DRIVELINE VIBRATION ANALYSIS OF A LAND ROVER FOUR WHEEL DRIVE VEHICLE

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Abstract
The presence of universal joints in a driveline can cause rotational velocity fluctuations between the crankshaft and final drive. These fluctuations can be a fundamental cause of driveline vibration. The problem is further enhanced if tolerances on geometry, assembly, mounting compliance, rotational component out of balance and universal joint phasing are not met.

On any vehicle, there are a significant number of variables which can be the cause of driveline vibration. For example an engine mount may be manufactured with a tolerance of +/-15% on stiffness. Thus it may be considered to have two or more levels of stiffness. Rover Group had a requirement to increase their understanding of the interaction of these variables to assist them with product design and development programmes. To perform a full investigation into the problem by analysing all combinations of all variable levels concerned would be too time consuming and expensive. By employing the technique of Design of Experiments(DOE), it is possible to reduce the amount of work required to identify the variables which have the greatest influence on driveline refinement.

It was decided that the Land Rover Discovery would be used to perform a trial investigation with the following objectives:
Objectives:

- To analyse the effects of out of balance forces, geometry assembly tolerances and mounting compliance tolerances on driveline vibration using a constant velocity crankshaft input.
- To identify the variables on the vehicle which are most significant in causing driveline vibration.
- To produce a parametric ADAMS model of a Land Rover Vehicle driveline which can be used by Rover engineers after completion of the project.

The operating condition chosen arbitrarily as a start point for the analysis was steady state cruising at approximately 60 miles per hour.

Design of experiments is an analysis technique which provides planning and analysis tools for running experiments (or computer simulations in this case), such that a fraction of the full permutations can be used to identify which variables are most significant with respect to the defined objective (e.g. to reduce driveline vibration). Action can then be taken to remedy the problem variables and time and expense is not wasted on those variables shown to be insignificant.

A computer model of the complete Land Rover Discovery vehicle was built using the mechanisms analysis software package ADAMS (Automatic Dynamic Analysis of Mechanical Systems). Twenty variables of interest were chosen and, using the current facilities within ADAMS, these were modelled in such a way that they could be automatically changed during a DOE run. For example, an engine mount would have two stiffness values either side of the design condition to represent the manufacturers tolerance on the component. As far as possible, this model was verified by comparing simulation natural frequencies with test.

Experiments, or sequences of simulations were designed using the statistical analysis software package Minitab. These experiments were used to provide ADAMS with information on how many simulations to perform and the level settings for each variable during each simulation.

ADAMS was then used to perform an initial screening experiment, where the most significant variables were selected for further analysis, and a further experiment which was the main DOE analysis.

The results from the ADAMS simulations were analysed within Minitab in order to identify the variables with the most influence on driveline refinement.

1.0 Introduction

1.1 Background

Relative rotational velocity fluctuations between the input and output shafts in a driveline are caused by the presence of the universal joints in the system. These fluctuations can only be completely overcome if three shafts are present in the system, namely an output shaft from the gear box, the main propeller shaft and the input shaft to the differential. There will be no relative velocity fluctuations provided that the input and output shafts are parallel, or the total angle at each joint is equal with appropriate phasing. When this is not the case, the situation can be improved by introducing phasing between the universal joints, but it does not completely eliminate the source of vibration.

It may well be that the vehicle driveline has been designed in such a way that this source of vibration should not be present. The presence of assembly tolerances on power train, chassis and suspension components, however, may introduce driveline angles which would could be a cause of vibration.

1.2 Land Rover Discovery Driveline

In the case of the current Land Rover Discovery vehicle, in its design condition, the rear transfer box output shaft, propeller and differential shafts are all in line with no universal joint (UJ) phasing. In the case of the front driveline, however, the propeller and transfer box shafts are angled relative to each other. The velocity fluctuations are not considered to be significant and are reduced by the introduction of a 45 degree UJ phasing, but the source of vibration is still present.

Manufacturing tolerances can also complicate the problem of driveline vibration. On the propeller shaft alone there are manufacturing tolerances on out of balance mass, UJ phasing, UJ swash and UJ eccentricity all of which could be significant causes of driveline vibration. A typical tolerance on an engine, chassis or suspension mount stiffness is +/- 15%. As the entire power train system on many vehicles is supported on these mounts, the problem could be further amplified by changing the system natural frequencies from the design targets.

1.3 Variables Considered

The number of variables to be considered when analysing the problem of driveline vibration is indeed vast. As a starting point to this exercise, a number of variables and corresponding levels were agreed upon during discussions between MDI and Rover engineers. A summary of the variable names, their abbreviations and levels is given in table 1.1.

Variable Description (Units)	Abbreviation	Level 1	Level 2
Chassis mount stiffness (N/mm)	chasmt	-15%*	+15%*
Engine lateral position (mm)	englat	-10	10
Engine longitudinal position (mm)	englon	-10	10
Engine pitch position (deg)	engpit	-1	1
Engine roll position (deg)	engrol	-1	1
Engine vertical position (mm)	engver	-10	10
Engine yaw position (deg)	engyaw	-1	1
Front prop. out of balance (kgmm)	foob	-tol.*	+tol.*
Front propshaft UJ phasing (deg)	fphase	-tol.*	+tol.*
Front engine mount stiffness (N/mm)	fengmt	-15%*	+15%*
Front differential pitch position (deg)	fdifpi	-1	1
Front propshaft front UJ swash (deg)	ffswsh	-tol.*	+tol.*
Front propshaft rear UJ swash (deg)	frswsh	-tol.*	+tol.*
Rear prop. out of balance (kgmm)	roob	-tol.*	+tol.*
Rear propshaft UJ phasing (deg)	rphase	-tol.*	+tol.*
Rear differential pitch position (deg)	rdifpit	-1	1
Rear propshaft front UJ swash (deg)	rfswsh	-tol.*	+tol.*
Rear propshaft rear UJ swash (deg)	rrswsh	-tol.*	+tol.*
Rear left engine mount stiff. (N/mm)	rlenmt	-15%*	+15%*
Rear right engine mount stiff. (N/mm)	rrenmt	-15%*	+15%*

Table 1.1: Variables and levels used in driveline vibration analysis

*Actual figures not published for reasons of confidentiality.

As a general rule in deciding the variable levels, the following tolerances were observed:

translation position: +/-10 mm
rotational position: +/-1 degree
mount stiffness: +/-15%

Other tolerances, e.g. UJ phasing, propshaft swash and out of balance were obtained from engineering drawings.

It was decided that only two levels of each variable were necessary since this is the requirement to perform a DOE analysis whose purpose is to identify variables which influence the objective most. In this case the objective was to minimise driveline vibration.

Having decided which variables and levels to employ, the project was then undertaken in three phases; model building, model verification, and model analysis.

2.0 Model Building

One of the deliverables of this project was a customised, parameterised full vehicle ADAMS model complete with driveline, and capable of being used for DOE analysis. The process of building this model is described in sections 2.1 to 2.4

2.1 Initial Model

The model building process began by obtaining a full vehicle ADAMS model of the Land Rover Discovery which had been used for previous ride and handling work at Rover. This model did not contain a representation of the driveline and so the initial tasks were as follows:

- To check all geometry and compliance associated with the driveline and update if necessary.
- To add a full powertrain model including power unit and transfer box, differentials, axles and all associated rotating shafts and appropriate gearing.

Geometry, mass and inertia figures and gear ratios for the power train were obtained from Rover engineers.

2.2 Parameterisation and Customisation

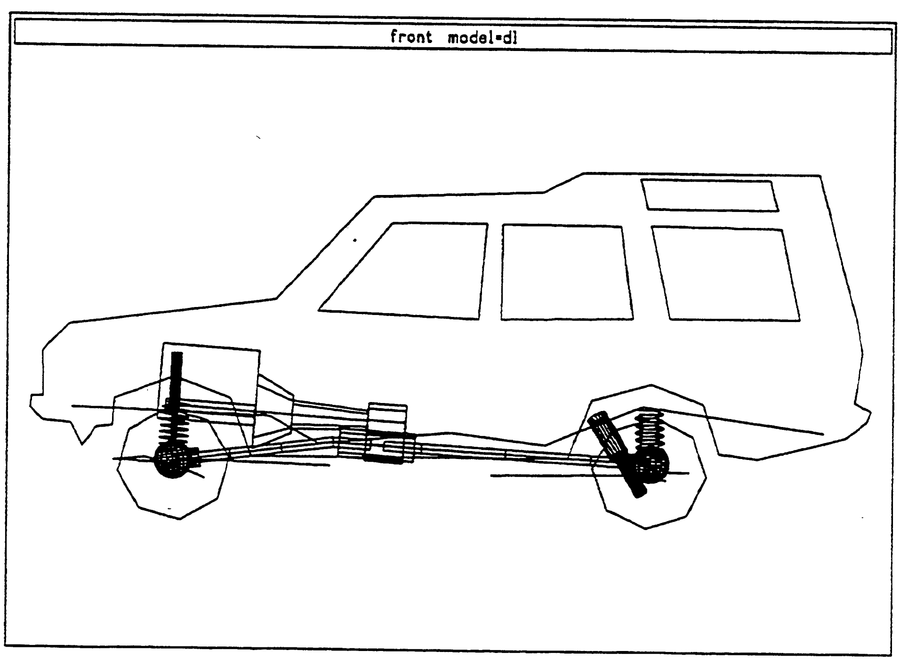


Figure 2.1a: Parametric Model Before Modification

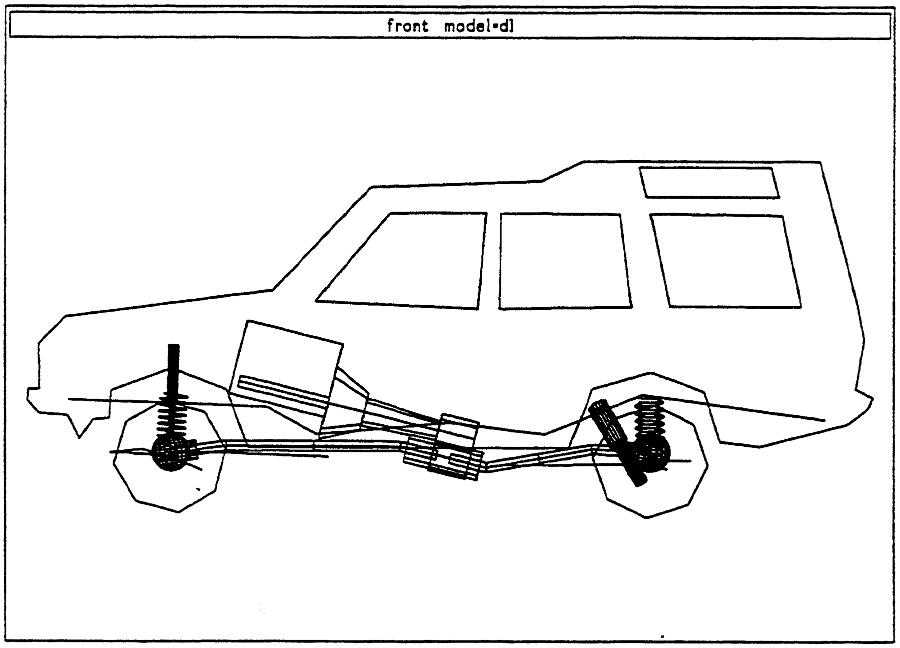


Figure 2.1b: Parametric Model After Modification

The current facilities within ADAMS/View enable parametric associativity to be included in the model. For example, if the vehicle power unit position had to be changed, the part could be placed in the correct position with one instruction. The transfer box to propshaft connections would change automatically, as would the lengths and positions of the propshafts themselves. This is demonstrated in Figures 2.1a and 2.1b, the power unit positional change being exaggerated for purposes of demonstration.

In order to enhance the ease of use of the model, custom menu buttons were added to the ADAMS/View environment. These buttons enable the model to be modified by users who have little or no experience of ADAMS at all. Such a system of menu buttons would be as shown in Figure 2.2. The custom buttons are shown in bold.

Figure 2.2: Custom Menu Buttons

2.3 Modelling the DOE Variables

The DOE analysis capability within ADAMS is made possible through the use of design variables. These variables are specified to have either a range of values, for use in optimisation, or to have a number of allowed values which are the levels used in DOE. A design variable can be used to represent any entity type of data, whether it be co-ordinate, dimension, mass, force, stiffness, damping, part initial conditions, etc. During a DOE analysis, the algorithm employed to perform the runs, selects the required variable (or factor) levels from the allowed values of each design variable. Thus only one ADAMS model is required to perform a whole range of DOE runs.

For each of the factors listed in table 1.1, design variables were created in the ADAMS model. In the cases where a positional change was required, for example in the case of the engine vertical and yaw movements, it was necessary to introduce a dummy part and an extra joint. This meant that instead of the engine being mounted directly to the chassis via the engine mounts, it was mounted to the dummy part. The dummy part was then connected to the chassis with a cylindrical joint whose translation/rotational axis was orientated vertically. This cylindrical joint then had two motion generators associated with it, one translational and one rotational. These motion generators then accessed the appropriate design variables, thus imposing the required vertical and yaw position levels.

In order to impose the phasing between the propshaft universal joints, each propshaft was split in two. The propshaft halves were then connected using a revolute (hinge) joint. This joint then had a motion generator associated with it which accessed the appropriate design variable to impose the required joint phasing.

Propeller shaft out of balance was modelled by attaching an extra mass to the centre of the propshaft via a translational joint whosé axis was perpendicular to the axis of the shaft. A motion generator then accessed the appropriate design variable to impose the out of balance position.

Engine and chassis mount stiffnesses accessed the appropriate design variables directly with all radial stiffnesses being set to one sixth of the axial stiffnesses as agreed with Rover engineers.

Before the main DOE analysis could be performed the model had to be verified and tested.

3.0 Model Verification and Testing

3.1 Model Verification

Before using the ADAMS model to perform the large numbers of runs associated with a DOE analysis, it was necessary to obtain sufficient confidence that the model was indeed capable of predicting the correct results.

A good method of doing this is to compare the computer model natural frequencies with any known corresponding natural frequencies on the actual vehicle. After discussions with Rover NVH personnel, it was agreed that, for such a driveline vibration study, it was imperative that the model should correlate to the driveline 'shunt' mode the actual vehicle. This mode, shown in figure 3.1, is where the engine and driveline vibrate forwards and backwards on engine mounts.

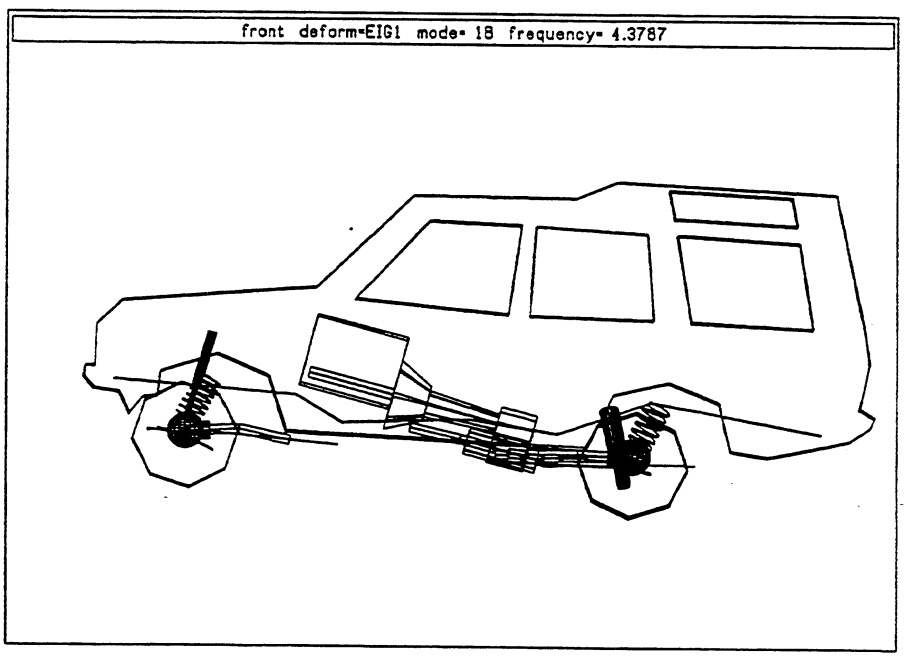


Figure 3.1: Driveline Shunt Mode

Mode Description	Model Freq. (% of actual)
Driveline shunt	-5%
Engine pitch/bounce	-10%
Front axle bounce/yaw	-11%/-6%
Front axle fore/aft	-5%
Rear axle bounce/yaw	-14%/-10%
Rear axle fore/aft	+5%
Front axle nod	-16%%
Rear axle nod	-13%

Table 3.1: Comparison of Model and Vehicle Modes

It should be noted that the fore/aft displacements shown in figure 3.1 have been exaggerated for purposes of demonstration.

Other modes of significance for such an exercise were considered to be engine pitch/bounce, front and rear axle bounce/yaw and front and rear axle nod. Table 3.1 shows a comparison of model and vehicle modes of interest.

It can be seen from table 3.1, that although good correlation was achieved on the shunt mode, correlation on some of the other modes was not so accurate.

It should be noted at this stage that to obtain good model correlation through a range of frequencies is a major task in itself. Many of the model modes obtained were, as expected, very sensitive to the tyre stiffnesses used. Good estimates of the tyre linear stiffnesses were made but these can easily change depending on the frequency range and amplitude of vibration in question. The correlation obtained in this exercise was considered good enough to perform the DOE exercise, particularly as the project was a study of the relative effects of the variables in question.

3.2 Model Testing

Before submitting a DOE analysis in which a sequence of 128 runs would be performed, the model robustness needed to be tested. In other words would the model run under the extreme conditions of all variables being set to their maximum levels. To test the model robustness a number of dummy runs would be performed with the required model inputs and the corresponding results would be examined for accuracy.

3.2.1 Model Input

Two model inputs were required to perform the simulations, a motion input on the crank shaft, and the application of rolling resistance or friction torques to each wheel. The rolling resistance torques were simply applied as external 'action only' torques to each wheel about each spindle axis. The motion input to the crank shaft was more complex to apply.

It had already been decided that the crank shaft speed would be representative of a vehicle speed of 60 mph in 5th gear. After the differential, transfer box and gear ratios had been accounted for, this meant a crank shaft speed of 2237.4 rpm or 37.29 rev/sec. To start each simulation with the crank rotating at this speed would have meant severe and complex initial conditions being applied to the model, bearing in mind that, many other motion generators were being used to apply positional levels of design variables. It was hence decided that each analysis would be performed in three stages as follows:

Stage 1: The first second of the simulation would be a quasi-static analysis during which the model would settle to its start position and positional DOE variables would be gradually applied.

Stage 2: The next five seconds would be a dynamic analysis in which the crank shaft would be accelerated up to the required operating speed using an ADAMS 'step' function.

Stage 3: The next two seconds would be a further dynamic analysis in which the crank shaft speed would be held constant but the number of output steps per second would be increased to ensure that the frequencies of interest could be identified in the model results.

The crank shaft motion input for each analysis would then be as shown in figure 3.2.

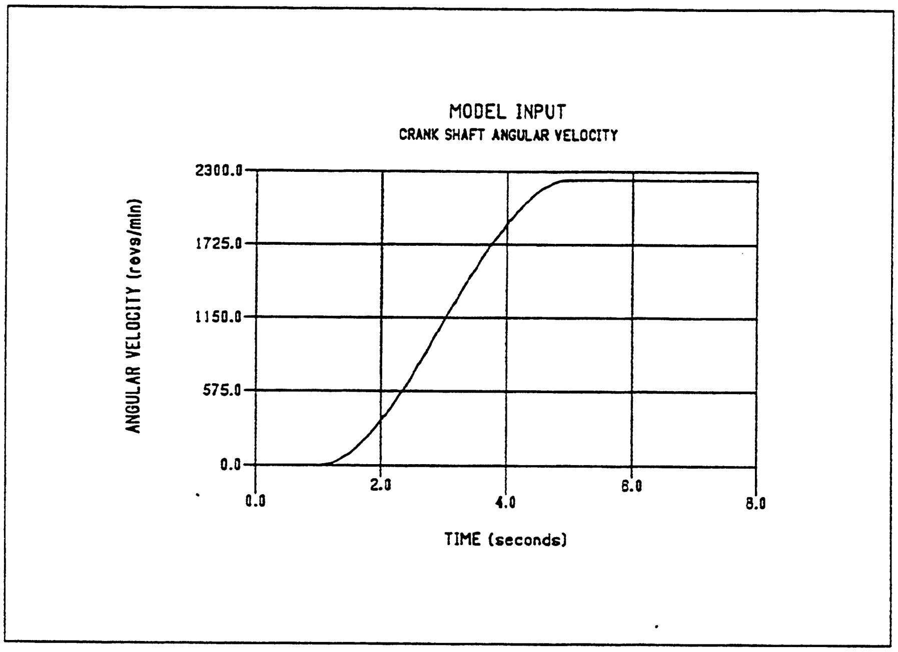


Figure 3.2: Model Input - Crank Shaft Angular Velocity

3.2.2 Model Output

To make sure the model was functioning as expected, the velocity of all rotating shafts was requested as output. This would enable checking that all gear ratios had been implemented correctly, and that the model was indeed picking up the velocity fluctuations characteristically associated with universal joints. The rotational velocity of the front differential shaft can be seen in figure 3.3. All DOE variables are set at their design levels.

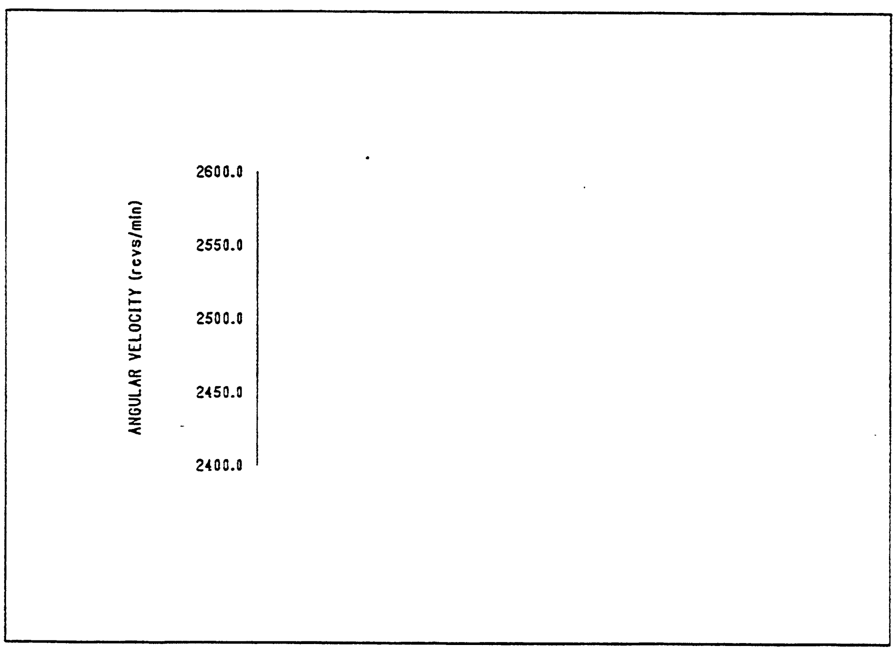
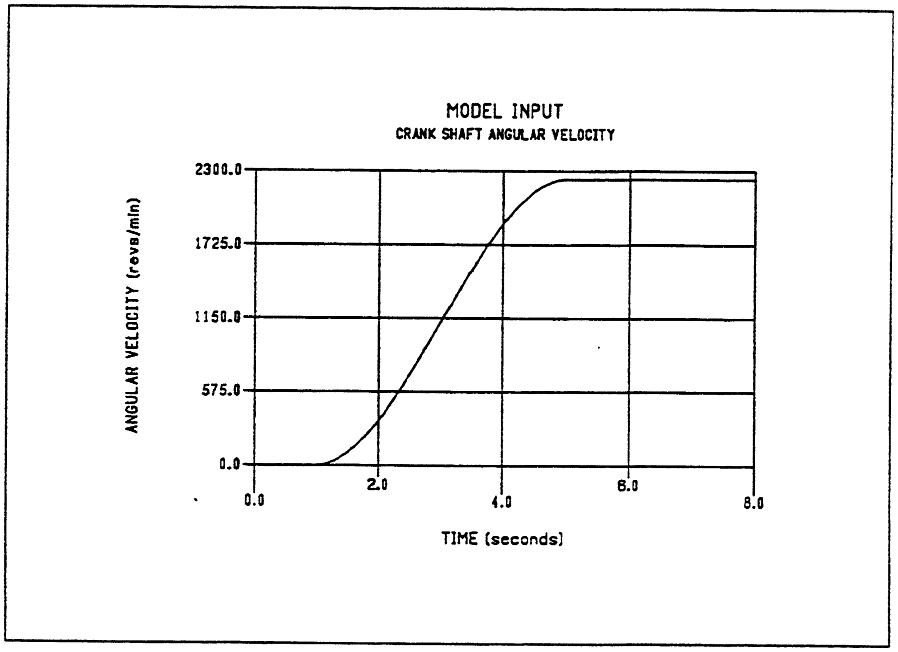


Figure 3.3: Front Differential Shaft Angular Velocity



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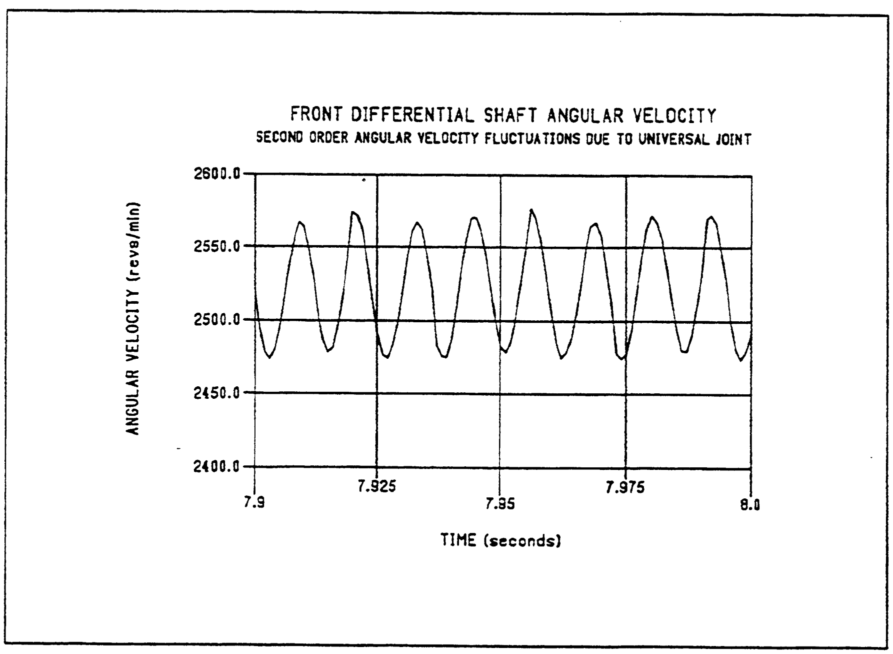

Figure 3.3: Front Differential Shaft Angular Velocity

The plot shows a 0.1 second window of time in which it is possible to identify approximately eight cycles of velocity fluctuation. This suggests that the front propshaft rotational speed is just over 40 Hz, since the velocity fluctuation is a second order characteristic. This value agreed with calculation.

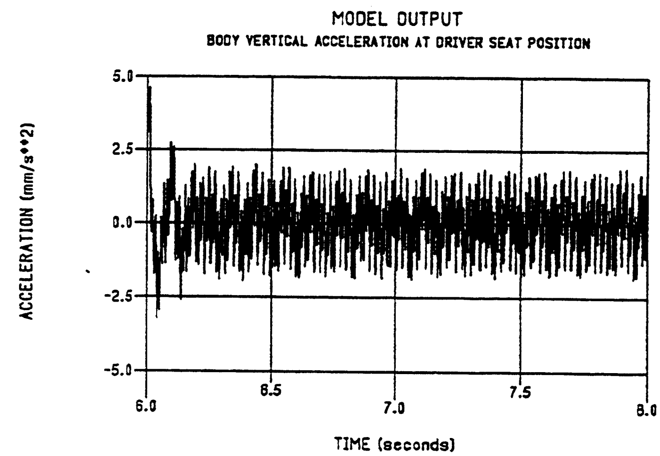

Figure 3.4: Body Acceleration at Driver Seat Position

The objective of each simulation was to measure the vibration felt at the driver seat position. This was achieved by placing a marker on the body at the assumed driver's seat "H" point. Another marker was placed on ground at the same position and the relative acceleration between the two requested. The acceleration output from the simulation was as shown in figure 3.4. This plot shows a time window on the last two seconds of the analysis. At six seconds the vibration is just settling down after the ramp up of crank shaft speed. By seven seconds the vibration has settled down to a consistent fluctuation. It was hence considered best to use the 7 to 8 second window to measure the severity of the vibration. The model proved to very robust running with all DOE variables set at both their design levels and at the maximum levels required for the experiment.

3.2.3 DOE Objective

In order for a true relative comparison of each of the DOE runs to be made, a single figure representation of the severity of the vibration had to be output from each simulation. It was hence decided that the last second of each analysis should be used as a sampling window to calculate the root mean square (RMS) acceleration over this period. A Fortran user subroutine was written to achieve this. The RMS acceleration figure was calculated using the magnitude of acceleration over the one second window so that the components in all translational degrees of freedom could be accounted for. Thus at the end of each DOE analysis, a single figure would be returned as the result or response. This is known as the objective of the analysis.

Having performed the possible model verification and testing, the main design of experiments analysis could begin.

4.0 DOE Analysis

The DOE analysis described throughout this section was performed using ADAMS in conjunction with a statistical computer software package called Minitab. Although ADAMS has a number of built in experimental design algorithms, these were not considered suitable for the processes required in the driveline vibration analysis. Hence Minitab was employed to design the experiments, ADAMS then used these designs to perform the simulations and Minitab was used to perform the DOE analysis of the simulation results.

4.1 Variable Selection

As already described in section 1.3, the variables (or factors) of interest in the driveline vibration analysis were chosen by joint agreement between MDI and Rover engineers. The names of these variables were abbreviated for use within Minitab. A summary of the variable names, their abbreviations and levels is given in table 1.1.

4.2 Initial DOE Screening

Because of the large number of variables to be analysed in the driveline vibration analysis (20 in total), an initial screening exercise had to be performed to reduce the DOE analysis size. This process involves early removal of variables from the analysis, whose influence on the objective is considered to be insignificant. The exercise was performed by using a Plackett-Burman experimental design consisting of 24 runs to analyse the 20 variables. This type of design is typically used for performing initial 'screening' on designs with a large number of factors. It should be noted, however, that the Plackett-Burman design will only show the significance of the main effects and not indicate the presence of factor interactions. The run matrix was generated using Minitab.

The results from the 24 ADAMS runs were then transferred back to Minitab and used to select the most significant factors. This was achieved using the main effects plot shown in figure 4.1.

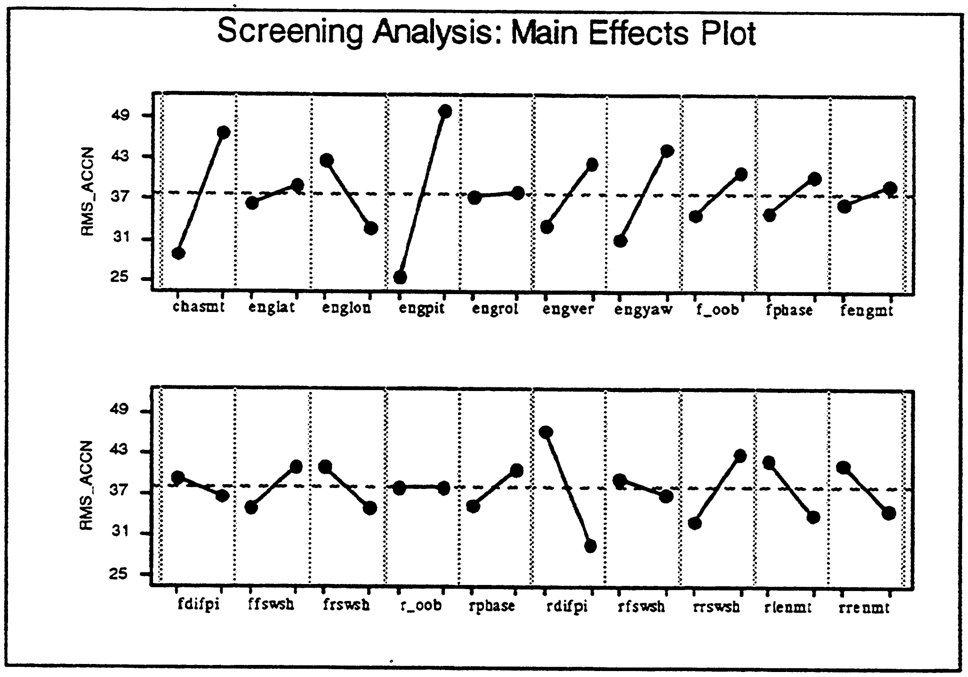

Figure 4.1: Main Effects Plot for Screening Analysis

Each point on the plot is obtained by summing all the RMS acceleration values corresponding to all the occurrences of a factor at the specified level, and then dividing by the number of those occurrences. This is, in effect, the mean acceleration for that factor level. The most significant factors are those which show the most marked variation in RMS acceleration between the high and low factor levels. This was then the criteria for selecting factors for further analysis. Using figure 4.1, the factors chosen were:

1. chassis mount stiffness (chasmt)
2. engine longitudinal position (englon)
3. engine pitch angle (engpit)
4. engine vertical position (engver)
5. engine yaw angle (engyaw)
6. front prop shaft out of balance (foob)
7. front prop shaft rear UJ swash (frswsh)
8. rear differential pitch angle (rdifpi)
9. rear prop shaft rear UJ swash (rrswsh)
10. rear left engine mount stiffness (rlenmt)
11. rear right engine mount stiffness (rrenmt)

4.3 Main DOE Analysis

With the model now reduced to eleven factors, a more specific analysis could be performed. The object of this analysis was to carry out a detailed investigation to identify the factors most likely to influence driveline vibration and to see if there were any significant interactions between the factors. Minitab was used to design a fractional factorial experiment consisting of 128 runs for the eleven factors at two levels. This is known as one sixteenth replicate since 128 = $2^{11}/16$. The design matrix generated in Minitab was then used to perform the runs in ADAMS. The RMS acceleration results from the ADAMS analyses were imported back into Minitab to perform further DOE analysis. The main effects plot for the reduced model was then produced as shown in figure 4.2.

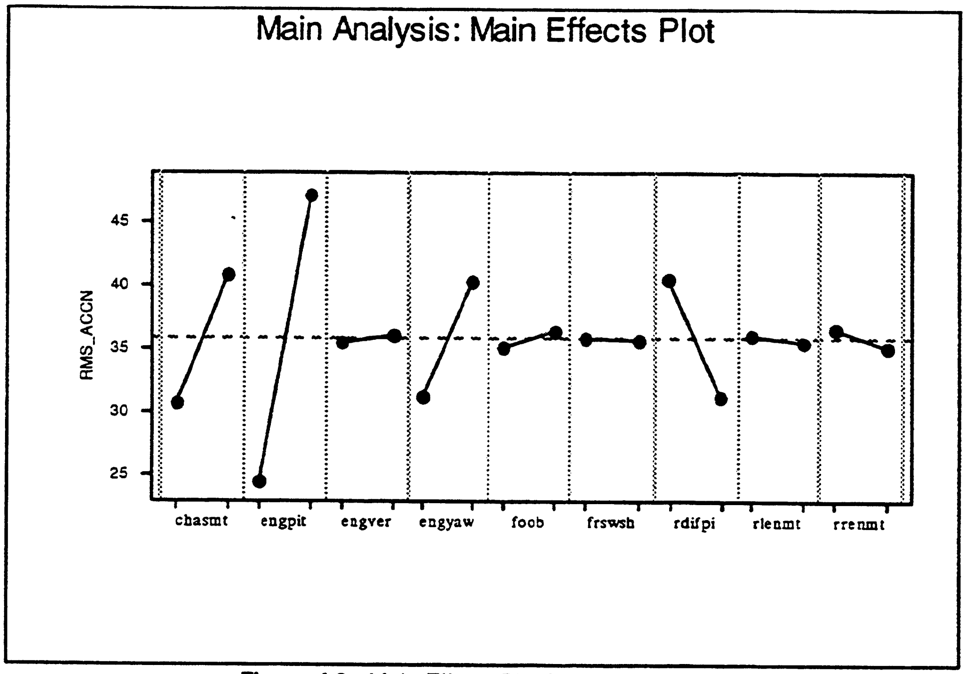

Figure 4.2: Main Effects Plot for Main Analysis

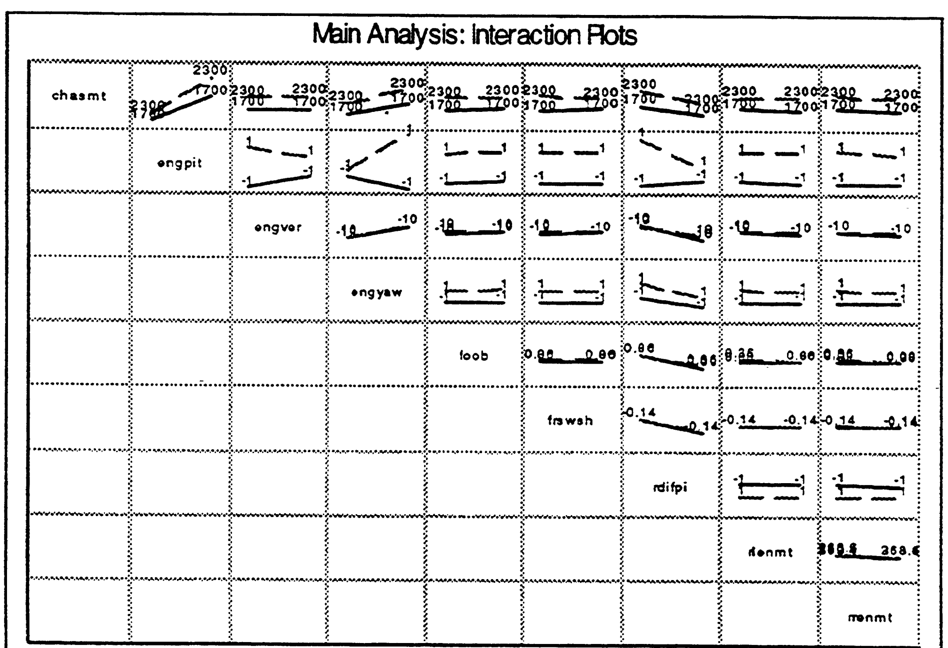

Figure 4.3: Interaction Plot Matrix

From the plot it can be seen that the most significant factor is the engine pitch position(engpit). The chassis mount stiffness(chasmt), engine yaw(engyaw) and rear differential pitch(rdifpi) positions also figure as significant factors.

Having performed the fractional factorial design it was now possible to investigate the interactions between the factors. The interaction plots matrix can be seen in figure 4.3. Interaction plots consist of two straight lines which, if parallel indicate no interaction, and if non-parallel or crossed, indicate an interaction. The relative angle between the lines is an indication of the severity of the interaction.

Although figure 4.3 is not clearly readable, it can be seen that three significant interactions are present in the model, namely between engpit and engver (engpit*engver), engpit and engyaw (engpit*engyaw), and engpit and rdifpi (engpit*rdifpi).

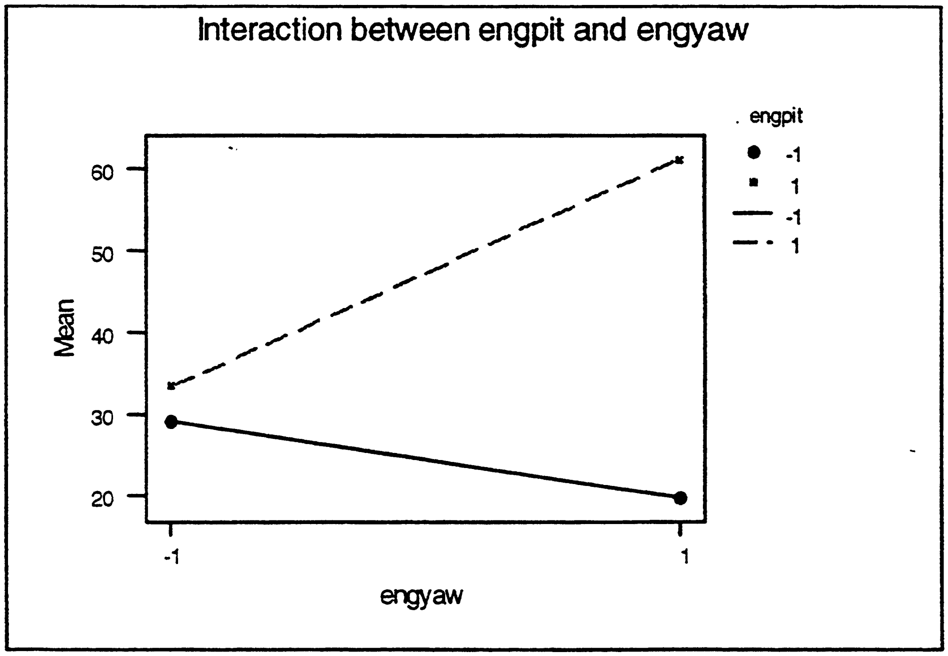

Figure 4.4: Engine Pitch and Yaw Position Interaction

The interaction between engine pitch and yaw positions can be seen in more detail in Figure 4.4 in which the low and high factor levels of each factor are represent by -1 and 1. This plot is effectively saying that the driveline vibration due to these factors is at a minimum when the engine pitch and yaw positions are at their levels of -1 and +1 degrees respectively.

4.4 Statistical Verification of Results

Within statistical packages such as Minitab there are numerical means of verifying the assumptions which have been made visually. So far it had been subjectively established the most significant effects were engpit, chasmt, engyaw and rdifpi. It had also been established that there were significant interactions present, namely engpit*engver, engpit*engyaw, engpit*rdifpi.

The first step in obtaining confirmation of the subjective assumptions was to fit a factorial model to the results. This is effectively fitting a polynomial to the results in order to indicate the significance of the main effects and the interactions by numerical means. Once this had been done it, was then possible to produce a normal probability plot as shown in figure 4.5.

Each point on the plot represents the effect of a main factor or an interaction between two factors. An easy way of interpreting this graph is to drop all points on to the 'x' axis and imagine a normal probability distribution plotted using frequency of occurrence of the effect as the 'y' axis. Thus all points on the normal probability line follow a normal distribution and are least likely to affect the objective. Points away from the line are significant with respect to the objective. The further away from the line, the more significant.

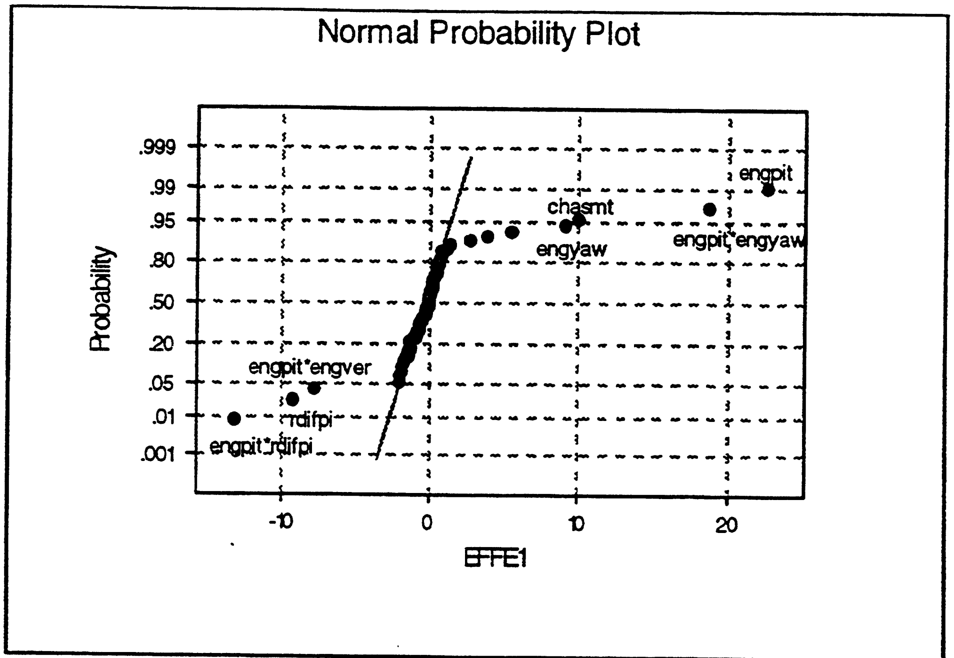

Figure 4.5: Normal Probability Plot for the Main DOE Analysis

Confirmation of the subjective assumptions was obtained from the plot. It was then decided to perform a further investigation on the main effects to check their significance and overall contribution to the driveline vibration.

Analysis of variance, or ANOVA, was performed on the effects of interest, namely engpit, chasmt, engyaw, rdifpi, engpit*engver, engpit*engyaw and engpit*rdifpi. This process would provide information on the relative significance of the effects using numerical means. Hence further data on the subjective assumptions could be made. The ANOVA revealed the most significant effects to be:

1. engpit
2. engpit*engyaw
3. engpit*rdfpi

By performing a linear regression analysis on these effects, it was revealed that 78% of the vibration during the main DOE analysis was indeed due to effects considered. It should be noted that where interactions of interest are present in linear regression, the single effects making up the interactions must also be included even though they may not be significant on their own. Hence the single effects of rdifpi and engyaw were also present in the analysis.

As 22% of the vibration had yet to be accounted for, it was decided to perform a further linear regression analysis on all the significant effects considered namely:

1. engpit
2. engpit*engyaw
3. engpit*rdifpi
4. chasmt
5. rdifpi
6. engyaw
7. engpit*engver

This analysis revealed that the effects considered accounted for 89% of the driveline vibration. At this stage it was considered appropriate to end the analysis.

5.0 Conclusions

From the trial investigation into driveline vibration on a Land Rover four wheel drive vehicle, the following conclusions can be drawn:

- A single ADAMS model can be used to analyse numerous combinations of parameters at different levels using ADAMS design variables.
- ADAMS provides the automatic job submission and objective storage required for effective DOE analysis.
- External DOE algorithms can easily be used via the design matrix file.
- Easy transfer of results from ADAMS to a DOE package enables fast determination of:

  Variable relative significance,

  Variable contribution to objective.