EFFECTS OF TEST CONDITIONS ON BEARING PERFORMANCE

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Abstract

This work explains the effects of test conditions on the performance of grease lubricated tapered bearings. The elastic modulus (G') of two NLGI 2 grade greases, A and B, was measured at temperatures from 25 to 150°C using a rotary viscometer. G' was consistently higher in B indicating that it was stiffer and more shear stable than A. All test conditions were generated with a KRL rig and were designed to simulate various lubrication regimes described by the Stribeck-Hersey curve. Six test sequences with specific speed and load combinations were implemented at four grease loadings and two operating times. In each sequence, bearing performance was evaluated by measured torques and temperatures and by completion of the operating time. The starved flow lubricant model in combination with the rheological data was used to describe grease behavior and assign the corresponding lubrication regime during each sequence. The tribological data indicates that both greases performed equally well in low to medium severity sequences. In the more severe sequences, all failures were temperature related and superior tribological performance was linked to the higher stiffness and greater shear stability of grease B.

Key-words: Grease; Rheology; Tribology.

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BACKGROUND

An effective description of lubrication regimes in grease lubricated bearings remains problematic because of inherent complexities in the rheological properties of the grease. It has been reported that these properties control the nature, supply and replenishment of grease in the tribological contact (1-4). However, unanimous agreement over which parameters best describe this behavior has yet to be reached. Typically, viscosity is selected in mathematical calculations and models used to describe tribological regimes. One such model, known as the Stribeck-Hersey curve, has successfully described the relationship between viscosity, torque, speed and load (5). A version of this curve modified to account for grease lubrication is shown in Figure **1**. This relationship is however obscured by the pressure dependency of the viscosity term. George et al. showed that a rheological parameter, the elastic modulus (G'), could be used to establish this relationship (6-8). G' measures grease stiffness, whereas loss of G' after shearing measures its shear stability. These authors showed that G' was strongly correlated to increasing temperature and also to bearing performance in different grease types. However, they did not report a formal mathematical description for these relationships.

In this work, we will use six test sequences to further validate their approach with two commercial greases. First, rheological data is obtained for two polyurea greases with a rotary viscometer operated at different temperatures. Second, for each grease and test sequence, temperature and torque responses are collected with a KRL thrust bearing tester designed to simulate bearing performance under the mild, medium and severe lubrication regimes depicted in Figure 1. Bearing performance is measured by these responses and depends on the grease type and its loading, the test sequence and its operating time. Finally, we will show that the correlation between these two techniques can be used to explain grease behavior and predict bearing performance across a wide range of mechanical applications.

EXPERIMENTAL

Two commercial NLGI 2 grade extreme pressure additized polyurea greases, A and B, with known performances in the KRL tester were chosen for this work (9). Their listed information indicated that A was formulated with a Group III and B with an ester base stock. No additional information was given for either material.

Rheological measurements were obtained with a controlled stress rheometer equipped with textured parallel plates (TA Instruments AR1000). Fundamentally, an oscillatory stress is applied to the top plate and the response of the bottom plate is periodically sampled (6-8). Immediately after loading a grease sample, a time sweep at constant frequency and strain is conducted to obtain its recovery. The amplitude and phase of the response relative to the applied stress are then used to calculate the elastic or storage modulus (G'). Torque and time sweeps at 25, 40, 100 and 150^{0}C, were also conducted at constant strain and angular frequency to monitor temperature effects on grease structure. The recorded G' was obtained by averaging the linear region of these plots (8). These four temperatures were selected because they closely approximate previously recorded bearing operating temperatures (9).

All tribological tests were conducted with a KRL thrust rig fitted with a 40-mm bore tapered bearing (9). Temperatures and torques were monitored and collected for the duration of each test sequence but only the maximum values were extracted from these traces to simplify the data analysis. The cut-off limits for motor shut-down or test failure were set at 200^0C and 10Nm for temperature and torque, respectively. Six test sequences, depicted in Table 1, and grease loadings of 5, 2.5, 1 and 0.5g were used to influence lubrication conditions and establish performance limits of these greases. To further influence severity, each sequence was designed to operate for either 2.5 hours (S) or 6.5 hours (L). The L operating time at similar load and speed settings was designed to be more severe than the S operating time. These mechanical settings were designed to simulate tribological conditions encountered in different bearing applications and described in Figure 1. The low load and high speed is mild, the high load and low speed is medium and the high load and high speed condition is severe. During the first half-hour of each sequence, a break-in step was implemented to smooth out the contacting surfaces and thus help scale and normalize torque and temperature responses. This step is a precaution against premature failure and is intended to insure that bearings successfully transition from boundary to mixed Elastohydrodynamic (mixed/EHD) lubrication. A total of forty-seven bearings with two bearings per sequence and grease loading were used in this work. Usually, the lowest grease loading was run first to confirm pass or fail. A pass was determined by the successful completion of the test sequence. If failure occurred, then progressively higher grease loadings were tested until a pass was attained.

RESULTS & DISCUSSIONS

Bulk and starved flow rheological models have been advanced to describe grease behavior in most bearing applications (1-4). In bulk flow, the oil and thickener are considered a single phase and flow continuously into the contact to lubricate the components. The starved flow model suggests that grease will gradually degrade as it passes through the contact and is subjected to high local stresses and very high temperatures. Degraded thickener then deposits on the tracks while an oil/thickener mixture bleeds to form a lubricating film. This model and experimental support from G' and the lubrication regimes described in Figure 1 will be used to explain our results.

The rheological data for these greases is shown in Figure 2. At each temperature, the higher G' values of grease B, in the stress plot indicate that B is stiffer than A. G' was higher at 150^0C than at 25^0C suggesting that it is also very shear stable. The time plot showed that recovery is almost instantaneous at each temperature. Thus grease B is unusually stiff and recovers very quickly at high temperature, which might be explained by oil loss from evaporation. In contrast, the stress trace for grease A decreases as temperature increases suggesting that it is not very shear stable. Its time plot indicates that it recovers very well at each temperature within 20 minutes of testing. George et al, observed that better tribological performance was always obtained when the stress and recovery sweeps of G' remained unaffected by high temperatures (6, 7). This information suggests that B should perform better than A in the KRL tester.

The performance of any grease is dictated by its ability to generate a lubricating film and to equilibrate input and output sources of frictional heat during the various lubricating regimes. Physical explanations for temperature and torque signals have already been advanced in prior publications (9-12). Briefly, and provided motor shutdown does not occur, these signals are caused by the rib-roller contact and by shear forces between the roller body, the working grease film and the cup-cone raceway. Rib-roller contact forces a bearing to operate in the boundary regime. It is always present at the onset of the test and always creates a sharp rise to a peak value followed by a decrease and leveling off of the torque and temperature signals as a working lubricant film is formed. If the signal decrease is not instantaneous, motor shut-down is quickly initiated. This interaction is also regenerated when the working film collapses in the presence of high temperature and shear and will likewise trigger motor shut-down. The second tribological condition deals with the effective maintenance of the working grease film created after the break-in step. Both torques and temperatures will randomly display transient signal increases for brief periods before slowly decreasing to lower values. These perturbations are directly related to intermittent variations in the thickness of the working film as the oil/thickener mixture unevenly flows into the contact.

Figures 3 and 4 show the maximum temperatures and torques extracted from traces associated with the forty-seven tested bearings. The figures include the break-in steps and show where passes and failures occurred with each grease type and grease loading in each test sequence. Maximum break-in values are enclosed in boxes for clarification. Note that test conditions are replicated for some or all of the four grease loadings within a sequence. Each test sequence and its relation to these plots will be described in more detail in the subsequent paragraphs.

Sequence I was the least severe and was designed to simulate lightly loaded and short duration applications such as high speed spindle bearings. In this sequence, the outer raceway is lightly loaded against the inner raceway which rotates at increasing speeds. Physically, it caused the gap between the rolling elements and the raceway to increase, thus allowing more grease to flow into the contact. It also raised the temperatures which induced oil/thickener mixture to bleed from the grease and push hot grease outward more easily with the high centrifugal forces. This behavior is, in part, explained, by the starved flow model. Effective equilibration of these opposite responses allows operation in the mixed/EHD regime depicted in Figure 1. Figures 3 and 4 show that the bearings passed with both greases loaded at 0.5g. The low average torques and temperatures observed during the break-in step confirmed a successful transition from boundary to mixed/EHD lubrication. The temperatures for both greases gradually decreased as sequence conditions moved from break-in to the next load and speed combination (300KN, 1500rpn). Beyond that combination, the average temperatures for both greases increased while their torques decreased. On average the values for B were higher than those of A. This suggests that Sequence I favored the grease with the lower shear stability.

Sequence II was more severe and was designed to simulate bearing applications in electric motors. The maximum temperatures and torques for both greases are shown in Figures 3 and 4, respectively. When the greases are loaded at 0.5g, one bearing fails and one passes with A, whereas both bearings pass with B. The data shows a successful transition from boundary to mixed/EHD lubrication with slightly higher torques than those recorded during the break-in of Sequence I. Even under controlled conditions, these variations cannot be entirely eliminated in bearing tests. Beyond the break-in step, replicate runs of grease A at 200KN and 3000 rpm indicate that the bearings were operating very close to 200°C. The traces show that with grease A the bearings had difficulty equilibrating the heat input during the L operating time. In the first test, the bearing operated at 194^0C and barely passed while in the second it reached the 200^0C mark around 3.5hr and failed. The average maximum torque values for A were 1.5Nm indicating operation in the mixed/EHD regime. Since failure and near failure with grease A were temperature rather than torque related, we think that hot oil/thickener mixture was continuously introduced into the contact as described by the starved flow model. However, the heat input was so high that it induced motor shut-down. The average maximum temperatures for B were less that 130^0C. Average torques around 1.5Nm indicate that operation was, as expected, in the mixed/EHD regime. Although the starved flow condition was also in effect during operation, we believe that grease B was successfully able to negotiate the heat input because of its superior recovery and shear stability. The performance difference between A and B indicates that the longer operating time favored the grease with the higher shear stability.

Sequence III was designed to simulate short duration in heavily loaded applications such as journal bearings found in mining applications. In this medium severity sequence, the outer raceway is heavily loaded against a slowly rotating inner raceway. This loading forces the gap between the two raceways to slowly close which in turn restricts grease motion. The decreasing speeds and temperatures reduce grease mobility and flow towards the contact. These two responses combine to force operation in the mixed to boundary regimes. The maximum temperatures and torques indicate that the bearings passed with both greases loaded at 0.5g. Again, on average, the values for B were slightly higher than those of A. At 300rpm and 65^0C, the average torque was 1.5Nm since more oil/thickener mixture could bleed into the contact. The torques gradually increased to 3.5NM when the speed and resulting temperatures decreased to 100rpm and 50^0C, respectively. The bearings were starting to move from the mixed/EHD to the boundary regime. These responses are in very good agreement with the starved flow model and also indicate that the grease with the lower shear stability had lower temperatures and torque values.

Sequence IV was designed to simulate heavily loaded bearings operating for long duration under steady state conditions. Again, for A and B loaded at 0.5g, Figures 3 and 4 suggest a successful break-in as temperatures and torques are below 80^0C and 4.5Nm. Similar to Sequence III, the maximum torque was rather high. Beyond the break-in, both replicates of B passed, but the instrument triggered a torque related failure with the first replicate of A. This was the only torque induced failure and it occurred after 2 hours of operation. The temperature traces showed the same gradual decrease from break-in to nearly 40^0C observed at the end of the previous sequence. The torque traces for both greases were very noisy, but nevertheless tracked the speed and load combinations. The bearings operated in the boundary regime since very little oil/thickener mixture got into the contact. Although the torque traces of B indicated that it also operated in the boundary regime, it passed because of its higher shear stability in good agreement with the starved flow model.

Sequence V was designed to simulate behaviors experienced by axle bearings during short driving conditions. In this high severity sequence, the outer raceway is heavily loaded against an inner raceway rotating at medium to high speeds. The high load again forces the gap between the rolling elements and the raceway to close thus decreasing grease flow into the contact and the high centrifugal forces resulting form the increasing speeds push the grease outward. The combination of these two effects favors starved flow behavior and film formation from the hot oil/thickener mixture. The successful equilibration of both effects allows operation in the mixed/EHD regime while the opposite causes a transition to the boundary regime which induces bearing failure. The maximum temperatures and torques for A and B loaded at 1.0g and 0.5g are shown in Figures 3 and 4, respectively. In this sequence, grease loading becomes a significant performance differentiating factor. Temperatures and torques below 80^0C and 4.5Nm suggested that bearing break-in was successful. Beyond that point, the maximum temperatures point to two failures for A at 1.0g and two passes for B at the 0.5g. With grease A, the temperatures for the two replicates gradually increased above 200^0C, causing the test to stop. The high temperatures and torques for A indicate that hot oil/thickener mixture flowed into the contact but was unable to maintain a working film. This lack of replenishment was most likely caused by high centrifugal forces which pushed the grease away from the contact. The absence of a working film gradually forced the bearings to operate in the boundary regime and caused their failure. The maximum temperatures for B indicate that the bearings successfully dissipated the heat and operated in the mixed/EHD regime. Average torques of 2Nm also confirmed that a hot oil/thickener mixture continuously flowed in the contact forming a working lubricating film. These responses suggest that the superior shear stability of B is sufficient to compensate for its lower grease loading.

Sequence VI was very severe and was designed to simulate behaviors experienced by axle bearings during long driving intervals. Figures 3 and 4 show a pass and a fail with grease A loaded at 5g and two failures with B when loaded at 0.5g. Grease loading was the significant performance differentiating factor. With a 1g loading of B both replicates pass. Hence, at least five times the loading of A is needed to match the performance of B. Temperatures and torques below 80^0C and 4.5Nm were recorded suggesting a successful bearing break-in. Beyond that point, the temperatures increased rapidly for both greases. With grease A, the passing bearing reached a maximum temperature of 195^0C, while the other reached 199^0C before triggering failure after 1hr of operation. The maximum torque for the passing bearing was 3.1Nm suggesting that enough oil-thickener mixture was delivered to the contact. Again, this mixture was very hot, but nevertheless managed to force operation in the mixed/EHD regime. Hence, grease was available to form a film in one case, but not in the other. A closer look at temperature traces of B loaded at 0.5g, indicate that the bearings successfully dissipated the heat input for about 3.0 hours. Beyond that, temperatures quickly increased to 192^0C and 199^0C, triggering motor shut-down. Individual torque values for these two failures were less than 1.5Nm and confirmed that hot oil/thickener mixture flowed into the contact during operation. Beyond that point, torques sharply rose suggesting that the existing lubricating film suddenly collapsed. This information was in very good agreement with the shear stability data collected for these two greases and strongly suggests that the starved flow model can be further refined once shear stability and grease volume are properly included in the original definition.

CONCLUSIONS

The rheological and tribological performance results of A and B were reported in this work. The stress plot indicates that B was stiffer and more shear stable than A since it had higher G' values at each measured temperature. This data predicts that B will have better tribological performance than A. The rheological data in combination with the starved flow lubricant model were used to describe grease behavior and assign the lubrication regime during the six tribological test sequences. The tribological results indicate that both greases performed equally well under the low to medium severity sequences and short operating with 0.5g grease loading. At the highest test severity and longer operating time, all failures were temperature related and at least five times the loading of A was needed to equal the performance of B. Finally, the data analysis showed that G' and shear stability can be successfully used to predict the behavior of grease lubricated bearings under a wide range of tribological conditions.

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Time (H)	Speed (rpm)	Load (KN)	Conditions
	Sequence I		
Time (H)	**Speed (rpm)**	**Load (KN)**	**Conditions**
0.5	1000	450	Break-in Time
0.5	1500	300	
0.5	2000	250	
0.5	3000	200	Short
	Sequence II		
0.5	1000	450	Break-in Time
6	3000	200	Long
	Sequence III		
Time (H)	**Speed (rpm)**	**Load (KN)**	**Conditions**
0.5	1000	450	Break-in Time
0.5	300	300	
0.5	200	600	
0.5	100	900	Short
	Sequence IV		
0.5	1000	450	Break-in Time
6	100	900	Long
	Sequence V		
Time (H)	**Speed (rpm)**	**Load (KN)**	**Conditions**
0.5	1000	450	Break-in Time
0.5	1500	300	
0.5	2000	600	
0.5	3000	900	Short
	Sequence VI		
0.5	1000	450	Break-in Time
6	3000	900	Long

Table 1. Mechanical test procedures.

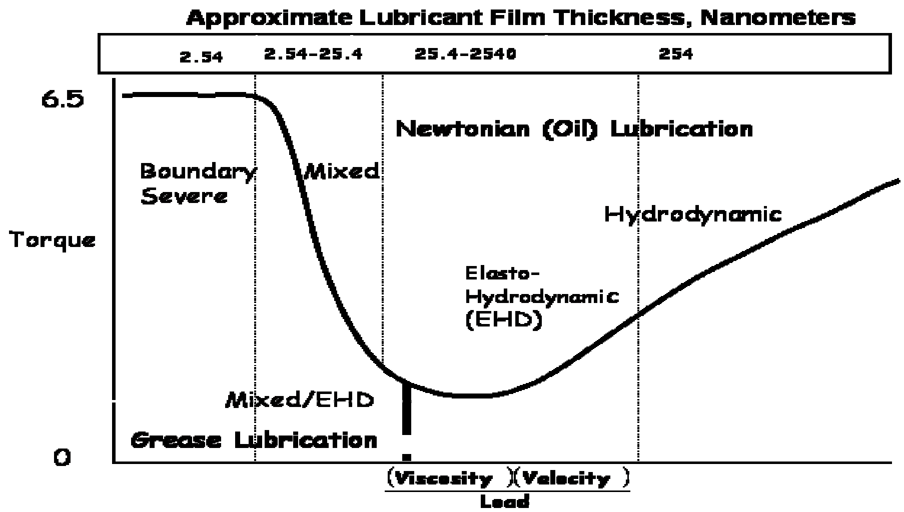

Figure 1. Modified Stribeck-Hersey curve.

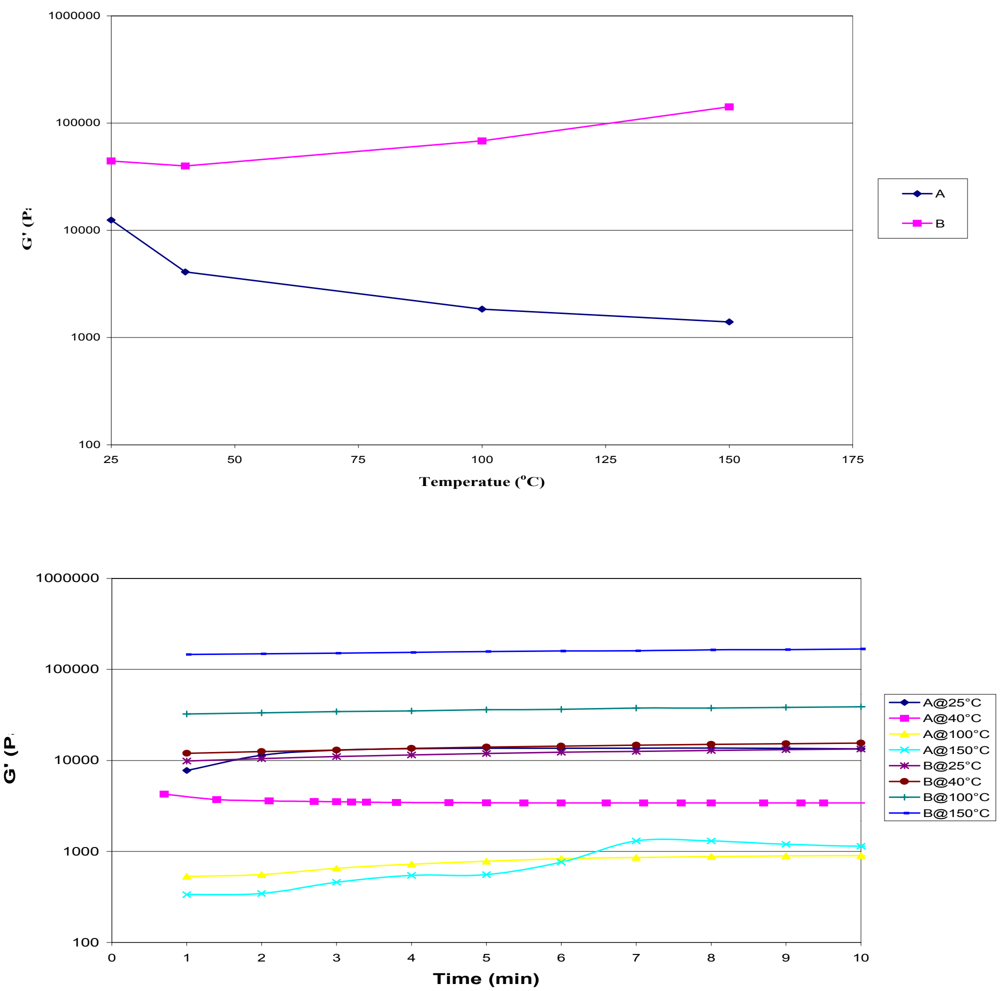

Figure 2. Stress and recovery curves for A and B.

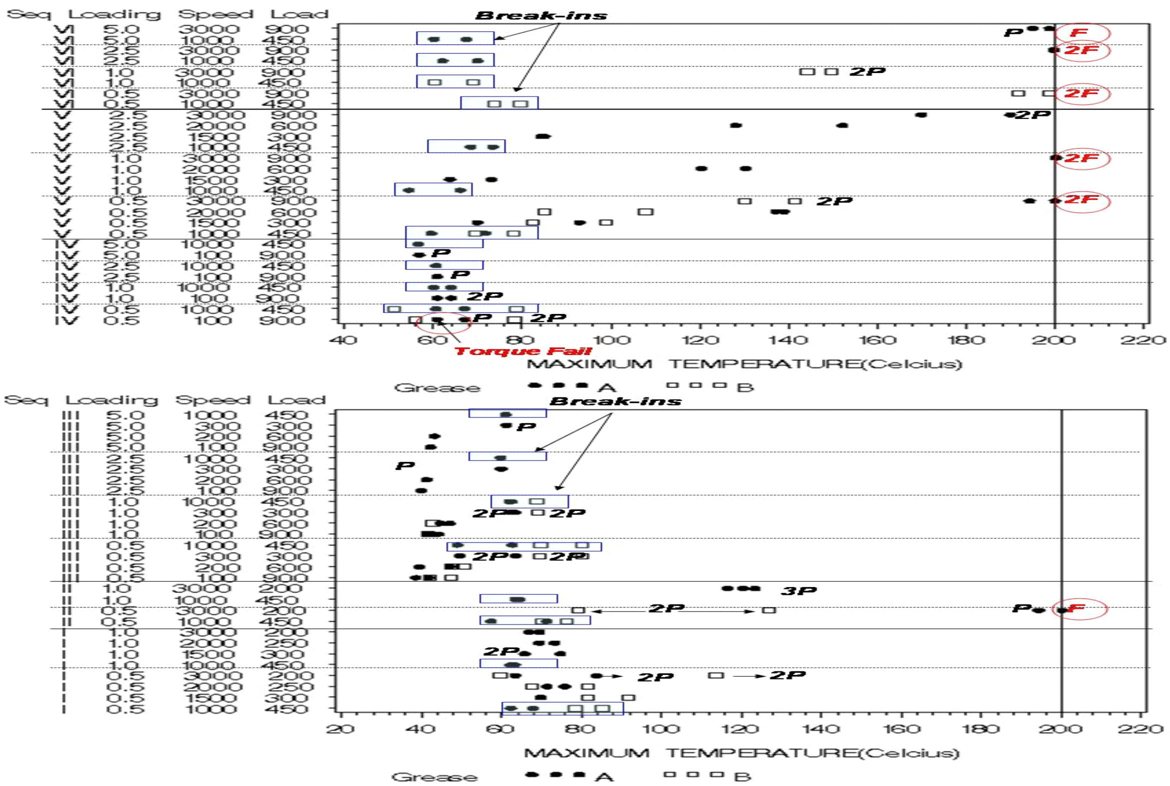

Figure 3. Maximum temperature responses for A and B.

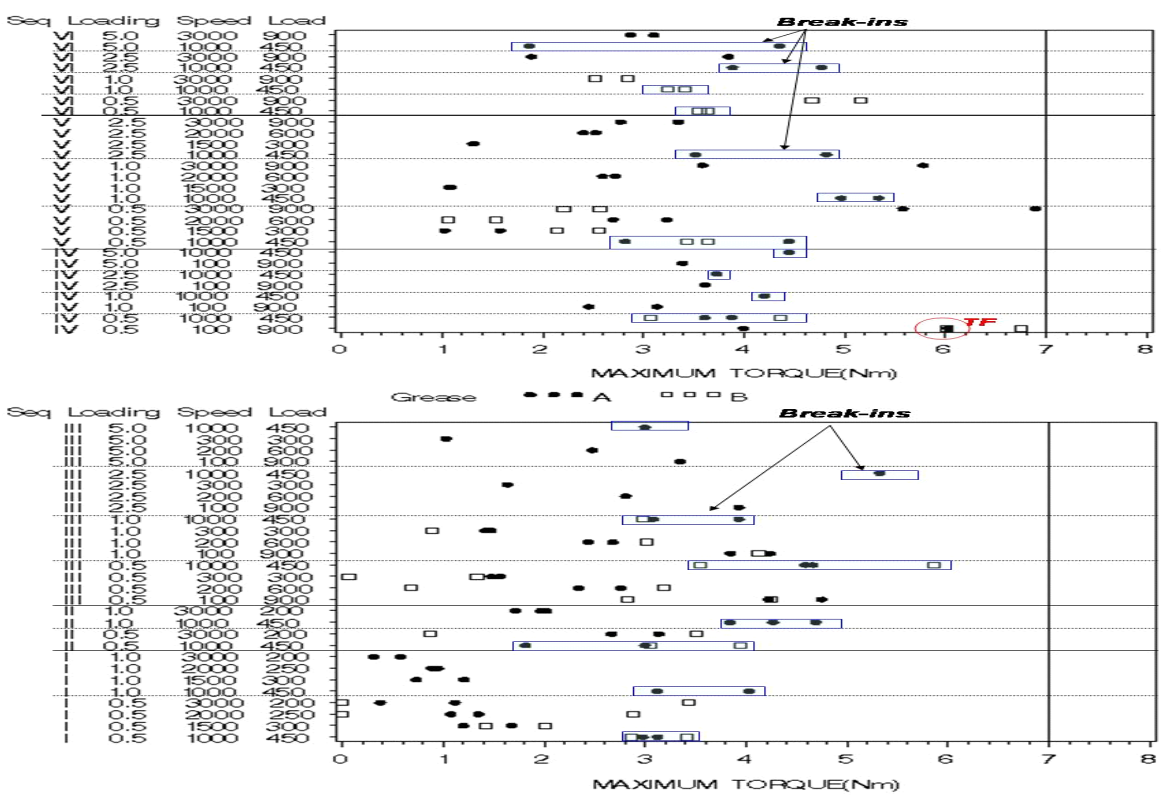

Figure 4. Maximum torque responses for A & B.