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STUDY REGARDING THE STRESS OF A HEAVY-VEHICLE CLUTCH

BY

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Abstract: The clutch of a heavy-vehicle is subjected to very important mechanical and thermal loads. For a close evaluation of the clutch operation conditions it was designed and used a Simulink model of the powertrain. For the same purpose, experimental measurements were also accomplished on a 38 ton tractor-semitrailer combination. Records obtained in normal and heavy road condition were used both to calibrate the model and to compare with the computer results. Other simulations were performed to find out the influences of design and functional parameters over the torsional moment and heat that affect the clutch. So, there were studied the influences of gear ratios, road slope or coupling rate and were made some evaluations of the clutch contact pressure, frictional mechanical work, wear and heating.

Key words: heavy-vehicle, clutch, simulation, experiment, mechanical and thermal loads

1. Introduction

The aim of this work was to determine the mechanical and thermal loads acting on the clutch of a heavy-vehicle. For that, combined studies, theoretical and experimental, were accomplished.

The object of study was a road train composed by a two-axle rear-drive tractor and a two-axle semitrailer. During road testing, the total mass of the combination was 38200 kg, from which 6500 kg and 10000 kg were distributed on the tractor axles.

The ROMAN 16320 FS tractor is powered by a direct-injection turbo-diesel engine developing 235 kW at 2600 rpm and 1000 Nm at 1600 rpm. The drivetrain includes a twin-disk dry clutch, a synchromesh gearbox with 8+2 speeds, a cardan driveshaft and an axle with locking differential and two-stage final drive (middle mounted and hub mounted). The gearbox includes a front splitter unit, realises a total ratios spread of 8.52 and 7th and 8th gears are, respectively, direct drive and overdrive.

2. Simulation Model

For the theoretical study of the powertrain dynamical behavior, it was firstly conceived a dynamical model consisting in seven inertial elements (equivalent flywheels), idealized shafts (with no mass) and four couplings. These couplings represent the clutch, two synchronizers (one for the main gearbox and one for the splitter) and the wheel-road interface. Based on this, a Matlab-Simulink model was designed [1], [2]. Its main module is presented in Figure 1. The blocks **Engine …Translation mass** compute the angular accelerations of the inertial elements and then obtain by integration the angular speeds and spaces. The torsional moments are computed as functions of flywheels kinematic values and actual positions of driver actuated pedals (accelerator, brake and clutch) and gearbox shifting lever.

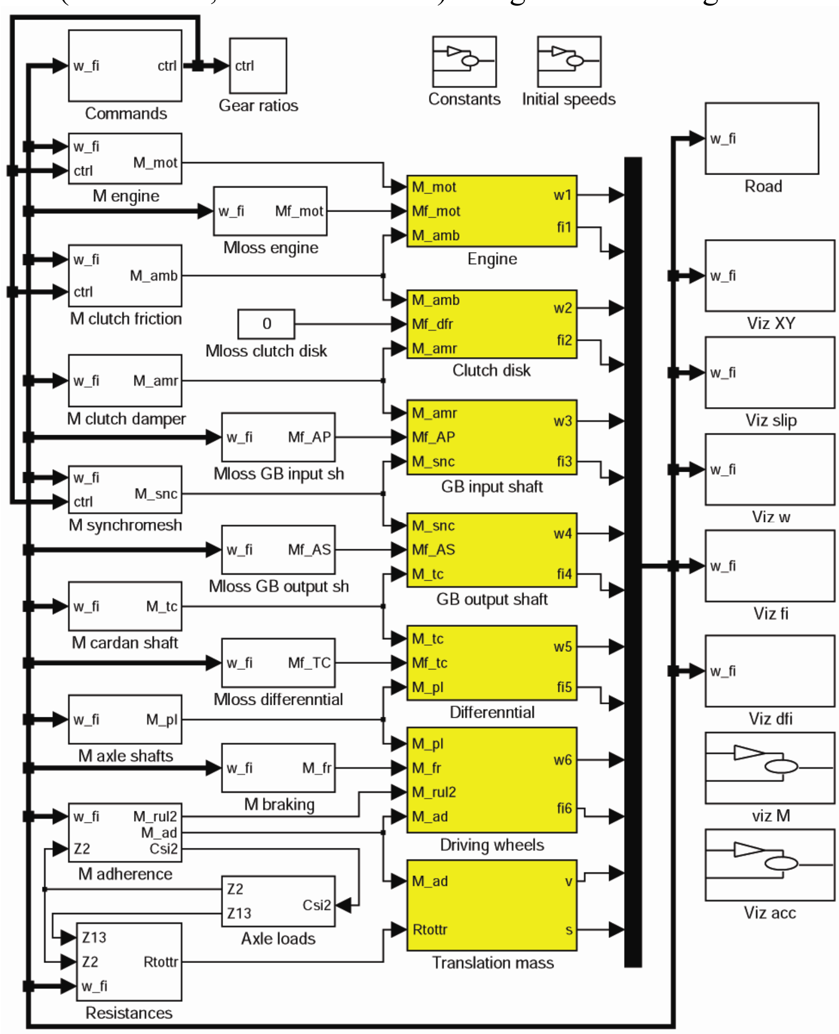

Fig. 1. Main module of the Matlab-Simulink model used to simulate the behaviour of a tractor-semitrailer drivetrain.

The model is complex enough to consider the quasi-static load redistribution between axles or the drivetrain losses (blocks **Axle loads**, respectively **Mloss…**). The block **M adherence** take into account the slip of the wheels, the tires load change and permits to input variable friction and rolling resistance coefficients as function of traveled distance.

Many problems needed to be surmounted in order to simulate the gear shifts. Part of these was caused by the change of model characteristics simultaneously with the transmission ratios. Others were due by the necessity to model the electro-pneumatic servo-mechanisms that actuate both the clutch and the synchromesh of the gearbox splitter unit.

3. Experimental Research

To obtain the loads stressing the tractor's drivetrain, extended experimental studies were also performed. In this sense, a measurement installation was designed and used (some sensors were adapted and others realized). The instrumentation permitted to record on magnetic tape the speeds of engine and gearbox output-shaft, the torque of gearbox output-shaft and the torques of right- and left driving-wheels half-shafts.

The testing program included short time records (starts from rest, gear shifting, engine braking), to obtain the load peaks, and long duration road tests. In this last case, a portable computer was used for real-time data processing in order to obtain the statistical load distributions in typical ground and traffic conditions (plain, hill and mountain roads). Simultaneously, a mechanical counter has given for each gear the number of shifts and the total time of operation. Indirectly resulted also the number of clutch engagements.

Figure 2 presents a set of data recorded during road train pulling away from rest on good, even surface, with successive passing through the first seven gears. The upper plots shows the temporal evolution of the real torques that load the gearbox output shaft and the two semi-axles, while the lower plots indicate the speeds of engine and gearbox output shaft. It can be seen clearly the way the vehicle is gearing up in 70 s, the duration of gear shifts or the shocks induced by the clutch and gear engagements.

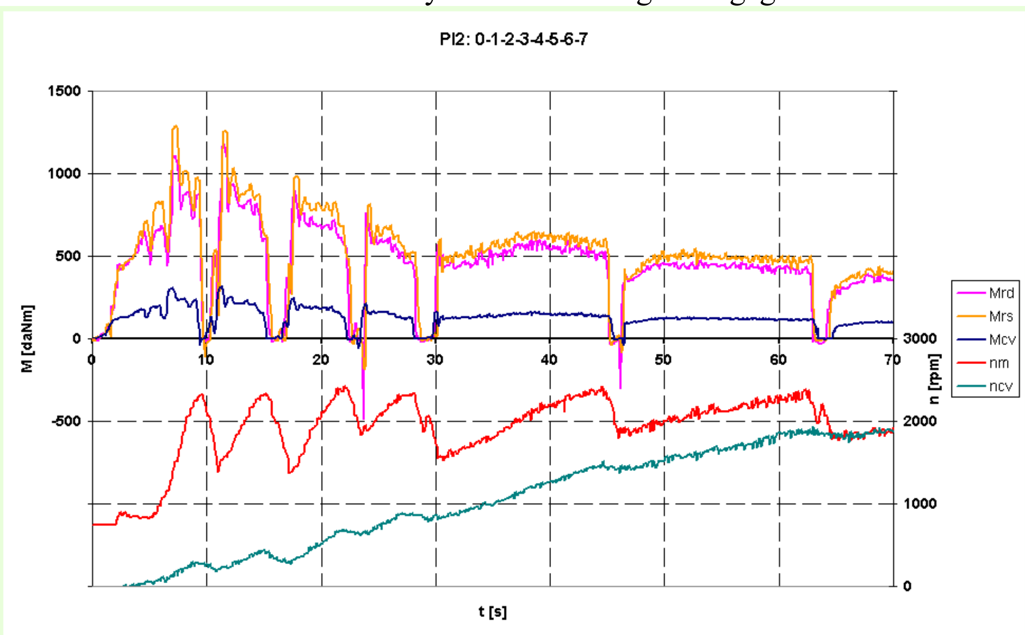

Fig. 2. Experimental record of acceleration process, from standstill to seventh gear upper side – torsional moments of gearbox output shaft (Mcv), left- and right half-shafts (Mrs, Mrd) lower side – engine speed (nm) and gearbox output shaft speed (ncv)

Using the measured maximal values of the torques, some transmission parts were analyzed by finite element method to find the corresponding stresses and deformations [3]. In addition, combining mileage estimated proportions for different road types, statistical loads of drivetrain parts were obtained. These values of torques were used to perform fatigue calculations for the vehicle's parts life span.

Lastly, the experimental data acquired from the road tests were used both for the fine tuning of the computer model and for the verification of the simulation results.

4. Simulation of Clutch Dynamic Behavior

The first kind of simulations presented here shows the clutch behavior during the first 16 s of the acceleration from standstill on an asphalt road – Figures 3, 4 and 5. The graphs in the left side correspond to a level road, while the graphs in the right side were obtained for a road with 4° (7%) ascending gradient.

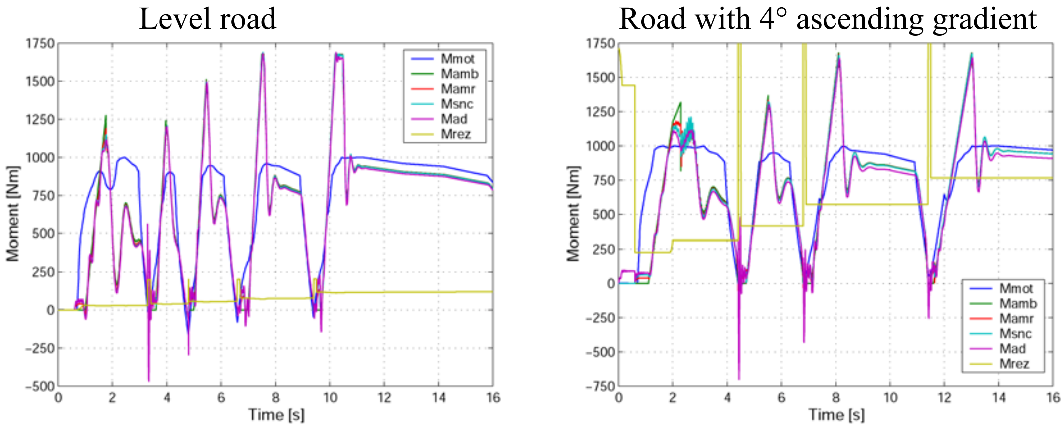

Fig. 3. Simulated torques during an acceleration from rest on asphalt road

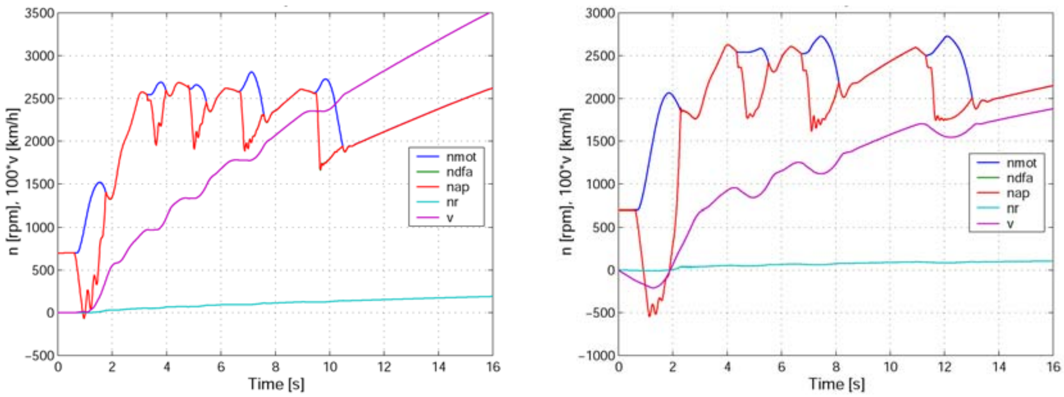

Fig. 4. Simulated speeds during an acceleration from rest on asphalt road

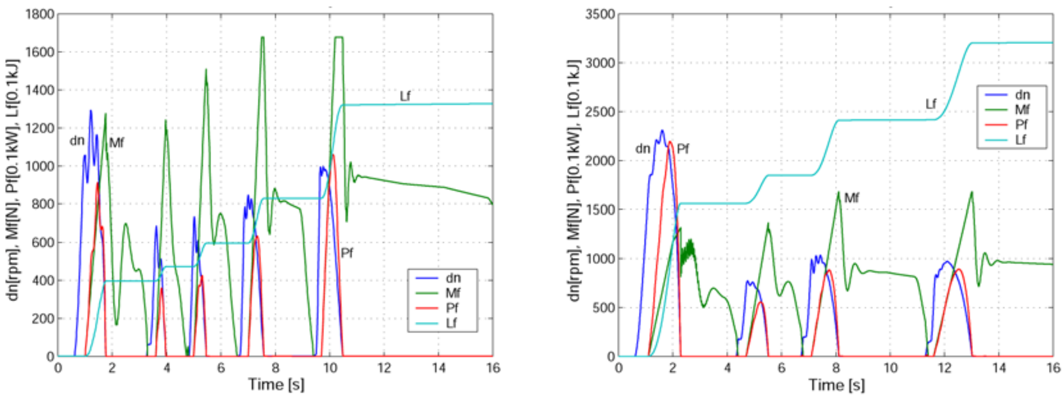

Fig. 5. Simulated parameters influencing clutch wear and heating during an acceleration from rest

At the time t=0 s the engine is idling and the gearbox is on neutral. At the time t=0.1 s the clutch starts disengaging and becomes fully opened at t=0.5. The synchronization process for the shift of the first gear initiates at t=0.6 s. Starting after one tenth of second, the accelerator pedal is integrally pressed in 0.5 s and the engine speed begins to increase. The synchronization finishes at t=1 s and the first gear is on. At the same time, the clutch starts to be engaged abruptly, with a constant rate that determines the total release in 1 s. The clutch slippage finishes at t=1.8 s (with 0.8 s coupling time), when the clutch disk speed equalizes the speed of the engine. At this moment, the engine's kinetic energy ends to be transferred (by the transmission and driveline) to the driving wheels so that the transmitted torque diminishes dramatically, inducing an important relaxation of the driveline. The torsional vibrations of the drivetrain and the fore-aft acceleration of the vehicle are dampened by the tires slip, by the dry-friction damper of the clutch and by the internal friction in the remaining mechanisms of drivetrain. The velocity of the vehicle increases until the engine reaches its rated speed that indicates the moment for a change in the next gear.

The successive passing from gears repeats in the same manner.

As can be seen in Figure 4, the torque transmitted by the clutch increases with the gear and, at the shift of third, it attains already his maximal value (the capable torque). This explains by the smaller quotient of the engine power that consumes to rev the engine itself in high gears, permitting a higher fraction of the power to be transmitted by the clutch.

The putting in movement of the vehicle is much easier on level road than on grade. For example, at the moment t=3 s the vehicle attains 9 km/h versus 7 km/h and at the moment t=14 s reaches 32 km/h (in the fifth gear) versus 17 km/h (in the fourth gear). Also, in the case of 4° ascending gradient, the vehicle starts firstly to roll back (presuming that the brake is released at the moment t=0 s) and only after 2 s starts to move uphill.

Although the pick values of the frictional torque aren't very different, the durations and speeds of the clutch slippage are bigger on grade, as can be seen from Figures 3 and 4, but much better from Figure 5. Here *dn* [rad/s] represents the difference between input and output clutch speeds, M_f [N] – the frictional torque, P_f [W] – the slippage power, L_f [J] – the slippage work.

(1) $$P_f = M_f\, dn = M_f\, |\omega_{mot} - \omega_{dfa}| \qquad L_f = \int_0^t P_f\, dt$$

The maximal slippage power, which transforms in heat and increases the temperature of the clutch parts, attains on level road 91 kW at the first gear engagement and 106 kW on fourth to fifth shifting, and reaches on grade 220 kW (more than half of the engine rated power) at the engagement of the first gear.

More evident differences it observes when analyse the mechanical work lost by friction: 40 kJ versus 160 kJ after the engagement of the first gear and 83 kJ versus 325 kJ after the engagement of the fourth gear.

The increase of temperature in the main parts of the clutch during the vehicle acceleration can be estimated well enough considering that all the energy developed by friction is accumulated by the pressure plates and the flywheel and neglecting the heat dissipation by ventilation:

(2) $$\Delta t_p = \frac{\alpha L_f}{c\, m_p}$$

Here α represents the quotient of the slippage-work absorbed by the part; c – the specific heat of the part material; m_p – the mass of the part.

In the case of the analyzed clutch, with α=0.25 (the heat is equally taken over by the engine flywheel and by three pressure plates), c=500 J/kg for cast iron, m_p=9.3 kg for the lightest plate, the temperature increases during the coupling of the first gear with 2.2°C on level road and with 7.1°C on the slope of 7%. The first value is much smaller than 8…15°C presented as acceptable in the literature. If considers the slippage work that corresponds to the successive coupling of the first four gears, the temperature increases with 8.1°C, respectively 17.4°C. The heat in this last case is enough to boil 1 kg of water having initially 20°C.

5. Influences over the clutch stress

The simulation program proved to be very useful to obtain valuable information, both qualitative and quantitative, about the drivetrain behaviour when different parameters are changed. Figures 6, 7 and 8 show the results obtained in two types of analyses, each figure containing three sets of plots, one for each value of the studied parameter:

- in Figure 6 are represented the graphs of the six torques, all reduced at the engine (engine torque – M_{mot}; frictional torque of clutch – M_{amb}; clutch damper torque – M_{amr}; torque of gearbox synchroniser – M_{snc}; adherence torque of the driving wheels – M_{ad}; global resistance torque – M_{rez}); with the exception of the engine torque, the others almost superposes most of the time;
- in Figure 7 are plotted five speeds (engine rpm – n_{mot}; clutch disk rpm – n_{dfa}; gearbox input shaft rpm – n_{ap}; driving wheels rpm – n_r; vehicle velocity – v);
- in Figure 8 are showed the same quantities as in Figure 5 (speed difference – dn; frictional torque of clutch – M_f; slippage power – P_f; slippage work – L_f).

A first type of simulations was performed to put in evidence the influence of the road conditions over the clutch stress. Three different grades were considered: -4° (downhill); 0° (level); 4° (uphill). The results of simulations are presented in the left side of the Figures 6, 7 and 8.

The clutch coupling rate was adopted constant and identical on the three cases – more precisely, the clutch pedal is completely released in 2 s, starting with the moment t=1.1 s. As can be seen from Figure 6, the maximal values of the frictional torque attained on the roads with -4°, 0° and 4° gradients represents respectively 48%, 75% and 82% from the capable torque of the clutch. The slippage durations can be studied from Figures 7 and 8 and are 1.1 s, 1.45 s and 1.65 s.

From Figure 8 results also that the slippage work on the level road is half of the uphill work and is four times bigger that the downhill work, which will induce wears at least in the same proportions.

The second type of simulations presented in the right side of Figures 6, 7, 8 tried to emphasize the influence of the clutch pedal releasing rapidity. The three cases studied correspond to the coupling duration of 1.33 s, 2 s (as considered in the previous simulations) and 3 s, the vehicle starting on a good level road.

The maximal values of the frictional torque attained in these situations are 79%, 75% and 69% from the capable torque of the clutch. The slippage durations are 1.05 s, 1.45 s and 1.95 s. Comparing with the slippage work of the intermediate coupling rate (2 s releasing time), the work of the faster coupling is with one third smaller and the work of the slower coupling is with one fourth bigger.

The conclusion that seems to be deducible from these results is that it is better to quickly engage the clutch to diminish its wear and heating. This is true, but such driving behavior has a reverse: the rapid coupling of clutch determines an important stress increase in all subassemblies of the drivetrain. In addition, a very quick clutch coupling rate makes spinning the driving wheels and generates strong torsional vibrations in the drivetrain. This can be seen in the both sides of the Figure 7 for the most stressing cases.

Influence of road gradient Influence of coupling rate

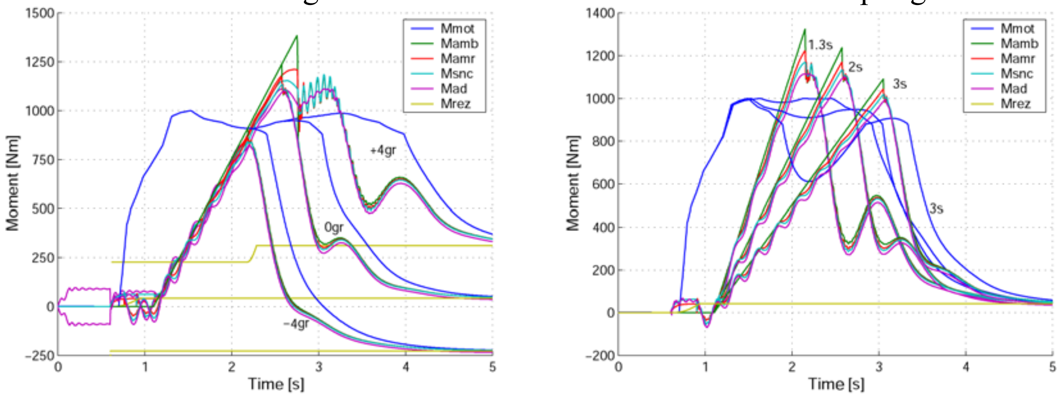

Fig. 6. Comparison of torques stressing the powertrain components during an acceleration from rest

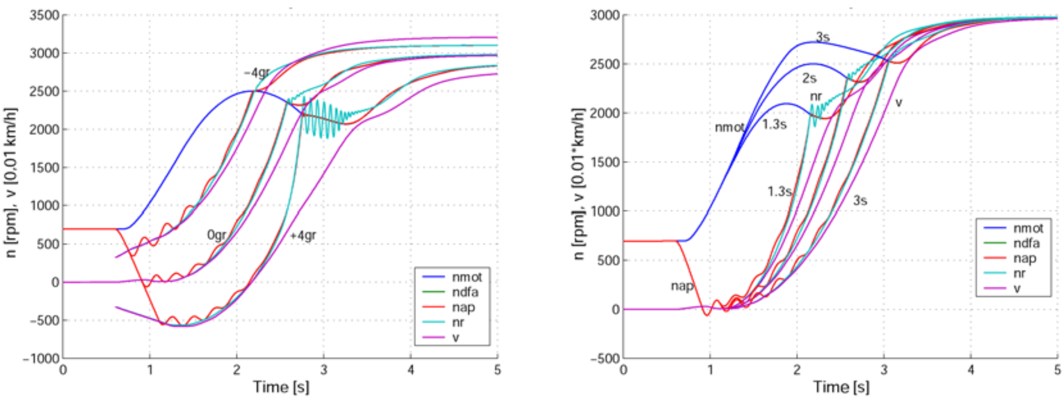

Fig. 7. Comparison of powertrain components' speeds during an acceleration from rest on asphalt road

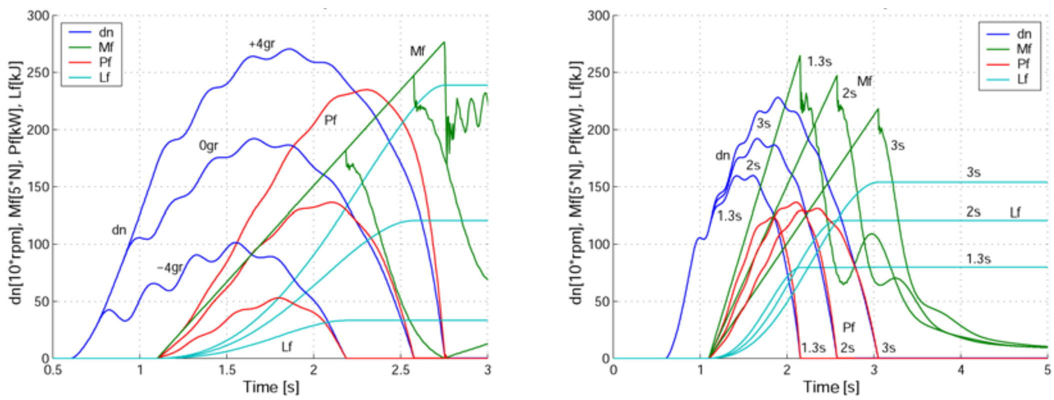

Fig. 8. Comparison of the parameters influencing clutch wear and heating during an acceleration from rest on asphalt road

Another noticeable aspect resulting from all the simulations is the apparition of the self-induced vibrations in the drivetrain as a consequence of the dry friction between the clutch's driving parts and disks.

Some other calculations were also performed that mainly regard the local pressure distribution on the friction surface and the wearing intensity of the disks. These permitted to conclude that the relative small contact pressure on the disk surfaces (approx. 0.1 MPa) and specific frictional work (approx. 0.24 J/mm^2) will allow estimating a life span of 100000 normal couplings.

6. Conclusions

The paper presented few results regarding the clutch behavior, extracted from a more complex study of a heavy-vehicle. Combining theoretical and experimental researches, the authors were able to determine the temporal evolution of the main quantities that influence the mechanical and thermal stress of the clutch. For that, road tests were made and an original computer model was conceived, tuned and used. The simulation performed permitted to analyze the influences of many constructive, road or operational parameters that can affect the clutch loads. The knowledge accumulated is useful for the better understanding of the operation of the motor-vehicle subassemblies and for the improvement of the outlining principles in the design activities.

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STUDIU PRIVIND SOLICITAREA AMBREIAJULUI UNUI VEHICUL GREU

(Rezumat)

Ambreiajul unui vehicul greu este supus unor foarte importante încărcări mecanice şi termice. Pentru o evaluare exactă a condiţiilor de funcţionare ale ambreiajului a fost conceput şi folosit un model Simulink al sistemului de propulsie. În acelaşi scop au fost efectuate de asemenea măsurători experimentale pe un autotren cu semiremorcă de 38 de tone. Înregistrările obţinute în condiţii de deplasare normale şi grele au fost folosite atât pentru calibrarea modelului cât şi pentru compararea cu rezultatele de la calculator. Alte simulări au fost realizate pentru a afla influenţele pe care le au parametrii constructivi şi funcţionali asupra momentului de torsiune şi energiei calorice care afectează ambreiajul. Astfel, au fost studiate influenţele treptei de viteze utilizate, pantei drumului sau vitezei de cuplare şi au fost făcute unele evaluări asupra presiunii de contact, lucrului mecanic de frecare, uzurii şi încălzirii ambreiajului.