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Design Optimization and Analysis of Crankshaft for Light Commercial Vehicle

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Abstract

Objective: There are enormous ways to reduce crankshaft weight, which includes web design optimisation, reducing the bearing diameters etc. This paper exploits the crankshaft optimization through conceptualizing hollow shaft design. **Methods**: The optimization of engine components for mass reduction should be carefully done ensuring the functionality of the components is within the safety limits. The optimization of the crankshaft is made viable through the detail study of crankshaft dynamics and advanced calculations such that the design is not only within safer limits but also it must be durable throughout the engine life. The crank pin and journal are made hollow and validation results are within the safer limits. **Findings**: The Automobile industries are currently in surge of meeting high customer needs and emission regulations which heuristics to develop innovative methods assuring less exhaust emissions and high power. As the mechanical losses are more, the pragmatic way for achieving this customer needs, government regulations and enhance the performance of the engine is by the development of light weight components. Lightweight engine components design is a method of improving fuel economy and dynamic behaviour of engine. Apart from these the other advantages that we can achieve by reducing the inertial mass is the decrease in power required for accelerating the moving components. Variety of studies have shown that further optimization of the crankshaft which is the prime rotating component within an engine is also possible. This paper demonstrates the crankshaft optimization through the hollow crank pins and journals which yields a weight reduction of 22% and also doubles the safety factor of the existing solid crankshaft design. **Applications**: This concept can be successfully implemented for light commercial vehicles which require more power in addition to the torque requirements. The racing cars which require high durable light weight components can use this concept to achieve high power vehicles.

Keywords: Crankshaft Dynamics, Hollow Crankshaft

1. Background

Honda R&D[1] first developed hollow crankshaft for Formula One engines. A hollow crankshaft of light weight and high quality was realized through the use of friction welding. They made the oil passage in the journal into hollow structure and this had the effect of increasing average oil pressure. The location and optimum shape of joins in the hollow were subjected to CAE verification. The bending stress, caused by explosion force, is mitigated by the hollow structure, so that the stress was lower than in a solid. This developed technology has provided the prospect of the first hollow crankshaft of 10 kg or less for practical use in the Honda Formula One. However, the International Automobile Federation (FIA) included a provision in its regulations that no welding would be permitted between the front and rear main bearing journals from 2006. This made impossible to use a welded hollow crankshaft in cars entered in FIA races.

In[2] in their SAE paper Light weight Crankshafts compared the performance of cast ductile iron, cast austempered ductile iron, cast MADI and forged steel. They

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also collected static and dynamic property data, component performance and machinability data and preliminary vehicle test data for crankshafts produced from this new material and compared this data to crankshafts produced from cast ductile iron, cast austempered ductile iron and forged steel. They showed the ductile iron, MADI developed matches the strength of forged steel.

This paper shows the methodology of hollow crankshaft design with hollow journal and hollow crank pin and the manufacturing process of this crankshaft. The comparison between stresses and safety limits of the solid and hollow structure is also highlighted.

2. Procedures

The crank mechanism in internal combustion engine converts the reciprocating motion of connecting rod and piston due the combustion force into rotary motion and torque. The determination of the dynamic forces of the crank mechanism involves the calculation of gas force and inertial forces. The forces acting on the crankshaft are:

- Gas pressure inside the cylinder,
- The reciprocating masses inertia forces,
- Centrifugal forces.

These forces acting in the engine are taken up by the useful resistance at the crankshaft, friction forces and supports of engine. For every operating cycle, the forces acting on the crank continuously vary in value and direction. Therefore, in order to determine the changes in these forces versus the crankshaft's rotation angle these values are determined for a number of crankshaft positions usually taken at 10-20 degree intervals.

2.1 Gas Pressure Force

The gas pressure force which acts on the piston area are replaced with a single force acting along the cylinder axis and it has to be applied on the piston pin axis to make the dynamic analysis easy.

2.2 Masses of Crank Mechanism Parts

The mass of the crank mechanism parts is divided into:

- Reciprocating masses (the piston and the connecting rod small end),
- Rotating mass (the crankshaft and the connecting rod big end) and
- The connecting rod shank.

The crank mechanism is replaced with system of equivalent concentrated masses. The piston pin axis will constitute the mass of the piston group. The connecting rod mass is replaced by two masses out of which one is concentrated on the piston pin axis and the other on the crank axis. The crank mass is also replaced with two masses, one concentrated on the crank axis and other on the main bearing journal axis.

Therefore, the system of concentrated masses is dynamically equivalent to the crank mechanism consists of mass:

- Concentrated mass which reciprocates at piston pin axis.
- Concentrated mass which rotates at crank pin axis.

2.3 Inertial force

Inertial forces acting in a crank mechanism constitute of inertial forces of:

- Reciprocating masses
- Inertial centrifugal forces of rotating masses

2.4 Total Force

The total forces on the crank mechanism are determined by algebraically adding the gas pressure forces to the forces of reciprocating masses. Force normal to the cylinder bore axis is called as normal force and it is absorbed by the cylinder bore walls. Force which is directed along the connecting rod acts upon it and is transmitted to the crank. Acting upon the crankpin, this force produces two components of forces

- Force directed along the crank pin radius.
- Force tangent to the crank radius circumference.

The torsional moment for one cylinder is calculated. The values of the torques to the crankshaft angle are the same for all the engine cylinders and it differ only in angle intervals between firing in individual cylinders. Since the engine has equal intervals between firing, the total torque will regularly change (i is the number of cylinders). From A. Kolchin's Design of Automotive Engines[1]:

$$\theta = \frac{720°}{i} \qquad (1)$$

2.5 Forces Acting on the Crank Pin

Resulting force acting on the crank pin is calculated by vectorial addition of force acting on the crank and tangential force.

2.6 Force Acting on the Main Journal

Resulting force acting upon the main bearing journal is then calculated by vectorial addition of forces transmitted from two adjacent throws.

2.7 Design of Crankshaft

The crankshaft is acted upon by the cyclic loads of gas pressure, inertial forces and their couples. These forces and their moments cause stresses of torsion, bending, tension and compression in the crankshaft material. And also periodically varying moments cause torsional vibration of the shaft with results in additional torsional stresses.

Since crankshaft shape is intricate and a lot of forces and moments acting on it computing the strength of crankshaft is not easy.

There are various methods to determine the stresses and safety factors for individual elements of a crankshaft. While designing a crankshaft, we assume that:

- A crank is assumed to be freely supported by supports;
- The supports and force points are assumed to be acting on the centre planes of the crank-pins and journals;
- This entire span between the supports represents an ideally rigid beam.

The crankshaft is generally designed for operating on the action of the various forces and moments and the calculated results are tabulated in table 1 for existing and proposed design.

2.8 Unit Area Pressures of Crankpins and Journals

The unit area pressure on a crank pin or a main journal determines the conditions under which the bearing has to operate and its service life. The design of crankshaft journals and crankpins is done on the action of average and maximum resultants of all forces loading the crankpins and journals.

2.9 Design of Main Bearing Journals

The main bearing journals are designed for torsion. The moments and torques of individual cylinders are summed up algebraically in the order of engine firing order starting with the first cylinder.

2.10 Design of Crank Pin

Crank pins are designed to withstand the bending and torsion stresses. Torsion of a crank pin occurs due to the effect of moments and its bending is due to bending moments acting in the crank plane and in a perpendicular plane. The maximum values of twisting and bending moments won't coincide at same time, the crank pin safety factors for twisting and bending stresses are determined separately and then added together to obtain the total safety margin.

2.11 Design of Crankwebs

The crankshaft webs are acted upon by alternating stresses: Tangential stresses due to torsion and normal stresses due to bending and push-pull. Maximum stresses occur where the crank pin fillets joining a crankweb.

2.12 Safety Factor for Crankshaft

Since the fatigue strength of crankshaft is dependent on variation of a load causing symmetric, asymmetric or pulsating stresses on fatigue limits and yield strength and of the part material, on part shape, size, machining and thermal treatment and case-hardening, the effects of all these factors should be accounted. The results for the four-cylinder direct injection diesel engine crankshaft is tabulated below

Table 1. Comparison of stress and safety factor

	Forged Solid Crankshaft	**Hollow ADI Crankshaft**
Unit Area Pressures On Crank pins And Journals MPa		
Mean unit area pressure on crankpin	10.61	10.61
Mean unit area pressure on main journal	10.67	10.67
Maximum unit area pressure on crankpin	55.38	55.38

Maximum unit area pressure on main journal	27.09	27.09
Main Journal Stresses , MPa		
Maximum tangential stress	24.23	22.5
Minimum tangential stress	-10.7	-9.9
Mean tangential stress	6.76	6.28
Tangential stress amplitude	17.47	16.22
Tangential stress amplitude with stress concentration	22.24	20.27
Tangential stress safety factor	8.37	15.73
Crank pin Stresses, MPa		
Maximum tangential stress	51.8	51.8
Minimum tangential stress	-22.44	-22.44
Mean tangential stress	14.69	14.69
Tangential stress amplitude	37.12	37.12
Tangential stress amplitude with stress concentration	47.26	46.40
Tangential stress safety factor	3.92	6.85
Maximum normal stress	89.47	89.47
Minimum normal stress	-44.38	-44.37
Mean normal stress	22.55	22.55
Normal stress amplitude	66.93	66.93
Normal stress amplitude with stress concentration	132.09	132.09
Normal stress safety factor	2	3.28
Total safety factor of the crankpin	1.8	2.96
Crank web Stresses, MPa		
Maximum tangential stress	21.52	21.52
Minimum tangential stress	-48.7	-48.70
Mean tangential stress	-13.59	-13.59
Tangential stress amplitude	35.11	35.12
Tangential stress amplitude with stress concentration	44.7	43.89
Tangential stress safety factor	8.84	11.29
Maximum stress due to bending	130.76	130.76
Minimum stress due to bending	-28.41	-28.41
Maximum stress due to normal force	21.84	21.84
Minimum stress due to normal force	-8.5	-8.51
Nominal bending stress	79.59	79.59
Nominal normal stress	15.17	15.17
Mean alternating stress	57.84	57.84
Alternating stress amplitude	187.02	187.02
Alternating stress safety factor	1.31	2.13
Total safety factor of the crankpin	1.5	2.1

Table 1 shows the comparison between the stresses and safety factors of forged solid crankshaft of SAE1548 material and hollow crankshaft of ADI 1050 material for the four-cylinder direct injection diesel engine of a light commercial vehicle. Figure 1 and Figure 2 shows the hollow crankshaft structure.

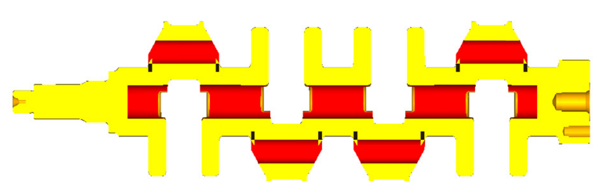

Figure 1. Cross sectional view of hollow crank pin and hollow journal.

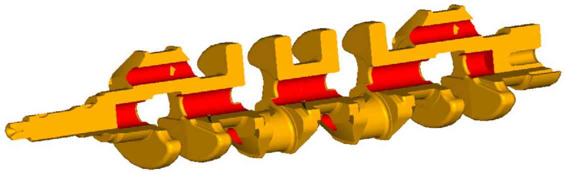

Figure 2. 3D model of hollow crankshaft.

3. Summary and Conclusion

- The methodology of hollow crankshaft design with hollow journal and hollow crank pin has been shown.
- The comparison between stresses and safety limits of the solid and hollow structure is highlighted for a four cylinder DOHC turbocharged engine of a Light commercial vehicle.
- A significant weight saving about 18% without compromising the safety factors has been achieved through cast iron crankshaft with hollow structure.

4. References

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