Thompson Couplings Limited - Orange

# INTERNAL CORRESPONDENCE

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| To: | TCL Board, CEO, (cc Tim Smith) |
| From: | David Farrell |
| Date: | July 7th 2008 |
| Subject: | Test results from UNSW dynamometer – June / July 2008 |

*Objective*: A series of engineering tests were performed at the ICE Lab of UNSW to quantify the efficiency effects and vibration characteristics of the Thompson coupling compared to existing Cardan and Rzeppa style shaft couplings. The tests consisted of back to back tests using a Cummins 7litre diesel engine driving a Froude eddy current dynamometer through the various shaft couplings. Approximately 70 tests were conducted over 2 separate events varying shaft angles, speed, torque and misalignment angles between flanges.

*Test methodology:*

There were two specific objectives of testing that were carried out to measure efficiency differences and vibration differences between all three couplings. The efficiency testing involved a fixed throttle setting and torque from the diesel engine in a straight line and varying the coupling angle to observe the resulting drop in engine speed as recorded from the dynamometer. The vibration results were recorded from a uniaxial accelerometer mounted on the dynamometer. In the following summary the term misalignment refers to the relative angle between the flanges of each coupling (in normal practice it is to be kept parallel)

*Summary of results:*

- In the first set of tests to measure efficiency differences the effect of shaft misalignment was discarded to represent an ideal world condition. Under these conditions there was virtually no detectable differences ( less that 0,3%) found between all three types of couplings. This result essentially negates the capability of the test equipment based on the premise that some degree of difference should have been detected on the Rzeppa style coupling.
- Testing of the Rzeppa coupling did result in friction failure at an early stage as has been previously experienced. Notably where the test data did not reflect the change in power whilst running at 2100 rpm and 100Nm torque the temperature of the Rzeppa joint rapidly reached over 150 degrees in less than 10 minutes while articulated at 14 degrees angle
- In the second series of tests the real world misalignment conditions were tested to compare Efficiency & NVH benefits by recording the vibration signatures of both the UJ and TC couplings only. (the premise being that efficiency losses equate directly to change in vibration levels). The Rzeppa style coupling was eliminated in these tests.
  - Based on a misalignment angle of 6 degrees the Thompson Coupling showed a significant improvement in vibration levels across all loads and speeds compared to the UJ coupling.
  - Based on a misalignment angle of 3 degrees the Thompson Coupling showed little improvement in vibration level across load and even less improvement across speed compared to a UJ coupling. It also showed that the UJ performed better as the shaft angle increased !
  - Based on a misalignment of 0 degrees the Thompson Coupling showed a very slight improvement in vibration levels with load and even less improvement with speed compared to a UJ coupling.

*Important points to note:*

- Validity of data is limited to the accuracy of the dynamometer and control of the diesel engine. This may account for the uncertainty in results for the efficiency tests especially when comparing the Rzeppa coupling to the standard UJ and Thompson Couplings.
- Vibration data for the 6 degree misalignment test is valid and indicative of the harshness also felt in the floor when testing both couplings. However during the 3 degree misalignment tests there occurred a step change in the Thompson Coupling resulting from either a centering mechanism bearing failure or looseness in the assembly (slop). It became apparent that the amount of slop in the centering mechanism changed at some stage through the 3 degree and subsequent 0 degree tests and this may explain why the TC results are somewhat variable and slightly higher than would be expected.
- However it does not negate the fact that the standard UJ vibration at 3 degrees and 0 degrees was significantly lower than at 6 degrees (nominally an order of magnitude less) and that the TC has little improvement in vibration reduction according to this data
- Both UJ and TC shafts were dynamically balanced at Sydney Driveline Services in a straight line and at 2100rpm. After balancing the runout of each coupling end was less than 0,005".However after running at the university it was observed that the spline connection of the Thompson shaft was imperfect resulting in some slop. The maximum recommended radial movement for a slip spline is 0,15mm which will be measured later.
- With regard efficiency the major contributing factor to energy loss in a cardan joint is friction losses in bearings and their seals. A standard UJ has 4 needle rollers and high efficiency seals per coupling which accounts for its rated 99% plus efficiency value.

*Summary table of relative benefits:*

If one compares the benefits of the Thompson Coupling to a UJ based on the test results and additional subjective sensory analysis in terms of NVH savings with respect to angles the following table could be concluded (where 1 is probably no commercial benefit and 10 is maximum benefit):

**Thompson Coupling Benefit vs UJ shaft**

Misalignment Angles

| Operating angles | 0 deg | 3deg | 6 deg |
|---|---|---|---|
| 0 deg | **1** | **3** | **1** |
| 2 deg | **1** | **3** | **2** |
| 4 deg | **1** | **3** | **4** |
| 6 deg | **1** | **2** | **6** |
| 8 deg | **2** | **2** | **8** |
| 10 deg | **2** | **2** | **9** |
| 12 deg | **2** | **1** | **10** |
| 14 deg | **2** | **1** | **10** |

*Discussion of NVH & drivetrain dynamics:*

According to manufacturers specifications (Dana Corp) "it is not required to precision align the driving shaft with the driven shaft, however cancellation of the non uniform motion characteristics of the cardan joint will occur when the angularity of each universal joint is equal and in the same plane. Deviations from this ideal cancellation should be limited to the motion that produces an angular acceleration of less than 300 rad/sec$^2$…. For maximum durability of the universal joint bearings, the true angularity $a^2=(a_i^2+a_o^2)$ , must be between 0,5 and 3 degrees."
Again regarding Dana specifications care must be taken to insure the dynamic torsional moments resulting from the non c-v do not exceed limits that will impose damage to the drivetrain components. These limits are a function of the angle of misalignment, speed of rotation and mass moment of inertia of the drivetrain."

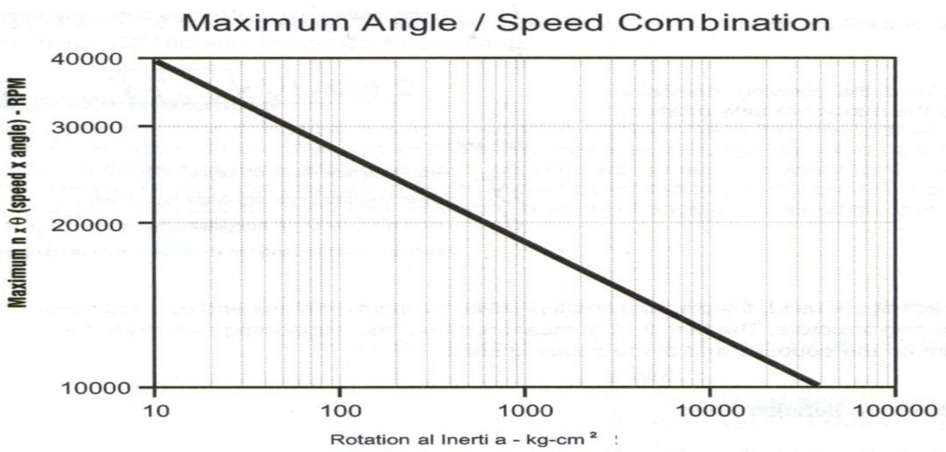



***Chart 5***

Rotational inertia (Kg-cm²) = component mass moment of inertia (Kg-cm²)+Tubing length (mm)/100 x tubing mass(Kg-cm²) / 100mm

By comparison Elbe express the maximum fluctuation rate U should be less or equal to 0,0027 where U = 1/cos b – cos b and b is the joint angle. Furthermore with a compound offset angle the total corresponding deflection angle should be less than 3 degrees.

Spicer also recommend that the ideal should be operating angles within 1 degree of each other, have a 3 degree maximum operating angle and have at least ½ degree continuous operating angle to achieve optimal bearing life. All angles should be measured with an accuracy of ¼ degree. Things that cause U joint angles to change include: suspension changes resulting from worn bushings in spring hangers, worn bushings in torque arms and incorrect airbag height. Other factors include revisions in driveline components and chassis modifications, additions to transmission cases and worn engine mounts.

Hardy Spicer recommend that each joint have at least 1 degree operating angle to allow the needles to roll slightly and thus prevent highly stressed contact area from being in the one position all the time.

There are 3 distinct forms of driveshaft disturbances that produce NVH problems and should conform to the following limits:

- Torsional Excitation (less than 400 rad/sec$^2$)
- Inertia Excitation (less than 1000 rad/sec$^2$)
- Secondary Couple Excitation

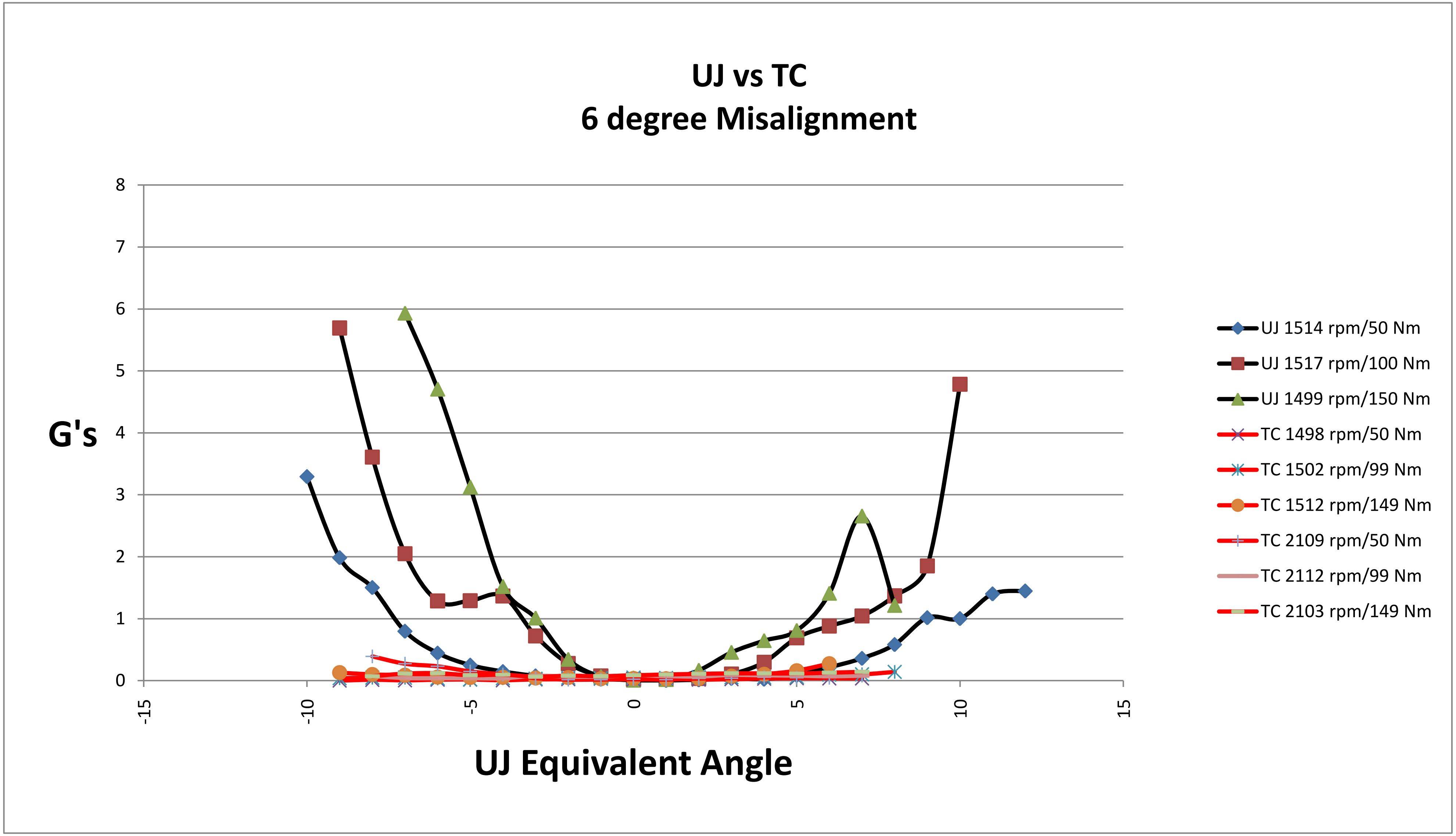
The torsional excitation occurs purely from the non cv oscillation of the cardan joint

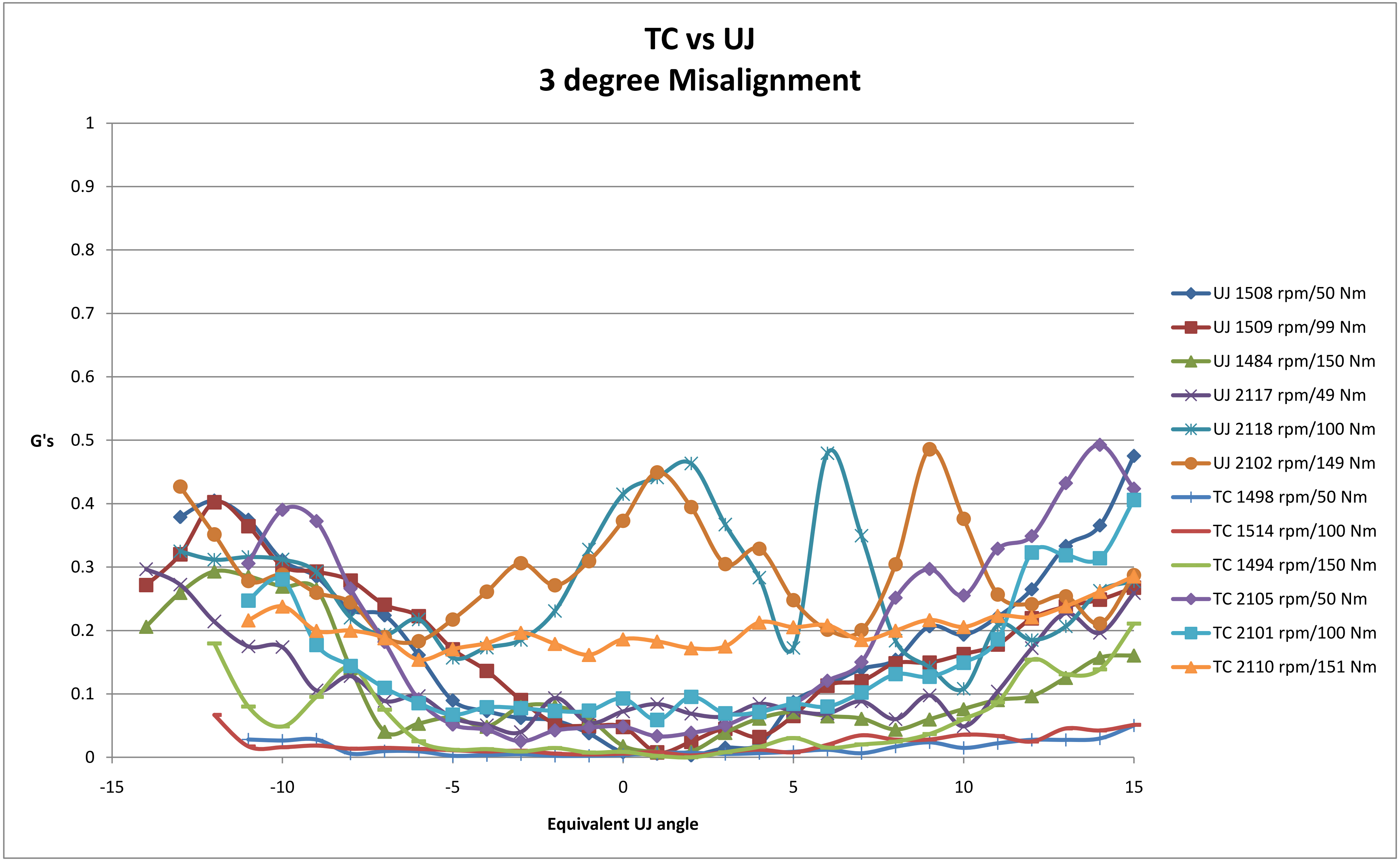
The inertia excitation is produced by the oscillating torque loads resulting from the driveshaft inertia being accelerated through the non cv motion.
Finally the secondary couple excitation occurs from torque transmission at an angle.
The SAE manual recommends that inertia excitation from the non CV driveshaft be limited to 1000 rad/sec$^2$ angular acceleration for passenger cars and light trucks. This corresponds to a maximum joint angle of 4,75 degrees operating at 3600 rpm. Larger trucks and off highway vehicles may generally tolerate higher levels of disturbances. When the shaft inertia is relatively small, as in the case of multipiece truck drivelines, the maximum angular acceleration can be increased to 2000 rad/sec$^2$. Based on the 1000 rad/sec$^2$ limit the following table shows the continuous condition for a driveshaft with equal joint angles:

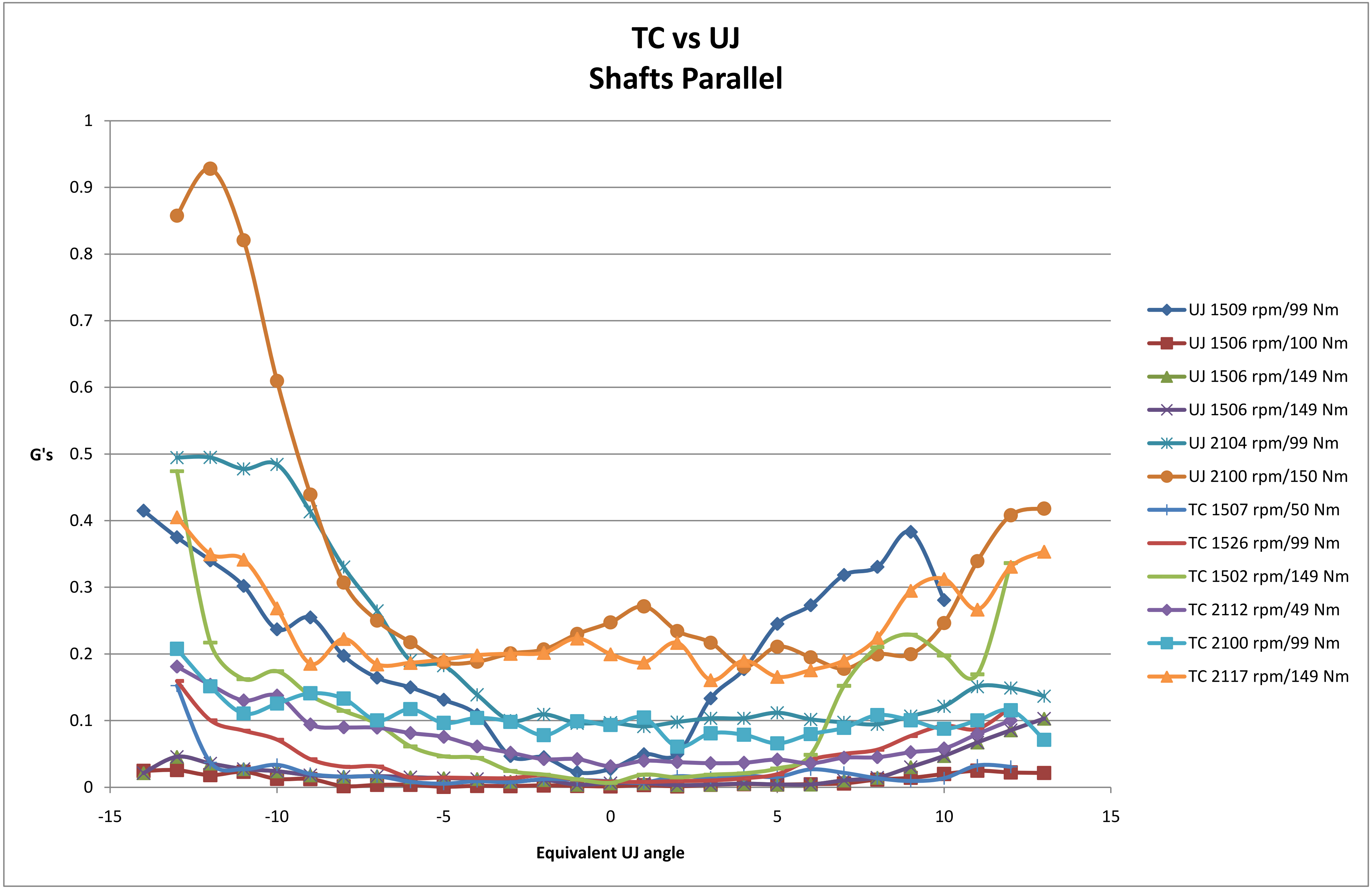
| **Driveshaft speed rpm** | **Maximum operating joint angle degrees** |
|---|---|
| 5000 | 3,5 |
| 4500 | 3,8 |
| 4000 | 4,3 |
| 3500 | 5 |
| 3000 | 5,75 |
| 2500 | 6,9 |
| 2000 | 8,66 |
| 1500 | 11,5 |

"It should be noted that the equivalent joint angle limits specified for acceptable levels of torsional and inertia excitation relate to idealized situations. In practice, the level of these disturbances have sometimes been found to be acceptable with joint angles that are relatively greater than those suggested."

Furthermore the amplitude of torsional excitation produced by the cardan joint oscillation which can be accepted without causing excessive disturbances depends on the operating speed of the driveshaft and the characteristics of the supporting structure and other units in the drivetrain. Experience has shown that if this form of excitation is maintained under 400 rad/sec $^2$ max angular acceleration in any continuous operating position and speed the driveshaft, in general, will perform satisfactorily in passenger cars and most truck applications.


UJ vs TC
6 degree Misalignment
G's
UJ Equivalent Angle
8
7
6
5
4
3
2
1
0
-15
-10
-5
0
5
10
15
UJ 1514 rpm/50 Nm
UJ 1517 rpm/100 Nm
UJ 1499 rpm/150 Nm
TC 1498 rpm/50 Nm
TC 1502 rpm/99 Nm
TC 1512 rpm/149 Nm
TC 2109 rpm/50 Nm
TC 2112 rpm/99 Nm
TC 2103 rpm/149 Nm


TC vs UJ
3 degree Misalignment
G's
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
-15
-10
-5
0
5
10
15
Equivalent UJ angle
UJ 1508 rpm/50 Nm
UJ 1509 rpm/99 Nm
UJ 1484 rpm/150 Nm
UJ 2117 rpm/49 Nm
UJ 2118 rpm/100 Nm
UJ 2102 rpm/149 Nm
TC 1498 rpm/50 Nm
TC 1514 rpm/100 Nm
TC 1494 rpm/150 Nm
TC 2105 rpm/50 Nm
TC 2101 rpm/100 Nm
TC 2110 rpm/151 Nm



TC vs UJ
Shafts Parallel
G's
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
-15
-10
-5
0
5
10
15
Equivalent UJ angle
UJ 1509 rpm/99 Nm
UJ 1506 rpm/100 Nm
UJ 1506 rpm/149 Nm
UJ 1506 rpm/149 Nm
UJ 2104 rpm/99 Nm
UJ 2100 rpm/150 Nm
TC 1507 rpm/50 Nm
TC 1526 rpm/99 Nm
TC 1502 rpm/149 Nm
TC 2112 rpm/49 Nm
TC 2100 rpm/99 Nm
TC 2117 rpm/149 Nm