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STUDY OF TOOTHED BELT TRANSMISSION IN CASE OF VARYING ANGULAR VELOCITY

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Abstract: *Toothed belt transmissions are widely used for internal combustion camshaft timing transmissions. Timing belt transmission behavior is studied by numerical simulations. Problems of force variations into the belt and belt tensioner behavior are discussed within a wide range of angular velocities. Effects of tensioner stiffness variations and various tensioner damping cases are considered. Optimum magnitude of tensioner stiffness and damping are given.*

Keywords: *timing belt, numerical simulation, tensioner stiffness and damping.*

1. INTRODUCTION

Toothed belt transmissions are often used for driving camshafts in automobile engines. Because of constraints issued from structural and manufacturing requirements, pulley axles are fixed, and use of a tensioner is necessary. The choice of tensioner spring stiffness and damping stays a real problem. Considering a layout of one transmission and with a given timing belt type, mechanical problems could be solved mainly through a good choice of tensioner spring stiffness and damping coefficient. In this paper, effects of stiffness and damping coefficient variations on the belt forces are presented.

2. MODEL OF THE STUDIED TRANSMISSION

A PSA DV4 engine timing transmission is modelized (Fig. 1) in order to make numerical simulations. A stiff and damped timing belt with HTD tooth profile connects six pulleys being the crankshaft pulley (1), idler pulley (2), camshaft pulley (3), injection pump pulley (4), tensioner pulley (5) and water pump pulley (6). The driving element of the transmission is the crankshaft pulley, having a given angular velocity. The whole characteristics of the model are given in the literature [4]. Theoretical bases of the simulation are also described in the literature [1], [3].

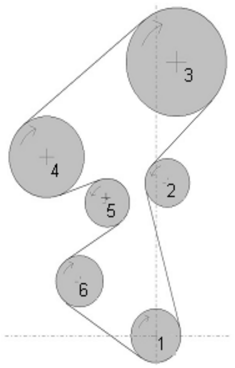

Fig. 1. Timing belt transmission of the DV4 engine

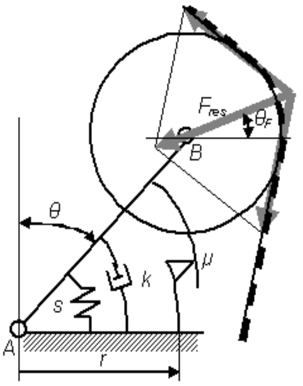

Fig. 2. Model of the belt tensioner

The model of the belt tensioner is presented in figure 2. More details can be found in the literature [1], [2a], [2b]. The dynamics equation for the tensioner arm torque equilibrium is:

$$M = s\theta + k\dot{\theta} + sign(\dot{\theta}) \cdot \mu \cdot r \cdot F_{res} \cdot \left|\sin(\theta - \theta_F)\right| \quad (1)$$

where s – torsional stiffness,
k – equivalent viscous damping,
θ – angular position of the tensioner arm,
F_{res} – resultant force on pulley bearing,
θ_F – angular position of the resultant force,
r – radius of the pivot “A”,
μ – friction coefficient on the pivot “A”.

Previous simulations have shown that the effect of the Coulomb friction can be neglected [2b]. Thus, damping by Coulomb friction is not included in these numerical simulations.

3. SIMULATIONS

Numerical simulations were realized in order to study the belt force variations at different angular velocities. Crankshaft angular velocity was increased from 800 rpm to 4800 rpm by steps of 100 rpm. The computed data are shown as a surface, where one horizontal axis is the rpm axis, the other horizontal axis is the crankshaft turning angle axis. Vertical axis gives the studied quantity (fig. 3), here for example the belt force in the crankshaft tight belt span is shown.

First, simulations with no damping into the belt tensioner were computed, with 5 Nm/rad tensioner stiffness. It can be seen how oscillations increase and disappear with varying angular velocity. First harmonics appear at very low angular velocities, and resonance frequencies appear near 2700 rpm. Then, oscillations decrease and disappear. Another harmonics appear at the same angular velocity, and become more and more important when the angular velocity increases (fig. 3). The highest value is obtained at the highest angular velocities (4800 rpm). Note also that oscillations increase in both ways: maxima are not only higher, but minima are also lower.

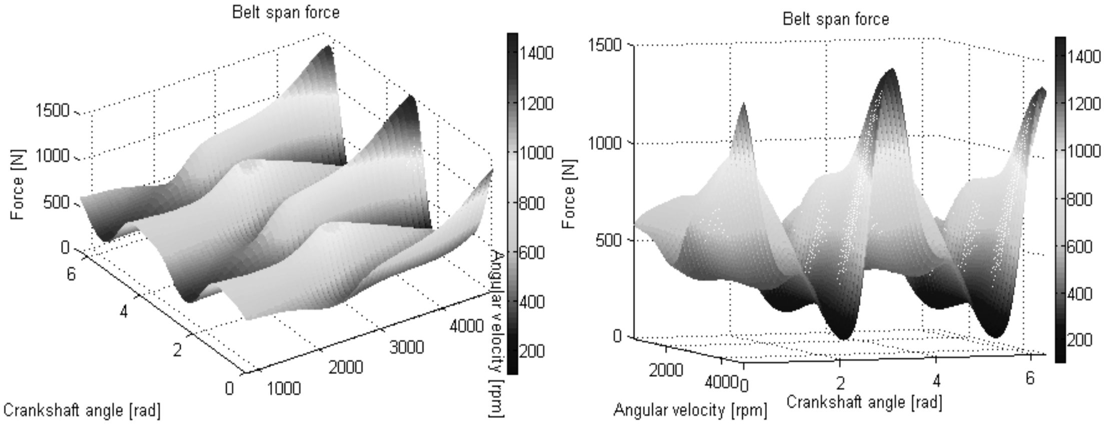

Fig. 3. Belt force in the crankshaft tight span, not damped case (two views of the same surface)

As maximum belt span forces are higher than 1000 N over 4000 rpm (fig. 3), values of the tensioner spring stiffness were varied in order to try to lower these belt span forces. First, the spring stiffness was decreased by 50% to 2,5 Nm/rad. Fig. 4 shows the belt span force variation regarding the original case. It can be seen, that less then 15 N (1%) force decrease and less than 10 N (0,7%) of force increase were obtained in the whole studied angular velocity domain. Then, a 50% spring stiffness increase (7,5 Nm/rad) was considered and compared to the original tested case. Here again, force differences are lower than 15 N (1%).

Then, simulations with viscous damping applied in the belt tensioner were computed with an original belt tensioner stiffness of 5 Nm/rad and with a viscous damping coefficient of 0,458 Nms/rad. The belt force variations regarding to the original case are shown in figure 5. The shape of the belt force surface is slightly transformed. The higher forces decreased with nearly 35% (500 N), while the lower forces increased by a factor of 21 % (300 N). This gave a smoother belt force surface compared to the original one. Thus, timing belt load variations are globally decreased. Note that the shape of the damped case belt force surfaces are modified comparing to the surface not damped (fig. 6). Thus, using appropriate viscous damping coefficient, it may be possible to decrease or remove the higher force peaks in an angular velocity zone where the engine is expected to turn most of the time.

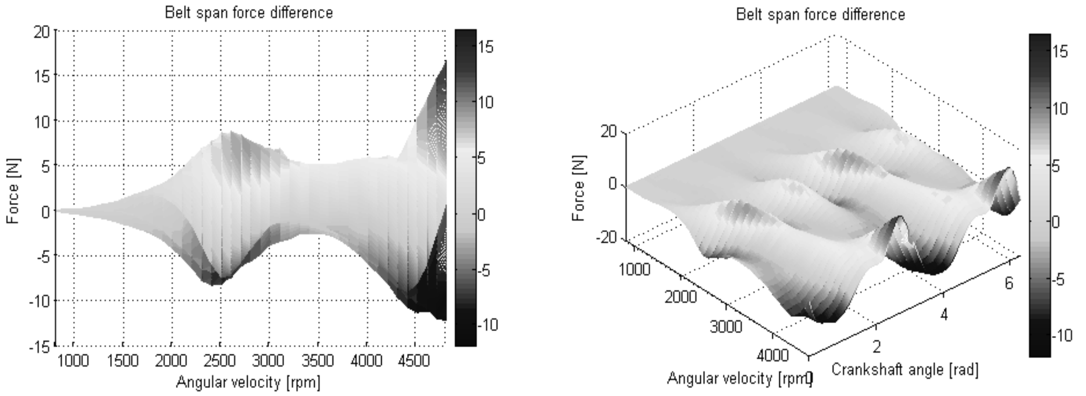

Fig. 4. Force difference in case of decreased tensioner stiffness, no damping (two views of the same surface)

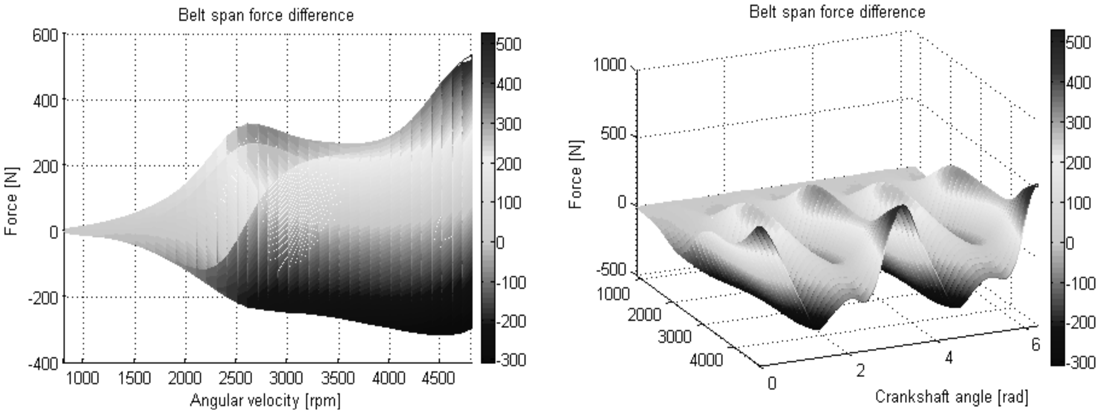

Fig. 5. Force difference in the damped case, original tensioner stiffness (two views of the same surface)

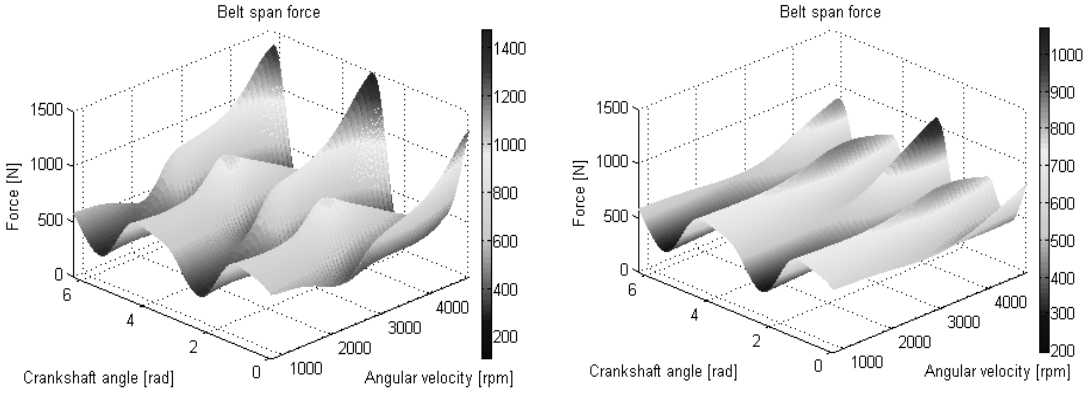

Fig. 6. Belt force in the crankshaft tight span at original stiffness, not damped case (left) and damped case (right)

4. DISCUSSION OF THE SIMULATION RESULTS

From the simulation results, it can be concluded that when no viscous damping is applied in the tensioner, the tensioner stiffness variation has little effect on the maximum and minimum of the belt span force for a given timing belt transmission. However, the use of viscous damping, gives an important decrease of the maximum force in the zones of the most important belt load. This effect observed earlier for some angular velocities [2b] can now be generalized to the whole studied angular velocity domain. A smoother belt force surface is obtained compared to the original one. Thus, belt load variation is decreased, which should have positive effect on belt lifetime.

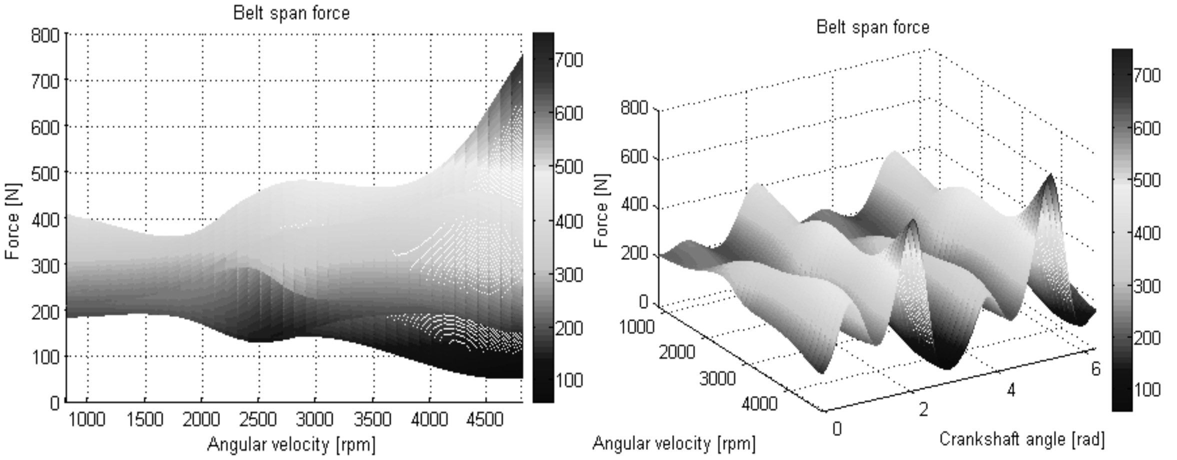

Fig. 7. Belt force in the crankshaft slack span, not damped case, original stiffness (two views of the same surface)

Attention must also be paid to the belt force in the crankshaft slack span. If the force is too low, for example at high angular velocities (fig. 7), tooth jump can happen. A tooth jump changes the relative position of the cam distribution and the crankshaft. Because of this, pistons can hit the opened valves provoking the failure of the engine. It can be seen in figure 8, that use of viscous damping increases force minima near 2500 rpm, thus improving slack belt span behavior. The surface bump is shifted to right, and provokes earlier force increase compared to the original case.

Note that tensioner arm angular displacement remains practically the same independently the arm stiffness and damping variations.

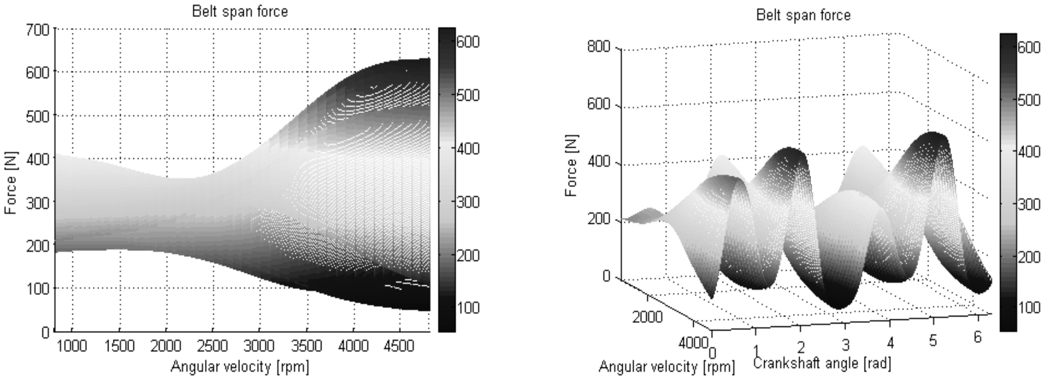

Fig. 8. Belt force in the crankshaft slack span, damped case, original stiffness (two views of the same surface)

5. CONCLUDING REMARKS

Systematic numerical simulations are made in order to understand in detail the effect of timing belt tensioner. The results show that tensioner stiffness variations have little effect on belt span forces. In contrary, to decrease high force values, the use of viscous damping is necessary. Tensioner viscous damping decreases belt forces not only at specific angular velocities, but also in a large angular velocity domain. Moreover, a well chosen damping may smooth force peaks for a large angular velocity domain of the transmission. When choosing a viscous damping coefficient value, it is also necessary to control its efficiency at high crankshaft angular velocities [2a]. In the same time, attention must be paid to the force in the crankshaft slack belt span to avoid tooth jump.

Further systematic studies are now needed to understand tooth belt meshing and tooth load shearing during the dynamic transmission in order to control in details the dynamic timing belt life.

6. REFERENCES

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