Experimental study on runaway characteristics of pump system

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Abstract. Experiments on runaway characteristics were conducted for two sets of tubular pump systems and two sets of vertical axial pump systems. The measurement results show that at different blade angles the unit runaway speeds are different and increase with the increase of the blade angle. For the same pump system at the same blade angle the unit runaway speeds decrease with decrease of reverse-water-head of the system. The different outflow passages of pumps also caused variation of the unit runaway speeds. Through the calculation of resistance torque the variable factors of runaway speed with same blade angle are analyzed under different operation conditions of reverse-water-head. It is evident that the unit runaway speed obtained from model pump system is applicable and safe for conversion to prototype pump system.

1. Introduction
When the pump suddenly stop because of malfunction, meanwhile, the water cut-off facilities of outlet passage don’t work as usual, for example value, sluice gate and so on, so that pump operates in the turbine conditions. If the reverse rotation speed of impeller is maximal and remains unchanged, the pump will operate in the runaway condition, the reverse rotation speed will be defined as the runaway speed of the pump in this condition[1]. For large and medium pumping station, it is very necessary to obtain the runaway speed of pump, so that the strength of the impeller parts is confirmed and the safety and operation stability of pump are ensured according to it. The model pump segment consist of three parts: impeller, guide vane and elbow pipeline, and the pump is consist of three parts: the model pump segment, inlet and outlet passage in the 《Code for model pump and its installation acceptance tests》 (SL140-2006). At now, the runaway rotation speed is obtained by the conversion of the model test results of pump system. Domestic and foreign scholars have made achievement in studying on runaway characteristics of hydraulic machinery, such as literature [1-6]. In this paper, four sets of pump system were selected as study object, and the methods of experimental study and theoretical analysis were used to study on the runaway speed of pump system.

2. Theoretical analysis
According to the regulations of article 6.3.1 in the 《Code for model pump and its installation acceptance tests》 (SL140-2006), the pump must operate at the turbine conditions and output torque is...
zero for the runaway characteristic test, the rotation speed is defined as the runaway speed of pump. The formula of the runaway speed is given in the article 6.3.2, and the formula is shown in equation 1:

\[ n'_{1,R} = \frac{n_R D}{\sqrt{H}} \]  \hspace{1cm} (1)

Where: \( n'_{1,R} \) is the unit runaway speed; \( n_R \) is the runaway speed of model test; \( D \) is the nominal diameter of impeller; \( H \) is the experimental head.

According to the definition of the runaway speed in the Ref.[7], the reverse speed of impeller is maximal and remains unchanged in certain operation condition, meanwhile, the reverse rotation speed is defined as the runaway speed. Compared with the impeller, the pump also has guide vane, inlet and outlet passage. When pump operates in the turbine condition, the impact action of water on impeller will be affected by the outlet flow pattern of guide vane. For the same impeller and different guide vane, the runaway speed is different. At now, the runaway speed of pump is calculated on the basis of the equation of the runaway speed of impeller. For the pump at the same blade angle, the pump meets the requirements of pump affinity law in the different reverse head. For the same pump, when the pump operates in the reverse head \( H_1 \), the rotation speed is \( n_1 \), and which operates in the reverse head \( H_2 \), the rotation speed is \( n_2 \), the unit runaway speeds were calculated by the equation (1) in different conditions.

\[ n'_{1,R1} = \frac{n_1 D}{\sqrt{H_1}} \]  \hspace{1cm} (2)

\[ n'_{1,R2} = \frac{n_2 D}{\sqrt{H_2}} \]  \hspace{1cm} (3)

According to pump affinity law, the equation (2) was divided by and equation (3), we can get the equation (4).

\[ \frac{n'_{1,R1}}{n'_{1,R2}} = \frac{n_1 D}{\sqrt{H_1}} \frac{\sqrt{H_2}}{n_2 D} = 1 \]  \hspace{1cm} (4)

According to equation (4), we can know the unit runaway speed is unique value for the pump at the same blade angle.

![Figure 1. Schematic diagram of test-bed.](image)

3. Model device and test methods
3.1. Test-bed and measuring instruments

Four sets of pump system were tested in high-precision hydraulic machinery test-bed of Jiangsu key lab of hydrodynamic Engineering in Yangzhou University. The profile sketch of the whole test-bed is shown in Fig.1, which is the vertical closed cycle system, the total length is 60.0 m, and the diameter of pipeline is 0.5 m and 0.4 m.

The comprehensive uncertainty of test-bed efficiency is ±0.39%, which meet the accuracy requirements of the industry standard《Code for model pump and its installation acceptance tests》(SL140-2006), which is established by the Ministry of Water Resources of the People’s Republic of China.

Head of pump is directly measured by differential pressure transmitter EJA110A. The torque of model pump is directly measured by torque meter ZJ type. The flow rate is measured by electromagnetic flowmeter E-mag type. Test equipments of test-bed and the calibration accuracy of them are shown in Tab.1

| Terms       | Equipment                        | Type         | Range for measuring | Calibration accuracy |
|-------------|----------------------------------|--------------|---------------------|----------------------|
| head        | differential pressure transmitter| EJA-110A     | 0 ~ 200 kPa         | ±0.113%              |
| Flow rate   | Electromagnetic flowmeter        | E-mag        | DN 400mm            | ±0.197%              |
| torque and  | Torque, speed sensor             | ZJ           | 100 N·m             | ±0.108%              |
| speed       | digital torque and speed routing | TS-800B      |                     | ±0.0203%             |

3.2. Test methods

According to the requirement of runaway characteristics test in《Code for model pump and its installation acceptance tests》(SL140-2006), four sets of pump system were done, which is consisted of two sets of tubular pump system and two sets of vertical axial-flow pump system, and mechanical seals were taken in the model test. Test parameters of each pump system are shown in Tab.2

| Terms                             | Nominal diameter of impeller (mm) | Blade angle   |
|-----------------------------------|-----------------------------------|---------------|
| Bulb tubular pump system          | 300                               | +2°, -2°      |
| Shaft tubular pump system         | 300                               | +2°           |
| Vertical axial-flow pump system   | 300                               | +2°           |
| (type A)                          |                                   |               |
| Vertical axial-flow pump system   | 300                               | 0°            |
| (type B)                          |                                   |               |

4. Experimental results and analysis

For bulb tubular pump system and shaft tubular pump system, the relation curves of the unit runaway speed and reverse head are shown in Fig.2 at the blade angle +2°. For two sets of vertical axial-flow pump system, the relation curves of the unit runaway speed and reverse head are shown in Fig.3 at the blade angle +2° and 0°.
We can know test results is not consistent with theoretical conclusions from Fig.2 ~ Fig.3. For the same pump with the decrease of reverse head, the unit runaway speed decrease gradually at the same blade angle, the regression analysis shows the relation of which is conical relationship. When the pump operates in the runaway condition, torque of water to impeller is equal to resistance torque. The resistance torque mainly is consist of friction torque $M_1$ between water and the out surface of impeller, friction torque $M_2$ of stuffing and friction torque $M_3$ of rolling bearing, the formula of each friction torque is shown in equation (5) ~ (7) according to Ref.[8].

$$M_1 = C_f r_1^5 \omega^2 \gamma$$  \hspace{1cm} (5)

Where: $C_f$ is friction coefficient; $r_1$ is the radius of impeller; $\omega$ is the angular velocity; $\gamma$ is the water severe.

$$M_2 = r_2^2 S \frac{\mu_1}{\mu_2} P_0 e^{-2 \frac{\mu_1}{\mu_2}} (1 - e^{-2 \frac{\mu_1}{\mu_2}}) \pi$$  \hspace{1cm} (6)

Where: $r_2$ is the radius of bearing; $S$ is the thickness of filler; $l$ is the length of filler; $\mu_1$ is sliding friction coefficient; $\mu_2$ is friction coefficient of axial force; $P_0$ is the residual pressure of pump.

$$M_2 = 2 \eta \frac{u}{\delta} r_3^2 l \pi$$  \hspace{1cm} (7)

Where: $\eta$ is absolute viscosity coefficient of lubricating oil; $u$ is the peripheral speed of the shaft; $r_3$ is the radius of shaft; $l$ is the length of shaft; $\delta$ is radial clearance of bearing. If the thickness of filler is immutable value, and wear and tear of filler is not considered, the ratio of each resistance torque is shown in equation (8) ~ (10) in the reserve condition 1 and 2.

$$\frac{M_{11}}{M_{12}} = \frac{n_1}{n_2}^2 \hspace{1cm} (8) \hspace{1cm} M_{21}/M_{22} = 1 \hspace{1cm} (9) \hspace{1cm} M_{31}/M_{32} = n_1/n_2 \hspace{1cm} (10)$$

The ratio of total resistance torque is shown in equation (11).

$$\frac{M_1}{M_2} = \frac{M_{11} + M_{21} + M_{31}}{M_{12} + M_{22} + M_{32}} \neq idem$$  \hspace{1cm} (11)

According to equation (11), the results is inevitable that unit runaway speed of pump is not equal at the same blade angle, which shows that the model test of the runaway characteristics of pump don’t meet the requirement of pump similarity law.
For prototype pump, the component parts of resistance torque are more than that of model pump system. Besides of above three parts, the total resistance torque also includes the resistance torque of thrust head and upper guide bearing, sliding rotor and lower guide bearing, thrust head mirror plate and thrust bearing, et al. It is difficult that the resistance torques of prototype and model pump meet the pump similarity law. The prototype resistance torque is larger than that of model, so that the actual runaway speed of prototype pump is safer, on the basis of the maximal runaway speed of model pump.

On the basis of the unit runaway speed of four sets of model pump, the actual maximal runaway speed was obtained at the same maximal reverse head for each prototype pump. The calculation results are shown in Tab.3.

| Terms                                      | Rated speed r/min | Maximal runaway speed r/min | Ration λ |
|--------------------------------------------|-------------------|-----------------------------|----------|
| Bulb tubular pump system                   | 120.0             | 155.4                       | 1.295    |
| Shaft tubular pump system                  | 189.5             | 273.7                       | 1.444    |
| Vertical axial-flow pump system type A     | 250.0             | 283.3                       | 1.133    |
| Vertical axial-flow pump system type B     | 125.0             | 179.2                       | 1.434    |

Inlet and outlet passage of tubular pump system is straight, of which internal flow pattern is steady, but the ratio of runaway speed and rated speed is not larger than that of vertical axial-flow pump system in the maximal water level difference. In order to study on the influence of different outlet passage on the runaway speed of pump, the shaft tubular pump was taken as a research object, and the runaway characteristics model was tested, the width and length of horizontal projection of two outlet passages are same. For shaft tubular pump, two outlet passages are respectively siphon outlet passage and straight tube outlet passage, which are shown in Fig.4. The test results are shown in Fig.5.

**Figure 4.** Shaft tubular pump system with different outlet passage

(a) Straight tube outlet passage
(b) Siphon outlet passage

**Figure 5.** The unit runaway speed of shaft tubular pump system with different outlet passages

The unit runaway speed of pump system with different outlet passages is different. In the same reverse head, the unit runaway speed of pump with siphon outlet passage is larger than that of pump with straight tube outlet passage, because the centerline of siphon outlet passage is longer than that of
straight tube outlet passage, the restriction effect of siphon outlet passage on water is slower than that of straight tube outlet passage, which consumes less total energy of water flow.

The magnitude of runaway speed of model pump is primarily depended on the structure of pump, characteristics of impeller and some friction, which includes the interaction of outer surface of impeller and water, stuffing and rolling bearings.

5. Conclusion
(1) For the runaway characteristics model test of the same pump system at the same blade angle, the unit runaway speed is not constant value in the different reverse head. With the decrease of reverse head, the reason is that resistance torque is not mathematical relation of fixed ratio. The actual runaway speed of prototype pump is safer, on the basis of the maximal runaway speed of model pump.
(2) The unit runaway speed of pump system with different outlet passage is different in the same reverse head, which shows that the unit runaway speed has a relationship with the type of pump.
(3) The runaway speed of the same pump is different at different blade angle. For tubular pump system, the ratio of runaway speed and rated speed has a little difference with that of vertical axial-flow pump system.

Nomenclature

- $n'_{\text{R}}$ — unit runaway speed of model test
- $n_R$ — runaway speed of model test
- $D$ — nominal diameter of impeller
- $H$ — experimental head
- $r_2$ — radius of bearing
- $r_3$ — radius of shaft
- $S$ — thickness of filler
- $\mu_1$ — sliding friction coefficient
- $\mu_2$ — friction coefficient of axial force
- $P_0$ — residual pressure of pump
- $\eta$ — absolute viscosity coefficient of lubricating oil
- $u$ — peripheral speed of the shaft
- $\delta$ — radial clearance of bearing

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