TORSIONAL VIBRATION ANALYSIS OF A MULTI-BODY SINGLE CYLINDER INTERNAL COMBUSTION ENGINE MODEL

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Abstract: This paper presents a detailed multi-body numerical nonlinear dynamic model of a single cylinder internal combustion engine. The model comprises all rigid body inertial members, support bearings, joints, couplers, and connections between the various engine components, as well as means of vibration damping. The detailed model is parameterized, thus enabling virtual prototype testing of various engine designs, as well as allowing the engine designer to carry out a comprehensive noise, vibration, and harshness (NVH) investigation of engine performance. This new approach in engine design reduces the conceptual design-to-development cycle time and removes the need for extensive engine testing, which accounts for a considerable cost in engine design and development process. The model incorporates the simultaneous solution of large displacement dynamics of engine components, infinitesimal vibrations of support bearings, and the trapped air-fuel cylinder transient pressures. The simultaneous solution of nonlinear inertial rigid body dynamics, the combustion process, and nonlinear vibrations of the crankshaft main journal bearing supports brings about a new approach in the numerical analysis of complex nonlinear multi-body dynamic problems. Frequency domain analysis of the results shows agreement with generally-known experimental spectra. The numerical results also agree with a closed-form analytical solution reported by various authors.

Keywords: engine dynamics, multi-body system simulation, engine NVH

1. INTRODUCTION

Torsional vibrations of reciprocating engines arise due to the application of periodic combustion forces in the cylinder and the associated inertial forces of rotating

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and articulating members such as the crankshaft, the camshaft, and the connecting rod. Further contributions occur as the result of the eccentric rotation of journal bearing supports and off-center rotation of the flywheel. There are clearly other sources of vibration, such as the gear teeth meshing vibrations of timing gears, and noise and vibrations generated as the result of transient contact dynamics of the cam/follower, as well as from the concentrated contact of rolling element bearings to raceways in camshaft support bearings.

At any instant of time, the radial component of the piston force is transmitted along the connecting rod and is applied to the crankpin. This force tends to pull the crankpin away and induces the instantaneous bearing support load. The tangential component of the piston force, on the other hand, is utilized in the rotation of the crankshaft. With every power stroke, the transmitted piston force is initially increased in magnitude and is subsequently reduced at the end of the stroke. This action results in an imparted twist-untwist action of the crankshaft, causing torsional vibrations to occur. In a four-stroke internal combustion engine, described in this paper, the fundamental frequency of the applied torque coincides with half speed of the crankshaft, and its harmonics are at whole or half orders of the same speed. Therefore there are an infinite number of critical speeds, and in principle high vibratory torques may be induced at every one of these frequencies. The generated vibration amplitudes can be high enough in extreme cases to cause crankshaft or other engine component failures. In practice, however, a few of the critical speeds produce seriously high vibratory torque amplitudes.

Some systems exhibit high vibration amplitudes at unexpected rotational speeds. This phenomenon was first identified by Draminsky. Hestermann and Stone have pointed out that the cause of unexpected multiples of engine speed is the variable inertial effects due to large displacement dynamics of rotating and articulating members or uneven cylinder firing. In multiple cylinder engines cylinder-to-cylinder combustion variation can account for some of these unexpected vibration contributions. In general, these secondary vibration peaks are attributed to the nonlinear dynamic behavior of the system's overall inertial variation and the distributed system stiffness.

A less obvious and less troublesome source of vibratory torque is the imperfection in gears. The errors in gears are much too small to produce serious torsional vibrations, unless their effect is amplified by resonance.

There is a large body of literature on crankshaft torsional vibrations at different operating conditions. Zeis-chka et al. highlight a multi-body elastodynamic model of the crankshaft and the engine block. In their model they made use of finite element models of the crankshaft and the engine block, which were subsequently imported to an overall four-cylinder engine model, comprising rigid body representation for the connecting rod and pistons. The hydrodynamic journal effects were implemented by an impedance method. The impedance charts provided journal forces as a function of parameters such as bearing dimensions, oil viscosity, and eccentricity.

Katano et al. developed a model for simulating crankshaft NVH characteristics. Their method included not only the resonance of the crankshaft but also the flexibility and the oil film characteristics of the main bearings. They verified that there is a significant correlation between crankshaft behavior and the rumbling noise, and estimated the rumbling noise levels for different operating conditions.

Hestermann and Stone developed a number of models to quantify the effect of variable inertia on a single cylinder engine in both time and frequency domains. Their work showed that the reciprocating engines have the potential to exhibit secondary critical speeds due to the frequency intermodulation between the engine speed, the natural frequency orders, and the forcing frequency orders.

Song et al. analyzed the coupled crankshaft vibration. The coupling effect generates high amplitudes of vibrations when the natural frequencies of the torsional and the axial vibrations are equal to one another or when the axial vibration frequency is double that of the torsional mode. They investigated the torsional-axial interactions to clarify some unexpected vibrations of crankshafts.

Lacy reported a torsional analysis of a four-cylinder gasoline engine. In this model the crankshaft nodes, representing the main bearings, were connected to the main bearing housing by an oil film model, first reported by Kikuchi. This oil film model describes the linear and the rotary stiffness and the damping of the oil film. The mean eccentricity of each bearing is calculated by a conventional bearing oil film analysis at the desired engine conditions.

Lacy assumed that a journal eccentricity is constant, resulting in an axisymmetric oil film constraint. This assumption would hold true at high speeds and under steady-state conditions. However, transient conditions at all speeds lead to small perturbations, which can be significant and, thus, a complete solution of journal bearing dynamics under rolling and squeeze film action together with large displacement dynamics of reciprocating members is desired. With the presence of a torsional damper at the front-end main bearing, torsional oscillations of the crankshaft will be different than those occurring at the flywheel end. Furthermore the distributed inertial effect exacerbates the nonlinear planar motion of the crankshaft at the rear main journal bearing (in the vicinity of the flywheel). These conditions can contribute to the gyroscopic whirl of the crankshaft at various rotational speeds. The uneven loading of the crankshaft main journal bearing supports can also introduce additional nodding vibrations of the flywheel, as shown by Boysal and Rahnejat. The model presented in this paper attempts to highlight the existence of this problem, even under rigid body dynamics of the crankshaft. For this purpose a simultaneous solution of the rigid body dynamics of all inertial components, hydrodynamics of journal bearing reactions under pure entraining motion, and rigid body squeeze effect is required. Such an approach is undertaken in this paper that also embodys together the instantaneous solution for the combustion piston gas force during the compression and expansion strokes in the cylinder. As far as the authors are aware such a holistic approach to engine dynamic modelling has not been reported hitherto.

2. THE ENGINE MODEL

A single cylinder internal combustion engine model, comprising a piston, connecting rod, crankshaft, flywheel, timing gear, torsional damper, and crankshaft main journal bearings, is presented in this paper.

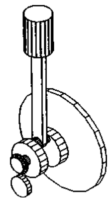

Fig. 1.Isometric view of the one-cylinder engine model.

The schematic model is illustrated in Figure 1. Unlike previous multi-body engine models in the literature the crankshaft main journal bearings' hydrodynamic action under rolling and squeeze film action are modeled. The "big-end" and "small-end" bearings are represented by scalar constraint functions imposed by a cylindrical and a universal joint, respectively. These constraint functions closely represent the articulation of the connecting rod without a further consideration for other journal bearings in these locations.

All rigid body inertial properties of the parts and stiffness/damping characteristics of the torsional damper are included in the model. The flywheel-to-crankshaft connection is represented by appropriate scalar constraint functions to ensure the torsional and lateral vibrations to take place.

The timing gear is modeled as a coupling constraint function, where the camshaft to crankshaft rotational relation is defined as 1-2. An elastomeric torsional damper is mounted on the front end of the crankshaft and forms the hub to which the crankshaft pulley is attached. The stiffness and damping properties of the elastomer are modeled as a torsional bush using a linear function.

The cylinder pressure can be calculated through the application of the first law of thermodynamics to the trapped air-fuel mass. The gas mixture can be treated as ideal with air properties.

Under the startup conditions the crankshaft must rotate fast enough to force an air-fuel mixture to enter the cylinder. For this purpose an initial torque must be applied over a short period of time. This starter motor torque is experimentally monitored and applied in the model as a step function in terms of the crankshaft angular velocity. The main crankshaft journal bearings are subjected to peak applied loads at intervals of every two revolutions of the crankshaft in this single cylinder four-stroke engine. In the absence of the big-end and small-end journals (in this model) and the elasticity of the crankpin these dynamics loads are entirely carried by the instantaneous hydrodynamic pressures generated in the main crankshaft journal bearings. The lubricant film thickness is then determined by the bearing pressure under the design clearance between the crank journal and the bearing shell. A typical clearance of the order of a thousandth of the journal radius is assumed in this analysis. The hydrodynamic reaction is also dependent on the bearing width-to-diameter ratio. The eccentric rotation of the journal generates the hydrodynamic entraining motion as well as the rigid body squeeze effect. Therefore a solution of the Reynold's equation, including the effect of squeeze film action, is required.

The mass fraction of the fuel burned is calculated as a Wiebe function, and the instantaneous heat transfer coefficient can be estimated by Woschini's correlation. The instantaneous piston force acts on the piston crown surface area and is obtained by the simultaneous solution of the differential equations (1) and the integration of the pressure distribution there.

The piston friction force that acts between the piston compression ring and the cylinder wall is formulated in terms of the instantaneous cylinder pressure in each stroke of the piston. The value of the friction force is then approximated using experimentally obtained cylinder pressure with respect to the crankshaft angular velocity and the instantaneous location of the contacting area given in terms of the piston translational velocity.

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