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Dyamic analysis of balancing of an automobile engine using ADAMS

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***Abstract*—This paper explain the primary balancing of single cylinder engine and optimizing the counter balancing mass to reduce the unbalanced forces coming on the crankshaft. After optimizing, the effect of primary balancing is validated. After primary balancing, the vibration is measured on the engine at various locations. Balancer shaft is designed for the unbalanced forces exist after primary balancing. Balancer shaft is introduced into the simulation model and simulated the running at 6000rpm and unbalanced forces on the crankshaft are measured. The vibration on engine assembly at previously measured location is measured. The effectiveness of primary balancing and balancer shaft is validated by comparing the acceleration (m/s^2) measured at various locations.**

***Keywords*— Inertia,mass properties,balancing,ADAMS,CATIA.**

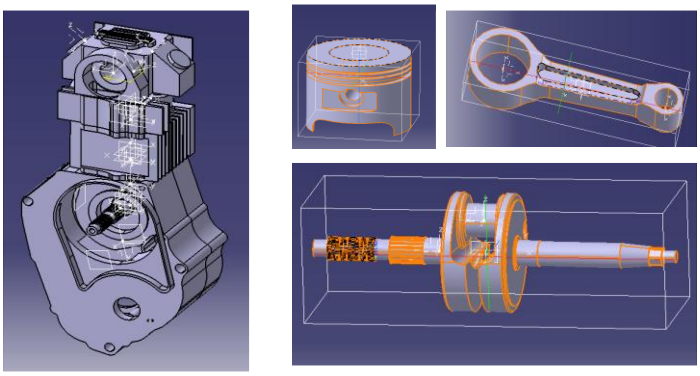
I. BALANCING METHODOLOGY

The methodology involved the following steps

- Building of CAD models of individual components of existing Engine model and checking against the actual model
- Converting the CAD model into simulation model (ADAMS)
- Building the Mechanism in ADAMS using the various joints
- Calculating the theoretical unbalanced force and comparing with the Simulation values
- Primary balancing and optimizing the counter mass
- Validating the impact of primary balancing by measuring the unbalanced forces coming the crankshaft.
- Designing the Balancer shaft to reduce the unbalanced forces after primary balancing.
- Introducing the balancer shaft into the simulation model and evaluating the impact by measuring the unbalanced forces on crankshaft
- The overall performance characteristics that is Acceleration is measured on the engine in without balancing, with primary balancing and with balancer shaft.

A. Building of CAD model of individual components:

The various engine components are modeled using the solid modeling techniques (Catia V5). The Engine consists of Piston, Connecting rod, Crankshaft, piston pin, Cylinder block, Cylinder head and Crankcase. All these parts are modeled and checked the mass properties against the actual model.



B. Converting CAD model into simulation model:

The engine model developed using the solid modeling software need to be converted into simulation model which involves following steps

Catia model (CAT file) – IGES format

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IGES to X_t format using the CAD converter

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Importing to ADAMS

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Inputting the Mass properties

↓

Checking the inertia properties

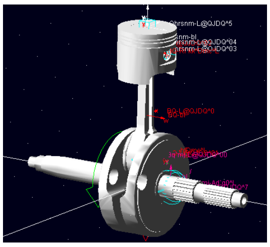
↓

Fixing the pointers at the C G points

↓

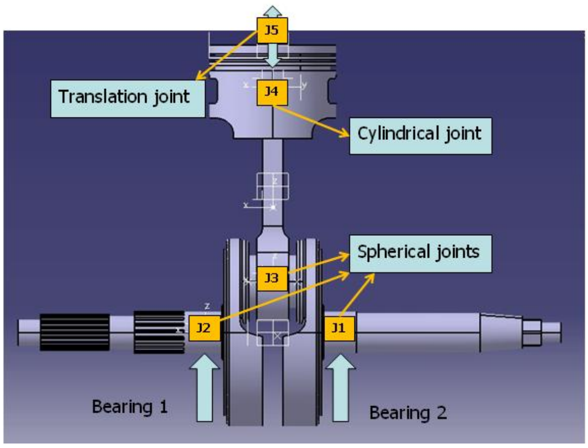
Fixing the Joints

The output file after converted to simulation model will be used in ADAMS for analysis.



C. Building the mechanism model in ADAMS:

The mechanism model is built in the ADAMS software by using the various joints.



Ref.	Components	Joint type
J1	Bearing & Crankshaft (Right side)	Spherical joint
J2	Bearing & Crankshaft (left side)	Spherical joint
J3	Crankshaft & Connecting rod	Spherical joint
J4	Connecting rod & Piston	Cylindrical joint
J5	Piston & cylinder block	Translation joint

In order to avoid the redundancy, the spherical joints are used instead of revolving joint.

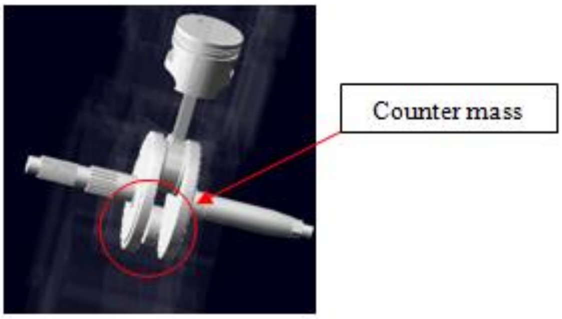
D. Primary balancing and optimising the counter mass:

As explained earlier, the primary balancing is the balance achieved by compensating for the eccentricities of the masses in the rotating system, including the connecting rods. At the design stage primary balance is improved by considering and adjusting the eccentricity of each mass along the crankshaft. In theory, any conventional engine design can be balanced perfectly for primary balance. Once the engine is built primary balance is controlled by adding or removing mass to or from the crankshaft, typically at each end, at the required radius and angle, which varies both due to design and manufacturing tolerances.

Mass of the counter mass is equal to m_c+c*$m_{p,}$

Where,

$m_{c\,=}$ mass of the crankpin
$m_{p\,=}$ mass of the connecting rod
c = Factor
c=0 when no mass of connecting rod is added
c=1 when the full mass of connecting rod is added



The model is simulated the condition of running at 6000rpm and the unbalanced forces coming on the crankshaft bearing are measured. The forces coming on the crankshaft is measured by varying the C factor in both Y & Z direction.

The unbalanced forces are minimum when C is 0.2 in Y direction and when C is 1.5 in Z direction. The unbalanced forces in Z direction is decrease when C is increasing from 0 to 1.5 and starts increasing but the forces in the Y direction is increasing from 0.2. When C is equal to 0.835, the unbalanced forces on the both Y and Z direction are same. This point taken as the optimized C.

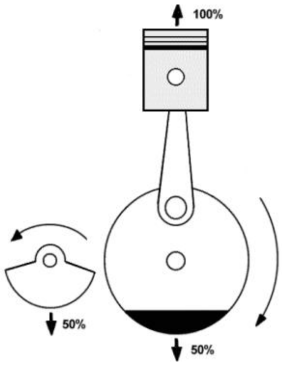
At C=0.835,

The mass of the counter mass is 0.2541kg

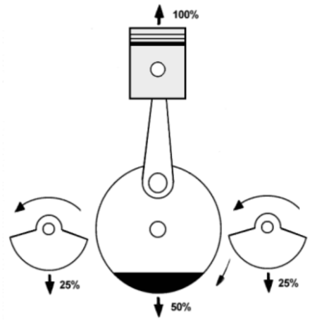
The unbalanced forces on crankshaft is both Y and Z direction are 750N, which needs to balanced by Balancing shaft.

E. Blancer shaft:

Many modern engines are fitted with balancer shafts. The principle behind these are originally proposed by Dr. Lanchester around hundred years ago.A constant rotating force can easily be balanced by another constant rotating force spaced 180° apart. However, it is not as easy as adding another counter weigh to crankshaft, because this force rotates in opposite direction, as we have seen, and therefore can only be balanced by a a counter weight rotating in opposite direction.This is the reason that balance shafts rotate in the opposite direction to the crankshaft.



The crankshaft has a 50%balance factor,and the balance shaft creates further 50% to the balance the reciprocating force when at TDC.At 90° rotation there is no reciprocating force but because the balance shaft rotates in the opposite direction the two centrifugal forces cancel out,leaving zero unbalance force through a full revolution.However,the spacing between the two shafts creates a rocking couple.As seen on the right,the use of two balancer shafts,which each balance 25% can eliminate that couple as shown below

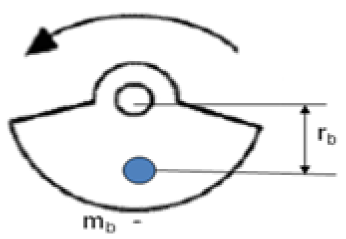


Cost,packaging and the weight are the reasons that it is more common to see only one balance shaft.

Prior to the common usage of the Lanchester balance shaft,it was usual to adjust the vibration characteristics of a particular motorcycle, by means of balance factor.

F. Design of balancer shaft:

Design of balancer shaft involves selection mass required to create the counter balancing for and the offset of its CG from the axis of balancer shaft.



m_b = Mass of the balancer shaft (kg)
r_b = Offset of CG from the axis of balancer shaft (mm)
ω^2 = Angular velocity (Rad/sec)
F = Unbalanced force which need to be balanced

Calculated balancer shaft is 0.1899kg at 10mm

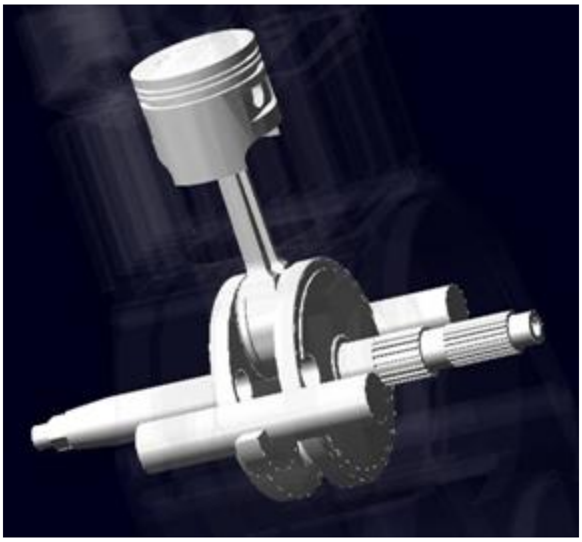
m_b = 0.1899 kg

II. RESULTS

The unbalanced forces coming on the crankshaft is measured in both Y and Z direction at

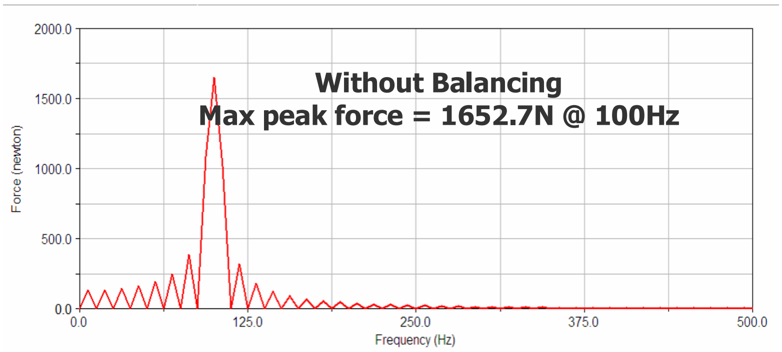
- Before balancing
- After primary balancing
- With balancer shaft

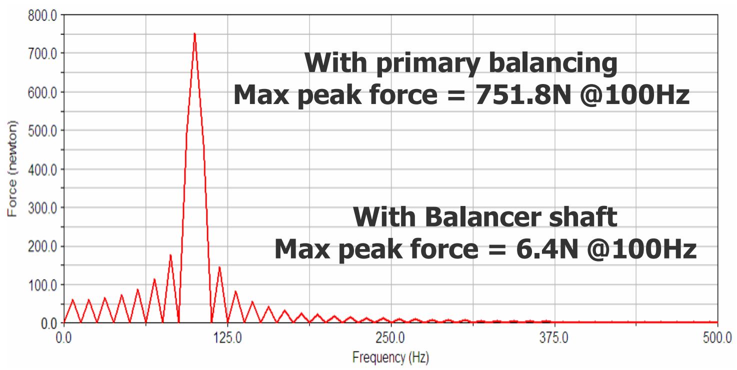
The forces in the Y and Z direction decreased from without balancing to primary balancing and it is further reduced with balancer shaft.



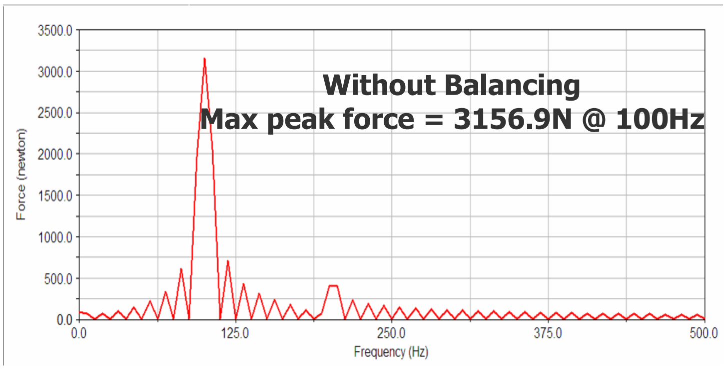
The forces are same as of single balancer shaft and the additional which caused due to introduction of balancer shaft has got reduced to normal condition. The results are shown in Frequency domain..

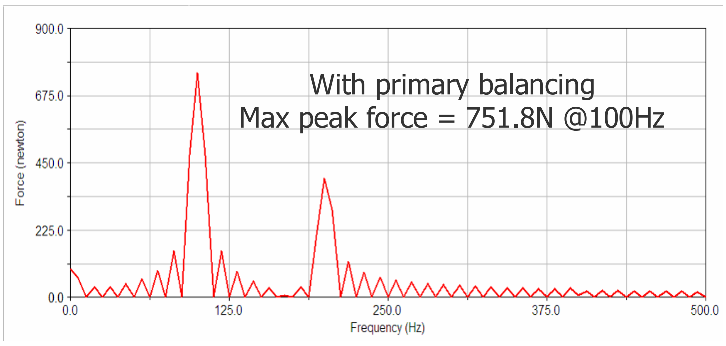
Forces in Y direction:

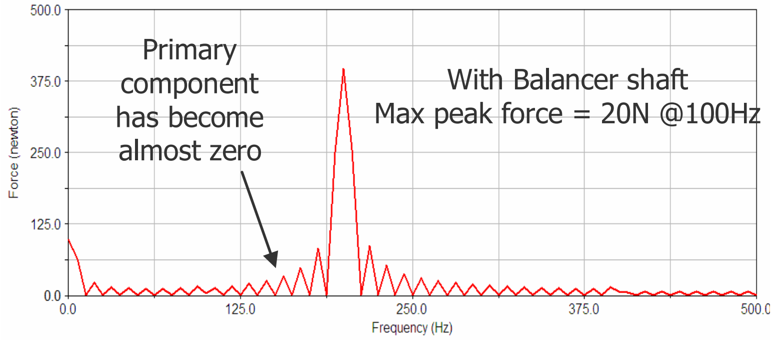


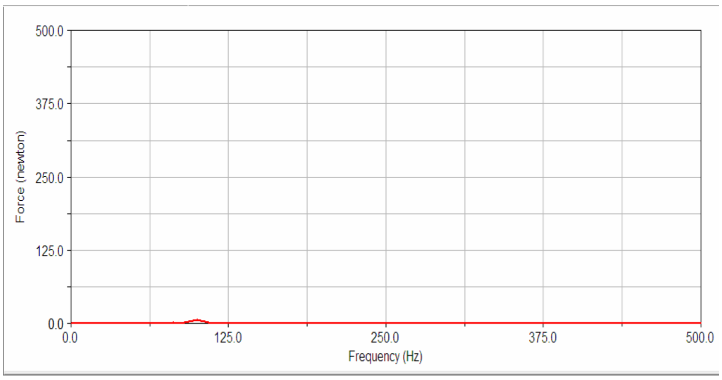


Forces in Z direction:



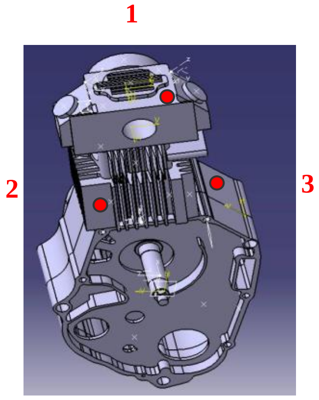






III. VALIDATION

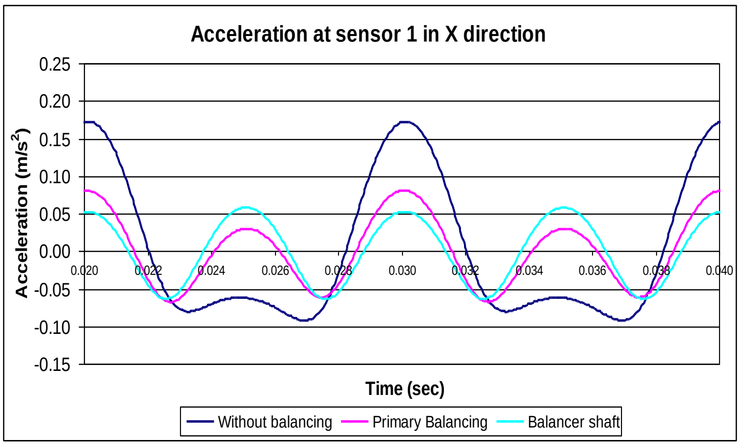
In order to validate the concept, Accelerometers are fixed on the engine assembly at various locations. The main performance character that is acceleration (m/sec^2) is measured in X, Y and Z directions. Location of sensors are shown below



To simulate the actual running of single cylinder engine, the real condition is simulated by mounting the engine of the engine mounting. The engine assembly is mounted on the vehicle mounts with dampers. The vibration coming on the engine at different locations are measured.

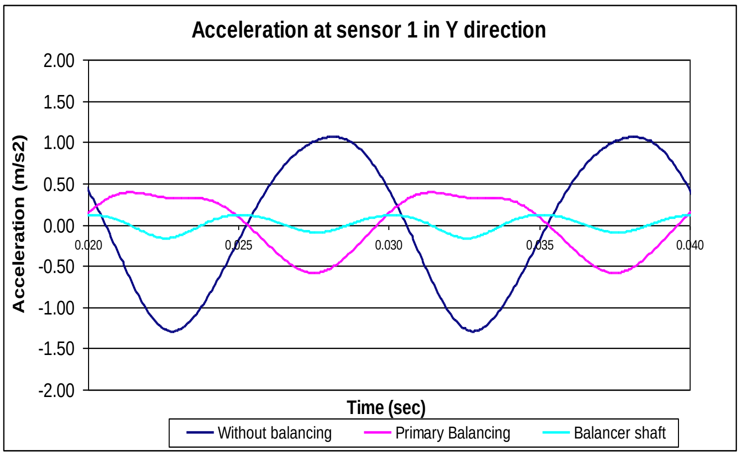
At sensor 1:

X direction:



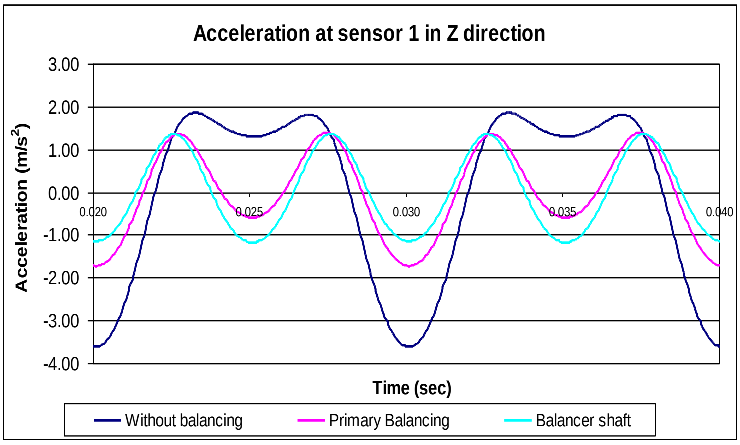
	No Balancing	Primary Balancing	Balancer shafts
Maximum Acceleration (m/s^2)	0.1724	0.0804	0.0578
Minimum Acceleration (m/s^2)	-0.0927	-0.0678	-0.0642

Y direction:



	No Balancing	Primary Balancing	Balancer shafts
Maximum Acceleration (m/s^2)	1.0591	0.388	0.116
Minimum Acceleration (m/s^2)	-1.2954	-0.5908	-0.1677

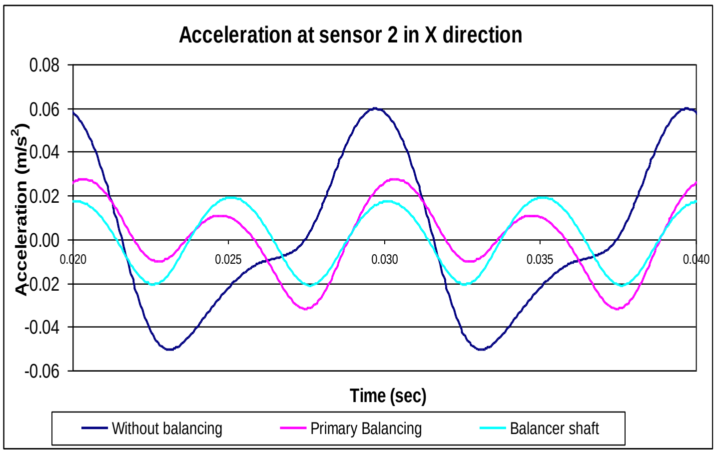
Z direction:



	No Balancing	Primary Balancing	Balancer shafts
Maximum Acceleration (m/s^2)	1.8462	1.3893	1.3608
Minimum Acceleration (m/s^2)	-3.6229	-1.7279	-1.1858

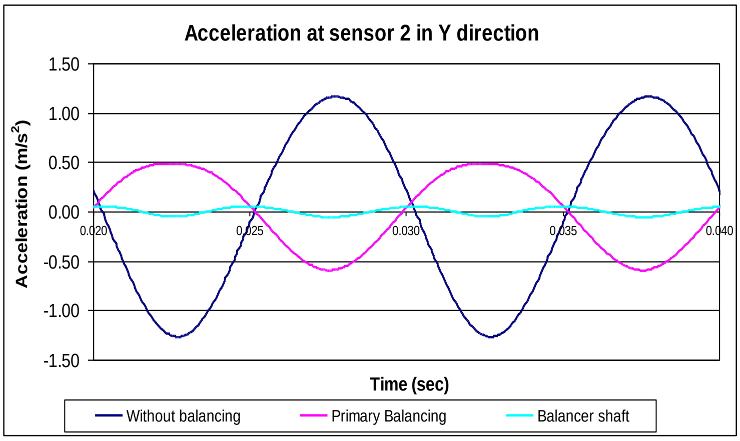
Sensor 2:

X direction:



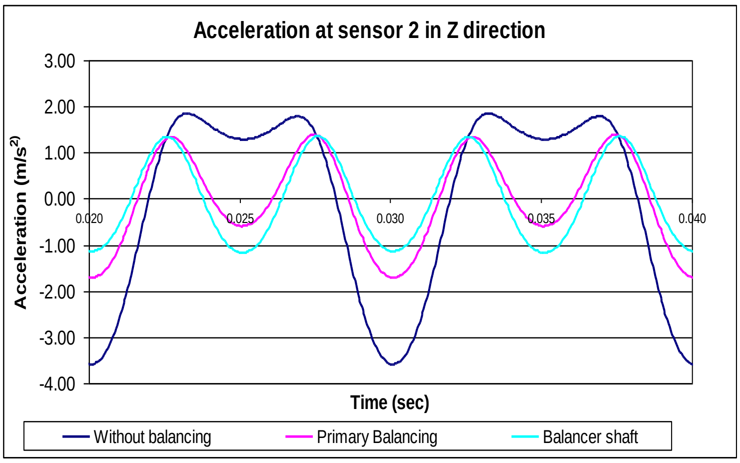
	No Balancing	Primary Balancing	Balancer shafts
Maximum Acceleration (m/s^2)	0.0596	0.0274	0.019
Minimum Acceleration (m/s^2)	-0.0506	-0.0319	-0.0211

Y direction:



	No Balancing	Primary Balancing	Balancer shafts
Maximum Acceleration (m/s^2)	1.1611	0.4852	0.0499
Minimum Acceleration (m/s^2)	-1.2707	-0.5955	-0.0585

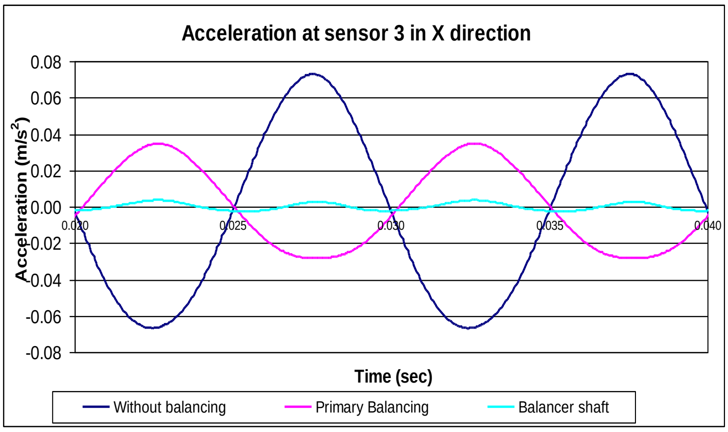
Z direction:



	No Balancing	Primary Balancing	Balancer shafts
Maximum Acceleration (m/s^2)	1.8394	1.3852	1.3522
Minimum Acceleration (m/s^2)	-3.5943	-1.7144	-1.1763

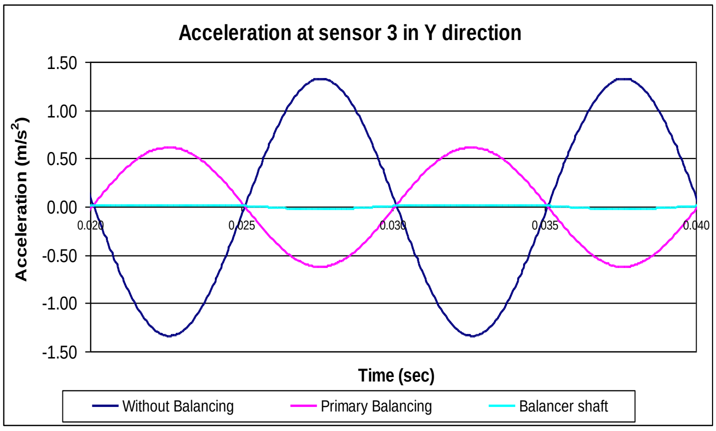
Sensor 3:

X direction



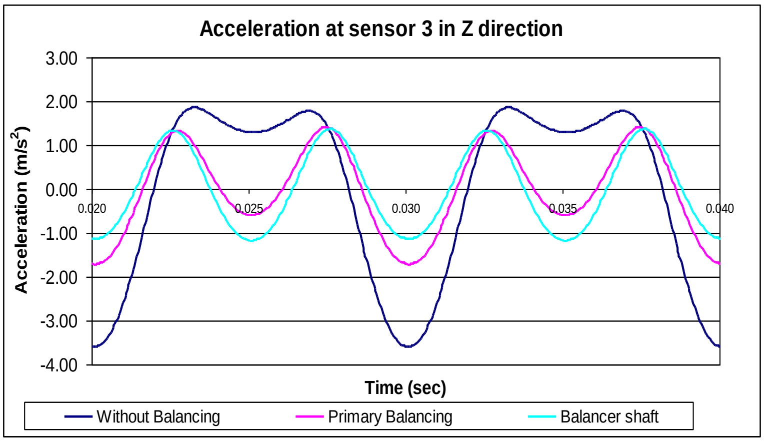
	No Balancing	Primary Balancing	Balancer shafts
Maximum Acceleration (m/s^2)	0.0728	0.0346	0.0036
Minimum Acceleration (m/s^2)	-0.0669	-0.0285	-0.0026

Y direction:



	No Balancing	Primary Balancing	Balancer shafts
Maximum Acceleration (m/s^2)	1.3244	0.6102	0.0168
Minimum Acceleration (m/s^2)	-1.3412	-0.6269	-0.0287

Z direction:



	No Balancing	Primary Balancing	Balancer shaft	Balancer shafts
Maximum Acceleration (mm/s^2)	1.8538	1.4059	1.3600	1.3653
Minimum Acceleration (mm/s^2)	-3.6054	-1.7191	-1.2346	-1.1803

From the above data, it is clear that the vibration levels are decreased from without balancing to primary balancing and its further reduced with balancer shaft.

IV. CONCLUSIONS

1. The unbalanced forces coming on the crankshaft is reduced with primary balancing and its further reduced with the balancer shaft
2. The torque was increased due to single balancer shaft and the has been controlled to normal condition by introducing the two balancer shafts of each of half the mass of single balancer shaft placed opposite to each other.
3. The performance parameter vibration (acceleration) coming on the engine is reduced.
4. The packaging of the two balancer shaft and the impact of additional mass need to studied.

5. This concept can be implemented for above 150CC motor cycles which prone for more vibration.

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