Bevel gears are essential components in the transfer of motion between intersecting axes. Typically, they are used in situations where the direction of a shaft's rotation needs to be changed. They are characterized by their conical shape, allowing them to mesh at various angles, most commonly at a 90-degree angle. This calculator focuses on simplifying the complex calculations involved in designing bevel gears, making it easier for engineers, designers, and hobbyists tospecify the necessary parameters for their gear systems.Historical BackgroundThe development of bevel gears can be traced back to the mechanics of ancient machinery, where they were used to change the direction of motion in mills and other mechanical devices. Over time, their design and application have evolved significantly, with modern usage ranging from automotive differentials to complex machinery in industrial applications.Calculation FormulaThe formulas provided are crucial for determining the specific dimensions and attributes of bevel gears. They include calculations for pitch diameter, whole depth, addendum, dedendum, and more, based on the diametral pitch (P) and the number of teeth (N) on both the gear and the pinion.Example CalculationFor a pinion with 15 teeth and a gear with 30 teeth at a diametral pitch of 50 teeth per inch, the calculations would yield specific dimensions for pitch diameters, addendum, dedendum, etc., providing a comprehensive overview of the gear's specifications.Importance and Usage ScenariosBevel gears are crucial in various applications, from automotive to industrial machinery, where the transmission of power at an angle is necessary. Understanding the calculations and specifications of these gears is vital for the design and functioning of such mechanisms.Common FAQsWhat is diametral pitch?Diametral pitch (P) is a measure of the fineness of a gear, defined as the number of teeth per unit of the pitch diameter.How do you calculate the pitch diameter?The pitch diameter (D) can be calculated using the formula \(D = \frac{N}{P}\), where \(N\) is the number of teeth and \(P\) is the diametral pitch.What determines the size of a bevel gear?The size of a bevel gear is determined by its diametral pitch, number of teeth, and the angle between the gear and pinion, which influence the pitch diameter, face width, and other parameters.This calculator streamlines the process of designing and specifying bevel gears, catering to the needs of professionals and enthusiasts in the field of mechanical engineering and design. The calculation is designed for geometric designs and checks of bevel and hypoid gear with straight, oblique and curved teeth according the ISO 23590 The programme gives solutions to the following tasks: Preliminary design of the gear size Detail geometrical design for gear: - Straight - Oblique - Spiral - Zerol - Hypoid (Gleason, Oerlikon, Klingelnberg) Automatic design of a transmission with the minimum number of input requirements. Generation of standard 2D drawings. Precision 3D model of the bevel gearing. Precision 3D model of the hypoid gearing. Support for all CAD systems. The calculations use procedures, algorithms and data from ISO 23509 and related standards AGMA ISO 23509 : A, ANSI, ISO, DIN, BS and specialized literature. Hint: The comparative document "Choices of transmission" can be helpful when selecting a suitable transmission type. User interface. Download. Purchase, Price list. Control, structure and syntax of calculations. Information on the syntax and control of the calculation can be found in the document "Control, structure and syntax of calculations". Information on the project. Information on the purpose, use and control of the paragraph "Information on the project" can be found in the document "Information on the project". Theoretical foundations. Estimated pinion / gear size (ISO 23509): Estimation is for a gear with a shaft angle of 90. For other axis angles, the estimation must be corrected. Pinion torque T1 Pinion torque is a convenient criterion for approximate rating of bevel gears, requiring conversion from power to torque by the relation: T1 = 9550 * P / n1 where: T1 ... pinion torque [N*m] P ..... power [kW] n1 ... pinion speed [/min] Pinion outer pitch diameter de1 Pinion outer pitch diameter versus pinion torque, transmission ratio, material, gear type, load type...: Pitting resistance de1 = (0 + 15 * T1^0.325) / (u^0.35) * KM * SZF * PFG [mm] Bending strength For T1 < 5000 [N*m] de1 = (0 + 14 * T1^0.35) / (u^0.5) * LTF * SZF [mm] For T1 >=5000 [N*m] de1 = (0 + 14 * T1^0.35) / (u^0.5) * u^0.14 * (T1/5000)^0.1 * LTF * SZF [mm] where: de1 ... Pinion outer pitch diameter [mm] T1 .... Pinion torque [N*m] u ...... Transmission ratio KM .... Material factor SZF.... Straight and zerol bevel factor PFG ... Precision-finished gears LTF ... Load type factor Preliminary hypoid pinion pitch diameter deplm1 For hypoid gears: deplm1 = de1 - a / u where: deplm1... Preliminary hypoid pinion pitch diameter [mm] de1 .... Pinion outer pitch diameter [mm] (larger value from previous calculation) a .... Hypoid offset [mm] u .... Transmission ratio Geometry All used geometry formulas can be found in ISO 23509. Below is the geometry nomenclature. Bevel gear nomenclature - Axial plane back angle back cone angle back cone distance clearance - c crown point crown to back dedendum angle - Thetaf1, Thetaf2 face angle - deltaa1, deltaa2 face width - b front angle mean cone distance - Rm mean point mounting distance outer cone distance - Re outside diameter - dae1, dae2 pitch angle - delta1, delta2 pitch cone apex crown to crossing point - txo1, txo2 outer pitch diameter - de1, de2 root angle - deltaf1, deltaf2 shaft angle - Sigma equivalent pitch radius mean pitch diameter - dm1, dm2 pinion wheel Bevel gear nomenclature Mean transverse section whole depth - hm pitch point clearance - c circular thickness circular pitch chordal addendum chordal thickness backlash working depth - hmw addendum - ham dedendum - hfm equivalent pitch radius Hypoid - nomenclature face apex beyond crossing point - tzF1 root apex beyond crossing point - tzR1 pitch apex beyond crossing point - tz1 crown to crossing point - txo1, txo2 front crown to crossing point - txi1 outside diameter - dae1, dae2 outer pitch diameter - de1, de2 shaft angle - Sigma root angle - deltaf1, deltaf2 face angle of blank - deltaa1, deltaa2 wheel face width - b2 hypoid offset - a mounting distance pitch angle - delta2 outer cone distance - Re pinion face width - b1 Bevel gear tooth tapers depth slot width thickness space width Root line tilt Bevel gear depthwise tapers A. Standard depth taper B. Constant and modified slot width C. Uniform depth mean whole depth mean addendum mean dedendum Machining Principle. Two basic principles of machining are primarily used for the calculation and generation of the 3D models of the gear wheels. A. For face hobbed gears The curvature is an extended epicycloid (locus), which is created by rolling the tool on the basic circle of the epicycloid. B. For face milling gears The curvature radius is equal to the radius of the tool. Input values for selected methode Symbol Symbol Description Method 0 Method 1 Method 2 Method 3 Sigma S Shaft angle X X X X a a Hypoid offset 0.0 X X X z1, z2 z1, z2 Number of teeth X X X X dm2 dm2 Mean pitch diameter of wheel - - X - de2 de2 Outer pitch diameter of wheel X X - X b2 b2 Wheel face width X X X X betam1 bm1 Mean spiral angle of pinion - X - - betam2 bm2 Mean spiral angle of wheel X - X X rc0 rc0 Cutter radius X X X X z0 z0 Number of blade groups (only face hobbing) X X X X Process of calculation. This calculation is intended primarily for geometry design. The calculation or a gear size design, respectively, is only indicative and needs to be checked with respect to the relevant standard or the documentation supplied by the manufacturer of the specific machining centre. Gear dimension design: Enter the power and operating parameters of transmission (transmitted power, rotation speed, type of gears, material, type of load, ...). [1] Enter the gear ratio or the number of teeth, respectively; the angle of shaft axes angle; the shaft offset; and the toothing width and angle [2] Specify the tooth profile parameters. [3] Based on the type of toothing, select the relevant method of calculation 0, 1, 2, or 3. [4,5,6,7] Using the relevant method, enter the input geometric parameter or run iteration of the calculation. Check the results. Use the graphic representation in paragraph [9.0]. If necessary, load the 2D drawing into the 2D CAD system, or generate data for creation of a precision 3D model. Options of basic input parameters. [1] Enter basic input parameters of the designed gearing in this paragraph. 1.1 Calculation units. Use the drop-down menu to choose the required system of units for calculation. After the units are switched, all values will immediately be re-calculated automatically. 1.2 Transferred power. Enter the power to the driven gear. Usual values are in the range 2 - 500 kW / 3-700 HP, in extreme cases up to 4000 kW / 6000 HP. 1.3 Speed (Pinion / Gear). Enter the speed of the driven gear. Extreme speed can reach 50 000 rpm. The speed of the driven gear is calculated using the number of teeth of both gears. 1.4 Torsional moment (Pinion / Gear). This is the result of the calculation and cannot be entered. 1.5 Gearing type This calculation allows users to solve a choice of gearing types. Select the type of toothing from the list. By making this selection, you also select a range of coefficients which affect the estimate of the gear assembly size and the coefficients for geometry calculations. Standard straight toothing - Methode 0 Standard oblique toothing - Methode 0 Spiral bevel gears (non hypoids) - Methode 0 Zerol bevels - Methode 0 Hypoid spiral bevel gears (Gleason) - Methode 1 Hypoid spiral bevel gears (Oerlikon) - Methode 2 Hypoid spiral bevel gears (Klingelnberg) - Methode 3 Note: To calculate the geometry, always use the relevant paragraph marked as Method 0, 1, 2 or 3. 1.7 Material of the pinion / gear For materials other than case-hardened steel at 55 minimum HRC, the pinion outer pitch diameter, as given in Figure B.2 ISO 23509, is to be multiplied by the material factor KM 1.9 Precision-finished gears. "Precision finish", refers to a machine finishing operation which includes grinding, skiving and hard cut finishing. 1.10 Load type. Statically loaded gears should be designed for bending strength rather than pitting resistance For statically loaded gears which are subject to vibration, the pinion outer pitch diameter, as given in Figure B.2 ISO 23509, is to be multiplied by 0,70. For statically loaded gears which are not subject to vibration, the pinion outer pitch diameter, as given in Figure B.2, B.3 ISO 23509, is to be multiplied by 0,60. 1.12 Accuracy grade - ISO1328 It only affects the design of the backlash. Preliminary / Approximate design of geometrical parameters. [2] In this paragraph, select the gear ratio, numbers of teeth, angle of axes, offset of axes, etc. Then you obtain a preliminary design of the size and shape of toothing. - The entered number of teeth and the angle of axes are used for all the methods of the geometry calculations. - The diameters, the axes offsets, the width of toothing, and the angle of toothing are then offered in a specific paragraph (Methods 0 to 3) as the recommended values (minimum and maximum). 2.1 Transmission ratio / from table The optimum transmission ratio varies in the range 1-5. In extreme cases this ratio can reach up to 10. The transmission ratio can be entered in the left input field using the keyboard. The right pop-up list contains recommended values of the transmission ratio and when selecting a value from this list, the chosen value is added to the field on the left automatically. 2.2 Recommended (minimum) number of teeth pinion / wheel The recommended (and minimum) number of pinion teeth is based on the ISO 23590 standard. The recommended number of the gear teeth is then calculated based on the gear ratio entered on the previous line. 2.3 Number of teeth pinion / wheel Enter the number of pinion teeth. The number of gear teeth is calculated based on the required gear ratio [2.1]. After checking the checkbox on the right, it is possible to enter the number of the gear teeth directly. 2.4 Actual transmission ratio / deviation. As the actual transmission ratio is the ratio of the number of teeth of both gears (integers), the actual transmission ratio will be mostly different from the desired (entered) one. The value of the "Actual transmission ratio" is displayed on the left; the percentage deviation from the desired transmission ratio is displayed on the right. This deviation for the transmission ratio should be in the range: i = 1 to 4.5 .... 2.5% i is greater than 4.5... 4.0% Hint: In case you need to design gearing with a transmission ratio as accurate as possible or need to distribute the transmission ratio among more gears in the gearbox, use "Calculation of the transmission ratio". 2.5 Angle of shaft axes. Enter the angle of axes of individual wheels (mostly 90). The calculation also enables options of other values. The case when the angle of the pitch cone exceeds 90 is indicated by a red cell. (This creates conical gearing that cannot be produced on usual machines). 2.6 Hypoid offset / max. value (25% of de2) In most cases, the hypoid offset is determined by the application. Pinion offset is designated as being positive or negative. Picture illustrates positive and negative pinion offsets as seen from the wheel apex. A. Left-hand pinion mated with right-hand wheel B. Right-hand pinion mated with left-hand wheel It is recommended that positive pinion offset be used because of the increasing diameter of the pinion, higher face contact ratio, pitting and bending load capacity. ln general, due to lengthwise sliding, the offset should not exceed 25 % of the wheel outer pitch diameter and, for heavy-duty applications, it should be limited to 12,5 % of the wheel pitch diameter. 2.7 Pitch diameter (outer) Based on power and geometry inputs, the outer pitch diameter is estimated. 2.8 Outer transverse module / Outer transverse DP Since tooling for bevel gears is not standardized according to module, it is not necessary that the module be an integer. However, it is possible in calculation to calculate the wheel size according to the required module. 2.11 Face width / max. recommended value After clearing the button the value of the toothing width can be entered. Ticking the button automatically selects the maximum value. Generally, the face width is 30% of the cone distance Re2 or 10*met, whichever is less. However, design parameters may require values to be larger or smaller. For zerol bevel gears, the face width should be multiplied by 0.83 and should not exceed 25 % of the Re2. For shaft angles less than 90 the face width larger than given may be used. For shaft angles greater than 90 the face width smaller than given should be used. The hypoid pinion face width is generally greater than the face width of the wheel. 2.12 Mean spiral angle Common design practice suggests that the spiral angle be selected to give a face contact ratio of approximately 2.0. For high-speed applications and maximum smoothness and quietness, face contact ratios greater than 2.0 are suggested, but face contact ratios less than 2.0 are allowed. For hypoid sets, the pinion spiral angle may be calculated using Formula: betam1 = 25 + (z2 / z1)^0.2 + 90 * a / de2 where: z1, z2 .... Number of teeth a .... Hypoid offset [mm] de2 .... Gear outer pitch diameter [mm] Initial data for tooth profile parameters (used in all calculation methodes) [3] Enter basic input parameters for tooth profile in this paragraph. 3.1 Nominal design pressure angle drive side / coast side There are three normal pressure angles that are to be considered. - Nominal design pressure angle alfad, is the start value for the calculation. It may be half of the sum of pressure angles or different on drive and coast side. - Generated pressure angle alfan, is the pressure angle of the generating gear; alfan can be found on the tooth flank in the mean normal section. - Effective pressure angle, alfae, is a calculated value. The most commonly used design pressure angle for bevel gears is 20. This pressure angle affects the gear design in a number of ways. Lower generated pressure angles increase the transverse contact ratio, reduce the axial and separating forces and increase the toplands and slot widths. The converse is true for higher pressure angles. Based on the requirements of the application, the engineer may decide to choose higher or lower design pressure angles. Lower effective pressure angles increase the risk of undercut. For hypoid gears, it could be reasonable to have unequal generated pressure angles on the coast and drive sides, in order to balance the mesh conditions. If full balance of the mesh conditions is recommended, the influence factor of limit pressure angle, falfalim, is set to "1". Then the limit pressure angle, alfalim, is added to the design pressure angle alfad, on the drive side and subtracted on the coast side in order to obtain the generated normal pressure angle, alfan. As for bevel (non-hypoid] gears, the limit pressure always is equal to zero, the nominal design pressure angles have the same values as the generated pressure angles. If the effective pressure angles have the same values, the mesh conditions on coast and drive side are equal. Straight bevels To avoid undercut, use a nominal design pressure angle of 20 or higher for pinions with 14 to 16 teeth and 25 for pinions with 12 or 13 teeth. Zerol bevels On zero bevels 22.5 and 25 nominal design pressure angles are used for low tooth numbers, high ratios, or both, to prevent undercut. Use a 22.5 nominal design pressure angle for pinions with 14 to 16 teeth and a 25 nominal design pressure angle for pinions with 13 teeth. Spiral bevels To avoid undercut a 20 design pressure angle or higher for pinions with 12 teeth or fewer teeth may be used. Hypoids To balance the mesh conditions on coast and drive side, the influence factor of limit pressure angle should be falfalim = 1. For the use of standard cutting tools, the value of falfalim may be different from "1". The nominal design pressure angles 18 or 20 may be used for light-duty drives; higher pressure angles such as 22.5 and 25 for heavy-duty drives. 3.4 Data type selection (I or II) Two alternative sets of input data, those used by American Gear Manufacturers Association AGMA - (data type ll) and those used in European Standards - (data type l) be available to calculate one or more of the four design methods shown. After selecting the type, the related values will also be calculated for the second set of input data and displayed in the right column. The data in the right column are then used for the calculation. In the green column, the recommended values resulting from the previous data (toothing type, gear ratio, number of teeth) are shown. 3.5 Profile shift coefficient Recommended values for gearings with the angle of axes 90: Gear ratio / xhm1 1..............0.00 1.12..........0.10 1.25..........0.19 1.6............0.27 2...............0.33 2.5............0.38 3...............0.40 4...............0.43 5...............0.44 6...............0.45 3.6 Basic crown gear addendum factor In common cases, the addendum factor khap is set to khap = 1.00 and the dedendum factor khfp is set to khfp = 1.25. 3.7 Basic crown gear deddendum factor In common cases, the addendum factor khap is set to khap = 1.00 and the dedendum factor khfp is set to khfp = 1.25. 3.8 Thickness modification coefficient (theoretical) Adjust the value of the unit change of the tooth thickness here. Recommended values for gearings with the angle of axes 90: Gear ratio / xsmn 10.0 1.120.010 1.250.018 1.60.024 20.030 2.50.039 30.048 40.065 50.082 60.100 3.9 Mean addendum factor of wheel This factor apportions the working depth between the pinion and wheel addendums. The pinion addendum is usually longer than the wheel addendum, except when the numbers of teeth are equal. Longer addendums are used on the pinion to avoid undercut. Suggested values for shaft angles Sum = 90 for cham are on the right. Other values based on sliding velocity, topland or point width limits, or matching strength between two members, can be used. Clause8 from ISO 23509 gives the limits for the mean addendum factor to prevent undercut on pinion and wheel. 3.10 Depth factor Normally, a depth factor kd = 2.000 is used to calculate mean working depth hmw, but it can be varied to suit design and other requirements. The value on right gives the suggested depth factors based on pinion tooth numbers. 3.11 Clearence factor While the clearance is constant along the entire length of the tooth, the calculation is made at mean point. Normally, the value of 0.125 is used for the clearance factor kc, but it can be varied to suit the design and other requirements. 3.12 Circular thickness factor The mean normal circular thickness is calculated at the mean point. Values of kt based on balanced bending stress are on the right. Other values of kt may be used if a different strength balance is desired. Results section [4,5,6,7] The following paragraphs describe the above-mentioned methods of calculation corresponding to the selected type of toothing [1.5]. Expand the corresponding method and specify the input parameters of toothing. The recommended values based on the draft design (see par. 1.0, 2.0, and 3.0) are shown on the right side. When using the methods 1, 2 and 3, it is usually required to perform an iteration to finish the calculation of the geometric results. Using the '