SAE International™ **SURFACE VEHICLE INFORMATION REPORT**

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Ship Systems and Equipment—Hydraulic Systems—Noise Control

1. Scope—Hydraulic systems are used on marine vehicles for steering, vehicle control, and utility services. System components that generate and transmit noise are of concern. This SAE Information Report (a) addresses noise requirements which may apply to the hydraulic systems of ships and submersibles, and (b) identifies noise sources and techniques which may be used to reduce system noise. Noise of power sources (e.g., electric motors) and end items (e.g., steering linkages) is beyond the scope of this document.

1.1 Purpose—The purpose of this document is to summarize design information on how to reduce noise of on-board hydraulic systems.

1.2 Application—This document is applicable to the marine vehicles using hydraulic systems. This document does not apply to recreational watercraft.

1.3 Rationale—This document has been reaffirmed to comply with the SAE 5-Year Review policy.

2. References

2.1 Applicable Publications—The latest issue of the referenced documents should be used. Nothing in this document supersedes applicable laws and regulations unless a specific exemption has been obtained.

2.1.1 INDUSTRY PUBLICATIONS—Industry publications that have been adopted by the Department of Defense (DoD) can be ordered by DoD activities only from DODSSP, Subscription Services Desk, Building 4D, 700 Robbins Avenue, Philadelphia, PA 19111-5094, Tel: 215-697-2179, http://assist.daps.mil or http://stinet.dtic.mil.

2.1.1.1 SAE Publications—Available from SAE, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or 724-776-4970 (outside USA), www.sae.org.

SAE J1778—Ship Systems and Equipment—Recommended Practice for Hydraulic Fluid Selection
SAE AS5440—Hydraulic Systems, Aircraft, Design and Installation Requirements for

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Tel: 724-776-4970 (outside USA)
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2.1.1.2 *ASA Publications*—Available from the Acoustical Society of America, Standards and Publications Fulfillment Center, P.O. Box 1020, Sewickley, PA 15143-9998, http://asa.aip.org.

ASA 38—Guide for the Evaluation of Human Exposure to Whole-Body Vibration (ANSI S3.18)
ASA 47—Specification for Sound Level Meters (ANSI S1.4)
ASA 65—Specification for Octave, Half-Octave and Third-Octave Band Filter Sets (ANSI S1.11)
ASA 107—Guidelines for the Specification of Noise of New Machinery (ANSI S12.16)
ASA 111—Acoustical Terminology (ANSI S1.1)

2.1.1.3 *IMO Publication*—Available from International Maritime Organization, 4 Albert Embankment, London SE1 7SR, United Kingdom, Tel: +44-0-20-7735-7611, www.imo.org.

IMO Resolution A.468(XII)—Code on Noise Levels on Board Ships (part of Publication IMO-814E)

2.1.1.4 *ISO Publications*—Available from ANSI, 25 West 43rd Street, New York, NY 10036-8002, Tel: 212-642-4900, www.ansi.org.

ISO 4412-1—Hydraulic fluid power—Test code for determination of airborne noise levels—Part 1: Pumps
ISO 4412-2—Hydraulic fluid power—Test code for determination of airborne noise levels—Part 2: Motors
ISO 4412-3—Hydraulic fluid power—Test code for determination of airborne noise levels—Part 3: Pumps—Method using a parallelepiped microphone array
ISO 10767-1—Hydraulic fluid power—Determination of pressure ripple levels generated in systems and components—Part 1: Precision method for pumps
ISO 10767-2—Hydraulic fluid power—Determination of pressure ripple levels generated in systems and components—Part 2: Simplified method for pumps
ISO/TR 11688-1—Acoustics—Recommended practice for the design of low-noise machinery and equipment—Part 1: Planning
ISO/TR 11688-2—Acoustics—Recommended practice for the design of low-noise machinery and equipment—Part 2: Introduction to the physics of low-noise design

2.1.1.5 *NFPA Publication*—Available from the National Fluid Power Association, 3333 North Mayfair Road, Suite 211, Milwaukee, WI 53222-3219, Tel: 414-778-3344, www.nfpa.com.

NFPA/T2.7.2—Hydraulic Fluid Power—Pumps—Determination of Fluid Pressure Fluctuation Characteristics (to be replace with ISO 10767-2)

2.1.1.6 *SNAME Publication*—Available from the Society of Naval Architects and Marine Engineers, Publication Sales, 601 Pavonia Avenue, Jersey City, NJ 07306, Tel: 201-798-4800, www.sname.org.

SNAME TRB 3-37—"Design Guide for Shipboard Airborne Noise Control," SNAME Technical and Research Bulletin 3-37

2.1.2 U.S. Government Publications

2.1.2.1 *Military Specifications and Standards*—Available from DODSSP, Customer Service, Building 4D, 700 Robbins Avenue, Philadelphia, PA 19111-5094, Tel: 215-697-2179, http://assist.daps.mil or http://stinet.dtic.mil.

MIL-STD-740-2—Structureborne Vibratory Acceleration Measurements and Acceptance Criteria of Shipboard Equipment
MIL-STD-1474—Noise Limits
MIL-M-24476—Mounts, Resilient: Pipe Support, Types 7M50, 6M150, 6M450, 6M900, and 5M3500

2.1.2.2 Other U.S. Government Publications

2.1.2.2.1 CFR Publication—Available from internet site http://www.osha.gov or from U.S. Government Printing Office, Superintendent of Documents, Mail Stop: SSOP, Washington, DC 20402-9328.

29 CFR 1910.95—Occupational Safety and Health Act, 29 CFR 1910.95, "Occupational Noise Exposure"

2.1.2.2.2 OPNAVINST Publications—Available from internet site http://neds.nebt.daps.mil.

OPNAVINST 5100.19—Navy Occupational Safety and Health (NAVOSH) Program Manual for Forces Afloat, Vol. 1 - NAVOSH and Major Hazards - Specific Programs, CH B4 Noise Preservation
OPNAVINST 9640.1—Shipboard Habitability Program

2.1.2.2.3 NAVSEA Publications—Available from NAVICP, 700 Robbins Avenue, Attention: Customer Service, Code 0862, Philadelphia, PA 19111, with approval from NAVSEA 05L3, 2531 Jefferson Davis Highway, Arlington, VA 22242-5160.

NAVSEA S9073-A2-HBK-010—Navy Resilient Mount Handbook: A Users Guide of Design, Installation, and Inspection Information (NSN 0910-LP-000-5250)
NAVSEA S9078-AA-HBK-010/DIM—Navy Distributed Isolation Material (DIM) Mount Design Handbook (NSN 0910-LP-049-7200)

2.1.3 OTHER APPLICABLE REFERENCES

Harris, C.M., "Handbook of Noise Control," 3rd Edition, McGraw-Hill, Inc., 1991
Harris, C.M., "Shock and Vibration Handbook," 4th Edition, McGraw-Hill, 1996
Johnston, D.N. and Edge, K.A., "Simulation of the Pressure Ripple Characteristics of Hydraulic Circuits," Proc. I. Mech. E., Vol. 203, 1989, pp. 275-282
Kwong, A.H.M. and Edge, K.A., "Towards the Design of Quiet Hydraulic Circuits," Proc. of the 1994 ASME ESDA Conf., Vol. 8, Part B, July 1994, pp. 431-440
Piper, G.E., "Active Feedback noise Control of a Magnetic Bearing Pump," Noise Control Eng., J. 45 (2), Mar.-Apr. 1997
Shorin, V.P., "The Correction of Dynamic Characteristics of Hydraulic and Gas Lines of Automatic Systems," Fluid Power Transmission and Control—Proceedings of the 2nd International Conference, International Academic Publishers (PRC)/Pergamon Press (outside PRC), 1989
Skaistis, S.J., "Noise Control of Hydraulic Machinery," Marcel Dekker, Inc., 1988
Smith, S.E., "FPRC Experimental Survey of Fluid Power Pump Sound Levels," Fluid Power Research Center BFPR Annual Report No. 8, Paper No. P74-5, Oct. 1974
Taylor, R., "The Effects of Fluid Viscosity and Bulk Modulus on the Flow Ripple Produced by an Axial Piston Pump," Fluid Power Research Center 4th Annual Fire Resistant Hydraulics Conference Papers, 1984, 4:33-38
Vickers, Inc., "Noise Control in Hydraulic Systems," Vickers Document No. 510, August 1991, Vickers, Inc., P.O. Box 101, Jackson, MS 39206
Yeaple, F.D., "Fluid Power Design Handbook," 3rd Edition, Marcel Dekker, 1996

3. ***Definitions***—For definitions of other terms, see ASA 111.

List of Symbols:

B	bulk modulus of fluid
c	speed of sound
C	capacitance
f	frequency of oscillations
I	intensity of sound power
L	length of tubing
P	pressure of fluid
Q	flow rate
t	time of valve closure
v	velocity of fluid
V	volume of fluid
λ	wave length
ρ	density of fluid
dB	decibel

3.1 Sound—Sound is a disturbance which propagates through an elastic medium of bulk modulus (B) and density (ρ) at a speed (c) characteristic of the medium. See Equation 1.

$$c = \sqrt{B/\rho} \quad \text{(Eq. 1)}$$

The disturbance consists of a variation in pressure (ΔP) and associated fluid particle displacement, at a velocity (Δv). The variation in pressure and the rate of change in momentum across the disturbance are in dynamic balance. See Equation 2.

$$\Delta P = \rho c \Delta v \quad \text{(Eq. 2)}$$

It follows that (see Equation 3)

$$\Delta P / \Delta v = \rho c \quad \text{(Eq. 3)}$$

The product ρc is called the characteristic impedance of the fluid.

Table 1 lists typical values in engineering units.

TABLE 1—FLUID CHARACTERISTICS

	c (m/s)	ρ (kg/m^3)	ρc (Pa/(m/s))
Air, 20 °C, Sea Level	343	1.2	412
Typical aircraft petroleum-base fluid	1360	860	1.16×10^6
Water-glycol fluid	1358	1060	1.44×10^6
Sea Water, 15 °C	1507	1027	1.55×10^6

NOTE—The speed of propagation can be reduced by the elasticity of the tubing as much as 10%. SAE J1778 provides additional information for numerous hydraulic fluids.

3.2 Sound Intensity—Sound power transmitted per unit area. See Equation 4.

$$I = \Delta P \Delta v = \Delta P^2 / (\rho c) \quad \text{(Eq. 4)}$$

In SI units, I is measured in watt per square meter (W/m^2). At the threshold of hearing (see Equation 5):

$$I_o = 10^{-12} W/m^2. \qquad \text{(Eq. 5)}$$

3.3 Sound Pressure—Sound pressure (ΔP) is measured directly. In air, the threshold of hearing is (see Equation 6):

$$\Delta P_o = 20 \mu Pa, \text{ that is } 0.0002 \text{ dyne/cm}^2. \qquad \text{(Eq. 6)}$$

Since Δv is proportional to ΔP, the product $\Delta P \Delta v$ is proportional to ΔP^2 (see Equation 7):

$$I_o = (\Delta P / \Delta P_o)^2 \qquad \text{(Eq. 7)}$$

3.4 Decibel—A logarithmic scale for sound intensity levels is defined by Equation 8:

$$dB = 10\log(I/I_o) \qquad \text{(Eq. 8)}$$

When adding noises from different sources, it is their sound intensities that are added. The combined dB is then calculated.

In terms of sound pressure (see Equation 9):

$$dB = 20\log(\Delta P / \Delta P_o) \qquad \text{(Eq. 9)}$$

Table 2 illustrates typical values of the decibel scale.

TABLE 2—DECIBEL SCALE

dB	60dB	80dB	100dB	120dB
I	10^{-6} W/m^2	10^{-4} W/m^2	10^{-2} W/m^2	1.0 W/m^2
P	0.02 Pa	0.2 Pa	2 Pa	20 Pa

3.5 Sound Power—Total sound power radiated from a source is determined by measurement of the sound intensity at a specified radial distance. For example, ISO 4412-1 specifies a radial distance of 1 m for hydraulic pump noise rating.

3.6 Octave Bands—An octave is a range or "band" of frequencies such that the upper frequency limit is two times the lower limit. ASA 65 defines standard octaves. One-third octave band noise measurement is standard where higher frequency resolution is required. Figure 1 shows the noise contribution of a hydraulic pump with a fundamental frequency of 500 Hz. The sources of noise can often be identified by their fundamental and harmonic frequencies.

One-third octave band data are shown in Figure 1 by squares and standard octave band data by hexagons. Sound intensity of each standard octave band is the sum of the intensities of the component one-third octave bands.

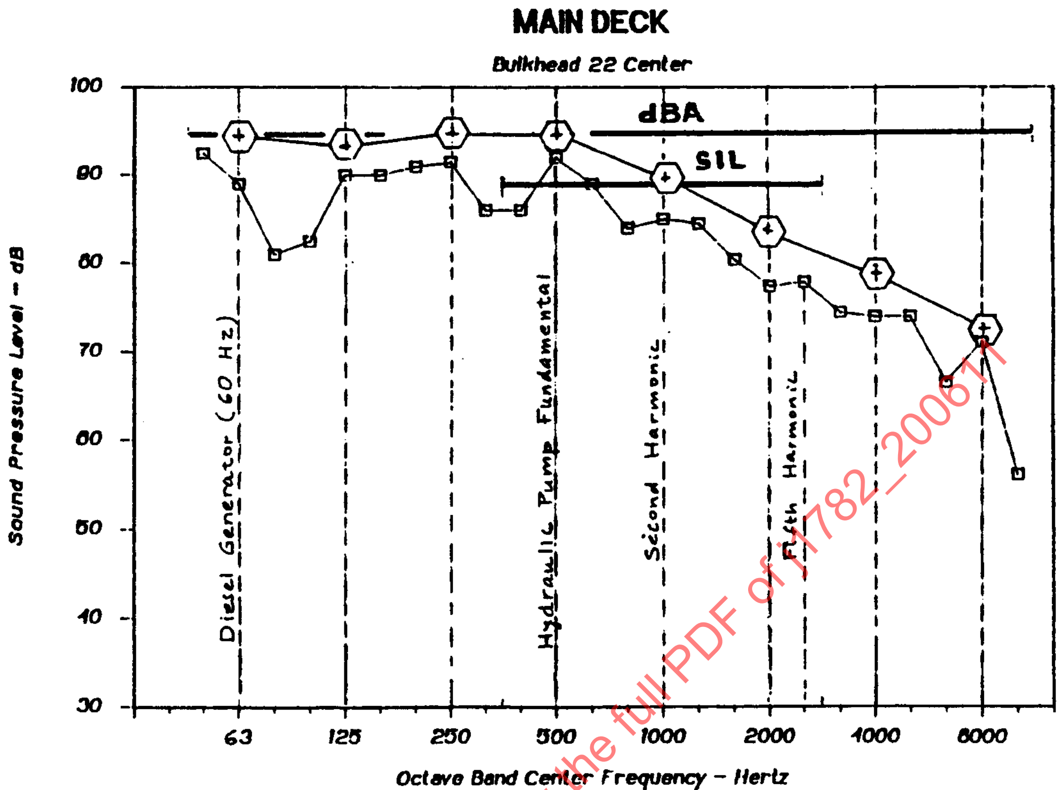

FIGURE 1—TYPICAL TEST DATA

3.7 Harmonics—Multiples of the fundamental frequency of oscillations. For example, the pump with the 500 Hz fundamental frequency also peaked at 1000 Hz, the second harmonic, and at 2500 Hz, the fifth harmonic. See Figure 1.

3.7.1 Subharmonics (for example, 1/2, 1/4, and 1/8 of the fundamental frequency of oscillations) sometimes exist.

3.8 Measurements in Audible Frequency Ranges

3.8.1 SPEECH INTERFERENCE LEVEL—SIL measures the effect of airborne background noise on intelligible speech communication. Numerically, SIL is the mean (not power average) of sound pressure levels (SPLs) in decibels for the octave bands of 500, 1000, 2000, and 4000 Hz. Refer to Figure 1. In the past, sometimes "PSIL" ("Preferred" SIL) was defined as the mean of SPLs for the octave bands of 500, 1000, 2000, and 4000 Hz whereas SIL was sometimes defines as the mean of SPLs for the octave bands of 500, 1000, and 2000 Hz.

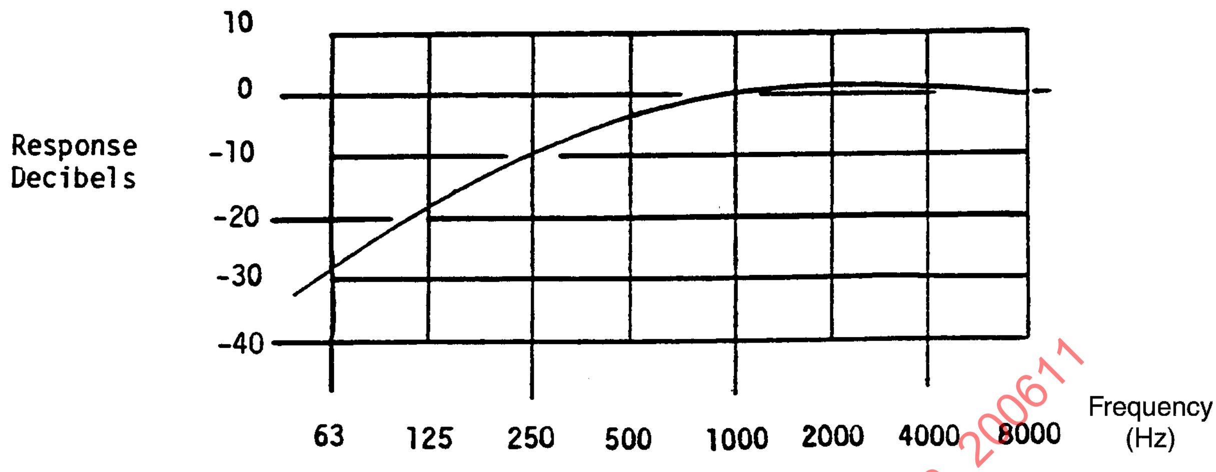

FIGURE 2—CHARACTERISTICS OF A STANDARD A FILTER

3.8.2 dB(A) SCALE—This is the sound level or volume scale providing a weighted average of audible sounds that reflects variable sensitivity of the human ear. The combined result is a single dB(A) number for specifying loudness. A filter is used for measuring sound on the dB(A) scale. See Figure 2. Attenuation correction factors are listed in ASA 47.

4. Requirements and Testing—Noise criteria vary with vehicle and application. It is important to design to well-defined noise requirements to avoid unnecessary penalties or redesign after the fact. Noise levels of hydraulic systems and components must support noise criteria for the applicable marine vehicle.

4.1 Fluidborne and Structureborne Noise—Where (a) structure-to-air noise coupling, (b) security of the vehicle from detection by radiated noise, or (c) the effects of noise on sonar performance (self-noise) is a concern, fluidborne (particularly for pumps) and structureborne noise requirements may be specified. ISO 10767-1, ISO 10767-2, NFPA/T2.7.2 (which is technically equivalent to ISO 10767-2), and MIL-STD-740-2 provide useful information for the verification of component fluidborne and structureborne noise requirements.

For vibrations transmitted from solid surfaces to the human body, ASA 38 provides exposure limits.

4.2 Airborne Noise—ASA 107, ISO 4412 (-1, -2, and -3), IMO Resolution A.468(XII), and MIL-STD-1474 provide useful information for the development and verification of vehicle compartment/space and component airborne noise requirements.

4.2.1 COMMERCIAL SHIPS

4.2.1.1 IMO Recommendations—IMO Resolution A.468(XII) provides guidance to Administrations regarding acceptable noise levels. See IMO Resolution A.468(XII) for scope of application. Examples of categories and noise limits are as follows:

Cabins, hospitals, and radio rooms (with radio equipment operating but not producing audio signals): 60 dB(A)
Navigating bridge, chartrooms, radar rooms, mess rooms, recreation rooms, and offices: 65 dB(A)
Listening post, including navigating bridge wings and windows: 70 dB(A)
Machinery control rooms, galleys without food processing equipment operating, serveries, pantries, and open recreation areas: 75 dB(A)
Workshops: 85 dB(A)

4.2.1.2 *Surface Effect Ships (SESs)*—For commercial SESs with displacements up to 145 Ltons, hydraulic system noise is usually not significant compared to drive system noise.

4.2.1.3 *Air Cushion Vehicles (ACVs)*—For commercial ACVs with displacements up to 30 Ltons, hydraulic system noise is usually not significant compared to drive system noise.

4.2.2 UNITED STATES MILITARY REQUIREMENTS

4.2.2.1 *Equipment Noise Limits*—MIL-STD-1474, Requirement 5 specifies equipment noise limits. These limits are specified in terms of sound pressure level measured close enough to the machine as to minimize the effect of the room in which the machine is tested. These limits are typically invoked for land-based (e.g., pre-installation) testing rather than shipboard testing.

4.2.2.2 *Compartment and Space Noise Limits*

4.2.2.2.1 Large Ships—For Navy ships and submarines over 150 ft in length or manned by 100 or more crewmembers, OPNAVINST 9640.1 specifies acceptable airborne noise levels. Older ships may have octaveband requirements in lieu of dB(A) requirements; newer ships may have requirements more stringent than OPNAVINST 9640.1.

4.2.2.2.2 Air Cushion Vehicles—Navy noise limits in the command module and personnel and equipment module are 92 dB(A) and 99 dB(A), respectively. Hydraulic system noise is usually not significant compared to drive or lift system noise.

4.2.3 NOISE EXPOSURE AND HEARING CONSERVATION—29 CFR 1910.95 is applicable to all ships operating under United States federal regulations. OPNAVINST 5100.19 is applicable to all United States Navy Ships. 29 CFR 1910.95 and OPNAVINST 5100.19 invoke, and IMO Resolution A.468(XLL) recommends, hearing conservation requirements for noise exposure. Harris' "Handbook of Noise Control" (2.1.3) addresses the effects of noise on speech, hearing loss, and physiology.

5. *Hydraulic Noise Sources*—Hydraulic system noise is typically dominated by pumps, motors driving the pumps, actuators, and fluid flow and cavitation. The main sources of noise should be identified first. A reduction of 3 dB results from reducing sound power level by 50%. ISO/TR 11688-1 and 11688-2 provide general guidance regarding the design of low-noise components.

5.1 Pumps

5.1.1 GENERAL—Hydraulic pumps are the main source of fluid power system noise on board marine vehicles, considering sound power and exposure duration. Factors affecting pump noise are:

a. Design requirements (pressure, flow rate, drive speed)
b. Type of design
c. Installation (inlet quality, outlet impedance, mounting)

Pump noise will significantly increase if cavitation occurs, for example, from insufficient suction pressure or excessive pump speed.

5.1.2 QUIET PUMPS—Helical screw pumps are generally, but not always, quieter than other pumps. These pumps require a rigid case and a viscous hydraulic fluid for minimum internal leakage. Some other pumps also have relatively low noise levels. See Smith (2.1.3) for a comparison of airborne and fluidborne noise levels of various pumps. See Yeaple (2.1.3) for an estimate of the effects of speed, pressure, and swept volume on the noise of axial piston pumps and vane pumps. See Taylor (2.1.3) for a discussion of fluid effects.

5.1.3 LIGHTWEIGHT PUMPS—Lightweight systems are of utmost importance for advanced marine surface vehicles. Typical system requirements call for a constant pressure of 21 MPa (3000 psi) and for a variable flow rate. Design solutions include:

a. Fixed-displacement pumps plus accumulators, or bypass regulation
b. Centrifugal pumps of 2 or 3 stages rotating at 18 000 to 30 000 rpm
c. Variable-displacement pressure compensated piston pumps at 1800 to 5400 rpm

Accumulator weight has caused rejection of fixed displacement pump systems on many advanced marine surface craft. Three-stage centrifugal pumps of 760 L/min (200 GPM) flow and 21 MPa (3000 psi) pressure were built for United States Navy Hydrofoil Program. The pumps were relatively silent, but the speed-increasing gears were a problem. Some surface effect craft are at present equipped with variable-displacement piston pumps.

Aircraft type piston pumps are noisy. Pump noise is due mainly to filling, compression, and discharge of discrete fluid volumes plus to a lesser extent to vibration of unbalanced rotating components. Poor porting can increase the severity of noise at each pumping cycle.

For a 9-piston pump operating at 3600 rpm, the shaft frequency is 3600 ÷ 60 = 60 Hz. The piston frequency is determined by the number of cylinders passing by the port plate in a given period of time: 9 pulse/rev x 60 rev/s = 540 Hz. The fundamental frequency of piston pump pressure ripple for a pump with an even number of cylinders is equal to piston frequency; for odd number of cylinders, pressure ripple fundamental frequency is two times the piston frequency (in this example, 1080 Hz). The piston noise is in the SIL frequency range where it is most objectionable.

A noise versus weight tradeoff exists in the selection of pump speed. A slower speed reduces noise and increases weight of the pump.

5.1.4 PUMP INSTALLATION—Mechanical isolation of pumps is particularly important to preclude “rebroadcasting” pump vibration. This should be in the form of:

a. Flexible hose at all hydraulic connections
b. Isolation mounts
c. Elastomeric section between coupling parts if possible.

5.2 Motors—Noise characteristics of hydraulic motors are similar to those of hydraulic pumps of the same type and power rating. Motors run generally only intermittently such as for turbine engine start, bow thrusters, propeller drive, or for winches. Intermittent-operating motors may need slow, quiet shutoff valves. Time on the order of 150 ms is typical for establishing pressure. Flow restrictors and servovalves can be used to reduce motor noise by reducing start-of-motion, end-of-motion, and steady-motion velocities.

5.3 Actuators—Bottoming hydraulic actuator pistons at end of stroke can cause a sharp percussion noise. The higher the actuation speed, the more probable the need for deceleration devices. These devices can reduce noise and also extend service life.

5.4 Valves—Valve noise increases as flow energy dissipation ($Q\Delta P$) in the valve increases. High-speed jets with pressure recovery downstream can result in gas-related or other broadband noise. High return-line pressure helps to prevent gas-related noise. Valves are relatively quiet when fully open or fully closed, with noise peaking at partial valve opening. Reduced flow velocities through any one metering path in the valve result in reduced noise.

Pressure-control valves may become unstable if fluid viscosity is greatly reduced, such as at high temperatures, or in the presence of entrained air.

5.5 Reservoirs—A hydraulic reservoir is an ideal noise sink. However, a gas-over-fluid type reservoir could become a source of entrained gas if not properly designed and pressurized. In addition, in a gas-over-fluid type reservoir, the fluid will become saturated with dissolved gas over time. Return fluid should be introduced below the lowest reservoir fluid level to be encountered in service. There should be adequate baffling to prevent vortex formation at the reservoir outlet, and possibly screens for gas bubble separation. A gasless reservoir which incorporates provisions for removal of dissolved and entrained gas or which is installed in conjunction with a gas-fluid separator will tend to avoid these possible problems. See Skaistis (2.1.3) for additional information.

5.6 Hydraulic Lines

5.6.1 WATER HAMMER—Fast-acting shutoff valves, such as direct-acting solenoid valves, can set up reverberating water hammer in long hydraulic lines. The incremental pressure (ΔP) which can be reached for instantaneous valve closure in a rigid pipe is shown in Equation 10:

$$\Delta P = \rho c \Delta v \qquad \text{(Eq. 10)}$$

where:

ΔP = overpressure (Pa)
ρ = density, per Table 1
c = speed of sound, per Table 1 (less for elastic walled tubing)
v = variation in flow velocity (m/s)

The instantaneous closure equation applies up to valve closure time (see Equation 11):

$$\Delta t = 2L/c \qquad \text{(Eq. 11)}$$

where:

L = length of pipe

It is recommended that velocities be kept under 5 m/s for ΔP_{max} = 5.8 MPa (840 psi).

The design criterion to preclude water hammer is shown in Equation 12:

$$\Delta t \geq 6L/c \qquad \text{(Eq. 12)}$$

Transient pressure is then given in semi steady-state conditions by Equation 13:

$$\Delta P = \rho L \frac{dv}{dt} \qquad \text{(Eq. 13)}$$

which yields pressures mild compared to water hammer ($\rho c \Delta v$).

5.6.2 LINE CAVITATION—Water hammer will occur if continuity of the hydraulic fluid is interrupted by a vapor cavity or entrained gas. Flow transients can be set up, for instance, in hydraulic lines of hydraulic flight control actuators on automatically stabilized vehicles, when the vehicle is subjected to turbulence. Transient pressures may be relatively mild, but sufficient for lowering return-line pressure below reservoir pressure to vapor pressure, and result in a vapor pocket. When the vapor pocket is pressurized it collapses suddenly. The resulting water hammer pressure is given by Equation 14:

$$\Delta P = \rho c \Delta v \qquad \text{(Eq. 14)}$$

same as for sudden valve closure. A sharp noise is heard.

5.7 Selection of System Pressure—Increased system pressure will increase pump and system noise. Valve, motor, and actuator transient noise will increase with system pressure due to higher differential pressure.

6. Noise Propagation and Attenuation—Noise may propagate from its source through fluid, structure, and air to inboard spaces (as airborne noise) or to the hull and into the water. Noise propagation through any path can be attenuated by reflection or absorption of acoustic waves. Where reflection is used, the designer should ensure that the reflective path is not an equally undesirable path of noise propagation. Noise attenuation devices and system components (e.g., reservoirs and accumulators) may be designed to attenuate noise, and distribution piping (a transmission path) must be designed with sufficient isolation. All contributing paths and acoustic interactions must be considered to balance the acoustic design of the system. Skaistis (2.1.3) and SNAME TRB 3-37, provide guidance regarding noise attenuation.

6.1 Fluid Borne Noise—Noise propagates in hydraulic fluid by pressure waves along the tubing.

6.1.1 PUMP RIPPLE—High noise levels can result from even relatively mild pump pressure ripple. Pump discharge may be analyzed as the sum of a steady flow plus a superimposed pulsating flow. Steady-state pressure is determined by load resistance, and pressure pulsation is determined by dynamic load impedance, with pipe and wave lengths and compressible volumes as design factors.

6.1.2 STANDING WAVES IN HYDRAULIC LINES—Standing waves can be avoided by design. Organ pipe resonance occurs if a line from the pump has an abrupt change at a one-quarter wave length, and resonance may recur at one-half wave length intervals. Wave length is given by Equation 15:

$$\lambda = c / f \qquad \text{(Eq. 15)}$$

6.1.3 CAPACITANCE—A compressible volume which accommodates an incremental volume of a fluid ΔV with a pressure rise ΔP is a “capacitance.”

By definition (see Equation 16):

$$C = \Delta V / \Delta P \qquad \text{(Eq. 16)}$$

Frequency response of a capacitance varies with type. A passive fluid volume is effective at higher frequencies only. A piston-type accumulator is effective for flow transients below audible frequencies only. A pneumatic charge (in-line flexible membrane or bladder type pulse damper) is effective over a broad frequency band. A given pipe arrangement tuned to a specific frequency will perform at discrete frequencies including the fundamental and its odd harmonics.

6.1.4 PASSIVE PULSATION ATTENUATORS—A flexible hose of good installation design is the simplest means of isolating noise at its source. However, flexible hose radiates noise and is undesirable for routing through occupied spaces. Vickers (2.1.3) identifies preferred flexible hose configurations. Coiled tubing attenuates pulsations and noise also. Skaistis (2.1.3) addresses fluidborne noise attenuation of flexible hoses and other devices.

A hydraulic fluid filter is an available means of passive ripple damping at no additional equipment weight. To be effective as a ripple damper, the filter should be near the pump, if possible.

A fluid-filled sphere built integrally with the pump case acts as a Helmholtz resonator. It has the advantage of stopping most of the pulsations at the source, and requires no maintenance. Diameter of the sphere, however, limits frequency band width. A larger diameter sphere would weigh more.

Suction, return, and case-drain line pulsations are lower in magnitude than pressure-line pulsations. Attenuation may be required depending on line routing.

Shorin (2.1.3) addresses in-line dampers utilizing both reactive (reflective) attenuation via damper geometry and dissipative (absorptive) attenuation via porous throttles.

Location of attenuation devices should be optimized with regard to peak resonance points of standing waves within the system.

6.1.5 GAS-BACKED PULSATION DAMPERS—A nitrogen-charged bladder-type accumulator with flow through its fluid chamber has been used on hydrofoils to reduce noise. Broad-band noise attenuation is excellent. Charge maintenance is periodic and bladder replacement unscheduled. Loss of charge reduces effectiveness to nil. Appendix A identifies computer software for pulsation damper sizing.

6.1.6 MUFFLERS—Pipe arrangements relying on multiple reflections and/or interference on noise attenuation are very effective at their design frequency. A QUINCKE tube splits flow in two branches, one branch one-half wave length longer than the other. At the junction of the two branches, pressure waves are 180 degrees out of phase, so ideally they cancel. See Johnston and Edge (2.1.3) for a performance assessment.

6.1.7 CAVITATION—Adequate positive pressure vapor pressure is mandatory to preclude cavitation. Options to prevent return line cavitation include accumulators, pressurized reservoirs, and adequate return line diameter-to-length ratio. Options to prevent pump suction line cavitation include pre-charge pumps and pressurized reservoirs. For pump suction lines, minimizing suction line length and maintaining suction velocities below 1.2 m/s (4 ft/s) are recommended unless an inlet charge system is employed.

6.2 Structure Borne Noise

6.2.1 NOISE TRANSMISSION—Noise, vibrating hull structures, and radiated noise induced by hydraulic systems are a source of concern. Hydraulic pressure fluctuations temporarily deform the walls of hydraulic lines and system components and disturb the air at the outside wall surface. Noise intensity transmitted to the air is attenuated because air has a density times speed constant some three thousand times less than the hydraulic fluid. Airborne noise-radiating area of hydraulic lines and components is relatively small. Most noise is in fact transmitted from the hydraulic system first to structure and then radiated from bulkheads and decks into occupied spaces. This is particularly true of the lightweight aluminum construction used on advanced marine vehicles.

For calculation of noise transmission through ship structures, empirical data vice theoretical analysis is often used. SNAME TRB 3-37 provides some relevant data.

6.2.2 BULKHEAD AND DECK PENETRATIONS—Bulkhead unions can transmit longitudinal oscillations of hydraulic lines unbending and stretching under internal pressure pulses. This can be a noise problem far more severe than transverse pulsation or hydraulic lines at clamps.

Numerous watertight penetrations exist on marine vehicles, with an added requirement of retaining watertight integrity during a fire. Acoustic isolation of the hydraulic lines must be accomplished without degrading the ship's fire resistance. 230 °C (450 °F) resistant elastomers or metal diaphragms are suggested for new design.

6.2.3 HYDRAULIC LINE CLAMPS—The function of hydraulic line clamps is to support piping or tubing. Resilient clamps are required for acoustic isolation. Soft mounting blocks are preferred for lines connected to high noise sources over simple clamps which are lighter in weight. SAE AS5440 identifies hydraulic line support spacing as a function of tube size and tube material for aircraft-type hydraulic systems. MIL-M-24476 specifies resilient mounts with proven shipboard experience in pipe hanger applications.

6.2.4 RESILIENT MOUNTS—Hydraulic pumps and motors should be vibration isolated from structure by resilient mounts whenever practicable.