Gear

v1.02

to **Kaveh**,
My “bestest” friend

DISCLAIMER
Gear v1.02
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Introduction.
Gear design is a relatively simple but comparatively tedious design task any mechanical engineer may encounter in their career. Simple because it's based on simple Lewis and Buckingham stress equations for bending and contact stress respectively; and tedious because, like virtually any other design task, it involves several iterations. Due to the presence of several correction factors applied to Lewis and Buckingham stress equations, and the correction factors required for correcting strengths, gear design can become quite unwieldy without a computer.
The purpose of this program is to help the mechanical engineer in this design process by providing them with a tool for designing *spur*, *helical* and *bevel* gears. It is based on AGMA stress and strength equations and is programmed using the material presented in the well-known book, "Mechanical Engineering Design," Joseph E. Shigley et al.

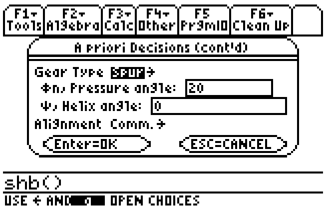
Installation
Create a folder named *gear* on your TI-89 handheld and send all of the *.89? files to that folder. (Remember, to choose the destination folder, you have to uncheck the *Retain Folder* option in the *Send files to TI-89* window.) Do not rename any files, or the program will not function properly if it does function at all.
You may run the program by going to the folder *gear* and typing shb(). (shb stands for Spur Helical Bevel; as you might have guessed already.)
If you opt to archive the program files, be careful *not* to archive the variables left in the program folder; the program needs to write new values to them. I could have defined those variables as local but that way for the next run, which may be only slightly different from the current run, you'd had to re-enter all the parameters. Another alternative was to create a file into which those variables could be written; pretty much like the data files used by FEM solvers. However, I wrote this program for my Machine Design exam when I was studying my BS, and I didn't have time for adding any bells and whistles at that time. If suggestions prove the need, I may add them in future versions.

I haven't tested this program on any handheld other than TI-89 but I suppose it should function with no problem on TI-89 Titanium and Voyage too.

Notation description
The program dialog boxes are well commented so herein we elaborate on only those points that really need discussion.
Life
N, Life in cycles, actually refers to the desired life of the pinion. Most books don't make out the difference between the life of the pinion and that of the gear. Since, according to the definition, the pinion is the smaller of the two gears in mesh, it always sees more loading cycles, both in bending and surface pitting. This results in its life getting less than that of the gear. This point is taken care of in the program and plays an important role when we are dealing with high reduction ratios. In such cases, bending strength stress-cycle factor and pitting resistance stress-cycle factor differ somewhat for the pinion and the gear. In short, you

type in the desired life for the pinion (in cycles of rotation), N, and the program computes that of the gear.

Normal vs. Transverse



The subscript *n* refers to *normal* everywhere in the program prompts. For instance, when the program requests the pressure angle, refers to it as ϕ_n meaning that it expects you to enter the normal pressure angle. For spur and straight bevel gears, this is of no concern, since we have only one type of quantities for them, be it referred to as *normal* or *transverse*. For helical gears, careful attention is required as to which type of quantity, normal or transverse, is requested. So ϕ_n, Pd_n refer, respectively, to the normal pressure angle and normal diametral pitch.

Alignment

Alignment dropdown menu has four choices: Open, Comm.(ercial), Precision, and Extra.(precision) It is used to select the empirical constants used in evaluating the load distribution factor, Km. The load distribution factor modifies the stress equations to reflect the non-uniform distribution of load across the line of contact. The ideal is to locate the gear "midspan" between two bearings at the zero slope place when the load is applied. However, this is not always possible, nor is it always desirable from the other aspects of design. Open gearing requires the most modification and hence are associated with it the largest empirical coefficients. Less modification is required as we select better alignment conditions, with the best being the Extra(precision) alignment. The mostly encountered alignment in gearing is Comm.(ercial) alignment. Except for the Open gearing, all other alignments apply to enclosed units.

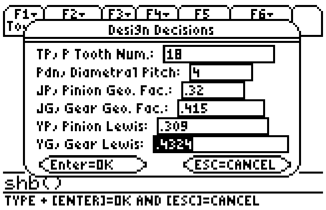
Helix angle

Helix angle applies only when the selected gear type is Helical. In the other two cases (spur and bevel gearing), its value is ignored.

TP

In contrast to the usual notation Np, used for referring to the pinion tooth count, we have selected TP to refer to this parameter only because nP was used before to refer to the pinion rotational speed.

Geometry Factors

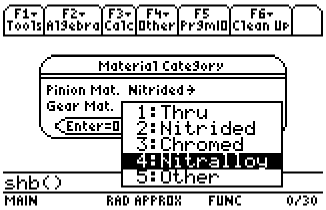


There are three kinds of geometry factors requested in the program prompts.

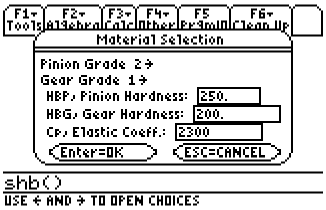
1. The first kind is J, bending-strength geometry factor. J can be obtained from ANSI/AGMA diagrams. Note that for helical gears AGMA represents a J' factor and a modifying factor to correct it. The multiplication of the two equals the J factor requested by the program for the case of helical gears.
2. The second kind is I, surface-strength geometry factor. This is only requested for bevel gears as it is internally evaluated for the other two types of gears, namely spur and helical gears.

3. The third and last kind is Y, the original Lewis geometry factor. This factor is used to evaluate the size factor Ks. AGMA suggests Ks=1, but following this suggestion in this manner is “a failure to bring all of your knowledge to bear,” in Shigley’s words. So we have provided the means for the program to calculate the size factors

Material Selection



In the material category window, you have five choices, which actually fall into two groups. You can either use any of the four predefined material types or select the fifth choice, *Other*, to type in your own material data. The first four choices are: *Thru* (referring to through hardened steel), *Nitrided* (referring to nitrided through hardened steel), *Chromed* (referring to 2.5% chrome steel), and *Nitralloy (*Nitralloy is a trademark of the Nitralloy Corp. New York, NY). The equations relating the core hardness to the allowable bending stress number for these four types are extracted from ANSI/AGMA 2001-C95, 2101-C95. Selecting any of these five material categories brings up a more detailed material selection window allowing you to choose either grade 1 or 2 for the selected material type.



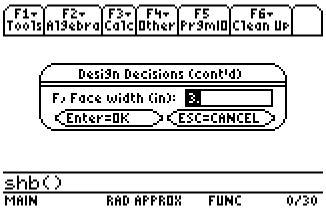
Even if you have selected “Other” to type in your own material strengths, you will still get this same window. In this case, the pinion and gear grades chosen in this dialog box will be ignored. Hardness is expected to be entered in Brinell (BHN, Brinell Hardness Number). In the case of using one of the four predefined material types, the entered HB would be used to calculate the pertaining strength in *bending* by feeding that HB to the appropriate equation. The equations relating HB to the *contact* strength are available only for *through hardened* steels (first choice in the material category window); for the other three choices, you have to type in the contact strength, Sc. If you select “Other” as the material type, you will be requested to type in the strengths of the pinion and/or the gear in bending, St, and in contact, Sc, *independently* of the respective harnesses you entered in the previous window. In this case, the hardness you entered is solely used to calculate the hardness ratio factor, CH. The pinion generally has a smaller number of teeth than the gear and consequently is subjected to more cycles of contact stress. If both the pinion and the gear are through hardened, then uniform surface strength can be obtained by making the pinion harder than the gear. A similar effect can be obtained when a surface-hardened pinion is mated with a through-hardened gear. The hardness ratio factor is used only for the gear. Its purpose is to adjust the surface strength for this effect.

Tips

- In spur and helical gears, a first guess for the face width can be made having in mind its range of variation with respect to the normal diametral pitch, 8/Pdn<F<16/Pdn. A reasonable initial guess would be F=12/Pdn which resides midway between the lower and upper limits of the permissible range. Some references suggest F=4π/Pdn. It really doesn’t matter which initial guess you use, because, after all, it’s only a guess; it’s most likely that you need to modify it in later iterations to obtain a satisfactory design.

- In bevel gears, bending strength is not linear with face width, because added material is placed at the small end of the teeth. Consequently, face width is roughly prescribed as
  F=min (A0/3, 10/Pdn), A0=(dG/3)*sin(Γ)
  and the nonlinearity is sidestepped. (Some designers use 0.3*A0 instead of A0/3.)

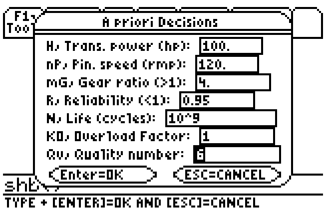
  These two tips are utilized in the program; so in the face width dialog box, you have an initial value pre-entered. You can either use that value or enter that of your choice. Being only a guess, it, most of the times if not always, needs to be modified in the next iterations.

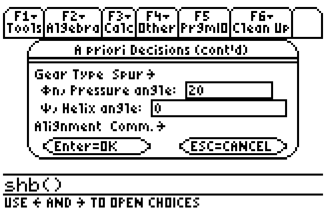


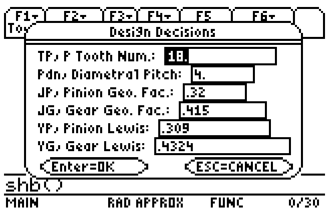
- If no more precise data is available, you may enter the same values for Y's as those of respective J's.
- Suppose you want to carburize and case harden grade 1 ASTM 1320 to
  Core 21 HRC (HB is 229 Brinell)
  Case 55-64 HRC (HB is 515 Brinell minimum)
  Since the selected material is not through hardened (i.e. it has different core and case hardness) you have to select *Other* for the material category and type in the appropriate St and Sc manually. (Currently the program does not allow you to enter different core and case hardness.)

Sample Problems

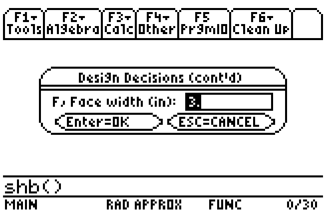
1. *Design a 4:1 spur gear reduction for a 100hp electric motor running at 1120rpm. The load is smooth, providing a reliability of 0.95 at 10^9 revolutions of the pinion.*



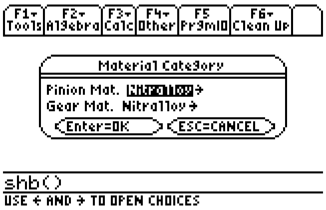




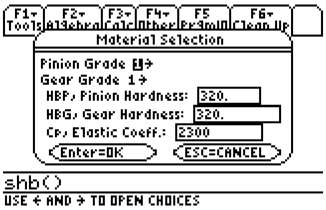
We have *assumed* a trial pinion tooth count, 18, and a normal diametral pitch of 4.



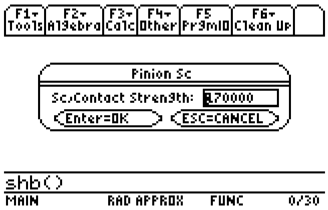
The trial face width calculated by the program. (We leave it intact.)



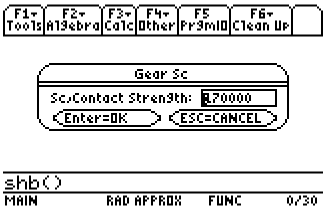
Nitralloy 135M for both pinion and gear.



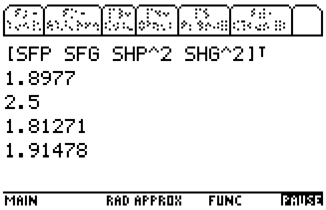
Grade 1 steel with 320 BHN for both pinion and gear.



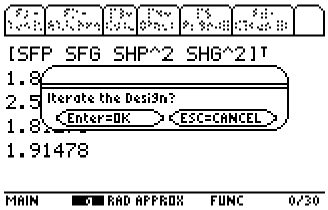
Contact strength for the pinion, 170 000 psi.



Contact strength for the gear, 170 000 psi.

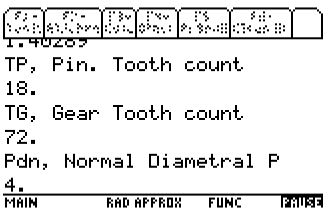


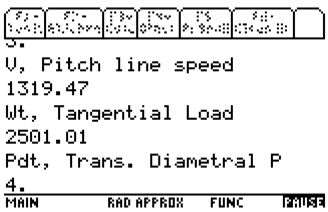
Resulting safety factors in bending and contact.

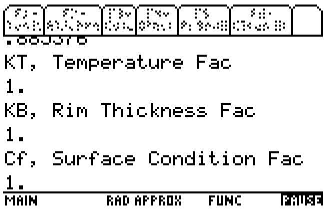


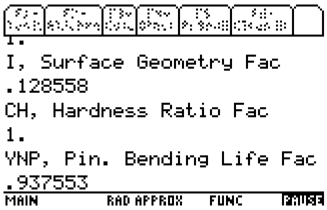
This window gives you two options. If the safety factors are satisfactory, you may press ESC to view the detailed "post processing" info of your design. Otherwise, you can press Enter to iterate the design.

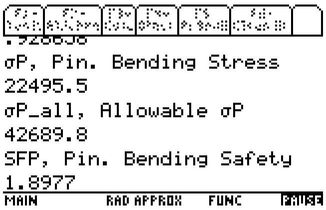
Note: Caution is required when comparing SF with SH in an analysis in order to ascertain the nature and severity of the threat to loss of function. Both SF and SH are strength over stress definitions but in the former the stress is *linear* with the transmitted load while in the latter the stress relation with the transmitted load is either *quadratic*, in the case of linear or helical contact, or *cubic*, in the case of spherical contact. That's why we have printed out SH^2's, not SH's, in the middle figure above. This way direct comparison is possible between the respective SH's and SF's.

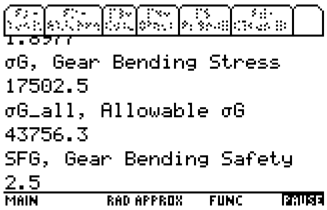








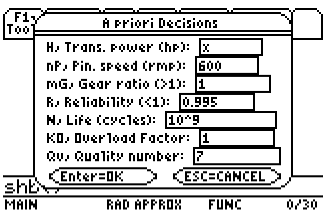


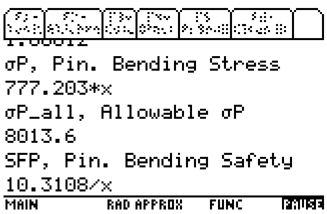


2. *A pair of straight tooth miter gears has a diametral pitch of 5 at the large end, 25 teeth, a 1.1-in face width, and a 20deg normal pressure angle; the gears have St=10 020 psi and Sc=85 000 psi. They have a quality number of 7. For a reliability of 0.995, a gear life of 10^9 revolutions, and a safety factor of 1.5, for both bending and contact, find the power rating for this gearset.*

We have presented this problem since finding the power rating of a gearset is a frequently encountered analysis task.

All we need do is enter the power as an unknown variable, say x, in the A Priori Decisions dialog box. Other parameters are entered as usual. Since the prescribed safety factor is 1.5, we set SFP=1.5 and SHP=1.5 and solve these equations for x. This yields,
x=6.87 from SFP Eq. and x=2.56 from SHP Eq. Hence H=min (6.87, 2.56) =2.56hp.





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Any suggestions or comments are highly welcome. Feel free to write to OmidLink@hotmail.com.