Introduction

This paper describes the concept of approach and recess action in gear meshes. The transfer of torque and rotational speed between interacting tooth surfaces is divided into two distinctive sections: approach and recess. The difference between the two sections is based on the direction of the friction force and sliding velocity vectors. For an unmodified set of gears, the only way the approach and recess sections can be equal is if the gear ratio is 1:1. For an unmodified speed reducing gear mesh, the approach section will be longer than the recess section. For an unmodified speed increasing gear mesh the approach section will be shorter than the recess section. Certain geometric values of the gear teeth can be modified in order to adjust the distances covered in each of the sections during approach and recess, leading to a change in the maximum relative sliding velocity. Designing a gear mesh with nonstandard proportions (approach and recess modifications) affects the system's pitting resistance, wear, scuffing, contact flash temperature, bending resistance, gear mesh efficiency, and noise. Although there are several approaches to modifying a gear mesh, a good gear design should not have more recess than approach action. In case of a gear mesh containing a series of idlers, the tooth proportion modifications may not be possible. This is directly related to the fact that a given idler is both driving and driven at the same time. This makes it difficult, or even impossible, to ensure that each mesh is recess-dominant. The same applies to the gear-sets with reversing direction of rotation.

While no specific recommendation can be made regarding the amount of recess and approach action in a given mesh, the designer should be aware of the influence that recess and approach action have on the functional parameters listed above.

Approach and recess defined

In meshing gears, a uniform transfer of torque is achieved as the active surfaces (involute profiles) of both members interact with each other. In a conventional gear design the pitch line divides the tooth into two sections called addendum and dedendum respectively. At the beginning of mesh, the portion of the driver's tooth near the root (the area defined by the involute function adjacent to the point where the root curvature starts) makes initial contact with the tip of the driven member. Contact then progresses until it reaches the tip of the tooth of the driver and near the root of the tooth of the follower. The action from initial contact to the pitch point (no sliding point) is known as **approach action** because this takes place ahead of the pitch point. The remainder of tooth travel, from the pitch point until the tooth leaves the engagement is known as **recess action.** As the members rotate, the point of contact between teeth in mesh travels along a straight line that passes through the pitch point. This line is known as the "line of action". (Refer to Figure 1.)

Equations

Gear tooth sliding velocity is defined as the difference between rolling velocities of teeth in mesh.

At a given point on the line of action, the product of the radius of curvature and rotational speed, calculated for pinion and gear respectively, are not equal, and therefore, the resultant rolling velocities are different. The pitch point, the location where the pitch circle intersects the line of action, is the only contact point between meshing teeth where there is pure rolling. At any other contact point during the interaction of tooth profiles, there is relative sliding. The amount of sliding increases as the contact point moves away from the pitch point. (Refer to Figure 2)

Rolling velocities at any point along the line of action (except for the pitch point): $v_{RpA} \neq v_{RGA}$

Rolling velocities at the pitch point: $v_{RpA} = v_{RGA}$

$$v_{RpA} = \frac{2\pi\rho_{pA}}{60} * n_P$$

$$v_{RGA} = \frac{2\pi\rho_{GA}}{60} * n_G$$

Where: $v_{RpA} -$ *rolling velocity at an arbitrary point* A, *pinion* [*in./sec.*]

$\rho_{pA} -$ *radius of curvature at point* A, *pinion* [*in.*]

$v_{RGA} -$ *rolling velocity at point* A, *gear* [*in./sec.*]

$\rho_{GA} -$ *radius of curvature at point* A, *gear* [*in.*]

$\phi_A -$ *pressure angle at point A*

Sliding velocity

For conjugate gear teeth (helical & spur gears), at an arbitrary point on the involute curve, sliding velocity is expressed as the difference of the two rolling velocities (gear and pinion) and can be determined using the following equation:

$$v_s = v_{RpA} - v_{RGA}$$

Where: $v_S -$ $sliding\ velocity\ [in./sec.]$
$v_{RpA} -$ $rolling\ velocity\ at\ an\ arbitrary\ point\ A, pinion\ [in./sec.]$
$v_{RGA} - rolling\ velocity\ at\ point\ A, gear\ [in./sec.]$

Process properties

The properties of approach and recess are quite different because of the process's dynamics
In a gear set with teeth of standard proportions, the length of the line of action is not equal on the approach and recess sides. For a speed reducing gear set the highest sliding velocity is on the start of the approach side. The direction of the sliding component of the action during approach is negative because the friction component of force is in the opposite direction of the working force vector; it opposes, rather than helps the action. During recess, the friction vector is in the same direction as the working force, aiding the rotation of the driven member. If no modifications are made to gears' tooth proportions, a speed reducing gear set will experience approach action and speed increasing gear set (larger member being the driver) will experience recess action.
By balancing the approach and recess, the sliding velocity at the start of approach is lowered and the sliding velocity at the end of recess is increased to make them equal. In the case of a gear design with recess action, the sliding velocity is greatest at the end of recess action rather than at the start of approach action.

A gear expert, Earle Buckingham claims that approach action has definite disintegrating effect on the tooth surface as compared with the recess action. According to Earle Dudley, more recess action on a driving member in a gear mesh would make it less susceptible to inaccuracies in the gears. Such a gear mesh would run more smoothly and quietly.

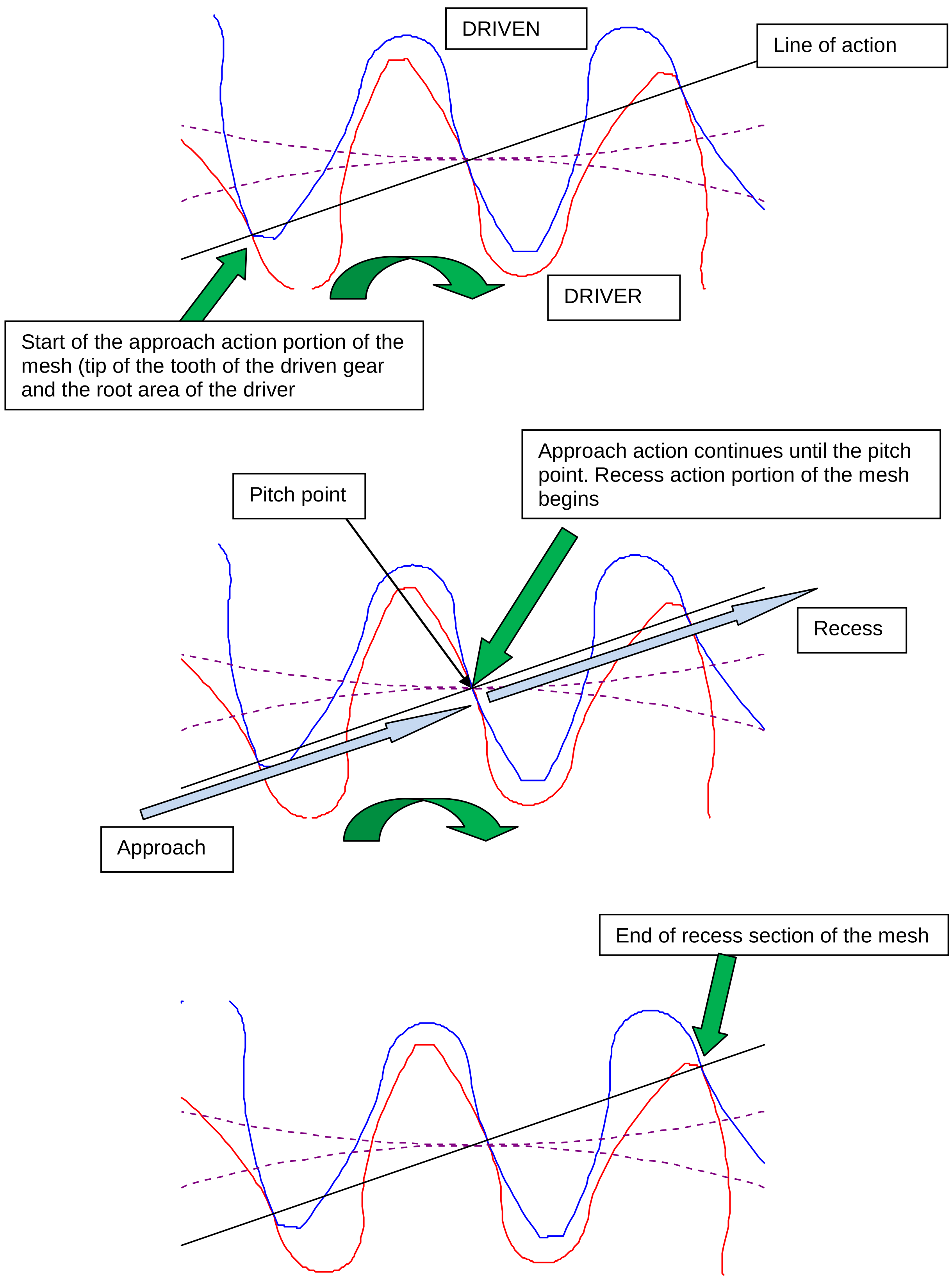
Techniques used to modify the recess and approach action

1. Tool shift - change in the pitch tooth thickness (Addendum modification) which also changes the outside diameter
    a. When equal positive and negative addendum modifications are applied to the pinion and gear respectively, there is no change to center distance. When unequal amounts of addendum modifications are applied to the driver and driven gears, the center distance changes by half the sum of the addendum modifications.
    b. The application of positive addendum modification to the driver (which increases its outside diameter) and negative addendum modification to the driven gear (reduces its outside diameter) produces an reduction in the amount of approach action and an increase in the amount of recess action

2. "OD change"- gear outside diameter modification without a change in tooth thickness at the pitch diameter
    a. Increasing the outside diameter of the driving gear without changing the tooth thickness at the pitch diameter reduces the approach action. Reducing the outside diameter of the driving gear without changing the tooth thickness increases the approach action.
    b. Increasing the driven gear outside diameter without changing the tooth thickness reduces the recess action. Reducing the driven gears outside diameter without changing the tooth thickness increases the recess action.

One should also be aware that many other gear parameters, including profile contact ratio and minimum tooth tip thickness, are affected when these methods are used to modify approach and recess action. When using addendum modification and OD change, a designer must weigh the effects on the overall gear design.

Fig. 1 Recess and approach action in gears



Generally, it's desirable to modify tooth proportions in order to decrease the negative effects associated with approach and promote the involute action during recess. Depending on the design requirements and other constraints, multiple solutions can be achieved based on the approach taken:

1. Balance flash temperature (scuffing resistance) and tip sliding velocity
   The goal, in this case, is to equalize the peak flash temperatures and sliding velocity at the high and low point of single tooth contact. This is generally achieved by balancing the approach and recess action. This approach is more common in very high speed gearing.
2. Balance bending strength (fatigue life)

When the shape of the trochoid of a pinion is considerably different from that of a gear, the pinion's tooth may be weaker than the gear's tooth. A positive profile shift is applied to the pinion and a negative profile shift to the gear. In this case, the pinion's tooth becomes stronger and the gear's tooth weaker, resulting in balanced bending strength. This technique is generally achieved for a speed reducing gear set by having more recess action than approach action.

In certain cases, a design optimization can't be achieved to the degree desired. A large number of gears in a given gear train can limit the extent to which the gears can be modified. This is due to the fact that idlers, within the string of gears in series, mesh with two members simultaneously, limiting the amount of profile shift that can be applied.
The other example would be a gear set operating in both directions of rotation, in which case approach and recess action should be balanced.
Profile shift is also used to modify gears to avoid undercut due to low tooth counts. This and the other reasons listed may conflict with one another. The gear designer must decide which parameters are most important for a specific design.

References:

- AGMA 913-A98 Methods of Specifying the Geometry of Spur and Helical Gears
- AGMA 917-B97 Design Manual for Fine Pitch Gearing
- Review of Gear Efficiency Equation and Force Treatment
  JSME International Journal. Series C, Vol.40(1997)1,p1-8 Tsuneji YADA
- "Manual of gear design" Earle Buckingham & Eliot K. Buckingham

Fig. 2 Geometric relationships of an involute tooth profile.