hole or balancing piston
I, O : at the inlet or the outlet of the control volume through which leakage flows
r, θ , z : radial, tangential or axial component
R, S, C : at the rotating, stationary or cylindrical walls

1. Introduction

It is not too much to say that the reliability and safety of pumps transferring combustible gas, such as LOX, LH₂, LPG or LNG, depend upon the axial thrust balance, and the adoption of a thrust balancing device is unavoidable. In thrust balancing devices is required strong stiffness not only against the static balance but against the dynamic balance. In this case a complicated sealing system is also adopted to prevent the leakage of combustible gases and accordingly the thrust balancing device becomes very complex.

The LOX pump of 20000 rpm is going to be used in the first stage of the next launch vehicle, H-II rocket, in Japan. It was developed by the National Aerospace Laboratory and showed fairly good thrust balance^{(1)~(3)}. This pump has a unique thrust balancing device and two types of seal systems, but the behavior of the axial thrust and the flow characteristics in the device is not yet well known.

One of the present authors has been studying the axial thrust of radial flow turbomachinery, and revealed that the accuracy of the thrust calculation is mainly dependent upon the leakage flow-rate and the boundary values of the pressure and the flow velocity^{(4)~(7)}. The former is usually determined from the gap flow analysis and the latter from the prediction of the impeller outlet flow characteristics, of which method has also been developed by one of the present authors^{(8)~(9)}.

The present study is aimed to develop the method of predicting the axial thrust of multi-stage pumps with complicated balancing devices, and to reveal the thrust behavior of the LOX pump in Japan.

2. Theory

Axial thrust is caused by the pressure acting on the rotating parts of a machine. Axial thrust analysis then consists of the flow analysis in the gap between a rotating and a stationary walls and the determination of boundary values. In the gap flow there are generally the following two types. One is the axial gap flow between a rotating disk and a stator such as that at the back of an impeller shroud, and the other the annular gap flow such as that in an annular seal. In the former the centrifugal force is dominant, while in the latter the wall shearing stress is dominant.

The gap flow analysis requires precise determination of the boundary values of flow velocity and pressure at the inlet and the outlet of all the gaps, that is at the entrance and the

* Professor, Department of Mechanical Eng.

** Head of Rocket Fluid System Section, Member of AIAA

*** Senior Researcher of Rocket Fluid System Section, Member of AIAA

exit of an impeller, an inducer and guide vanes. The boundary values at an impeller outlet are predicted by the present author's method ^{(8),(9)}, and those at the outlet of an inducer and guide vanes are predicted by using an ordinal two-dimensional cascade theory.

2.1 Analysis of Axial Gap Flow

As shown in Fig.1, the flow in the axial gap between the rotating and the stationary disks consists of the boundary layer flows along both disk walls and the core flow in which the tangential velocity v_θ is expressed as $Kr\omega$ and no radial flow is induced. The flow characteristics are determined from the equations of momentum, angular momentum and continuity shown as

$$\frac{\partial}{\partial r} \left\{ r \int_0^s v_r dz \right\} - \int_0^s v_\theta dz = -\frac{r}{\rho} \frac{dp}{dr} s - \frac{r}{\rho} (\tau_{Rr} + \tau_{Sr}) \quad (1)$$

$$\frac{\partial}{\partial r} \left\{ r^2 \int_0^s v_r v_\theta dz \right\} = -\frac{r^2}{\rho} (\tau_{R\theta} + \tau_{S\theta}) \quad (2)$$

$$2\pi r \int_0^s v_r dz = q \quad (3)$$

Assuming the 1/7th power law for the velocity distribution in the boundary layers and shearing stress of Blasius type for hydraulically smooth wall, the above equations are transformed to two ordinary differential ones as for the core velocity ratio K and the nondimensional pressure P as a function of radius ratio r/r_2 . The governing equations are also obtained for rough wall case, by assuming the logarithmic velocity profile ⁽⁶⁾, and for a very narrow gap such that the boundary layers on both walls interfere with each other ⁽⁷⁾.

The analyzed results reveals that the main influencing parameters upon the characteristics of axial gap flow are the leakage flow parameter Cq , the tangential velocity ratio K_1 of the leakage before entering into the gap, the axial space ratio s/r_2 , the relative surface roughness and the Reynolds number Re defined as follows;

$$Cq = (q/2\pi r_2^3 \omega) Re^{1/5}, \quad Re = r_2 u_2 / \nu \quad (4)$$

The analysis of the axial gap flow requires the accurate value of the leakage parameter Cq , which is to be determined from the pressure drop at balancing holes or annular seals, but the pressure drop also varies with the leakage. The trial-and-error method is then needed to determine the higher approximation of leakage by assuming the 1st one.

Determination of boundary values. The boundary value of K is determined from the angular momentum balance in the control volume supposed at the boundary of the flow field. There are various types in boundary conditions and the typical case is illustrated in Fig.2. The leakage entering from an annular gap into the control volume ABCDE surrounded by the rotating wall (R) and the stationary wall (S) brings angular momentum AM_1 and takes out AM_0 per unit time, and the friction torque M_R and M_{CR} are supplied from the rotating wall and M_S and M_{CS} are consumed by the stationary wall. Thus the boundary condition is

$$AM_0 - AM_1 = M_R - M_S + M_{CR} - M_{CS} \quad (5)$$

The right hand of Eq.(5) is calculated by assuming the 1/7th power law for boundary layer velocity distribution and Blasius type searing

Fig.1 Flow in axial gap and symbols

Fig.2 Angular momentum balance at the boundary

stress. The outlet angular momentum AM_0 is calculated considering the leakage exit manner, such as into an axial gap, into a stationary annular gap or into a rotating annular gap. The inlet angular momentum AM_1 is also calculated by considering the leakage entrance manner and pre-rotation ^{(4),(7)}. The boundary equation is thus obtained, and the boundary values of K are determined at all sections where the dimensions and the configurations of the axial gap change.

2.2 Analysis of Annular Gap Flow

The friction coefficient λ of an annular gap flow is usually determined under the assumption that the gap fluid rotates at a half of the rotating wall velocity ⁽¹⁰⁾. However, the fluid rotational velocity $Kr\omega$ is influenced largely by the leakage pre-rotation $K_1 r\omega$. The momentum and the angular momentum balance in an annular gap are expressed as follows using the symbols shown in Fig.3 and the pressure difference Δp ;

$$AM_0 - AM_1 = 2\pi l \{ r^2 \tau_{R\theta} - (r+b)^2 \tau_{S\theta} \} \quad (6)$$

$$b(2r+b)\Delta p = 2l \{ r \tau_{Rz} + (r+b) \tau_{Sz} \} \quad (7)$$

Assuming 1/7th power law of velocity and Blasius-type shearing stress, the above equations becomes

$$0.0267 [(1-K) \{ (1-K)^2 + Y^2 \}^{3/8} - K \{ K^2 + Y^2 \}^{3/8}] / Re^{0.24} = (q/2\pi r^2 l \omega) \{ 1 + 16(K-K_1) \} / 18 \quad (8)$$

$$\Delta p = (\lambda l / 2b + 1.5) v_z^2 / 2g \quad (9)$$

Fig.7 Whole view of the LOX turbopump used in H-II rocket

3. Dimensions of LOX-Pump and Input Data

The whole view of the LOX pump of the H-II rocket is shown in Fig.7. A part of LOX discharged from the main impeller is again pumped by the smaller impeller (preburner impeller) into the preburner, is mixed with LH_2 and is vapourized. The mixed gas drives the turbine and then is lead to the combustion chamber. The LOX pump tested consists of the main and the preburner impellers arranged back to back, and two types of seals are used to isolate the LOX and the mixed gas. At the main impeller entrance the inducer and the guide vanes are installed. The slinger with radial vanes is arranged to cool the rear bearings.

The thrust balancing device consists of the variable orifices A and B shown in Fig.4, which adjust the gap at the back of the main impeller automatically. If the axial thrust toward suction becomes larger, the gap at A becomes smaller, which causes a decrease in pressure and the axial thrust toward suction becomes again smaller. On the contrary if the thrust force toward turbine becomes larger, the orifice B works.

As the gaps at the orifice A and B are very small as shown in Fig.4, the deformation due to the centrifugal force and the inner pressure must be taken into account besides the assembling tolerance. The measured gap at the assembly was $e_A = 38\mu$ but became 30μ at the operation because of the elongation of 22μ of the inner wall and 30μ of the outer wall. The gap of the orifice B was $e_B = 14\mu$ at the pump operation.

The pump tested has 8 leakage loops as shown in Fig.8, in which ① and ⑦ are the bearing cooling and lubrication loops, ② and ⑥ the leakage loops through the impeller front shroud gap into the suction side, and ③, ④ and ⑤ through the impeller back shroud gap and the balancing holes into the main impeller eye. The rear bearing cooling loop branches to ⑦ and ⑧, joins to one loop and flows through the center axis holes. Among them the independent loops are ①, ② and ⑥, and in the other there is interference among several loops.

Calculations are performed in each loop by assuming the first approximation of leakage parameter C_q , and then the pressure distribution is calculated. The leakage of the higher accuracy is asymptotically determined. The calculation of the interference loop is to be initiated from the shortest loop.

4. Comparison of Theory with Measurements

The following three cases were tested here; Test No.1(LN2) : 20,600rpm, 95% of the design flow-rate, with balancing hole B and two fixed orifices.

Test No.2(LN2) : 20,160rpm, 97% of the design flow-rate, with balancing holes A and B and one fixed orifice.

Test No.3(LOX) : 20,100rpm, 102% of the design flow-rate, with balancing hole A and B, one fixed orifice.

The calculations described in Sec.2 are performed for the above three cases. The property of the liquid used is shown in Table 1.

Fig.8 Leakage loops of the LOX turbopump

4.1 Some Problems Arising from Analyzed Results

The calculated pressure distribution at every rotating part showed fairly good agreement with the measurements except for one part, i.e. in the rear shroud gap of the main impeller. The full examination of the impeller and casing revealed that the disagreement was due to the annular groove with a row of fastening bolts machined on the casing wall shown in Fig.9.

In this case the flow around the groove can be modeled to the rotational flow with a row of cylinders surrounded with two annular walls. The core velocity ratio here can also be determined from Eq.(5) by subtracting the following angular momentum loss due to the drag of the cylinder row from the right hand of Eq.(5)⁽¹³⁾;

$$\rho r d_F n_F C_D \int_0^{h_F} (v_r^2 + v_\theta^2)^{0.5} v_\theta dz / 2 \quad (16)$$

where the symbols used are shown in Fig.9. The calculated results reveal that the fluid rotates at less than 0.1ω at the groove and the flow with small angular momentum enters into the inner radii, which results that the fluid rotates very slowly and the pressure is almost uniform in the inner radii of the groove. This fact suggests that the existence of such resistance makes axial thrust considerably large not only in the thrust balancing device but also in the usual rotational flow field.

4.2 Comparison of Impeller Outlet Flow

The calculated results show that the outlet flow from the inducer has still high tangential velocity with strong radial pressure gradient, and that this rotating flow enters into the guide vanes and leaves with weak rotation. Accordingly there remains the flow rotation at the main impeller entrance. This causes the decrease in pumping head about 5.3% of total head.

The predicted head curve of the main impeller is compared with the measurements in Fig.10, which includes the pressure head at the outlet of inducer and inlet guide vane. They agree very well and the inducer pressure head amounts to about 23% of the main impeller head at the designed flow-rate.

4.3 Axial Thrust Behavior

The axial thrust working on the pump side varies largely with the gap s_A at the variable orifice A, and the calculations are performed with varying this gap.

The calculated axial thrust and the leakage characteristics are shown in Fig.11, in which the thrust toward suction side is taken positive. It is seen that the axial thrust decreases considerably with a decrease in the gap s_A . The main difference between Test No.1 and No.2 (and No.3; LOX test) is that the balancing hole A is installed in the latter, and the comparison of the thrust curves of Test No.1 and No.2 (and No.3) reveals that the installation of the balancing hole A decreases the total axial thrust considerably. It is also recognized in Fig.11 that the leakage do not increase by the installation of the balancing hole A in the range $s_A < 0.15$ mm, while for the wider range of s_A the leakage becomes much larger. The above described characteristics are the typical work of this type of balancing piston.

Table 1 Physical properties of LOX and LH₂

	p(MPa)	Temp(K)	ρ (kg/m ³)	μ (N·s/m ²)
LH ₂	0.1013	77.4	808	1.63×10^{-4}
LOX	0.1013	90.2	1140	1.96×10^{-4}

Fig.9 Bolt grooves at the back of the main impeller and the flow around the groove

Fig.10 Pressure head of main impeller and inducer

Fig.11 Total axial thrust and leakage vs. axial gap at the balancing piston A

The stability of the thrust balancing device can be estimated by the gradient dT/ds_A of the thrust curve. In case of no balancing hole A (Test No.1), it needs about 8 tons to move the axis by 0.1 mm around the point of $T=0$, which results that this device has strong stiffness and is stable against thrust variation. But the installation of balancing hole A (Test No.2) makes the thrust curve very gentle.

To determine the operating gap s_A of the balancing device the value of the thrust force working on the turbine rotor is needed, but it changes largely depending upon the turbine operating condition. As the thrust force in the turbine side is not known, the balancing gap s_A is determined so that the calculated pressure at the balancing chamber of the variable orifice A is equal to the measured one. The balancing gap s_A so determined is indicated by ϕ in Fig.11 and is 0.10mm for Test No.1 and 0.17mm for Test No.2. It is then concluded that the operating gap becomes 1.7 times wider by the installation of the balancing hole A, which promotes the pump reliability.

The axial thrust is composed of axial forces working on every rotating part, and the variation of each axial force is shown in Fig.12. The axial forces on the inducer and the slinger are seen nearly equal in magnitude and reverse in direction, and then they cancel with each other. Accordingly, the total axial thrust behavior is determined as the balance between the axial force on the main impeller and on the preburner impeller. For the variation of s_A the axial force on main impeller varies largely but little on the preburner impeller. This is because the change of leakage with the gap s_A is large in the main impeller, while it is small in the preburner impeller. It is interesting that the direction of the axial force of the preburner impeller is reverse to that of ordinary pumps. This is because the leakage in the rear shroud gap is 43% larger than that in the front shroud gap.

To make clear the thrust variation with the gap s_A , the variation of pressure P at the balancing chamber A is shown in Fig.13 together with the leakage q_2 at the back of the main impeller. The pressure curve is seen to be very similar to those of the total thrust in Fig.11 and the axial force on main impeller in Fig.12, and the leakage curve is also very similar to that of the total leakage in Fig.11. This reveals that the variations of total thrust and leakage are mainly caused by the balancing piston A.

4.4 Comparison of Pressure Distribution

The calculated pressure distributions around the main and the preburner impellers are compared with the measured ones in Figs.14(a) and (b), in which the solid lines show the rear shroud side and the dotted lines the front shroud side. The present theory is seen to give satisfactory pressure distributions including the boundary values at the impeller outlet. The pressure distribution at the back of the main impeller rear shroud is seen to be very uniform in spite of large leakage, which is due to the effect of the bolt groove as mentioned in Sec 4.1. This reveals that the annular groove with obstacles on the casing wall has very effective to reduce the

Fig.12 Each component of axial thrust

Fig.13 Pressure and leakage variation at the back of the main impeller ($s_{A0}=0.5\text{mm}$)

(a) Test No.1 (no balancing hole A)
Fig.14 Pressure distribution around the main impeller and the preburner impeller

axial thrust, even though its depth and width is not large.

As mentioned above, the variable orifice A has dominant influence on the total thrust behav-

ior, the calculation is further performed to examine the resistance of the variable orifice by changing the orifice loss coefficient ζ_p from 1.0 to 1.5. The calculated result is shown by the chain dotted line in Fig.14(a). It is recognized that the pressure hardly change in spite of considerable change in the loss coefficient. This is because the increase in orifice resistance results in a decrease in leakage to 83%, and as a result the pressure loss in the orifice A hardly change. Such a case corresponds to the gap blockage by obstacles or gap damage, and the present self-balancing device is proved to have strong stiffness and high reliability.

Comparison of Figs.14(a) and (b) reveals the effect of balancing hole A, and it is seen that the pressure in the rear gap of the main impeller decreases about 10% by the balancing hole A. As the difference of the rotational speed in Figs. 14(a) and (b) causes 5% of pressure difference, the rest 5% is due to the balancing hole. That is to say, the effect of balancing hole A is only 5% decrease in pressure, but it amounts to large decrease in axial thrust as shown in Fig.11, which makes the variable orifice gap wider to the safety side. In this case the total leakage increases to 1.2 times.

The pressure distributions around the slinger and the inducer are compared in Fig.15 in case of Test No.1. The calculated results agree fairly well with the measured ones, which implies that the inducer outlet flow can be predicted by a simple cascade theory around the design flow-rate. The slinger vane is seen to resist leakage into the vane. The labyrinth at the back of the slinger is seen to work effectively with considerable pressure drop, which suggests that it is very effective for the reduction of leakage to decrease the annular gap or to increase the stage number of the labyrinth.

4.5 Effects of Swirl Breaker

A swirl breaker installed on the front casing wall of the main impeller decreases the radial pressure drop, and increases the axial force toward discharge, which results in large decrease in total axial thrust. The calculated pressure drop from $r/r_2=1.0$ to 0.71 is compared with the measured ones in Fig.16 in case that two types of swirl breakers are installed on the casing wall. It is recognized that the radial grooves of only 0.5mm depth decreases the pressure drop considerably and those of 1mm depth makes the radial pressure distribution almost uniform, and that the present analysis gives accurate estimation for the effects of a swirl breaker. It is then concluded that the work of a swirl breaker is mainly due to the angular momentum loss of the fluid entering into the radial grooves, and that its effect is mainly dependent on the total sectional area of the grooves. The angular momentum loss is especially large at the outer radii, because the rotating velocity here is very large. Accordingly, the short grooves have sufficient effects of reducing total axial thrust, if it is installed at the outer radii.

4.6 Thrust Performance at Off-Design Operation

The above-described thrust performances are around the designed point of pump operation. At the off-design operation the axial thrust curve varies depending on the pumping head as shown in

(b) Test No.2 (balancing hole A installed)
Fig.14 Pressure distribution around the main and the preburner impellers

Fig.15 Pressure distributions around the slinger and at the inducer outlet

Fig.16 Pressure drop at the main impeller front shroud and the effect of swirl breaker

Fig.17 for two cases of flow-rates of $1.2Q_0$ and $0.8Q_0$. The operating gap s_A becomes a little larger at higher flow-rates and smaller at lower flow-rates. In Fig.17 the thrust curves at different rotational speed N are also shown, and it is recognized that the nondimensional quantities of axial thrust and leakage become slightly

larger with the Reynolds number, and that the operating gap s_A changes little for the change of the pump rotational speed.

4.7 Some Improvement in Thrust Balancing Device

As the present analysis is confirmed to give satisfactory results, the calculations are further performed to improve the thrust balancing device. Several variations of the main influencing parameters upon total axial thrust are supposed, such as changing the diameter of the balancing hole A and that of the labyrinth seal at the back of the preburner impeller, installing a swirl breaker on the front casing wall, or removing the bolt groove at the back of the main impeller. The calculated results are shown in Fig.18 for Test No.1 and in Fig.19 for Test No.2.

The calculated results in Fig.18 show that the operating gap s_A can be made a little wider by reducing the labyrinth diameter at the back of the preburner impeller or by installing a swirl breaker on the front casing wall without losing the stiffness of the balancing device. The installation of the balancing hole A makes the operating gap much wider, though the thrust curve becomes very gentle and the stiffness of the balancing device decreases largely. In the transient operation the balancing hole A has an important role of attaining rapid decrease of pressure at the back of the main impeller.

It is to be remarked that the removal of the bolt groove at the back of the main impeller makes the thrust curve very unstable, and that the operating gap s_A becomes negative. This implies that the bolt groove has an important role of balancing the axial thrust of this pump, and that it is difficult to ensure sufficient operating gap without the bolt groove, even if several improvements are attained.

In Fig.19 it is seen that the reduction of the diameter of the balancing hole A from 4.5mm to 2mm makes the thrust curve much stiffer, though the operating gap becomes much smaller. But the combination of reduction of the balancing hole diameter with the installation of a swirl breaker can make the thrust curve much stiffer and ensure almost the same operating gap s_A as the case of no modification ("original").

5. Conclusions

The theoretical analysis of predicting axial thrust and leakage of multistage centrifugal pumps with complex thrust balancing devices is established. Using this method, the thrust and leakage performances of the LOX turbopump of H-II rocket are revealed, and the results are summarized as follows;

(1) The comparison of the present prediction with the measurements revealed that the angular momentum balance presented here gives satisfactory results including the boundary values and also the accurate prediction of axial thrust behavior. It is here very important to predict correctly the effects of annular and axial grooves installed in the casing wall.

(2) The axial thrust of the LOX pump varies largely with the gap of the balancing piston, and the present balancing device has significant stiffness against the thrust variation, but the operating gap is as narrow as 0.10mm. When the

Fig.17 Thrust performance at off-design operation and effect of the Reynolds number

Fig.18 Thrust curve variation due to the variation in dimensions for Test No.1

Fig.19 Thrust curve variation due to the variation in dimensions for Test No.2

balancing hole A (of 4.5mm dia.) is installed, the operating gap increases to 1.7 times and the reliability of the device increases, though the stiffness decreases largely. To increase the stiffness, it is effective to reduce the diameter of the balancing hole A.

(3) The operating gap of the balancing piston varies little for the variation of the impeller rotational speed. It decreases a little in the smaller pump discharge and increases a little in the larger discharge. The adoption of balancing holes usually increases leakage, but in the present device the leakage changes little at the operating gap.

(4) The balancing piston adopted has high reliability for gap blockage and damage, and the operating gap hardly changes for large variation of leakage, as the pressure distribution changes little.

(5) The existence of obstacles, such as a row of bolts in an annular groove on the casing, decreases the fluid peripheral velocity largely and makes the radial pressure distribution uniform, which causes considerable axial force. However, if it were not for the annular groove in the present device, it should be very difficult to ensure the enough operating gap.

(6) The prediction of a swirl breaker is also predated. A swirl breaker is effective to increase the negative axial thrust and the operating gap of the balancing piston, but the effect is limited to some extent up to zero pressure drop. The effect is mainly dependent upon the total sectional area of the grooves.

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