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International Journal of Current Research
Vol. 7, Issue, 04, pp.15001-15009, April, 2015

INTERNATIONAL JOURNAL OF CURRENT RESEARCH

ISSN: 0975-833X

RESEARCH ATICLE

ENGINE WITH DUAL HELICAL CRANKSHAFT SYSTEM OPERATING AT ANOVERDRIVE GEAR RATIO

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ARTICLE INFO

Article History:
Received 23rd January, 2015
Received in revised form
02nd February, 2015
Accepted 26th March, 2015
Published online 30th April, 2015

Key words:

Lowrevving, High revving, ANSYS, modal analysis, Alternating, Bending stress, Torsional stress, Von-misses stress, Von-misses strain, Secondary shaft, New crankshaft, Crankweb, Helical gear, Meshing, Fuel economy, Overdrive gear ratio, Partial meshing, Two shaft system.

ABSTRACT

This paper suggests a new design of the crankshaft system that would help to improve the fuel economy of the vehicle as it will enable the usage of a low revving engine for applications requiring the use of a high revving engine operating at the same power by converting the extra or unnecessary torque obtained from a low revving engine into angular velocity of the crankshaft of the engine. This paper also shows how the proposed design will improve the fuel economy of the vehicle. This will be achieved by changing the design of the crankweb in such a way that it functions both as a counterweight as well as a helical gear and incorporating another gear shaft in the crankshaft system which will be operating at an overdrive ratio. This paper also analyses the proposed crankshaft design for safety against fatigue failure in accordance to the guidelines given by Germanischer Lloyd. Finite element analysis of the crankshaft has been done using ANSYS 14.0 and the resultant stresses have been tabulated.

INTRODUCTION

The biggest problem faced by the people in the present world, be it a taxi driver or a truck driver or an auto-rickshaw driver or any person whomsoever owns a vehicle, is that the prices of the fuels are increasing and they are spending a considerable amount of their income in buying fuel for their locomotive. Moreover, there are around 1.2 billion vehicles on the roads in 2014 and this number will swell up to two billion by 2020. Taking into account the alarming rise in the need of petrol/diesel by the human population and the available crude oil deposits on earth, the estimated number of years these resources will last is less than 60 years moreover the pollution caused by the combustion of these lead to release of greenhouse gasses which are major players causing both health and environmental hazards. Since the needs of the human race cannot be compromised, it is of utmost importance to modify the existing machinery to make their functioning smarter and hence save fuel for sustainability of the environment and for ensuring that fuel lasts for the future generations to use.Areas like measurement of crankshaft positioning have been worked upon by (Rizzoni and Ribbens, 1989). Three dimensional vibrational analysis of the crankshaft was done by (Okamura *et al.,* 1995). Transient analysis of a diesel engine was performed by (Rakopolous and Giakoumis, 1998). Dynamic analysis of an Inrernal combustion engine crankshaft was done by (Mourelatos, 2001). Wok on improvement of accuracy of crankshaft balancing was carried out by (Kang *et al*., 2003). Fatigue analysis of crankshaft sections under bending was performed by (Chien *et al*., 2005).Analysis of crankshaft motion with a crack in crankpin fillet was done by Lie *et al*., 2006). Non-linear dynamics of engine-propeller system was studied by (Yu *et al*., 2009). Torsional stiffness of an engine crankshaft was studied by (Zhao and Jiang, 2009). Detection of diesel engine crankshaft vibration for combustion engine diagnosis was carried out by (Charles *et al.,* 2009). Intension analysis of 3-D finite element analysis of a diesel engine crankshaft was done by (Meng *et al*., 2010). Strength analysis of a diesel engine crankshaft was done by (Gu and Zhou, 2011). FEM based fracture mechanics technique to estimate life of an automotive forged steel crankshaft of a single cylinder diesel engine was evaluated by (Metkar *et al*., 2013). A novel crankshaft mechamism and a regenerative braking system was proposed by (Boretti and Scalzo, 2013). A simplified coupled crankshaft-engine-block model was proposed by (Bellakhdhar *et al*., 2013). Fatigue failure analysis on a large scale reciprocating compressor was studied by(J. Zhao *et al*., 2014).Till now there has not been any significant work where the crankshaft has been designed to improve the

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fuel economy of the vehicle. Hence there is a need to formulate a design of the crankshaft that will not only serve its intended purpose but will also improve the fuel economy of the vehicle in which it is functioning. In this paper the theoretical calculations and analysis have been done in order to confirm the benefits that would be obtained from the new design of the crankshaft. Modelling of the proposed crankshaft system has been done using professional CAD software *Solidworks* and the finite element analysis has been done using commercial FEA software ANSYS Workbench. Verification of the dimensions and measurements of the designed crankshaft has been done in accordance to the guidelines issued by technical supervisory organisation, Germanischer Lloyd.

There was no experimental testing of the crankshaft system to prove the benefits of the design or to ensure that the designed crankshaft system will be safe against fatigue failure or the loads it will be subjected to while the engine is operating. In this paper, a new design of crankshaft has been suggested along with how it can be implemented to improve the fuel economy of the vehicle. The dimensions of the crankshaft system have been decided and its safety against loads acting on it has been verified using finite element analysis and the guidelines given by Germanischer Lloyd. Modelling of the crankshaft has been done using Solidworks. Modal analysis of the crankshaft has been done using ANSYS Workbench to extract the modal frequency of the crankshaft hence ensuring that it does not match with the frequency at which the engine is operating. In this paper, there is a comparative study between a low revving and a high revving engine has been done to prove that after implementation of the proposed design of the crankshaft system in the low revving engine, it will give the same speed and torque as that given by a high revving engine but with much better fuel economy.

Design of the new Crankshaft system

The new crankshaft system was designed using commercial 3-D modelling software, Solid works. The dimensions of the crankshaft system were decided with a goal of having minimum weight and volume of the crankshaft system without any compromise on the performance and safety of the crankshaft system for which it has been designed.

S.No	Name of Parameter	Unit	Value
1	Cylinders	-	Straight four
2	Capacity	cc	1598
3	Bore	mm	79.5
4	Stroke	mm	80.5
5	Bore/Stroke Ratio	-	.99
6	Maximum Power Output	Kw	77 at 4400RPM
7	Maximum Torque	N-m	250 at 1500RPM
8	Compression Ratio	-	16.5
9	Air-Fuel Ratio	-	13.5
10	Cut-off Ratio	-	2.75
11	Efficiency	-	.3
12	Length of Connecting Rod	mm	180
13	Width of Connecting Rod	mm	22
14	Adiabatic Constant	-	1.4
15	Cycle of Combustion	-	Diesel Cycle
16	Maximum Pressure in a Cycle	KPa	1966
17	Firing Order	-	1-4-3-2

Engine Specifications

The specifications of the four cylindered turbocharged diesel engines for which the crankshaft system was designed are given in Table.2.1.1.

Assumed Dimensions of the Crankshaft System

The dimensions of the crankshaft were assumed and then optimised to reduce the volume and weight. The assumed dimensions are given in Table 2.2.1.

S.No	Name of Parameter	Unit	Value
1	Length	mm	402
2	Crankweb Width	mm	18
3	Crankpin Width	mm	22
4	Journal Bearing Width	mm	34
5	Teeth on Complete Crankweb Circle	nos	43
6	Module	mm	2.25
7	Pressure Angle	Degrees	20
8	Helix Angle	Degrees	45
9	Crank Radius	mm	37
10	Diameter of Crankpin	mm	44
11	Diameter of Journal Bearing	mm	50
12	Fillet Radius on Crankpin	mm	2
13	Fillet Radius on Journal Bearing	mm	2
14	Diameter of Relief Bore on Crankpin	mm	17.6
15	Diameter of Relief Bore on Journal Bearing	mm	20
16	Diameter of Crankweb	mm	70
17	Diameter of Oil Bore	mm	2
18	Length of Secondary Gear Shaft	mm	402
19	Face Width of Gear	mm	18
20	Width of Bearing	mm	34,22
21	Teeth on Complete Gear	nos	24
22	Pressure Angle	Degrees	20
23	Helix Angle	Degrees	45
24	Diameter of Bearing	mm	50
25	Diameter of Relief Bore	mm	20
26	Outer Diameter of Gear	mm	80.86

Material and Manufacturing Process Selected for the Crankshaft System

The material of the crankshaft system was decided to be ASTM 182 F11 with induction hardening of the fillets on the crankpin and the journal bearing. Chrome-moly alloy was chosen because this alloy is light, strong and resistant to both scaling and oxidation. It also hardens through heat treatment. Forging process was selected for manufacturing of the crankshaft system over casting because the crankshaft manufactured from the forging process will be stronger than that manufactured by casting process.

S.No	Name of Property	Units	Value
1	Density	g/cc	7.9
2	Ultimate Tensile Strength	MPa	520
3	Yield Tensile Strength	MPa	310
4	Elongation at Break	%	20
5	Reduction of Area	%	40
6	Modulus of Elasticity	GPa	200
7	Specific Heat Capacity	$J/g\text{-}^0C$	.460
Component Elements Properties			
8	Carbon,C	%	.05-.15
9	Chromium,Cr	%	1-1.5
10	Manganeese,Mn	%	0.3-0.8
11	Molybdenum,Mo	%	0.3-0.6
12	Phosphorous,P	%	<=0.03
13	Silicon,Si	%	0.5-1
14	Sulfur,S	%	<=0.03

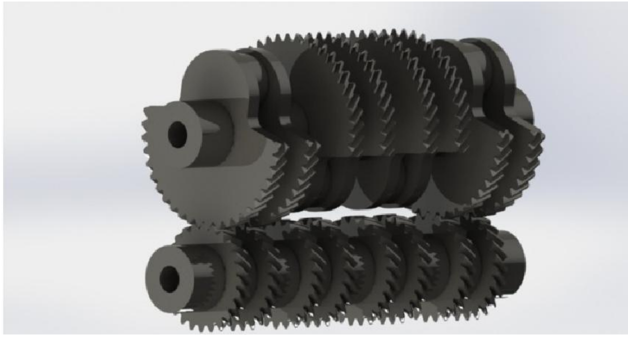
Moreover the defects are refined during preworking in case of forging and the manufactured crankshaft will respond better to heat treatment which is required for hardening of the fillets on the crankpin and journal. Finally the forging process is more reliable and cheap as compared to casting process. The specific type of forging, F11 has been selected because the crankshaft will be operated at high temperatures in a non-corrosive environment that is in oil. The material data is given in Table.2.3.1.

Modelling of the New Crankshaft System

Figure.2.4.1. shows the new design of the crankshaft along with the relief bores and geared section

Figure.2.4.2. shows the design of the secondary shaft which will be meshed with the new crankshaft and will operate at an overdrive ratio. Hence, the angular velocity of the secondary shaft will be more than that of the crankshaft but the torque available at the secondary shaft will be lesser than that available at the new crankshaft.

The crankshaft system has been designed in such a way that at any instant of time only half the number of crankwebs of the crankshaft meshes with the secondary shaft. If crankweb numbers one, two, seven and eight mesh with the secondary shaft for first half of revolution then in the second half of the revolution, crank web numbers three, four, five and six will mesh with the secondary shaft. In this way the crankshaft will not only be meshed with the secondary shaft but it will also reduce the vibrations that may arise due to meshing of the gears of the crankshaft and the secondary gear shaft because only four crankwebs of the crankshaft are meshing with the secondary shaft as mentioned above. Fig.2.4.3. shows the crankshaft system suggested in this paper. The crankweb has been designed in such a way that the geared part subtends an angle of 190^0 at the centre of the crankweb so that there is no sudden disengagement of the crankshaft and the secondary shaft at any instant of time, hence eliminating jerks and sudden increase in vibration of the crankshaft system. That is, when the first set of crankwebs which engaged with the secondary shaft are about to dis-engage then the other set of four crankwebs will mesh with the secondary shaft, so there will be a duration of 10^0 rotation of the crankshaft when all the crankwebs will be meshed with the secondary shaft.



Design Verification

The proposed design of the crankshaft was verified for safety against fatigue with the dimensions that were assumed.

Modal Analysis

Modal analysis of the crankshaft was performed using ANSYS Workbench to extract the modal frequencies and hence verify that they were not matching that of the engine. The number of modal frequencies that were extracted was six. Prestress effects were not included in the analysis. Fig. 3.1. shows the modal frequencies that were extracted.The analysis gives us 6.8875Hz, 627.11Hz, 769.91Hz, 853.94Hz, 945.54Hz and 1413.3Hz as the six modal frequencies. Hence it is verified that the crankshaft will never resonate with the engine as the frequencies at which the engine operates, which is from 41.66Hz to 73.33Hz confirming that the crankshaft will never resonate with other components of the engine.

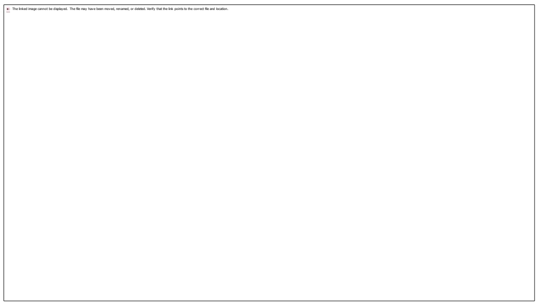
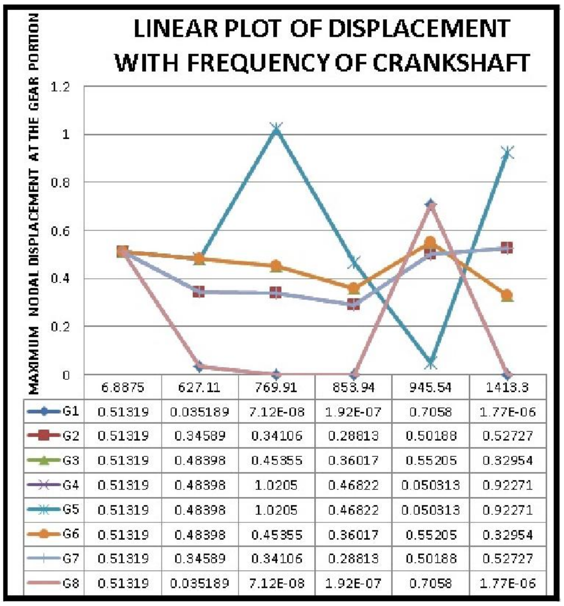
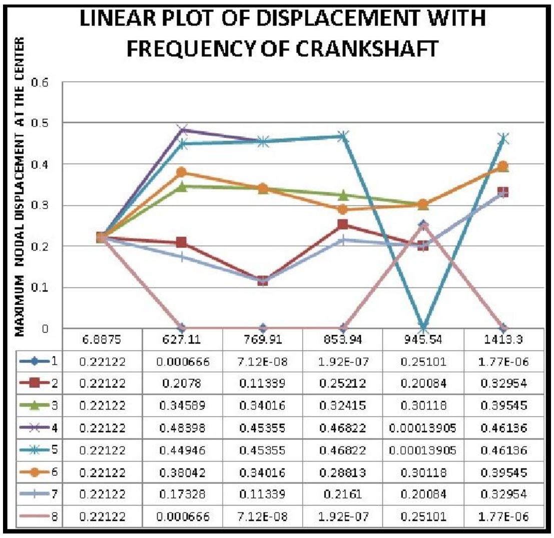


Table 3.1.1. shows the statistics for meshing

S.NO	Parameter	Value
1	Nodes	1415347
2	Elements	948224
3	Type of Element	SOLID187
4	Shape of Element	10 Node Tetrahedron
5	Type of Meshing	Free

Fig.3.1.2. shows the maximum nodal displacement that occurs in the geared section of the crankweb and Fig.3.1.3. shows the maximum nodal displacement that occurs in the centre of the crankweb. The nodal displacement plots are given in Appendix.3.





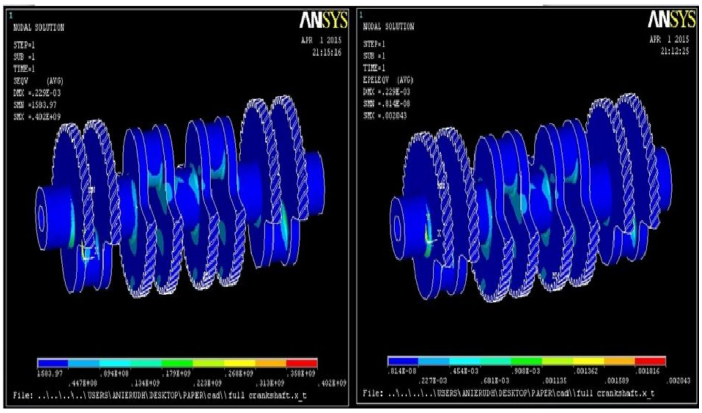
Verification Using Alternsting Bending and Torsional Stresses and ANSYS Stress-Strain Analysis

Values of maximum alternating bending and torsional stresses were calculated to ensure that the designed crankshaft system is safe against fatigue failures. Stress concentration factors were calculated for calculation of the same which are given in Table.3.2. The formulae that were used and the constants that were defined for the calculation of these stress concentration factors are given in Appendix.1.

Table.3.3. gives the values of the alternating bending stresses that were used to satisfy the acceptability criterion which has been defined by Germanischer Lloyd as factor Q whose value has to be greater than 1.15.

S.NO	Stress Concentration Factor	Value
1	α_B	2.476069426
2	β_B	2.66791746
3	β_Q	3.004285999
4	α_T	2.27666446
5	β_T	2.27666446
6	γ_B	2.805465
7	γ_T	3.79075

Clearly the value of Q corresponding to each pin is greater than 1.15 proving that the crankshaft will not fail under bending or torsion. The values of radial and transverse bending moments corresponding to each section of the crankshaft and the formulae used for calculation of the alternating stresses are given in Appendix.2. Fig.3.3. shows the values of nodal stress and strain that were obtained after the static analysis performed on ANSYS APDL. Shear stress, shear strain and nodal displacement plots are given in Appendix.3.



Working of the Propesed Design

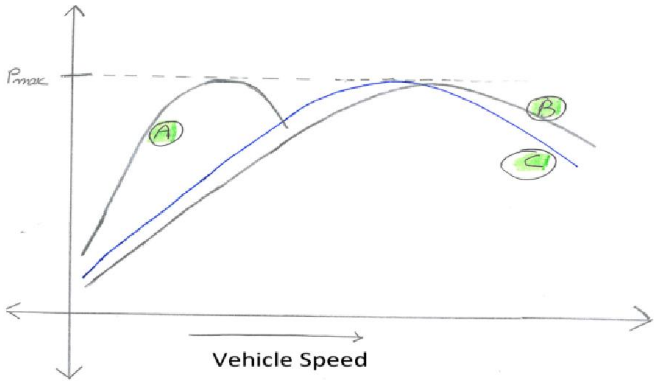
The crankshaft system has been designed in such a way that the crankweb will function as a counterweight as well as a helical gear that will transmit power to the secondary shaft that will be meshed with the new crankshaft. Minimum torque required to move the vehicle forward from rest when the engine is operating at its maximum torque configuration is given by Eqn.4.1.

Minimum torque= (d*g*r*f)/(2*R_1*R_2)=139.81N-m - (4.1)

Torque at which the engine is operating=250N-m
Torque that is getting wasted=110.19N-m

S.No	Crankpin Number	M_{BON}	σ_{BON}	σ_{BO}	σ_{TO}	σ_V	σ_B	Q= σ_B / σ_V
1	P1	59.245	7.22	20.2751	13.2	25.6512739	231.13	9.010469
2	P2	187.275	22.84	64.0768	13.2	66.06975887	231.13	3.498272
3	P3	278.10	33.92	95.1613	13.2	96.52006965	231.13	2.394632
4	P4	210.277	25.71	72.1285	13.2	73.90736823	231.13	3.127293

This torque can be converted into angular velocity of the crank shaft, hence increasing the speed of the vehicle. The resultant R.P.M. of the engine can be 4470 revolutions per minute. So when the engine is operating at the maximum torque configuration the RPM of the engine crankshaft can be increased for increasing the speed of the vehicle without compromising on the required acceleration. Hence the low revving engine is giving the same acceleration and speed to the vehicle as given by a high revving engine after implementing the new design of the crankshaft. The low revving engine can operate at lean air fuel mixtures hence reducing the fuel consumption, results in low wear and tear of the engine hence increasing the life of the engine and causing less pollution, most importantly improving the fuel economy of the vehicle. In Fig.4.1.three engines have been considered, engine A which is a low revving engine, engine B which is a high revving engine and engine C which has the proposed design of the crankshaft and the secondary shaft.



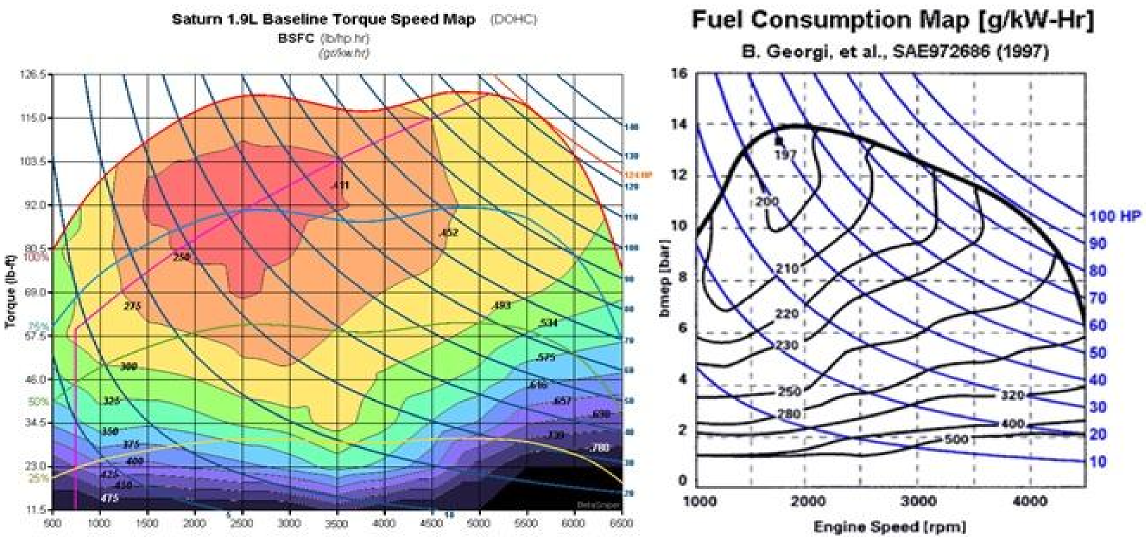
By using the overdrive gear ratios, the entire engine curve C has shifted towards the right increasing the speed of the vehicle but reducing the effective torque at the tyres and this change has been brought about by using overdrive ratios in the engine itself without touching the transmission system. But still the torque delivered by the engine C is higher than the torque given by engine B. The crankshaft system will operate at an overdrive ratio of 0.56:1; this means that if the crankshaft is rotating at a R.P.M. of 1000 then the secondary shaft will rotate at a R.P.M. of 1778 with 0.56 times torque available as that available at the crankshaft. Fig.4.2. shows two engine maps, one of a high revving engine and the other of a low revving engine

We make the following assumptions before analysing the two maps for the sake of simplicity of analysis:-

1. Both the engines are functioning at the same power of 60 HP
2. The time frame taken into consideration is one hour
3. The low revving engine is having a R.P.M. of 3000
4. Both engines run on diesel
5. The cars are running at a constant gear ratio of 1:1 throughout the duration of study.
6. The total diameter of the tyre of both the vehicles is 600mm
7. The final drive ratio in both the vehicles is 3.937:1.
8. Density of the diesel used is 845kg/m^3
9. One gallon=3.785L(USA standards)
10. All vehicles (trucks, busses, trains, cars, twowheelers and all other commercial vehicles)run on only two types of engines as specified above.

The gear ratio in the proposed design is 1:1.788; so if the low R.P.M. is revving at an RPM of 3000 then the output that is obtained from the proposed design is approximately 5364. At 3000 RPM and power of 60HP the bsfc is approximately 220g/Kw-hr and at the RPM of 5364 and engine power of 60HP the bsfc is around 310g/Kw-hr. if the engine runs for one hour then the fuel consumed by the low revving engine is approximately 9.834Kg of fuel and the high RPM engine will consume around 13.869Kg of fuel.Equation 4.2.gives a formula for obtaining car speed from the engine R.P.M. Carspeed(mph)=(RPM*cir)/(gear*final*88) (4.2)RPM=engine RPMCir= tyre circumference in feetGear= gear ratio of the carFinal= final drive ratio of the car88=combines several conversion factors

For an engine RPM of 5364, gear ratio of 1:1, final drive ratio of 3.937:1 and tyre circumference of 6.18 feet the vehicle speed will be approximately 95.7. So in one hour both the vehicles will travel a distance of 95.7 miles. For the same distance travelled by both the vehicles the high revving engine will consume 4.33 gallons of diesel whereas the low revving engine will consume around 3.07 gallons of diesel. So the mileage given by the car running on high revving engine will be around 22.10mpg and the mileage given by the low revving engine will be around 31.17mpg, hence serving the purpose of the design.



Applications / Expeted Benefits

The design of the crankshaft system proposed in this paper aims at using the low revving engines instead of high revving engines providing the same acceleration and speed as provided by the high revving engines but giving much better fuel economy by the conversion unnecessary torque into angular velocity of the crankshaft. This design can be implemented in the engines of all the automobiles, ships, aircrafts, railway engines and all other defence vehicles. This idea can be extended to all the systems which involve the use of reciprocating mechanisms. This design can be used in diesel generators used for generating electricity in markets, marine propulsion, construction equipment and backup power applications. In all these applications, high angular velocity of the crankshaft can be achieved using low revving engines and power generation can be more efficient and clean. Benefits of a low revving engine and a high revving engine are now available in a single engine having the proposed design of the crankshaft. Also the lubrication system of the engine can be made fewer complexes by employing the lubrication system of the transmission box which may reduce the weight of the engine hence compensate for the increase in the weight due to addition of another shaft in the crankshaft system.

Conclusion

The proposed design of the crankshaft will enable to make an engine capable to deliver the benefits expected from a high revving engine and a low revving engine. So we will get high speed, better fuel economy and lower wear and tear of the piston rings; all in a single engine

REFERENCES

Journals

Bellakhdhar *et al*. 2013. A simplified coupled crankshaft-engine block model, ComptesRendus – Mecanique, 341, 11-12, 743-754.

Boretti, A. and Scalzo J. 2013. Novel crankshaftmechanism and regenerative braking system to improve the fuel economy of passenger cars, Lecture Notes in Electrical Engineering, 189 LNEE, vol.1, 19-44.

Charles. P. *et al*. 2009. Detecting the crankshaft torsional vibration of diesel engines for combustion related diagnosis, *Journal of Sound and Vibration*, 321, 3-5, 1171-1185.

Gu, Y and Z. Zhou 2011. Strength analysis of diesel engine crankshaft based on PRO/E and ANSYS, Proceedings - 3rd *International Conference on Measuring Technology and Mechatronics Automation*, ICMTMA 2011, 3, 362-364.

Lie, X. *et al*. 2006. Simulation on the motion of crankshaft with crack in crankpin-web fillet region, 295, 3-5, 890-905.

Meng, J. *et al*. 2010. Intension Analysis of 3-D Finite Element Analysis on 380 diesel crankshaft, Proceedings - 2010 *International Conference on Computational and Information Sciences,* ICCIS 2010*, 1269-1272.*

Metkar, R. *et al*. 2013. Evaluation of FEM based fracture mechanics technique to estimate life of an automotive forged steel crankshaft of a single cylinder diesel engine Procedia Engineering, 51, 567-572.

Mourelatos, Z. 2001. A crankshaft system model for structural dynamic analysis of internal combustion engines, Computers & Structures, 79, 20-21, 2009-2027.

Okamura, H. *et al*. 1995. Simple Modeling and Analysis for Crankshaft Three-Dimensional Vibrations, Part 1: Background and Application to Free Vibrations *Journal of Vibration and Acoustics*, 117, 1, 1-70.

Rakopolous, C. and Giakoumis E. 1998. Simulation and analysis of a naturally aspirated IDI diesel engine under transient conditions comprising the effect of various dynamic and thermodynamic parameters, Energy Conversion and Management, 39, 5-6, 465-484.

Rizzoni, G. and W.Ribbens 1989. Crankshaft position measurement for engine testing, control, and diagnosis, IEEE 39th Vehicular Technology Conference.

Chien W. *et al*. (2005).Fatigue analysis of crankshaft sections under bending with consideration of residual stresses, International Journal of Fatigue, 27, 1, 1-19.

Y.Kang *et al*. 2003. An accuracy improvement for balancing crankshaftsMechanism and Machine Theory, 38, 12, 1449-1467.

Yu, S. *et al*. 2009. Nonlinear dynamics of a simplified engine-propeller system, Communications in Nonlinear Science and Numerical Simulation, 14, 7, 3149-3169.

Zhao, G. and Jiang, Z. 2009. Study on torsional stiffness of engine crankshaft, IFCSTA 2009 Proceedings - 2009 *International Forum on Computer Science-Technology and Applications*, 3, 431-435.

Zhao, J. *et al*. 2014. Analysis for fatigue failure causes on a large-scale reciprocating compressor vibration by torsional vibrationProcedia Engineering, 74, 170-174.

BOOKS

Budynas G. Richard and J. Keith Nisbett. 2008 *Shigley's Mechanical Engineering Design*. 9th edn.

Mcgraw Hill Education (India) Private Limited, New Delhi.

Ghosh Amitabha and Mallik Asok Kumar. 1998. *Theory of Mechanisms and Machines*. 3rd edn. East-West press Private Limited, New Delhi.

WEBSITE

Georgi, B. *et al*., SAE972686 1997. http://www.pics.tdiclub.com/showphoto.php?photo=11263 &title=fuel-consumption-model-fit&cat=500

Saturn DOHC LL0 1999. http://ecomoddercom/wiki/ index.php/Brake_Specific_Fuel_Consumption_(BSFC)_Ma ps# Saturn_1.9L_DOHC

REPORTS AND USER GUIDE

Germanischer Lloyd 2012. Rules for Classification and Construction VI-Part Additional Rules and Guidelines. GL 2012

Appendix-1

1. $\alpha_B = 2.6914 \cdot f(s,w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f$ (recess)

2. $f(s,w) = -4.1883 + 29.2004 \cdot w - 77.5925 \cdot w^2 + 91.9454 \cdot w^3 - 40.0416 \cdot w^4 + +(1-s) \cdot (9.5440 - 58.3480 \cdot w + 159.3415 \cdot w^2 - 192.5846 \cdot w^3 + 85.2916 \cdot w^4) + (1-s)^2 \cdot (-3.8399 + 25.0444 \cdot w - 70.5571 \cdot w^2 + 87.0328 \cdot w^3 - 39.1832 \cdot w^4)$
3. $f(w) = 2.1790 \cdot w^{0.7171}$
4. $f(b) = 0.6840 - 0,0077 \cdot b + 0.1473 \cdot b^2$
5. $f(r) = 0.2081 \cdot r^{-0,5231}$
6. $f(d_G) = 0.9993 + 0.27 \cdot d_G - 1.0211 \cdot d_G^2 + 0.5306 \cdot d_G^3$
7. $f(d_H) = 0.9978 + 0.3145 \cdot d_H - 1.5241.d_H^2 \cdot + 2.4147 .d_H^3$
8. $f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$
9. $\alpha_T = 0.8 \cdot f(r,s) \cdot f(b) \cdot f(w)$
10. $f(r,s) = r^{[-0.322 + 0.1015 \cdot (1-s)]}$
11. $f(b) = 7.8955 - 10.654 \cdot b + 5.3482 \cdot b^2 - 0.857 \cdot b^3$
12. $f(w) = w^{-0.145}$
13. $\beta_B = 2.7146 \cdot f_B(s,w) . f_B(w) \cdot f_B(b) . f_b(r) . f_B(d_G)\, f_{B(d_H)} \cdot f(recess)$
14. $f_B(s,w) = -1.7625 + 2.9821 \cdot w - 1.5276 \cdot w + (1-s) \cdot (5.1169 - 5.8089 \cdot w + + 3.1391 \cdot w) + (1-s^2) \cdot (-2.1567 + 2.3297 \cdot w - 1.2952 \cdot w^2)$
15. $f_B(w) = 2.2422 \cdot w0.7548$
16. $f_B(b) = 0.5616 + 0.1197 \cdot b + 0.1176 \cdot b2$
17. $f_B(r) = 0.1908 \cdot r(-.5568)$
18. $f_B(d_G) = 1.0012 - 0.6441 \cdot d_G + 1.2265 \cdot d_G^2$
19. $f_B(d_H) = 1.0022 - 0.1903 \cdot d_H + 0.0073 \cdot d_H^2$
20. $f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$
21. $\beta_Q = 3.0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(recess)$
22. $f_Q(s) = 0.4368 + 2.1630 (1-s) - 1.5212 (1-s)^2$
23. $f_Q(b) = -0.5 + b$
24. $f_Q(r) = 0.5331 \cdot r^{(-0.2038)}$
25. $f_Q(d_H) = 0.9937 - 1.1949 \cdot d_H + 1.7373 \cdot d_H^2$
26. $f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$
27. $f_Q(w) = w/(0.0637+0.9369.w)$
28. $\beta_T = \alpha_T$
29. $\gamma_B = 3 - 5.88 \cdot d_0 + 34.6 \cdot d_0^2$
30. $\gamma_T = 4 - 6 \cdot d_o + 30 \cdot d_o^2$

Appendix-1

1. $\alpha_B = 2.6914 \cdot f(s,w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(recess)$
2. $f(s,w) = -4.1883 + 29.2004 \cdot w - 77.5925 \cdot w^2 + 91.9454 \cdot w^3 - 40.0416 \cdot w^4 + +(1-s) \cdot (9.5440 - 58.3480 \cdot w + 159.3415 \cdot w^2 - 192.5846 \cdot w^3 + 85.2916 \cdot w^4) + (1-s)^2 \cdot (-3.8399 + 25.0444 \cdot w - 70.5571 \cdot w^2 + 87.0328 \cdot w^3 - 39.1832 \cdot w^4)$
3. $f(w) = 2.1790 \cdot w^{0.7171}$
4. $f(b) = 0.6840 - 0,0077 \cdot b + 0.1473 \cdot b^2$
5. $f(r) = 0.2081 \cdot r^{-0,5231}$
6. $f(d_G) = 0.9993 + 0.27 \cdot d_G - 1.0211 \cdot d_G^2 + 0.5306 \cdot d_G^3$
7. $f(d_H) = 0.9978 + 0.3145 \cdot d_H - 1.5241.d_H^2 \cdot + 2.4147 .d_H^3$
8. $f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$
9. $\alpha_T = 0.8 \cdot f(r,s) \cdot f(b) \cdot f(w)$
10. $f(r,s) = r^{[-0.322 + 0.1015 \cdot (1-s)]}$
11. $f(b) = 7.8955 - 10.654 \cdot b + 5.3482 \cdot b^2 - 0.857 \cdot b^3$
12. $f(w) = w^{-0.145}$
13. $\beta_B = 2.7146 \cdot f_B(s,w) . f_B(w) \cdot f_B(b) . f_b(r) . f_B(d_G)\, f_{B(d_H)} \cdot f(recess)$
14. $f_B(s,w) = -1.7625 + 2.9821 \cdot w - 1.5276 \cdot w + (1-s) \cdot (5.1169 - 5.8089 \cdot w + + 3.1391 \cdot w) + (1-s^2) \cdot (-2.1567 + 2.3297 \cdot w - 1.2952 \cdot w^2)$
15. $f_B(w) = 2.2422 \cdot w0.7548$
16. $f_B(b) = 0.5616 + 0.1197 \cdot b + 0.1176 \cdot b2$
17. $f_B(r) = 0.1908 \cdot r(-.5568)$
18. $f_B(d_G) = 1.0012 - 0.6441 \cdot d_G + 1.2265 \cdot d_G^2$
19. $f_B(d_H) = 1.0022 - 0.1903 \cdot d_H + 0.0073 \cdot d_H^2$
20. $f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$
21. $\beta_Q = 3.0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(recess)$
22. $f_Q(s) = 0.4368 + 2.1630 (1-s) - 1.5212 (1-s)^2$
23. $f_Q(b) = -0.5 + b$
24. $f_Q(r) = 0.5331 \cdot r^{(-0.2038)}$
25. $f_Q(d_H) = 0.9937 - 1.1949 \cdot d_H + 1.7373 \cdot d_H^2$
26. $f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$
27. $f_Q(w) = w/(0.0637+0.9369.w)$
28. $\beta_T = \alpha_T$
29. $\gamma_B = 3 - 5.88 \cdot d_0 + 34.6 \cdot d_0^2$
30. $\gamma_T = 4 - 6 \cdot d_o + 30 \cdot d_o^2$

$$\sigma_{BH} = \pm(\alpha_B \cdot \sigma_{BN}), \text{ [MPa]}$$

$$\sigma_{BG} = \pm(\beta_B \cdot \sigma_{BN} + \beta_Q \cdot \sigma_{QN}), \quad \text{[MPa]}$$

α_B– stress concentration factor for bending in the crankpin fillet
β_B– stress concentration factor for bending in journal fillet
β_Q– stress concentration factor for compression due to radial force in journal fillet

2.2. Alternating Bending Stresses in Outlet of Crankpin Oil Bore

$$\sigma_{BO} = \pm(\gamma_B \cdot \sigma_{BON}), \quad \text{[MPa]}$$

γ_B– stress concentration factor for bending in crankpin oil bore

$$\tau_N = \pm\frac{M_{TN}}{W_p} \cdot 10^3, \quad \text{[MPa]}$$

$$M_{TN} = \pm\frac{1}{2}\left[M_{T\max} - M_{T\min}\right], \quad \text{[Nm]}$$

$$W_p = \frac{\pi}{16}\left(\frac{D^4 - D_{BH}{}^4}{D}\right), \quad \text{or}$$

$$W_p = \frac{\pi}{16}\left(\frac{D_G{}^4 - D_{BG}{}^4}{D_G}\right), \quad \text{[mm}^3\text{]}$$

M_T– nominal alternating torgue in the crankpin or journal, [Nm]
W_p– polar section modulus related to cross-section of axially bored crankpin or bored journal, [mm^3],
$M_{T\max}$, $M_{T\min}$– maximum and minimum values of the torque, [Nm]

$$\tau_H = \pm(\alpha_T \cdot \tau_N), \quad \text{[MPa]}$$

$$\tau_G = \pm(\beta_T \cdot \tau_N), \quad \text{[MPa]}$$

α_T– stress concentration factor for torsion in crankpin fillet
β_T– stress concentration factor for torsion in journal fillet

$$\sigma_{TO} = \pm(\gamma_T \cdot \tau_N),\ [\text{MPa}]$$

γ_T– stress concentration factor for torsion in outlet of crankpin oil bore.

Type of engine	σ_{add} [N/mm²]
Crosshead engines	± 30 *)
Trunk piston engines	± 10

$$\sigma_v = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3 \cdot \tau_H^2}$$

$$\sigma_v = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3 \cdot \tau_G^2}$$

$$\sigma_v = \pm\frac{1}{3}\sigma_{BO} \cdot \left[1 + 2\sqrt{1 + \frac{9}{4}\left(\frac{\sigma_{TO}}{\sigma_{BO}}\right)^2}\right]$$

$$\sigma_{DW} = \pm K \cdot (0.42 \cdot \sigma_B + 39.3) \cdot \left[0.264 + 1.073 \cdot D^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{R_X}}\right]$$

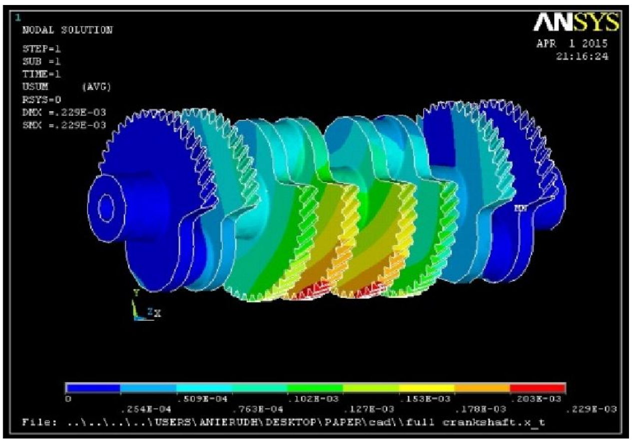
$R_X = R_H$ - in the fillet area
$R_X = D_o/2$ - in the oil bore area
σ_{DW}– allowable fatigue strength of crankshaft for bending, [MPa]
K – factor for different types of crankshafts without surface treatment
Values greater than 1 are only applicable to fatigue strength in fillet area
= 1.05 for continuous grain flow forged or drop-forged crankshafts,
= 1.0 for free form forged crankshafts
K – factor for cast steel crankshafts with cold rolling treatment in fillet area
= 0.93 for cast steel crankshafts manufactured by companies using a PRS approved cold rolling process
σ_B– minimum tensile strength of crankshaft material, [MPa]

$$Q = \frac{\sigma_{DW}}{\sigma_v}$$

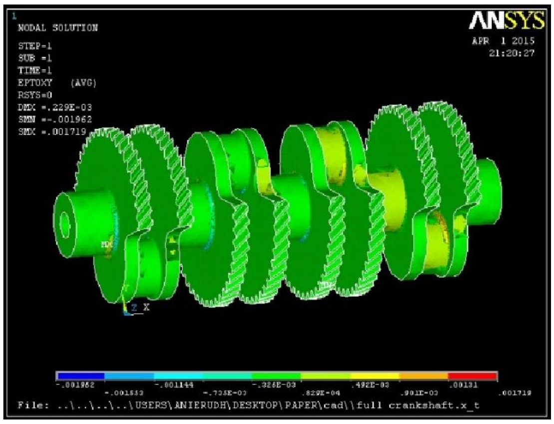
$$Q \geq 1.15$$

Appendix-3

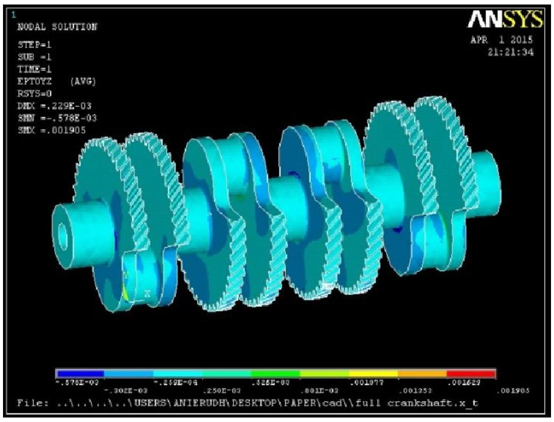
Finite Element Analysis Using ANSYS
Nodal Displacement



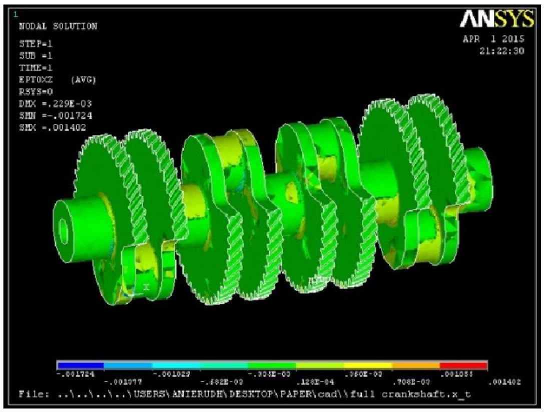
X-Y shear mechanical strain



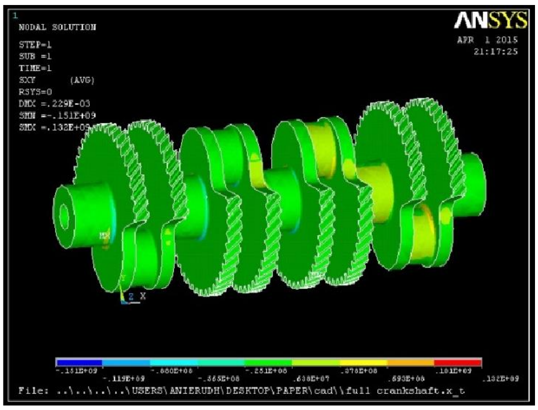
Y-Z shear mechanical strain



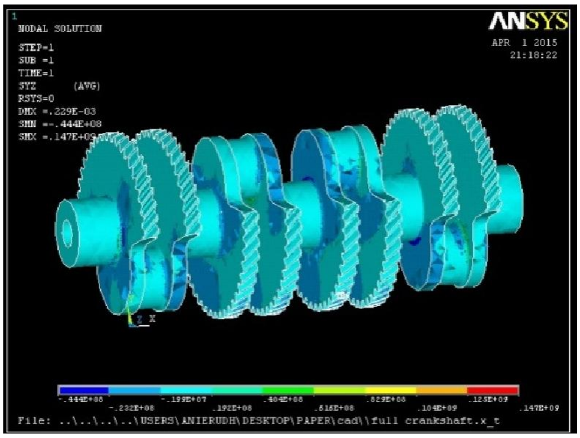
Z-X shear mechanical strain



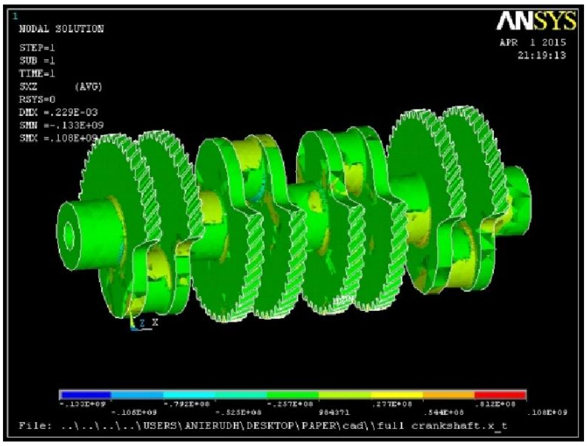
X-Y shear mechanical stress



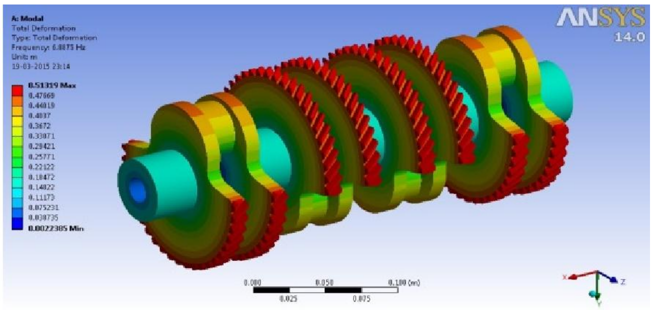
Y-Z shear mechanical stress



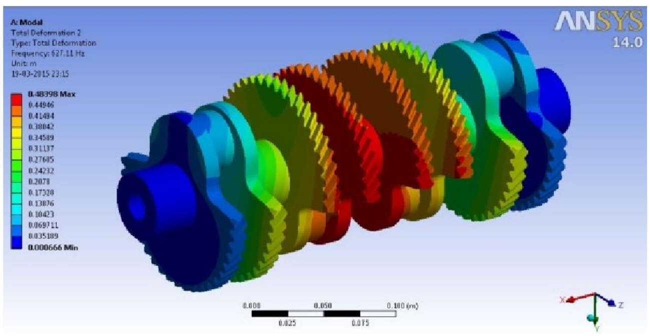
X-Z shear mechanical stress



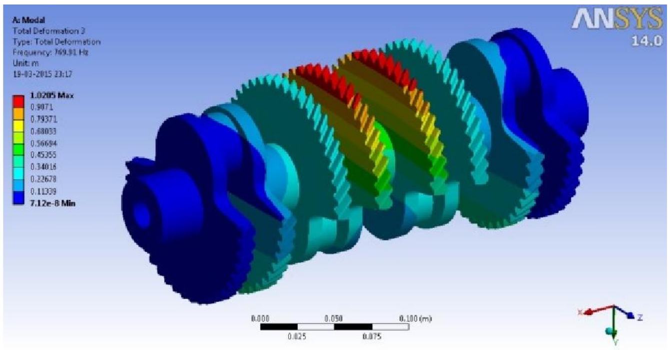
Nodal displacement for 6.8875Hz



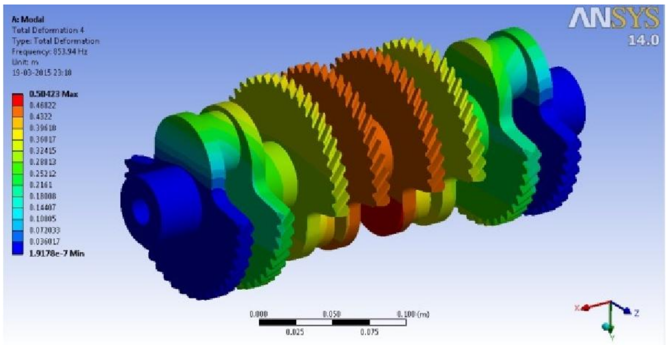
Nodal displacement for 627.11Hz



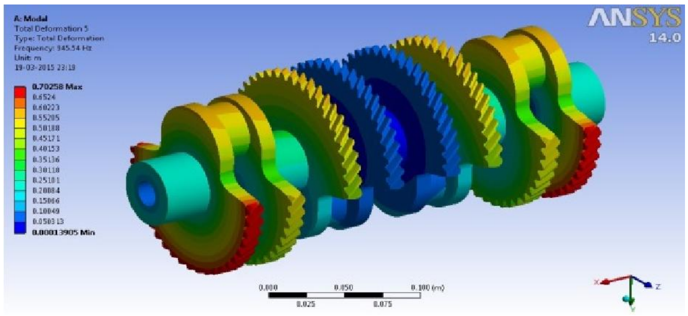
Nodal displacement for 769. 91Hz



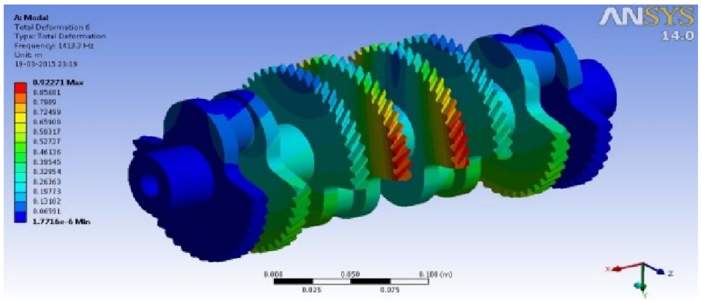
Nodal displacement for 853.94Hz



Nodal displacement for 945.54Hz



Nodal displacement for 1413.13Hz



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