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Improving Military Ground Vehicle Fuel Efficiency through the Identification of More Fuel Efficient Gear Oils (FEGOs) – Development of a Stationary Axle Efficiency Test Stand and Test Procedure

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ABSTRACT

For existing vehicle fleets there are few ways to reduce fuel consumption that do not involve expensive retrofitting. Replacing standard lubricants with those that achieve greater efficiency through superior formulation is one practical and inexpensive way to reduce fleet fuel consumption. In an effort to identify axle lubricants that reduce fuel consumption, the U.S. Army has developed a stationary axle efficiency test stand and test procedure using data from vehicle testing and simulation. Test method developmental work was initiated using hardware representative of light and medium tactical vehicles. Results indicate that the stationary test stand can differentiate and map efficiency changes between lubricants. The test stand has been used to test fuel efficient axle lubricants, which proved to be in good agreement with prior vehicle testing. Stationary testing has been shown to offer a higher degree of accuracy than full-scale vehicle testing at lower cost.

INTRODUCTION

The U.S. Army Tank Automotive Research Development and Engineering Center (TARDEC) is currently investigating the use and impact of fuel efficient gear oils (FEGO) in its ground vehicle fleet as a means to increase fleet fuel efficiency. FEGO products are formulated specifically to maximize efficiency through the use of low viscosity base oils, shear thinning polymers, and low friction additive chemistry. Studies have shown that optimized axle gear lubricants aimed at increasing mechanical efficiency and reducing viscous losses have the ability to reduce overall vehicle fuel consumption [1-3]. This is advantageous to the U.S. Army, as new or updated lubricants are relatively easy to implement into the existing vehicle fleet without requiring expensive or infeasible vehicle hardware retrofitting.

To determine the benefits of using FEGO, the U.S. Army conducted on-track vehicle testing in light, medium, and heavy tactical wheeled vehicles. Testing was completed following a modified SAE J1321 [4] procedure, and the vehicles were evaluated following a city and highway driving cycle using two fuel efficient gear oil candidates, an SAE 75W-90 and an SAE 75W-140. Both were evaluated against a baseline SAE 80W-90 qualified against SAE J2360 [5], and representative of axle gear oil currently fielded by the U.S. Army. This testing demonstrated the potential to reduce vehicle fuel consumption by as much as 2% in steady-state highway driving and over 4% for a more transient cycle, but results indicated that lubricant selection could vary depending on operating cycle and vehicle application [6].

Although effective in investigating gross vehicle fuel consumption changes, full-scale vehicle testing often lacks the required accuracy needed to discriminate between smaller changes in efficiency. This is due to inherent variability of full scale vehicle tests which adds additional uncertainty in the fuel consumption measurement data, potentially masking real fuel consumption changes. Full-scale testing also limits the ability to obtain a more in-depth understanding of why specific oils perform as they do, as only a top level fuel consumption result is attained. In addition, full scale vehicle tests are costly to conduct, especially in the case of specialized military equipment which can be difficult to obtain for testing and requires that testing be conducted on a closed course. Because of these limitations and the need to support additional future testing for continued product qualification efforts, a stationary axle efficiency test was desired.

Several research papers were identified that have attempted to study and develop stationary axle testing methods [7-9], but evaluated hardware generally focused on light duty passenger vehicle applications. As the ability to study medium and heavy duty hardware was desired to represent the full breadth of the military vehicle fleet, the requirement for a new test stand was created. The following lists the goals outlined for the new stationary test stand:

- To the greatest extent possible, be modular in design and provide sufficient motoring and absorption capabilities to test light to heavy duty military axles.
- Provide improved testing accuracy and precision when assessing lubricants to improve the ability to discriminate between similarly performing oils.
- Provide a lower cost alternative for quantifying efficiency impact from new lubricants compared to full-scale vehicle testing.
- Support the future development of a Federal Test Method (FTM) intended to be used for product qualification by the U.S. Army.

Based on these goals a new test stand was designed and constructed, and developmental work was initiated using light and medium tactical wheeled vehicle hardware in an effort to relate results to vehicle testing and develop a test method for use in future product qualification.

VEHICLE FUEL CONSUMPTION TESTING

To quantify the potential impact on overall vehicle fuel consumption from the use of fuel efficient gear oils, vehicle testing was conducted on three classes of military vehicles representative of light, medium, and heavy tactical wheeled vehicles [6, 10, 11]. Testing was conducted on up-armored M1151A1 High Mobility Multipurpose Wheeled Vehicle (HMMWV), the M1083A1 5-ton cargo variant of the Family of Medium Tactical Vehicles (FMTV), and the M1070 Heavy Equipment Transporter (HET). Photos of these vehicles are shown in Figure 1, Figure 2, and Figure 3, respectively. Table 1 provides a brief description of these vehicles.

Figure 1: M1151A1 HMMWV

Figure 2: M1083A1 FMTV

Figure 3: M1070 HET

Parameter	HMMWV	MTV	HET
Model	M1151A1	M1083A1	M1070
Engine	GEP 6.5L(T)	CAT C7	DDC 8V92TA
Transmission	GEP 4sp Auto	Allison MD3070PT	Allison CLT-754
Power	142 kW (190 hp) @ 3400 rpm	246 kW (330 hp) @ 2800 rpm	372 kW (500 hp) @ 2100 rpm
Torque	515 Nm (380 lbft) @ 1700 rpm	1154 Nm (851 lbft) @ 1600 rpm	1992 Nm (1470 lbft) @ 2100 rpm
Axle Type	Frame mounted, independent hypoid differential, wheel end portal reduction	Beam type, hypoid differential, planetary wheel end reduction	Beam type, hypoid differential, planetary wheel end reduction
Axle Ratio	5.91:1 overall (3.08:1 differential 1.92:1 wheel)	7.8:1 overall (3.9:1 differential 2:1 wheel)	7.36:1 overall
Curb Weight	4731 kg (10,430 lb)	9606 kg (21,178 lb)	17,649 kg (38,910 lb)
GVWR	6123kg (13,500 lb)	14,061 kg (31,000 lb)	39,009 kg (86,000 lb) – tractor 104,961 kg (231,400 lb) – +trailer
Approx. Tested Weight	13,000 lb	31,000 lb	44,900 lb
No. of Drive Axles	2	3	4
Approx. Axle Oil Sump Vol.	1.9L (2 qt)	11.5L (12 qt)	17L (18 qt)

Table 1: Vehicle Descriptions

Two candidate fuel-efficient gear oils, an SAE 75W-90 and an SAE 75W-140, were evaluated against a baseline SAE 80W-90 to determine potential fuel consumption improvement. The baseline oil was selected because it represented a typical axle gear oil currently utilized in the military fleet. With the exception of the MTV, evaluations were conducted following two different driving cycles. The first driving cycle, referred to as the “highway cycle,” consisted of steady state operation at constant speeds to represent highway or convoy operations. Specific speeds and distances for this cycle are shown in Table 2 for each of the vehicles. The second driving cycle, referred to as the "city cycle," consisted of a combination of stop-and-go driving and limited duration medium and high speed operation to simulate general mixed use. An example of this cycle for the light tactical vehicle is shown in Figure 4 (Note: for the MTV and HTV, the city cycle remained the same except for the maximum speed, which was adjusted to be consistent with the maximum speeds shown for the highway cycle). Overall testing of the vehicles was based on procedures outlined by SAE J1321 Fuel Consumption Test Procedure - Type II, but some deviations were made from the method that do not comply with its most recent 2012 revision (i.e., generally related to overall test route length and greater leniency regarding weather conditions). In general, SAE J1321 testing requires a minimum of two trucks, one test and one control, which are operated over a desired test cycle. Fuel consumption is measured gravimetrically and used to create a test to control, or T/C fuel consumption ratio. The relative changes in the T/C ratios between baseline and test segments are used to determine fuel consumption changes as a function of the changing variable (in this case, axle gear oil).

Conditions		**Vehicle Speed**	**Distance**
LTV	1	40.2 km/h (25 mph)	36.2 km (22.5 miles)
	2	88.5 km/h (55 mph)	36.2 km (22.5 miles)
MTV	1	40.2 km/h (25 mph)	36.2 km (22.5 miles)
	2	80.5 km/h (50 mph)	36.2 km (22.5 miles)
HTV	1	40.2 km/h (25 mph)	36.2 km (22.5 miles)
	2	64.4 km/h (40 mph)	36.2 km (22.5 miles)

Table 2: Highway Cycle Profile

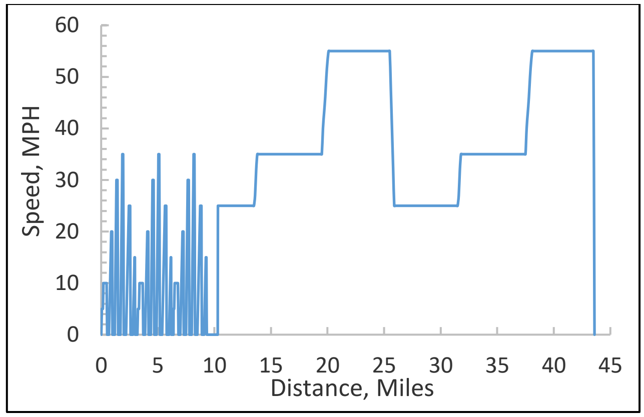

Figure 4: City Cycle Profile

Table 3 summarizes the fuel consumption improvement results for the 75W-90 and 75W-140 gear oils for the vehicle testing (Note: The city cycle is not shown for MTV using 75W-140 fluid. Other vehicle testing was conducted with the MTV that supported fuel efficiency gains from lubricant changes, but results were combined with other powertrain fluid changes, so only the highway cycle result for the MTV is shown).

Vehicle, Driving Cycle	**SAE 75W-90**	**SAE 75W-140**
HMMWV, city	+0.71% +/- 1.82%	+2.17% +/- 1.09%
HMMWV, highway	+0.57% +/- 0.35%	+1.41% +/- 0.83%
MTV, city	Axle Only Evaluation Not Completed	
MTV, highway	+1.82% +/- 1%*	-0.84% +/- 1%*
HET, city	+1.89% +/- 1.1%	-2.75% +/- 1.62%
HET, highway	+0.75% +/- 0.77%	-1.89% +/- 0.31%

*Confidence interval based on standard +/- 1% specified in pre-2012 SAE J1321 procedure. All other confidence intervals reported were calculated following post-2012 J1321 revision statistical analysis procedures.

Table 3: Fuel Consumption Improvement Over Baseline SAE 80W-90

As seen in Table 3, an indication of positive improvement (i.e., fuel consumption reduction) was noted for the SAE 75W-90 oil for the

HMMWV and HET vehicles on both the highway and city driving cycles. This suggests that the SAE 75W-90 is likely a good candidate to provide fleet wide fuel savings. However, for the SAE 75W-140, a different response was noted between the light duty vehicle and the medium and heavy tactical vehicles. In this case, the HMMWV results showed a positive improvement from the use of SAE 75W-140 on both driving cycles, while the MTV and HET results showed an almost equal detriment in fuel consumption for both. This result was unexpected because both the SAE 75W-90 and SAE 75W-140 had been advertised (i.e., by the formulator or OEM) to be fuel efficient gear oils. The response from the SAE 75W-140 was believed to be mainly the result of greater churning losses due to the higher viscosity. These results brought into question why the light and heavier class vehicles differentiated the higher viscosity fluid differently and indicated that the HMMWV had a larger degree of mechanical gear mesh losses than the medium and heavy tactical vehicles. This was to be investigated further with the stationary axle efficiency test stand.

TEST STAND – DESIGN AND CONSTRUCTION

The following sections outline the design and construction of the stationary axle efficiency test stand.

Determination of Stand Loading Requirements

The first step in creating the test stand was to identify all required loading conditions for the axles to be tested. For the MTV axle, J1939 CAN-BUS data was captured during on-track vehicle testing. Transmission shaft output speed, gear selection, and driver requested torque were recorded at all operating conditions for the city and highway driving cycles. Using this data along with detailed powertrain power/torque data and the MTV's advertised 30/70 front/rear torque split, axle pinion input conditions were calculated for each mode of the driving cycles. All conditions were calculated for the rear-most axle of MTV.

For the heavy and light tactical vehicles, direct data from vehicle testing was not available. Therefore, to establish conditions for the test stand design TARDEC conducted computer based vehicle simulations for a Palletized Load System (PLS) vehicle that defined required wheel output torque values over the same specified driving profiles (Note: The PLS platform simulated shares many powertrain component similarities with the HET including engine, transmission and axle supplier). These simulated output torque values and speed profiles were then used to back calculate the required input pinion condition at the axle. Similar to the MTV, conditions were identified specifically for the rear-most axle. For the light duty HMMWV, maximum possible powertrain loading was determined to be well below the nominal requirements of the medium and heavy vehicles, so only verification of required input pinon speed (based on vehicle speed, wheel diameter, and gear ratio) was needed.

Component Selection and Layout

Based on the test conditions identified, all stationary axle stand components could be selected. It was desired to configure the stand in a T-type arrangement. In this arrangement both wheel end outputs of the axle are coupled together and absorbed.

For the input motor a variable-frequency drive (VFD) controlled 186 kW (250 hp) AC motor was identified to be sufficient in providing the required loading conditions for each of the desired test axles. As part of the T-type test arrangement, two 7.259:1 speed-increasing gear boxes were specified to convert the low speed high torque output of the tested axle into a higher speed but lower torque output that could then be coupled to an absorbing unit. Based on the output speed and torque of the gear boxes, a second 186 kW (250 hp) AC motor (configured as a generator) was specified for absorption. Figure 5 shows the general layout of the stationary axle efficiency test stand.

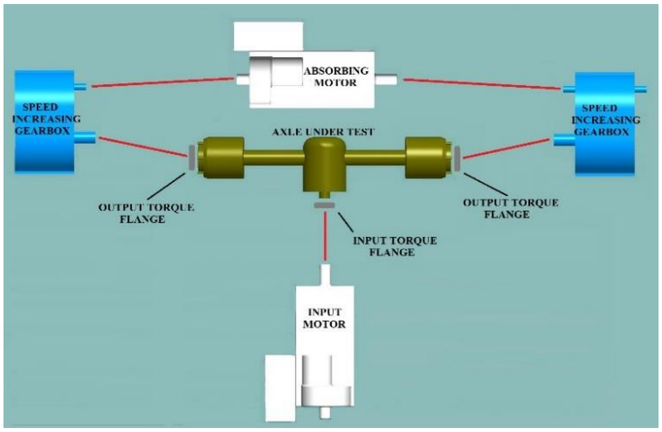

Figure 5: Layout of the Axle Efficiency Test Stand

For input and output speed measurement, incrementing encoders were integrated into the input motor and absorber units. For torque measurement, inline measurement directly at the axle's input and outputs (where applicable) was targeted to reduce any influence from the rest of the test stand components. To accomplish this, flange type digital torque meters were specified, and special couplings were machined for each axle for mounting the torque meters. Figure 6 shows a photo of the completed stationary axle test stand.

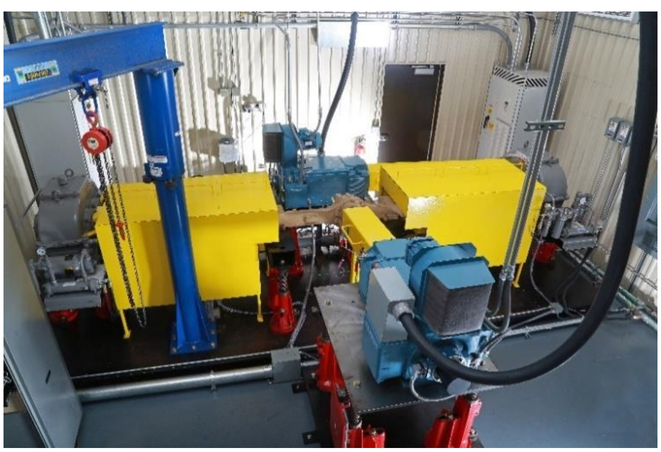

Figure 6: Completed Axle Efficiency Test Stand

Temperature Control

The temperature of the gear oil in an axle varies greatly depending on ambient temperature, duty cycle, and oil properties. The gear oil is heated by internal friction and viscous dissipation and cooled by external convective heat transfer. Therefore, in normal operation the oil is in dynamic equilibrium between these heating and cooling processes and naturally stabilizes to a temperature that depends on ambient temperature, axle efficiency, and the oil being used. One method of testing axle efficiency would be to allow the axle to obtain its own steady-state temperature during the testing process. This would require the ability to supply a fixed cooling rate to the axle oil during testing, while still allowing the oil temperature to seek its own steady-state. Although this approach more realistically reproduces actual operating conditions it is generally believed to reduce the repeatability of the results which is undesirable for identifying relatively small efficiency differences between oils. An alternative approach to letting the axle oil temperature naturally stabilize is to carefully control its temperature to a set value, providing system cooling or heating depending on demand. Both approaches have merit and both approaches were explored but it was decided to adopt the latter approach and control temperature precisely in order to maximize the sensitivity of the test stand and repeatability of the results. Therefore, a recirculation control loop was implemented that circulated lubricant during the test operation from the lower portion of the axle differential housing through the external control system and returning back to the axle's fill port. This set-up was similar to the control system used successfully by Anderson et al. during axle efficiency testing [7]. The external control system consisted of a small fixed displacement gear pump, a circulation heater, and a liquid-to-liquid heat exchanger to provide cooling. The gear pump was sized to provide nominally five gallons per minute of flow during operation. The axle lubricant temperature was measured at the drain port of the axle with a closed tip thermocouple protruding approximately one inch into the differential housing.

EFFICIENCY TEST DEVELOPMENT

After the completion of the test stand design and construction, it was necessary to understand if the results from the test stand would correlate with on-track test results.

Drive Cycle Replication – MTV

The first goal of the test development process was to replicate the driving cycle previously discussed. Using the J1939 CAN-BUS data, pinion speeds and load points were identified to develop the test cycle shown in Table 4.

Approximate Vehicle Velocity	Pinion Input Speed (rpm)	Pinion Input Torque	Pinion Input Power
88.5 km/h (55 mph)	3207	141 Nm (104 lbft)	47 kW (64 hp)
56 km/h (35 mph)	2033	91 Nm (67 lbft)	19 kW (26 hp)
48 km/h (30 mph)	1723	89 Nm (66 lbft)	16 kW (22 hp)
40 km/h (25 mph)	1469	73 Nm (54 lbft)	11 kW (15 hp)
32 km/h (20 mph)	1157	61 Nm (45 lbft)	7 kW (10 hp)
24 km/h (15 mph)	865	61 Nm (45 lbft)	6 kW (7 hp)
16 km/h (10 mph)	684	76 Nm (56 lbft)	5 kW (7 hp)
8 km/h (5 mph)	294	108 Nm (80 lbft)	3 kW (5 hp)

Table 4: MTV Replicated Highway Cycle - Axle Input Conditions

The horse power ranges from 5 to 64 hp. These relatively low power requirements roughly represent the road-load power at each speed. This is the power needed to overcome rolling resistance and aerodynamic drag to keep the vehicle moving at a steady speed over a relatively level road.

Since the MTV axle being tested was new, a large amount of initial operation was conducted primarily for test stand shakedown and axle break-in. All of this work was completed using the baseline 80W-90, and calculated efficiency results were tracked to observe axle stabilization. During this time an investigation into temperature effects on measured efficiency was conducted. Testing was completed comparing results between test runs targeting different temperatures and were used to develop a more simplified temperature profile. Based on the findings, it was decided to conduct efficiency testing at a single temperature set point of 79.4°C +/- 0.6°C (175°F +/- 1°F) considered typical and representative of average temperature for the rear-most MTV axle.

Once initial investigative work and break-in was completed, testing following the vehicle replicated driving cycle was conducted. Multiple evaluations were completed using each oil to establish trends and to observe the repeatability of results. Between repeated runs the axle and heater control loop were double flushed. Figure 7 shows the plotted results.

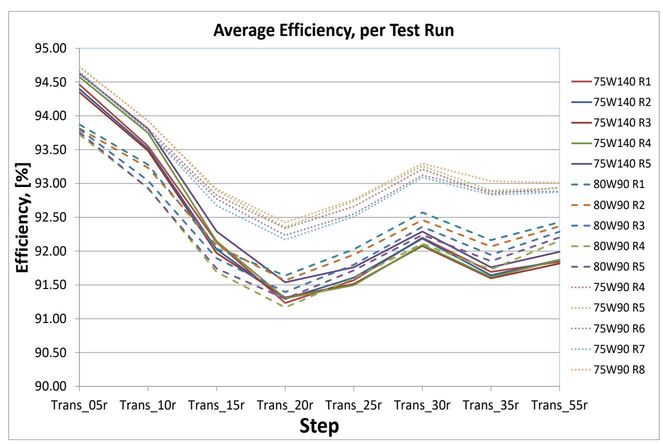

Figure 7: MTV Axle Efficiency Results

Efficiency was calculated using the following

$$Power_{In} = \frac{(Speed_{In} * Torque_{In})}{5252}$$

$$Power_{Out} = \frac{Speed_{Out} * (Torque_{Left} + Torque_{Right})}{5252}$$

$$Efficiency = \frac{Power_{Out}}{Power_{In}} * 100$$

where power [hp], speed [rpm], and torque [lbft].

The steps along the x-axis refer to a transient cycle (see Table 4) followed by the approximate vehicle speed in miles per hour. For example, the step labeled "Trans_15r" represents the transient cycle conducted at 15 mph. Compared to the baseline 80W-90, the 75W-90 oil resulted in greater efficiency at all operating conditions. The 75W-140 oil provided improved efficiency at the lower speeds (i.e., 5 to 15 mph) but lower or equivalent efficiency at higher speeds. Axle power losses are usually separated into two classes, speed dependent losses called spin or churning losses and load dependent losses also called mechanical losses. Jeon [12] investigated the churning losses of hypoid axle gears and pinions operating with a low and high viscosity lubricant and found that the churning losses of the high viscosity lubricant were higher than that of the low viscosity lubricant at low speeds, but roughly equivalent at high speeds. Based on these findings it would be expected that the 75W-140 should result in more losses and lower efficiency at low speeds, but this is not the case in Figure 7. This suggests that the higher churning losses are being off-set by a larger reduction in mechanical loss which is surprising given the relatively low power requirements tested. This was the first insight into how efficiency response might differ based on operating conditions.

From the initial results, much consideration was given to what direction the development path should take. If the primary goal was to create a laboratory based test that reproduced the exact results of the J1321 city and highway driving cycles, weighted averages of each of these operating points based on the duration of time that the vehicle operated in them could be calculated. However to be applicable to real-world field use, the operating cycle being replicated on the stationary axle stand must be representative of the full spectrum of real-world operation. Confidence in this was less clear, as estimating a "typical" military duty cycle is difficult. To ensure that the stationary test stand results could be applicable over a wider range of vehicle operating conditions, it was decided that additional investigation over a more broad range of input speed and loading conditions was required.

Efficiency Mapping

To investigate other input pinion conditions an efficiency mapping exercise was conducted to determine axle and lubricant response over a wider range of conditions. In order to ensure that the mapping and resulting efficiency test did not turn into a hardware durability test, the maximum pinion input torque was limited to 677 Nm (500 lbft). This was based on approximately 50% of the maximum torque the powertrain package could deliver to the rear-most axle under typical operating conditions, and was expected to be well within the capabilities of the axle so durability would not be affected.

Prior to mapping, additional break-in was conducted to run-in the axle up to the new maximum input load. Once the efficiency response stabilized, mapping was conducted for each of the three oils. Figure 8, Figure 9, and Figure 10 show the resulting efficiency maps for the 80W-90, 75W-90, and 75W-140, respectively. For each test the oil temperature was maintained at 79.4°C (175°F). (Note: The dotted line overlay at the bottom of the map represents the original operating points of the vehicle driving cycle.)

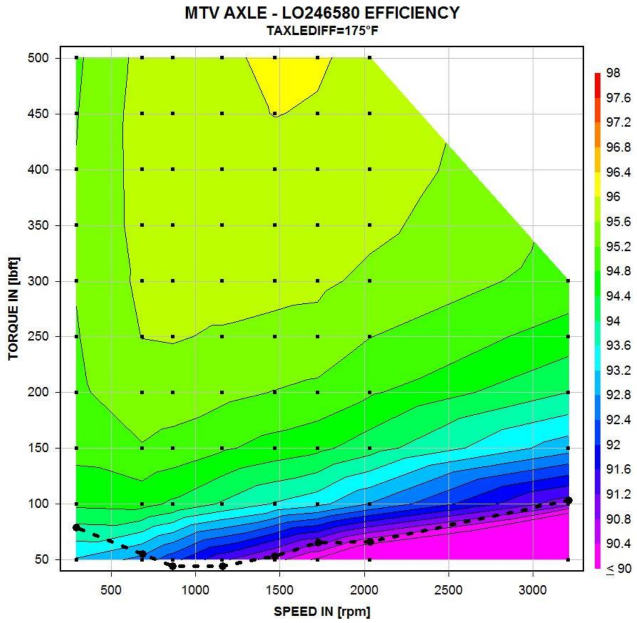

Figure 8: MTV Axle Efficiency Map – 80W-90

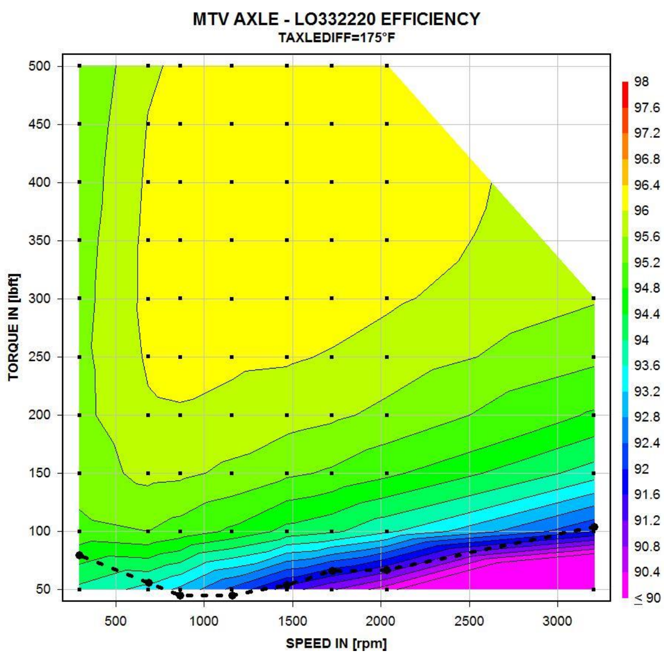

Figure 9: MTV Axle Efficiency Map – 75W-90

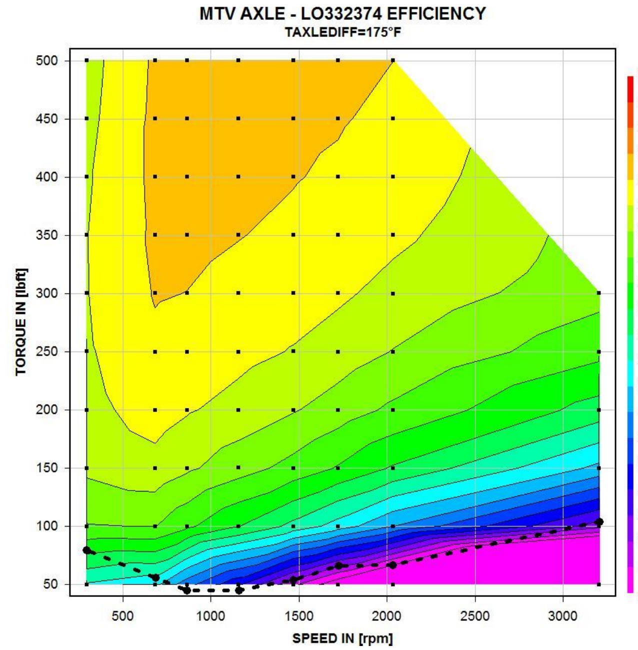

Figure 10: MTV Axle Efficiency Map – 75W-140

From the maps it can be observed that the three lubricants provide differing efficiency responses. Compared to the baseline 80W-90, the 75W-90 resulted in improved efficiency over the entire operating range measured. The 75W-140 provides additional efficiency benefits over the 75W-90 in regions of low speed and high torque. This response is attributed to its viscosity profile. At the high loads and low speeds, the gear mesh is expected to be in mixed or boundary type lubrication regime, and it is theorized that the higher viscosity of the 75W-140 is resulting in an increased film thickness within the gear mesh, reducing the severity of surface to surface contact and its resulting friction. However, as the load decreases and speed increases, churning losses in the differential and planetary wheel end reduction become the larger driver in efficiency and the higher viscosity becomes a detriment.

Proposed Test Cycle - Results for Medium Axle

Based on the findings from the efficiency maps, new test points were investigated for inclusion into the proposed efficiency test cycle for the MTV axle. To limit the length of the overall test cycle, the existing drive cycle replication points were also reviewed to determine their necessity. It was desired that the test cycle would be representative of the full spectrum of efficiency response that the MTV axle exhibits, so new test points were identified and selected from the original driving cycle and the remainder of the map to coincide with major efficiency contours present. The points identified are listed in Table 5 and shown graphically in Figure 11.

Test Step	**Approximate Vehicle Speed**	**Pinion Input Speed (rpm)**	**Pinion Input Torque**	**Pinion Input Power**
FTM_50_100	40 km/h (25 mph)	1469	610 Nm (450 lbft)	94 kW (126 hp)
FTM_30_150	56 km/h (35 mph)	2033	338 Nm (250 lbft)	72 kW (97 hp)
FTM_40_100	40 km/h (25 mph)	1469	440 Nm (325 lbft)	68 kW (91 hp)
FTM_50_75	72 km/h (45 mph)	2600	237 Nm (175 lbft)	65 kW (87 hp)
FTM_15_200	24 km/h (15 mph)	865	542 Nm (400 lbft)	49 kW (66 hp)
FTM_50_50	88 km/h (55 mph)	3207	141 Nm (104 lbft)	47 kW (64 hp)
FTM_50_35	56 km/h (35 mph)	2033	91 Nm (67 lbft)	19 kW (25 hp)
FTM_20_75	40 km/h (25 mph)	1469	73 Nm (54 lbft)	11 kW (15 hp)
FTM_20_50	24 km/h (15 mph)	865	61 Nm (45 lbft)	6 kW (8 hp)
FTM_20_35	8 km/h (5 mph)	294	108 Nm (80 lbft)	3 kW (4 hp)

Table 5: Proposed Test Cycle Points for MTV Axle

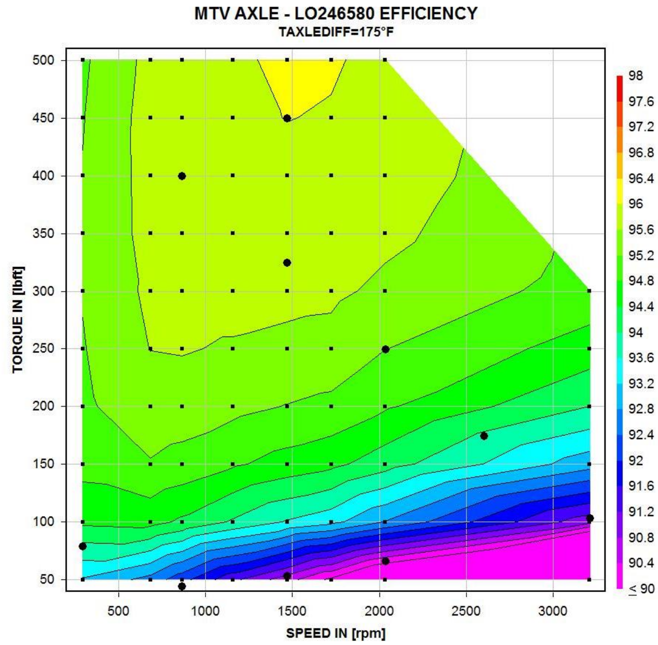

Figure 11: MTV 80W-90 Axle Efficiency Map with Proposed Final Test Points

With the new proposed test points, additional evaluations were conducted for each of the three oils. Figures 12 and 13 show the plotted results for the 75W-90 and 75W-140 versus the baseline 80W-90.

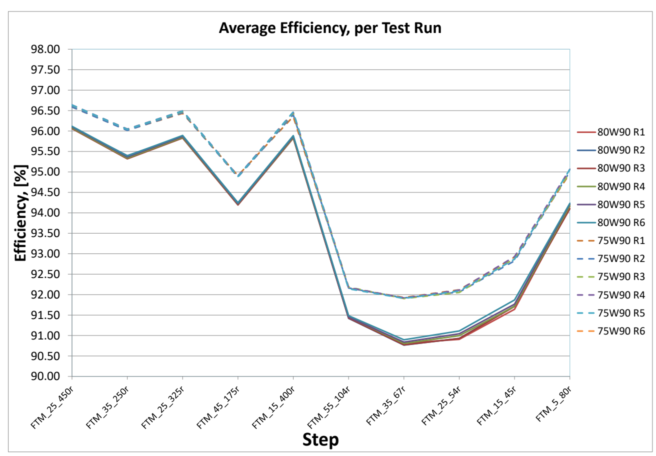

Figure 12: MTV Efficiency Results - Proposed Final Test Points 75W-90

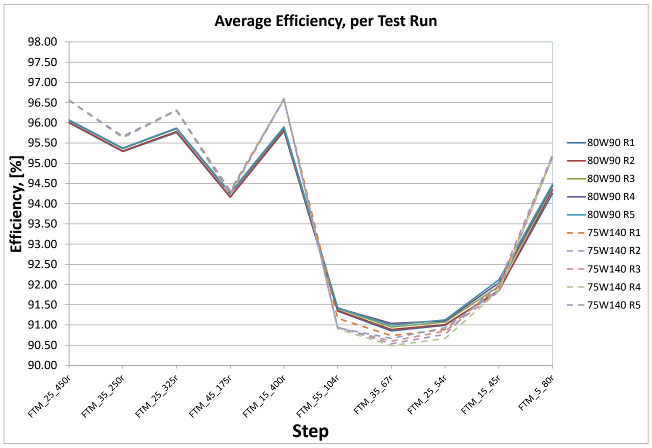

Figure 13: MTV Efficiency Results - Proposed Final Test Points 75W-140

As before, the 75W-90 continued to show improved efficiency compared to the 80W-90 at all conditions. However, the 75W-140 now showed efficiencies that demonstrated its full propensity to improve, decrease, or match the 80W-90 performance based on the specific operating condition. With this greater diversity in the test cycle, a more applicable weighting system could be developed to better predict resulting efficiency changes over a wide range of real world operation.

To establish the statistical significance in the changes between each oil with regards to stand repeatability, additional analysis was conducted on the resulting data. For each data set the variance was calculated and used in a statistical F-test model to determine if the variances between the two compared data sets were equal. Based on that result, an appropriate T-test model was conducted to determine if the mean results between the two compared oils were different, and determine the upper and lower 95% confidence interval bounds. Figures 14 and 15 show the plotted improvement of each of the oils with their appropriate confidence intervals for each step.

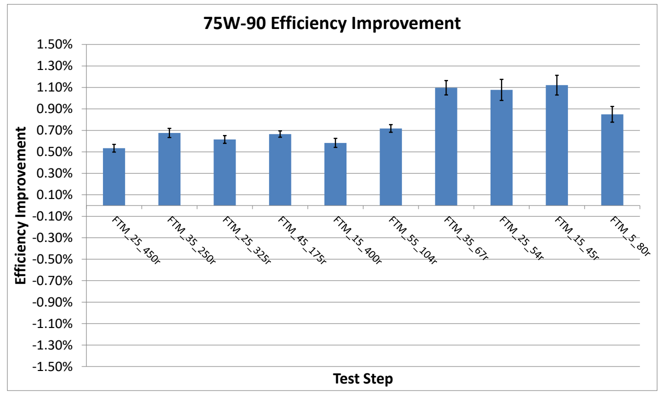

Figure 14: MTV Efficiency Improvement with 95% Confidence Bounds – 75W-90

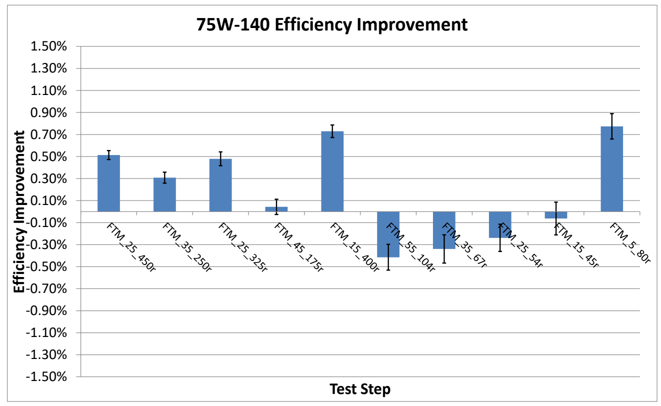

Figure 15: MTV Efficiency Improvement with 95% Confidence Bounds – 75W-140

The 75W-90 improved efficiency from +0.6% to just over +1.1%, with an average overall improvement of +0.79% over the baseline 80W-90 with the resulting confidence intervals being +/- 0.1% or less. For the 75W-140, improvement ranged from +0.3% to just over +0.7% for the more highly loaded or extreme low speed points, and detriments ranging from approximately -0.2% to -0.4% where low loads or high speeds where prevalent. Overall average improvement was +0.18%, with all but two of the operating points showing statistically significant differences in efficiency from the 80W-90.

Light Tactical Vehicle (LTV)

The next step of the development process moved to the light tactical wheeled HMMWV axle. Detailed loading information from the driving cycles was not available, so exact replication of the driving cycle was impossible. As a result, the investigation moved directly to a mapping exercise like that conducted for the MTV hardware. For the lighter HMMWV, an efficiency map test matrix was constructed to investigate axle response over a wide range of pinion loading conditions. Similar to the MTV, the maximum HMMWV input torque was limited to 339 Nm (250 lbft) to ensure differential durability was not affected. As the HMMWV axle was also new at the start of testing, a break-in test cycle was conducted to ensure that efficiency response from the differential was stabilized before testing.

Figures 16, 17, and 18 show the resulting efficiency maps for the HMMWV at a differential fluid temperature of 79.4°C (175°F). It was immediately observed that the overall efficiency level of the HMMWV differential was much lower than the MTV axle, however, the general efficiency trends between the oils persisted. The 75W-90 showed an improved efficiency response across the entire map compared to the 80W-90, while the 75W-140 showed even greater gains, especially at loads greater than 74 Nm (100 lbft). Interestingly, the 75W-140 did not appear to cause the same kind of detriment at low-load high speeds as seen in the MTV. This was attributed to the differences in differential oil capacity. The HMMWV's capacity is 1.9 L (2 qts), while the MTV's capacity is 13.2 L (3.5 gal), approximately seven times greater.

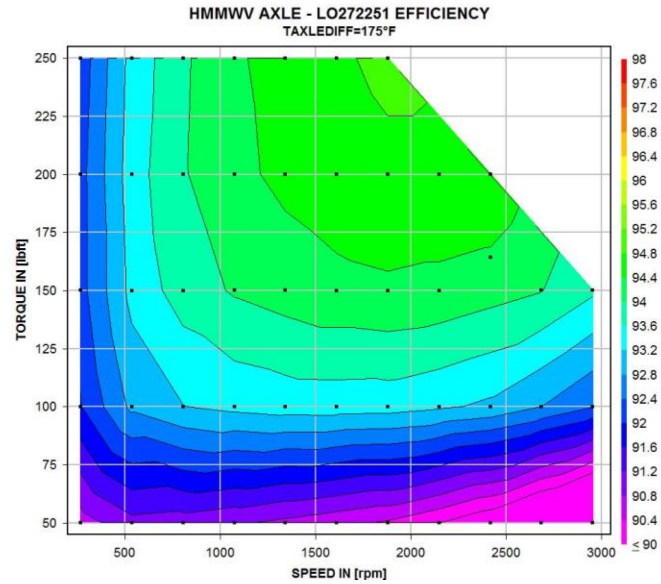

Figure 16: HMMWV Axle Efficiency Map – 80W-90

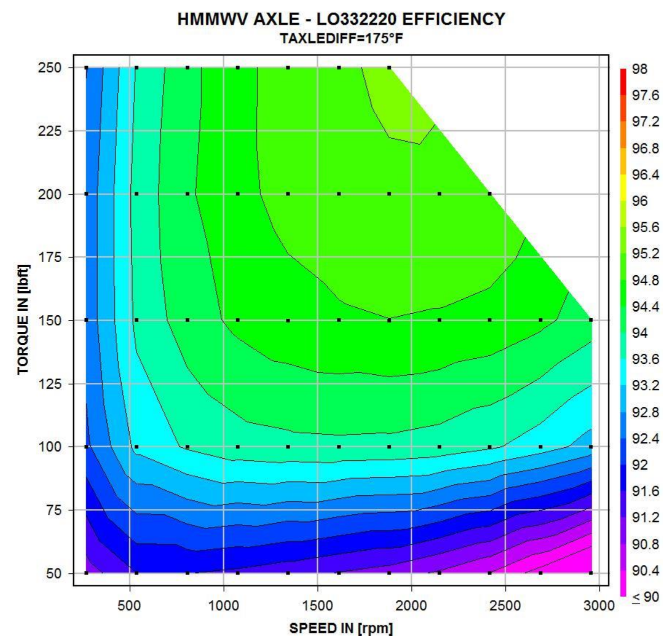

Figure 17: HMMWV Axle Efficiency Map – 75W-90

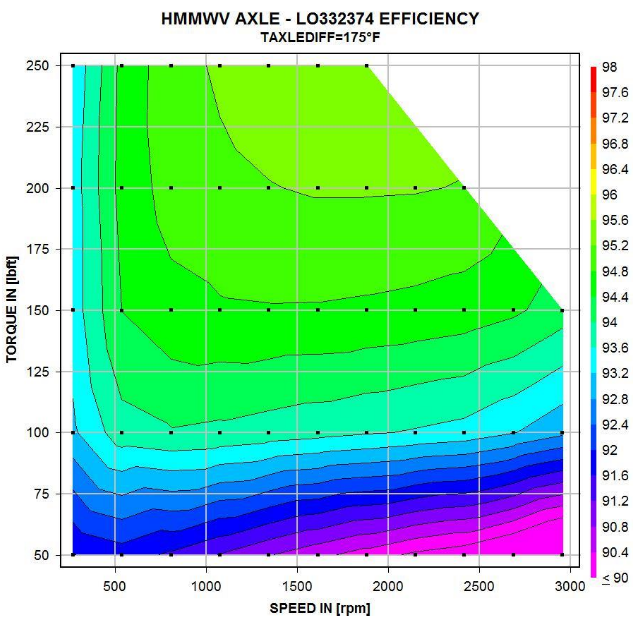

Figure 18: HMMWV Axle Efficiency Map – 75W-140

With the efficiency mapping data in hand, an investigation into test points for an efficiency test cycle was conducted. Since driving cycle data for the HMMWV did not exist, other sources of vehicle operating data were investigated. In previous work supported by the U.S. Army and Southwest Research Institute (SwRI), an investigation into HMMWV differential efficiency testing was conducted for super-finished hypoid gear sets over a Peacetime – Operational Mode Summary/Mission Profile (OMS/MP) [13] developed for the HMMWV. In this work, detailed axle pinion torque and speed were identified through vehicle simulation for peacetime operation. Since this was considered a good representation of potential HMMWV use, this same data set was revisited to determine applicable points for an efficiency test cycle.

The operating points were trimmed to the core usable area for the efficiency test, eliminating speeds lower than 5 mph and torque higher than 338 Nm (250 lbft). With the remaining data, the input torque conditions were binned based on discrete vehicle speed and torque ranges to determine the frequency of operation under particular conditions (shown in Table 6).

Pinon Input Torque Bins (lbft)	**Vehicle Speed Bins (mph)**						
	15	20	30	35	40	45	50
25	57	116	13	17	15	0	24
50	2	129	7	4	16	0	21
75	5	24	11	3	8	0	32
100	4	14	9	6	18	0	34
125	5	5	2	2	7	5	16
150	8	0	10	2	4	7	11
175	6	2	0	7	3	0	0
200	6	1	2	1	0	0	0
225	6	1	0	0	0	1	0
250	1	1	0	0	0	0	0

Table 6: HMMWV Peace Time Duty Cycle – Torque/Speed Frequency Bins

From this data, the speed and load points in Table 7 were proposed for the HMMWV efficiency test cycle.

Approximate Vehicle Velocity	Pinion Input Speed (rpm)	Pinion Input Torque	Pinion Input Power
80 km/h (50 mph)	2686	136 Nm (100 lbft)	38 kW (51 hp)
48 km/h (30 mph)	1611	203 Nm (150 lbft)	34 kW (46 hp)
64 km/h (40 mph)	2149	135 Nm (100 lbft)	30 kW (40 hp)
80 km/h (50 mph)	2686	102 Nm (75 lbft)	29 kW (39 hp)
24 km/h (15 mph)	806	271 Nm (200 lbft)	23 kW (31 hp)
80 km/h (50 mph)	2686	68 Nm (50 lbft)	19 kW (25 hp)
80 km/h (50 mph)	2686	47 Nm (35 lbft)	13 kW (17 hp)
32 km/h (20 mph)	1074	102 Nm (75 lbft)	11 kW (15 hp)
32 km/h (20 mph)	1074	68 Nm (50 lbft)	8 kW (11 hp)
32 km/h (20 mph)	1074	47 Nm (35 lbft)	5 kW (7 hp)

Table 7: Proposed Test Cycle Points for LTV Axle

To establish a test temperature, differential temperature data collected during the vehicle testing was reviewed. Unique to the recent variants of the HMMWV, the rear-most differential incorporates an integrated liquid cooling system to control differential oil temperature. This effectively causes the differential temperature to stay at a fairly consistent temperature during all operation. In review, all of the on-track vehicle testing conducted on the HMMWV showed an average rear differential temperature of approximately 79.4°C (175°F). As a result, this temperature was again selected for the remaining efficiency test runs for the HMMWV.

Proposed Test Cycle - Results for Light Axle

The HMMWV axle was operated over the proposed efficiency test cycle for all three oils. The plotted results showing efficiency improvement compared to baseline 80W-90 and resulting confidence interval at each point can be seen in Figure 19 and Figure 20.

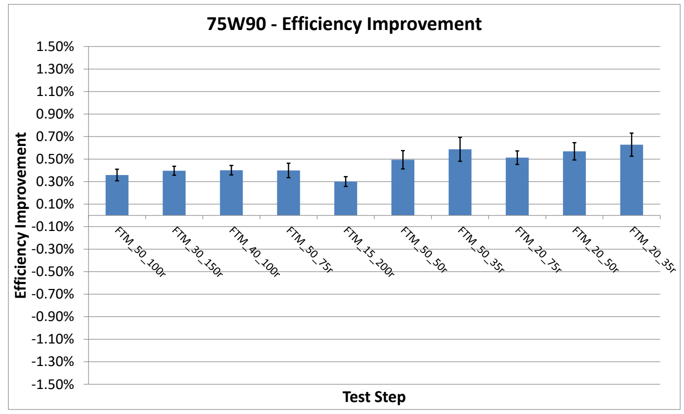

Figure 19: LTV Efficiency Improvement with 95% Confidence Bounds – 75W-90

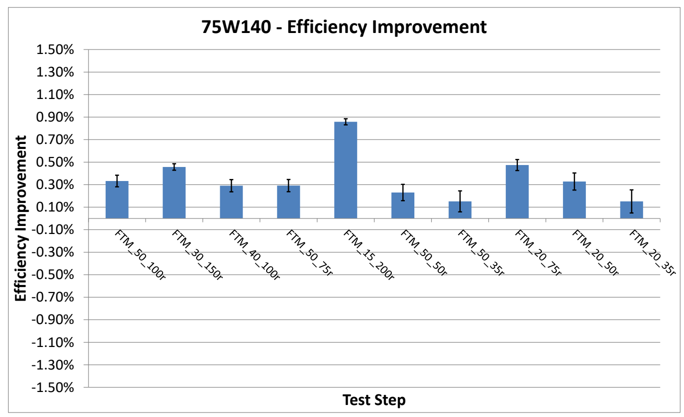

Figure 20: LTV Efficiency Improvement with 95% Confidence Bounds – 75W-140

As suggested from the mapping data and the on-track vehicle testing, both the 75W-90 and 75W-140 showed improvements over the 80W-90. The 75W-90 resulted in improvements ranging from +0.3% to +0.6%, with an overall average improvement of +0.47%. Repeatability in the results was again excellent, with resulting confidence intervals of +/- 0.1% or less. For the 75W-140 results ranged from approximately +0.15% to about +0.85%, with an overall average improvement of +0.36%, and confidence intervals at +/- 0.1% or less.

Also clearly evident in the results is the impact of the higher viscosity 75W-140 on efficiency related to load. This can be quantified by taking the ratio of the speed to the torque. Such a ratio is sometimes referred to as a quasi-Stribeck number and provides a representation of the lubricating film thickness, such that, higher speeds result in larger film thickness (i.e., larger quasi-Stribeck number) while higher loads result in reduced film thicknesses (i.e., smaller quasi-Stribeck number). Table 8 provides the quasi-Stribeck number for each test step. The quasi-Stribeck numbers range from 3 at 806 rpm and 271 Nm (200 lbft) torque to 57 at 2686 rpm and 47 Nm (35 lbft) torque. A lower number implies conditions that would promote a reduced film thickness and thus favor a higher viscosity lubricant, resulting in less surface–to-surface contact.

Test Step	Pinion Input Speed (rpm)	Pinion Input Torque	Quasi-Stribeck Number (rpm/torque)
FTM_50_100	2686	136 Nm (100 lbft)	20
FTM_30_150	1611	203 Nm (150 lbft)	8
FTM_40_100	2149	135 Nm (100 lbft)	16
FTM_50_75	2686	102 Nm (75 lbft)	26
FTM_15_200	806	271 N (200 lbft)	3
FTM_50_50	2686	68 Nm (50 lbft)	40
FTM_50_35	2686	47 Nm (35 lbft)	57
FTM_20_75	1074	102 Nm (75 lbft)	11
FTM_20_50	1074	68 Nm (50 lbft)	16
FTM_20_35	1074	47 Nm (35 lbft)	13

Table 8: LTV Test Cycle and Quasi-Stribeck Number

Temperature Control – Revisited

As discussed in an earlier section, there are two basic schools of thought when it comes to controlling the axle lubricant temperature during an axle efficiency test. With a focus on maximizing test stand sensitivity and repeatability it was decided to carefully control the axle lubricant temperature during testing. Thus, a test procedure was developed that carefully controlled the lubricant temperature resulting in excellent sensitivity to changes in efficiency and was very repeatable. In an effort to ensure the best possible test procedure had been identified, it was decided to modify the axle efficiency test stand to be able to control the cooling rate to the axle and investigate axle efficiency impact while allowing the test lubricant to stabilize at its natural steady-state temperature. Therefore, the heating and cooling loop of the test stand was modified by the installation of an appropriately sized Coriolis flow meter and the flow rate data was used to calculate the cooling rate. Cooling rate control targets were based on the maximum cooling rate identified from previous fixed temperature MTV axle efficiency testing and scaled with speed. The 80W-90, 75W-90, and 75W-140 were retested in the MTV axle using the fixed cooling rate targets. Figures 21 and 22 are plots of the temperature response and efficiency of the 80W-90 and 75W-90 when tested under the conditions previously defined in Table 5 using a fixed cooling rate. Figures 23 and 24 are plots of the temperature response and efficiency of the 80W-90 and 75W-140.

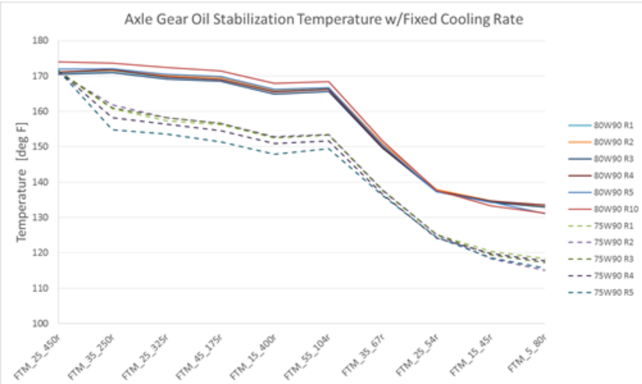

Figure 21: MTV Axle Lubricant Temperature with Fixed Cooling Rate – 80W-90 vs. 75W-90

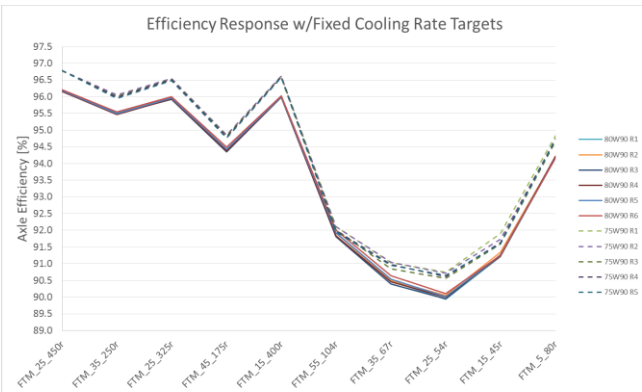

Figure 22: MTV Axle Efficiency Response with Fixed Cooling Rate – 80W-90 vs. 75W-90

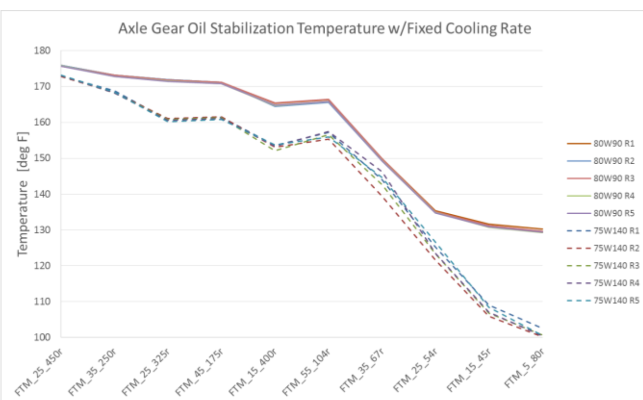

Figure 23: MTV Axle Lubricant Temperature with Fixed Cooling Rate – 80W-90 vs. 75W-140

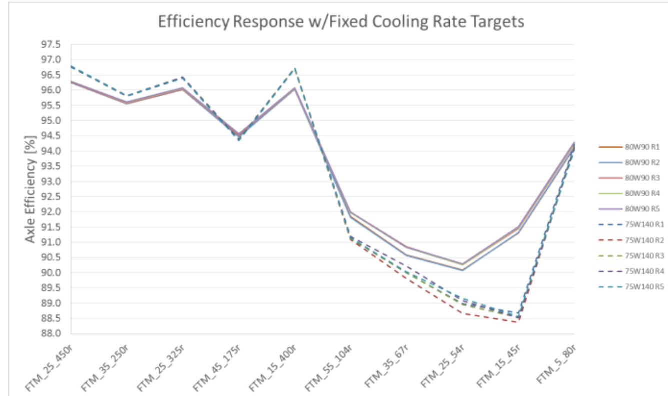

Figure 24: MTV Axle Efficiency Response with Fixed Cooling Rate – 80W-90 vs. 75W-140

From Figure 21 it can be seen that the 75W-90 results in a reduction of temperature of approximately 10 to 15 degrees over the majority of the operating conditions. Comparing these efficiency results shown in Figure 22 to the results generated using a fixed temperature of 175°F (see Figure 12), it is demonstrated that the use of a fixed cooling rate does not significantly change the results from the first five higher power and load steps of the cycle. For the five lower power and load steps, the fixed cooling rate resulted in a reduction in efficiency compared to those generated using fixed temperature. This response can be rationalized by noting that the reduced operating temperature of the 75W-90 results in a slightly higher viscosity which increases churning losses. These losses have a greater impact at the lower power and load points where churning losses are a higher percentage of total losses. From Figure 23 it can be seen that the 75W-140 also results in lower temperatures over the entire cycle, while for efficiency shown in Figure 24, the first five higher power and load steps of the cycle resulted in very similar results compared to the fixed temperature (see Figure 13). For the five lower power and load steps generated by the fixed cooling rate, the results indicate a significant reduction in efficiency compared to the results generated using a fixed temperature. This suggests that the efficiency results at fixed temperature are over estimating the efficiency. A possible explanation for this is that the temperature of the axle lubricant during the fixed cooling rate test is significantly lower than the fixed temperature test (conducted at 175°F), approaching 75 degrees Fahrenheit by the last step of the test. The lower operating temperature results in higher viscosity and more churning losses. These results, as well as the results from the 75W-90 test at fixed cooling rate, indicate that the use of the fixed operating temperature of 175°F is unrealistic for the lower power and load steps.

SUMMARY AND CONCLUSIONS

A stationary axle efficiency test stand was constructed and a test procedure was developed to identify gear oils that provide improved efficiency and reduced fuel consumption. The test stand and procedure will allow the U.S. Army to evaluate current and future gear oils during the qualification process, selecting only those with the capability to provide improved fuel efficiency. It was shown that compared to standard 80W-90, fuel efficiency can be improved by more than 2% and operating temperatures may be reduced by 10 – 30 degrees Fahrenheit. For large fleet owners even modest reductions in fuel consumption can result in major fuel savings. Furthermore, the lower operating temperatures reduce lubricant degradation allowing oil drains to be extended and maintenance to be minimized.

The 75W-90 tested in this paper provided improvements in efficiency over a large range of operating conditions. It was demonstrated that efficiency did vary with axle size and duty cycle. In the case of the Army’s light tactical vehicle, on-track testing results showed that the 75W-140 provided greater fuel economy than the 75W-90, particularly under conditions of low speed and high loads. Under these conditions, results from the stationary axle efficiency test showed that efficiency almost tripled, going from 0.30% improvement for the 75W-90 to 0.85% improvement for the 75W-140 compared to standard 80W-90. Thus, based on an analysis of a vehicles duty cycle different gear oil grades might be selected to optimize fuel economy benefits.

An investigation of the efficiency and temperature response under conditions of fixed cooling rate indicates a dynamic equilibrium exists between the effects of efficiency, temperature and oil viscosity. Improved efficiency leads to lower gear oil temperatures, but lower temperatures result in increased viscosity and churning losses. Thus an axle and its oil are in constant dynamic response to input speed, load and cooling level. The fixed cooling rate procedure highlighted the need to carefully consider operating temperature and suggested that our original operating temperature used for fixed temperature testing was unrealistic for the high speed and low load operating conditions. Therefore, for future gear oil efficiency evaluations we have modified the fixed temperature procedure to operate at 175°F for steps 1 – 5 (i.e., high load) and 140°F for steps 6 – 10 (i.e., low load). Additional testing using these revised temperatures is underway.

The final step in the development of the stationary axle efficiency procedure, and implementation as a Federal Test Method, will be to finalize a method to distill the efficiency results down to a single number (or small set) that can be used for qualification.

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