11.4 Chordal measurement with a coordinate measurement machine (CMM)

The use of the chordal technique for data gathered by a CMM is discouraged. While a CMM can obtain points that are in a transverse plane and are on opposite sides of a gear tooth, this data can easily be used to directly find the transverse circular arc tooth thickness. A CMM should always measure the datum surfaces so the datum axis can be determined. All other measurements should reference the datum axis, so points on opposite sides of a tooth will be measured at a known diameter referenced to the datum axis. After determining the location of these points, it is strongly recommended that rather than finding the straight line distance between these points, the arc length between the points be found. This arc length is the functional transverse tooth thickness at the measurement diameter.

12 Backlash in gear meshes

12.1 General

Backlash in a gear mesh is the circular arc length at the operating pitch diameter through which one gear can freely rotate while holding its mate in a fixed position. In Figure 24, two spur gears are in mesh contacting on their flanks at the pitch point. The lower gear in the figure has an additional involute clockwise from its right flank representing the free rotation of the lower gear while the upper gear is held in a fixed position. The backlash is the circular arc length along the operating pitch diameter circle between point A and A'. The magnitude of the circular arc length of the backlash of the upper gear along its own operating pitch diameter is the same as that of the lower gear.

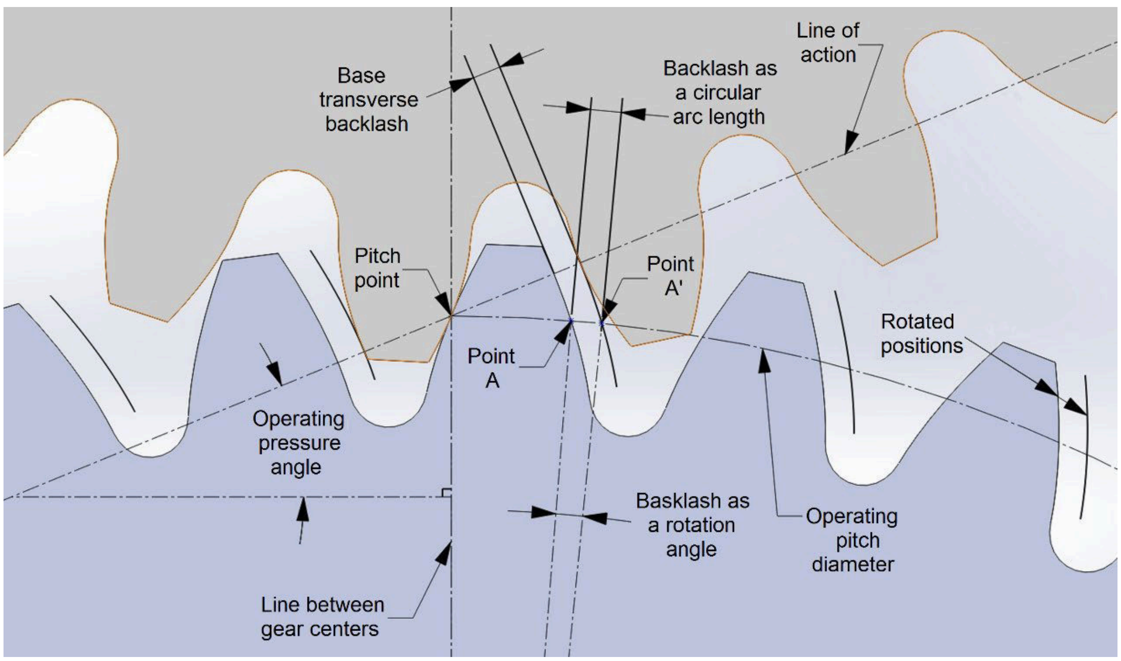

Figure 24 – Backlash in a spur gear mesh

The definition of backlash is very simple, but accurately predicting backlash is often difficult. The way to properly account for variations in operating center distance and in functional tooth thickness is not obvious. The method presented here will give a good estimate of the range of operating backlash, provided the range of operating center distance and the range of operating functional tooth thickness is known. A method of estimating the functional tooth thickness from a combination of the nominal tooth thickness range and the elemental tolerances is presented.

Backlash in a gear mesh is essential for smooth and efficient operation. Insufficient backlash can result in power loss, heat buildup, wear and noise. Too much backlash can result in excessive lost motion and impact noise upon direction reversal. Applications where gear rotation is unidirectional can usually tolerate a larger backlash range than reversing drives.

> **NOTE:** In some applications, allowing a larger range of tooth thickness tolerance and therefore a larger range of operating backlash might not adversely affect the performance or load capacity of gears while allowing for more economical manufacturing. A tight tooth thickness tolerance should not be used unless necessary, since it has a strong influence on manufacturing cost.

An individual gear does not have backlash. Backlash only exists in a meshed condition. A single gear can have many different mates at many different center distances, each with its own unique operating backlash range.

NOTE: A manufacturing drawing of a single gear should not specify backlash and should only refer to a predicted backlash range if the mating gear and operating center distance range are specified. The recommended practice for tooth size control is to specify the tooth thickness measurement method with the corresponding measurement limits, and to additionally list, as reference dimensions, the maximum and minimum normal or transverse circular arc tooth thickness at the reference diameter.

For both spur and helical gears, backlash is usually expressed as a circular arc length along the operating pitch cylinder in the transverse plane. This transverse circular backlash is used in this standard unless otherwise stated. It is rarely directly measured, but is often used during the design of gear pairs.

12.2 Factors that influence backlash

Backlash is affected by:

– the maximum and minimum operating center distance of the mesh;
– the maximum and minimum tooth thickness, either nominal or functional, for both the pinion and the gear;
– the total composite variation for both the pinion and the gear when tooth thickness is specified in the nominal system;
– other changes, such as those that only occur during operation, see 12.5.

When taking into account the stack up of these influences and the tolerances associated with them, backlash in a mesh is not considered to have a single value, but has a range of values. In an external gear pair, the minimum backlash condition occurs at the smallest operating center distance when the pinion and gear simultaneously mesh where the teeth appear to be the thickest. This is at their maximum double flank tight mesh center distance condition. Conversely, the maximum backlash occurs at the largest operating center distance when the pinion and gear simultaneously mesh where the teeth appear to be the thinnest. This is at their minimum double flank tight mesh center distance condition. As a result, gears that exhibit large total composite deviations (which includes runout) tend to have a large range between maximum and minimum backlash values.

In those cases where maximum backlash needs to be closely controlled, a careful study of the factors discussed here should be made and the geometric accuracy of the gears, range of operating center distance, range of tooth thickness and measurement method should be carefully specified. It may be necessary to specify smaller tolerances to hold backlash within the desired limits, since the lowest possible value for maximum backlash will increase as composite deviation increases.

NOTE: Total composite deviations include the effects of runout and other tooth form deviations. In all cases, on an individual product gear, the magnitude of the total composite deviation will equal or exceed the runout.

Most measurement methods only measure specific points on the tooth flank, and might not reveal the effect on backlash over the entire tooth surface. The entire functional surface of the product gear can be checked with an appropriate master gear in a double flank composite action test. When practical, this can give a more accurate prediction of the range of backlash compared to other methods.

12.3 Backlash in the functional system

For gear teeth specified in the functional system, the total composite variation is already accounted for in the functional tooth thickness specification itself, and so does not appear in the following calculations.

The influence of the operating center distance is accounted for through the definition of the operating pitch diameters. The maximum and minimum operating pitch diameters of the pinion and gear are:

$$d_{\mathrm{wi\ max}} = \left| \frac{2\, z_{\mathrm{i}}\, a_{\mathrm{w\ max}}}{\left(z_1 + z_2\right)} \right| \tag{169}$$

$$d_{\text{wi min}} = \left| \frac{2\, z_i\, a_{\text{w min}}}{(z_1 + z_2)} \right| \tag{170}$$

where

i is a subscript which denotes either the pinion (subscript 1) or gear (subscript 2);

NOTE: In this calculation, for internal gear meshes, the internal gear will always have subscript 2.

$d_{\text{wi max}}$ is operating pitch diameter, pinion or gear, maximum, mm;

z_i is number of teeth on the pinion or gear, depending on subscript i;

$a_{\text{w max}}$ is operating center distance, maximum, mm;

z_1 is number of teeth, pinion;

z_2 is number of teeth, gear;

$d_{\text{wi min}}$ is operating pitch diameter, pinion or gear, minimum, mm;

$a_{\text{w min}}$ is operating center distance, minimum, mm.

The maximum and minimum operating transverse pressure angles, which apply to both the pinion and the gear, at these operating pitch diameters are:

$$\alpha_{\text{wft max}} = \cos^{-1}\left(\frac{d_{\text{b1}}}{d_{\text{w1 max}}} \right)$$ (functional tooth thickness only) (171)

$$\alpha_{\text{wft min}} = \cos^{-1}\left(\frac{d_{\text{b1}}}{d_{\text{w1 min}}} \right)$$ (functional tooth thickness only) (172)

where

$\alpha_{\text{wft max}}$ is functional operating transverse pressure angle, maximum, degrees;

d_{b1} is base diameter of the pinion, mm;

$\alpha_{\text{wft min}}$ is functional operating transverse pressure angle, minimum, degrees.

NOTE: The operating transverse pressure angles for both the pinion and the gear are equal to each other at their operating pitch diameters. Therefore this calculation is only done for the pinion.

In order to account for the influence of functional tooth thickness, convert the functional normal circular tooth thickness values at the reference diameter to values at the operating pitch diameters.

The maximum and minimum functional transverse circular tooth thickness at the operating pitch diameters for an external gear pair are:

$$s_{\text{wtfi max}} = d_{\text{wi min}} \left(\frac{s_{\text{nfi max}}}{z_i\, m_n} + \operatorname{inv}\alpha_t - \operatorname{inv}\alpha_{\text{wft min}} \right)$$ (external only) (173)

$$s_{\text{wtfi min}} = d_{\text{wi max}} \left(\frac{s_{\text{nfi min}}}{z_i\, m_n} + \operatorname{inv}\alpha_t - \operatorname{inv}\alpha_{\text{wft max}} \right)$$ (external only) (174)

where

$s_{\text{wtfi max}}$ is functional transverse circular tooth thickness at the operating pitch diameter for the pinion or gear, for maximum tooth thickness, mm;

$s_{\text{nfi max}}$ is functional normal circular tooth thickness at the reference diameter for the pinion or gear, maximum, mm;

$s_{\text{wtfi min}}$ is functional transverse circular tooth thickness at the operating pitch diameter for the pinion or gear, for minimum tooth thickness, mm;

$s_{\text{nfi min}}$ is functional normal circular tooth thickness at the reference diameter for the pinion or gear, minimum, mm.

For a pinion mated with an internal gear, the maximum and minimum functional transverse circular tooth thickness at the operating pitch diameters for the pinion and ring gear are:

$$s_{\text{wtfi max}} = d_{\text{wi max}} \left(\frac{s_{\text{nfi max}}}{|z_i| m_n} + \frac{z_i}{|z_i|} \left(\operatorname{inv} \alpha_t - \operatorname{inv} \alpha_{\text{wft max}} \right) \right)$$ (pinion and internal only) (175)

$$s_{\text{wtfi min}} = d_{\text{wi min}} \left(\frac{s_{\text{nfi min}}}{|z_i| m_n} + \frac{z_i}{|z_i|} \left(\operatorname{inv} \alpha_t - \operatorname{inv} \alpha_{\text{wft min}} \right) \right)$$ (pinion and internal only) (176)

Note: With internal gears, increased center distance leads both to decreased backlash and to increased operating pitch diameters. On the pinion, the increased operating pitch diameter results in a decrease in apparent tooth thickness, and in some internal gear sets this affect is larger than the tolerance on tooth thickness. Therefore, $s_{\text{wtf1 max}}$ can be smaller than $s_{\text{wtf1 min}}$ for pinions that are mated with an internal gear.

The maximum and minimum transverse circular backlash at the operating pitch diameter based on functional tooth thickness for an external gear pair are:

$$j_{\text{wt max}} = d_{\text{w1 max}} \frac{\pi}{z_1} - s_{\text{wtf1 min}} - s_{\text{wtf2 min}}$$ (external only, functional thickness only) (177)

$$j_{\text{wt min}} = d_{\text{w1 min}} \frac{\pi}{z_1} - s_{\text{wtf1 max}} - s_{\text{wtf2 max}}$$ (external only, functional thickness only) (178)

where

$j_{\text{wt max}}$ is transverse circular backlash at the operating pitch diameter, maximum, mm;

$j_{\text{wt min}}$ is transverse circular backlash at the operating pitch diameter, minimum, mm.

For a pinion mated with an internal gear, the maximum and minimum transverse circular backlash at the operating pitch diameter based on functional tooth thickness are:

$$j_{\text{wt max}} = d_{\text{w1 min}} \frac{\pi}{z_1} - s_{\text{wtf1 min}} - s_{\text{wtf2 min}}$$ (pinion with internal only, functional thickness only) (179)

$$j_{\text{wt min}} = d_{\text{w1 max}} \frac{\pi}{z_1} - s_{\text{wtf1 max}} - s_{\text{wtf2 max}}$$ (pinion with internal only, functional thickness only) (180)

12.4 Backlash in the nominal system

In the nominal system, since tooth thickness is independent of the datum axis, the total composite deviation shall be taken into account when calculating backlash. Proper use of the nominal system requires that either a total composite tolerance be defined in the specification of the gears, or elemental tolerances be defined.

In the calculation of backlash, the total composite tolerance is needed. If only elemental tolerances are available, it is possible to estimate a total composite tolerance. While this estimate might not precisely predict the total composite errors, it is generally helpful for the prediction of backlash.

NOTE: The total composite error is measured with a master gear that will have contact across its face width while simultaneously being in contact with more than one pair of teeth. Sometimes errors are hidden due to bridging over teeth. Conversely, elemental measurements are typically only taken over a few teeth and only on a few lines across each tooth meaning that the entire profile is not scanned either. As a result, the actual total composite error can be higher or lower than that predicted by elemental measurements.

The estimate of total composite tolerance from elemental tolerances is based mainly on the accumulated pitch deviation (or runout), the single pitch, profile and helix tolerances.

$$F_{\text{idTi}} \approx F_{\text{rc}} F_{\text{rTi}} + \frac{\sqrt{f_{\text{pTi}}^2 + F_{\alpha\text{Ti}}^2 + F_{\beta\text{Ti}}^2}}{1.5 \cos \alpha_t}$$ (181)

where

F_{idTi} is total composite tolerance for the pinion or gear, µm;

F_{rTi} is runout tolerance for the pinion or gear, µm;

f_{pTi} is single pitch tolerance for the pinion or gear, µm;

$F_{\alpha Ti}$ is total profile tolerance for the pinion or gear, µm;

$F_{\beta Ti}$ is total helix tolerance for the pinion or gear, µm

F_{rc} is a multiplier to adjust conversion from runout to total composite tolerance.

The multiplier to adjust conversion from runout to total composite tolerance, F_{rc}, should be based on the user's experience, and should result in an accurate prediction of backlash. When an appropriate value based on experience is not available, a value of 1.0 may be used for F_{rc}.

If the runout tolerance is not defined, then the following equation may be used although it adds additional uncertainty:

$$F_{idTi} \approx F_{pc}\, F_{pTi} + \frac{\sqrt{f_{pTi}^2 + F_{\alpha Ti}^2 + F_{\beta Ti}^2}}{1.5 \cos \alpha_t} \quad (182)$$

where

F_{pTi} is total cumulative pitch (index) tolerance for the pinion or gear, µm;

F_{pc} is a multiplier to adjust conversion from total cumulative pitch to total composite tolerance.

The multiplier to adjust the conversion from total cumulative pitch tolerance to total composite tolerance, F_{pc}, should be based on the user's experience, and should result in accurate prediction of backlash. When an appropriate value based on experience is not available, a value of 0.9 may be used for F_{pc}.

NOTE: There is no technical correlation between the elemental tolerances of any accuracy standard and the composite tolerances of any companion standard. The equations presented here are just to assist in obtaining a rough estimate for the calculation of backlash, and should not be used for other purposes.

Once the total composite tolerances have been estimated, the calculation of backlash can proceed in a similar manner as was presented in the functional approach. The operating pitch diameters should be converted into functional operating pitch diameters using the equations:

$$d_{wfi\,max} = \left| \frac{z_i \left(2\, a_{w\,max} + \dfrac{F_{idT1} + F_{idT2}}{1000} \right)}{\left(z_1 + z_2 \right)} \right| \quad (183)$$

$$d_{wfi\,min} = \left| \frac{z_i \left(2\, a_{w\,min} - \dfrac{F_{idT1} + F_{idT2}}{1000} \right)}{\left(z_1 + z_2 \right)} \right| \quad (184)$$

where

$d_{wfi\,max}$ is functional operating pitch diameter, pinion or gear, maximum, mm;

$d_{wfi\,min}$ is functional operating pitch diameter, pinion or gear, minimum, mm.

The operating transverse pressure angle is the same on the pinion as it is on the gear. The maximum and minimum functional operating transverse pressure angles at these operating pitch diameters are:

$$\alpha_{\text{wft max}} = \cos^{-1}\left(\frac{d_{\text{b1}}}{d_{\text{wf1 max}}}\right) \quad \text{(nominal tooth thickness only) (185)}$$

$$\alpha_{\text{wft min}} = \cos^{-1}\left(\frac{d_{\text{b1}}}{d_{\text{wf1 min}}}\right) \quad \text{(nominal tooth thickness only) (186)}$$

The maximum and minimum nominal transverse circular tooth thickness for a pinion or an external gear at the functional operating pitch diameters are:

$$s_{\text{wti max}} = d_{\text{wfi min}}\left(\frac{s_{\text{ni max}}}{z_{\text{i}}\, m_{\text{n}}} + \operatorname{inv}\alpha_{\text{t}} - \operatorname{inv}\alpha_{\text{wft min}}\right) \quad \text{(external only) (187)}$$

$$s_{\text{wti min}} = d_{\text{wfi max}}\left(\frac{s_{\text{ni min}}}{z_{\text{i}}\, m_{\text{n}}} + \operatorname{inv}\alpha_{\text{t}} - \operatorname{inv}\alpha_{\text{wft max}}\right) \quad \text{(external only) (188)}$$

where

$s_{\text{wti max}}$ is nominal transverse circular tooth thickness at the functional operating pitch diameter for the pinion or gear, for maximum tooth thickness, mm;

$s_{\text{wti min}}$ is nominal transverse circular tooth thickness at the functional operating pitch diameter for the pinion or gear, for minimum tooth thickness, mm;

$s_{\text{ni max}}$ is nominal normal circular tooth thickness at the reference diameter, pinion or gear, maximum, mm;

$s_{\text{ni min}}$ is nominal normal circular tooth thickness at the reference diameter, pinion or gear, minimum, mm.

Or for a pinion mated with an internal (ring) gear:

$$s_{\text{wti max}} = d_{\text{wfi max}}\left(\frac{s_{\text{ni max}}}{|z_{\text{i}}|\, m_{\text{n}}} + \frac{z_{\text{i}}}{|z_{\text{i}}|}\left(\operatorname{inv}\alpha_{\text{t}} - \operatorname{inv}\alpha_{\text{wft max}}\right)\right) \quad \text{(pinion with internal only) (189)}$$

$$s_{\text{wti min}} = d_{\text{wfi min}}\left(\frac{s_{\text{ni min}}}{|z_{\text{i}}|\, m_{\text{n}}} + \frac{z_{\text{i}}}{|z_{\text{i}}|}\left(\operatorname{inv}\alpha_{\text{t}} - \operatorname{inv}\alpha_{\text{wft min}}\right)\right) \quad \text{(pinion with internal only) (190)}$$

Note: With internal gears, increased center distance leads both to decreased backlash and to increased operating pitch diameters. On the pinion, the increased operating pitch diameter results in a decrease in apparent tooth thickness, and in some internal gear sets this affect is larger than the tolerance on tooth thickness. Therefore, $s_{\text{wtf1 max}}$ can be smaller than $s_{\text{wtf1 min}}$ for pinions that are mated with an internal gear.

The equations for the maximum and minimum transverse circular backlash at the functional operating pitch diameter based on nominal tooth thickness for an external gear pair are:

$$j_{\text{wt max}} = d_{\text{wf1 max}}\frac{\pi}{z_1} - s_{\text{wt1 min}} - s_{\text{wt2 min}} \quad \text{(external only, nominal thickness only) (191)}$$

$$j_{\text{wt min}} = d_{\text{wf1 min}}\frac{\pi}{z_1} - s_{\text{wt1 max}} - s_{\text{wt2 max}} \quad \text{(external only, nominal thickness only) (192)}$$

For a pinion mated with an internal gear, the equations for the maximum and minimum transverse circular backlash at the functional operating pitch diameter based on nominal tooth thickness are:

$$j_{\text{wt max}} = d_{\text{wf1 min}}\frac{\pi}{z_1} - s_{\text{wt1 min}} - s_{\text{wt2 min}} \quad \text{(pinion with internal only, nominal thickness only) (193)}$$

$$j_{\text{wt min}} = d_{\text{wf1 max}}\frac{\pi}{z_1} - s_{\text{wt1 max}} - s_{\text{wt2 max}} \quad \text{(pinion with internal only, nominal thickness only) (194)}$$

12.5 Other potential influences on backlash

There are additional influences on backlash that can affect some applications. They are mentioned here to make the user aware that they can exist and that they might need to be considered relative to a specific application. Many of these occur during operation, and so will not be detected during a backlash check performed when the gear set is not operating. The potential other influences on backlash that should be considered include but are not limited to:

– **Measurement errors.** Measurement errors will affect the backlash, and the designer may account for them by adding some additional marginal backlash into the design.

– **Tooth deflection.** Gear teeth deflect under load, leading to unloaded teeth being closer than expected to the loaded teeth. In extreme cases, with insufficient tip relief, this can result in collisions between teeth at the start of meshing action.

– **Misalignment.** Misalignment of gears will decrease the backlash of the set. Misalignment can be caused by distortion of the gearbox housing due to either internal or external forces, differential thermal growths, and shaft position changes within journal bearings due to bearing reaction or other forces.

– **Torsional twist.** Twisting of the gear under loading changes the helix angle. This is a particular problem for highly loaded gears with a high ratio of face width to reference diameter. Gears that have been modified for optimal contact at full load can actually improve their helical shape at full load thereby increasing backlash during operation.

– **Thermal change.** Thermal growth or shrinkage of gears and housings can affect backlash depending on the nature of the gear materials, housing materials, and bulk temperatures of each of the components. In many cases, the gears operate at a higher bulk temperature than ambient or even housing conditions due to frictional heat buildup. Many gear housings need to operate over a wide temperature range, therefore this potential influence on backlash can warrant serious consideration (i.e., automotive electronic throttle body gears operating under the hood of a vehicle where the gears are made of plastic and the housing is made of die cast zinc).

– **Moisture change.** Some materials (i.e., some plastics) have a potential to expand over time as they absorb moisture, or shrink as they dry out. The effect of such changes can take place over an extended period of time. In some applications, moisture can lead to a significant loss of backlash (i.e., pump gears).

– **Phase change.** A phase change, such as from martensite to austenite in steel, can cause the gear to grow or shrink, and can occur after parts have left the manufacturer.

– **Centrifugal growth.** High tip speed gears can experience considerable growth due to centrifugal forces, leading to reduced backlash.

To avoid interference between teeth, all of these factors should be considered in the magnitude of the final backlash allowance in the design. Consideration should be given to the potential difference between the backlash measured at assembly and the operating backlash. The application of these factors and the calculation of the backlash resulting from these factors is beyond the scope of this standard.

12.6 Ways to express backlash

12.6.1 Transverse circular backlash

Transverse circular backlash is the basic backlash used in this standard. It is rarely directly measured, so this clause provides conversions to quantities that can easily be measured.

12.6.2 Base transverse backlash

Backlash can be expressed as a straight line length in the transverse plane along the line of action tangent to the gear's base circle, see Figure 24. It can be measured using a dial indicator positioned perpendicular to surface of the gear flank in the transverse plane. With one gear held stationary, the other gear is rocked back and forth to indicate the backlash. The indicator may contact any tooth, but shall be positioned so it contacts unmodified portions of the tooth flank as the gear is rocked.

NOTE 1: Minor deviations in perpendicularity generally have an insignificant effect on the readings.

NOTE 2: It is good practice to use a second indicator in contact with a tooth of the stationary gear to verify that it does not move.

For an external gear pair, the base transverse backlash is:

$$j_{bt\ max} = j_{wt\ max} \cos \alpha_{wft\ max} \quad \text{(external only) (195)}$$

$$j_{bt\ min} = j_{wt\ min} \cos \alpha_{wft\ min} \quad \text{(external only) (196)}$$

where

$j_{bt\ max}$ is base transverse backlash, maximum mm;

$j_{bt\ min}$ is base transverse backlash, minimum mm.

For a pinion mated with an internal gear, the base transverse backlash is:

$$j_{bt\ max} = j_{wt\ max} \cos \alpha_{wft\ min} \quad \text{(pinion mated with internal only) (197)}$$

$$j_{bt\ min} = j_{wt\ min} \cos \alpha_{wft\ max} \quad \text{(pinion mated with internal only) (198)}$$

12.6.3 Base normal backlash, feeler gage backlash

Backlash can also be expressed as a straight line length in the normal plane along the line of action tangent to the gear's base circle. This happens to be the shortest distance between the non-working flanks of the teeth in the gear pair and is often referred to as normal backlash, although the term base normal backlash is more suitable since it is actually in the normal plane of the base helix, not the reference circle helix. Another common term for it is feeler gage backlash because it can be directly measured with a feeler gage, or in some cases by putting a piece of soft solder in the gear mesh as it rolls through. This latter technique will give an indication of backlash over the whole profile, so mismatch in pressure angle will become very obvious. Another technique involves using a dial indicator positioned perpendicular to surface of the gear flank. With one gear held stationary, the other gear is rocked back and forth to indicate the backlash. The indicator may contact any tooth, but shall be positioned so it contacts unmodified portions of the tooth flank as the gear is rocked.

NOTE 1: Minor deviations in perpendicularity generally have an insignificant effect on the readings.

NOTE 2: It is good practice to use a second indicator in contact with a tooth of the stationary gear to verify that it does not move.

The base normal or feeler gage backlash is:

$$j_{bn\ max} = j_{bt\ max} \cos \beta_b \quad (199)$$

$$j_{bn\ min} = j_{bt\ min} \cos \beta_b \quad (200)$$

where

$j_{bn\ max}$ is base normal (feeler gage) backlash, maximum, mm;

$j_{bn\ min}$ is base normal (feeler gage) backlash, minimum, mm.

12.6.4 Axial backlash

In cylindrical worms used in crossed axis helical systems where the shafts are mounted perpendicular to each other, it is customary to define the axial backlash in a direction along the axis of the worm. The calculation of axial backlash is beyond the scope of this standard.

12.6.5 Angular backlash

The backlash may be expressed as a free rotation angle for each gear of an external pair while holding its mate in a fixed position using:

$$j_{\theta i\ max} = \frac{360\, j_{bt\ max}}{\pi\, d_{bi}} \quad (201)$$

$$j_{\theta i\ min} = \frac{360\, j_{bt\ min}}{\pi\, d_{bi}} \quad (202)$$

where

$j_{\theta i\ max}$ is angular backlash of the pinion or gear, maximum, degrees;

d_{bi} is base diameter of the pinion or gear, mm;

$j_{\theta i\ min}$ is angular backlash of the pinion or gear, minimum, degrees.

NOTE: The maximum and minimum angular backlash of the pinion differs from those of the gear, the ratio of the gear angular backlash to the pinion angular backlash is equal to the gear ratio.

12.7 Variations in backlash

The backlash is not constant; it varies with rotation of the gears. Variations can be due to eccentricity, differences in pressure angle between the two gears, pitch errors, helix errors, profile errors, etc. If a single flank test is done first with one set of flanks in contact and then with the other set in contact, subtracting one trace from the other should give the backlash as a function of rotation (see Clause 6). For a complete picture, the gears should be rotated a sufficient number of times so that all possible tooth pairs have gone through mesh.

Annex A
Tooth thickness measurement using analytical machines

[The foreword, footnotes, and annexes are provided for informational purposes only, and should not be construed as a part of AGMA 2002-DXX, *Tooth Thickness and Backlash Measurement of Cylindrical Involute Gearing*.]

Functional tooth thickness can be measured using analytical machines. The measurement must be done relative to the datum axis for the results to have meaning.

There are generally three different types of measurement machines that are used to measure the circular pitch or index on a gear. Each type of machine has its strengths and limitations in measuring tooth thickness on a gear. These types include:

– CNC gear measuring machines with a precision rotary table and specialized gear measurement software; they are designed specifically for measurement of gears.

  These machines use a generative method to create an involute. Coordinated rotational movement of the gear and linear movement of the probe positioned along a line tangent to the base circle will naturally trace an involute relative to the gear. This type of machine can be used to measure functional transverse circular tooth thickness directly, provided they have a properly calibrated ball stylus, a defined gear datum axis and correction for the probe tip contact position and spacing errors. These machines should measure the datum surfaces to establish the datum axis and correct for any deviation of that axis from the rotary table axis. At the specified measurement diameter, similar to the way transverse circular pitch can be measured directly based on the rotation angle between two consecutive left or right flanks, the functional tooth thickness can be measured based on the rotation angle between the left and right flanks of each single tooth. The average, maximum and minimum values of functional tooth thickness (either normal or transverse) are commonly reported.

– **3-axis co-ordinate measuring machines (CMM) with a precision rotary table and specialized gear measurement software.**
  These machines generally take point to point measurements, involute shapes are calculated. They might perform the measurement in a similar manner as the CNC machine, finding the circular arc length between points on opposite flanks of the gear teeth. However, if they are programmed to measure the straight line distance between points, then they are measuring chordal thickness and do not fall into this measurement by pitch category. See Clause 11 for chordal measurement.

– **3-axis co-ordinate measuring machines (CMM) with software adapted for gear measurement.**
  Depending on how the software is written, these machines may or may not properly measure circular pitch or index. If programmed to calculate a chordal straight line tooth thickness measurement, then they do not fall into the measurement by pitch category. See Clause 11 for chordal measurement.

  NOTE: The calculation of the straight line distance between points on opposite flanks of a tooth is discouraged. The circular arc distance between the points can be calculated as the angle, in radians, between the points times the radius from the datum axis to the points.

Use of CMM machines for gear inspection require both sufficient repeatability and measurement resolution to measure the feature of interest. Ensure that the method of data collection and reduction to result allows for the successful measurement of tooth thickness. It is common for the software of both CNC and CMM machines to convert the measured functional tooth thickness result to an estimate of indirect methods such as span, dimension over balls/pins (between balls/pins on internal gears) or chordal thickness. Conversion to one of the indirect methods is primarily done because this is what is specified on the gear drawing. It should be recognized that the measuring machines only approximate the expected measurements based on the measured circular tooth thickness at the measurement diameter. Since these other methods are nominal in nature (i.e., not related to the datum), the software needs to take into account the difference between the center of a best fit pitch cylinder and the datum axis to properly estimate the nominal tooth thickness measurement.

If measurement results from an indirect method are compared to the results from a CNC gear measuring machine or CMM, significant differences are often noted. The primary causes of these differences are:

– Indirect tooth thickness measuring methods, such as over balls, span, or chordal, measure nominal tooth thickness, whereas a CNC or CMM, when properly set up, will measure a functional tooth thickness. Note that the amount of runout is often a main contributor to the difference between a nominal and functional measurement.

– Improper setup can lead to measurement errors:
  - The CMM or CNC gear measuring machine probe needs to be accurately calibrated for size and position relative to the gear datum axis;
  - Runout of the datum axis will result in measurement errors.
– The contact position for the measurements on the CNC or CMM machine is defined by the pitch measurement diameter and this can be different from the contact diameter for dimension over pins or span dimension flank contact positions. Each method is susceptible to different helix, profile and pitch errors, and also susceptible to deliberate flank corrections, such as crowning and tip and root relief, which changes the tooth thickness value.
– Chordal thickness measurements often use the tip diameter as a datum and this is rarely measured as part of the tooth thickness measurement process on CNC and CMM machines.

It is recommended that the following steps be taken to minimize tooth thickness measuring errors using CNC machines and CMMs:

– Calibrate the probe frequently. The maximum interval should be defined by testing because there are many influence factors, but at least once per working shift is recommended.
– The effects of removing the probe and replacing it (without re-calibration) should also be checked. Many manufacturers claim that there is no need to re-calibrate the stylus but this should be verified by the user.
– When it is known that a CNC or CMM will be used for measurement, then it is recommended that drawings specify circular tooth thickness as the primary measurement method.

It is recommended that runout compensation only be used to compensate for the position of the datum surfaces in relation to the measuring machine, and not for eccentricity in the gear. While it is possible to determine a pitch axis for a gear, this should only be used for analysis and correction of the machining setup, or for estimation of one of the indirect methods of tooth thickness measurement. The pitch axis is that axis from which there is minimum runout of the gear tooth flanks. Ideally the pitch axis should be coincident with the datum axis; when it is not coincident the gear will exhibit eccentricity. Confirm the method of runout compensation if present. The reported values of tooth thickness should include the effects of runout in relation to the datum surfaces. The reporting method shall be consistent with the specification.

– When gears are to be measured both with a measuring machine and with actual balls or pins, ensure that the pitch measurement diameter is the same as the contact diameter for the specified measuring ball diameter.
– During initial installation of the machine and after any software upgrades, the accuracy of the measuring machine for tooth thickness measurement should be independently verified with a calibrated tooth thickness artifact which is similar to the manufacturer's product gear geometry.