CZECH TECHNICAL UNIVERSITY IN PRAGUE
Faculty of Mechanical Engineering
12120 - Department of Automotive, Combustion Engine and Railway Engineering
Technická 4, 166 07 Prague 6, Czech Republic

REVIEWER'S REPORT ON THE MASTER'S THESIS

Master's thesis title .... **Power Transmission Torsional Analysis for a Tilting Test Stand**

Author .... **Mehmet Ekmekci**

Reviewer .... **Ing. Václav Zoul, CSc.**

Evaluation criteria and their classification

Fulfilment of the thesis requirements and goals.. E (sufficient)

Methodology a its application.. D (satisfactory)

Application of knowledge gained
by self-study and from professional literature .... E (sufficient)

Usage of groundwork and data from practice .... E (sufficient)

Professional level and contribution of the thesis .... E (sufficient)

Formal aspects of the thesis .... E (sufficient)

Further comments to the thesis:

In the diploma thesis was found plenty of weaknesses, both technical and formal. It is not possible within this opponent review to list these all deficiencies in detail. Therefore only remarks to some of them will be presented here.

Language and way of expression:

In the thesis a deficiency and lack of experience in reading and writing of technical texts in English is obvious. Plenty of explanation is complicated, plenty of words are used in complicated way and it needs too much effort to understand, what the student wanted to say. Expressions are often used, which are not common in technical texts.

The first example:

The paragraph 3.1 for example could be written simply and shortly this way:

"To carry a torsional vibration analysis the test stand was describe by a mass-elastic system composed of discrete masses connected by mass-loss springs, as usually in such cases. Such a schema allows to calculate the natural frequency as well as the vibration caused by periodical excitation (forced vibration).

The natural frequencies show, at which engine speeds the resonances of vibration could occur, the results of forced vibration calculation show, how high the load in various parts of the propulsion could be awaited."

and not so complicated, as it was written in the thesis.

The second example:

Only small part of the dual mass flywheel is used for the test ring. It is the part consisting the springs which should act more or less as a torsional flexible coupling. Therefore the description of the function of the DMF (pages 5 – 11) is full useless for the purpose of the test stand vibration analysis.

"Inertia calculation for the DMF" (in reality an experimental way for determination of the inertia moment of the DFW)

Firstly, there is presented no detail drawing or a schema of the DMF, which is concretely used for the test stand (the Fig. 2.1 is only a popular figure taken from some publication). Therefore it is not understandable, how the masses and springs are arranged in the device.

Second. In reality for the evaluation of the moment of inertia only one formula is necessary, especially when the stiffness of the string is known (page 12). Therefore almost all formulas on the page 13, which, by the way, are written rather confusingly, are unnecessary.

"Torsional stiffness of the DMF"

Again, a scheme of the spring arrangement, together with the necessary dimensions is missing. Therefore the calculation of the stiffness becomes confusing. It is for example not clear, if the both springs are inserted in the DMF with preload (a preload could change the stiffness substantially) or not. It is also not quite clear, whether the DMF has or has not different stiffness in the both directions of the twist.

Using of the term "phase difference" for the twist angle between two discs is a bit uncommon and obscure, similar as the term "perimeter" there, where the term "circumference" is more common (page 16).

Proclamation that for the calculation of torsional stiffness a differential equation was used (page 15) is a bit excessive, looking on the formulas on the page 16.

Similar, it will be necessary to explain the last sentences on the page 16 (to generate a critical torque results) and why so many formulas are necessary (page 17) to express the simple equation showing that the torque [Nm] is stiffness [Nm/rad] multiplied by twist angle [rad].

“Planetary gear set”

Totally unnecessary and confusing is description of the entire ZF gear box, when only one gear (with gear ratio 2,077) from the whole gearbox was used.

On the other side the moments of inertia of the wheels, which are also an important parameter for the mass-elastic system, are not calculated or presented. When these inertias were neglected (mentioned until in the next paragraph 2.2.2, page 19), it could be good to know, how big or small they are.

“Inertia and stiffness calculation for the input shaft”

Totally chaotic. What are the elements 1 ÷ 6, marked in the table page 20 and where these elements are located? Why the Poisson’s ratio is mentioned (page 19)?

“Parker hydraulic pump”.

As the maximum speed of the pump on the test stand will be 3000 rpm, deep enough under the maximum continuous pump speed 4000 rpm it is not surprising that the “all values of flow” (page 23) are “within the limits”. The Tab. 2.6 and the diagram Fig. 2.16 are not necessary to confirm it.

More important are the pressures (the pressures are connected with the transmitted torque). In this context it will be suitable to explain the meaning of the terms the “operating pressure” (Tab.2.5) and the “differential pressure” (page 23).

In description of the hydraulic connection between the pump and the hydromotor is essential to describe this arrangement with all accessories as a valve block with pressure relief valves. These valves protect the system, pump and motor from high pressure spikes as well as they help to prevent cavitation.

Adjustment of relief pressure is an important parameter in relation to the dynamic pulsation of the oil pressure in the system.

“Mayr Roba DS Coupling – Size 64”

It is difficult to understand, why in this paragraph the weight, moment of inertia and stiffness are calculated (I am not sure, if correctly), when these all values are presented in the catalogue of the company Mayer, inclusive other important data, as a permissible torque, misalignment and radial and axial stiffness (data, which are also important when the load of the coupling in operation should be assessed).

Using of unit [m] in Tables 2.7, 2,8, 2,9 for dimensions of relative small couplings is not very common.

Please, explain the formulas in last lines of the paragraph 2.4 (before the chapter 2.5). Are the both springs (stiffnesses) connected parallel or in serie?

“Building the Analytical Model”

The reduction of the moments of inertia in the Table 3.1 is not correct. The values of I must be divided by quadrat of the gear ration, not multiplied.

“Setting Up the Matrixes and Explanation of Method”

The results of the subroutine eig are eigenvalues and eigenvectors.

The presented results are not eigenvectors (as written) but the eigenvalues. Maybe, it will be useful to know also the eigenvalues which enable to plot normal elastic curves.

“Campbell”

The most important area in the diagram (red rectangle) should be plotted in a separate diagram to be better for reading. Also the idle speed should be marked.

As the engines intended for testing are four stroke engines, the lines for half-harmonic orders should be plotted.

Please, explain this part of text in detail:

The natural frequencies 150 and 841,2 RPM cross with each order, while most importantly on the 1st order of the engine (which can also be referred to as natural or 0th order) the system goes into resonance 2 times, 1 being in between 0-100 RPM and the other being in 900-1000 RPM.

“Assessment of Natural Frequencies and Dynamical Variables Using the Dynamic Model”

Also this Chapter is written very chaotic and poorly arranged.

In the input data no information about power and torque characteristics and idle speed of the engine are presented. More symbols written on the pages 36 and 38 are without explanation. Diagrams Figs. 4.2 and the others are without units.

The headline “4.3 Calculating the Engine Motor and Idle Torque” is not to understand.

It is possible that it is intended to test the engines on the test ring also when the engine will turned over in “motoring regime”. But this type of run was not mentioned before and also not in the assignment. Why in this case the diagrams 4.10 – 4.16?

It could be assumed that it is intended to test the engines on the test ring even when the engine is turned over powered by the dynamometer in a "motoring mode". But these types of testing were not mentioned before and also not in the specification. Why in this case the diagrams 4.10 to 4.16?

When the motoring regime was still a subject of interest, where are the results of calculation?

Please some comments and explanation to the diagrams Fig.4.1 and 4.8.

“4.4. Setting Up the Dynamic Model Using GT-Suite”

In this chapter the goal of the entire calculation is defined:

"The goal here is, …. the dynamic torque between first and secondary masses of the dual-mass flywheel, the torque between the hydro generator and the hydro motor, and the frequency analysis of the whole system to monitor the critical resonance areas."

More precise it could be said that the final goal of the thesis should be a comparison between the calculated values of dynamical load determined for:

• the springs in the DMF

• the hydrostatic transmission between the pump and hydromotor

and the permissible values.

For the springs the load was primarily defined as a deformation of the springs (or the twist angle between the both flanges of the DMF)

For the hydrostatic transmission the pressure pulsation and the pressure peaks are deciding.

An important factor for such calculation is the damping. The damping is not so much important in connection with the time which is necessary for stabilization of simulation results, as discussed in the thesis, page 46 (last lines), but because of calculation of resonance peaks.

Please, explain the way in which the damping was used and the using of the trail-failure method in this case.

“Defining a linearly interpolated torque profile for the whole simulation”

For description in the schema Fig. 4.17 too small fonts are used, it complicates readability.

In the schema Fig. 4.17 the block for the torque frequency analysis is drawn, but for DMF only, not for the hydrostatic connection.

If I understand correctly, based on the 6 torque profiles available before (Figs. 4.2 ÷ 4.7) using interpolation 56 profiles for excitation was prepared (page 51) for a speed region between 500 and 6000 rpm (page 49).*) It means, between each from the simulation calculation is a step 100 rpm. If the resonant peaks should be found with sufficient accuracy, then step 5 rpm, or highest 10 rpm are necessary.

*) in reality the calculation started at 100 rpm, see the Fig.4.22.

I have not found information, how long is one integration step in every simulation calculation carried out for every speed step. These step influences the quality and accuracy of depiction of calculated waveforms. Looking on the Figs.5.15 and 5.16 I guess that this step is too long.

There are two figures signed as 4.24. There are also two different formulas for Tcrit (page 54).

No units for Output in the diagrams Figs. 5.1, 5.2 and 5.4.

Comparing the Fig. 5.1 with the 5.3 the mean value of the torque could be really about 120 Nm (at 800 rpm). But mean value from the Fig. 5.2 does not correspond with the 5.3 (for 200 rpm).

What does it mean that the torque values below 300 rpm are negative? (Fig.5.3).

“5.2. Hydro Motor and Hydro Generator”

Where was found and taken over or how was derived the relationship describing the correlation between torque and oil pressure?

Why the results of dynamic oil pressures (part of which is plotted in the Fig.5.4) are not plotted in the Fig. 5.10, in place of the dynamical torque?

The mean value of the dynamical oil pressure from the Fig. 5.4 (about 90) does not correspond with the mean value from the Fig. 5.5 (for 100 rpm about 35 bar).

It is hard to believe that the highest peak of the dynamical pressure oscillations occurs directly at 159 rpm, it means directly where the 1st order excitation corresponds with the 1st natural frequency. The highest peak could be awaited at 80 rpm, where the resonance of the 1st NF with the 2nd order of the excitation could be awaited. Regrettably the diagrams Figs. 5.7 ÷ 5.10 start at 100 rpm.

In every case for the depiction of the dynamical torques or pressures is not usual and appropriate to use such type of diagrams which are presented in the Figs. 5.7 ÷ 5.10.

The form of frequency spectrum (dynamic amplitudes in dependency on the rotational speed) gives a better overview and offers the possibility to compare the results of calculation with the permissible values.

Also the diagram in Fig. 5.1, 5.2 and 5.4 show very good vibration characteristic. They say that nothing at frequencies that are contained in the oscillations.

“5.4. Transient Behavior for Engine Start“

The start and stop are problems more complicated as described in the thesis. Firstly, during starting period the engine is running in the motoring regime usually only shortly, till the working (combustion) regime starts to work. Depending on the type of engine the engine starts to work at speed significantly lower (here probably to180 rpm – see the Fig.5.12) than the idle speed (here 1000 rpm) is. After it the engine is running with its own power and the intensity of the excitation depends partly on the regulation (how much fuel come to engine). These aspects are not considered in the thesis. In every case, the motoring regime cannot be used for the whole period of starting.

Also the form how the torques are plotted is not appropriate. Usually and properly the torque-time diagram is presented together with a diagram, where the dependence of the engine speed on the time is plotted. This brings better orientation in the problem.

In the diagrams, where the coupling’s torque during start and stop are plotted, a crossing of resonances are usually good to see. Nothing like this could be found in the

Figs. 5.13, 5.15 and 5.16. The all three diagrams show an oscillation at almost constant frequency, approximately 2,3 Hz, which is near to the 1st NF. During stop or start the frequency of excitation and with it also the frequency of torque oscillations change. The presented diagrams remind an impulse response, not a frequency response excited by excitation with changing frequency.

“6. Conclusion”

In the conclusion is stated that the aim of the thesis were fulfilled.

“For the parts dual mass flywheel and the hydro elements, the results seem to always stay below the critical values even when the system goes through resonant frequencies.”

As seen in the above submitted reviewer’s comments, this statement has a number of omissions, that should be discussed eventually disproved during the defense.

Reviewer’s conclusions:

In the conclusion of the thesis is stated that the goals were met and proposed test stand is free from risk of increased loads caused by torsional vibrations.

It is necessary to add that the thesis is rather confusing, loaded by more serious weaknesses both technical and formal. The expectations of the supervisor were not fully fulfilled.

Nevertheless I recommend the master’s thesis for the presentation and defense, to give to the student possibility to clarify his arguments.

Summary:

I **recommend** the master's thesis for the defence.

Summary classification of the master's thesis .... E (sufficient)

Ing. Václav Zoul, CSc.
....
reviewer’s name reviewer’s signature

In Prague............ 5th of February 2017