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Testing Items During Gearbox Developement

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Abstract: Many elements are mentioned which are necessary to get a gearbox which fulfil the requirements of a European car manufacturer to get build up in their vehicles. The problems and solutions for different shifting items and misused tests will be illustrated and explained. Overviews over the problems which are occurring if a new transmission will be developed are described. Vehicle tests and their results are mentioned.

Keywords*: shifting behaviour, gearbox development, down shift force, cross shift, vehicle tests*

I. INTRODUCTION

The shifting behaviour plays a big role in a gearbox development process for a vehicle. During shifting many different elements are included. Improvements can be done on all the elements that are mentioned in this paper. To get a good transmission in a vehicle, investigations on the selecting behaviour, the synchronization of each gear, the shifting and cross shift behaviour are necessary. All drivers want to have good static and dynamic performance of a gearbox, therefore all possible build ups for a gearbox have to be analyzed and checked. Only vehicles with a good working gearbox will be bought. The requirements of the European car manufacturer to a new gearbox are increased through the last years. All investigations are done in cooperation with a vehicle manufacturer. The final transmission will be build up in a new type of vehicle of this car manufacturer. A lot of tests were done to fulfill the requirements and a lot of problems have to be solved.
The shift effort (operating force) is the result of a summation of the static shift effort, due to the ramp profile and the dynamic shift effort, due to the synchronization action and is also the load that the driver has to apply at the shift lever (Sharma, 2012).

Shift force Fa:

$$F_a = \frac{2 * sin\alpha * J * \Delta\omega}{n_c * \mu * d_m * T_F}$$ (Back, 2013)

The tooth stiffness is composed of the stiffness of all the engaged tooth pairs together (tooth pair stiffness), depending on the profile overlap are gearing up for a different number of teeth in contact. For a time-constant torque load is therefore a constant force toothing, so to change the load and the deformation of the wheel pairs located in each case in engagement. The variation of tooth stiffness depends significantly on the choice of coverings, while the other tooth sizes have a minor impact (Knauer, 1988). This can also explain why usually considerably higher excitation level of spur gears in comparison to helical gears occur (Heider, 2012).
Roof angles of sliding sleeve and synchronizer ring teeth, pointing angle, chamfer angle determine the way the sleeve and synchronizer ring are in contact. This affects the components of the applied forces. With a lower angle the braking capacity increases when the circumferential force increases. On the contrary, if the axial component will be decreased then this could lead to a gear clash due to an inefficient braking. Besides, the lower the angle the higher the shift effort will be because of a reduction of the axial force (Vettorazzo, et al., 2006).

Gearbox testing

In testing the gear to calculated fatigue is practically verified. Firstly, the gears are installed in the vehicle for continuous operation tested in a driving test. Here, the gears are used with different application profiles and routes, which largely correspond to the expected customer profile. Further driving tests are performed under difficult conditions, which should lead to a reduction of testing time. Parameters that can be varied herein are inter alia the route profile, the road profile and the characteristic of driver or the vehicle weight. In order to reduce the high time and expense of vehicle testing, transmission are investigated on test benches. The advantages of bench testing across from driving tests are the possibility to perform endurance tests among other things like independent of weather conditions, union regulated working hours or statutory provisions such as maximum driving hours. Further, experiments carried out with the maximum expected loads, such as abuse testing. As the basis of bench tests can serve measured or synthetically generated load spectra. Measurements of customer profiles or internal company standardized driving test routes are performed to determine the measured load spectra. Synthetic load spectra can be generated by driving simulation or statistical methods (Weidler, 2005).

Pitting

For the ultimate limit state according to DIN 3990 dimples the apparent surface pressure on the pinion wheel and the permissible surface pressure is compared and used to calculate the safety regarding dimples. The equations show an excerpt from the calculation according to DIN 3990 equations

$$S_H = \frac{\sigma_{HP}}{\sigma_H}$$

$$\sigma_H = Z_{B/D} \cdot \sigma_{H0}\sqrt{\cdot K_A \cdot K_V \cdot K_{H\beta} \cdot K_{H\alpha}}$$

$$\sigma_{H0} = Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t}{d_1 \cdot b} \cdot \frac{u+1}{u}}$$

$$\sigma_{HP} = \sigma_{Hlim} \cdot Z_{NT} \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_W \cdot Z_X$$

S_H	$[-]$	dimple safety	Z_β	$[-]$	chamfer factor
σ_{HP}	$[\frac{N}{mm^2}]$	allowed edge compression	F_t	$[-]$	nominal tangential force
σ_H	$[\frac{N}{mm^2}]$	working edge compression	d_1	$[mm]$	pitch diameter bevel
$Z_{B/D}$	$[-]$	bevel-/ gear contact ratio	b	$[mm]$	tooth width
σ_{H0}	$[\frac{N}{mm^2}]$	nominal edge compression	u	$[-]$	gear ratio = z_2/z_1
K_A	$[-]$	application factor	σ_{Hlim}	$[\frac{N}{mm^2}]$	endurance limit edge
K_V	$[-]$	dynamic factor	Z_{NT}	$[-]$	durability factor
$K_{H\beta}$	$[-]$	longitudinal load distribution factor for edge compression	Z_L	$[-]$	lubricant factor
$K_{H\alpha}$	$[-]$	front factor for edge compression	Z_V	$[-]$	velocity factor
Z_H	$[-]$	spread factor	Z_R	$[-]$	surface quality factor
Z_E	$[-]$	flexibility factor	Z_W	$[-]$	material pairing factor
Z_ε	$[-]$	contact ratio factor	Z_X	$[-]$	size factor

(Mulzer, 2009)

Figure 1: Pitting

II. FACTORS FOR SHIFT FORCE

The graph below shows elements on which improvements for a shifting behaviour can be done. All the elements have tolerances and can be produced in different ways and shapes.

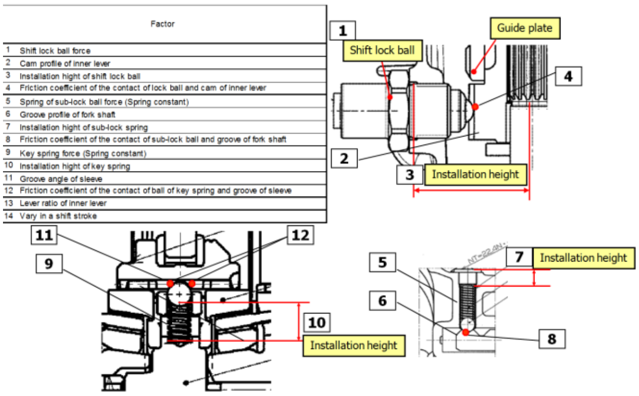

Figure 2: Factor and tolerances for shift and select

The peak position of shift force can be moved by lever stiffness. The graph below shows two examples:

· On the shift tower: The peak position is at 18% of total stroke.

· On the shift knob: The peak position moved to 32% position of total stroke.

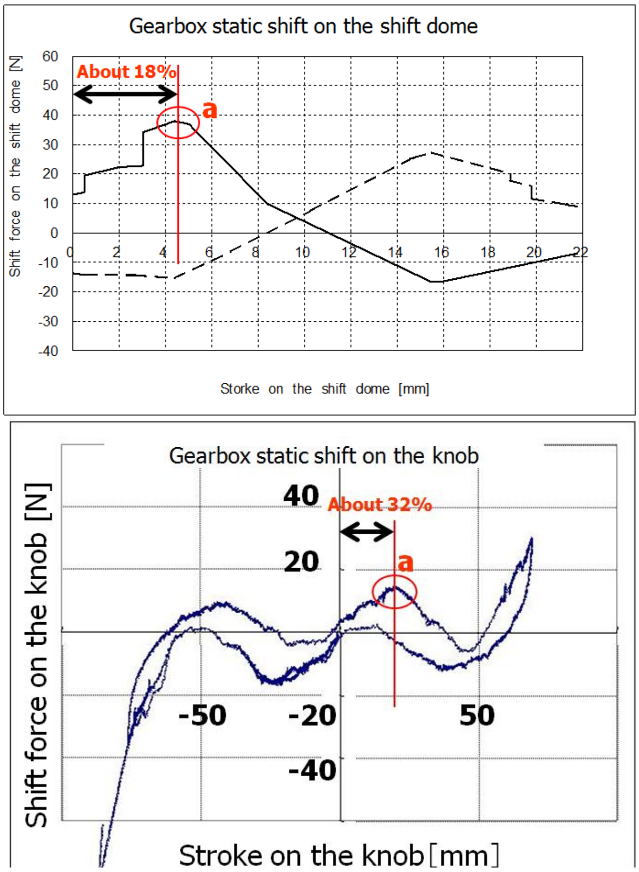

Figure 3: Shift force

III. SHIFTING CHARACTERISTIC

Just adjusting spring load is able to create different upper and lower limits because the shift force can then change. So therefore change the shift lock ball to an adjustable spring load, then the wanted upper and lower limit shift conditions can be adjusted by the spring load. With that method many different shifting forces and behaviours are possible.

Maximum force-stroke characteristic during shifting

The graph below shows the shifting behaviour at maximum condition of each gear. There is always an in and out direction shown in the graph below, that makes it easy to follow the engage and disengage process of each gear. The maximum forces are between 55 N and 70 N.

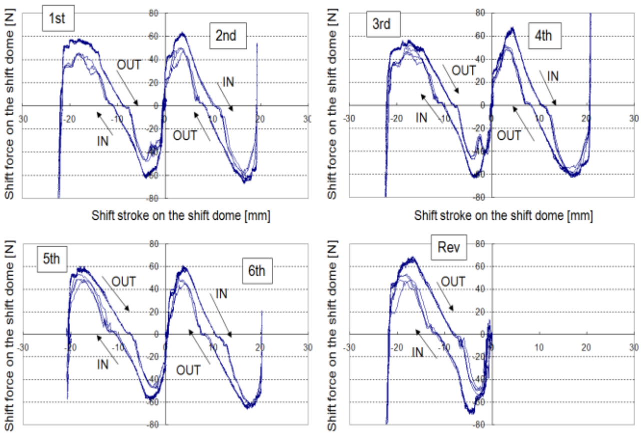

Figure 4: Maximum shifting condition

Minimum force-stroke characteristic during shifting

The graph below shows the shifting conditions under minimum forces for each gear. An in and out direction is always shown in the graph below for the engage and disengage process of the gears. The minimum forces are between 25 N and 45 N.

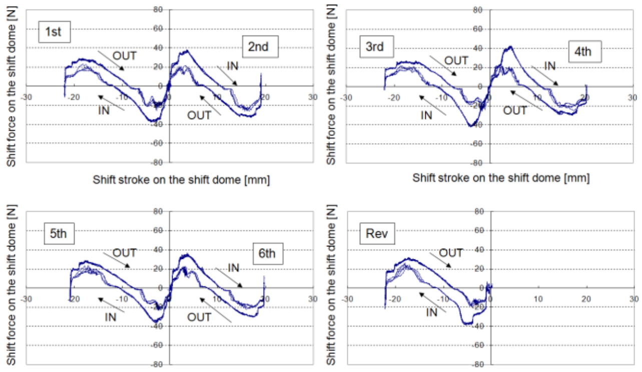

Figure 5: Minimum shifting condition

Maximum force-stroke characteristic during selecting

The graph below shows the selecting behaviour at maximum condition. The force to come to the reverse gear is over 200N. Therefore it is then not so easy to come unwanted in the reverse gear.

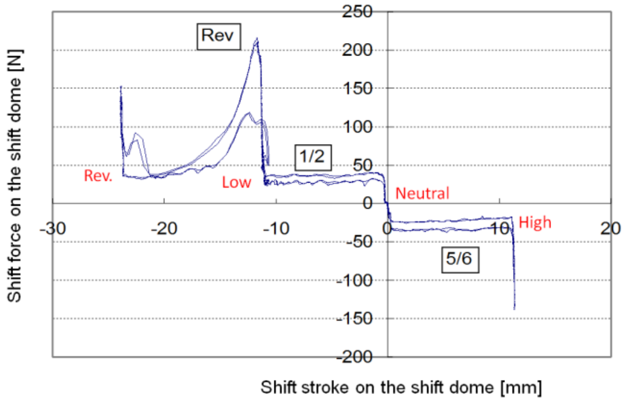

Figure 6: Maximum selecting condition

Minimum force-stroke characteristic during selecting

The graph below shows the select feeling in a vehicle under minimum force conditions. The force to come to the reverse gear is over 180N. In this graph the other selecting forces are lower than above, so therefore the same safety factor exists.

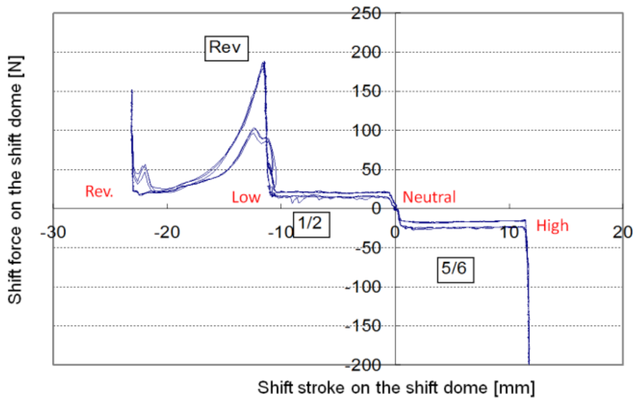

Figure 7: Minimum selecting condition

IV. MISUSED VEHICLE TEST

This chapter starts with pitting on gearbox teeth and ends with chattering in a gearbox.

Gear pitting

The left side root of tooth has marks of strong tooth contact and pitting has occurred only in the left side root of tooth. A possible way to overcome pitting is to improve the teeth surface strength. Therefore the teeth surface design will change and an additional production step like shot peening is necessary. Also the pressure angle and crowning are changed and left side root of teeth in high torque is reduced. The graph below shows on the left side teeth with pitting and on the right side the solution for the problem.

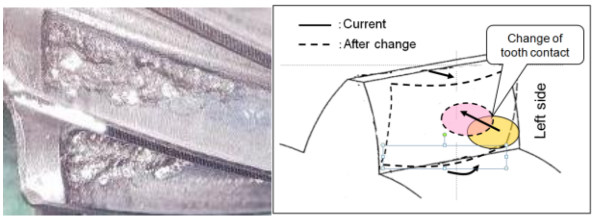

Figure 8: Pitting

Concerning the teeth surface strength, the teeth contact load of left side root of teeth is reduced by 30%. The graph below shows the reduction by teeth surface design change. Also the acoustic has to be checked after design changes like these.

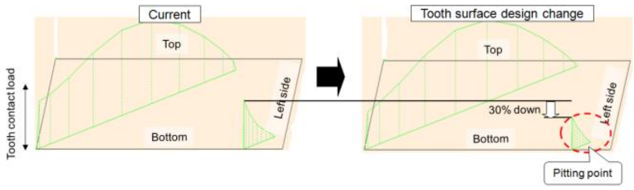

Figure 9: Teeth surface analysis

After these changes the NVH performance was good, so the changes can be implemented in the gearboxes. The pitting problem was solved. This is shown on the left side of the graph below. The right side of the graph shows the acoustic performances after the changes and the acceptance line. There is a big distance between the NVH performance of the gearbox and the acceptance line for the vehicle.

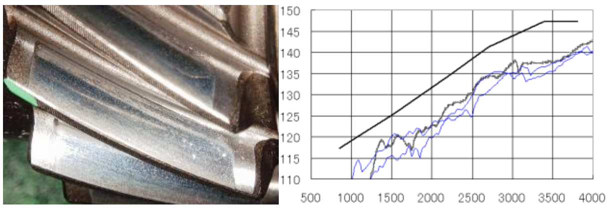

Figure 10: Pitting solution

Chattering noise in a vehicle

The graph below shows the vibration in a vehicle, when the engine revolution is changing. The biggest vibrations are between 1800 and 2200rpm. The vibrations were measured on the shifting knob. It can be felt sometimes in all gears and sometimes only in a few gears.

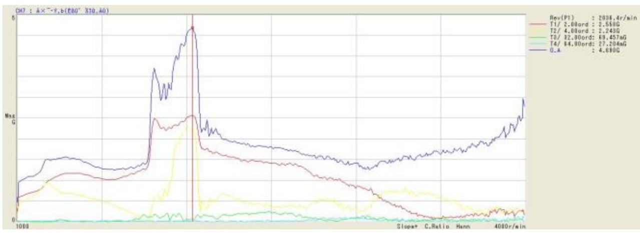

Figure 11: Chattering noise analysis

The graph below shows that the selecting force on the shifting tower has an influence to the vibration on the selecting shaft. It is shown that reducing the force will reduce the vibration.

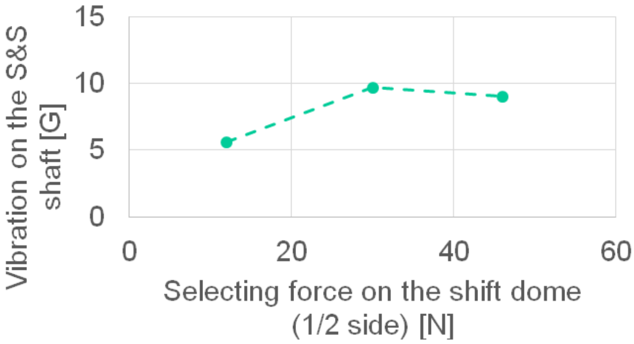

Figure 12: Selecting force

The way to eliminate the chattering noise issue is to optimize the intersection between the guide plate pin and the guide plate. This is shown in the graph below, where a permanent contact takes place.

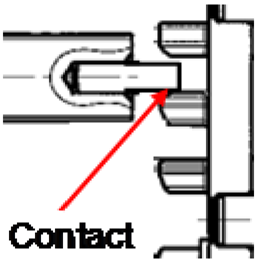

Figure 13: Intersection of guide plate and guide plate pin

V. CONCLUSION

This paper showed many different items for which the best solution has to be found to get a perfect gearbox in a vehicle. Also the result of vehicle tests were shown and how to solve the problems. Different elements come often from different suppliers so also a good cooperation between the companies is necessary. All the items were solved and the vehicles can be built up with these transmissions.

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