Using Optimization in MSC/NASTRAN to Minimize Response to a Rotating Imbalance

by

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Abstract:

In any applications of rotating equipment, it is common for an engineer to try to minimize the response of a structure with a rotating imbalance. This paper demonstrates how to perform this minimization using MSC/NASTRAN. A practical example problem is used. This sample minimizes the response at the driver's seat of a car model with a wheel out-of-balance. The problem will begin by demonstrating how to perform frequency response analysis of the car model with a rotating imbalance, followed by dynamic sensitivity of the response, followed by minimization of the response by tuning the dampers (shock absorbers) and springs.

During the process, special features in MSC/NASTRAN will be used to assist in understanding the dynamic problem and in determining the best approach to minimizing the response.

Introduction:

MSC/NASTRAN contains many tools for dynamic analysis and design. This paper is intended to demonstrate application of some of these tools to a specific problem, that of a structure with a rotating imbalance. The sample used is an automobile with a tire out-of-balance. Solutions 103, 111, and 200(optimization) will be used to perform the analysis, determine modal contributions to the response, determine the sensitivity of the response to changing several design variables, and finally to minimize the response over the operating frequency range of the problem.

1. Dynamic Analysis of a Model with a Rotating Imbalance

The model is shown in figure 1. It is an automobile frame model with springs and damping elements used to model the suspension. In a real analysis, there would be additional structure modeled and additional masses modeled to represent the drive-train and other components. The loads and frequency range are chosen simply for demonstration purposed and are not intended to model a realistic environment.

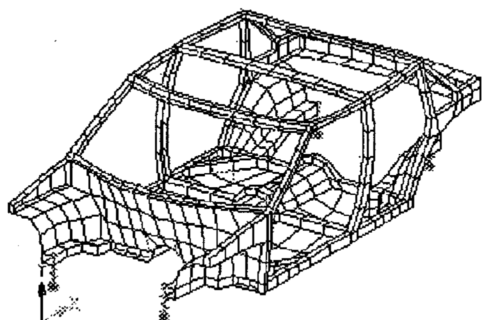

Figure 1 - Car Model

The front left tire is assumed to be out of balance and the operating frequencies are assumed to occur between 0 and 50 Hz. Since the analysis is for a rotating imbalance and the steady-state solution is wanted, SOL 108 (direct frequency response) or 111 (modal frequency response) will be used for the initial analysis. SOL's 101 and 103 will be used to assist in understanding the problem. SOL 200 will then be used to perform the final optimization..

First, the dynamic loading must be defined. The loading is a rotating imbalance acting at frequency ω and may be described as shown in figure 2.

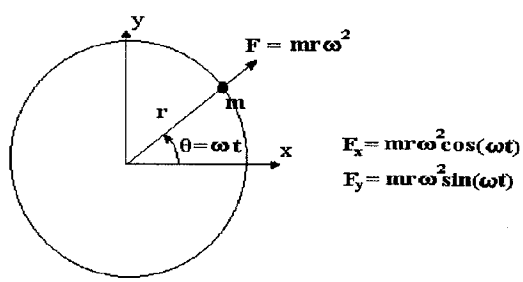

Figure 2 - Rotating Load

At any point in time, the force can be described as a combination of the x and y components. RLOAD1 entries will be used to define each component of the applied loading. The applied load has a constant term (mr) and a frequency-dependent term (ω^2). The constant term will be entered by using DAREA entries and the frequency-dependent term will be entered using a TABLED4 entry. The 90 degree phase angle between the x- and y-components will be entered using a DPHASE entry. These terms will be combined using a DLOAD entry. The following describes how these entries will be filled out for this problem.

First, let us define values for r and m. Since this is a demonstration, let us arbitrarily set r=10 inches and m = 0.3. Therefore, mr = 3.0 will be used on the DAREA entries. As mentioned, the phase angle between the x- and y-components is 90 degrees and will be entered on the DPHASE entry. The ω^2 term is more complicated to describe. This will be done by using a TABLED4 entry, which is best described as a Taylor Series.

The TABLED4 entry uses the algorithm:

$$Y = \sum_{i=1}^{n} Ai\left[\frac{X - X1}{X2}\right]^{i}$$

Where X is input to the table and Y is returned. If the input value of X is less than X3, X3 will be used. If the input value of X is greater than X4, X4 will be used. For this sample, all that is desired is to return with a value equal to ω^2. Therefore, only the term for I=2 will be used, X3 will be set to 0.0 and X4 will be given a value larger than the frequency range of interest. It should be noted at this point, that the input frequencies will be in Hz, not in radians/sec. Therefore, it is necessary to convert the frequencies to radians per second for the equation. This will be done by entering a value of X2 = $(2\pi)^2$ or 39.478.

The load is applied at GRID point 358, which we will assume is the center of the wheel. Using this information, the dynamic load will be entered using the following bulk data entries:

```
DLOAD,1,1.,1.,11,1.,12

RLOAD1,11,20,,,,111

RLOAD1,12,30,,,40,111

DPHASE,40,358,2,90.

DAREA,20,358,1,3.0

DAREA,30,358,2,3.0

TABLED4,111,0.,1.,0.,100.

,0.,39.478,ENDT
```

These bulk data entries are described as follows:

The DLOAD (set 1) instructs the program to apply the loading described by combining RLOAD1 entries 11 and 12, both with a scaling factor of 1.0.

RLOAD1 number 11 applies DAREA 20 (the x load) and uses TABLED4 number 111 to describe the frequency content of the load.

RLOAD1 number 12 applies DAREA 30 (the y load) with a phase angle of 90 degrees (DPHASE set 40) and also uses TABLED4 number 111 to describe the frequency content of the load.

The frequency range of interest is from 0. to 50 Hz. Since 0 Hz is a static solution (not of interest), we will start at a frequency of 0.5 Hz and perform our analysis using a frequency increment of 0.5 Hz until 50.0 Hz is reached. The following FREQ1 entry describes this frequency range. (Actually it goes up to 50.5 Hz, just beyond the range of interest)

```
FREQ1, 1,.5,.5,100
```

In the interest of efficiency, a modal approach will be used for the solution. Modes up to 100 Hz will be obtained and used in the solution. The following EIGRL instructs the program to find those modes.

```
EIGRL  1     -1.    100.   40
```

The following input file combines all of these entries with the model definition to solve this problem.

```
ID    MSC-XL, MSC-NASTRAN
SOL 111 $ Modal Frequency Response Analysis
TIME 55
CEND
TITLE = Sample dynamic analysis model
SUBTITLE = Rotating force due to tire out of balance
set 999 = 358,471
DISP(phase) =  999
SUBCASE 1
method = 1
```

```
DLOAD = 1
METHOD = 1
FREQ = 1
BEGIN BULK.
include 'car.dat'
include 'springs.dat'
$
EIGRL   1     -1.    100.   40
$
DLOAD   1     1.     1.     11     1.     12
RLOAD1  11    20                   111
RLOAD1  12    30            40     111
DPHASE  40    358    2      90.
DAREA   20    358    1      3.
DAREA   30    358    2      3.
TABLED4 111   0.     1.     0.     100.
        0.    39.478 ENDT
FREQ1   1     .5     .5     100
ENDDATA
```

Input file 'car.dat' is the model car (courtesy of LAPCAD Engineering) and 'springs.dat' is the input file with the suspension model. The springs are modeled using CROD and CELASi elements and the shock absorbers are modeled using CVISC elements. The results of this run are shown in figure 3. As expected, the motion of the wheel is greater than the motion of the driver, however, it is possible that the magnitude of the motion at the driver's seat is not acceptable. The next sections will describe how to find which modes of the model are contributing to the solution, how to identify the design variables for optimization, how to evaluate the sensitivity of the responses to changes in the design variables, and (finally) perform optimization to attempt to minimize the response at the driver's seat.

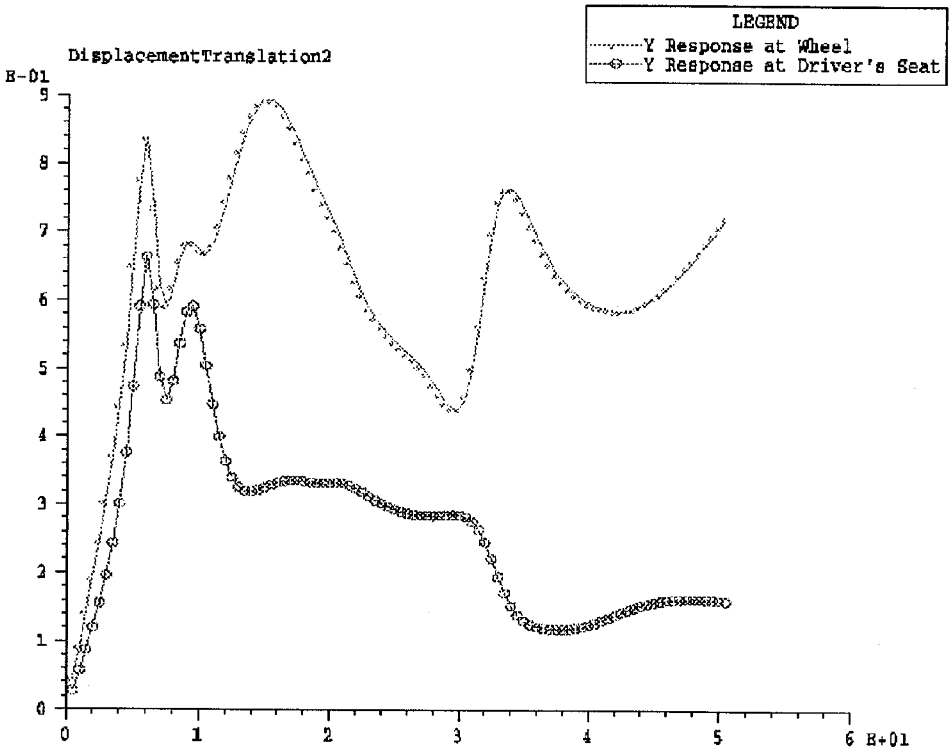

Figure 3 - Response of Initial Model

2. Evaluate Which modes are Dominant in the Response

This step is actually the recommended initial step in the problem. It is not recommended to start a dynamic analysis without first understanding the structure, it's modes, and how we expect the modes to contribute to the solution.

There are a number of tools used in dynamic analysis to assist in understanding the response. At this point, some of the tools provided with MSC/NASTRAN will be used to help understand how the car model responds to the rotating imbalance.

Using DMAP alters provided in the "sssalter" directory (delivered with MSC/NASTRAN since version 67.5) I will demonstrate how to better understand the response. The alters I will use are 1) checka.v68, 2) pchdispa.v68, 3) modevala.v68, and 4) mfreqea.v68. Each of these will be used in a separate run to demonstrate how each one works.

First a brief description of the sssalter directory. Beginning with version 67.5, an additional directory of files has been provided on the MSC/NASTRAN delivery tape. These alters are described as follows in the MSC/NASTRAN COMMON QUESTIONS AND ANSWERS[1].

> *"Miscellaneous alters, primarily for the Structured Solution Sequences (SSS). These alters provide new capabilities and error corrections. The alters were tested on several, but not all, Version 67.5 and 68 platforms, so you are cautioned about their use. . . . Note that these alters may not be maintained for future versions of MSC/NASTRAN; in other words, they are currently supplied for use only with Versions 67.5 and 68.*

"The alters in this directory are provided for illustrative purposes and have not been fully tested for all applications. Please exercise caution in their use."

A table, included in the above mentioned reference lists most of the DMAP alters provided in the directory with a description of what they do. Each alter has a set of files with it. The file names are used to describe the purpose of each file.

For example, If we were to look at the DMAP alter checka.v68, the following files are provided:

checka.v68 - the DMAP alter

check1.dat - sample input file to demonstrate how to use the alter

checkr.rdm - "read-me" file describing the DMAP alter and how to use it. This file will contain a list of reference papers (if they exist) which are available upon request from your local MSC office.

Results of checka.v68

The model was run using SOL 103 with the checka.v68 DMAP alter. In this run, the model was checked for any improper constraints, then, after the eigenvalue solution, the modal effective weight[2] was used to identify which modes might be important to the solution.

The chart shown in Figure 4 was created using the results from this run.

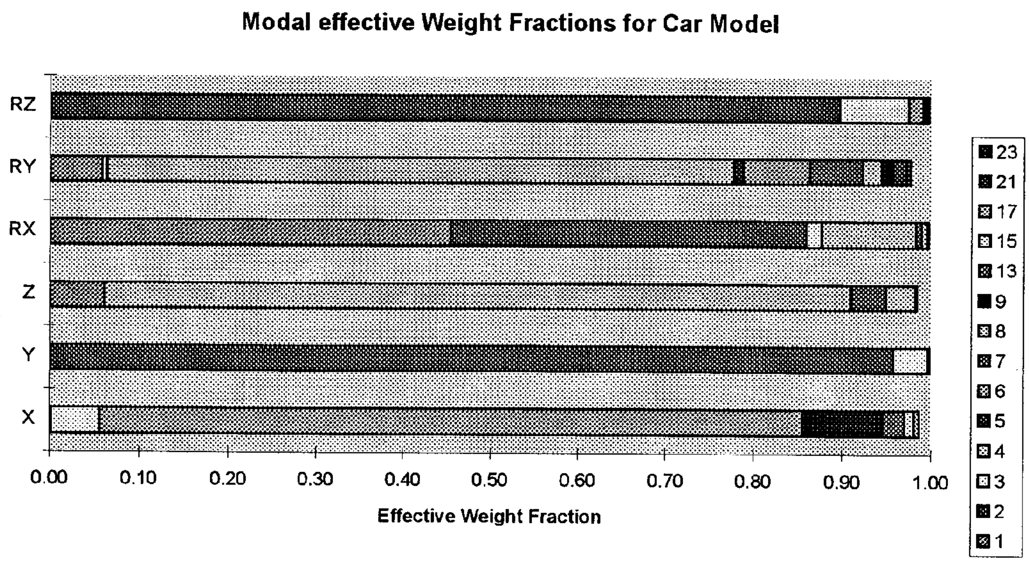

Figure 4 - Modal Effective Weight

The chart shows the modal effective weight of each mode as a fraction of the total weight of the system. Each bar represents either a translational weight of a rotational inertia. The chart has been edited to show only the modes which displayed significant values.

Looking at the chart helps us to identify "important" modes of the model. For example, by looking at the bar for the x-direction, we see that modes 1, 3, 6, 9, and 13 have noticeable contributions, with mode 6 being the dominant mode in that direction. If we look at the y-direction, we see that mode 2 is dominant with some contributions from mode 3.

Since the input loading is in the x- and y-directions, we may now assume that modes 2,3, and 6 will be dominant in the response.

However, there is an old expression in dynamics: "If the modes can represent the static solution to an applied loading, they should be adequate to represent the solution if that load is applied dynamically."

To evaluate this, we will use two of the DMAP alters in the sssalter directory, pchdispa.v68 and modevala.v68[3]. First a static solution will be run applying the loads statically. The results will be "punched" to a file by the alter pchdispa.v68. Those results will be used as input in SOL 103 with the modevala.v68 alter and we will see how well the modes can represent the static solutions.

Results of modevala.v68

Figures 5 and 6 show the results of those runs.

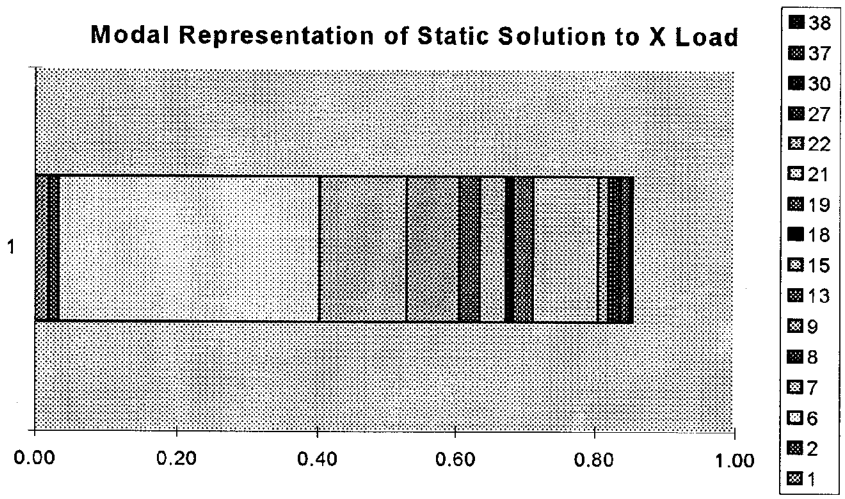

Figure 5

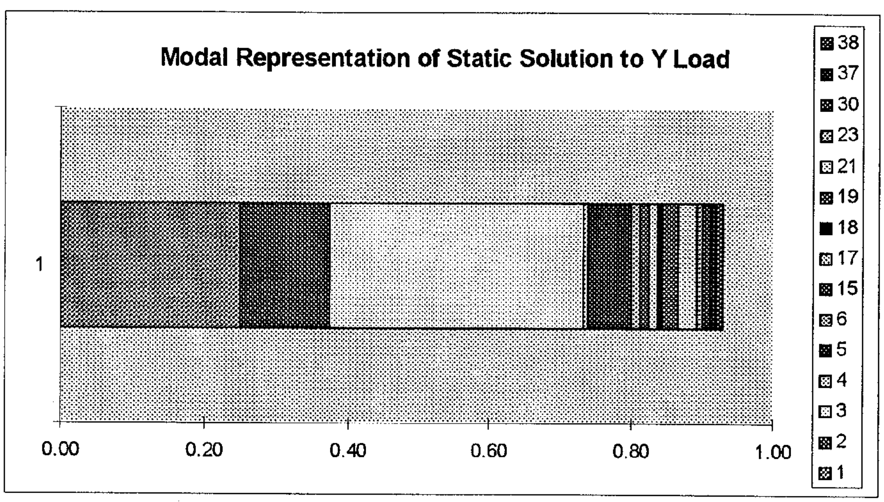

Figure 6

The output from the run informs us that the modes are capable of representing 90% of the static solution to a load in the x-direction and 96% of the static solution to a load in the y-direction. What fraction is acceptable is a personal choice for each engineer. Since this is a demonstration problem, no judgments will be made as to whether more modes should be obtained or not.

Once again, only the modes with the largest contributions are shown in the figures. From the results of the x-direction loading (shown in Figure 5), we see that mode 6 is the dominant mode, with modes 7,9, and 12 also contributing. For the y-direction loading (shown in Figure 6), we see that modes 1 and 3 are dominant in the response to a y-direction loading.

Based on this, we expect that modes 1, 3, and 6 will be the dominant modes in the final solution.

Now that we have a good idea of which modes will be dominant in the solution, we might begin to select design variables for optimization based on information we can obtain using strain energy contributions for elements obtained from SOL 103. However, one more step should be performed before the optimization process begins. That is the dynamic analysis under loading should be performed to verify which modes are dominant in the actual dynamic solution. Once again a DMAP alter from the sssalter directory will be used. This alter is mfreqea.v68[4]. This DMAP alter will determine how much each mode contributes to the total strain energy in the solution at each input frequency. This will enable us to determine which modes are important in the actual dynamic solution. At this point, the initial input file is submitted with the DMAP alter included. Figures 7 and 8 show the results of the DMAP alter.

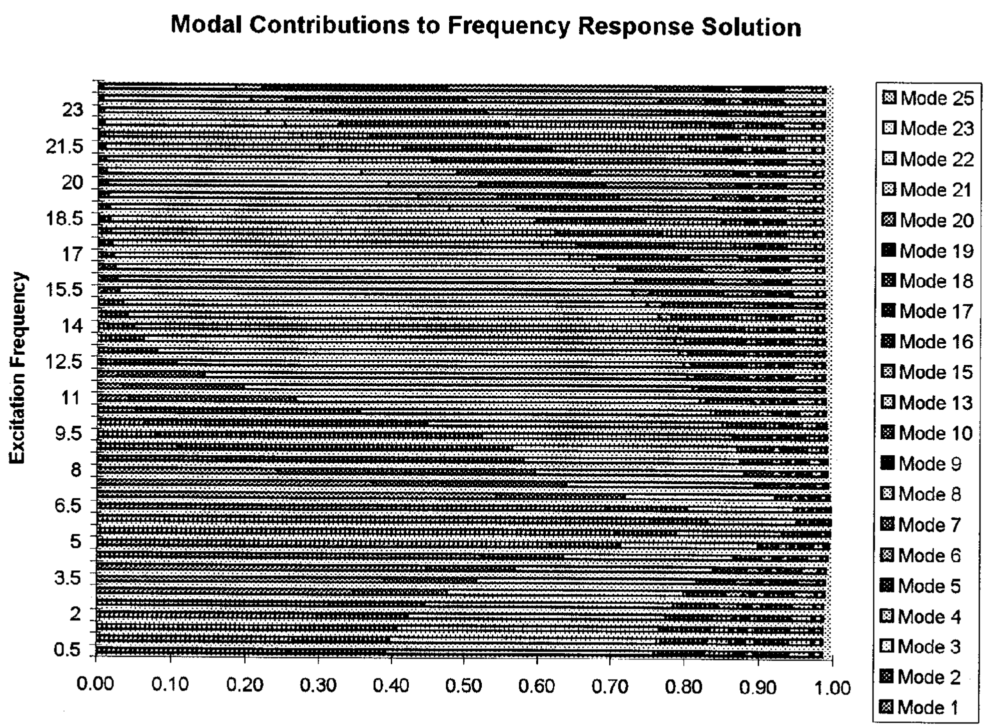

Figure 7

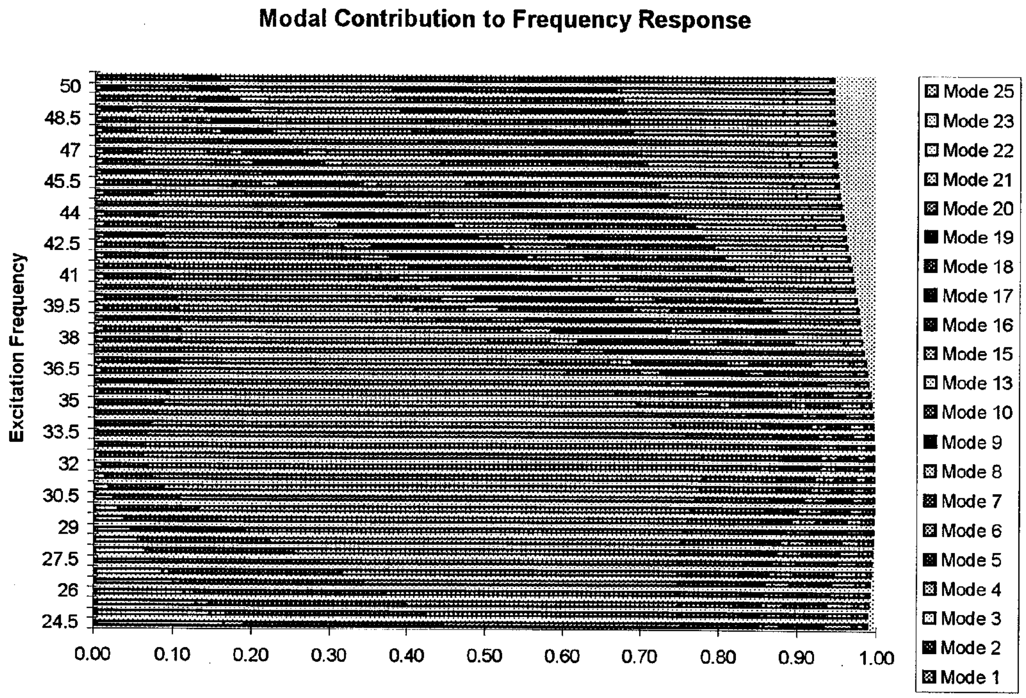

Figure 8

From Figures 7 and 8, we determine that modes 1 and 3 are dominant in the lower frequency region and mode 6 is dominant in the higher frequencies. Other modes contribute to the response, but these appear to be the dominant modes.

Selection of Design Variables for Optimization

Dynamic analysis is an art, not a science. This statement is a simple summary of dynamics. When performing dynamic analysis, nothing is obvious. Normally, to reduce dynamic response one would increase the damping of the shock absorbers, or try adjusting the spring stiffness. The initial optimization run in this paper does just that, and arrives at a reasonable design. However, it is possible that, with a little insight, we might improve on that design. This will be tried after the initial optimization.

3. Design Sensitivity

At this point, we have the response based on the original design. Assuming that the response levels are more than is acceptable, we need to modify the design. The question is, "should we increase or decrease the stiffness of the springs and the damping of the shock absorbers?" A sensitivity analysis will give us valuable information as to which is better. The following input will perform the sensitivity analysis.

```
ID    MSC-XL, MSC-NASTRAN
SOL 200
TIME 5500
CEND
TITLE = Sample dynamic analysis model
SUBTITLE = Rotating force due to tire out of balance
```

```
LABEL = perform optimization to minimize driver response
set 999 = 358,471
DISP(phase) =  999
spc = 99
SUBCASE 1
  ANALYSIS = MFREQ
  DESSUB = 100 $ constraints
  DESOBJ(min) = 300 $ design objective - minimize driver response
  DLOAD =  1
  METHOD =  1
  FREQ  =  1
BEGIN BULK
PARAM.OPTEXIT,4                $ Calculatc sensitivites and exit
spc1,99,123,1402,1825,1358,1869
doptprm,desmax,25              $ allow as many as 25 Design Cycles
include 'car.dat'               $ car model
include 'springsfm.dat'         $ suspension model
$
$ data for design sensitivity
$
$ define design variables
$
desvar,1,frntdamp,10.,1.,100.
desvar,2,reardamp,5.,1.,100.
desvar,3,frntstif,10.,4.,20.
desvar,4,rearstif,8.,4.,20.
$
$ relation between properties and variables
$
dvprel1,101,pvisc,2001,3,1.,,,,+dv101
+dv101,1,1.
dvprel1,102,pvisc,2002,3,1.,,,,+dv102
+dv102,2,1.
dvprel1,103,prod,1001,4,400.,,,,+dv103
+dv103,3,100.
dvprel1,104,prod,1002,4,400.,,,,+dv104
+dv104,4,100.
$
$ select displacemcnt Y at driver seat and wheel mount point as
$  response quantities
$
$ wheel mount point - no frequency is spccified, indicating that all frequencies are used
$
dresp1,200,disp,frdisp,,,2,,358
$
$ define driver's seat disp as a responsc - a separate dresp1 is entered for each input frequency
$
drcsp1,201,driver,frdisp,,,2,.5,471
=,*(1),=,=,=,=,=,*(.5),==
=(49)
$
$ add constraints
$
$ require that maximum tire displacement be .5 inches
```

```
$
dconstr,101,200,-.5,.5
$
$ require that maximum driver displacement be less than .25 inches
$ (over the lower half of the frequency range)
dconstr,102,201,-.25,.25
=,*(1),*(1),==
=(49)
$
$ combine constraints into set 100
$
dconadd,100,101,102,103,104,105,106,107,+dc100a
+dc100a,108,109,110,111,112,113,114,115,+dc100b
*(1),*(8),*(8),*(8),*(8),*(8),*(8),*(8),*(8),*(8)
=(3)
+dc100f,148,149,150,151,152
$
$ define objective = minimize srss of response over the first half of the input range
$
dresp2,300,srss, 301,,,,,,+dr300a
+dr300a,dresp1,201,202,203,204,205,206,207,+dr300b
+dr300b,,208,209,210,211,212,213,214,+dr300c
*(1),=,*(7),*(7),*(7),*(7),*(7),*(7),*(7),*(7)
=(4)
+dr300h,,250,251
$
deqatn 301     resp(a,b,c,d,e,f,g,h,i,j,k,l,m,n,o,p,
     q,r,s,t,u,v,w,x,y,z,aa,bb,cc,dd,ee,ff,gg,hh,ii,
     jj,kk,ll,mm,nn,oo,pp,qq,rr,ss,tt,uu,vv,ww,xx,yy)=
     sqrt(a**2+b**2+c**3+d**2+e**2+f**2+g**2+h**2+
     j**2+i**2+l**2+m**2+n**2+o**2+p**2+q**2+r**2+
     s**2+t**2+u**2+v**2+w**2+x**2+y**2+z**2+
     aa**2+bb**2+cc**2+dd**2+ee**2+ff**2+gg**2+
     hh**2+ii**2+jj**2+kk**2+ll**2+mm**2+nn**2+oo**2+
     pp**2+qq**2+rr**2+ss**2+tt**2+uu**2+vv**2+ww**2+
     xx**2+yy**2)
$
$ end of optimization input
$
```

In the preceding input, the DESVAR entries are used to define four design variables. These variables are named frntdamp, reardamp, frntstif, and rearstif. On each entry, a starting value is given and a minimum value and maximum value is provided.

DESVAR entries are not sufficient by themselves, there needs to be a relationship provided between the design variables and the properties of the model. This relationship is provided by the DVPREL1 entries. Each DVPREL1 describes how a design property (for example, the spring stiffness) is related to the design variables. For example, DVPREL1 number 103 states that the area of PROD number 1001 (the front springs) is 100 times design variable number 3 and has an initial value of 400.0.

Now that we have defined the design variables and their relation to the model, it is time to define the response quantities of interest. these may be almost any response quantity of the model, or a combination of responses. In our case, we will use the magnitude of the displacement at the wheel and at the driver's seat as responses we are interested in. These are defined by using DRESP1 entries. Since there are 100 input frequencies, we may choose the response at any set of these frequencies, or all of them. Since we wish to minimize the "overall" response at the driver's seat, a response quantity is chosen at each frequency. At this point, one might ask, "Why not use the frequency with the largest response and ignore the others?". This seems reasonable, until you remember that this is a dynamic response analysis. If we were to minimize the response at 4.0 Hz (where the peak occurs), the simplest way to do this is to simply shift the peak response to another frequency, perhaps 8.0 Hz, by shifting the resonant frequency of the underlying mode. Although the response at 4.0 Hz may be minimized by doing this, it is possible that the new peak may be larger than the old peak. Therefore, the response over a range of input frequencies is chosen as our desired set of response quantities.

Now we need to define constraints and a design objective. These are required by the optimizer (even although we are not performing the optimization yet). Our constraints will simply be limits on the magnitude of the displacements at the driver's seat. Let us assume that there is a requirement that the motion at the driver's seat must be less than .25 inches. In order to apply this constraint to all of the response variables representing the driver's seat motion will need to be constrained. These constraints are defined by DCONSTR entries 102 through 152. Each DCONSTR entry points to a response quantity at a specific frequency and the minimum and maximum values that quantity may have. In this case, $\pm$ 0.25 inches for each driver's seat response quantity.

We will also constrain the wheel motion not to exceed .5 inches. This is done with DCONSTR 101.

All of the DCONSTR entries are combined into set 100 with a DCONADD entry. Although all of the DCONSTR entries could belong to the same set, for this example, each belongs to a separate set, therefore, the DCONADD is used to combine them into one set.

As for the design objective, we would like to instruct the program to minimize the maximum displacement, but that is not possible at this time. Only one design objective is allowed, and that objective may not change during the optimization (remember, as the model is modified, the excitation frequency with the peak response may shift). Therefore, we will instruct the program to minimize the Square Root of the Sum of the Squares of the responses to input frequencies between .5 and 25 Hz. This will require the use of a DRESP2 and a DEQATN entry. The DRESP2 is used to define a response which is a combination of individual responses. The DRESP2 points to a DEQATN, which provides the equation to define the function.

Since we want the program to calculate the sensitivities and stop, PARAM,OPTEXIT,4 will be used in this run.

Figure 9 shows a part of the results from this run.

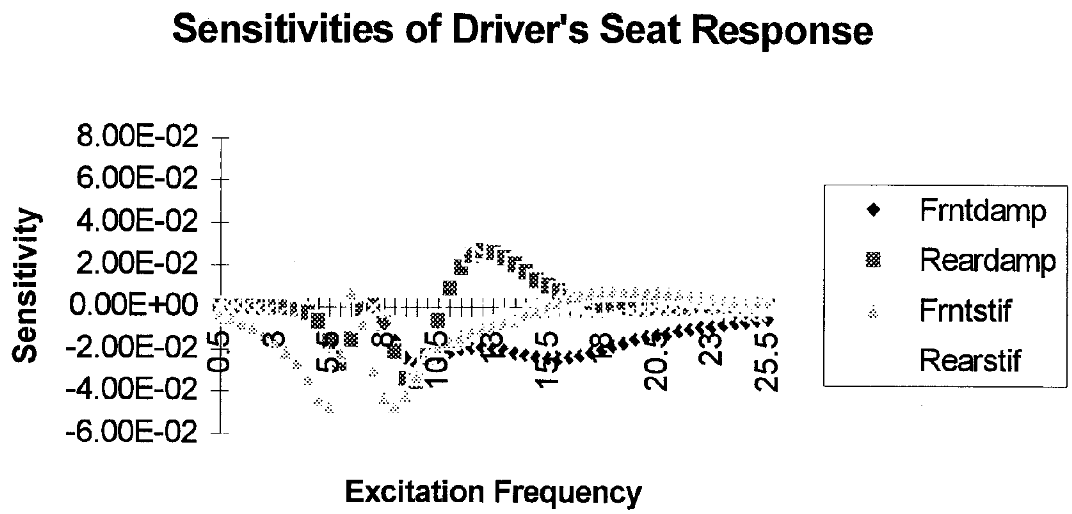

Figure 9

From this, we see that the change in the response (sensitivity) changes as the excitation frequency changes. It is obvious that increasing the damping of the front shock absorbers will decrease the response across the frequency range, but increasing the damping of the rear shock absorbers will actually increase the response at the driver's seat a number of frequencies. Changing the spring stiffnesses also will either increase or decrease the response, depending on the excitation frequency.

4. Dynamic Optimization

Now it is time to allow the optimizer to improve the design. The input file shown above is ready to perform the optimization in SOL 200. All that has to be done is the PARAM,OPTEXIT,4 entry should be removed. Now the program can perform the optimization

Figure 10 shows the responses before and after the optimization. The optimizer took 12 design cycles to converge.

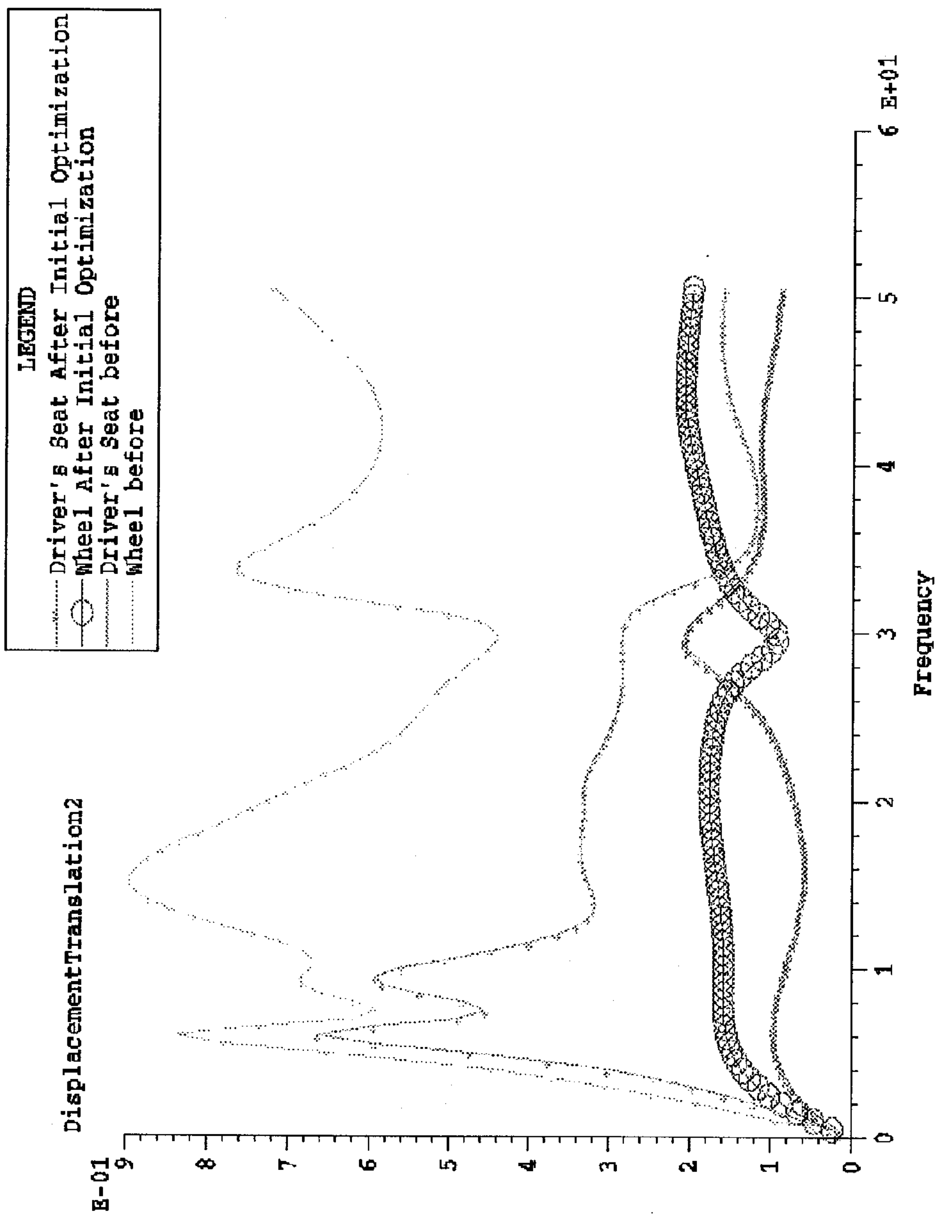

Figure 10 - Dynamic Response Before and After Optimization

At this point, we have improved significantly on the original design, but the question arises. Could the results be better if we used the information we obtained from the earlier runs.

5. Improved Optimization Using Information we Obtained Earlier

From Figures 7 and 8 we determined that modes 1, 3, and 6 appear to be the dominant modes in the response. We can now use that information to improve on the optimization analysis. Originally the vertical spring stiffnesses and the damping rates of the shock absorbers were used as design variables. Are there any other design variables we could use to improve the design?

A simple method to determine which areas of the model are the best to change to modify a mode is to look at the element strain energy output which we can obtain from SOL 103 by requesting ESE=ALL in the Case Control.

Using this simple rule and looking at the element strain energy results for modes 1, 3, and 6, it was determined that not only the vertical spring rates are important, but also the horizontal spring rates in the x-direction (forward). The optimization was run once more adding the x-direction stiffness of the suspension as design variables. The sensitivities of the response to these new variables (Frntstfx and Rearstfx) are shown in figure 11.

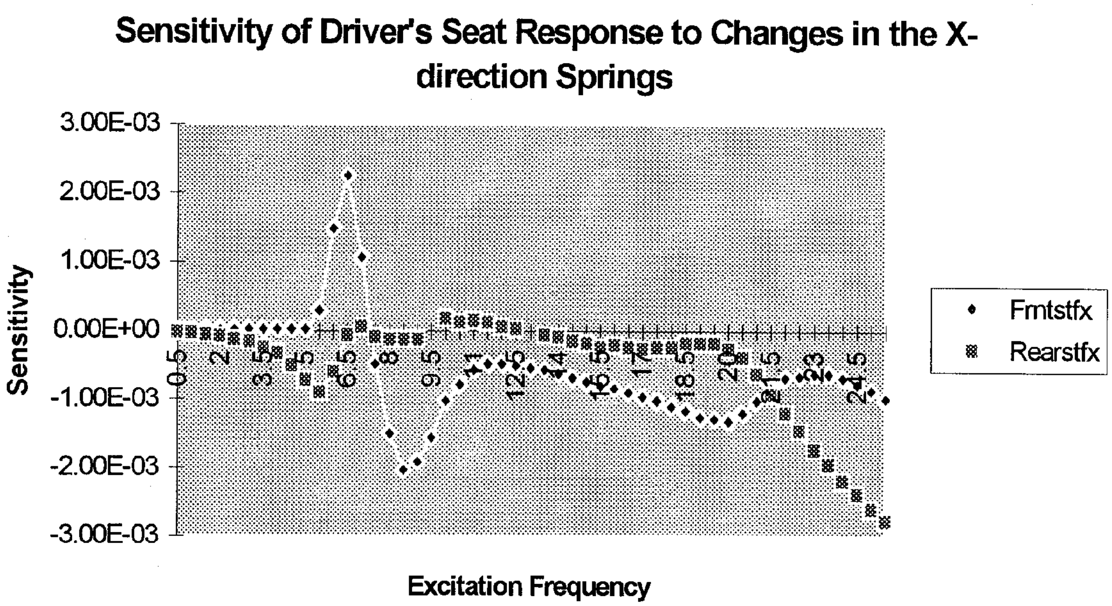

Figure 11

Once again, the sensitivities depend on the excitation frequency. The results of the optimization analysis including these two additional design variables are shown in figure 12.

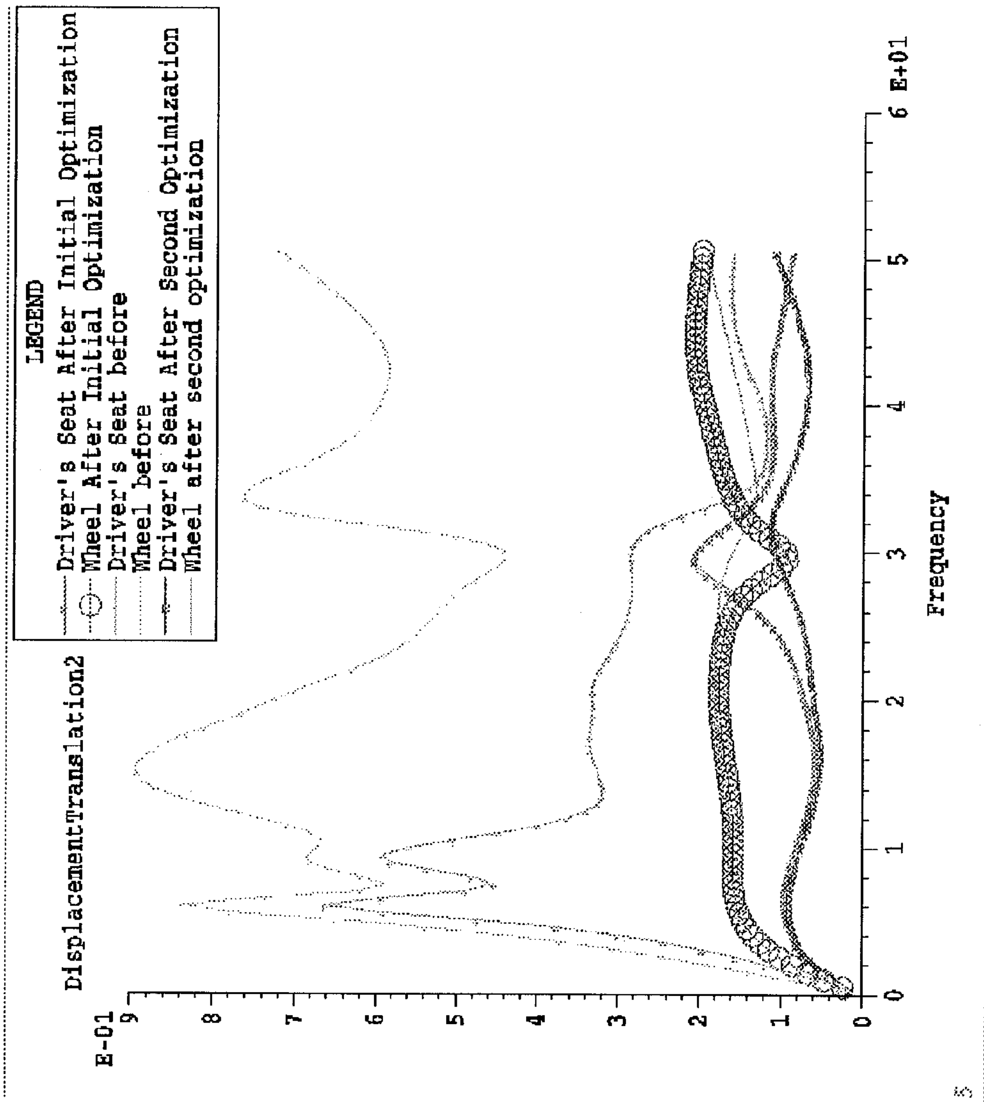

Figure 12

From Figure 12, we can see that the addition of the two additional design variables has improved the final design. The response of the driver's seat has been reduced across the entire frequency range of interest, which is impressive, when one considers that the design objective was only in the lower half of the frequency range.

Conclusions

The goal of this paper is to demonstrate a number of the capabilities of MSC/NASTRAN. Those demonstrated include dynamic analysis (normal modes and frequency response), a number of "tools" provided with MSC/NASTRAN to assist in understanding the dynamic behavior of the structure (the sssalter directory), and the optimizer, which improved on the original design with a minimum of effort from the user.

The goal of reducing the response at the driver's seat to the input excitation achieved across the frequency range of interest. A reasonable insight into the dynamic behavior of the model was obtained, and by using this insight, the response reduced even further.

REFERENCES:

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