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6. AUTHOR(S)
-HAYES HOBOLTH
-HUGH LARSEN
-TOM JOHNSON
-CHARLIE PARKER

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
MICHIGAN SCIENTIFIC CORPORATION
730 BELLEVUE AVENUE
MILFORD, MI 48381

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14. ABSTRACT
EVALUATIONS OF CANDIDATE DRIVESHAFTS FOR THE FMTV PROGRAM FROM DANA, ARVIN MERITOR (2 DESIGNS), AND GKN WERE CONDUCTED. THE EVALUATIONS INCLUDED INSTRUMENTED ROAD TESTING, A 1000-MILE DURABILITY TEST THAT INCLUDED HIGH SPEED DRIVING AND SUBMERSION, LABORATORY MEASUREMENTS, AND DADS MODELING. THE DRIVESHAFTS WERE (UN)BALANCED TO THEIR MANUFACTURERS PRODUCTION LIMITS PRIOR TO EVALUATION AND THEN TESTED ON THE VEHICLE USING A SPECIAL TRANSFER CASE TRANSDUCER CAPAPBLE OF MEASURING DRIVESHAFT DYNAMIC FORCES. A PRINCIPLE FOCUS OF THE PROGRAM INCLUDED SEVERAL METHODS OF MEASURING INSTALLED CRITICAL SPEED.

15. SUBJECT TERMS
FMTV, LMTV, DRIVESHAFT, CARDAN, CV, CONSTANT VELOCITY, FLYWHEEL HOUSING, BELL HOUSING, CRITICAL SPEED, DRIVELINE, DRIVELINE DYNAMICS, DRIVELINE MODEL, POWERPACK, POWERPACK DYNAMICS, TRANSFER CASE TRANSDUCER, DRIVESHAFT BALANCE, DADS MODEL.

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Evaluation of

Alternative Driveshaft Configurations

For the

Family of Medium Tactical Vehicle

(FMTV) Program

TACOM Contract Number DAAE07-98-C-M012

November 8, 2002

Michigan Scientific Corporation

Tom Johnson
Hayes Hobolth
Hugh Larsen
Charles Parker

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Overview

Background:

The FMTV provides a challenging environment for driveshaft applications. Long driveshafts are of concern because of critical speed issues. Large operating angles on cardan joint driveshafts are also of concern because of torsional acceleration issues. Both concerns are exacerbated by high driveshaft rotational speeds. The FMTV has driveshafts that exceed the limits of the traditional design envelopes on these three design features: length, operating angle, and rotational speed.

The FMTV was designed to meet a wide array of state-of-the-art performance requirements that expanded the mobility and utility of the vehicle, but also led to the need to push components and sub systems to the limits of practibility. Some of the performance requirements that challenged the design include:

- The FMTV was required to meet the low speed/high torque requirements of the cooling test. It also had to meet high speed trailering and grade climbing requirements. These combined performance expectations drove the need for a seven speed automatic transmission. The seven-speed Allison MD-D7 transmission with two overdrive gears (derived from a production six speed transmission) was developed in order to meet these requirements.
- The FMTV trucks are required to climb up a twenty-four inch high vertical step. This requirement leads to the need for a high clearance transfer case. The only way to connect the high clearance transfer case to center input axles is with large driveshaft angles that exist some of the FMTV driveshafts.
- The vehicle is required to operate on a 60% grade. Tilting the engine slightly off horizontal helped the oil lubrication performance in this condition. This tilt, also improved ground clearance for the step test and off-road performance. This tilt, however, increased the rear driveshaft-operating angle.

The original design of the FMTV as provided by Steyr at the beginning of the FMTV program, used double cardan constant velocity driveshafts in the front positions on all vehicles and on the LMTV rear position. The design experienced durability issues associated with these driveshafts on test vehicles. This prompted a change to single cardan joint driveshafts.

The Society of Automotive Engineers (SAE) publishes a design guide for driveshafts called: Advancements in Engineering No. 7 (AE-7) "Universal Joint and Driveshaft Design Manual". On pages 270 and 271, this manual states, "...speed factors associated with vehicle application experience (for passenger cars and light duty trucks,...and medium and heavy duty trucks) are related to the following generally accepted (cardan joint) driveshaft installation parameters:

Universal Joint Angles	6 deg maximum continuous operating angle
Driveshaft Length	60 in maximum installed center to center distance of universal joints"

Additionally, in medium duty truck applications, the governed speed of diesel engines often limits the maximum rotational speed of the driveshafts. Most driveshafts in medium duty trucks have speeds limited by a governor, to less than 3000 rpm.

The FMTV has driveshafts that exceed these traditional guidelines.

- The universal joint operating angles on all FMTV front driveshafts and LMTV rear driveshafts exceed 10 degrees.
- The LMTV rear driveshaft has an installed length of 61.5 inches
- The FMTV powerpack has a transmission with two overdrive speeds, resulting in a driveshaft speed that is 1.28 times engine speed in seventh gear which is over 3300 driveshaft rpm at the engine governed speed of 2600 rpm.

SAE AE-7 goes on to state that, "Successful applications exceeding [these] parameters have been and are continuing to be made". However, when designs are released with parameters that lie outside generally accepted common practice, the effort in validating these designs must be ever more vigilant.

Rzeppa style constant velocity universal joints have undergone expanded vehicle usage since the time of the original FMTV designs. Millions of units of annual production of Rzeppa style universal joints are currently manufactured for half-shafts on front wheel drive vehicles. Increased usage in driveshafts in cars and trucks is occurring as well. US military vehicles that use this type of universal joint include the HMMWV and MTVR. European military vehicles also commonly use this design for driveshafts. Driveshaft application was not common at the time of the FMTV original design.

Rzeppa style joints are routinely used with articulation angles well in excess of 35° on the steer axles of front wheel drive cars and trucks. They are also used in high-speed (greater than 10,000 rpm) racecar driveshafts. For wheel speed applications (usually well under 2000 rpm), convoluted boots are used for joint protection. High-speed driveshaft applications (usually well over 5000 rpm) usually use single fold can-style boots. Single fold can-style boots usually have joint angles limited to less than 8°. Convoluted boots usually have rpm limited to less than 3000 rpm.

The FMTV driveshaft pushes a can-style boot beyond its traditional angle limit, or it pushes the convoluted style boot beyond its traditional rpm limit. The Rzeppa joint itself, has no difficulty with either the angle or the rpm.

In the field, the FMTV has experienced a very limited number of powerpack and driveshaft incidents in vehicles with retrofitted A1 driveshafts. These incidents have generally had minor vehicle, occupant, and bystander involvement. However, costly powerpack and driveshaft repairs and vehicle downtime have occurred in some of these vehicles. These incidents, combined with an engineering review of design opportunities to take advantage of state-of-the-art hardware, encouraged this program to evaluate alternative driveshaft designs.

Candidate driveshafts from three suppliers were evaluated. Arvin Meritor (ARM) supplied a single cardan candidate driveshaft designated by their series number RPL-20, as well as the single cardan A1 driveshafts, which were included in the program to measure baseline performance. Dana supplied their SPL-140 series single cardan driveshafts. GKN supplied driveshafts that were constant velocity type designs for all positions except the MTV rear-rear, which was a single cardan Kempf design.

At the conclusion of testing, driveshafts were returned to the suppliers for inspection and analysis. Each manufacturer wrote a detailed report on their individual results. These reports are available from TACOM to approved individuals for more detailed investigations. See the PM-FMTV to request copies. The driveshafts were returned to MSC for storage. They are also available for inspection by individuals approved by PM-FMTV.

The testing plan is covered in detailed in the section titled "Testing Overview" beginning on page 19. The testing included 1000 miles of driving for each submission on the MTV and FMTV, under conditions severe to the driveshafts and powerpack. Emphasis in the testing was on recorded data in the vehicles that included driveshaft forces and powerpack accelerations, temperatures, and strains. This was done to enable objective measurement of the influence of each driveshaft submission on the vehicle.

This report covers only technical and engineering aspects of the driveshaft applications relating to information derived from this project only. It is recognized that final selection must also consider business, cost, supply, manufacturing and other issues as well.

General Comments on Test Results

A primary focus of this program was to down select from three candidate driveshaft concepts to a single one. In an ideal environment, more than one supplier would meet all requirements, and the selection could be left to business issues. However, there was no candidate driveshaft submission that clearly met all of the test conditions in all of the positions on the vehicle. Thus, no single supplier could be recommended for production without additional development work. Some of the submissions had better performance in certain vehicle positions than others, but no submission had overall satisfactory performance in all positions. For this reason, the driveshafts in this report are ranked from highest to lowest for each position based on test performance, with comments on likely development that would be required to obtain satisfactory performance on the test.

Single Cardan Joint Driveshafts

Single cardan joint driveshafts inherently produce torsional accelerations. The amount of acceleration is determined solely by operating angle and rpm, which cannot be influenced by driveshaft design. A measure of the oscillating torque produced by this acceleration is the product of torsional acceleration and torsional inertia of the driveshaft. The oscillating torque therefore increases proportionally, as the driveshaft torsional inertia increases. The term "torsional inertia" is used to describe the mass moment of inertia about the rotating axis of the driveshaft. See the section titled "Driveshaft Inertial Acceleration" on page 52 for more detailed information on this topic.

Both cardan joint submissions for this program had substantially higher torsional inertia compared with the original AØ driveshafts that were tested and validated on the vehicle. The Dana SPL-140 submissions were approximately 41% higher, and the Meritor RPL-20 submissions were approximately 72% higher. Incidentally, the A1 driveshafts, which were tested as baseline for this program, had approximately 8% higher torsional inertia than the AØ driveshafts that were originally released on the vehicle.

It is not known what long-term effect these increased torsional vibration levels might have. The levels are well above SAE recommended practice. It is beyond the scope of this project to assess the durability effects of these higher torsional vibration levels, other than to quantify them. It would be expected that the manufacturers of components interfacing with the driveshafts, as well as the vehicle supplier, should address the issue of higher torques associated with higher driveshaft inertias before any decision to increase driveline inertias could be made.

SAE Guidelines limit driveshaft inertial accelerations to less than 1000 rad/sec^2. None of the cardan driveshafts on the vehicle meets this criterion. SAE Guidelines allow inertial accelerations as high as 2000 rad/sec^2 for certain low inertia driveshafts. The MTV intermediate and rear-rear driveshafts have inertial accelerations at, or below this level. Submissions for the front driveshaft position and the LMTV rear position exceeded this criterion, as well.

For this reason, the cardan joint driveshafts with higher inertias are not recommended for the front driveshaft position or for the LMTV rear position without additional vehicle development work.

Driveshaft Critical Speed

Driveshafts operated at or near critical speed can produce destructive vibrations in the vehicle. For this reason, SAE recommends that any driveshafts have a critical speed at least 33% higher than top vehicle speed. There has been much discussion about what is considered the top speed of the vehicle. Three numbers are often discussed: 1) Full load governed speed is 57 mph. 2) Speed on a typical expressway has been recorded at 63 mph. 3) Down hill speed on a 7% grade can reach 70 mph. The 133% limit recommended by SAE would be 1) 76, 2) 84, and 3) 93 mph, respectively, for each of these speeds. With this background, 90mph was presented early in this program as a desired driveshaft critical speed.

The testing revealed that the actual mph value indicated for critical speed in a vehicle can vary through a range of 2-4 mph since conditions such as driveline torque (accelerating, coasting, or braking), temperatures, driveshaft hinging, and spline lock, effect the actual critical speed. Additionally, the resonance condition that multiplies driveshaft unbalance forces to destructive levels begins to occur at speeds well below critical. Forces higher than would be expected from simple unbalance, begin at speeds 10-20 mph or more below critical speed.

The driveshaft position with a critical speed issue is the LMTV rear. The Arvin Meritor RPL-20 LMTV Rear driveshaft does not meet the 90 mph criterion. It cannot be recommended without additional development.

The GKN and Dana LMTV rear designs meet this criterion. They can be considered satisfactory from a critical speed perspective.

Water Intrusion

The water intrusion exposure on this test was very severe, with immersion in water for one minute every 55 miles. Water Intrusion performance by manufacturer and by vehicle position were varied.

All manufacturers provided spline-sealing systems that experienced no water intrusion. The Arvin Meritor A1 spline was not sealed and experienced water intrusion and grease washing.

All manufacturers provided universal joint sealing systems that completed the testing with no water intrusion. The A1 universal joints exhibited evidence of water inside universal joint seals.

The boots on the GKN Fixed-Fixed Joints (On all CV joints except the LMTV Axle Joint) were inadvertently provided with two air vents on each boot, and experienced significant water intrusion. Conventional sealed systems, with venting to the driveshaft tube, would be expected to eliminate this problem.

The boot protecting the GKN Plunge Joint (LMTV Rear Axle Joint) ruptured at about 870 miles into the test. It is expected that alterations to the boot design would be required before this problem would be eliminated.

Maintenance and Serviceability

Field experience has demonstrated that serviceability is an important aspect of the design of FMTV driveshafts. There have been a number of field incidents attributed to improper driveshaft removal and replacement. Certain vehicle service procedures like flat towing over 100 miles, lift towing, and transmission filter service, require driveshaft removal and replacement.

The driveshafts submitted for this program were considered "Easy Service" and "Permanent Lubrication" by their manufacturer.

Driveshaft service often involves removal and replacement of the driveshaft mounting bolts. Field replacement of attachment bolts may not involve the same scrutiny for torque and fastener quality as in the factory. Several incidents of driveshaft attachment irregularities in the field have been attributed to improper maintenance related driveshaft fastener installation. Therefore, attachment integrity and susceptibility to improper bolt installation in the field was considered.

The RPL-20 design submitted by Arvin Meritor was said by Arvin Meritor personnel to be less susceptible to bolt installation irregularities than the A1 design.

Dana provided no opinions on the susceptibility of the Dana design to improper bolt installation.

The design submitted by GKN was said by GKN to have redundant fasteners and that a missing bolt or improperly torqued bolt was judged to be not critical.

General Summary

FMTV Front position on all vehicles: The GKN submission would meet all test criteria with the development of boot venting systems that do not allow water intrusion.

LMTV Rear position: The GKN submission would meet all test criteria with the development of boot sealing and venting systems that do not allow water intrusion.

MTV Intermediate: The GKN submission would meet all test criteria with the development of boot seal systems that do not allow water intrusion. The Dana and Arvin Meritor submissions would require concurrence by the vehicle manufacturer and by the suppliers of adjacent components that the higher dynamic torques associated with higher rotational inertias in these submissions could be acceptable.

MTV Rear-Rear: All submissions had essentially similar performance.

Summary Charts of the Test Performance for each of the suppliers for each driveshaft location are included in Tables 1 to 4 below.

Table 1

Driveshaft Performance Summary

	LMTV Rear Driveshaft					
Over-all Rank	**Supplier Model**	**Measured Dynamic Radial Forces**	**Calculated Dynamic Inertial Torques**	**Water Intrusion**	**Serviceability**	**Summary and Comments**
NA	Arvin Meritor A0	Forces not measured 66 mph Critical Speed	184 ft-lbs @ 58 mph			
NA	Arvin Meritor A1 Baseline	+/-850 lb. 0-p at 58 mph 68 mph Critical Speed	199 ft-lbs @ 58 mph	Water found in Tube No Spline Seals Universal Joint Grease Possibly Discolored from Water	Most difficult submission to service: Bearing Caps must be pressed out Driveshafts must be rotated to properly torque bolts One Missing Bolt Critical	Baseline for Dynamic Radial Force Measurements Low Critical Speed Major Development Required to increase Critical Speed 8% increase above A0 inertial torques may have slight detrimental effect on adjacent components
NA	Arvin Meritor RPL-20	+/-1000 lb. 0-p at 58 mph 68 mph Critical Speed	317 ft-lbs @ 58 mph	Water found in Tube Splines and Universal Joints OK	"Easy Service" Four Bolts per End Bolts are accessable Missing Bolt Assessed by Supplier as less Critical than A1	Highest Radial Forces Low Critical Speed Major Development Required to increase Critical Speed 72% increase above A0 inertial torques may have detrimental effect on adjacent components
2	Dana SPL-140	+/-800 lb. 0-p at 58 mph 90 mph Critical Speed - est.	261 ft-lbs @ 58 mph	No Water found any Locations	"Easy Service" Four Bolts per End Bolts are accessable	Dynamic Radial Forces not significantly less than Baseline 41% increase above A0 inertial torques may have detrimental effect
1	GKN Constant Velocity	+/-400 lb. 0-p at 58 mph 100+ mph Critical Speed -est.	No Inertial Acceleration and Torque	Transfer Case Boot not Sealed Axle Boot Failed	"Easy Service" Six Bolts per End Bolts are accessable Missing Bolt Assessed by Supplier as Not Critical	Only Submission with significant improvement in Dynamic Forces and Inertial Accelerations and Torques Requires Lab Test and Road Test to Validate seals

Table 2

Driveshaft Performance Summary						
	All FMTV Front Driveshafts					
Over-all Rank	**Supplier Model**	**Dynamic Radial Forces**	**Calculated Dynamic Inertial Torques**	**Water Intrusion**	**Serviceability**	**Summary and Comments**
NA	Arvin Meritor A0	Radial Forces not Measured	137 ft-lbs @ 58 mph			
NA	Arvin Meritor A1 Baseline	Radial Forces not Measured	148 ft-lbs @ 58 mph	Water found in Tube No Spline Seals Universal Joint Grease Possibly Discolored from Water	Most difficult submission to service: Bearing Caps must be pressed out Driveshaft must be rotated to properly torque bolts One Missing Bolt Critical	8% increase above A0 inertial torques may have slight detrimental effect on adjacent components
3	Arvin Meritor RPL-20	Radial Forces not Measured	250 ft-lbs @ 58 mph	Water found in Tube Splines and Universal Joints OK	"Easy Service" Four Bolts per End Bolts are accessable Missing Bolt Assessed by Supplier as less Critical than A1	72% increase above A0 inertial torques may have detrimental effect on adjacent components
2	Dana SPL-140	Radial Forces not Measured	172 ft-lbs @ 58 mph	No Water found any Locations	"Easy Service" Four Bolts per End Bolts are accessable	41% increase above A0 inertial torques may have detrimental effect on adjacent components
1	GKN Constant Velocity	Radial Forces not Measured	No Inertial Acceleration and Torque	Boots not Sealed Splines OK	"Easy Service" Six Bolts per End Bolts are accessable Missing Bolt Assessed by Supplier as Not Critical	Significant improvement in Inertial Accelerations and Torques Requires Lab Test and Road Test to Validate seals

Table 3

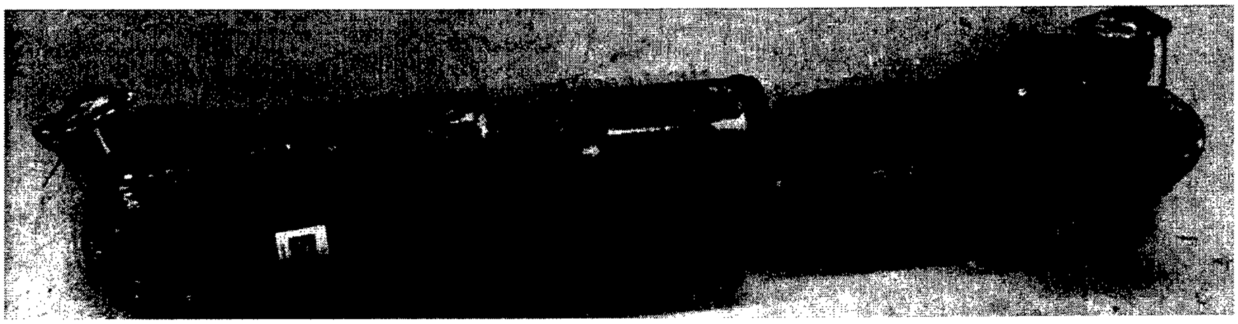
Driveshaft Performance Summary						
	MTV Intermediate Driveshaft					
Over-all Rank	**Supplier Model**	**Dynamic Radial Forces**	**Calculated Dynamic Inertial Torques**	**Water Intrusion**	**Serviceability**	**Summary and Comments**
NA	Arvin Meritor A0	Radial Forces not Measured	147 ft-lbs @ 58 mph			
NA	Arvin Meritor A1 Baseline	+/-600 lb. 0-p at 58 mph	159 ft-lbs @ 58 mph	Water found in Tube No Spline Seals Universal Joint Grease Possibly Discolored from Water	Most difficult submission to service: Bearing Caps must be pressed out Driveshaft must be rotated to properly torque bolts One Missing Bolt Critical	Baseline for Radial Forces - Second Lowest Measured 8% increase above A0 inertial torques may have slight detrimental effect on adjacent components
3	Arvin Meritor RPL-20	+/-800 lb. 0-p at 58 mph	264 ft-lbs @ 58 mph	Water found in Tube Splines and Universal Joints OK	"Easy Service" Four Bolts per End Bolts are accessable Missing Bolt Assessed by Supplier as less Critical than A1	Highest Radial Forces 72% increase above A0 inertial torques may have detrimental effect on adjacent components
2	Dana SPL-140	+/-750 lb. 0-p at 58 mph	191 ft-lbs @ 58 mph	No Water found any Locations	"Easy Service" Four Bolts per End Bolts are accessable	Dynamic Radial Forces higher than Baseline 41% increase above A0 inertial torques may have detrimental effect
1	GKN Constant Velocity	+/-400 lb. 0-p at 58 mph	No Inertial Acceleration and Torque	Boots not Sealed Splines OK	"Easy Service" Six Bolts per End Bolts are accessable Missing Bolt Assessed by Supplier as Not Critical	Only Submission with significant improvement in Dynamic Forces Requires Lab Test and Road Test to Validate seals Zero Inertial Torques

Table 4

Driveshaft Performance Summary						
	MTV Rear-Rear Driveshaft					
Over-all Rank	**Supplier Model**	**Dynamic Radial Forces**	**Calculated Dynamic Inertial Torques**	**Water Intrusion**	**Serviceability**	**Summary and Comments**
NA	Arvin Meritor A0	Not Measured	138 ft-lbs @ 58 mph			
NA	Arvin Meritor A1 Baseline	Not Measured Not Expected to be significant in this position	149 ft-lbs @ 58 mph	Water found in Tube No Spline Seals Universal Joint Grease Possibly Discolored from Water	Most difficult submission to service: Bearing Caps must be pressed out Tires must be rotated to properly torque bolts One Missing Bolt Critical	Approximate 8% increase above A0 Inertial Torques may have slight detrimental effect on adjacent components
1	Arvin Meritor RPL-20	Not Measured Not Expected to be significant in this position	251 ft-lbs @ 58 mph	Water found in Tube Splines and Universal Joints OK	"Easy Service" Four Bolts per End Bolts are accessable Missing Bolt Assessed by Supplier as less Critical than A1	Approximate 72% increase above A0 Inertial Torques may have detrimental effect on adjacent components
1	Dana SPL-140	Not Measured Not Expected to be significant in this position	261 ft-lbs @ 58 mph	No Water found any Locations	"Easy Service" Four Bolts per End Bolts are accessable	Approximate 41% increase above A0 Inertial Torques may have detrimental effect on adjacent components
1	Kempf Cardan by GKN	Not Measured Not Expected to be significant in this position	Not Analyzed Expected to be highest based on mass	No Water Probable - See page 39 for details	"Easy Service" Four Bolts per End Bolts are accessable Missing Bolt Assessed by Supplier as Not Critical	Inertial Torques Expected to be highest of all submitted driveshafts for this position based on measured driveshaft mass May have detrimental effect on adjacent components

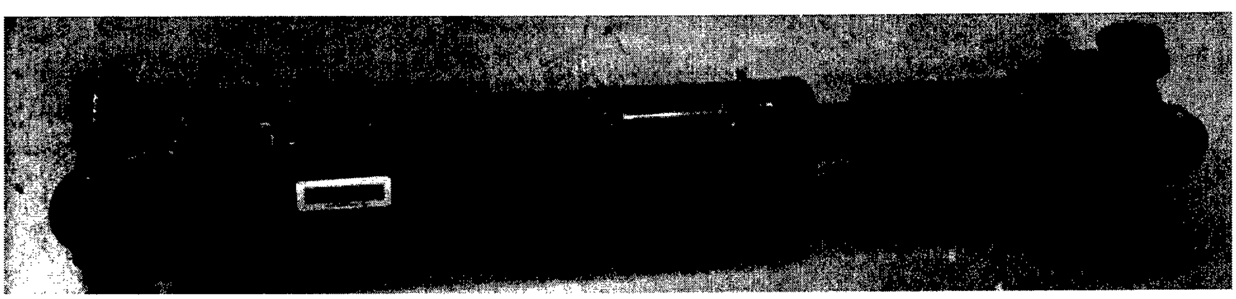
Conclusions for each Driveshaft Supplier and Design

Arvin Meritor A1 (Front Driveshaft is Illustrated)



- The Arvin Meritor A1 LMTV rear driveshafts reached transfer case dynamic forces that exceeded 2000 lbs., and exhibited characteristics representative of driveshaft critical speed at about 69 mph. The dynamic forces were the second highest measured on the four candidate driveshafts
- Water intrusion was noted inside the driveshaft tubes which degraded driveshaft balance and increased driveshaft dynamic transfer case forces
- Water intrusion was observed to discolor universal joint grease
- Brinelling of universal joint cross journals at needle bearing spacing was observed on several universal joint journals
- Water and washing away of grease was observed in the unsealed slip joints
- The driveshafts meet SAE published guidelines in the MTV intermediate and rear-rear positions. They do not meet SAE published guidelines in the front and LMTV rear positions

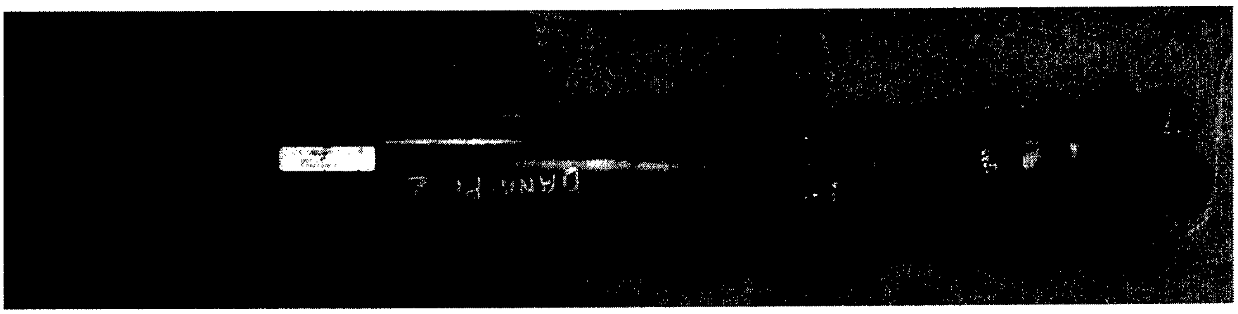
Arvin Meritor RPL-20 (Front Driveshaft is Illustrated)



- The Arvin Meritor RPL-20 LMTV rear driveshafts reached transfer case dynamic forces that exceeded 2000 lbs., and exhibited characteristics representative of driveshaft critical speed at about 68 mph. These were the highest dynamic forces measured on the four candidate driveshafts
- Water intrusion was noted inside the driveshaft tubes which degraded driveshaft balance and increased driveshaft dynamic transfer case forces

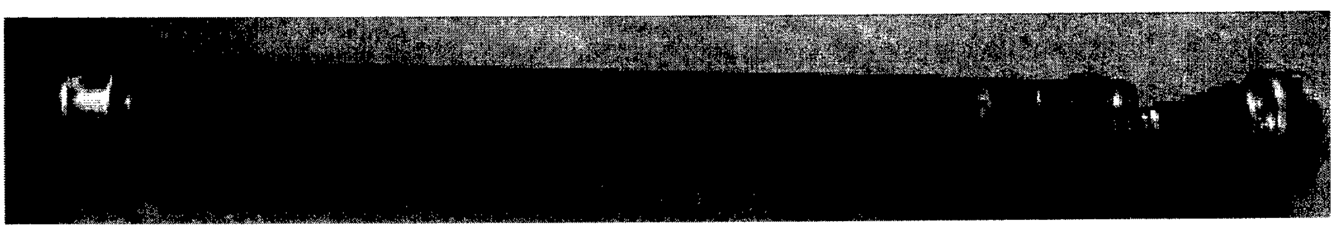
- No water intrusion was noted in the universal joint grease. However, some seal damage was noted on the universal joint seals.
- The driveshafts can meet SAE published guidelines in the MTV intermediate and rear-rear positions. They do not meet SAE published guidelines in the front and LMTV rear positions

Dana SPL-140 (Front Driveshaft is Illustrated)



- The Transfer case dynamic forces were second lowest of the tested driveshafts
- No test incidents were observed
- The driveshafts had little to no degradation observed in the post-test teardown
- The driveshafts can meet SAE guidelines in the MTV intermediate and rear-rear positions. They do not meet SAE published guidelines in the front and LMTV rear positions

GKN (LMTV Rear Driveshaft is Illustrated)



- The Transfer case dynamic forces were lowest of the tested driveshafts
- The driveshafts had no incidents relating to driveshaft integrity on the test
- Water intrusion was observed in all universal joint positions in the CV joint universal joints
- The LMTV rear driveshaft experienced a ruptured boot
- No water entry was observed in the Kempf rear-rear cardan joint positions
- One of the five slip joints tested (the MTV rear-rear) experienced beads of water intrusion. The supplier reviewed this observation in a post-test report and indicated that the water entry was likely due to improper treatment of the driveshaft after initial manufacturing due to a requirement to adjust the length of the driveshaft. See the supplier's report (available, with approval, from TACOM) for additional details
- The supplied CV driveshafts met SAE published guidelines in all positions.

Recommended Driveshafts in Rank Order

LMTV Rear Driveshaft

1. The GKN driveshaft is conditionally recommended first choice.
   - This was the only design that meets SAE guidelines for both Critical Speed and Inertial Acceleration
   - This driveshaft had the lowest transfer case dynamic forces of all designs tested. This is attributed to:
     - High critical speed of the driveshaft
     - Low unbalance forces in the ISO standard
     - Absence of Inertial Acceleration in a CV design
   - GKN must demonstrate acceptable boot performance before the conditional provision is lifted.
2. The Dana driveshaft is conditionally recommended as second choice
   - This design meets SAE guideline for Critical Speed
   - This design does not meet the SAE guideline for Inertial Acceleration
   - Concurrence required by the vehicle manufacturer and by the suppliers of adjacent components that the higher dynamic torques associated with higher rotational inertias in this submission could be acceptable
3. The Meritor RPL-20 is not recommended for this application without significant additional development
   - The driveshaft reached critical speed at 68 mph
   - The driveshaft exceeds SAE guideline for Inertial Acceleration
   - Significant redesign and testing would be required in order to meet traditional critical speed guidelines for this design and demonstrate robust performance
   - If radial forces and critical speeds were corrected, concurrence required by the vehicle manufacturer and by the suppliers of adjacent components that the higher dynamic torques associated with higher rotational inertias in this submission could be acceptable

All FMTV Front Driveshafts

1. The GKN driveshaft is conditionally recommended as first choice.
   - This was the only design that meets SAE guideline for Inertial Acceleration
   - This driveshaft would be expected to have the lowest transfer case dynamic forces of all designs analyzed. This is attributed to:
     - Low unbalance forces in the ISO standard
     - Absence of Inertial Acceleration in a CV design
   - GKN must demonstrate acceptable boot performance before the conditional provision is lifted.
2. The Dana driveshaft is conditionally recommend as second choice

- This design does not meet the SAE guideline for Inertial Acceleration
- Concurrence required by the vehicle manufacturer and by the suppliers of adjacent components that the higher dynamic torques associated with higher rotational inertias in these submissions could be acceptable

3. The Arvin Meritor RPL-20 driveshaft is conditionally recommend as third choice
    - This design had water entry issues that altered the driveshaft balance
    - This design does not meet the SAE guideline for Inertial Acceleration
    - Concurrence required by the vehicle manufacturer and by the suppliers of adjacent components that the higher dynamic torques associated with higher rotational inertias in this submission could be acceptable
    - Arvin Meritor must demonstrate acceptable water entry exclusion before the conditional provision is lifted.

MTV Intermediate Driveshaft

1. The GKN driveshaft is conditionally recommended as first choice.
    - This was the only design that meets SAE recommended guideline (1000 rad/sec^2) for Inertial Acceleration (this design has zero inertia acceleration)
    - This driveshaft was measured to have the lowest transfer case dynamic forces of all designs analyzed. This is attributed to:
        - Low unbalance forces in the ISO standard
        - Absence of Inertial and Torsional Acceleration in a CV design
    - GKN must demonstrate acceptable boot performance before the conditional provision is lifted.
2. The Dana driveshaft is conditionally recommend as second choice
    - This design meets the SAE conditional guideline (2000 rad/sec^2) for Inertial Acceleration (this design has 2000 rad/sec^2 inertial acceleration)
    - Concurrence required by the vehicle manufacturer and by the suppliers of adjacent components that the higher dynamic torques associated with higher rotational inertias in this submission could be acceptable
3. The Arvin Meritor RPL-20 driveshaft is conditionally recommend as third choice
    - This design had water entry issues that altered the driveshaft balance
    - This design meets the SAE conditional guideline (2000 rad/sec^2) for Inertial Acceleration Inertial Acceleration (this design has 2000 rad/sec^2 inertial acceleration)
    - Arvin Meritor must demonstrate acceptable water entry exclusion before the conditional provision is lifted.
    - Concurrence required by the vehicle manufacturer and by the suppliers of adjacent components that the higher dynamic torques associated with higher rotational inertias in this submission could be acceptable

MTV rear-rear Driveshaft

- All candidate designs had essentially equal performance in this position and are given equal ranking for conditional approval
- The designs meet the SAE conditional guideline (2000 rad/sec^2) for Inertial Acceleration (the designs have 1800 rad/sec^2 inertial acceleration)
- Concurrence required by the vehicle manufacturer and by the suppliers of adjacent components that the higher dynamic torques associated with higher rotational inertias in these submissions could be acceptable
- Arvin Meritor must also demonstrate acceptable water entry exclusion before the conditional provision is lifted.

Testing Overview

Three manufacturers submitted hardware for the testing program. The current production A1 was also tested, as a baseline. The tested hardware is designated as follows:

Manufacturer	Designation	Type
Arvin Meritor	A1 Current Production Baseline	Single Cardan
Arvin Meritor	RPL-20	Single Cardan
Dana	SPL-140	Single Cardan
GKN	CV (with Kempf Cardan in MTV rear-rear)	Constant Velocity (Rzeppa)

Note: all GKN driveshafts except the LMTV rear were provided with Kempf manufactured spline sections. The GKN LMTV rear driveshaft utilized a plunge type CV joint in lieu of a slip spline. The MTV rear-rear driveshaft provided by GKN was completely manufactured by Kempf.

General Features Comparisons

Each manufacturer provided measurably larger slip spline diameters compared with the A1 driveshaft. This feature would be expected to improve critical speed, reduce spline wear, reduce propensity for spline lock, and reduce changes in balance due to spline wear.

Each manufacturer provided complete spline sealing systems, compared with the A1 spline section, which was unsealed. This would be expected to reduce grease maintenance and prevent grease washing from the spline section.

Each manufacturer provided driveshaft and universal joint torque capacity that was least one standard size larger compared with the A1 driveshaft. This feature would be expected to reduce propensity for field failures, and reduce universal joint distress on the high torque conditions.

Arvin Meritor and GKN provided "Lubricated for Life" driveshaft lubrication systems. Dana has an optional "Lubricated for Life" system. The tested Dana driveshafts have a recommended 25,000 mile or 6 month lubrication interval.

The tested driveshafts were manufactured by the respective suppliers and shipped to MSC at Milford, MI. The driveshafts were measured for hinging, straightness, and balance, on a balance machine, using production yokes centered on the yoke spline ID. The driveshafts were then set to 90 to 100% of the limit of balance as specified by the manufacturer. The raw data from these measurements are listed in Appendix IV.

Balancing

The two manufacturers of the cardan driveshafts stated that the balance specifications were the SAE specification for medium and heavy-duty vehicles, which balances each end of the driveshaft to within one in-oz of unbalance per 10 lbs of driveshaft end weight. The manufacturer of the CV driveshaft indicated that these driveshafts were balanced to the ISO specification, which is roughly one third that of the SAE specification. A Nomogram of the ISO balance specification is included in Appendix V.

The driveshafts were set to the unbalance limits as specified. Since balance can change during testing due to wear in the universal joints and slip splines, balance and runout was rechecked at the end of testing.

It should be noted that each manufacturer initially balanced the driveshafts using their own actual production, or simulated production, balancing procedure. Production procedures usually do not include production flanges. The balance and runout measurement procedures used by MSC included production flanges, which were centered on the flange spline inner diameter. This method, though realistically simulating actual vehicles, would be expected to yield slightly different results than the production process.

When the testing was completed, each manufacturer was provided the opportunity to examine the tested components and make written observations on their findings. Each manufacturer elected to perform this inspection and provide a written report. These reports are available through the PM-FMTV to approved requestors.

Test Plan

A test route was developed that represented a severe, yet possible, scenario that would bound the envelope of expected field operation. The testing included the following:

- 55 miles at 57 mph (governed speed, ≈ 3370 driveshaft rpm)
- Accelerate to 70 mph (≈ 4100 driveshaft rpm) and hold for 10 seconds
- Drive into three foot depth submersion tank, and dwell for 1 minute (Figure 1)
- Repeat for 1000 miles
- Ballast = curb weight (most severe driveshaft angles)
- Conduct High Torque/Low Speed "Cooling Test" (See Figure 1)
    - Ballast to GVW
    - 13,500 lb Drawbar Load on LMTV
    - 20,800 lb Drawbar Load on MTV (Simulated tractor by over ballasting and by omitting front driveshaft)
    - 90 Minutes Duration

During this testing the following channels of drivetrain data were continuously recorded:

- Rear Transfer Case Housing Vertical and Lateral Forces
- Rear Transfer Case Vertical and Lateral Accelerations
- Both Driveshaft Speeds and Displacements at the TC Output Position
- Engine Speed and Displacement
- Left and Right Engine Block Strains
- Lateral and Vertical Accelerations at the TC Housing

- Cabin Sound Pressure Level
- Temperatures: Ambient Air, Exhaust Gas, Transfer Case Housing

High-Torque Low Speed Tests

All candidate driveshafts were subjected to a test equivalent to the Cooling Test normally conducted at the Army's Yuma Proving Ground. This was accomplished at General Motors Milford Proving Ground using Allison Division's Terex Towing Dynamometer.

The cooling test is typically a ninety-minute long low-speed high tractive effort test. It is run at a sustained drawbar pull that is 60% of the weight of the vehicle. Sixty percent is considered the maximum sustainable drawbar pull that is realistically achievable. A drawbar pull higher than this can result in weight transfer that unloads the front axle to the point that the vehicle cannot be reliably steered.

The testing was designed to simulate the only high torque conditions of the cooling test; maximizing the torque loads in the driveshafts. No attempt was made to simulate the thermal conditions of the test.

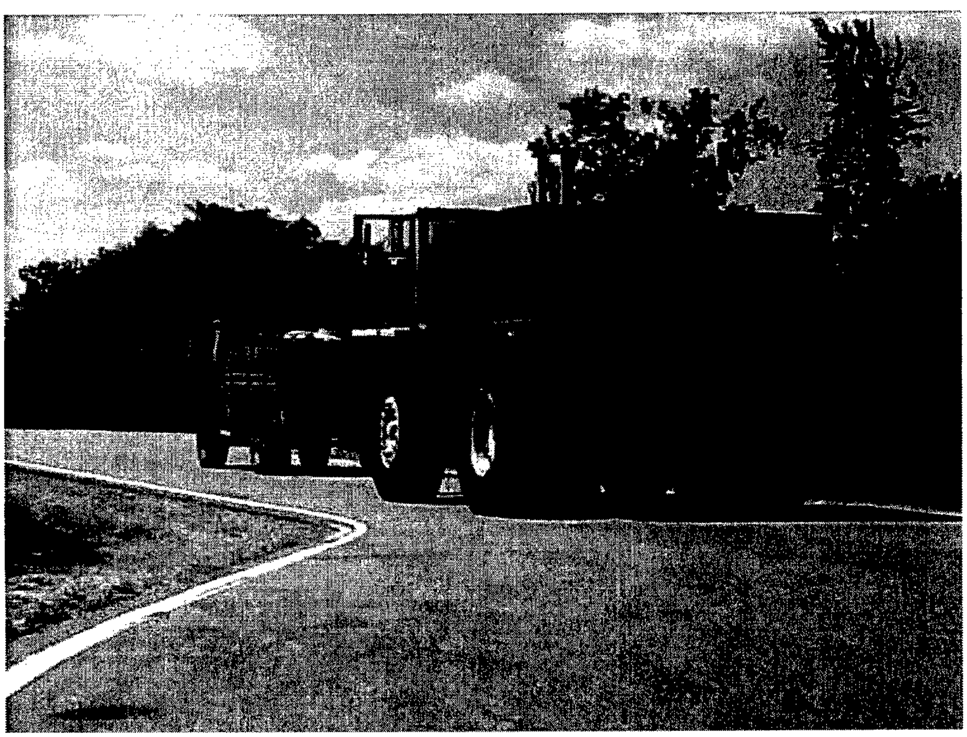
MTV: The heaviest variant (a tractor with maximum trailer load) was simulated, using the available M1083 (cargo) vehicle as the actual test vehicle. The maximum torque of concern was that transmitted by the intermediate driveshaft. The M1083 was ballasted to a rear bogie weight of 30,300 lbs. The drawbar pull was set to 20,800 lbs. By operating in "mode" without a front driveshaft, the required 4923 lb. ft. of calculated torque was achieved at the intermediate driveshaft. The ballast required was 20,000 lbs. and the speed was around 3 mph for the ninety minutes of test. See Appendix VI for the method used to calculate these torques.

High tractive effort tests do not challenge the front driveshaft due to weight transfer from the front to the rear during the test. Since the intermediate and front driveshafts shared the same design universal joint in all submissions, removing the front driveshaft to achieve the much higher torques in the intermediate shaft was considered reasonable. The front driveshaft was tested to realistic torques in the LMTV test.

LMTV: The LMTV was tested at GVW of 23,300 lbs. with drawbar pull of 13,500 lbs. The speed was typically 3 mph for the ninety minutes of test.

Figure 1
Submersion Test (MTV)

Figure 2
Low Speed/High Torque Dynamometer Test (LMTV)



Transfer Case Force Transducer

A critical portion of the measurement system is the transfer case force transducer. This transducer was designed to measure the static and dynamic forces imparted by the driveshaft onto the transfer case during the entirety of the testing. The transducer was designed to accurately perform this measurement without significantly altering the structure of the transfer case. Finite Element Modeling (FEM) and Dynamic Modeling (DADS) of the system were used to verify that these goals were achieved.

Figure 3

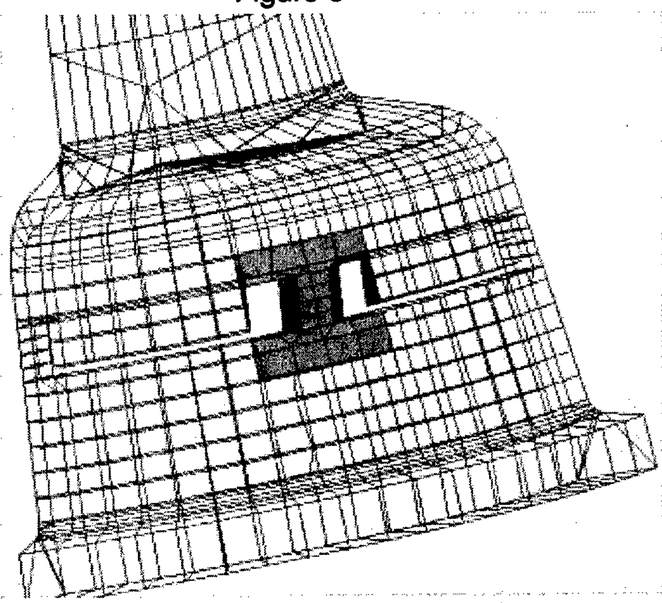


Figure 3 shows the ¼ section FEM of the Transfer Case Housing in the final machining configuration. The FEM was used for extensive redesign of this configuration in order to achieve maximum sensitivity, minimum crosstalk, and maximum stiffness of the housing. The final shape required the strain gage surfaces to be perpendicular to the direction of measured loading in order to meet these goals. The total housing stiffness was calculated in the model and compared with the unmodified housing stiffness. Appendix I contains a summary of the FEM analyses.

This information was then used in a DADS model of the LMTV Drivetrain system (See Appendix III for a complete description of this DADS model). The critical speed of an A1 driveshaft was actually increased by 1 to 2 Hz. when run using the modified transfer case housing transducer, compared with the critical speed when run on an unmodified housing. This change was not considered significant enough to alter the conclusions based on data taken from a vehicle using the transducer. See Appendix II for a summary of this analysis. Figure 4 shows a photograph of the machined part, while figure 5 shows the transducer mounted on an LMTV.

Figure 4

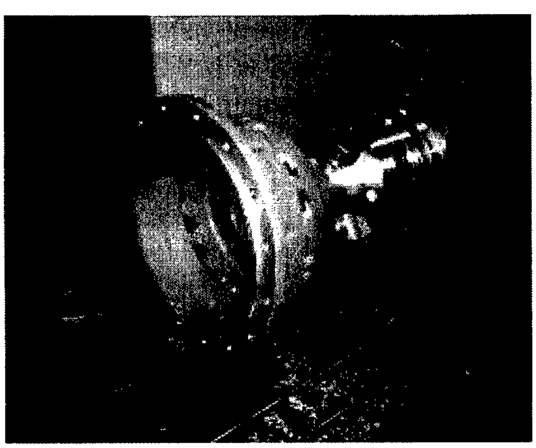
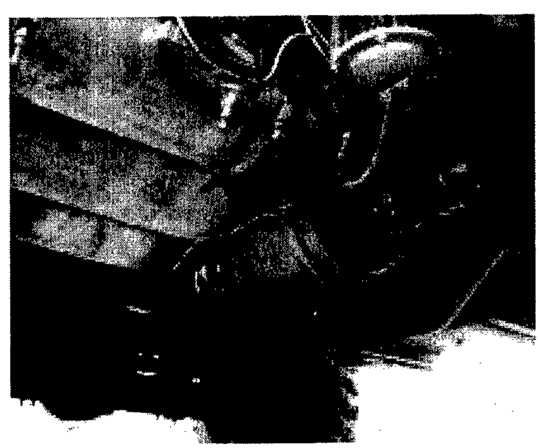


Figure 5



The transducer was designed with a maximum force capability of 2000 lb. This number was chosen as a maximum sustained dynamic force that the vehicle would be expected to incur during normal operation. Forces higher than 2000 lb would not be expected to permanently yield the transducer, but normal limits of linearity, hysteresis and fatigue strength would be exceeded. The yield strength of the transducer would be expected to be several times the 2000 lb operating limit. The transducer can be mounted in either the forward or aft housing position on the transfer case. The rear position was used for all tests since that position was considered to be worst case for this testing.

Transfer Case Housing Forces

Since a significant portion of the conclusions of this study are based on the measured transfer case housing forces, some discussion of the nature of these forces is in order. The initial check out of the vehicle measurement system was made using the Dana driveshafts on the LMTV and the data from this check out will be used for this discussion.

Several sources of forces in the housing have been identified:

- First Order (once per revolution) relating to driveshaft unbalance and rpm, as well as due to unbalanced hardware in the transfer case housing.
- Second Order (twice per revolution) relating to a cardan joint phenomenon called 'secondary couple' as well as driveshaft inertial accelerations, relating to torque, rpm and joint angles.
- Higher frequency gear rattle, relating to driveshaft inertial accelerations and drive torque.

Figure 6 shows a chart of unfiltered lateral force data in the transfer case housing during a run to 70 mph followed by a throttle release and partial coast down. The force peaks in this data exceed 1200# zero-to-peak. Exhaust temperature is included on this chart in order to give an indication of engine power.

Figure 7 shows a portion of data low-pass filtered at 200 Hz. The filtering removes the gear rattle and other high frequency components, but retains the first and second order components of the force. It can be seen that the force peaks exceed 900# zero-to-peak.

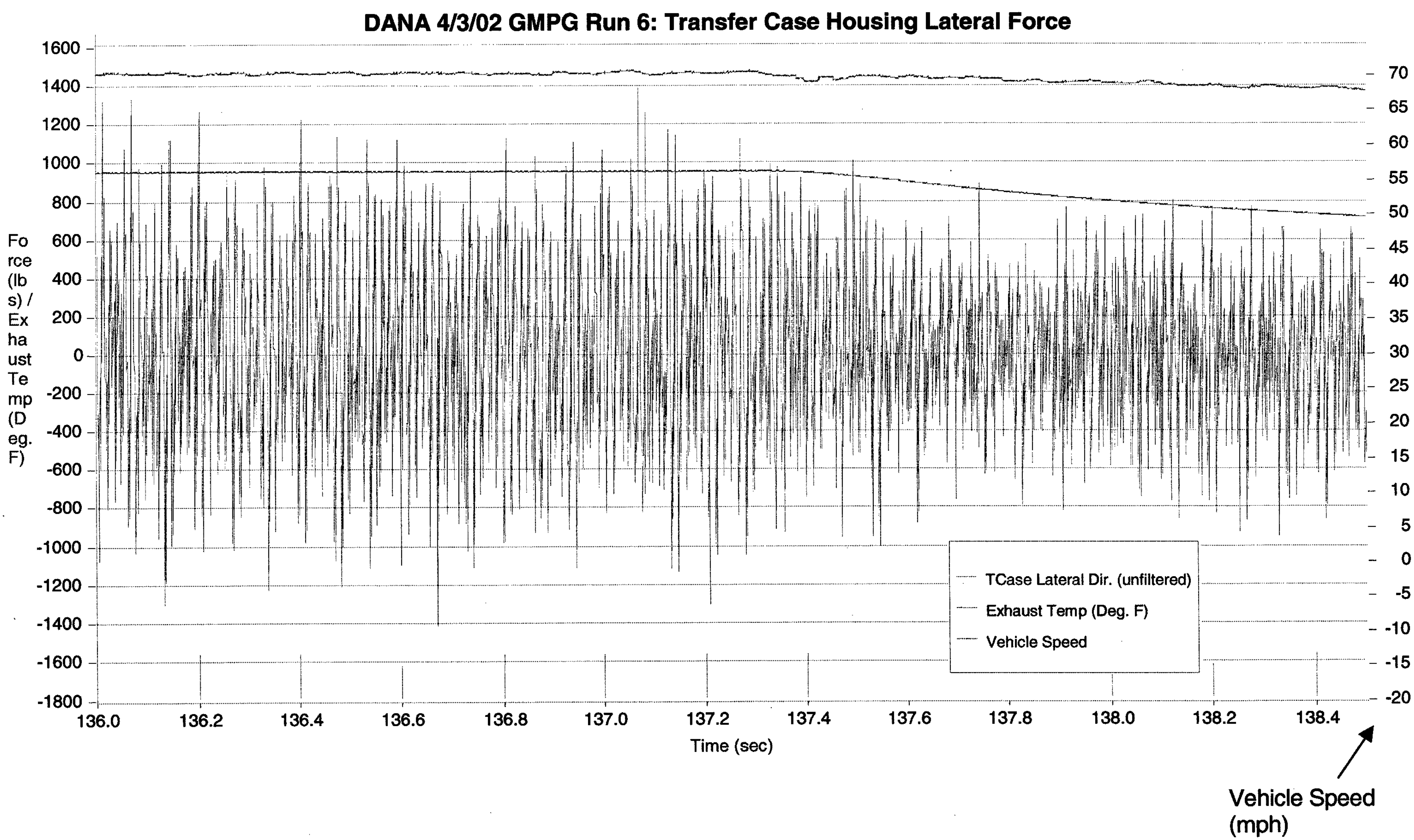
In order to better understand the nature of this force, figure 8 was plotted, showing an expanded time scale plotted along with a once per revolution driveshaft speed sensor. It shows that the forces are primarily first order, but there are also higher frequency components.

This driveshaft has a specified unbalance limit of 4.2 oz-in at the transfer case rear position. This driveshaft was intentionally unbalanced to be near this limit. The centrifugal force caused by 4.2 oz-in unbalance operating at 70 mph (4077 rpm), without any resonances, would impart a zero-to-peak first order force on the flywheel housing of 124 pounds.

The difference between the measured 900 pound force and the calculated force of 124 pounds can be explained by the effect of driveshaft critical speed. Measurements of the critical speed of this driveshaft indicate are that the critical speed is near 5400 rpm. However, modeling of this driveshaft in the vehicle drivetrain indicates that force amplification due to critical speed effects commences well below the actual critical speed of the driveshaft and could easily be 6 times higher by 4080 rpm (70 mph). This would explain the measured forces in the transfer case housing.

It would be expected that if the speed were to be increased to achieve the actual critical speed of the driveshaft, the forces would further increase and, perhaps, reach a destructive level. The 900# zero-to-peak measured value of first order force is not considered destructive in the short term, given proper grease maintenance and driveshaft hinging levels in the vehicle. Forces this high, however, could diminish the durability of vehicle and powertrain components with continued operation.

Figure 6



DANA 4/3/02 GMPG Run 6: Transfer Case Housing Lateral Force
Force (lbs) / Exhaust Temp (Deg. F)
1600
1400
1200
1000
800
600
400
200
0
-200
-400
-600
-800
-1000
-1200
-1400
-1600
-1800
136.0
136.2
136.4
136.6
136.8
137.0
137.2
137.4
137.6
137.8
138.0
138.2
138.4
Time (sec)
70
65
60
55
50
45
40
35
30
25
20
15
10
5
0
-5
-10
-15
-20
TCase Lateral Dir. (unfiltered)
Exhaust Temp (Deg. F)
Vehicle Speed
Vehicle Speed (mph)

Figure 7

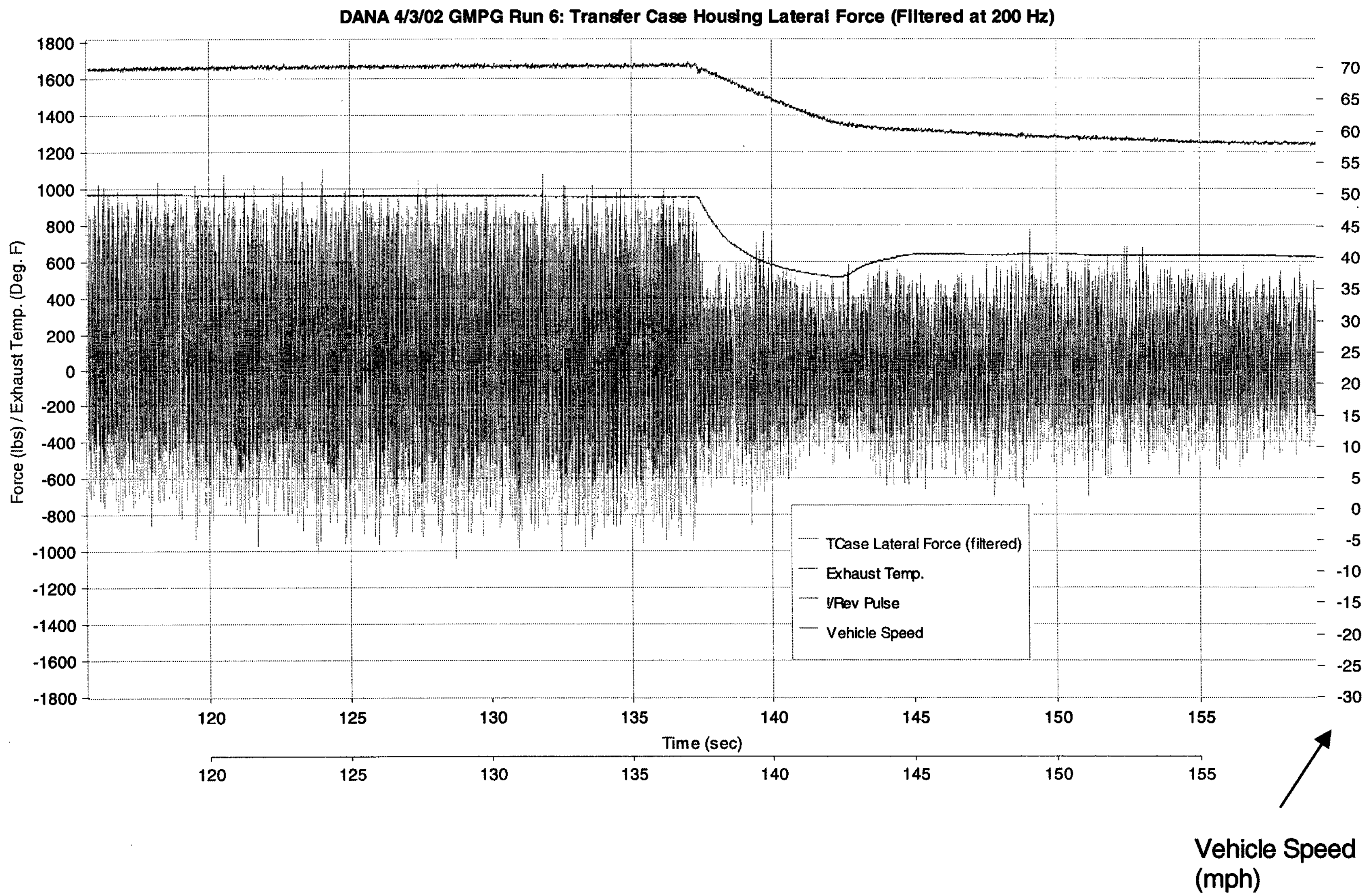
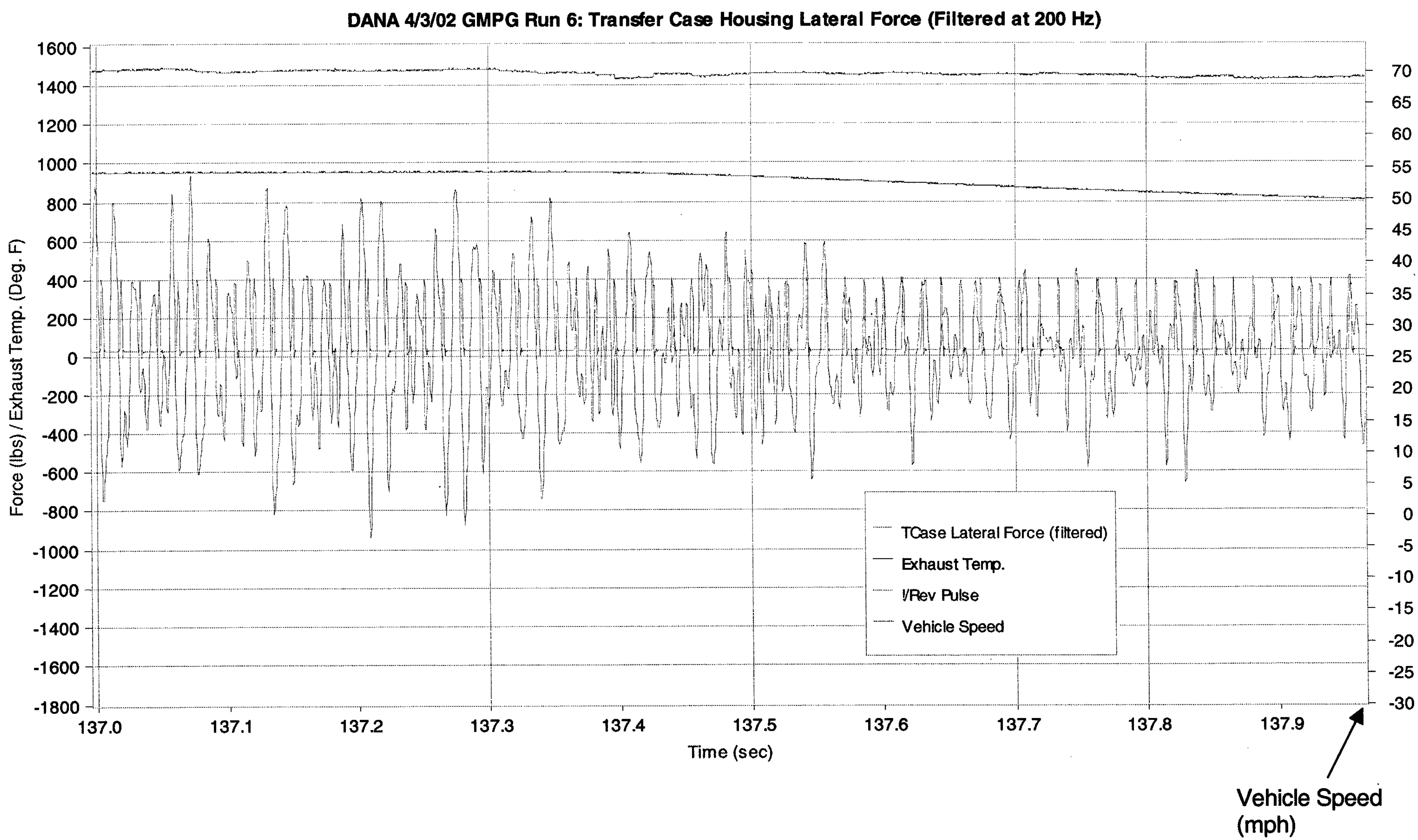

DANA 4/3/02 GMPG Run 6: Transfer Case Housing Lateral Force (Filtered at 200 Hz)
Force (lbs) / Exhaust Temp. (Deg. F)
Time (sec)
TCase Lateral Force (filtered)
Exhaust Temp.
I/Rev Pulse
Vehicle Speed
Vehicle Speed (mph)

Figure 8


DANA 4/3/02 GMPG Run 6: Transfer Case Housing Lateral Force (Filtered at 200 Hz)
Force (lbs) / Exhaust Temp. (Deg. F)
1600
1400
1200
1000
800
600
400
200
0
-200
-400
-600
-800
-1000
-1200
-1400
-1600
-1800
70
65
60
55
50
45
40
35
30
25
20
15
10
5
0
-5
-10
-15
-20
-25
-30
137.0
137.1
137.2
137.3
137.4
137.5
137.6
137.7
137.8
137.9
Time (sec)
TCase Lateral Force (filtered)
Exhaust Temp.
!/Rev Pulse
Vehicle Speed
Vehicle Speed
(mph)

Critical Speed Measurements on the Vehicle

Data Verification

During initial check out testing on the Arvin Meritor RPL-20 driveshafts in the LMTV, forces were measured that exceeded 2000 lbs. zero-to-peak. It was observed that driveshaft critical speed was reached at about 70 mph (68 Hz.). Similar measurements and data were also observed with the Arvin Meritor A1 driveshaft. In order to establish confidence in these observed results, several checks to verify the integrity of the vehicle and data were performed. These tests are summarized as follows:

Transfer Case Force Transducer Calibration

The original transducer calibration was performed in the vehicle with a reference load cell with a maximum calibration force of 1000 lbs. This value was rechecked in a partial recalibration after the RPL-20 measurements by hanging dead weights of 50, 100, and 150 lbs. on the transducer. The transducer was shown to be linear, within calibration specification, and within the hysteresis specification.

Repeatability of Data

Prior to the RPL-20 measurements, the vehicle transfer case forces had been measured with a GKN CV joint driveshaft at a sustained speed of 60 mph. A similar recording with a GKN CV joint driveshaft was made after the RPL-20 measurements. Analysis of the data verified that there was no measured change in performance of the vehicle or the data analysis system.

Truck Bearings

The transfer case output shaft and rear axle input shaft endplay were measured after the RPL-20 testing had been completed. The endplay of both shafts was within specification.

Measured Driveshaft Resonant Frequency in the Vehicle

The first mode resonant frequency of the driveshaft installed in the vehicle was to be measured to estimate driveshaft critical speed. This testing was expanded to verify the measurements at several conditions of driveshaft torque and force excitation levels. The resonant frequency was measured by sweeping an excitation force from an electrodynamic shaker through the resonant frequency and measuring the response with an accelerometer on the driveshaft. The resonant frequency was determined by finding the frequency where the maximum acceleration per unit input force occurs. This frequency of the RPL-20 driveshaft varied from 68 to 78 Hz, depending on the amount of excitation force, and torque applied through the driveshaft.

Driveshaft Modeling

The MSC Dynamic Analysis and Design System (DADS) model of the RPL-20 driveshaft indicated a critical speed of 72 Hz. This compares quite favorably with the measured critical speed of 69 Hz, and the in vehicle shaker testing which indicated a 68-78 Hz resonance range.

Transducer Modeling

The transducer modeling was discussed earlier. The transducer modifications were shown to slightly raise the critical speed of the driveshaft by approximately 1 to 2 Hz. The magnitude of this change is not considered significant enough to alter any conclusions.

Analysis of RPL-20 data

There were three circumstances in the RPL-20 data runs where the shaft appeared to reach, or exceed critical speed. These data were analyzed for a phase shift of 180 degrees of force vs. driveshaft angular position, which is characteristic of a driveshaft passing through critical speed. In each case, a 180-degree phase shift was noted. Simultaneous with the phase shift, a sharp reduction of amplitude was noted. Both phenomena are classical critical speed behavior. A typical plot of this data is shown in Figure D

Summary

After careful analysis of the road test data, laboratory test data, driveshaft modeling, and analysis, and verification of the vehicle and data integrity, it became apparent that the LMTV rear driveshafts discussed here reached critical speed at around 70 mph (68 Hz.). It is believed that this characteristic of the driveshaft was related to its fundamental design, not to manufacturing variability nor test conditions peculiar to this measurement.

Dynamic System Modeling

Dynamic Modeling was used throughout the program to provide analysis and insight into the FMTV powertrain system and guide the testing process. Dynamic Analysis and Design System (DADS) modeling was used. A summary of this effort is provided in Appendix III. Results from this effort are used and referenced throughout this report. An overall summary of this effort is provided here.

MSC has previously reported on modeling of the LMTV powertrain for TACOM in a report titled " An Investigation of Driveline Incidents of the US Army's Model M1078 Light Medium Tactical Vehicle (LMTV)" under contract Number DAAE07-98-M012. This model included the powerpack, rear driveshaft, and other dynamic elements of the vehicle. This model was expanded to include the front driveshaft, and appropriate elements for each supplier's submissions. A balance machine model was also developed. Each of these models and elements was validated that the model accurately reproduced the physical system by extensive checking of modeled results that were compared with measured results. The reader is referred to the report for detailed descriptions of this original model.

A drawing of the Powerpack that was modeled is shown in Figure 9. For modeling purposes, a large block finite element model of the important mass and stiffness elements of the power pack, as well as front and rear driveshaft elements was generated. A drawing of this model is illustrated in Figure 10. The dynamic properties driveshaft end supports were also included. Appropriate vehicle properties were added when necessary for acceleration and constant torque studies.

The Balancing Machine model was used to develop the balance techniques and procedure for setting the appropriate unbalance limits. The balancing machine model used appropriate driveshaft properties, with constraints similar to balance machine dynamics. This model was used to study the effects of driveshaft straightness on balance measurements. It was used to develop the strategy for unbalancing each driveshaft to the limit of its specification.

The two-driveshaft system model was used to establish appropriate transducer locations and force limits, as well as to study the interactions among the various components. This model, plus analysis, determined that the front driveshaft dynamic forces were always lower than the rear driveshaft dynamic forces, for any driveshaft configuration, negating the need to simultaneously measure both front and rear driveshaft support forces simultaneously. This meant that it was necessary to measure only the rear forces, and that two transfer case force transducers for each vehicle were unnecessary. The actual difference between the front and rear force magnitude was not proportional to speed. At low speeds, the forces were approximately proportional to driveshaft mass and unbalance. At higher speeds they were related to mass, unbalance, and closeness to critical speed.

The model of each driveshaft design was used to evaluate critical speeds and balance methods. This information was used to supplement the measurement program, where detailed forces were not always available. The reader is referred to Appendix III for more detail.

Figure 9
Actual Vehicle Powerpack

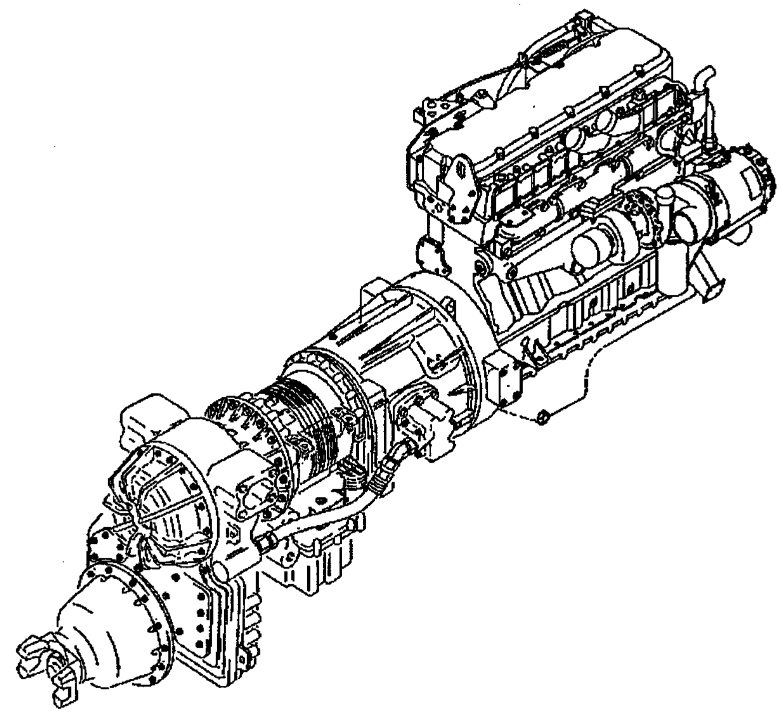
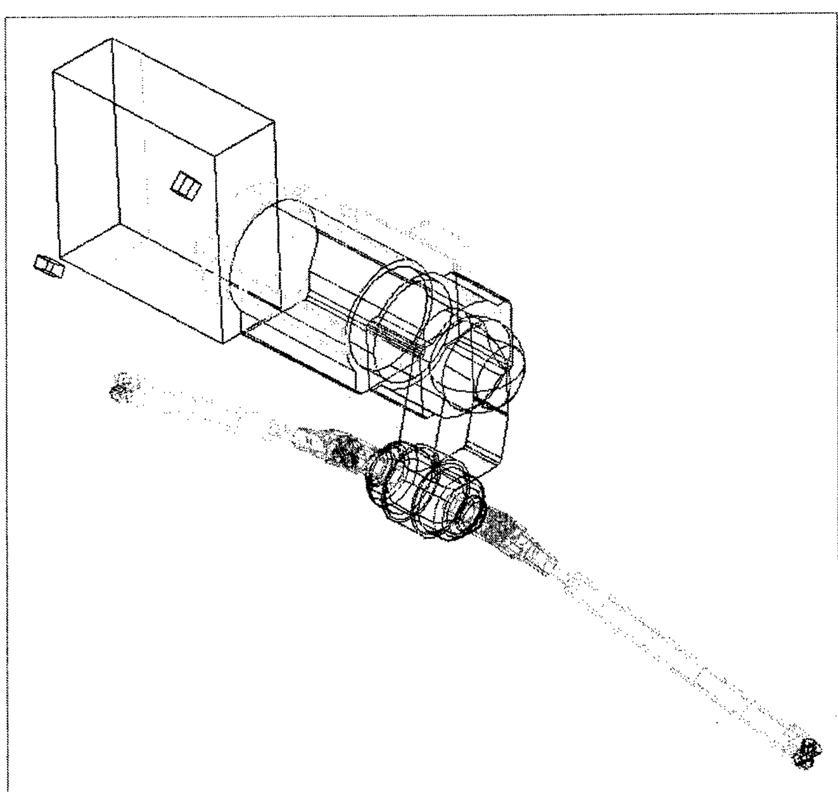


Figure 10
Modeled Elements in Drivetrain



Balancing and inspections

Prior to testing, the candidate driveshafts were weighed and examined. They were then delivered to the balancing facility for measurements of end play, hinging, straightness, and balance. They were set to their balance tolerance limit.

Upon test completion, they were re-examined, re-measured, and re-checked for end-of-test balance.

Tables 5, 6. 7,. And 8 summarize the measurement data and report observations.

Table 5
Summary Balance Data and End-of-Test Inspections
Arvin Meritor A1

MSC ID#	Installed Length	Usage	Total Mass Lbs	Slip End Lbs	Weld End Lbs
PI-2	39.6	MTV Intermediate	54	29	25.5
PR-L-2	61.5	LMTV Rear	60.5	39	31.5
PRR-2	33.9	MTV Rear-Rear	51	29	22
PF-2	33.5	LMTV Front	54	29	24
PF-3	33.5	MTV Front	54	29	24

Driveshaft Balance Conditions
In-oz.

End of Test	Start of Test	Slip End Spec	MSC ID#		Weld End Spec	Start of Test	End of Test
2.6	2.1	2.9	PI-2	MTV Intermediate	2.6	2.4	1.7
2.2	3.8	3.9	PR-L-2	LMTV Rear	3.2	2.9	1.7
1.1	NA	2.9	PRR-2	MTV Rear-Rear	2.2	NA	.4
16.8	2.9	2.9	PF-2	LMTV Front	2.5	2.4	3
4.1	2.9	2.9	PF-3	MTV Front	2.5	2.4	3.1

Initial checkout on the LMTV indicated transfer case forces were exceeding 2000 lbs. Analysis of data taken during these runs showed that the driveshaft was reaching critical speed at about 68 mph. Instrumentation was installed on the vehicle to indicate with warning lamps whenever transfer case forces reached or exceeded 2000 lb.

The full 1000-mile test was run on this driveshaft. A modified test procedure was followed. When the test driver reached the point in the procedure calling for acceleration to 70 mph, with a 10 second dwell at 70 mph, he substituted the following procedure: "Slowly accelerate to no more than 70 mph. If a 2000 lb warning lamp illuminates at any time during this acceleration, release the throttle, and resume 58 mph." The driver reported that each time during this portion of the procedure, the 2000 lb warning lamp activated.

Subsequent analysis of the data indicated that the approach near critical speed of the driveshaft was the cause of the warning lamp activation.

At the completion of the 1000-mile high-speed portion of the test, inspection revealed that the rear differential end of the rear-rear MTV driveshaft had increased endplay in the universal joint thrust washers. Since no other A1 driveshafts were available for testing, the universal joint was replaced with a new universal joint kit in order to complete the low-speed/high-torque testing.

The driveshafts, otherwise, completed the testing without incident. Subsequent teardown inspection of the driveshafts by Arvin Meritor indicated no visible deterioration of the slip splines. There was a change in slip joint grease color, probably due to water in the grease.

Several of the universal joints were observed to have brinell marks on the cross journals probably due to high torques during the testing. The universal joint grease was also discolored, probably due to a small amount of water ingression during the testing.

There was water found in the driveshaft tubes. Water likely entered through the vent hole in the yoke end of the tube. One tube was cut apart revealing that the water had deteriorated the cardboard damping tube, and could possibly explain a change in post-test driveshaft balance measurements.

For more detail, see the Arvin Meritor report on the driveshaft examination Available through the PM-FMTV.

Table 6
Summary Balance Data and End-of-Test Inspections
Arvin Meritor RPL-20

MSC ID#	Installed Length	Usage	Total Mass Lbs	Slip End Lbs	Weld End Lbs
PI-3	39.6	MTV Intermediate	78	40	38
PR-L-1	61.5	LMTV Rear (#1)	90.5	50.	40.5
PR-L-2	61.5	LMTV Rear (#2)	90.5	50.	40.5
PRR-4	33.9	MTV Rear-Rear	78	40	38
PF-5	33.5	LMTV Front	76	40	36
PF-4	33.5	MTV Front	76	40	36

Driveshaft Balance Conditions
In-oz.

End of Test	Start of Test	Slip End Spec	MSC ID#		Weld End Spec	Start of Test	End of Test
.6	4	4	PI-3	MTV Intermediate	3.7	3.7	.7
No meas.	4.8	5	PR-L-1	LMTV Rear (#1)	4.1	3.9	No meas.
16.2	4.8	5	PR-L-2	LMTV Rear (#2)	4.1	3.9	22.6
4.7	4	4	PRR-4	MTV Rear-Rear	3.8	3.6	9.4
3.4	3.8	4	PF-5	LMTV Front	3.6	3.6	10.4
3.9	3.8	4	PF-4	MTV Front	3.6	3.5	5.1

Initial checkout of the LMTV (approximately 150 total miles of driving) indicated transfer case forces were exceeding 2000 lb. Testing of this driveshaft (PR-L-1) was suspended at this point. No post-test measurements were made on this driveshaft.

Instrumentation was added to the vehicle to indicate with warning lamps whenever transfer case forces exceeded 2000 lb. A new driveshaft was installed (PR-L-2), and the full 1000-mile test was run. The test procedure was modified for this driveshaft. When the test driver reached the point in the procedure calling for acceleration to 70 mph, with a 10 second dwell at 70 mph, he substituted the following procedure: "Slowly accelerate to no more than 70 mph. If a 2000 lb warning lamp illuminates at any time during this acceleration, release the throttle, and return to 58 mph." The driver reported that each time during this portion of the procedure, the 2000 lb warning lamp activated.

Subsequent analysis of the data indicated that the approach near critical speed of the driveshaft was the cause of the warning lamp activation. It is also likely that the high forces generated in this condition lead to permanent deformation of the driveshaft, explaining the increase in end of test unbalance measured on both ends of the LMTV rear driveshaft (#2).

The driveshafts, otherwise, completed the testing without incident. Subsequent teardown inspection of the driveshafts by Arvin Meritor indicated no visible deterioration of the slip splines and universal joints due to the testing. There was a slight change in universal joint grease color, though no water intrusion was noted. There was seal damage noted on the outer lip of several universal joint seals. There was water found in the driveshaft tube. It likely entered through the vent hole in the end of the tube. There was a measured increase in post-test driveshaft unbalance that was likely due to this water intrusion.

For more detail, see the Arvin Meritor report on the driveshaft examination available through the PM-FMTV.

Table 7
Summary Balance Data and End-of-Test Inspections
Dana

MSC ID#	Installed Length	Usage	Total Mass Lbs	Slip End Lbs	Weld End Lbs
PL-1	39.6	MTV Intermediate	62	35	27
PRL-1	61.5	LMTV Rear	73	42.5	31.5
PRR-2	33.9	MTV Rear-Rear (High Speed)	59	32	27
PRR-3	33.9	MTV Rear-Rear (High Torque)	59	32	27
PF-6	33.5	MTV Front	59	32	27

				Driveshaft Balance Conditions In-oz.			
End of Test	Start of Test	Slip End Spec	MSC ID#		Weld End Spec	Start of Test	End of Test
6.5	3.3	3.5	PL-1	MTV Intermediate	2.7	2.5	2.3
2.5	4.2	4.3	PRL-1	LMTV Rear	3.2	2.9	1
2.8	3	3.2	PRR-2	MTV Rear-Rear (High Speed)	2.7	2.6	1.9
4.6		3.2	PRR-3	MTV Rear-Rear (High Torque)	2.7		4.2
3.6	3.1	3.2	PF-6	MTV Front	2.7	2.5	2.8

The Dana driveshafts completed testing with no incidents. Upon final tear down inspection on 9/13/02 at the Dana Driveshaft Engineering lab in Toledo, no water intrusion, nor excessive wear nor Brinelling was found. The driveshaft grease was serviceable.

The MTV driveshafts had been removed from the vehicle at the end of the 1000-mile high-speed portion of the testing, as a scheduling conflict prevented immediate low-speed/high-torque testing. Discolorations were noted on the bearing end caps in the rear-rear driveshaft at this time. Since the discoloration possibly indicated overheating of the bearings, this driveshaft was replaced with a new driveshaft for the low-speed/high-torque test.

Subsequent analyses of this driveshaft indicated that the discolorations were due to fretting, not overheating. The thrust surfaces, needle surfaces, and grease were serviceable. Since the High Torque driveshaft was not scheduled to have any high-speed tests, no start of test driveshaft balance measurements were made on this driveshaft.

For more detail, see the Dana report on the driveshaft examination in available through the PM-FMTV.

Table 8

Summary Balance Data and End-of-Test Inspections

GKN

MSC ID#	Installed Length	Usage	Total Mass Lbs	Slip End Lbs	Weld End Lbs
PI-3	39.6	MTV Intermediate	76	48	35
PR-L-3	61.5	LMTV Rear	67	36.5	30.5
PRR-2	33.9	MTV Rear-Rear	89.5	49.5	40
PF-4	33.5	LMTV Front	71	37	34
PF-6	33.5	MTV Front	71	37	34

Driveshaft Balance Conditions In-oz.

End of Test	Start of Test	Slip End Spec	MSC ID#		Weld End Spec	Start of Test	End of Test
2.8	1.4	1.4	PI-3	MTV Intermediate	1.2	1.2	.5
.7	.9	1	PR-L-3	LMTV Rear	.9	.9	1.35
1	1.4	1.4	PRR-2	MTV Rear-Rear	1.4	1.4	2.2
.5	.8	1.3	PF-4	LMTV Front	1.2	1.6	2.1
2.9	1.2	1.3	PF-6	MTV Front	1.2	1.2	.9

The GKN driveshafts completed testing without structural compromise. However, intra test inspections had indicated water intrusion in all of the CV joints. The LMTV rear driveshaft had a visually damaged boot. In spite of the subsequent water and grit contamination, all of the driveshafts had bearing surfaces that were considered to be serviceable at the end of the test.

Subsequent teardown of the driveshafts by GKN revealed CV joint wear due to contaminant intrusion caused by the boot seal issues, but the CV joints were otherwise serviceable. The Kempf MTV rear-rear driveshaft was found to have no visible deterioration due to the testing, but droplets of water were visible on the slip spline teeth inside the sealed area. Subsequent inspection by Kempf revealed that the water intrusion was a result of modifications made to lengthen the shaft after its initial manufacture. The water intrusion was not attributed to the inherent design of the slip spline seal.

For more detail, see the GKN report on the driveshaft examination in available through the PM-FMTV.

Driveshaft Critical Speed

One important aspect of driveshaft selection is driveshaft critical speed. A driveshaft reaches critical speed when it rotates at a speed approximately equal to its first mode bending resonant frequency. When this happens, vibrating forces at the driveshaft supporting structure get extremely high and damage can occur. This damage may manifest as fracture or fatigue of supporting elements, permanent bending of the driveshaft, universal joint overheating, or excessive vibration. Actual forces for any particular vehicle system are influenced by many factors including driveshaft mass, unbalance, straightness, damping, hinging, etc.

The SAE "Universal Joint and Driveshaft Design Manual" Advances in Engineering No. 7 (AE-7) lists guidelines for designing to accommodate driveshaft critical speeds. Several of these guidelines, listed in the section titled "Safe Operating Speed" on pages 270 and 271 referring to driveshafts with cardan joints, are as follows:

1) "...the maximum safe operating speed of a shaft is 75 (percent of) Critical speed"
2) "60 inch maximum installed center to center distance of universal joints."
3) "6 deg. Maximum continuous operating angle"

Since the applications discussed here exceed some, or all of these guidelines, an in depth discussion of each of these items will be made.

Maximum Operating Speed is 75% of Critical Speed

This statement can be rephrased to be: "The critical speed of the driveshaft is to be at least 33% higher than the maximum vehicle operating speed". This phrasing is more appropriate for designing a driveshaft since the maximum operating speed is a vehicle-derived parameter independent of the driveshaft.

There has been considerable discussion on the definition of maximum operating speed of the FMTV vehicles. Since the LMTV is the vehicle with the longest driveshaft, and the one most likely to be influenced by critical speed conditions, discussion will be limited to this vehicle and driveshaft.

The LMTV engine is governed at about 57 mph. However, in actual service, speed can substantially exceed the governed speed. As a vehicle is operated near its maximum speed, the governor controls the actual fuel delivered to the engine. Assuming the operator is applying wide-open fuel control, the governor will dispense up to maximum available fuel if the vehicle speed is somewhat less than 58 mph, and it will dispense down to zero fuel if the vehicle speed is somewhat greater than 58 mph. The amount of fuel actually delivered is determined by the difference between the actual speed and the governed speed.

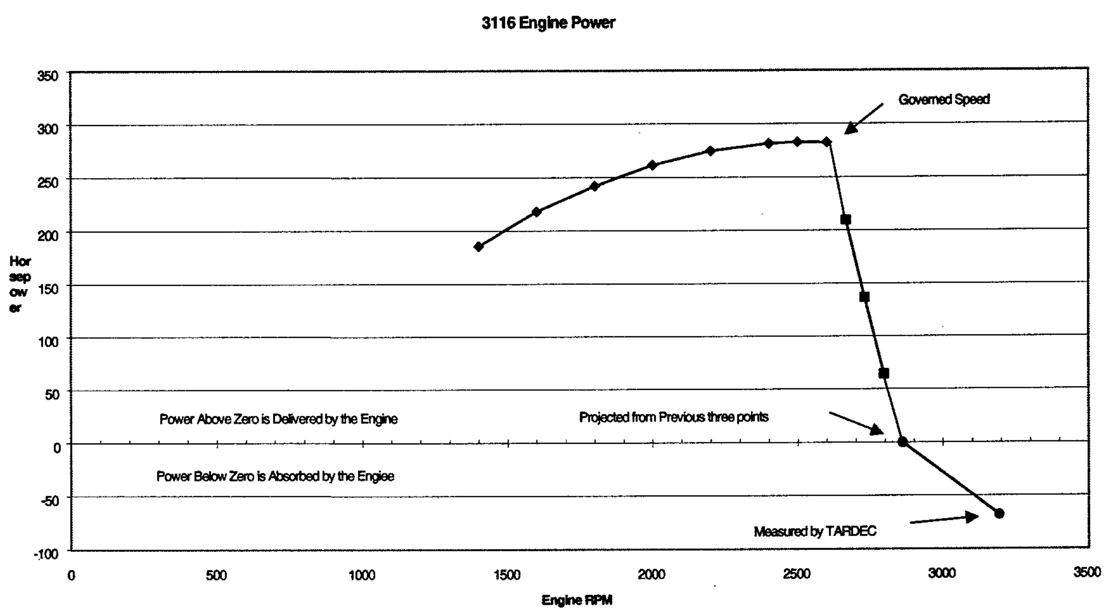
When the vehicle is driving uphill, the governor will not command full fuel until the difference between actual speed and governed speed is about 200 engine rpm. Similarly, if the vehicle is rolling down hill, the governor will not command zero fuel, until the vehicle exceeds governed speed by about 200 engine rpm.

The rpm at which the governor commands minimum fuel is called "high idle" speed. For the Caterpillar 3116 engine, it is about 224 rpm above governed speed, or about 2824 rpm.

For a vehicle rolling down hill, the governor will actually provide additional fuel to the engine up to high idle speed, somewhat helping the vehicle to roll faster than governed speed. It is only when the engine speed is above high idle speed that the engine provides retardation to help slow the vehicle.

Note that a diesel engine does not have a throttle that can limit airflow into the engine. Thus, it has little air pumping loss when it is driven faster than governed speed, even at minimum fuel delivery. The major retardation effect due to driving the engine at higher than governed speed is frictional loss. TARDEC had measured that this loss to be about 68 horsepower at 70 mph. This data for the Caterpillar 3116 engine is summarized in Figure 11.

Figure 11

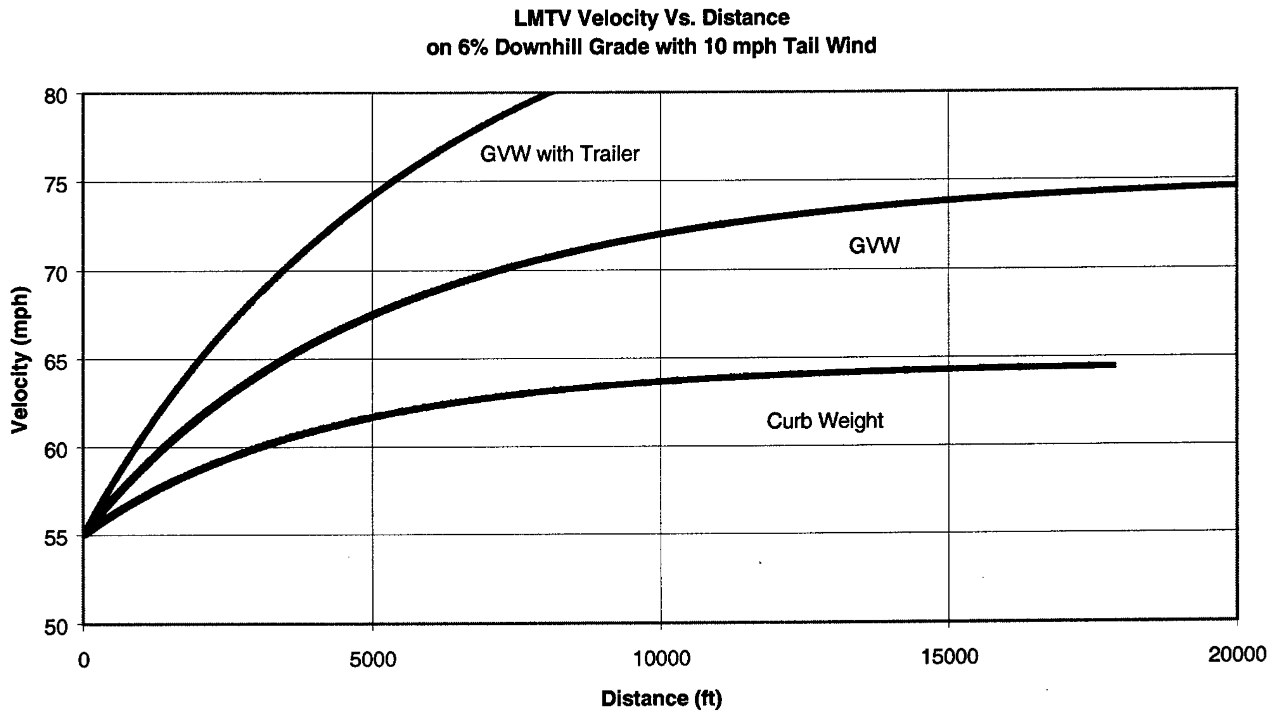


Downhill Speed Modeling

In order to evaluate the potential top speed of the vehicle, a model of a vehicle on a grade was written to calculate vehicle speeds while driving downhill. Modeled effects included tire rolling resistance, aerodynamic drag, wind speed, engine power and losses, road grade, and vehicle weight, including a trailer.

A typical output chart from the model is shown in Figure 12.

Figure 12



It can be seen that an LMTV at GVW, can easily achieve 70 mph on grades found on the US interstate highway system. The AØ vehicles were built with no engine braking hardware. The A1 vehicles have an engine exhaust brake, which can be turned off by the driver. This analysis is representative of all AØ vehicles as well as the A1 vehicles with the exhaust brake off. The use of a trailer further increases speeds.

No sustained road grades have been found in the immediate vicinity of the testing area. However, there is a typical expressway (Interstate 96) within several miles. This portion of the highway has no grades greater than two or three percent slope. The A0 LMTV loaded to GVW with no trailer, reached 64 mph on this road.

Since the analysis showed that 70 mph was an achievable maximum speed under realistic sustained grade conditions occasionally found on US expressways, it was chosen as the top test speed for this program. This was chosen to represent a severe, though not impossible, condition.

60 Inch maximum installed Center-to-Center Distance of Universal Joints

In setting a 60-inch recommended maximum length for driveshafts, the SAE obviously picked an arbitrary value that does not represent the ultimate limit of driveshaft technology.

The authors even go on to say, "Successful applications exceeding (this) parameter have been and are continuing to be made." In fact, the LMTV rear driveshaft at 61.5 inches exceeds this guideline by only 2.5%.

Nevertheless, designers that develop vehicles outside the guidelines of traditional engineering practice have the added responsibility to prove that their designs are sufficiently robust. In general, acknowledgement that the design is outside guidelines, and demonstration of analysis and testing to prove adequacy is appropriate.

Six Degree Maximum Continuous Operating Angle

A cardan joint coupled driveshaft with an operating angle offset from the output shaft, requires that the universal joint cross must articulate at double the offset angle, twice per revolution of the driveshaft. When the driveshaft is operated at a substantial percentage of critical speed, the resulting driveshaft supporting forces must be restrained by the universal joint cross thrust washers, which also articulate through this double angle at twice per revolution. This can result in considerable energy absorption in the thrust washers, leading to heating of the joint. This heating can be minimized by limiting the driveshaft operating angle, as well as by operating at speeds well below critical speed.

Critical Speed Measurements

The most accurate measurement of critical speed is to monitor driveshaft mounting forces or accelerations and to actually spin the driveshaft through critical speed. This provides the most accurate conditions of damping, vibration nodes, and constraints. There are very recognizable observations that can be made when a driveshaft spins through its critical speed. The most obvious are: 1) The forces or accelerations increase exponentially with rpm as critical speed is approached. The only limitations on actual maximum force are set by the damping and other losses in the driveshaft supporting structure. 2) There is an abrupt drop in force to a relatively constant value independent of rpm just after the driveshaft passes through critical. This can be contrasted with a resonant frequency mode, where the drop in force is less abrupt and it continues to drop as frequency is increased. 3) There is an abrupt 180° phase shift in the angle of the force relative to the driveshaft angular position, just as the driveshaft passes through critical.

Most driveshaft manufacturers design driveshafts for multiple vehicle applications, and often don't have vehicles for measuring driveshafts as installed. Thus, driveshafts are often measured in the free-free state by the manufacturers. This is considered a good first approximation for estimating and comparing driveshafts, as long as the limitations of this measurement are considered. It is generally understood that if the free-free frequency is too low, installation in a vehicle is likely to make the frequency even lower.

Almost all driveshaft resonant frequency measurements are made with a non-rotating driveshaft, while the critical speed condition results while the driveshaft is rotating. Many of the causes of the differences between critical speed frequency and resonant speed frequency are related to this disparity.

In most conditions, it is not possible to actually measure a driveshaft as it passes through critical speed. Usually this happens at a speed higher than is achievable by the vehicle. If it is achievable, there is a concern that the forces will become destructive in nature leading to fatigue or fracture of supporting structures, or to separation of the driveshaft from the vehicle.

Thus, in most cases, driveshaft critical speeds are estimated by measuring the driveshaft first mode resonant frequency. The underlying equations of motion are nearly identical for first mode resonant frequency and for critical speed. The actual in vehicle operating conditions, though, are quite different, and some measurements of first mode resonant frequency may not be very good estimates of driveshaft critical speed.

First Mode Resonant Frequency Measurements Methods

Free-Free Hammer Test

In this test, the driveshaft is suspended from elastic constraints and impacted with a hammer. Response of the driveshaft is measured with accelerometers mounted on the driveshaft in one or more locations. A Fourier analyzer or oscilloscope is used to measure the frequency. This is one of the least accurate methods, as the driveshaft is not rotating, and is not constrained in the same manner as it is in the vehicle. This usually measures frequencies that are higher than critical speed by ten to 100 percent. It is difficult to provide a simulation of driving torque while conducting this test. This test is considered a first approximation for critical speed and can exclude driveshafts that measure too low, but does not necessarily approve driveshafts that measure to have adequate margin.

In Vehicle Hammer Test

This test is similar to the free-free test, except that the driveshaft is mounted in its actual vehicle environment, though not rotating. This test usually gives an accurate estimate of resonant frequency. It is generally considered to give inaccurate damping estimates. The damping observed in this test, and other resonant frequency tests, relates to the internal driveshaft damping. The actual critical speed damping is derived more from external damping in the vehicle powertrain, and driveshaft internal damping is a minor factor. This type of test also has vibration amplitudes that are usually quite reduced from amplitudes of motion at actual critical speed conditions since a hammer blow generally excites only one or two thousandths of an inch of vibration amplitude while the actual critical speed condition often has several tenths of an inch of driveshaft bending. It is possible to apply an appropriate level of static torque on the driveshaft while conducting this test.

In Vehicle Shaker Test

This test has the driveshaft properly mounted in the vehicle, but differs from an actual critical speed test in that the vibratory force is provided by a shaker that is swept through the resonant frequency. The driveshaft is not rotating. This test can have appropriate vibration amplitudes given that an adequate shaker force level is available. It has the same errors in assessing damping that other resonant frequency tests exhibit. It provides a good estimate of the correct frequency. It is also possible to apply an appropriate level of static torque on the driveshaft while conducting this test. See Figure 13 for a photograph of this test setup.

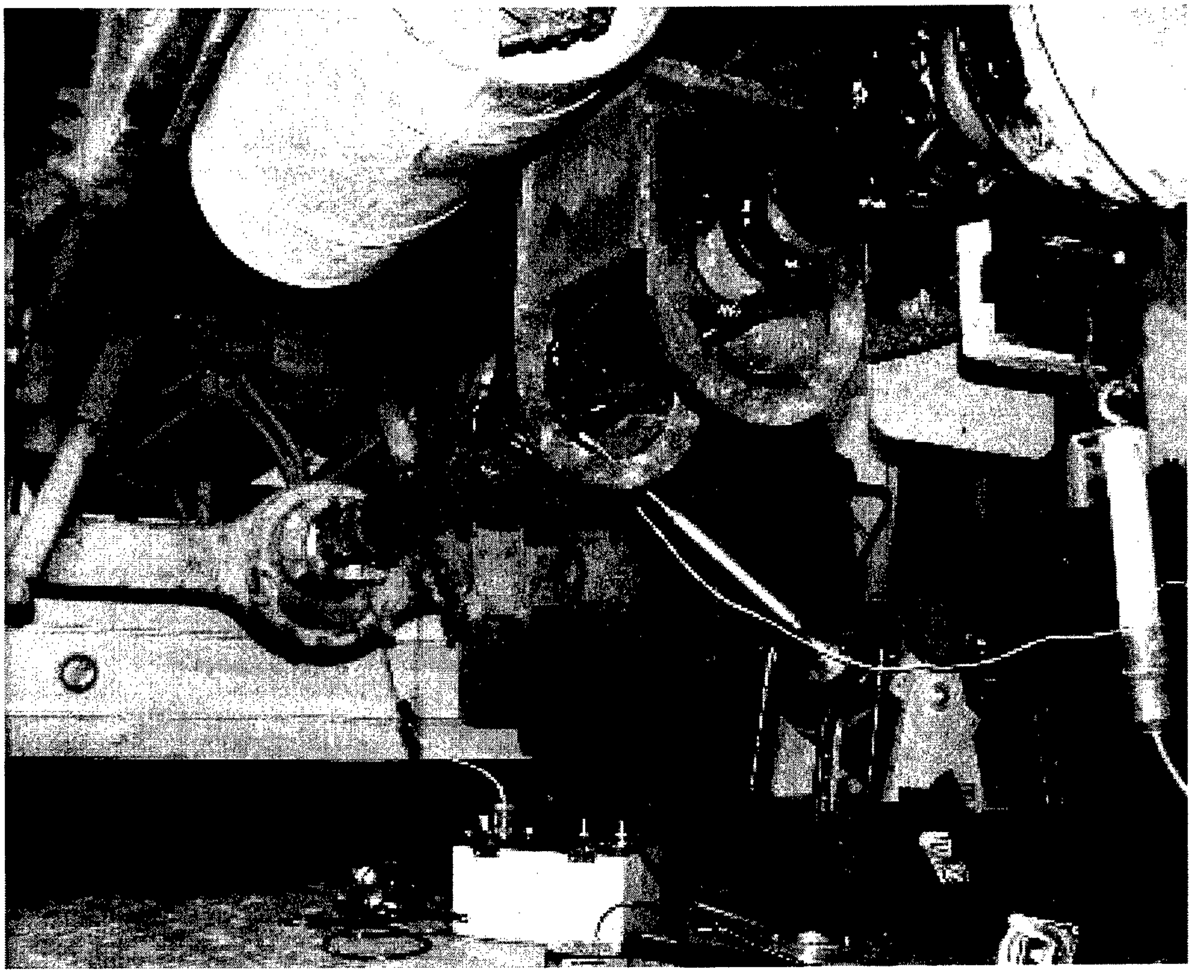
MSC DADS Model

Since the DADS model can be run at any speed, it was used to run each driveshaft through critical speed. The DADS model used accurate measurements and calculations of driveshaft components, and it used measured data for damping, so it produces reasonably accurate estimates of both critical speed and forces.

Critical Speed Measured in the Vehicle

The most accurate measurement of critical speed is to actually drive up to, and through critical speed in the vehicle and observe the response of the driveshaft and vehicle with data measurements. In vehicles that meet SAE design guidelines, though, the vehicle will not achieve speeds high enough to reach driveshaft critical speed.

Figure 13 In-Vehicle Shaker Test



It is necessary to provide guards to prevent excessive vehicle damage if the vehicle is to be driven near critical speed. The LMTV was fitted with appropriate guards and measurement channels for this test.

Note that even actual critical speed observations on a vehicle are not always a single number. Variability in actual measured frequency occurs due to driveline torque differences and slip spline location differences, as well as other reasons.

Since the LMTV rear driveshafts were the most important from a critical speed perspective, all methods were used to provide critical speed data this driveshaft. The only actual critical speed measurements were made on the Arvin Meritor A1 and RPL-20 driveshafts, which exceeded critical speed during testing. Figures 14 to 17 illustrate these data. Note the 180° phase shift (relative to the driveshaft position sensor) in the transfer case force transducer before and after the critical speed occurrence in figures 15 and 17. The summary of the critical speed and resonant speed test measurements is listed in Tables 9 and 10

Figure 14

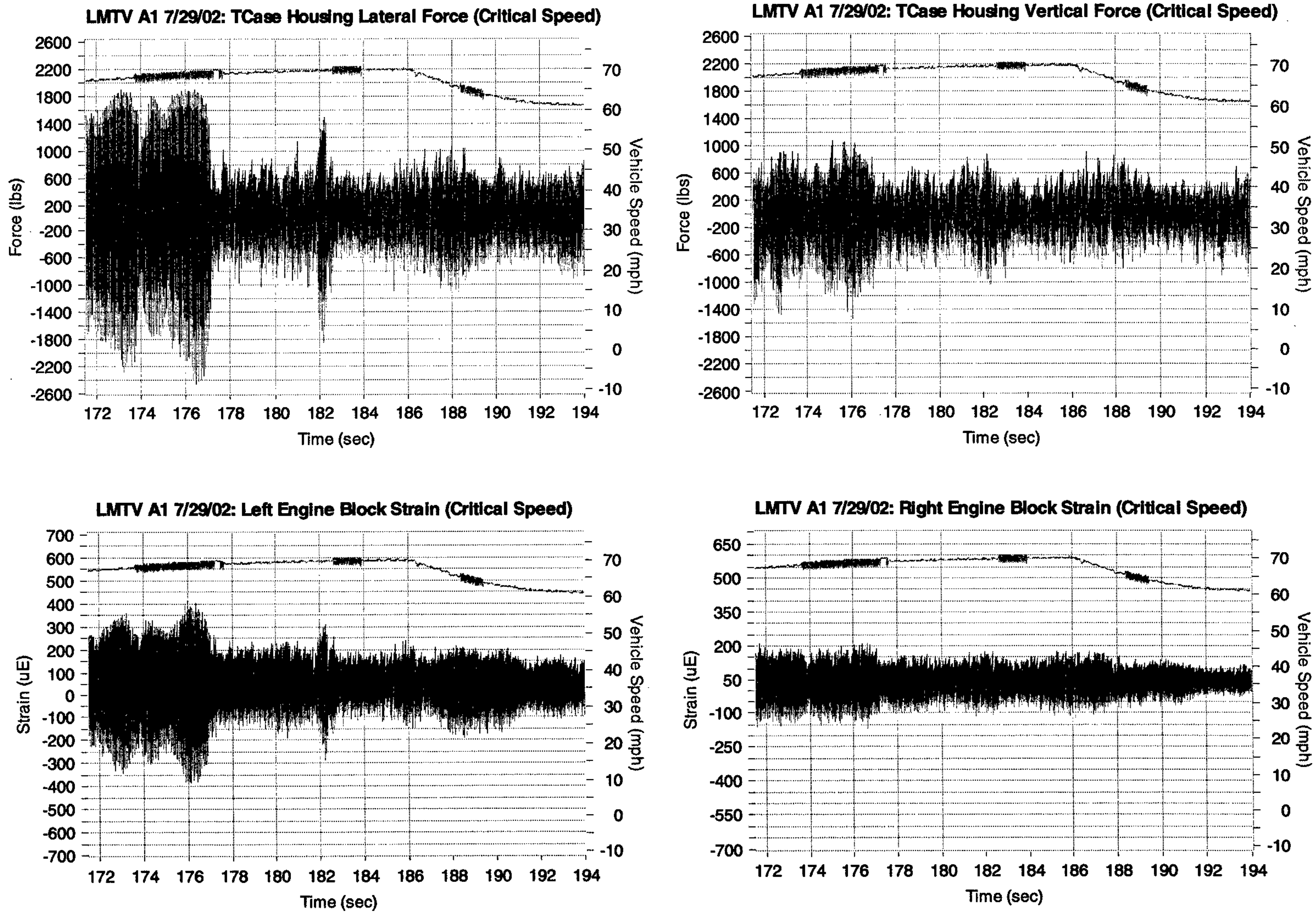
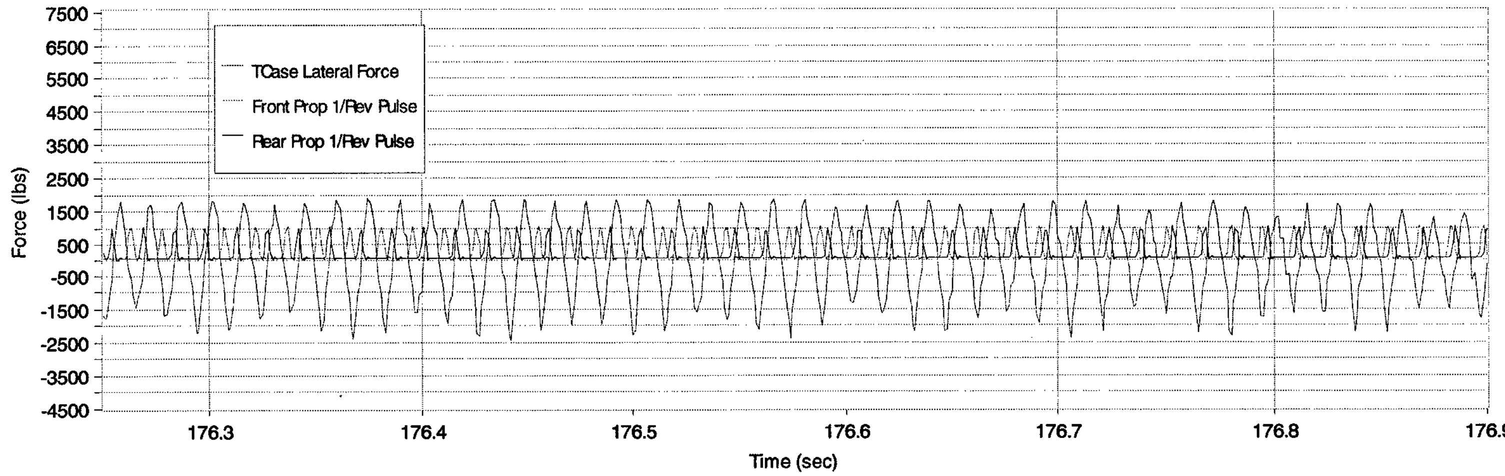

LMTV A1 7/29/02: TCase Housing Lateral Force (Critical Speed)
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV A1 7/29/02: TCase Housing Vertical Force (Critical Speed)
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV A1 7/29/02: Left Engine Block Strain (Critical Speed)
Strain (uE)
Vehicle Speed (mph)
Time (sec)
LMTV A1 7/29/02: Right Engine Block Strain (Critical Speed)
Strain (uE)
Vehicle Speed (mph)
Time (sec)

Figure 15

LMTV A1 7/29/02: Critical Speed at 176.25 to 176.9



LMTV A1 7/29/02: Critical Speed at 176.9 to 177.5

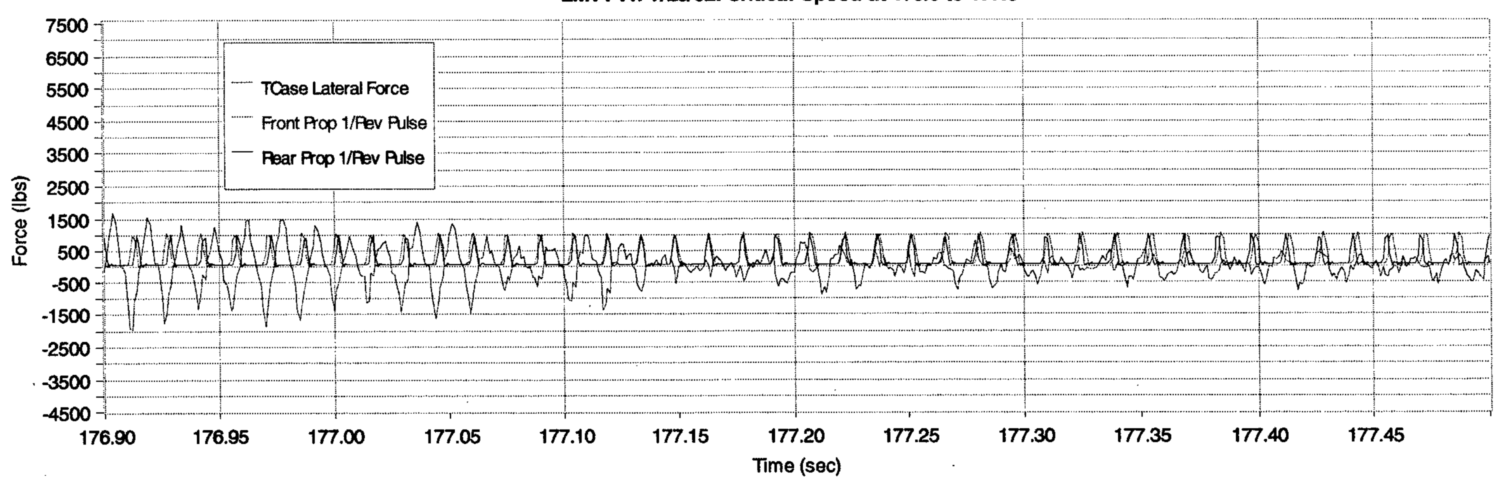
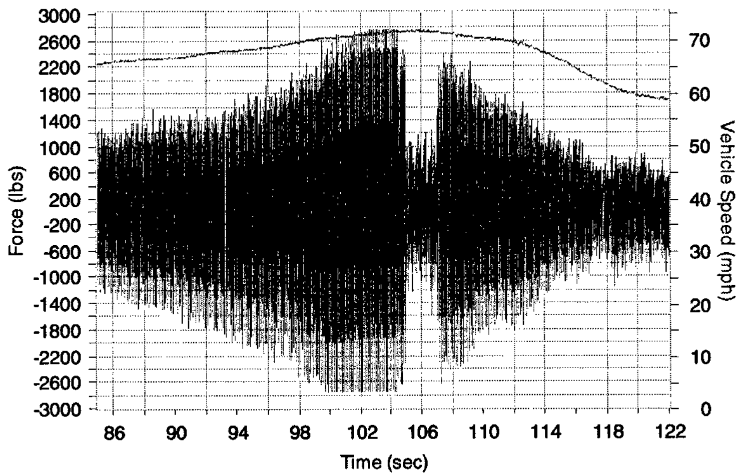
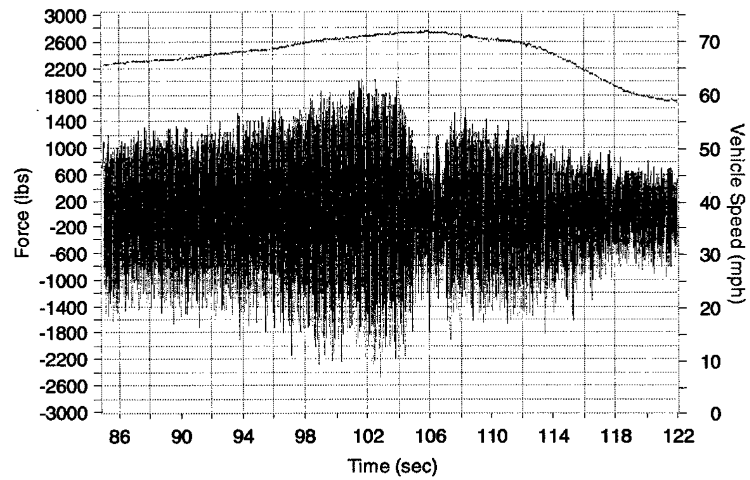


Figure 16

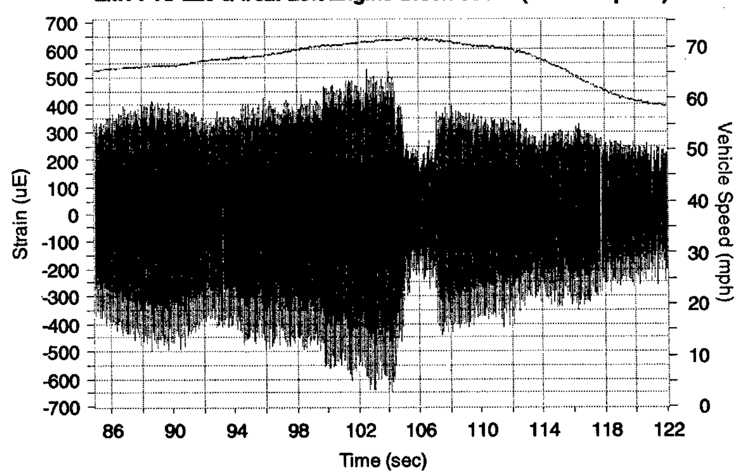
LMTV RPL20 5/1/02: TCase Housing Lateral Force (Critical Speed)



LMTV RPL20 5/1/02: TCase Housing Vertical Force (Critical Speed)



LMTV RPL20 5/1/02: Left Engine Block Strain (Critical Speed)



LMTV RPL20 5/1/02: Right Engine Block Strain (Critical Speed)

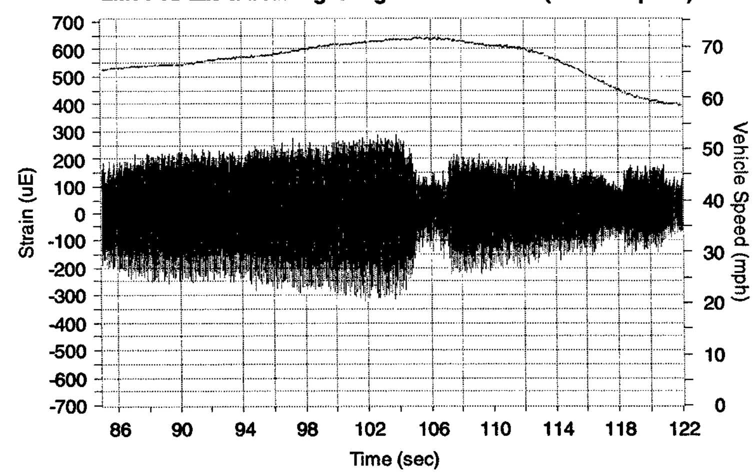
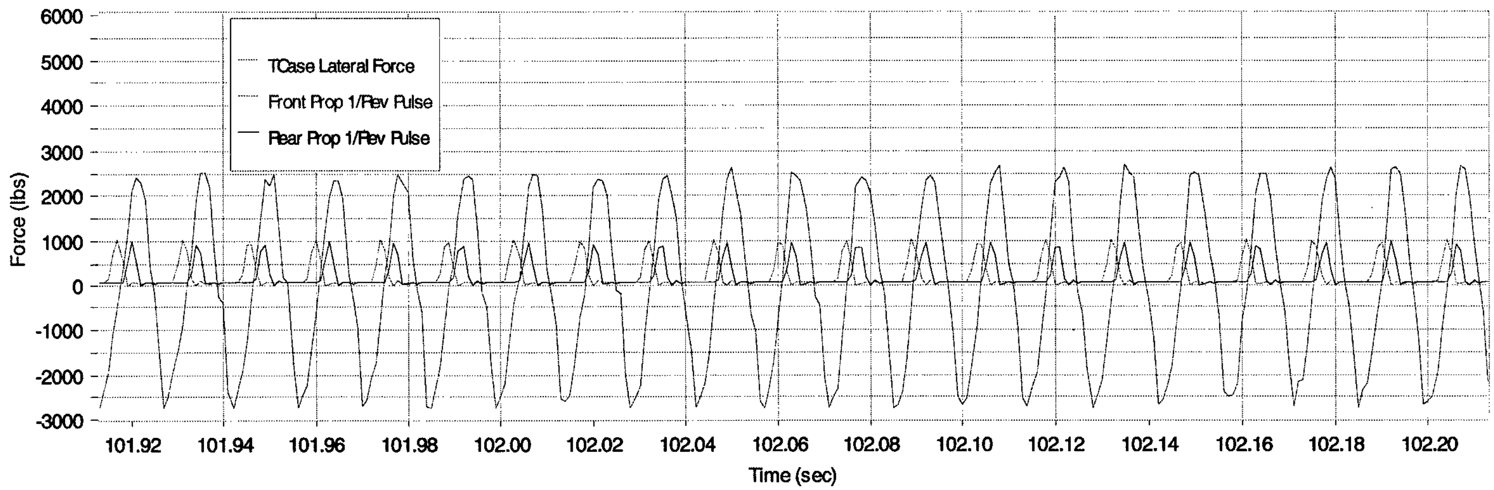


Figure 17

LMTV RPL20 5/1/02: Critical Speed at 101.92 to 102.2 seconds



LMTV RPL20 5/1/02: Critical Speed at 104.88 to 105.08 seconds

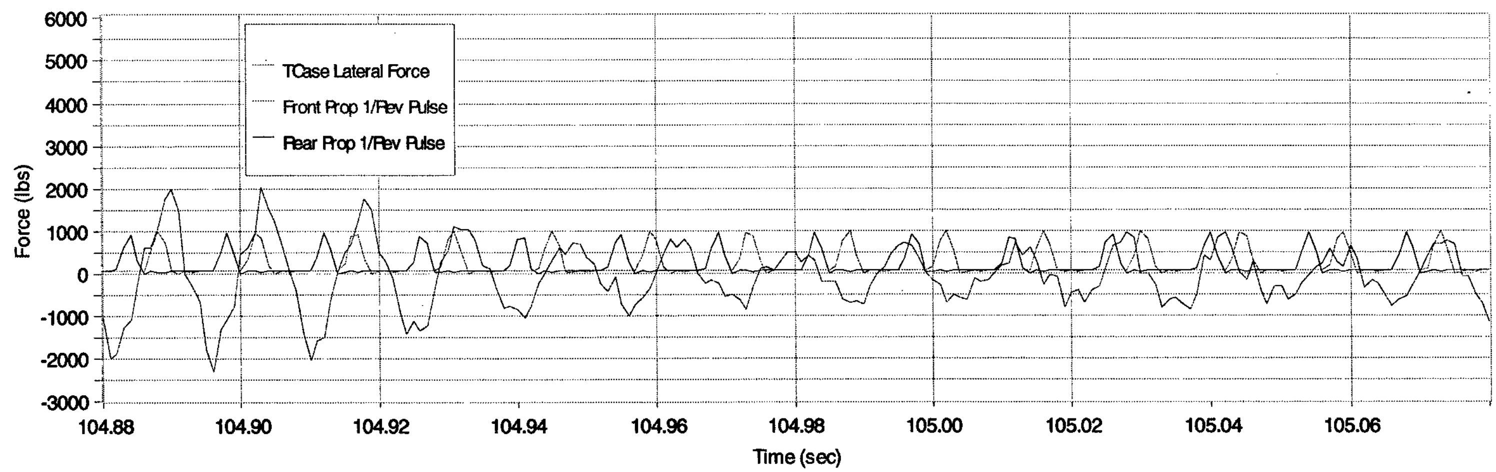


Table 9
Resonant Frequency Test Data - Hz

Driveshaft	Free-Free Hammer	In-Vehicle Hammer	In-Vehicle Shaker	In-Vehicle Critical Speed	MSC DADS Model
ARM A1	147	69	68	67	68
ARM RPL-20	139	64-70	68	67	69
Dana SPL-140	156	92	87-90	*	87
GKN CV	160	106	102	*	101

*Not reached in actual testing

In order to provide a method of comparing all of the driveshafts, a free-free resonant frequency test was conducted on a driveshaft submitted for each position. The data are reported in Table 10

Table 10

Driveshaft Free-Free Hammer Test Resonant Frequency

	Arvin Meritor A1	Arvin Meritor RPL-20	Dana SPL 140	GKN
FMTV Front	227Hz	227Hz	278Hz	250Hz
LMTV Rear	147Hz	139Hz	156Hz	160Hz
MTV Intermediate	178Hz	179Hz	277Hz	208Hz
MTV Rear-Rear	208Hz	208Hz	202Hz	*

* Not available at time of test

Driveshaft Inertial Acceleration

As previously stated, a cardan joint coupled driveshaft with an operating angle offset from the output shaft, requires articulation at double the offset angle, twice per revolution of the driveshaft. This articulation results in acceleration and deceleration of the rotating parts twice per revolution. The term "Inertial Acceleration" is used to identify this phenomenon. Note that Constant Velocity (CV) universal joints have zero inertial acceleration. This is inherent to the fact that they are, by design, constant velocity.

The inertial acceleration is approximated by squaring the operating angle and multiplying by the square of the rpm. When these units are in radians, the resulting torsional acceleration is in units of radians per second squared (rad/sec^2). The SAE AE-7 manual limits the inertial acceleration to "approximately 1000 rad/sec^2 in any continuous operating position". The manual goes on to state: "Certain highway, as well as off-highway vehicles may tolerate…higher levels of excitation." SAE AE-7 goes on to say that 2000 rad/sec^2 may be a "reasonable limit" in those cases. (Page 61)

The FMTV has driveshafts that operate at considerably higher Inertial Accelerations. Measurements on early build vehicles yielded the following observed average driveshaft angles. These angles result in the indicated Inertial Accelerations for cardan joint couplings at governed speed of 2600 engine rpm (3320 driveshaft rpm in seventh gear lockup). Note that observed maximum angles (due to build variation) result in Inertial Accelerations more than 1000 rad/sec^2 higher than the mean values.

Position	Average Angle	Inertial Acceleration (rad/sec^2)		
		Mean	SAE Recommended	SAE Maximum
All Vehicle Front	10.9°	4400	1000	2000
LMTV Rear	11.5	4900	1000	2000
MTV Intermediate	7.4	2000	1000	2000
MTV Rear-Rear	6.9	1800	1000	2000

There are several consequences for having large Inertial Accelerations. The driveshafts always turn at an rpm related to tire speed, and thus, have high Inertial Accelerations at high vehicle speeds regardless of the transmission gear. This can result in annoying "noise periods" or "buzz" in the vehicle at highway speeds (typically occurring at frequencies from 80 to 120 Hz). These noise periods often "tune-up" or worsen at certain speeds as resonances in the suspension or frame are excited. These are often occupant annoyances rather than serious mission jeopardy problems.

However, Inertial Accelerations, acting upon the rotational inertia of the driveshafts, result in oscillating torques in the drivetrain. These high frequency oscillating torques can effect the durability and reliability of other drivetrain components. Powertrain component suppliers indicated some reluctance to give application approval to components subjected to the high torques resulting from these high Inertial Accelerations.

As was discussed with other SAE guidelines, meeting, and/or exceeding, SAE guidelines is left to the responsibility of the appropriate engineering organizations. There are many successful designs that exceed SAE recommended practice. However, it is the responsibility of the engineering group that provides these designs to take note of that fact, and to demonstrate the adequate robustness of the application.

This is important in light of the driveshafts tested for this program. Calculated values, as determined form dimensional analysis, for rotational inertia about the axis of rotation for the tested LMTV rear driveshafts are as follows:

Driveshaft	Torsional Inertia	Percent Increase over AØ
Arvin Meritor AØ	.452 in-lb-sec^2 per radian	0%
Arvin Meritor A1	.488 in-lb-sec^2 per radian	8%
Arvin Meritor RPL-20	.776 in-lb-sec^2 per radian	72%
Dana SPL-140	.639 in-lb-sec^2 per radian	41%

The GKN driveshaft is not listed since it has zero Inertial Acceleration.

Since the Inertial Acceleration is determined solely by the angle and rpm, the torque resulting from the acceleration is directly proportional to the Torsional Inertia. Thus, it would be expected that the Arvin Meritor A1 driveshaft would have 8% higher oscillating torques compared with the originally approved application. The Arvin Meritor RPL-20 would have 72% higher oscillating torques, and the Dana SPL-140 would have 41% higher oscillating torques. It is not known what affect these higher oscillating torques may have on supplier application approvals.

Summary of Test Data

Transfer Case Forces

A principal focus of this project was driveshaft support force data as measured by the transfer case force transducer. This transducer was designed and modeled to assure that it accurately measures driveshaft-supporting forces, and to assure that it does not significantly alter the system dynamics. See page 21 for details on the transducer

Figures 18 and 19 show transfer case supporting forces and engine strains for the A1 driveshaft on the LMTV early in the testing (39 miles) compared with the same forces and strains late in the test (1027 miles). Note how the forces and strains have increased as the testing progressed. The transfer case forces exceeded the 2000 lb limit in Figure 19, even though 70 mph was not reached.

Figures 20 an21 show transfer case supporting forces and engine strains for the RPL-20 driveshaft on the LMTV early in the testing (308 miles) compared with the same forces and strains later in the test (768 miles). Note how the forces and strains have increased as the testing progressed. The transfer case forces exceeded 2000 lb. in the high-speed runs. Data from the high speed runs cannot be compared from run to run since the actual speed reached in each trial was different.

Figures 22 to 25 show similar measurements for the Dana and GKN driveshafts. Note that the GKN driveshafts had the lowest forces, and the Dana was next lowest.

The Arvin Meritor RPL-20 and Arvin Meritor A1driveshafts produced transfer case dynamic forces that exceeded the 2000 lb transducer limit during the high-speed portion of the testing. The Dana and GKN driveshafts produced roughly similar transfer case forces during 58 mph runs. The GKN driveshaft produced lower forces than the Dana driveshaft, during the runs above 58 mph.

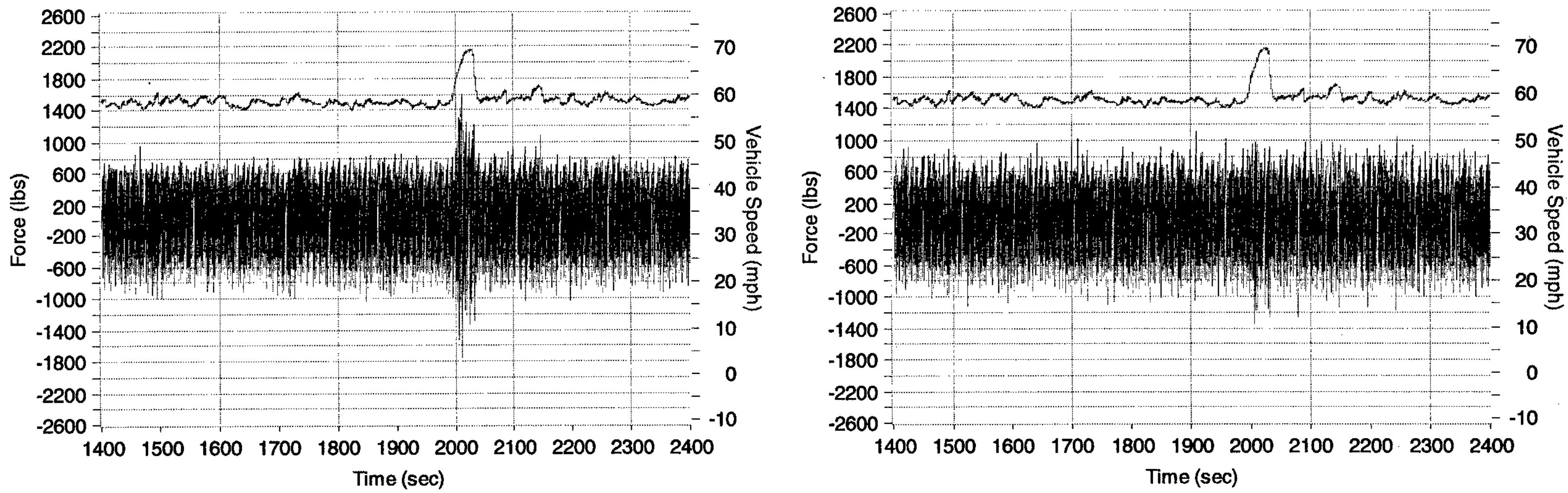
Figures 25 to 28 illustrate the transfer case forces from the four driveshaft candidates on the MTV during the 58 mph portion of the runs. This driveshaft position is not influenced by driveshaft dynamic forces so the 58 mph comparisons are similar to the high-speed comparisons. Note that the trends are similar. The lowest force driveshaft is the GKN, followed by the Arvin Meritor A1, the Dana SPL-140, and the Arvin Meritor RPL-20, respectively.

Approximately 80 gigabytes of test data were recorded during the testing. The complete data files are available on removable hard drives stored in MSC archives. Summaries of data were plotted for analysis and organized in three ring binders. These are also stored in MSC archives.

Figure 18

LMTV A1(39 miles into test): TCase Housing Lateral Force

LMTV A1(39 miles into test): TCase Housing Vertical Force



LMTV A1(39 miles into test): Left Engine Block Strain

LMTV A1(39 miles into test): Right Engine Block Strain

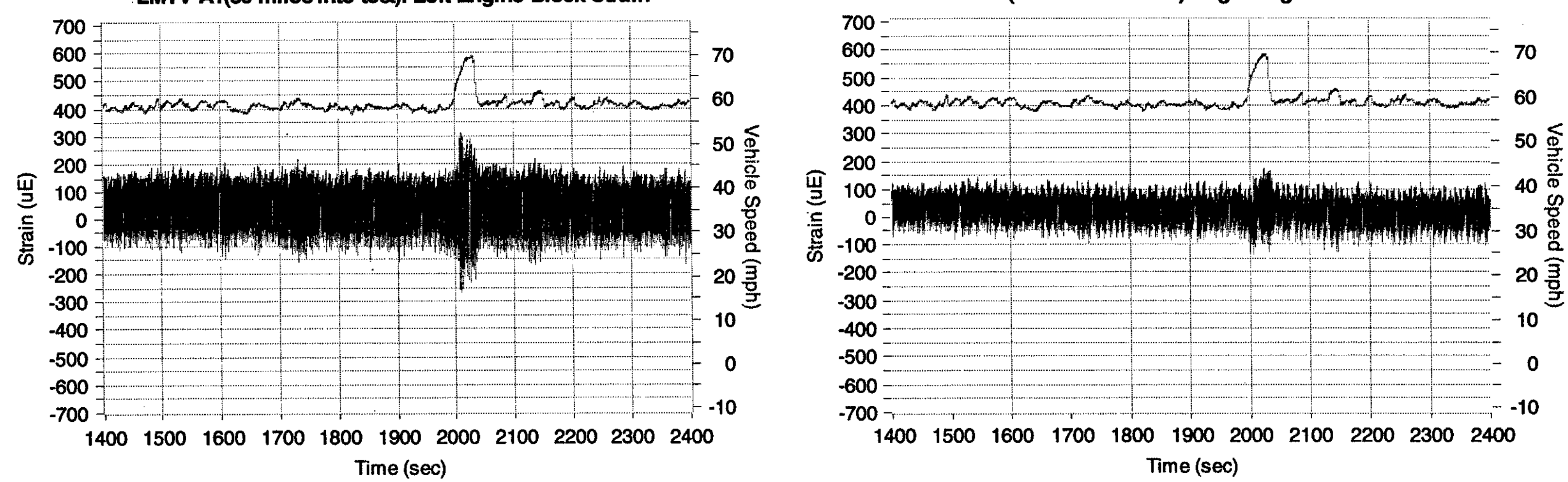
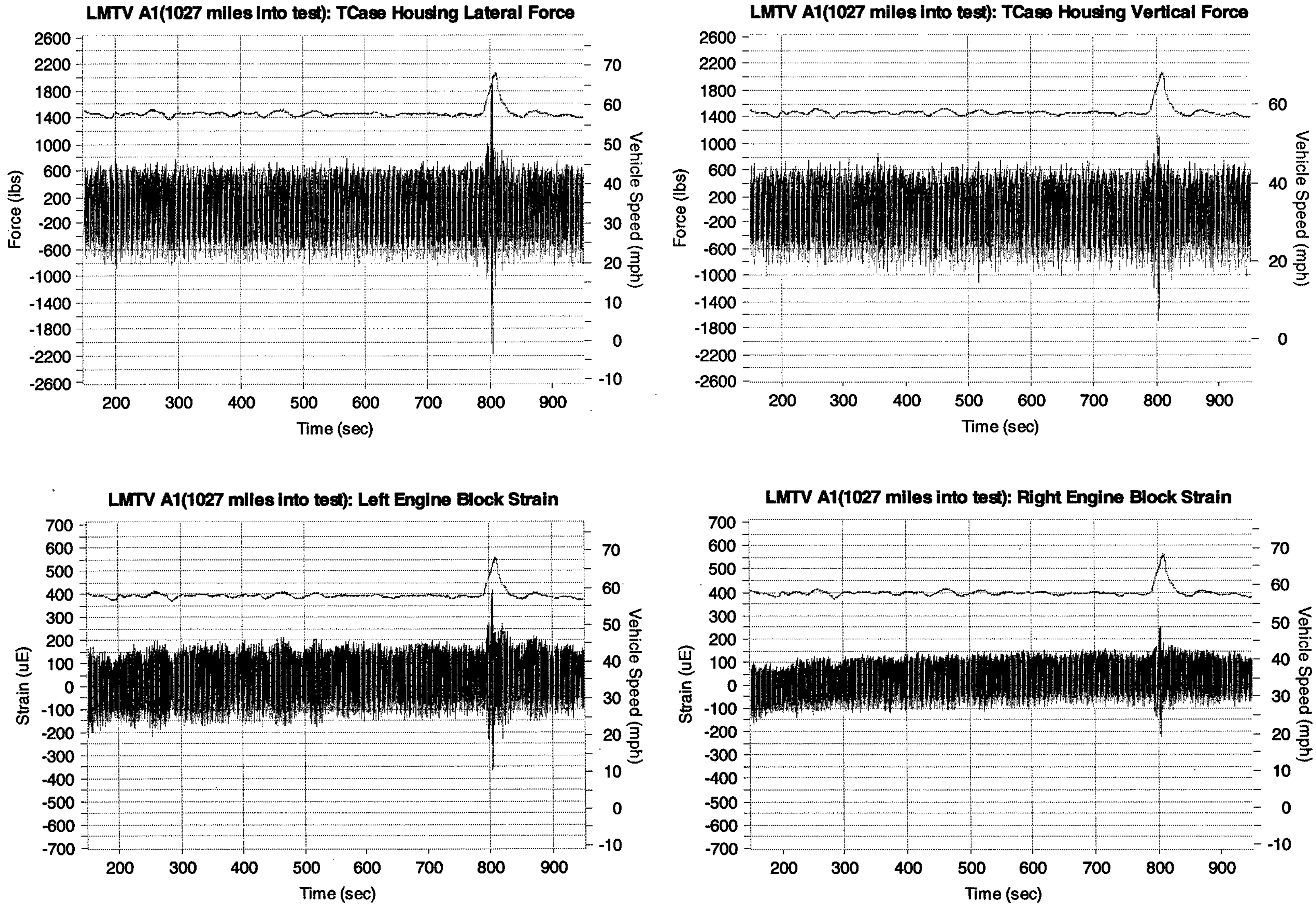
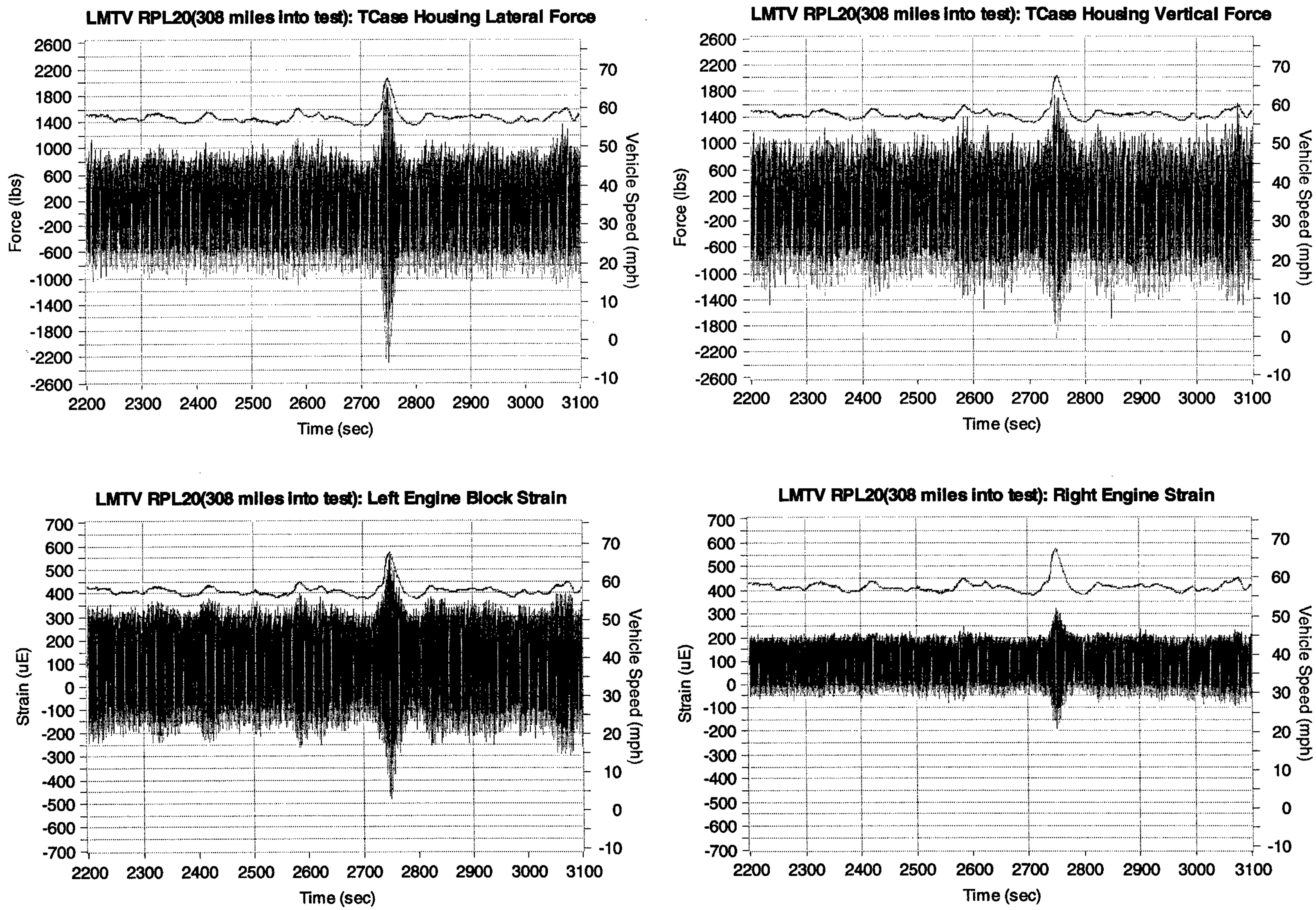


Figure 19



LMTV A1(1027 miles into test): TCase Housing Lateral Force
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV A1(1027 miles into test): TCase Housing Vertical Force
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV A1(1027 miles into test): Left Engine Block Strain
Strain (uE)
Vehicle Speed (mph)
Time (sec)
LMTV A1(1027 miles into test): Right Engine Block Strain
Strain (uE)
Vehicle Speed (mph)
Time (sec)

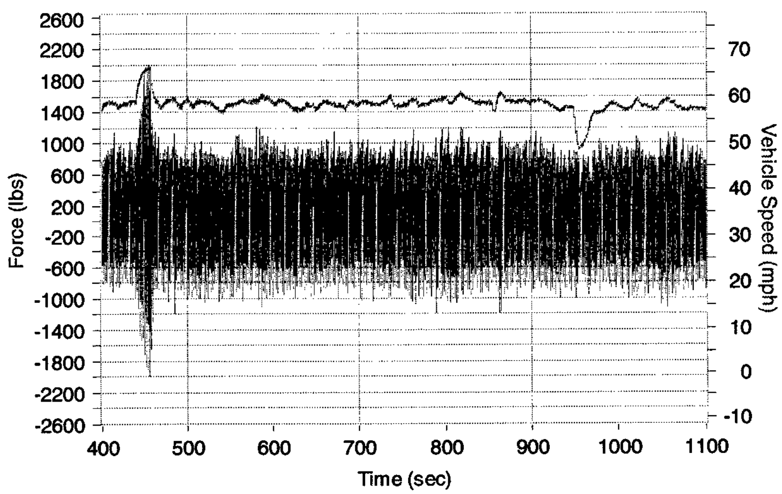
Figure 20



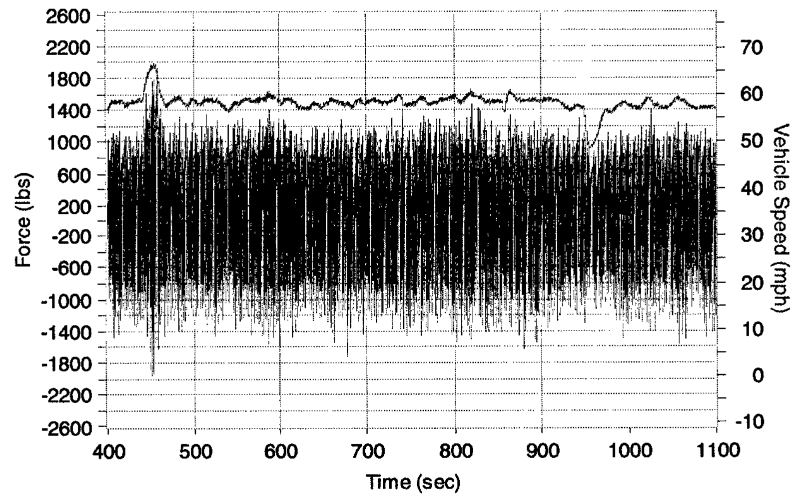
LMTV RPL20(308 miles into test): TCase Housing Lateral Force
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV RPL20(308 miles into test): TCase Housing Vertical Force
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV RPL20(308 miles into test): Left Engine Block Strain
Strain (uE)
Vehicle Speed (mph)
Time (sec)
LMTV RPL20(308 miles into test): Right Engine Strain
Strain (uE)
Vehicle Speed (mph)
Time (sec)

Figure 21

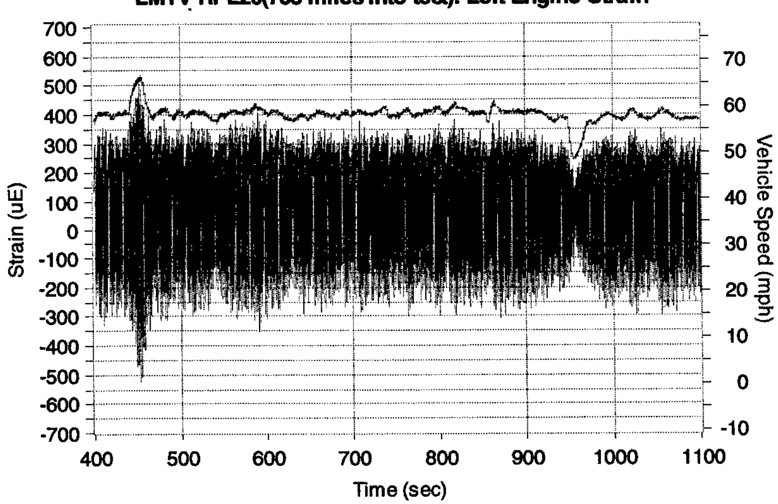
LMTV RPL20(768 miles into test): TCase Housing Lateral Force



LMTV RPL20(768 miles into test): TCase Housing Vertical Force



LMTV RPL20(768 miles into test): Left Engine Strain



LMTV RPL20(768 miles into test): Right Engine Strain

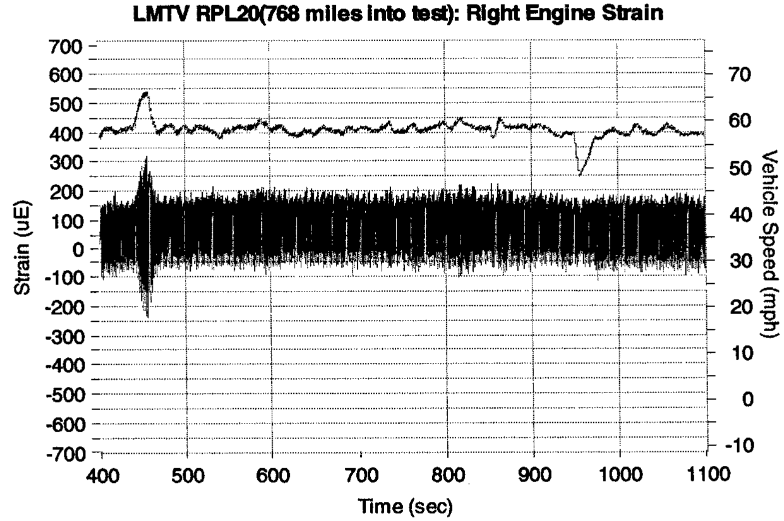
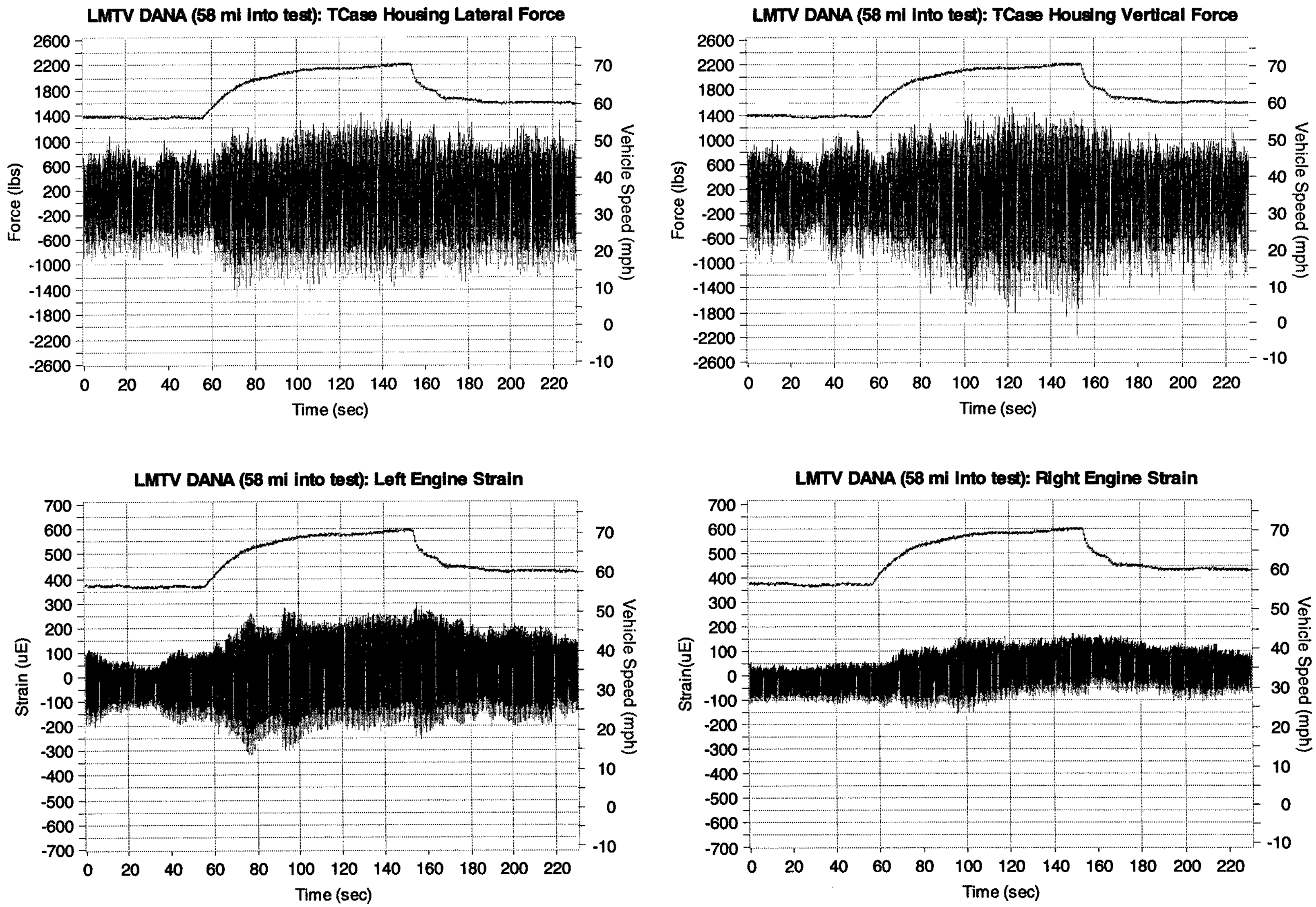


Figure 22



LMTV DANA (58 mi into test): TCase Housing Lateral Force
LMTV DANA (58 mi into test): TCase Housing Vertical Force
LMTV DANA (58 mi into test): Left Engine Strain
LMTV DANA (58 mi into test): Right Engine Strain
Force (lbs)
Strain (uE)
Strain(uE)
Vehicle Speed (mph)
Time (sec)

Figure 23

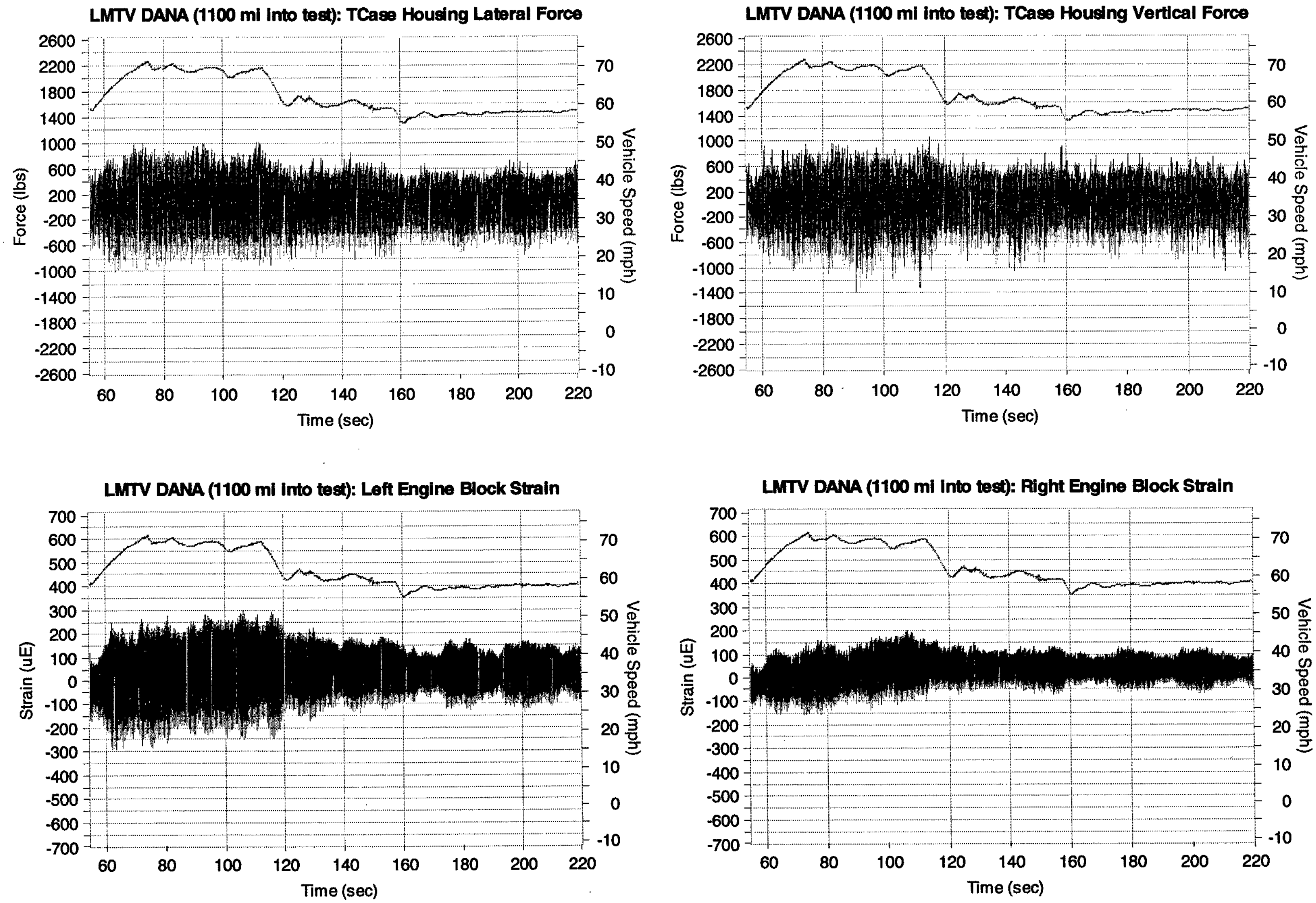

LMTV DANA (1100 mi into test): TCase Housing Lateral Force
LMTV DANA (1100 mi into test): TCase Housing Vertical Force
LMTV DANA (1100 mi into test): Left Engine Block Strain
LMTV DANA (1100 mi into test): Right Engine Block Strain
Force (lbs)
Strain (uE)
Vehicle Speed (mph)
Time (sec)

Figure 24

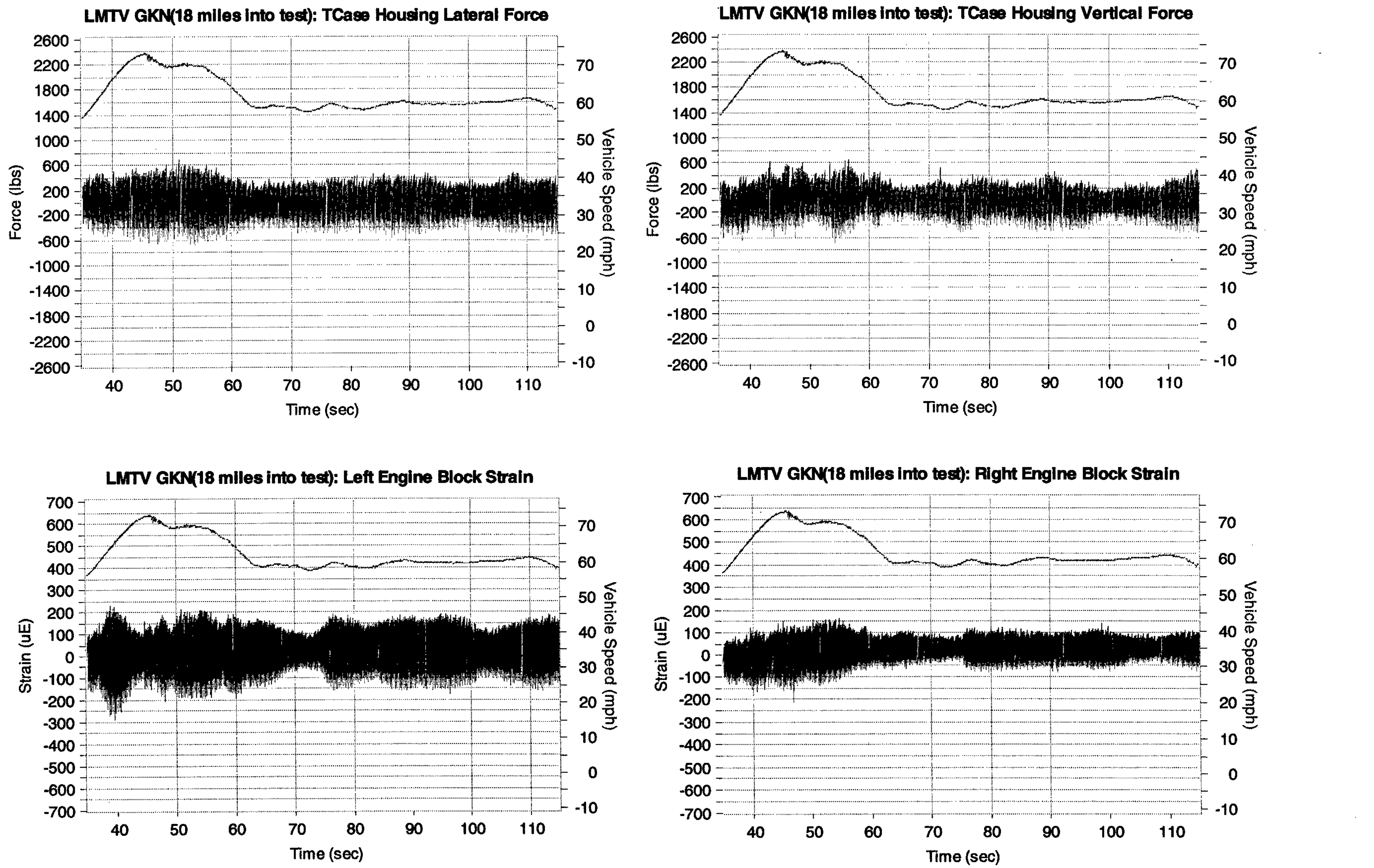

LMTV GKN(18 miles into test): TCase Housing Lateral Force
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV GKN(18 miles into test): TCase Housing Vertical Force
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV GKN(18 miles into test): Left Engine Block Strain
Strain (uE)
Vehicle Speed (mph)
Time (sec)
LMTV GKN(18 miles into test): Right Engine Block Strain
Strain (uE)
Vehicle Speed (mph)
Time (sec)

Figure 25

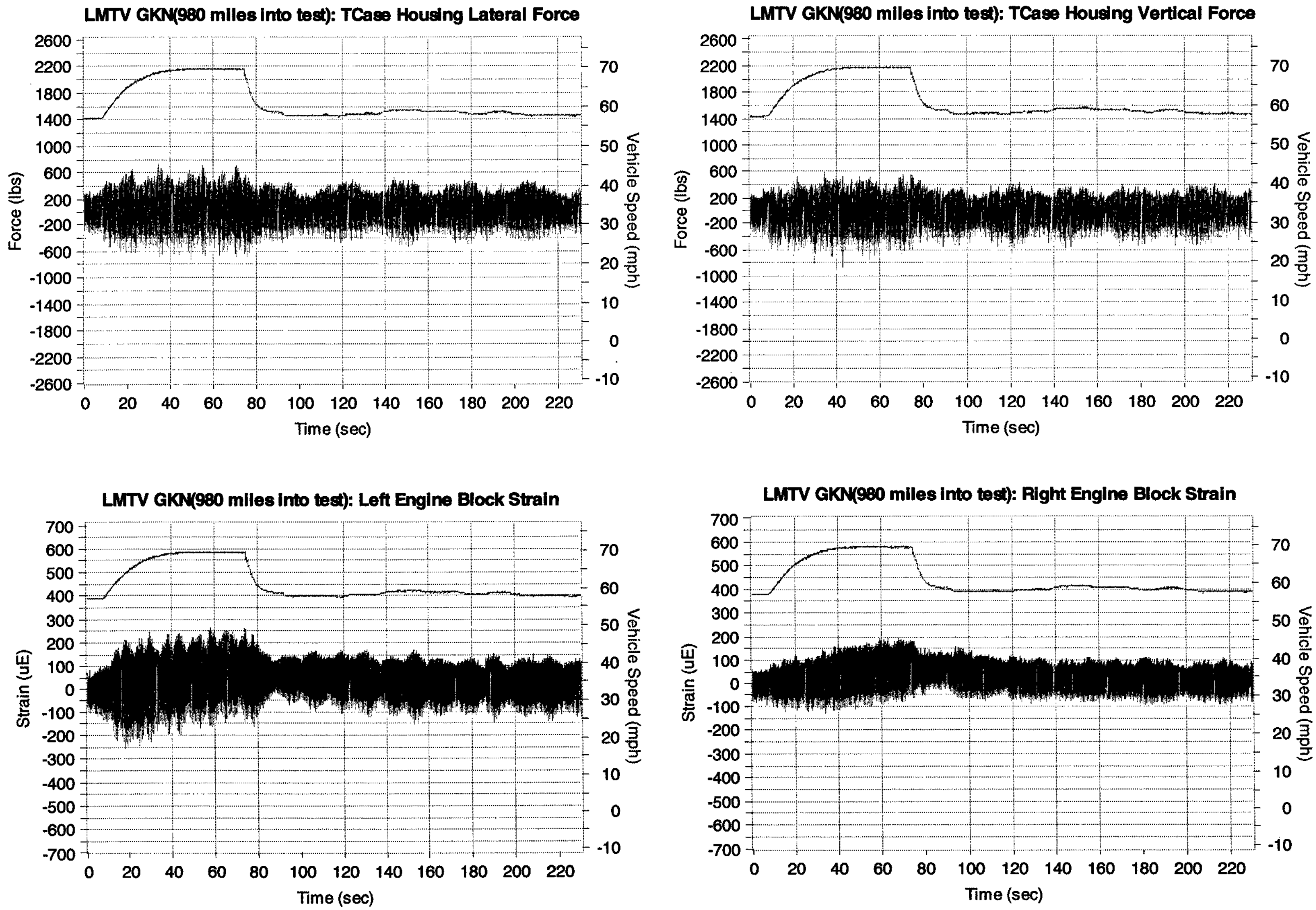
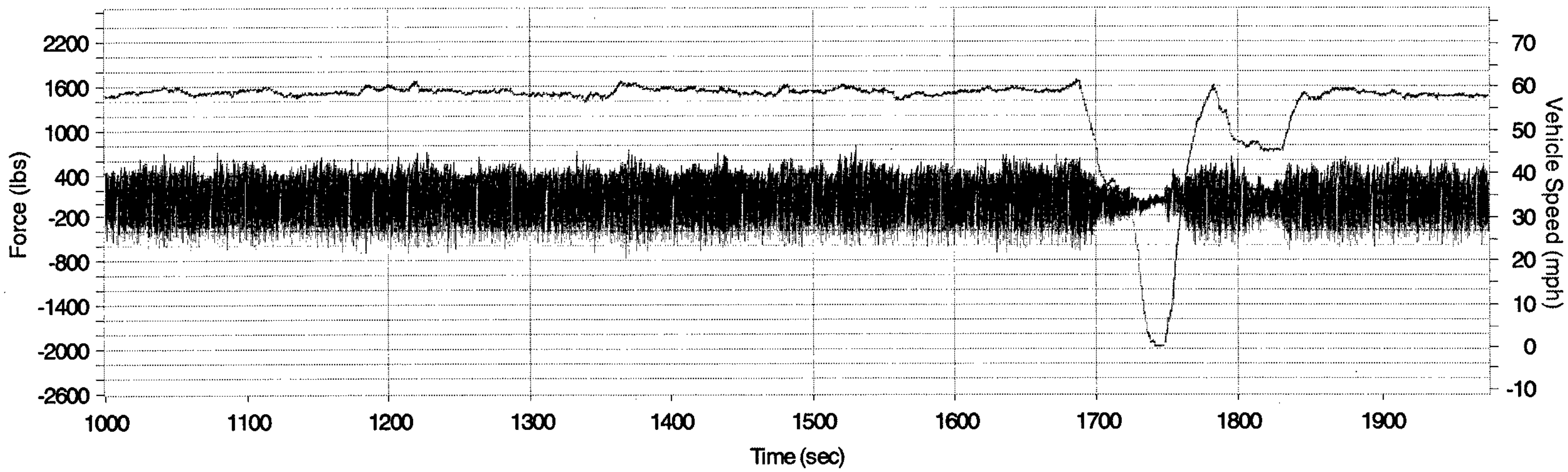

LMTV GKN(980 miles into test): TCase Housing Lateral Force
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV GKN(980 miles into test): TCase Housing Vertical Force
Force (lbs)
Vehicle Speed (mph)
Time (sec)
LMTV GKN(980 miles into test): Left Engine Block Strain
Strain (uE)
Vehicle Speed (mph)
Time (sec)
LMTV GKN(980 miles into test): Right Engine Block Strain
Strain (uE)
Vehicle Speed (mph)
Time (sec)

Figure 26

MTV A1 (8/21/02 test): Transfer Case Housing Lateral Force



MTV A1 (8/21/02 test): Transfer Case Housing Vertical Force

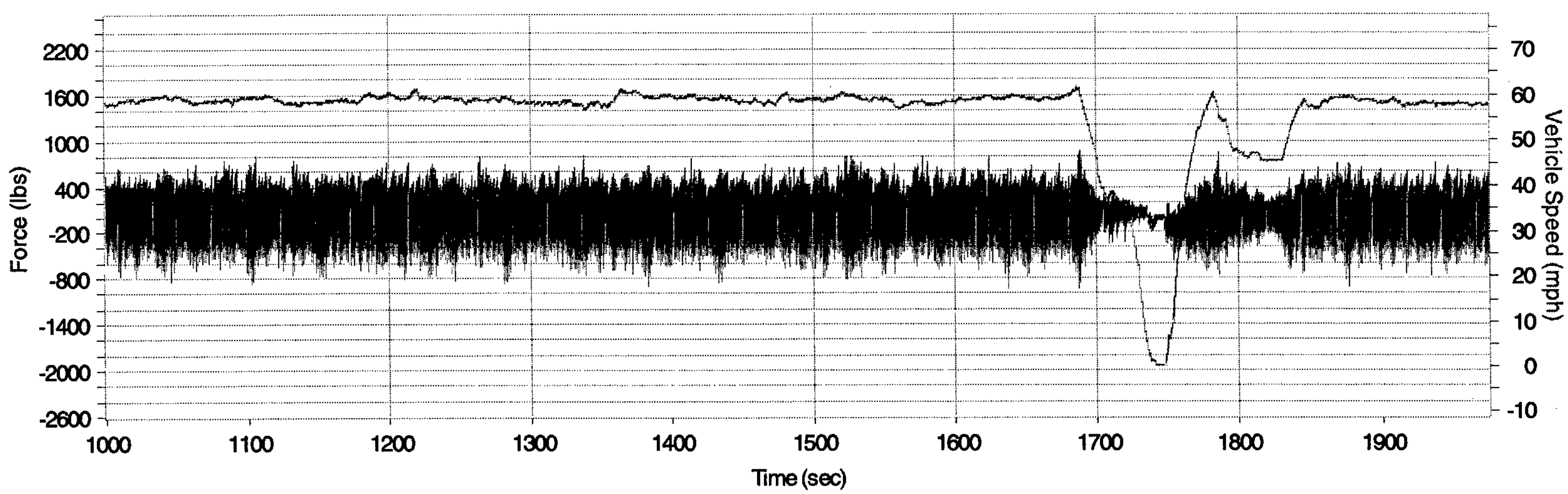
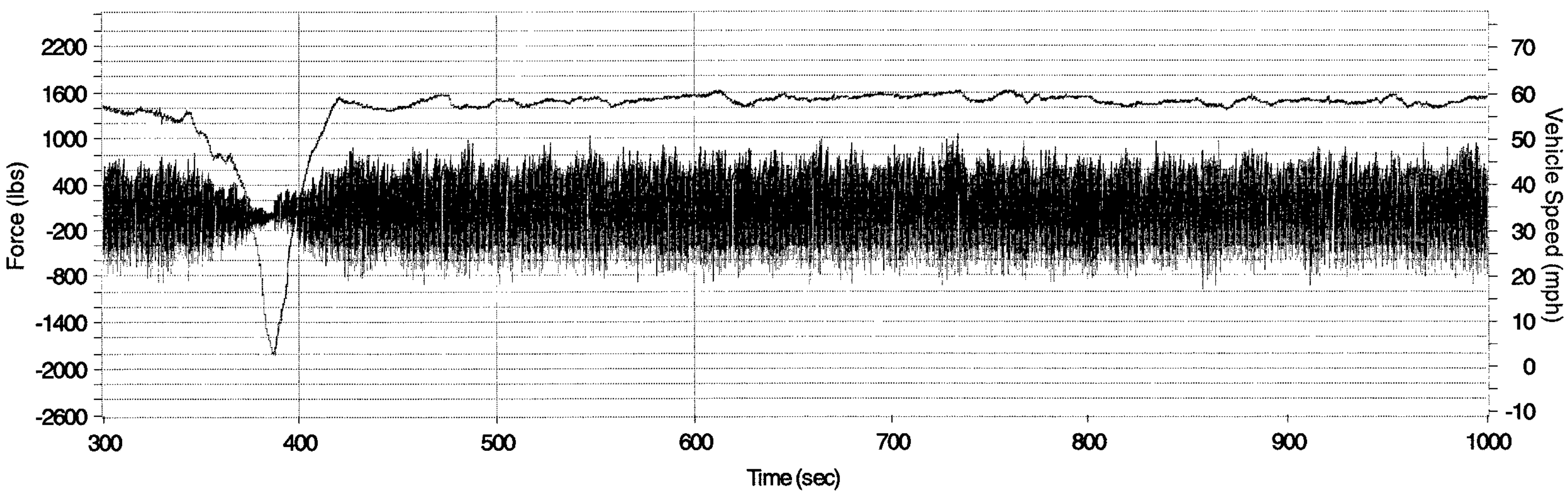


Figure 27

MTV RPL20 (8/13/02 test): Transfer Case Housing Lateral Force



MTV RPL20 (8/13/02 test): Transfer Case Housing Vertical Force

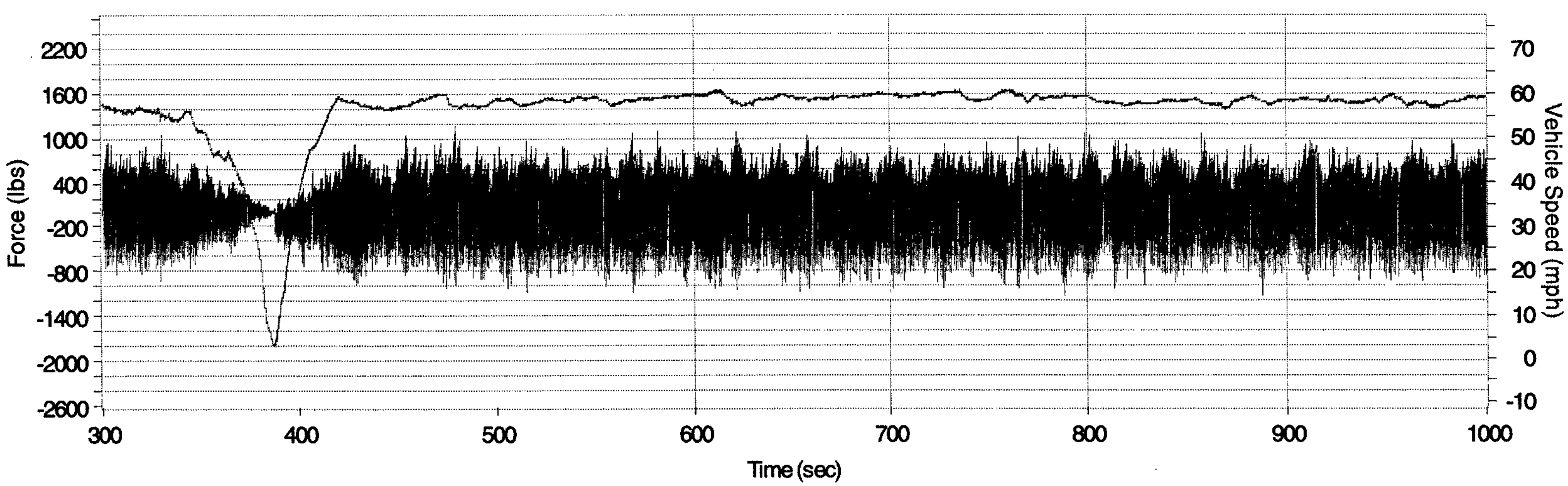
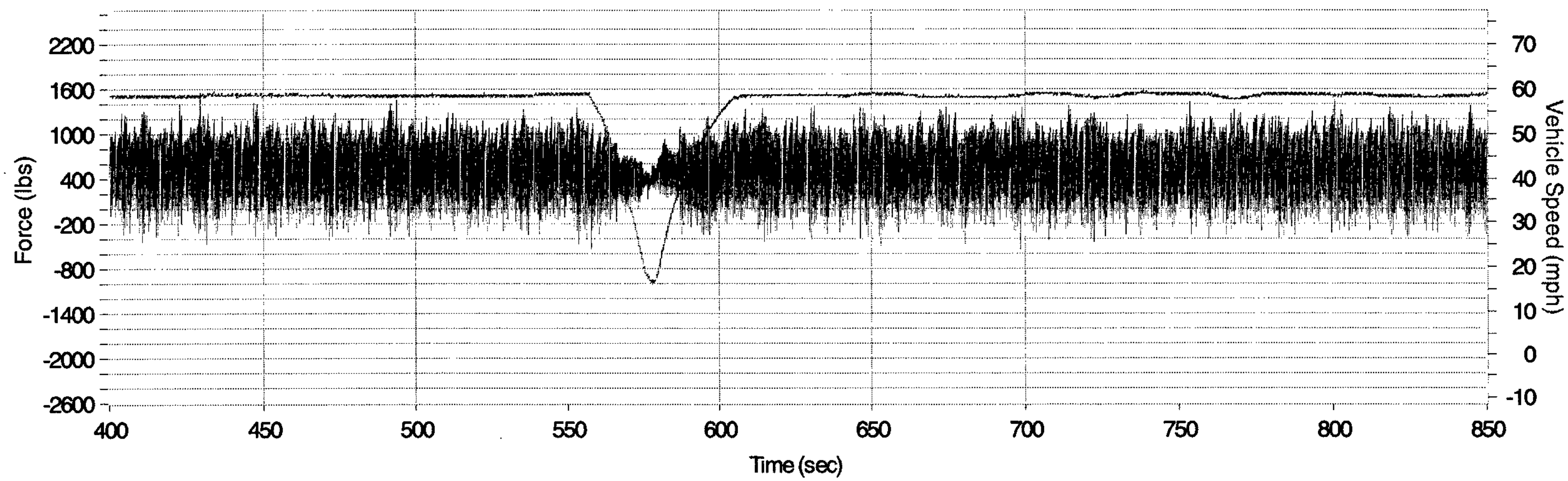


Figure 28

MTV DANA (7/26/02): Transfer Case Housing Lateral Force



MTV DANA (7/26/02): Transfer Case Housing Vertical Force

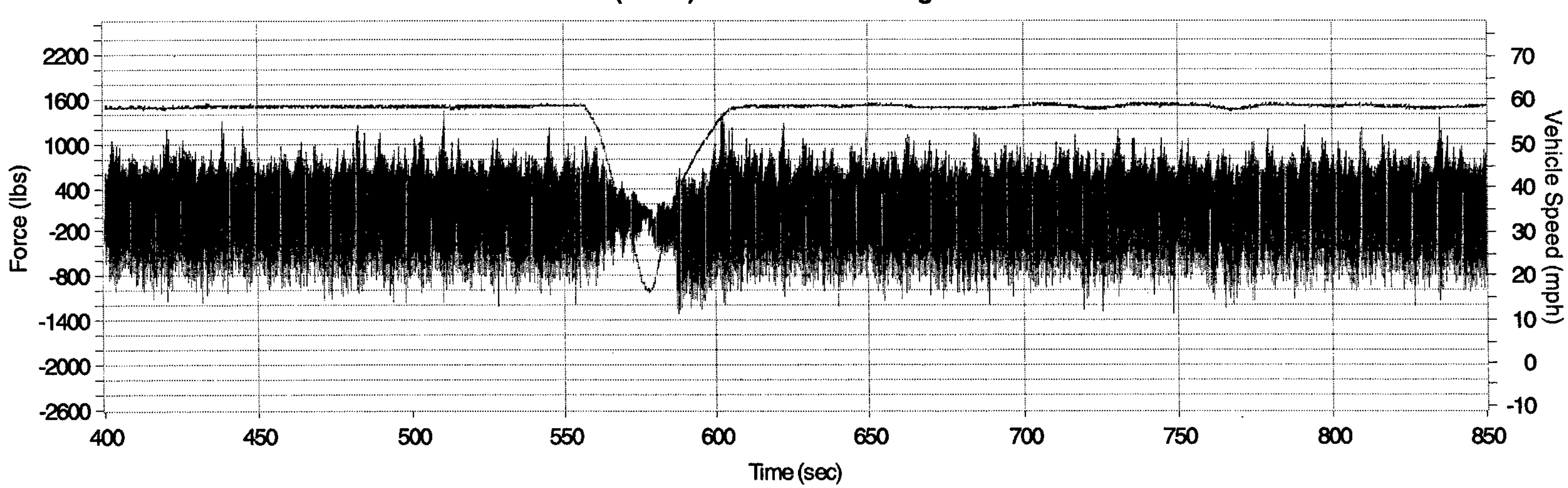
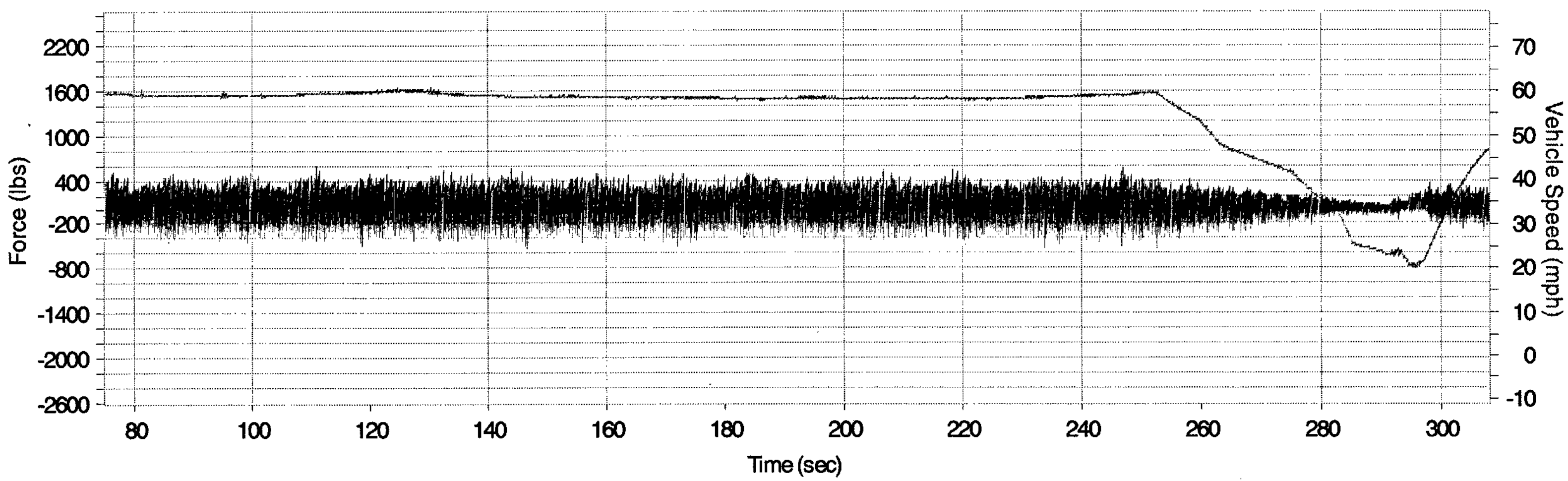
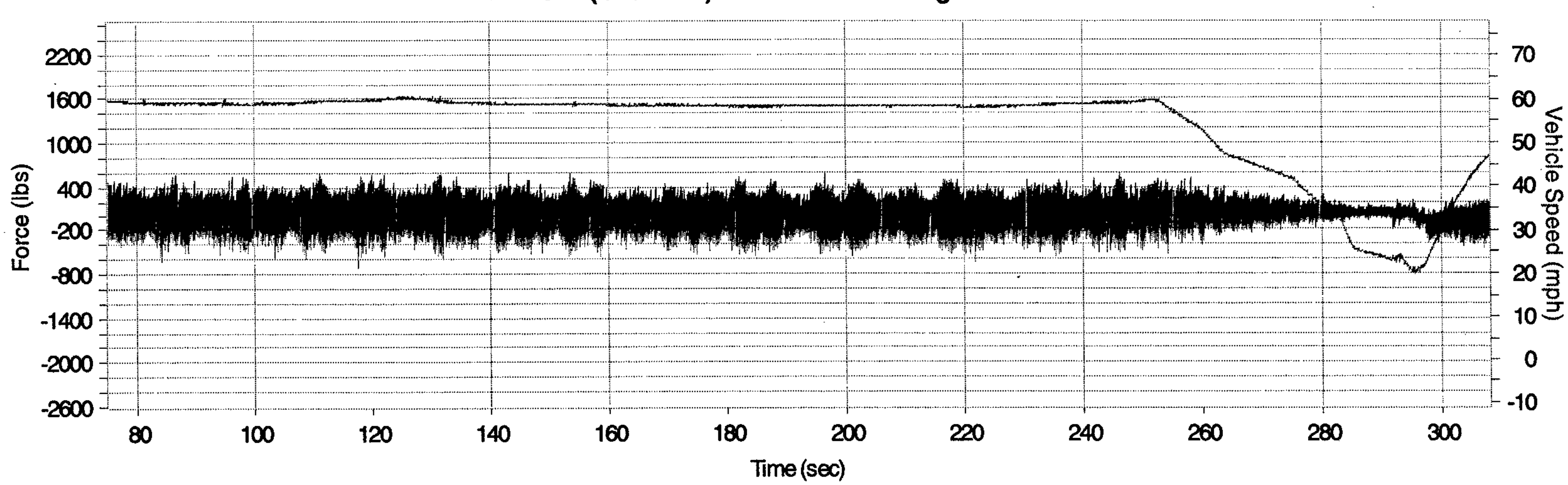


Figure 29

MTV GKN (7/17/02 test): Transfer Case Housing Lateral Force



MTV GKN (7/17/02 test): Transfer Case Housing Vertical Force



Engine Strains

Engine strains were recorded during this program on the LMTV. Locations for measurements were chosen from observations of engine block cracks that had occurred in the field. Gauges were located on the left and right side of the block, near the dipstick tube bosses. Figure E1 shows the location of one of the strain gauges on the right side of the engine block.

Engine strains are influenced by items in addition to transfer case forces. However, at similar engine speeds and vehicle speeds, differences in engine strains measured with different driveshafts quantify the effect of transfer case force on engine strain level. Table E1 lists the engine block strain gauge values for the driveshafts tested at 58 and 70 mph.

Figure E1
Right Side Engine Block Strain Gauge Site

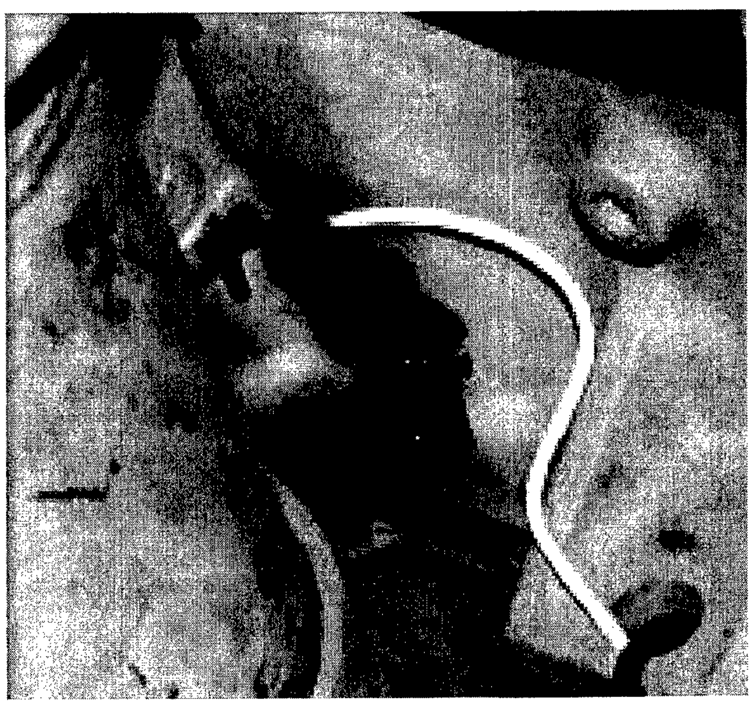


Table E1

LMTV Left Engine Block Strains

Zero to Peak (+/-) Strain-microstrain

Supplier	Model	58 mph	70 mph
Arvin Meritor	A1	150	275
Arvin Meritor	RPL-20	300	500*
Dana	SPL-140	150	225
GKN	CV	125	190

* 67 mph

Torque Fight

The dual drive pair of rear axles in the MTV is equipped with a differential clutch on the intermediate axle to lock out the inter-axle differential when the Mode condition is selected. This prevents differential action while in Mode and can improve vehicle traction when the friction conditions vary among the wheels.

The type of differential clutch in the axle used in the MTV requires that a compressed spring must slide a clutch element over a spline in order to release the clutch. This type of device may not immediately release when under torque due to friction in the spline. This condition can delay release of the lockup clutch when the mode command is turned off.

When Mode is on, the vehicle speed is limited by the electronic governor to less than 40 mph. If the stuck clutch condition occurs after the Mode command is released, torque can build up in the two axles due to slightly different radii of the tires, and the speed is no longer limited by the governor. If this happens, this torque can further exacerbate the tendency for the clutch to avoid releasing. Thus, there is a theoretical possibility that the vehicle can reach highway speeds with a significant amount of torque built up in the rear axles. This condition is termed "Torque Fight" or "Axle Fight". This condition can result in axle torques that are two or more times their normal level.

Inspections of two of the MTV rear-rear driveshaft candidates after testing indicated possible evidence of torque levels in the driveshaft that may have been higher than expected. The driveshaft connecting the two rear axles (the MTV Rear-Rear driveshaft) was instrumented to measure torque to determine if the torque fight condition was occurring. The Dana configuration was chosen for this measurement.

This driveshaft was strain gauged to measure torque and equipped with a telemetry system to transmit the measured torque on the rotating driveshaft to the non-rotating truck instrumentation system. A rotary transformer was used to provide power to the strain gauge. This is propriety MSC equipment, and is shown in Figure M.

The vehicle, equipped with this system was driven under various conditions of speeds, torques, and mode selection states. No "torque fight" was observed during this testing. A typical plot of this data is shown in Figure N where the MTV is accelerating onto a curved freeway entrance ramp at full throttle. Note between 132 and 135 seconds, the vehicle is in mode and the speed is limited by the mode speed-governor. When mode was released, just before 135 seconds, the torque increased, due to the higher available torque as the engine speed dropped. If the suspected "torque fight" condition were occurring, the torque would not be expected to vary so directly with available engine torque.

Though "torque fight" was not observed during this testing, it cannot be ruled out as a contributor to field incidents. Factors too numerous to study in this limited measurement effort, such as tire wear, tire pressure, temperatures, road friction, driveshaft parameters, and others might also contribute to the phenomenon. Additional measurements would be required before it could be completely ruled out as a contributor to high driveshaft torques.

Figure M
MTV Rear-Rear Driveshaft Telemetry System

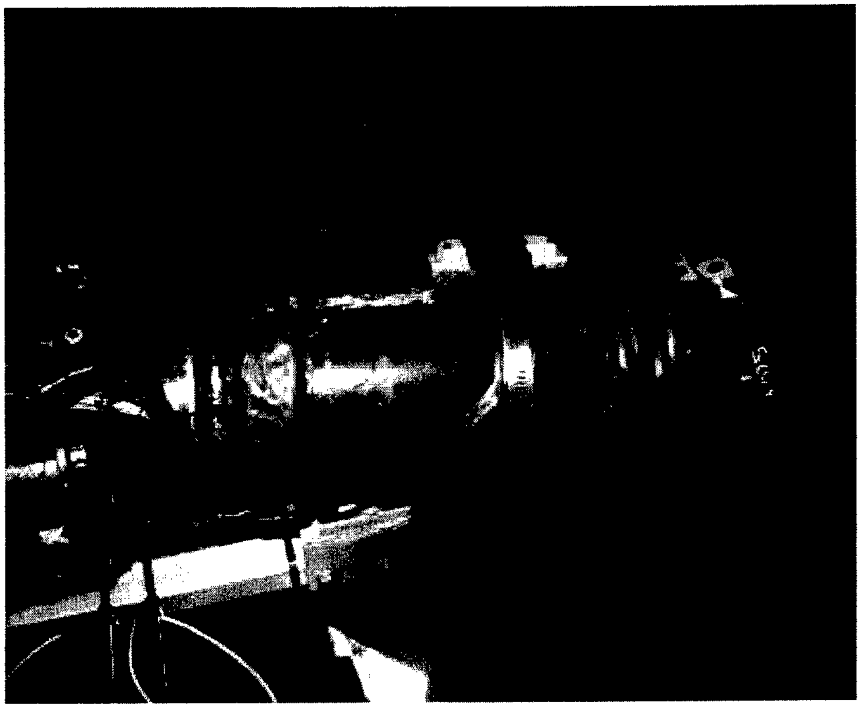
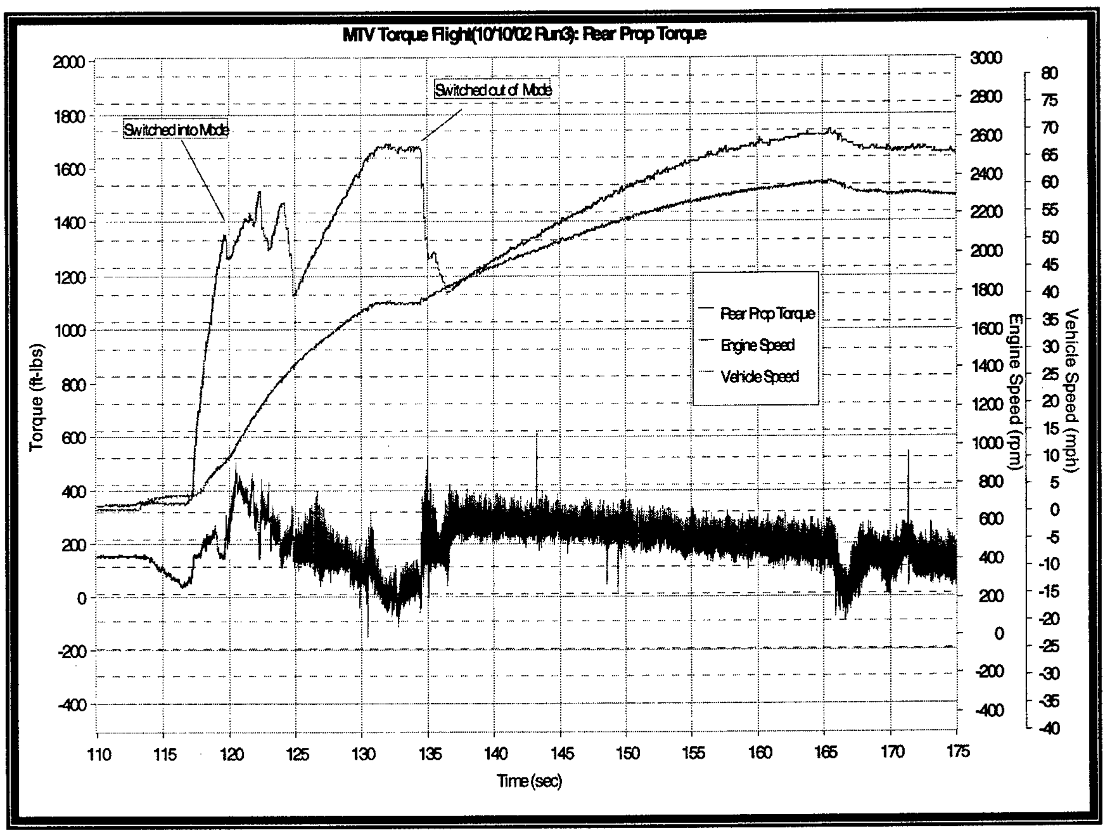


Figure N
Typical MTV Rear-Rear Driveshaft Data



Appendices

Appendix I

Finite Element Model

Of

Transfer Case Force Transducer

May 7, 1999

To: H. Hobolth, R. Washnock

From: C. Talaski

Subject:: FMTV Transfer Case Modifications to Measure Radial Forces

Modifications were made to an alumimum a transfer case housing to measure radial forces generated by the drive shaft. The initial attempt was unsuccessful because of a sensitivity to the location at which the forces were applied along the output shaft (sensitivity to moments in addition to radial forces).

A finite element model was made for the original aluminum housing that supports the rear output shaft bearings. The shape of the housing was approximated by cutting cardboard sheets to fit the inside and outside profiles of the housing (darwings of the housing were not available). It was assumed that the large end of the housing was constrained by an infitely rigid interface and that a solid shaft with the outside diameter of the bearing races was the load path for radial forces. Radial forces were imposed at two locations along the solid shaft to determine sensitivity to forces and moments. The maximum principle stress and the deflection were also determined. Several configurations were evaluated and the results are tabulated on the next page.

The first modification modeled represented the unsuccessful physical component. It had 4 beams 1.33 inches wide and 0.75 inches long. The thick ness of the beams was 0.236. The finite element model (B90429S) showed considerable bending in the relatively flat area between the beams and the shaft.

Next a model (A90430S) was constructed with the beams moved down the cylindrical section halfway between the flange at the large diameter end and the relatively flat area. This showed less sensitivity to the location of the applied radial forces but was judged to unsatisfactory.

A model (B90430S) with eight 0.22 inch square beams spaced uniformily around the perimeter was constructed. It responded the radial force in an acceptable manner (insensitive to the location of the radial force application) but had excessive principle stress.

An eight beam model (A90505S) was constructed with 0.5 inch wide by full thickness sections. This model was only useful for measuring the radial forces along one axis due to a lack of symetry. Its performance relative to the location of where the force was imposed along the shaft was excellent. Principle stress was acceptable.

Finally an eight beam model (A90506S) was constructed with four rectangular beams 0.5 inch wide by the full thickness. Four other six sided beams with approximately the same crossectional area but with some surfaces parallel to the sides of the first four beams were intersperced around the perimeter. This model has the advantage of measuring radial force along two axes. The sensitivity to location of application of the radial force and the maximum principle stress are acceptable.

FMTV Transfer Case Modifications to Measure Radial Forces

	1000 lb. radial on shaft 1 inch from housing				1000 lb radial on shaft 6.4 inch from housing			
Finite Element Model	Shear inside Surface (u-strain)	Shear outside Surface (u-strain)	Deflection (inch)	Maximum principle stress (psi)	Shear inside surface (u-strain)	Shear outside Surface (u-strain)	Deflection (inch)	Maximum principle stress (psi)
A90429S unmodified							0.0029	514
B90429S 1st modified 4 beams	167	150	0.0054		134	3	0.0146	6000
A90430S moved half way 4 beams	65	136	0.0029		29	99	0.058	5350
B90430S 8 square beams 0.22	384	382	0.0055		347	343	0.0089	31380
A90505 8 beams 0.5 one axis only	67 avg.	67 avg.	0.0018		67 avg.	67 avg.	0.0045	3990
A90506S 8 beams 4 – 0.5 wide 4 – six sided	69 avg.	69 avg.	0.0018		71 avg.	71 avg.	0.0045	4059

Appendix II

Dynamic Analysis and Design System

(DADS)

Model of Transfer Case Transducer

Mounted in an LMTV Powerpack

ENGINEERING MEMORANDUM FOR FILE

To: TACOM Project File, Hayes Hobolth, Tom M. Johnson, and Hugh W. Larsen

From: Chas. D. Parker

Subject: Feasibility Study, Making the Transfer Case Rear Output Housing a Force Transducer

Date: 5/16/01

Brief

Covers work done to investigate possible modification of the transfer case rear output housing of an US ARMY MODEL M1078 LIGHT MEDIUM TACTICAL VEHICLE (LMTV). The modification would enable transducers placed on the housing to measure forces generated by the driveshaft and applied to the housing. The Michigan Scientific report "AN INVESTIGATION OF DRIVELINE INCIDENTS OF THE US ARMY'S MODEL M1078 LIGHT MEDIUM TACTICAL VEHICLE (LMTV)" dated March 3, 1999, Contract Number DAAE07-98-M012 is incorporated in this memorandum by reference. This memorandum assumes a basic understanding of that report's content, technology and terminology.

Introduction and Objectives

The cited report discusses issues impacted by driveshaft dynamics. Proposed design changes to the flywheel housing and to the rear driveshaft were evaluated with a computer based mathematical model described in the cited report. Experimental verification of some of the proposed design changes was also done and discussed in the report. Subsequent to that report Michigan Scientific has expended effort to quantify the dynamic effects of the combination of driveshaft unbalance and hinging. That work was done using an updated version of the mathematical model. The findings suggest additional design changes may be needed to the prop shaft to further improve vehicle reliability.

While mathematical modeling can assist in the evaluation of driveshaft designs that may significantly reduce or eliminate the adverse dynamics, experimental validation of promising designs is most important. Measurement of the radial forces generated by the driveshaft and applied to the housing would provide optimal validation. Direct measurement of these forces can be achieved by modification of the rear portion of the transfer case enabling placement of strain gages on the housing.

The modification of production parts to develop force transducers is common practice. It is important that the modification **not significantly effect the performance of the production part.** This report covers the measurements and analysis done to assure that the planned modification of the rear portion of the transfer case would leave its performance unaffected.

Methodology and Discussion

I chose, for this report, to define the transfer case rear output housing as the removable element attached to the rear of the transfer case that contains the rear output yoke bearing. The radial compliance between the transfer case rear output yoke and the transfer case was measured as part of the work covered in the cited report (See page 23 and Appendix D). That compliance, .008 inches per 1000 pounds load, is the **sum of several compliances including that of the transfer case rear output housing.** This **summed** compliance represents a stiffness of 125,000 lbf/in.

Finite element models of the transfer case rear output housing were constructed. They are covered in the attached memo from C. Talaski. The model for the unmodified housing showed a radial compliance of .0029 inches per 1000 pounds load. Additional models were constructed, based on the first, to optimize machining of the housing for strain gage placement. The intended output of the gages, after calibration and appropriate signal processing, will be the desired radial force values without moment generated crosstalk. The modified housing, acting as a transducer, will report the forces applied to the housing through its bearing from the rear output yoke. The selected configuration showed a radial compliance of .0045 inches per 1000 pounds load compared with .0029 in per 1000 lbf for the unmodified housing.

The compliance of several springs in series is the sum of the compliance's of each spring. Using this relationship we have:

C<sum> = C<other> + C<tcroh> where C<sum> is the sum of the compliances between the transfer case

rear output yoke and the transfer case, C<tcroh> is the transfer case rear output housing compliance and

C<other> is the sum of the balance of the in-series compliances.

Then before modification we have C<sum> = .008/1000, c<tcroh> = .0029/100 giving C<other> = (.008 - .0029)/1000. After modification we have c<tcroh> = .0045/100 and C<other> = (.008 - .0029)/1000 giving
C<sum> = (.0045 + .008 - .0029)/1000 = .0096/1000 in per 1000 lbf. This is a series stiffness of 104,167 lbf/in. This stiffness was used in modified versions of the mathematical model cited above.

Four model runs were executed to compare the dynamic behavior of the "as modified" version of the transfer case rear
output housing versus the original. In all runs the driveshaft speed is swept through its critical speed. Critical speed is displayed along with the peak to peak radial forces applied to the housing by the rear output yoke at the critical speed. Both directions Fy (lateral) and Fz (vertical) are displayed. The compliance for the Caterpillar CAT 2 flywheel housing is used in all runs.

Results are presented in the attached plots. Each figure set consists of three pages. The first page (A) of each set plots the force vector magnitude vs. time for the full speed sweep. Its respective model generates each of the two traces. The second page (B) of each set displays a subset in time of the same data. The subset includes the critical speed. The third page (C) of each set plots is an overplot of the two time histories. It is a time subset taken from the second page and expanded to show the phase relationships. It is taken at or in the neighborhood of the critical speeds of the two runs.

Figures 1 and 2 display the results from the baseline run (model dm25cat2ver1) and the "as modified" (model dm30cat2a1_tcase_xdcr_v1). The properties of the revised Meritor A1 shaft were used in all data runs for Figures 1 and 2. Figures 3 and 4 display the results from the baseline run (model dm22cat2ver2) and the "as modified" (model dm31cat2msc_tcase_xdcr_v1). The baseline model for Figures 3 and 4 is similar for the baseline used in Figures 1 and 2 with the exception of the driveshaft properties. Driveshaft properties representative of the MSC Revised Design Shaft discussed on page 22 of the cited report were used in all data runs for Figures 3 and 4. The higher critical speed shaft was tested in anticipation of usage of the transducer on additional shafts whose properties will provide a higher critical speed. The following table summarizes the results. "ORG" in the "Figure, Model and Vector" cell indicates that the unmodified stiffness of 125,000 lbf/in was used in that model run. "MOD" indicates that the modified stiffness of 104,167 lbf/in was used in the run.

Figure, Model and Vector	Driveshaft Critical Speed (RPS)	Peak To Peak Force (lbf)
Figure 1, dm25, ORG, Fy	69	10333
Figure 1, dm30, MOD, Fy	71	9756
Figure 2, dm25, ORG, Fz	69	10574
Figure 2, dm30, MOD, Fz	69	10370
Figure 3, dm22, ORG, Fy	80	10951
Figure 3, dm31, MOD, Fy	83	10590
Figure 4, dm22, ORG, Fz	80	7305
Figure 4, dm31, MOD, Fz	80	7568

Conclusion

Differences between the original and modified stiffness runs are minor and understood. Consequently a modified transfer case rear output housing can be used in the experimental validation of driveshaft designs with confidence that the results are not significantly impacted by the modification.

All machine-readable files, which include the model definition files, will be included with the balance of the project documentation package.

tfr_case_hsng_xducer.doc

Appendix III

Dynamic Analysis and Design System

(DADS)

Model of

FMTV Drivetrain Systems

And

Balance Machines

To: TACOM Project File, Hayes Hobolth, Tom M. Johnson, and Hugh W. Larsen

From: Chas. D. Parker

Subject: Effects of FMTV Drive Shaft Parameter Changes on the Dynamic Behavior of Universal Joints

Date: 9/27/02

Brief

Covers work done to demonstrate changes in the dynamics of drivetrains containing Cardan joints as drive line parameters change. While the findings are general in nature, they were generated in the context of a US Army vehicle. The Michigan Scientific report "AN INVESTIGATION OF DRIVELINE INCIDENTS OF THE US ARMY'S MODEL M1078 LIGHT MEDIUM TACTICAL VEHICLE (LMTV)" dated March 3, 1999, Contract Number DAAE07-98-M012 is incorporated in this memorandum by reference. This memorandum assumes a basic understanding of that report's content, technology and terminology.

Introduction and Objectives

The cited report discusses issues impacted by driveshaft dynamics. Proposed design changes to the flywheel housing and to the rear driveshaft were evaluated with a computer based mathematical model described in the cited report. Significant effort was expended to validate the model. Experimental verification of some of the proposed design changes was also done and is discussed in that report. Subsequent to that report Michigan Scientific has expended effort to quantify the effects of changes to the drivetrain. That work is being supported using updated versions of the mathematical model.

The dynamic forces transmitted by the Cardan joints are strongly influenced by the parameters of the drive shaft, its encompassing system and operating conditions. The joints and shaft assembly impact on its encompassing system can range from minimal to destructive. The complexity of the total system and its non-linearities makes it difficult to fully evaluate the impact of parameter changes on the dynamic behavior of Cardan joints with limited experimental data.

Compliances seen by the shaft that supports the Cardan joint at the transfer case rear are different for all three force directions and all three moments. As a result the rotating force vector acting on that yoke sees a mechanical impedance (compliance and inertia) that varies as it rotates. That, in turn, effects the magnitude of the force vector and the forces acting on all of the degrees of freedom of the joint yoke. Shaft "critical speed" is also a function of the mechanical impedance seen by the shaft. "Critical speed" therefore has a range set by the impedance as a function of shaft rotational angle. At shaft speeds in that range there are order related forces that have non-sinusoidal waveforms with significant harmonics. The majority of the forces seen by the shaft at the transfer case rear are transmitted through the Cardan joint. As a result they are trigonometric functions of the instantaneous angles (and vector force) for the degrees of freedom of the joint. Sum and difference frequencies will be present with excitation and non-linearities. The SOCIETY OF AUTOMOTIVE ENGINEERS (SAE) publication "UNIVERSAL JOINT and DRIVESHAFT DESIGN MANUAL" (AE-7) extensively details Cardan joint theory and practice and is recommended to readers of this report.

The determination of cause and effect relationships from the review of experimental data is often difficult because of the complexity of the total system. One parameter variation experiments are difficult and costly to execute on systems as complex a US Army truck. Test schedules that traverse a wide range of terrain further complicate such experiments. Mathematical modeling provides a quick and precise tool to do parameter variation studies on complex systems. Such studies often provide insight into the system's dynamics as well as providing information that can help in the analysis of experimental data.

Michigan Scientific, in response to requests from TACOM, has and is continuing to expand the model. Its current level of fidelity is adequate to do the studies documented in this memo. The parameter variations covered in this memo are limited to drive shaft parameters, the mechanical impedances external to the shaft and transmitted torque. The "measured" effects are limited to those "seen" by the drive shaft, ujoints and their interfaces with its supporting system. Other parameter variation studies will be covered in separate memos.

Methodology and Discussion

This version of the mathematical model includes a Meritor A1 rear drive shaft and Cat 2 housing The model was reduced to eliminate confounding responses from selected degrees of freedom in the encompassing system. This was done to meet the objectives outlined above and is technically sound for these comparison studies. The majority of the modifications will not be covered in this report. Exceptions will be described as needed. Sixteen out of fifty-six runs have been selected for this report. Output from several of the runs will be used in more than one set of comparisons. Some of the comparisons will be pairs, representing two conditions. Others will have two or more levels of a varied parameter.

One model modification, which is used in thirteen runs, relates to interfaces between the drive shaft with its ujoints and the encompassing system. The ujoints are supported by bearings in the rear of the transfer case on one end and in the differential on the other. Those bearings are then supported by other structures that have mass and compliance. Those bearings are "grounded" for those twelve runs. "Grounding" in the model means they are not allowed to move. The remaining three runs frees those degrees of freedom while modifying other constraints or parameters. The text detailing results will explicitly identify important modifications. Results of the comparisons will be presented in graphical and/or text format. The graphical figures are included with the body of the text describing them. They are also repeated in an appendix to allow easy A/B comparison.

Most of the model runs documented below go from zero rotational speed to sixty revolutions per second (60 RPS) at time zero. The resulting startup transient decays within a few seconds due to system damping. However most of the runs below are terminated and have their data taken for analysis before the residual response has fully decayed to zero. A physical vehicle will generate broad band noise input to the drive train. This noise will excite the responsive system elements causing motions analogous in kind and amplitude to the residual response from the startup transient. Complete removal of the residual response would display unrealistic results. However selected runs **DO** have the response removed to better document results from certain conditions. This is accomplished by allowing the run to execute for a relatively long time or adding system damping. These cases are well identified.

FIRST COMPARISON GROUP

The two runs compared were V20srNOS1ubrsFN.def and V20srNOSgb1ubgbrsFN.def. The shaft, in both runs, was not rotating. Both runs were given an initial vertical displacement at the approximate center of gravity location of the shaft (including it's ujoints). Both runs had a mass placed at the same location that causes an unbalance of nearly four inch-ounces when rotating. One run, V20srNOSgb1ubgbrsFN.def, had the bearings grounded as described above, while the other allowed "normal" connection to the balance of the system. The response to the initial condition is an approximation of the shaft's critical speed. The difference between the results stems from the effects of the supporting impedances. V20srNOS1ubrsFN.def approximates the critical speed in the full vehicle and is about 66 Hz. V20srNOSgb1ubgbrsFN.def is about 71 Hz.

SECOND COMPARISON GROUP

The next comparison was between the pair lmtv01c2a1_FNTA.def and lmtv01c2a1_FNTB.def. The purpose of the test was to estimate the effects of the compliances of the spline near the forward end of the shaft on the shaft torsional critical speed. Lash was not allowed and spline hinging was set to zero in lmtv01c2a1_FNTA.def. In lmtv01c2a1_FNTB.def the spline's compliances were changed to the values for a shaft segment. Again the shaft was not rotating in either run. Damping was eliminated at most locations to allow all modes to be easily identified in the run output. The driveshafts, in both runs, were reverted to "perfect" shafts, i.e. no unbalance or runout. An initial angular displacement condition was applied to the spring representing the compliance of the axle. **Neither run had their bearings grounded as discussed above.** The first run, lmtv01c2a1_FNTA.def, showed a first torsional mode at 160 Hz while the first torsional mode for lmtv01c2a1_FNTB.def was 266 Hz.

THIRD COMPARISON GROUP

The next comparison was between the pair V20srVCSgb0ubgbrs.def and V20srVCSgb1ubgbrs.def. It was designed to demonstrate the dynamic differences between a shaft with **no** unbalance and one with the maximum allowed unbalance (per the manufacturers specs) for the shaft studied. V20srVCSgb0ubgbrs.def was the "perfect" shaft while V20srVCSgb1ubgbrs.def had a mass placed as described above in the FIRST COMPARISON GROUP. Both shafts had their ujoint bearings grounded as described above. Both had a drag torque of 240 in-lbs. applied at the differential pinion. Both shafts were spun at 60 RPS and both had a critical speed of 71 Hz (see above).

Below are selected graphical results for both runs. The graphs titled "VERTICAL FORCE APPLIED TO TRANSFER CASE BEARING BY UJOINT" have two time histories displayed. One is vertical force, while the second, which is symmetric around the zero axis, is a pure sine wave at once per revolution (first order) to help in the identification of order related events. Note that the vertical force scaling is significantly different between the two runs. The second graph in the comparison pair is as titled, "FFT OF VERTICAL FORCE APPLIED TO TRANSFER CASE BEARING BY UJOINT". It is the Fourier transform of the force signal. Both runs have a decaying transient response to the startup impulse at the critical speed of 71 Hz.

The vertical force trace for V20srVCSgb0ubgbrs.def, figure1, shows a complex waveform with elements related to the rotational speeds and with a signal changing faster than the rotational speed. Its FFT, figure2, shows a signal at the critical speed of 71 HZ, the residual response to the startup transient. It shows weaker signals at the second and fourth orders of rotational speed. These are generated by the twice per revolution rotational acceleration and deceleration of the shafts inertia due to ujoint's operating angle. Both of these signal sources will be treated in more detail later. The addition of unbalance (V20srVCSgb1ubgbrs.def) adds a very strong component at first order of revolution. The force trace, figure3, shows a beat between that and the residual response to the startup transient. The amplitude of the residual response and that due to the rotational acceleration/deceleration remains about the same as shown by both FFTs. The amplitude of the once per revolution caused by the unbalance is the dominant signal in the second FFT, figure 4.

V20srVCSgb0ubgbrs.def

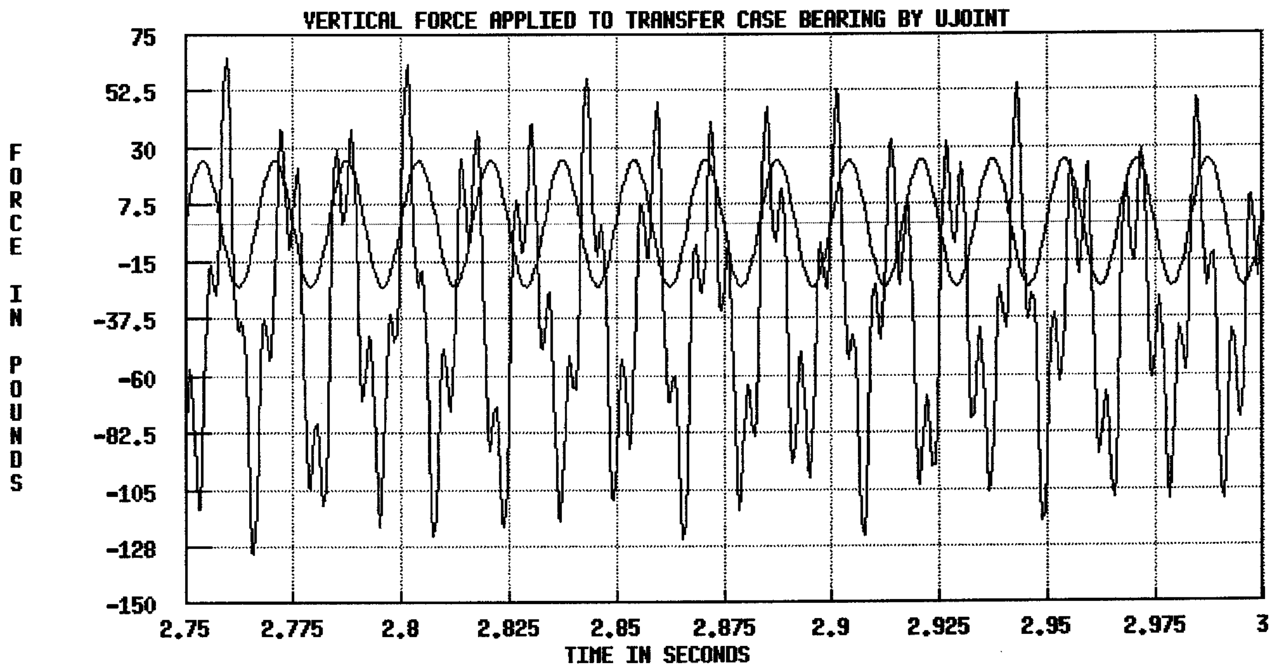


FIGURE 1

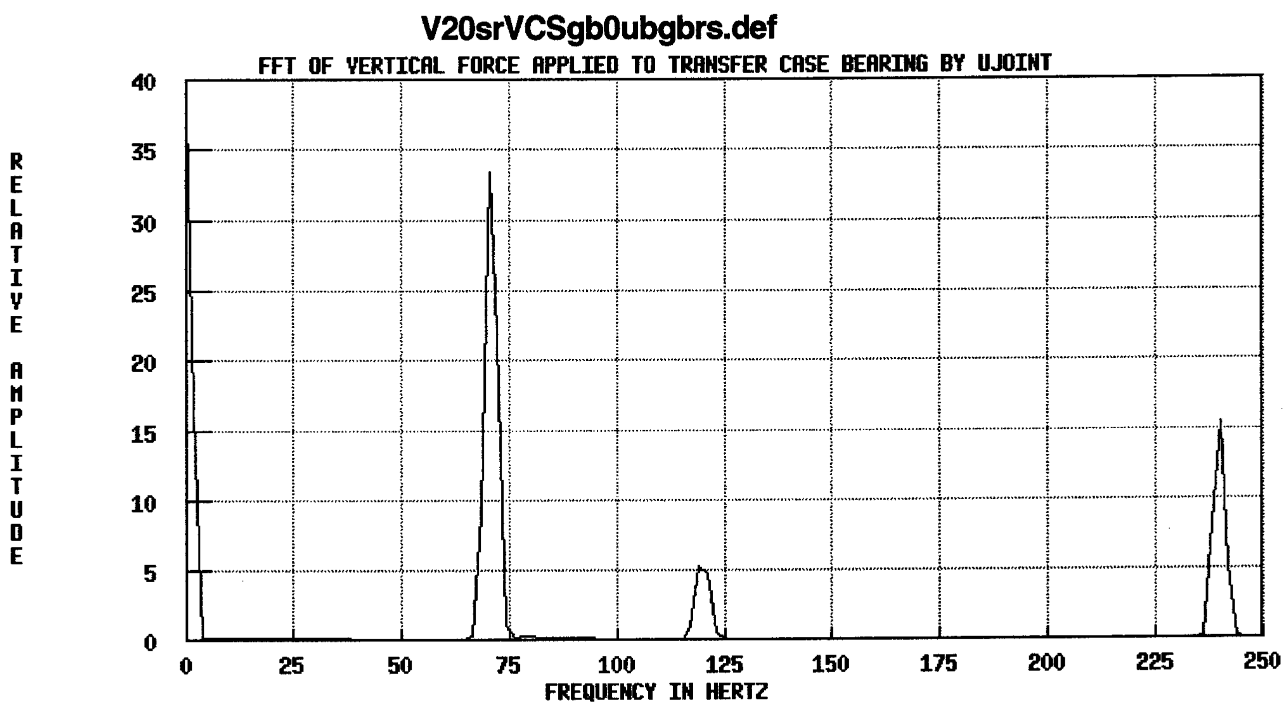


FIGURE 2

V20srVCSgb1ubgbrs.def

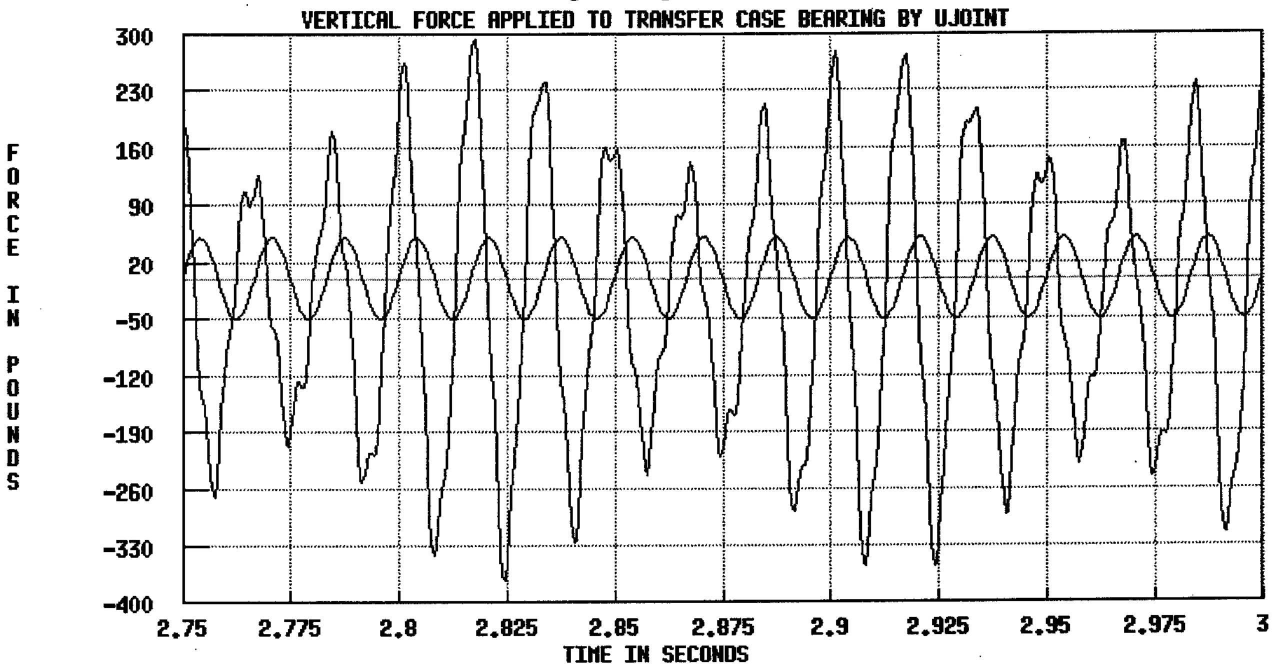


FIGURE 3

V20srVCSgb1ubgbrs.def

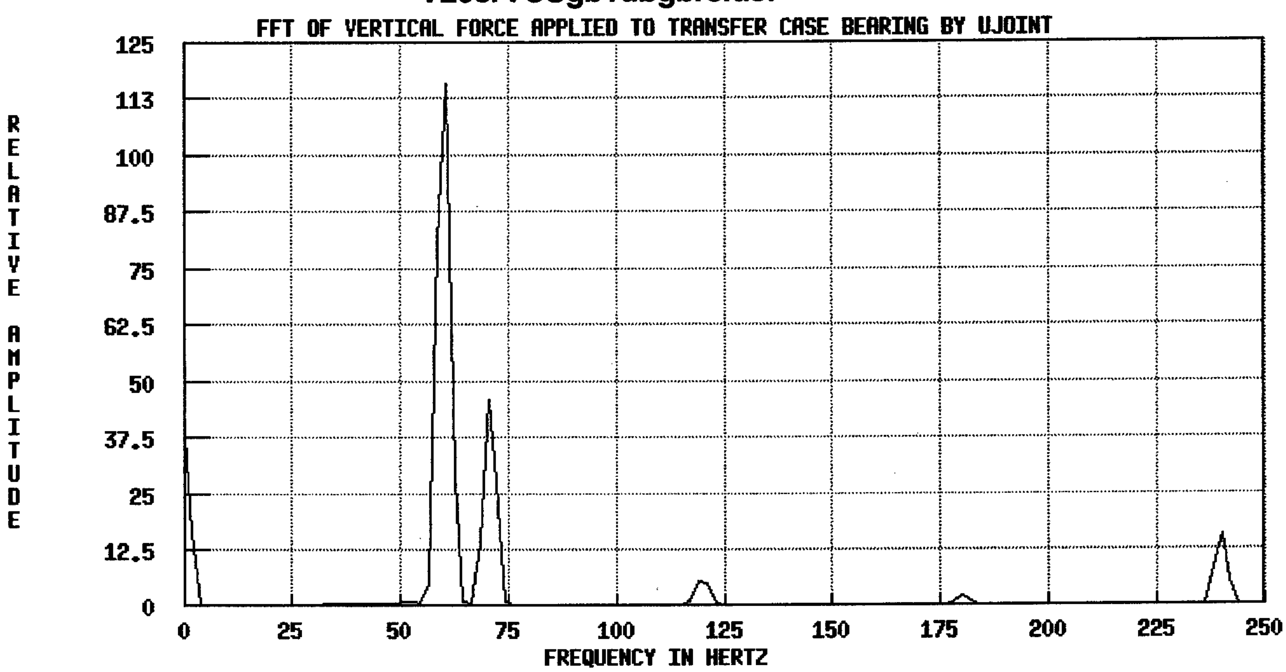


FIGURE 4

FOURTH COMPARISON GROUP

This next comparison is between the pair V20srVCSgb0ubgbrs.def (figures 1 and 2 in the previous group) and V20vlrVCSgb0ubgbrs.def. The run time was increased for the run

V20vlrVCSgb0ubgbrs.def. All other run parameters were unchanged. This enabled the damping to remove the residual response from the startup transient. No other parameters were changed. The first graph, figure 5, contains two traces, vertical force and a first order signal which is the sine wave around the zero axis. The vertical force trace at the end of the long run is much less complex and is clearly order related. Removal of the residual response is confirmed by the FFT, figure 6. The amplitude of the second and fourth order signals is unchanged from the short run.

V20vlrVCSgb0ubgbrs.def

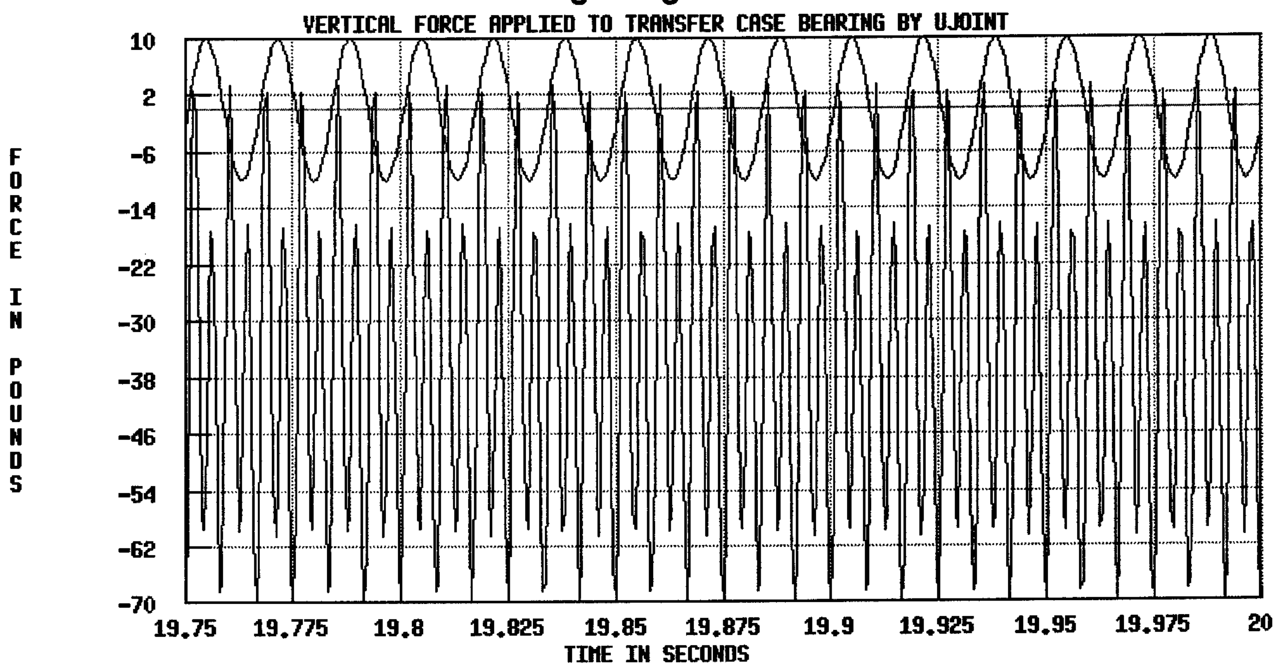


FIGURE 5

V20vlrVCSgb0ubgbrs.def

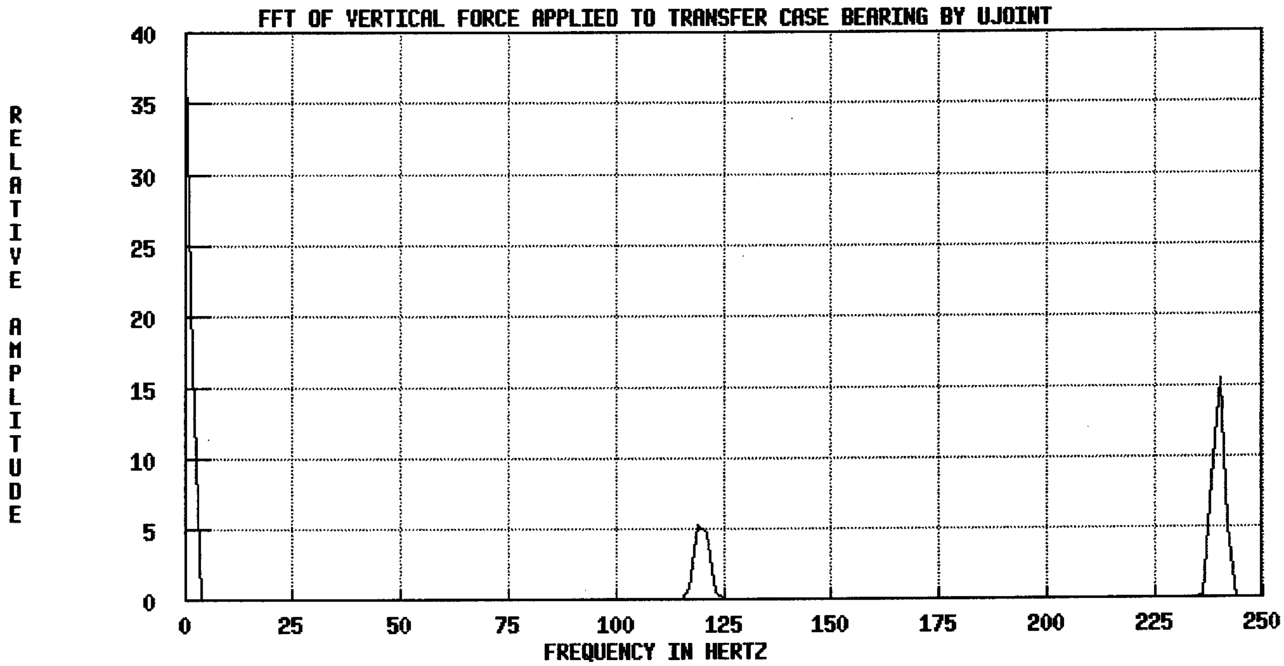


FIGURE 6

FIFTH COMPARISON GROUP

This group utilizes three runs designed to investigate the effects of rear axle load torque. The load torque is analogous to the drag torque described earlier. It is applied at the pinion through its connection with the differential and rear axle element. Run V20srVCSgb0ubgbrs0RT.def had no load torque, V20srVCSgb0ubgbrsLRT.def's load was 1,240 in-lbs. and V20srVCSgb0ubgbrsHRT.def's was 10,240 in-lbs. All shafts had their ujoint bearings grounded as described above. All shafts were spun at 60 RPS and had a critical speed of around 71 Hz. Three graphs for each run are presented. The first has the vertical force trace along with the first order signal as described above. The second graph is the Fourier transform of the force signal as above. The third graph in each group is the torque applied by the u-joint at the rear of the transfer case to the drive shaft. This is the driving torque required to hold constant speed. It also has a first order trace. Additional discussion will follow the graphs.

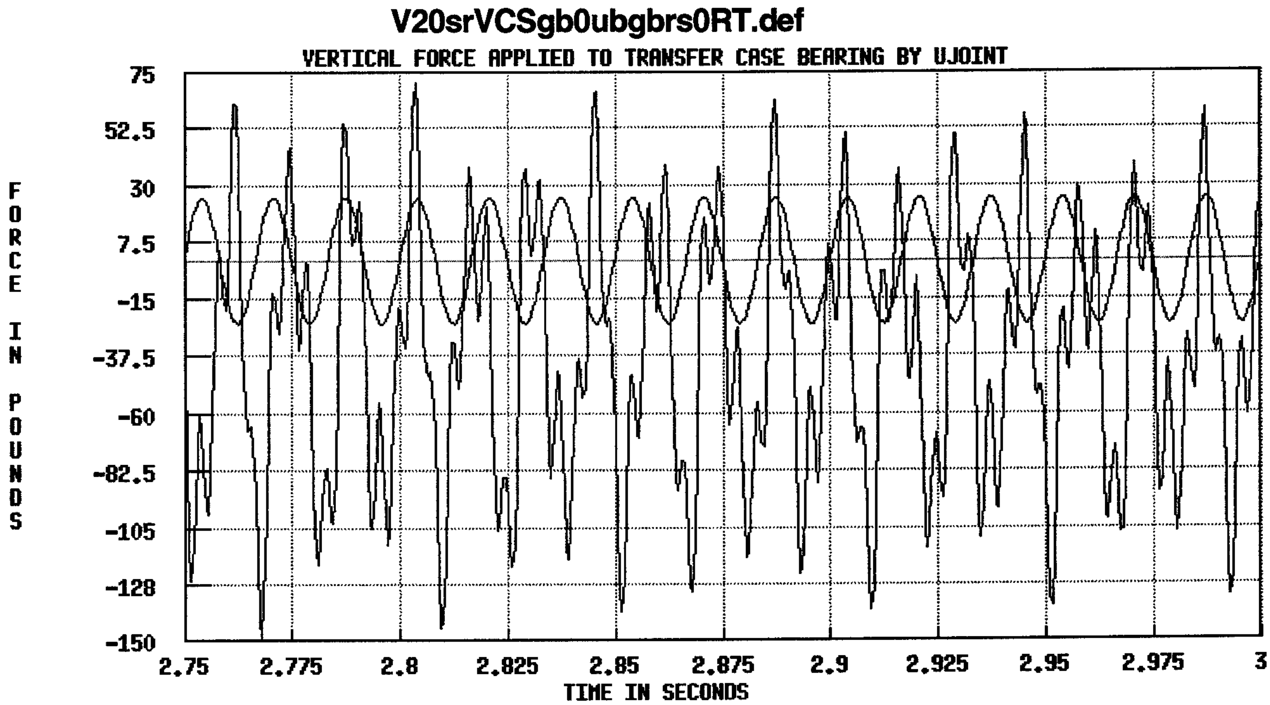


FIGURE 7

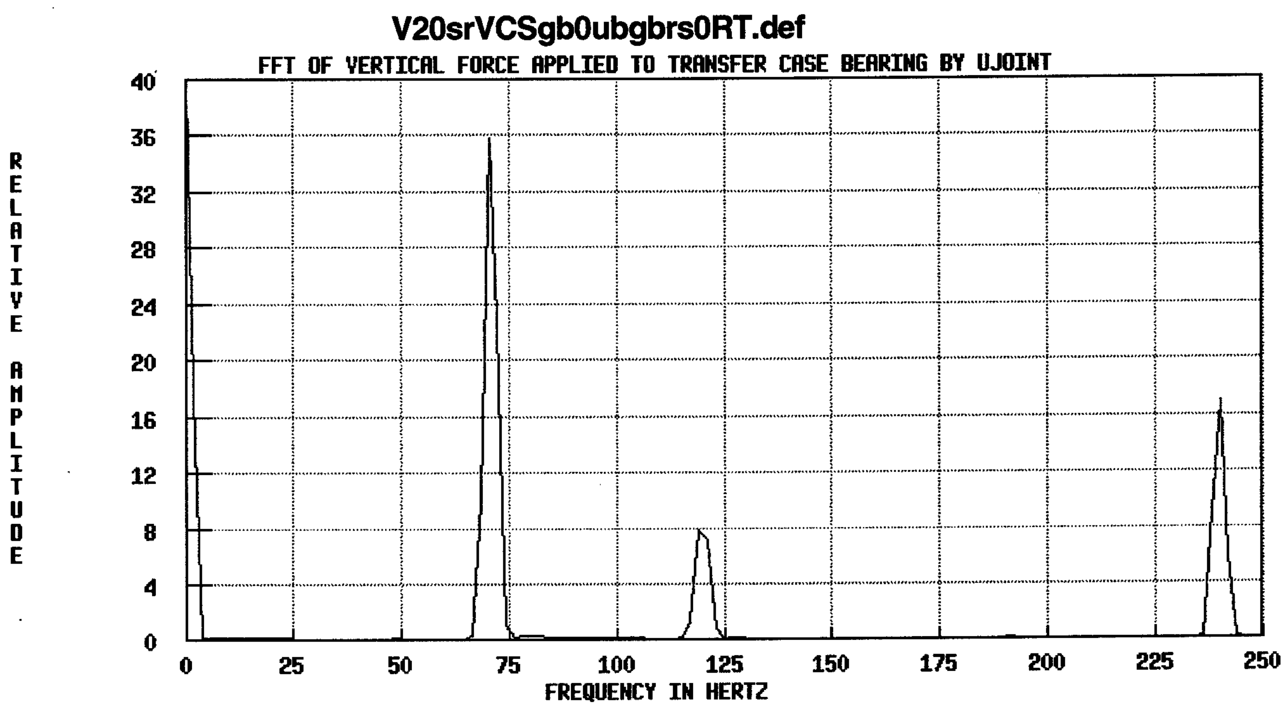


FIGURE 8

V20srVCSgb0ubgbrs0RT.def

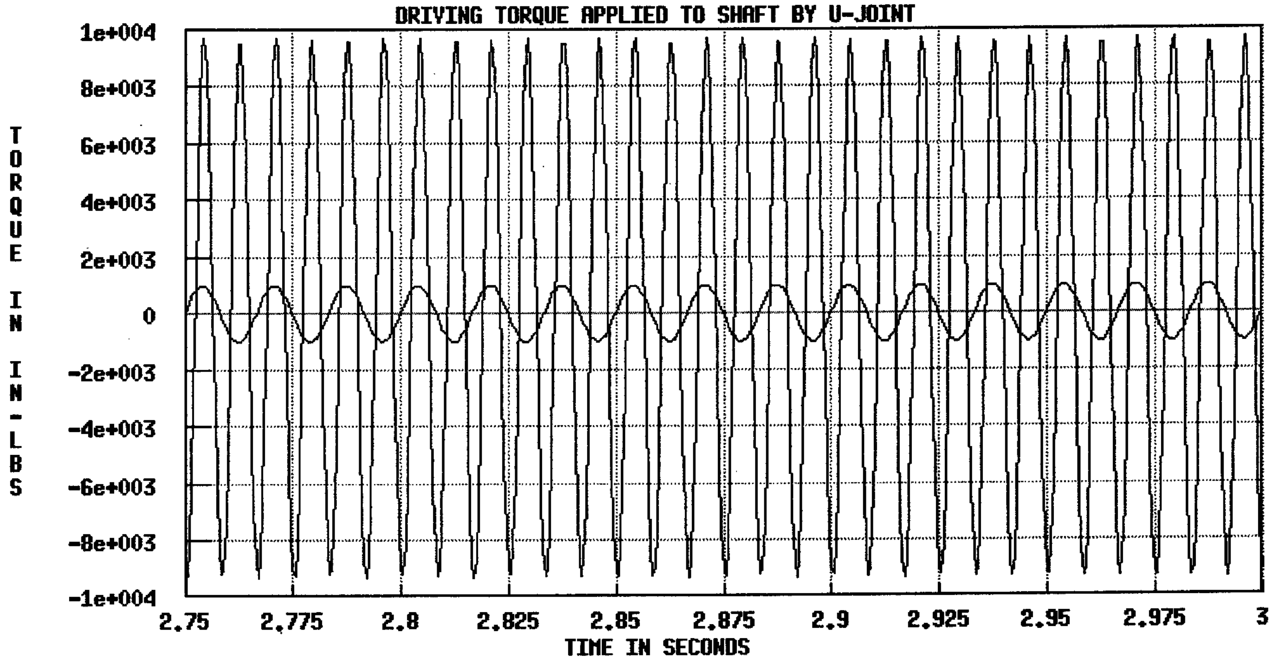


FIGURE 9

V20srVCSgb0ubgbrsLRT.def

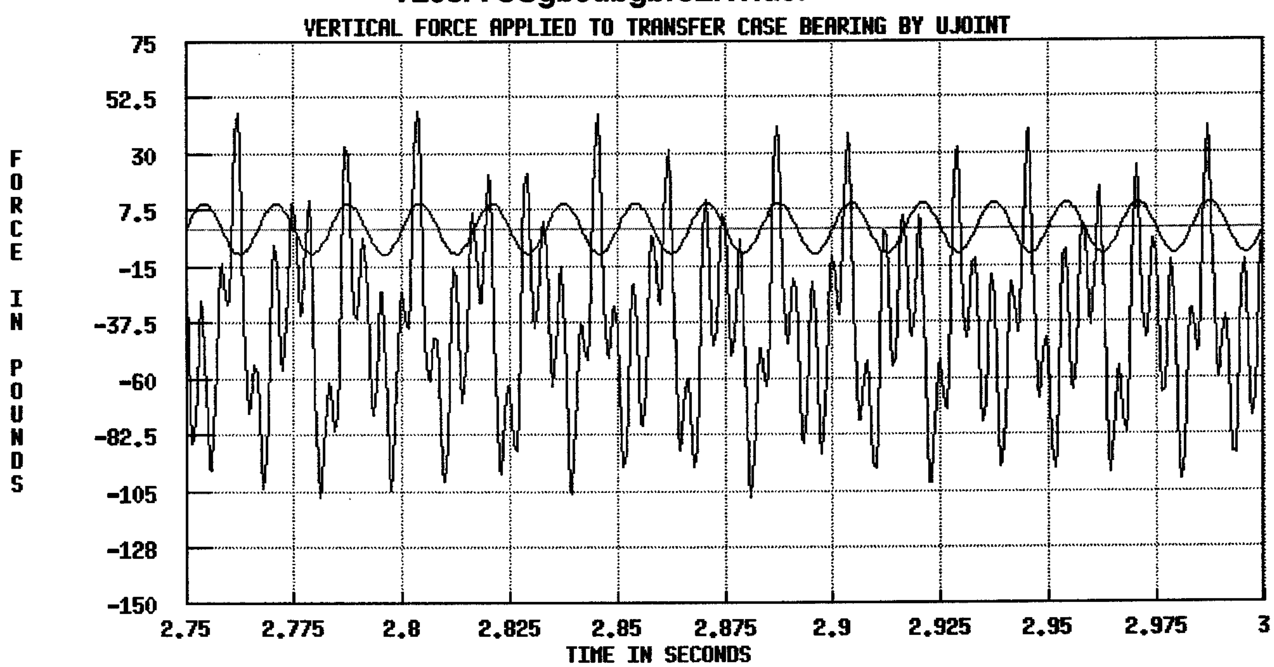


FIGURE 10

V20srVCSgb0ubgbrsLRT.def

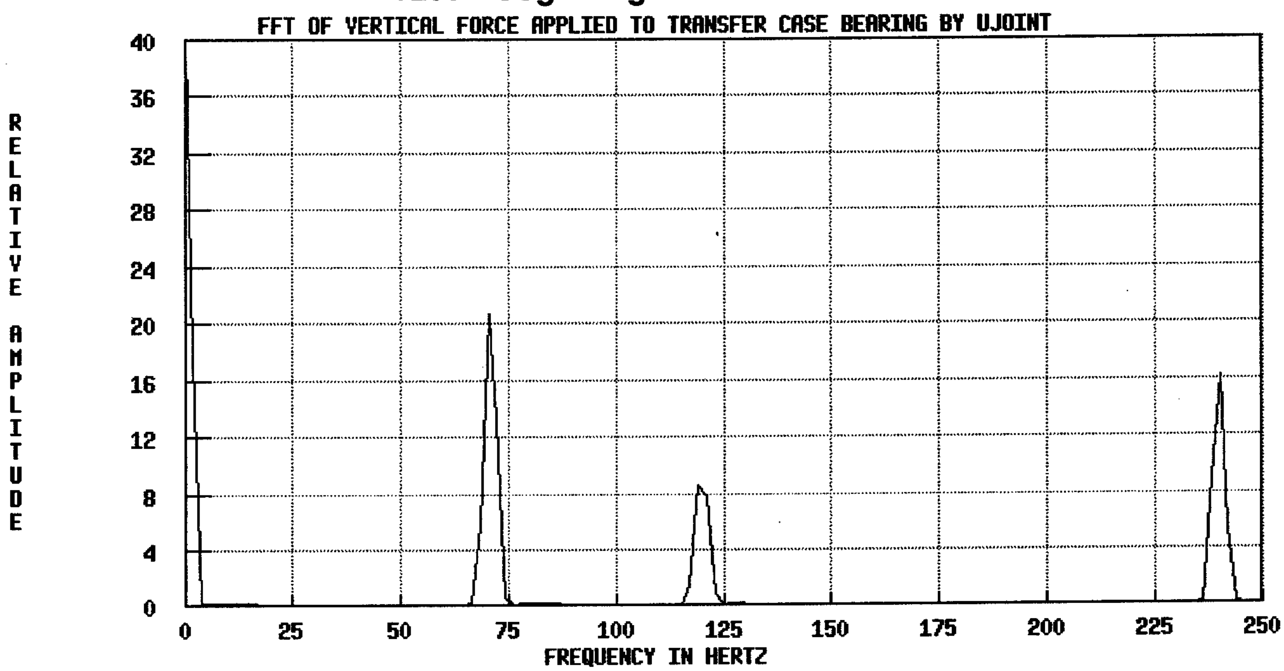


FIGURE 11

V20srVCSgb0ubgbrsLRT.def

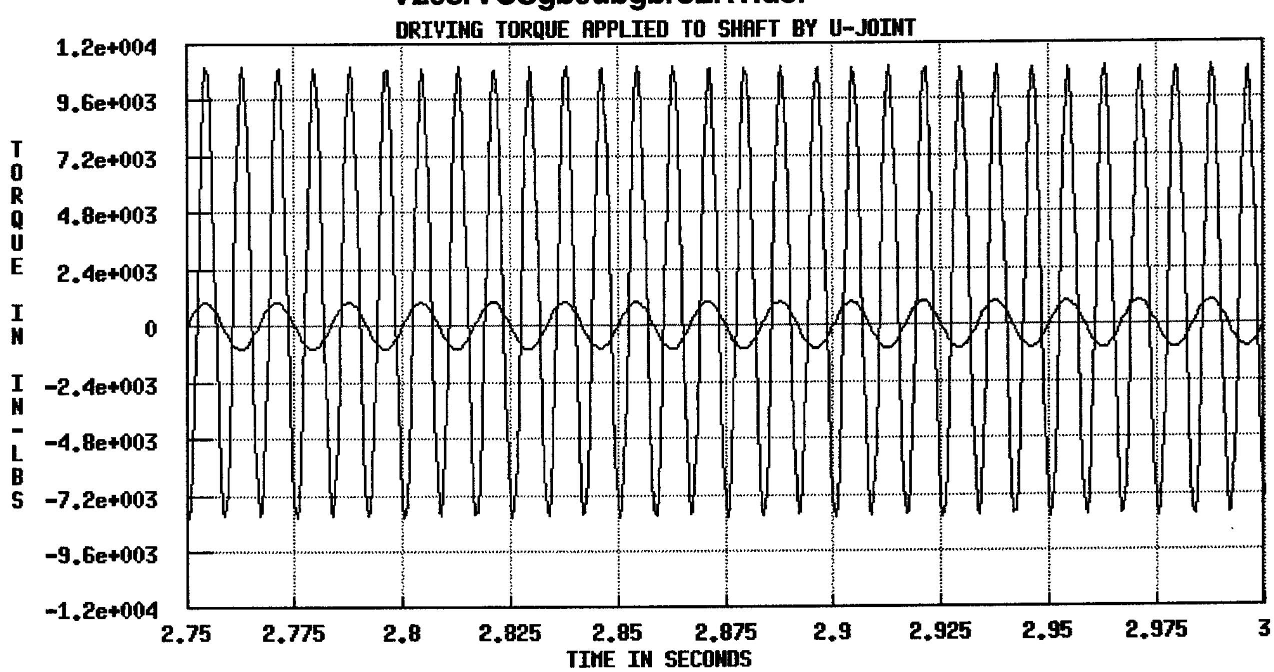


FIGURE 12

V20srVCSgb0ubgbrsHRT.def

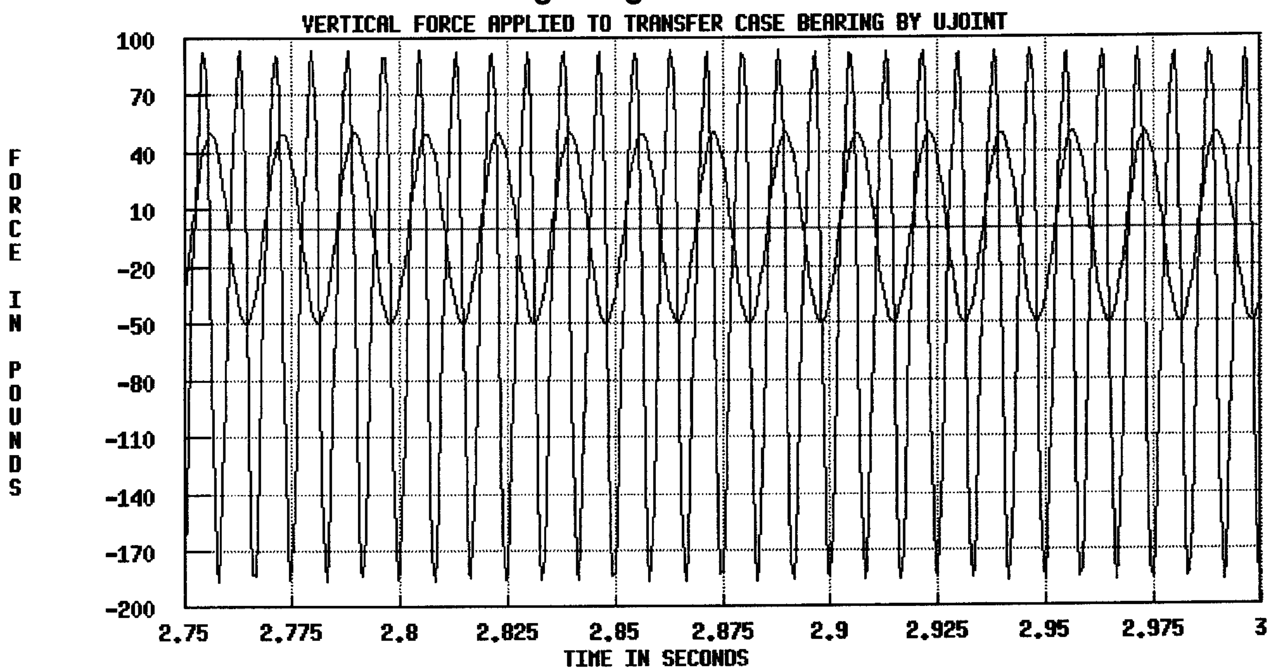


FIGURE 13

V20srVCSgb0ubgbrsHRT.def

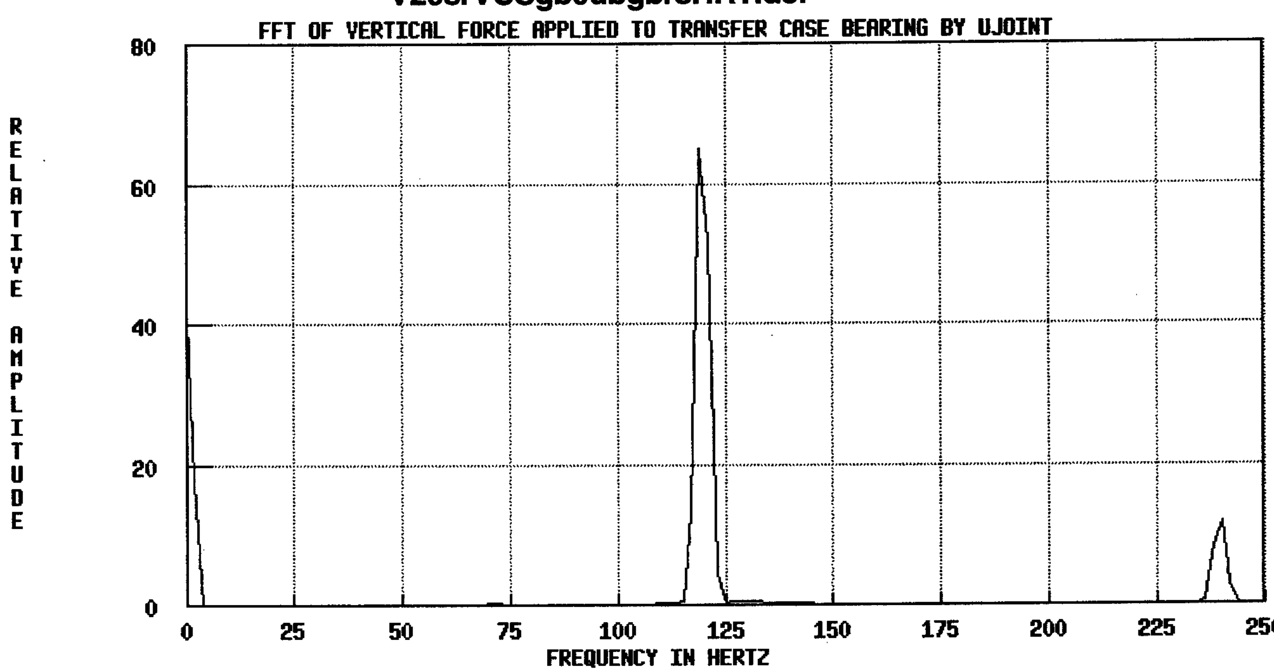


FIGURE 14

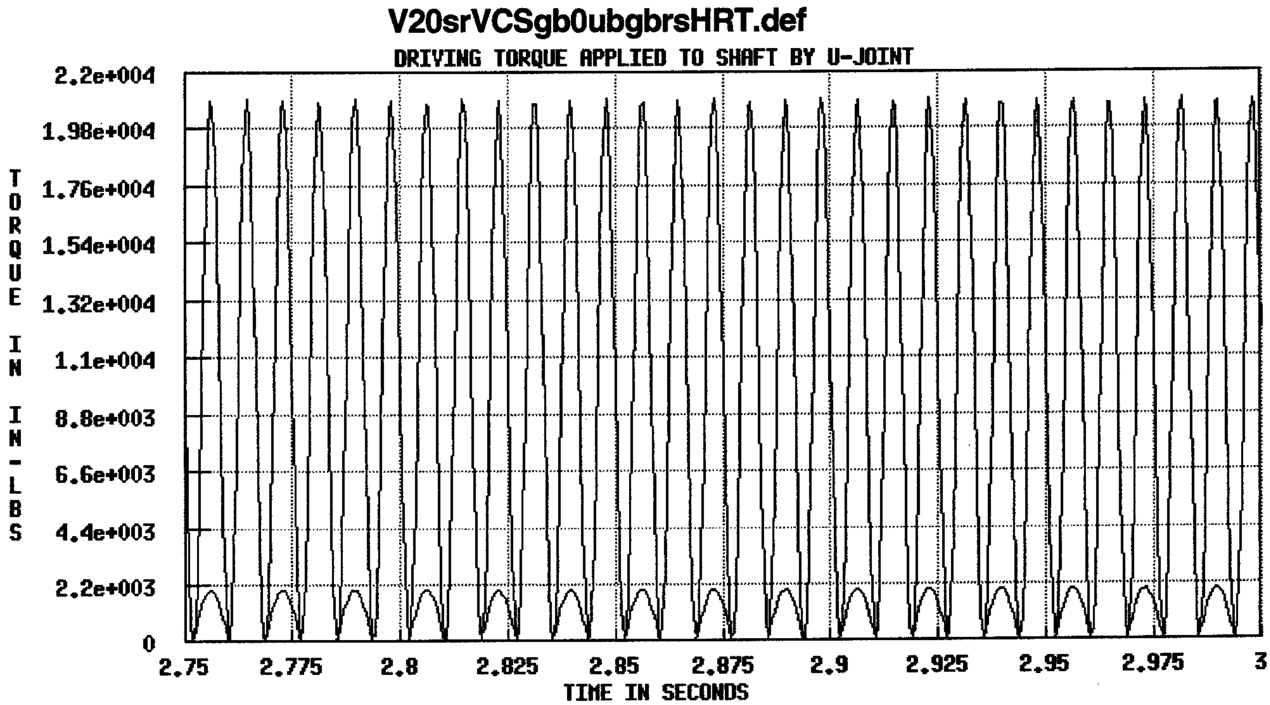


FIGURE 15

The vertical force trace for V20srVCSgb0ubgbrs0RT.def, figure 7, (no load torque) shows a complex waveform. It's FFT, figure 8, shows it is comprised of signals at 70.3 Hz @ 35.8 units of amplitude, 120 Hz @ 7.8 units and 240 Hz @ 17 units. The 70.3 Hz signal is, as described earlier, the residual response from the startup transient. The 120 Hz and 240 Hz signals are from the second and fourth order forces generated by the twice per revolution rotational acceleration and deceleration of the shaft. The driving torque graph, figure 9, shows that twice per revolution torque. Note that it is symmetric around the zero axis.

The vertical force trace for V20srVCSgb0ubgbrsLRT.def, figure 10, (low load torque) again shows a complex waveform. The FFT, figure 11, shows that it is also comprised of signals at 70.3 Hz @ 20.7 units, 120 Hz @ 8.5 units and 240 Hz @ 16.2 units. The second and fourth order signals are near the magnitude of the no load case. The second order now contains energy contributed by the reaction forces generated from a secondary couple, a function of shaft operating angle and transmitted load torque. The driving torque graph, figure 12, shows the twice per revolution torque. Note that it is **no longer** symmetric around the zero axis but is about the same peak to peak amplitude as the in no load case. The bias is a direct result of the load torque. The 70.3 Hz signal is much lower than the magnitude in the no load case. The load torque causes the system geometry to change and, in turn, impacts the damping.

The vertical force trace for V20srVCSgb0ubgbrsHRT.def, figure 13, (high load torque) does not have the complex waveform of the other two runs. It is a relatively clean second order signal. Its FFT, figure 14, has the same frequency content with radically different amplitudes, 70.3 Hz @ 0.2 units, 120 Hz @ 70.3 units and 240 Hz @ 12 units. The comments for the low load torque case apply here as well, but with greater impact.

SIXTH COMPARISON GROUP

This group utilizes three runs designed to investigate the effects of drive shaft inertia. The shaft inertia was adjusted by changing **only the moment of inertia** around its axis of rotation. This change has slight impact on critical speed and on the response to the startup transient. Run V20srVCSgb0ubgbLlrs.def's inertia was lowered by a factor of 100 from the standard, V20srVCSgb0ubgbrs.def's inertia was the standard shaft inertia and V20srVCSgb0ubgbHlrs.def's was increased by a factor of 10. All shafts had their ujoint bearings grounded as described above. All shafts were spun at 60 RPS and had a critical speed of 70 to 80 Hz. Three graphs for each run are presented. The first has the vertical force trace along with the first order signal as described above. The second graph is the Fourier transform of the force signal as above. The third graph in each group is the torque applied by the u-joint at the rear of the transfer case to the drive shaft with a first order of rotation trace added. The graphs for V20srVCSgb0ubgbrs.def from above are repeated here for convenience. Additional discussion follows the graphs.

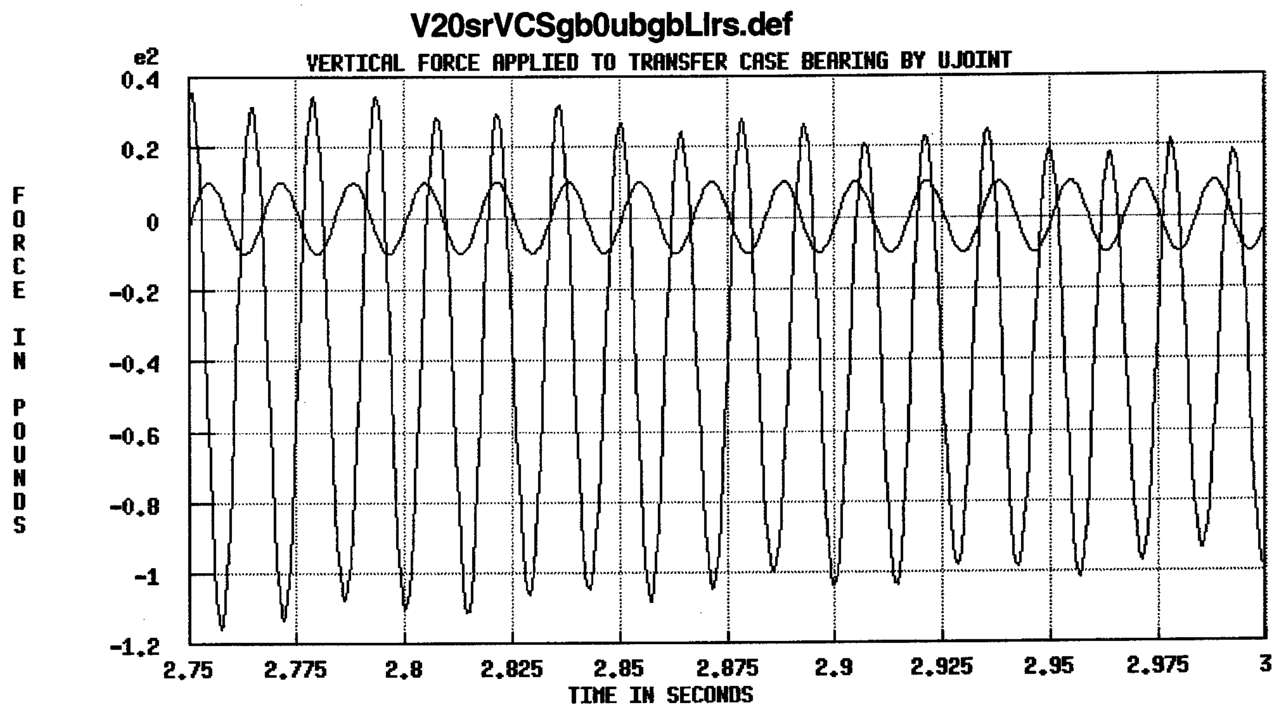


FIGURE 16

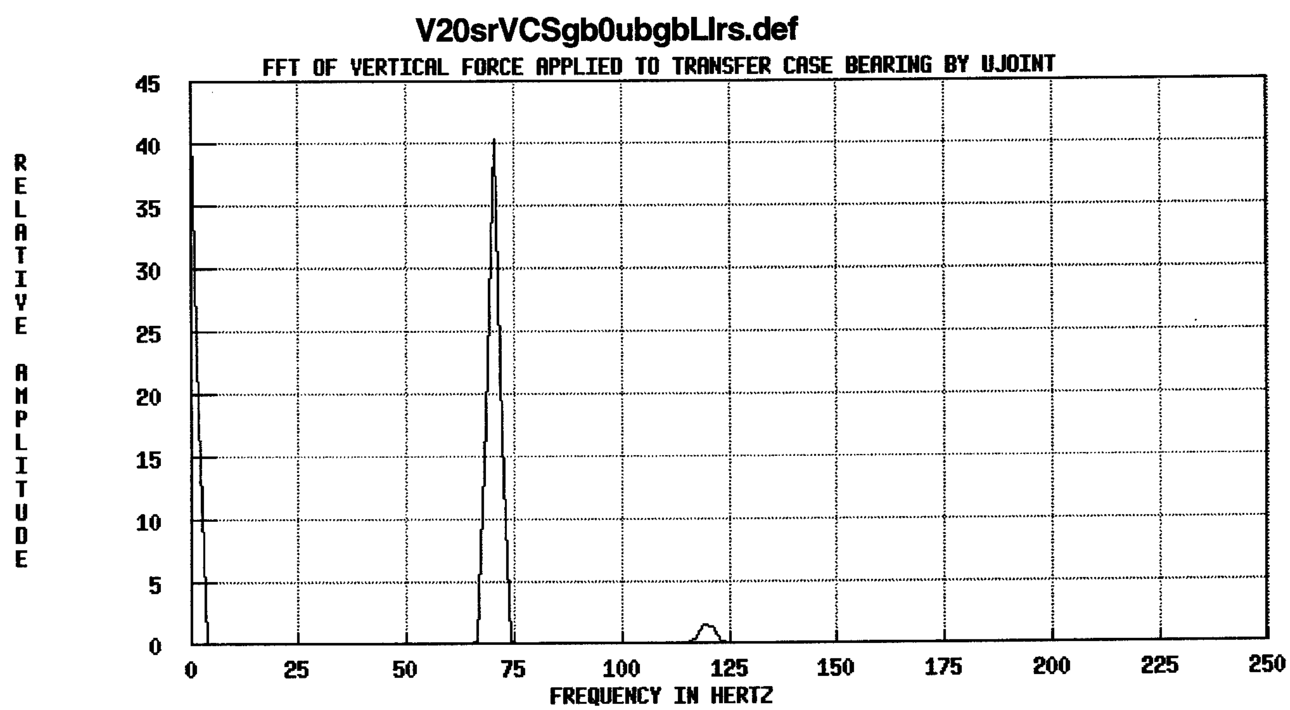


FIGURE 17

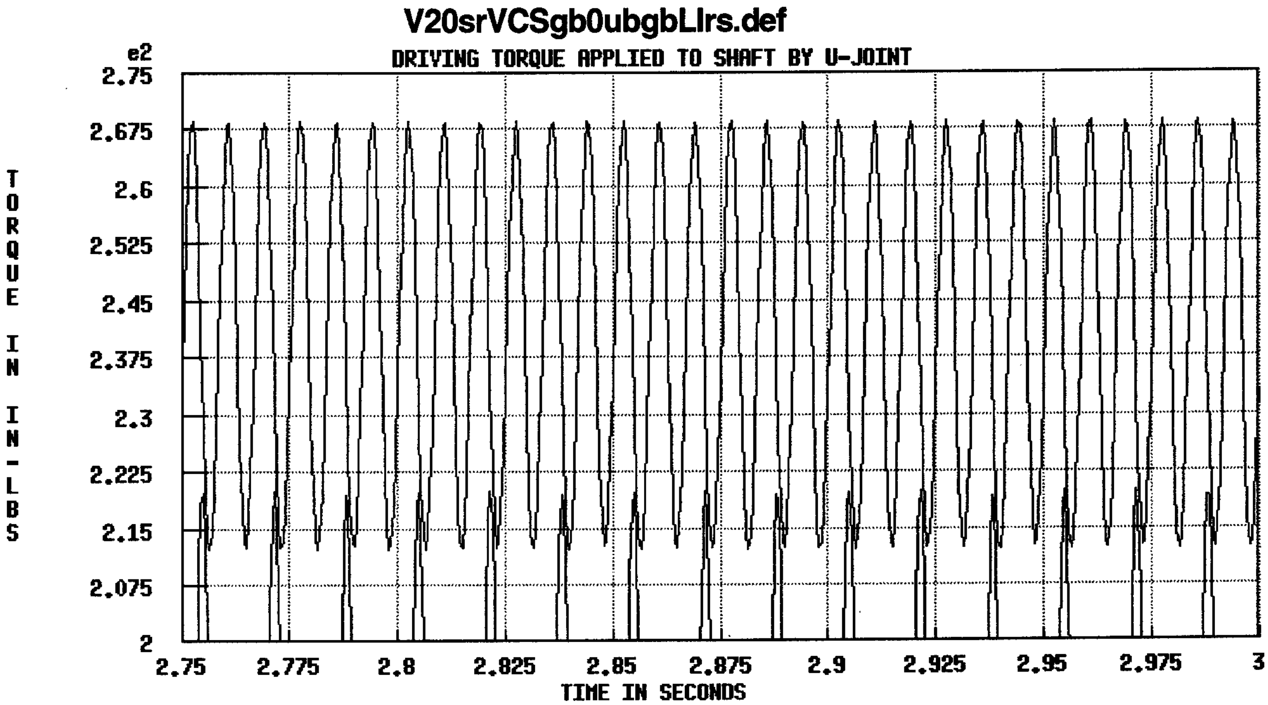


FIGURE 18

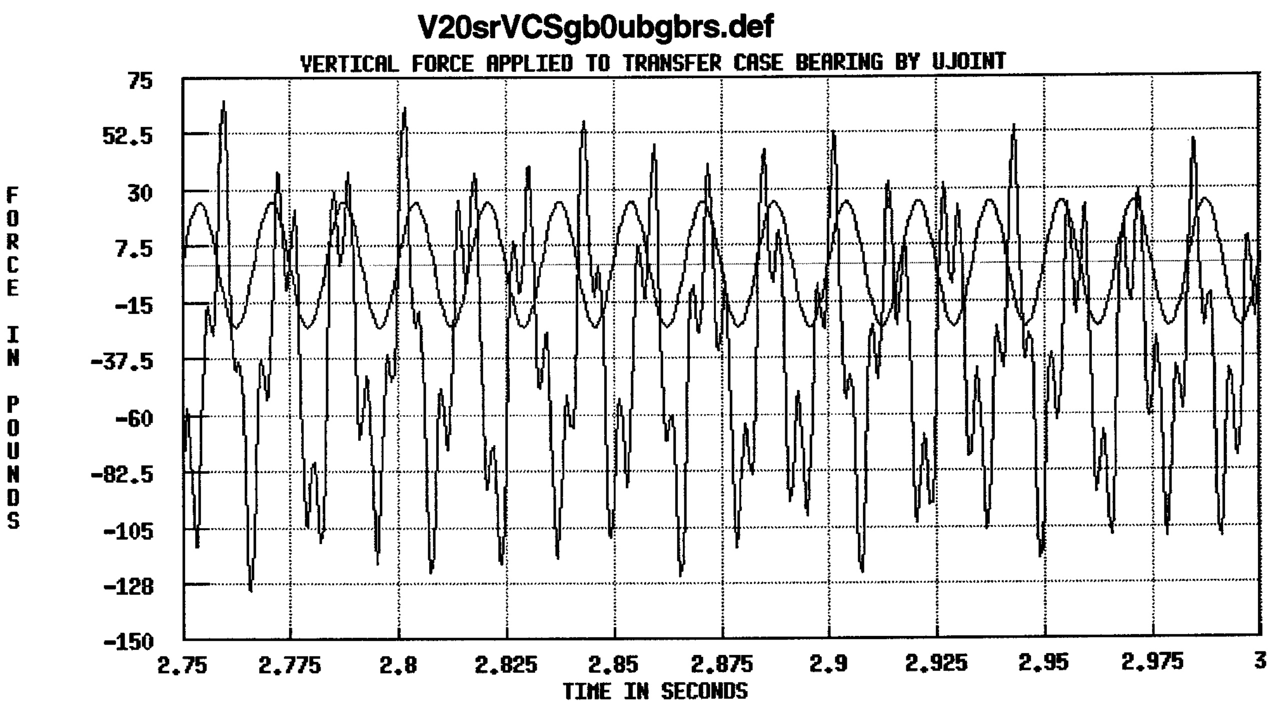


FIGURE 19

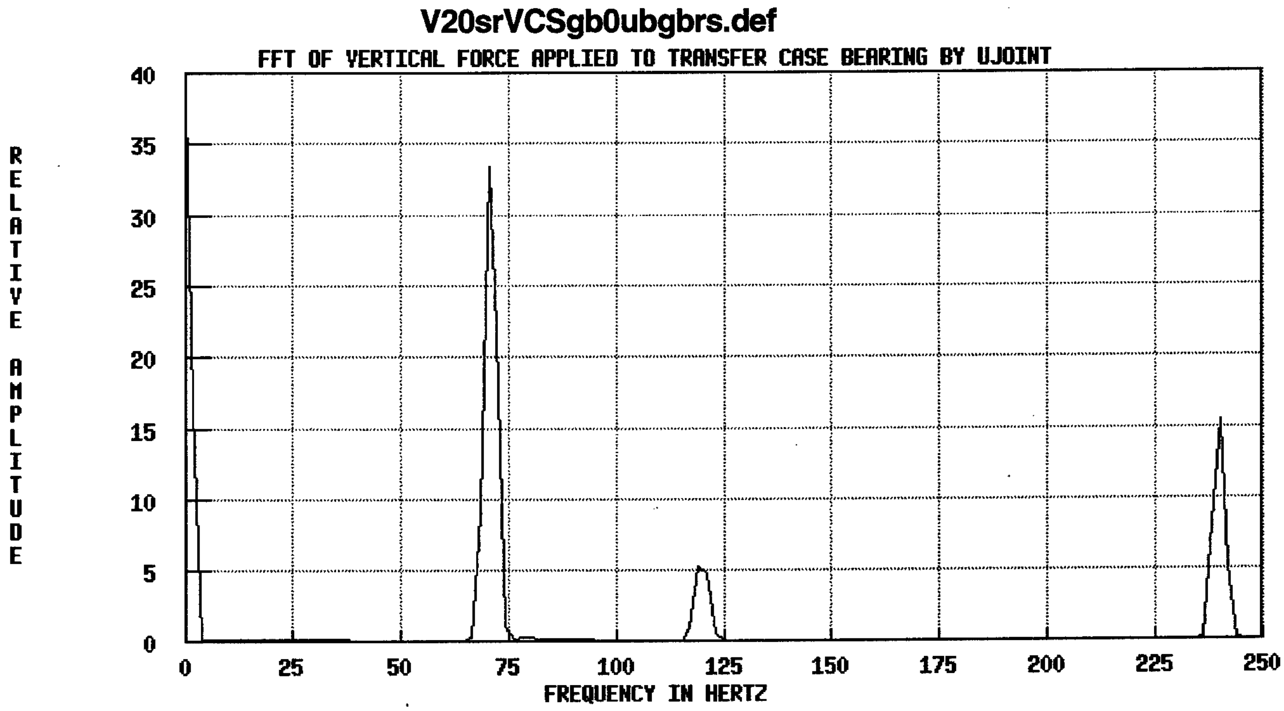


FIGURE 20

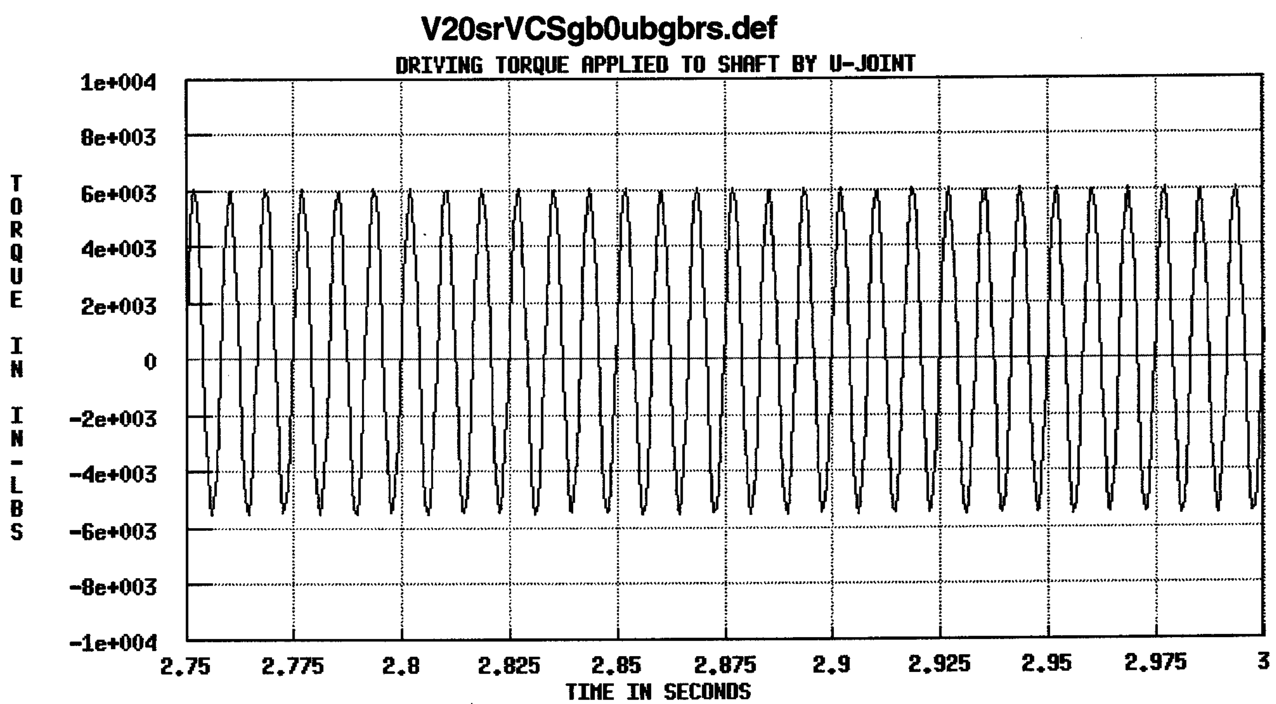


FIGURE 21

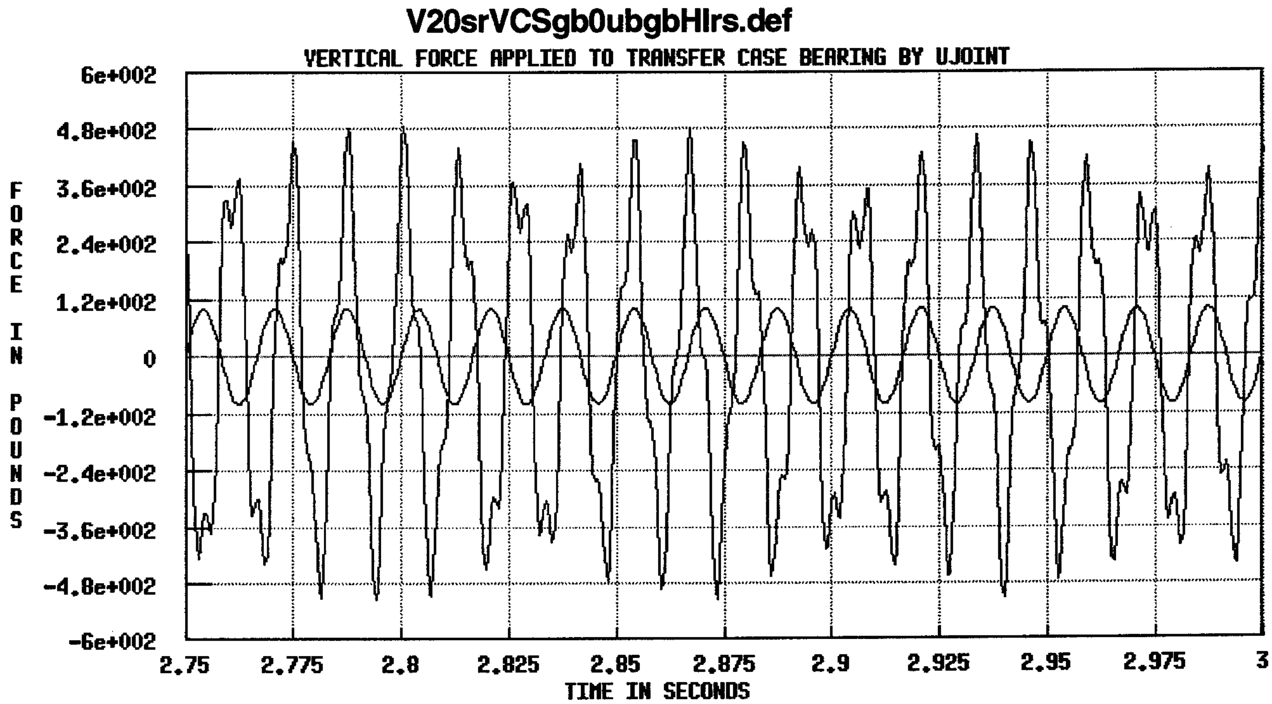


FIGURE 22

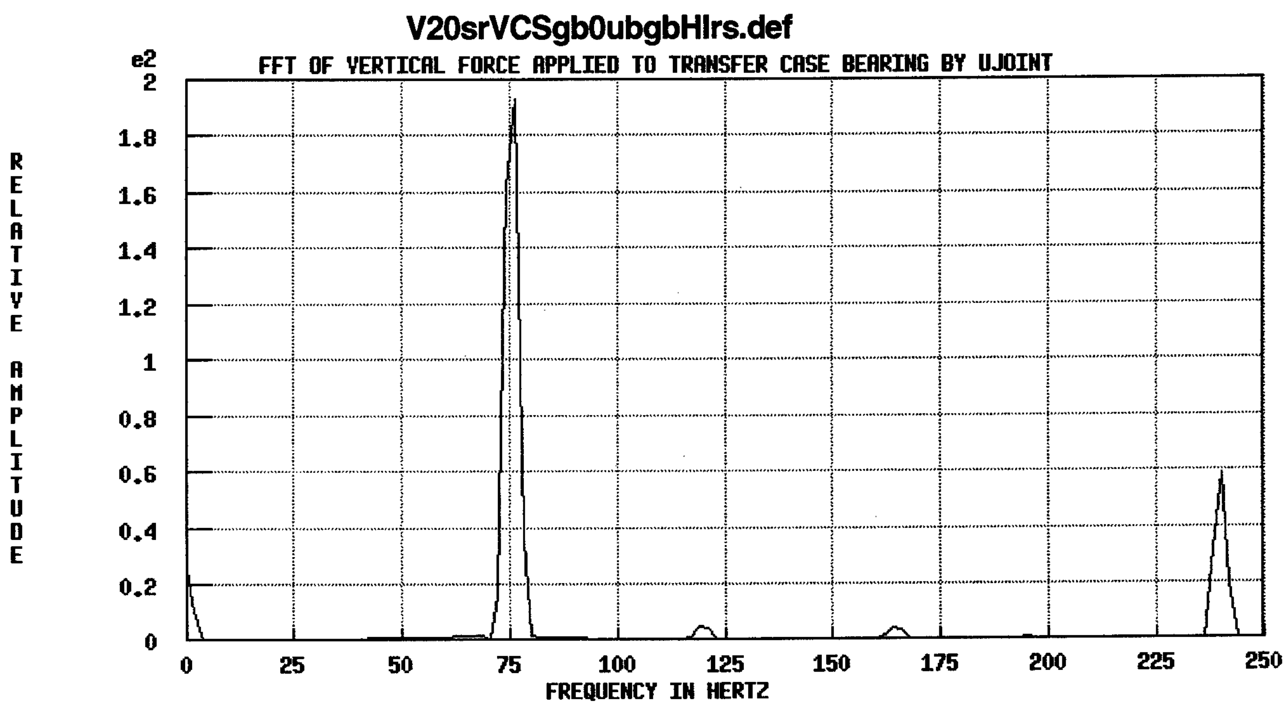


FIGURE 23

V20srVCSgb0ubgbHIrs.def

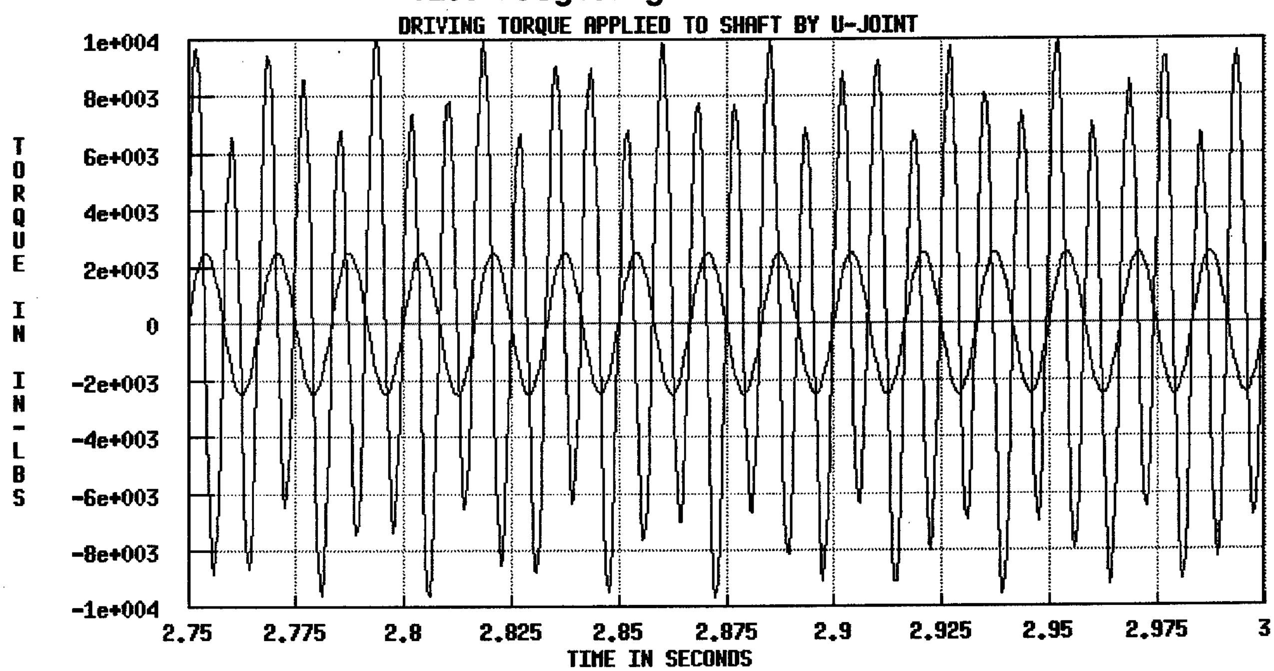


FIGURE 24

In run V20srVCSgb0ubgbLIrs.def, figure 16, (low shaft inertia) the vertical force is about 140 lbs. peak to peak. This is somewhat lower than with the standard inertia. The force is biased negatively due to the static weight of the shaft. Note that the force is not locked to rotation. The force FFT, figure 17, shows a peak at 70.3 Hz from the residual response to the startup transient. It also shows a weak peak at 120 Hz, second order of rotation. The driving torque, figure 18, applied to the tube is about 56 in-lbs. peak to peak biased upwards by the 240 in-lbs. drag torque. It is a second order signal.

The vertical force is about 175 lbs. peak to peak in run V20srVCSgb0ubgbrs.def, figure 19 (standard shaft inertia). The weight bias remains. The force waveform is more complex due to the stronger second and fourth order content as shown by the FFT, figure 20. The required driving torque, figure 21, is much higher but continues to show the drag torque bias.

The force trace for V20srVCSgb0ubgbHIrs.def, figure 22, is about 960 lbs. peak to peak with a strong beat frequency signature present. The FFT, figure 23, shows peaks at approximately 76 Hz, 120 Hz, 164 Hz, 195 Hz and 240 Hz. Other components are also present but at much lower amplitude. The significant change (times 10) in the torsional moment of inertia of the shaft has shifted the response to the startup transient upwards to 76 RPS. It has also increased in amplitude. The 120 and 240 Hz peaks are from the second and fourth order of shaft rotation. The 164 and 195 Hz energy results from non-linearities. The strong beat frequency results from the decaying startup transient and the order related responses. The driving torque signal, figure 24, again has increased and also shows the effects of the startup transient.

SEVENTH COMPARISON GROUP

The run pair, V20vlrVCSgb0ubgbLlrs.def and V20vlrVCSgb0ubgbLlrs0RT.def, had their run times increased to 20seconds. This enabled the damping to remove the residual response from the startup transient without impacting the amplitude of the signals of interest. This comparison demonstrates the effects of drag (load) torque without the confusing effects of the residual response to the startup transient. Please see the comments after the graphs. **V20vlrVCSgb0ubgbLlrs.def**

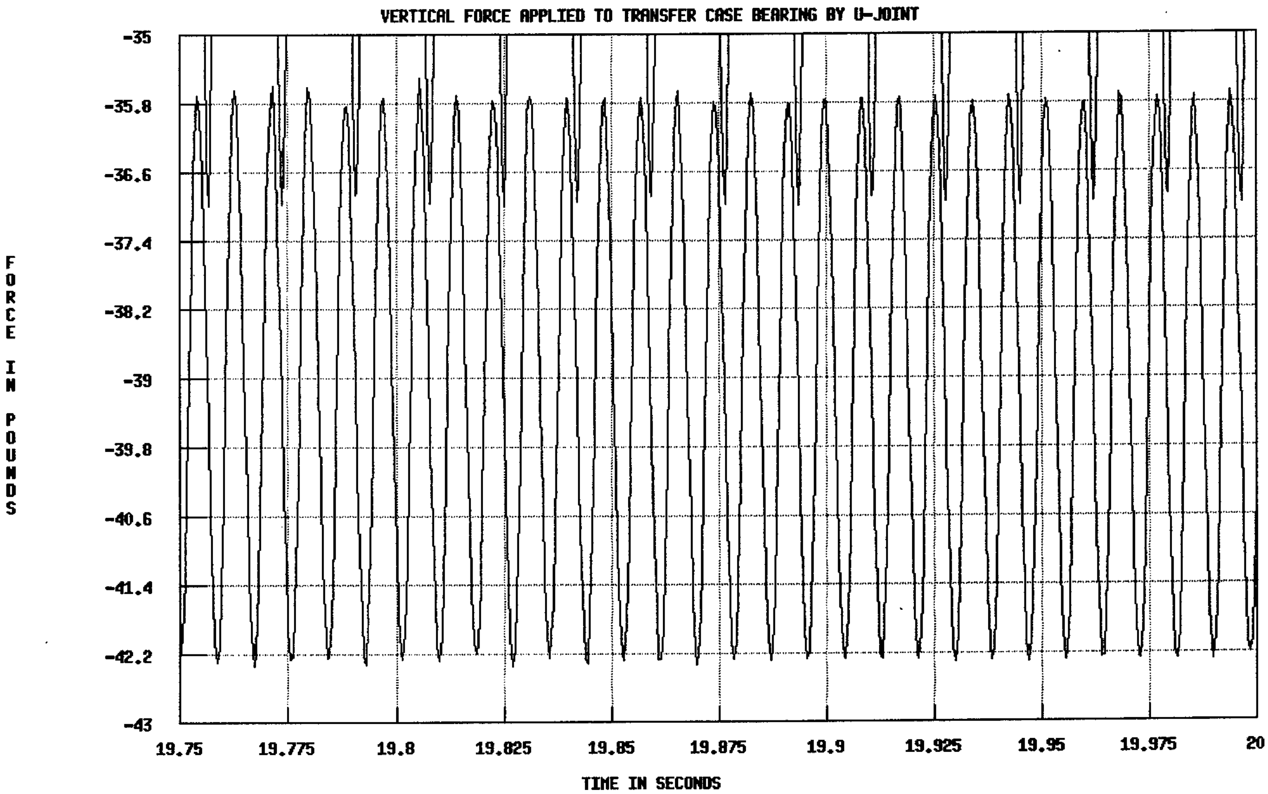
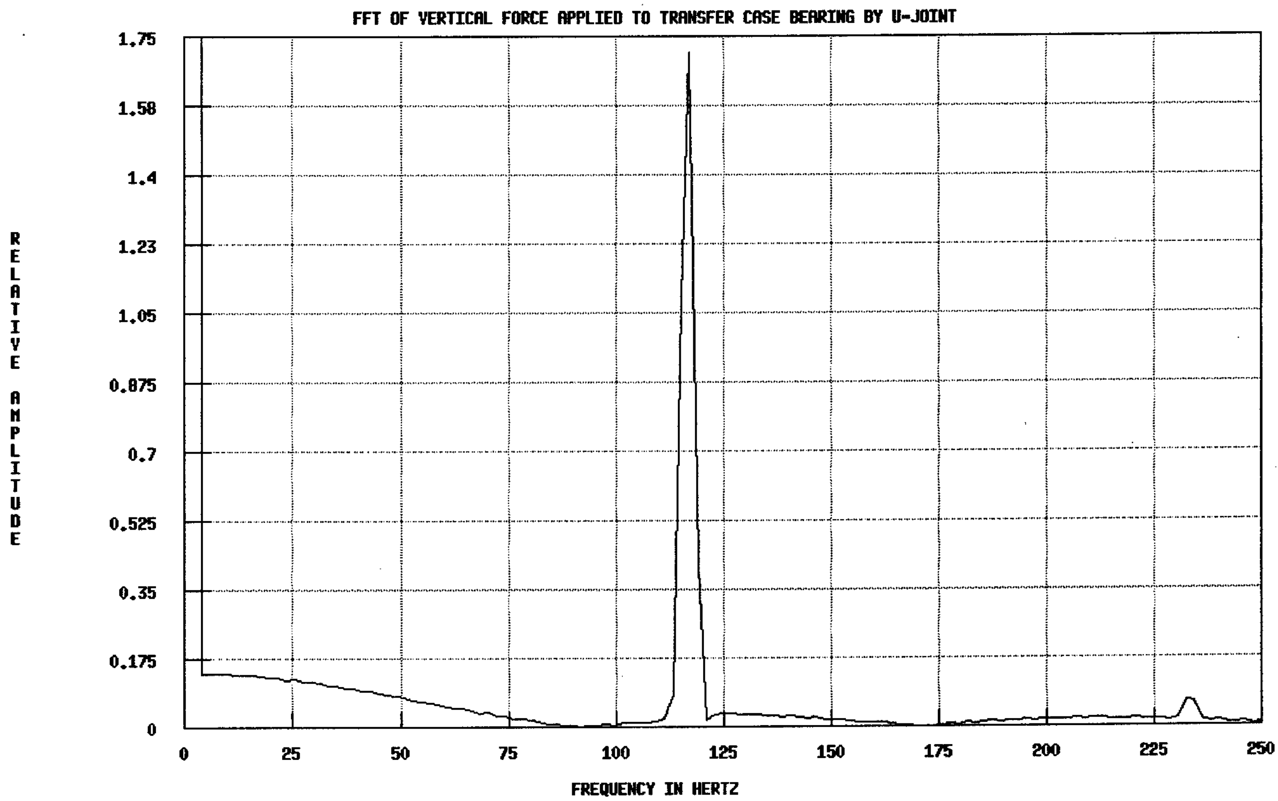


FIGURE 25

V20vlrVCSgb0ubgbLlrs.def



FFT OF VERTICAL FORCE APPLIED TO TRANSFER CASE BEARING BY U-JOINT
RELATIVE AMPLITUDE
1.75
1.58
1.4
1.23
1.05
0.875
0.7
0.525
0.35
0.175
0
0
25
50
75
100
125
150
175
200
225
250
FREQUENCY IN HERTZ

FIGURE 26

V20vlrVCSgb0ubgbLlrs.def

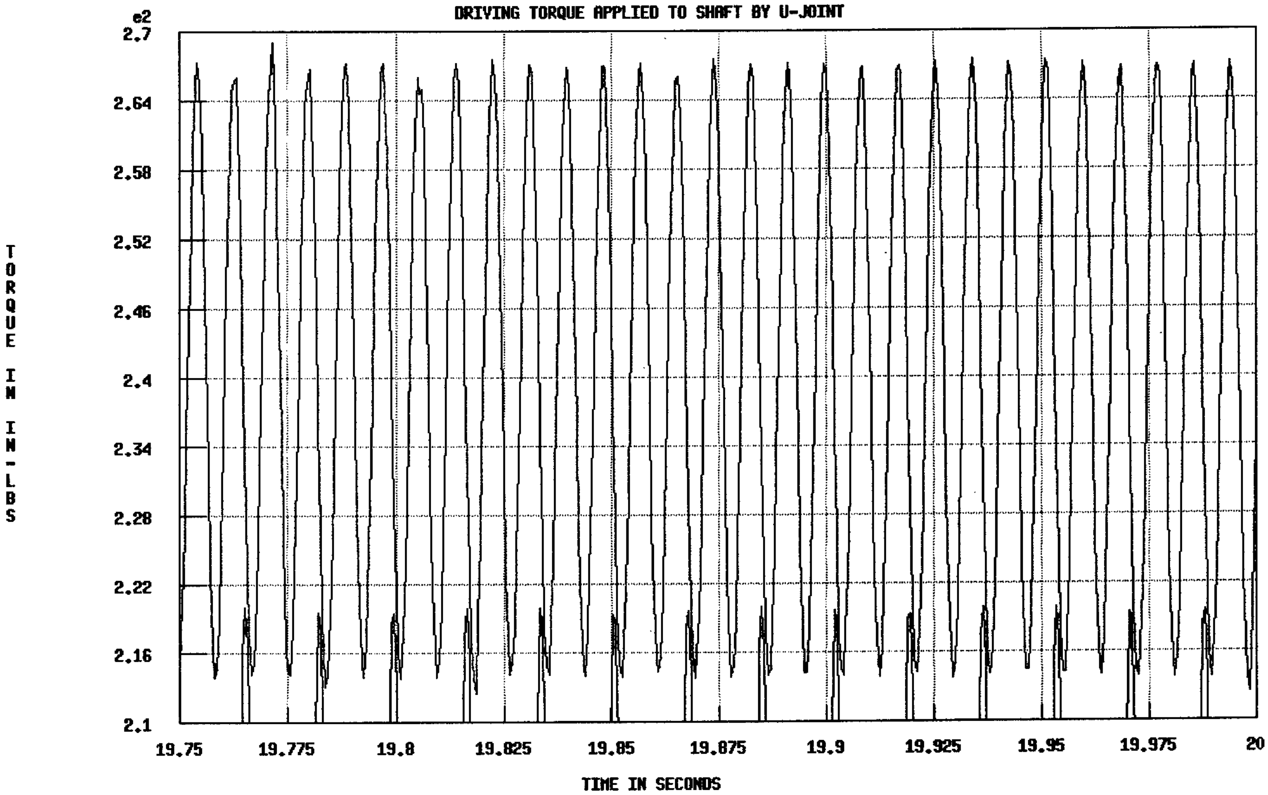


FIGURE 27

V20vlrVCSgb0ubgbLlrs0RT.def

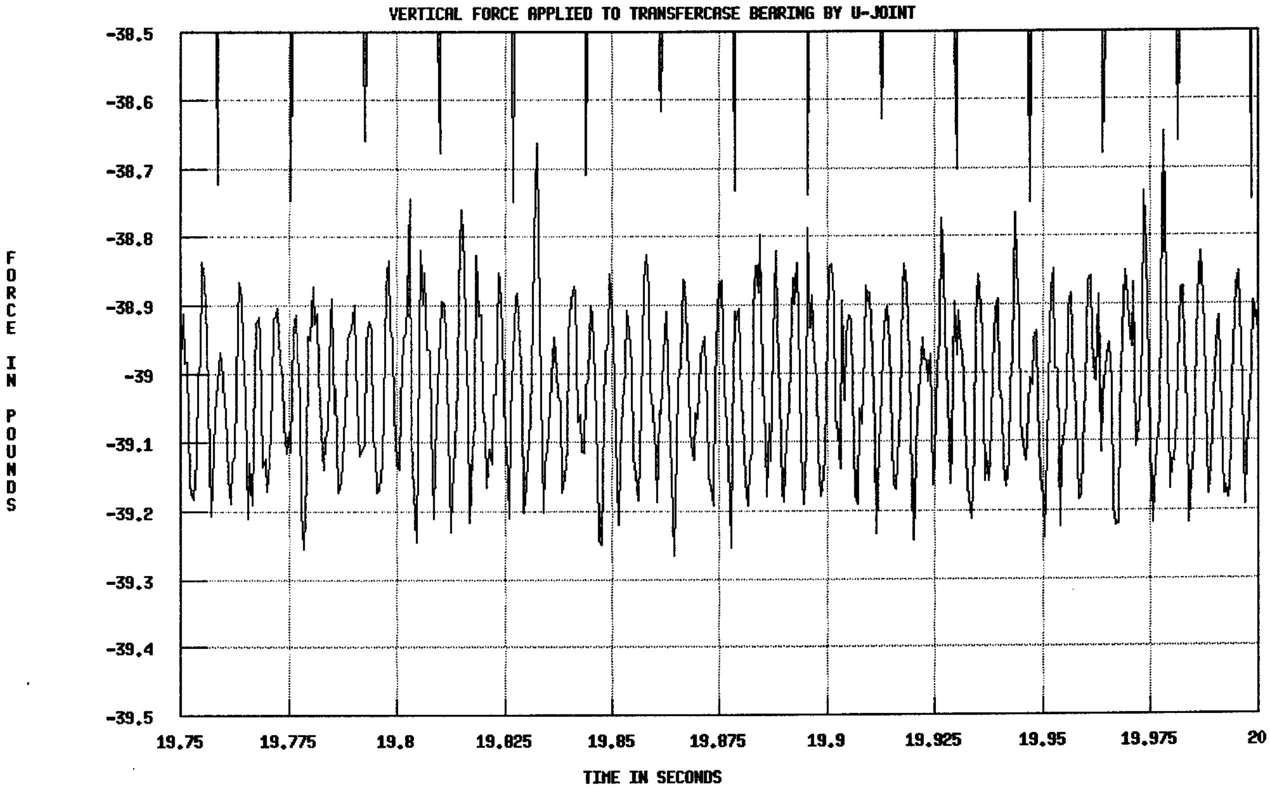

VERTICAL FORCE APPLIED TO TRANSFERCASE BEARING BY U-JOINT
FORCE IN POUNDS
-38.5
-38.6
-38.7
-38.8
-38.9
-39
-39.1
-39.2
-39.3
-39.4
-39.5
19.75
19.775
19.8
19.825
19.85
19.875
19.9
19.925
19.95
19.975
20
TIME IN SECONDS

FIGURE 28

V20vlrVCSgb0ubgbLlrs0RT.def

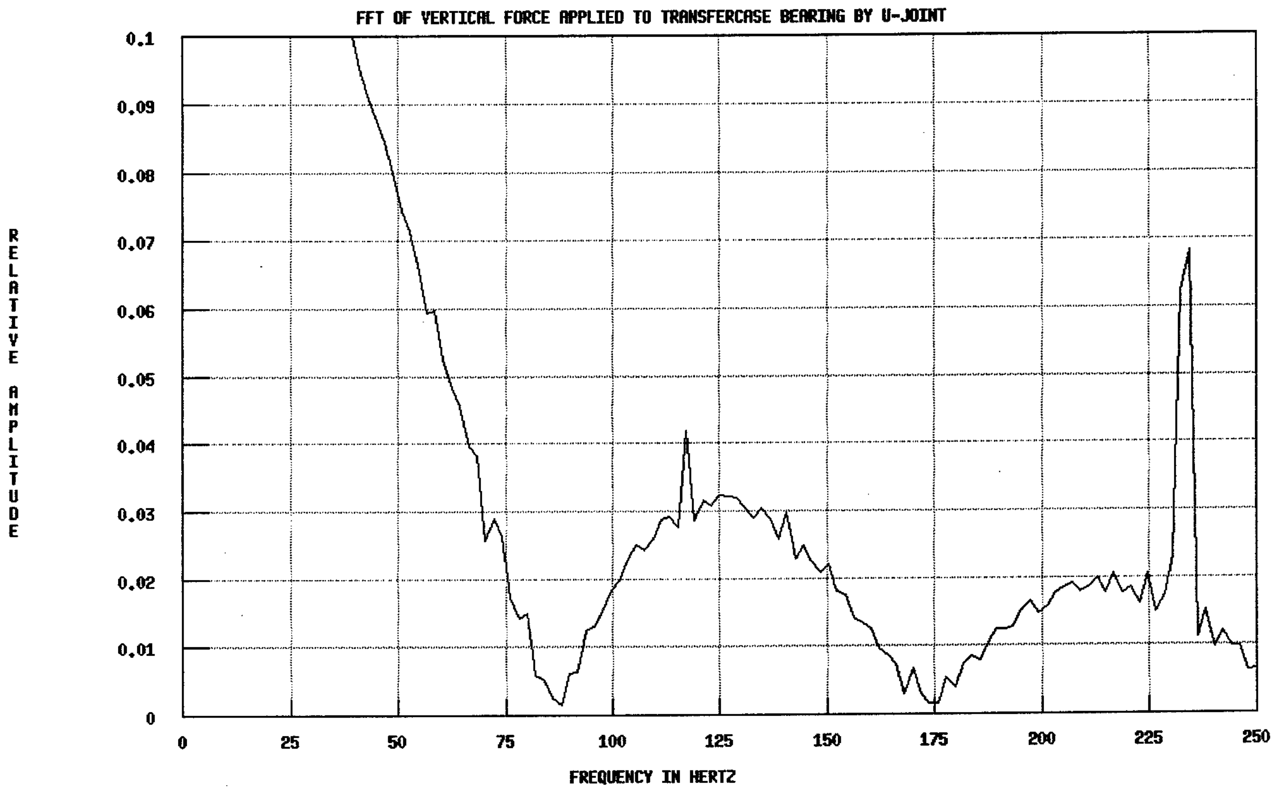


FIGURE 29

V20vlrVCSgb0ubgbLlrs0RT.def

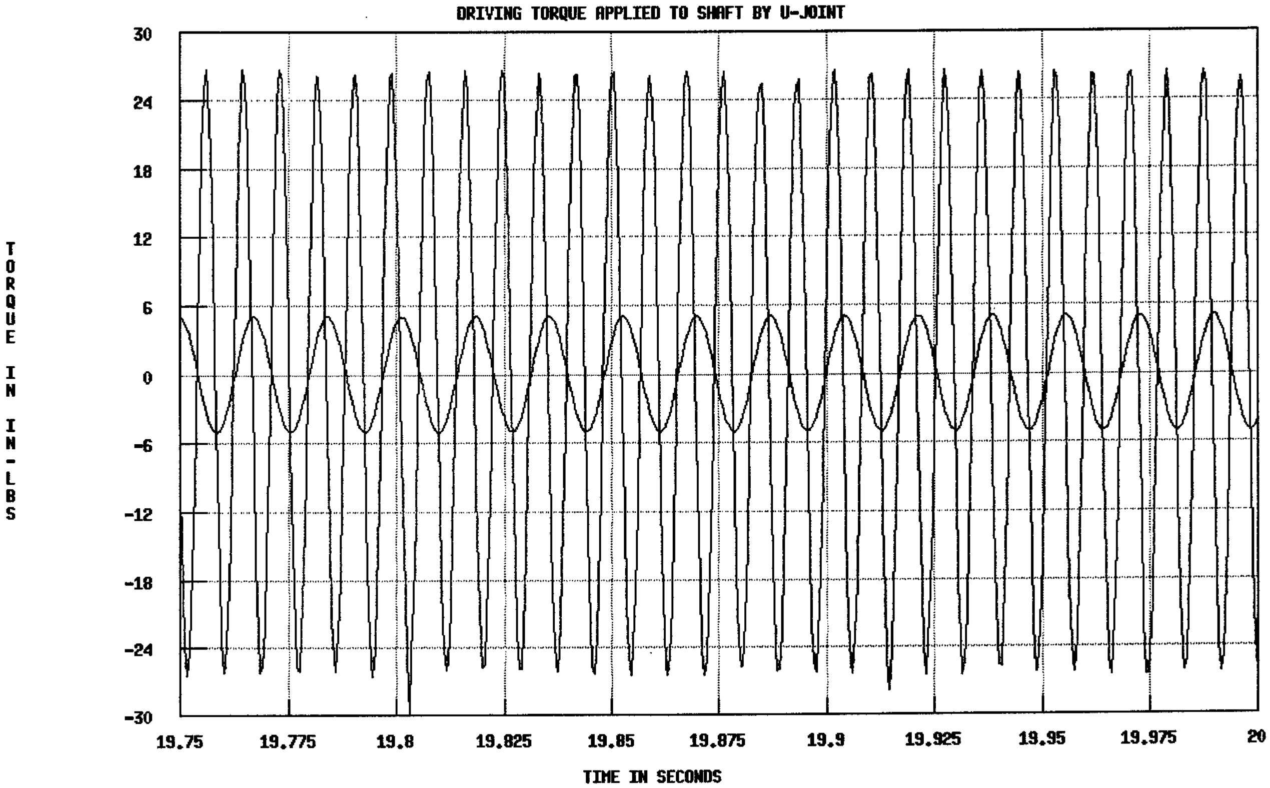


FIGURE 30

Both vertical force traces show the vertical force bias due to gravity of about 39 plus lbs. The force trace for the run with drag torque, figure 25, is the sum of the reaction components to both drag torque and the twice per rev shaft velocity change. The AC force is about 7-lbs. peak to peak, predominately second order. The AC force for the run without drag torque, figure 28, is about 0.4 lbs. peak to peak, predominately fourth order. The required driving torque, figure 27, for the run with drag has a 56 in-LB peak to peak second order AC component added to a 240 in-lbs. DC component. The required driving torque without drag, figure 30, is also 56 in-lbs. peak to peak second order AC without a DC component. The FFT for the run with drag, figure 26, shows a second order component and a weaker fourth order. The no drag FFT, figure 29, shows weak second and fourth order components from the low shaft inertia being accelerated twice per rev. However the fourth order is about twice the second order component.

The shaft's inertia requires a second order driving torque that is symmetric around zero. However, the reaction forces are second and fourth order with fourth order dominant. The drag/load torque adds, as expected, a driving torque requirement. This adds a DC or low frequency AC bias (based on the current mission profile) to the driving torque. The reaction forces to a DC torque addition are second order. An AC drag/load torque will generate sum and difference frequencies with the second order reaction forces.

Figures 5 and 6 document V20vlrVCSgb0ubgbrs.def, a long run to allow the startup transient to decay. It has a shaft with no unbalance, nominal drag torque and standard inertia. Figures 25 and 26 above (V20vlrVCSgbOubgbLlrs.def) are from a similar run with only the shaft inertia changed. The reaction force trace in figure 5 shows strong fourth relative to second order and is confirmed by the FFT in figure 6. Reducing the shaft inertia by a factor of 100 (figures 25 and 26) reduces the fourth order much more than second. This confirms that shaft inertia contributes second and fourth order reaction forces with the fourth being dominant. This also confirms that the reaction force to a DC torque addition is second order.

EIGHTH COMPARISON GROUP

The run pair, V20srVCSgb0ubgbLIrs.def and V20srVCSgb1ubgbLIrs.def, shows the effects of adding unbalance to a "perfect" shaft with low inertia and minimal drag/load torque. The graphs from the V20srVCSgb0ubgbLIrs.def run were displayed above but are repeated below.

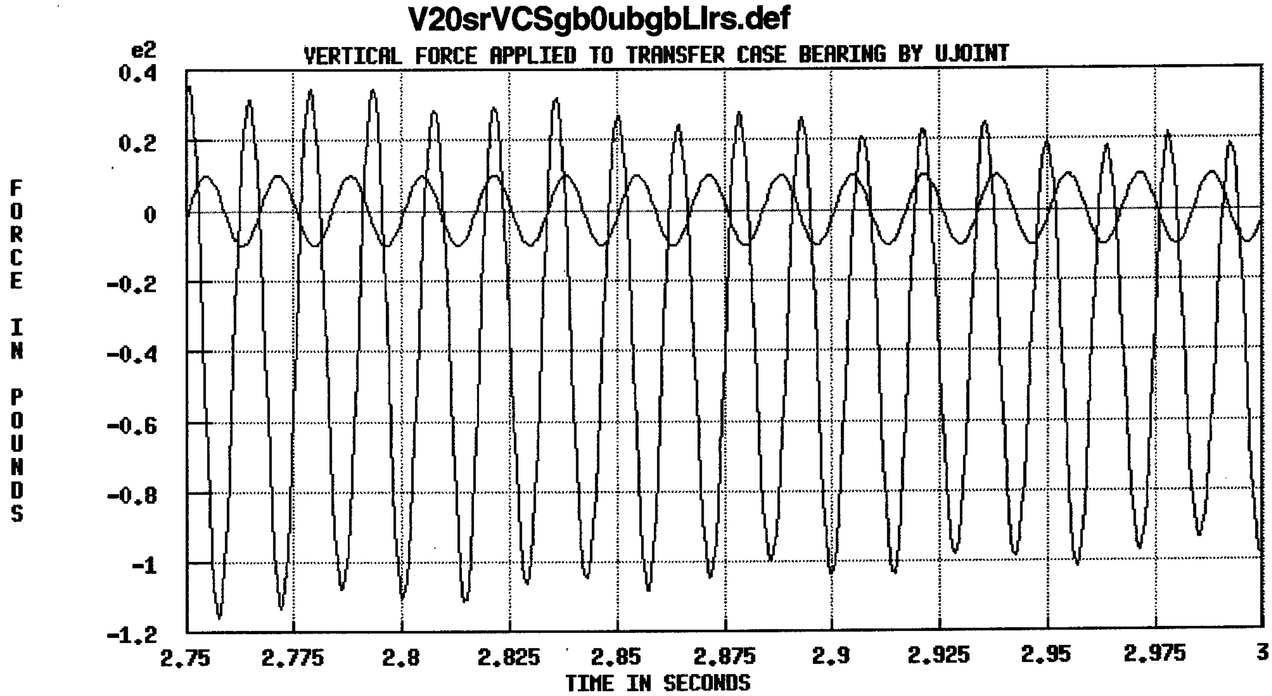


FIGURE 31

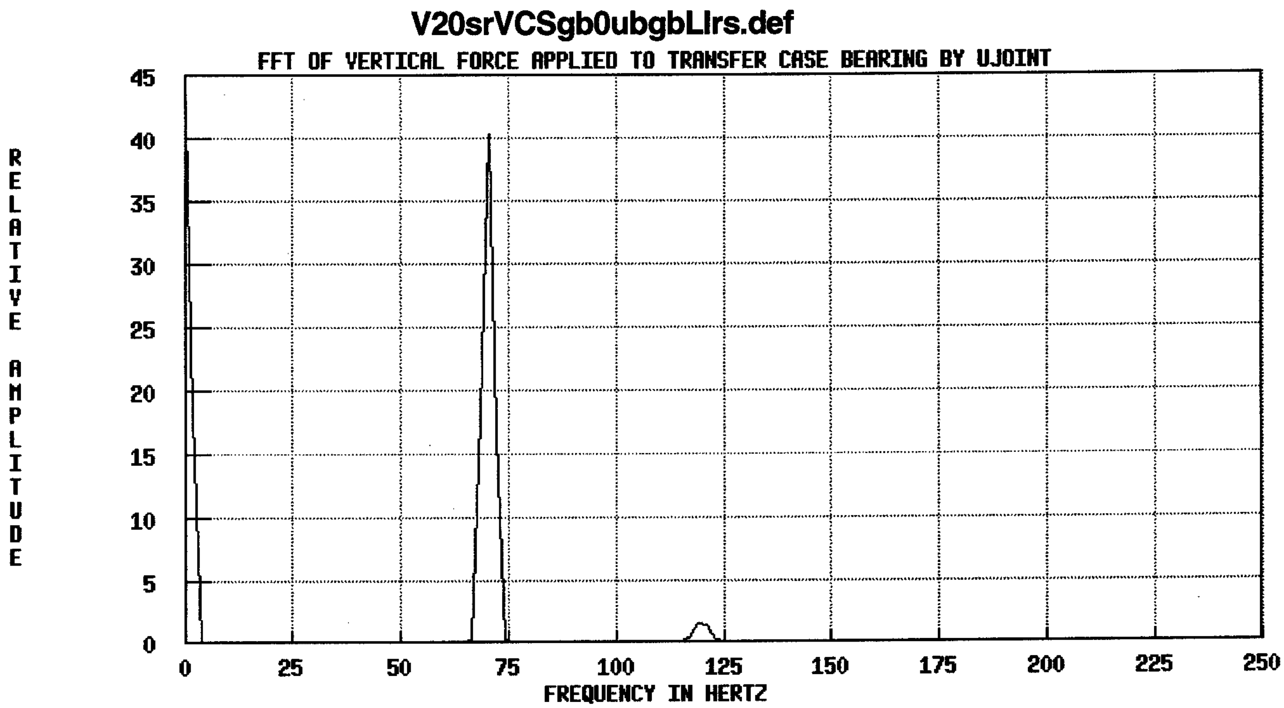


FIGURE 32

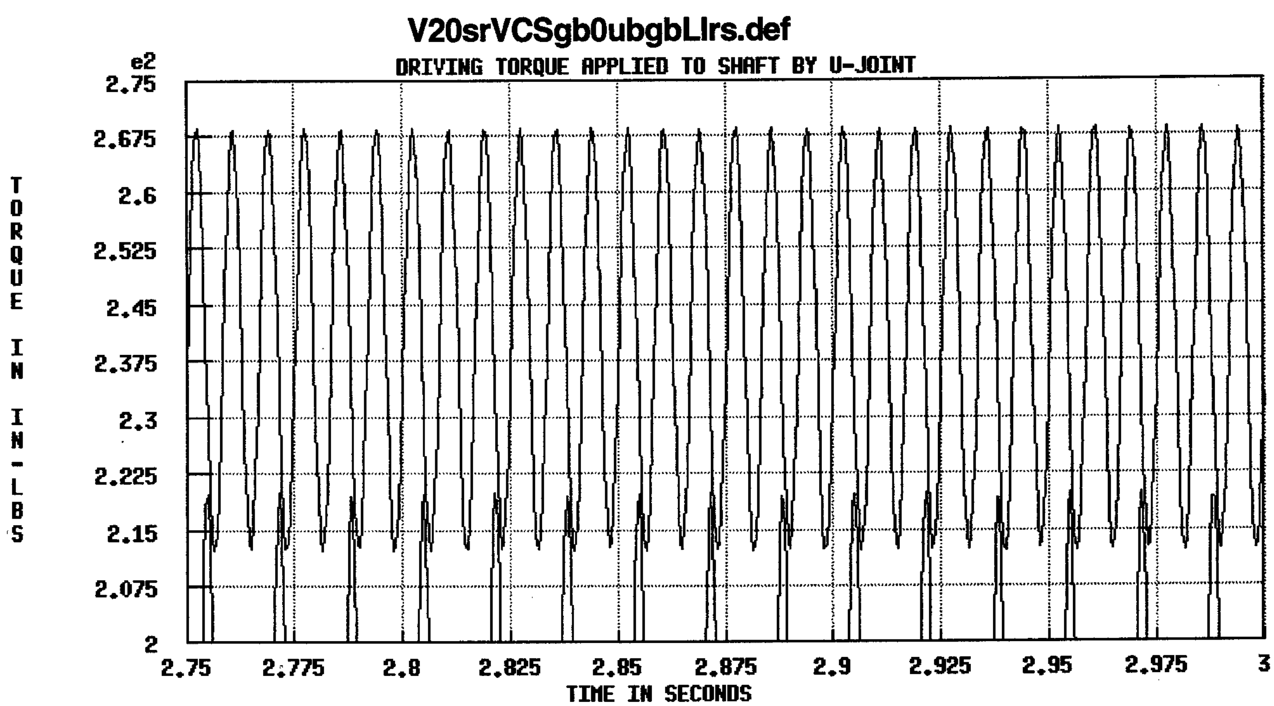


FIGURE 33

V20srVCSgb1ubgbLlrs.def

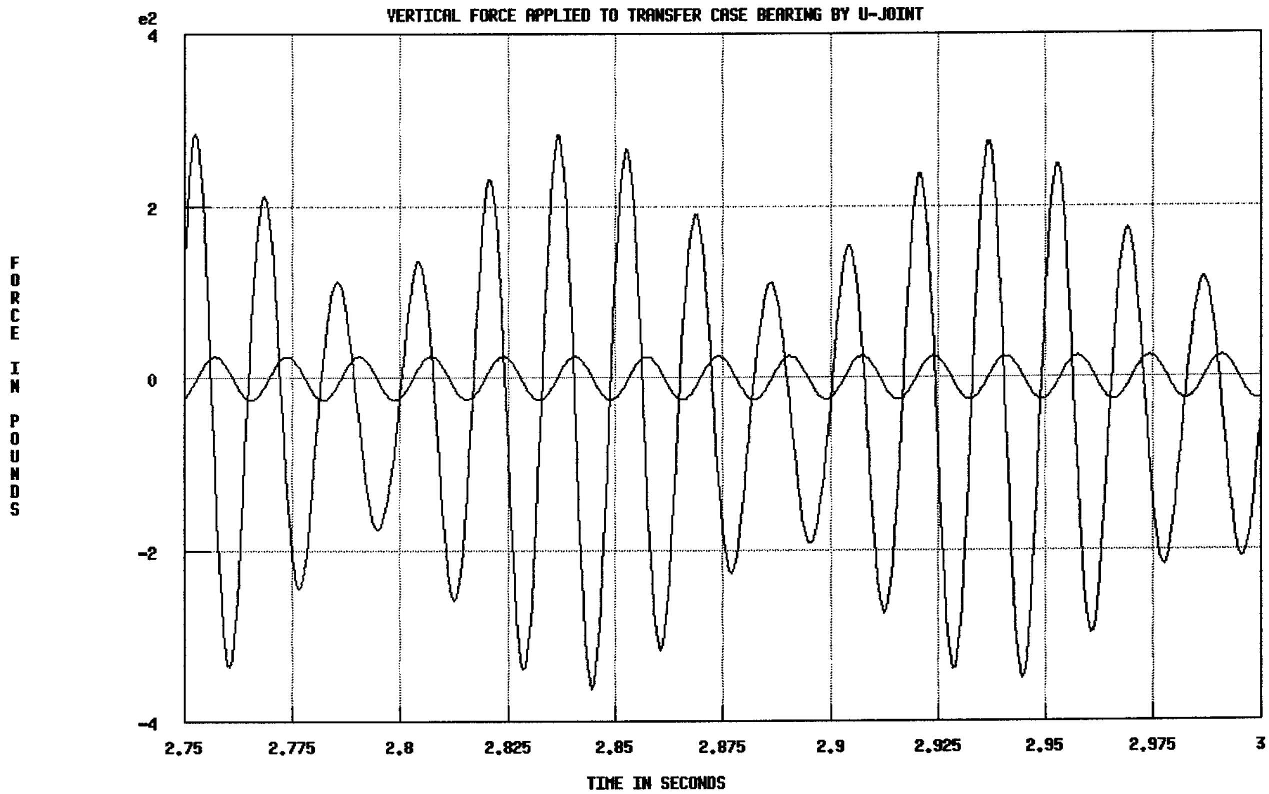


FIGURE 34

V20srVCSgb1ubgbLlrs.def

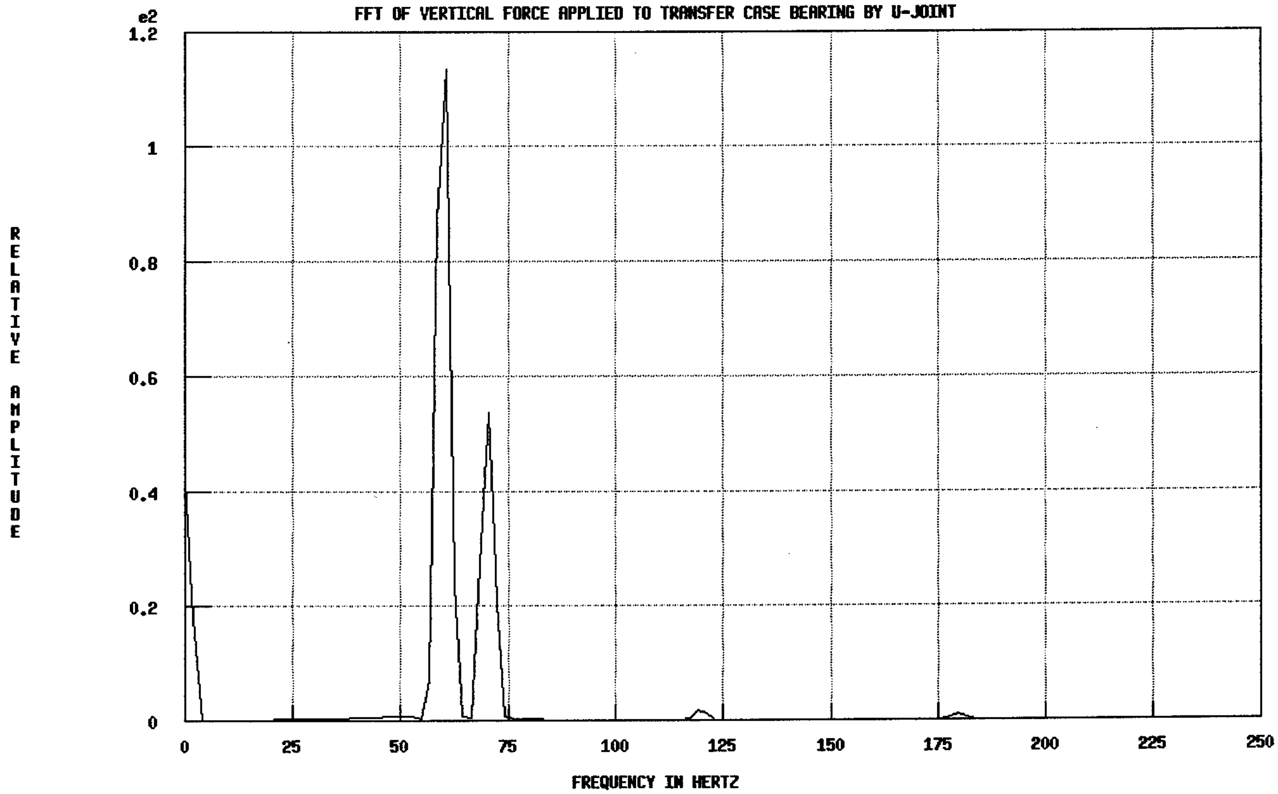

FFT OF VERTICAL FORCE APPLIED TO TRANSFER CASE BEARING BY U-JOINT
e2
1.2
1
0.8
0.6
0.4
0.2
0
RELATIVE AMPLITUDE
0
25
50
75
100
125
150
175
200
225
250
FREQUENCY IN HERTZ

FIGURE 35

V20srVCSgb1ubgbLlrs.def

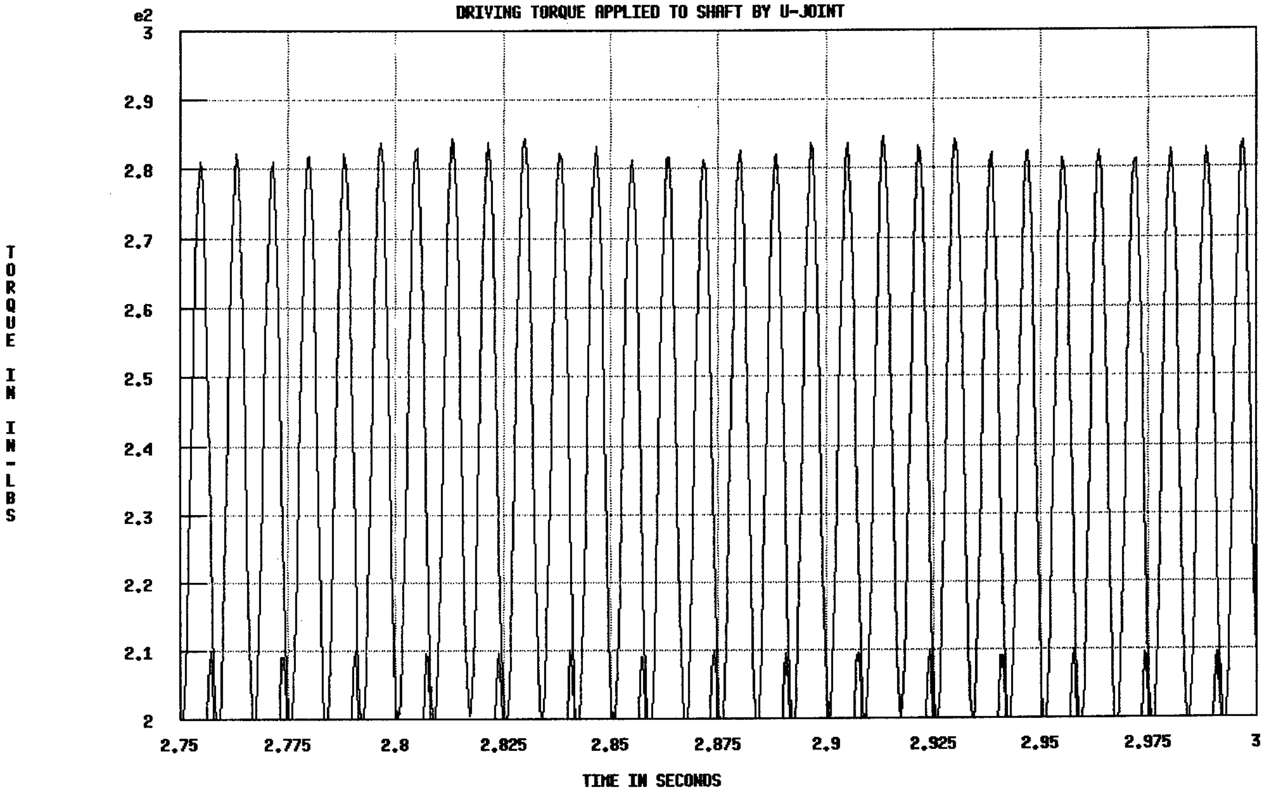


FIGURE 36

Figure 31 shows a decaying response to the startup transient with an additional second order force. This is also shown by its FFT, figure 32. Figure 33 shows the second order driving torque required by the drag torque and small shaft inertia. Figure 34 displays a more complex force waveform with a strong beat pattern. Figure 35, its FFT, shows a large component at 60 Hz caused by the unbalance. The component at 70 + Hz from the decaying response to the startup transient has increased by 25 % but is less than half of the 60 Hz component. These two are the source of the beat. The magnitude of the residual response is consistent with that generated by broad band noise input to the drive train. The second and fourth order components are nearly unchanged between the two runs, however addition of the unbalance has added a slight third order component.

Figures 3 and 4 document a run with standard shaft inertia and maximally permitted imbalance. Figures 34 and 35 above document a run with low shaft inertia and maximally permitted imbalance. The FFT's figures 4 and 35 show that the addition of shaft inertia (figure 4) increases the second, third and fourth order components. The fourth order component shows the largest increase. This mirrors the results discussed in the sixth comparison group that deals with inertial effects.

NINTH COMPARISON GROUP

Comparison group four used a shaft with standard inertia and drag torque. One run had some residual response to the startup transient. (Again this decaying response approximates the effects of broad band noise input to the drive train.) In the other run the residual response had fully decayed. This comparison group is designed to highlight the signature of just the decaying response.

The run that generated figures 37, 38 and 39 used low shaft inertia and no drag torque. The decaying response is present. Figures 40, 41 and 42 below are repeated from comparison group seven. Their generator also had low shaft inertia and no drag torque. However the residual response is fully decayed. Additional comments follow the graphs.

V20srVCSgb0ubgbLIrs0RT.def

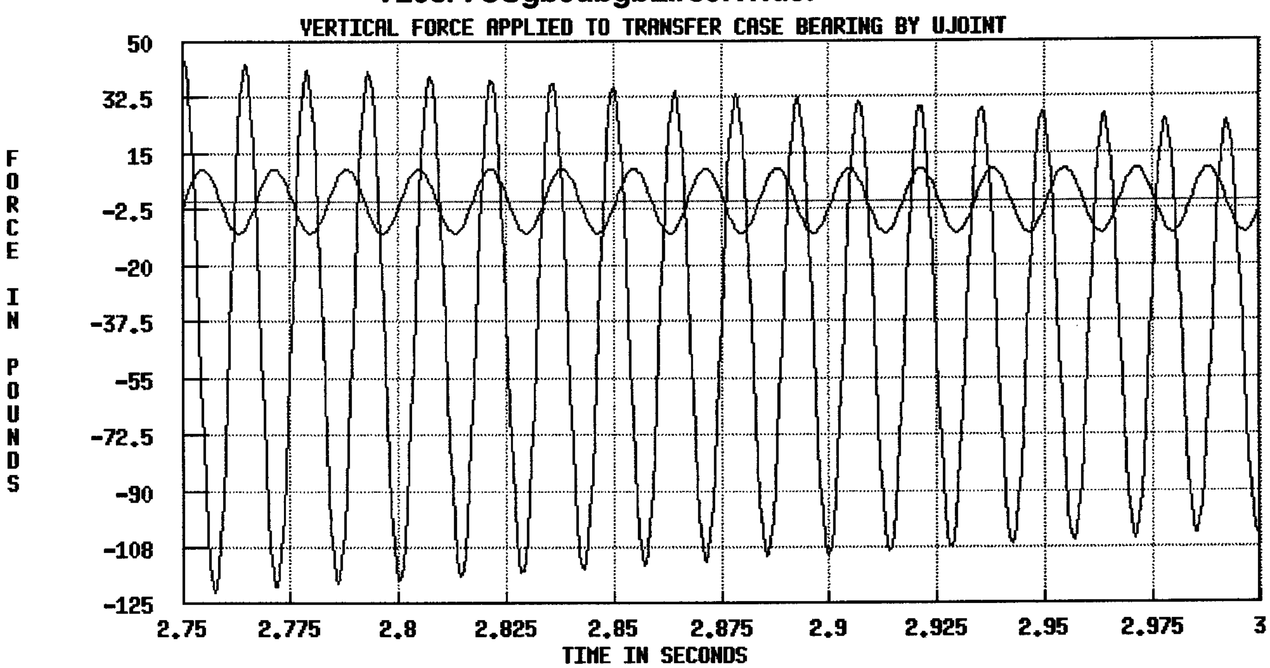
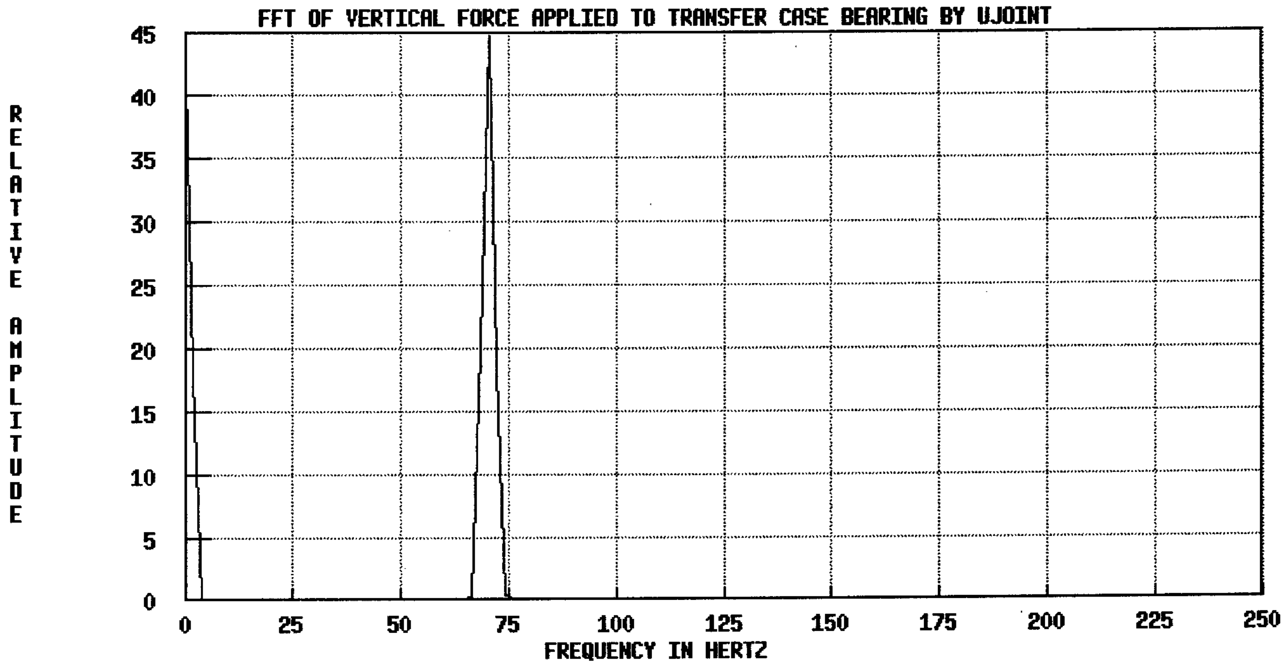


FIGURE 37

V20srVCSgb0ubgbLIrs0RT.def



FFT OF VERTICAL FORCE APPLIED TO TRANSFER CASE BEARING BY UJOINT
RELATIVE AMPLITUDE
45
40
35
30
25
20
15
10
5
0
0
25
50
75
100
125
150
175
200
225
250
FREQUENCY IN HERTZ

FIGURE 38

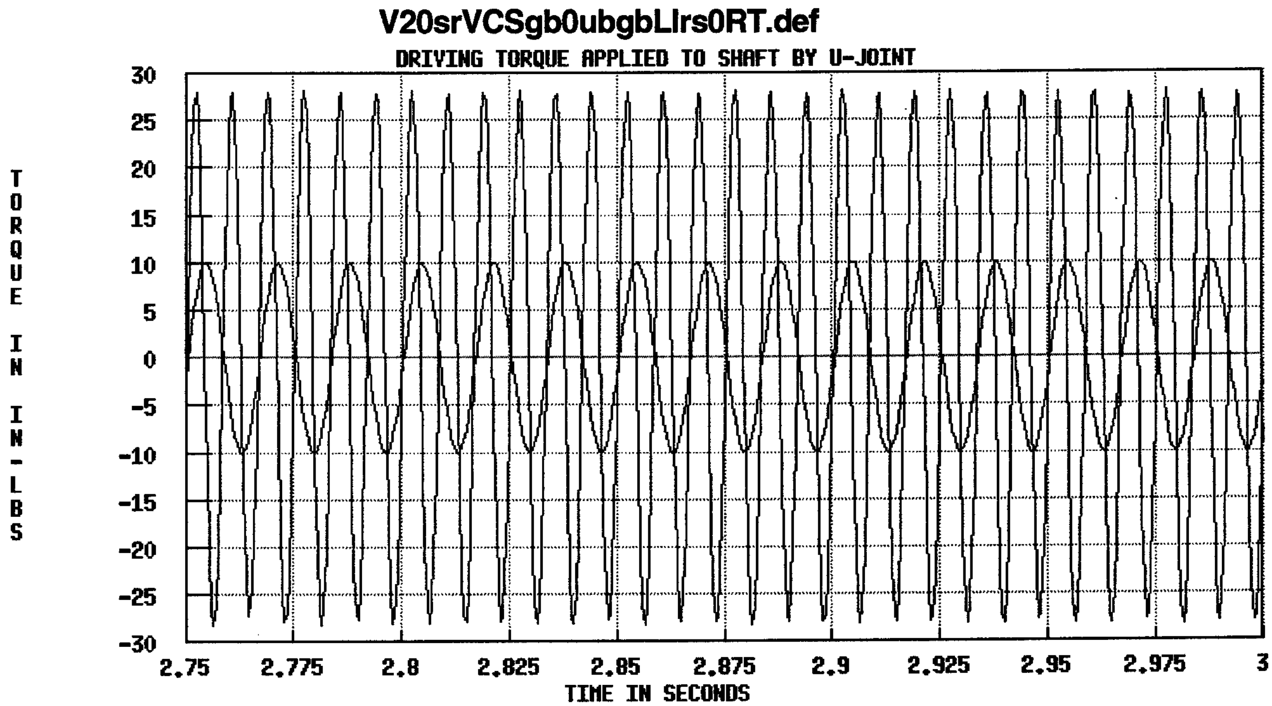

V20srVCSgb0ubgbLIrs0RT.def
DRIVING TORQUE APPLIED TO SHAFT BY U-JOINT
TORQUE IN IN-LBS
30
25
20
15
10
5
0
-5
-10
-15
-20
-25
-30
2.75
2.775
2.8
2.825
2.85
2.875
2.9
2.925
2.95
2.975
3
TIME IN SECONDS

FIGURE 39

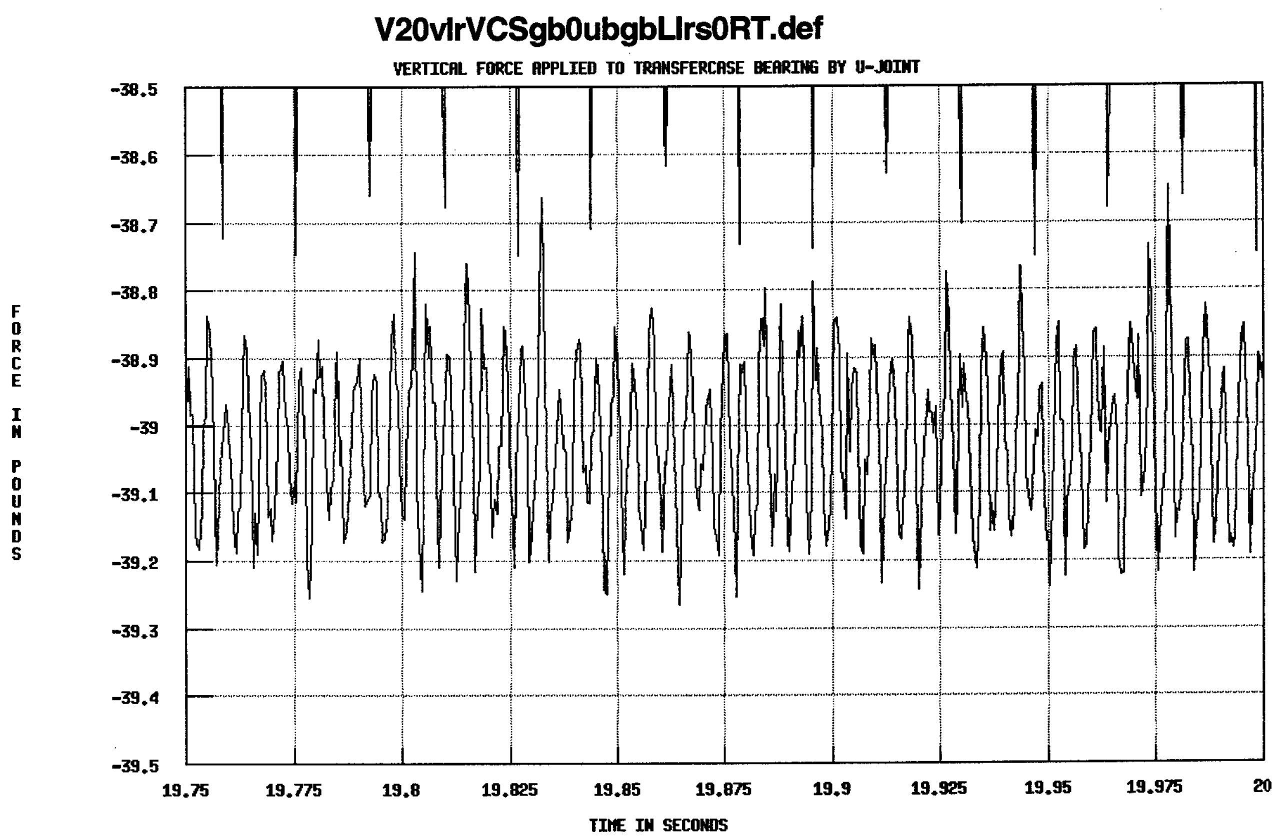

V20vlrVCSgb0ubgbLIrs0RT.def
VERTICAL FORCE APPLIED TO TRANSFERCASE BEARING BY U-JOINT
FORCE IN POUNDS
-38.5
-38.6
-38.7
-38.8
-38.9
-39
-39.1
-39.2
-39.3
-39.4
-39.5
19.75
19.775
19.8
19.825
19.85
19.875
19.9
19.925
19.95
19.975
20
TIME IN SECONDS

FIGURE 40

V20vlrVCSgb0ubgbLlrs0RT.def

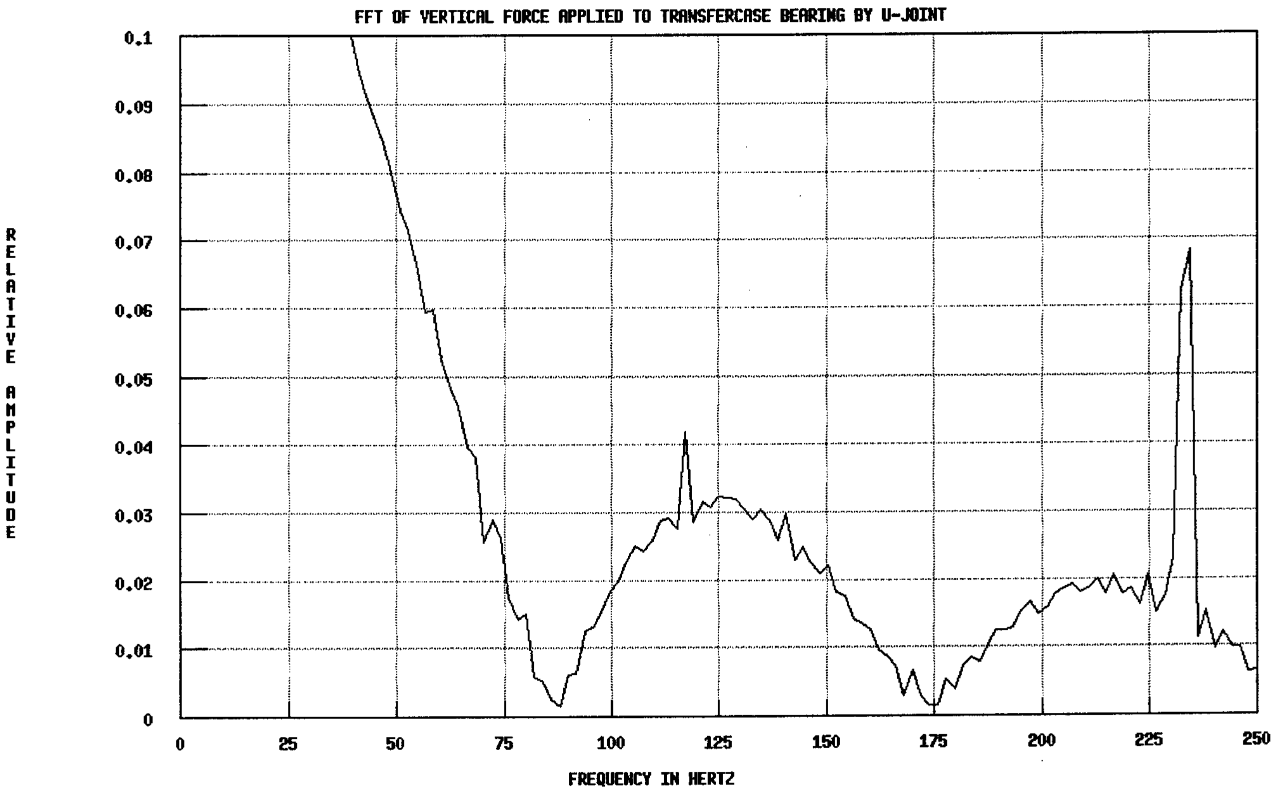


FIGURE 41

V20vlrVCSgb0ubgbLlrs0RT.def

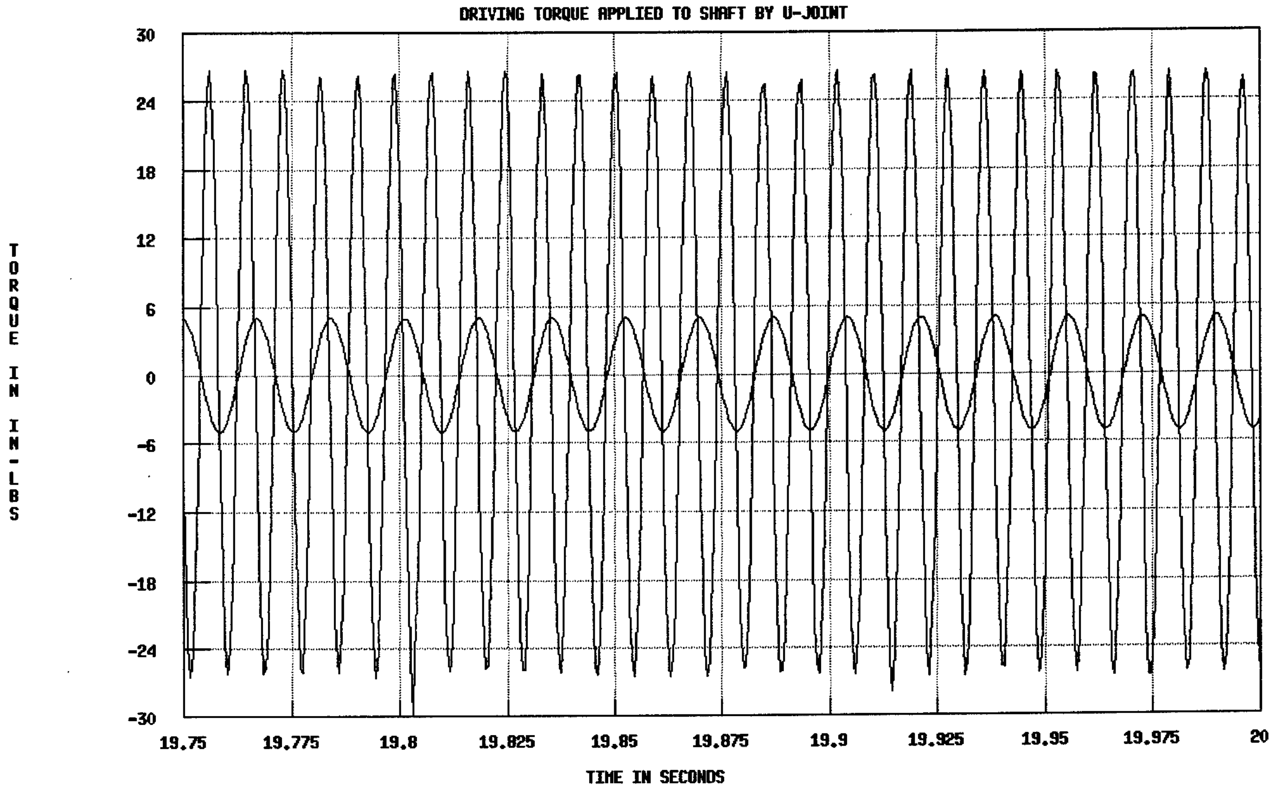


FIGURE 42

Figure 37, the force trace, shows the signature of the decaying transient. Its FFT, figure 38, confirms that it the strongest signal present. A zoomed FFT, not shown, shows small levels of signal at second and fourth order. A small signal also appears at slightly above third order. Figure 39, the driving torque signal, is clearly second order, not related to the transient. It is about 56.1 in-lbs. peak-to-peak. It is the torque required to accelerate and decelerate the small shaft inertia as well as replace energy dissipated in damping. Figure 40 shows the forces after the startup transient has fully decayed. It is a mixture of second and fourth order at low magnitude. A zoomed FFT, figure 41, shows low levels of second and fourth order. These are about half of their counterparts discussed above. Also there is no energy around third order. The driving torque signal, figure 42, is about 52.7 in-lbs. peak-to-peak. System geometry changes as the startup transient decays and less energy is dissipated in damping.

Conclusions

The first comparison group demonstrates that the encompassing system impacts the shaft's critical speed. A shaft with bending resonance (critical speed) measured in the "Free Free" state will have a lower critical speed when installed in a vehicle.

The second group illustrates the effects of a slip spline on torsional compliance and in turn torsional resonances. Prior work showed effects of a slip spline on bending resonance (critical speed).

Group three illustrated that shaft imbalance results in first order reaction forces at the joint shaft bearings. This first order phenomenon will interact with other vibrations to cause complex force signatures including beats.

Comparison group four employs a decaying response to a startup transient to simulate the system response caused by random noise inputs to the driveline. Examples are firing frequency components, gear chatter and road inputs. It shows interaction between that response and responses due to drag torque and shaft inertia. Typical beat signals are shown.

Comparison group five uses three drag/load torque levels to qualify the impact of loading on the force signatures. The signature is second order but its interaction with other force components will cause complex signals. As load torque changes frequency, such as in vehicle acceleration both frequency domain and order tracking analysis may be required to sort out cause and effect.

The sixth group employs three shaft polar moment of inertia levels to highlight shaft inertial effects. Non-zero joint angles cause twice per revolution acceleration deceleration events. The relative shaft VS encompassing system inertia ratio will determine who does the dance. In most systems it is primarily the shaft. Shaft inertia "creates" second and fourth order responses. The required driving torque is second order while the vertical joint reaction force is primarily fourth order.

Group seven was designed to illustrate the signature of drag/load torque without the confusing interactions with inertial effects and decaying startup transient present in group five. The startup transient has fully decayed in both runs in the group. Low shaft inertia was also used in both runs. The resultant FFT's clearly showed drag/load torque caused primarily second order reaction forces.

Comparison group eight was designed to illustrate the signature of beating caused by the interaction of reaction forces from shaft imbalance, the decaying response to the startup transient (simulating the system response caused by random noise inputs) and low drag torque. The low shaft inertia used in both runs enabled a clearer beat pattern to be seen.

Comparison group nine highlighted the signature of just the decaying response.

All machine-readable files, which include the model definition files and graphical data will be included with the balance of the project documentation package.

Ujoint_comp_runs.doc

Appendix IV

Balance Measurements

Straightness Measurements

Hinging Measurements

End Play Measurements

Raw Data

STATIC DATA

Supplier: Meritor A1 LMTV Rear statdynA1

Date	Shaft No.	Weights Slip-end	Weights Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
1/11	Mer-PR-L-A1-1	39	31.5	61.5	0.009	0.003	0.013	0.014	0.0075	0.002	0.001	0.001	0.005	
1/9	Mer-PR-L-A1-2	39	31.5	61.5	0.011	0.013	0.013	0.033	0.002	0.002	0.002	0.002	0.005	
1/15	Mer-PR-L-A1-3	39	31.5	61.5	0.011	0.012	0.015	0.015	0.004	0.015	0.002	0.005	0.005	Would not balance-used shaft

DYNAMIC DATA

Supplier:

Date	Shaft No.	Balance As Rec'd Slip-end oz.in/spec	angle	Weld-end oz.in/spec	angle	Balance Minimum Slip-end oz.in/spec	angle	Weld-end oz.in/spec	angle	Balance Final Slip-end oz.in/spec	angle	Weld-end oz.in/spec	angle	Remarks
1/11	Mer-PR-L-A1-1	5.31/3.9	200.3	6.67/3.15	299.7	5.49/3.9	78	4.82/3.15	127.1					Would not meet spec
1/9	Mer. PR-L-A1-2	6.13/3.9	72.1	0.5/3.15	69.9	5.22/3.9	282.3	2.29/3.15	74.6					Would not meet spec.
1/15	Mer-PR-L-A1-3	4.75/3.9		6.75/3.15										Would not meet spec

Date	Shaft No.	Weights Slip-end	Weights Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
2/28	DANA-PI-1	35	29	39.63	0.026	0.03	0.012	0.012	0	0.001	0.001	0.001	0.001	Hinging by Mer. Procedure
2/8	DANA-PI-2	35	29	39.63	0.016	0.013	0.011	0.005	0	0	0	0	0.003	Hinging by Mer. Procedure
2/8	DANA-PI-3	35	29	39.63	0.019	0.019	0.008	0.017	0	0	0	0	0.001	Hinging by Mer. Procedure
9/7	DANA-PI-1 (po:	35	29	39.63	0.032	0.031	0.012	0.013	0	0.001	0.001	0.001	0.007	TIR lash @ slip end of tube

DYNAMIC DATA

Supplier: DANA-PI

Date	Shaft No.	Balance As Rec'd Slip-end oz.in	angle	Weld-end oz.in	angle	Balance Minimum Slip-end oz.in.	angle	Weld-end oz.in.	angle	Balance Final Slip-end oz.in.	angle	Weld-end oz.in.	angle	Remarks
2/28	DANA-PI-1	6.96	198.7	4.43	126					3.33	127.9	2.51	192.9	
2/8	DANA-PI-2	16.31	320.7	1.67	143.6					3.54	149.2	2.9	317.6	
2/8	DANA-PI-3	9.43	278	1.3	36					3.52	201	2.57	98.6	
9/9	DANA-PI-1 (po:	6.37	207.5	2.23	37.6									

STATIC DATA

Supplier: DANA PRR

Date	Shaft No.	Weights Slip-end	Weights Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
	DANA PRR-1	32	27	33.9										
2/20	DANA -PRR-2	32	27	33.9	0.018	0.022	0.023	0.024	0	0.001	0.001	0	0.002	Hinging by Meritor Procedure
	DANA-PRR-3	32	27	33.9										
9/9	DANA-PRR-2(p	32	27	33.9	0.018	0.022	0.023	0.024	0.001	0.001	0.001	0.001	0.01	Hinging TIR last @slip end tube
9/9	DANA-PRR-3(p	32	27	33.9	0.028	0.036	0	0.026	0	0	0	0	0.01	Hinging TIR last @slip end tube
		32	27	33.9										

DYNAMIC DATA

Supplier: DANA-PRR

Date	Shaft No.	lance As Rec'd Slip-end oz.in	angle	Weld-end oz.in	angle	Balance Minimum Slip-end oz.in.	angle	Weld-end oz.in.	angle	Balance Final Slip-end oz.in.	angle	Weld-end oz.in.	angle	Remarks
	DANA-PRR-1													

STATIC DATA
Supplier: DANA PR =L

Date	Shaft No.	Weights Slip-end	Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
2/19	DANA PR-L 1	42	31.5	61.5	0.031	0.032	0.014	0.013	0	0	0	0	0.002	Tested @ GMPG
	DANA PR-L 2	42	31.5	61.5	0.022	0.03	0.011	0.028	0.001	0.011	0.001	0.004	0.003	Would not balance to spec
	DANA PR-L 3	42	31.5	61.5	0.032	0.025	0.025	0.022	0.001	0.001	0	0	0.002	
9/9	DANA PR-L 1 F	42	31.5	61.5	0.031	0.032	0.014	0.013	0	0.001	0	0	0.018	Hinging TIR lash at slip end tub

DYNAMIC DATA
Supplier: DANA-PR-L

Date	Shaft No.	Balance As Rec'd Slip-end oz.in	angle	Weld-end oz.in	angle	Balance Minimum Slip-end oz.in.	angle	Weld-end oz.in.	angle	Balance Final Slip-end oz.in.	angle	Weld-end oz.in.	angle	Remarks
2/19	DANA PR-L-1	3.77	66	0.68	290					4.16	73.6	2.86	302	In spec (bal) as received
	DANA PR-L-2	11.5	92.9	3.99	67.6					10.14	216.2	1.06	177.5	Would not bal. Thrust clearan
	DANA PR-L-3	5.54	23.8	3.43	139.1					4.17	23.6	3.12	135.1	
9/9	DANA PR-L-1 F	2.53	252.9	1.08	148.1									Tested on road (LMTV)

STATIC DATA
Supplier: Mer-PI-A1

Date	Shaft No.	Weights Slip-end	Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
	Mer-PI-A1-1													
3/8	Mer-PI-A1-2	29	25.5	39.63	0.012	0.017	0.015	0.014	0	0	0.001	0	0.005	
	Mer-PI-A1-3													
9/24	Mer PI-A1-2	29	25.5	39.63	0.012	0.017	0.015	0.014	0	0	0.001	0.001	0.005	Post test audit

DYNAMIC DATA
Supplier: Mer-PI-A1

Date	Shaft No.	Balance As Rec'd Slip-end oz.in	angle	Weld-end oz.in	angle	Balance Minimum Slip-end oz.in.	angle	Weld-end oz.in.	angle	Balance Final Slip-end oz.in.	angle	Weld-end oz.in.	angle	Remarks
	Mer-PI-A1-1													
3/8	Mer-PI-A1-2	1.4	126.5	2.48	70.2					2.89	112	2.43	68.9	
	Mer-PI-A1-3													
9/24	Mer PI-A1-2									2.57	120	1.69	256.9	Post test audit

STATIC DATA
Supplier: Meritor RPL20-L

Date	Shaft No.	Weights Slip-end	Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
4/2	Mer-PR-L-RPL2	50	40.5	61.5	0.013	0.01	0.015	0.017	0.001	0.002	0.001	0.001	0.003	Tested @ GMPG (early)
4/2	Mer-PR-L-RPL2	50	40.5	61.5	0.015	0.018	0.019	0.016	0.002	0.002	0.001	0.001	0.004	Road tested -8/13
	Mer-PR-L-RPL2	50	40.5	61.5										
9/24	Mer PR-L-RPL2	50	40.5	61.5	0.021	0.026	0.027	0.018	0.002	0.002	0.001	0.001	0.008	Post test audit

DYNAMIC DATA
Supplier: Meritor RPL20-L

STATIC DATA

Supplier: Dana - PF (front)

Date	Shaft No.	Weights		Length	Runout				Thrust Clearance				Hinging	Remarks
		Slip-end	Weld-end		Neck	Slip-end of tube	Center of tube	Weld-end of tube	Slip-end yoke	shaft	Weld-end yoke	shaft		
	Dana PF-1	32	27	33.5							0			
2/19	Dana PF-2	32	27	33.5	0.034	0.042	0.025	0.012	0	0	5E-04	0	0.001	Hinging by Meritor Procedure
2/19	Dana PF-3	32	27	33.5	0.018	0.019	0.022	0.022	0	0	0	0	0.001	Hinging by Meritor Procedure
2/27	Dana PF-4	32	27	33.5	0.011	0.021	0.025	0.017	0.001	0	0	0.001	0.001	Run @ MPG
2/27	Dana PF-5	32	27	33.5	0.048	0.045	0.025	0.005	0	0	0.001	0	0.001	Hinging by Meritor Procedure
2/28	Dana PF-6	32	27	33.5	0.043	0.047	0.037	0.022	0	0	0	0.001	0.001	Hinging by Meritor Procedure
9/7	DANA PF-6(pos	32	27	33.5	0.046	0.047	0.037	0.022	0	0	0.001	0.001	0.003	TIR lash @ slip end of tube

DYNAMIC DATA

Supplier: Dana - PF (Front)

Date	Shaft No.	Balance As Rec'd				Balance Minimum				Balance Final				Remarks
		Slip-end		Weld-end		Slip-end		Weld-end		Slip-end		Weld-end		
		oz.in	angle	oz.in	angle	oz.in	angle	oz.in	angle	oz.in.	angle	oz.in	angle	
	Dana PF-1													
2/19	Dana PF-2	6.29	40.9	2.09	200.9					3.15	50.1	2.41	209.5	
2/19	Dana PF-3	2.62	161	3.58	222.3					2.91	34.9	2.57	40.3	
2/27	Dana PF-4	3.67	36.8	2.05	102.7					3.13	37.9	2.63	109.6	Run at MPG (LMTV)
2/27	Dana PF-5	2.82	215.6	1.72	30.9					3.18	217.1	2.52	13.6	
2/28	Dana PF-6	3.97	251.7	0.69	351.9					3.08	251.2	2.49	324.3	
9/7	DANA PF-6 (po	3.7	239.5	2.82	316.4									Run on road (MTV)

STATIC DATA

Supplier: Meritor PF-RPL20

Date	Shaft No.	Weights		Length	Runout				Thrust Clearance				Hinging	Remarks
		Slip-end	Weld-end		Neck	Slip-end of tube	Center of tube	Weld-end of tube	Slip-end yoke	shaft	Weld-end yoke	shaft		
	Mer-PF-RPL20-	40	36	33.5										
3/19	Mer-PF-RPL20-	40	36	33.5	0.013	0.008	0.009	0.011	0	0	0.002	0.001	0.007	Run @MPG(early) LMTV
3/19	Mer-PF-RPL20-	40	36	33.5	0.016	0.016	0.007	0.006	0.002	0	0.002	0.001	0.005	
4/26	Mer-PF-RPL20-	40	36	33.5	0.019	0.013	0.013	0.014	0.001	0	0.001	0	0.007	Run @ MPG-MTV
4/26	Mer-PF-RPL20-	40	36	33.5	0.01	0.007	0.01	0.015	0.001	0	0.001	0.001	0.007	Run @MPG -LMTV (last)
4/26	Mer-PF-RPL20-	40	36	33.5	0.013	0.014	0.01	0.019	0.001	0	0.001	0.002	0.007	
	Mer-PF-RPL20-	40	36	33.5	0.01	0.01	0.012	0.017	0.007	0.003	0.003	0.01	0.007	Post test audit

DYNAMIC DATA

Supplier: Meritor PF-RPL20

Date	Shaft No.	Balance As Rec'd				Balance Minimum				Balance Final				Remarks
		Slip-end		Weld-end		Slip-end		Weld-end		Slip-end		Weld-end		
		oz.in	angle	oz.in	angle	oz.in.	angle	oz.in.	angle	oz.in.	angle	oz.in.	angle	
	Mer-PF-RPL20-1													
3/19	Mer-PF-RPL20-	3.32	43.4	1.92	52.2					3.85	43.7	3.68	50.7	Run@MPG-LMTV(Early)
3/19	Mer-PF-RPL20-	5.45	2.8	2.6	291.3					3.92	351.1	3.43	287.1	
4/26	Mer-PF-RPL20-	1.33	342.9	1.85	29.3					3.84	67.7	3.46	61	Run @MPG-MTV
4/26	Mer-PF-RPL20-	1.31	265.6	2.87	181.8					3.8	271.5	3.59	174.2	Run @MPG-LMTV (last)
4/26	Mer-PF-RPL20-	0.4	287.2	0.78	321.9					3.72	225.3	3.42	18.2	
9/24	Mer PF-RPL20-	3.36		10.4										Post test audit

STATIC DATA
Supplier: GKN-PF

Date	Shaft No.	Weights Slip-end	Weld-end	Length	Runout Neck	Thrust Clearance Slip-end of tube	Center of tube	Weld-end of tube	Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
	GKN-PF-1	31	34	33.5	NA									
3/20	GKN-PF-2	31	34	33.5	NA	0.006	0.015	0.01	NA	NA	NA	NA	NA	
3/20	GKN-PF-3	31	34	33.5	NA	0.006	0.022	0.006	NA	NA	NA	NA	NA	
4/1	GKN-PF-4	31	34	33.5	NA	0.01	0.017	0.022	NA	NA	NA	NA	NA	Tested at MPG - LMTV
4/1	GKN-PF-5	31	34	33.5	NA	0.004	0.016	0.005	NA	NA	NA	NA	NA	
4/1	GKN-PF-6	31	34	33.5	NA	0.01	0.016	0.005	NA	NA	NA	NA	NA	Tested @ MPG-MTV
7/26	GKN PF-4 (Auc	31	34	33.5	NA	0.022	0.017	0.014	NA	NA	NA	NA	0.008	Post test audit
8/26	GKN-PF-6 audi	31	34	33.5	NA	0.012	0.002	0.005	NA	NA	NA	NA	0.001	Post tes audit

DYNAMIC DATA
Supplier: GKN-PF

Date	Shaft No.	Balance As Rec'd Slip-end oz.in	angle	Weld-end oz.in	angle	Balance Minimum Slip-end oz.in.	angle	Weld-end oz.in.	angle	Balance Final Slip-end oz.in.	angle	Weld-end oz.in.	angle	Remarks
	GKN -PF-1													
3/20	GKN -PF-2	1.63	272.2	1.06	282.2					1.16	271.1	1.09	277.9	
3/20	GKN -PF-3	2.63	53.6	0.75	12.6					1.24	52.8	1.15	12.9	
4/1	GKN -PF-4	0.23	105.1	1.99	191.5					1.28	94.4	1.2	153.6	Tested at MPG
4/1	GKN -PF-5	2.57	78.1	1.12	143.7					1.2	80.2	1.1	143.6	
4/1	GKN -PF-6	2.82	265.2	2.07	263.8					1.23	263.2	1.21	258.8	
8/26	GKN-PF-6 audit									2.89		1.87		Post test audit
7/26	GKN -PF-4 Auc	0.49	68.4	2.15	359.6					0.46		2.1		Post test audit

STATIC DATA
Supplier: Meritor A1 PF

Date	Shaft No.	Weights Slip-end	Weld-end	Length	Runout Neck	Thrust Clearance Slip-end of tube	Center of tube	Weld-end of tube	Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
	Mer -PF-A1-1	29	25	33.5										
3/6	Mer -PF-A1-2	29	25	33.5	0.015	0.026	0.036	NR	0.001	0.001	0.002	0.001	0.01	Tested @MPG-LMTV
3/8	Mer -PF-A1-3	29	25	33.5	0.01	0.015	0.015	0.01	0.001	0	0.002	0	0.006	Tested @MPG-MTV
	Mer -PF-A1-4	29	25	33.5										
	Mer -PF-A1-5	29	25	33.5										
	Mer -PF-A1-6	29	25	33.5										
9/25	Mer-PF-A1-2	29	25	33.5	0.015	0.026	0.062	NA	0.003	0.001	0.003	0.001	0.008	Post test audit LMTV front
9/25	Mer PF-A1-03	29	25	33.5	0.003	0.006	0.007	0.004	0.002	0	0.001	0.001	0.006	Post test audit MTV front

DYNAMIC DATA
Meritor-PF-A1

Date	Shaft No.	Balance As Rec'd Slip-end oz.in	angle	Weld-end oz.in	angle	Balance Minimum Slip-end oz.in.	angle	Weld-end oz.in.	angle	Balance Final Slip-end oz.in.	angle	Weld-end oz.in.	angle	Remarks
	Mer-PF-A1-1													
3/6	Mer-PF-A1-2	7.13	280.3	2.3	131.8					2.81	24.1	2.38	188.2	LMTV test
3/8	Mer-PF-A1-3	6.25	66.2	3.41	245.6					2.84	55.2	2.4	244.3	MTV test
	Mer-PF-A1-4													
	Mer-PF-A1-6													
9/25	Mer PF-A1-2									16.75	253.1	2.98	41.1	Post test Audit LMTV front
9/25	Mer PF-A1-03									4.07	182.9	3.06	0	Post test Audit MTV front

DYNAMIC DATA
Supplier: Meritor-PI-RPL20

Date	Shaft No	Slip-end oz.in	angle	Weld-end oz.in	angle	Slip-end oz.in.	angle	Weld-end oz.in.	angle	Slip-end oz.in.	angle	Weld-end oz.in.	angle	
	Mer-PI-RPL20-1													
3/18	Mer-PI-RPL20-:	4.53	125.1	1.62	17.7					4.28	126.4	3.57	356.7	
4/26	Mer-PI-RPL20-:	3.01	200.1	2.44	207.7					3.96	234.8	3.7	195.1	Run @ MPG MTV
9/24	Mer-PI-RPL20-3									0.61		0.68		Post test audit

STATIC DATA
Supplier: GKN-PI

Date	Shaft No.	Weights Slip-end	Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
	GKN-PI-1													
4/1	GKN-PI-2	41	35	39.63	0.008	0.012	0.006	0.005	NA	NA	NA	NA	NA	DTL
4/1	GKN-PI-3	41	35	39.63	0.01	0.01	0.004	0.006	NA	NA	NA	NA	NA	MTV Truck
8/8	GKN PI-03	41	35	39.63	0.015	0.016	0.004	0.019	NA	NA	NA	NA	NA	Post test audit

DYNAMIC DATA
Supplier: GKN-PI

Date		Balance as received Slip-end oz.in	angle	Weld-end oz.in	angle	Slip-end oz.in.	angle	Weld-end oz.in.	angle	Balance Final Slip-end oz.in.	angle	Weld-end oz.in.	angle	Remarks
	GKN-PI-1													
4/1	GKN-PI-2	0.95	337.8	0.84	25.1					1.33	336.8	1.17	27.9	DTL
4/1	GKN-PI-3	1.62	57.9	0.35	328.7					1.35	56.4	1.21	334.5	MTV Truck
8/8	GKN-PI-3	2.79		0.54										Post test audit

STATIC DATA
Supplier: Mer-PRR-RPL20

Date	Shaft No.	Weights Slip-end	Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
	Mer-PRR-RPL-:	40	38	33.9										
3/20	Mer-Prr-RPL-2(	40	38	33.9	0.013	0.008	0.009	0.011	0	0	0.002	0.001	0.007	
4/26	Mer-PRR-RPL2	40	38	33.9	0.007	0.008	0.004	0.012	0	0.001	0.001	0.002	0.006	
4/26	MER-PRR-RPL	40	38	33.9	0.01	0.009	0.013	0.016	0.002	0.001	0.002	0.004	0.004	Run @ MPG-MTV
9/24	Mer-PRR-RPL2	40	38	33.9	0.01	0.009	0.013	0.016	0.002	0.003	0.005	0.006	0.005	Post test audit

DYNAMIC DATA
Supplier: Meritor

Date		BALANCE AS RECEIVED Slip-end oz.in	angle	Weld-end oz.in	angle	Slip-end oz.in.	angle	Weld-end oz.in.	angle	BALANCE FINAL Slip-end oz.in.	angle	Weld-end oz.in.	angle	REMARKS
3/20	MER-PRR-RPL	3.32	43.4	1.92	52.2					3.85	43.7	3.68	50.7	
4/26	MER-PRR-RPL	2.51	150.3	1.92	252.3					3.94	141.6	3.76	242.5	
4/26	MER-PRR-RPL	6.09	241.8	2.07	169.8					3.96	187	3.64	196.6	Run @ MPG MTV
9/24	Mer-PRR-RPL20-4									4.7		9.35		Post test audit

STATIC DATA
Supplier: GKN-PRR Kempf

DYNAMIC DATA

Supplier: GKN-PRR Kempf

Date	Shaft No.	ance as received Slip-end oz.in	angle	Weld-end oz.in	angle	Slip-end oz.in.	angle	Weld-end oz.in.	angle	Balance Final Slip-end oz.in.	angle	Weld-end oz.in.	angle	
	GKN-PRR-1													
5/20	GKN-PRR-2	3.36	324.5	1.1	143.3					1.4	318.4	1.4	137.4	
5/20	GKN-PRR-3	2.85	310.9	0.85	133.7					1.32	313.7	1.39	129.7	
	GKN-PRR-1(not receiv)													
6/18	GKN-PRR-2	NR		NR						1.4	282.5	1.4	73.3	Extended 6 inches from original
9/8	GKN-PRR-2 audit									1.03		2.17		Post test audit

STATIC DATA

Supplier: Meritor A1 LMTV Rear statdynA1

Date	Shaft No.	Weights Slip-end	Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
1/11	Mer-PR-L-A1-1	39	31.5	61.5	0.009	0.003	0.013	0.014	0.0075	0.002	0.001	0.001	0.005	
1/9	Mer-PR-L-A1-2	39	31.5	61.5	0.011	0.013	0.013	0.033	0.002	0.002	0.002	0.002	0.005	LMTV test at MPG
1/15	Mer-PR-L-A1-3	39	31.5	61.5	0.011	0.012	0.015	0.015	0.004	0.015	0.002	0.005	0.005	Would not balance-used shaft
4/28	Mer PR-L-A1-3	39	31.5	61.5	0.011	0.012		0.015					0.005	
9/24	Mer PR-L-A1-2	39	31.5	61.5	0.022	0.032	0.005	0.038	0	0	0	0	0.005	Post test. Water intrusion

DYNAMIC DATA

Supplier: Meritor PR-L-

Date	Shaft No.	Balance As Rec'd Slip-end oz.in/spec	angle	Weld-end oz.in/spec	angle	Balance Minimum Slip-end oz.in/spec	angle	Weld-end oz.in/spec	angle	Balance Final Slip-end oz.in/spec	angle	Weld-end oz.in/spec	angle	Remarks
1/11	Mer-PR-L-A1-1	5.31/3.9	200.3	6.67/3.15	299.7	5.49/3.9	78	4.82/3.15	127.1					Would not meet spec
1/9	Mer. PR-L-A1-2	6.13/3.9	72.1	0.5/3.15	69.9	5.22/3.9	282.3	2.29/3.15	74.6					Would not meet spec.
1/15	Mer-PR-L-A1-3	4.75/3.9		6.75/3.15										Would not meet spec
4/27	Mer PR-L-A1-2	1.96	139	0.72	250.6					3.76	136.5	2.93	252	LMTV Test @ MPG
4/28	Mer PR-L-A1-3	4.59	303	1.63	39.7					3.85	303	2.67	15.3	
9/24	Mer PR-L-A1-2									2.24		1.72		Post test audit

STATIC DATA

Supplier: GKN-PR-L

Date	Shaft No.	Weights Slip-end	Weld-end	Length	Runout Neck	Slip-end of tube	Center of tube	Weld-end of tube	Thrust Clearance Slip-end yoke	shaft	Weld-end yoke	shaft	Hinging	Remarks
2/4	GKN PR-L-2	36.5	30.5	61.5		0.019	0.026	0.013	NA	NA	NA	NA	NR	
5/20	GKN-PR-L-3	36.5	30.5	61.5		0.013	0.17	0.1	NA	NA	NA	NA	*.007	Tested at MPG
7/26	GKN-PR-L-3	36.5	30.5	61.5		0.013	0.015	0.02					0.012	Post test results-audit

DYNAMIC DATA

Supplier: GKN-PR-L

Date		Balance as received Slip-end oz.in	angle	Weld-end oz.in	angle	Slip-end oz.in.	angle	Weld-end oz.in.	angle	Balance Final Slip-end oz.in.	angle	Weld-end oz.in.	angle	
2/4	GKN-PR-L-2	0.3	268	0.17	292.2					1.07	289	0.84	300.4	
5/20	GKN-PR-L-3	2.25	188.2	2.14	167					0.87	188.2	0.88	167	Tested at MPG
7/26	GKN-PR-L-3	0.79	202.5	1.31	259.7					0.9		1.35		Post test results- audit

STATIC DATA

Appendix V

Balance Quality Grades for Various Groups of Representative Rigid Rotors

in Accordance with

ISO 1940/1 and ANSI S2.19

Table 5.2

Pages 48 and 53

Balance Quality Grade G 16

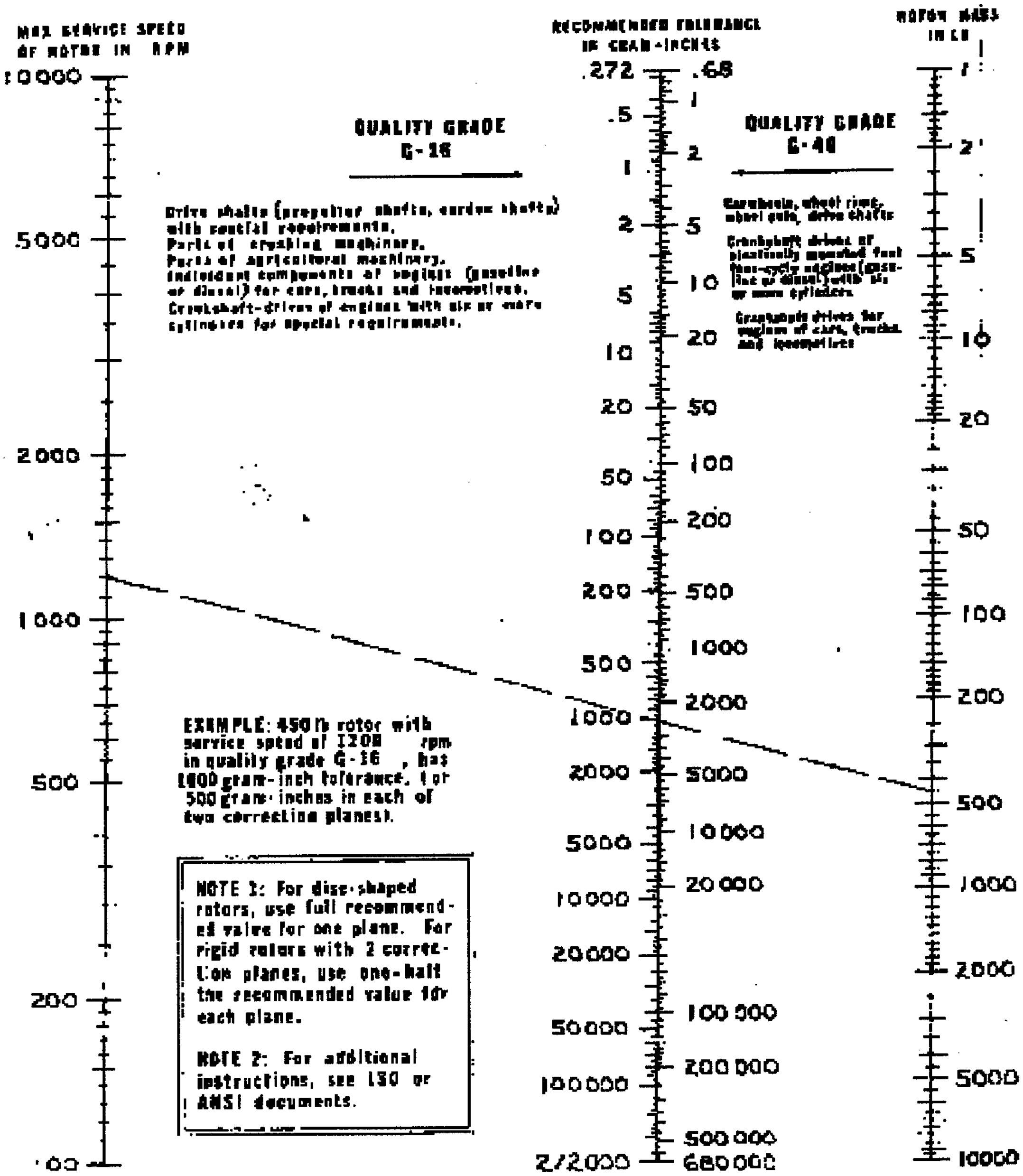
Driveshafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. Parts of agricultural machinery. Individual components of engines (gasoline or diesel) for cars, trucks, and locomotives. Crankshaft drives of engines with six or more cylinders under special requirements.

Balance Quality Grade G 40

Car wheels, wheel rims, wheel sets, driveshafts. Crankshaft-drives of elastically mounted, fast four-cycle engines (gasoline or diesel) with six and more cylinders. Crankshaft drives for engines of cars, trucks, and locomotives.

Fig 5-9 BALANCE TOLERANCE NOMOGRAM FOR G 16 & G 40

Based on ISO 1940, and ANSI S2.19



1 g in = .0353 oz-in
1 oz-in = 28.35 g-in

Appendix VI

Loads and ballast for MTV high torque test

The following are calculations based on current ATC and YPG procedures for cooling tests of MTV Tractors. Reference report # ATC-8266 (Volume one)

	Curb Wt.	Test wt. (static)
Total	19,740	44,700
Front	10,720	10,980
Bogey	9,000	33,720

Wt transferred: (23,800)(3)/14.5 = 5308 lb.

Dynamic rear bogey load: 33720+5308 = 39028 lb.

Rear/front split = 39028/44700 = 87.3% rear

Rear driveshaft torque:
(23,800)(.873)(21.3/12)(1/.98)(1/.98)(1/ 7.8)= 4923 lb.-ft.

To get this torque in Cargo truck:

Need Drawbar pull of:

4923(.98*2)(7.8)/ (21.3/ 12) = 20777 lbs. **with front shaft omitted**

Bogey load must be 34,629 lb at 0.6 TE/WT dynamic

Transferred load would be : 20777*(3/ 14.5)= 4298 lbs.

Static bogey load: 30,330 lbs