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SUMMARY

A study has been completed to determine the magnitude of the velocities existing in the various slurry pumps currently in use and to determine possible methods of reducing the velocities in order to obtain longer life.

The maximum velocities of the 100A, 200A and 300A pumps are between 39 and 53.6 fps with the exception of the 100A absolute discharge velocity which is 80.6 fps.

For large slurry pumps, it is recommended that lower speeds be used in order to reduce the impeller relative velocities.

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In order to build long lived slurry pumps, the erosion of the hydraulic parts must be reduced. It has been shown that the erosion rate in slurry is proportional to the square of the velocity, and therefore one method of reducing the erosion is by reducing the velocity.

Recent experience indicates that considerably lower attack rates are obtained with Zircaloy II impellers, but for extremely long life, further improvements will probably be necessary.

A velocity survey has been completed on several pumps used to handle slurry to determine the velocity range in which the pumps are being run and to determine possible methods of reducing the velocity.

PROCEDURE

There are several factors which determine the velocities within an impeller. At the pump suction, the relative velocity of the slurry with the leading edge of the vane is the vector sum of the radial fluid velocity and the tangential velocity of the vane leading edge. The inlet velocity is therefore determined by the capacity, the impeller suction diameter, and the pump speed.

$$W_1 = \sqrt{C_{m1}^2 + U_1^2} \quad \text{neglecting prerotation}$$

For most normal impeller designs, the change in radial velocity from the inlet to the outlet is small, and the radial component of velocity can be assumed to be the axial velocity in the impeller suction.

The exit velocities are determined by the head, capacity, pump speed, discharge vane angle, and the width of the impeller channel. The velocities of either an existing impeller design or a prospective design may be calculated using the method outlined by Stepanoff¹. For an existing impeller, the dimensionless specific speed (W_s) can be calculated from the performance specific speed (N_s) and the impeller width to diameter ratio (b_2/D).

$$W_s = \frac{N_s}{9675(b_2/D)^{1/2}}$$

A plot of various values of W_s can be made vs. head coefficient ($x = Hg/U_2^2$) and capacity coefficient ($\phi = C_{m2}/U_2$).

$$W_s = \frac{\phi^{1/2}}{x^{3/4}}$$

The intersection of the appropriate W_s curve and the appropriate discharge angle line determines a point representing the design point of the impeller. From this point, a vector diagram of the actual impeller exit velocities can be constructed as shown in Figure 1, with the length of the head coefficient line representing the tangential velocity of the impeller outside diameter.

For a new impeller design, the head and capacity coefficients may be obtained from Figure 9.15 of Stepanoff². These values can then be plotted on Figure 1 and the velocity vector diagram drawn as above. The value of the tangential velocity (U_2) can be calculated from the head coefficient and the desired head at the design point.

$$U_2 = \sqrt{Hg/x}$$

FIGURE 1

DISCHARGE VELOCITY DIAGRAM

DISCUSSION OF RESULTS

It is difficult to draw any specific conclusions on the erosion of the existing pump because of the wide variation in slurries, concentrations, and operating conditions. A visual examination of 100A, 200A, and 300A impellers when compared to Table I does seem to show a trend of increasing erosion at the vane inlet with increasing velocity. The same trend does not seem to exist at the vane exit, possibly because inactive or turbulent areas in the flow passages of the 300A impeller result in local velocities higher than those calculated. If impellers of optimum hydraulic design were available, the attack rate would probably be a function of velocity.

Table II illustrates the advantage of going to a lower speed pump. The relative velocity at the inlet is reduced considerably due to the decrease in tangential velocity in direct proportion to the pump speed. The relative discharge velocity is also reduced significantly at the lower pump speeds. The relative discharge velocities are also reduced by increasing the vane discharge angle, but this results in higher absolute velocities which would increase the casing problem. Since the discharge angle controls the shape of the head-capacity curve, and since pump manufacturers seem to prefer the lower angles (20° - 30°), it is uncertain whether the vane angle should be used to reduce the impeller relative velocity. It is interesting to note that the relative and absolute discharge velocities in the lower speed pump shown in Table II would be balanced at a vane angle slightly above 20° .

At higher specific speeds (higher RPM at fixed head and capacity), the total pump head is developed more by vane reaction and less by conversion of absolute discharge velocity which results in higher relative velocities and lower absolute velocities. This trend is not clearly illustrated in Table II because both pumps have relatively low specific speeds, but a comparison of the B-J 4000 pump and the Reliance 6000 pump (specific speeds of 2320 and 6770, respectively) in Table I will give a fairly good illustration.

RECOMMENDATIONS AND CONCLUSIONS

The following general observations were made in completing the above velocity study:

1. For a given head and capacity, lower relative velocities will exist at the impeller inlet and discharge at lower pump speeds.
2. For a given head and capacity, the absolute velocity will decrease with increasing pump speed. This is significant only at higher values of specific speed.
3. The vane discharge angle can be varied to control the relative and absolute discharge velocities to a limited extent.
4. The vane inlet velocity can be minimized by properly sizing the suction diameter of the impeller, but the total effect on the pump performance is uncertain at this time.

There is a clear advantage in operating slurry pumps at the lower speeds in regard to the erosion of the hydraulic parts. However, the lower pump speed should not be used as a substitute for proper hydraulic design since there is evidence that highly turbulent flow areas are as erosive as high velocity areas.

TABLE I

<u>Pump</u>	<u>Head Ft.</u>	<u>Relative Velocity at Vane Inlet ft/sec</u>	<u>Relative Velocity at Vane Discharge ft/sec</u>	<u>Absolute Velocity at Vane Discharge ft/sec</u>	<u>RPM</u>
100A	190	35.6	53.6	80.6	3550
200A	100	41.6	42.6	47.5	"
300A	67	48.6	39.0	40.2	"
B-J 4000 12-in. Impeller	144	69.7	44.2	53.4	1750
Reliance 6000	135	121	89.0	45.4	3450

TABLE II

3450 RPM

<u>B₂</u>	<u>Relative Velocity at Vane Inlet ft/sec</u>	<u>Relative Velocity at Vane Discharge ft/sec</u>	<u>Absolute Velocity at Vane Discharge ft/sec</u>
20	52.7	55.7	38.2
30	"	48.5	41.4
40	"	44.4	44.2
45	"	42.6	45.2

1750 RPM

20	29.2	41.2	40.8
30	"	37.0	42.6
40	"	34.7	44.1
45	"	33.6	44.6

The disadvantages of the lower speed pumps are: (1) higher initial cost due to the larger physical size of the pump and motor, and (2) the bearing film thickness for a given size bearing will be lower for the lower speeds. The bearing problem does not seem too important, however, since the larger size of the motor will permit the use of larger bearings. There is also the possibility that over a long period of time the higher initial cost can be offset by lower operating costs due to a higher overall efficiency. For example, the overall efficiency obtained from a 4000 gpm 100 ft. head pump at 1800 RPM would possibly be 5-7% higher than could be obtained at 3600 RPM. This is possible because of a higher motor efficiency³ and because the 1800 RPM unit would be approximately at the maximum efficiency range of specific speed⁴.

In view of the above information, lower speed pumps are recommended for large slurry pumps to be used in full scale reactor plants. It may be advantageous to base the speed selection on both velocity and economic considerations.

For test loops where extremely long life is not required, 3600 RPM pumps will probably give satisfactory service using Zircaloy II hydraulic parts.

NOMENCLATURE

b_2	Impeller width at discharge, ft.
C_1	Absolute fluid velocity at inlet, fps
C_2	" " " " exit, fps
C_{m1}	Radial component of fluid velocity at inlet, fps
C_{m2}	" " " " " exit, fps
D	Outside diameter, ft.
g	Gravitational constant
H	Head, ft.
N	Pump speed, RPM
n_s	Performance specific speed, $n_s = N \sqrt{Q} / H^{3/4}$
Q	Capacity, gpm
U_1	Impeller tangential velocity at inlet, fps
U_2	" " " " exit, fps
W_1	Relative velocity between impeller and fluid at inlet, fps
W_2	" " " " " exit, fps
β_2	Vane discharge angle
ϕ	Capacity coefficient $\phi = C_{m2} / U_2$
x	Head coefficient $x = \frac{Hg}{U_2^2}$
W_s	Dimensionless specific speed $W_s = \phi^{1/2} / x^{3/4}$

APPENDIX: Velocity Calculation for 400 GPM 100 Ft. Head Pump

The performance specific speed is calculated from the desired head, capacity, and pump speeds.

$$\text{At 1750 RPM} \quad n_s = N \sqrt{Q/H^{3/4}} = \frac{1750 \sqrt{400}}{100^{3/4}} = 1100$$

$$\text{At 3450 RPM} \quad n_s = 2170$$

Values of ϕ and x for each specific speed and vane discharge angle are obtained from Figure 9.15, p. 176 of Ref. 4 and listed in Table III. The value of the impeller tangential velocity (U_2) can be calculated for each value of x .

$$x = Hg/U_2^2$$

$$U_2 = \sqrt{Hg/x} = \sqrt{\frac{100 \times 32.2}{x}} = 56.7/\sqrt{x}$$

The values of ϕ and x are then plotted on Figure 2, and the exit velocity diagram is completed. The values of the relative and absolute velocities listed in Table IV are calculated by multiplying the tangential velocity by the relative length of the appropriate velocity vector.

TABLE III

β_2	$n_s = 1100$		$n_s = 2170$		U_2	
	x	ϕ	x	ϕ	1750	3450
20	.497	.098	.40	.15	80.5	89.9
30	.537	.125	.455	.193	77.5	84.0
40	.565	.146	.499	.229	75.5	80.2
45	.578	.156	.518	.243	74.6	78.8

TABLE IV

β_2	l_c	1750 RPM		Relative Discharge Velocity	
		$C_{2=1} U_2$	l_w	$W_{2=1} U_2$	
20	.507	40.8	.512	41.2	
30	.55	42.6	.478	37.0	
40	.585	44.1	.46	34.7	
45	.598	44.6	.45	33.6	
β_2	l_c	3450 RPM		Relative Discharge Velocity	
		$C_{2=1} U_2$	l_w	$W_{2=1} U_2$	
20	.425	38.2	.62	55.7	
30	.493	41.4	.577	48.5	
40	.55	44.2	.553	44.4	
45	.573	45.2	.54	42.6	

Values of l_c and l_w are obtained from Figure 2.

REFERENCES

1. A. J. Stepanoff, "Centrifugal and Axial Flow Pumps," 2nd Ed., John Wiley and Sons; pp 172-176.
2. Ibid, p 176.
3. W. M. Wepfer, "A Guide to the Selection of Canned Motor Pumps," 1958 Nuclear Engineering and Science Conference, Preprint 77, Session II.
4. A. J. Stepanoff, "Centrifugal and Axial Flow Pumps," 2nd Ed., John Wiley and Sons, p 76.

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