Recommended Practice on Electric Submersible Pump System Vibrations

API RECOMMENDED PRACTICE 11S8 (RP 11S8)
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SECTION 1
GENERAL

1.1 Introduction. This recommended practice provides guidelines to establish consistency in control and analysis of ESP system vibrations. These recommended practices are those generally considered appropriate for the acceptance testing of ESP systems and subsystems for the majority of ESP applications.

1.2 Scope. This RP covers the vibration limits, testing, and analysis of electric submersible pump systems and subsystems.

SECTION 2
TERMINOLOGY

Acceleration (a). Acceleration is a vector quantity that specifies the time rate of change of velocity; both linear and angular. Common units are in/sec^2 (or cm/sec^2) and radians/sec^2.

Amplitude. Amplitude is the maximum value of a periodic quantity.

Angular Frequency (Circular Frequency). The angular frequency of a periodic quantity, in radians per unit time, is the frequency multiplied by 2π.

Balancing. Balancing is a procedure for adjusting the mass distribution of a rotor so that rotating imbalance, as seen by vibration of the journals or the forces on the bearings at once-per-revolution, is reduced or controlled.

Critical Speed. Critical speed is a speed of a rotating system that corresponds to a natural frequency of the system.

Damping. Damping is the dissipation of energy with time.

Displacement (d). Displacement is a vector quantity that specifies the change of position of a body or particle and is usually measured from its position of rest. Displacement is expressed in mils (1 mil = 0.001 inch) or millimeters (1mm = 10^{-3} meter).

Excitation. Excitation is an external force (or other input) applied to a system that causes the system to respond in some way.

Filter. A filter is an analog or digital device for separating signals on the basis of their frequency. It introduces relatively small loss to signals in one or more frequency bands and relatively large loss to signals at other frequencies.

Forced Vibration. The oscillation of a system is forced if the response is imposed by continuous excitation.

Foundation (Support). A foundation is a structure that supports the loads of a mechanical system. It may be fixed in space, or it may undergo a motion that provides excitation for the supported system.

Frequency (f). The frequency of a function periodic in time is the reciprocal of the period. The unit is cycle per unit time. The unit cycle per second is called Hertz (Hz).

Fundamental Frequency. The fundamental frequency of an oscillating system is the lowest natural frequency.

g. The quantity g is the acceleration produced by the force of gravity, which varies with latitude and elevation of the point of observation. By international agreement, the value 32.1739 ft/sec^2 = 386.087 in./sec^2 = 980.665 cm/sec^2 has been chosen as the standard acceleration due to gravity.

G. The ratio of local acceleration to the acceleration of gravity. For example, an acceleration of 38.6 in./sec^2 or 98.1 cm/sec^2 is written 0.1G.

Isolation. Isolation is a reduction in the response of a system to an external excitation; attained by the use of a resilient support.

Natural Frequency (fn). Natural frequency is a frequency of free vibration of a system. There are a large number of natural frequencies in complicated systems, though normally only a few have energy contents high enough to be of concern.

Oscillation. Oscillation is the time variation of the magnitude of a quantity (e.g., displacement).

Peak-to-peak Displacement (d_{pp}). The peak-to-peak displacement of a vibrating quantity is the algebraic difference between the extremes of the displacement. This is twice the amplitude of the sinusoidal displacement.

Peak Velocity. Housing or case vibration is normally measured in units of peak velocity, the maximum velocity occurring during the normal sinusoidal displacement.

Period. The time interval to complete one cycle of sinusoidal oscillation, usually expressed in seconds.

Resonance. Resonance of a system in forced vibration exists when the forcing frequency is at or near a system natural frequency. Any change, however small, in the frequency of excitation results in a decrease in the response of the system.

Rotating Speed (N). Rotating speed is the frequency at which the mechanical system rotates; generally expressed in revolutions per minute (RPM).

Simple Harmonic Motion. A simple harmonic motion is a motion such that the displacement is a sinusoidal function of time; sometimes it is designated merely by the term harmonic motion.

Steady-state Vibration. Steady-state vibration exists when period and amplitude are constant.

Stiffness. Stiffness is the ratio of change of force (or torque) to the corresponding change in translational (or rotational) deflection of an elastic element.

Subharmonic. A subharmonic is a sinusoidal quantity having a frequency that is a fraction of the fundamental frequency.

Synchronous Vibration. The filtered component of vibration at the frequency that corresponds to the machine rotating speed.

Transducer (Pickup). A transducer is a device which converts vibration or shock motion to an optical, a mechanical, or most commonly to an electrical signal that is proportional to a parameter of the experienced motion.

Velocity (v). Velocity is a vector quantity that specifies the time rate of change of displacement with respect to a reference time frame.

Vibration. Vibration is a term that describes oscillation in a mechanical system.

Vibration Meter. A vibration meter is an apparatus for the measurement of displacement, velocity, or acceleration of a vibrating body.

SECTION 3
VIBRATION ANALYSIS

3.1 Harmonic Motion. A system that experiences simple harmonic motion follows a displacement pattern defined by:

$$d = d_o \sin(2\pi ft) = d_o \sin(\omega t) \quad \text{[Eq. 3.1]}$$

where f is the frequency of the simple harmonic motion, $\omega=2\pi f$ is the corresponding angular frequency, t is time, and d_o is the amplitude of the displacement.

The velocity v and acceleration a of the system are found by differentiating the displacement d once and twice, respectively:

$$v = d_o(2\pi f)\cos(2\pi ft) = d_o\omega\cos(\omega t) \quad \text{[Eq. 3.2]}$$

$$a = -d_o(2\pi f)^2\sin(2\pi ft) = -d_o\omega^2\sin(\omega t) \quad \text{[Eq. 3.3]}$$

The maximum absolute values of the displacement, velocity, and acceleration of the system undergoing harmonic motion occur when the trigonometric functions in Eqs.[3.1] to [3.3] are numerically equal to unity. These values are known, respectively, as displacement, velocity, and acceleration amplitudes; they are defined mathematically as follows:

$$d_o = d_o \quad v_o = (2\pi f)d_o \quad a_o = (2\pi f)^2 d_o \quad \text{[Eq. 3.4]}$$

It is common to express the displacement amplitude d_o in mils when the English system of units is used and in millimeters when the metric system is used. The velocity amplitude v_o is expressed in inches per second in the English system (centimeters per second in the metric system). The acceleration amplitude a_o usually is expressed in Gs.

Factors for converting values of rectilinear velocity and acceleration to different units are given in Table A.1. Similar factors for angular velocity and acceleration are given in Table A.2.

For certain purposes in analysis, it is convenient to express the amplitude in terms of peak value, the peak-to-peak value, the average or the root-mean-square (rms) value. These conversion factors are set forth in Table A.3 for ready reference. Peak-to-peak displacement and peak velocity amplitude as a function of frequency are shown graphically, in English units, in Fig. A.1 and, in metric units, in Fig. A.2.

The peak-to-peak displacements in terms of simple harmonic motion are given below:

$$d_{pp} = 2\left(\frac{V_p}{2\pi f}\right) \qquad d_{pp} = 2\left(\frac{a_p}{(2\pi f)^2}\right)$$

$$d_{pp} = \left(\frac{V_p}{\pi f}\right) \quad \text{[Eq. 3.5]} \qquad d_{pp} = \left(\frac{a_p}{2(\pi f)^2}\right) \quad \text{[Eq. 3.6]}$$

In English units:

$$d_{pp} = 318.3\left(\frac{V_p}{f}\right) \quad \text{[Eq. 3.7]}$$

$$d_{pp} = 50.66\left(\frac{a_p}{f^2}\right) \quad \text{[Eq. 3.8]}$$

where,

d_{pp} = peak-to-peak displacement, [mils]
v_p = peak velocity, [in./sec]
a_p = peak acceleration, [in/sec^2]
f = frequency, [Hz]

In metric units:

$$d_{pp} = 3.183\left(\frac{V_p}{f}\right) \quad \text{[Eq. 3.9]}$$

$$d_{pp} = 0.5066\left(\frac{a_p}{f^2}\right) \quad \text{[Eq. 3.10]}$$

where,

d_{pp} = peak-to-peak displacement, [mm]
v_p = peak velocity, [cm/sec]
a_p = peak acceleration, [cm/sec^2]
f = frequency, [Hz]

The relationship between displacement, velocity and acceleration is shown graphically in Appendix B.

3.2 Concepts of Vibration. Vibration is a term that describes oscillation in a mechanical system. It is defined by the frequency (or frequencies) and amplitude. An excitation or oscillating force applied to the system is vibration in a generic sense. Conceptually, the ensemble or time-history of vibration may be considered sinusoidal or simple harmonic in form. Although vibration encountered in practice often does not have this regular pattern, it may be a combination of several sinusoidal quantities, each having a different frequency and amplitude. If the vibration ensemble repeats itself after a determined interval of time, the vibration is termed periodic. Mechanical systems experiencing forced vibrations continue under steady-state conditions because energy is supplied to the system continuously to compensate for that dissipated by damping in the system. In a free vibrating system, there is no energy added to the system but rather the vibration is the continuing result of an initial disturbance. In the absence of damping, free vibration is assumed to continue indefinitely.

In general, the frequency at which energy is supplied (i.e., the forcing frequency) appears in the vibration of the system. The vibration of the system depends upon the relation of the excitation or forcing function to the properties of the system. This relationship is a prominent feature of the analytical aspects of vibration. The technology of vibrations embodies both theoretical and experimental facets. Thus, methods of analysis and instruments for the measurement of vibration are of primary significance. The results of analysis and measurement are used to evaluate vibration environments, to devise testing procedures and instruments, and to design and operate equipment and machinery. The objective is to eliminate or reduce vibration severity or, alternately, to design equipment to withstand its influences.

3.3 Sources of Vibration. Several potential sources of vibration are discussed in the following paragraphs. The correspondence between observed vibratory response frequencies and the likely cause of that response is shown in Table 3.3.

3.3.1 Mass Unbalance

a. Dissymmetry due to core shifts in casting, rough surfaces on forging, or unsymmetrical configuration.

b. Nonhomogeneous Material. General observations include blowholes in castings, inclusions in rolled or forged materials, slag inclusions or variations in material density.

c. Eccentricity. Sources of vibration due to eccentricity include:

   1. Journals not circular or concentric to shaft.
   2. Bent or bowed shafts.
   3. Tolerances or clearances of rotating parts may allow eccentricities that result in unbalance.
   4. Non-concentric shaft and coupling interfaces.
   5. Non-uniform thermal expansion.

3.3.2 Misalignment

There are two basic types of misalignment: angular misalignment, where the center lines of the two shafts meet at an angle, and offset misalignment, where the shaft centerlines are parallel but displaced from one another.

Misaligned couplings or shaft bearings can result in transverse vibration (vibration perpendicular to the shaft). Flexible couplings with angular misalignment may produce an axial mode of vibration. This is especially prominent in slender, long shafts. Misalignment may result in large axial vibration.

A characteristic of misalignment and bent shafts is that vibration will occur in both radial and axial directions. In general, whenever the amplitude of axial vibration is greater than 50% of the highest radial vibration, then misalignment or a bent shaft should be suspected.

3.3.3 Flow induced. Pump vibration can occasionally be caused by flow through the system. The amplitude usually depends upon where the pump is operated on the head-capacity curve, and the cause of the vibration is usually turbulence. In diffuser type pumps certain combination of impeller blades and diffuser vanes are more likely to produce vibration than others. Although this phenomenon can produce vibration amplitudes that are unacceptable, especially at rates conducive to cavitation problems, testing indicates that when the pump is operated within its recommended operating range, the impact of turbulence is minimal. Non-symetrical fluid passages in a pump can induce hydraulic imbalance which may be seen as a once per revolution vibration. Multi-phase flow can also induce vibration.

3.3.4 Journal Bearing Oil Whirl. A condition caused by hydrodynamic forces in lightly loaded journal bearings that results in a vibration at slightly less than one-half (42-48%) the rotating frequency.

3.3.5 Bearing Rotation. Journal bearings that are not properly secured can rotate with the shaft and produce vibration at one-half rotating frequency.

3.3.6 Mechanical Rub. Contact between the rotating and stationary surfaces results in a vibration at a frequency normally 1/3 or 1/2 the operating speed. Natural frequencies may be excited.

3.4 Control of Vibration. Methods of vibration control may be grouped into three broad categories: reduction at the source, isolation of external sources, and reduction of the response.

Table 3.3 Vibration Analysis of ESP Phenomena

ESP (Machine) parts	Response Frequency relative to ESP rotating speed (rpm)	Probable causes of problem
Rotor and shafts	1 or 2 x rpm	Bent shaft
All rotating parts	1 x rpm	Mass or hydraulicunbalance or off center rotor
Couplings, shafts, bearings	Often 1 to 2 x rpm, sometimes 3 x rpm	Misaligned coupling and/or shaft bearing.
Sleeve bearing	Less than 1/2 x rpm	Oil whirl, lightly loaded bearing. More prominent in seal chamber section
Anti-friction bearing	Relatively high, > 5 x rpm	Excessive friction, poor lubrication, too tight fit
Mechanical rub	1/3 or 1/2 x rpm	Contact between stationary and rotating surfaces
Journal bearing rotation	1/2 x rpm	Journal rotating with shaft
Armature and electric motors	1 x rpm	Eccentric armature (either OD or journals)

3.4.1 Reduction at the Source

a. **Balancing of Rotating Masses.** Where vibration results from the unbalance of rotating components, the magnitude of the vibratory forces, and hence the vibration amplitude, can often be reduced by balancing.

b. **Balancing of Magnetic Forces.** Vibratory forces arising in magnetic effects of electrical machinery are minimized by proper design and fabrication of the stator and rotor; details of which are beyond the scope of this RP.

c. **Control of Clearances.** Vibration can result when ESP system components and parts, operating within the clearances that exist between them, strike each other or otherwise come into impact-type contact during operation. Vibrations from this source can be minimized by avoiding excessive bearing clearances and by ensuring that dimensions of manufactured parts are within acceptable tolerances.

d. **Straightness of Rotating Shaft.** Rotating shafts should be as straight as practical since lack of shaft straightness will have a large effect on system vibrations.

3.4.2 Isolation of External Sources.

Other machines or equipment, unless properly isolated, may transmit vibration to an ESP under test or in operation. For example, a horizontal pump delivering high pressure water may experience vibration interference from neighboring pumps and drivers through the foundation). Accepted practice is to avoid the structure's natural frequency by approximately 25% above or below.

Isolation of equipment being tested is the responsibility of the tester. Isolation of equipment in service is the responsibility of the user.

3.4.3 Reduction of the Response

a. **Alteration of Natural Frequency.** If a natural frequency of the system coincides with the frequency of the excitation, the vibration condition may be made much worse as a result of resonance. Under such circumstances, if the frequency of the excitation is substantially constant, it often is possible to alleviate the vibration by changing the natural frequency of such system. This generally involves modifying mass and/or stiffness of the system.

b. **Operating at non-Resonant Frequencies.** Sometimes ESPs are operated with variable speed drives. Operation at a frequency corresponding to a critical speed should be avoided to minimize damage to the system.

c. **Additional Damping.** The vibration response of a system operating at resonance is strongly related to the amount of damping present. Techniques are available to increase the amount of damping. The addition of damping decreases unit efficiency.

3.5 Vibration in ESP Systems. The potential for vibrational problems is inherent with any rotating equipment having an extreme shaft length-to-diameter ratio such as an ESP system; consisting of a motor, seal chamber section, gas separator, and pump(s) all connected by a small diameter, high strength, coupled shaft. Recognizing that all ESP machinery operates in some state of unbalance, a reasonable displacement amplitude for new equipment should be established to allow a margin for deterioration in service. Guidelines are set forth in the section following.

3.5.1 Vibration Modes. Vibration modes can be axial, lateral (transverse), torsional, or combinations of all three. Torsional vibration is known to be a potential problem particularly when starting and when changing speeds Axial and transverse vibrations on shaft seals and thrust bearings may be important under certain circumstances.

3.5.2 Critical Speeds. Torsional and lateral critical speeds exist in ESP systems. If possible, operation of the ESP near a critical speed for an extended period of time should be avoided. When this problem is identified over specific, planned rotating frequencies, alteration of the response may be in order and should be addressed. This problem may be particularly acute when the ESP is operated over a wide speed range or during start-up.

Torsional critical speeds for the ESP system are generally calculated for the system as an entity by the Holtzer method (Shock and Vibration Handbook 3rd Edition McGraw-Hill, page 38-9), and can be furnished by the manufacturer for a specific system configuration.

Lateral critical speeds for the ESP system "can be calculated using either eigenvalue or Myklestadt-Prohl Methods ("Theory of Vibration", William T. Thompson, Prentice-Hall Inc., Englewood, N. N., 1965, pg. 243.

SECTION 4
RECOMMENDATIONS

4.1 Vibration Limits. It is generally acknowledged that severe vibration can decrease ESP system run life. Appendix C contains general industrial guidelines for assessing vibration severity. Vibration limits for ESPs are given below.

For ESP systems or components, a maximum velocity amplitude of 0.156 in/sec (0.396 cm/sec) — peak at the intended synchronous operating frequency or over the range of intended operating frequencies, with no other individual frequency component greater than 0.100 in/sec (0.254 cm/sec) — peak as measured on the housing or case, is recommended. For pumps the vibration limit should be applied over the manufacturer's recommended operating flow range (Ref. API RP11S2, sec. 2.1.10)

Velocity measurements should be made in accordance with Section 4.2, following. The relationship between displacement, velocity, and acceleration amplitudes is given by equation 3.4.

Accuracy and calibration of the measuring system should be considered when establishing vibration test acceptance limits.

During testing, the frequencies of critical speeds should be recorded for future reference. Operation at critical speeds is not recommended.

4.2 Measurement of Vibration. The first step in establishing a condition-monitoring program for rotating machinery is to determine what shall be measured: shaft vibration or bearing vibration. When the shaft is accessible, then the displacement amplitude at the shaft is measured with a displacement transducer such as a proximity probe. For ESP system components, accessibility to the shaft is difficult and therefore one must rely on measurement of acceleration or velocity amplitudes at a suitable location on the component housing (bearing). Transducers have moving parts and thus may be orientation sensitive. They should be mounted in a manner that will provide accurate, repeatable measurements.

4.2.1. Transducers

4.2.1.1 Accelerometers generally exhibit excellent linearity of electrical output vs. input acceleration under normal usage; a dynamic range of 10,000 to 1 or more is not uncommon. Accelerometers are relatively insensitive to temperature and magnetic influences, and generally have a wide frequency response range (10-10,000 Hz). It is common to obtain the velocity and displacement amplitudes by mathematically integrating the acceleration signal. Below 15 Hz, integrated displacement values may be suspect.

4.2.1.2 Velocity probes measure velocity directly. These transducers have a useful frequency range of approximately 10-3000 Hz.

4.2.1.3 Proximity probes. Proximity probes, which could be mounted within the machinery housing to measure the relative displacement between the shaft and the housing, are not usually used to measure ESP vibration because of the difficulties associated with mounting them.

4.2.2 Selection of Measurement Location. Vibration measurements should be taken at several bearing locations along the ESP system component.

4.2.2.1 Pump

It may be advantageous to conduct the vibration test along with the pump acceptance test recommended in API RP-11S2 on ESP pumps. At a minimum measurements should be taken at the mid-point on the housing, top radial bearing location, and bottom radial bearing location. Pump rate should be held constant while measurements are being taken.

4.2.2.2 Gas Separator/Intake

At a minimum, measurements should be taken at the mid-point on the housing, top radial bearing location, and bottom radial bearing location.

4.2.2.3 Seal Section

At a minimum, measurements should be taken at the mid-point on the housing, top radial bearing location, and bottom radial bearing location

4.2.2.4 Motor

At a minimum, measurements should be taken at the mid-point on the housing, top radial bearing location, and bottom radial bearing location.

4.2.3 Drive Motor. Caution must be exercised in component testing. The drive motor will contribute to the measured vibration amplitude. To minimize this effect, a well balanced test driver should be used.

APPENDIX A
UNITS CONVERSIONS

Table A.1 Conversion Factors For Translational Velocity and Acceleration

Multiply Value in => or => By To Obtain value in	g–sec g	ft/sec ft/sec^2	in/sec in/sec^2	cm/sec cm/sec^2
g–sec g	1	0.0311	0.00259	0.00102
ft/sec ft/sec^2	32.16	1	0.0833	0.0328
in/sec in/sec^2	386	12	1	0.3927
cm/sec cm/sec^2	980	30.48	2.54	1

Table A.2 Conversion Factors For Rotational Velocity and Acceleration

Multiply Value in => or => By To Obtain value in	rad/sec rad/sec^2	degree/sec degree/sec^2	rev/sec rev/sec^2	rev/min rev/min/sec
rad/sec rad/sec^2	1	0.01745	6.283	0.1047
degree/sec degree/sec^2	57.30	1	360	6
rev/sec rev/sec^2	0.1592	0.00278	1	0.0167
rev/min rev/in/sec	9.549	0.1667	60	1

Table A.3 Conversion Factors For Simple Harmonic Motion

Multiply numerical Value in terms of => By to obtain value in terms of	Amplitude	Root-mean-Square value (ms)	Peak-to-peak value
Amplitude	1	1.414	0.5
Root-mean-square value (rms)	0.707	1	0.354
Peak-to-peak value	2	2.828	1

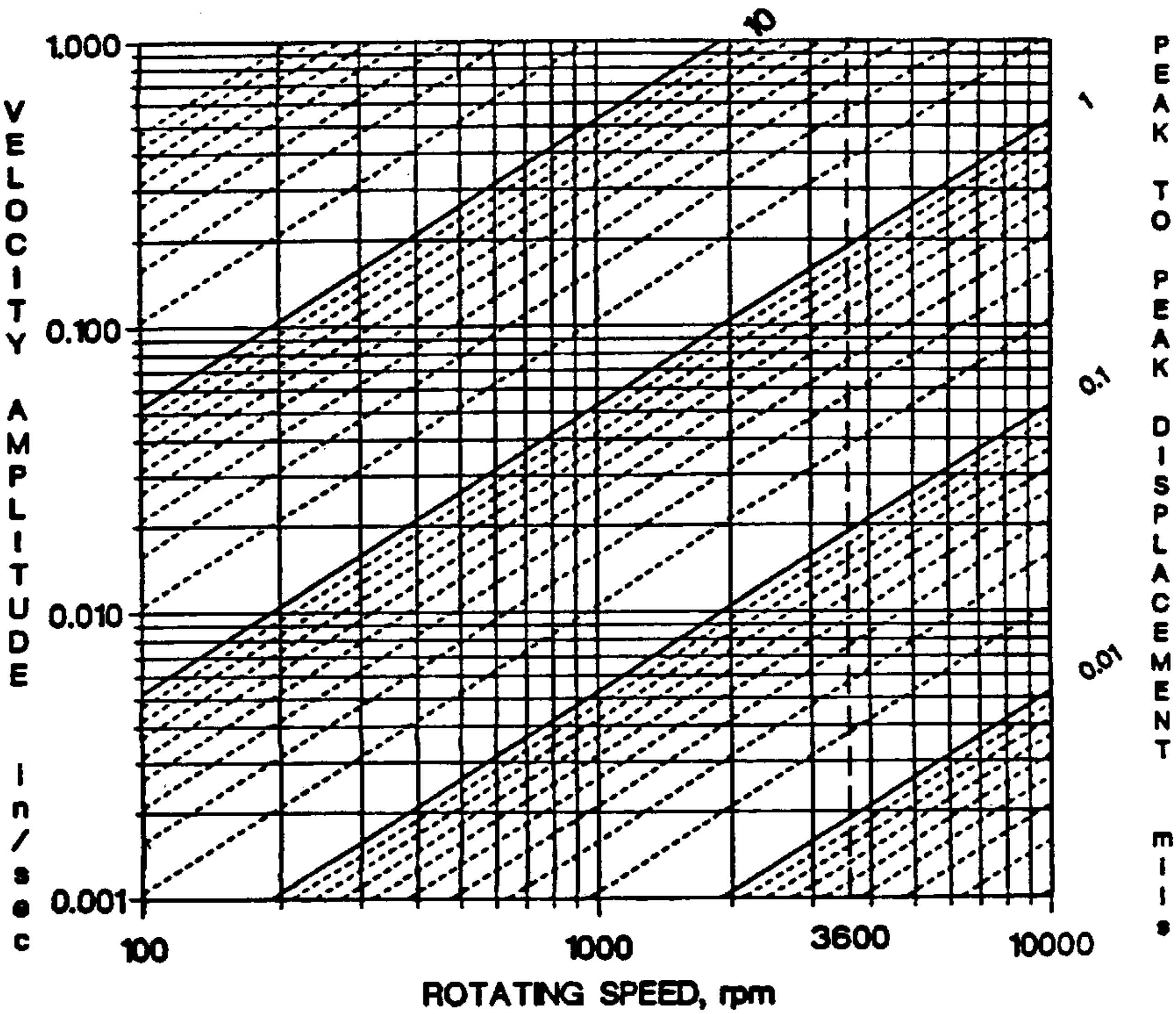


FIGURE A.1
RELATION OF FREQUENCY TO THE AMPLITUDES OF DISPLACEMENT AND VELOCITY
ENGLISH SYSTEM OF UNITS

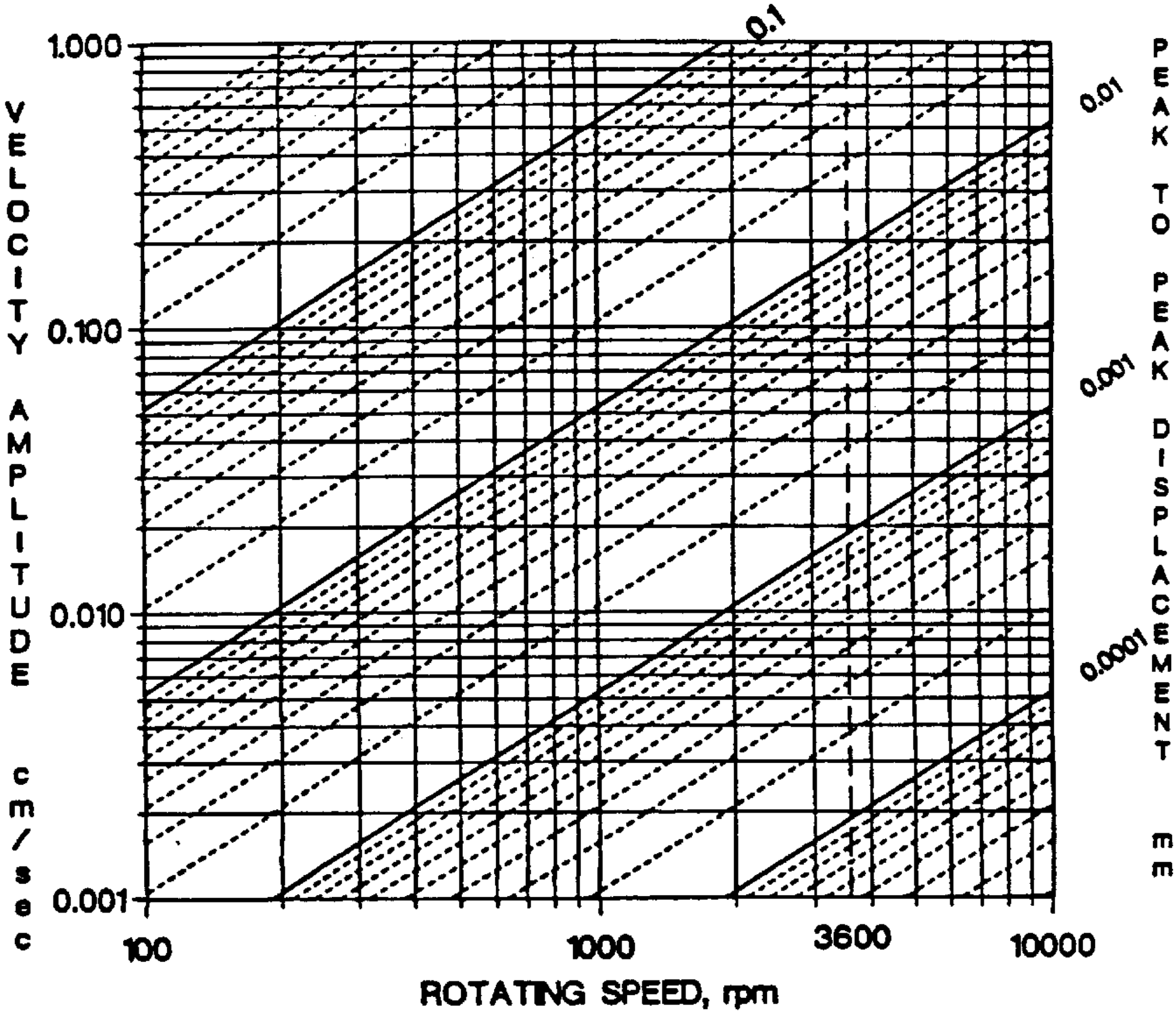
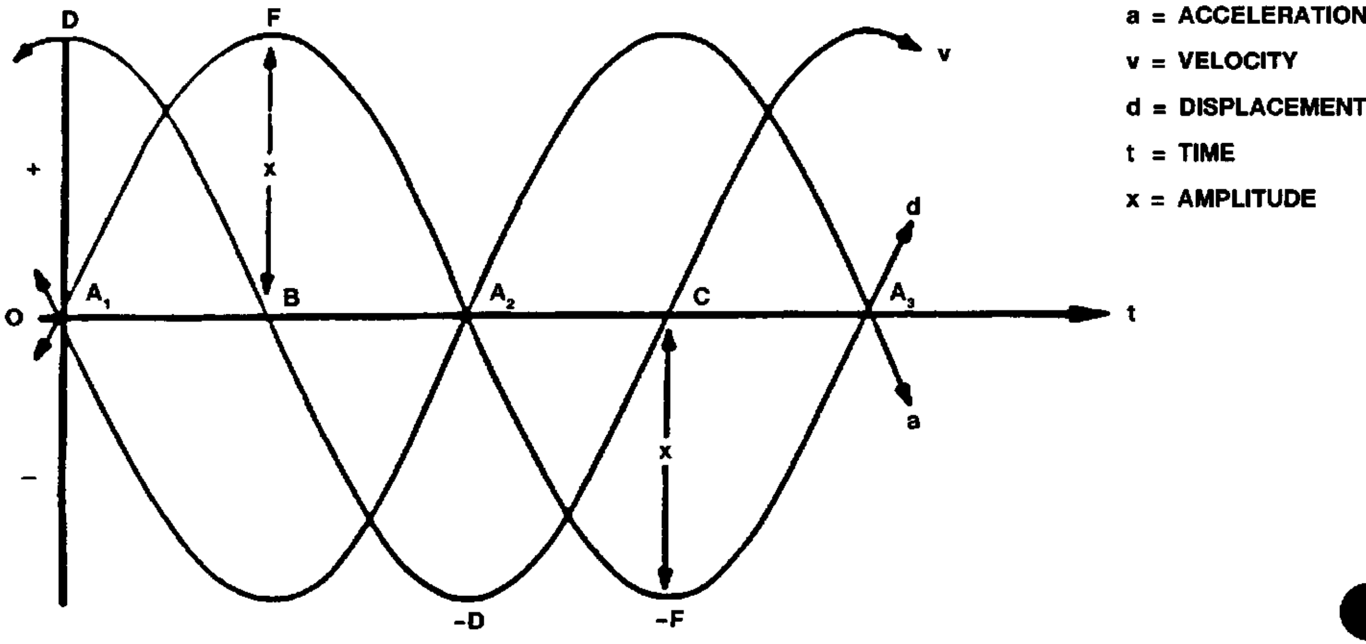


FIGURE A.2
RELATION OF FREQUENCY TO THE AMPLITUDES OF DISPLACEMENT AND VELOCITY
METRIC SYSTEM OF UNITS

APPENDIX B
RELATIONSHIP BETWEEN DISPLACEMENT, VELOCITY, AND ACCELERATION



At position **D** the velocity (v) is at its maximum; it approaches position **B** where the velocity is zero and continues in a negative direction to position **-D**, another maximum, then toward position **C**, where its velocity is zero. Then it continues on toward **D** at maximum velocity. The time interval for this total motion from **D** to **B** to **-D** to **C** to **D** is one **period**. Notice the **displacement (d)** curve. At A_1, the **displacement amplitude** (d_0) is zero; it approaches maximum amplitude at **F** and reverses direction toward A_2 again where it approaches zero. The displacement continues toward position **-F** or maximum amplitude again, and finally reaches one complete **period at** A_3. Notice that the **acceleration (a)** curve follows the same pattern as displacement, except that the curve is generated in the negative direction, because of the change in velocity direction **at B** and **C**. The **peak velocity** (v_p) is denoted each time it passes through position **D**. Displacement has two peak **amplitudes (x)**, namely at positions **F** and **-F**, and therefore, is measured in terms of **peak-to-peak displacement** ($d_{pp} = 2x$).

APPENDIX C
CLASSIFICATION OF SEVERITY OF MACHINERY VIBRATION

In the classification of severity of machinery vibration, the amplitude variable that is used (displacement, velocity, or acceleration) depends on the type of standard, the frequency range, and other factors. The International Standards Organization (ISO) has a special measure, **vibration severity**, which is defined as the highest value of the broad-band, velocity amplitude in the frequency range from 10 to 1,000 Hz as evaluated on the structure at prescribed points (bearing caps or pedestals).

ISO IS 2372. ISO standard 2372, applies to rotating machinery having rigid rotors and to those machines having flexible rotors in which bearing cap vibration is a measure of the shaft motion. Table C.1 lists the allowable vibration severity along with a corresponding peak-to-peak displacement amplitude calculated at 3,600 rpm.

IRD Mechanalysis Severity Criteria. An accepted guide for vibrational evaluation is shown in Table C.2 and is based on filtered reading taken on the machine structure or bearing cap.

Table C.1 Vibration Severity Criteria
(After ISO IS 2372,1974)

Peak velocity ranges of vibration severity		Vibration Severity	Peak-to-peak displacement amplitude @ 3,600 rpm	
(in/sec)	(cm/sec)		(mils)	(mm)
<0.014	<0.036	Extremely smooth	<0.074	<0.0019
0.028	0.071	Very smooth	0.148	0.0038
0.042	0.107	Smooth	0.233	0.0057
0.057	0.145	Very good	0.302	0.0077
0.099	0.251	Good	0.525	0.0133
0.156	0.396	Fair	0.828	0.0210
0.255	0.648	Sightly rough	1.353	0.0344
0.396	1.006	Rough	2.101	0.0534
0.622	1.580	Very rough	3.300	0.0838
>0.622	>1.580	Extremely rough	>3.300	>0.0838

C.2 Vibration Severity Criteria
(After Training Manual IRD Mechanalysis, Columbia, Ohio)

Peak velocity ranges of vibration amplitude		Vibration Severity	Generalization	Peak-to-peak displacement amplitude @ 3,600 rpm	
in/sec	cm/sec			mils	mm
<0.0049	<0.0124	Extremely smooth	Well balanced System	<0.026	<0.0007
0.0097	0.0246	Very smooth		0.051	0.0013
0.0195	0.0495	Smooth	Normal for new equipment	0.103	0.0026
0.0391	0.0993	Very good		0.207	0.0053
0.0784	0.1991	Good	Minor faults	0.416	0.0106
0.1560	0.3962	Fair	Correct to save wear	0.828	0.0210
0.3130	0.7950	Slightly rough	Cause major wear	1.661	0.0422
0.6270	1.5925	Rough	Premature breakdown	3.326	0.0845
>0.627	>1.5925	Very rough	Shut down	>3.326	>0.0845

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