PEERLESS PUMP™

TECHNICAL INFORMATION BULLETIN

NUMBER TWENTY-FIVE

Analysis of Torsional Vibrations in Vertical Centrifugal Pumps

Vibration is perhaps the greatest enemy of rotating machinery. A certain amount of it can be safely tolerated. An excessive amount will result in an equipment failure which might produce losses running into thousands of dollars.

This paper presents a simplified discussion of torsional vibrations in vertical pumps. It also suggests types and sizes of the systems which should be analyzed for potential problems.

Vibrations

Vibration is the periodic oscillation about, or relative to, a fixed plane, line or axis.

The three basic types of vibrations which can occur in any vertical pump system are: lateral, torsional, and longitudinal.

Lateral vibrations are characterized by linear displacement of mass, usually at right angles to the reference plane. The reference plane is also called neutral or equilibrium plane.

Although excessive torsional vibrations are less common than lateral, they are very dangerous, especially in large and complex systems. They are characterized by angular displacement of mass about a reference axis, which almost always is the axis or rotation.

Longitudinal vibrations are not usually dangerous. They are also very seldom noticed. However, they do occur especially in pumps having long settings and subjected to water hammer. Longitudinal vibrations, occasionally called axial vibrations, are characterized by linear displacement of mass parallel to the centerline of the body with reference to any fixed horizontal plane.

Pump Systems

Torsional vibrations are present in greater or smaller degree in every type of pump drive-train. Systems consisting of an internal combustion engine, a right angle gear drive, and a pump are much more susceptible to torsional vibrations than those consisting only of a motor and a pump. Critical torsional vibrations occur when the speed of any of the drive-train components is equal to the frequency of drive-train natural torsional vibrations. To determine the magnitude of their vibrations special instrumentation is necessary. The measurements normally taken with pickups attached to the driver housing, or the discharge head, are of the lateral type.

It is impossible to predict which systems will or will not develop destructive torsional vibrations. However, our past field experiences indicate that systems shown in Fig. 1 through Fig. 4 present potential field problems and should be analyzed at the design stage. In all probability, as the years go by, the sizes of these systems will be changed to reflect future field experiences, and the growing tendency to run pumps and drivers at higher and higher speeds.

Torsional Vibration Analysis

Natural frequencies of free torsional vibrations depend on the inertias of rotating masses, their location with respect to each other, and the elastic characteristics of shafts and couplings. Pump drive-train components and the data required for the analysis are shown in Fig. 5 through Fig. 7. The accuracy of the analysis will depend on the accuracy of these data.

Normal objectives of a torsional vibration analysis are:

1. Determination of natural frequencies, critical speeds, and mode shapes.
2. Calculation of vibration amplitudes, and stresses.

Peerless Pump's program for torsional vibration analysis called "HOLZERAM" is limited to computation of the first 19 natural frequencies of any drive-train comprising a maximum of 20 rotating masses. It is based on the tabulation method for free torsional vibrations as proposed by H. Holzer. Description of this method can be found in practically every textbook on mechanical vibrations.

Program input consists of:

1. Mass moments of inertia in lb. in. sec.^{2}
2. Torsional stiffness in in. lb./radian

Program output consists of:

1. Natural frequency of each mode, in cycles per minute.
2. Relative magnitude and direction of the vibration angular amplitude of each mass for each mode. It is dimensionless.

The amplitudes are used for plotting normal elastic curves from which location of the nodes is determined. This is often of importance where location of certain types of vibration damping devices, or gears, has to be considered. However, extreme sensitivity of the mode shape to frequency changes, and the limiting accuracy of the computer, limit the usable shapes to the fourth mode.

Whether or not it will be necessary to calculate actual amplitudes, and torsional stresses, depends on the outcome of the first part of the analysis and the cost of any proposed modification. If, for example, the first natural frequency falls within 25% of the operating speed and the cost of the proposed modification is substantial, stress analysis might help to reduce it. A designer, or a field engineer, knowing the intensity of stresses, and their location, will be able to estimate the risk factor involved, or to propose a less expensive solution.

Peerless Pump is prepared to analyze and give a complete report of any prospective or present vertical centrifugal pump system.

Costs and Report Content

Cost of a vibration analysis involving determination of natural frequencies, critical speeds and mode shapes depends on:

1. Complexity of the torsional rotating system.
2. Availability and form of the mechanical data of each component of the system.

If all the mechanical data is available as specified, the analysis time can be reduced by as much as 50%.

The final written report provided by Peerless Pump consists of:

1. Summary of input data and conversions including schematic drawing of pump drive-train.
2. Tabulation of natural frequencies of each mode converted to critical speeds in RPM of the driver.
3. A plot of normal elastic curve for each mode up to four modes.
4. Comment on performance of pump drive-train under operating conditions and recommendations for reducing vibrations, if any.

Address your inquiries to

Peerless Pump Company
Engineering Department
1441 Peerless Way
Montebello, CA 90640

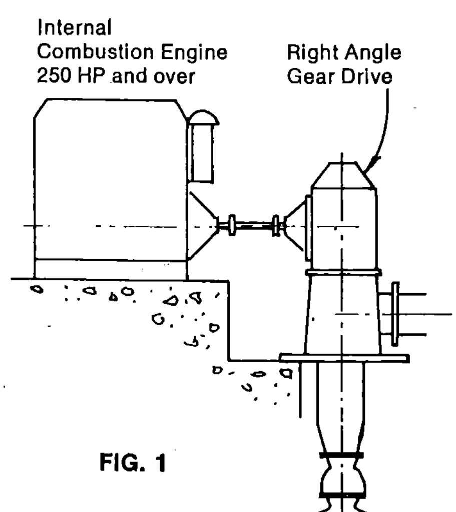


FIG. 1

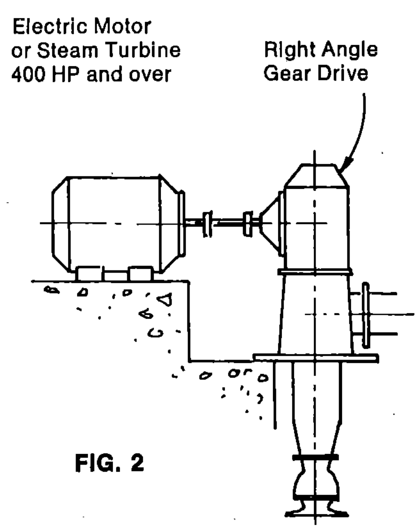


FIG. 2

Pump Systems Requiring Torsional Vibration Analysis

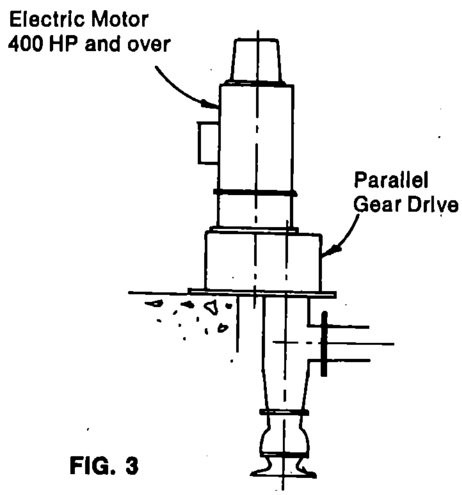


FIG. 3

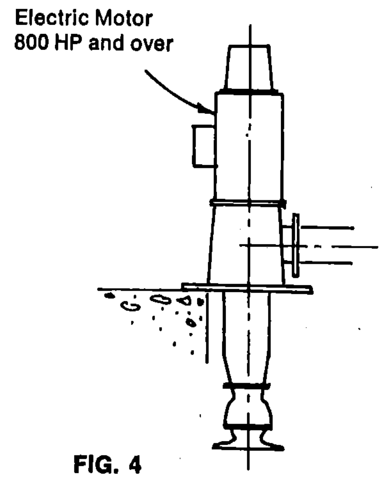
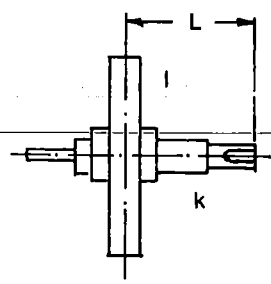


FIG. 4

Pump Systems Requiring Torsional Vibration Analysis

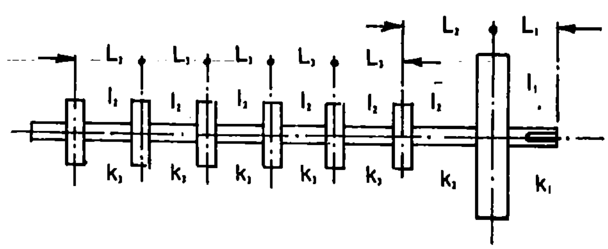


I —Moment of Inertia of Rotor-Shaft Assembly (WR^2), lb. in.^{2}

L —Distance between rotor and end of output shaft, inches

K—Torsional Stiffness of Length L, in. lb./radian

FIG. 5 Steam Turbine or Horizontal Electric Motor



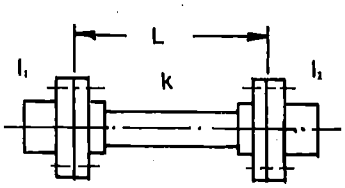
I_1—Moment of Inertia of Flywheel (WR^2), lb. in.^{2}

I_2—Equivalent Moment of Inertia of crank, connecting rod, and piston (WR^2), lb. in.^{2}

L_1, L_2, L_3—Lengths of equivalent shaft sections, inches.

k_1, k_2, k_3—Torsional stiffnesses of equivalent shaft sections, in. lb./radian.

FIG. 6 Internal Combustion Engine—Equivalent Rotating System
(Diagram illustrates 6-cylinder engine with a flywheel)



I_1—Moment of Inertia of Input Coupling Assembly (WR^2), lb. in.^{2}

I_2—Moment of Inertia of Output Coupling Assembly (WR^2), lb. in.^{2}

L—Length of Spacer, inches.

k—Torsional Stiffness of the Assembly, in. lb./radian.

FIG. 7 Flexible Shaft Coupling

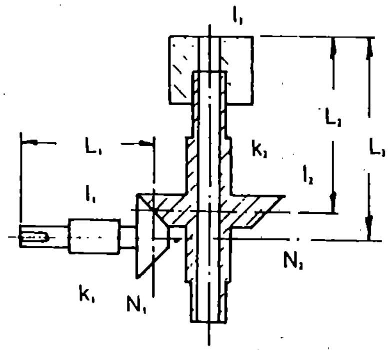


FIG. 8 VHS Right Angle Gear Drive

I_1—Moment of Inertia of High Speed Shaft Assembly (WR^2), lb. in.^{2}

I_2—Moment of Inertia of Low Speed Shaft Assembly without coupling (WR^2), lb. in.^{2}

I_3—Moment of Inertia of Coupling Assembly (WR^2), lb. in.^{2}

L_1, L_2—Effective Lengths of High and Low Speed Shaft Assemblies, inches

k_1, k_2—Torsional Stiffnesses of High and Low Speed Shaft Assemblies, in. lb./radian

N_1, N_2—Number of Teeth in Pinion and Gear

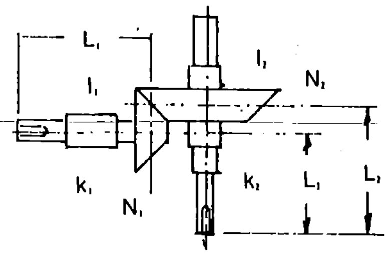


FIG. 9 VSS Right Angle Gear Drive

I_1, I_2—Moments of Inertia of High and Low Speed Shaft Assemblies (WR^2), lb. in.^{2}

L_1, L_2—Effective Lengths of High and Low Speed Shaft Assemblies, inches

k_1, k_2—Torsional Stiffnesses of High and Low Speed Shaft Assemblies, in. lb./radian

N_1, N_2—Number of Teeth in Pinion and Gear

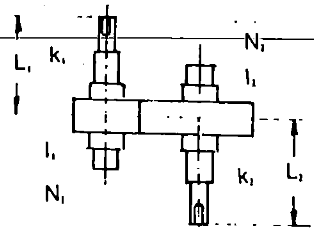


FIG. 10 Parallel Gear Drive

I_1, I_2—Moments of Inertia of High and Low Speed Shaft Assemblies (WR^2), lb. in.^{2}

L_1, L_2—Effective Lengths of High and Low Speed Shaft Assemblies, inches

k_1, k_2—Torsional Stiffnesses of High and Low Speed Shaft Assemblies, in. lb./radian

N_1, N_2—Number of Teeth in Pinion and Gear

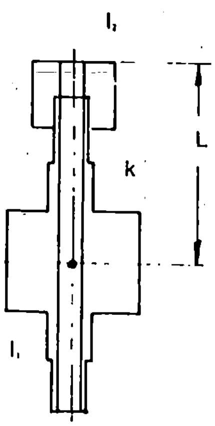


FIG. 11 VHS Electric Motor

I_1—Moment of Inertia of Rotor-Shaft Assembly without Coupling (WR^2), lb. in.^{2}

I_2—Moment of Inertia of Coupling Assembly (WR^2), lb. in.^{2}

L—Distance from Rotor C.G. to Top of Coupling

k—Torsional Stiffness of Length L, in. lb./radian

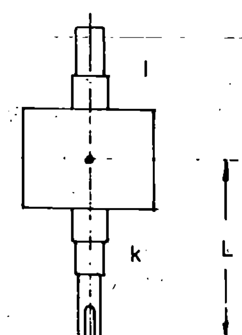


FIG. 12 VSS Electric Motor

I_1—Moment of Inertia of Rotor-Shaft Assembly (WR^2), lb. in.^{2}

L—Distance between Rotor C.G. and End of Output Shaft, inches

k—Torsional Stiffness of Length L, in. lb./radian

PEERLESS PUMP™

1441 Peerless Way, Montebello, California 90640 (213) 726-1232 Telex 67-4156 FAX (213) 726-0814
P.O. Box 7026, Indianapolis, Indiana 46207-7026 (317) 925-9661 Telex 27-355 FAX (317) 924-7388

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