Hand-In: Module Modelling – Group 9

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Abstract

We used the one shown in figure 1 as the model of the machine (see attachments). In the first section we will analyze the dynamics of our model after the insertion of a drive shafts stiffness, a delay in the generation of torque and wheel slip ('2.1.1 Model more physical phenomena'). In the second section we will analyze the behavior of the system to the variation of the stiffness parameter ('2.1.2 requirement Setting'). In the last section we will improve the model by adding more details ('2.2 Task 2')

1. Model more physical phenomena

We introduced the time constant parameter and used it to calculate the derivative of the torque generated by the Prime Mover. To make this calculation we used the approximation of the incremental ratio.

```
parameter Modelica.Units.SI.
          RotationalSpringConstant C_Drv=23000;

der(T_PmInternal)=
           (T_PmReq2-T_PmInternal)/TimeConstant;
```

As regards the drive shaft, we inserted the piece between the transmission and the wheel assuming that the torque exiting the transmission and entering the wheel is the same. The speed changes and that's why we added w_Drv, which is the rotational speed coming out of the drive shaft.

```
der(T_WhlRot)=C_Drv*(w_Drv-w_WhlRot);
```

2. Requirement Setting and Preparations for Virtual Verification

For this section we have decided to use the overshoot in acceleration as a fraction of the average acceleration as a measurement function. This was done as it seemed easier to alter this value by only modifying the stiffness of the drive shafts when compared with the acceleration time. On the contrary, this parameter is not affected by the change of $C_{DriveShafts}$. This can be seen in figure n ° 1.

This analysis allows us to understand that:

Model Validity: The initial model shows a somewhat rudimental and unrealistic response due to it's simplicity. The instantaneous changes in torque and acceleration and the smooth velocity are not expected in reality. The addition of a time delay to the input shows in the slower response of T_Pm to changes in the input torque as is the expected result of the change. The addition of the model of the driveshaft as a torsional spring shows the expected result of creating an oscillation in the response to the torque request, this is somewhat inconsistent with reality as there should be some dampening effect in the driveshaft in addition to the spring. The model shows the expected increase and decrease of oscillation with the decrease and increase in the driveshaft stiffness respectively. The addition of the tyre slip model shows an expected reduction in the oscillation of the response as when the peaks are delivered to the wheel, the wheel slips instead of translating the full force peak to acceleration. This has an overall damping effect on the system.

Improvements: Adding losses to the system, especially transmission would be an improvement as well as a better driveshaft stiffness model to include dampening. Adding friction and drag. Adding power and torque curves instead of max power and torque. Inertia of parts, especially wheel.

In the table in figure n° 2 it is possible to see the various values of the measurement function for each single analysed value of $C_{DriveShafts}$. In this way we can compare the design parameters. To meet the requirements of the previous point, it is necessary to increase the value of $C_{DriveShafts}$ by about five times. In this way, our measurement function will be less than 0.25. In figure n ° 3 it is possible to see the plots of the results obtained. It is also possible to note that:

Are there other values on the design parameter $cDriveShafts$ which you would propose to try? Why?

Extreme stiffness to see the effect on the response and if there are diminishing returns to extreme stiffness. Intermediate values to see that affect in more detail.

Would you recommend varying other design parameters? Why? Which?

Varying other parameters would cloud the results of an objective measure of driveshaft stiffness, though it would be interesting to see if other parameters reduce the oscillation of the response due to the spring behaviour of the driveshaft, such as the gear ratio, wheel diameter and max power/ torque. If these are a major factor, then a compromise could be made to reach an acceptable response for a lower driveshaft stiffness.

Which operational parameters (defining the maneuver) could be interesting to vary?

Changing the static input to a varied input would show the response of the model to other conditions. Also changing to an engine braking scenario would be interesting to see.

Are the function measures suitable?

The time for acceleration feature measure is unsuitable as the driveshaft stiffness does not significantly change the acceleration over a large timescale. The acceleration as a fraction of mean acceleration is an appropriate measure as it shows the effect on the driver for both comfort and motion sickness.

Are the required values of the requirements suitable?

The first requirement is unreachable at any driveshaft stiffness and as described above the acceleration does not change significantly according to driveshaft stiffness over a large timescale. The second requirement is quite abstract and difficult to understand what 25% deviation from mean acceleration would actually mean or feel like in the vehicle. Physical testing would be required to understand the suitability of this requirement. The frequency of acceleration oscillation can also play a significant role in the comfort of the response.

3. Task 2: Load Transfer + Suspension

For these two parts we have made the following assumptions:

- there is no vertical acceleration of the machine and therefore the suspension springs are not affected by this effect.
- the center of mass, calculated thanks to the data provided, is also the center of rotation.
- the suspensions have a vertical movement only and not a horizontal one in order not to allow a pitch movement of the machine.

Taking into consideration the figure n ° 4, the load transfer was modeled in code '_1D_Task2_1' by adding

```
F_rz*lr=F_zWhl*lf+F_xWhl*h;
```

So, adding a new balance of torque considering the center of mass as the center for the balance.

In the code "_1D_Task2_2" we modelled the suspensions of the vehicle by taking into consideration the figure n ° 44 which best represents the model we want to create. The dynamics of the suspension is defined by:

```
//Suspension
der(F_sr)=C_z*v_zr;
F_dr=D_z*v_zr;
der(F_sf)=C_z*v_zf;
F_df=D_z*v_zf;
J_y*der(w_b)=-(F_sr+F_dr)*(lf+lr)-m*der(v_x)*h-m*g*lf;
v_zf=-w_b*lf;
v_zr=w_b*lr;
```

The interaction between the new suspension system and the vehicle model is defined by:

```
F_rz*lr=F_zWhl*lf+F_xWhl*h+J_y*der(w_b);
```

To confirm that the new phenomena have been modeled correctly we used figures n ° 5 and n ° 6 respectively for the load transfer and for the suspensions.

Fig.1

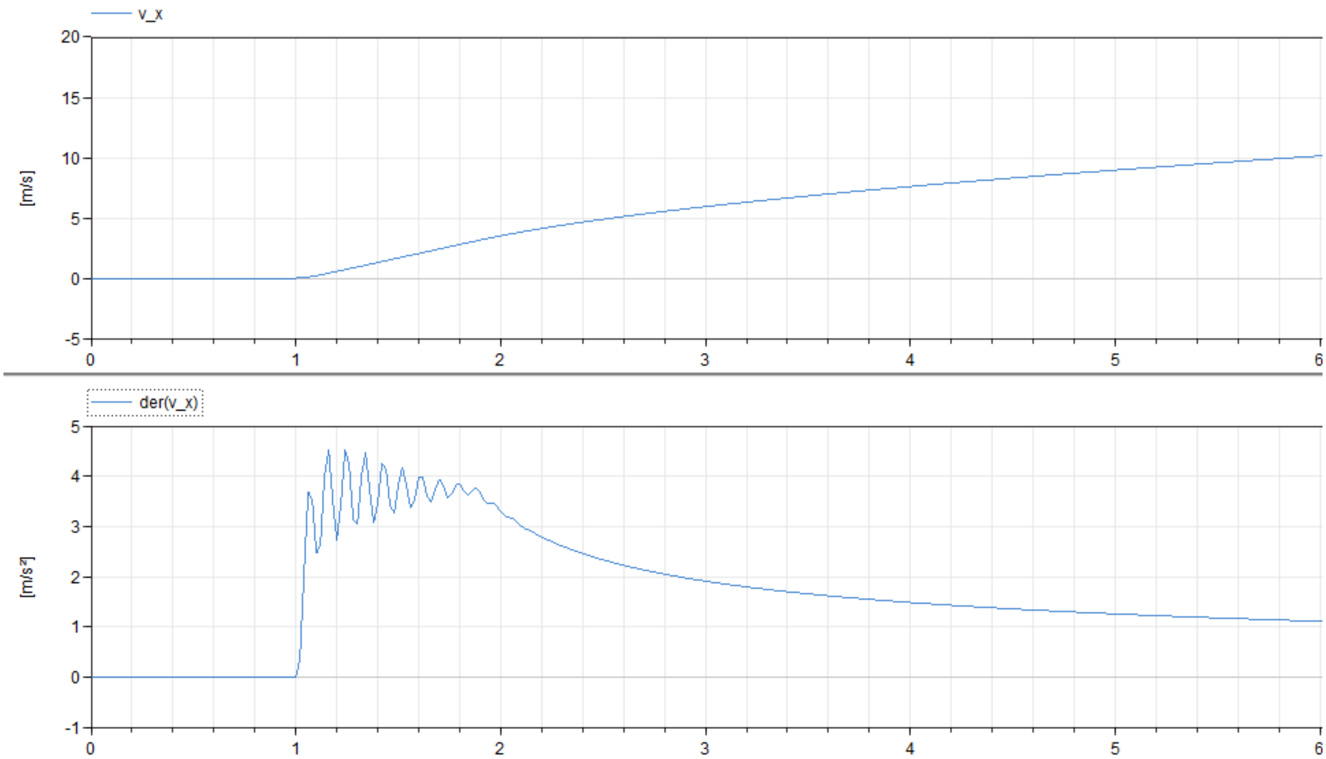

Fig.2

C_Dr (10^3)	Avg ax	D ax / Avg ax
23	3.75	0.47
30	3.775	0.43
46	3.75	0.36
60	3.68	0.31
100	3.65	0.26
110	3.625	0.24
120	3.65	0.23

Fig.3

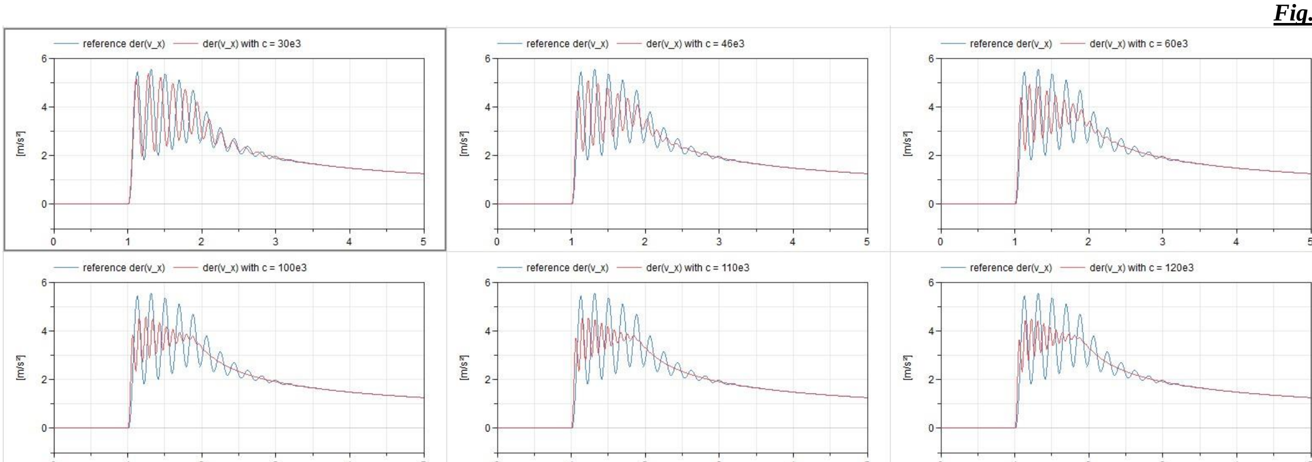

Fig.4

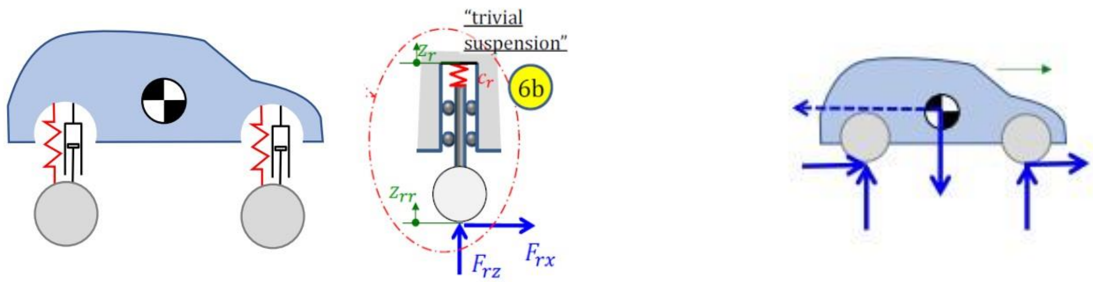

Fig.5

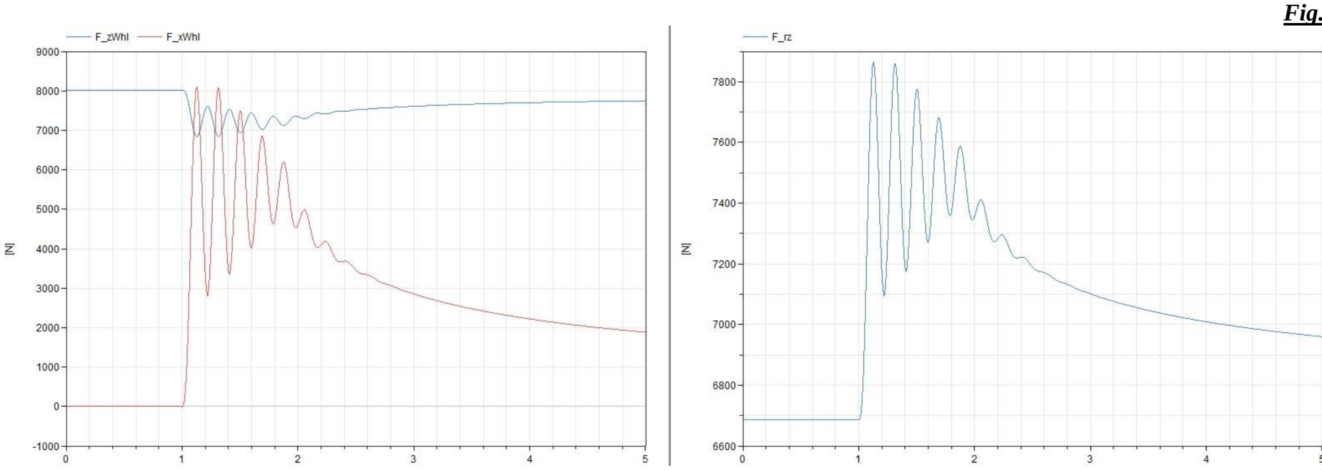

Fig.6