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ABSTRACT

KEY WORDS: FEAD, belt, accessory, engine, vibration.

Serpentine belt drives with long, flat, multi-ribbed belts are nowadays widely used in the automobile industry for passenger vehicles and heavy duty trucks. With a single belt, the engine power is delivered from the crankshaft to all of the individual accessories such as air conditioner, alternator and power steering pump. To turn the belt some idler pulleys are used and to maintain proper belt tension as accessory loads and engine speed varies, a spring loaded tensioner is utilized. This system including accessories, belt and tensioner is commonly called Front Engine Accessory Drive (FEAD). In such systems some issues like belt vibration, pulley hubload, belt slip and tensioner pulley displacement are of great importance. These are effective in durability of belt, decreasing engine NVH, torque delivery to accessories without losses and increasing the lifetime of components.

The study on these systems has been considered by several researches. Apart from treating the belt-pulley systems mostly analytically [1, 2] for the developments of the engine FEAD systems more sophisticated software models are implemented. Beikmann et al [3] focused on the design parameters that determine how effectively the tensioner maintains a constant tractive belt tension, despite belt stretch due to accessory loads and belt speed. A nonlinear model predicting the operating state of the belt/tensioner system was derived, and solved using numerical and approximate closed-form methods. According to the results of the closed-form solution they concluded that a single design parameter, referred to as the “tensioner constant,” can measure the effectiveness of the tensioner. Tension measurements on an experimental drive system also confirmed the theoretical predictions.

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In FLR, the speed of engine run up in 120 seconds from 800 to 6000 rpm and in PLR the engine torque is 30 Nm and speed run up in 120 seconds from 800 to 6000 rpm. For modeling of the system a poly V belt with 6 ribs is used. A Torsional Vibration Damper (TVD) for crankshaft pulley and an automatic tensioner are also included. In Figure 2, the variation of crankshaft non uniformity for a turbocharged engine and a natural aspirated engine in full load ramp and part load ramp are compared. This figure shows that the non uniformity of crankshaft in low rpm of engine is very high because TVD is effective in high rpms. Figure 3 shows torque demands for the accessories versus engine speed. The specifications for all parts of accessory drive system including tensioner and belt stiffness and damping are provided in Table A of Appendix.

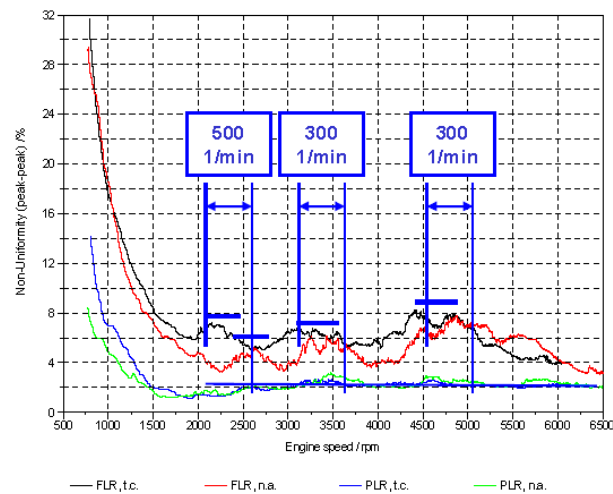


Figure 2. Non uniformity of crankshaft

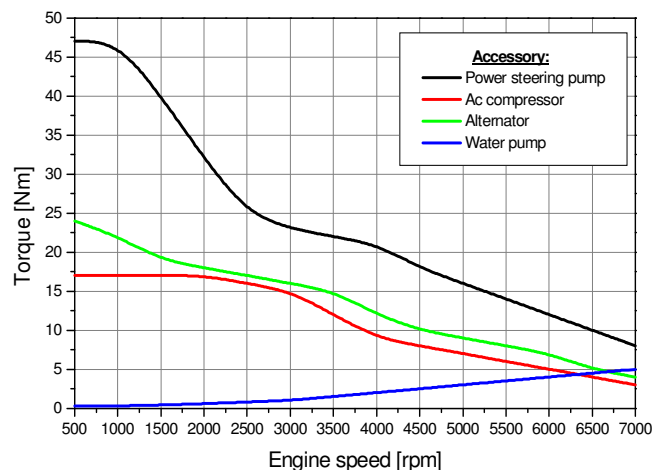


Figure 3. Torque consumption

3. VALIDATION OF FEAD SYSTEM MODEL

After modeling of FEAD system with ADAMS software, validation with the test data is performed. Test is carried out with real engine in FLR and PLR conditions. In Figure 4, one of the results for belt vibration between A/C compressor and crankshaft can be seen. The maximum difference between results of the model and the test is 5 percent. This difference is believed to be because of tolerances in position of accessories and decreasing of belt elasticity with increasing temperature in engine ramp up. Also this difference in low rpm and high rpm is more than middle rpm. In low rpm the difference is because of initial acceleration and in high rpm the difference is because of time step of software. In ADAMS software time step is fixed, so with increasing rpm, step angle increase and error is more than low rpm.

Other design factors such as pulley hubload, tensioner displacement and belt slip were also compared between test data and model results and similar agreements were observed.

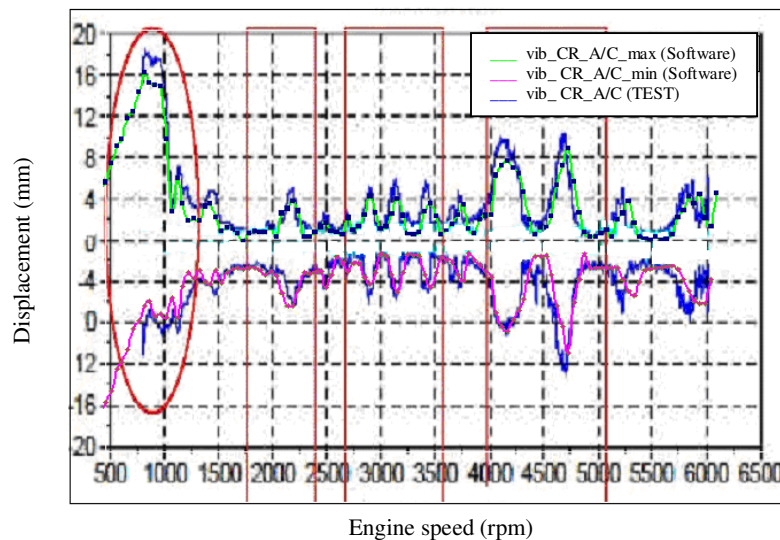


Figure 4. Belt vibration (comparison between results of test and developed model)

4. APPLICATION OF MODEL TO NEW DESIGNS

With the inspection of the belt vibration between A/C compressor and crankshaft, one can notice the amplitude of belt vibration is too high (Fig. 4). For decreasing belt vibration between A/C compressor and crankshaft pulley the first proposal is using an Over-running Alternator Pulley (OAP) with schematic shown in Figure 5. This kind of pulley is used for disconnection of alternator shaft and pulley when the belt forces the pulley for reverse rotation, because of fluctuations of the crankshaft. So with using this concept belt vibration and also dynamic belt tension is expected to decrease.

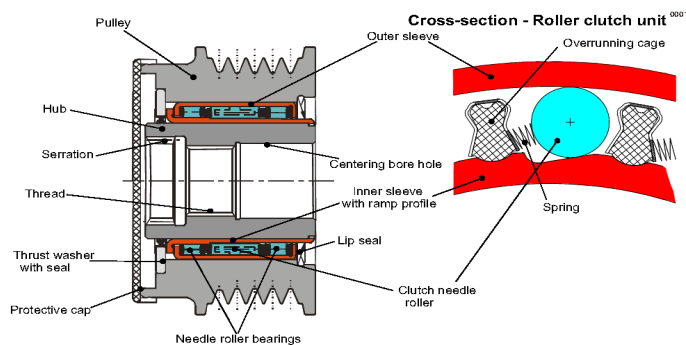


Figure 5. Schematic of an Over-running Alternator Pulley (OAP)

After using OAP in accessory belt drive, belt vibration between A/C compressor and crankshaft pulley are according to Figure 6. In this figure belt vibration for original system and new system with using OAP has compared together. According to the figure belt vibration has decreased some 40 percent in rpms lower than 2200 rpm and with increasing rpm of engine, effect of OAP has declined. The reason for decreasing effect of OAP in high rpms is that the spring of OAP is designed to act in low rpms. In fact in low rpms, fluctuation of crankshaft is higher than high rpms, so using OAP is more important at low rpms.

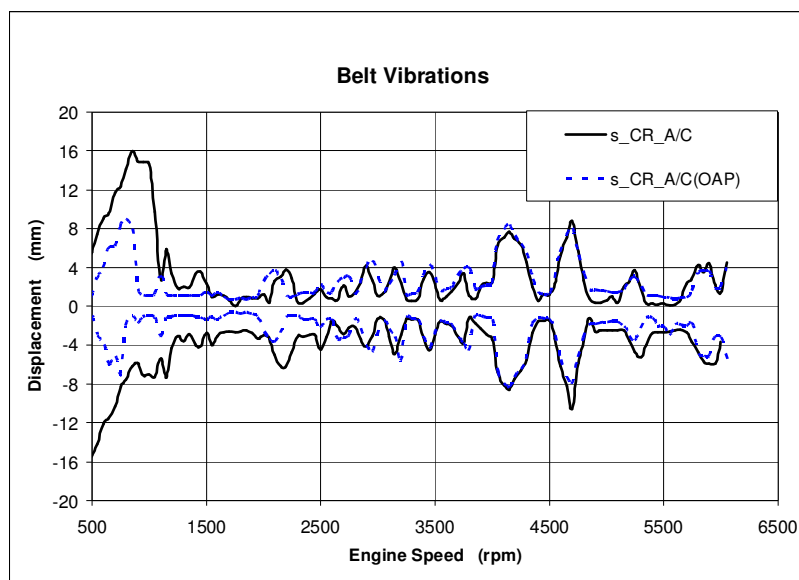


Figure 6. Belt vibration (comparison between result with OAP and without OAP)

The next item for comparing accessory layout is maximum hubload of idlers which is effective to the lifetime of idlers. In Figures 7 and 8, maximum hubload for idlers 2 and 3 are shown. The condition of this test is full engine load with A/C compressor ON, alternator 80A and power steering pump 50 bar. According to figures, the maximum hubload of idlers have decreased some 25 percent in low rpms. This pulley hubload decrease is effective on lifetime of ball bearings.

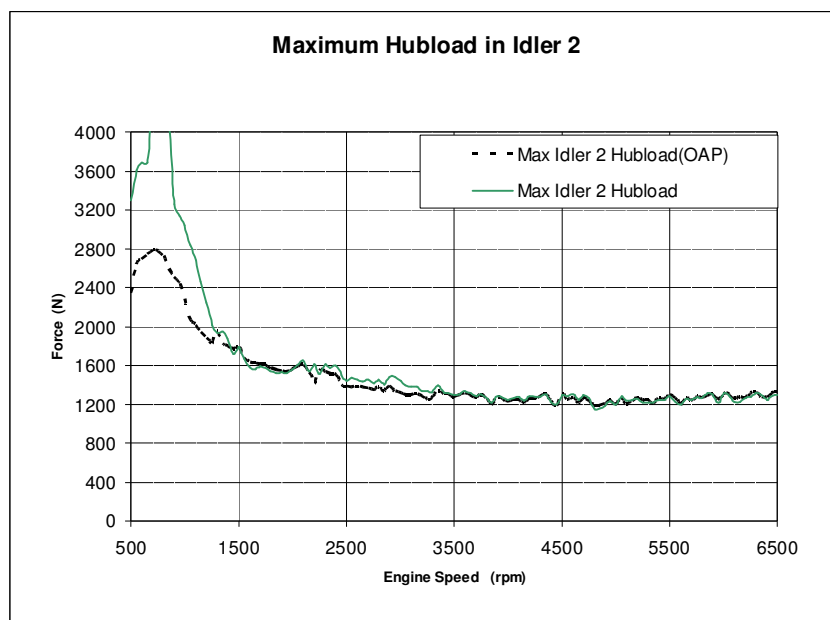


Figure 7. Maximum hubload in idler 2 (comparison between result with OAP and without OAP)

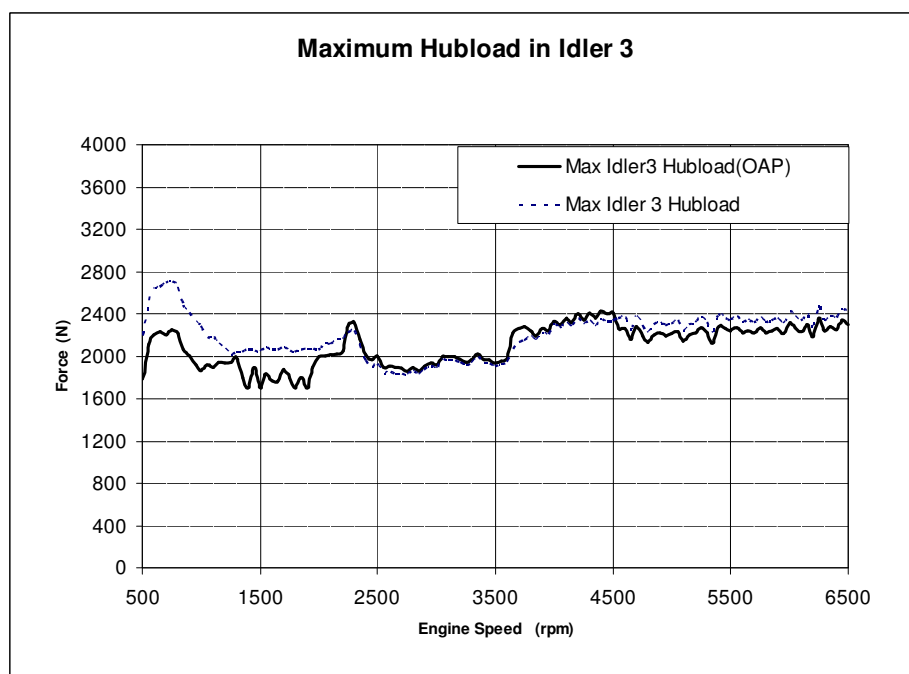


Figure 8. Maximum hubload in idler 3 (comparison between result with OAP and without OAP)

Another item in accessory drive is belt slip which is important because of effect to life time of belt and torque delivery to accessory component. Investigation of this item for alternator is more important relative to other auxiliary drives, because the diameter of alternator pulley is less than others and its moment of inertia is larger than others. In Figure 9 belt slip in alternator pulley with engine full load condition, A/C compressor ON, alternator 80A and

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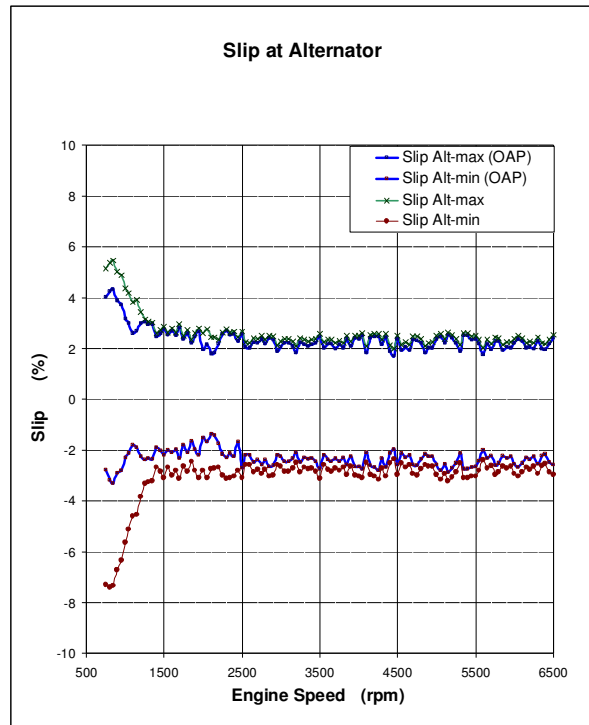


Figure 9. Slip at alternator (comparison between result with OAP and without OAP)

5. APPLICATION OF MODEL TO NEW DESIGNS

After achieving improvements in accessory drive by the use of OAP, yet another effect can be investigated by changing the layout of the system. In order to reduce belt vibration and pulley hubload, the layout can be modified by changing the position of tensioner and accessories as shown in Figures 10. In Figure 10 (a) wrap angle of A/C compressor, water pump and tensioner have increased. In Figure 10 (b) wrap angle of A/C compressor has increased and the tensioner is in fact positioned nearer to the crankshaft. With this concept, non uniformity of crankshaft is expected to damp out rapidly in FEAD system and the tension in slack side of belt to become more suitable, so that the belt vibration is expected to decrease with this layout.

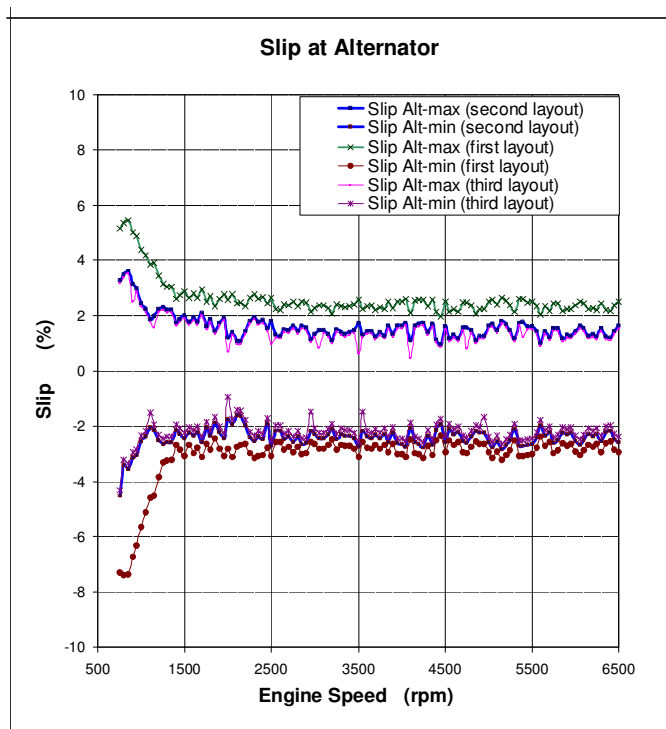


Figure 11. Slip at alternator (comparison between three layouts)

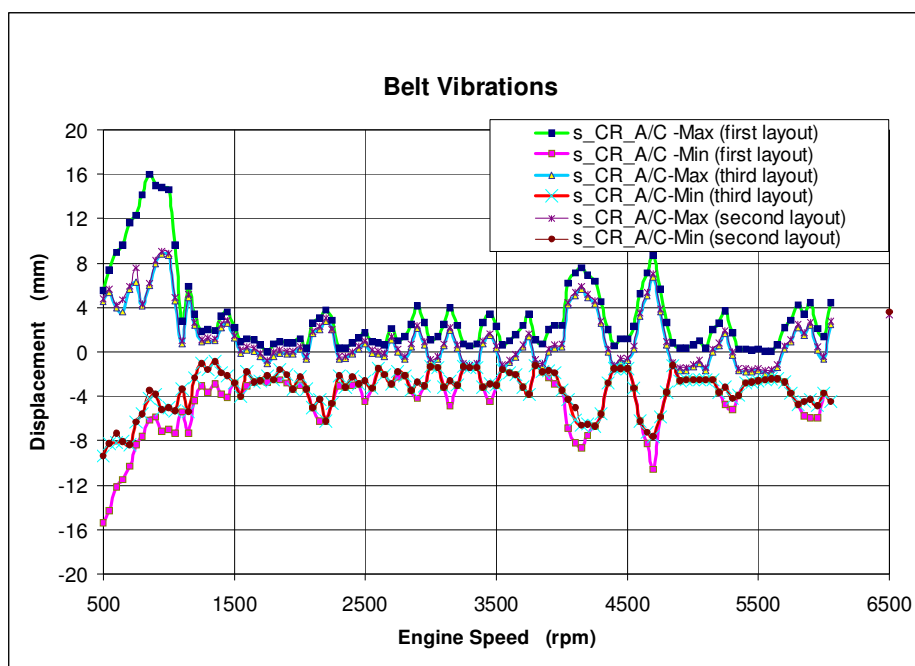


Figure 12. Belt vibration (Comparison between three layouts)

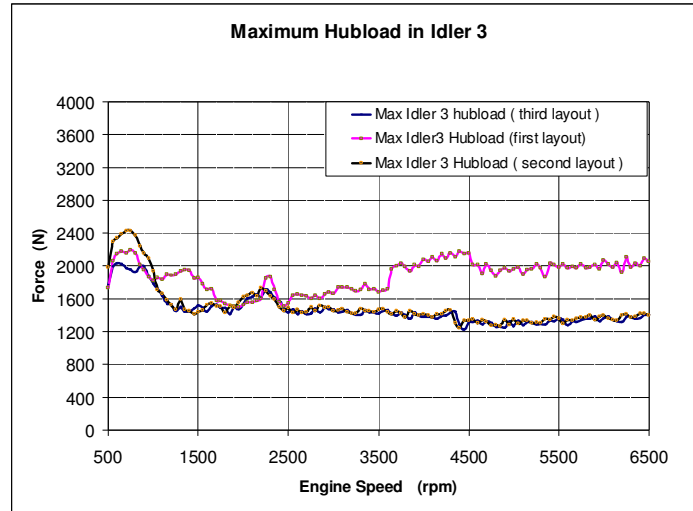


Figure 13. Maximum hubload in idler 3(comparison between three layouts)

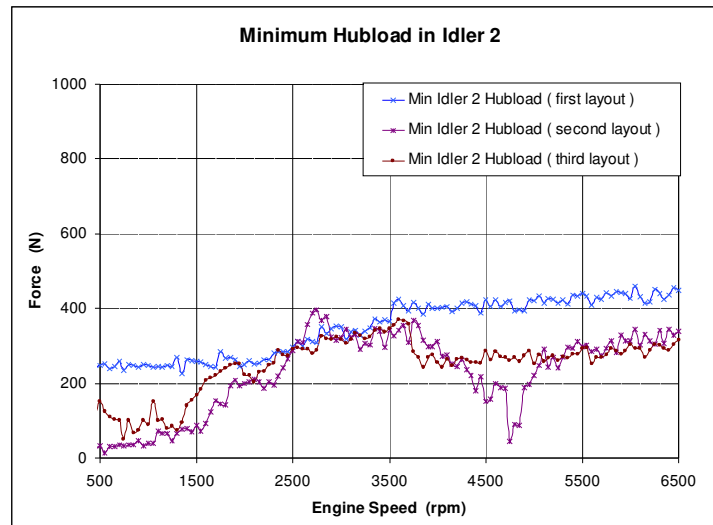


Figure 14. Minimum hubload in idler 2(comparison between three layouts)

Table 1 below is constructed in order to summarize the comparison made among the three layouts. According to this table, the third layout is of suitable characteristics. This layout has benefit in all items.

Table 1. Summary of comparisons among the characteristics of the three layouts

	Parameter	First layout	Second layout	Third layout
1	Wrap angle between belt and water pump	+	-	+
2	Belt vibration between A/C compressor and crankshaft	-	+	+
3	Minimum hubload of idler 2	-	-	+
4	Maximum hubload of idler 3	-	+	++
5	Slip of alternator pulley	-	+	+

6. APPLICATION OF MODEL TO THE THIRD LAYOUT

According to Table 1, the third layout was chosen among the proposed layouts. The last investigation in this paper would be using this layout together with OAP. This system is compared with original system for clearer comparison.

In Figure 15 the maximum hubload for idler 2 is shown. According to this figure with using this new layout maximum hubload can be reduced in low rpms up to 40 percent and in high rpms up to 20 percent. In Figure 16 is shown the maximum hubload for idler 3 that demonstrates a near 20 percent decrease in the load.

In Figure 17 the belt slip at crankshaft is shown to decrease close to 25 percent. This is because of using OAP, reducing the belt length and damping vibration of the belt by approaching the tensioner to crankshaft pulley.

The last item for comparison between initial layout and improved layout is belt vibration between A/C compressor and crankshaft pulley which is plotted in Figure 18. According to this figure belt vibration has reduced up to 60 percent for below 2500 rpm and up to 10 percent for high speeds.

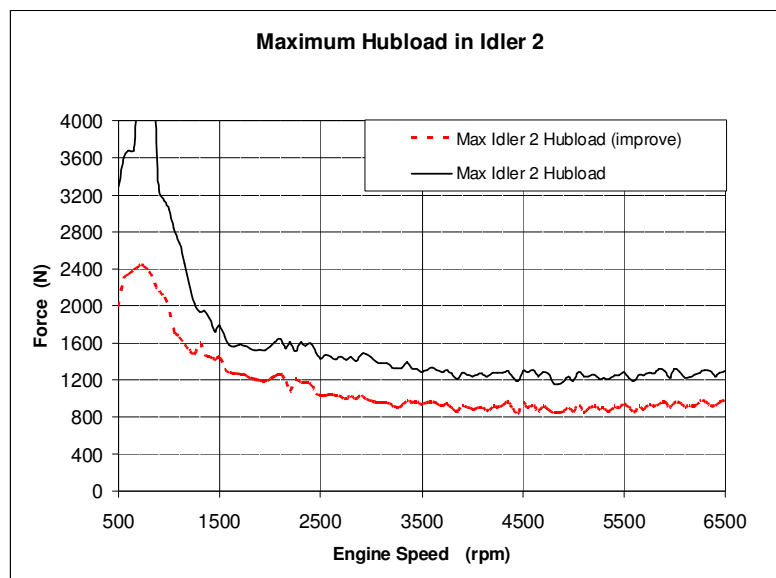


Figure 15. Maximum hubload in idler 2(comparison between initial layout and improve layout)

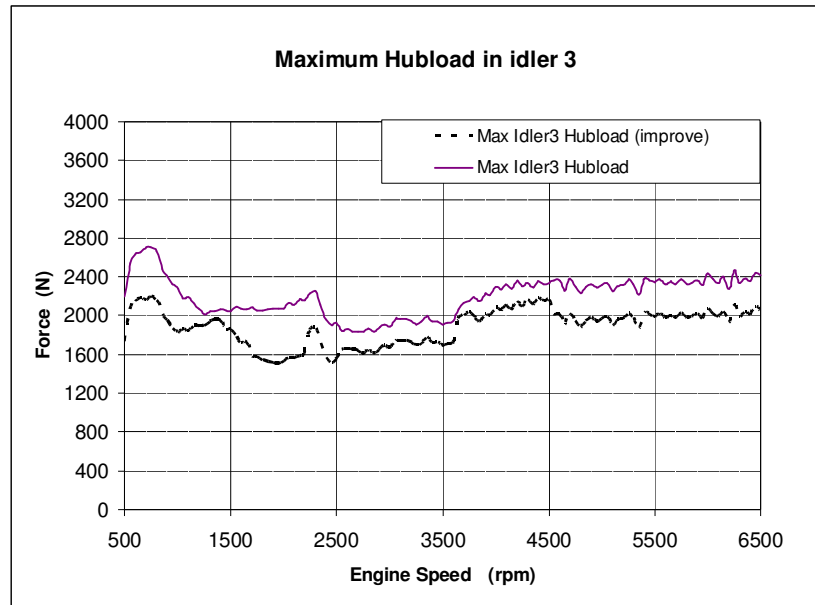


Figure 16. Maximum hubload in idler3(comparison between initial layout and improve layout)

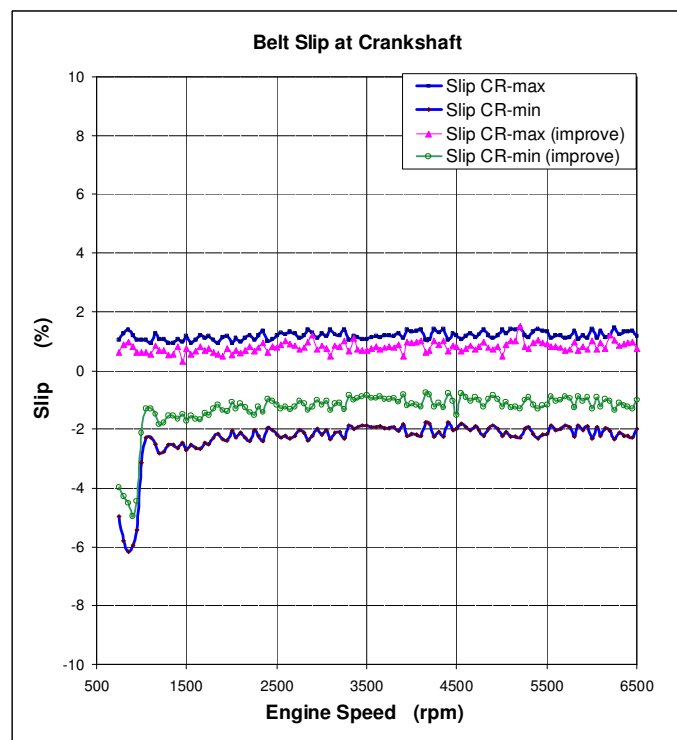


Figure 17. Belt slip at crankshaft(comparison between initial layout and improve layout)

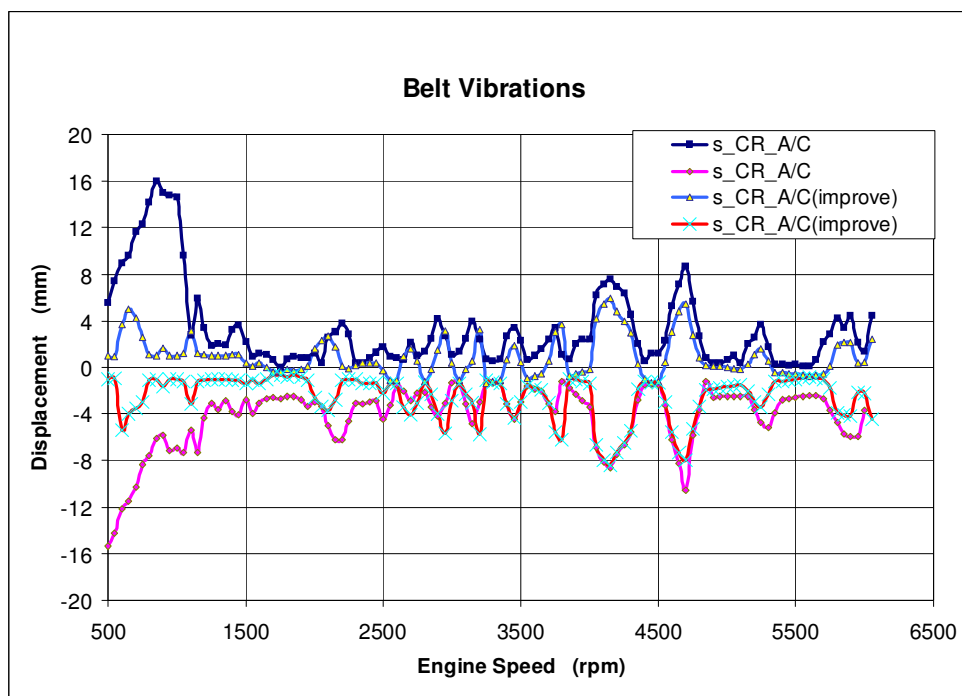


Figure 18. Belt vibration(comparison between initial layout and improve layout)

7. CONCLUSIONS

In this paper a sophisticated FEAD model was developed in ADAMS environment and generated results were validated by actual engine test results with close agreements of maximum 5 percent differences. The model was used to investigate the effect of different design issues and in one case with application of modifications to the layout, several benefits were achieved.

The results of the model generated in FLR and PLR for belt vibration, pulley hubload and belt slip have shown improvements for the chosen layout relative to original design of the engine FEAD system. These can summarized as:

- Reducing the hub load of idler 2 up to 40 percent at low speeds and up to 20 percent at high speeds. Also decrease in hub load of idler 3 up to 20 percent has been achieved. This improvement helps increasing the life time of idler bearing.
- Reducing up to 25 percent in belt slip with using the enhanced layout. This parameter helps to increase life time of belt and reduce the noise and increase the torque delivery to the consumers.
- Reducing the belt vibration between A/C compressor and crankshaft pulley up to 60 percent below 2500 rpm and up to 10 percent at high speeds. This item decreases risk of belt jumping and improves belt life time and noise.

Based on these results and with further consideration of the cost reduction inherent in the improved layout due to fewer idlers and smaller belt length, this layout has several advantages relative to original layout.

8. REFERENCES

- [1] Lingyuan Kong and Robert G. Parker, Coupled Belt-Pulley Vibration in Serpentine Drives With Belt Bending Stiffness, *Journal of Applied Mechanics*, 2004, Vol. 71, pp. 109-120.
- [2] LI Xiao-jun and Chen Li-qun, Modal analysis of coupled vibration of belt drive systems, *Journal of Applied Mathematics and Mechanics*, 2008, Vol. 29(1), pp. 9–13.
- [3] R. S. Beikmann, N. C. Perkins and A. G. Ulsoy, Design and Analysis of Automotive Serpentine Belt Drive Systems for Steady State Performance, *Journal of Mechanical Design*, June 1997, Volume 119, Issue 2, 162, doi:10.1115/1.2826231
- [4] Motoyasu Sakaguchi, Tomoaki Nishio, Toshimitsu Shinohara and Hiroshi Takagishi, Study of the Mechanism of Accessory Drive Belt Noise, *SAE International Journal of Passenger Cars- Mechanical Systems* October 2009 vol. 2 no. 1 434-439
- [5] H Zhu, Finite element modal analysis of the engine front end accessory drive systems, *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, Volume 208, Number D1 / 1994, pp 49-53

Information used in the ADAMS model of the FEAD system is provided in Table A.

Table A. Model information

No.	Part Name	Specifications	
1	Vibration Damper Pulley	1-Pitch Diameter	140 mm.
		2-Mass	6 Kg.
		3-Ixx, Ixy, Ixz, Iyy, Iyz, Izz	1000,0,0,1000,0,500
		4.Speed fluctuation	According to figure (2)
2	Water Pump Pulley	1-Pitch Diameter	140 mm.
		2-Mass	0.05Kg.
		3-Ixx, Ixy, Ixz, Iyy, Iyz, Izz	100,0,0,100,0,50
		4- Torque consumption	According to figure(3)
3	Power Steering Pump Pulley	1-Pitch Diameter	141.5 mm.
		2-Mass	0.1 Kg.
		3-Ixx, Ixy, Ixz, Iyy, Iyz, Izz	200,0,0,200,0,100
		4-Torque consumption	According to figure(3)
4	Alternator Pulley	1-Pitch Diameter	54.3 mm.
		2-Mass	0.15 Kg.
		3-Ixx, Ixy, Ixz, Iyy, Iyz, Izz	600,0,0,600,0,300
		5-Torque consumption	According to figure(3)
5	AC-Compressor Pulley	1-Pitch Diameter	119 mm.
		2-Mass	0.15 Kg.
		3-Ixx, Ixy, Ixz, Iyy, Iyz, Izz	200,0,0,200,0,100
		5-Torque consumption	According to figure(3)
6	Tensioner Pulley	1-Width	26 mm
		2- Radius	35 mm
		3-Mass	0.1Kg.
		4-Ixx, Ixy, Ixz, Iyy, Iyz, Izz	0.2,0,0,0.2,0,0.1
7	Tensioner	1-Length	58 mm
		2-Installation Angle	17.8 °
		3-Damping	0.06
		4-Stiffness	514.71
		6-Mass	0.05 Kg.
		7-Ixx, Ixy, Ixz, Iyy, Iyz, Izz	20,0,0,20,0,10
8	Idler-1	1-Width	26 mm
		2- Radius	32.5 mm
		3-Mass	0.05Kg.
		4-Ixx, Ixy, Ixz, Iyy, Iyz, Izz	20,0,0,20,0,10
9	Idler - 2	1-Width	26 mm
		2- Radius	32.5 mm
		3-Mass	0.05Kg.
		4-Ixx, Ixy, Ixz, Iyy, Iyz, Izz	20,0,0,20,0,10

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