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SPHERICAL VS CYLINDRICAL ENGINE BEARINGS

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ABSTRACT

Spherical bearings have been used successfully in engines for some years. The spherical bearing geometry allows a simplified axisymmetric piston design; gradual rotation of the piston and rings results in axisymmetric sidewall wear, temperature distribution and thermal expansion. A previously-described concept engine design incorporating a spherical joint piston was based on an existing production engine with a conventional cylindrical piston pin. Previously-developed finite element lubrication analysis methods are applied to both designs and predictive comparisons made.

INTRODUCTION

Spherical bearings have been used successfully in engines for some years. The large (400 mm bore) Sulzer Z40 series medium speed 4-stroke Diesel engine has a spherical piston bearing which has been well-documented by its maker [1-2]. Nearly 2000 Z40 variants have been built for marine and stationary applications.

The spherical bearing design allows a simplified axisymmetric piston design; gradual rotation of the piston and rings results in axisymmetric sidewall wear, temperature distribution and thermal expansion.

Development of a low heat rejection Diesel engine sponsored by the Department of Energy (DOE) and monitored by NASA-Lewis began at Cummins Engine Company in 1989 and ended around 1995 [3-8]. The LE-55 (*L*ow *E*mission, *55*% thermal efficiency) concept engine design incorporating a spherical joint piston was based on an existing Cummins L10 production engine with a conventional cylindrical piston pin. The Cummins L10 was a 10 liter 6 cylinder in-line Diesel engine produced for 15 years ending around 1998. The SCE (*S*ingle-*C*ylinder *E*ngine) L10 version used for test purposes formed the basis for successful life tests of LE-55 hardware. Fig. 1 shows an exploded view of the LE-55 piston and connecting rod; Fig. 2 compares L10 and LE-55 rods.

ANALYSIS

Relatively few analyses have been reported for *cylindrical* piston bearings; none at all are known for *spherical* ones (or for comparison of the two types). The present work seeks to fill these gaps.

Problem specification

Parameters for the LE-55 design were taken from published sources [3-4] or assumed. Parameters for the L10 design were supplied by the manufacturer.

Operating conditions were based on successful life tests of the LE-55 engine [3-4], with peak cylinder pressure reduced to a typical value for the production L10 engine.

Solution methods

Finite-element-based methods for lubrication analysis of cylindrical engine bearings are well-documented [9-10]; modification for spherical geometries is relatively straight-forward [11]. The present work considers mass-conserving cavitation but neglects surface elasticity.

Fig. 3(a) shows a 3-D view of the finite element oil film mesh for the spherical LE-55 piston bearing. Note that the mesh (like the oil film itself) consists of two non-contiguous parts. The mesh and coordinates X,Y,Z are fixed to the piston; shown is a (moving) oil feed always at or near the top of the mesh. Also shown are dead-ended spiral grooves in the lower segment of the mesh.

Fig. 3(b) shows a 3-D view of the finite element oil film mesh for the cylindrical L10 piston bearing. Note that the mesh (like the oil film itself) is tapered from bottom to top. The mesh and coordinates X,Y,Z are fixed to the moving rod; not shown is a (fixed) oil feed at the bottom. (The 'floating' pin is assumed fixed to the piston.)

RESULTS

Fig. 4 shows instantaneous film pressure distributions at a crank angle 30 degrees before firing top-dead-center (TDC). Involved are mappings of the 3-D meshes with coordinates X,Y,Z of Fig. 3 into the 2-D meshes with coordinates X',Y' of Fig 4. The mapping of the spherical mesh into Fig. 4(a) is distorted (but equal-area); the equatorial circle is indicated for reference. The mapping of the (developable) cylindrical mesh into Fig. 4(b) is undistorted.

Both Figs. 4(a) and 4(b) show superimposed effects of oil feed and lubricant cavitation which can be examined in more detail though instantaneous plots (not shown) of lubricant mean density (or liquid volume fraction).

Studies of cyclic extremes for both bearings (not shown) predict the spherical LE-55 bearing to have a somewhat smaller minimum film thickness together with a substantially lower peak pressure than the cylindrical L10 bearing.

CONCLUSIONS

It is obviously difficult to generalize from the admittedly preliminary results reported here. Exploration of the present model can be expected to give insight into optimal choices of clearance and inlet geometry for both bearings. Inclusion of elastic and thermal effects and consideration of possible mixed lubrication conditions are obvious next steps towards realistic design guidance.

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Issue date: 23 September 1997
Inventors: Paul D. Wiczynski et al.
Assignee: Cummins Engine Co.
Title: *"Spherical joint holder rings"*

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Issue date: 11 November 1997
Inventors: Paul D. Wiczynski et al.
Assignee: Cummins Engine Co.
Title: *"Spherical joint connecting rod"*

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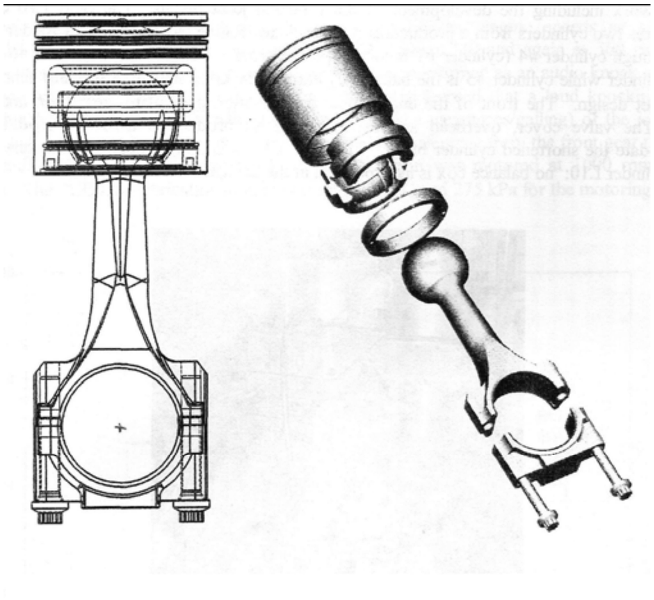


Fig. 1 LE-55 exploded view (from [4])

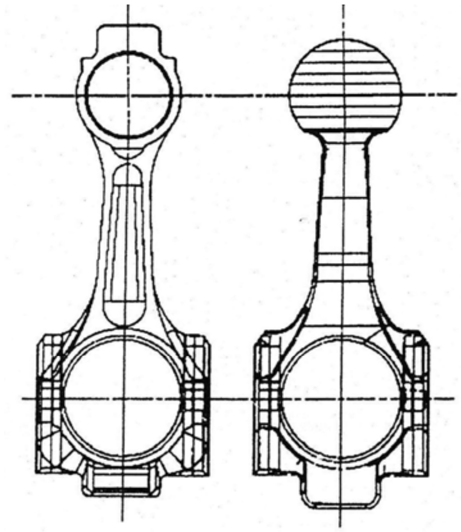
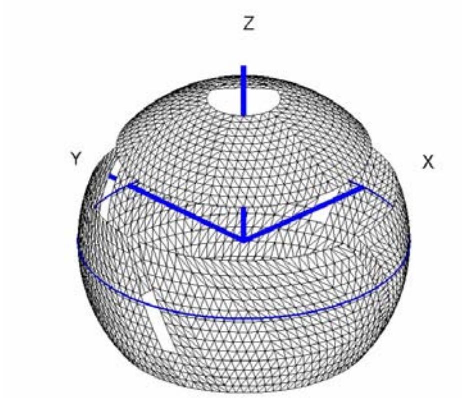
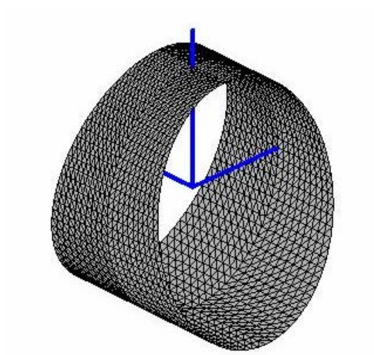


Fig.2 L10 and LE-55 rods (from [4])

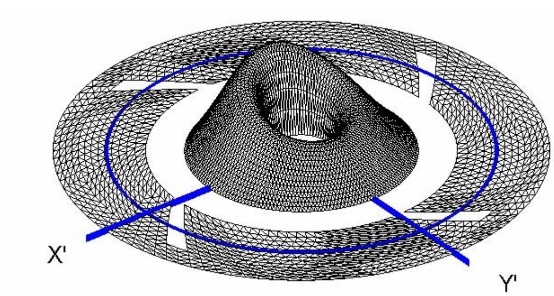


(a) LE-55

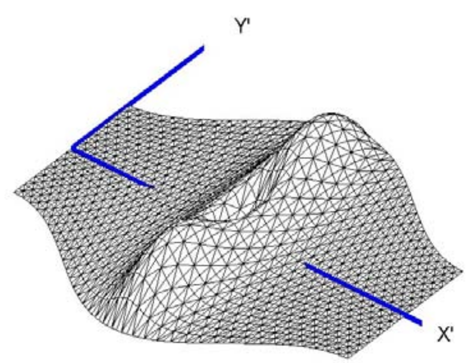


(b) L10

Fig. 3 3-D meshes



(a) LE-55



(b) L10

Fig. 4 2-D meshes & pressure (30 deg before TDC)