EFFECT OF MECHANICAL UNBALANCE ON VIBRATIONS, FORCES AND RELIABILITY OF A SINGLE-STAGE CENTRIFUGAL PUMP

by

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ABSTRACT

The effect of rotor mechanical unbalance on pump vibrations was tested for a typical ANSI pump. It was found that significant damping of vibrations occurs at actual (wet) operations as compared to dry run. Rotor forces and deflections were also studied and determined to be small, compared to hydraulic unbalance for a wide range of unbalances. Shaft deflections were confirmed by testing a range of flows from runout to shutoff, to be within allowable limits, as governed by applicable industrial specifications.

BACKGROUND

In the recent years, much attention has been drawn by pumps users and manufacturers to rotor (and component) balancing. It is commonly perceived that the better the balance, the lower are the vibrations, forces, and deflections, leading to better pump reliability and longer life. Indeed, it is known, that for grossly unbalanced rotors, the vibrations are high and noticeable even without special measuring equipment. The first reaction usually is to attempt to balance the rotor to as fine as possible, to eliminate the problem. However, upon closer data analysis, it becomes obvious that a point of diminishing return may be quickly reached. Other factors, such as supports stiffness, hydraulic unbalance, and others also have an effect on vibrations.

Several internationally recognized specifications cover the methods and quality of balancing. ISO 1940/1 specification [1] presents various "G-grades" of allowable unbalance for different types of rotating machinery. For pumps, a standard value has been G6.3. The "G" grade designations represent the magnitude of the product (e_{per} x ω)—permissible rotor eccentricity times rotational speed expressed in mm/sec., i.e., if the product of e_{per} x ω is 6.3 mm/sec., the balance quality grade is G6.3. Finer levels are often required by users, such as G2.5 for fan pumps in paper industry, or 4W/N (equivalent to G0.67) by API 610, W = rotor weight and N = speed, 7th Edition [2].

ISO recognizes and states the difficulties associated in balancing below G1.0 level, since such factors as shaft and bearing face roundness become important. At G0.4, ISO actually states in a subnote, that such fine balance can be achieved only in a pump's own housing and bearings, at operating conditions (and temperatures!) of a particular application.

INTRODUCTION

The objective of this work was to correlate the level of unbalance to pump vibrations, measured at the bearing housing, at inboard and outboard bearing locations, in vertical and horizontal directions. Axial vibrations were also measured, but not discussed here, since they have not shown correlation to unbalance, which causes radial, rather than axial, forces on the rotor.

The test pump was a standard ANSI single stage overhung open impeller design, as shown on Figure 1.

At 3550 rpm, the pump's best efficiency point (bep) was 275 gpm, producing 275 feet of head. This pump was mounted on a

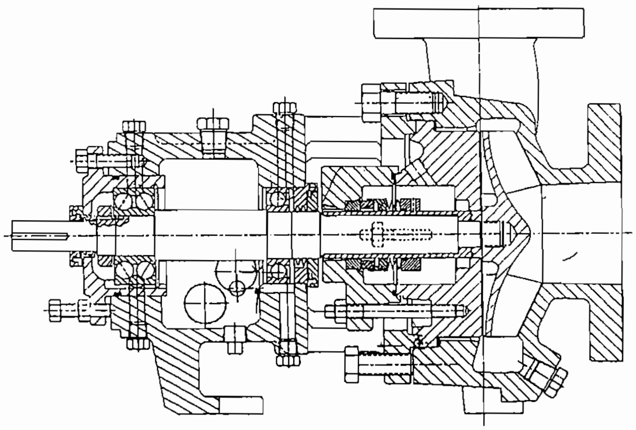


Figure 1. ANSI Single Stage Pump (2 in discharge, 3 in suction)

heavy base which was sitting on solid foundation, as shown on Figure 2 and Figure 3.

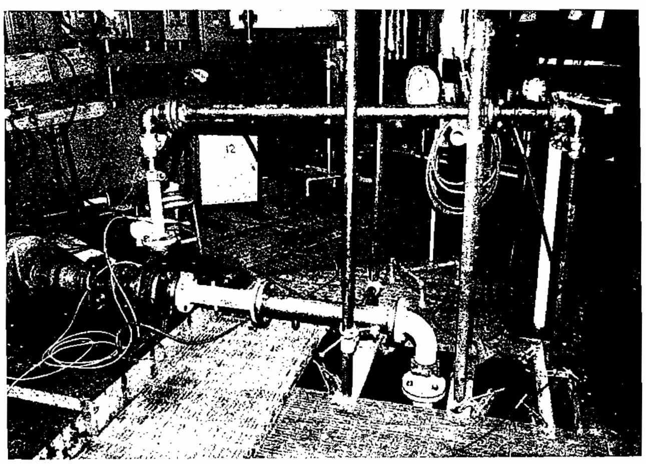


Figure 2. Test Pump, Mounted on Heavy Bedplate.

The impeller was then balanced on an arbor to produce low residual unbalance of the rotor (G0.9).

By removing one or several screws, a known rotor unbalance could be simulated, resulting in the following six tested G-values: 18.0, 13.8, 10.2, 6.2, 2.3, 0.9, using ISO formula:

$$\frac{6.015 \times G}{rpm} = \frac{oz.\ in}{lb_{ROTOR}}$$

No pump disassembly was done between the tests. The access to screws was through a special opening in the casing, allowing to remove the screws without pump disturbance, as shown on Figure 4.

VIBRATIONS

Dry Operation

The effect is shown in Figure 5 of rotor unbalance on overall unfiltered vibrations, as imposed by varying number of screws in impeller blade.

"Averaged" values are shown in Figure 6 of above vibrations from Figure 5 for both bearings and for both directions. The averaged value is used in the rest of the presentation, as being most illustrative of a trend in vibrations *vs* unbalance.

The vibration signature was filtered at running speed (~60hz), to isolate the effect of unbalance from other mechanical effects, such as casing components and supports contributions, ball bearings effects and others. This is also shown on Figure 6.

The filtered (1×) lower curve of Figure 6 looks somewhat undramatic. The filtered 1×, IB horizontal (to take the foundation out-of-play) might be more representative of unbalance force effect. It was not surprising that the horizontal vibrations were highest in Figure 5. even though unfiltered, i.e., unbalance is at synchronous 1×.

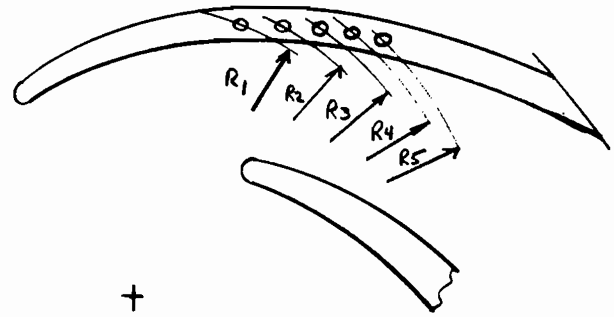


Figure 3. (5) 10-24 1/2 in Screws (0.040 oz Each/Set Axially into Impeller Blade.

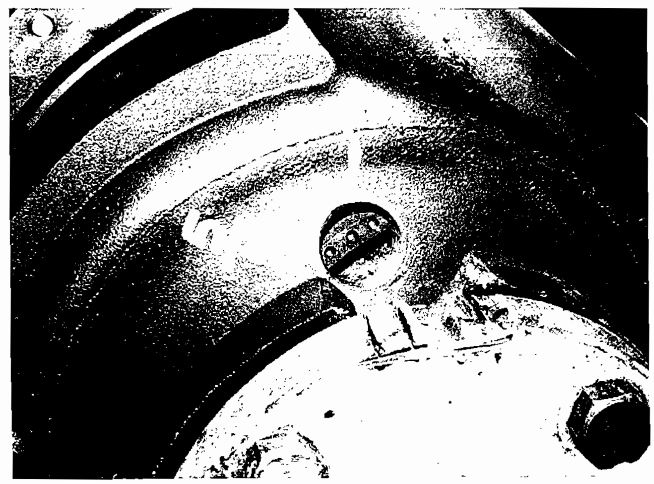


Figure 4. Special Access Hole in Casing for Screw Change.

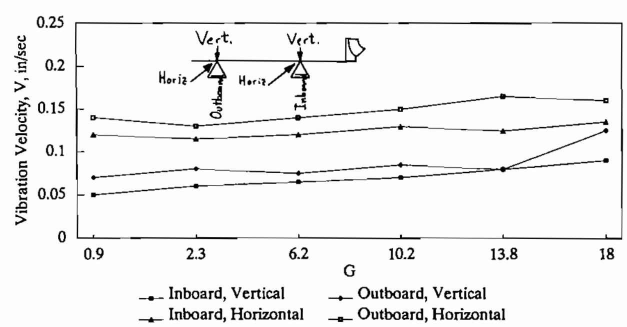


Figure 5. Dry Operation. Effect of unbalance (G) on vibrations (unfiltered), shown for each bearing at each plane (direction).

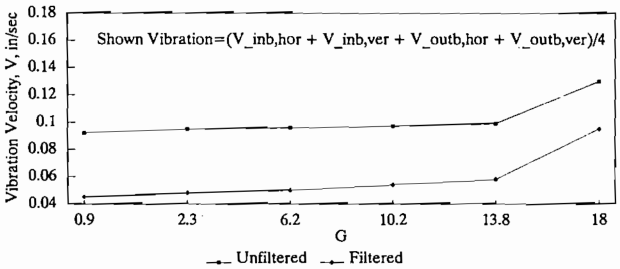


Figure 6. Dry Operation. Effect of unbalance (G) on averaged *unfiltered vibration.*

Wet Operation

A series of tests was performed at different flow rates, namely: 75 gpm (27 percent bep); 150 gpm (55 percent bep); 225 gpm (87 percent bep); 300 gpm (109 percent bep); 360 gpm (131 percent bep). The resultant averaged unfiltered vibrations are plotted *vs* unbalance on Figure 7.

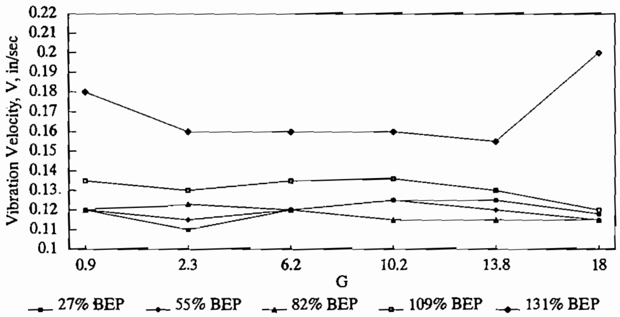


Figure 7. Wet Operation. Effect of unbalance (G) on averaged *vibration, at different flowrates.*

Filtered vibrations (at ~60hz running speed) are plotted on Figure 8 for 109 percent flow. Unfiltered vibrations for the same flow is also shown on the same figure for comparison.

INTERPRETATION OF VIBRATION DATA

Tests were performed starting at levels of unbalance significantly greater than maximum allowable by ISO and API specifications. The highest tested unbalance was G18.0, which is almost three times greater than a typically accepted G6.3 of ISO specification for pumps, or 30 times greater then API 610, 7th Edition level.

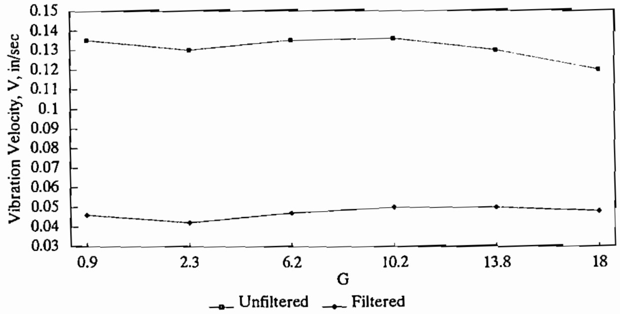


Figure 8. Comparison Between Unfiltered and Filtered Vibrations at Wet Operation at 109 percent Flow.

As shown on Figure 6 for the case of dry operation, the level of vibration was quickly reduced from approximately 0.13 in/sec. at G18.0, to 0.10 at approximately G14, and essentially levelled-off from there all the way to G0.9.

The filtered vibration level stayed approximately half of unfiltered, following similar trend as unfiltered.

When the pump operated in wet condition, (Figure 7), the level of vibration stayed relatively constant at 0.12 in/sec at all tested values of unbalance, and for all flows except high flow. Interestingly, the high rise in vibrations experienced at unbalance greater the G14 (at dry running, Figure 6) was eliminated when operating in wet conditions, apparently due to liquid damping effect.

When flow significantly exceeded the bep point, the vibration levels increased rapidly, indicating the hydraulic nature of additive unbalance effect at high flows.

Overall, the data indicated that for the type of the pump tested (ANSI), the "betterment" of unbalance below approximately G14.0 has little effect on vibrations. Certainly, a generally accepted value of G6.3 should be a conservative target for this type of pumps.

However, unbalance greater than approximately G14 leads to a rapid increase in vibrations. This confirms the experiences of many users, that the *grossly* unbalanced rotors lead to high vibration, but at certain G-level, a further betterment of unbalance may not produce measurable vibration reduction.

RADIAL FORCES AND DEFLECTIONS

The pressure distribution around impeller periphery results in unbalanced radial force except at bep, theoretically. This force consists of steady and fluctuating components. The issue of radial volute pressure distribution, and radial force associated with it, is well documented in literature, such as Agostinelli, et al. [3]. What is not documented sufficiently is the resulting shaft deflections, with respect to the allowable limits on these deflections.

A series of proximity probes were installed in the same pump, as described earlier, as shown on Figure 9.

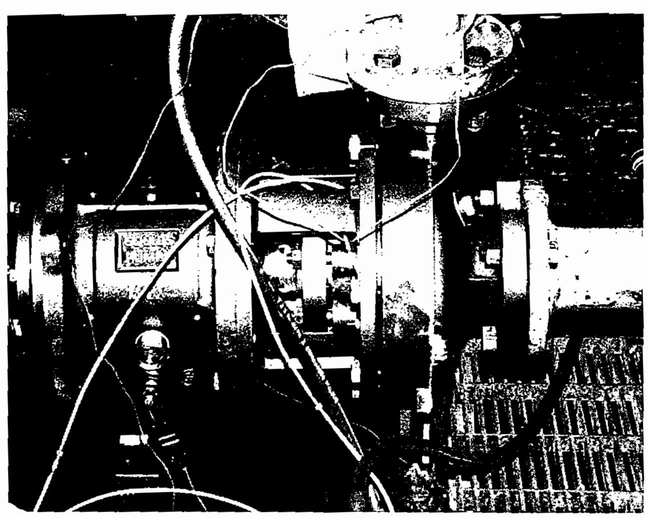


Figure 9. Pump Instrumented with Proximity Probes for Deflections and Forces Test.

Prior to testing, a calibration procedure was devised to calculate deflection of the rotor at impeller centerline as a function of applied static load, using weight, as shown on Figure 10.

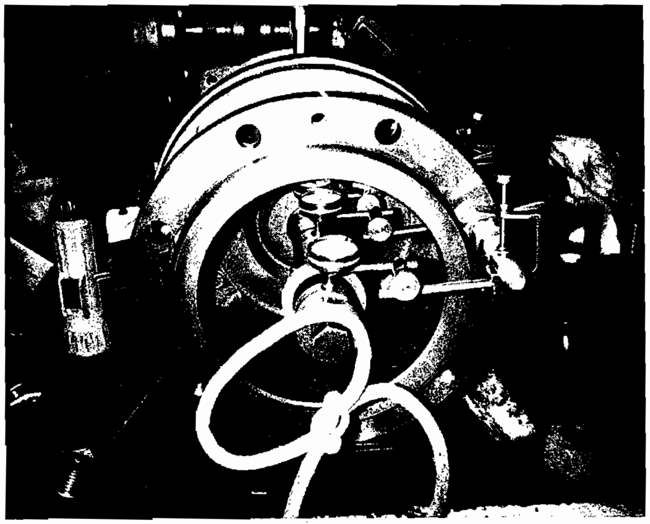


Figure 10. Setup for Calibrating Deflections at Impeller Centerline Vs *Radial Load.*

To eliminate nonlinearities associated with free sagging of the rotor due to clearances in the bearings, a "zero-load" force, applied by hand was imposed, and these deflection (2.0 mils at impeller), were assumed to be a result of free movement of the rotor towards the bottom of bearing clearance. Several other loads were imposed (100 lb, 200 lb, and 300 lb) and the resulting rotor positions are illustrated in Figure 11. Dial indicators were installed at five locations along impeller shaft, to establish shaft shape for different impeller static loads. At the same time, two proximity probes were also installed to read deflection of the shaft near seal face location for horizontal and vertical planes. The readings of these probes were used to relate to forces during pump actual running test.

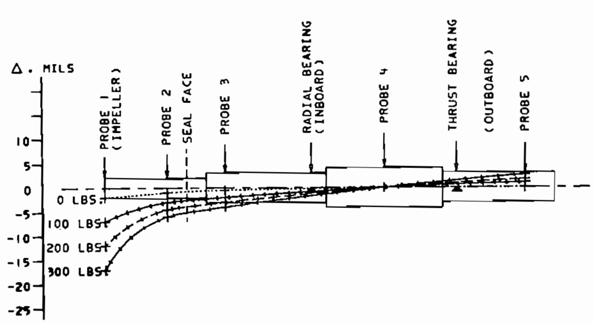


Figure 11. Shaft Deflection Shapes at Different Static Loads Imposed at Impeller Centerline Location.

Having established correlations to determine radial load via direct reading of shaft deflections at the seal, the pump was then run at several flows and the radial load magnitude and direction was plotted against flow (Figures 12 and 13). Then, using the correlation from Figure 11, a shaft deflection at the impeller centerline and at the seal face were calculated, Figure 14.

An allowable minimum continuous recommended flow (MCF) for this pump is 60 gpm. At this flow, shaft deflection at impeller centerline is 4.8 mils, and at the seal face, 1.9 mils. The 4.8 mils is less than allowable 5.0 mils per ANSI specification [4]. Maximum deflection at the impeller is specified by the latest version of ANSI specification; however, a previous revision of

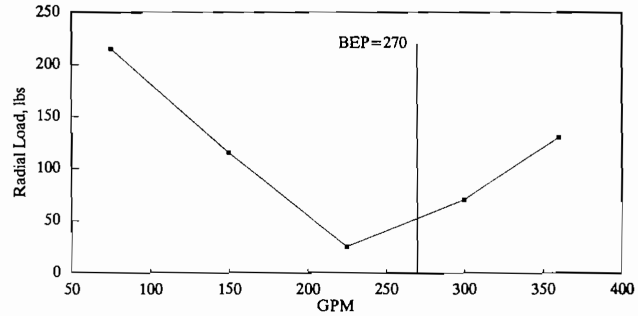


Figure 12. Radial Load Magnitude Vs *Flow.*

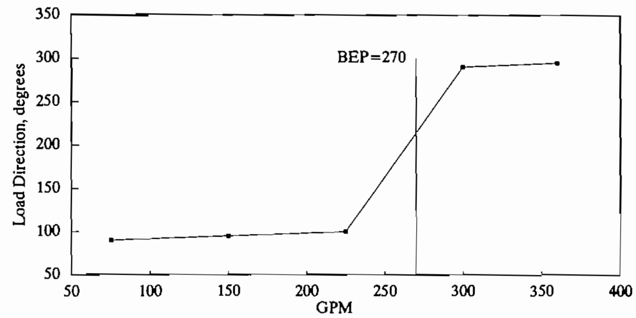


Figure 13. Radial Load Direction Vs *Flow.*

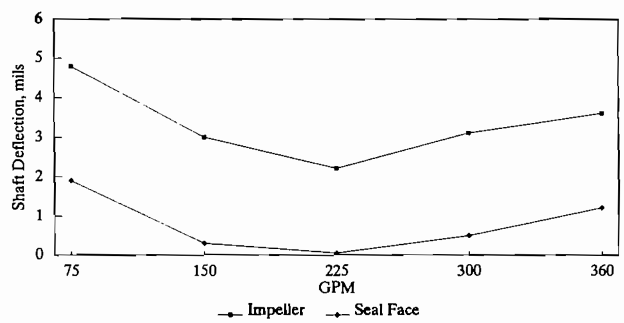


Figure 14. Shaft Deflection at Impeller Centerline and Seal Face.

ANSI specification had maximum allowable deflection of the seal face (2.0 mils). In either case, the pump is in compliance with either revision of ANSI specifications.

SIMPLIFIED CALCULATIONS FOR SHAFT DEFLECTIONS

It was of interest to compare the test results for the shaft deflections with a simplified calculation method. If close agreement was found, such simplified method could be a useful tool for designers to compare shaft deflections for similar pump designs, at least within some reasonable tolerance. A simplified model and calculations are shown in Figure 15.

This compares closely with tested 1.9 mils. The difference may be attributed to simplifications in bearing stiffness and shaft geometry.

LOADS COMPARISON: HYDRAULIC *VS* UNBALANCE

The mechanical unbalance results in the following centrifugal force:

$$F = \frac{m\omega^2 R}{g} = \frac{mR\left(\frac{\pi N}{30}\right)^2}{g} = \underset{(\text{lb} \times \text{ft})}{mR \times 4320} = \underset{(\text{oz} \times \text{in})}{22.5 \times U}$$

Where m = mass (weight, lbs.) of unbalance, R = radius location of unbalance (inches).

The impeller weight is 6.0 lb, and the shaft is 12 lb, and the total rotor weight is then 18 lb. For a typical allowable unbalance value of G6.3, this translates into:

$$\frac{6.015 \times 6.3}{3550} \times 18 = 0.192 \text{ oz} \times \text{in of allowable unbalance}$$

Of that value, only impeller weight participates in causing centrifugal force of unbalance. In other words, actual unbalance is less, by the ratio of the impeller weight to rotor weight:

$$0.192 \times \frac{6}{18} = 0.064 \text{ oz. in, which results}$$

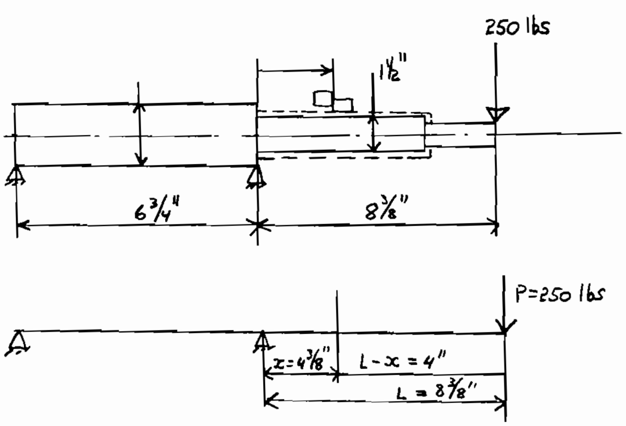
in force F = 22.5 × 0.064 = 1.44 lbs, which represents

$$\frac{1.44}{250} \times 100\% \approx 0.6\%, \text{ as compared to hydraulic force,}$$

which is small comparing to 250 lb hydraulic force at minimum flow, or even at bep flow (25 lb).

Even a possibility of impeller offset in allowable clearance does not add significantly to unbalance force. For example, for 1.0 mil radial offset in clearance, the unbalance is (6 × 16) × 0.001 = 0.1 oz × in, and the force is 22.5 × 0.1 = *2.3 lb.*

It should be noted briefly that the balancing arbor dimensions should be very close to those of a shaft, resulting in negating any fit clearance effect. For other types of pumps, such as key driven, the same dimensional tolerances should apply to the key and arbor of the balancing machine, as to pump shaft and impeller, resulting in practically neglible unbalance, due to such offsets.



$$y_{@SEAL} = \frac{P\chi^2}{6EI}(3L - \chi),\ I = \frac{\pi d^4}{64} = \frac{\pi \times 1.5^4}{64} = 0.25 \text{ in}^4,$$

$$E = 30E6 \text{ psi (carb. steel, 4140)}\ y_{@SEAL} = \frac{250 \times 4.375^2}{6 \times 30E6 \times 0.25}$$

$$(3 \times 8.38 - 4.38) = 0.0022 \text{ in} = 2.2 \text{ mils}$$

Figure 15. Simplified Method for Shaft Deflection Estimate.

USER PERSPECTIVE OF UNBALANCE SIGNIFICANCE AND HYDRAULIC EFFECTS

It is important to understand the current and upcoming updates of the industrial specifications governing balance issue, along with user practices. Many users feel that the hydraulic effects are the most important factors, as relating to rotor deflection, vibrations and reliability. Some of these effects are reviewed in this section. A summary of such hydraulic concerns is shown of Figure 16 [5]: Hydraulic Considerations for Centrifugal Pumps (after Gopalakrishnan, et al. [5])

Many feel that cavitation, temperature rise, bearing selection, seal selection, material, and operation far from bep are well discussed and fairly straightforward. However, there are some significant factors that require more attention: radial thrust, suction/discharge recirculation, balance procedure and standards, impeller fitting, coupling selection and fitting, and excessive deflections at the seal faces. These will be discussed briefly to put each in some perspective, specifically to unbalance.

Stable Region of Operation

It is known that pumps operate most stable (i.e., low vibrations, longer life), when operating near the bep. Most operations people want to operate in that stable region. However, in practice, when the pump must operate under varying flow conditions in response to a system demand, these guidelines are violated. In attempting to qualify and quantify the vibration acceptance levels, a proposed upcoming Eighth Edition of API 610 will include Figure 17.

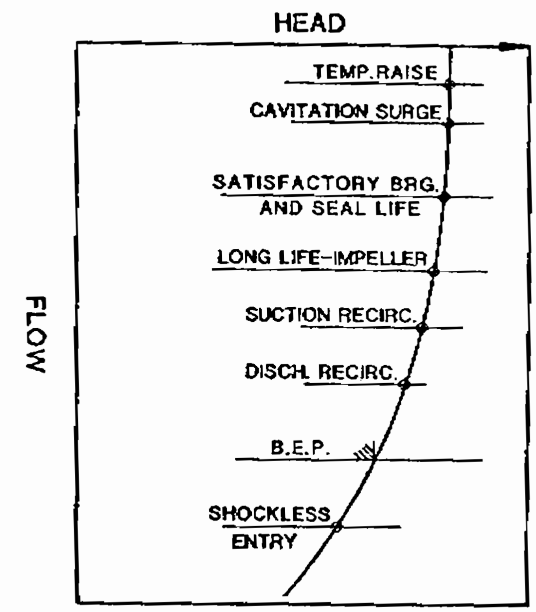


Figure 16. Hydraulic Considerations for Centrifugal Pumps (after Gopalakrishnan, et al.) [5].

Below a certain flow, usually to the left of bep, one must consider an important hydraulic phenomenon, namely suction (and discharge) recirculation. It was found by experience of many users, that above certain value of suction specific speed, these effects become very pronounced. The suction specific speed is defined as:

$$N_{SS} = \frac{\text{rpm} \times \sqrt{Q}}{\text{NPSH}_R^{0.75}},$$

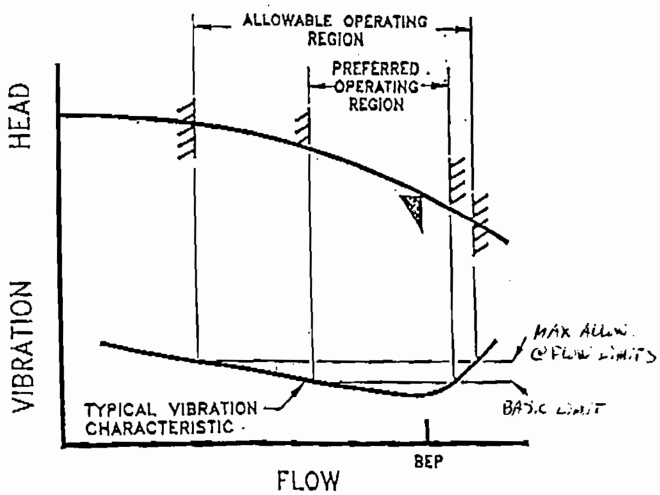


Figure 17. Proper Regions of Operation; Vibrations Vs *Flow (flow proposed API 610, Eighth Edition, [6].*

where

Q = flow, gpm (half total for double suction pumps)

$NPSH_R$ = required NPSH at BEP, maximum diameter.

The users self-imposed limits on Nss vary from 9000 to 12000, depending on particular users experience. Pump manufacturers recognize these concerns and developed internally used relationships, charts and formulas, to relate suction specific speed, flow, power, and other factors. An example of such chart is shown on Figure 18 [7], and A, B, C designate different pump sizes, i.e., different power levels.

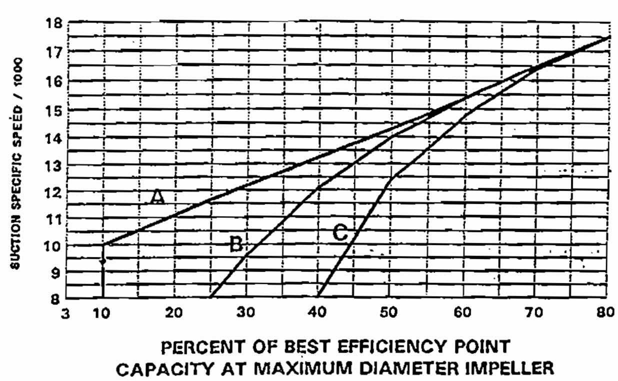


Figure 18. Minimum Stable Flow Vs *Suction Specific Speed [7].*

Radial Thrust

Another strong concern to relate to unbalance has been already evaluated in an earlier section. That concern is radial thrust. This has been a misnomer for years, as it refers to a radial load, pounds, (not thrust) transverse to the shaft at the impeller due to the differential pressures that the impeller sees in the *projected profile area of the impeller, i.e., diameter times effective width.* One way to understand this condition is to visualize the volute as having a series of pressure gauges tapped into volute OD at various spots in the periphery. There is only one flow-head condition (= bep), where equal pressures would exist at all points in the volute, thus resulting in theoretically zero load. A diagrammatic presentation is shown in Figure 19 taken from internal training material at a Gulf Coast petrochemical plant in 1972 [8].

As the discharge flow is throttled back towards "shutoff" (or the system head curve increases against the pump), the pressures near the discharge will be greater than the pressures on the opposite side of the pump volute. Conversely, if the pump is operated past the bep point, a radial load in the opposite vector direction would develop.

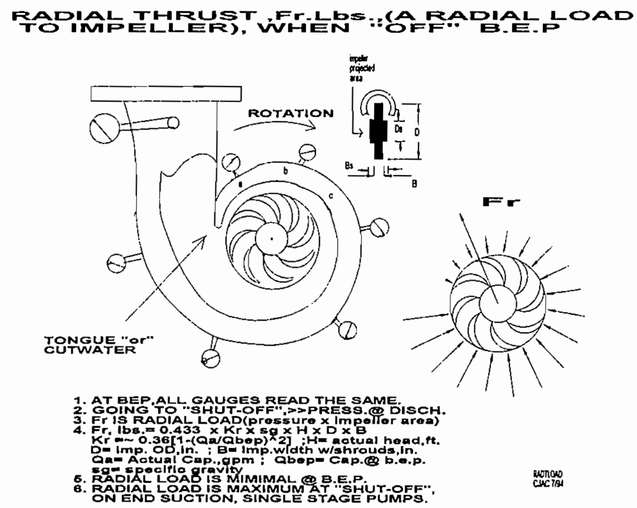


Figure 19. Concept of Radial Thrust Loads to an Impeller and Shaft (after C. Jackson, [8]).

This radial force can be determined as:

$$F_R = 0.433 \times K_R \times SG \times H \times D_2 \times b_2, \text{ where}$$

SG = specific gravity

H = pump head, ft

D_2 = outside diameter of impeller, in

b_2 = width of impeller at discharge, including shrouds, in

K_R = experimental coefficient, which depends on percent flow and type of the volute.

As can be seen from the formula, this radial force is smaller for lighter liquids, e.g., pumping chemical with SG = 0.8 would result in 20 percent lower force than on water. For conventional single volute, Stepanoff [9] gives $K_R = 0.36\ [1 - (Q/Q_{BEP})^2]$, whereas for circular volutes, $K_R = 0.36\ Q/Q_n$, i.e., circular volutes reduce the radial thrust significantly at the off peak flows.

The pump designer has several ways of reducing this unbalance of forces acting vectorially on the impeller. Double voluting or twin voluting can be introduced into the design wherein two tongues or "cutwaters" are oriented 180 degrees out of phase across the volute configuration. In this manner, unbalanced forces are cancelled out by similar (out-of-phase) forces.

As the reader can see, there is a similarity between the hydraulic radial force, and centrifugal force of unbalance—both cause shaft deflection, resulting in reduced components life, [10] vibrations, etc. However, several studies have shown, including this one, that the magnitude of hydraulic force can be five to ten times greater than the force of unbalance.

The shock loads (Figure 16) coming from too close a proximity of the cutwater to impeller flow emitting from each vane can be reduced by applying the proper Gap "B" between vane and cutwater tongue. Some recommendations call for $(D_{cw}-D_2)/D_2 >$ 6.0 percent to be used as Gap "B." This effect is more severe in high energy pumps, i.e., 250 to 300 hp per impeller.

The special design of the suction approach to the impeller, along with impeller inlets, can further improve pump operation, in terms of noise and vibrations.

Balancing: Experiences and Specifications

The described testing program deals with an ANSI pump. While the conclusions are, therefore, limited to these types of pumps, it is, nevertheless, possible to extend some conclusions beyond ANSI, into an API class of pumps.

In this regard, the authors felt helpful to touch on an apparent controversy or lack of consensus among pumping community, with regard not as much *method* of balancing, as the actual *values* of allowable unbalance, to be used in appropriate specifications formulas.

As the spirit of API 610, 7th Edition covers mostly balancing of *rotors* per infamous formula 4W/N, the *component* balancing criteria is unclear. Is W a component or journal reaction weight, which automatically refers to a rotor? And if so, which reaction-balancing machine or assembled pump? And, what about overhung pumps—which journal?

Hopefully, the 8th Edition will clarify this confusion once and for all.

In the meantime, the authors wanted to present the unbalance criteria in both ways—using either the rotor weight and impeller weight, so that the reader can pick and choose whatever criteria with which he or she is most comfortable.

This pump was unbalanced to six ISO Grades - 0.9, 2.3, 6.2, 10.2, 13.8, and 18 (mm/sec). These values can be seen in a different form of expression in Table 1. In light of the previous discussion, the comparison of residual unbalance is shown at 6.0 lb impeller and 18 lb rotor. In addition, the eccentricities are shown in mm and micro-inch (μin.). One should understand that an ISO chart has grades (e ω = mm/sec.). If a person divides the grade in mm/sec by the speed in radians/sec, then that person obtains the eccentricity or radial mass-center-displacement. If an individual compares this with a balancing machine's warranty, which might typically be 25 μin, then some sense of comparative value can be learned.

Table 1. Evaluation of Unbalance Values for a Tested 2x3-8 ANSI Pump.

Speed = 3550 rpm (ω = 371.75 radians/sec)
Weights: Impeller = 6 lbs.; Shaft = 12 lbs.; Rotor = 18 lbs.

ISO Grade $e\omega$, mm/sec	Eccent. e, mm (1)	Eccent. e, μinch. (2)	Force, F, lbs.	"g's" = F/W_{imp}	U_{res} of 6 lb. imp., gm-in	U_{res} of 18 rotor, gm-in	Comments
0.9	0.0024	95	0.204	0.034	0.259	0.777	Nearing API 610
2.3	0.0062	244	0.523	0.087	0.6646	1.994	
6.2	0.0167	657	1.410	0.235	1.7897	5.391	ANSI ISO
10.2	0.0274	1080	2.317	0.386	2.9419	8.757	
13.8	0.0371	1461	3.135	0.523	3.980	11.94	
18.0	0.0484	1906	4.090	0.682	5.192	15.576	F=2/3 of impeller weight

(1) $e_{mm} = \frac{e\omega}{\omega}$ (2) $e_{\mu\text{-inch}} = \frac{e_{mm}}{25.4}$ (3) $F = \frac{W_{imp}}{g}\omega^2 e_{\mu\text{-inch}}$ where g=386 in²/sec.

The unbalance created during tests was equivalent to the column under U_{res} (18 lb, rotor), i.e., 3 × the U_{res} using the impeller weight only.

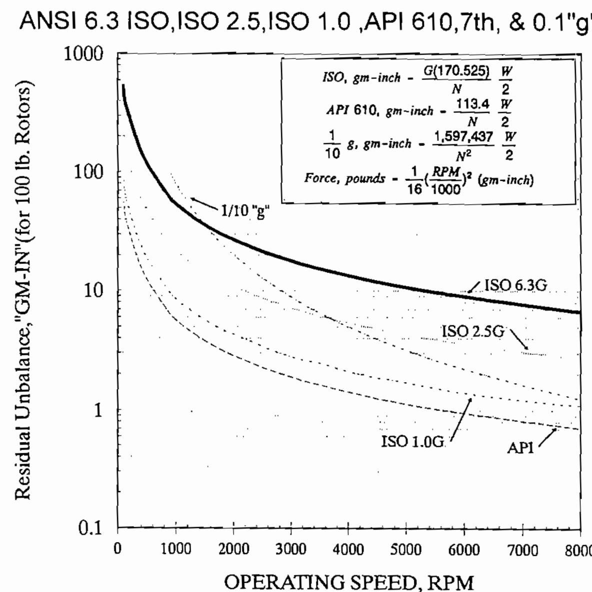
Finally, the unbalance due to fitting the impeller loose on the pump shaft by 1 mil (see LOADS COMPARISON: HYDRAULIC *VS* UNBALANCE) eccentricity was shown as 2.3 lb, whereas the force created by unbalance weight at G6.3 value, was 1.44 lb, e.g., the clearance effect was 2.3/1.44 = 1.6 times greater than G6.3 *pure* balancing effect! If allowable unbalance was chosen at API level, G0.67, then the same clearance effect would be as much as 11 times greater! From a very practical point, this says that if one cannot fit the balanced impeller onto the shaft better that one mil eccentricity; then the force could be 10 times API balance limit!

"SOMETIMES THE *DRAGON* WINS!" = Shop practices, fitting, keys, etc., are important.

The same applies to couplings. If the coupling is balanced to 4W/N, but cannot be assembled within 0.0003 in concentricity,; then 20W/N is exceeded [11].

From the standpoint of balancing machine operators, a useful effort would be to develop a simple to use chart, to quickly determine allowable unbalance, per a particular specification. An example of such a chart is shown on Figure 20, along with tabulation for users' reference, in Table 2.

ROTOR BALANCE TOLERANCES COMPARISON FOR 5 SPECIFICATIONS:



	100	150	900	1000	1200	1500	1800	2000	3000	3600	4000	5000	6000	8000
ISO 6.3 G —	537.2	358.1	59.7	53.7	44.8	35.8	29.8	26.9	17.9	14.9	13.4	10.7	9	6.7
ISO 2.5 G ·	214.9	143.2	23.9	21.5	17.9	14.3	11.9	10.8	7.2	6	5.4	4.3	3.6	2.7
ISO 1.0 G - -	85.3	56.8	9.5	8.5	7.1	5.7	4.7	4.3	2.8	2.4	2.1	1.7	1.4	1.1
API 610,7th - -	56.7	37.8	6.3	5.67	4.73	3.78	3.15	2.84	1.89	1.58	1.42	1.13	0.95	0.71
10% "g" - -			98.61	79.87	55.47	35.5	24.65	19.97	8.87	6.16	4.99	3.19	2.22	1.25

Figure 20. User Chart to Select Allowable Unbalance Vs *Speed, per 100 Lb of Rotor.*

CONCLUSIONS

- The effect of balancing on pump vibrations was quantified for a single stage overhung ANSI pump, mounted on good solid baseplate, relating unbalance to vibrations.

- With proper foundation, vibration levels at dry running operation were found to depend on the value of unbalance greater then approximately G14. For values below G14, the vibrations level off, and practically do not get much lower than that at G14. These levels of vibration are low, namely approximately 0.13 in/sec unfiltered and 0.06 in/sec filtered.

- In actual (wet) operation, hydraulic damping tends to suppress the vibrations even for unbalance above G14, providing the flows are at and lower than bep. However, the overall level

Table 2. Balance Specifications.(Table I) Vs. *GM-IN Residual* Vs *Centrifugal Force.*

N, rpm	ISO 6.3	Force, #	ISO 2.5	Force, #	ISO 1.0	Force, #	API 610	Force, #	1/10 "g"	Force,#
100	537.2	0.3	214.9	0.1	85.3	0.05	56.7	0.04	7,987.2	5.0
150	358.1	0.5	143.2	0.2	56.8	0.08	37.8	0.05	3,549.8	5.0
900	59.7	3.0	23.9	1.2	9.5	0.48	6.3	0.32	98.61	5.0
1,000	53.7	3.4	21.5	1.4	8.5	0.54	5.67	0.35	79.87	5.0
1,200	44.8	4.0	17.9	1.6	7.1	0.63	4.73	0.43	55.47	5.0
1,500	35.8	5.0	14.3	2.0	5.7	0.79	3.78	0.53	35.50	5.0
1,800	29.8	6.0	11.9	2.4	4.7	0.95	3.15	0.64	24.65	5.0
2,000	26.9	6.7	10.8	2.7	4.3	1.06	2.84	0.71	19.97	5.0
3,000	17.9	10.1	7.2	4.0	2.8	1.60	1.89	1.06	8.87	5.0
3,600	14.9	12.1	6.0	4.8	2.4	1.92	1.58	1.28	6.16	5.0
4,000	13.4	13.4	5.4	5.4	2.1	2.13	1.42	1.42	4.99	5.0
5,000	10.7	16.7	4.3	6.68	1.7	2.65	1.13	1.77	3.19	5.0
6,000	9.0	20.3	3.6	8.1	1.4	3.22	0.95	2.13	2.22	5.0
8,000	6.7	26.8	2.7	10.7	1.1	4.25	0.71	2.84	1.25	5.0
10,000	5.4	33.8	2.2	13.5	0.9	5.37	0.57	3.54	0.80	5.0
15,000	3.58	50.4	1.4	20.2	0.57	8.00	*0.38	5.32*	*0.35	5.0*

* 1/10 "g" has crossed as $1/N^2$ becomes the smaller value between 14,000 & 15,000 rpm

of vibrations increases significantly at flows higher than bep. These levels are higher than at dry running testing, but still low, namely 0.15 in/sec unfiltered and 0.08 in/sec filtered (with proper foundation).

- Centrifugal forces caused by unbalance, are significantly smaller than the hydraulic radial thrust at low flows. This unbalance force is less than one percent of hydraulic force at minimum flow and less than five percent at near bep. (However, the magnitudes of both unbalance force and hydraulic radial thrust are low near bep.)

- Shaft deflections at the impeller centerline and seal faces are within ANSI specifications (less than five mils at impeller per latest specification, and less then two mils at seal faces per previous specification revision).

- Although vibrations are low at unbalance of G14 and less, and increase for unbalance greater than G14, it is recommended that a conservative value of G6.3 should continue being a standard for these types of pumps, as it has been an accepted practice for many years by pump manufacturers and users.

- More testing on other pump types will help better determine a similar critical value of G-levels, below which the further betterment of unbalance leads to a diminishing return with respect to vibrations, forces, and life. An economical, reasonable compromise can thus be determined between the level of unbalance and technically feasible/reasonable efforts to achieve it.

- A good balance specification is necessary. ANSI uses ISO 6.3G. API uses ISO 0.665G (4W/N). Perhaps these should be reviewed. Not necessarily because of the actual value, but because of all the mechanical fitting practices, that may not be proper. One important practice is balancing impellers on solid mandrels with proper fit, and taking TIR face readings on the impellers and TIR radial readings on the impeller wear ring and the shaft near the fit step; these measures assuring that the impeller is straight, the shaft is not bowed and the key is not "high centered" (= key radially too high). Since hydraulic forces usually exceed the unbalance forces by about 5:1, or more, there should be more concern for proper overall pump selection and operation regions.

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