FILLING SOME GAPS IN SPLINE DESIGN GUIDELINES: CENTERING, FRICTION, AND MISALIGNMENT
This paper provides an accurate method for calculating radial loads transmitted by straight-sided splines by means of the effective pressure angle calculation, enabling more accurate hoop-stress calculation for these splines.

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International spline standards such as ISO 4156, ISO 14, ANSI B92.2, and SAE J499a have detailed definition of two-dimensional spline geometry but cover only the most basic axial effects such as helix error. Other widely used published documents, such as those written by Dudley, and Cedoz & Chaplin, provide information about stresses, including some axial effects such as misalignment. Many recent studies of load distribution have been published, but a general approach is not defined. Even armed with all these documents, the engineer is not provided with adequate guidance regarding several factors that influence how a splined interface functions. Additionally, for some characteristics, small-diameter spline interfaces behave very differently from larger diameter splines.

This paper presents information that is not found — or is not satisfactorily covered — in current standards and existing papers. These include: how to calculate the effective pressure angle of straight-sided splines that is needed to accurately determine normal and radial loads; how to calculate the effective centering force of a spline pair; an update to the Dudley misalignment factors that can be applied to splines of any size; and an update to the calculation of the maximum axial force that a spline can transmit via friction.

Results from analytical studies of centering forces and misalignment factors are provided, and an experimental study of centering force is discussed.

The effective pressure angle calculation method is based on the tooth thickness and profile angle and can be included in future standards and guidelines. The centering force analysis is supported by results from prior published measurement data (Medina, Olver) and new measurements. The misalignment factors are an update to the published table (Dudley) that covers a very limited — and undefined — spline size range. The friction force calculation method is a correction to widely published (Maag, and others) formulae that do not include the cam effect of normal force.

1 INTRODUCTION

This paper presents some practical approaches to some details of spline design that are not covered in widely used spline-design guidelines or are not clearly covered in standards. The results provide insight into how splines behave in non-dynamic conditions.

All splines described in this paper have a side fit with positive backlash and consist of a shaft and a hub as the interfacing parts. Tip-to-root clearances are larger than the backlash and form-clearance is large enough to prevent tip-to-root fillet contact.

Straight-sided splines come in many varieties, from parallel straight-sided splines defined by ISO14 [1] to non-parallel straight-sided splines that are not defined by an industry standard. Square and hex drives can even be considered and analyzed as splines. ISO 14 defines geometry but does not describe how to determine the radial loads associated with transmitting torque. A definition of, and calculations for, the effective pressure angle of a straight-sided spline is provided, and backlash effects are discussed to enable effective design and analysis of these splines.

The radial centering force provided by both involute and straight-sided splines is studied, including the behavior of off-center rotation. A description of the centerline motion during off-center rotation is provided. The effect of unequal tooth loading on centering for a hub with an offset load is explored. An update is provided for the formula for centering torque to include the effects of the shaft-rotation angle.

Angular misalignment affects spline contact width and stresses. A study of misalignment shows additional factors beyond those described by Dudley [2] in his load-distribution factor table are important.

Spline torque lock is discussed, and a formula is provided that considers the cam effect of the pressure angle, friction, and applied-axial load.

2 EFFECTIVE PRESSURE ANGLE OF STRAIGHT SIDED SPLINES

Anyone who has had the experience of a flat-head screwdriver slipping out of the screw head slot while tightening the screw should be able to appreciate the importance of having the driving and driven members rotate on the same center while transmitting torque. We call this “centering.” Since external splines essentially are a set of integral keys in a shaft, and internal splines are the mating set of keyways in the bore of the mating hub, a flat-head screwdriver can be considered a small diameter shaft with two radially tall straight-sided spline teeth. A Phillips-head screwdriver can be considered a four-tooth splined shaft with parallel straight-sided splines. The Phillips-head design ensures centering of the drive torque on the screw axis. These screwdriver examples both have an unusual characteristic for a spline: The minor diameter is nearly equal to the tooth thickness. Continuing with fasteners, square and hex shapes often used in bolts and nuts (as well as
other polygon drives) also can be considered variants of splines. In
the square and hexagon examples, the minor diameter is tangent to
the tooth flanks.

A common characteristic of straight-sided splines, whether par-
allel straight-sided splines as defined by ISO14 or non-parallel-sided
flat-tooth splines that have a non-zero included angle between drive
and coast flanks, is that they do not have a defined pitch diameter
that is useful for calculating tangential and radial forces. Another
characteristic is that, although the external and internal tooth
geometry is defined without regard to backlash, the contact zone
is highly dependent on backlash. These splines have a contact zone
that diminishes (becoming more concentrated toward the major
diameter) as backlash increases. As the profile contact zone de-
creases, the effective pitch radius used for force calculations increases
— whether that is considered the center of the profile contact zone
or the location of mean compressive stress in the profile pressure
distribution. The profile-contact zone size and effective-itch radius
also are affected by the tip radius.

2.1 NOMINAL PROFILE ANGLE FOR STRAIGHT-SIDED SPLINES
The nominal profile angle for various examples of straight-sided
splines is defined as the half-included angle from the center of the
external tooth to the flank of the tooth. The same angle exists in the
mating internal spline between the center of the tooth space and the
tooth flank. Parallel straight-sided splines have a zero-degree pro-
file angle. Figure 1 contains examples of straight-sided splines and
polygon drives showing their nominal profile angles: a) a parallel
straight-sided spline (one tooth shown) such as those defined by ISO14,
b) a non-parallel straight sided spline, c) a square drive, and d) a hex
drive. Although square and hex drives are not commonly thought of
as splines, they function essentially as straight-sided splines since the
drive faces are an integrated set of driving features and the width of
the contact zone in the profile direction is determined by the backlash
with the mating part.

2.2 EFFECTIVE PRESSURE ANGLE
FOR STRAIGHT SIDED SPLINES
Once a reference diameter is selected for a straight-sided spline,
whether it is the mid-tooth height or an arbitrary diameter within
the contact zone, tangential force can be calculated. But to calculate
the normal tooth force and its radial component correctly, the effec-
tive pressure angle, not the nominal profile angle, must be used. The
radial component of the normal force is important for understanding
hoop stress and radial deflections of thin wall parts.

The effective pressure angle is the sum of the nominal profile angle
(zero for parallel straight-sided splines) and the half-tooth thickness
in degrees (Figure 2). For the effective-pressure angle of the internal
spline space, substitute the space width in place of the tooth thickness
in the following formulas:

For parallel straight sided splines (zero-degree nominal profile
angle) with tooth thickness defined as a chordal distance:

\[ \alpha_e = \arcsin \left( \frac{s_a}{d_m} \right) \]

Equation 1

where

- \( \alpha_e \) = effective pressure angle (degrees)
- \( s_a \) = chordal tooth thickness (mm)
- \( d_m \) = mean diameter (mm)

For straight sided splines defined with a circular tooth thickness:

\[ \alpha_e = \alpha + \frac{360s}{2\pi d_R} \]

Equation 2

where

- \( \alpha \) = nominal profile angle (degrees)
- \( s \) = circumferential tooth thickness at the reference diameter (mm)
- \( d_R \) = reference diameter (mm)

2.3 MISMATCH IN EFFECTIVE PRESSURE ANGLE
Since straight-sided spline geometry is defined by the profile angle
(half-included angle) and width of the tooth or space, backlash (diff-

erence between internal-tooth space width and external-tooth thickness)
causes a difference in effective pressure angle between the external and internal teeth. Figure 3 shows the relationship of this effective pressure angle difference as the diameter of the spline increases. The four curves shown, from lowest to highest, are for 50 microns, 100 microns, 150 microns, and 200 microns of backlash.

Since the normal force on the external tooth must be of the same magnitude and angle as the corresponding normal force on the mating internal tooth, this difference in angle and its effect on the normal force must be properly considered. In a straight-sided spline with a small diameter and large backlash, the difference between using the external-tooth thickness vs. internal-space width to calculate the effective pressure angle, and therefore the normal force, is greatest. Since the external spline flank-to-tip radius likely contacts the flat flank of the internal spline, it is most appropriate to use the internal-spline space width for the calculation. Using the space width results in a slightly larger normal force, so this should be conservative.

For small diameter splines, this mismatch in angles causes a small profile-contact zone, so careful attention should be paid to the backlash specification for small-diameter straight-sided splines.

3 RADIAL CENTERING FORCE OF A SPLINE INTERFACE UNDER TORQUE LOADING

Involute splines are commonly understood to provide a centering force — but there are few references that provide comprehensive information regarding the amount of centering and the behavior of off-center motion. ISO 4156-1 [3] (Figure 3, page 12) shows that the centering force at any tooth is the radial component of the normal tooth force, which suggests that a spline carrying an applied radial load from imbalance or the weight of a supported hub results in unequal tooth forces (unequal load sharing). A free body diagram leads to the same conclusion. There are few, if any, explanations for whether straight (e.g. parallel straight-sided splines) provide any centering force, and if so, whether the amount differs from involute splines. A free body diagram analysis of a straight-sided spline system would show the physics is the same — an applied radial load will be supported by a centering force, resulting in unequal tooth load sharing, as with an involute spline. Thus, straight-sided splines should provide centering similar in magnitude to an involute spline that has a similar effective pressure angle and friction coefficient. Straight-sided splines, however, due to their tendency to have edge contact, may have a higher friction force than an involute spline.

Guo, et al. [4] provide a formula for restoring torque, which is the torque required to restore the supported part to the center of rotation. (In their system, the hubs on each end of the shaft are supported with bearings, and the shaft is supported by the hub splines.) Their formula includes the number of teeth in contact but does not distinguish between coast-side contact and drive-side contact. Coast-side contact is a reaction force that only exists when the hub is not centered, but only drive-side contact can provide centering. Their formula also does not account for the fact that not all drive-side teeth in contact have equal load due to the direction of their normal-tooth contact force relative to the load. While the fact the number of drive teeth in contact is important, a restoring torque formula should include a factor for the percentage of offset load carried by each drive side tooth in contact.

3.1 BEHAVIOR OF A HUB ROTATING OFF-CENTER

We return to the square drive as a simple spline system to investigate the behavior of a splined hub as it rotates off center due to its own weight on a shaft. This evaluation assumes motion is slow enough that there are no dynamic effects from hub inertia. Figure 4 shows several time steps of rotation. The hub has substantial backlash, allowing the motion of its centerline to be visualized. As the prior section demonstrated, this square drive is essentially a straight-sided spline with 45-degree nominal profile angle. In each time step, the shaft (black square) defines the center of rotation. The brown square represents the hub internal spline. The hub is forced downward by its own weight. Backlash is represented by clearance between the two squares. The center of the hub is a brown dot near, but not on, the center of rotation.

One quadrant has the number “1” in it to show shaft rotation in the counter-clockwise direction. Normal forces in the positive torque direction on the hub are in black. Reaction forces on the coast side of the tooth are in red. Drive flank forces that support the hub weight are higher in magnitude than those that do not (e.g. in view “f” the normal force shown in the upper right quadrant has a vertical component that supports the hub weight and provides most of the force to rotate the hub. The normal force shown in the upper left quadrant only contributes to hub rotation.

The center of the hub is not static — in view “a” it is below the center of shaft rotation. In view “b” it has slid to the left and below the shaft center. In view “c” it is again below the center. In views “d” through “f” it is below and to the right of the center. When the rotation reaches an angle at which the friction force is overcome by the hub weight, the hub will slide from its position in view “f” to a position similar to view “b” (but with the next tooth in the upper right quadrant). This pattern of motion indicates that there is more off-center hub motion normal to the direction of load (horizontal in this case) than motion aligned with the direction of load. The hub centerline moves in an arc from view “b” to view “f” (while tooth #1 has both drive and coast-side contact) and
then slides to contact the next tooth. This cycle of rotation followed by sliding occurs once per pitch rotation. The amount of centerline motion and sliding can be reduced by increasing the number of teeth or reducing backlash (or both). Once the torque is sufficient to lift the weight of the hub at all rotation angles (Figure 5), the rotation will be smooth since the hub spline is centered on the shaft spline, which is the center of rotation, and all drive flanks will be in contact.

Since the rotation of the hub is affected by friction (i.e. higher friction would cause it to slide at a larger angle than low friction) any comprehensive model of centering behavior would need to include friction.

### 3.2 TORQUE REQUIRED TO PROVIDE CENTERING

There may be an intermediate state during which the hub is lifted and dropped depending on the rotation angle of the spline. To investigate this potential interim state, consider the three shaft rotations in Figure 5 at the instant when the hub is lifted. In view “a” there is one tooth carrying the weight of the hub, and the force shown at tooth 1 is the vertical force reacting the weight (and providing all the torque). In view “b,” which is rotated 45 degrees from view “a,” there are two teeth in contact, and we can determine from the sum of forces in each direction that they carry equal load. The sum of the vertical components of their normal force will support the hub weight. In view “c” the vertical component of the normal force from tooth 1 will only carry a small fraction of the hub weight, so the next tooth carries most of it. Table 1 shows an example of a square drive (45-degree nominal profile angle spline) with tooth forces and torques that are required at several rotation angles through one pitch rotation to support the weight of a hub. The tooth load sharing (second and third columns) assumes tooth loads rise and fall in a sinusoidal pattern through 180 degrees of rotation, reaching the maximum load at the rotation angle where the normal force vector is opposite the direction of the hub weight. Since there is not a single torque value that lifts the same weight at all rotation angles, it is evident that if the torque is at a constant value between the minimum and maximum values in the table, the hub will lift and fall during each pitch rotation. Since this spline varies from one to two drive flanks in contact during off-center rotation and has 90 degrees of shaft rotation per pitch, it is an extreme case intended to demonstrate the phenomenon of slow-speed operation during the transition to (or from) centering. More typical splines used to support hubs would be expected to have a smaller torque range during the transition from onset of lift to the onset of complete centering.

Based on this analysis, the torque to provide centering is expressed by Equation 3, which is a modification of the formula provided by Guo, et al. [4]. In this formula, coast-flank forces should be zero, since the formula describes the onset of centering, and there are no drive forces in the same direction of the hub force (vertically downward in this case since the hub force is due to its own weight). Only drive flank forces that oppose the hub force are non-zero.

\[
T = \frac{wd\sum_{i=1}^{j} \left( T_i \cos(\gamma_i - \alpha_e) \right)}{2000 \cos \alpha_e}
\]

where

- \( T \) = shaft torque (Nm).
- \( T_i/T \) = fraction of load carried by drive flank of tooth \( i \).
- \( w \) = radial load (N).
- \( d \) = pitch diameter (mm).
- \( i \) = tooth number.
- \( j \) = number of teeth in 180 degrees.
- \( \gamma_i \) = shaft angle at tooth \( i \) contact point with hub, with zero defined as normal to the direction of the hub offset force (see Figure 6), (degrees).
- \( \alpha_e \) = effective pressure angle (degrees).

This formula can be used to determine the centering torque with any number of lifting teeth (up to the limit of half of the teeth lifting the hub), and to plot the theoretical lifting force vs. shaft-rotation angle. Figure 6 shows an involute spline with the tooth normal forces that contribute to centering (supporting the weight of the hub). The horizontal components of these tooth forces sum to zero net horizontal forces.
force, and the sum of tangential tooth forces times the pitch radius equals the shaft torque (and is opposite to the hub reaction torque). At the onset of centering, the sum of the vertical components of the lifting tooth forces equals the hub weight.

### 3.3 EXPERIMENTAL MEASUREMENT

When conducting fully-reversed or partially-reversed torsional testing of shafts with splines that have backlash, it is common to see a rotation discontinuity in the output compared to the sine wave torque vs. time input (Figure 7). The cause of this discontinuity is that the shaft has free motion through backlash, which takes a finite amount of time, and occurs when the torque changes from positive to negative and vice versa. During this transition, the shaft, which is the part that needs to have its weight supported by the test fixtures, falls off-center until the torque is sufficient to lift it again.

A measurement activity was undertaken to determine the torque required to center a shaft with an applied radial load near the spline at one end. The spline for this shaft has 21 teeth, 0.75 mm module, 30-degree pressure angle.

Testing was done on an MTS servo hydraulic rotary actuator with a controller running MTS 793 software with 100 Hz data collection and a rotary displacement transducer. Radial load was applied by hanging a known weight on one end of the shaft using a ball bearing to minimize rotational friction. Dial indicators were used as a secondary measure when the weighted shaft was centered or off-center. The test set-up is shown in Figure 8. Early in the testing, it became clear the calibration of the torque meter was not at a low enough torque to be able to detect the onset of centering with sufficient accuracy that the analytical model could be confirmed or rejected. Figure 9 shows a magnified trace of the transition from negative to positive torque, showing the difference between commanded torque from the actuator (black dashed line) and measured torque at the fixed end of the test stand (solid blue line). There appear to be two sources of the difference: a relatively stable offset due to wind-up of the system and the discontinuity both before and after passing through zero torque.

### 4. MISALIGNMENT LOAD FACTORS

Different methods are used to account for how misalignment affects spline stress. Some people use a measurement of the observed contact pattern — or an analytical estimate of it — and calculate contact pressure over the face width that seems to be in contact. A review of many published papers found that several authors present examples of stress vs. face width for splines under load, in some cases just from torque loading and in other cases including misalignment. Analytical studies were done by Volfson [5], Medina et al. [6, 7], Hong, et al. [8], Guo, et al. [4]; and experimental measurements by Dudley [2, 9], and Hong, et al. [8]. The stress vs. face width curves generally follow an exponential slope, and the maximum stress often is greater than three times the average value, but rarely greater than five. Dudley [2] provides a load-distribution factor table that indicates the stress increases as a function of two factors: misalignment and face width. Dudley’s load factor table covers a limited face width range of about 12 mm to 100 mm, and he does not indicate what pitch diameter, module, or pressure angle he used. A plot of the data from his chart is shown in Figure 10. The data follows a power function.

### 4.1 ANALYTICAL STUDY USING SPLINE LDP

An analytical study was done using Spline LDP to check whether splines with a range of diameters exhibit the same pattern of misalignment load factors vs. face width that Dudley published. Another purpose of the study was to determine whether this tool could be useful for estimating misalignment load factors. Several splines were evaluated: three pitch diameters and two length/diameter ratios. All are 1.0 module involute splines with a 30-degree pressure angle. The results of the study are in Table 2 and Figure 11. The data falls into two patterns: splines with a length/diameter ratio of 1.0 follow a power function (top three curves). Data with length/diameter ratio 0.2 are nearly linear over the range studied (lower three curves). It is apparent from the different slopes and shapes of the curves that L/D is a relevant factor, and diameter may be a relevant factor. Spline LDP appears to be a useful tool for estimating load distribution factors based on stress...
A series of shaft splines was designed to enable comparison of the effects of pitch diameter and face width. The basic spline designs are all 1.0 module and have three different pitch diameters. The nominal contact pressure based on a simple force/area calculation with uniform pressure was set at 360 MPa for the designs with 1.0 length/diameter ratio. Spline LDP was used to determine the peak stress for the baseline designs and for designs with 0.2 length/diameter ratio. The misalignment load factors were calculated based on the peak contact stress in the misaligned condition vs. the condition with perfect alignment (which included the non-uniform stress caused by torsional stiffness difference between the shaft and hub). The contact stress values are highly influenced by edge effects: For aligned splines, there is a stress peak at the inner contact diameter (corresponding to the internal spline minor diameter), and it is concentrated at one end. A 5-micron profile crown was added to the base designs to attempt to duplicate the stress profiles shown by Hong.

4.2 USING SPLINE LDP
Any finite element-based stress calculation method produces a very different stress value than the traditional force/area calculations available in Dudley’s time, so the specific stress values from Spline LDP cannot be compared to Dudley allowable stresses.

While using Spline LDP, some issues were encountered that led us to only consider patterns rather than exact results. The first issue is the pressure angle shown as an output did not match the angle used as an input. The second issue was the stress patterns for unmodified splines did not match those in Hong. Third, the current version of Spline LDP has no flexibility to define a different constraint on the outer part than the OD nor an option to vary the geometry from a full width hub. These issues limit the utility of the tool for predicting accurate contact stress, but they do not prevent it from being useful for comparison studies.

5 THRUST LOAD FROM SPLINE TORQUE LOCK
Crease [10] discusses axial tooth forces transmitted by a spline due to misalignment, axial sliding, and a combination of the two. He describes conditions such as small misalignments causing sliding, which result in effectively lower global-friction coefficients, and conditions such as perfect alignment causing lubricant to be pushed out of the interface, which results in higher global-friction coefficients. The paper includes a set of formulas for tooth friction force — but not a clear formula for total shaft axial force due to spline torque lock.

Transmitted thrust force from spline-torque lock can be from two phenomena: transmittal of an applied axial force on one part to the mating part through a spline or axial force generated in the spline coupling interface due to misalignment. In the first case, the force transmitted is the smaller of two values: the axial force applied to the first part or the amount of force that can be transmitted through the spline by friction (Equation 4). If the applied force exceeds the amount that can be transmitted, the spline experiences relative axial sliding, and another feature, such as a bearing, reacts on the remaining force. In the second case, the force transmitted through the spline joint according to Crease is a moment with a net axial force of zero.

\[
F_S = \min \left[ \frac{2000\mu T}{d \cos(\alpha)} F_A \right]
\]

where
\(F_S\) is spline torque lock axial force (N).
\(\mu\) is coefficient of friction.
\(T\) is torque (Nm).

\(d\) is pitch diameter (mm).
\(\alpha\) is pressure angle (degrees).
\(F_A\) is applied axial force (N).

The coefficient is static until motion occurs, and it is dynamic during sliding motion. If a spline has angular misalignment or a centerline offset that causes constant sliding during shaft rotation, the dynamic coefficient of friction (or an even smaller value per Crease) would be most appropriate. If the spline is static or rotating in very good alignment, the static coefficient (or possibly even the static-friction coefficient for dry contact) should be used. In either case, since friction coefficients vary depending on many application factors such as surface texture, lubrication type, lubrication amount, contact stress, temperature, and vibration, the user is advised to measure the friction force and then back-calculate an observed friction coefficient, then select a suitably conservative coefficient to use for future calculations of that system. In many applications, the user only needs to know the maximum friction force (the maximum axial force that a spline can transmit) so other parts of the system can be designed. In this case, a conservative (high) value for the friction coefficient can be used.

For an involute spline, the normal force is calculated from the torque, pitch diameter, and pressure angle. Along the profile contact zone, there is a range of pressure angles. In splines with a small number of teeth, the pressure angle at the tip can be multiple degrees of roll.
larger than at the pitch diameter. A calculus approach could be used to integrate loads along the profile to get a more accurate prediction, but this typically is not done since the friction coefficient has such a large uncertainty that the error from using the nominal pressure angle is not noticed.

For a straight-sided spline, the normal force calculation should be based on the effective pressure angle (a parallel sided spline per ISO 14 with six teeth has an effective pressure angle of approximately 13 degrees, not zero degrees).

6 CONCLUSION
Calculations for the effective pressure angle of straight-sided splines are established. Two formulas are provided: one for splines defined by a chordal tooth thickness and another for those defined by circumferential tooth thickness. This effective pressure angle, rather than the halfinclined angle, should be used to calculate the normal force on the tooth. The effective pressure angle for the internal spline tooth surface, rather than the external tooth, should be used to determine normal forces for both internal and external teeth if the backlash is large and/or the spline diameter is small. Square, hex, and other polygon drives can be analyzed as though they are straight-sided spline teeth if the contact width is known or can be estimated.

Centering behavior, even without considering dynamic effects at high speed, is complex. While rotating off-center, the center of the hub has a predictable, but irregular motion affected by friction. During the transition from eccentric rotation to centered rotation, the hub may oscillate between being centered and not centered until the torque is sufficient to maintain its center through all rotation angles. Straight-sided splines provide centering fundamentally similar to involute splines of a similar effective pressure angle.

Misalignment load factors are not simply a function of face width, they also are affected by spline-length-to-diameter ratio and probably by other factors. They can be estimated using tools such as Spline LDP.

Spline LDP is useful for comparative stress studies but not for absolute stress values, since it does not always produce the correct geometry and does not have the flexibility to adjust the hub constraints.

A formula for spline-torque lock is provided, which includes the cam effect of the spline pressure angle.

This paper provides an accurate method for calculating radial loads transmitted by straight sided splines by means of the effective pressure angle calculation. This enables more accurate hoop stress calculation for these splines. The centering-torque formula provides a useful calculation to determine the shaft torque required to center a mating hub without needing to know how many teeth are in contact. The method suggested here for determining misalignment factors covers splines in a wide range of sizes and L/D ratios.

Dudley’s table of misalignment factors is shown to under-predict the misalignment factor for some long L/D splines. Finally, the torque-lock formula provides a calculation of the maximum thrust that can be transmitted through a spline interface. This can be important for designing mating parts that react the thrust load transmitted through the spline.

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