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THIS PROGRESS REPORT DEFINES ROOT CAUSES OF THE DRIVELINE AND POWERPLANT COMPONENT FAILURES OF LMTV VEHICLES. THESE FAILURES HAVE RESULTED IN A SAFETY OF USE MESSAGE Restricting operating speeds to 30 MPH maximum. POSSIBLE CORRECTIVE ACTIONS ARE DISCUSSED.

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FMTV, LMTV, DRIVESHAFT, CARDAN, CV, CONSTANT VELOCITY, FLYWHEEL HOUSING, BELL HOUSING, CRITICAL SPEED, DRIVELINE, DRIVELINE DYNAMICS, DRIVELINE MODEL, POWERPACK, POWERPACK DYNAMICS, TRANSFER CASE TRANSUDER, DRIVESHAFT BALANCE, DADS MODEL

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Prescribed by ANSI Std. 239.18
PROGRESS REPORT
OF
LMTV FLYWHEEL HOUSING
AND
DRIVELINE
FAILURE INVESTIGATION

Contract No. DAAE07-98-C-M012

DISTRIBUTION STATEMENT A
Approved for Public Release
Distribution Unlimited

Hayes Hobolth
17 August 1998
INTRODUCTION: Low mileage failures of flywheel housings and related driveline components became unacceptable by mid 1997. A Safety of Use Message was issued restricting speeds of operation for both LMTV and MTV, which accelerated the search for solutions. Caterpillar issued statements indicating that unbalance in the driveshafts was the primary issue and that the strain levels in the flywheel housings would be acceptable if unbalance was kept low. A contract was awarded to Michigan Scientific Corporation in April of 1998, part of which was to identify the cause(s) and recommend corrective actions for the deficiencies noted. Mr. Gary Schultz was named as Contracting Officer’s Technical Representative, and later, Mr. Donald Dismang as his Alternate.
PHASE I: A Start of Work meeting was held at Michigan Scientific Corporation's Milford, Michigan facility on April 21, 1998, and was attended by Cpt. Brent Thomas and Mr. James Lim. A significant investigation and information gathering had already been done by Mr. Hayes Hoboth as a consultant to the PM-FMTV as an employee of SAIC. A summary statement of the cause of the problems was dated April 14, 1998 and is based on analysis of data presented by Caterpillar, US Army ATC, EG &G Automotive Research (contracted by Tactical Vehicle Systems, the prime contractor) and numerous field trips, and examination of failed parts. In short, the cause of the problems was attributed to a resonant configuration involving the two main masses of the engine and the transmission/transfer case assembly separated by an elastic flywheel housing which causes the system to have a resonant response at 42-48 Hz. This frequency is also that of the first order of the driveshaft speed at about 47 mph, where the majority of the on-road time is spent. When the vehicle is driven at its maximum speed of 58-60 mph, the first order of the engine speed also excites the resonant response, by virtue of the overdrive gear ratio. Depending on the particular vehicle, either source may be dominant at either speed. It was agreed at the Sealy Texas PM review meeting that the resonant system must be tuned out of the driving range, if possible.

Since the primary failures were cracks in the flywheel housing, and it was believed to be the frequency setting spring in the system, MSC, with TACOM approval, ordered experimental castings of higher strength nodular iron and with increased section as a diagnostic tool. The material had a specified 80,000 psi ultimate strength compared to the Caterpillar specification of 35, 200 psi.

The preliminary results of tests on the first housing produced, were presented to TACOM at Aberdeen Proving Ground on July 15, and again at Caterpillar on July 23, 98. Strains were well below design limits for that material, and the apparent frequency of the vertical bending mode was 65-68 Hz, which was consistent with the measured 360-400 percent static stiffness increase measured at the MSC laboratory. This investigation was primarily diagnostic in its intent, but now is considered a possible corrective action.

The driveline unbalance includes the internal rotating parts in the transfer case and a significant unbalance of the torque converter and engine cooling fan, both of which run at engine speed. Some early LMTVs had a fan design which was found to contribute as much as 12 ounce inches of unbalance*.

An important relationship is shown in Figure 1, which is a mode shape of the vertical bending mode of the resonant response at its peak excited by the an unbalance at the transfer case output. The dynamic first order vertical velocity was probed at 10 inch increments along the length of the powertrain and is plotted on a side view. It clearly shows that the elastic energy is stored in the deformation of the flywheel housing.

* Determined by EG &G and presented at Sealy, Tx
PHASE II- Identify corrective actions: Diagnostic tests often point to corrective actions as is demonstrated above. A flywheel housing having stiffness and strength properties comparable to the experimental MSC flywheel housing would correct the flywheel housing failure problem.

Tests were also performed in an attempt to develop a tuned-mass damper. The first attempt used a 94 lb. Mass and showed about a 30 percent reduction in flywheel housing strain. A less massive version was tested and failed to show any significant benefit. This approach was abandoned.

MSC attended several meetings where results of a Cradle stiffener, and an added center mount were discussed. While both of these approaches demonstrate some benefit, neither is currently recommended.

DRIVELINE INFLUENCES: In addition to the unbalance as manufactured, the driveshafts are susceptible to the following:

1. Runout due to assembly. The U-joint seats in the Transfer case and axle yokes allow off center installation, which causes the entire mass of the driveshaft to orbit the center of rotation.
2. Bending of the 3 \frac{1}{2} inch driveshaft tube.
3. Bending of the slender portion of the slip yoke.
4. Deformation and wear of the plastic coating of the slip-yoke spline. Data being accumulated by ATC indicates a steady increase of the force with miles.

All of these contribute to the cumulative total unbalance force vector which is often hundreds of pounds and clearly challenges the fatigue strength of the production flywheel housing. This underscores the necessity of tuning the primary resonance out of the driving speed range.

Future Progress Reports will deal more with driveshaft diagnoses and corrective actions.

Attachments:

b. Start of Work Meeting – handout
c. Preliminary Results of Ductile Iron Flywheel Housing Tests.
d. Figure 1. Vertical Bending Mode Shape.

Hayes M. Hobolth
April 14, 1998

MEMORANDUM FOR RECORD

Subject: Summary statement of root cause of Flywheel housing and driveshaft failures LMTV

To: Cpt. Brent Thomas, Mr. James Lim, Gary Schultz

Gentlemen:

To summarize the status of the knowledge I have been able to gain on the above open issue, please consider the following:

1. The root cause of the problems is cumulative unbalance of components which rotate at between 40 and 60 times per second. The relative contributions of the components and the sensitivities of the system to these are to be quantified by Michigan Scientific Corporation (MSC). The components include but are not limited to the driveshafts, transmission and transfer case, engine and fan. As much as is feasible, they will be quantified separately and in combination in terms of response and measured strains.

2. The principle mitigating factor is the distribution of mass and stiffness of the engine, flywheel housing and the transmission/transfer case assembly which forms a spring-mass system having one or more natural frequencies in this range. The dynamic behavior of the system is being quantified by TVS (E G & G) and Caterpillar. Experimental changes to the system will be created and/or evaluated by E G & G, MSC and others in terms of the response of the system and likelihood of success.

Cooperation and sharing of knowledge between parallel investigations is understood as imperative for an early solution.

Hayes Hobolth
PRELIMINARY RESULTS
OF
DUCTILE IRON FLYWHEEL
HOUSING TESTS

Hayes Hobolth
July 15, 1998
Figure 1

EXPERIMENTAL STIFFENED FLYWHEEL HOUSING
Vertical Displacement vs. Vertical Load at the Flywheel Housing - Original and Revised Housings

- Revised Housing
- Original Housing

Static Stiffness Increase 362%
Axial Deflection at Bottom of Hsg For Original and Revised Housings vs. Vertical Load

- Revised Housing
- Original Housing

Apparent Stiffness Increase 400%

Vertical Load (lbs)

Axial Deflection (in)
Impact Test of Natural Frequency 65-68 Hz  (Original – 47.6 Hz)
Preliminary Results of First Mode Vertical Bending

- Impact test agrees with Caterpillar projection for ductile iron and increased section.

- Speed sweeps do not show a clearly defined resonance.

- Probing for the characteristic first mode vertical bending suggest that the new bending mode may be critically damped
## Maximum Recorded Flywheel Housing Strains at Gage Locations

### Old Housing - Maximum Strain (Right Side Lower Gages)

<table>
<thead>
<tr>
<th>Cases</th>
<th>Maximum Peak to Peak Strains (uE)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Neutral</td>
<td>970</td>
</tr>
<tr>
<td>6th gear - no imbalance</td>
<td>920</td>
</tr>
<tr>
<td>7th gear - no imbalance</td>
<td>875</td>
</tr>
<tr>
<td>6th gear - 2.7 in-oz imbalance</td>
<td>1396</td>
</tr>
<tr>
<td>7th gear - 2.7 in-oz imbalance</td>
<td>1057</td>
</tr>
<tr>
<td>6th gear - 5.9 in-oz imbalance</td>
<td>1293</td>
</tr>
<tr>
<td>7th gear - 5.9 in-oz imbalance</td>
<td>1037</td>
</tr>
<tr>
<td>7th gear - 10.8 in-oz imbalance</td>
<td>1645</td>
</tr>
<tr>
<td>7th gear - 25.5 in-oz imbalance</td>
<td>2300</td>
</tr>
</tbody>
</table>

### New Housing - Maximum Strain (Left Side Upper Gages)

<table>
<thead>
<tr>
<th>Cases</th>
<th>Maximum Peak to Peak Strains (uE)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Neutral</td>
<td>65</td>
</tr>
<tr>
<td>6th gear - no imbalance</td>
<td>92</td>
</tr>
<tr>
<td>7th gear - no imbalance</td>
<td>133</td>
</tr>
<tr>
<td>6th gear - 2.7 in-oz imbalance</td>
<td>93</td>
</tr>
<tr>
<td>7th gear - 2.7 in-oz imbalance</td>
<td>122</td>
</tr>
<tr>
<td>6th gear - 5.9 in-oz imbalance</td>
<td>87</td>
</tr>
<tr>
<td>7th gear - 5.9 in-oz imbalance</td>
<td>121</td>
</tr>
<tr>
<td>6th gear - 8.5 in-oz imbalance</td>
<td>125</td>
</tr>
<tr>
<td>7th gear - 8.5 in-oz imbalance</td>
<td>137</td>
</tr>
<tr>
<td>6th gear - 10.8 in-oz imbalance</td>
<td>119</td>
</tr>
<tr>
<td>7th gear - 10.8 in-oz imbalance</td>
<td>169</td>
</tr>
</tbody>
</table>
Upper Left Strain vs. Time

10.5 oz in unbalance at rear yoke

Rear Shaft Speed vs. Time
Rear Prop Shaft Rotational velocity vs. Time

Housing Right Strain vs. Time

10.8 oz in unbalance at rear yoke
To: Mark E. Padesky, Thomas D. Keating
Subject: Lab Paper MS00002018 - HOUSING-FLYWHEEL
Retain Until: 07/19/98 Retention Category: G90

This test bar meets ASTM 80-80-03 ductile iron. As to which spec of Caterpillar it meets, it fits better with our 1E598 which is a ASTM 80-55-08 modified. The tensile strength of the test bar is 96.4 ksi. Mossville met lab does not have extensometer big enough to run on a .506 tensile bar so we did not obtain the yield or elongation. Looking at the microstructure, it is almost all pearlite (85%) similar to a typical 1E596 microstructure. 1E590 is a higher tensile strength grade than 1E356 which is a more ferritic grade with a tensile of 60.2 ksi. Chemistry meets either spec. If you other question please feel free to get a hold of me.

Tom Majewski
86269
START OF WORK MEETING

FMTV FLYWHEEL

INVESTIGATION

April 21, 1998
**Hypothesis:**

1. The vehicle is OK if built to the design tolerances.

   **Test:**
   - Locate out of spec parts
   - Replace or repair
   - Stack worst case probability
   - Retest
   - Determine limit unbalance

2. Design included a resonant configuration that will not tolerate the probable shaker magnitudes without failure.

   **Test:**
   - Establish mode
   - Develop means of improving the stiffness/mass ratio
   - Test for strain at Flywheel housing and accelerations
   - Includes experimental Flywheel housing bracing and/or dynamic absorber
   - Use limit stack unbalance and higher
   - Determine margin for a robust design
   - Check for “New Problems”
This rather paradoxical statement is not quite so important as it sounds. In the first place, the bad effect of damping is not great and can be easily offset by making the springs somewhat weaker, i.e., by moving somewhat more to the right in Fig. 52. On the other hand, though it is not our intention to run at the resonance point \( \omega/\omega_n = 1 \), this unfortunately may sometimes occur, and then the presence of damping is highly desirable. Thus in spite of the dictum of Fig. 52, some damping in the springs generally is of advantage.

Fig. 52.—Showing that damping in the spring support is advantageous for \( \omega < \omega_n \sqrt{2} \), but is detrimental for \( \omega > \omega_n \sqrt{2} \).

20. Application to Single-phase Electrical Machinery.—Practical cases of isolation by means of springs occur in many machines. The main field of application, however, lies in apparatus which is inherently unbalanced or inherently has a non-uniform torque. Among the latter, single-phase electric generators or motors and internal-combustion engines are the most important.

First, single-phase machines are to be discussed. As is well known, the torque in any electric machine is caused by the pull of the magnetic field on current-carrying conductors. The magnetic field itself is caused by a current flowing through the field coils. If the machine is operated by single-phase alternating
### Exciters

<table>
<thead>
<tr>
<th>Sources</th>
<th>Frequencies</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel out of round</td>
<td>1/rev, 2/rev</td>
</tr>
<tr>
<td>Tire nonuniformity</td>
<td>1/tread lug...</td>
</tr>
<tr>
<td>Engine</td>
<td>1,2,3,6</td>
</tr>
<tr>
<td>Fan</td>
<td></td>
</tr>
<tr>
<td>Accessories</td>
<td></td>
</tr>
<tr>
<td><strong>Transmission</strong></td>
<td></td>
</tr>
<tr>
<td>Converter</td>
<td>1/rev input</td>
</tr>
<tr>
<td>Clutches</td>
<td>Individual speeds</td>
</tr>
<tr>
<td>T-case –gears, output yokes</td>
<td>Driveshaft speed</td>
</tr>
<tr>
<td>Driveshafts</td>
<td>1/rev</td>
</tr>
<tr>
<td>U-Joints</td>
<td>2/rev</td>
</tr>
<tr>
<td>Road Inputs</td>
<td>Varied &amp; random</td>
</tr>
</tbody>
</table>
• A responding system will "phase-lock" with whatever is exciting it.

• The responding body may vibrate at the principal frequency of the shaker and also at harmonics of that frequency.

  Example: Cat data showing 2\textsuperscript{nd} and 3\textsuperscript{rd} orders.

• The responding body will not respond at sub-harmonic frequencies.

  Example: A U-joint will not cause first order response.

• We see clear evidence of both first order driveline and first order engine frequencies in somewhat similar magnitudes.
Strain Location 1
Paved

FMTV Bell Housing
Strain Location 1

[AVE]  rms = 383.8

Psd, m/s^2/Hz

2.0 x 10^4
4.0 x 10^4
6.0 x 10^4
8.0 x 10^4
1.0 x 10^5
1.2 x 10^5

Frequency, Hz

0  20  40  60  80  100

RUN 309: Paved, max mph, engine fan on, loaded
FMTV Bell Crank Housing
6 Averages; Delta f = 0.20 Hz; Block size = 2048; Hanning Window Enabled
Strain Location 1

[AVE]  \( \text{rms} = 310.44 \)

---

RUN 303: Paved, max mph, engine fan off, loaded
FMTV Bell Crank Housing
6 Averages; Delta \( f = 0.20 \text{ Hz} \); Block size = 2048; Hanning Window Enabled
Max RPM Static, NO Engine Fan

Run 337, CPT Thomas
37 MPH, in Mode, NO eng Fan

Run 336, mode, no eng fan, u-joints aliiigned, loaded
cpt thomas
Plan of Attack:

1. Determine the state of well being of the test vehicle.
   a. Strain gage the flywheel housing
   b. Install accelerometers
   c. Record strain levels over the speed range of the vehicle
   d. Compare with other investigations

2. Separate the exciters
   a. Run known unbalances at the transfer case yokes. In and out of phase
   b. Run known unbalances at front of the engine (fan)
   c. Run with torque converter removed
   d. Draw conclusions – rank order critical components and limits

3. Simultaneously fabricate a stiffened stronger retuned system.

4. Design and fabricate a dynamic absorber to cancel the exciting forces from 40-60 Hz.

5. Test and conclude.