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FATIGUE LIFE PROGNOSIS FOR TRANSMISSIONS BASED ON CRITICAL COMPONENT SPECTRUM

Körner, Tillmann[1]
Depping, Herbert[1]
Häckh, Jakob[2]
Willmerding, Günter[2]*

[1]Voith Turbo GmbH, Heidenheim, Germany
[2]Steinbeis Transfer Centre New Technologies in Traffic Engineering, Ulm, Germany

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INTRODUCTION

A number of demands are made on modern commercial vehicle transmissions. Low fuel consumption, small size and light weight have to be considered more and more in the development phase, along with a defined fatigue life and the cheapest possible manufacturing costs.
The most important characteristics of a transmission are fatigue life and reliability. If it is over-rated it will be too heavy, too large and too expensive. If it is under-rated the reclamation costs will be high and the reputation endangered.

Because fatigue life is influenced by so many aspects, it is impossible to carry out tests under all the peripheral conditions one can think of. The method “critical component spectrum“ described here uses all the fatigue life factors and all the conditions of use to calculate the component spectrum which leads to the greatest damage and the fatigue life which results from this. Based on this information a specific constructive adjustment and a component test can be carried out. This means that these demands can already be taken into account during the development stage of a transmission. This method can be used when developing new products as well as when adjusting transmissions to meet changed demands.

Voith Turbo, an important manufacturer of transmissions for public busses has many years experience in the design, adjusting and use of transmissions. For the past 15 years the Steinbeis Transfer Centre has been developing simulation programs to simulate drive lines and calculate fatigue life. These programs are in use all over the world. The process of defining the critical component spectrum is an important methodical help and is of assistance when analysing and assessing fatigue life analyses.

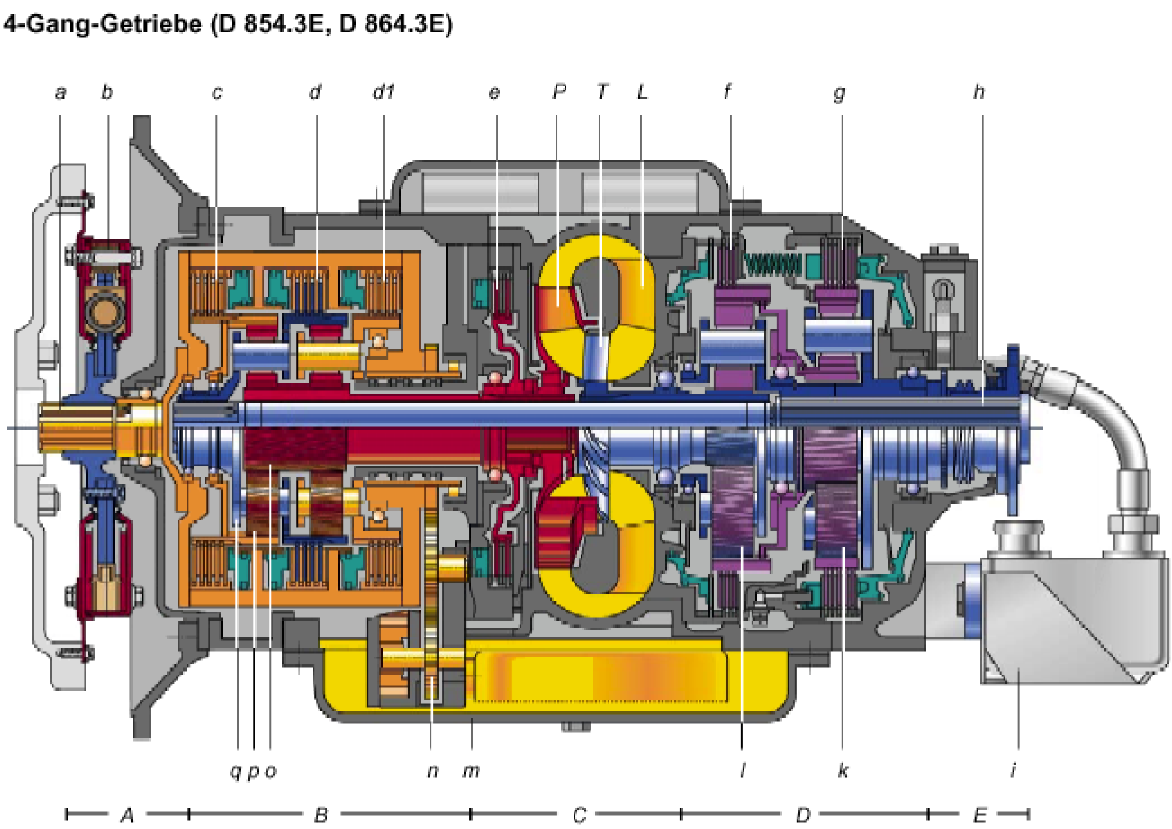
FATIGUE PROCESSES IN A TRANSMISSION AND CALCULATION MODELS

A transmission (fig. 1) for public busses – normally an automatic transmission – consists of, among other things, a hydrodynamic torque converter and planetary gears. These can be shifted under load with the help of multi-plate clutches.

The following machine elements are used here and they have to be dimensioned according to their fatigue:

- shafts	fracture
- gear wheels	pittings or fracture
- bearings	pittings or fracture
- clutches and brakes	wear

Fatigue calculation models are available for dimensioning the machine elements. They can be used to estimate the fatigue life and are described briefly below:



A=Input	B=Differential	C=Torque Converter	D=Planetary gears	E=Output

a=input shaft	d1 fast clutch	h output shaft	m oil sump
b Hydrodamp	d pump brake PB	i heat exchanger	n drive gear pump
c input clutch EK	f turbine brake	k planetary gear	o sun wheel
d lockup clutch	g reverse gear brake	l turbine gear	p planetary gear

Figure 1: Four speed automatic transmission DIWA 864 from Voith

FATIGUE LIFE ESTIMATION OF GEAR WHEELS

To analyse the fatigue life of gear wheels the different causes of damage have to be considered:

- local stresses act on the surface of the flank (Hertz` pressure) as a result of local contact (fig. 3). This can lead to pittings in the flank

- in the base of the tooth the bending moment applies stresses which, combined with the local notch, can lead to fatigue and finally to fracture (fig, 2)

To carry out a fatigue life analysis for a gear wheel, the S-N-curves according to DIN are used (fig. 8). This data is only valid for a simple gearwheel configuration as shown in figure 4, where only two wheels are acting together and only one loading occurs for each revolution. Furthermore this data is only valid for the wheel type used to establish these curves. The data regarding the module, radius, number of teeth and gradient angle acquired from the test wheel must be transferred to the data for a real wheel. When calculating the gear wheel, DIN 3990 [3-6] and FVA-research reports [12-15] were taken into consideration with the aid of the program ZAR1 [7].

In the case of a planetary gear, additional effects must be taken into account. The number of load reversals is different from a two wheel configuration as figures 5 and 6 show. Additionally the situation alters again, if the acting and driven wheels change or if the torque direction changes while the transmission is in operation.

To record the influence of these stress reversals, the alternating load Y_A is used. This reduces the fatigue life of the S-N-Curve accordingly [7].

FATIGUE LIFE ESTIMATION OF SHAFTS

The fatigue life of a shaft can be estimated in a number of different ways. The local strain approach or the nominal stress method is possible. Figure 9 shows the procedure using the Nominal Stress Method, where a component S-N-curve is needed for the fatigue life prediction. To predict a component S-N-curve of a notched shaft the theory of “Synthetic Woehler Curves” can be used [2]. If a S-N-Curve is available as the result of an experiment, then this should be used. The loading – typically the acting torque – is used to calculate the damage according to Miner`s law.

FATIGUE LIFE ESTIMATION OF BEARINGS

The fatigue life of bearings can be calculated according to data provided by the manufacturer of the bearings. The acting load in the bearing is needed and the number of revolutions at an acting load is used to estimate the fatigue life (figure 10).

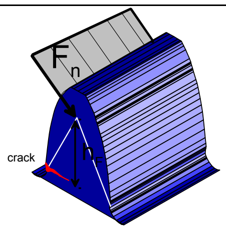


Figure 2 : Bending of the tooth and resulting notch stresses

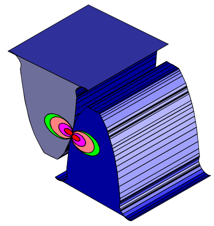


Figure 3 : Local pressing in the flank and resulting local pressure in the surface

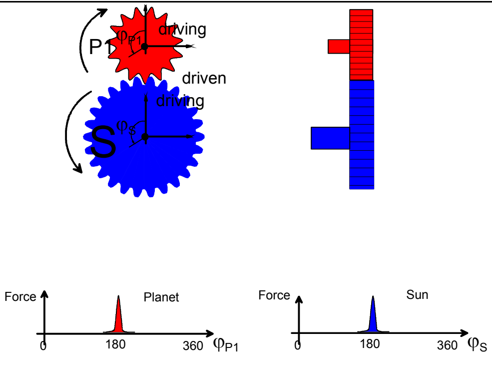


Figure 4: Two gear wheels in a simple configuration and resulting forces in a tooth

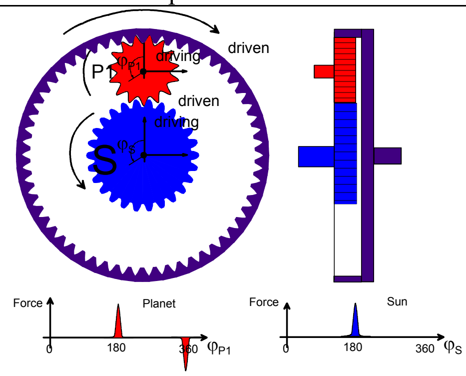


Figure 5: Simple planetary gear and forces in the planet gear and sun gear at one revolution

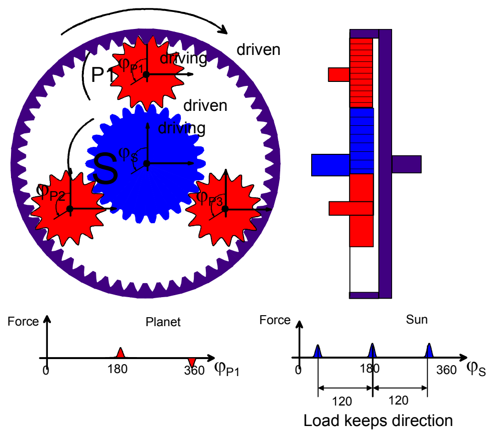


Figure 6: Planetary gear with 3 planet gear wheels and forces in planet and sun at one revolution

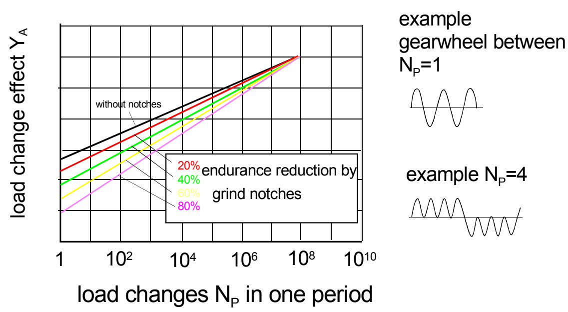


Figure 7: Load change effect factor Factor Y_A according to Niemann

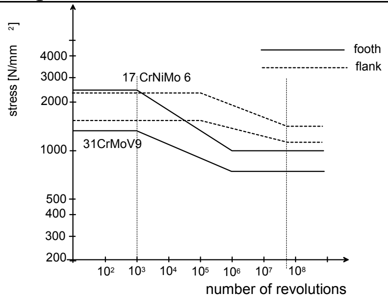


Figure 8: S-N-Curve similar to DIN 3990 for fatigue life against base bending and pittings

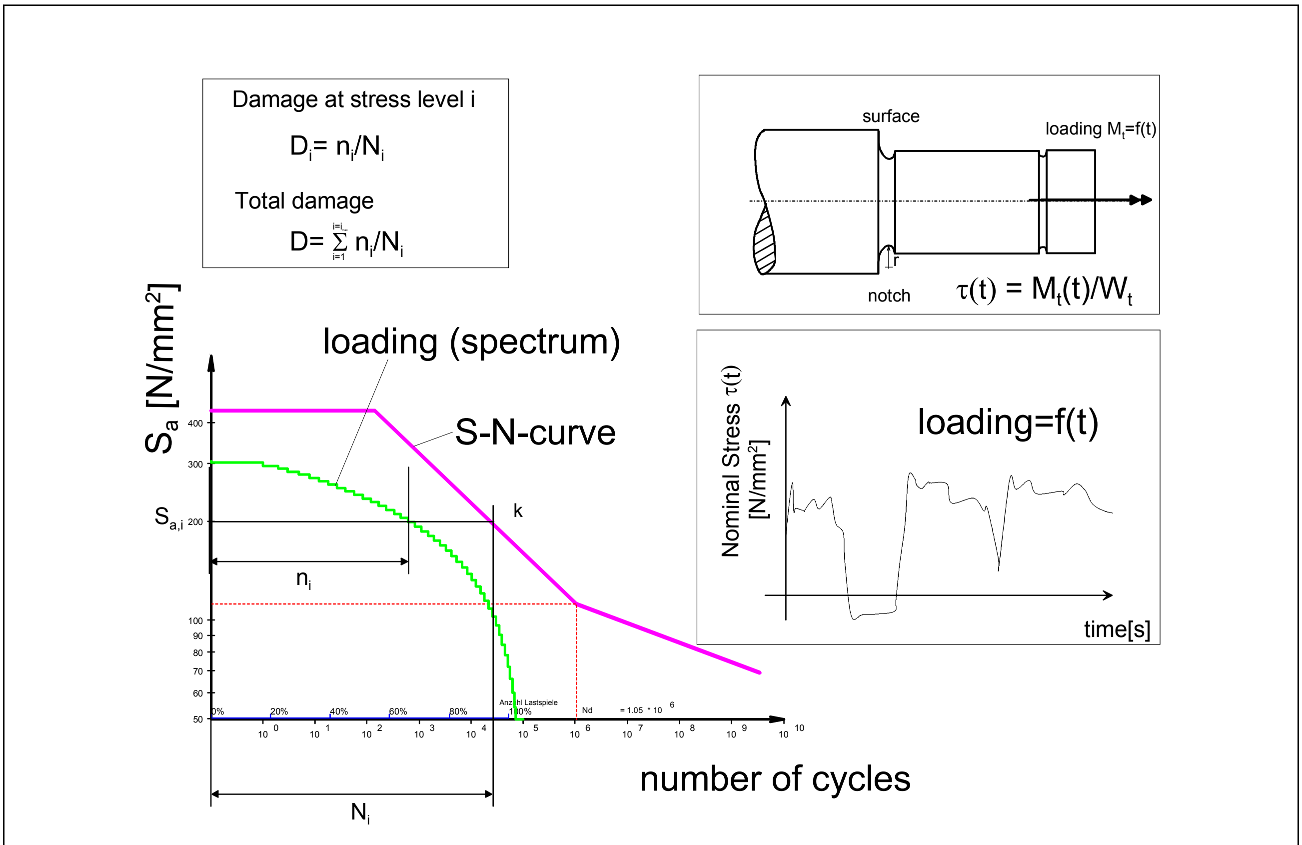


Figure 9: Schematic procedure of fatigue life prediction according to the nominal stress method

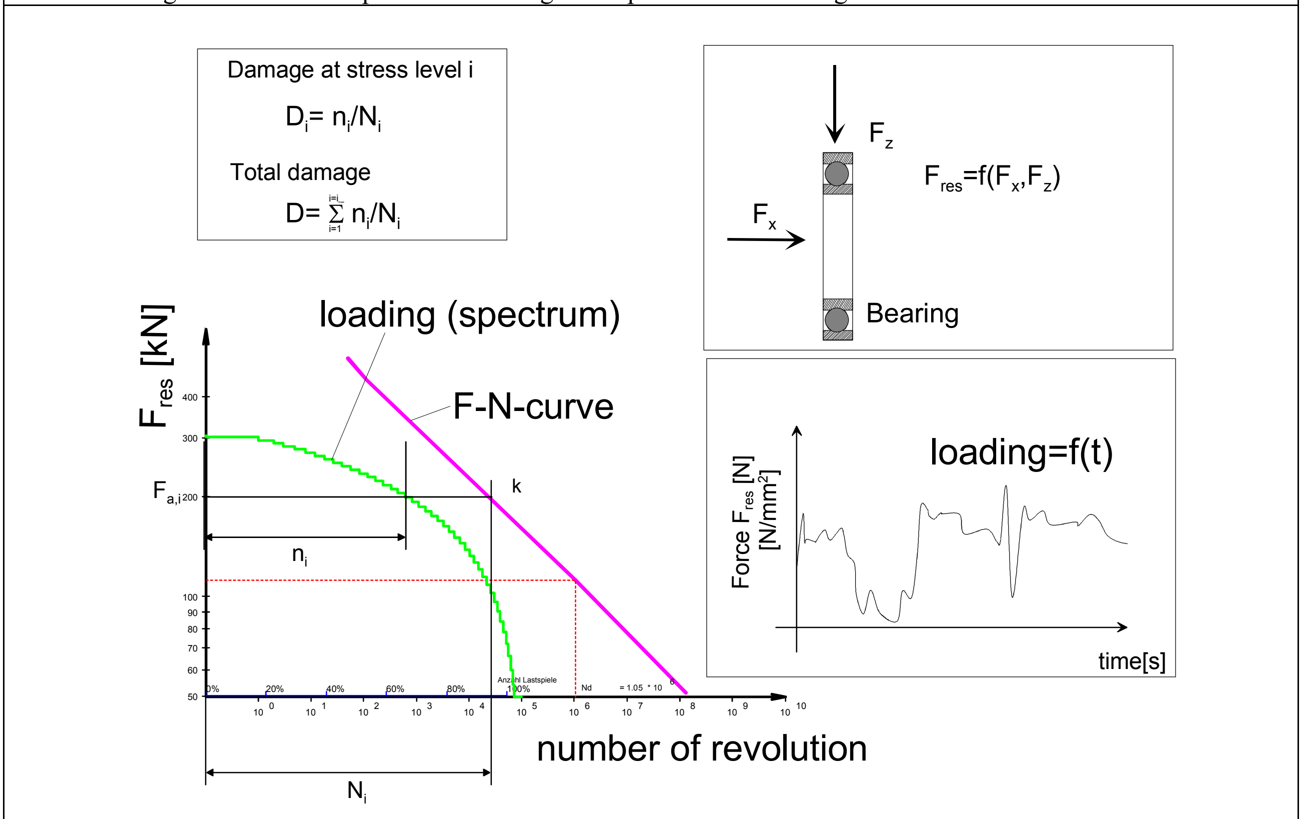


Figure 10: Data for Fatigue Life prediction of bearings

FATIGUE LIFE ESTIMATION OF CLUTCHES AND BRAKES

Clutches and brakes in an automatic transmission are highly loaded, because the power is transmitted while the gear is shifted. For fatigue life calculation the conditions of shifting such as moment of inertia, shifting-time, torque, speed before and after shift, as well as the temperature of the plates are relevant. The heat energy resulting from rubbing has been shown to be a characteristic value which can be calculated for each gear change. figure 12 shows the sudden change of important variables such as torque and speed.

CALCULATING THE LOADING OF THE POWER TRAIN

In order to make a useful fatigue life prediction, an exact knowledge of the component loadings arising is required. Establishing the torque, number of revolutions and forces in a transmission by experimenting on individual components is very tricky and only possible up to a certain extent.

Instead, we can use a simulation program for the power train (figure 11). We have been developing this program for more than 15 years and have gained a lot of experience in comparing measured results and calculating. The components of the power train are modelled in detail and transformed to a multi-body system. The model detail can vary and be adapted as required. For a fatigue life prognosis, the engine excitation, the vibrations resulting from the elasticity and inertia and the dynamics of the gear changes must be taken into consideration.

The calculation of the power-train can be carried out using a measured topography and speed cycles. The way the driver drives is related to the fuel consumption. The model therefore provides various types of driver.

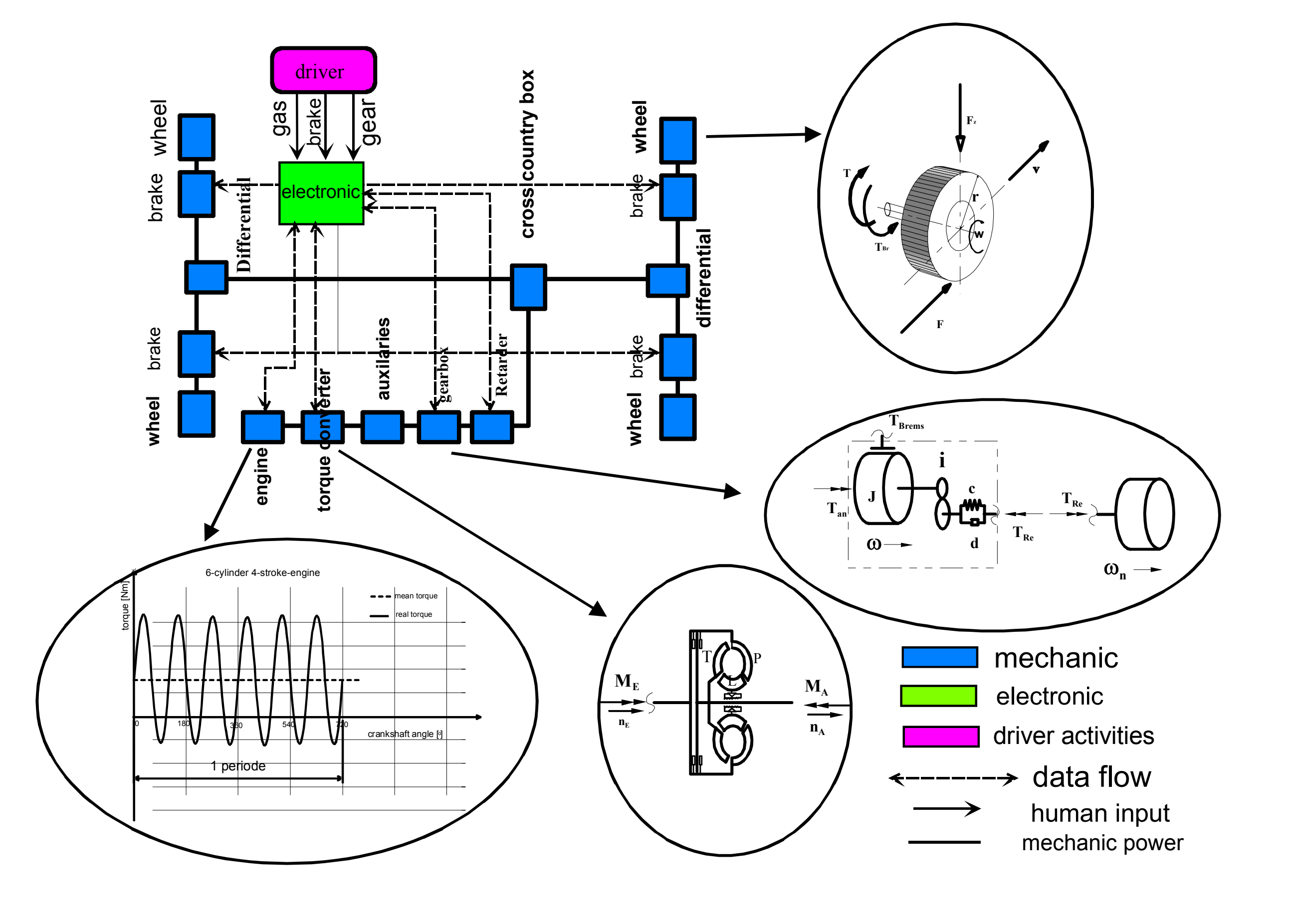


Figure 11: Schematic structure of the power train model

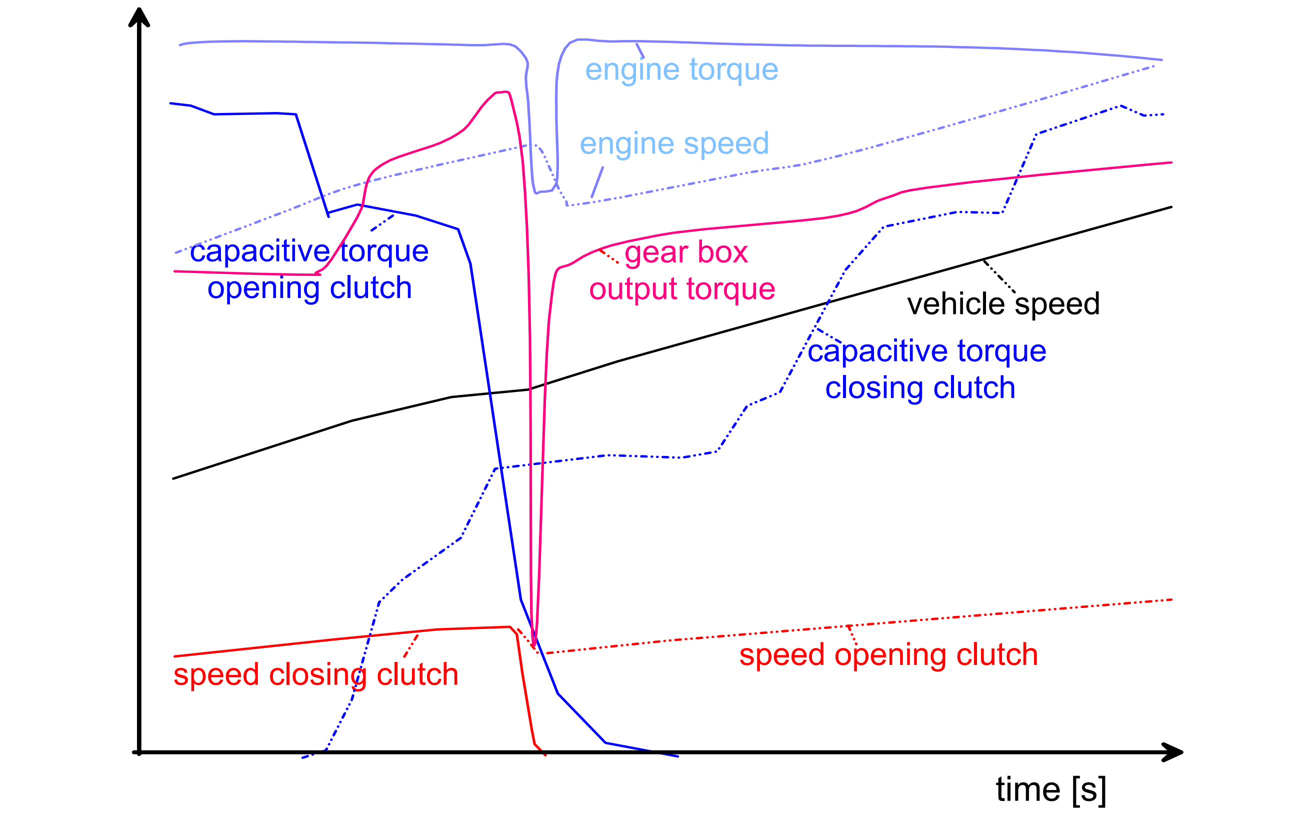


Figure 12: Results of a simulation of a city bus during gear change

OBTAINING REALISTIC DRIVING CYCLES OF BUS TRANSMISSIONS

The results of a simulation can only be as good as the computer input. For many years now, we have been recording the most important characteristics of usage, among other things by measuring speed and topography. This information forms the basis of the simulation calculations (fig. 13). The measurements are taken with an easy to use GPS-based system. In this way, a large empirical data-base has been built up, being an important basis for fatigue life prognoses.

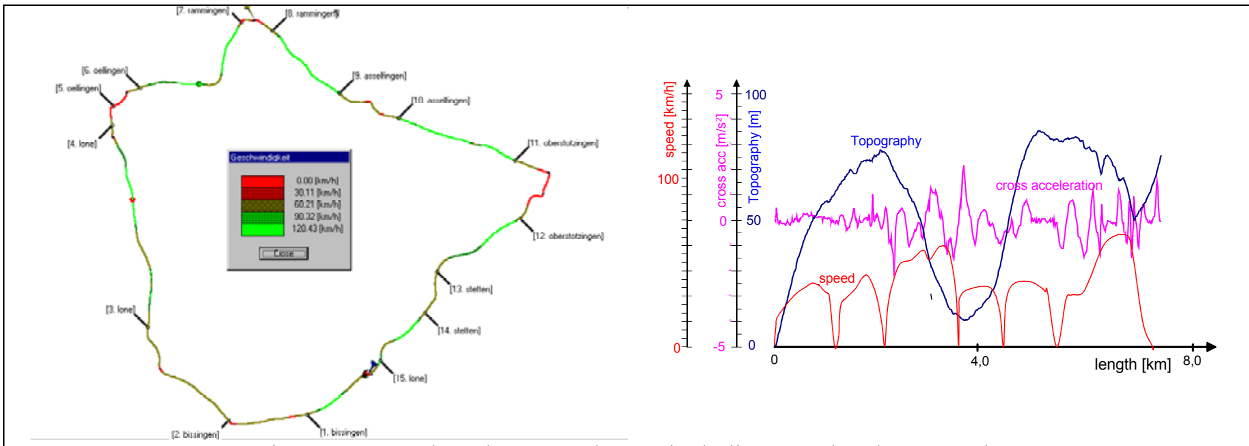


Figure 13: GPS-based measured route including speed and topography

DEFINING AN OBJECTIVES LIST FOR FATIGUE LIFE

For each route that is of interest and for the vehicle to be driven an objective is set i.e. the required minimum length of the route l_r in 10^3km and the total driving time t_r in 10^3h. Information about the type of route (inner city, feeder road, or cross country) and the type of vehicle (rigid bus, articulated bus) later helps to analyse the results. The objectives are set by the transmission manufacturer based on their experience and with the agreement of their customers. The top part of figure 14 shows a much shortened example of a data-input under the title objectives. Such a list would normally consist of 20 to 100 lines.

Firstly, the desired fatigue life is defined in driving hours and total length of route for the various types of use chosen (inner city traffic, cross country, mountainous etc) and types of vehicle (rigid bus, articulated bus, various loadings). This information is generally known from experience or the customer's requirements. These peripheral conditions should cover the whole spectrum of transmission loadings – even the area of slight loading.

If either the distance l_D or the driving duration t_D until damage occurs is known, then it is possible to use this information to define the damage D_l or D_t based on the required distance or the required driving duration as follows:

$$D_l = \frac{l_r}{l_D}$$

$$D_t = \frac{t_r}{t_D}$$

D_l expected damage after required target distance
D_t expected damage after required target duration
l_r required distance
t_r required duration
l_D distance until damage $= 1$ (means failure occurs)
t_D duration until damage $= 1$ (means failure occurs)

When dimensioning, the conditions with the greatest damage are relevant. The aim is then that the damage should not be greater than 1 for any of the given routes.

CALCULATING THE CRITICAL LOAD SPECTRUM AND FATIGUE LIFE FOR EACH ROUTE
For each line (see figure 14 above), a vehicle dynamics simulation is now carried out for each of the given combinations (route, vehicle) with the multi-body system previously described. The results of all the mechanic values of interest in the transmission (torque, rpm, forces)) needed for a later fatigue life calculation are saved. (Heading: Vehicle Dynamics Results in fig. 14) .

Afterwards a fatigue life calculation for every component is carried out based on the results of the vehicle dynamic simulation. An important result is the time or length until damage occurs to any transmission component. The whole drive line is shown in detail and the stress and strain as well as the fatigue life resulting from this can be calculated for every component which is of interest. This can then be compared with the requirements. The aim is now to look for the load spectrum for each component leading to the shortest fatigue life in relation to the required fatigue life. We have called this *the critical load spectrum of a component* in short, *the critical component spectrum*.

The critical component spectrum can be used to test the component. This is to make sure that it is always the least favourable conditions which are tested, although these are based on realistic input. Then there is one important result for each component (see critical component spectra in fig. 14)

Using the fatigue simulation results from chart 14 as an example, the critical component spectrum for the transmission shaft 1 is London, for the gear wheel 1 Paris. In Paris, making the matter more difficult, the intended fatigue life has not been achieved. It is necessary to find out the reason for this and possibly to make alterations to the component.

Calculations for the critical component spectrum are carried out in a batch process. This procedure is constantly being amended to take into consideration any current queries resulting from research and development.

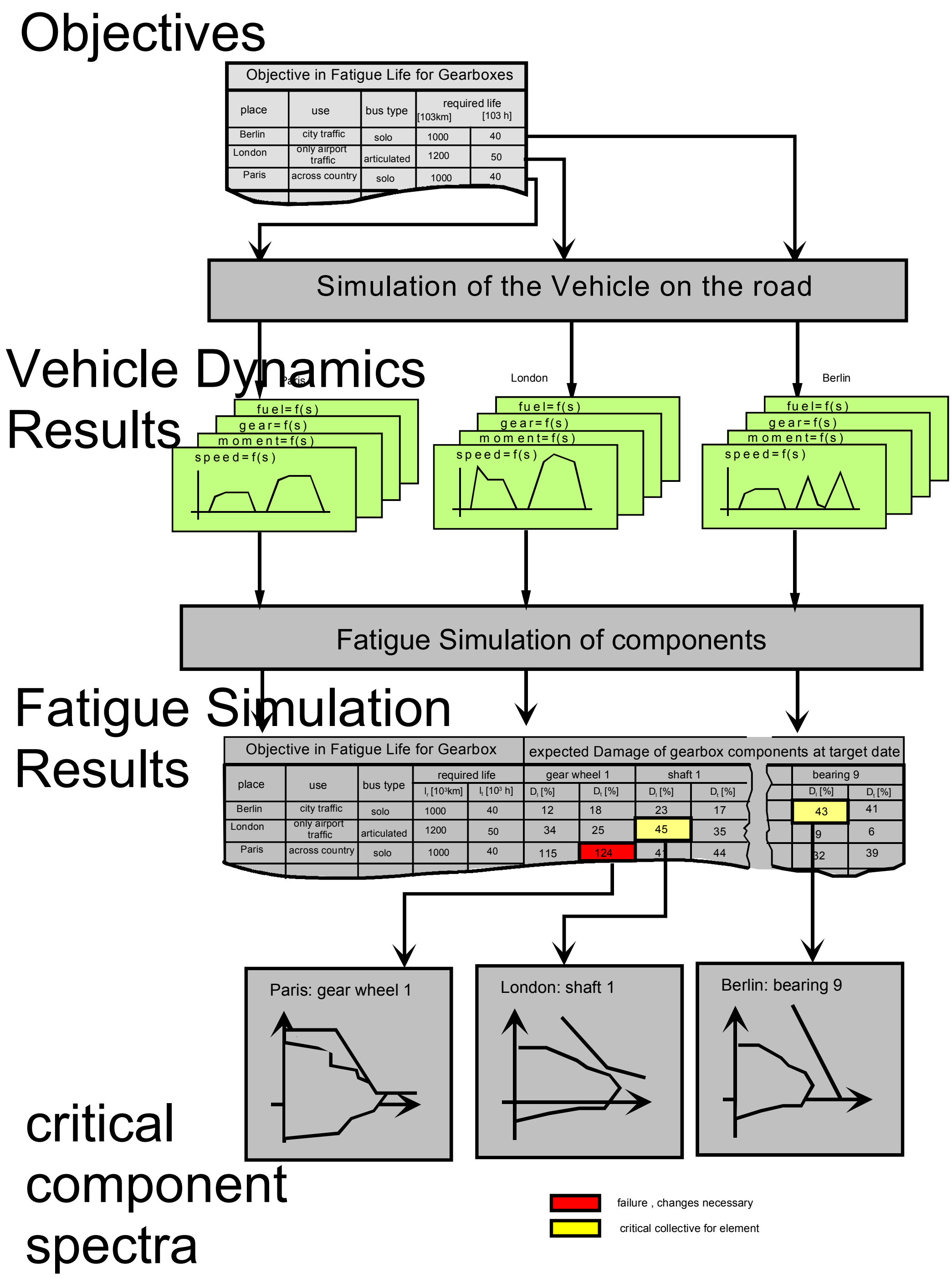


Figure 14: Process to ascertain the critical component spectrum

EXAMPLES OF USE AND ASPECTS DISCOVERED

Figures 15 and 16 show the results of damage to two different gearwheels (flank and base) and the mainbox gear shaft of an articulated bus. It can be seen that the various routes have a significant influence on the damage

Furthermore, it can be seen that the maximum damage to individual components of the transmission does not occur on the same route. Figure 15 shows this. The shaft is most damaged on the HDH_41 route whereas the other components are most damaged on bus route 15_HDH.

Shown in this way, it becomes very clear that the conditions of use for an individual component are reflected in the fatigue life.

The influence of individual influencing variables on the fatigue life is equally easy to recognise and is often surprising. Figure 16 shows the results of the damage for the same vehicle as in figure 15. Only the shifting point of the automatic transmission has been changed. The result is a considerably longer fatigue life for gear wheel 3.

The results of the case above show that a damage sum of 1 or greater occurs to the base of the tooth of gear wheel 3 on the bus route HDH 15. This shows that it is under-rated. The flank of gear wheel 3, on the other hand, is hardly affected. Gear flank 2 is almost at its tolerance level while the base of the tooth only shows a very small amount of damage.

Our aim must be to dimension the component in such a way that, taking into consideration all relevant conditions of use, the intended life expectancy is achieved. Earlier failures should not be allowed to occur. although over-rating should also be avoided as this leads to increased weight and costs.

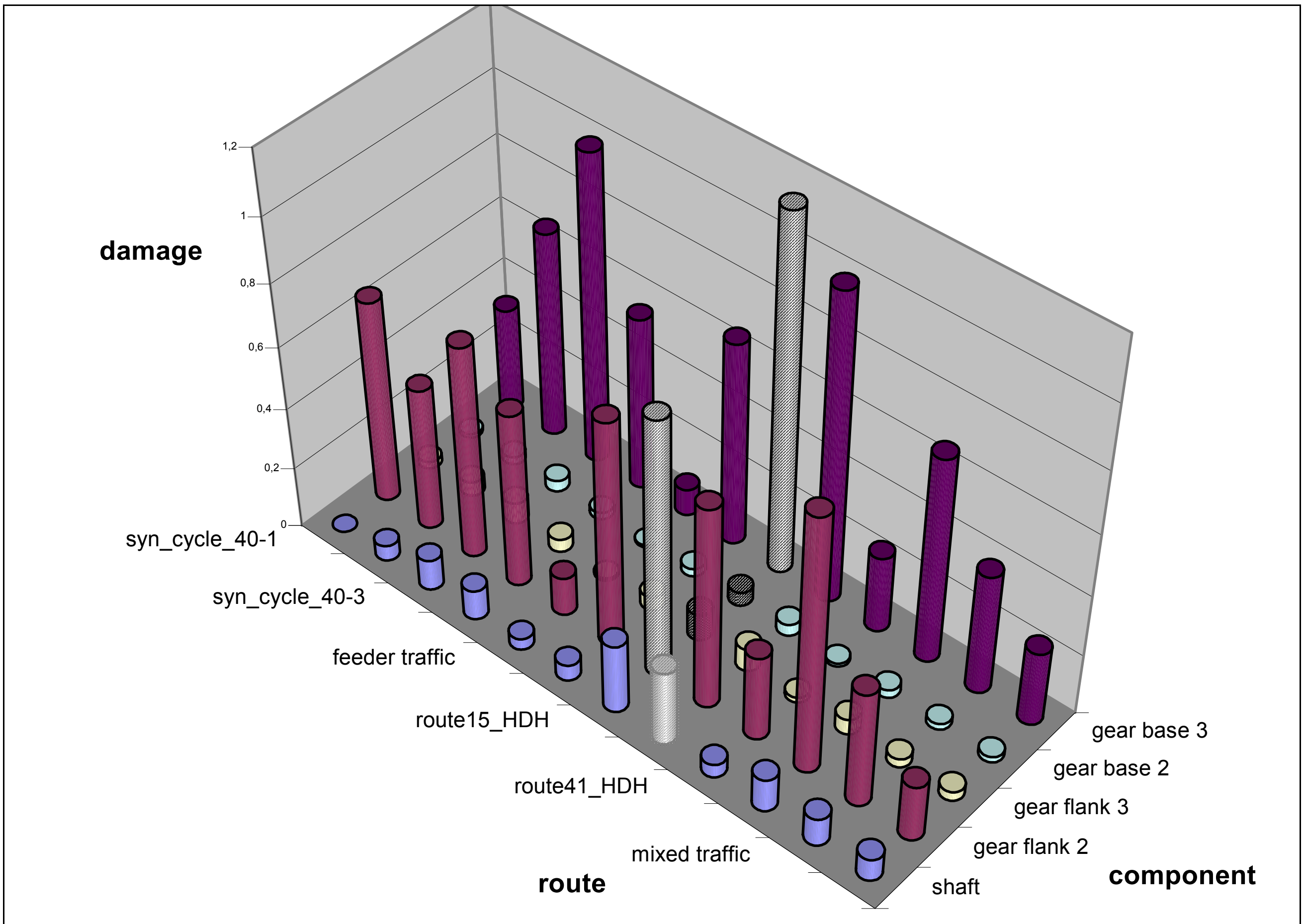


Figure 15: Results of the damage on selected transmission components of an articulated bus on different routes.
Vehicle data: Articulated bus 28 t, High engine torque, typical gear shift speed 1->2 1700 1/min

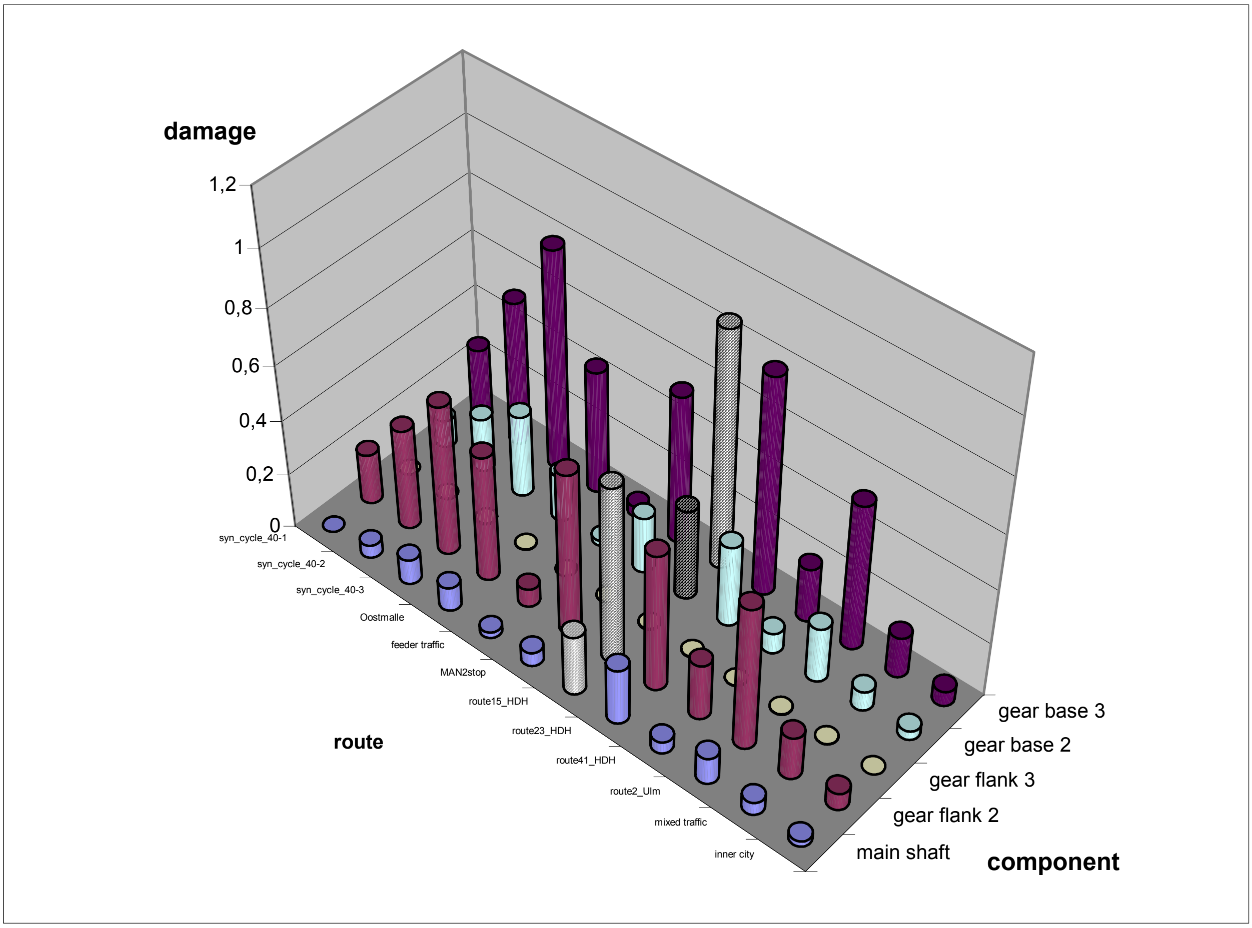


Figure 16: Results of the damage on selected transmission components of an articulated bus on different routes. Vehicle data: Articulated bus 28 t, High engine torque, typical gear shift speed 1->2 1400 1/min

It is very difficult to calculate a fatigue life prognosis exactly and only a combination of experimental and statistical analyses can form a basis of knowledge with which an increasingly certain statement can be made regarding the dimensioning (figure 17.).

The calculated analysis and the ascertaining of the critical component spectrum is one of the most important factors, since the costs are relatively low and it becomes clear which influence variables are important for the fatigue life and which are not.

CONCLUSIONS

When dimensioning and testing components it is helpful to carry out fatigue life calculations to ascertain the critical component spectrum. The statistical phenomena of the failure behaviour in the component, the problems arising because the calculation methods used for ascertaining fatigue life prognoses are poor and the difficulties in estimating a S-N component curve make it very difficult to forecast the quantitative fatigue life. However, after many years of use and comparisons between test and actual results it is possible to improve the calculation tools to such an extent that even a quantitative fatigue life prognosis with relatively exact results can be achieved.

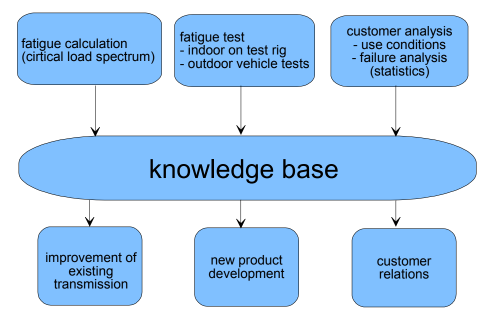


Figure 17: Fatigue life optimisation as part of the development process

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