# A REVIEW OF DESIGN ANALYSIS METHODS FOR A HORIZONTAL END SUCTION CENTRIFUGAL PUMP

by

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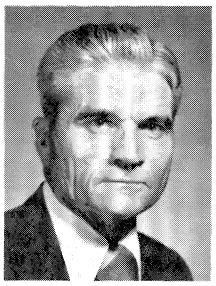

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## ABSTRACT

Users of standard production model centrifugal pumps are frequently faced with the task of evaluating a group of very competitive bids. A thorough evaluation will involve both technical and commercial aspects of the submittals.

As energy costs increase, the commercial value of a pump's hydraulic efficiency becomes more significant. The vendor's characteristic curve typically shows efficiencies which can be used to estimate power costs for a particular pump. Frequently, the type of pump discussed in this paper is a key component in a refinery or chemical process. Mechanical ruggedness may be more important than efficiency.

This paper reviews some of the technical evaluation methods available to the user. The mechanical components of a horizontal end suction centrifugal pump are subjected to various hydraulic and/or mechanical forces. A determination of the magnitude and direction of such forces at various points along the characteristic curve permits further evaluation of the pump's mechanical design for operation at best efficiency, minimum flow, runout, etc.

An understanding of the principles involved herein should assist the user both in the initial equipment selection as well as troubleshooting after installation. The cause of a problem must normally be identified prior to its effective removal by modification of the equipment or the service.

## INTRODUCTION

This paper presents an analytical method for mechanical design review of horizontal end suction centrifugal pumps. A cross-sectional view of such a pump, as shown in Figure 1, shows areas of greatest interest.

Two unbalanced hydraulic forces, radial and axial, are acting on the impeller. The radial force produces a shaft deflection which is a classic fatigue cause for a rotating shaft. It also causes reactions at the bearings and affects bearing life. Shaft deflection through the stuffing box must be limited to permit adequate packing and/or mechanical seal life with acceptable leakage rates.

Torsional loading of the shaft adds to the bending and axial loads to produce a combined stress. In this type of pump, the axial load is usually the least significant with respect to total shaft stress. The torsional load and unbalanced hydraulic forces vary, depending on the impeller and case design and point of operation along the pump characteristic curve. Methods for determining the various forces will be individually presented for a hypothetical pump followed by a discussion of their combined effect on pump operation.

The paper is divided into three main sections. 1) A review of calculation methods for determination of the unbalanced hydraulic forces at various points on the operating curve; 2) The mechanical forces and reactions in pump components resulting from the hydraulic forces in 1) above; 3) A sample evaluation is conducted for a hypothetical pump design and application.

## DETERMINATION OF UNBALANCED HYDRAULIC FORCES

The radial unbalanced force for a volute style case, see Figure 2, may be calculated as follows [1]:

$$F = \frac{KHD_2B_2\,(S.G.)}{2.31} \qquad (1)$$

A nomenclature at the end of the paper defines the terms of this equation and those that follow.

For a concentric style case, see Figure 3, the same equation for F may be used with a new value for K; i.e.,

$$K = .28\ \frac{Q}{Q_n}$$

In the latter equation, the constant .28 is used in lieu of .36 as a result of experimentally measuring radial unbalance in a concentric case pump.

The direction of the radial force varies with the flow rate and may add to or subtract from the weight of the overhung shaft and impeller. For purposes of comparison and evaluation, however, the direction of the radial force unbalance may be neglected. When radial force unbalance is significant, the static weight of the impeller and shaft will not be significant since all pumps of a similar case style will have similar sized components and direction of the force. If, however, volute style cases are compared with concentric style cases, it may be well to consider both the direction of the force and the weight of the rotor. An explanation for determining thrust direction may be found in reference [1].

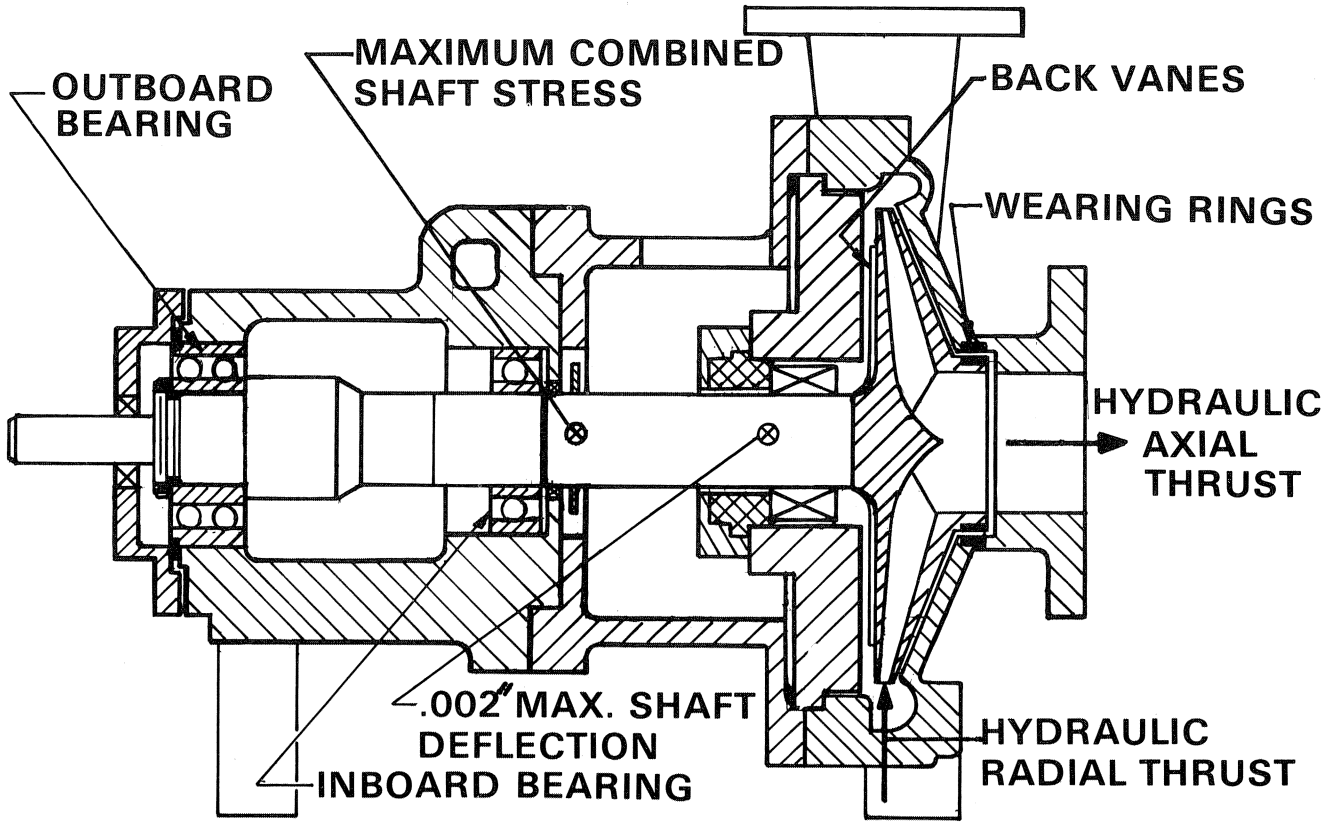


*Figure 1. Sectional View of Typical Horizontal End Suction Pump.*

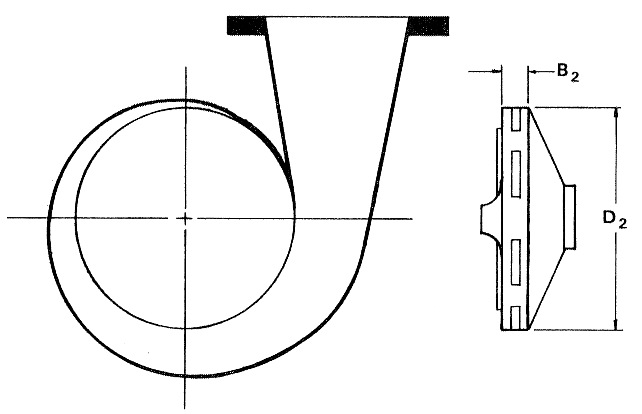


*Figure 2. Volute Style Case and Impeller.*

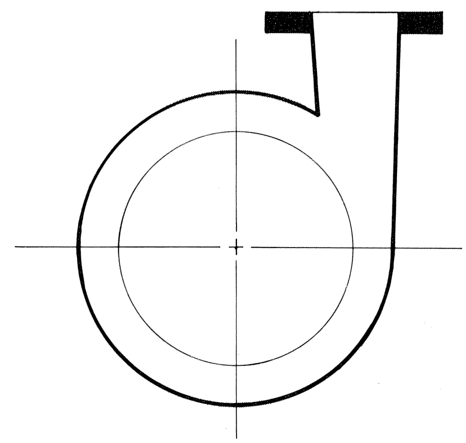

*Figure 3. Concentric Style Case.*

A word of caution is in order at this point. At design, the value of K is unity for a concentric case pump and approaches zero as flow is reduced. The opposite is true for the volute style case. This means that a concentric style pump will see a substantial radial thrust at its normal operating condition and requires a heavier shaft and bearing design than a volute style which has zero radial thrust at its normal operating condition.

## AXIAL THRUST UNBALANCE

The axial thrust unbalance for a single suction impeller is influenced by the following design features:

A. Impeller style: enclosed, semi-enclosed, open.

B. Existence or non-existence of back vanes and their diameter.

C. Balancing ring and vents.

D. Running clearances.

Figures 4 through 6 show the various impeller styles and balancing details.

For an enclosed impeller, Figure 7, the unbalanced axial thrust may be calculated as follows [1]:

$$T = (A_1 - A_s)(P_2 - P_1) \tag{2}$$

The value of $(P_2 - P_1)$ can be approximated as follows [1]:

$$(P_2 - P_1) = \frac{3}{4}\,\frac{(U_2^2 - U_1^2)}{2g}\left(\frac{S.G.}{2.31}\right) \tag{3}$$

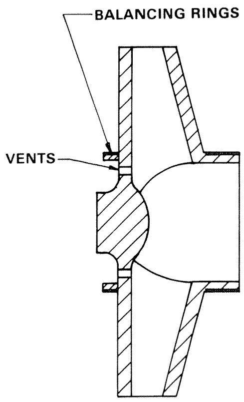


*Figure 4. Enclosed Impeller with Balancing Rings and Vents.*

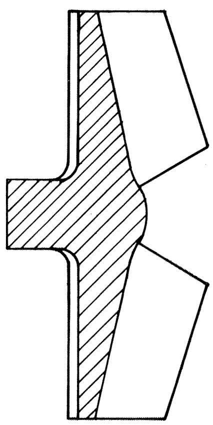

*Figure 5. Semi-Enclosed Impeller with Back Vanes.*

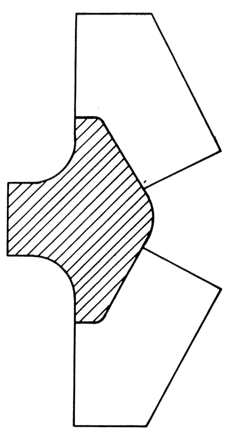

*Figure 6. Open Impeller.*

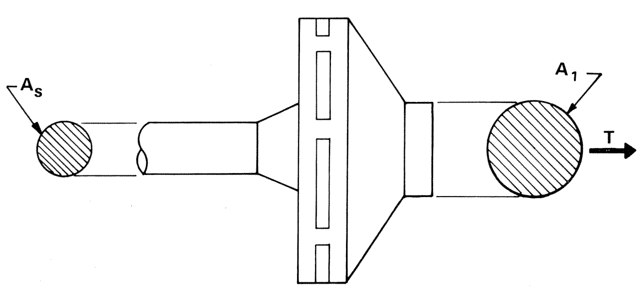


*Figure 7. Enclosed Impeller.*

Back vanes, as shown in Figure 8, will alter the unbalanced force. Their effect may be calculated as follows [1]:

$$T_{br} = \frac{3}{8}(A_r - A_s)\frac{(U_r^2 - U_s^2)}{2g}\left(\frac{S.G.}{2.31}\right) \tag{4}$$

For perfect balance, T must equal $T_{br}$.

Balance rings and vent holes may be used, as shown in Figure 4, for clean fluids. The use of balancing rings and vents is a common practice for the type of pump under discussion. The principle involved is to provide a set of wearing rings on the inboard side of the back shroud and to communicate the area between the balancing rings and the shaft with the eye of the impeller through vents in the shroud. Such an arrangement adds to the leakage loss and disturbs the incoming flow to the impeller eye. As wearing rings erode unevenly, the unbalance increases, leading to reduced thrust bearing life.

For semi-enclosed impellers, as shown in Figure 9, the unbalanced axial thrust may be calculated as follows [1]:

$$T = T_b - T_{bi} \tag{5}$$

where,

$$T_b = (A_2 - A_s)\left[H_2 - \frac{1}{8}\frac{(U_2^2 - U_s^2)}{2g}\right]\left(\frac{S.G.}{2.31}\right) \tag{6}$$

and,

$$T_{bi} = (A_2 - A_1)\left(\frac{1}{2}\right)(H_2)\left(\frac{S.G.}{2.31}\right) \tag{7}$$

The effect of back vanes is the same as with closed impellers. Therefore, for perfect balance, T must equal $T_{br}$ and:

$$T_b - T_{bi} - T_{br} = 0 \tag{8}$$

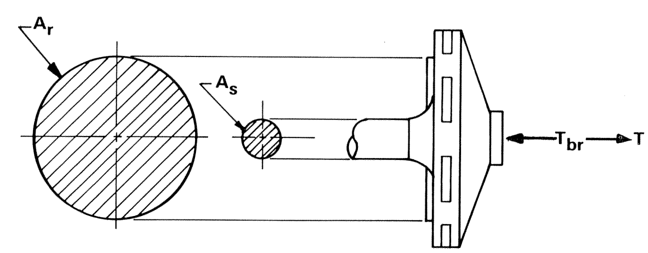


*Figure 8. Enclosed Impeller with Back Vanes.*

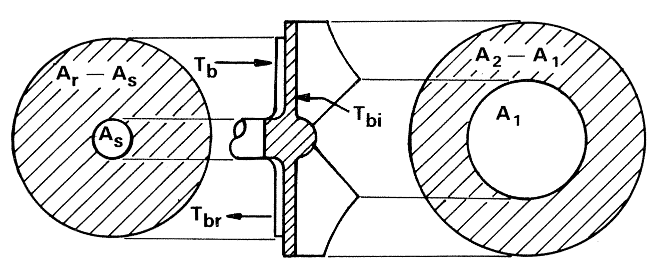


*Figure 9. Thrust Generation Areas of a Semi-Enclosed Impeller with Back Vanes.*

Substituting equations (4), (6) and (7) gives:

$$(A_2 - A_s)\left[H_2 - \frac{1}{8}\frac{(U_2^2 - U_s^2)}{2g}\right]\left(\frac{S.G.}{2.31}\right) - (A_2 - A_1)\left(\frac{1}{2}\right)\left(H_2\right)\left(\frac{S.G.}{2.31}\right) - \frac{3}{8}(A_r - A_s)\frac{(U_r^2 - U_s^2)}{2g}\left(\frac{S.G.}{2.31}\right) = 0 \tag{9}$$

## EFFECTS OF UNBALANCED FORCES ON MECHANICAL COMPONENTS

### *Shaft Deflection*

Figure 10 shows a typical shaft arrangement for a horizontal end suction centrifugal pump. The hydraulic radial unbalance will cause a deflection in the shaft which may be calculated by using the following expanded version of the beam formula [2]:

$$Y = \frac{F}{3E}\left[\frac{N^3}{I_N} + \frac{M^3 - N^3}{I_M} + \frac{L^3 - M^3}{I_L} + \frac{L^2X}{I_X}\right] \tag{10}$$

If the effects of film support are neglected at the wearing rings, one can determine whether or not a positive clearance exists by comparing the calculated deflection to one-half of the diametral clearance between the impeller and case wearing rings. For semi-enclosed or open impellers, an analysis of the running clearance between the vane edge and the suction cover may be made.

Should the calculated deflection exceed .004″ to .005″ at the impeller, a detailed solution for deflection across the entire shaft length may be in order. Of particular interest will be the shaft deflection through the stuffing box area. For good mechanical seal and/or packing life, it is recommended that the shaft deflection not exceed .002″ at the sealing face of the stuffing box.

A sample graphical solution for a non-uniform diameter shaft deflection is described in "Centrifugal and Axial Flow Pumps," by Stepanoff [1] or in most structural analysis texts. Some users may have access to computer programs for the solution.

### *Shaft Stress*

Shaft stress will result from a combination of axial thrust, bending moment, torsion and stress concentration. Axial thrust stress is normally the least significant of these.

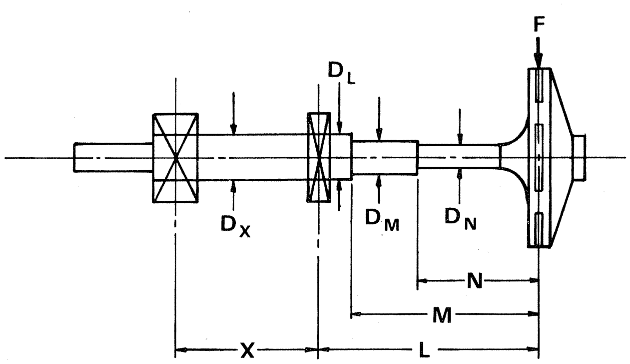


*Figure 10. Typical Shaft Arrangement.*

Referring again to Figure 10, the bending moment will usually be a maximum at the inboard bearing. For the cantilevered impeller, the bending moment is calculated simply as:

$$F \cdot L = (MO) \tag{11}$$

The horsepower input to the shaft is taken from the vendor's performance curve, corrected for specific gravity, at the same capacity point used to calculate F.

$$TORQUE = (TO) = \frac{63025 \times BHP}{RPM}\ \text{in.-lb.} \tag{12}$$

The combined maximum shear stress in the shaft may be determined as follows:

$$S_{s\,max} = \frac{16}{\pi D^3}\sqrt{(1.5\,MO)^2 + (1.2\,TO)^2}\ \text{psi} \tag{13}$$

Most pump manufacturers follow good design practice in filleting shafts at steps to avoid excessive stress raisers. Nevertheless, it is good practice to determine the geometry involved and apply a torsional stress concentration factor to (TO) in equation (13). Such factors are generally available in machine design texts or handbooks. They usually begin at approximately 1.2 and increase to approximately 3.2. The calculated stress should provide a minimum design factor of 2 when compared to the shear yield and endurance limit of the material.

Other shaft areas of interest are at the impeller and coupling. Since very little bending moment exists at the latter positions, the stress is primarily torsional. The magnitude may be calculated as follows:

$$S_s = \frac{16\,(TO)}{\pi D^3} \tag{14}$$

Good practice dictates a maximum wetted shaft stress of 7500 psi for common shaft materials to allow for possible stress corrosion effects.

The coupling end of the shaft may see a bending moment in addition to torsional load if the coupling used has a significant angular stiffness and is not perfectly aligned. Should this be the case, the equation for combined bending and torsion stress should be used, equation 13, and some allowance for the keyway made.

### *Bearing Life*

Referring to Figure 10, the inboard bearing reaction is approximately:

$$F\,\frac{(X + L)}{X}\ \text{lbs.} \tag{15}$$

Outboard bearing reaction is approximately:

$$F \cdot \frac{L}{X}\ \text{lbs.} \tag{16}$$

The outboard bearing usually serves as the thrust bearing. It has a combined radial and axial loading.

The expected life of the bearings may be calculated by methods described in most bearing catalogs or pertinent ANSI standards such as B 3.15 or B 3.16.

The competitive nature of the pump business has resulted in similar shaft and bearing designs by the various manufacturers. For a comparative analysis between pumps of different manufacturers, the general rule for anti-friction bearing life, all else being equal, is:

Identical bearings have a life inversely proportional to load cubed.

## SAMPLE EVALUATION

As an example, the following hypothetical acid pump will be analyzed using vendor supplied data sheet information:

## GENERAL DATA

Nominal size = 1½″ × 3″ × 8″

Discharge = 1½″

Suction = 3″

Impeller O.D. = 8⅜″

Impeller tip width less back vanes = .562″

Case style = volute

Back vane O.D. = 6″

Impeller style = semi-enclosed

## SHAFT DATA

At impeller = .875″ Dia. × 1.5 ″

In stuffing box = 1.125″ Dia. × 5″

Under bearing = 1.379″ Dia. × .750″

Between bearings = 1.5″ Dia. × 4.125″

Sleeve O.D. = 1.375″

At coupling = .875″ Dia.

## BEARING DATA

Radial = 207K

Thrust and radial = 5306W

Bearing span = 4.125″

Shaft overhang = 6.125″

## HYDRAULIC PERFORMANCE DATA

RPM = 3500

$Q_n$ = 200 GPM @ 254 ft. @ 20 HP (Water)

$Q_{max}$ = 300 GPM @ 58 ft. @ 26 HP (Water)

$Q_{shut\ in}$ = 0 GPM @ 290 ft. @ 10 HP

S.G. = 1.83

## UNBALANCED AXIAL THRUST T AT $Q_n$

$A_2 = .785\,(8.375)^2 = 55.06\ \text{in}^2$

$A_s = .785\,(1.375)^2 = 1.48\ \text{in}^2$

$$U_2^2 = \left[\left(\frac{8.375}{12}\right)(\pi)(3500)(1/60)\right]^2 = 16358\ \text{ft.}^2/\text{sec.}^2$$

$$U_s^2 = \left[\left(\frac{1.375}{12}\right)(\pi)(3500)(1/60)\right]^2 = 441\ \text{ft.}^2/\text{sec.}^2$$

$A_1 = .785\,(3)^2 = 7.07\ \text{in}^2$

$A_r = .785\,(6)^2 = 28.26\ \text{in}^2$

$$U_r^2 = \left[\left(\frac{6}{12}\right)(\pi)(3500)(1/60)\right]^2 = 8396\ \text{ft.}^2/\text{sec.}^2$$

$$T = (55.06 - 1.48)\left[254 - 1/8\,\frac{(16358 - 441)}{2(32.17)}\right]\left(\frac{1.83}{2.31}\right)$$

$$- (55.06 - 7.07)(1/2)(254)\left(\frac{1.83}{2.31}\right)$$

$$- 3/8(28.26 - 1.48)\left[\frac{(8396 - 441)}{2(32.17)}\right]\left(\frac{1.83}{2.31}\right)$$

$$= 3657\ \text{lbs.}$$

## RADIAL THRUST F AT $Q_n$

For volute case at $Q_n$, F = 0

The bearing reactions at $Q_n$ will be negligible at the inboard bearing and practically all of the thrust will be at the outboard bearing. Referring to Fafnir Bearing Catalog 68, page 51, the following life for a 5306W bearing at $Q_n$ is:

$$R_E = 1.24(3657) = 4535\ \text{lbs.}$$

$$L_{10} = \left(\frac{2041}{4535}\right)^3 (1500) = 137\ \text{hours.}$$

Obviously, there is a problem and the pump is unacceptable as is.

If the back vane outside diameter is increased from 6″ to 8.375″, the reduction in the unbalanced axial thrust will be:

$$T_{br\ 8.375''} - T_{br\ 6''} = 3938 - 984 = 2954\ \text{lbs.}$$

The new unbalanced axial thrust will be:

$$3657 - 2954 = 703\ \text{lbs.}$$

The new calculated bearing life for a 5306W is:

$$R_E = 1.24(703) = 872$$

$$L_{10} = \left(\frac{2041}{872}\right)^3 (1500) = 19234\ \text{hours.}$$

It is safe to assume a satisfactory shaft deflection and inboard bearing life at $Q_n$.

## RADIAL THRUST F AT $Q_{max}$

Radial Thrust = F

$$= \frac{.36\left[1 - \left(\frac{300}{200}\right)^2\right](58)(8.375)(.562)(1.83)}{2.31}$$

$$= -97.3\ \text{lbs.}$$

Using the absolute value of 97.3 lbs., the bearing reactions are:

$$\text{INBOARD BRG.}\quad \frac{97.3\,(4.125 + 6.125)}{4.125} = 242\ \text{lbs.}$$

$$\text{OUTBOARD BRG.}\quad \frac{97.3\,(6.125)}{4.125} = 144\ \text{lbs.}$$

## UNBALANCED AXIAL THRUST T AT $Q_{max}$

Axial Thrust = T

$$= (55.06 - 1.48)\left[58 - \tfrac{1}{8}\frac{(16358 - 441)}{2(32.17)}\right]\left(\frac{1.83}{2.31}\right)$$

$$- (55.06 - 7.07)(\tfrac{1}{2})(58)\left(\frac{1.83}{2.31}\right)$$

$$- \tfrac{3}{8}(55.06 - 1.48)\left[\frac{(16358 - 441)}{2(32.17)}\right]\left(\frac{1.83}{2.31}\right)$$

$$= -3890 \text{ lbs.}$$

The negative sign indicates a thrust reversal between $Q_n$ and $Q_{max}$. Use the absolute value in bearing life calculations.

## INBOARD BRG. LIFE AT $Q_{max}$

$$= \left[\frac{.214(4620)}{242}\right]^3 (1500) = 102285 \text{ hours}$$

## OUTBOARD BRG. LIFE AT $Q_{max}$

$$= \left[\frac{.214(9450)}{4976}\right]^3 (1500) = 101 \text{ hours}$$

## MAXIMUM SHAFT DEFLECTION AT $Q_{max}$

$$Y = \frac{97.3}{3 \times 30 \times 10^6}\left[\frac{3.375}{.0287} + \frac{275 - 3.375}{.0786} + \frac{381 - 275}{.1775} + \frac{217}{.2485}\right]$$

$$= 1.08 \times 10^{-6}\,[117.59 + 3498.72 + 597.18 + 873.23]$$

$$= .0055 \text{ in.}$$

## MAXIMUM SHAFT STRESS AT $Q_{max}$

Bending Moment = MO = 97.3 × 6.125 = 596 in.-lb

$$\text{Torque} = \text{TO} = \frac{63025 \times 26 \times 1.83}{3500} = 857 \text{ in.-lb}$$

$$S_s \text{ Max at Inboard Brg.} = \frac{16}{\pi(1.379)^3}$$

$$\sqrt{(1.5 \times 596)^2 + (1.2 \times 857)^2} = 2646 \text{ psi}$$

$$S_s \text{ at Impeller} = \frac{16 \times 857}{\pi(.875)^3} = 6515 \text{ psi}$$

$$S_s \text{ at Coupling} = \frac{16 \times 857}{\pi(.875)^3} = 6515 \text{ psi}$$

The hypothetical pump is not suited for continuous flow at the runout condition because of excessive shaft deflection, low bearing life and high wetted shaft stress. It will suffice for operation near $Q_n$ and could probably withstand short periods of runout without significant problems. The back vane diameter could be further improved.

These same methods can be used to investigate the suitability for reduced flow operation.

## CONCLUSION

Fundamental methods exist for the determination of the hydraulic forces acting on the impeller of a single stage end suction centrifugal pump. The user's inquiry data sheet should provide spaces for vendor input such as impeller and shaft sketch showing necessary nominal dimensions. Proprietary design data are not required.

An individual pump may be evaluated for minimum acceptable design criteria or a bid tabulation can be made to establish a mechanical ranking for several different pumps. Although the competitive nature of the pump business has resulted in a high degree of similarity in pump designs among the various manufacturers, a careful analysis of hydraulic and mechanical features can lead to significant conclusions with regard to optimum selection.

## REFERENCES


1. Stepanoff, A. J., "Centrifugal and Axial Flow Pumps," Second Edition, John Wiley & Sons, Inc., 1957.
2. "Pump Engineering Manual," The Duriron Company, Inc., 1960. Credit for the several articles is given to: J. R. Birk, T. Bramlage, A. B. Brinkel, D. W. Duffey, J. B. Freed and R. M. Shields.


## NOMENCLATURE

$$F = \frac{KHD_2B_2(S.G.)}{2.31} \qquad (1)$$

where:
- F = radial resultant force, pounds
- H = head, feet
- $D_2$ = impeller outlet diameter, inches
- $B_2$ = impeller outlet width including shrouds, inches
- K = $0.36\left[1 - \left(\frac{Q}{Q_n}\right)^2\right]$, dimensionless
- Q = operating point GPM
- $Q_n$ = best efficiency GPM
- S.G. = Specific gravity (water = 1.0)

$$T = (A_1 - A_s)(P_2 - P_1) \qquad (2)$$

where:
- T = unbalanced thrust, pounds, acting outside of back shroud
- $A_1$ = area of impeller wearing ring circle, $in^2$
- $A_s$ = area of shaft cross section, $in^2$
- $P_2$ = pressure at wearing ring diameter, psig
- $P_1$ = suction pressure, psig

$$(P_2 - P_1) = \frac{3}{4}\,\frac{(U_2^2 - U_1^2)}{2g}\left(\frac{S.G.}{2.31}\right) \qquad (3)$$

where:
- $U_2$ = linear velocity of impeller O.D., ft./sec.
- $U_1$ = linear velocity of impeller vane entrance tip, ft./sec.

S.G. = Specific gravity (relative to water = 1.0)

$$T_{br} = \frac{3}{8}(A_r - A_s)\frac{(U_r^2 - U_s^2)}{2g}\left(\frac{S.G.}{2.31}\right) \quad (4)$$

where: $T_{br}$ = back vane thrust effect, pounds
$A_r$ = area of back vane O.D. circle, $in^2$
$A_s$ = area of shaft cross section, $in^2$
$U_r$ = linear velocity of back vane O.D., ft./sec.
$U_s$ = linear velocity of shaft O.D., ft./sec.

$$T = T_b - T_{bi} \quad (5)$$

where: T = unbalanced thrust, pounds
$T_b$ = thrust outside of back shroud

$$= (A_2 - A_s)\left[H_2 - \frac{1}{8}\frac{(U_2^2 - U_s^2)}{2g}\right]\left(\frac{S.G.}{2.31}\right) \quad (6)$$

$T_{bi}$ = thrust inside of back shroud

$$= (A_2 - A_1)\left(\frac{1}{2}\right)(H_2)\left(\frac{S.G.}{2.31}\right) \quad (7)$$

$A_1$ = area of circle of impeller eye diameter, $in^2$
$A_2$ = area of circle of impeller O.D., $in^2$
$H_2$ = discharge head, ft.

$$T_b - T_{bi} - T_{br} = 0 \quad (8)$$

$$(6) - (7) - (4) = 0 \quad (9)$$

$$Y = \frac{F}{3E}\left[\frac{N^3}{I_N} + \frac{M^3 - N^3}{I_M} + \frac{L^3 - M^3}{I_L} + \frac{L^2X}{I_X}\right] \quad (10)$$

where: Y = shaft deflection at impeller centerline, inches
F = hydraulic radial unbalance, pounds
M,N = distances from impeller centerline to the steps on the shaft, inches
L = distance from impeller centerline to centerline of inboard bearing, inches
X = span between bearing centerlines, inches.
$I_L$, $I_M$, $I_N$, $I_X$ = moments of inertia of the various diameters, $in^4$
E = modulus of elasticity of shaft material, psi

$$F \cdot L = (MO) \quad (11)$$

where: F = radial force from equation (1)
L = distance between inboard bearing centerline and line of action of F, inches
MO = moment, in.-lb.

$$\text{TORQUE} = (TO) = \frac{63025 \times BHP}{RPM} \text{ in.-lb.} \quad (12)$$

$$S_{s\,max} = \frac{16}{\pi D^3}\sqrt{(1.5\,MO)^2 + (1.2\,TO)^2} \text{ psi} \quad (13)$$

where: $S_{s\,max}$ = maximum shaft shear stress
D = shaft diameter, inches

$$S_s = \frac{16\,(TO)}{\pi D^3} \quad (14)$$

where: Terms are as previously defined.

$$\text{Inboard bearing reaction} = F\frac{(X + L)}{X} \text{ lbs.} \quad (15)$$

where: X and L are defined in Figure 10
F as previously defined

$$\text{Outboard bearing reaction} = F \cdot \frac{L}{X} \quad (16)$$