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Axial Hydraulic Thrust Caused by Pump Starting[1]

JOHN T. KEPHART, JR.[2] The authors of this paper have taken an interesting approach in combining the use of pump characteristics to compute transient hydraulic conditions and knowledge of steady-state thrust data to show that transient thrust may be predicted.

The results of the experiments described in this paper clearly show that thrust transients may be a problem when starting a pump which discharges into an empty system. It should be recognized that this is not an unusual situation for vertical pumps, but that it is probably more common than starting a vertical pump to discharge into a filled system.

It is agreed that equations (1) and (2) are valid for calculating the static head as the vertical section of the discharge line becomes filled after an empty start; however, these equations describe only a portion of the system behavior under the stated conditions, and do not represent system behavior when starting a pump with a filled discharge line. Nor do they represent system behavior when starting a pump with an air filled (otherwise empty) discharge line with a closed discharge valve. The inertial effects of the liquid in the discharge line must be accounted for in order to describe system behavior under these conditions.

Perhaps the following assumptions oversimplify the situation, but they may be reasonable:

1 Elastic effects may be ignored (influence of the pressure wave).

2 Pipeline friction may be ignored.

3 Friction at the orifice and discharge valve are lumped at the end of the discharge line.

4 Inertial effects of the liquid may be ignored until the discharge line is filled when starting a pump into an empty system with an open discharge valve.

With these assumptions, the authors' equations (1) and (2) may be used to calculate static head when starting into an empty system with an open discharge valve. Then pump head remains constant until the entire discharge line is filled. Then, as the authors state, head across the lumped friction at the end of the pipeline increases from essentially zero to a magnitude determined by the flow as in equation (5).

$$H_f = kQ^2 \tag{5}$$

In equation (5) H_f represents the head loss at the lumped friction and k is a constant determined by orifice size and extent of valve opening.

If H_s is taken as the static head (H in the authors' equation (1)), and H_p as the pump head, then the system head acting to influence the flow rate is given by equation (6)

$$H_t = H_s + H_f - H_p \tag{6}$$

and the discharge line flow, which from the pump characteristics then determines the value of pump head H_p, is governed by equation (7)

$$dQ = -\frac{gAH_t}{L}\,dt \tag{7}$$

in which L is the total discharge line length.

It would be interesting if the authors would utilize the suggested equations to complete the plot of computed points in Fig. 14. It would also appear possible then to compute points for comparison with the test points of Fig. 7(b), where equation (7) would govern for the entire transient, having a constant value of H_s equal to the vertical length of the discharge line.

In Fig. 6(a) with $S = 0$, the fact that pump shut-off head was not reached until a considerable time had elapsed after the pump reached rated speed suggests that air in the discharge line was being compressed (closed discharge valve) during the transient. For this case, the system head in equation (6) could be calculated by dropping H_f, the friction term, and substituting a term accounting for the air compression, H_c, as given in equation (8)

$$H_c = 10.4\left[\left(\frac{LA}{LA - \int Q dt}\right)^n - 1\right] \tag{8}$$

where the constant 10.4 is the absolute ambient atmospheric pressure in meters of water, and n is a polytropic constant, conventionally assumed as 1.2 in this type problem.

In conclusion, the authors have done an excellent experimental job. Moreover, the authors have shown the importance of accounting for the characteristics of the system in which a pump is to be applied. It is hoped that the recommendations in this discussion will enable even better prediction of the hydraulic transients and hence the transient thrust and its time duration. The duration as well as the magnitude of the starting thrust transient seems important in selecting a restraining device.

KEN-SHOU FANG.[3] As a serious student in seeking the nature of axial thrust of vertical turbine pumps for many years, this writer wishes to express his heartiest appreciation for the excellent experimental work conducted by the authors. He enjoyed the paper so much that he only asks for more after reading it. Lacking facilities to measure the thrust during the transient period and the pressure distribution around the impeller, this writer has drawn conclusions by analyzing the thrust data available to him. These conclusions can now be checked against the experimental results of the authors. The sketchy discussion on the upthrusting phenomenon during pump starting, as given in reference [6] is confirmed and greatly detailed by the authors' work. The momentum change in the flow through the impeller has been considered by many as the sole cause of upthrust. But calculated upward force due to this momentum change is

[1] By H. Miyashiro and K. Takada, published in the September, 1972, issue of the JOURNAL OF BASIC ENGINEERING, TRANS. ASME, Series D, Vol. 94, pp. 629–636.

[2] Senior Design Consultant, Engineering Department, E. I. du Pont de Nemours & Co., Inc., Wilmington, Del. Mem. ASME.

[3] Pearless Pump Division, FMC Corporation, Los Angeles, Calif.

always small in comparison with that measured by scale, load cell or other device. This leads one to a speculation that pressure distribution must also contribute the upthrust. Fig. 9(*a*) verifies this conclusion.

Following are a few comments about this paper:

1 Nearly all the tests were conducted using impellers with holes drilled inside the water channels. To begin with, impellers with balance holes are not common. And to drill holes inside the water channel appears strange, as there is apparently no practical usage for this kind of impeller. Therefore, the purpose of this paper is apparently obscured by the use of an odd impeller configuration.

2 The authors did not show formulas for the thrust calculations, nor did they state that their findings would be applicable to all kinds of centrifugzl pumps. Since thrust depends a great deal on the specific speed of the pump, caution should be exercised in extending the results to pumps with specific speeds far different from those used in the experiments.

3 The third sentence of the paper, "This upward thrust is not observed under steady operating conditions," is not a true statement. Fig. 6 shows upthrust under a steady condition. And field observation of continuous upthrusting is rather common.

4 Referring to Fig. 9(*b*), can the authors provide some explanation of why the pressure in the balance chamber is lower than that at the impeller eye?

Authors' Closure

The authors wish to thank Messrs. Fang and Kephart for their discussions. The authors have the following opinions on four valuable comments from Mr. Fang:

1 Impellers with balance holes are widely used, since the balance hole is one of simple and practical methods of balancing the axial hydraulic thrust. The balance holes are drilled usually outside the water channels of the impeller, unless structural restrictions compell designers to drill the holes inside the water channels.

2 In this paper the authors do not intend to present formulas for calculating axial thrust under steady operating conditions. They try to calculate the transient axial thrust from the axial thrust measured under steady operating conditions. The authors agree with Mr. Fang that the axial thrust depends on the specific speed of pumps.

3 As Mr. Fang points out, the expression of the third sentence of the paper is not correct. The upward thrust is observed in the high discharge range even under steady operating conditions.

4 The difference between the pressure in the balance chamber and that at the impeller eye may be related with the pressure distribution in the impeller. However, the authors have not investigated this relation.

The hydraulic transients are better predicted by Mr. Kephart's adequate suggestions. Equations (5), (6), and (7) suggested by him enable to calculate transient conditions after the empty discharge pipe was filled with water. In case of the closed discharge valve the effect of the air compression in the empty pipe has to be taken into account, as Mr. Kephart points out.