

The CFD Analysis of Pressure Pulsation in the Aircraft Engine and Control Systems Lubrication Pump

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ABSTRACT

Fluid pressure pulsation in a fluid system is an inherent consideration in applications such as aircraft engine and control systems where mechanical component fatigue life and flow performance are critical. Positive displacement pumps transmitting fluid through hydraulic lines under high pressure impart periodic flow pulses to the fluid which can induce undesirable pressure ripple. Some failures of advanced aircraft prototype hardware were traced to a break in the hydraulic component of the control system due to severe localized responses to periodic pressure pulsations produced by a pump flow-induced ripple at the system resonant frequency. This response is associated with a strong structural fluid resonance that is not sufficiently damped by fluid leakage internal to the aircraft hydraulic system. In the case of pumps or hydraulic motors the main source of pulsation energy is in the flow-induced pressure wave associated with the system plumbing pressure pulsations. The pressure wave may propagate in both the pumping wall and fluid as a result of fluid/structure coupling. For a high level of pressure pulsation, the resulting pumping wall motion can then cause mechanical fatigue and unwanted radiated noise and cavitation. This report describes the application of CFD analysis in an aircraft engine control system's vane pumps with a non-intrusive fluid wave in the system. The ultimate goal is to understand the root cause of the pressure pulses and to define improvements or corrective action.

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INTRODUCTION

During the design phase of prototype engine control systems there is a Design Requirements Document (DRD) that defines the hydraulic system pressure pulsation constraint to prevent any mechanical components structural fatigue failure. Modern aircraft trends are to design smaller, lighter weight, and higher efficiency mechanisms which create growing challenges in the development process. Most of the hydraulic pumps applied in aircraft engine and control systems are positive displacement pumps. These pumps deliver flow with inherent flow ripple containing kinetic energy and compressible fluid with finite stiffness from some amount of entrapped air/fluid. This flow ripple can interact with the pumping paths and create a pressure wave which propagates through the connected pipeline system and if the section of the plumbing path changes through control servo valves or other intermittencies which can interrupt the pressure wave. This undesired wave will be partially reflected back in the flow source in conjunction with the pressure wave is created. The pressure pulsation adding in pumping element

hydraulic force may create severe pressure spikes as well as cavitation. This can occur when the local lower bound dynamic pressure falls below the true vapor pressure. The air and gas vapor will then be released from the fluid. When the downstream pressure subsequently rises, the air/gas bubbles implode and pressure shock waves are created causing major fatigue and erosion damage. Cavitation can occur in any area of the pumping system and is rather difficult to analyze without applicable numerical transient formulation and computational tools. However, the pressure ripple can be measured with appropriate pressure probes to identify the possibility of cavitation in the system.

A vane pump is a positive displacement rotary pump which has gained widespread acceptance for engine oil systems due to its simplicity and versatility in design and manufacturing. The pumping chambers between rotor and cam in the vane pump introduce flow ripple leading to fluid pressure pulsations. The vane and port plate suction and discharge port geometry may also have an influence on the performance, fluid power, and volumetric efficiency of the pump. Although several theoretical and experimental

investigations [1], [2], [3] have been reported in literature about these factors, a detailed design and analysis is still necessary to examine the principle drivers of system pressure ripple pulsations. Though some of the basic principles may be analogous, dual-vane flow-induced system pulsations are not comparable to small pressure perturbations described by basic acoustic theory.

This present work, a study was carried out to investigate the flow dynamic and pressure ripple in a Lube and Scavenge Pump (LSP) utilized for a generator. The oil lubrication system schematic diagram is shown in [Figure 1](#). This oil system consists of an oil reservoir, a dual-vane lube pump with pressure regulating valve, a filter, three pressure transducers (P1, P2, and P3), a Fuel Oil Heat Exchanger (FOHE), an electric generator with stator and rotor, and a scavenge pump. The dual-vane pump is driven through an engine mounted accessory drive gearbox. The transient CFD model was developed in this dual-vane pumping system from an oil reservoir through a fuel-oil heat exchanger then to the generator. In order to understand the fluid flow dynamic performance, to prevent any pressure pulsation damage and to reduce the product development cycle time becomes a challenge; since the distinguishing pressure pulsations can be excited by the dual-vane pumps which cause flow-induced pressure waves that may resonate the pulsation amplitudes with the response of the system modules.

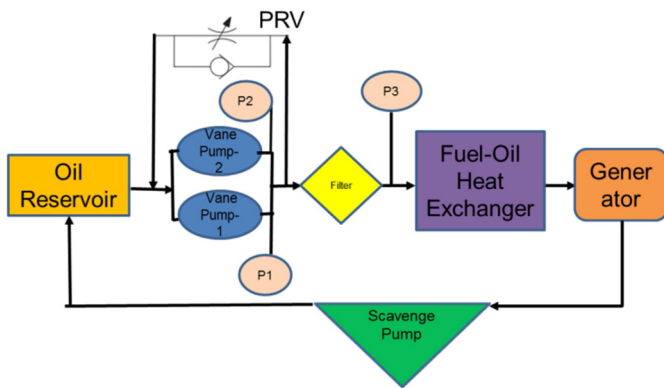


Figure 1. Schematic diagram of oil lubrication system

The Fluid Equation in Oil Lubrication System

For many years the aerospace industry has relied on the basic equation of motion [Equation-\(1\)](#) and mass continuity [Equation-\(2\)](#) with partial acoustic-wave theory [Equations-\(3\)](#) and [\(4\)](#) to form the basis of prediction of flow-induced pulsations with an empirical technique to determine hydraulic control system resonances [4]. This was initially accomplished using an assumption of the acoustic theory, frequency wave equation [Equation-\(5\)](#) with those of a fully fluid dynamic model of volumetric flow rates seen in [Equations-\(6\)](#) and [\(7\)](#). Explicitly, these transient pressure wave equations were difficult to solve using analytical

methods or a one-dimensional numerical tool simulated based on the functionally governing equations when 3D geometry shape are included in the factors of the pressure waves.

$$m \frac{d^2 y}{dt^2} + c \frac{dy}{dt} + k y(t) = F(t) \quad (1)$$

Where:

m : mass of the fluid including gas and liquid.

$F(t)$: the force induced by flow rate, or inertia, or any other effect which causes flow force.

$$m_i - m_o = \frac{d}{dt} (\rho V) \quad (2)$$

Where:

m_i, m_o : Inlet and outlet mass flow in controlled system

V : Controlled volume

ρ : Fluid density

$$c = \sqrt{\frac{\beta}{\rho}} \quad (3)$$

Where:

C : Acoustic velocity

β : Fluid adiabatic bulk modulus

ρ : Fluid density

$$c_M = \frac{c_o \left(1 + \frac{V_{gas}}{V_L}\right)}{\sqrt{1 + \frac{\beta_e V_{gas}}{P_L V_L}}} \quad (4)$$

Where:

c_M : Acoustic velocity in mixture

c_o : Acoustic velocity in liquid

β_e : Liquid bulk modulus

P_L : Line pressure

V_{gas} : Gas volume

V_L : Liquid volume

$$f_N = N \frac{c}{2L} \quad (5)$$

Where:

f_N : Nth natural frequency

L : Length of pipe

C : Acoustic velocity in pipe

N : 1,2,3, etc.

$$Q_i - Q_o = \frac{dV}{dt} + \frac{V}{\beta} \frac{dP}{dt} \quad (6)$$

$$\beta = - \frac{dP}{dV/V} \quad (7)$$

Where:

Qi, Qo: Inlet flow and outlet in controlled system

V: Volume

P: Pressure

In the oil lubrication fluid system model, there are five inherent factors affecting the flow and pressure ripple level. (1) The vane pump cam contour, arrangement of suction and discharge port configuration, and the dimension of a timing notch in the discharge port, (2) the presence of entrapped air in the suction fluid, (3) the pump operating speed and temperature, (4) piping length and stiffness, and (5) pressure regulating valve stability. Since the system operating conditions have been mostly defined by prior engineering detail design, we are content with items (1) and (2) for pumping system design performance evaluation. Therefore, we developed experimentally the 3D computation fluid dynamic numerical study relation between the pumping mechanism, the pressure ripple, and the reduction method of the pressure ripple.

Fluid property in hydraulic system

The oil system geometry, pump operating speed, fluid viscosity bulk modulus and density are important features in solving these hydraulic system governing equations; hence they can influence the dynamic behaviors of the system. The most common fluids used in the oil system are Mil-PRF-7808 or Mil-PRF-23699 [5] oil which fluid properties are shown in Table 1. The bulk modulus of non-aerated hydraulic oil is a function of temperature and pressure and for oil Mil-PRF-23699 ranges from 738 to 1910 MPa at temperatures between 149 °C and 37.8 °C.

Table-1. Fluid properties of Mil_PRF_23699

Temperature °C	True Vapor Pressure (Pa)	Density (kg/m ³)	Kinematic Viscosity (cStoke)	Bulk Modulus (Pascal) 0.1 MPa	Bulk Modulus (Pascal) 20 MPa	Bulk Modulus (Pascal) 34.5 MPa	Bulk Modulus (Pascal) 55.2 MPa
37.8	0.069	988	27.5	1434	1613	1737	1910
93.3	0.379	946	5.9	1082	1269	1386	1572
148.9	3.172	903	2.5	738	924	1048	1227

Aircraft can fly at an attitude with pitch and roll angles that may introduce air into the hydraulic system oil even with a fully functional LSP. Air-bubbles in the fluid can cause system performance issue and introduce cavitation, which erodes the system mechanisms, and cause unfavorable pressure ripple. Moreover, out of control system pressure due to the entrapped air was mostly affected by the bulk modulus value and density. The effective bulk modulus shown in Equation-(8) from Noah D. Manring [6] and Herbert E. Merritt [7] gives the effective bulk modulus β_e of fluid-air mixture in a flexible container.

$$\frac{1}{\beta_e} = \frac{1}{\beta_f} + \frac{1}{\beta_h} + \frac{V_a}{V_t} * \frac{1}{\beta_a} \quad (8)$$

Where the subscripts a, f and h refer to air, fluid, and hose respectively.

Normally, the bulk modulus of the fluid β_f can be obtained from the manufacturer, but the effective bulk modulus will not be available from the supplier database due to its complexity. Applying adiabatic bulk modulus used for air by $(C_p/C_v)P = 1.4P$, Equation-(8) can be rewritten as in Equation-(9),

$$\frac{1}{\beta_e} = \frac{1}{\beta_f} + \frac{1}{\beta_h} + \frac{s}{1.4 * P} \quad (9)$$

Where:

s: is entrapped air percent in the total volume ($s = V_a/V_t$).

Neglecting the stiffness term in the steel housing and piping structures, the effective bulk modulus in Equation-(9) varies from 0% to 5% of entrapped air in oil and is plotted in Figure-2. This plot shows that a small amount entrapped air severely impacts the effective bulk modulus. As Noah's [6] comments, there are several unknowns in Equation-(9) that must be discussed further, in particular, useful expressions and measurement for the effective bulk moduli must be developed and were discussed by many of researchers [8] [9], [10]. Nevertheless, applying of Equation-(9) into a CFD boundary condition can produce a faulty solution and deceive the component design engineer. The composite effective bulk modulus formulation with density function is presented in the next section of "solving the system equation and computer simulation". The evaluated aerated fluid density Mil-PRF-23699 oil is shown in Table:2.

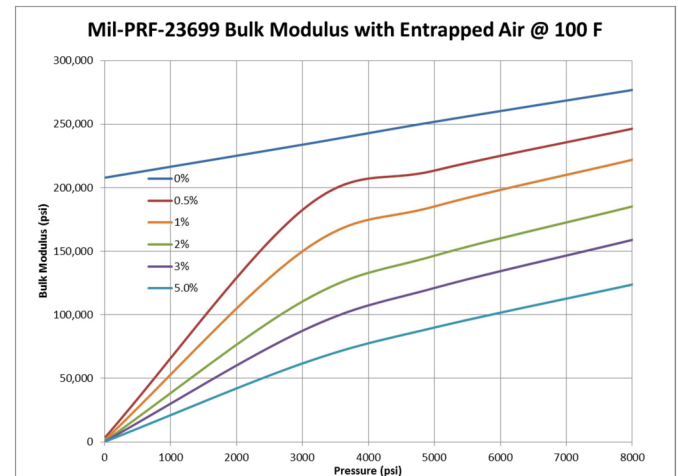


Figure 2. oil-23699 bulk Modulus with Fraction of Entrapped Air