A No-Nonsense Guide to Mech 325 (Dog Edition)

by:

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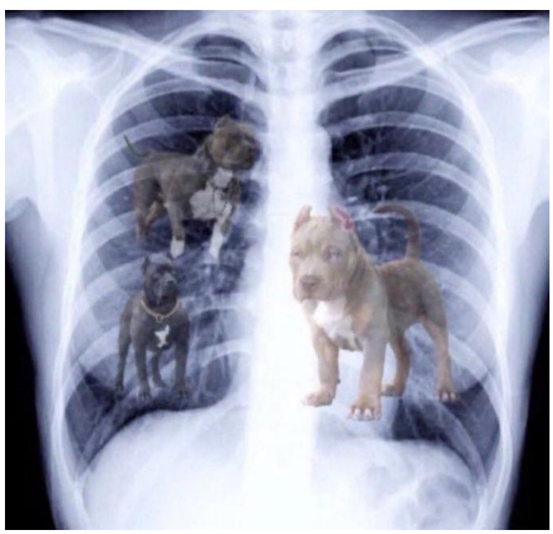
Note:

This guide is DOG certified. Please only proceed if you have that dog in you.



Version 1.1

Warning: Explicit Content, You May Lose Brain-cells

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1 Introduction

This document, in its entire vastness, is meant to help you anticipate the amazing Mech 325 and what is yet to come. In no shape or form will it save your ass on a Midterm or Final Exam, so if you are reading this for the first time on a test, you are more or less, screwed. Unless Jon has allowed communications on your test, you will have a PDF version of this document, which is completely useless. Anyways, it is more likely that you are reading this for the first time shortly after I have shared it with the DOGs on discord. I hope this document remains between us, but just in-case you are a cheeky DOG, like myself, I will allow you to share this with ONE friend. However, I expect that you will more or less keep this document within Mech (get a better grade and a better scale). Also, please uphold academic integrity while taking this course, as it is not that hard, and Jon will definitely catch you if you try cheating!

To keep things real with you, Mech 325 is an interesting course, and definitely more interesting than Mech 368, Mech 328, Mech 360 or probably whatever else you'll take during your degree (except Mech 358). The workload is definitely high, which is reflected by it being a 4 credit course. If you are interested in Salmon and Fishing, please make sure to attend the opening lectures as Jon is quite passionate on that subject.

You might be wondering what I would want in return for blessing you with this amazing document. Well, I would just like you to pray for my success in the future. I believe that is a fair trade for me helping you to succeed in this course. With this said, I wish you the best in Mech 325!

2 Disclaimer

Moving on, I will now explain what exactly this document will include and what it will not. Starting off, this document will:

- Display the potential struggles of Mech 325 and help you prepare for them
- Give you a good idea of what to expect from attending lectures and tutorials
- Help you do better in the course, by studying smarter not harder!
- Make sure you stay above the class average!
- Provide Access to an extensive google Drive with important documents
- Provide Access to the best online calculators on earth, made by yours truly

To be clear, none of what is mentioned above is 100% guaranteed. Although the exams are open book, your performance in this course will come down to how much effort you put in (i.e. I cannot carry your ass!). However, taking my advice, you will probably become more efficient at learning this course's content. To further assert my point, this document will NOT:

- Provide Solutions of any sort (Assignments, Tests, Tutorials, etc.)
- Help you do your Design Project[1]

In addition to all this, I cannot guarantee that the stuff I present are 100% correct. I could make mistakes, so I would appreciate it if you could please let me know if you find a mistake somewhere!

[1] Just a quick heads up, although you have the entire semester to fuck around and then finish this in a day or two, I would suggest starting this project ASAP (I know you won't anyways). Nevertheless, once started, the project will take around 2-3 days to complete, and you'll hand in a 50 page report that Jon will love. Just don't fuck around too much, or else you'll have to pull an all-nighter or two.

3 Section Overview

Now that I've formally explained to you what you should expect from this document, I hope that you do not put too much hope into the vast knowledge that I am about to throw at your head. Before moving on to the real shit, I will give you a breakdown of how I will write each section.

In each section, I will start off with a memory of class (or course in general) when this subject was taught. You can skip this portion if you want, as it won't contribute much to your understanding, but I am sure you are bound to experience something similar yourself.

I will then move on to the Do's and Don'ts, where I will tell you what to worry about and what is TRIVIAL[2].

In the next section, I will give you a no-nonsense guide to solving some of the relevant questions. In later topics, this section may not exist, and it will be YOUR job to learn it yourself and perhaps I will let you write it out yourself.

The final section, perhaps the most important, will be figures, definitions, and questions (with answers or without) that will be presented with not much further explanation. Why you might ask? Because this what you'd have to expect on the closed book portion of the midterm or final, **so it might be good to look at these the night before an exam (or the morning of!)**.

[2] When I mention that a certain topic is trivial, it is not necessarily something that you should ignore, as it will most likely be on an exam. However, it is sometimes not worth the amount of illness you will go through to understand this concept, so I've deemed it as trivial.

4 Textbook(s)

Depending on your preference, there exists two textbooks for you to choose in this course. They will both work (Jon will probably say this in the opening class as well), but which one you'll use comes down to personal preference. These are the textbooks:

- Machine Elements in Mechanical Design by Robert L. Mott (here)
- Shigley's Mechanical Engineering Design by J. Keith Nisbeth and Richard G. Budynas (figure it out here)

Now you're probably wondering: "**Which one is better?**" since you don't want to go through two textbooks at a time. Well, there isn't an easy answer to this question, so I've instead asked fellow Mech 325 survivors for their opinions. You can find them below. *Personally, I feel like you will need a combination of both textbooks to succeed, as some later topics (e.g. splines), as you'll come to find out, will only be covered by one textbook. Additionally, you will find that one textbook often does a better job explaining certain topics than the other.* Don't worry, Jon will make sure to let you know when this is the case. Now, onto the reviews:

> *"Honestly they are both very similar, I think reading both is beneficial just to see the similarities and differences. I cant remember all of it but for gears I used Mott mainly and for pulleys ropes (earlier chapters), I used shigley"*
>
> – Andy Zhu, ***Fellow DOG***

> *"When comparing between the Mott and Shigley textbook, there is certainly something to say for both of them. While Shigley has the pedigree, including decades as the standard reference for mechanical design, Mott has the new-comer advantage. Shigley has equations and processes built around tables and tables of coefficients, carrying the baggage of decades of backwards compatibility. Mott often simplifies to the key factors that affect mechanical design, leading to quicker and more intuitive calculations"*
>
> – Felix Wilton, ***Missiles Expert***

> *"I feel like there's not one that's fully better, depending on the topic and assignments questions one textbook may be easier to follow"*
>
> – Jules Turries, ***Fellow DOG***

All in all, if I were taking this course all over again, I'd certainly suggest you figure out which book is more convenient to you. For example, you might find that you have a physical copy of one at home, so it would be stupid of you to not use it. Anyways, onto the more important sections now.

5 Course Overview

Mech 325 is probably not like any of the previous courses you have taken. In this class, the content you learn can actually be applied to work you do in real life. What is even better is that you do not need to memorize anything. All you need to have is a good idea of what to do and where to look and a decent calculator, as the steps and formulae will all be given to you within the textbooks.

Mech 325 Consists of the Following:

- 6 Assignments (5 graded)
- 5-6 RAP Quizzes
- 1 Design Project
- 1 Midterm Exam
- 1 Final Exam

Below, I will let you know what to expect from each.

5.1 Assignments

In simple words, some of the assignments are quite fucked. As in, expect to spend hours upon hours on a single question. Some questions can be solved using the steps outlined in the example problems in the book. For others, you will have to use your brain. Overall, doing the assignments will help you feel ready for the tests and get a better feel for the book content and where to find what you need. **The term gets hectic quite quickly though, so if you don't have time for a few questions of an assignment, don't worry, you'll do fine!**

5.2 RAP Quizzes

The RAP quizzes directly test the readings that Jon assigns you at the start of each topic. If you attend class and take notes (impossible), you will likely not need to read the textbook as much. The RAP quizzes only include more theoretical questions that test your understanding of the topics and are all MCQ (No calculations). There's a trick to reading the text book efficiently[3]. The exams are closed book but the questions are pretty reasonable. **If you miss a reading, you'll probably still pass, and if you don't, they're 1% anyways!** If you have a hard time keeping up with the topics, let me know, and I may be able to guide you in doing better, but no guarantees!

[3]Study Smart, not Hard, IYKYK

5.3 Design Project

The design project is probably the most unwanted portion of Mech 325. Given the heavy course-load you have this semester, the last thing you want to worry about is this project. So what you will likely do is form your teams early and leave this until the last week of class. Groups consist of 6-7 members. My top suggestion would be finding a group of Fizz kids and letting them carry you through the course, but making a group with your friends is also a viable option. For your topic, choose something simple, preferably designing something that is almost fully mechanical. Generally, gears, belts, chains, shafts, etc. are much easier to work with than motors, actuators, pumps, etc. For inspiration, you may have a look at our project's report here. If you run out of project ideas, here are 2 GREAT things to design:

- A Filament Extruder for UBC Rapid
- A Filament Spooler for UBC Rapid

Anyways, good luck on finding the perfect team and choosing your project!

5.4 Midterm Exam

The midterm exam will cover both gears and belts. You will need to have a good idea of the types of questions to expect (the assignments provide this). There is a closed book portion of the exam that tests theory, with both MC and Short Answer questions. They should be mostly straightforward and easy, and are often recycled from previous midterms Jon gives you. However, there are some questions that you will likely sell on, so don't fuss about it too much.

5.5 Final Exam

The final exam covers everything (I think?), with more emphasis on shafts, fluid power, and springs and fasteners. It is basically a longer version of the midterm. There is also a no-calculator section for unit conversion which also contains a few cheeky bonus questions as well. In our final exam, I left one whole long question blank (there were 3 in total) and still managed to get a decent final mark, so don't expect the final to be smooth sailing, but also don't worry about failing if you fuck up.

6 Calculator

Being the DOG that I am, I worked with fellow Mech 325 victims to bring to you what I call "**The Greatest Calculator to Grace The Earth**", which is a calculator made in excel (download it here). Although it does not encompass all topics of Mech 325, it still does a great job of making life easier in most stages of the course. We even included a linear interpolator since you will most likely get gaped by the amount of times you need to do that.

Now given that I am indeed the Matlab DOG, especially after my recent endeavour in Mech 358, you mustn't be surprised when I tell you that I have created Matlab Live scripts as well. As I said at the beginning of the document, I will not be spoon-feeding you throughout this course, but rather guiding you to success, therefore, the scripts will not cover all question variations for Mech 325 (no Screws & Fasteners unfortunately!), so it will be your job to figure those out. It should still help you calculate some exact values instead of just reading from the tables. The script files can be found here.

7 Unit 0: Unit Conversions

In the first ever lecture of 325, Jon will tell you a bit about himself and then probably show you a slide with different unit conversions. He may even write them on the board. He will then tell you one thing: "These will be on the final in a NO Calculator Closed Book portion of the exam". I can't quite remember which ones he shows in class, but I'll put some relevant ones below for you to look out for. Note that some of the conversion values may be rounded!

Metric and Imperial Lengths (in/ft/mi ↔ km/m/cm)

$$1\ in = 2.54\ cm$$

$$1\ ft = 0.305\ m$$
$$1\ ft = 12\ in$$
$$1\ Yard = 3\ ft$$
$$1\ Mile = 1760\ Yards = 1.609\ km$$
$$1\ Nautical\ Mile = 1852\ m$$

Metric and Imperial Mass (lb/slug ↔ kg/tonne)

$$1\ Slug = 32.2\ lb$$

$$1\ kg = 2.2\ lb$$
$$1\ Tonne = 1000\ kg$$
$$1\ Imperial\ Ton = 2240\ lb$$
$$1\ US\ Ton = 2000\ lb = 907.2\ kg$$ [4]

Metric and Imperial Volume (Gallons/mL/Oz/m^3/in^3)

$$1\ Gallon = 231\ in^3$$

$$1\ Gallon = 3.785\ L$$
$$12\ Fluid\ Oz = 355\ mL$$
$$12\ Dry\ Oz = 341\ mL$$ [5]
$$1\ m^3 = 1000\ L$$

Power Conversions (hp/W/ft-lb/s)

$$1\ hp = 746\ W$$ [6]

[4] In Mott, they use US Tons

[5] This could be wrong!

[6] Note that some sources say 1 hp = 735.5 W but I believe Jon uses 746 W. But look out for it nonetheless.

$$1\ hp = 550\ ft - lb/s$$

Temperature Conversion (°F/°C/K)

$$(\frac{9}{5})^{\circ}C + 32 = {}^{\circ}F$$

$$K = 273.15 + {}^{\circ}C$$

Pressure Conversion (Pa/lb/in^2/atm

$$1\ lb/in^2 = 6895\ Pa$$

$$1\ atm = 101.325\ kPa$$

Energy Conversion (BTU ↔ ft-lb)

$$1\ BTU = 778.2\ ft - lb$$

Know all of these by heart and you are guaranteed to get full marks on the first portion of the final exam. You may also grab a few bonus marks in the process! Also, Jon lets you do the conversions in whatever direction you like, so don't worry about the different variations, remember whatever is easier!

8 Unit 1: Belts, Chains, and Ropes

This unit is what you will start off with in Mech 325. It may seem a bit difficult at first, but you will soon realize how easy understanding these topics are compared to what is yet to come in the course.

8.1 Class Memory

I do not remember the exact date, but I think it was maybe the 3rd or 4th 325 lecture in the term, and Jon had to leave for something, leaving us with his TA, Jose. Now I am not sure if you will have the pleasure of meeting Jose this year, but I can say he was definitely one of the better TAs overall. Cool accent too. Anyways, Jose comes in and begins going over Jon's lecture slides. Needless to say this was the first time the class was moving at normal pace, since Jose wasn't really in a rush to get through the lecture slides, so I could for once actually make sense of what was going on. Unfortunately, this was also when I found out that I had understood jack shit about the topic even though I had attended all prior lectures. This definitely made me lose interest in the lectures, which eventually led to me not attending them at all by the later half of the semester.

8.2 Do's and Don'ts

Do's:

- Familiarize yourself with both imperial and metric formulae, since assignments and exams may ask for either.
- Focus on basic question types (design questions), with heavy emphasis on:
  - V-belts
  - Chain Drives
  - Timing (Synchronous) Belts

Don'ts:

- Don't use Shigley for exam questions, except for:
  - Flat Belt Questions
  - Mine Hoist Problems (Wire Ropes)
- Don't put too much effort into wire ropes[7]
- Don't try to learn ALL the different procedures by heart. Instead try to get a good idea of where to look and what you need to solve questions

Also there is lots of theory and readings in this unit, so don't expect to know everything on demand. Practice makes perfect!

[7]I came to learn this the hard way, but there was actually a question on wire ropes in the final, so maybe don't skip out on it too much.

8.3 Unit Guide

Since this unit is quite extensive, I will give you a clear guide for sizing each type of component, independent of one another, despite there being alot of repetition. In each guide, I will explain in a way that even a child will understand. This is to make your life easier.

I hope you can tell for yourself that all these guides are for 2 pulley systems. For systems of 3 or more pulleys, solve for 2 pulleys at a time, and you should be Gucci.

Also, when it comes to both textbooks, they do things considerably differently. In this case, Shigley is a bit more complicated with a fuck-load of correction factors and what not, so I will primarily use Mott. It does however have a lot of good figures. It is very much possible that some homework questions in this section will require one textbook or the other, so in case I don't have it covered, I'll let you figure it out yourself.

Make sure to check out this unit's calculators (here), they're useful! Here's the list of calculators available within the Mlx file:

- Flat-Belt Calculator
- Automatic Standard Belt Length Calculator (for V-belt)
- Chain Drive Calculator[8]
- Wire Rope Calculators (Mine Hoist[9] and Shigley's Design Guide)

For force calculations, check the excel as well.

[8] Very Good
[9] Spectacular

8.3.1 Flat Belts

**This guide follows Shigley and so should you.*

1. **In case of a flat belt, you will need the following specifications to start the question/design:**
   - C [in.], the centre to centre distance between each pulley
   - Pulley diameters d (driving) and D (driven) [in.], or at least a velocity ratio
   - n [rpm], rotational speed
   - H_{nom} [hp], nominal horsepower
   - b [in.], belt width
   - t [in.], belt thickness
   - Belt Material: Leather, Polyamide, or Urethane (most common)
   - w [lbf/ft], weight of a foot of belt
   - f, Coefficient of Friction

   If the question requires you to DESIGN a flat belt drive, you will only be given the first 4 specifications, and you'll have to choose a belt yourself. Use the figures below to do so.

Table 17–2 Properties of Some Flat- and Round-Belt Materials. (Diameter = d, thickness = t, width = w)

Material	Specification	Size, in	Minimum Pulley Diameter, in	Allowable Tension per Unit Width at 600 ft/min, lbf/in	Specific Weight, lbf/in^3	Coefficient of Friction
Leather	1 ply	$t = \frac{11}{64}$	3	30	0.035–0.045	0.4
		$t = \frac{13}{64}$	$3\frac{1}{2}$	33	0.035–0.045	0.4
	2 ply	$t = \frac{18}{64}$	$4\frac{1}{2}$	41	0.035–0.045	0.4
		$t = \frac{20}{64}$	6^a	50	0.035–0.045	0.4
		$t = \frac{23}{64}$	9^a	60	0.035–0.045	0.4
Polyamide[b]	F–0[c]	$t = 0.03$	0.60	10	0.035	0.5
	F–1[c]	$t = 0.05$	1.0	35	0.035	0.5
	F–2[c]	$t = 0.07$	2.4	60	0.051	0.5
	A–2[c]	$t = 0.11$	2.4	60	0.037	0.8
	A–3[c]	$t = 0.13$	4.3	100	0.042	0.8
	A–4[c]	$t = 0.20$	9.5	175	0.039	0.8
	A–5[c]	$t = 0.25$	13.5	275	0.039	0.8
Urethane[d]	$w = 0.50$ in	$t = 0.062$	See Table 17–3	5.2^e	0.038–0.045	0.7
	$w = 0.75$ in	$t = 0.078$		9.8^e	0.038–0.045	0.7
	$w = 1.25$ in	$t = 0.090$		18.9^e	0.038–0.045	0.7
	Round	$d = \frac{1}{4}$	See Table 17–3	8.3^e	0.038–0.045	0.7
		$d = \frac{3}{8}$		18.6^e	0.038–0.045	0.7
		$d = \frac{1}{2}$		33.0^e	0.038–0.045	0.7
		$d = \frac{3}{4}$		74.3^e	0.038–0.045	0.7

Some things to clarify:

- Velocity Ratio:

$$Velocity\ Ratio = \frac{V_{driver}}{V_{driven}} = \frac{D}{d}$$

- Belt Speed [ft/min]:

$$V = \frac{\pi dn}{12}$$

where d [in.] is pulley diameter, and n [rpm] is rotational speed

- w [lbf/ft], weight per foot:

$$w = 12\gamma bt$$

where γ [lbf/in^3] is weight density[10]*, b [in.] is belt width, and t [in.] is belt thickness*

2. **Next, you may want to find some of these parameters, based on the question being asked** [11]**:**

- Wrap Angles[12] [rad]:

$$\phi_d = \pi - 2sin^{-1}(\frac{D-d}{2C})$$

$$\phi_D = \pi + 2sin^{-1}(\frac{D-d}{2C})$$

- Belt Length[13] [in.]:

$$L = \sqrt{4C^2 - (D-d)^2} + \frac{1}{2}(D\phi_D + d\phi_d)$$

- H_d [hp], Design Horsepower:

$$H_d = H_{nom} K_s n_d$$

where K_s is a service factor and n_d is a design factor (also called safety factor at times). For flat belts, they are usually given to you

3. **After this, you will want to find Torques and Forces:**

- T [lbf · in], required torque:

$$T = \frac{63025 H_d}{n}$$

[10] or specific weight in the figure above

[11] If it's "design" question, find all to be safe

[12] These equations are for open belts. for crossed belts, $\phi = \pi + 2sin^{-1}(\frac{D+d}{2C})$ and both pulleys have the same angle of wrap

[13] For crossed belts, $L = \sqrt{4C^2 - (D+d)^2} + \frac{1}{2}(D+d)\phi$

- F_c [lbf], centrifugal force:

$$F_c = \frac{w}{g}(\frac{V}{60})^2$$

where w [lbf/ft] is weight per foot, g is 32.2 ft/s^2, and V [ft/min] is belt speed

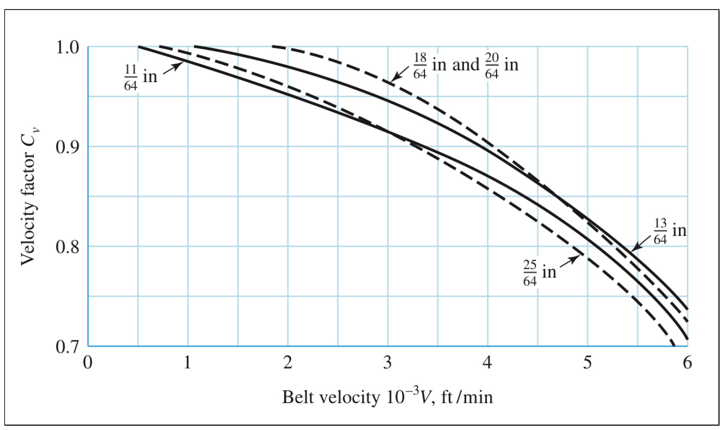
- F_{1a} [lbf] allowable largest tension (tight side tension):

$$F_{1a} = bF_aC_pC_v$$

where b [in] is belt width, F_a [lbf/in] is manufacturer's allowed tension (see figure above), C_p is the pulley correction factor (1 for urethane belts; see figure below for others), and C_v is the velocity correction factor (1 for urethane and polyamide belts; see figure below for leather belts of different thicknesses)

Table 17–4 Pulley Correction Factor C_P for Flat Belts*

	Small-Pulley Diameter, in					
Material	**1.6 to 4**	**4.5 to 8**	**9 to 12.5**	**14, 16**	**18 to 31.5**	**Over 31.5**
Leather	0.5	0.6	0.7	0.8	0.9	1.0
Polyamide, F–0	0.95	1.0	1.0	1.0	1.0	1.0
F–1	0.70	0.92	0.95	1.0	1.0	1.0
F–2	0.73	0.86	0.96	1.0	1.0	1.0
A–2	0.73	0.86	0.96	1.0	1.0	1.0
A–3	—	0.70	0.87	0.94	0.96	1.0
A–4	—	—	0.71	0.80	0.85	0.92
A–5	—	—	—	0.72	0.77	0.91



- $F_{1a} - F_2$ [lbf]:

$$F_{1a} - F_2 = \frac{2T}{d}$$

where T [lbf $\cdot$in] is Torque, and d [in.] is the driver pulley diameter

- F_2 [lbf], slack side tension:

$$F_2 = F_{1a} - (F_{1a} - F_2) = F_{1a} - \frac{2T}{d}$$

- F_i [lbf], initial tension:

$$F_i = \frac{F_{1a} + F_2}{2} - F_c$$

4. **Finally, wrap up with a few concluding calculations:**

- H_a [hp], Transmitted Power:

$$H_a = \frac{(F_{1a} - F_2)V}{33000}$$

This should equal to your design horsepower...

- n_{fs}, Safety Factor:

$$n_{fs} = \frac{H_a}{H_{nom}K_s}$$

- Check friction development:

$$f' = \frac{1}{\phi_d} ln(\frac{F_{1a} - F_c}{F_2 - F_c})$$

make sure f' < f or else you will need to change the design!!!!!

- dip [in.]:

$$dip = \frac{12(C/12)^2 w}{8F_i}$$

where C [in.] is the centre to centre distance of the pulleys, w [lbf/ft] is the weight per foot of the belt, and F_i [lbf] is initial tension

Check out the Matlab Live script calculator to make life easier!

8.3.2 V-Belts

**This guide follows Mott*[14].

1. **To start designing a V-Belt drive, you must know:**

   - H_{nom} [hp], Nominal Transmitted Power
   - n [rpm], Input Rotational Speed
   - Velocity Ratio (or some way to find it, like output speed). Recall:

   $$VR = \frac{V_{driving}}{V_{driven}} = \frac{D}{d} = \frac{n_{driving}}{n_{driven}}$$

   d is driving sheave's diameter, while D is driven sheave's diameter

2. **From there, you must choose your belt type. To do so, first find your relative service factor (K_s or SF) from the figure below**

TABLE 7–1 V-Belt Service Factors[1]

	Driver type					
	AC motors: Normal torque[2] DC motors: Shunt-wound Engines: Multiple-cylinder			AC motors: High torque[3] DC motors: Series-wound, or compound-wound Engines: 4-cylinder or less		
Driven machine type	**<6 h per day**	**6–15 h per day**	**>15 h per day**	**<6 h per day**	**6–15 h per day**	**>15 h per day**
Smooth loading	1.0	1.1	1.2	1.1	1.2	1.3
Agitators, light conveyors, centrifugal pumps fans and blowers under 10 hp (7.5 kW)						
Light shock loading	1.1	1.2	1.3	1.2	1.3	1.4
Generators, machine tools mixers, fans and blowers over 10 hp (7.5 kW) gravel conveyors						
Moderate shock loading	1.2	1.3	1.4	1.4	1.5	1.6
Bucket elevators, piston pumps textile machinery, hammer mills heavy conveyors, pulverizers						
Heavy shock loading	1.3	1.4	1.5	1.5	1.6	1.8
Crushers, ball mills, hoists rubber mills, and extruders						
Machinery that can choke	2.0	2.0	2.0	2.0	2.0	2.0

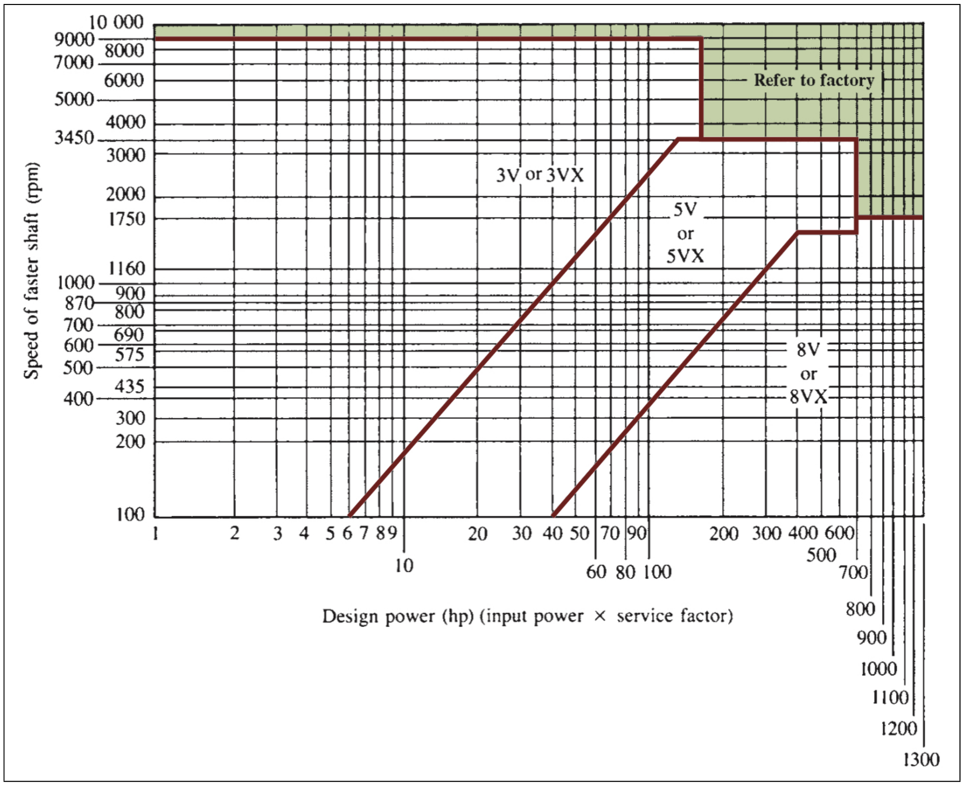
[1]Factors given are for speed reducers. For speed increases, multiply listed factors by 1.2.
[2]Synchronous, split-phase, three-phase with starting torque or breakdown torque less than 175% of full-load torque.
[3]Single-phase, three-phase with starting torque or breakdown torque greater than 175% of full-load torque.

Then, find the design power:

$$H_d = H_{nom} K_s$$

[14]Shigley also provides a good V-belt guide, but you're better off to use Mott in exam. If the question asks for Tension per Strand (T_{st}), F_{min} , or deflection, have a look in Shigley.

Finally, choose the belt based on which region your specs lie in the graph below:



3. **After figuring out your belt, you will have to find the Driven and Driver sheave diameters**. To do so, assume a belt speed of $V = 4000\ ft/min$[15], and solve for d:

$$d = \frac{12V}{\pi n}$$

Then choose a standard Sheave Diameter from the figures below (7-15, 7-16, 7-17) which is close to your d value (based on belt type), and use the VR to solve for D:

$$D = VR \times d$$

Once again, choose the closest standard sheave diameter from the figures below (7-15, 7-16, 7-17), based on your belt type. Finally, solve for your

[15]PLEASE NOTE: the velocity value is a place holder. I'm not exactly sure why 4000 ft/min is used, maybe ask Jon if you're curious!

new velocity ratio, and as a result, your final output rotational speed and belt speed. Make sure your final velocity is within 5% of 4000 ft/min.[16]

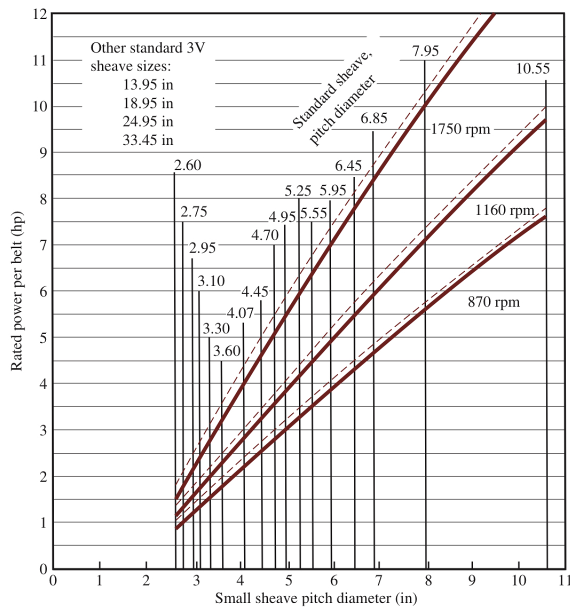


FIGURE 7–14 Power rating: 3V belts

[16]In this step where you're finalizing your sheave diameters, you might have to perform a few "trials" where you choose different d values and calculate D. In some cases it will be easy, in some it might take a while. Be patient.

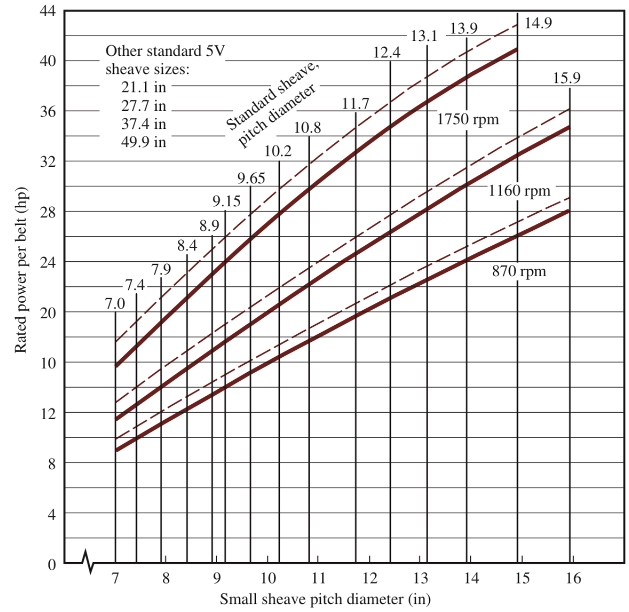


FIGURE 7–15 Power rating: 5V belts

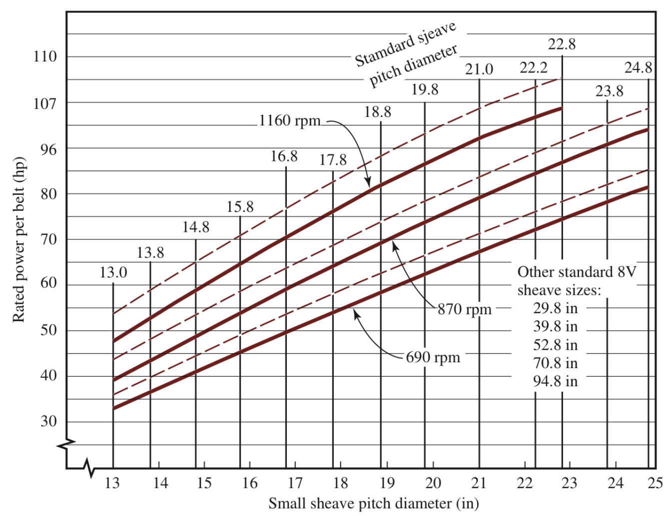


FIGURE 7–16 Power rating: 8V belts

4. **Once you've finalised your diameters, you can use the same figures above to get the rated power of your belt** (use your d value). Looking at the figures, the red curves represents the power rating at a VR

of 1 and the dashed curves are for a VR of 3.38[17]. The dashed line shows the "added power" amount that gives you your true power rating. For 5V belts, the figure below shows the amount you have to add. However, for 3V and 8V belts, you'll need to interpolate for the "true" rating using the figures. Since this is probably a bit hard to understand, I will give a simple rundown:

- **If you choose a 3V or 8V belt**, read off the solid and dashed curve power values (P_1 and P_2), then use the following equation to find your rated Power:

$$H_r = P_2 + \frac{VR - 3.38}{3.38 - 1}(P_2 - P_1)$$

- **If you choose a 5V belt**, read off the value of the solid line (H_s), then use the figure below to get the "added power" (H_{added}) based on your VR. Then:

$$H_r = H_s + H_{added}$$

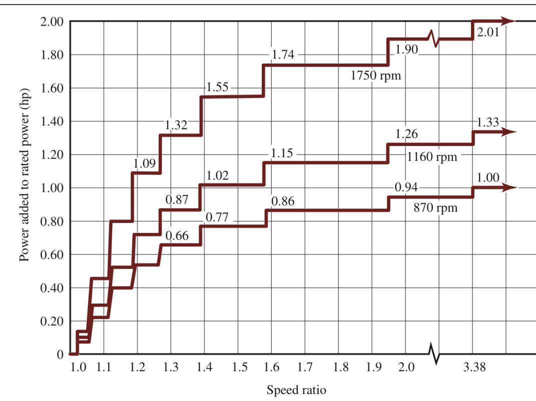


FIGURE 7–17 Power added versus speed ratio: 5V belts

5. **Now you need to finalize your belt length (L, in.) and Centre Distance (CD, in.)**. Begin by specifying a trial centre distance using the following range:

$$D < CD < 3(D + d)$$

Try being closer to D. After that, compute the belt length using the equation below, then find the closest standard length for your belt type from the figure below.

$$L_p = 2CD + 1.57(D + d) + \frac{(D - d)^2}{4CD}$$

[17]For a VR greater than 3.38, assume the dashed line doesn't increase further, and your rated power doesn't change much.

TABLE 7–2 Standard Belt Lengths for 3V, 5V, and 8V Belts (in)

3V only	3V and 5V	3V, 5V, and 8V	5V and 8V	8V only
25	50	100	150	375
26.5	53	106	160	400
28	56	112	170	425
30	60	118	180	450
31.5	63	125	190	475
33.5	67	132	200	500
35.5	71	140	212	
37.5	75		224	
40	80		236	
42.5	85		250	
45	90		265	
47.5	95		280	
			300	
165			315	
			335	
			355	

Once you have your standard length, re-solve for your CD (in.):

$$CD = \frac{B + \sqrt{B^2 - 32(D-d)^2}}{16}$$

where,

$$B = 4L_p - 6.28(D+d)$$

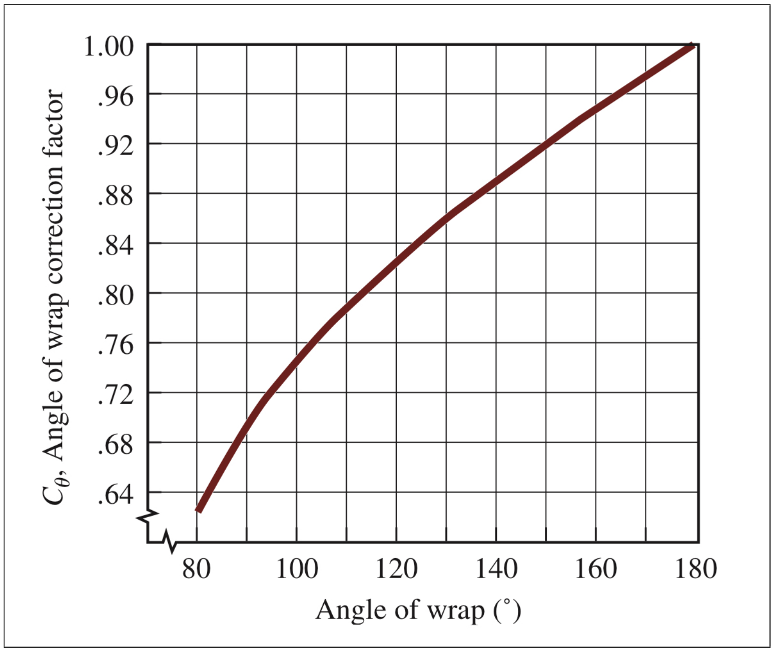
6. **Next, you need to find the corrected rated power** (H_c, hp). First compute the angle of wraps for both sheaves:

$$\theta_d = \pi - 2sin^{-1}(\frac{D-d}{2CD})$$

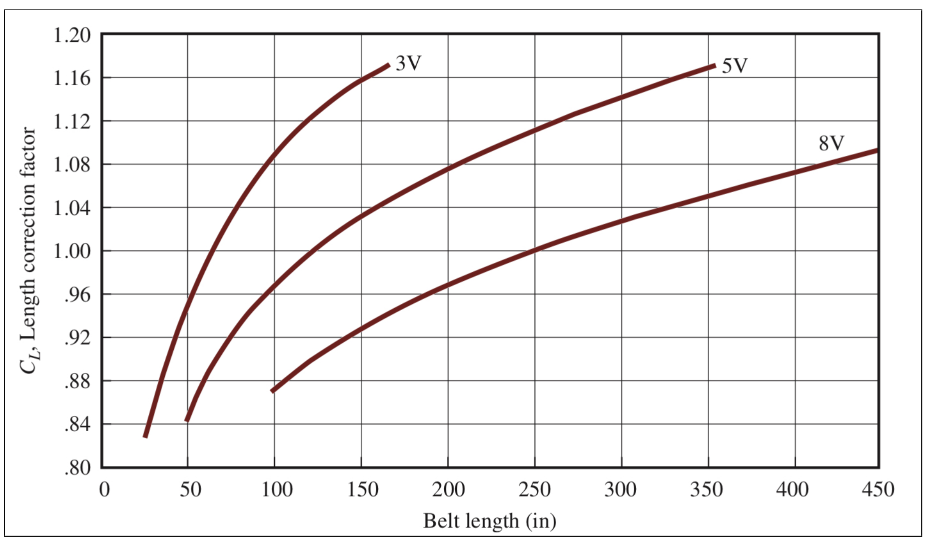
$$\theta_D = \pi + 2sin^{-1}(\frac{D-d}{2CD})$$

Then, find from the figures below:

- **Angle of wrap correction factor,** C_θ



- **Belt Length Correction Factor,** C_L



Then, calculate the corrected power rating:

$$H_c = H_r C_\theta C_L$$

7. **Finally, calculate how many belts you need**, using:

$$Minimum\ No.\ of\ Belts = \frac{H_d}{H_c}$$

Always round up to the next whole number![18]

You're done!

EXTRA: For this section, there is no calculator since V-belts require quite a bit of table lookup. However, these might help

- for wrap angles, you can use the flat belt code
- for finalizing length and CD, use the designated code at the end of the Matlab live script.[19]
- For the rated power of a 3V or 5V belt, you can use the interpolator on excel.

[18]For example, if you get 2.01, you'll need 3 belts.

[19]I tried making this part automated so just give it d and D and it will do its thing. You can adjust the CD yourself if you need a desired value

8.3.3 Timing Belts

**This guide follows Mott*

1. **To start off, you will be given the following specifications:**

   - H_{nom} [hp], nominal horsepower
   - n [rpm], Input speed
   - VR, velocity ratio. Recall:

   $$VR = \frac{V_{driving}}{V_{driven}} = \frac{D}{d} = \frac{n_{driving}}{n_{driven}}$$

   - CD [in.], nominal center distance[20]
   - Other minor details, such as:
     - Maximum Sprocket diameter [in.]
     - Operation time [hr]
     - Shaft diameters [in.]

2. **Next, determine the service factor (SF) from the figure in the next page and find the design power (H_d).**

   $$H_d = H_{nom} \times SF$$

[20] The CD is usually given as a range with some $\pm$ tolerance in these cases

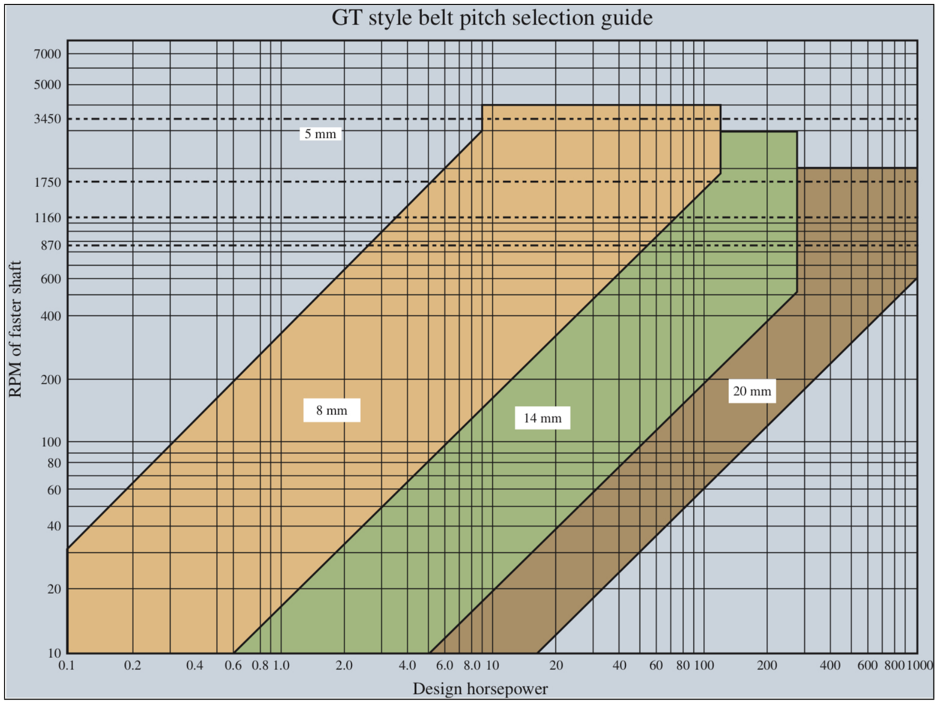
TABLE 7–8 Service Factor

DriveN machine	DriveR					
The driveN machines listed below are representative samples only. Select a driveN machine whose load characteristics most closely approximate those of the machine being considered.	AC Motors: Normal Torque, Squirrel Cage, Synchronous, Split Phase, Inverter Controlled DC Motors: Shunt Wound Stepper Motors Engines: Multiple Cylinder Internal Combustion			AC Motors: High Torque, High Slip, Repulsion-Induction, Single Phase, Series Wound, Slip Ring DC Motors: Series Wound, Compound Wound Servo Motors Engines: Single Cylinder Internal Combustion, Line Shafts, Clutches		
	Intermittent Service (Up to 8 Hours Daily or Seasonal)	Normal Service (8–16 Hours Daily)	Continuous Service (16–24 Hours Daily)	Intermittent Service (Up to 8 Hours Daily or Seasonal)	Normal Service (8–16 Hours Daily)	Continuous Service (16–24 Hours Daily)
Display, Dispending Equipment Instrumentation Measuring Equipment Medical Equipment Office, Projection Equipment	1.0	1.2	1.4	1.2	1.4	1.6
Appliances, Sweepers, Sewing Machines Screens, Oven Screens, Drum, Conical Woodworking Equipment (Light): Band Saws, Drills Lathes	1.1	1.3	1.5	1.3	1.5	1.7
Agitators for Liquids Conveyors: Belt, Light Package Drill Press, Lathes, Saws Laundry Machinery Wood Working Equipment (Heavy): Circular Saws, Jointers, Planers	1.2	1.4	1.6	1.6	1.8	2.0
Agitators for Semi-Liquids Compressor: Centrifugal Conveyor Belt: Ore, Coal, Sand Dough Mixers Line Shafts Machine Tools: Grinder, Shaper, Boring Mill, Milling Machines Paper Machinery (except Pulpers): Presses, Punches, Shears Printing Machinery Pumps: Centrifugal, Gear Screens: Revolving, Vibratory	1.3	1.5	1.7	1.6	1.8	2.0
Brick Machinery (except Pug Mills) Conveyor: Apron, Pan, Bucket, Elevator Extractors, Washers Fans, Centrifugal Blowers Generators & Exciters Hoists Rubber Calender, Mills, Extruders	1.4	1.6	1.8	1.8	2.0	2.2
Centrifuges Screw Conveyors Hammer Mills Paper Pulpers Textile Machinery	1.5	1.7	1.9	1.9	2.1	2.3
Blowers: Positive Displacement, Mine Fans Pulvertzers	1.6	1.8	2.0	2.0	2.2	2.4
Compressors: Reciprocating Crushers: Gyratory, Jaw, Rol Mils: Ball, Rod, Pebble, etc. Pumps: Reciprocating Saw Mill Equipment	1.7	1.9	2.1	2.1	2.3	2.5

These service factors are adequate for most belt drive applications. Note that service factors cannot be substituted for good engineering judgment. Service factors may be adjusted based upon an understanding of the severity of actual drive operating conditions.

3. **Find the required pitch for the belt** from your H_d and n values using the figure below.[21]

[21] I could bet my entire fortune that it will be 8 mm, but do check anyways



4. **From your VR, use the figures in Appendix A to choose combinations of driver and driven sprocket teeth sizes (N_d and N_D) that work.**[22] Now check to see which combinations work (first three columns in table only for now!). To do this, check for all possible limitations, such as

   - Check bushing style and maximum bore sizes in case you are given shaft information. Use the Tables (7-4 and 7-5) below and compare to your given diameter. Make sure

$$Min.\ Bore\ Diameter < Shaft\ Diameter < Max.\ Bore\ Diameter$$

   - Check sprocket sizes using table 7-4 below. Make sure they pass whatever limitations is imposed by the question.

[22]The figures took up too much space here, so I had to put them in an appendix. From now on, this will be the case for some figures as well

TABLE 7–4 Sprockets with 8 mm Belt Pitch

Dim's all widths			20-mm Wide belt		30-mm Wide belt		50-mm Wide belt		85-mm Wide belt	
No. of teeth	Pitch dia.	Flange dia.	Sprocket number	Bushing size	Sprocket number	Bushing size	Sprocket number	Bushing size	Sprocket number	Bushing size
22	2.206	2.559	P22-8MGT-20	1108	P22-8MGT-30	1108	N/A	N/A	N/A	N/A
24	2.406	2.756	P24-8MGT-20	1108	P24-8MGT-30	1108	N/A	N/A	N/A	N/A
26	2.607	2.953	P26-8MGT-20	1108	P26-8MGT-30	1108	N/A	N/A	N/A	N/A
28	2.807	3.15	P28-8MGT-20	1108	P28-8MGT-30	1108	P28-8MGT-50	MPB	N/A	N/A
30	3.008	3.346	P30-8MGT-20	1210	P30-8MGT-30	1210	P30-8MGT-50	1210	N/A	N/A
32	3.208	3.543	P32-8MGT-20	1210	P32-8MGT-30	1210	P32-8MGT-50	1210	N/A	N/A
34	3.409	3.819	P34-8MGT-20	1610	P34-8MGT-30	1610	P34-8MGT-50	1610	P34-8MGT-85	1615
36	3.609	3.937	P36-8MGT-20	1610	P36-8MGT-30	1610	P36-8MGT-50	1610	P36-8MGT-85	1615
38	3.810	4.134	P38-8MGT-20	1610	P38-8MGT-30	1610	P38-8MGT-50	1610	P38-8MGT-85	1610
40	4.010	4.331	P40-8MGT-20	1610	P40-8MGT-30	2012	P40-8MGT-50	2012	P40-8MGT-85	2012
44	4.411	4.764	P44-8MGT-20	2012	P44-8MGT-30	2012	P44-8MGT-50	2012	P44-8MGT-85	2012
48	4.812	5.157	P48-8MGT-20	2012	P48-8MGT-30	2012	P48-8MGT-50	2012	P48-8MGT-85	2012
56	5.614	5.945	P56-8MGT-20	2012	P56-8MGT-30	2012	P56-8MGT-50	2517	P56-8MGT-85	2517
64	6.416	6.772	P64-8MGT-20	2012	P64-8MGT-30	2517	P64-8MGT-50	2517	P64-8MGT-85	2517
72	7.218	7.598	P72-8MGT-20	2012	P72-8MGT-30	2517	P72-8MGT-50	2517	P72-8MGT-85	3020
80	8.020	8.386	P80-8MGT-20	2517	P80-8MGT-30	2517	P80-8MGT-50	2517	P80-8MGT-85	3020
90	9.023	N/A	P90-8MGT-20	2517	P90-8MGT-30	2517	P90-8MGT-50	3020	P90-8MGT-85	3020
112	11.229	N/A	N/A	N/A	P112-8MGT-30	2517	P112-8MGT-50	3020	P112-8MGT-85	3020
144	14.437	N/A	N/A	N/A	P144-8MGT-30	2517	P144-8MGT-50	3020	P144-8MGT-85	3535
192	19.249	N/A	N/A	N/A	N/A	N/A	P192-8MGT-50	3020	P192-8MGT-85	3535

TABLE 7–5 Taper-Lock Bushing

Bushing size	Min bore	Max bore
1008	0.500	0.875
1108	0.500	1.000
1210	0.500	1.250
1610	0.500	1.500
1615	0.500	1.500
2012	0.500	1.875
2517	0.500	2.250
3020	0.875	2.750
3525	1.188	3.250
3535	1.188	3.250
4030	1.438	3.625
4040	1.438	3.625
4535	1.938	4.250
4545	1.938	4.250
5040	2.438	4.500
6050	4.438	6.000
7060	4.938	7.000

Once you've determined the sprocket combination given your velocity ratio, go back to Appendix Table 7-4 above and determine the pitch diameters PD_d and PD_D [in.] for your sprockets.

5. **After that, go back to Appendix A and choose a belt given your sprocket combination and your desired CD range**

6. **Now, you have to select the belt width**[23]. First, find the belt length correction factor (C_L) from the figure below.

TABLE 7–11 8M GT Style Belt Length Correction Factor

Pitch/Length designation	No. of feeth	Correction factor	Pitch/Length designation	No. of teeth	Correction factor	Pitch/Length designation	No. of teeth	Correction factor	Pitch/Length designation	No. of teeth	Correction factor
384-8MGT	48	0.70	920-8MGT	115	1.00	1440-8MGT	180	1.10	2600-8MGT	325	1.20
480-8MGT	60	0.80	960-8MGT	120	1.00	1512-8MGT	189	1.10	2800-8MGT	350	1.20
560-8MGT	70	0.80	1040-8MGT	130	1.00	1584-8MGT	198	1.10	3048-8MGT	381	1.20
600-8MGT	75	0.80	1064-8MGT	133	1.00	1600-8MGT	200	1.10	3280-8MGT	410	1.20
640-8MGT	80	0.90	1120-8MGT	140	1.00	1760-8MGT	220	1.10	3600-8MGT	450	1.20
720-8MGT	90	0.90	1160-8MGT	145	1.00	1800-8MGT	225	1.20	4400-8MGT	550	1.20
800-8MGT	100	0.90	1200-8MGT	150	1.00	2000-8MGT	250	1.20			
840-8MGT	105	0.90	1224-8MGT	153	1.00	2200-8MGT	275	1.20			
880-8MGT	110	0.90	1280-8MGT	160	1.10	2400-8MGT	300	1.20			

7. **Next, use the two figures**[24] **in the next pages with your values of N_d and n to find the rated power of your belt**. Then, get the adjusted power using

$$P_{adjusted} = P_{rated} \times C_L$$

Compare this value to your design horsepower and make sure it is well above it![25]

[23] You basically have two choices here, 30 mm or 50 mm.
[24] Again, try either or both to check. If you are given a weird n value, you need to interpolate
[25] Tip: if 30 mm doesn't work, 50 mm will probably work, so starting off with 50 mm may be smart!

TABLE 7–9 8M GT Style Belt Power Rating Table—30-mm Belt Width

RPM of faster shaft	Base rated horsepower for small sprocket (Number of grooves and pitch diameter, inches)															
	22	24	26	28	30	32	34	36	38	40	44	48	56	64	72	80
	2.206	2.406	2.607	2.807	3.008	3.208	3.409	3.609	3.810	4.010	4.411	4.812	5.614	6.416	7.218	8.020
10	0.10	0.12	0.13	0.15	0.16	0.17	0.19	0.20	0.22	0.23	0.26	0.29	0.34	0.40	0.45	0.51
20	0.20	0.22	0.25	0.28	0.31	0.33	0.36	0.39	0.42	0.44	0.50	0.55	0.66	0.76	0.87	0.98
40	0.37	0.43	0.48	0.53	0.59	0.64	0.69	0.75	0.80	0.85	0.96	1.06	1.27	1.47	1.68	1.88
60	0.54	0.62	0.70	0.78	0.86	0.94	1.01	1.09	1.17	1.25	1.40	1.55	1.86	2.16	2.46	2.76
100	0.87	1.00	1.12	1.25	1.38	1.51	1.63	1.76	1.89	2.01	2.26	2.51	3.00	3.49	3.98	4.47
200	1.64	1.89	2.13	2.38	2.63	2.87	3.12	3.36	3.60	3.84	4.33	4.80	5.76	6.70	7.64	8.58
300	2.37	2.74	3.10	3.46	3.82	4.18	4.54	4.90	5.25	5.61	6.32	7.02	8.42	9.80	11.2	12.5
400	3.08	3.56	4.04	4.51	4.99	5.46	5.93	6.40	6.87	7.33	8.26	9.18	11.0	12.8	14.6	16.4
500	3.77	4.36	4.95	5.54	6.13	6.71	7.29	7.87	8.45	9.02	10.2	11.3	13.6	15.8	18.0	20.2
600	4.45	5.15	5.85	6.55	7.25	7.94	8.63	9.31	10.0	10.7	12.0	13.4	16.1	18.7	21.4	24.0
700	5.11	5.93	6.74	7.54	8.35	9.15	9.95	10.7	11.5	12.3	13.9	15.5	18.6	21.6	24.7	27.7
800	5.77	6.69	7.61	8.52	9.44	10.3	11.2	12.1	13.0	13.9	15.7	17.5	21.0	24.5	27.9	31.4
870	6.22	7.22	8.22	9.20	10.2	11.2	12.2	13.1	14.1	15.1	17.0	18.9	22.7	26.5	30.2	33.9
1000	7.05	8.19	9.33	10.5	11.6	12.7	13.8	14.9	16.0	17.1	19.3	21.5	25.8	30.1	34.3	38.5
1160	8.06	9.37	10.7	12.0	13.3	14.5	15.8	17.1	18.4	19.6	22.2	24.7	29.6	34.5	39.4	44.2
1200	8.31	9.66	11.0	12.3	13.7	15.0	16.3	17.6	19.0	20.3	22.9	25.4	30.6	35.6	40.6	45.6
1400	9.54	11.1	12.7	14.2	15.7	17.3	18.8	20.3	21.8	23.3	26.3	29.3	35.2	41.0	46.8	52.4
1600	10.7	12.5	14.3	16.0	17.8	19.5	21.2	23.0	24.7	26.4	29.8	33.1	39.8	46.3	52.8	59.1
1750	11.6	13.6	15.5	17.4	19.3	21.2	23.0	24.9	26.8	28.6	32.3	36.0	43.2	50.3	57.2	64.1
2000	13.1	15.3	17.5	19.6	21.8	23.9	26.0	28.1	30.2	32.3	36.5	40.6	48.7	56.7	64.5	72.1
2400	15.4	18.0	20.5	23.1	25.6	28.1	30.7	33.1	35.6	38.1	43.0	47.8	57.3	66.6	75.6	84.4
2800	17.6	20.6	23.6	26.5	29.4	32.3	35.2	38.0	40.9	43.7	49.3	54.8	65.6	76.1	86.2	96.0
3200	19.8	23.2	26.5	29.8	33.1	36.4	39.6	42.8	46.0	49.2	55.4	61.6	73.6	85.2	96.2	
3450	21.1	24.7	28.3	31.9	35.4	38.9	42.3	45.8	49.2	52.5	59.2	65.7	78.4	90.6	102.2	
4000	24.0	28.1	32.2	36.2	40.3	44.2	48.1	52.0	55.9	59.7	67.1	74.5	88.5			
4500	26.6	31.1	35.6	40.1	44.5	48.9	53.2	57.5	61.7	65.9	74.0	82.0				
5000	29.0	34.0	39.0	43.8	48.7	53.4	58.1	62.8	67.3	71.8	80.6	89.1				
5500	31.4	36.8	42.2	47.5	52.7	57.8	62.9	67.8	72.7	77.5	86.8					

TABLE 7–10 8M GT Style Belt Power Rating Table—50-mm Belt Width

RPM of faster shaft	Base rated horsepower for small sprocket (Number of grooves and pitch diameter, inches) 28 2.807	30 3.008	32 3.208	34 3.409	36 3.609	38 3.810	40 4.010	44 4.411	48 4.812	56 5.614	64 6.416	72 7.218	80 8.020
10	0.25	0.28	0.30	0.33	0.35	0.38	0.40	0.45	0.50	0.59	0.69	0.78	0.88
20	0.49	0.53	0.58	0.63	0.68	0.72	0.77	0.86	0.96	1.14	1.33	1.51	1.70
40	0.93	1.02	1.11	1.21	1.30	1.39	1.48	1.66	1.84	2.20	2.56	2.92	3.27
60	1.35	1.49	1.63	1.76	1.90	2.03	2.17	2.43	2.70	3.23	3.75	4.28	4.80
100	2.18	2.40	2.62	2.84	3.06	3.28	3.50	3.93	4.36	5.22	6.08	6.92	7.77
200	4.14	4.57	4.99	5.42	5.84	6.26	6.68	7.52	8.35	10.0	11.7	13.3	14.9
300	6.02	6.65	7.27	7.90	8.52	9.14	9.75	11.0	12.2	14.6	17.0	19.4	21.8
400	7.85	8.67	9.49	10.3	11.1	11.9	12.7	14.4	16.0	19.2	22.3	25.5	28.6
500	9.63	10.7	11.7	12.7	13.7	14.7	15.7	17.7	19.7	23.6	27.5	31.4	35.2
600	11.4	12.6	13.8	15.0	16.2	17.4	18.6	20.9	23.3	28.0	32.6	37.2	41.7
700	13.1	14.5	15.9	17.3	18.7	20.1	21.4	24.2	26.9	32.3	37.6	42.9	48.2
800	14.8	16.4	18.0	19.6	21.1	22.7	24.2	27.3	30.4	36.5	42.6	48.6	54.5
870	16.0	17.7	19.4	21.1	22.8	24.5	26.2	29.5	32.9	39.5	46.0	52.5	58.9
1000	18.2	20.1	22.1	24.0	25.9	27.9	29.8	33.6	37.4	44.9	52.4	59.7	67.0
1160	20.8	23.1	25.3	27.5	29.7	32.0	34.1	38.5	42.9	51.5	60.0	68.5	76.8
1200	21.5	23.8	26.1	28.4	30.7	33.0	35.2	39.8	44.2	53.1	61.9	70.6	79.2
1400	24.7	27.4	30.0	32.7	35.3	38.0	40.6	45.8	51.0	61.2	71.3	81.3	91.2
1600	27.9	30.9	33.9	36.9	39.9	42.9	45.9	51.8	57.6	69.2	80.6	91.8	102.9
1750	30.2	33.5	36.8	40.1	43.3	46.6	49.8	56.2	62.5	75.0	87.4	99.5	111.4
2000	34.1	37.8	41.5	45.2	48.9	52.6	56.2	63.4	70.6	84.7	98.5	112.1	125.4
2400	40.2	44.6	48.9	53.3	57.6	62.0	66.2	74.7	83.1	99.7	115.8	131.5	146.8
2800	46.1	51.2	56.2	61.2	66.2	71.1	76.0	85.7	95.3	114.1	132.3	149.9	166.9
3200	51.9	57.6	63.2	68.9	74.5	80.0	85.5	96.4	107.1	128.0	148.1	167.4	
3450	55.4	61.5	67.6	73.6	79.6	85.5	91.3	102.9	114.3	136.4	157.5	177.7	
4000	63.0	70.0	76.9	83.7	90.4	97.1	103.7	116.8	129.5	154.0			
4500	69.7	77.4	85.0	92.6	100.0	107.3	114.5	128.7	142.5				
5000	76.2	84.7	92.9	101.1	109.1	117.1	124.9	140.1	154.9				
5500	82.5	91.6	100.5	109.3	117.9	126.4	134.7	150.9					

8. **Finally, calculate the belt speed and make sure it is less than 6500 fpm:**

$$V = \frac{\pi n P D_d}{12}$$

9. **Now, display your results** and make sure to specify the belt names and what not clearly!

You're done!

Timing belts are probably the easiest to size, and require even less calculations than V-belts. For that reason, I do not have a calculator made for them.

8.3.4 Chain Drives

**This guide follows Mott and Shigley, but mostly Mott*

8.3.4.1 Chain Drives (Mott)

1. **To start off, you will be given the following information**:

   - H_{nom} [hp], nominal horsepower
   - n_1, input speed ($n_{driving}$)
   - n_2, nominal output speed (n_{driven})[26]
   - VR, velocity or speed ratio:

   $$VR = \frac{V_{driven}}{V_{driving}} = \frac{D}{d} = \frac{n_{driving}}{n_{driven}}$$

2. **Find the design horsepower H_d.** Use

   $$H_d = H_{nom} \times SF$$

   where SF is the service factor given in the figure below

   TABLE 7–17 Service Factors for Chain Drives

Load type	Type of driver: Hydraulic drive	Electric motor or turbine	Internal combustion engine with mechanical drive
Smooth Agitators; fans; generators; grinders; centrifugal pumps; rotary screens; light, uniformly loaded conveyors	1.0	1.0	1.2
Moderate shock Bucket elevators; machine tools; cranes; heavy conveyors; food mixers and grinders; ball mills; reciprocating pumps; woodworking machinery	1.2	1.3	1.4
Heavy shock Punch presses; hammer mills; boat propellers; crushers; reciprocating conveyors; rolling mills; logging hoists; dredges; printing presses	1.4	1.5	1.7

3. **Use the tables in Appendix B to choose a chain** (No. 40, 60, or 80) given your H_d and n_1 values.[27]

   After choosing your chain type, you will also be given the the number of teeth on your driving sprocket[28], or N_d, from the same figure (left-most column).

   If you do not find the desired power rating, consider using multiple strands instead! The multiple strand factor is explained at the end of this guide.

[26] Usually a range. For VR calculations, take mid-point to be conservative

[27] Tip: you most certainly can never go wrong with choosing a No. 60 chain. No. 40 often does not work, and No. 80 is usually an overkill.

[28] 17 teeth or above is best

Also, note down the pitch[29], p [in.] for your chosen chain (from the figures) and the rated power value (H_r).

4. **Use N_d and your VR to calculate N_D:**

$$N_D = N_d \times VR$$

Round N_D up to the nearest whole number, and recalculate your VR and output speed, n_2[30].

5. **Calculate the pitch diameters**, PD_d and PD_D [in.] using the following equation:

$$PD_d = \frac{p}{sin(180°/N_d)}$$
$$PD_D = \frac{p}{sin(180°/N_D)}$$

6. **Next, find CD and L_c in pitches**. Start with a trial CD (30 - 50 pitches is a good)[31], and calculate L_c:

$$L_c = 2CD + \frac{N_2 + N_1}{2} + \frac{(N_2 - N_1)^2}{4\pi^2 CD}$$

Round up your L_c value to the nearest **even whole number**. Then recalculate CD:

$$CD = \frac{1}{4}\left[L_c - \frac{N_2 + N_1}{2} + \sqrt{\left[L_c - \frac{N_2 + N_1}{2}\right]^2 - \frac{8(N_2 - N_1)^2}{4\pi^2}}\right]$$

To find L_c and CD in inches, simply multiply by your pitch!

7. **After that, find the wrap angles** [degrees]:

$$\phi_d = 180 - 2sin^{-1}(\frac{PD_D - PD_d}{2CD})$$
$$\phi_D = 180 + 2sin^{-1}(\frac{PD_D - PD_d}{2CD})$$

Note: CD is in inches here!

[29] 0.5 in for No. 40, 0.75 in for No. 60, and 1 in for No. 80
[30] Once again, if the output speed was given as a range, make sure you are still within it
[31] I usually go for 40 pitches

8. **Finally, check how many strands you need**, which is given by:

$$\frac{H_d}{H_r \times Strand\ Factor} < 1$$

where the strand factors are:

- **Strand Factor = 1** for 1 strand
- **Strand Factor = 1.7** for 2 strands
- **Strand Factor = 2.5** for 3 strands
- **Strand Factor = 3.3** for 4 strands

Try them one by one, and see which one works, or rearrange for the strand factor:

$$\frac{H_d}{H_r} < Strand\ Factor$$

The calculator does this for you if you give it the H_r value as well.

You're Done, Good job! Check out the calculator for amazing magic!

Extra note: *When using Shigley, the process is very similar, you just need* N_d *to be 17 to not have to do too much work. Also, the* H_r *value requires to be corrected with two different factors*

8.3.4.2 Chain Drive (Shigleys)

Now looking more in-depth into Shigley's method of chain drive calculations

1. **Here we are starting at step X of the Mott Process**
2. **There are some differences in the calculation method for Nominal horsepower**

8.3.5 Wire Ropes

**This guide follows both Mott and Shigley, but mostly Mott*
This section will have multiple guides for wire ropes. I would suggest that for exam situations, use the Mott Guide[32], unless it is a Mine Hoist problem, then use the corresponding Mine Hoist guide, which is from Shigley. Only use the Mine Hoist guide for sizing wire ropes with an acceleration or deceleration occurring. If you just need to find the Maximum Working Load, use the Mott Guide. For Wire Rope Sizing in Shigley, I will present a brief portion on it as well just in-case.

As for the calculator, I have created one for the mine hoist problems only. Otherwise, the other designs simply require way too many table look-ups, and use very simple equations outside of the look-ups. Therefore, I believe that the guides are clear and should suffice in getting you through as fast as possible.

8.3.5.1 Wire Rope Design (Mott)

1. **In a typical design question, you will have to start with your rope type and rope diameter (d)**. If these are given, skip to the next step. If not, see below.

   - If the diameter is not given, consult the figure below for some possible nominal diameters.

TABLE 7–19 Nominal Wire Rope Diameter

inches	mm	inches	mm
1/4	6.5	2 1/8	54
5/16	8	2 1/4	58
3/8	9.5	2 3/8	60
7/16	11.5	2 1/2	64
1/2	13	2 5/8	67
9/16	14.5	2 3/4	71
5/8	16	2 7/8	74
3/4	19	3	77
7/8	22	3 1/8	80
1	26	3 1/4	83
1 1/8	29	3 3/8	87
1 1/4	32	3 1/2	90
1 3/8	35	3 3/4	96
1 1/2	38	4	103
1 5/8	42	4 1/4	109
1 3/4	45	4 1/2	115
1 7/8	48	4 3/4	122
2	52	5	128

[32] Our final exam asked for a Warrington Wire Rope design, which is only mentioned in Mott, but designing using Shigley should also be completely possible

- If the rope type is not given, you will be given "Wires per Strand" and "Max. No. of outer wires in Strand" instead. Use the next figure to determine your rope classification.

TABLE 7–20 Wire Rope Classification

Classification	Wires per strand	Maximum number of outer wires in strand
6×7	7 through 15	9
6×19	16 through 26	12
6×36	27 through 49	18
6×61	50 through 74	24

*Classifications are the same in 7 and 8 strand wire ropes

2. **Next, use the table below to find the (D/d) ratio** where D is the Tread diameter and d is the rope diameter. Use the ratio alongside your rope diameter to find the tread diameter.

$$D = Ratio \times d$$

TABLE 7–22 Sheave and Drum Diameter Factors

Construction	Suggested *D/d* ratio	Minimum *D/d* ratio
6×7	72	42
6×19 Seale	51	34
6×21 Filler Wire	45	30
6×25 Filler Wire	39	26
6×31 Warrington Seale	39	26
6×36 Warrington Seale	35	23
6×41 Seale Filler Wire	20	20
6×41 Warrington Seale	32	21
6×42 Tiller	21	14
8×19 Seale	41	27
8×25 Filler Wire	32	21

3. **Start by designing the Drum.** Find the minimum groove depth, h, for the drum using

$$h_{drum} = 0.374d$$

Then, the drum diameter is

$$Drum\ Diameter = D + 2h$$

The **fleet angle** should be at least 0.5° but no more than 2°

The pitch distance, p, has a range:

$$2.065r < p < 2.18r$$

where r is the groove radius, given in the figure below for different wire rope diameters. Use "new" unless specified otherwise![33]

[33]EXTRA: if the groove radius is less than the "worn" specification, it must be replaced!!

TABLE 7–23 Recommended Sheave and Drum Groove Radius

Nominal wire rope diameter		Groove radius			
		New		Worn	
inches	mm	inches	mm	inches	mm
1/4	6.5	0.137	3.48	0.129	3.28
3/8	9.5	0.201	5.11	0.190	4.83
1/2	13	0.271	6.88	0.256	6.50
5/8	16	0.334	8.48	0.320	8.13
3/4	19	0.401	10.19	0.380	9.65
7/8	22	0.468	11.89	0.440	11.18
1	26	0.543	13.79	0.513	13.03
1 1/4	32	0.669	16.99	0.639	16.23
1 1/2	38	0.803	20.40	0.759	19.28
1 3/4	45	0.939	23.85	0.897	22.78
2	52	1.070	27.18	1.019	25.88
2 1/2	64	1.338	33.99	1.279	32.49
3	77	1.607	40.82	1.538	39.07
3 1/2	90	1.869	47.47	1.794	45.57
4	103	2.139	54.33	2.050	52.07
4 1/2	115	2.396	60.86	2.298	58.37
5	128	2.663	67.64	2.557	64.95

4. **Next, design the sheave.** Every sheave has a hub and bore for a bearing (don't worry about this yet). You already have determined the tread diameter, D, and the groove radius, r. The **throat angle** must be between 35° and 45°.

   The sheave groove depth is given by a range:

$$1.5d \leq < h_{sheave} \leq 1.75d$$

   The pitch diameter and rim diameter can also be found:

$$Pitch\ Diameter = D + 2d$$

$$Rim\ Diameter = D + 2h$$

5. **Choose the Wire Rope Grade** [34] between Plow Steel (PS), Improved Plow Steel (IPS) and Extra Improved Plow Steel (XIP). Use the table below for some strength references.

TABLE 7–24 Grades of Wire Rope

Grade	Tensile strength
Plow Steel	1570 N/mm^2
Improved Plow Steel	1770 N/mm^2
Extra Improved Plow Steel	1960 N/mm^2

6. **Calculate the maximum working load**. Firsts, find the minimum breaking force of the wire rope based on your diameter and rope grade from the figure on the next page. The force is given in US Tons (2000 lb/ton). The weight per foot is also given in case you need to find the total weight of the rope!

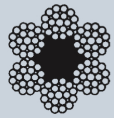
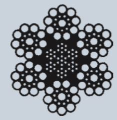
   The Maximum working load is given by

$$Max.\ Working\ Load = \frac{Min.\ Breaking\ Force}{SF}$$

   Where SF is a service factor. A minimum SF of 5 is required for activities concerning human safety such as overhead cranes, gantry cranes, hoists, ... etc.[35]

[34] this may already be given, so skip it
[35] Just use 5 if not sure

TABLE 7–25 6×19 and 6×36 Classes Technical Data

	Fiber core 			IWRC 		
Diameter	Weight per foot	Min breaking force		Weight per foot	Min breaking force	
		IPS	XIP		IPS	XIP
in	lb/ft	tons	tons	lb/ft	tons	tons
1/4	0.105	2.74	3.02	0.116	2.94	3.4
5/16	0.164	4.26	4.69	0.18	4.58	5.27
3/8	0.236	6.1	6.72	0.26	6.56	7.55
7/16	0.32	8.27	9.1	0.35	8.89	10.2
1/2	0.42	10.7	11.8	0.46	11.5	13.3
9/16	0.53	13.5	14.9	0.59	14.5	16.8
5/8	0.66	16.7	18.3	0.72	17.9	20.6
3/4	0.95	23.8	26.2	1.04	25.6	29.4
7/8	1.29	32.2	35.4	1.42	34.6	39.8
1	1.68	41.8	46	1.85	44.9	51.7
1 1/8	2.13	52.6	57.8	2.34	56.5	65
1 1/4	2.63	64.6	71.1	2.89	69.4	79.9
1 3/8	3.18	77.7	85.5	3.5	83.5	96
1 1/2	3.78	92	101	4.16	98.9	114
1 5/8	4.44	107	118	4.88	115	132
1 3/4	5.15	124	137	5.67	133	153
1 7/8	5.91	141	156	6.5	152	174
2	6.72	160	176	7.39	172	198
2 1/8	7.59	179	197	8.35	192	221
2 1/4	8.51	200	220	9.36	215	247

**IWRC = Independent Wire Rope Core*

You are done!

8.3.5.2 Wire Rope Design (Shigley)

**Shigley doesn't provide guides on sizing sheaves and drums, so use the Mott guide instead for those type of questions*

1. **If asked to design a wire rope system via shigley, start by using the 2 figures below** (and whatever is given in the problem, such as rope type) to determine:
   - d [in.], rope diameter
   - D [in.], sheave diameter (use minimum or recommended values if not given one)
   - Weight per foot [lbf]
   - If needed:
     - Material
     - Size of outer wires
     - Modulus of Elasticity
     - Wire Strength
     - A_m [in^2], Area of metal (given in 2nd figure for some types)

Table 17–24 Wire-Rope Data

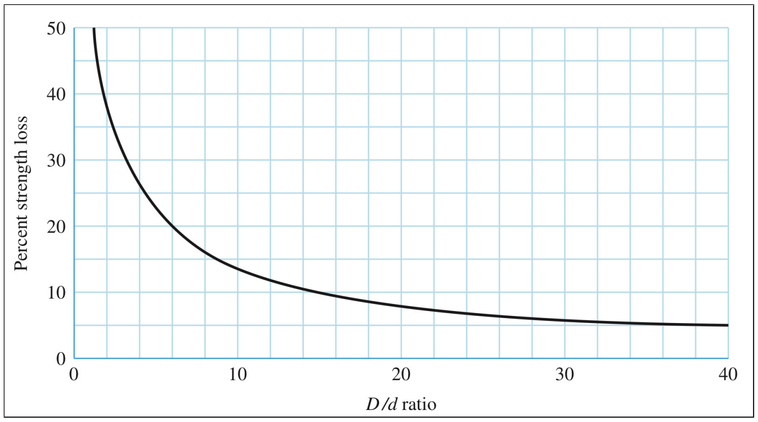
Rope	Weight per Foot, lbf	Minimum Sheave Diameter, in	Standard Sizes d, in	Material	Size of Outer Wires	Modulus of Elasticity,* Mpsi	Strength,† kpsi
6 × 7 haulage	$1.50d^2$	$42d$	$\frac{1}{4}$–$1\frac{1}{2}$	Monitor steel	$d/9$	14	100
				Plow steel	$d/9$	14	88
				Mild plow steel	$d/9$	14	76
6 × 19 standard hoisting	$1.60d^2$	$26d$–$34d$	$\frac{1}{4}$–$2\frac{3}{4}$	Monitor steel	$d/13$–$d/16$	12	106
				Plow steel	$d/13$–$d/16$	12	93
				Mild plow steel	$d/13$–$d/16$	12	80
6 × 37 special flexible	$1.55d^2$	$18d$	$\frac{1}{4}$–$3\frac{1}{2}$	Monitor steel	$d/22$	11	100
				Plow steel	$d/22$	11	88
8 × 19 extra flexible	$1.45d^2$	$21d$–$26d$	$\frac{1}{4}$–$1\frac{1}{2}$	Monitor steel	$d/15$–$d/19$	10	92
				Plow steel	$d/15$–$d/19$	10	80
7 × 7 aircraft	$1.70d^2$	—	$\frac{1}{16}$–$\frac{3}{8}$	Corrosion-resistant steel	—	—	124
				Carbon steel	—	—	124
7 × 9 aircraft	$1.75d^2$	—	$\frac{1}{8}$–$1\frac{3}{8}$	Corrosion-resistant steel	—	—	135
				Carbon steel	—	—	143
19-wire aircraft	$2.15d^2$	—	$\frac{1}{32}$–$\frac{5}{16}$	Corrosion-resistant steel	—	—	165
				Carbon steel	—	—	165

*The modulus of elasticity is only approximate; it is affected by the loads on the rope and, in general, increases with the life of the rope.
†The strength is based on the nominal area of the rope. The figures given are only approximate and are based on 1-in rope sizes and $\frac{1}{4}$-in aircraft-cable sizes.
Source: Compiled from *American Steel and Wire Company Handbook.*

Table 17–27 Some Useful Properties of 6 × 7, 6 × 19, and 6 × 37 Wire Ropes

Wire Rope	Weight per Foot w, lbf/ft	Weight per Foot Including Core w, lbf/ft	Minimum Sheave Diameter D, in	Better Sheave Diameter D, in	Diameter of Wires d_w, in	Area of Metal A_m, in^2	Rope Young's Modulus E_r, psi
6 × 7	$1.50d^2$		$42d$	$72d$	$0.111d$	$0.38d^2$	13×10^6
6 × 19	$1.60d^2$	$1.76d^2$	$30d$	$45d$	$0.067d$	$0.40d^2$	12×10^6
6 × 37	$1.55d^2$	$1.71d^2$	$18d$	$27d$	$0.048d$	$0.40d^2$	12×10^6

2. **Next you will find the the sum of the static loads on your wire rope, and then use the figure below to reduce the strength based on the (D/d) ratio.** The final number will be your F_u, or the ultimate wire load.



Use this value for your safety factor:

$$n = \frac{F_u}{F_t}$$

where F_t is the largest working tension. Often, you will determine a safety factor for the operation and find F_t instead of n by rearranging the equation above. The safety factor is 5 for average operations, but see the figure below for some specific operations.

Table 17–25 Minimum Factors of Safety for Wire Rope*

Track cables	3.2	Passenger elevators, ft/min:	
Guys	3.5	50	7.60
Mine shafts, ft:		300	9.20
Up to 500	8.0	800	11.25
1000–2000	7.0	1200	11.80
2000–3000	6.0	1500	11.90
Over 3000	5.0	Freight elevators, ft/min:	
Hoisting	5.0	50	6.65
Haulage	6.0	300	8.20
Cranes and derricks	6.0	800	10.00
Electric hoists	7.0	1200	10.50
Hand elevators	5.0	1500	10.55
Private elevators	7.5	Powered dumbwaiters, ft/min:	
Hand dumbwaiter	4.5	50	4.8
Grain elevators	7.5	300	6.6
		500	8.0

*Use of these factors does not preclude a fatigue failure.

Source: Compiled from a variety of sources, including ANSI A17.1-1978.

3. **Next, you can calculate the bearing pressure, P [psi], and choose a suitable Sheave Material** for your rope type using the figure below.

$$P = \frac{2F}{dD}$$

Where F [lbf] is the tensile force on the rope

Table 17–26 Maximum Allowable Bearing Pressures of Ropes on Sheaves (in psi)

	Sheave Material				
Rope	Wood[a]	Cast Iron[b]	Cast Steel[c]	Chilled Cast Irons[d]	Manganese Steel[e]
Regular lay:					
6 × 7	150	300	550	650	1470
6 × 19	250	480	900	1100	2400
6 × 37	300	585	1075	1325	3000
8 × 19	350	680	1260	1550	3500
Lang lay:					
6 × 7	165	350	600	715	1650
6 × 19	275	550	1000	1210	2750
6 × 37	330	660	1180	1450	3300

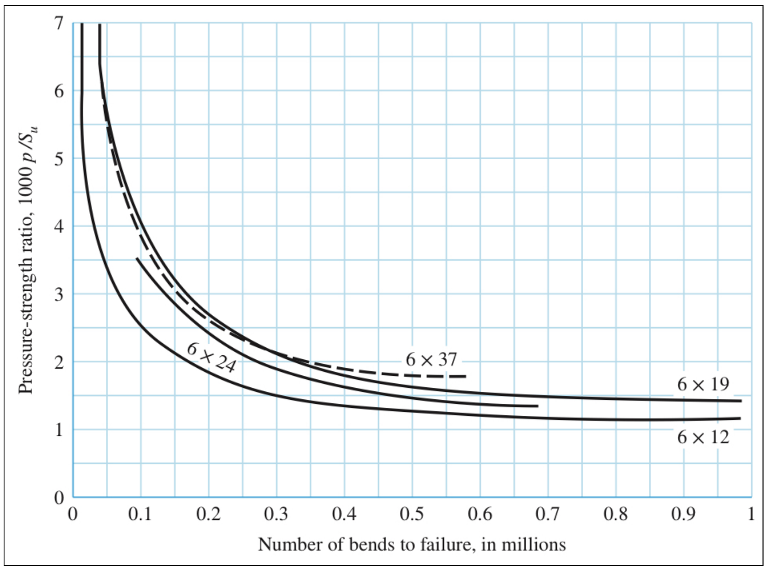
[a]On end grain of beech, hickory, or gum.
[b]For H_B (min.) = 125.
[c]30–40 carbon; H_B (min.) = 160.
[d]Use only with uniform surface hardness.
[e]For high speeds with balanced sheaves having ground surfaces.
Source: Wire Rope Users Manual, AISI, 1979.

4. After that, you can calculate the allowable fatigue tension under flexing, F_f:

$$F_f = \frac{(p/S_u)S_u dD}{2}$$

where (p/S_U)[36] *is the pressure strength ratio from the figure on the next page, and* S_u *is the ultimate tensile strength of the wire, given in ranges for these three materials*[37]*:*

Improved plow steel (monitor)	$240 < S_u < 280$ kpsi
Plow steel	$210 < S_u < 240$ kpsi
Mild plow steel	$180 < S_u < 210$ kpsi



5. Calculate Safety Factors:

   - In Fatigue:

$$n_f = \frac{F_f - F_b}{F_t}$$

   - For Static Loading:

$$n_s = \frac{F_u - F_b}{F_t}$$

[36]Note that the figure gives 1000(p/S_u), so divide whatever you get by 1000. Also, if the number of bends to failure is not given, be as conservative as you can. For example, use 1 million bends for 6x19 ropes

[37]Use the lower bound values to be more conservative. If for some reason you have to find the S_u value, an equation in the book gives it as $S_u = \frac{2000F}{dD}$ for a rope with long life

Where F_t was the largest working tension you determined, and F_b is the bending load, given by:

$$F_b = \frac{E_r d_w A_m}{D}$$

- E_r is the Modulus of Elasticity of the Rope
- d_w is the diameter of the WIRE (not ROPE)
- A_m is the area of the metal

These values are given in the figure from step 1

Calculate whatever you need and you should be done!

8.3.5.3 Mine Hoist Problems (Shigley)

**NOTE: Usually you will not be given d in these types of questions and your goal is to find the best value at the end. Therefore, leave most equations in terms of d!*

1. In mine hoist problems, you will almost always be given the following:
   - a [ft/s^2], maximum acceleration or deceleration experienced
   - W [lbf], weight at the end of the rope
   - L [ft], maximum suspended length of rope
   - m, number of ropes supporting the load
   - Rope Type, one of three[38]:
     - 6 x 7
     - 6 x 19
     - 6 x 37

   In some case, you'll also be given:
   - D [in.], sheave diameter
   - Extra information, such as neglecting bending loads

   In cases where D is not given, use the figure below.

   Table 17–27 Some Useful Properties of 6 × 7, 6 × 19, and 6 × 37 Wire Ropes

Wire Rope	Weight per Foot w, lbf/ft	Weight per Foot Including Core w, lbf/ft	Minimum Sheave Diameter D, in	Better Sheave Diameter D, in	Diameter of Wires d_w, in	Area of Metal A_m, in^2	Rope Young's Modulus E_r, psi
6 × 7	$1.50d^2$		$42d$	$72d$	$0.111d$	$0.38d^2$	13×10^6
6 × 19	$1.60d^2$	$1.76d^2$	$30d$	$45d$	$0.067d$	$0.40d^2$	12×10^6
6 × 37	$1.55d^2$	$1.71d^2$	$18d$	$27d$	$0.048d$	$0.40d^2$	12×10^6

2. Do some initial calculations (or just write down the values) from the figure above. Find:
   - d_w [in.], Diameter of Wires
   - A_m [in^2], Area of Metals
   - E_r [psi], Young's modulus of rope
   - w [lbf/ft], Weight per foot (No core, unless asked!)

3. Next, find different Tension Values (likely in terms of d):

[38] I hope only these three are possible, since it will be hard for you otherwise

- F_t [lbf], Rope Tension

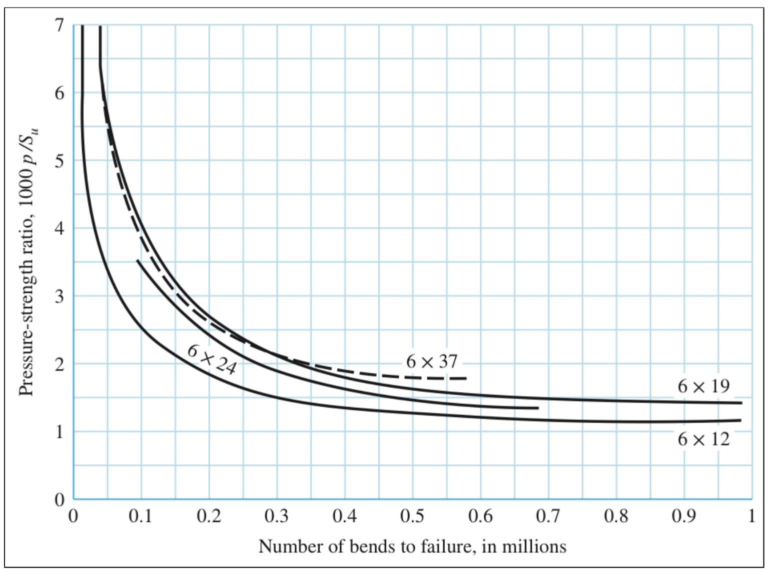
$$F_t = (\frac{W}{m} + wL)(1 + \frac{a}{32.2})$$

- F_f [lbf], Fatigue Tension

$$F_f = \frac{(p/S_u)S_u dD}{2}$$

where (p/S_U)[39] *is the pressure strength ratio from the figure on the next page, and* S_u *is the ultimate tensile strength of the wire, given in ranges for these three materials*[40]*:*

Improved plow steel (monitor)	$240 < S_u < 280$ kpsi
Plow steel	$210 < S_u < 240$ kpsi
Mild plow steel	$180 < S_u < 210$ kpsi



- F_b [lbf], Bending Tension

$$F_b = \frac{E_r d_w A_m}{D}$$

[39]Note that the figure gives 1000(p/S_u), so divide whatever you get by 1000. Also, if the number of bends to failure is not given, be as conservative as you can. For example, use 1 million bends for 6x19 ropes

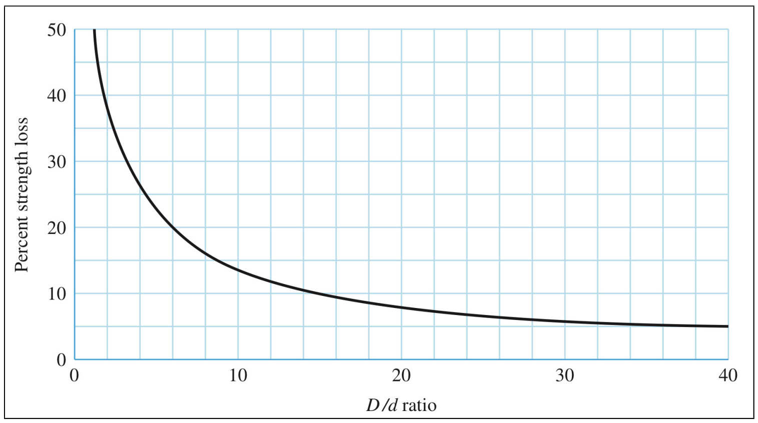
[40]Use the lower bound values to be more conservative.

4. After that, solve the safety factor equations (see which you need):

   - n_s, static safety factor:

$$n_s = \frac{F_u - F_b}{F_t}$$

     sometimes written as only F_u/F_t.For this factor, F_u, the ultimate wire load, is the sum of the static load, with the percent strength loss of the figure below applied[41].



   - n_f, fatigue factor of safety:

$$n_f = \frac{F_f - F_b}{F_t}$$

   With these expressions in terms of d, you will likely solve for the diameter value that will give you the maximum (or desired) safety factor. Use Desmos to easily figure it out. After you find the best diameter, choose the closes standard wire rope size. For standard wire rope diameters, use the figure below, albeit from Mott, or search on the internet!

[41] We were never asked for F_u, but I think maybe the value of W would suffice, since you'll likely not have the value of d anyways

TABLE 7–19 Nominal Wire Rope Diameter			
inches	mm	inches	mm
1/4	6.5	2 1/8	54
5/16	8	2 1/4	58
3/8	9.5	2 3/8	60
7/16	11.5	2 1/2	64
1/2	13	2 5/8	67
9/16	14.5	2 3/4	71
5/8	16	2 7/8	74
3/4	19	3	77
7/8	22	3 1/8	80
1	26	3 1/4	83
1 1/8	29	3 3/8	87
1 1/4	32	3 1/2	90
1 3/8	35	3 3/4	96
1 1/2	38	4	103
1 5/8	42	4 1/4	109
1 3/4	45	4 1/2	115
1 7/8	48	4 3/4	122
2	52	5	128

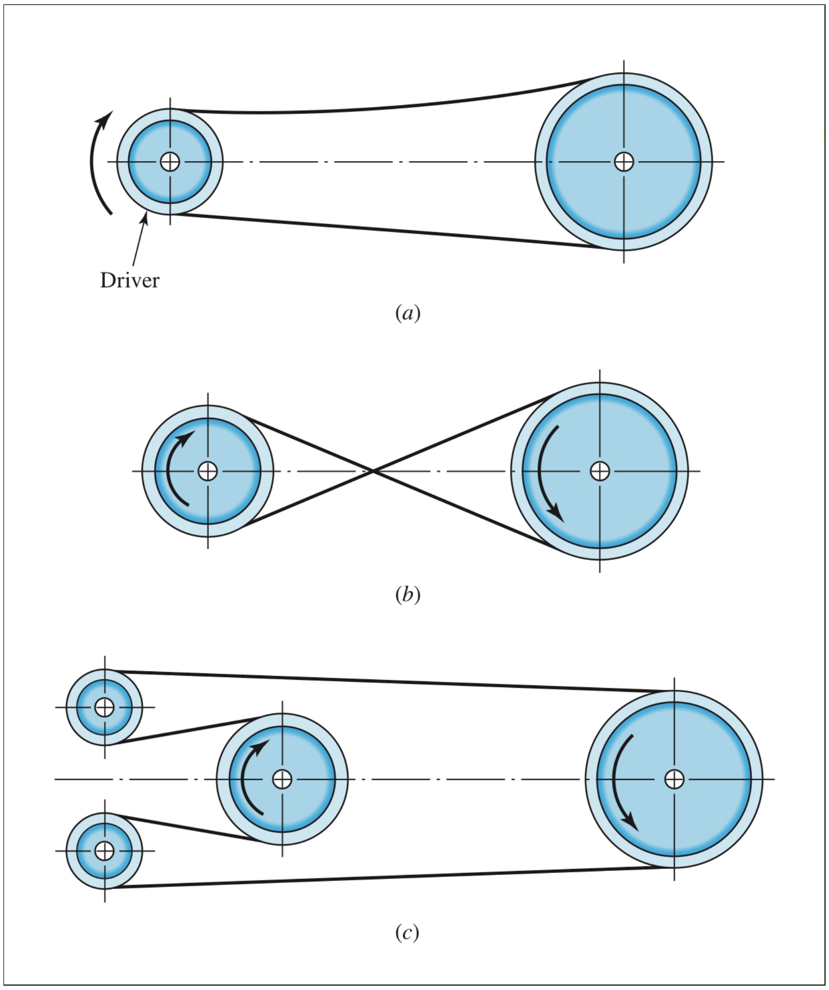
You're done!

For Wire Rope problems, there are 2 calculators, one where it will find you the best d value in a mine hoist question, and one where you give it some initial values and it will calculate things like the safety factors, forces, and what not for you in a regular design question. Only the fatigue safety factor will be used to find the best d value. It will choose the closest standard diameter for you as well.[42]

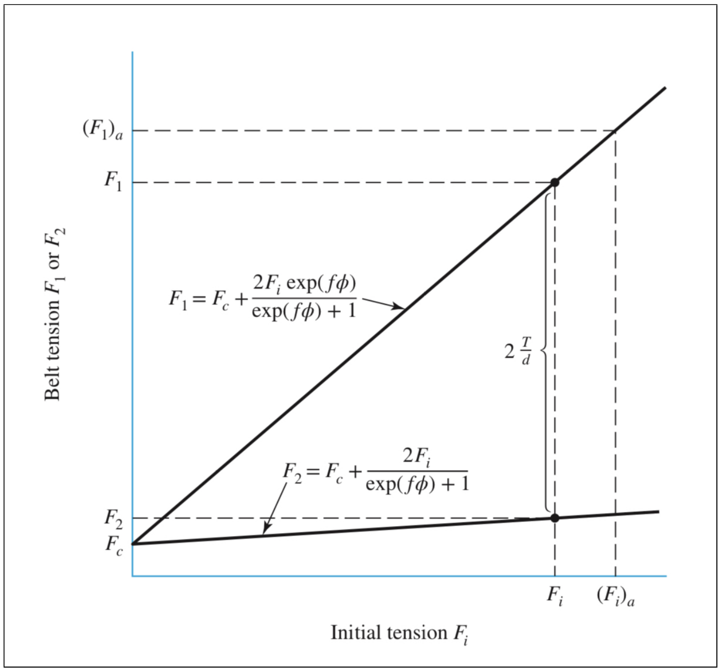
[42] Make sure to choose on of the three ropes I mentioned (6x7, 6x19, and 6x37) or else the calculator will not work, since you need to input the type

8.4 Figures, Definitions, and Relevant Questions

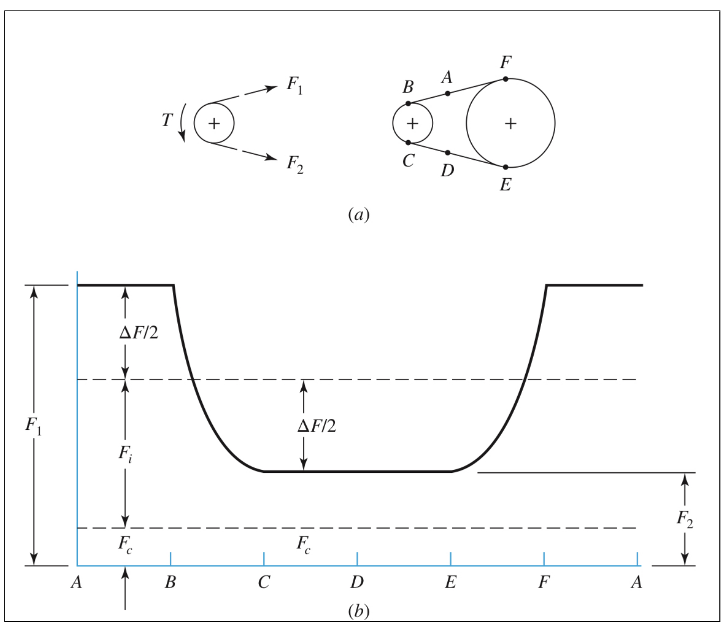
8.4.1 Figures



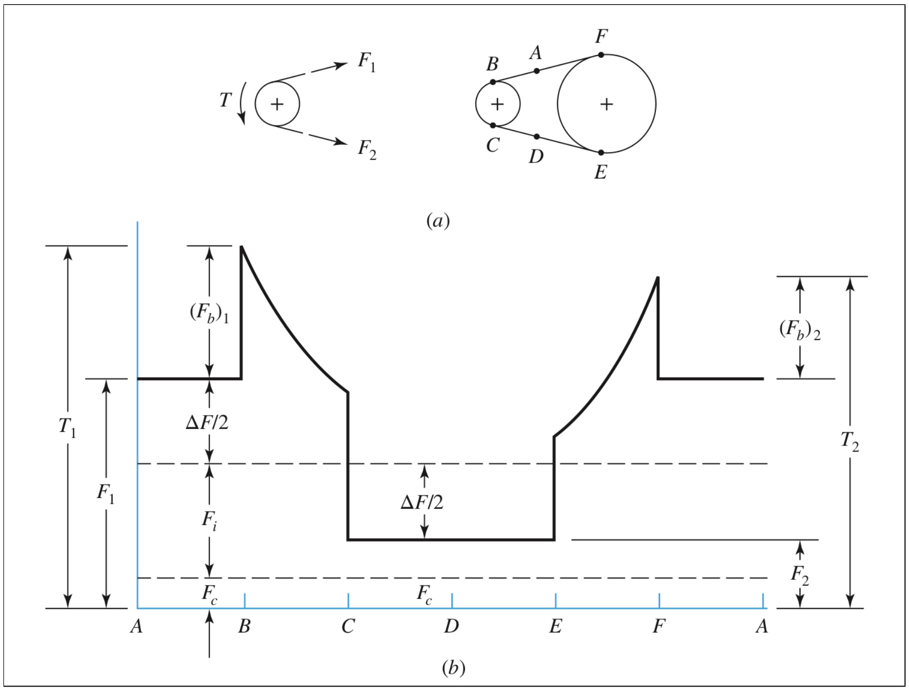
(a) Non-reversing open belt (b) Reversing crossed belt (c) Reversing open belt



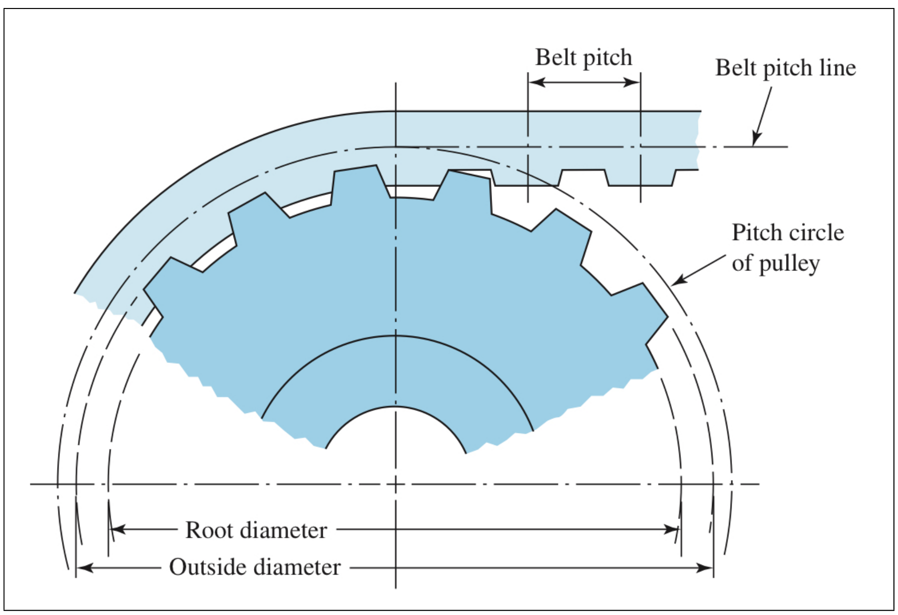
F_1 is tight side tension, F_2 is slack side tension, F_{1a} is maximum allowable tension, and F_c is centrifugal tension



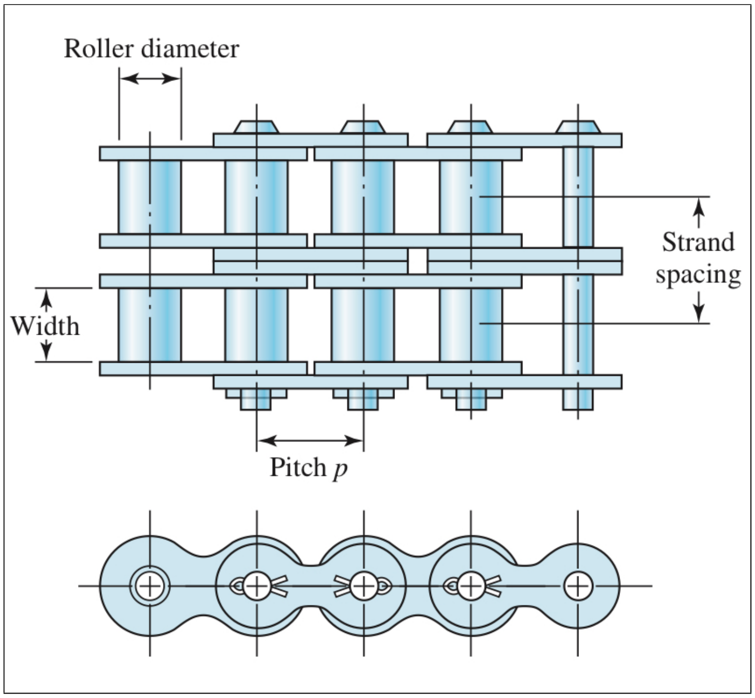
Flat Belt Tensions



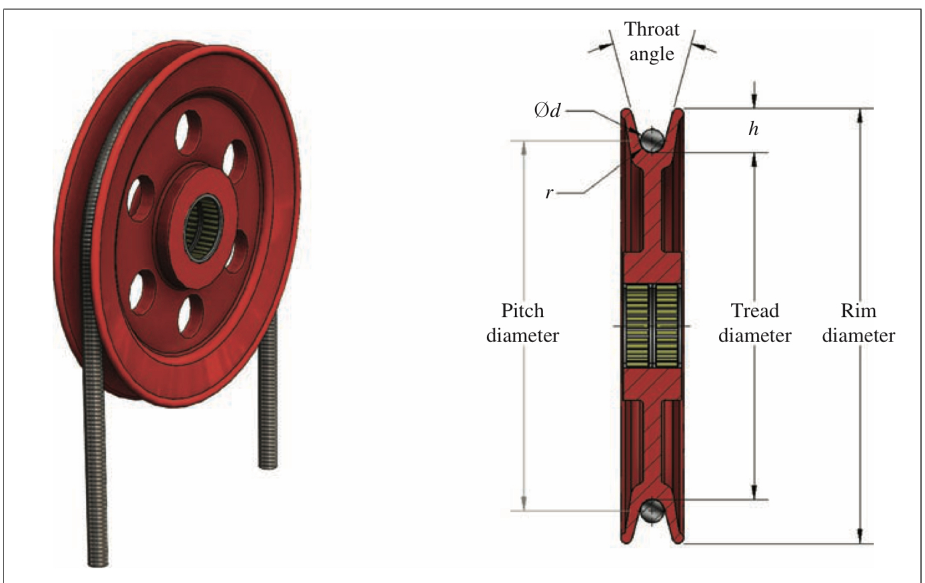
V-Belt Tensions



Timing Belt Drive



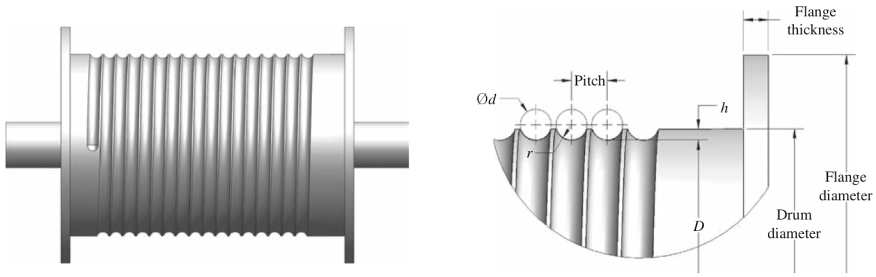
Roller Chain Components (double strand)



(a) Sheave with roller bearing

(b) Sheave dimensional specifications

Wire Rope Sheave

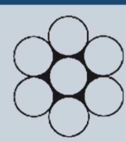
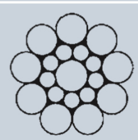
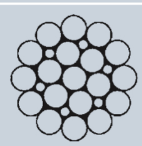
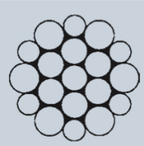
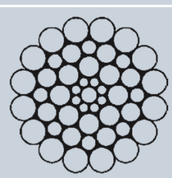


(*a*) Grooved drum with flanges and journal ends

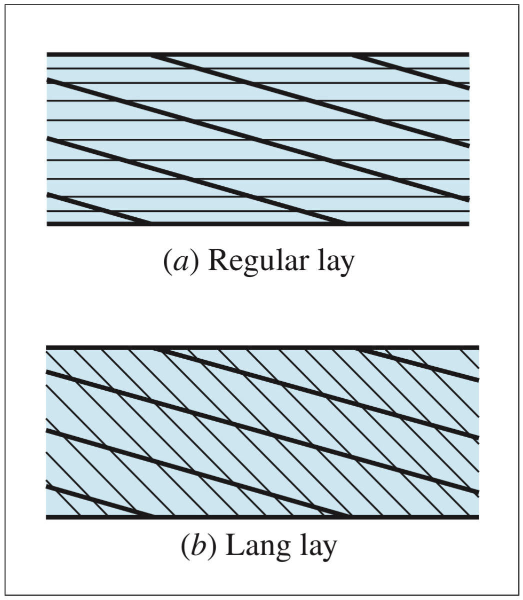
(*b*) Grooved drum dimensional specifications

Wire Rope Drum

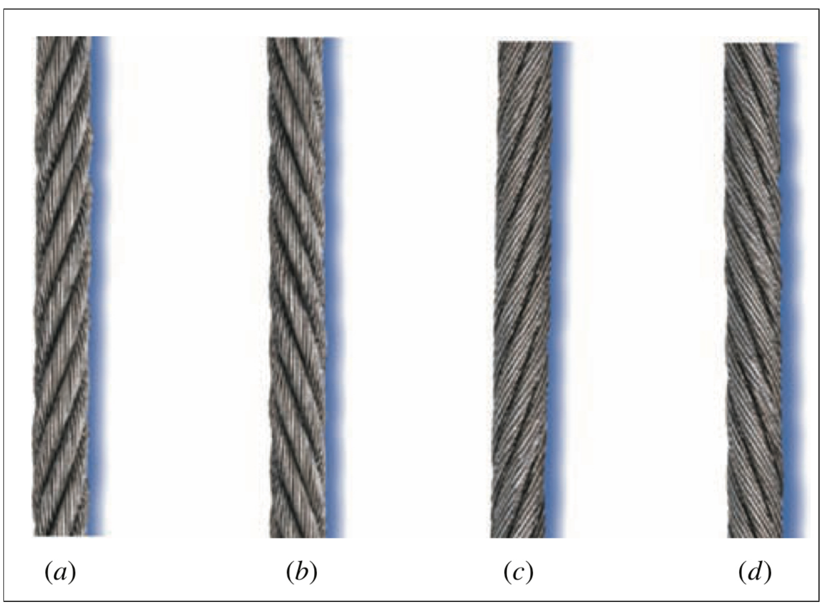
TABLE 7–21 Strand Construction

Single layer		Single wire in the center with six wires of the same diameter
Seale		Equal number of wires in each layer All wires in layer are the same diameter Large outer wires rest in the valley between the small inner wires
Filler wire		Inner layer having half the number of wires as outer layer Smaller filler wires equal in number to the inner layer are laid in the valleys of the inner layer
Warrington		One diameter of wire in the inner layer Two diameters of wire alternating large and small in the outer layer The large outer layer wires rest in the valleys The smaller wires rest on the crowns of the inner layer
Combined pattern		Strand is formed using two or more of the above constructions

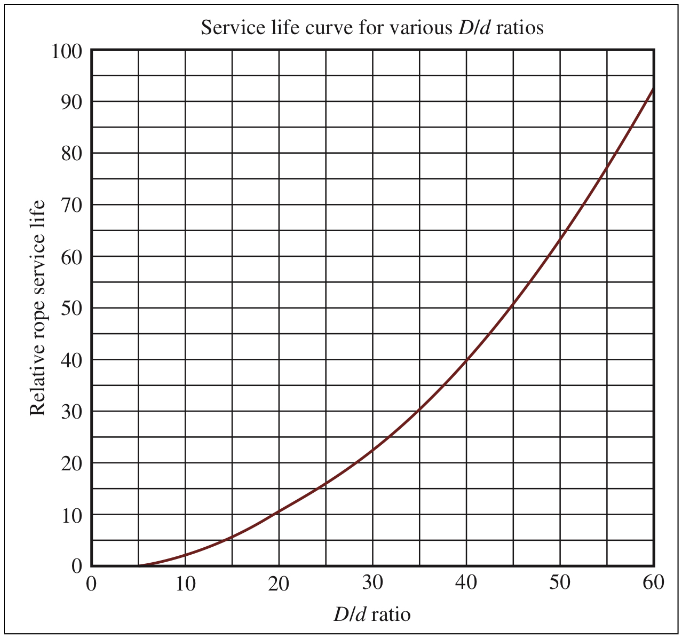
Wire Rope Strand Classification



Types of Wire Ropes (Both are Right Lay)



(a) Right Lay - Regular Lay, (b) Left Lay - Regular Lay, (c) Right Lay - Lang Lay, (d) Left Lay - Lang Lay



Service Life vs. D/d Ratio

The graph shows that as we increase the ratio, the rope service life increases substantially. For example, D/d = 20 gives a relative service life of 10, whilst D/d = 40 gives a life of 40! This is almost a 4x increase.

This concludes unit 1! Good Job!

9 Unit 2: Gears

9.1 Class Memory

With gears, life was still sort of manageable. I remember spending many hours in the ESC study rooms trying to figure this unit's assignment with fellow peers. I think those were times where we were actually managing to learn the content for the first time. In class, Jon used to go quite in detail and spent a good amount of time covering gears. I must say that I do not remember much, except that I was spending countless hours during his Gear lectures trying to fix my group's Gantt Chart for 328.[43]

9.2 Do's and Don'ts

Do's:

- Always be aware of the difference between using normal or transverse diametral pitch/pressure angle in questions.
- Understand the directions of the forces.[44]
- Do your best to properly understand how each gear type differs from the other.

Don'ts:

- Don't do all the calculations by hand (use the Matlab calculators).[45]
- Don't spend too much time on this unit's assignment questions. Some can be quite time consuming, so start early and do the ones you know first.

[43] A bit off-topic, but Gantt Charts are a scam. You can do a much better job without one. My advice would be to create it as you progress, so you don't have to change it much every now and then.

[44] This will be useful and basically essential in the coming units!

[45] Sometimes, it might be easier for you to memorize some essential equations, such as converting the diametral pitch to pitch diameter for example.

9.3 Unit Guide

Well, let's cut to the chase for this one. There are usually 2 types of questions in this section, one where you spec a gear type, find speed ratios, diameters, etc. similar to what we had with belts basically. The 2nd type will be stress calculations. For these, I would highly suggest using the calculators, since a calculation mistake is very easy to make.

For this section, I believe both textbooks will usually do the deed, but in some cases, one textbook is more simplified than the other. I will point out which textbook I will be using for each guide as always.

A few things before we begin:

1. A pinion is the smaller gear (p) and the bigger gear is just called Gear (G). The guide will follow this naming scheme. Exceptions will be noted

2. All calculations you make in this section must be written to four decimal points. This is the way to avoid mistakes my friends.

Make sure to check the calculator as well (here), you will find it to be quite a time saver!

9.3.1 Design Specification Questions

These questions are usually pretty straightforward. Let's get right into it!

9.3.1.1 Spur Gears **using Mott*

In a question involving a spur gear pair, you will be given the following:

- Diametral Pitch (P_d)[46] [teeth/inch]
- Pressure Angle (ϕ)
- Number of teeth for both the Pinion (N_P) and Gear (N_G)

In return, you will have to specify the following:

- Pitch Diameter for Pinion (D_P) and Gear (D_G)
- Circular Pitch (p)
- Addendum (a)
- Dedendum (b)
- Clearance (c)
- Outside Diameters for Pinion (D_{oP}) and Gear (D_{oG})
- Root Diameters for Pinion (D_{RP}) and Gear (D_{RG})
- Whole Depth (h_t)
- Working Depth (h_k)
- Tooth Thickness (t)
- Center Distance (C)
- Base Circle Diameter for Pinion (D_{bP}) and Gear (D_{bG})

I know they are a lot of things to calculate, but the calculator does all of these in 1 click. All units are generally in inches[47]. Anyways, let's begin.

1. **Start by using your given information to calculate the pitch diameters.**[48]

$$D_P = \frac{N_p}{P_d}$$

$$D_G = \frac{N_G}{P_d}$$

[46] Same for both gears

[47] If you are given module (m) in mm instead of P_d, convert using $P_d = \frac{25.4}{m}$

[48] You might also be given one of the diameters in some circumstances. In that case, solve for diametral pitch

2. **Calculate the circular pitch using any of the equations below.**

$$p = \frac{\pi}{P_d}$$

$$p = \pi \frac{D_P}{N_P}$$

$$p = \pi \frac{D_G}{N_G}$$

3. **Calculate the Addendum, Dedendum, and Clearance.**

$$a = \frac{1}{P_d}$$

For Coarse Pitch ($P_d < 20$ teeth/in):

$$b = \frac{1.25}{P_d}$$

$$c = \frac{0.25}{P_d}$$

For Fine Pitch ($P_d > 20$ teeth/in):

$$b = \frac{1.2}{P_d} + 0.002$$

$$c = \frac{0.2}{P_d} + 0.002$$

4. **Find Outside Diameters.**

$$D_{oP} = \frac{N_P + 2}{P_d}$$

$$D_{oG} = \frac{N_G + 2}{P_d}$$

5. **Calculate Root Diameters.**

$$D_{RP} = D_P - 2b$$

$$D_{RG} = D_G - 2b$$

6. **Calculate Whole Depth and Working Depth.**

$$h_t = a + b$$

$$h_k = 2a$$

7. **Calculate Tooth Thickness.**

$$t = \frac{\pi}{2P_d}$$

8. **Calculate Center Distance.**

$$C = \frac{N_G + N_P}{2P_d}$$

9. **Finally, find Base Circle Diameter**

$$D_{bP} = D_P \ cos(\phi)$$
$$D_{bG} = D_G \ cos(\phi)$$

10. **You're done!**

9.3.1.2 Helical Gear **using Mott*

In helical gear questions, you will usually be specifying a single gear only. You will likely be given the following:

- Diametral Pitch (P_d)[49] *or* Normal Diametral Pitch (P_{nd})
- Transverse Pressure Angle (ϕ_t) *or* Normal Pressure Angle (ϕ_n)
- Number of Teeth (N)
- Face Width (F)
- Helix Angle (ψ)

In return, you will need to specify the following:

- Transverse Circular Pitch (p_t)
- Normal Circular Pitch (p_n)
- Axial Pitch (p_x)
- Pitch Diameter (D)
- Whichever diametral pitch and pressure angle you are not given
- Number of Axial Pitches in the Face Width

These questions are very straight-forward. You'll be zoooooming using the calculator! Steps are below.

1. **Start by calculating the normal diametral pitch[50].**

$$P_{nd} = \frac{P_d}{cos(\psi)}$$

2. **Find the Transverse and Normal Circular Pitch.**

$$p_t = \frac{\pi}{P_d}$$

$$p_n = p_t \ cos(\psi)$$

3. **Calculate Axial Pitch.**

$$p_x = \frac{p_t}{tan(\psi)}$$

[49] Sometimes also called the Transverse Diametral Pitch

[50] Use the equation to solve for P_d in-case the question gives you the transverse diametral pitch instead

4. **Calculate Pitch Diameter.**

$$D = \frac{N}{P_d}$$

5. **Find the Normal (or Transverse) Pressure Angle**

$$\phi_n = tan^{-1}[tan(\phi_t)cos(\psi)]$$

$$\phi_t = tan^{-1}\left[\frac{tan(\phi_n)}{cos(\psi)}\right]$$

6. **Find Number of Axial Pitches in Face Width.**

$$No.\ of\ Axial\ Pitches = \frac{F}{p_x}$$

**Full Helical Action if greater than 2*

That's basically it! Good job!

9.3.1.3 Bevel Gears *using Mott*

For bevel gears, you will likely encounter a set of straight bevel gears in a question. In such a question, you will be given the following:

- Diametral Pitch (P_d)
- Number of Teeth in pinion and gear (N_P and N_G)
- Pressure Angle (ϕ)

In return, you will need to provide the following geometric features:

- Gear Ratio (m_G)
- Pitch Diameter for gear and pinion (D and d)
- Pitch Cone angles for pinion and gear (γ and Γ)
- Outer Cone Distance (A_O))
- Face Width[51] (F)
- Mean Cone Distance (A_m)
- Mean Circular Pitch (p_m)
- Mean Working Depth (h)
- Clearance (c)
- Mean Whole Depth (h_m)
- Mean Addendum Factor (c_1)
- Gear Mean Addendum (a_G)
- Pinion Mean Addendum (a_P)
- Gear Mean Dedendum (b_G)
- Pinion Mean Dedendum (b_P)
- Gear Dedendum Angle (δ_G)
- Pinion Dedendum Angle (δ_P)
- Gear Outer Addendum (a_{OG})
- Pinion Outer Addendum (a_{OP})
- Gear Outside Diameter (D_O)

[51] Usually a range rather than a discrete value

- Pinion Outside Diameter (d_O)

As you can see, there is an absolute shit ton of things to calculate. Luckily, the calculator does all of this immediately, so don't be a dummy and use it. I will provide the different equations below, but will not present these as proper steps. It is easier for both of us that way.
*All units are in inches, and angles are in degrees.

1. **Start off by gathering everything that is given and calculating the gear ratio and pitch diameters.**

$$m_G = \frac{N_G}{N_P}$$

$$D = \frac{N_G}{P_d}$$

$$d = \frac{N_P}{P_d}$$

2. **Next, calculate Pitch Cone angles.**

$$\gamma = tan^{-1}(\frac{N_P}{N_G})$$
$$\Gamma = tan^{-1}(\frac{N_G}{N_P})$$

3. **Find outer cone distance.**

$$A_O = \frac{0.5D}{sin(\Gamma)}$$

4. **Determine the face width. Find F_{nom}, then F_{max} using both equations, and choose something (preferably a whole number within the range.**[52]

$$F_{nom} = 0.3A_O$$
$$F_{max} = A_O/3$$
$$F_{max} = 10/P_d$$

**Check both F_{max} equations!*

Then,

$$F_{nom} \leq F \leq F_{max}$$

[52]For the Matlab code, I use the midpoint between the nominal and the max, but you are better off just taking a whole number if applicable

5. **Find the Mean Cone Distance and Mean Working Depth.**

$$A_m = A_O - 0.5F$$
$$p_m = (\frac{\pi}{P_d})(\frac{A_m}{A_O})$$

6. **Calculate Mean Working Depth, Clearance, and Mean Whole Depth.**

$$h = (\frac{2}{P_d})(\frac{A_m}{A_O})$$
$$c = 0.125h$$
$$h_m = h + c$$

7. **Calculate the Mean Addendum and Dedendum for the Gear and Pinion.**

$$c_1 = 0.21 + \frac{0.29}{(m_g)^2}$$
$$a_G = c_1 h$$
$$a_P = h - a_G$$
$$b_G = h_m - a_G$$
$$b_P = h_m - a_P$$

8. **Find the Dedendum Angles for the Gear and Pinion**

$$\delta_G = tan^{-1}(\frac{b_G}{A_m})$$
$$\delta_P = tan^{-1}(\frac{b_P}{A_m})$$

9. **Find the Outer Dedendums for the Gear and Pinion**

$$a_{OG} = a_G + 0.5F\ tan(\delta_P)$$
$$a_{OP} = a_P + 0.5F\ tan(\delta_G)$$

10. **Find Outside Diameters for Gear and Pinion**

$$D_O = D + 2a_{OG}\ cos(\Gamma)$$
$$d_O = d + 2a_{OP}\ cos(\gamma)$$

That's basically it! Good job!

9.3.2 Gear Forces (Not Force Analysis)

This section will cover the forces induced by gears **but does not include the stress calculations, which is instead covered in the next section**. You may need to calculate these forces for these gears before beginning with some foul stress calculations.

9.3.2.1 Spur Gears **using Mott*

We will obviously start with spur gears. These formulas, especially for the tangential force, will translate into the other gears too, so you will probably use one of these formulas no matter the type of gear.

The Torque and the Tangential force:

$$T = \frac{63000P}{n} = \frac{W_t D}{2} \tag{1}$$

$$W_t = \frac{33000P}{\nu_t} = \frac{126000P}{nD}$$

The Radial and Normal forces now:

$$W_r = W_t \tan(\phi)$$

$$W_n = \frac{W_t}{\cos(\phi)} = \sqrt{W_t^2 + W_r^2}$$

9.3.2.2 Helical Gears **using Mott*

Now for helical gears we use the same Torque formula above in Equation (1).

The pitch line speed, which you will need for the tangential force, is given by:

$$\nu_t = \frac{\pi D n}{12}$$

Now, we find the tangential, radial, and the axial forces:

$$W_t = \frac{33000P}{\nu_t} = \frac{126000P}{nD}$$

$$W_r = W_t \tan(\phi_t)$$

$$W_x = W_t \tan(\psi)$$

9.3.2.3 Bevel Gears **using Mott*

Bevel gears will use the pitch cone angles for the pinion and gear (γ & Γ) and will also use the mean radius r_m

$$\nu_t = \frac{\pi D_G n_G}{12} = \frac{\pi D_P n_P}{12}$$
$$W_t = \frac{33000P}{\nu_t} = \frac{126000P}{nD}$$
$$r_m = \frac{d}{2} - \frac{F}{2}\sin(\gamma)$$
$$R_m = \frac{D}{2} - \frac{F}{2}\sin(\Gamma)$$

Now we can find our transmitted (W_t), radial (W_r), and axial (W_x) loads:

$$W_t = \frac{T_P}{r_m} = \frac{T_G}{R_m}$$
$$W_t = \frac{63,000P}{r_{m,p} n_p}\,(\text{lbf})$$
$$W_r = W_t \tan(\phi)\cos(\Gamma) = W_t \tan(\phi)\cos(\gamma)$$
$$W_x = W_t \tan(\phi)\sin(\Gamma) = W_t \tan(\phi)\sin(\gamma)$$

Note that there are two formulas for transmitted force loads which are different from each other. Just use the second one to be honest.

9.3.3 Stress Calculations

Stress Analysis (This guide Follows Mott)

- Typically design using pinion values as the gear will be the same material
- Always use 4 decimal places for your values

9.3.3.1 Spur Gears **using Mott*

Here is the bending stress formulas for a pinion and a gear.

$$s_{tP} = \frac{W_t P_d}{F J_p} K_o K_s K_m K_B K_v$$

$$s_{tG} = s_{tP} \frac{J_P}{J_G}$$

The contact stress formula is:

$$s_c = C_p \sqrt{\frac{W_t K_o K_s K_m K_B K_v}{F D_p I}}$$

The allowable bending stress formulas are:

$$s_{at,P} > s_{tP} \frac{K_R(SF)}{Z_{NP}}$$

$$s_{at,G} > s_{tG} \frac{K_R(SF)}{Z_{NP}}$$

Follow steps 1 - 24 to calculate the bending stress values and to specify a material and SF for it.

Follow steps 25 - 30 to calculate the contact stress value and to specify a material and SF for it.

Let the fun begin!

1. **Find your shock for the input and output using the following:**

> ***Uniform:*** Electric motor or constant-speed gas turbine
>
> ***Light shock:*** Water turbine, variable-speed drive
>
> ***Moderate shock:*** Multicylinder engine
>
> Examples of the roughness of driven machines include the following:
>
> ***Uniform:*** Continuous-duty generator, paper, and film winders.
>
> ***Light shock:*** Fans and low-speed centrifugal pumps, liquid agitators, variable-duty generators, uniformly loaded conveyors, rotary positive displacement pumps, and metal strip processing.
>
> ***Moderate shock:*** High-speed centrifugal pumps, reciprocating pumps and compressors, heavy-duty conveyors, machine tool drives, concrete mixers, textile machinery, meat grinders, saws, bucket elevators, freight elevators, escalators, concrete mixers, plastics molding and processing, sewage disposal equipment, winches, and cable reels.
>
> ***Heavy shock:*** Rock crushers, punch press drives, pulverizers, processing mills, tumbling barrels, wood chippers, vibrating screens, railroad car dumpers, log conveyors, lumber handling equipment, metal shears, hammer mills, commercial washers, heavy-duty hoists and cranes, reciprocating feeders, dredges, rubber processing, compactors, and plastics extruders.

2. **Find your overload factor (K_o) based on the shocks for the driven machine and the power source using Table 9-1 below**

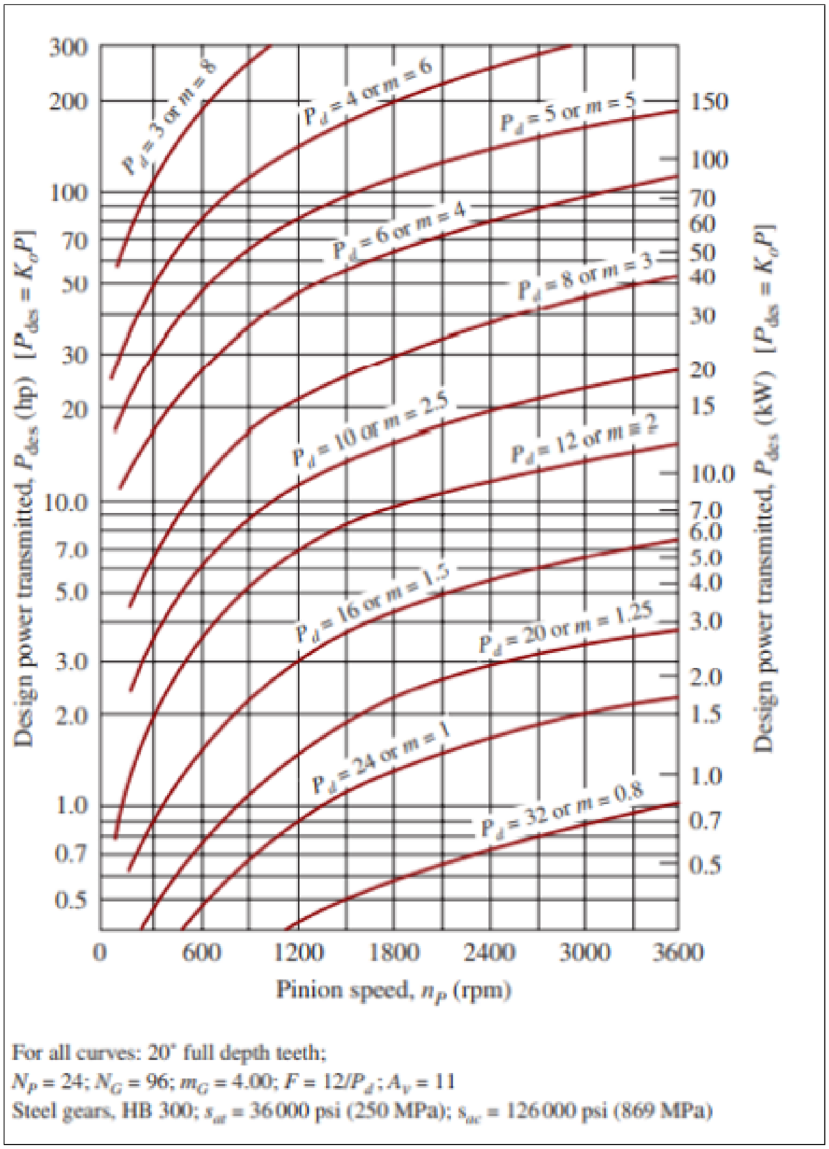
TABLE 9–1 Suggested Overload Factors, K_o

	Driven Machine			
Power source	Uniform	Light shock	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.50	1.75
Light shock	1.20	1.40	1.75	2.25
Moderate shock	1.30	1.70	2.00	2.75

3. **Using your K_o and your input power, calculate your design power (HP) using the following formula:**

$$P_{design} = PK_o$$

4. **Using your design power and the pinion speed, find your diametral pitch (P_d) using the following figure 9-11)**[53]



[53]Round to the smallest number!

5. **Now, choose the number of teeth for the pinion (N_P), using the following range:**

$$17 < N_P < 20$$

6. **Choose the speed of the gear based on the range given in the problem. Typically want to assume a value that is in the middle of the range.**

7. **Calculate the velocity ratio using the following formula:**

$$VR = \frac{n_P}{n_G}$$

Where "n_P" is the rotational speed of the pinion given in the question and "n_G" is the rotational speed of the gear

8. **Calculate the number of teeth in the output gear (N_G) using the following formula:**

$$N_G = N_P(VR)$$

Round to the nearest integer

9. **Now recalculate your velocity ratio using the number of teeth in the pinion and gear:**

$$VR = \frac{N_G}{N_P}$$

10. **Calculate the actual output speed (n_G) using this formula:**

$$n_G = n_P\left(\frac{N_P}{N_G}\right)$$

Make sure this value is within the specified range given in the question, if not, test a new value for N_P.

11. **Find the pitch diameters in inches for the two gears:**

$$D_P = \frac{N_P}{P_d}$$

$$D_G = \frac{N_G}{P_d}$$

12. **Calculate the Centre Distance (CD in inches), pitch line speed, v_t $[ft/min]$, transmitted load W_t $[lbs]$, and radial load W_r (lbs) using the following formulas:**

$$C = \frac{N_P + N_G}{2P_d}$$

$$v_t = \frac{\pi}{12} D_P n_P$$
$$W_t = \frac{33,000P}{v_t}$$
$$W_r = W_t \tan\phi$$

13. **Calculate your face width value, F (inches), given the following ranges:**

    (a) Lower Limit $F = \frac{8}{P_d}$

    (b) Nominal $F = \frac{12}{P_d}$

    (c) Upper Limit $F = \frac{16}{P_d}$

    Just go for the nominal value if the question doesn't mention anything specific

14. **If the quality number A_v is not given you can calculate it using this table:**

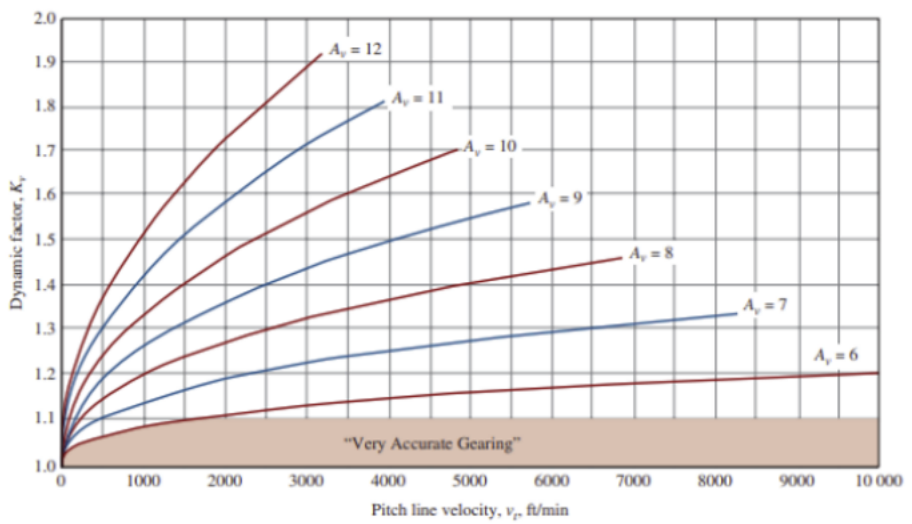
TABLE 9–5 Recommended AGMA Quality Numbers

Application	Quality number	Application	Quality number
Cement mixer drum drive	A11	Small power drill	A9
Cement kiln	A11	Clothes washing machine	A8
Steel mill drives	A11	Printing press	A7
Grain harvester	A10	Computing mechanism	A6
Cranes	A10	Automotive transmission	A6
Punch press	A10	Radar antenna drive	A5
Mining conveyor	A10	Marine propulsion drive	A5
Paper-box-making machine	A9	Aircraft engine drive	A4
Gas meter mechanism	A9	Gyroscope	A2

Machine tool drives and drives for other high-quality mechanical systems

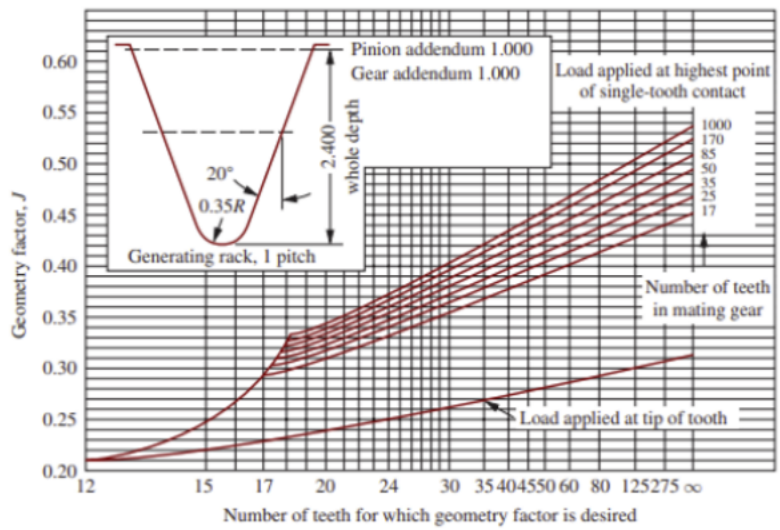
Pitch line speed (fpm)	Quality number	Pitch line speed (m/s)
0–800	A10	0–4
800–2000	A8	4–11
2000–4000	A6	11–22
Over 4000	A4	Over 22

15. **Find the the dynamic factor K_v in the following figure using A_v and v_t:**

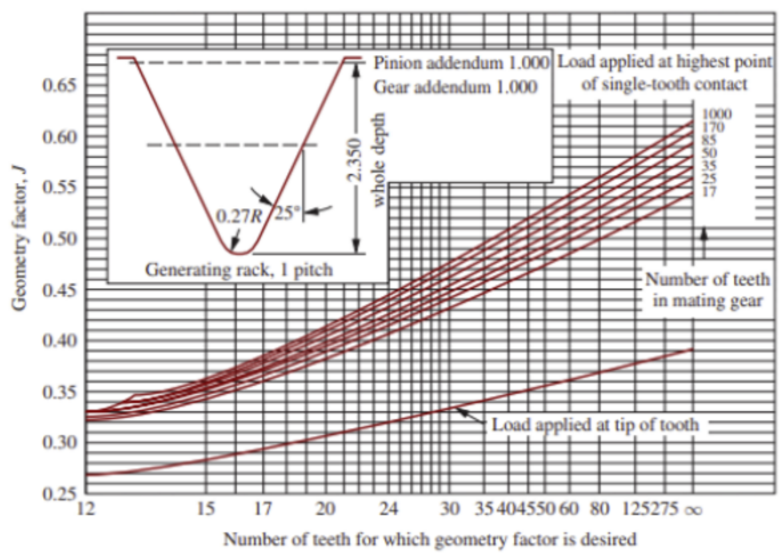


(*a*) U.S. Customary units

16. **Using N_P and N_G, Find the geometry factor "J" using the following graphs. Typically we assume a 20 degree pressure angle spur gear so use that graph and <u>don't dog it too much</u>.**



(a) 20° spur gear: standard addendum



(b) 25° spur gear: standard addendum

17. **Now to find the size factor, Ks, using the following table using your diametral pitch value**[54]

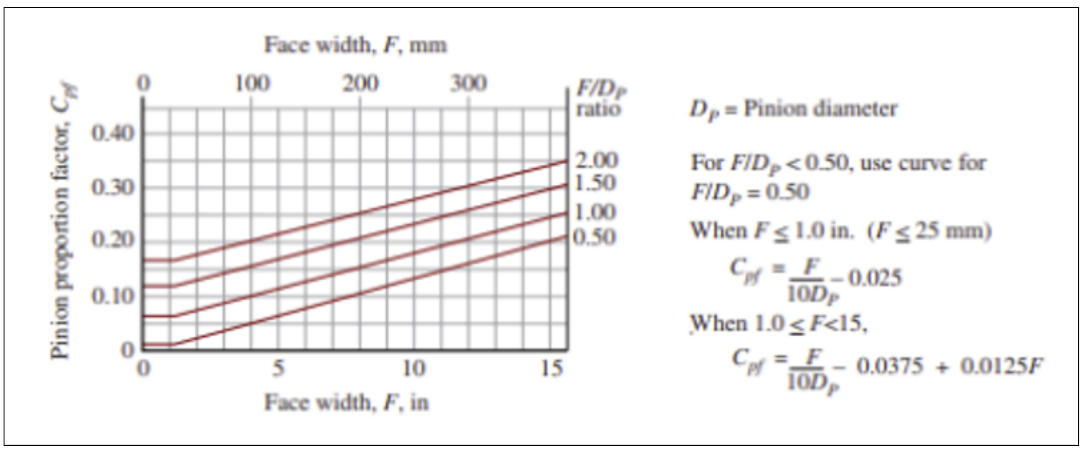
TABLE 9–2 Suggested Size Factors, K_s

Diametral pitch, P_d	Metric module, m	Size factor, K_s
≥ 5	≤ 5	1.00
4	6	1.05
3	8	1.15
2	12	1.25
1.25	20	1.40

18. **In order to find your load-distribution factor, Km, perform the following steps given this formula for K_m.**

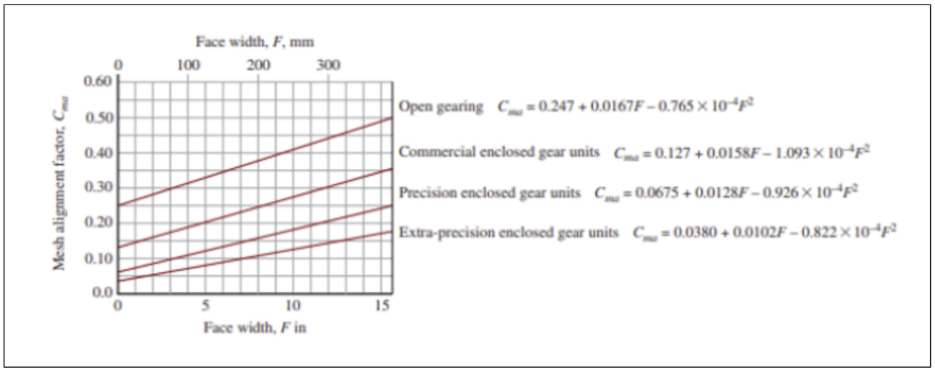
$$K_m = 1 + C_{pf} + C_{ma}$$

Using your face width value and $\frac{F}{D_p}$ ratio, find the pinion proportion factor (C_{pf}) using figure 9-12 below:



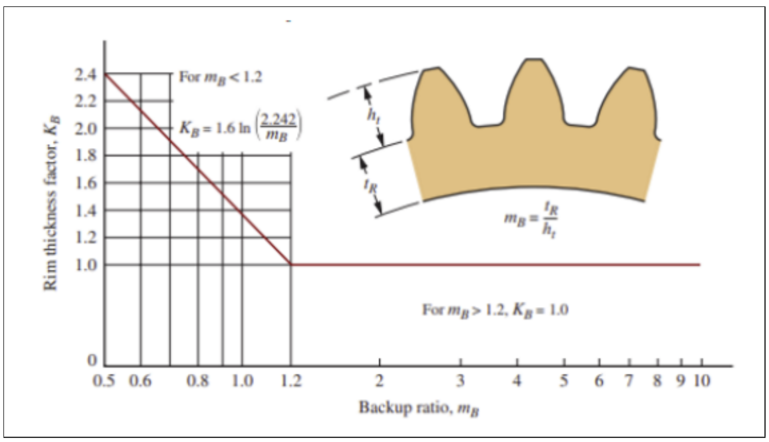
Now we need to find the mesh alignment factor, C_{ma}, using figure 9-13 below. Typically assume that the gears are "Commercial Enclosed Gear Units"

[54] or if you're enough of a dog, you can check the module



Finally find K_m using the formula given in this step.

19. **Find the rim thickness factor, K_b.** Typically this value is 1 for commercially made gears. If this is not the case then use the following figure 9-14 below:



20. **We now have all the factors needed to calculate the bending stress**. Plug everything back into the formula we initially introduced:

$$s_{t,P} = \frac{W_t P_d}{F J_p} K_o K_s K_m K_B K_v$$

$$s_{t,G} = s_{t,P} \frac{J_P}{J_G}$$

21. **You can also calculate the allowable bending stress using the following formula:**

$$s_{at\ P} > s_{tP}\frac{K_R(SF)}{Y_{NP}}$$
$$s_{at\ G} > s_{tG}\frac{K_R(SF)}{Y_{NG}}$$

(a) In order to find the Reliability factor, K_R, use this table:

TABLE 9–11 Reliability Factor, K_R

Reliability	K_R
0.90, one failure in 10	0.85
0.99, one failure in 100	1.00
0.999, one failure in 1000	1.25
0.9999, one failure in 10 000	1.50

Typically, Jon tell us to assume 99% Reliability, so use $K_R = 1$

(b) For the safety factor (SF), assume 1 for now. General range is between 1 and 1.5

(c) To find the stress cycle factor Y_N, we need to first calculate the number of loading cycles, N_c using this formula:

$$N_{cP} = 60 \times \text{lifetime} \times n_P$$
$$N_{cP} = 60 \times \text{lifetime} \times n_G$$

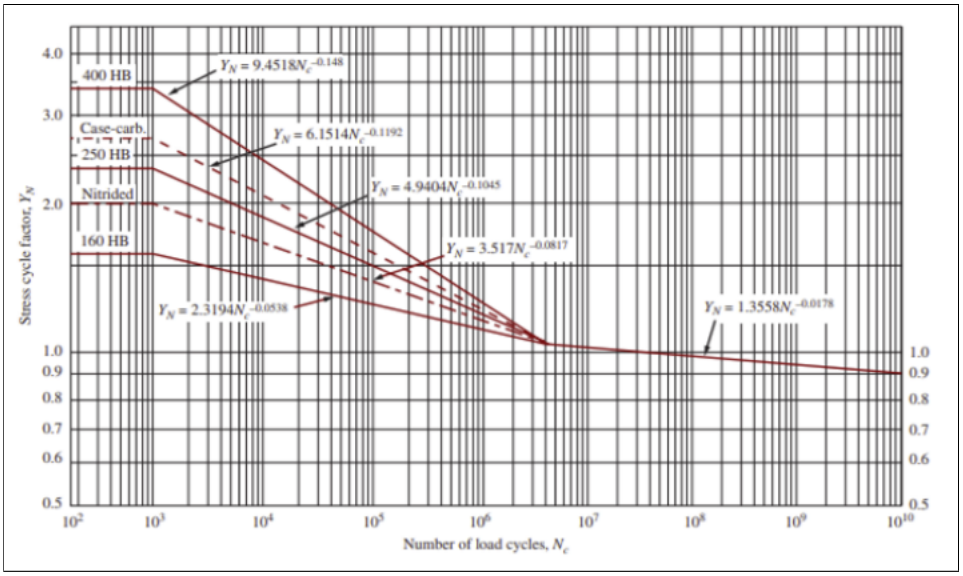
(d) The lifetime could be specified in the question in hours, or can be found using the following table:

TABLE 9–12 Recommended Design Life

Application	Design life (h)
Domestic appliances	1000–2000
Aircraft engines	1000–4000
Automotive	1500–5000
Agricultural equipment	3000–6000
Elevators, industrial fans, multipurpose gearing	8000–15 000
Electric motors, industrial blowers, general industrial machines	20 000–30 000
Pumps and compressors	40 000–60 000
Critical equipment in continuous 24-h operation	100 000–200 000

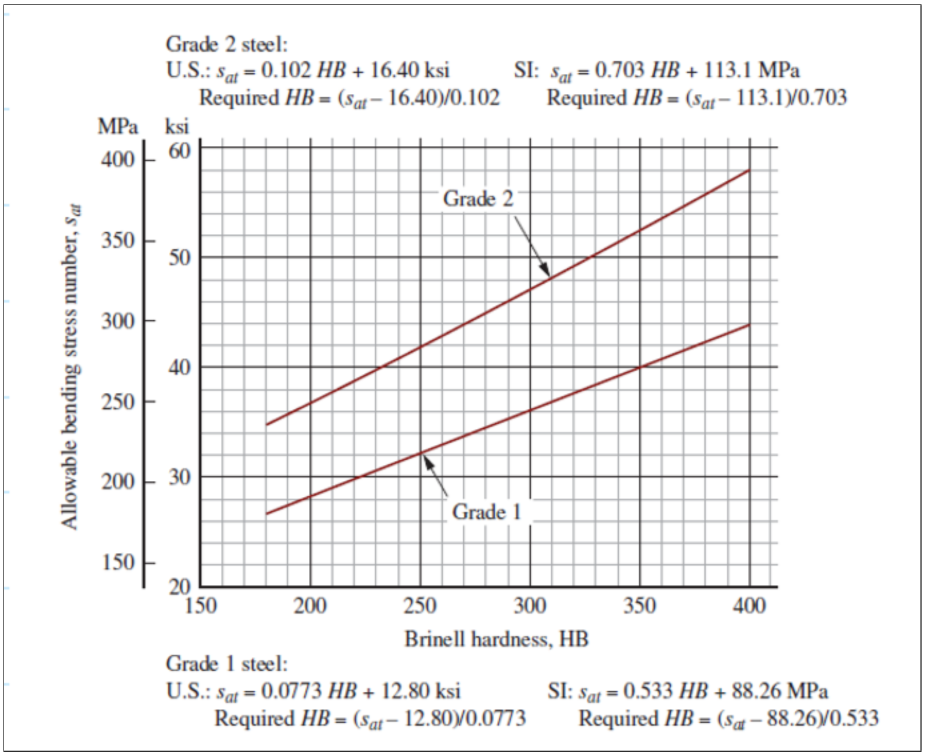
Source: Eugene A. Avallone and Theodore Baumeister III, eds. *Marks' Standard Handbook for Mechanical Engineers.* 9th ed. New York: McGraw-Hill, 1986.

(e) Now use N_c to calculate Y_N using figure 9-21:



22. **Now you can plug everything in to get your allowable bending stress, S_{at}.**

23. **We now have to choose a material. Use figure 9-18 below to find the required Brinell Hardness (HB) for the material.**



We typically use grade 1 steel values. Use ksi for every calculation here.

Now that we have the required hardness, look at appendix 3 or 5 in the textbook and choose any material that has a hardness above the required one. We normally use Appendix 3.

APPENDIX 3 Design Properties of Carbon and Alloy Steels

Material designation (SAE number)	Condition	Tensile strength (ksi)	Tensile strength (MPa)	Yield strength (ksi)	Yield strength (MPa)	Ductility (percent elongation in 2 in)	Brinell hardness (HB)
1020	Hot-rolled	55	379	30	207	25	111
1020	Cold-drawn	61	420	51	352	15	122
1020	Annealed	60	414	43	296	38	121
1040[1]	Hot-rolled	72	496	42	290	18	144
1040	Cold-drawn	80	552	71	490	12	160
1040	OQT 1300	88	607	61	421	33	183
1040	OQT 400	113	779	87	600	19	262
1050	Hot-rolled	90	620	49	338	15	180
1050	Cold-drawn	100	690	84	579	10	200
1050	OQT 1300	96	662	61	421	30	192
1050	OQT 400	143	986	110	758	10	321
1117	Hot-rolled	65	448	40	276	33	124
1117	Cold-drawn	80	552	65	448	20	138
1117	WQT 350	89	614	50	345	22	178
1137	Hot-rolled	88	607	48	331	15	176
1137	Cold-drawn	98	676	82	565	10	196
1137	OQT 1300	87	600	60	414	28	174
1137	OQT 400	157	1083	136	938	5	352
1144[1]	Hot-rolled	94	648	51	352	15	188
1144	Cold-drawn	100	690	90	621	10	200
1144	OQT 1300	96	662	68	469	25	200
1144	OQT 400	127	876	91	627	16	277
1213	Hot-rolled	55	379	33	228	25	110
1213	Cold-drawn	75	517	58	340	10	150
12L13	Hot-rolled	57	393	34	234	22	114
12L13	Cold-drawn	70	483	60	414	10	140
1340[1]	Annealed	102	703	63	434	26	207
1340	OQT 1300	100	690	75	517	25	235
1340	OQT 1000	144	993	132	910	17	363
1340	OQT 700	221	1520	197	1360	10	444
1340	OQT 400	285	1960	234	1610	8	578
3140	Annealed	95	655	67	462	25	187
3140	OQT 1300	115	792	94	648	23	233
3140	OQT 1000	152	1050	133	920	17	311
3140	OQT 700	220	1520	200	1380	13	461
3140	OQT 400	280	1930	248	1710	11	555
4130	Annealed	81	558	52	359	28	156
4130	WQT 1300	98	676	89	614	28	202
4130	WQT 1000	143	986	132	910	16	302
4130	WQT 700	208	1430	180	1240	13	415
4130	WQT 400	234	1610	197	1360	12	461
4140[1]	Annealed	95	655	54	372	26	197
4140	OQT 1300	117	807	100	690	23	235
4140	OQT 1000	168	1160	152	1050	17	341
4140	OQT 700	231	1590	212	1460	13	461
4140	OQT 400	290	2000	251	1730	11	578
4150	Annealed	106	731	55	379	20	197
4150	OQT 1300	127	880	116	800	20	262
4150	OQT 1000	197	1360	181	1250	11	401
4150	OQT 700	247	1700	229	1580	10	495
4150	OQT 400	300	2070	248	1710	10	578

(Continued)

APPENDIX 3 (*Continued*)

Material designation (SAE number)	Condition	Tensile strength (ksi)	Tensile strength (MPa)	Yield strength (ksi)	Yield strength (MPa)	Ductility (percent elongation in 2 in)	Brinell hardness (HB)
4340[1]	Annealed	108	745	68	469	22	217
4340	OQT 1300	140	965	120	827	23	280
4340	OQT 1000	171	1180	158	1090	16	363
4340	OQT 700	230	1590	206	1420	12	461
4340	OQT 400	283	1950	228	1570	11	555
5140	Annealed	83	572	42	290	29	167
5140	OQT 1300	104	717	83	572	27	207
5140	OQT 1000	145	1000	130	896	18	302
5140	OQT 700	220	1520	200	1380	11	429
5140	OQT 400	276	1900	226	1560	7	534
5150	Annealed	98	676	52	359	22	197
5150	OQT 1300	116	800	102	700	22	241
5150	OQT 1000	160	1100	149	1030	15	321
5150	OQT 700	240	1650	220	1520	10	461
5150	OQT 400	312	2150	250	1720	8	601
5160	Annealed	105	724	40	276	17	197
5160	OQT 1300	115	793	100	690	23	229
5160	OQT 1000	170	1170	151	1040	14	341
5160	OQT 700	263	1810	237	1630	9	514
5160	OQT 400	322	2220	260	1790	4	627
6150[1]	Annealed	96	662	59	407	23	197
6150	OQT 1300	118	814	107	738	21	241
6150	OQT 1000	183	1260	173	1190	12	375
6150	OQT 700	247	1700	223	1540	10	495
6150	OQT 400	315	2170	270	1860	7	601
8650	Annealed	104	717	56	386	22	212
8650	OQT 1300	122	841	113	779	21	255
8650	OQT 1000	176	1210	155	1070	14	363
8650	OQT 700	240	1650	222	1530	12	495
8650	OQT 400	282	1940	250	1720	11	555
8740	Annealed	100	690	60	414	22	201
8740	OQT 1300	119	820	100	690	25	241
8740	OQT 1000	175	1210	167	1150	15	363
8740	OQT 700	228	1570	212	1460	12	461
8740	OQT 400	290	2000	240	1650	10	578
9255	Annealed	113	780	71	490	22	229
9255	Q&T 1300	130	896	102	703	21	262
9255	Q&T 1000	181	1250	160	1100	14	352
9255	Q&T 700	260	1790	240	1650	5	534
9255	Q&T 400	310	2140	287	1980	2	601

Notes: Properties common to all carbon and alloy steels:
Poisson's ratio: 0.27.
Shear modulus: 11.5×10^6 psi; 80 GPa.
Coefficient of thermal expansion: $6.5 \times 10^{-6}\,°F^{-1}$.
Density: 0.283 lb/in^3; 7680 kg/m^3.
Modulus of elasticity: 30×10^6 psi; 207 GPa.
[1]See Appendix 4 for graphs of properties versus heat treatment.

APPENDIX 5 Properties of Carburized Steels

Material designation (SAE number)	Condition	Core properties: Tensile strength (ksi)	Tensile strength (MPa)	Yield strength (ksi)	Yield strength (MPa)	Ductility (percent elongation in 2 in)	Brinell hardness (HB)	Case hardness (HRC)
1015	SWQT 350	106	731	60	414	15	217	62
1020	SWQT 350	129	889	72	496	11	255	62
1022	SWQT 350	135	931	75	517	14	262	62
1117	SWQT 350	125	862	66	455	10	235	65
1118	SWQT 350	144	993	90	621	13	285	61
4118	SOQT 300	143	986	93	641	17	293	62
4118	DOQT 300	126	869	63	434	21	241	62
4118	SOQT 450	138	952	89	614	17	277	56
4118	DOQT 450	120	827	63	434	22	229	56
4320	SOQT 300	218	1500	178	1230	13	429	62
4320	DOQT 300	151	1040	97	669	19	302	62
4320	SOQT 450	211	1450	173	1190	12	415	59
4320	DOQT 450	145	1000	94	648	21	293	59
4620	SOQT 300	119	820	83	572	19	277	62
4620	DOQT 300	122	841	77	531	22	248	62
4620	SOQT 450	115	793	80	552	20	248	59
4620	DOQT 450	115	793	77	531	22	235	59
4820	SOQT 300	207	1430	167	1150	13	415	61
4820	DOQT 300	204	1405	165	1140	13	415	60
4820	SOQT 450	205	1410	184	1270	13	415	57
4820	DOQT 450	196	1350	171	1180	13	401	56
8620	SOQT 300	188	1300	149	1030	11	388	64
8620	DOQT 300	133	917	83	572	20	269	64
8620	SOQT 450	167	1150	120	827	14	341	61
8620	DOQT 450	130	896	77	531	22	262	61
E9310	SOQT 300	173	1190	135	931	15	363	62
E9310	DOQT 300	174	1200	139	958	15	363	60
E9310	SOQT 450	168	1160	137	945	15	341	59
E9310	DOQT 450	169	1170	138	952	15	352	58

Notes: Properties given are for a single set of tests on 1/2-in round bars.
SWQT: single water-quenched and tempered.
SOQT: single oil-quenched and tempered.
DOQT: double oil-quenched and tempered.
300 and 450 are the tempering temperatures in °F. Steel was carburized for 8 h. Case depth ranged from 0.045 to 0.075 in.

Anything that has a hardness over 400 should use flame or induction hardening techniques over through hardening, since they provide the strength required for high contact stress because their inside is still ductile to prevent failure.

TABLE 9–8 Examples of Gear Materials

Heat treatment	Typical alloys (SAE numbers)
Through-hardened or Case-hardened by flame or induction hardening	1045, 4140, 4150, 4340, 4350
Carburizing, case hardened	1020, 4118, 4320, 4820, 8620, 9310

24. **We now calculate our actual safety factor**. Use the formula in figure 9-18 to calculate the allowable bending stress value using the hardness for the material you chose. This formula is also what you will use to calculate your Safety factor in the end:

$$SF = \frac{s_{at} Y_n}{s_t K_r}$$

25. **Now we move to calculating the contact stress** using the following formula:

$$s_c = C_p \sqrt{\frac{W_t K_o K_s K_m K_B K_v}{F D_p I}}$$

We have most of these constants from the bending stress calculations. Key differences are C_p and I, which will be shown below

26. **For the elastic coefficient**, C_p, we generally assume a material of steel which has a C_p of 2300. There are other materials shown here as well:

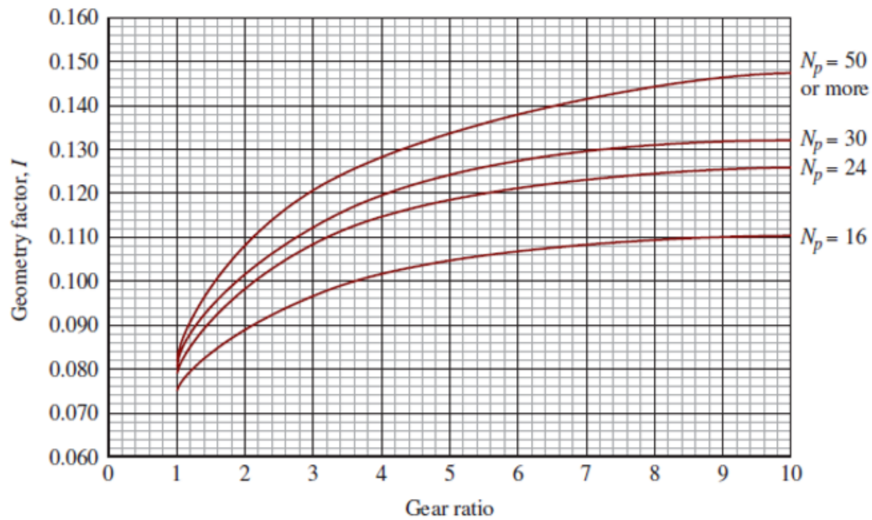
TABLE 9–7 Elastic Coefficient, C_p

		Gear material and modulus of elasticity, E_G, lb/in² (MPa)					
Pinion material	Modulus of elasticity, E_P, lb/in² (MPa)	Steel 30×10^6 (2×10^5)	Malleable iron 25×10^6 (1.7×10^5)	Nodular iron 24×10^6 (1.7×10^5)	Cast iron 22×10^6 (1.5×10^5)	Aluminum bronze 17.5×10^6 (1.2×10^5)	Tin bronze 16×10^6 (1.1×10^5)
Steel	30×10^6	2300	2180	2160	2100	1950	1900
	(2×10^5)	(191)	(181)	(179)	(174)	(162)	(158)
Mall. iron	25×10^6	2180	2090	2070	2020	1900	1850
	(1.7×10^5)	(181)	(174)	(172)	(168)	(158)	(154)
Nod. iron	24×10^6	2160	2070	2050	2000	1880	1830
	(1.7×10^5)	(179)	(172)	(170)	(166)	(156)	(152)
Cast iron	22×10^6	2100	2020	2000	1960	1850	1800
	(1.5×10^5)	(174)	(168)	(166)	(163)	(154)	(149)
Al. bronze	1.75×10^6	1950	1900	1880	1850	1750	1700
	(1.2×10^5)	(162)	(158)	(156)	(154)	(145)	(141)
Tin bronze	16×10^6	1900	1850	1830	1800	1700	1650
	(1.1×10^5)	(158)	(154)	(152)	(149)	(141)	(137)

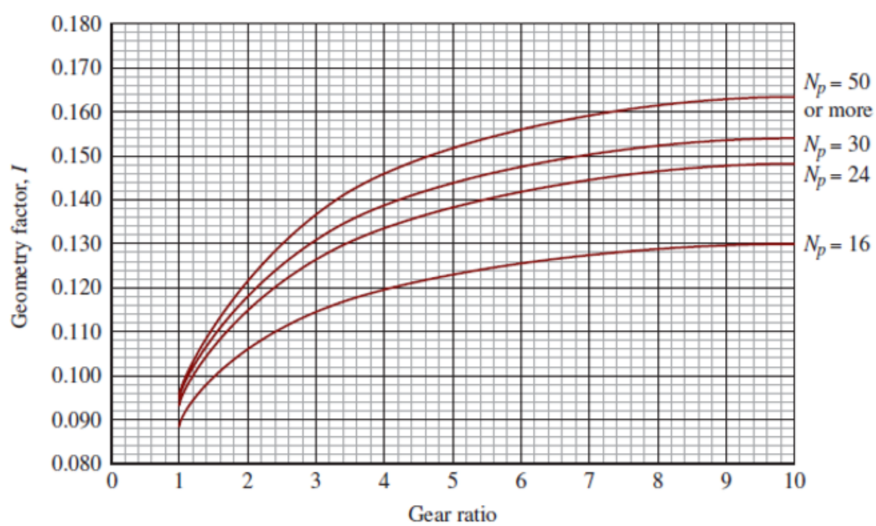
Source: Extracted from AGMA Standard 2001-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1001 North Fairfax Street, 5th floor, Alexandria, VA 22314.

Note: Poisson's ratio = 0.30; units for C_p are $(\text{lb/in}^2)^{0.5}$ or $(\text{MPa})^{0.5}$.

27. **Using the actual gear ratio and Np, Find the pitting geometry factor, I**, using the following graphs. Again, assume a 20 degree pressure angle, so use that graph. Figure 9-17



(a) 20° pressure angle, full-depth teeth (standard addendum = $1/P_d$)



(b) 25° pressure angle, full-depth teeth (standard addendum = $1/P_d$)

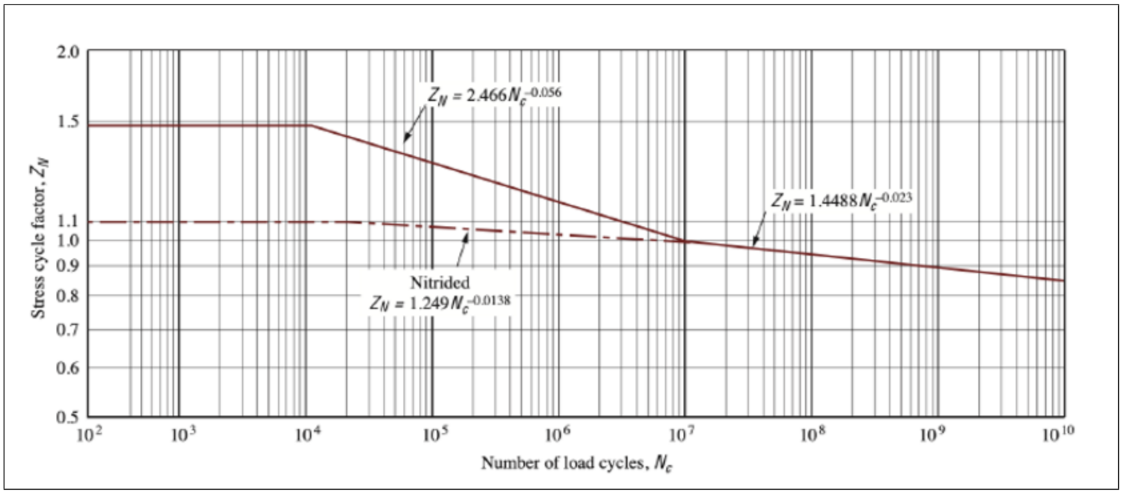
28. **We can now plug everything in to find the contact stress,** s_c

29. **In order to find the allowable contact stress,** s_{ac}**, we use this formula:**

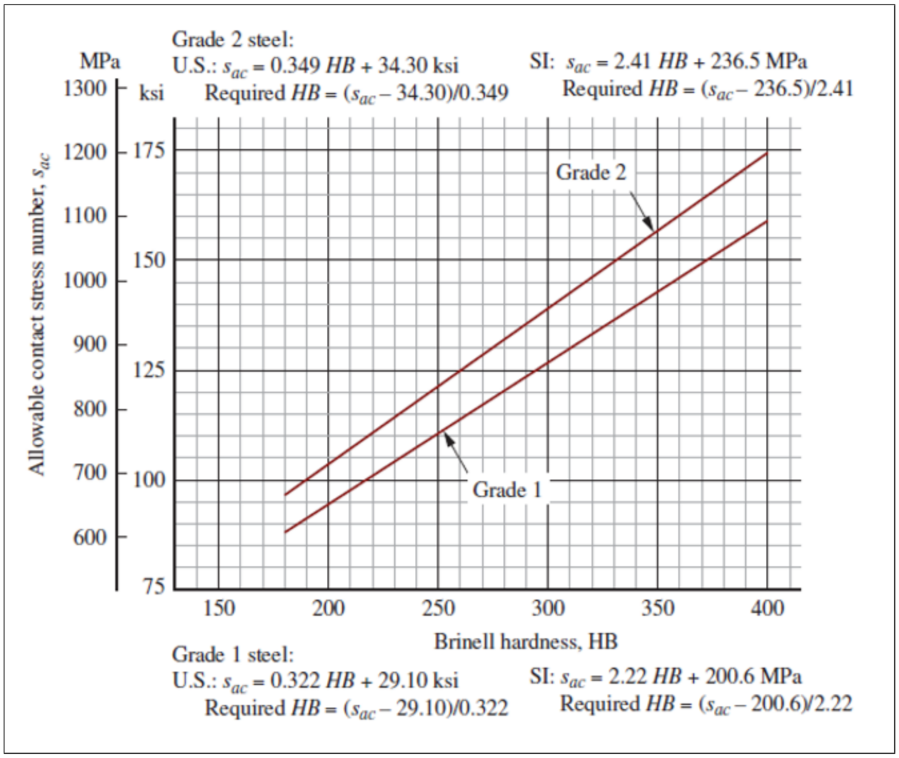
$$s_{ac,P} > s_{cP}\frac{K_R(SF)}{Z_{NP}}$$

$$s_{ac,G} > s_{cG}\frac{K_R(SF)}{Z_{NG}}$$

We already assumed a K_R of 1 and a safety factor of 1 in step 21. Now we need to find Z_N which is the stress cycle factor. We use the following figure 9-22:



30. **We follow a similar procedure to specify the material and to find the safety factor using the contact stresses**. Although the formulas change slightly, use figure 9-19 below



Follow the same procedure for bending stress material specification and safety factor calculations. This is what you will use to calculate your Safety factor in the end:

$$SF = \frac{s_{ac}Z_n}{s_c K_r}$$

9.3.3.2 Helical Gears **using Mott*

The process for calculating helical gear stress is very similar to spur gears, with the key differences being that J and I change!. Here's how it goes:

1. **Find your shock for the input and output using the following**:
2. **Find your overload factor (K_o) based on the shocks for the driven machine and the power source using Table 9-1 below**
3. **Using your K_o and your input power, calculate your design power (HP) using the following formula**:

$$P_{des} = PK_o$$

4. **In order to find the normal diametrical pitch and number of teeth in the pinion, just use these random numbers $P_{nd} = 12$ and $Np = 24$**

5. **Calculate the diametrical pitch using this formula**:

$$P_d = P_{nd} \cos(\psi)$$

   Use the helix angle given in the question

6. **Calculate the axial pitch p_x using this formula**:

$$p_x = \frac{\pi}{P_d \tan(\psi)}$$

7. **Assume that n_G is given, if not then use a similar process in the spur gear section to find it. Use the gear ratio to get the number of teeth in the gear, N_G**

$$VR = \frac{N_G}{N_P} = \frac{n_p}{n_G}$$

8. **Calculate the tangential pressure angle or the normal pressure angle**:

$$\phi_t = \arctan(\frac{\tan(\phi_n)}{\cos(\psi)})$$

$$\phi_t = \arctan(\tan(\phi_t)\cos(\psi))$$

9. **Calculate the diameters of the gears cause fuck it why not**:

$$D_p = \frac{N_P}{P_d}$$

$$D_G = \frac{N_G}{P_d}$$

   Look the P_d we are using is not the Normal one its the new one just keep that in mind

10. **Find the nominal face width**:

$$F_{nom} = 2p_x$$

11. **Find Center distance (inch), pitch line speed (feet/min), and transmitted load (lbs)**:

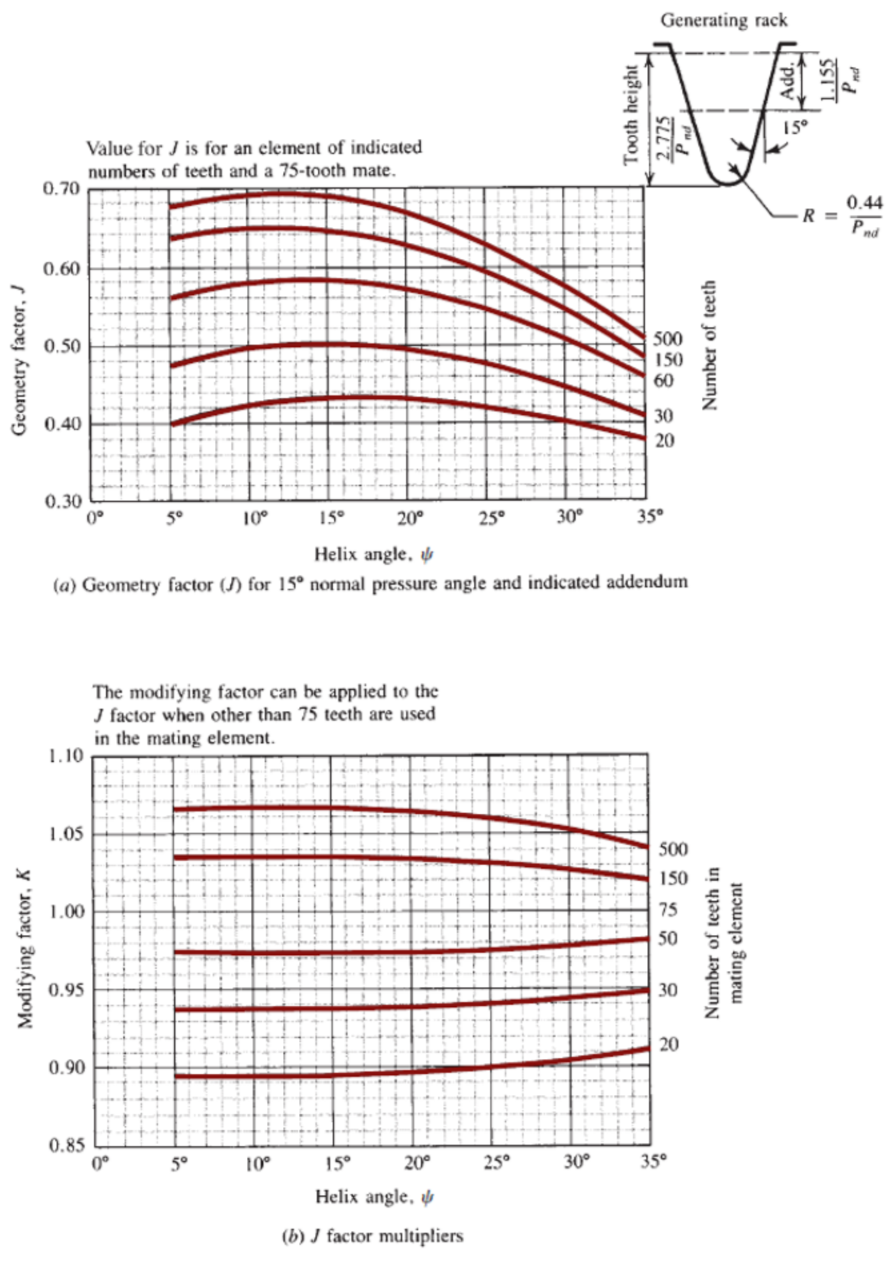
$$C = \frac{N_P + N_G}{2P_d}$$

$$v_t = \frac{\pi}{12} D_p n_p$$

$$W_t = \frac{33000P}{\nu_t}$$

12. **Now follow the same procedure as the spur gears to find the stresses, J and I will be different though**.

13. **Choose J from the graphs below depending on the normal pressure angle**



(a) Geometry factor (J) for 15° normal pressure angle and indicated addendum

(b) J factor multipliers

FIGURE 10–5 Geometry factor (J) for 15° normal pressure angle

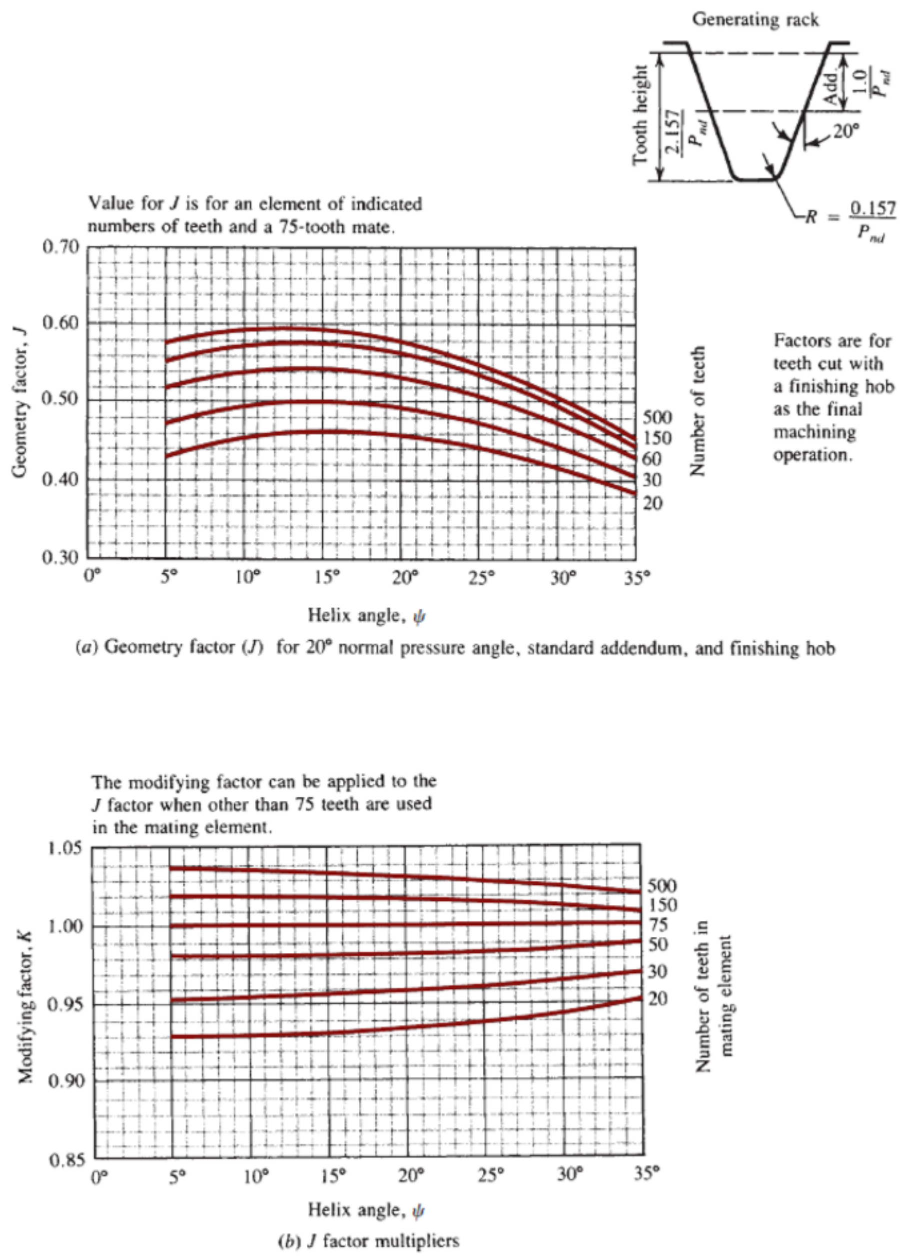
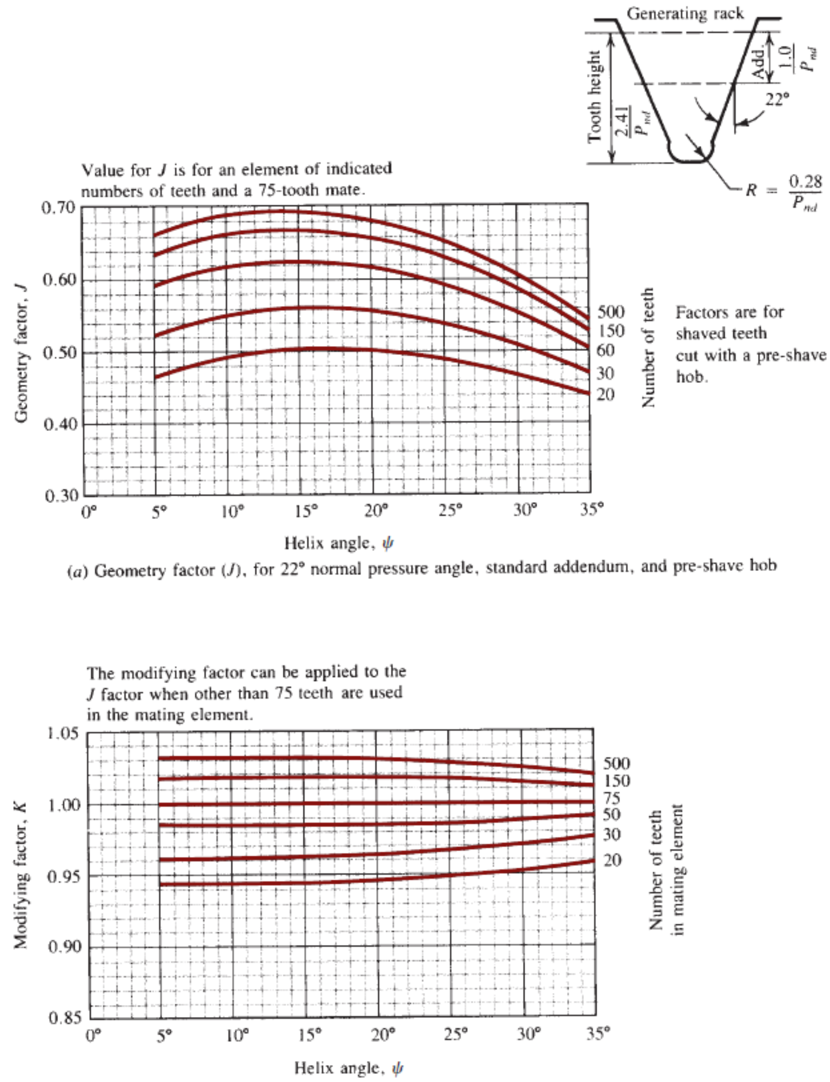


FIGURE 10–6 Geometry factor (J) for 20° normal pressure angle



(a) Geometry factor (J), for 22° normal pressure angle, standard addendum, and pre-shave hob

(b) J factor multipliers

FIGURE 10–7 Geometry factor (J) for 22° normal pressure angle

I swear he will never give you a 22° one but if he does it is what it is.

14. **Now calculate I using the tables below:**

TABLE 10–1 Geometry Factors for Pitting Resistance, I, for Helical Gears with 20° Normal Pressure Angle and Standard Addendum

A. Helix angle ψ = 15.0°

Gear teeth	Pinion teeth				
	17	**21**	**26**	**35**	**55**
17	0.124				
21	0.139	0.128			
26	0.154	0.143	0.132		
35	0.175	0.165	0.154	0.137	
55	0.204	0.196	0.187	0.171	0.143
135	0.244	0.241	0.237	0.229	0.209

B. Helix angle ψ = 25.0°

Gear teeth	Pinion teeth					
	14	**17**	**21**	**26**	**35**	**55**
14	0.123					
17	0.137	0.126				
21	0.152	0.142	0.130			
26	0.167	0.157	0.146	0.134		
35	0.187	0.178	0.168	0.156	0.138	
55	0.213	0.207	0.199	0.189	0.173	0.144
135	0.248	0.247	0.244	0.239	0.230	0.210

Source: Extracted from AGMA Standard 908-B89 (R 1999), *Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1001 North Fairfax Street, 5th floor, Alexandria, VA 22314.

TABLE 10–2 Geometry Factors for Pitting Resistance, I, for Helical Gears with 25° Normal Pressure Angle and Standard Addendum

A. Helix angle ψ = 15.0°

Gear teeth	Pinion teeth					
	14	**17**	**21**	**26**	**35**	**55**
14	0.130					
17	0.144	0.133				
21	0.160	0.149	0.137			
26	0.175	0.165	0.153	0.140		
35	0.195	0.186	0.175	0.163	0.143	
55	0.222	0.215	0.206	0.195	0.178	0.148
135	0.257	0.255	0.251	0.246	0.236	0.214

B. Helix angle ψ = 25.0°

Gear teeth	Pinion teeth						
	12	**14**	**17**	**21**	**26**	**35**	**55**
12	0.129						
14	0.141	0.132					
17	0.155	0.146	0.135				
21	0.170	0.162	0.151	0.138			
26	0.185	0.177	0.166	0.154	0.141		
35	0.203	0.197	0.188	0.176	0.163	0.144	
55	0.227	0.223	0.216	0.207	0.196	0.178	0.148
135	0.259	0.258	0.255	0.251	0.246	0.235	0.213

Source: Extracted from AGMA Standard 908-B89, *Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1001 North Fairfax Street, 5th floor, Alexandria, VA 22314.

9.3.3.3 Bevel Gears *using Mott

Let's get right to it:

1. **Find the type of shock and K_o from the spur gear guide**

2. **Calculate any missing values such as N or F using the spur gear guide**

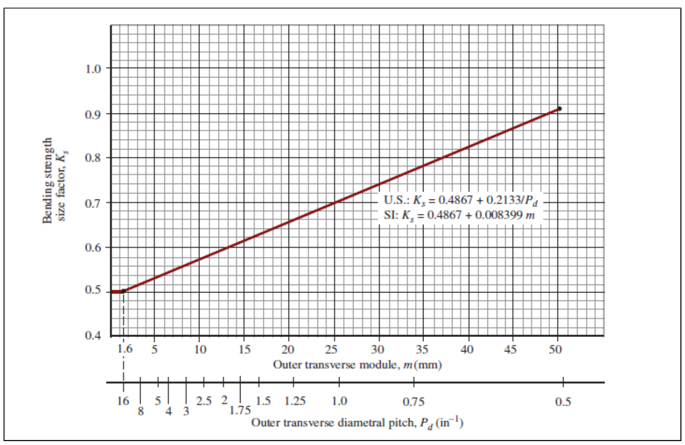
3. **Find the load transmitted, and the pitch line velocity**

$$\nu_t = \frac{\pi D n}{12}$$

$$W_t = \frac{33000P}{\nu_t}$$

4. **Find the size factor, K_s, using the equations below or the table:**

$$K_s = \begin{cases} 0.5 & P_d \geq 16 \\ 0.4867 + \frac{0.2133}{P_d} & P_d < 16 \end{cases}$$



5. **Find $K_m b$ where:**

   $K_{mb} = 1$ for both gears straddle mounted
   $K_{mb} = 1.1$ for one gears straddle mounted
   $K_{mb} = 1.25$ for neither gears straddle mounted

6. **Find K_m using $K_m b$ and F:**

$$K_m = K_{mb} + 0.0036F^2$$

7. **If the quality number, Av, is given use that. If not, use this table:**

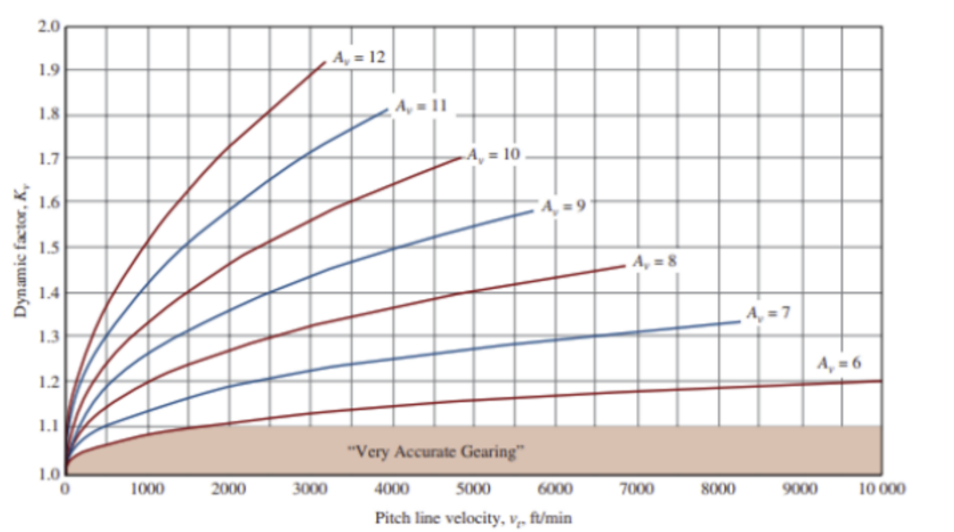
TABLE 9–5 Recommended AGMA Quality Numbers

Application	Quality number	Application	Quality number
Cement mixer drum drive	A11	Small power drill	A9
Cement kiln	A11	Clothes washing machine	A8
Steel mill drives	A11	Printing press	A7
Grain harvester	A10	Computing mechanism	A6
Cranes	A10	Automotive transmission	A6
Punch press	A10	Radar antenna drive	A5
Mining conveyor	A10	Marine propulsion drive	A5
Paper-box-making machine	A9	Aircraft engine drive	A4
Gas meter mechanism	A9	Gyroscope	A2

Machine tool drives and drives for other high-quality mechanical systems

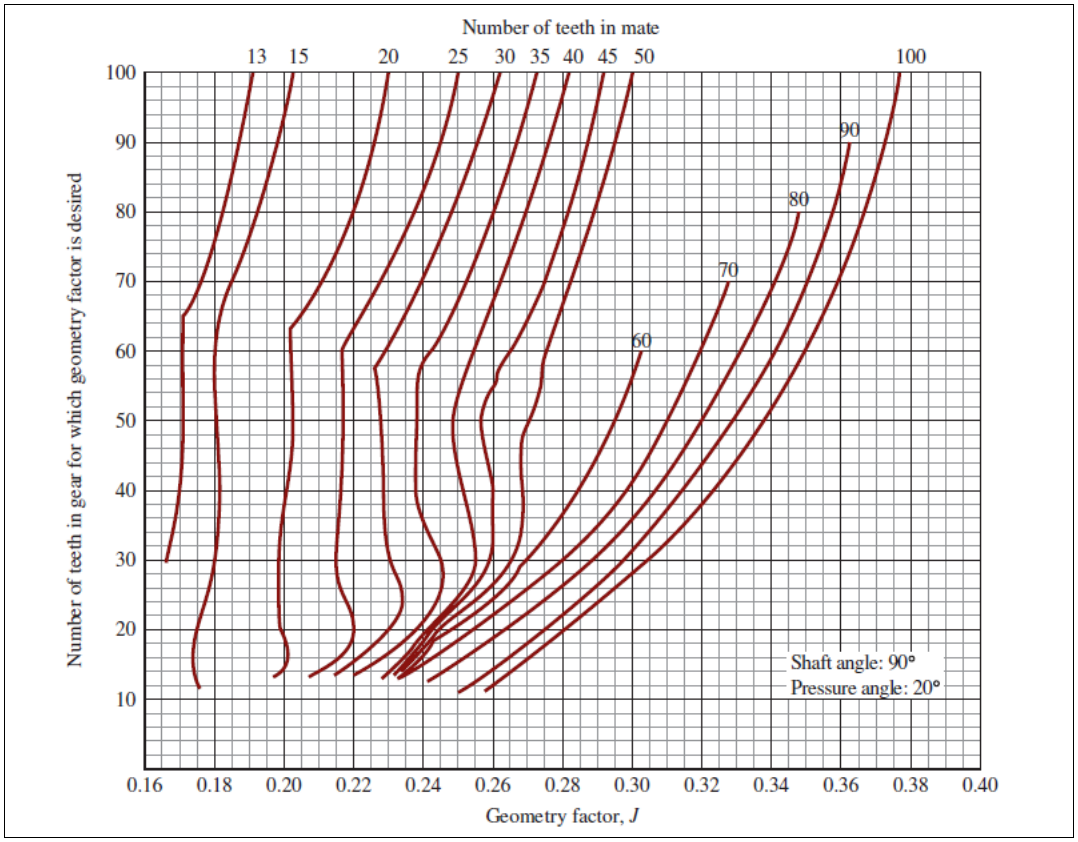
Pitch line speed (fpm)	Quality number	Pitch line speed (m/s)
0–800	A10	0–4
800–2000	A8	4–11
2000–4000	A6	11–22
Over 4000	A4	Over 22

8. **Find K_v from the graphs below:**



(*a*) U.S. Customary units

9. **Find J using the graph shown below:**



10. **Calculate the bending stress number:**

$$s_t = \frac{W_t P_d K_O K_s K_m K_v}{FJ}$$

11. **Choose a safety factor of 1 for now:**

12. **Assume** 99% **reliability and thus choose** K_R **as 1, make a note of what** C_R **is, also going to typically be 1 just like every other one of these stupid ass factors, anyway if you have a reason for higher or lower reliability use the table below:**

TABLE 10–3 Reliability Factors for Allowable Bending and Contact Stresses

Reliability R	Interpretation	Reliability factors: Bending K_R	Contact C_R
0.9	Fewer than one failure in 10	0.85	0.92
0.99	Fewer than one failure in 100	1.00	1.00
0.999	Fewer than one failure in 1000	1.25	1.12
0.9999	Fewer than one failure in 10 000	1.50	1.22

Source: Adapted from AGMA 2003-C10, *Rating the Pitting Resistance and Bending Strength of Generated Straight Bevel, Zerol Bevel and Spiral Bevel Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1001 North Fairfax Street, 5th Floor, Alexandria, VA.

13. **Using the lifetime of the machine, calculate the number of load cycles, N_L**

$$N_{LP} = (60)(lifetime)n_P$$

$$N_{LG} = (60)(lifetime)n_G$$

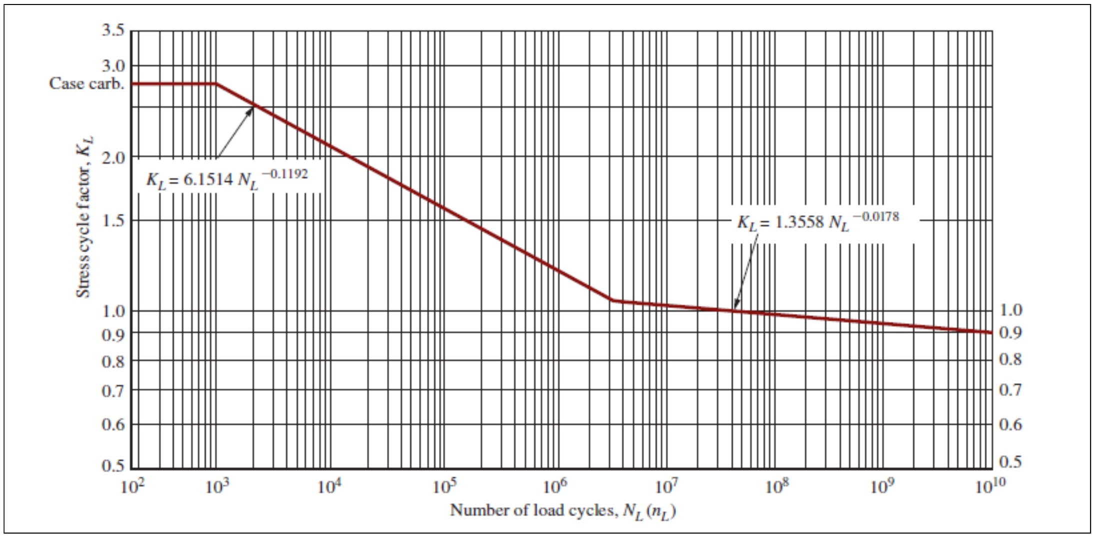
If lifetime is not specified then use this table:

TABLE 9–12 Recommended Design Life

Application	Design life (h)
Domestic appliances	1000–2000
Aircraft engines	1000–4000
Automotive	1500–5000
Agricultural equipment	3000–6000
Elevators, industrial fans, multipurpose gearing	8000–15 000
Electric motors, industrial blowers, general industrial machines	20 000–30 000
Pumps and compressors	40 000–60 000
Critical equipment in continuous 24-h operation	100 000–200 000

Source: Eugene A. Avallone and Theodore Baumeister III, eds. *Marks' Standard Handbook for Mechanical Engineers.* 9th ed. New York: McGraw-Hill, 1986.

14. **Find the stress cycle factor, K_L, from the graph below:**



15. **Find the allowable bending stress:**

$$s_{at} = \frac{s_t(SF)K_R}{K_L}$$

16. **Perform a similar analysis to spur gears for specifying a material and calculating a safety factor**

17. **To find the critical stress, choose $C_p = 2300$ for steel**

18. **Calculate the pitting resistance size factor, Cs, using this formula/table:**

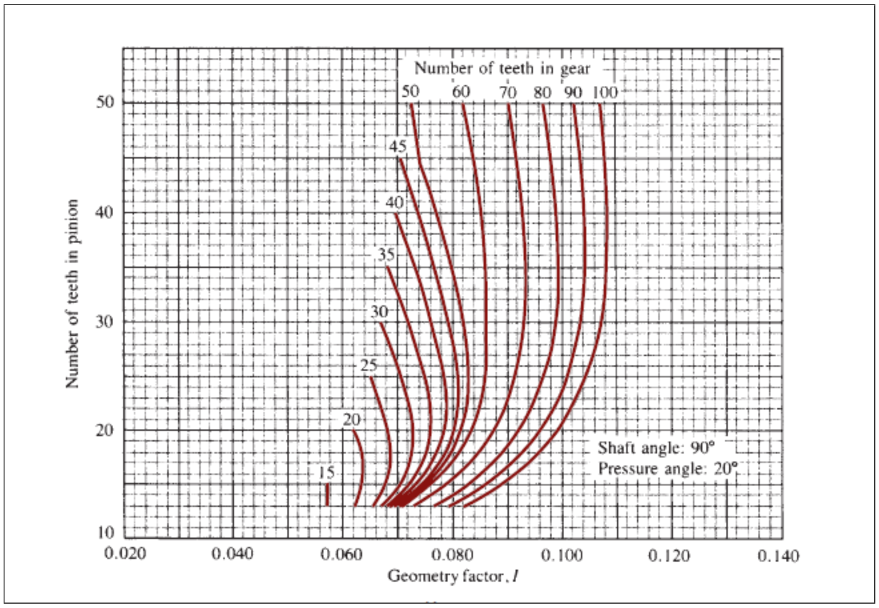
$$C_s = 0.125F + 0.4375$$

19. **Calculate the crown factor, C_{xc}, to be one of the following:**

$C_{xc} = 1.5$ for properly crowned teeth (this is the assumption we use)

$C_{xc} = 2$ for non-crowned teeth

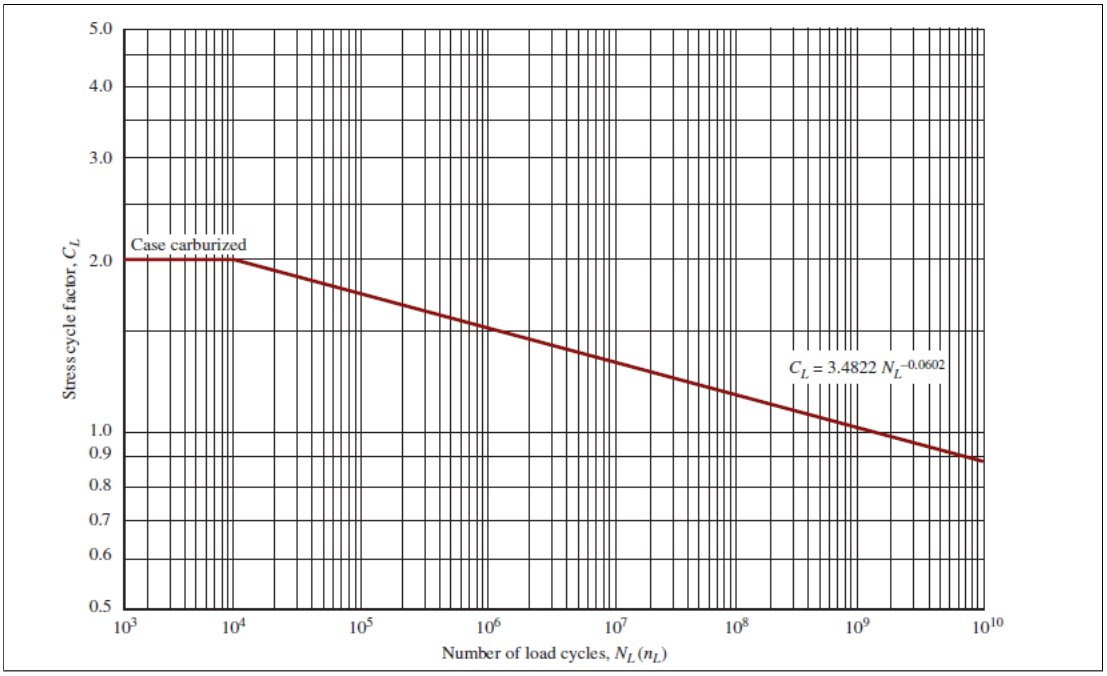
20. **Calculate I from the graph below:**



21. **Calculate the contact stress number:**

$$s_c = C_p \sqrt{\frac{W_t K_O K_m K_\nu C_a C_{xc}}{F D_p I}}$$

22. **Calculate the stress cycle factor, C_L, from the graph below:**



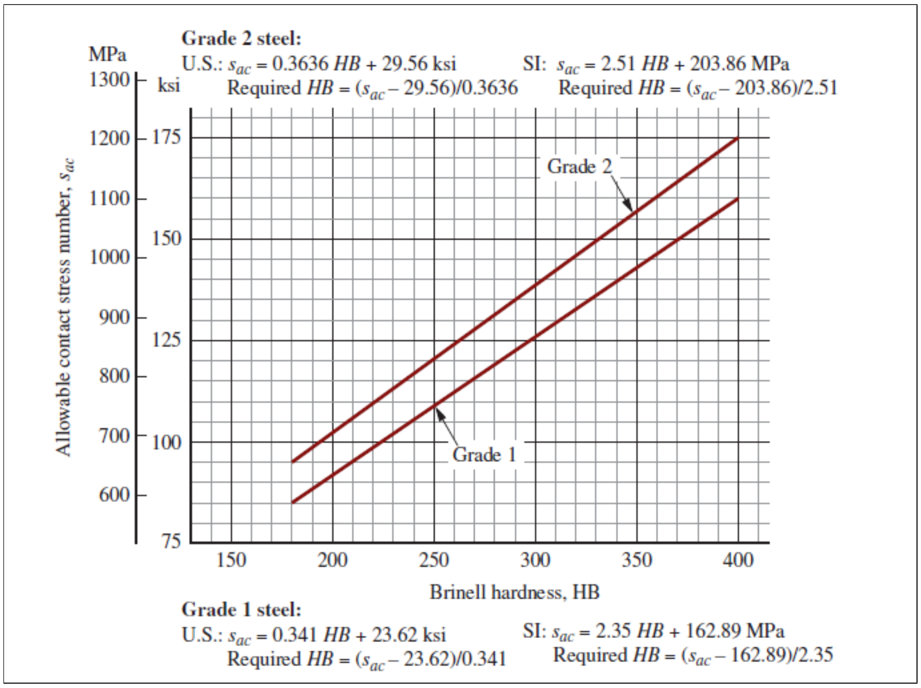
23. **Calculate the allowable contact stress number using the following formula:**

$$s_a c = \frac{s_c(SF)C_R}{C_L}$$

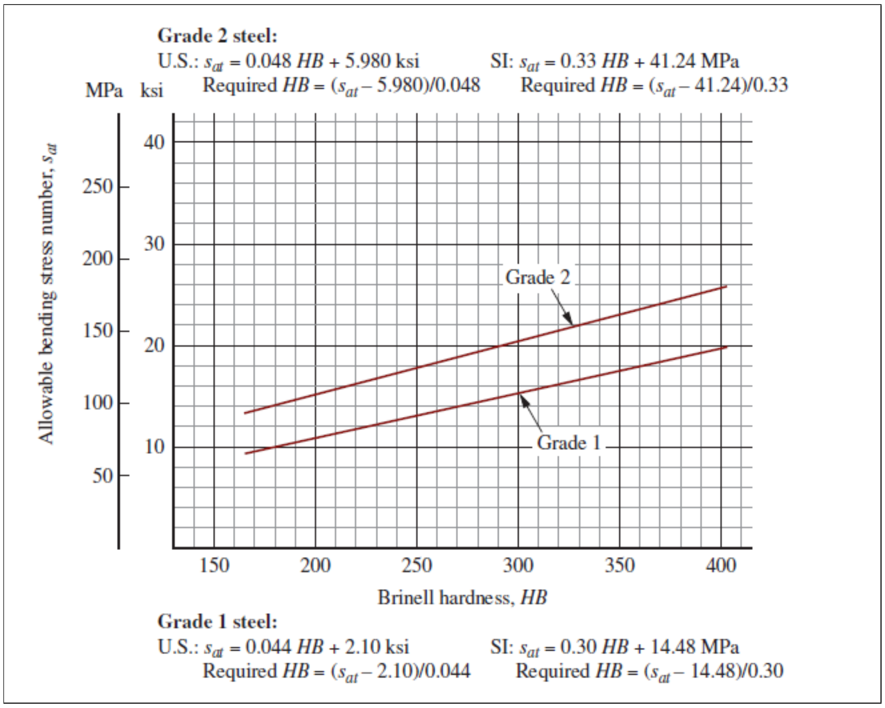
24. **Perform a similar analysis to spur gears for finding the material and associated safety factor**. However you will use these graphs below and a different formula for the safety factor which is:

$$SF = \frac{s_{ac}C_L}{s_c C_R}$$

$$SF = \frac{s_{at}K_L}{s_t K_R}$$



Note that these graphs are different from the ones used in the spur gears they have different numbers so be sure to use the ones here.



Grade 2 steel:
U.S.: $s_{at} = 0.048\,HB + 5.980$ ksi
Required $HB = (s_{at} - 5.980)/0.048$
SI: $s_{at} = 0.33\,HB + 41.24$ MPa
Required $HB = (s_{at} - 41.24)/0.33$
MPa
ksi
Allowable bending stress number, s_{at}
Grade 2
Grade 1
Brinell hardness, HB
Grade 1 steel:
U.S.: $s_{at} = 0.044\,HB + 2.10$ ksi
Required $HB = (s_{at} - 2.10)/0.044$
SI: $s_{at} = 0.30\,HB + 14.48$ MPa
Required $HB = (s_{at} - 14.48)/0.30$

9.3.4 Other Shit

9.3.4.1 Worm Gears **Using Mott*

Some useful Shit:

$$N_G = \text{Number of teeth on the gear}$$

$$N_W = \text{Number of worm threads}$$

$$D_G = \text{pitch diameter of the gear (in)}$$

$$D_W = \text{Pitch diameter of the worm}$$

$$p = \text{circular pitch (in)}$$

$$P_d = \text{diametral pitch (teeth/inch)}$$

$$m = \text{module}$$

$$L = \text{Lead angle}$$

$$C = \text{Center distance (in)}$$

$$\phi_n = \text{Normal pressure angle}$$

$$\phi_t = \text{transverse pressure angle}$$

$$a = \text{addendum (in)}$$

$$h_t = \text{whole depth (in)}$$

$$h_k = \text{working depth (in)}$$

$$b = \text{dedendum (in)}$$

$$D_{rW} = \text{Root diameter of worm (in)}$$

$$D_{oW} = \text{Outside diameter of worm (in)}$$

$$D_{rG} = \text{Root Diameter of gear (in)}$$

$$F_G = \text{Face width of wormgear (in)}$$

$$F_W = \text{face length of worm (in)}$$

$$n_W = \text{speed of worm (rpm)}$$

$$n_G = \text{speed of gear (rpm)}$$

$$\nu_{tW} = \text{Pitch line speed for worm (ft/min)}$$

$$\nu_{tG} = \text{Pitch line speed for gear (ft/min)}$$

$$VR = \text{Velocity ratio}$$

Now for some of the formulae to solve for these values:

circular pitch: $p = \frac{\pi D_G}{N_G}$

diametral pitch: $P_d = \frac{N_G}{D_G}$ or $P_d = \frac{\pi}{p}$

module: $m = \frac{D}{N}$

axial pitch: $P_x = p$

lead: $L = N_W P_x$

lead angle: $\tan(\lambda) = \frac{L}{\pi D_W}$

center distance: $C = \frac{D_W + D_G}{2} = \frac{N_W + N_G}{2P_d}$

angle relationship: $\tan\phi_n = \tan(\phi_t)\cos(\lambda)$

addendum: $a = 0.318P_x = \frac{1}{P_d}$

whole depth: $h_t = 0.6866P_x = \frac{2.157}{P_d}$

working depth: $h_k = 2a$

dedendum: $b = h_t - a$

root diameter of worm: $D_{rW} = D_W - 2b$

outside diameter of worm: $D_o W = D_W + 2a = D_W + h_k$

root diameter of gear: $D_r G = D_G - 2b$

throat diameter of gear: $D_t = D_G + 2a$

face width of wormgear: $F_G = \sqrt{D_{oW}^2 - D_W^2} = 2p \approx \frac{6}{P_d}$

face length of worm: $F_W = 2\sqrt{(\frac{D_t}{2})^2 - (\frac{D_G}{2-a})^2}$

Speed:

- pitch line speed for worm: $\nu_{tW} = \frac{\pi D_W n_W}{12}$ lb-ft or $\nu_{tW} = \frac{\pi D_W n_W}{60000}$ m/s
- pitch line speed for gear: $\nu_{tG} = \frac{\pi D_G n_G}{12}$ lb-ft or $\nu_{tW} = \frac{\pi D_G n_G}{60000}$ m/s
- Velocity Ratio: $VR = \frac{n_W}{n_G} = \frac{N_G}{N_W}$
- Sliding speed: $\nu_s = \frac{\nu_{tG}}{\sin(\lambda)} = \frac{\nu_{tW}}{\cos(\lambda)}$

Forces:

- Force relationship: $W_{tG} = W_{xW}$, $W_{xG} = W_{tW}$, $W_{rG} = W_{rW}$
- output torque: $T_o = \frac{63000 P_o}{n_G} = \frac{W_{tG} D_G}{2}$
- transmitted force: $W_{tG} = \frac{2T_o}{D_G}$
- axial force: $W_{xG} = W_{tG} \frac{\cos(\phi_n)\sin(\lambda) + \mu\cos(\lambda)}{\cos(\phi_n)\cos(\lambda) - \mu\sin(\lambda)}$
- radial force: $W_{rG} = \frac{W_{tG}\sin(\phi_n)}{\cos(\phi_n)\cos(\lambda) - \mu\sin(\lambda)}$
- friction force: $W_f = \frac{\mu W_{tG}}{\cos(\lambda)\cos(\phi_n) - \mu\sin(\lambda)}$
- power loss due to friction: $P_L = \frac{\nu_s W_f}{33000}$
- input power: $P_i = P_o + P_L$
- $\eta = \frac{P_o}{P_i} = \frac{\cos(\phi_n) - \mu\tan(\lambda)}{\cos(\phi_n) + \frac{\mu}{\tan(\lambda)}}$

Now design selection for worm gears:

1. **Calculate the lead L and the lead angle λ**

$$p = \frac{\pi}{P_d} = \frac{\pi D_G}{N_G}$$

$$P_x = p$$

$$L = N_W P_x$$

$$\lambda = \arctan(\frac{L}{\pi D_W}$$

2. **Find Center Distance CD**

$$CD = \frac{D_G + D_W}{2}$$

3. **Calculate the pitch line speed of the gear**

$$\nu_{tG} = \frac{\pi D_G n_G}{12}$$

4. **Calculate the sliding speed**

$$\nu_s = \frac{\nu_{tG}}{\sin(\lambda)} = \frac{\nu_{tW}}{\cos(\lambda)}$$

5. **Calculate the coefficient of friction:**

$$\mu = \begin{cases} 0.15 & , \nu = 0 \\ 0.124e^{-0.074\nu_s^{0.645}} & , 0 < \nu_s < 10 \\ 0.103e^{-0.11\nu_s^{0.45}} + 0.012 & , \nu_s > 10 \end{cases}$$

6. **Find the forces on the gear:**

$$T_o = \frac{63000P_o}{n_G} = \frac{W_{tG}D_G}{2}$$

$$W_{tG} = \frac{2T_o}{D_G}$$

$$W_{xG} = W_{tG}\frac{\cos(\phi_n)\sin(\lambda) + \mu\cos(\lambda)}{\cos(\phi_n)\cos(\lambda) - \mu\sin(\lambda)}$$

$$W_{rG} = \frac{W_{tG}\sin(\phi_n)}{\cos(\phi_n)\cos(\lambda) - \mu\sin(\lambda)}$$

7. **Find the friction force:**

$$W_f = \frac{\mu W_{tG}}{\cos(\lambda)\cos(\phi_n) - \mu\sin(\lambda)}$$

8. **Find the power loss due to friction:**

$$P_L = \frac{\nu_s W_f}{33000}$$

9. **Compute the input power:**

$$P_i = P_o + P_L$$

10. **Calculate efficiency:**

$$\eta = \frac{P_o}{P_i} = \frac{\cos(\phi_n) - \mu\tan(\lambda)}{\cos(\phi_n) + \frac{\mu}{\tan(\lambda)}}$$

11. **Find the Lewis form factor y from the table below using your ϕ_n value:**

TABLE 10-5 Approximate Lewis Form Factor for Wormgear Teeth

ϕ_n	y
$14\frac{1}{2}°$	0.100
20°	0.125
25°	0.150
30°	0.175

12. **Find the normal circular pitch:**

$$p_n = p\cos(\lambda) = \frac{\pi\cos(\lambda)}{P_d}$$

13. **Find the dynamic factor K_v:**

$$K_v = \frac{1200}{1200 + \nu_{tG}}$$

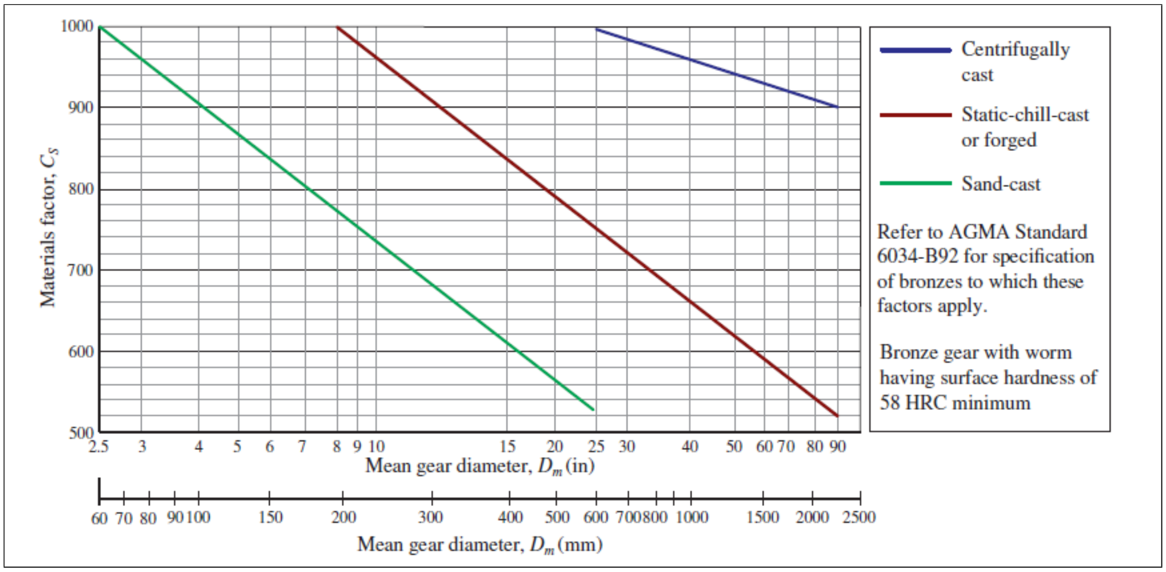
14. **Find the dynamic load, W_d:**

$$W_d = \frac{W_{tG}}{K_v}$$

15. **Find the stress σ in the gear teeth:**

$$\sigma = \frac{W_d}{yFp_n}$$

16. **Find the materials factor C_s using this figure:**



For sand-cast bronzes:

$$C_s = \begin{cases} 1189.636 - 476.545\log_{10}(D_p) & D_G > 2.5 \\ 1000 & D_G < 2.5 \end{cases}$$

For static-chill-cast or forged bronze:

$$C_s = \begin{cases} 1411.651 - 455.825\log_{10}(D_p) & \\ 1000 & D_G < 8 \end{cases}$$

For centrifugally cast bronze:

$$C_s = \begin{cases} 1251.291 - 179.75\log_{10}(D_p) & \\ 1000 & D_G < 25 \end{cases}$$

17. **Find the ration correction factor C_m:**

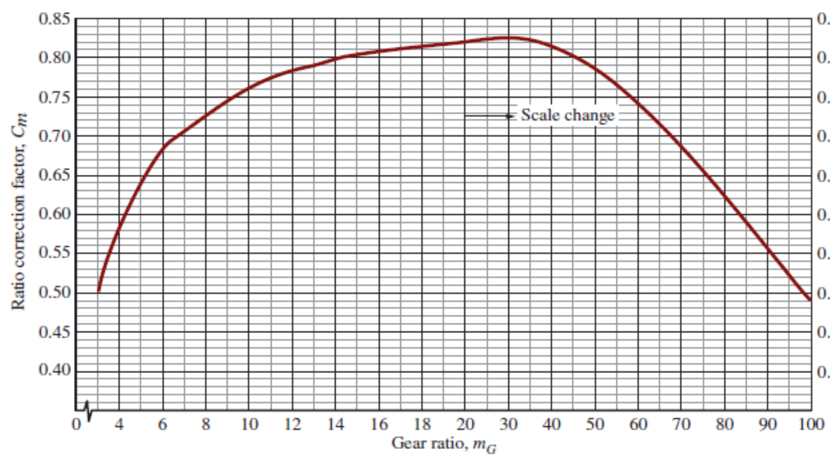


FIGURE 10–28 Ratio correction factor, C_m, versus gear ratio, m_G

Remember that m_G is the gear ratio which is $m_G = \frac{N_G}{N_p}$

$$C_m = \begin{cases} 0.02\sqrt{-m_G^2 + 40m_G - 76} + 0.46 & 6 < m_g < 20 \\ 0.0107\sqrt{m_G^2 + 56m_G + 5146} & 20 < m_G < 76 \\ 1.1483 - 0.00658m_G & m_G > 76 \end{cases}$$

18. **Find the Worm gear velocity factor C_v:**

$$C_v = \begin{cases} 0.659e^{-0.0011\nu_s} & 0 < \nu_s < 700 \\ 13.31\nu_s^{-0.571} & 700 < m_G < 3000 \\ 65.52\nu_s^{-0.774} & \nu_s > 3000 \end{cases}$$

19. **Find the effective face width F_e:**

$$F_e = \begin{cases} F & , F < \frac{D_W}{3} \\ \frac{D_W}{3} & , F > \frac{D_W}{3} \end{cases}$$

20. **Find the rated tangential load W_{tR}:**

$$W_{tR} = C_s D_G^{0.8} F_e C_m C_v$$

21. **Check if the design is satisfactory to resist pitting:**
If $W_{tR} > W_{tG}$ then the design is satisfactory

9.3.4.2 Rack and Pinion **Using Mott*

Nomenclature

P_D = diametral pitch (teeth/in)

N_P = number of teeth on the pinion

D_P = Circular Pitch Diameter of the pinion (in)

n_p = angular speed of the pinion (rpm)

v_t = pitch line velocity of the pinion

B = distance from back of the rack to pitch (in) (Table 8-10)

$B - C$ = distance from back of the rack to pinion centerline (in)

V_{rack} = speec of the rack (ft/min)

s_{rack} = distance the rack travels (ft)

t = time (s)

θ_p = number of revolution of the pinion (rev)

General Equations

pitch line speed: $v_t = \frac{D_p n_p}{2}$

speed of rack: $V_{rack} = \frac{\pi D_p n_p}{12}$

distance the rack travels: $S_{rack} = \frac{D_p \theta_p}{2}$

Design Steps

1. **Find the Pitch Diameter,** D_p

$$D_p = \frac{N}{P_D}$$

2. **Find distance from the pitch line to the back of the rack from the following table:**

TABLE 8–10 Example rack specifications

Diametral pitch	Pitch line to back (*B*)	Overall thickness	Face width	Nominal length [ft]
64	0.109	0.125	0.125	2
48	0.104	0.125	0.125	2
32	0.156	0.187	0.187	4
24	0.208	0.250	0.25	4
20	0.450	0.500	0.5	6
16	0.688	0.750	0.75	6
12	0.917	1.000	1	6
10	1.150	1.250	1.25	6
8	1.375	1.500	1.5	6
6	1.333	1.500	2	6
5	1.300	1.500	2.5	6
4	1.750	2.000	3.5	6

3. **Find the distance from the back of the rack to the pinion centerline (B-c)**

$$B - C = B + \frac{D_P}{2}$$

4. **Find velocity of the rack,** V_{rack}

$$V_{rack} = \left(\frac{\pi}{6}\right)\left(\frac{D_P n_P}{2}\right)$$

5. **Find the time it takes the rack to move some distance:**

$$t = 60\left(\frac{s_{rack}}{V_{rack}}\right)$$

6. **Find the number of revolutions required to move that rack that far:**

$$\theta_P = \left(\frac{6}{\pi}\right)\left(\frac{2s_{rack}}{D_P}\right)$$

10 Unit 3: Fluid Power

10.1 Class Memory

I can't quite recall much about fluid power in class. However, I do recall a fluid power lab we had to attend in KAIS 1180 where we did some interesting shenanigans with a hydraulic training unit with multiple valves, actuators, pumps, and motors. The lab went as follows: imagine a group of 15-20 students listening to a TA for about an hour or two. Every now and then, one pair of us had to get up and mimic a hydraulic circuit from the board onto the actual unit. It is safe to say that we all had absolutely no idea what to do, but the TA was pretty nice, so we didn't suffer much thanks to him. There was also a fluids tutorial before/after the lab, depending on your luck. You will attend the lab with your design project group.

I am sure you will find this lab fun, so look forward to it!

10.2 Do's and Don'ts

Do's:

- Learn how hydraulic circuits work (obviously!)
- Get familiar with the Design Engineers Handbook[55]

Don'ts:

- Don't focus on sizing Motors too much during questions. That is not the purpose of this unit.
- Don't depend on the handbook for everything (e.g. when calculating working load). Sometimes you need to use common sense![56]

[55]This is a handbook that Jon will give out in class for the Fluid Power unit. You will basically need it for most of the questions within the unit. For a digital (and somewhat clearer) version than the one given in class, find it here. Also, FYI, I will use the clearer graphs (that I will manually edit myself) in the guide, so you theoretically do not need the handbook for solving questions.

[56]In simpler words, don't be a Donkey.

10.3 Unit Guide

In this unit, the handbook treats you like an idiot, so there will be minimal calculations on your end. As a result, no calculator exists, and there is only really one type of question for me to guide you through, so it will be rather quick.

On the other hand, I will try my best to make this walk-through as detailed as possible to help you understand every bit.

10.3.1 General Question Guide

Straight off the bat, the question itself should provide you with some basic information, such as:

- Operating pressure[57] [psi]
- Stroke Length [in.]
- Friction Coefficient
- Piston/Rod Diameters [in.]
- Acceleration/Deceleration condition[58]
- Extension Speed [ft/min or in/sec]
- Pump Flow Rate [gpm]
- Maximum Load[59] [lbf]

You will likely not get all of these, but most design questions will definitely include the working pressure and stroke length to start you off.

Now, in most conventional fluid power design questions, you will need to determine the following:

- Sizing the Piston/Rod (if not given)
- Basic Length
- Flow Rate (if not given)
- Pipe diameter
- Motor Type/Size

Below are steps to solving a fluid power question from scratch.

[57] This pressure is what the relief valve is set to
[58] For example: "Your load must be accelerated to X ft/sec in X inches..."
[59] Either given or found using statics

1. First, start off by writing down what you have been given, which will likely be all or some of the following:
    - Relief Pressure[60]
    - Stroke Length
    - Acceleration/Deceleration condition
    - Extension Speed [ft/min or in/sec]
2. Next, depending on whether this is given or not, determine the maximum load on your cylinder. You might need to do some trig if the load is being pushed up a ramp at an angle or something like that.

    Once you do that, use the table below to choose a piston (bore) with a stroke force that exceeds your maximum force rating. Remember to use your operating pressure![61]

CUSTOMARY U.S. UNITS

Cyl. Bore or Piston Rod. Dia. (in.)	Cyl. Bore Size (ϕ mm)	Area (sq. in.)	CYLINDER PUSH STROKE FORCE IN POUNDS AT VARIOUS PRESSURES (PSI)										Displacement per inch of Stroke (gallons)
			50	80	100	500	750	1000	1500	2000	2500	3000	
5/8	15.9	.307	15	25	31	154	230	307	461	614	768	921	.0013
1	25.4	.785	39	65	79	392	588	785	1,177	1,570	1,962	2,355	.0034
1-3/8	34.9	1.490	75	119	149	745	1,118	1,490	2,235	2,980	3,725	4,470	.0065
1-1/2	38.1	1.767	88	142	177	885	1,325	1,770	2,651	3,540	4,425	5,310	.00765
1-3/4	44.5	2.410	121	193	241	1,205	1,808	2,410	3,615	4,820	6,025	7,230	.0104
2	50.8	3.140	157	251	314	1,570	2,357	3,140	4,713	6,280	7,850	9,420	.0136
2-1/2	63.5	4.910	245	393	491	2,455	3,682	4,910	7,364	9,820	12,275	14,730	.0213
3	76.2	7.070	354	566	707	3,535	3,502	7,070	10,604	14,140	17,675	21,210	.0306
3-1/4	82.6	8.300	415	664	830	4,150	6,225	8,300	12,450	16,600	20,750	24,900	.0359
3-1/2	88.9	9.620	481	770	962	4,810	7,215	9,620	14,430	19,240	24,050	28,860	.0416
4	101.6	12.570	628	1,006	1,257	6,285	9,428	12,570	18,856	25,140	31,425	37,710	.0544
5	127.0	19.640	982	1,571	1,964	9,820	14,730	19,640	29,460	39,280	49,100	58,920	.0850
5-1/2	139.7	23.760	1,188	1,901	2,376	11,880	17,820	23,760	35,640	47,520	59,400	71,280	.1028
6	152.4	28.270	1,414	2,262	2,827	14,135	21,203	28,270	42,406	56,540	70,675	84,810	.1224
7	177.8	38.490	1,924	3,079	3,849	19,245	28,868	38,490	57,736	76,980	96,225	115,470	.1666
8	203.2	50.270	2,513	4,022	5,027	25,135	37,703	50,270	75,406	100,540	125,675	150,810	.2176
8-1/2	215.9	56.750	2,838	4,540	5,675	28,375	42,563	56,750	85,125	113,500	142,875	170,250	.2455
10	254.0	78.540	3,927	6,283	7,854	39,270	58,905	78,540	117,810	157,080	196,350	235,620	.3400
12	304.8	113.100	5,655	9,048	11,310	56,550	84,825	113,100	169,650	226,200	282,750	339,300	.4896

table b-1 NOTE: Deduct Force of Poston Rod Size from Bore Size for Pull Applications.

3. Find the basic (effective) length, given by:

$$\textit{Basic Length} = \textit{Actual Stroke} \times \textit{Stroke Factor}$$

    The actual stroke is given through the question, and the stroke factor is found based on the mounting style and rod end connection using the table below.

[60] If not given, I'm pretty sure you just assume 1000 psi

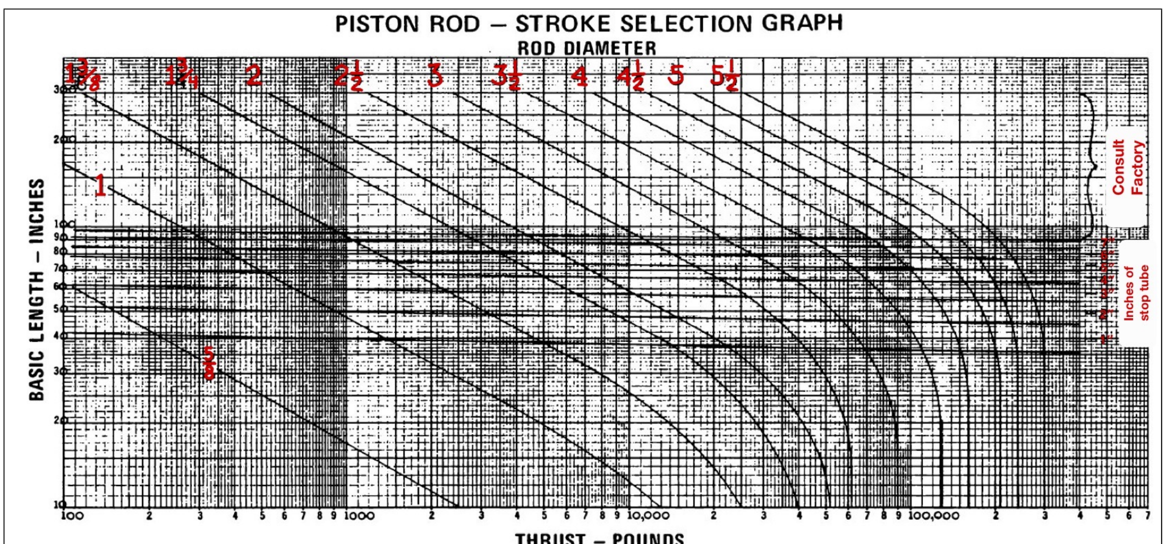
[61] There's a mistake in one of the force values in this graph, let's see if one of you can find it

piston rod — stroke selection table

RECOMMENDED MOUNTING STYLES FOR MAXIMUM STROKE AND THRUST LOADS	ROD END CONNECTION	CASE	STROKE FACTOR
CLASS 1 – GROUPS 1 OR 3 Long stroke cylinders for thrust loads should be mounted using a heavy-duty mounting style at one end, firmly fixed and aligned to take the principle force. Additional mounting should be specified at the opposite end, which should be used for alignment and support. An intermediate support may also be desirable for long stroke cylinders mounted horizontally. Machine mounting pads can be adjustable for support mountings to achieve proper alignment.	FIXED AND RIGIDLY GUIDED.	I	.50
	PIVOTED AND RIGIDLY GUIDED	II	.70
	SUPPORTED BUT NOT RIGIDLY GUIDED	III	2.00
CLASS 2 – GROUP 2 Style – Trunnion on Head	PIVOTED AND RIGIDLY GUIDED	IV	1.00
Style – Intermediate Trunnion	PIVOTED AND RIGIDLY GUIDED	V	1.50
Style – Trunnion on Cap or Style – Clevis on Cap	PIVOTED AND RIGIDLY GUIDED	VI	2.00

4. Choose a rod diameter using the graph[62] below (round up!).[63]

   "Thrust" is the maximum force that is given or calculated and "Basic Length" is what we calculated in the previous step.



graph b-1

5. Once you choose both the cylinder bore and rod diameters, use the tables below to check that the following combo is available.

[62]I TRIED to make it clearer. Don't worry about the stop tube stuff....

[63]In-case the writings are too small, access the full size graphs here

CYLINDER BORE-INCHES	PISTON ROD DIA.-INCHES	PISTON ROD AREA SQ. IN.	CYLINDER NET AREA SQ. IN.	FLUID DISPLACEMENT AT 10 FT. PER MINUTE PISTON VELOCITY G.P.M.	C.F.M.	FLUID VELOCITY (IN FEET PER SECOND) THROUGH EXTRA HEAVY PIPE AT 10 F.P.M. PISTON SPEED. 1/4	3/8	1/2	3/4	1	1-1/4	1-1/2	2	2-1/2
1	0	0	0.785	0.41	0.054	1.82	0.92	0.56	0.30	0.183	0.102	0.074	0.045	.....
	1/2	0.196	0.589	0.30	0.041	1.33	0.68	0.41	0.21	0.134	0.075	0.055	0.033	.....
	5/8	0.307	0.478	0.16	0.033	0.71	0.36	0.22	0.12	0.071	0.040	0.029	0.017	.....
1½	0	0	1.77	0.92	0.123	4.09	2.09	1.259	0.680	0.410	0.230	0.167	0.100	.....
	5/8	0.307	1.46	0.76	0.101	3.38	1.73	1.040	0.562	0.338	0.190	0.138	0.082	.....
	1	0.785	0.98	0.51	0.068	2.27	1.16	0.699	0.378	0.228	0.128	0.093	0.055	.....
2	0	0	3.14	1.63	0.218	7.27	3.71	2.238	1.209	0.728	0.408	0.296	0.177	.....
	5/8	0.307	2.84	1.48	0.197	6.56	3.35	2.019	1.091	0.657	0.368	0.267	0.160	.....
	1	0.785	2.36	1.23	0.164	5.45	2.79	1.678	0.907	0.546	0.306	0.222	0.133	.....
	1-3/8	1.485	1.66	0.86	0.115	3.84	1.96	1.180	0.638	0.384	0.215	0.156	0.094	.....
2½	0	0	4.91	2.55	0.341	11.36	5.80	3.496	1.890	1.138	0.638	0.463	0.277	.....
	5/8	0.307	4.60	2.39	0.319	10.65	5.44	3.278	1.771	1.067	0.598	0.434	0.260	.....
	1	0.785	4.12	2.14	0.286	9.54	4.87	2.937	1.587	0.956	0.536	0.389	0.233	.....
	1-3/8	1.485	3.42	1.78	0.237	7.93	4.05	2.439	1.318	0.794	0.445	0.323	0.193	.....
	1-3/4	2.405	2.50	1.30	0.174	5.96	2.96	1.783	0.963	0.580	0.325	0.236	0.141	.....
3¼	0	0	8.30	4.31	0.576	19.20	9.81	5.909	3.193	1.923	1.078	0.783	0.468	.....
	1	0.785	7.51	3.90	0.521	17.38	8.88	5.349	2.891	1.741	0.976	0.708	0.424	.....
	1-3/8	1.485	6.81	3.54	0.473	15.77	8.05	4.851	2.622	1.579	0.885	0.642	0.384	.....
	1-3/4	2.405	5.89	3.06	0.409	13.64	6.96	4.196	2.268	1.366	0.765	0.556	0.333	.....
	2	3.142	5.15	2.68	0.357	11.93	6.09	3.671	1.984	1.195	0.670	0.486	0.291	.....
4	0	0	12.57	6.53	0.872	29.09	14.85	8.95	4.84	2.91	1.63	1.19	0.709	.....
	1	0.785	11.78	6.12	0.818	27.27	13.93	8.39	4.54	2.73	1.53	1.11	0.665	.....
	1-3/8	1.485	11.08	5.76	0.769	25.65	13.10	7.89	4.27	2.57	1.44	1.05	0.625	.....
	1-3/4	2.405	10.16	5.28	0.705	23.52	12.01	7.24	3.91	2.36	1.32	0.96	0.574	.....
	2	3.142	9.42	4.89	0.654	21.82	11.14	6.71	3.63	2.19	1.22	0.89	0.532	.....
	2-1/2	4.909	7.66	3.98	0.532	17.73	9.05	5.45	2.95	1.78	1.00	0.72	0.432	.....
5	0	0	19.64	10.20	1.363	45.45	23.21	13.99	7.56	4.55	2.55	1.85	1.108	.....
	1	0.785	18.85	9.79	1.308	43.64	22.28	13.43	7.26	4.37	2.45	1.78	1.064	.....
	1-3/8	1.485	18.15	9.43	1.260	42.01	21.45	12.93	6.99	4.21	2.36	1.71	1.024	.....
	1-3/4	2.405	17.23	8.95	1.196	39.88	20.37	12.27	6.63	3.99	2.24	1.63	0.973	.....
	2	3.142	16.49	8.57	1.144	38.18	19.50	11.75	6.35	3.82	2.14	1.56	0.931	.....
	2-1/2	4.909	14.73	7.65	1.022	34.09	17.41	10.49	5.67	3.41	1.91	1.39	0.831	.....
	3	7.069	12.57	6.53	0.872	29.09	14.85	8.95	4.84	2.91	1.63	1.19	0.709	.....
	3-1/2	9.621	10.01	5.21	0.695	23.18	11.84	7.13	3.86	2.32	1.30	0.95	0.565	.....
6	0	0	28.27	14.69	1.962	65.45	33.42	20.14	10.88	6.55	3.67	2.67	1.596	.....
	1-3/8	1.485	26.79	13.92	1.859	62.01	31.67	19.08	10.31	6.21	3.48	2.53	1.512	.....
	1-3/4	2.405	25.87	13.44	1.795	59.88	30.58	18.43	9.96	5.60	3.36	2.44	1.460	.....
	2	3.142	25.13	13.06	1.744	58.18	29.71	17.90	9.67	5.83	3.27	2.37	1.418	.....
	2-1/2	4.909	23.37	12.14	1.622	54.1	27.6	16.64	8.99	5.42	3.04	2.20	1.32	.....
	3	7.069	21.21	11.02	1.472	49.1	25.1	15.10	8.16	4.92	2.76	2.00	1.20	.....
	3-1/2	9.621	18.65	9.69	1.294	43.2	22.1	13.29	7.18	4.32	2.42	1.76	1.05	.....
	4	12.566	15.71	8.16	1.090	36.4	18.6	11.19	6.05	3.64	2.04	1.48	0.89	.....

table b-5

CYLINDER BORE-INCHES	PISTON ROD DIA.-INCHES	PISTON ROD AREA SQ. IN.	CYLINDER NET AREA SQ. IN.	FLUID DISPLACEMENT AT 10 FT. PER MINUTE PISTON VELOCITY G.P.M.	FLUID DISPLACEMENT C.F.M.	FLUID VELOCITY (IN FEET PER SECOND) THROUGH EXTRA HEAVY PIPE AT 10 F.P.M. PISTON SPEED. 1/4	3/8	1/2	3/4	1	1-1/4	1-1/2	2	2-1/2
7	0	0	38.49	20.00	2.671	89.1	45.5	27.41	14.81	8.92	5.00	3.63	2.17	.....
	1-3/8	1.485	37.00	19.22	2.568	85.7	43.7	26.35	14.24	8.58	4.81	3.49	2.09	.....
	1-3/4	2.405	36.08	18.74	2.504	83.5	42.7	25.70	13.89	8.36	4.69	3.40	2.04	.....
	2	3.142	35.34	18.36	2.453	81.8	41.8	25.17	13.60	8.19	4.59	3.33	2.00	.....
	2-1/2	4.909	33.58	17.44	2.330	77.7	39.7	23.92	12.92	7.78	4.36	3.17	1.90	.....
	3	7.069	31.42	16.32	2.181	72.7	37.1	22.38	12.09	7.28	4.08	2.96	1.77	.....
	3-1/2	9.621	28.86	14.99	2.003	66.8	34.1	20.56	11.11	6.69	3.75	2.72	1.63	.....
	4	12.566	25.92	13.47	1.799	60.0	30.6	18.46	9.98	6.01	3.37	2.45	1.46	.....
	4-1/2	15.904	22.58	11.73	1.567	52.3	26.7	16.08	8.69	5.23	2.93	2.12	1.28	.....
	5	19.635	18.85	9.79	1.308	43.6	22.3	13.43	7.26	4.37	2.45	1.78	1.06	.....
8	0	0	50.27	26.12	3.489	116.4	59.4	35.80	19.35	11.65	6.53	4.74	2.84	1.977
	1-3/8	1.485	48.78	25.34	3.385	112.9	57.7	34.74	18.78	11.31	6.34	4.60	2.75	1.918
	1-3/4	2.405	47.86	24.86	3.321	110.8	56.6	34.09	18.42	11.09	6.22	4.51	2.70	1.882
	2	3.142	47.12	24.48	3.270	109.1	55.7	33.56	18.14	10.92	6.12	4.45	2.66	1.853
	2-1/2	4.909	45.36	23.57	3.149	105.0	53.61	32.31	17.46	10.51	5.892	4.278	2.560	1.784
	3	7.069	43.20	22.44	2.998	100.0	51.06	30.77	16.63	10.01	5.612	4.074	2.438	1.699
	3-1/2	9.621	40.65	21.12	2.821	94.1	48.04	28.95	15.65	9.42	5.279	3.834	2.294	1.598
	4	12.566	37.70	19.59	2.616	87.3	44.56	26.85	14.51	8.74	4.897	3.556	2.128	1.483
	4-1/2	15.904	34.36	17.85	2.385	79.5	40.62	24.47	13.23	8.20	4.464	3.241	1.939	1.351
	5	19.635	30.63	15.91	2.126	70.9	36.21	21.82	11.79	7.10	3.979	2.889	1.729	1.205
	5-1/2	23.758	26.51	13.77	1.840	61.4	31.33	18.88	10.20	6.15	3.444	2.500	1.496	1.043
10	0	0	78.54	40.80	5.451	181.8	92.84	55.94	30.23	18.21	10.203	7.408	4.433	3.089
	1-3/4	2.405	76.14	39.56	5.284	176.2	89.99	54.23	29.31	17.65	9.890	7.181	4.297	2.994
	2	3.142	75.40	39.17	5.233	174.5	89.12	53.70	29.02	17.48	9.795	7.112	4.255	2.965
	2-1/2	4.909	73.63	38.25	5.110	170.4	87.03	52.44	28.34	17.07	9.565	6.945	4.156	2.896
	3	7.069	71.47	37.13	4.960	165.4	84.48	50.91	27.51	16.57	9.284	6.741	4.034	2.811
	3-1/2	9.621	68.92	35.80	4.783	159.5	81.47	49.09	26.53	15.98	8.953	6.501	3.890	2.710
	4	12.566	65.97	34.27	4.578	152.7	77.98	46.99	25.39	15.29	8.570	6.223	3.724	2.595
	4-1/2	15.904	62.64	32.54	4.347	145.0	74.04	44.61	24.11	14.52	8.137	5.908	3.535	2.463
	5	19.635	58.91	30.60	4.088	136.4	69.63	41.96	22.67	13.65	7.652	5.556	3.325	2.317
	5-1/2	23.758	54.78	28.46	3.802	126.8	64.75	39.02	21.09	12.70	7.116	5.167	3.092	2.154
	6	28.274	50.27	26.12	3.489	116.4	59.42	35.80	19.35	11.65	6.530	4.741	2.837	1.977
	6-1/2	33.183	45.36	23.57	3.148	105.0	53.6	32.31	17.46	10.52	5.89	4.278	2.560	1.784
	7	38.485	40.06	20.81	2.780	92.7	47.4	28.53	15.42	9.29	5.20	3.778	2.261	1.575
12	0	0	113.10	58.76	7.849	261.8	133.7	80.55	43.53	26.22	14.69	10.668	6.383	4.448
	2	3.142	109.96	57.12	7.631	254.5	130.0	78.32	42.32	25.49	14.28	10.371	6.206	4.324
	2-1/2	4.909	108.19	56.21	7.508	250.4	127.9	77.06	41.64	25.08	14.05	10.205	6.106	4.255
	3	7.069	106.03	55.08	7.359	245.4	125.3	75.52	40.81	24.58	13.77	10.001	5.984	4.170
	3-1/2	9.621	103.48	53.76	7.182	239.5	122.3	73.70	39.83	23.99	13.44	9.760	5.840	4.069
	4	12.566	100.53	52.23	6.977	232.7	118.8	71.60	38.70	23.30	13.06	9.482	5.674	3.954
	4-1/2	15.904	97.19	50.49	6.745	225.0	114.9	69.23	37.41	22.53	12.63	9.168	5.486	3.822
	5	19.635	93.46	48.55	6.486	216.4	110.5	66.57	35.98	21.67	12.14	8.816	5.275	3.676
	5-1/2	23.758	89.34	46.41	6.200	206.8	105.6	63.63	34.39	20.71	11.61	8.427	5.042	3.513
	6	28.274	84.82	44.06	5.887	196.4	100.3	60.42	32.65	19.66	11.02	8.001	4.787	3.336
	6-1/2	33.183	79.92	41.52	5.547	185.0	94.5	56.92	30.76	18.53	10.38	7.538	4.510	3.143
	7	38.485	74.61	38.77	5.179	172.7	88.2	53.14	28.72	17.30	9.69	7.038	4.211	2.934
	7-1/2	44.179	68.92	35.80	4.783	159.5	81.5	49.09	26.53	15.98	8.95	6.501	3.890	2.710
	8	50.266	62.83	32.64	4.360	145.4	74.3	44.75	24.19	14.57	8.16	5.926	3.546	2.471
	8-1/2	56.745	56.35	29.27	3.911	130.5	66.6	40.14	21.69	13.06	7.32	5.315	3.181	2.216
14	0	0	153.94	79.97	10.683	356.3	182.0	109.6	59.25	35.68	20.00	14.52	8.688	6.054
	2-1/2	4.909	149.03	77.42	10.343	345.0	176.2	106.2	57.36	34.55	19.36	14.06	8.411	5.861
	3	7.069	146.87	76.30	10.193	340.0	173.6	104.6	56.53	34.05	19.08	13.85	8.289	5.776
	3-1/2	9.621	144.32	74.97	10.016	334.1	170.6	102.8	55.55	33.45	18.75	13.61	8.145	5.676
	4	12.566	141.37	73.44	9.811	327.3	167.1	100.7	54.42	32.77	18.37	13.33	7.979	5.560
	4-1/2	15.904	138.03	71.71	9.579	319.5	163.2	98.3	53.13	32.00	17.93	13.02	7.791	5.428
	5	19.635	134.30	69.77	9.320	310.9	158.8	95.7	51.70	31.13	17.45	12.67	7.580	5.282
	5-1/2	23.758	130.18	67.63	9.035	301.3	153.9	92.7	50.11	30.18	16.91	12.28	7.347	5.120

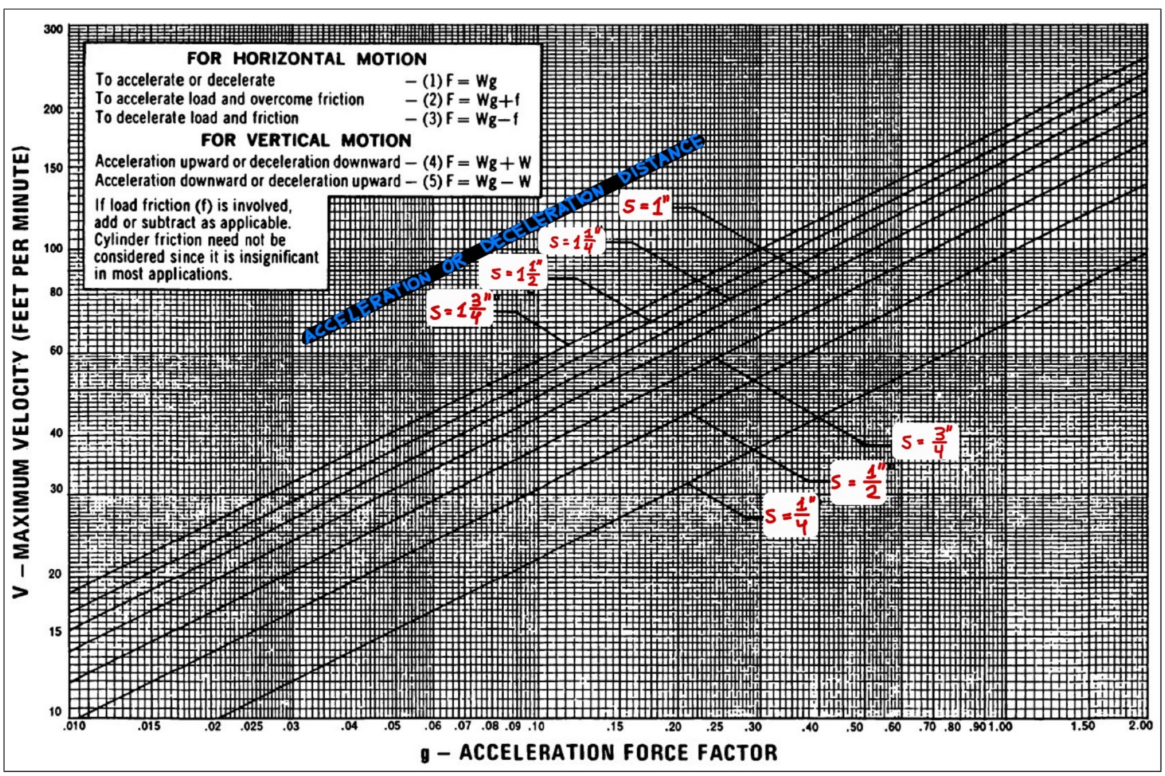
**If the rod is not available for the bore that you chose, choosing a larger bore and using the same rod works most of the time*

6. Check the acceleration/deceleration condition against your cylinder/rod selection. Use the graph below to find the value of the Acceleration Force Factor "g" and calculate the total force to make sure it doesn't exceed the allowed bore rating. The way to calculate F is given for some conditions in the top-left portion of the below graph.

   If you prefer a more reliable result, use the equation below instead:

$$\frac{V^2}{S} \times 0.0000517$$

   Here, V is the velocity you need to accelerate to, and S is the distance the acceleration must occur over.



7. Find the flow rate [GPM]. Use the given extension speed (convert to ft/min if needed) and find the flow rate [GPM] using the tables above in step 5 (use your piston diameter with 0 inch rod diameter for extension problems!).[64]

   The tables present the flow rate at 10 ft/min piston velocity, therefore, use this equation to get the "true value" for your piston speed [ft/min]:

   $$Q_{true} = \frac{Piston\ Speed}{10} \times Q_{table}$$

   This will be the flow rate required for your piping!

8. Choose Pipe Diameter. Again, use the tables in step 5, which give the fluid velocity for different pipe diameters at 10 ft/min piston speed (Use 0 rod diameter for extension case!). Make sure the fluid velocity is less than 15 ft/sec. In other words:

   $$Fluid\ Velocity \times \frac{Piston\ Speed}{10} < 15 ft/s$$

   In some cases you may be able to calculate it, but I suggest just sticking to the table and choosing the smallest piping possible (to cut down on costs of course!).

9. Next, you must choose a motor. First, find your minimum horsepower. Now there is a table to look at below, but I'd suggest just using the equation below:

   $$HP_{min} = \frac{GPM \times PSI}{1714 \times 0.85}$$

   Then, using the minimum horsepower you calculated, choose a motor from the tables below (usually 3-phase).

[64] Some questions may just give you the flow rate, so skip this step if that's the case

3 PHASE MOTOR STARTERS

1/2 TO 20 H.P. MOTOR H.P. 3ø	1/2		3/4		1		1-1/2		2		3		5		7-1/2		10		15		20	
VOLTAGE	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440
Nema Starter Size	00	00	00	00	00	00	00	00	0	00	0	0	1	0	1	1	2	1	2	2	3	2
⊕ Full Load Current	2.0	1.0	2.8	1.4	3.5	1.8	5.0	2.5	6.5	3.3	9.0	4.5	15	7.5	22	11	28	14	40	20	52	26
Fuses – Amps Std. N.E.C.	15	15	15	15	15	15	15	15	20	15	25	15	40	20	60	30	70	35	100	50	150	70
Fuses – Amps Dual Element	5	4	5	4	8	4	8	5	12	8	15	10	25	15	30	20	45	25	60	30	80	40
Circuit Breaker Max. Amps.	15	15	15	15	15	15	15	15	15	15	20	15	30	15	50	20	50	30	70	40	100	50
Minimum Wire Sizes R, RW, T, TW	14	14	14	14	14	14	14	14	14	14	14	14	12	14	10	14	8	12	6	10	4	8
Minimum Wire Sizes RH	14	14	14	14	14	14	14	14	14	14	14	14	12	14	10	14	8·	12	6	10	6	8
Always specify voltage and frequency.																						

3 PHASE MOTOR STARTERS

25 TO 200 H.P. MOTOR H.P. 3ø	25		30		40		50		60		75		100		125		150		200	
VOLTAGE	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440	220	440
Nema Starter Size	3	2	3	3	4	3	4	3	5	4	5	4	5	4	6	5	6	5	6	5
⊕ Full Load Current	64	32	78	39	104	52	125	63	150	75	185	93	246	123	310	155	360	180	480	240
Fuses – Amps Std. N.E.C.	175	80	200	100	300	150	350	175	400	200	500	250	600	350	–	400	–	450	–	600
Fuses – Amps Dual Element	100	50	125	60	175	80	200	100	225	125	300	150	400	200	450	250	600	300	–	400
Circuit Breaker Max. Amps.	125	-50	100	70	175	100	200	125	225	125	300	150	400	200	–	250	–	300	–	400
Minimum Wire Sizes R, RW, T, TW	3	8	1	6	00	4	000	3	0000	2	300	0	500	000	–	0000	–	300	–	500
Minimum Wire Sizes RH	4	8	3	6	1	6	00	4	000	3	0000	1	350	00	–	000	–	0000	–	350
Always specify voltage and frequency.																				

SINGLE PHASE MOTOR STARTERS

1/6 TO 5 H.P. MOTOR H.P. 1ø	1/6		1/4		1/3		1/2		3/4		1		1-1/2		2		3		5	
VOLTAGE	115	230	115	230	115	230	115	230	115	230	115	230	115	230	115	230	115	230	115	230
⊕ Full Load Current	4.4	2.2	5.8	2.9	7.2	3.6	9.8	4.9	13.8	6.9	16	8	20	10	24	12	34	17	56	28
Fuses – Amps. Std. N.E.C.	15	15	20	15	25	15	30	15	45	25	50	25	60	30	80	40	100	60	–	90
Circuit Breaker Max. Amps.	15	15	15	15	15	15	30	15	40	20	40	20	50	30	70	30	100	50	–	70
Min. Wire Sizes – R, RH, RW, T, TW	14	14	14	14	14	14	14	14	12	14	12	14	10	14	10	14	6	10	–	8

Don't worry too much about the voltages and other specs other than the HP, so choose them rather arbitrarily.

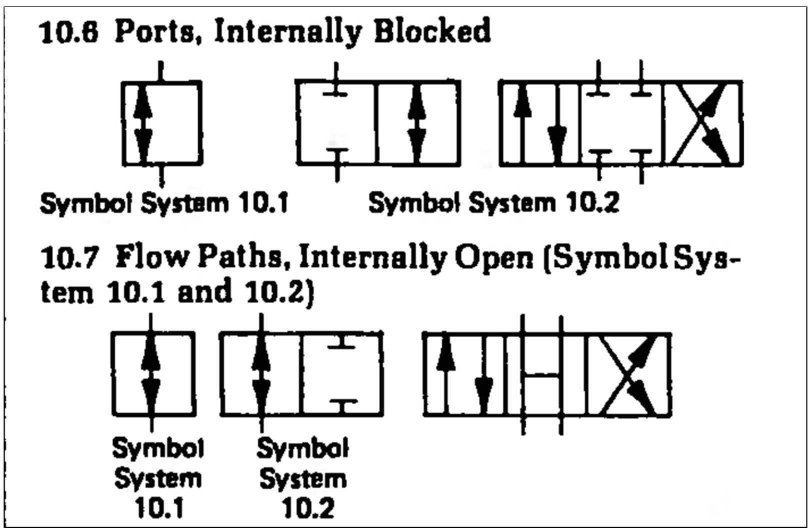
You should now have a fully specified system!

Good work!

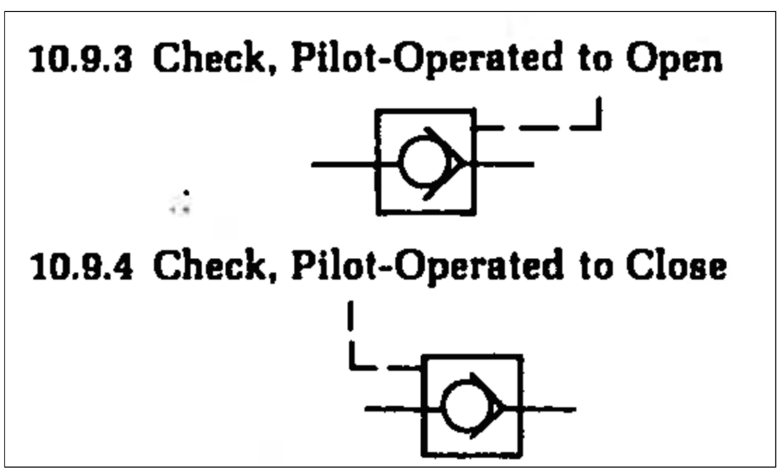
10.4 Figures, Definitions, and Relevant Questions

Below is an overview of fluid symbols, introducing what you'll generally need to know to understand what's happening when looking at a fluid circuit.[65]

We will start off with valves. Valves are composed of one or more envelopes (square boxes) with lines inside that indicate fluid flow. **Directional Control Valves** are shown in the figure below, which you'll probably see in almost every fluid circuit you'll analyze.

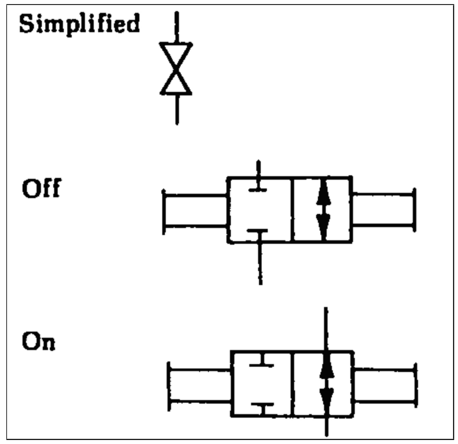


Check valves are valves where the flow to the right is blocked, but flow to the left is permitted. They are used to **prevent backflow**.

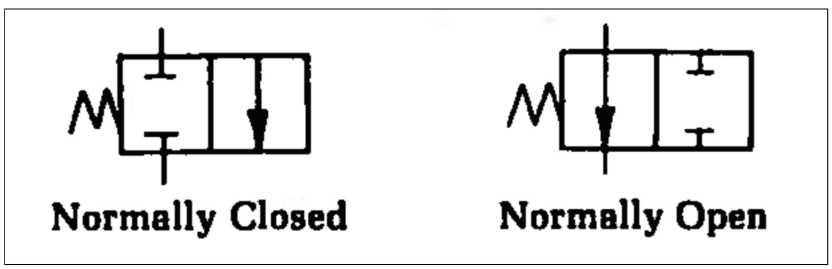


[65] I will list good-to-know ones only

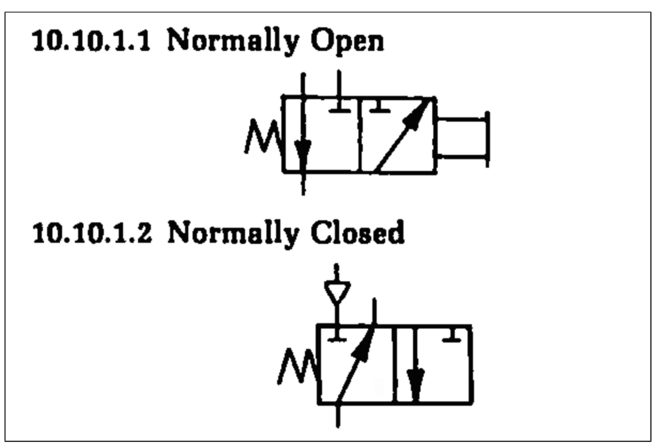
On-Off valves (manual) are as simple as it gets, either fluid flows or it doesn't.



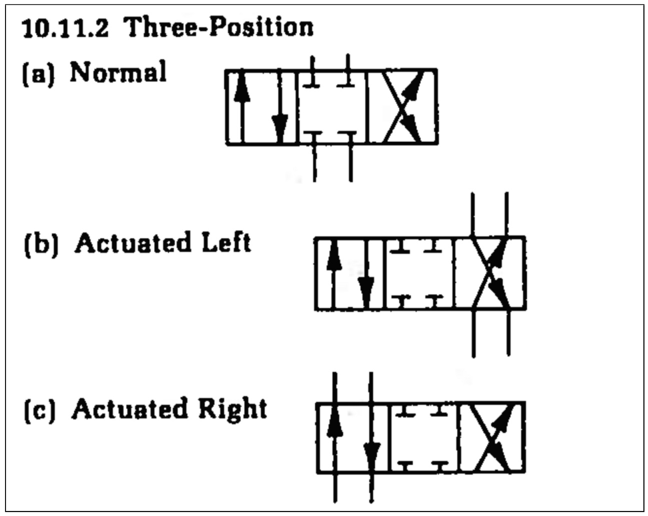
Two-way valves need not necessarily be manual. In some cases, a **spring** (squiggly line in the figure below) is used to switch between the two positions.



We can have 3-way or 4-way valves too. I have provided some examples below.

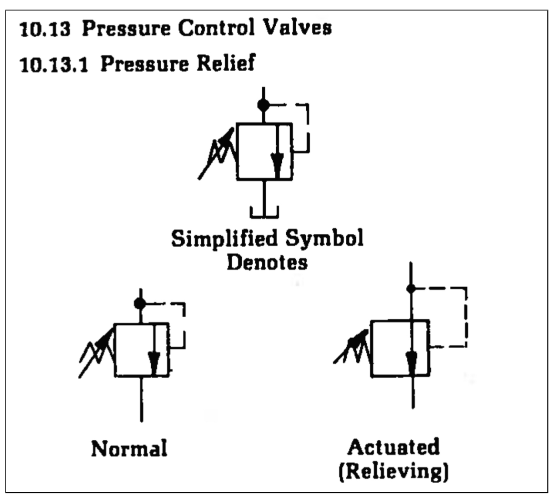


2-position 3-way valve



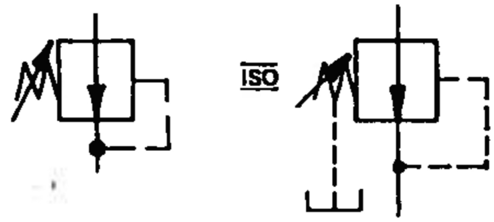
3-position 4-way valve

Now we move to **Pressure Control Valves**, and mainly the **Pressure Relief Valve**, which is used for safety and literally helps relief the system by diverting flow as the operating pressure is reached. This helps keep the system at a certain operating pressure.



Another pressure control valve is the **Pressure Reducing Valve**, which reduces the inlet pressure to a controlled and constant pressure at the outlet.

10.13.3 Pressure Reducing



Basically, the following valves should be enough for you to describe circuits in Mech 325 when asked to. There may be some that I missed, but I am sure the ones I've included are essential for your understanding of this unit.

11 Unit 4: Bushings and Bearings

11.1 Class Memory

I'll keep this short. I attended Jon's lecture on Bushings, and kid you not, it was probably the only lecture where I found it possible to fully follow up with him. He went through an example or two as well, which I thought was great. I am not sure if he had extra time or something that day, but he definitely went in a more than acceptable pace during that lecture. Overall, bushing selection is not that difficult, but I feel like that lecture made it crystal clear. I would suggest making it top priority to attend that lecture if possible.

11.2 Do's and Don'ts

11.3 Unit Guide

11.4 Figures, Definitions, and Relevant Questions

12 Unit 5: Shaft Analysis and Accessories

12.1 Class Memory

The highlight of our design project was trying to finalize the shaft for our shredder. In class, Jon provided good insight into Shaft Analysis and different Shaft Accessories (Keys, Splines, Pins, ... etc.). I personally found sizing shafts to be a bit more straightforward than the previous topics, but let me tell you one thing: you will NEED to draw moment, shear, and even torque diagrams (Mohr's Circles too), so let the 360 PTSD kick in for a little bit here. I hope it will be alright, as I've thrown in a code that automatically graphs the shear and moments diagrams, courtesy of Mr Felix Wilton himself (here).

12.2 Do's and Don'ts

12.3 Unit Guide

12.4 Figures, Definitions, and Relevant Questions

13 Unit 6: Springs and Fasteners

13.1 Class Memory

I didn't really attend class anymore at this point of the semester. The other courses were getting quite hectic, and I remember not even taking a look at this unit's assignment. As a result, I will leave this portion to you to continue on your own. Find out how to contact me in the final section.

I really hope you can do better!

13.2 Do's and Don'ts

13.3 Unit Guide

13.4 Figures, Definitions, and Relevant Questions

14 Closing Remarks

If you are planning to fail this course, DON'T. I would imagine that after spending countless hours putting together this spectacular document for you, you would at least give me the courtesy of passing this course. With this powerful weapon[66] in your hands, I expect you to score above average with minimal effort.

Regardless of when you are reading this, whether it be before the final, at the beginning of the term, or after classes are over, I hope you do/did well in Mech 325!

Writing this document has been a blast! From the bottom of my heart, I hope you found this document useful!

[66] this document...

15 Acknowledgements

I would like to thank the text book reviewers as well as all fellow Mech 325 survivors who in some shape and form helped make this document possible.

Of course, if it wasn't for Jon's Any% WR 100 slide speed-run attempt lectures, this document would've never been made, so I would like to thank Professor Jon Mikkelsen as well.

As of October 24th, 2023, I would also like to thank all the dogs for their splendid efforts at progressing what we now call the DOGument. I shall not name any specific names but y'all are all legends, and deserve to be acknowledged.

16 Contact Me (Please Leave Me Alone)

If you'd like to contact me for any reasons, just let me know in person. I am very easy to reach. Alternatively, you could also send me an email on

yasiinranjbar@gmail.com

Appendices

A Timing Belt Tables

TABLE 7–7 8-mm Pitch GT Drive Selection Table

Sprocket combinations			Center distance (inches)															
			920-8MGT	960-8MGT	1040-8MGT	1064-8MGT	1120-8MGT	1160-8MGT	1200-8MGT	1224-8MGT	1280-8MGT	1440-8MGT	1512-8MGT	1584-8MGT	1600-8MGT	1760-8MGT	1800-8MGT	2000-8MGT
			P.L. 36.220	P.L. 37.795	P.L. 40.945	P.L. 41.890	P.L. 44.094	P.L. 45.669	P.L. 47.244	P.L. 48.189	P.L. 50.394	P.L. 56.693	P.L. 59.528	P.L. 62.362	P.L. 62.992	P.L. 69.291	P.L. 70.866	P.L. 78.740
Driver	Driven	Velocity ratio	115 teeth	120 teeth	130 teeth	133 teeth	140 teeth	145 teeth	150 teeth	153 teeth	160 teeth	180 teeth	189 teeth	198 teeth	200 teeth	220 teeth	225 teeth	250 teeth
22	22	1.000	14.65	15.43	17.01	17.48	18.58	19.37	20.16	20.63	21.73	24.88	26.30	27.72	28.03	31.18	31.97	35.90
24	24	1.000	14.33	15.12	16.69	17.17	18.27	19.06	19.84	20.32	21.42	24.57	25.98	27.40	27.72	30.87	31.65	35.59
26	26	1.000	14.02	14.80	16.38	16.85	17.95	18.74	19.53	20.00	21.10	24.25	25.67	27.09	27.40	30.55	31.34	35.28
28	28	1.000	13.70	14.49	16.06	16.54	17.64	18.43	19.21	19.69	20.79	23.94	25.35	26.77	27.09	30.24	31.02	34.96
30	30	1.000	13.39	14.17	15.75	16.22	17.32	18.11	18.90	19.37	20.47	23.62	25.04	26.46	26.77	29.92	30.71	34.65
32	32	1.000	13.07	13.86	15.43	15.91	17.01	17.80	18.58	19.06	20.16	23.31	24.72	26.14	26.46	29.61	30.39	34.33
34	34	1.000	12.76	13.54	15.12	15.59	16.69	17.48	18.27	18.74	19.84	22.99	24.41	25.83	26.14	29.29	30.08	34.02
36	36	1.000	12.44	13.23	14.80	15.28	16.38	17.17	17.95	18.43	19.53	22.68	24.09	25.51	25.83	28.98	29.76	33.70
38	38	1.000	12.13	12.91	14.49	14.96	16.06	16.85	17.64	18.11	19.21	22.36	23.78	25.20	25.51	28.66	29.45	33.39
40	40	1.000	11.67	12.46	14.03	14.50	15.61	16.39	17.18	17.65	18.76	21.91	23.32	24.74	25.06	28.21	28.99	32.93
44	44	1.000	11.18	11.97	13.54	14.02	15.12	15.91	16.69	17.17	18.27	21.42	22.83	24.25	24.57	27.72	28.50	32.44
48	48	1.000	10.55	11.34	12.91	13.39	14.49	15.28	16.06	16.54	17.64	20.79	22.21	23.62	23.94	27.09	27.87	31.81
56	56	1.000	9.29	10.08	11.65	12.13	13.23	14.02	14.80	15.28	16.38	19.53	20.95	22.36	22.68	25.83	26.61	30.55
64	64	1.000	8.03	8.82	10.39	10.87	11.97	12.76	13.54	14.02	15.12	18.27	19.69	21.10	21.42	24.57	25.35	29.29
72	72	1.000	-	-	9.13	9.61	10.71	11.50	12.28	12.76	13.86	17.01	18.43	19.84	20.16	23.31	24.10	28.03
80	80	1.000	-	-	-	-	9.45	10.24	11.02	11.50	12.60	15.75	17.17	18.58	18.90	22.05	22.84	26.77
24	30	1.250	13.85	14.64	16.22	16.69	17.79	18.58	19.37	19.84	20.94	24.09	25.51	26.93	27.24	30.39	31.18	35.12
32	40	1.250	12.43	13.22	14.80	15.27	16.37	17.16	17.95	18.42	19.52	22.67	24.09	25.51	25.82	28.97	29.76	33.70
64	80	1.250	-	-	9.10	9.57	10.68	11.47	12.26	12.73	13.84	16.99	18.41	19.83	20.14	23.29	24.08	28.02
72	90	1.250	-	-	-	-	9.25	10.04	10.83	11.30	12.41	15.56	16.98	18.40	18.72	21.87	22.66	26.60
24	32	1.333	13.70	14.48	16.06	16.53	17.63	18.42	19.21	19.68	20.78	23.93	25.35	26.77	27.08	30.23	31.02	34.96
30	40	1.333	12.59	13.38	14.95	15.42	16.53	17.32	18.10	18.58	19.68	22.83	24.25	25.66	25.98	29.13	29.92	33.85

(*continued*)

TABLE 7–7 (*continued*)

Sprocket combinations			Center distance (inches)															
			920-8MGT P.L. 36.220 115 teeth	960-8MGT P.L. 37.795 120 teeth	1040-8MGT P.L. 40.945 130 teeth	1064-8MGT P.L. 41.890 133 teeth	1120-8MGT P.L. 44.094 140 teeth	1160-8MGT P.L. 45.669 145 teeth	1200-8MGT P.L. 47.244 150 teeth	1224-8MGT P.L. 48.189 153 teeth	1280-8MGT P.L. 50.394 160 teeth	1440-8MGT P.L. 56.693 180 teeth	1512-8MGT P.L. 59.528 189 teeth	1584-8MGT P.L. 62.362 198 teeth	1600-8MGT P.L. 62.992 200 teeth	1760-8MGT P.L. 69.291 220 teeth	1800-8MGT P.L. 70.866 225 teeth	2000-8MGT P.L. 78.740 250 teeth
Driver	Driven	Velocity ratio																
36	48	1.333	11.48	12.27	13.85	14.32	15.42	16.21	17.00	17.47	18.57	21.72	23.14	24.56	24.87	28.03	28.81	32.75
48	64	1.333	9.26	10.05	11.63	12.10	13.20	13.99	14.78	15.25	16.36	19.51	20.93	22.35	22.66	25.81	26.60	30.54
24	36	1.500	13.36	14.15	15.73	16.20	17.30	18.09	18.88	19.35	20.46	23.61	25.02	26.44	26.76	29.91	30.70	34.63
32	48	1.500	11.78	12.57	14.15	14.62	15.73	16.52	17.30	17.78	18.88	22.03	23.45	24.87	25.18	28.34	29.12	33.06
48	72	1.500	8.58	9.37	10.96	11.43	12.54	13.33	14.12	14.60	15.70	18.86	20.28	21.70	22.01	25.17	25.96	29.90
22	44	2.000	12.87	13.66	15.24	15.71	16.81	17.60	18.39	18.87	19.97	23.12	24.54	25.96	26.28	29.43	30.22	34.16
24	48	2.000	12.38	13.17	14.75	15.23	16.33	17.12	17.91	18.39	19.49	22.65	24.06	25.48	25.80	28.95	29.74	33.68
28	56	2.000	11.41	12.20	13.79	14.26	15.37	16.16	16.95	17.42	18.53	21.69	23.11	24.53	24.84	28.00	28.79	32.73
32	64	2.000	10.43	11.22	12.81	13.29	14.40	15.19	15.98	16.46	17.56	20.73	22.15	23.57	23.88	27.04	27.83	31.77
36	72	2.000	9.43	10.24	11.83	12.31	13.42	14.22	15.01	15.49	16.60	19.76	21.18	22.61	22.92	26.08	26.87	30.81
40	80	2.000	8.42	9.23	10.84	11.32	12.44	13.23	14.03	14.51	15.62	18.79	20.22	21.64	21.96	25.12	25.91	29.85
56	112	2.000	-	-	-	-	-	9.18	10.00	10.49	11.63	14.85	16.29	17.73	18.05	21.23	22.03	25.99
72	144	2.000	-	-	-	-	-	-	-	-	-	-	12.22	13.70	14.02	17.26	18.06	22.07
32	80	2.500	8.97	9.78	11.40	11.88	13.01	13.81	14.61	15.08	16.20	19.38	20.81	22.23	22.55	25.71	26.51	30.46
36	90	2.500	7.71	8.55	10.19	10.68	11.82	12.62	13.43	13.91	15.03	18.22	19.66	21.09	21.40	24.58	25.37	29.32
24	72	3.000	10.27	11.08	12.69	13.17	14.29	15.08	15.88	16.36	17.47	20.65	22.07	23.50	23.82	26.98	27.77	31.72
30	90	3.000	8.10	8.94	10.60	11.09	12.23	13.04	13.85	14.33	15.46	18.65	20.09	21.52	21.84	25.02	25.81	29.77
48	144	3.000	-	-	-	-	-	-	-	-	-	12.29	13.81	15.31	15.64	18.92	19.73	23.76

B Chain Drive Tables

TABLE 7–14 Horsepower Ratings—Single-Strand Roller Chain No. 40

No. of teeth	0.500-in pitch				Rotational speed of small sprocket, rev/min																			
	10	25	50	100	180	200	300	500	700	900	1000	1200	1400	1600	1800	2100	2500	3000	3500	4000	5000	6000	7000	8000	**9000**
11	0.06	0.14	0.27	0.52	0.91	1.00	1.48	2.42	3.34	4.25	4.70	5.60	6.49	5.57	4.66	3.70	2.85	2.17	1.72	1.41	1.01	0.77	0.61	0.50	0.00
12	0.06	0.15	0.29	0.56	0.99	1.09	1.61	2.64	3.64	4.64	5.13	6.11	7.09	6.34	5.31	4.22	3.25	2.47	1.96	1.60	1.15	0.87	0.69	0.57	0.00
13	0.07	0.16	0.31	0.61	1.07	1.19	1.75	2.86	3.95	5.02	5.56	6.62	7.68	7.15	5.99	4.76	3.66	2.79	2.21	1.81	1.29	0.98	0.78	0.00	
14	0.07	0.17	0.34	0.66	1.15	1.28	1.88	3.08	4.25	5.41	5.98	7.13	8.27	7.99	6.70	5.31	4.09	3.11	2.47	2.02	1.45	1.10	0.87	0.00	
15	0.08	0.19	0.36	0.70	1.24	1.37	2.02	3.30	4.55	5.80	6.41	7.64	8.86	8.86	7.43	5.89	4.54	3.45	2.74	2.24	1.60	1.22	0.97	0.00	
16	0.08	0.20	0.39	0.75	1.32	1.46	2.15	3.52	4.86	6.18	6.84	8.15	9.45	9.76	8.18	6.49	5.00	3.80	3.02	2.47	1.77	1.34	0.00		
17	0.09	0.21	0.41	0.80	1.40	1.55	2.29	3.74	5.16	6.57	7.27	8.66	10.04	10.69	8.96	7.11	5.48	4.17	3.31	2.71	1.94	1.47	0.00		
18	0.09	0.22	0.43	0.84	1.48	1.64	2.42	3.96	5.46	6.95	7.69	9.17	10.63	11.65	9.76	7.75	5.97	4.54	3.60	2.95	2.11	1.60	0.00		
19	0.10	0.24	0.46	0.89	1.57	1.73	2.56	4.18	5.77	7.34	8.12	9.66	11.22	12.64	10.59	8.40	6.47	4.92	3.91	3.20	2.29	0.09	0.00		
20	0.10	0.25	0.48	0.94	1.65	1.82	2.69	4.39	6.07	7.73	8.55	10.18	11.81	13.42	11.44	9.07	6.99	5.31	4.22	3.45	2.47	0.00			
21	0.11	0.26	0.51	0.98	1.73	1.91	2.83	4.61	6.37	8.11	8.98	10.69	12.40	14.10	12.30	9.76	7.52	5.72	4.54	3.71	2.65	0.00			
22	0.11	0.27	0.53	1.03	1.81	2.01	2.96	4.83	6.68	8.50	9.40	11.20	12.99	14.77	13.19	10.47	8.06	6.13	4.87	3.98	2.85	0.00			
23	0.12	0.28	0.56	1.08	1.90	2.10	3.10	5.05	6.98	8.89	9.83	11.71	13.58	15.44	14.10	11.19	8.62	6.55	5.20	4.26	3.05	0.00			
24	0.12	0.30	0.58	1.12	1.98	2.19	3.23	5.27	7.28	9.27	10.26	12.22	14.17	16.11	15.03	11.93	9.18	6.99	5.54	4.54	0.87	0.00			
25	0.13	0.31	0.60	1.17	2.06	2.28	3.36	5.49	7.59	9.66	10.69	12.73	14.76	16.78	15.98	12.68	9.76	7.43	5.89	4.82	0.00				
26	0.13	0.32	0.63	1.22	2.14	2.37	3.50	5.71	7.89	10.04	11.11	13.24	15.35	17.45	16.95	13.45	10.36	7.88	6.25	5.12	0.00				
28	0.14	0.35	0.67	1.31	2.31	2.55	3.77	6.15	8.50	10.82	11.97	14.26	16.53	18.79	18.94	15.03	11.57	8.80	6.99	5.72	0.00				
30	0.15	0.37	0.72	1.41	2.47	2.74	4.04	6.59	9.11	11.59	12.82	15.28	17.71	20.14	21.01	16.67	12.84	9.76	7.75	6.34	0.00				
32	0.16	0.40	0.77	1.50	2.64	2.92	4.31	7.03	9.71	12.38	13.66	16.30	18.89	21.48	23.14	18.37	14.14	10.76	8.54	1.41					
35	0.18	0.43	0.84	1.64	2.88	3.19	4.71	7.69	10.62	13.52	14.96	17.82	20.67	23.49	26.30	21.01	16.17	12.30	9.76	0.00					
40	0.21	0.50	0.96	1.87	3.30	3.65	5.38	8.79	12.14	15.45	17.10	20.37	23.62	26.85	30.06	25.67	19.76	15.03	0.00						
45	0.23	0.56	1.08	2.11	3.71	4.10	6.06	9.89	13.66	17.39	19.24	22.92	26.57	30.20	33.82	30.63	23.58	5.53	0.00						
	Type A				Type B									Type C											

Type A: Manual or drip lubrication
Type B: Bath or disc lubrication
Type C: Oil stream lubrication

TABLE 7–15 Horsepower Ratings—Single-Strand Roller Chain No. 60

No. of teeth	0.750-in pitch											Rotational speed of small sprocket, rev/min													
	10	25	50	100	120	200	300	400	500	600	800	1000	1200	1400	1600	1800	2000	2500	3000	3500	4000	4500	5000	5500	6000
11	0.19	0.46	0.89	1.72	2.05	3.35	4.95	6.52	8.08	9.63	12.69	15.58	11.85	9.41	7.70	6.45	5.51	3.94	3.00	2.38	1.95	1.63	1.39	1.21	0.00
12	0.21	0.50	0.97	1.88	2.24	3.66	5.40	7.12	8.82	10.51	13.85	17.15	13.51	10.72	8.77	7.35	6.28	4.49	3.42	2.71	2.22	1.86	1.59	1.38	0.00
13	0.22	0.54	1.05	2.04	2.43	3.96	5.85	7.71	9.55	11.38	15.00	18.58	15.23	12.08	9.89	8.29	7.08	5.06	3.85	3.06	2.50	2.10	1.79	0.00	
14	0.24	0.58	1.13	2.19	2.61	4.27	6.30	8.30	10.29	12.26	16.15	20.01	17.02	13.51	11.05	9.26	7.91	5.66	4.31	3.42	2.80	2.34	0.41	0.00	
15	0.26	0.62	1.21	2.35	2.80	4.57	6.75	8.90	11.02	13.13	17.31	21.44	18.87	14.98	12.26	10.27	8.77	6.28	4.77	3.79	3.10	2.60	0.00		
16	0.27	0.66	1.29	2.51	2.99	4.88	7.20	9.49	11.76	14.01	18.46	22.87	20.79	16.50	13.51	11.32	9.66	6.91	5.26	4.17	3.42	1.78	0.00		
17	0.29	0.70	1.37	2.66	3.17	5.18	7.65	10.08	12.49	14.88	19.62	24.30	22.77	18.07	14.79	12.40	10.58	7.57	5.76	4.57	3.74	0.00			
18	0.31	0.75	1.45	2.82	3.36	5.49	8.10	10.68	13.23	15.76	20.77	25.73	24.81	19.69	16.11	13.51	11.53	8.25	6.28	4.98	4.08	0.00			
19	0.33	0.79	1.53	2.98	3.55	5.79	8.55	11.27	13.96	16.63	21.92	27.16	26.91	21.35	17.48	14.65	12.50	8.95	6.81	5.40	0.20	0.00			
20	0.34	0.83	1.61	3.13	3.73	6.10	9.00	11.86	14.70	17.51	23.08	28.59	29.06	23.06	18.87	15.82	13.51	9.66	7.35	5.83	0.00				
21	0.36	0.87	1.69	3.29	3.92	6.40	9.45	12.46	15.43	18.38	24.23	30.02	31.26	24.81	20.31	17.02	14.53	10.40	7.91	6.28	0.00				
22	0.38	0.91	1.77	3.45	4.11	6.71	9.90	13.05	16.17	19.26	25.39	31.45	33.52	26.60	21.77	18.25	15.58	11.15	8.48	0.00					
23	0.40	0.95	1.85	3.61	4.29	7.01	10.35	13.64	16.90	20.13	26.54	32.88	35.84	28.44	23.28	19.51	16.66	11.92	9.07	0.00					
24	0.41	0.99	1.93	3.76	4.48	7.32	10.80	14.24	17.64	21.01	27.69	34.31	38.20	30.31	24.81	20.79	17.75	12.70	9.66	0.00					
25	0.43	1.04	2.01	3.92	4.67	7.62	11.25	14.83	18.37	21.89	28.85	35.74	40.61	32.23	26.38	22.11	18.87	13.51	10.27	0.00					
26	0.45	1.08	2.09	4.08	4.85	7.93	11.70	15.42	19.11	22.76	30.00	37.17	43.07	34.18	27.98	23.44	20.02	14.32	10.90	0.00					
28	0.48	1.16	2.26	4.39	5.23	8.54	12.60	16.61	20.58	24.51	32.31	40.03	47.68	38.20	31.26	26.20	22.37	16.01	0.00						
30	0.52	1.24	2.42	4.70	5.60	9.15	13.50	17.79	22.05	26.26	34.62	42.89	51.09	42.36	34.67	29.06	24.81	17.75	0.00						
32	0.55	1.33	2.58	5.02	5.98	9.76	14.40	18.98	23.52	28.01	36.92	45.75	54.50	46.67	38.20	32.01	27.33	19.56	0.00						
35	0.60	1.45	2.82	5.49	6.54	10.67	15.75	20.76	25.72	30.64	40.39	50.03	59.60	53.38	43.69	36.62	31.26	1.35	0.00						
40	0.69	1.66	3.22	6.27	7.47	12.20	18.00	23.73	29.39	35.02	46.16	57.18	68.12	65.22	53.38	44.74	38.20	0.00							
45	0.77	1.86	3.63	7.05	8.40	13.72	20.25	26.69	33.07	38.39	51.92	64.33	76.63	77.83	63.70	53.38	12.45	0.00							
	Type A				Type B							Type C													

Type A: Manual or drip lubrication
Type B: Bath or disc lubrication
Type C: Oil stream lubrication

TABLE 7–16 **Horsepower Ratings—Single-Strand Roller Chain No. 80**

No. of teeth	1.000-in pitch				Rotational speed of small sprocket, rev/min																				
	10	25	50	75	88	100	200	300	400	500	600	700	800	900	1000	1200	1400	1600	1800	2000	2500	3000	3500	4000	4500
11	0.44	1.06	2.07	3.05	3.56	4.03	7.83	11.56	15.23	18.87	22.48	26.07	27.41	22.97	19.61	14.92	11.84	9.69	8.12	6.83	4.96	3.77	3.00	2.45	0.00
12	0.48	1.16	2.26	3.33	3.88	4.39	8.54	12.61	16.82	20.59	24.53	28.44	31.23	26.17	22.35	17.00	13.49	11.04	9.25	7.90	5.65	4.30	3.41	2.79	0.00
13	0.52	1.26	2.45	3.61	4.21	4.76	9.26	13.66	18.00	22.31	26.57	30.81	35.02	29.51	25.20	19.17	15.21	12.45	10.43	8.91	6.37	4.85	3.85	3.15	
14	0.56	1.35	2.63	3.89	4.53	5.12	9.97	14.71	19.39	24.02	28.62	33.18	37.72	32.98	28.16	21.42	17.00	13.91	11.66	9.96	7.12	5.42	4.30	3.52	
15	0.60	1.45	2.82	4.16	4.86	5.49	10.68	15.76	20.77	25.74	30.66	35.55	40.41	36.58	31.23	23.76	18.85	15.43	12.93	11.04	7.90	6.01	4.77	0.00	
16	0.64	1.55	3.01	4.44	5.18	5.86	11.39	16.81	22.16	27.45	32.70	37.92	43.11	40.30	34.41	26.17	20.77	17.00	14.25	12.16	8.70	6.62	5.25	0.00	
17	0.68	1.64	3.20	4.72	5.50	6.22	12.10	17.86	23.54	29.17	34.75	40.29	45.80	44.13	37.68	28.66	22.75	18.62	15.60	13.32	9.53	7.25	0.00		
18	0.72	1.74	3.39	5.00	5.83	6.59	12.81	18.91	24.93	30.88	36.79	42.66	48.49	48.08	41.05	31.23	24.78	20.29	17.00	14.51	10.39	7.90	0.00		
19	0.76	1.84	3.57	5.28	6.15	6.95	13.53	19.96	26.31	32.60	38.84	45.03	51.19	52.15	44.52	33.87	26.88	22.00	18.44	15.74	11.26	0.36	0.00		
20	0.80	1.93	3.76	5.55	6.47	7.32	14.24	21.01	27.70	34.32	40.88	47.40	53.88	56.32	48.08	36.58	29.03	23.76	19.91	17.00	12.16	0.00			
21	0.84	2.03	3.95	5.83	6.80	7.69	14.95	22.07	29.08	36.03	42.92	49.77	56.58	60.59	51.73	39.36	31.23	25.56	21.42	18.29	13.09	0.00			
22	0.88	2.13	4.14	6.11	7.12	8.05	15.66	23.12	30.47	37.75	44.97	52.14	59.27	64.97	55.47	42.20	33.49	27.41	22.97	19.61	14.03				
23	0.92	2.22	4.33	6.39	7.45	8.42	16.37	24.17	31.85	39.46	47.01	54.51	61.97	69.38	59.30	45.11	35.80	29.30	24.55	20.97	15.00				
24	0.96	2.32	4.52	6.66	7.77	8.78	17.09	25.22	33.24	41.18	49.06	56.88	64.66	72.40	63.21	48.08	38.16	31.23	26.17	22.35	15.99				
25	1.00	2.42	4.70	6.94	8.09	9.15	17.80	26.27	34.62	42.89	51.10	59.25	67.35	75.42	67.20	51.12	40.57	33.20	27.83	23.76	8.16				
26	1.04	2.51	4.89	7.22	8.42	9.52	18.51	27.32	36.01	44.61	53.14	61.62	70.05	78.43	71.27	54.22	43.02	36.22	29.51	25.20	0.00				
28	1.12	2.71	5.27	7.77	9.06	10.25	19.93	29.42	38.78	48.04	57.23	66.36	75.44	84.47	79.65	60.59	48.08	39.36	32.98	28.16	0.00				
30	1.20	2.90	5.64	8.33	9.71	10.98	21.36	31.52	41.55	51.47	61.32	71.10	80.82	90.50	88.33	67.20	53.33	43.65	36.58	31.23					
32	1.28	3.09	6.02	8.89	10.36	11.71	22.78	33.62	44.32	54.91	65.41	75.84	86.21	96.53	97.31	74.03	58.75	48.08	40.30	5.65					
35	1.40	3.38	6.58	9.72	11.33	12.81	24.92	36.78	48.47	60.05	71.54	82.95	94.29	105.58	111.31	84.68	67.20	55.00	28.15	0.00					
40	1.61	3.87	7.53	11.11	12.95	14.64	28.48	42.03	55.40	68.63	81.76	94.80	107.77	120.67	133.51	103.46	82.10	40.16	0.00						
45	1.81	4.35	8.47	12.49	14.57	16.47	32.04	47.28	62.32	77.21	91.98	106.65	121.24	135.75	150.20	123.45	72.28	0.00							
	Type A				Type B								Type C												

Type A: Manual or drip lubrication
Type B: Bath or disc lubrication
Type C: Oil stream lubrication

C Cross Product Review

To start off, imagine you want to find the cross product of two 3D vectors V_1 and V_2 in Cartesian coordinates (i, j, and k), such that

$$V_1 = A\,\hat{\imath} + B\,\hat{\jmath} + C\,\hat{k}$$
$$V_2 = D\,\hat{\imath} + E\,\hat{\jmath} + F\,\hat{k}$$

In this example, we want to find C, which is given by:

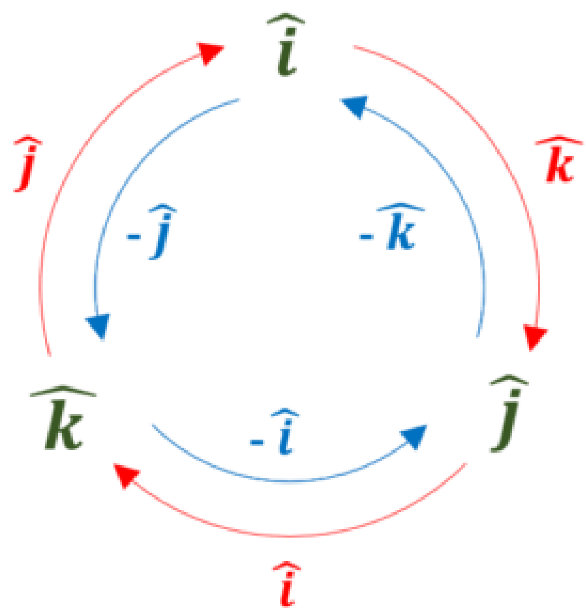
$$C = V_1 \times V_2$$

Before doing so, let's remember two important things about cross products:

1. **Crossing two vectors that point in the exact same direction results in zero**. This means that:

$$\hat{\imath} \times \hat{\imath} = 0$$
$$\hat{\jmath} \times \hat{\jmath} = 0$$
$$\hat{k} \times \hat{k} = 0$$

2. **The cross product sign convention**. This is given by the figure below. Further explanation will also be provided below.



Source: DCBA Online

Now that we've presented the following vital information, let's reintroduce ourselves to the matrix notation for cross products, given by the following matrix.

$$\begin{bmatrix} \hat{i} & \hat{j} & \hat{k} \\ A & B & C \\ D & E & F \end{bmatrix}$$

The top row presents the directions in the Cartesian coordinate system, whereas the next 2 rows present the 3D components of the vectors V_1 and V_2, respectively.

The cross product is then given by finding the determinant of the matrix above using the cofactor method. This can be written as:

$$C = V_1 \times V_2 = (BF - CE)\hat{i} - (AF - CD)\hat{j} + (AE - BD)\hat{k}$$

Note that the '$-$' sign in the $\hat{j}$ component is due to the "place signs" of each elements in a square matrix, which is introduced when finding the determinant of a "$n \times n$" matrix using the co-factor method. See below:

EXTRA:

What is beautiful about the cross product in Cartesian coordinates is that it will always follow the abovementioned formula and method. In case you need to check your answer when doing this method, use the cross function in Matlab. Here are the steps:

1. Define your vectors as two 1x3 matrices. For example:

$$A = [1; 2; 3]$$
$$B = [4; 5; 6]$$

2. Use the cross function in Matlab to find the cross product:

$$C = cross(A, B)$$

3. Your answer will be outputted as a 1x3 column vector!

Hope this helps!