University of Technology
Department of Mechanical Engineering
Branch of General Mechanics Engineering

Mechanical Engineering Design II
Fourth year

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Lecturers: Design Group
2014-2015

REFRENCES:

1. Machine Elements in Mechanical Design, by: Robert L. Mott.
2. Machine Design, by: Black & Adams.
3. Standard Handbook of Machine Design, by: Joseph E. Shigley & D.N. Mischke.
4. Machine Design, by: D.N. Reshetov.
5. Design of Machine Elements, by: Virgil M. Faries.
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7. Machine Design, by: Roberts H. Creamer.
8. Machine Elements, by: V. Dobrovolskey.
9. Machine Design, by: Hall, Holowenko, (Schaum's Series).
10. Design Methods, by: G. Jones.
11. Engineering Design Methods, by: Nigel Cross.
12. Optimization Methods for Engineering Design, by: Richard L. Fox.
13. Optimization of Mechanical Elements, by: Ray Johnson.

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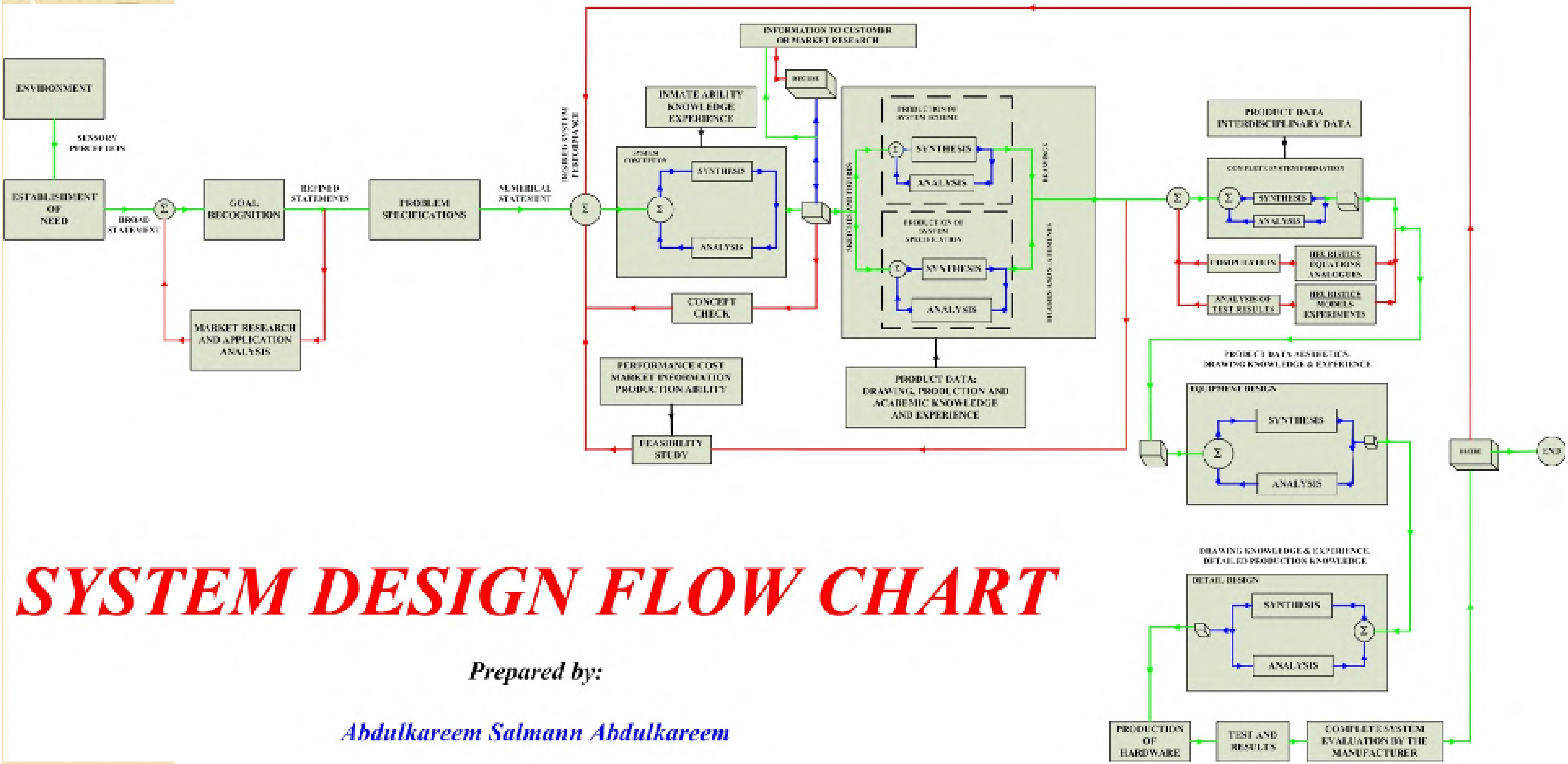
Mechanical Engineering Design II

First Lecture

Structure of Lectures



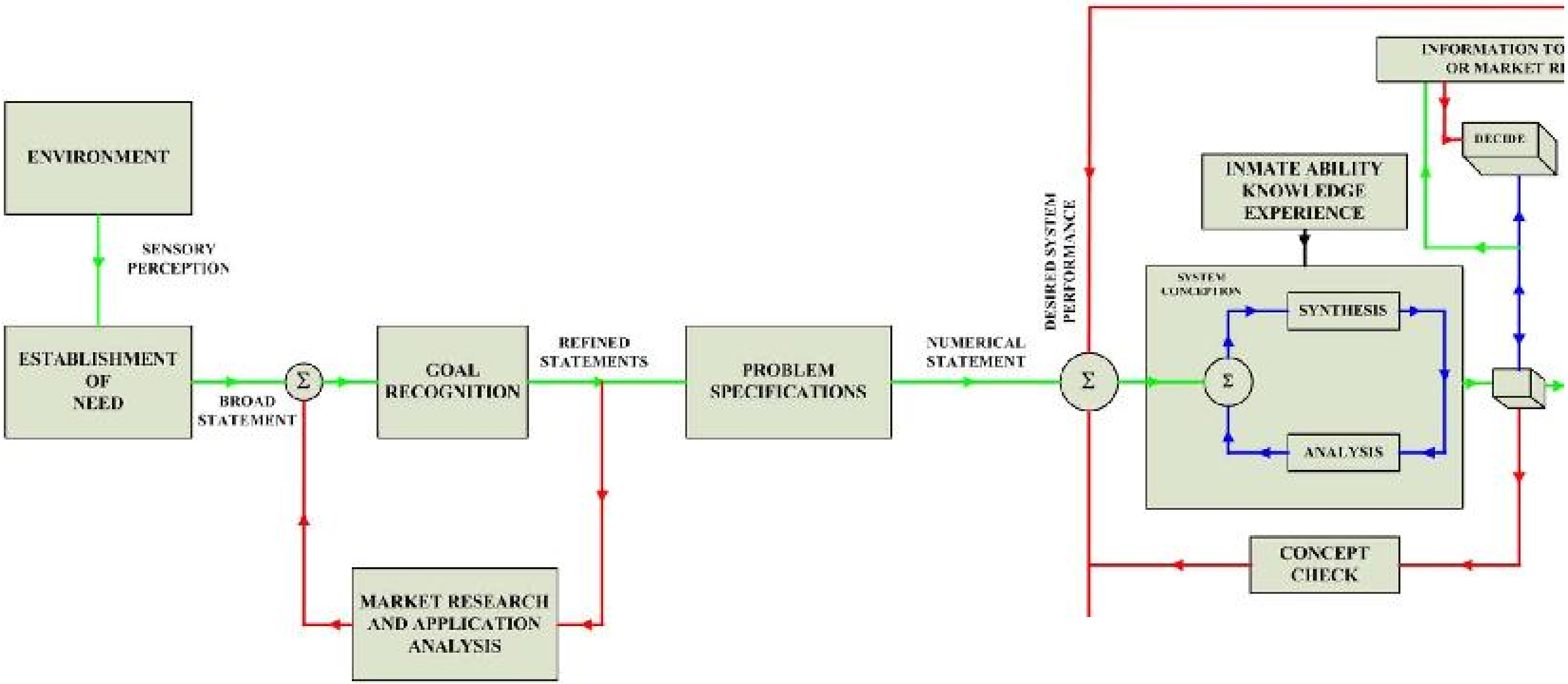
SYSTEM DESIGN FLOW CHART

Prepared by:

Abdulkareem Salmann Abdulkareem

First Semester

Part 1

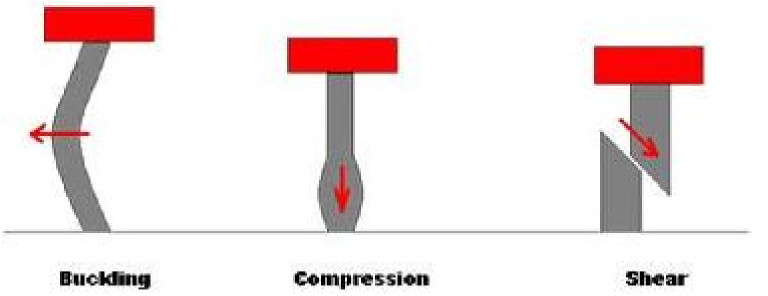


10% Mark

Part 2

Mechanical Elements Design

- **Column Analysis and Design**



- **Chain Design**

- **Introduction to Mdesign Program and Parametric Analysis Problems**

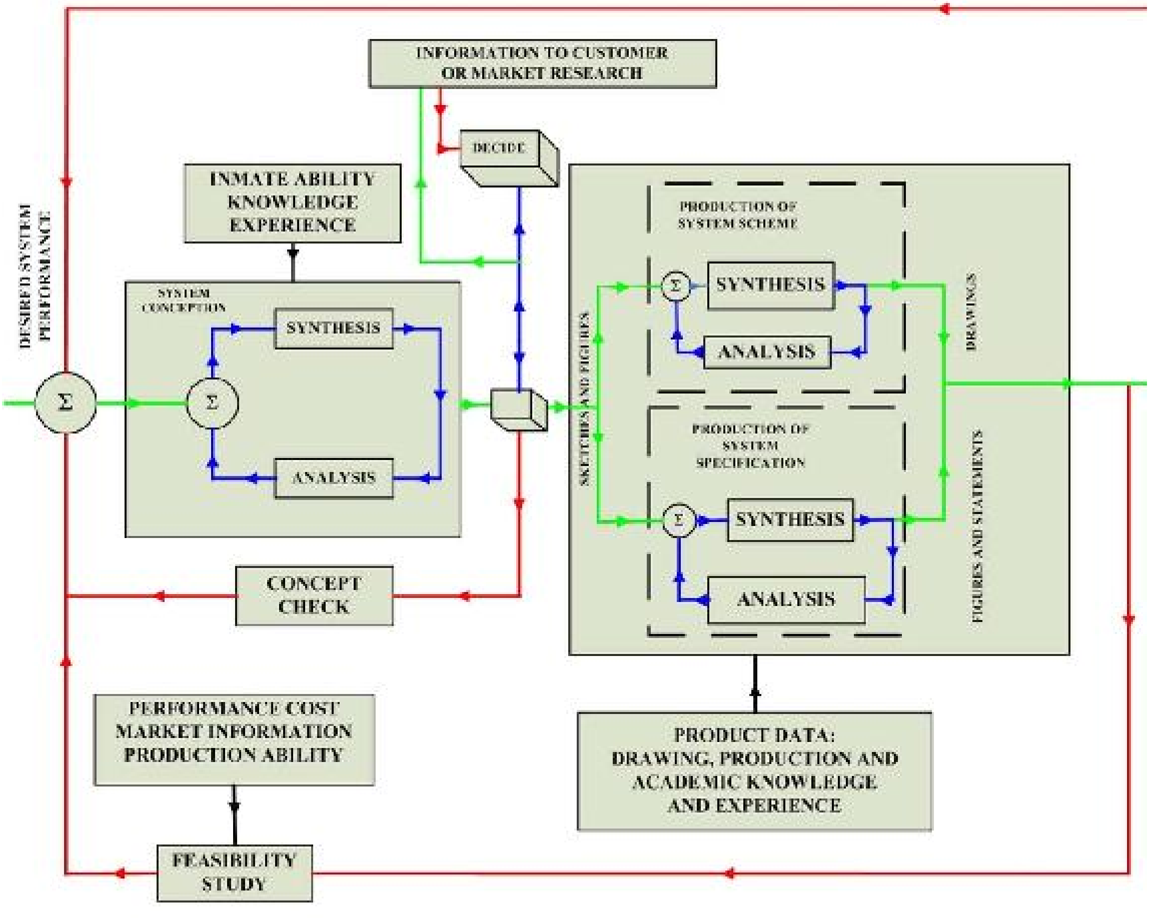
5% Mark

First Semester Examination

10% Mark

Second Semester

Part 1



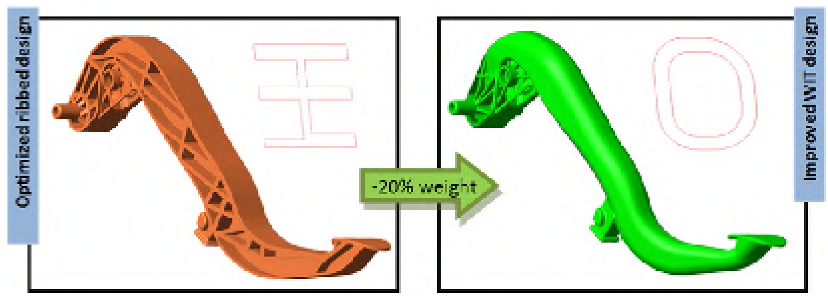
10% Mark

Part 2 **Mechanical Elements Design**

- **Belts Design**

- **Spur, Helical, Bevel and Worm Gear Design**

- **Introduction to Optimum Design**



5% Mark

Second Semester Examination 10% Mark

Final Examination 50% Mark

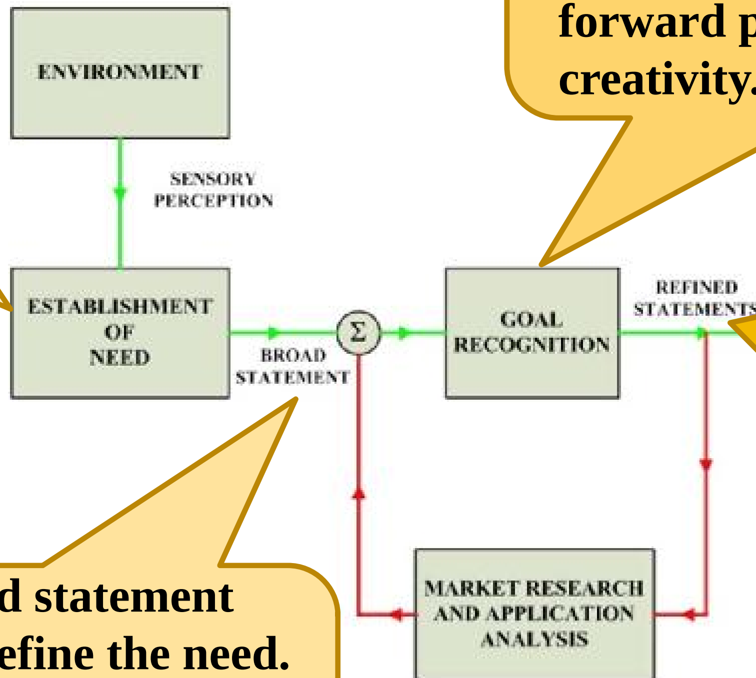
Mechanical Engineering Design II

Second Lecture

Introduction to the system design flow chart

Summary of the flow chart

1- The entry to the flow chart starts with the Establishment of need for the particular system.



2- Then the Broad statement can be made to define the need.
These statements are analyzed to obtain a goal, which will satisfy the need.

3- The forward path, which is a Goal-recognition, is a Synthesis and the output from the forward path is a product of creativity.

4- The feed back function, (Market research and Application analysis), is essentially an Analysis. The results of this analysis are compared with the Desired Input, (the comparison element), which is the Broad statement from the Establishment of need.

5- The Refined statement then made to specify the particular solution of the problem, but describe the goal toward, which we are heading.

Application of the items from no.1 to no.5 is as follows, using the following example:

Design Equipment to convert waste

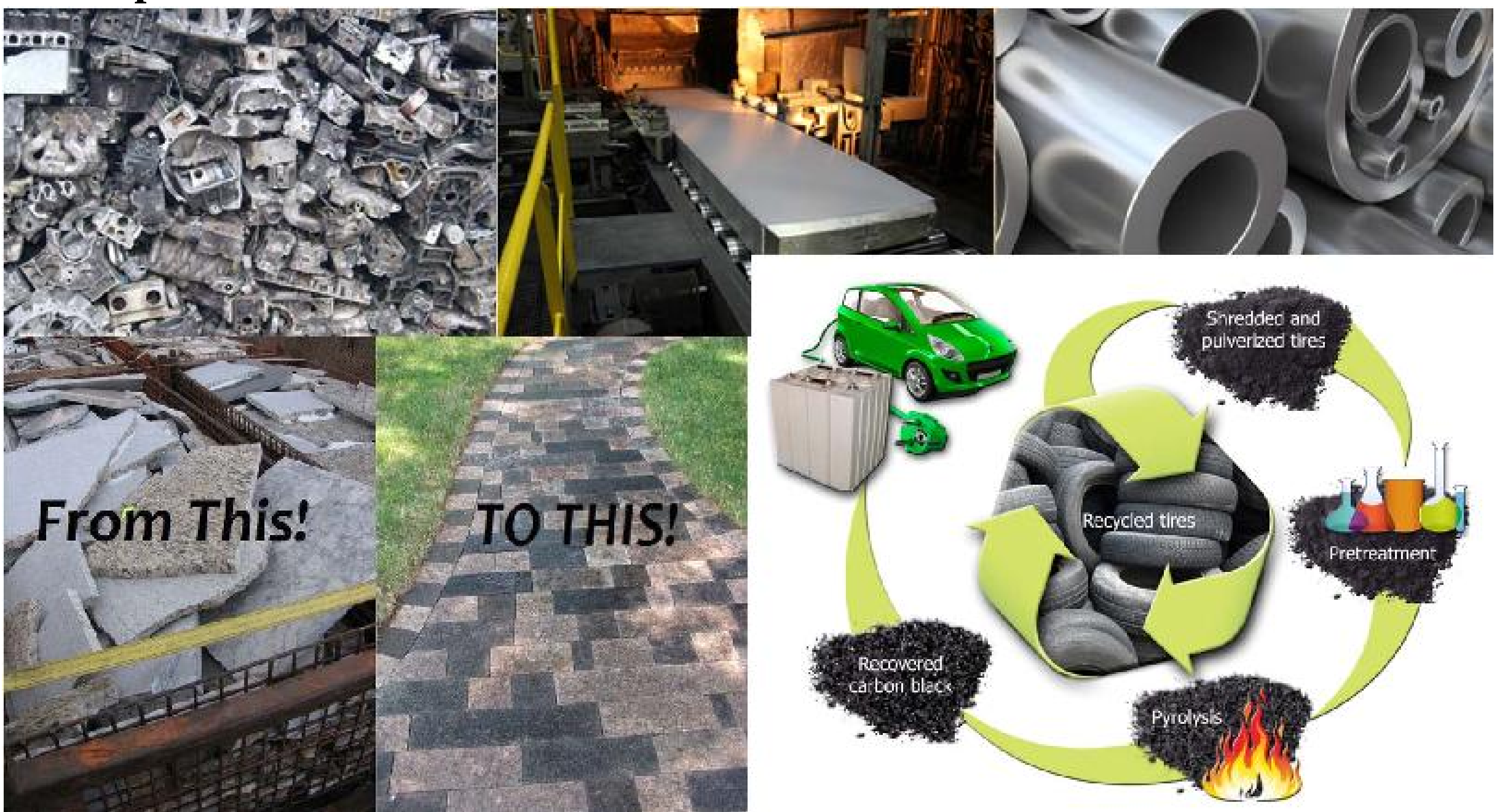
1- Environment:

Humankind has always produced pollution and waste. The industrial revolution saw a major increase in activities producing waste as well as useful manufactured goods. Nowadays, with the world population increased and new awareness of what he is doing to his environment, the need for an efficient ways of handling waste materials has become important.

2- Establishment of need:

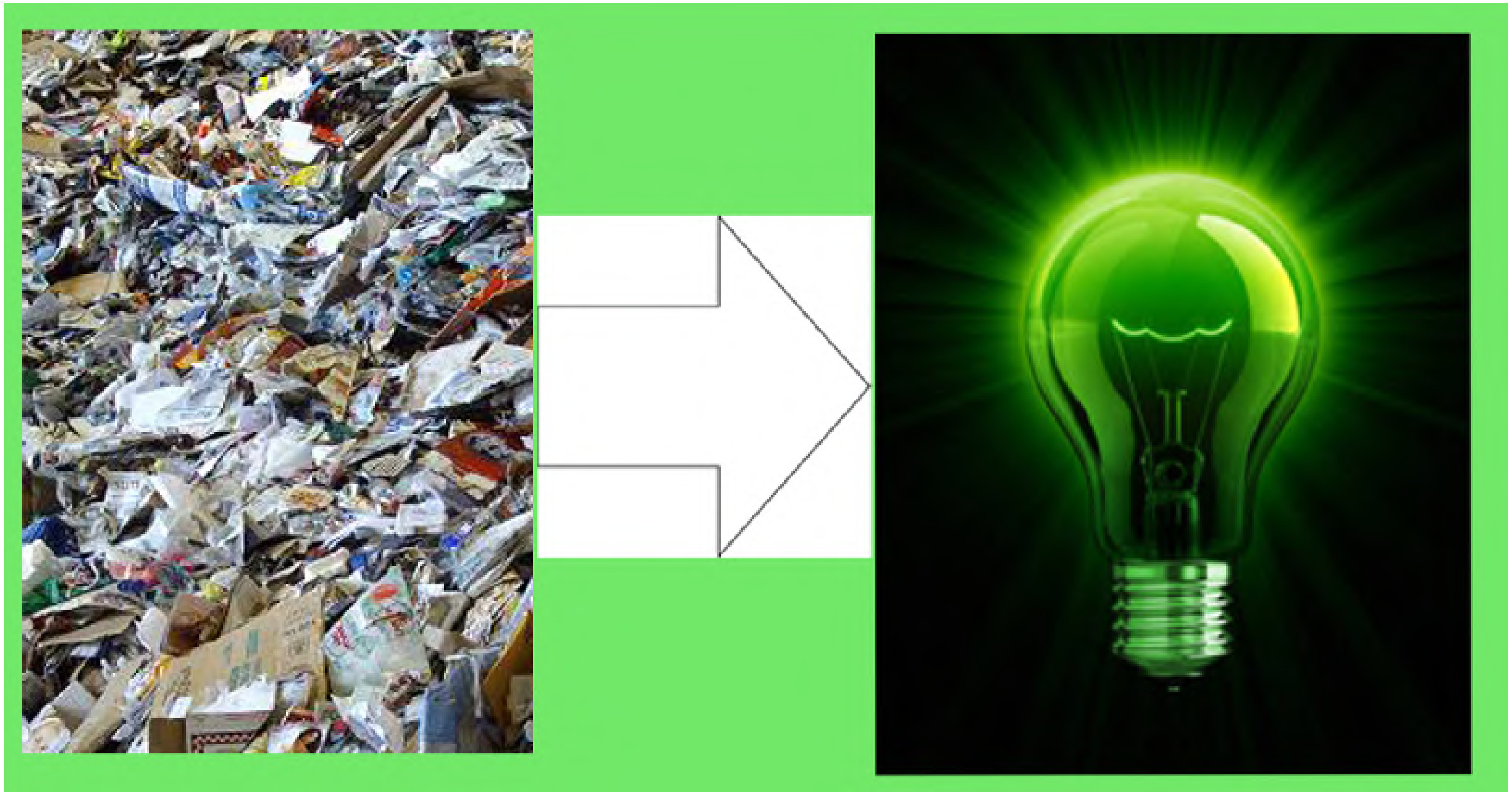
Waste materials may be broadly categorized under the following headings:

- **Waste that can be recycled to its original form, re-melted and reused such as scrap materials.**
- **Waste that can be used as it is to manufacture some other useful items.**
- **Waste that can be converted to something useful in another form such as wood-waste that converged to chipboard.**
- **Waste that can be recycled, reused, or converted and then should be dumped.**

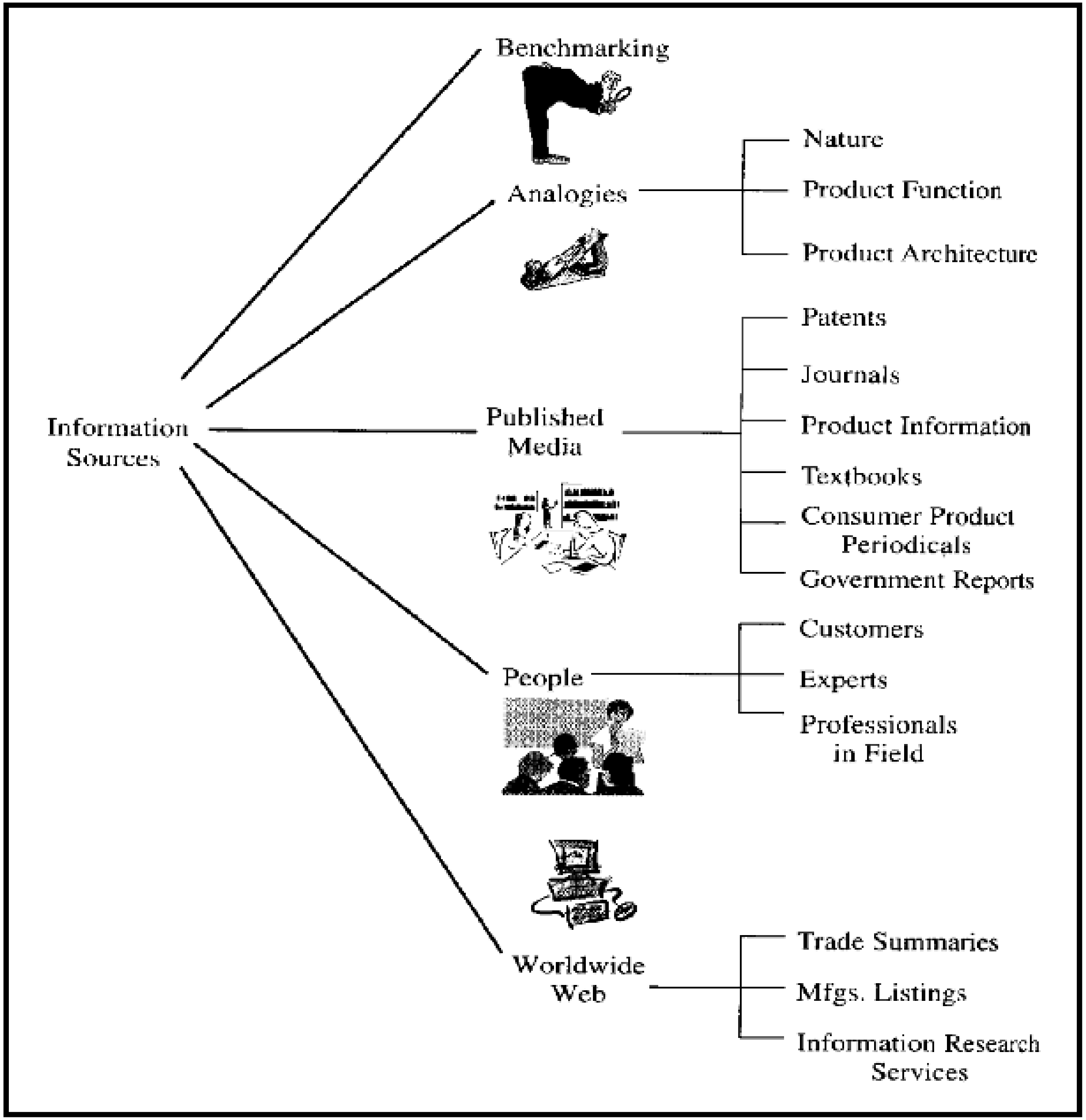


3- Goal Recognition:

The company would be appearing to be in an apposition to manufacture and market equipment, which would convert waste of a certain category in to a form of energy.



4- Market Research:



Writing Questionnaire: (as an application of market research)

Questionnaires are used to collect usable information from the number of large population. They can be represented by two methods:

- **Direct contact (face –to-face) situation; where the researcher can explain the purpose of the study, clarifying points and answering questions that arise. However, bringing a group to full the questionnaire is difficult and takes time.**

- **Mailed questionnaire; where it can reach many peoples in widely scattered areas quickly and easily in the same time and at low cost. However, the return of their answers may be half of the questions that had been sending.**

Questionnaires are in three forms:

Closed form; facilitate process of tabulated and analysis. It consists of a prepared list of questions and a multiple choices of possible answer, to indicates his reply, respondent marks YES or NO, checks, circles, and etc.

Open form.

Pictorial form; this questionnaire presents responders with drawings or photographs rather than writing statement from which to choose answer.

The following is an example of a how you can start writing a questionnaire in our case.

Please tick the box or the boxes that are relevant to your situation:

Q.1) are you:

a) A government manufacturing company ☐
b) A service organization ☐
c) A public company ☐
d) A hospital ☐

Q.2) do you employ

a) Less than 500 person ☐
b) Between 500 to 10000 person ☐
c) More than 10000 person ☐

Q.3) do you produce waste materials

a) Yes ☐
b) No ☐

Q.4) do you

a) Dispose of your own waste ☐
b) Have it collected ☐
c) Both ☐

Q.5) if your waste is collected

a) Does this cost you money ☐
b) Have it collected free of charge ☐
c) Receive a payment for it ☐

Q.6) do you recover any heat from your waste

a) Yes ☐
b) No ☐

Q.7) if the answer of (Q.6) is YES, do you use the heat for any of the following

a) Space heating ☐
b) Hot water service ☐
c) Process requirement ☐

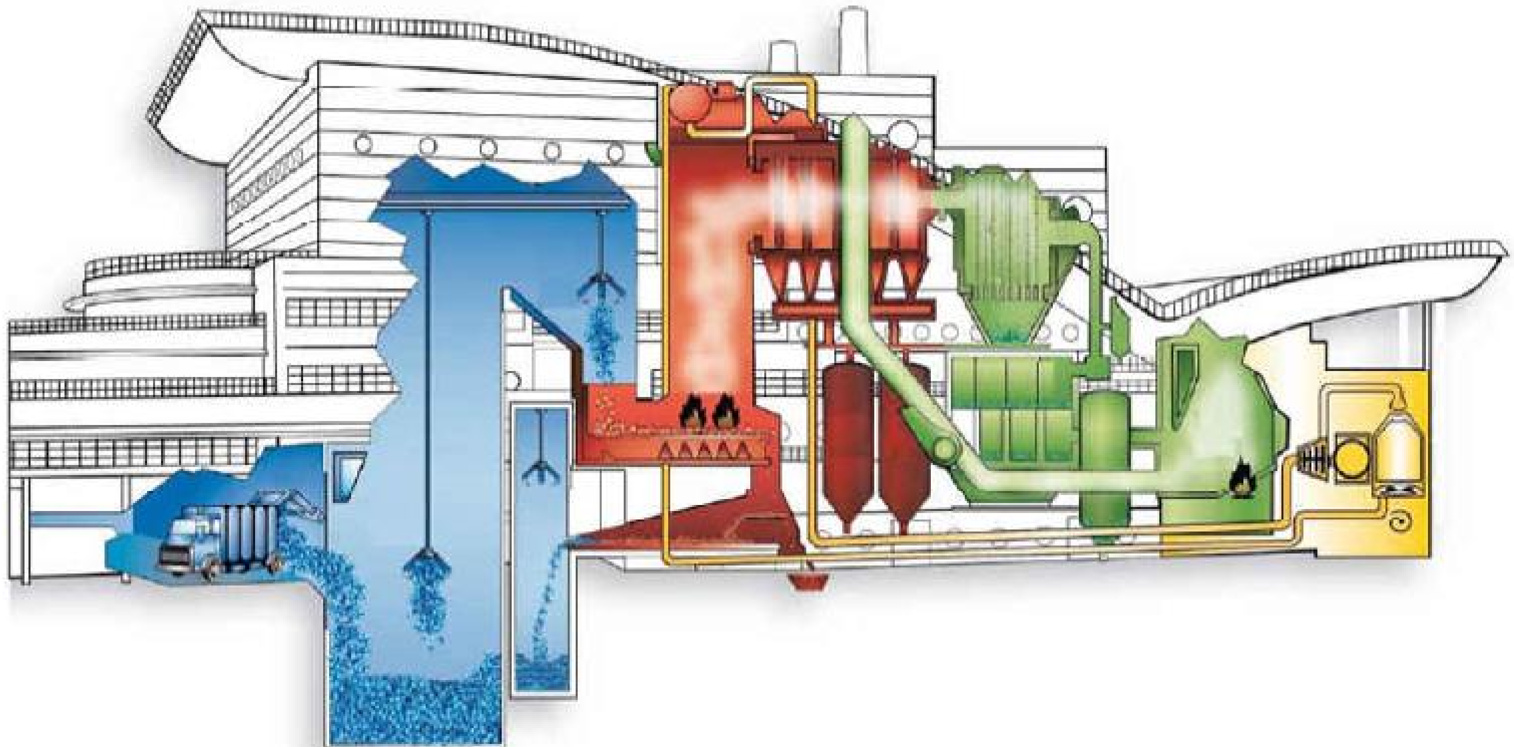
Q.8) has the quantity of the waste materials that you produce

a) Increased in the recent years ☐
b) Decreased in the recent years ☐
c) Remain unchanged ☐

5-Refined Statement:

Based on the returned answers, the system must provide:

- Suitable waste storage
- Automatic handling of waste
- Since there may be an energy requirement when waste is not available, then some alternative means of providing energy must be offered as an extra.
- Some means of converting the waste materials into a readily usable form of energy.
- The plant must provide safety acts and clean air acts.
- Small packaged unit is preferred.



Mechanical Engineering Design II

Third Lecture

Advantages of Questionnaire and how to write an Initial Specification of the System

Advantages of questionnaire:

- Identifying the design decision that is to be influenced by replies to the questionnaire.
- Identifying the kind of information that is critical to the taking of these decisions.

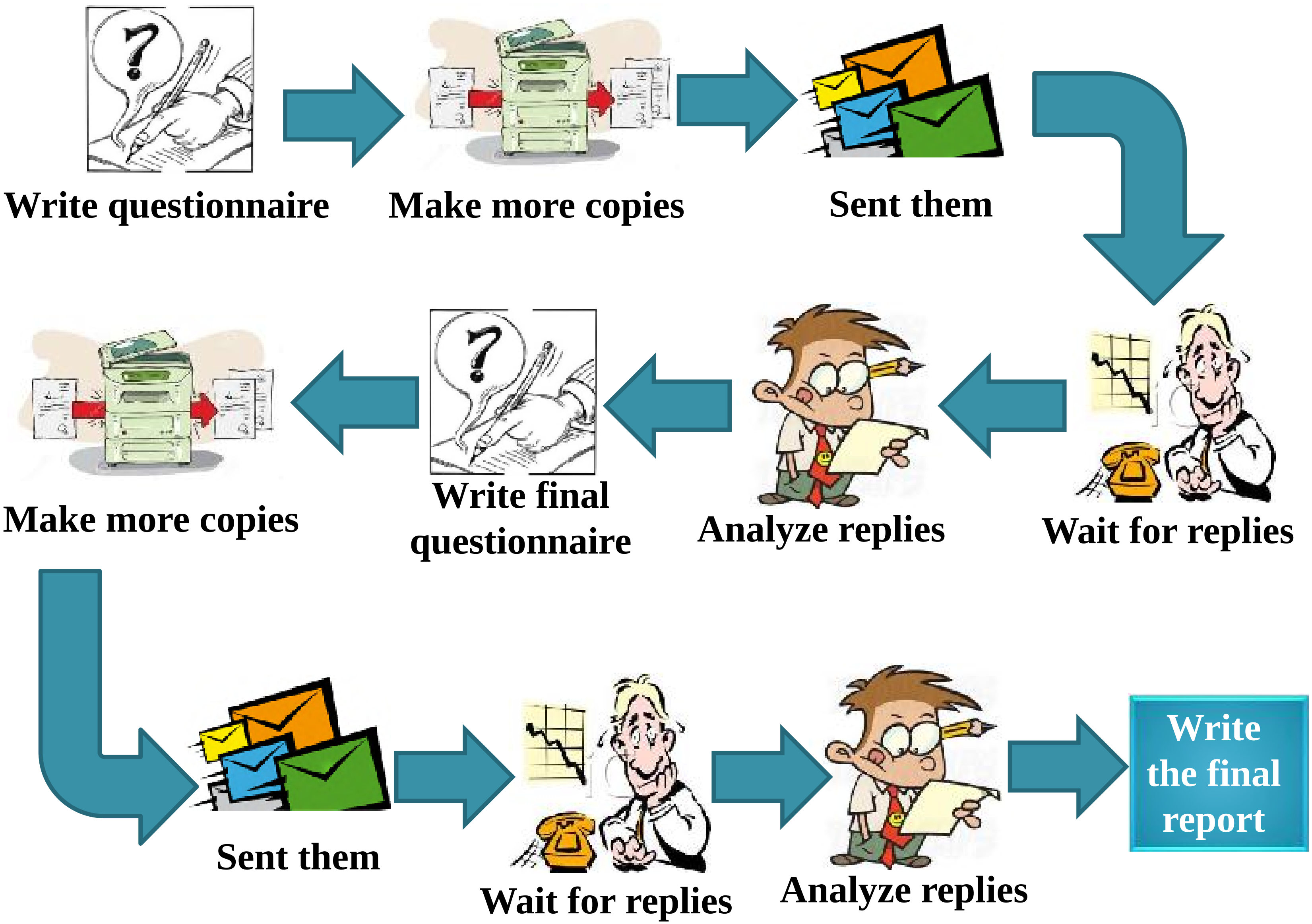
- Identifying the kind of people who having a rapid accept to the kind of information needed.
- The appropriate sample was selected.

- From the replies, taking the most helpful data.

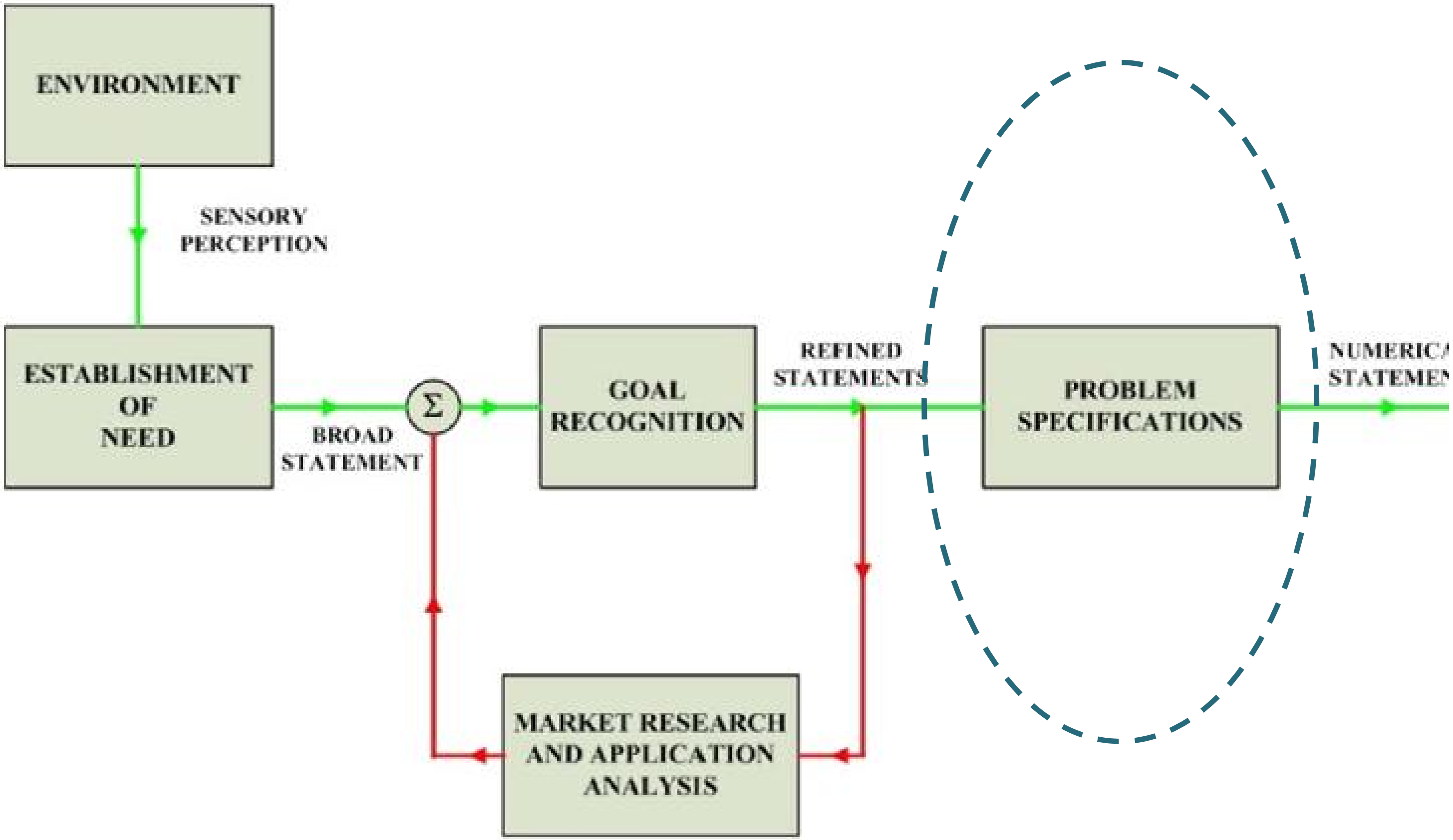
How to write the questionnaire:

1. Ask for minimum information needed for t
   purpose.
2. Be those, which the information is able to
   answer.
3. Require an answer of YES or NO, or a simp
   one, or something equally definite and prec
4. Be such as will be answered truthfully and
   without bias.

Procedure:

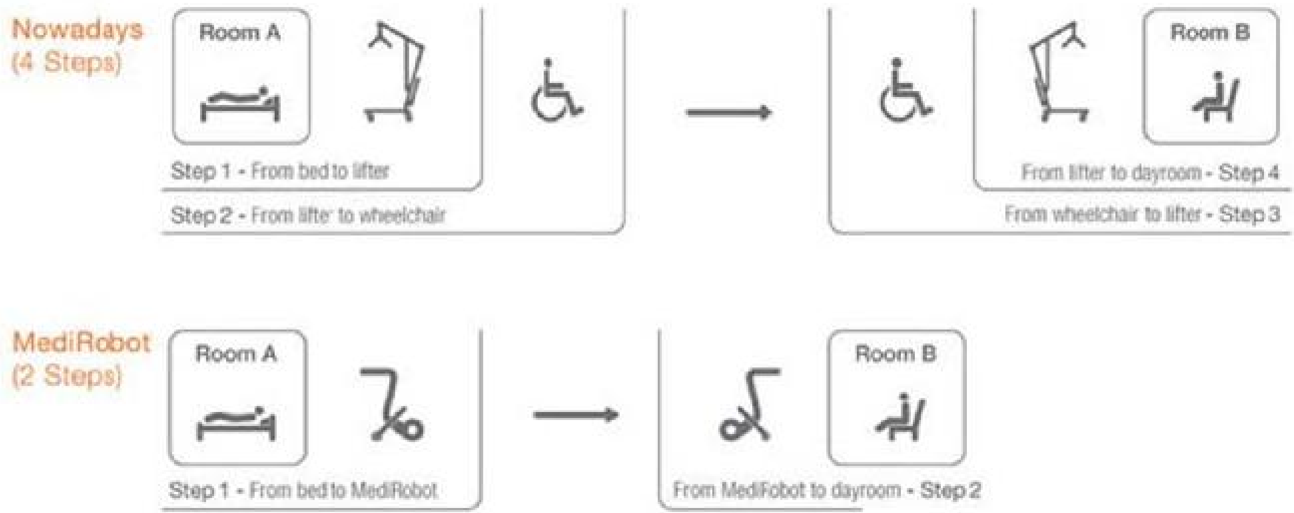

Write questionnaire
Make more copies
Sent them
Wait for replies
Analyze replies
Write final questionnaire
Make more copies
Sent them
Wait for replies
Analyze replies
Write the final report

Problem Specifications (Initial Specification):



List of items which may be required in Specifications:

1. Title of specifications.
2. Forward or introduction.
3. The role of equipment or material.



4. Related documents and references.

5. **Condition in which the item is to be used, manufactured or stored**

- **Environmental features including for example temperature, humidity, pressure, shock, vibration, noise, dust, etc.**
- **Condition of use, power requirements, supply services,**
- **Servicing requirements.**

6. **Characteristics:**

- **Samples, drawings, models, tests, etc**
- **Properties such as strength, dimensions, weight, safety, degree of purity.**
- **Interchange ability.**
- **Appearance, finish, color, protection.**

7. **Performance:**

- **Performance under specified conditions.**
- **Test method and equipment for assessing performance.**

8. **Life:**

- **Period of useful life.**
- **Total life.**
- **The method and equipment for assessing life.**

9. **Reliability and control.**

10. **Control of quality checking.**

Mechanical Engineering Design II

Fourth Lecture

Examples of Design (Initial) Specifications

Part I

Ex.1: Design a power unit for a lorry:

Design specification for a power unit:

This specification is for the design of a power unit which may be used as a primary power unit in a lorry and for other purposes of similar power size. The power unit is to use normal road fuels and to have a rotating shaft output.

Introduction:

This specification is for the design of a power unit of approximately 250 hp. The power unit is intended to be marketed as a basic power source for a variety of uses, the main use is as a lorry power unit. The design is to be in the form of a gas turbine.

The power unit is to be designed for quantity production.

Markets:

The possible markets for the power unit are as follows:-

- Lorries
- Small aircraft
- Helicopters
- Hovercraft & Marine uses
- Small electrical generators
- Aircraft servicing trucks
- Pump units & compressor units
- Cranes

Related Specifications:

It is suggested that gearboxes should be used as speed reducers to change the turbine shaft speed to the output shaft speed which is quoted later in this specification.

Performance:

Consideration is to be made for uprating to 300 hp for short duration runs.

Weight:

The weight of the unit excluding the gearbox is to be less than 100 lb.

Instrumentation:

Modern instrumentation is to be incorporated to measure the jet-pipe temperature and the other relevant temperatures. Electrical pules tachometers are to be used to measure speeds.

Failsafe Devices:

Failsafe devices are to be supplied with the unit so that malfunctioning causes the power unit to shut down. Correction of the fault should then enable immediate restart.

Starter:

A conventional vehicle engine starter is to be used for starting purposes.

Operation:

The power unit should be capable of being used for long periods without maintenance.

Fuel:

The unit must be able to be run using conventional vehicle fuels, aircraft fuels and natural gas.

Multiple Unites:

Consideration is to be given to the possibility of using a standard, quantity production power unit on a special gearbox to give power units up to 1000hp limit.

Maintenance & Spares:

The unit is to be designed so that the assembly of critical high speed parts is carried out on the manufacturers premises. Assemblies requiring less skilled fitting work are to be supplied as spares. The spares are to be designed so that the value/weight ratio is high so as to make air freight a reasonable proposition for spares transportation.

Noise Level:

The noise level is to be equal to or lower than that from other gas turbines of comparable size.

Transportation:

Facilities are to be provided for easy transportation of the unit.

Price:

The selling price is to be less than ($) per horsepower output (continuous rating), including the gearbox.

Life:

The overhaul life of the unit shall be 500 hrs or greater.

Overall Size:

The unit must fit inside a rectangular box 15in x 20in x 40in.

Measure of Value for Some Items:

Priority shall be given to various parts of this specification by utilizing the measures of value given below:

Performance	20	Noise Level	10
Failsafe Devices	15	Overall Size	10
Price	15	Diversity of Market	5
Life	12	Instrumentation	5
Maintenance & Spares	8		

Mechanical Engineering Design II

Fifth Lecture

Examples of Design (Initial) Specifications

Part II

Ex.2: Design of Household Carpet Cleaner:

Environment:

Since the cleaner will be housed and used indoors weather protection is not required but the finish should be non-corrosive. Although it is expected that the carpet area being cleaned should not emit fumes or noise that will cause a nuisance to other occupants of the building.

Normal vacuum cleaning will have been carried out prior to the rinsing process and it may be assumed that the carpet is satisfactorily fixed to the floor.

Supply Services:

240 V, 13 A, 50 Hz electrical power outlets will normally be available.

Hot or cold water may well be available from taps in the vicinity of the cleaning operations but provision will need to be made for instances when this is not the case.

Performance:

- **The cleaner, when set up and operated by one person, should be capable of cleaning a carpet area of 140 m^2 in 60 minutes.**
- **The degree of soil removal achieved must be substantially better than that obtained by shampooing.**
- **The cleaning operation should include soil and fluid extraction to the extent that in normal room temperature and humidity the carpet may be reusable within two hours of completion of cleaning.**
- **The width of the cleaning path should be of the order of 0.5 m.**
- **Areas of carpet under fixtures such as radiators should be accessible to the cleaner.**
- **A special feature of this machine will be powered driving wheels/roller to reduce operator fatigue. Power will be available for both forward and reverse motions.**
- **Provision is to be made to enable carpets with different types and length of pile to be cleaned affectively.**

Control:

- **Variable speed control of the powered wheels/roller will be provided for both directions of motion.**
- **Independent control of rinsing fluid application rate shall be possible.**
- **In the event of the use of a chemical agent to cope effectively with badly soiled carpets a setting must be provided to control the amount of agent mixed with the rinsing fluid.**
- **Provision shall be made for heating and temperature control of the rinsing fluid to ensure it is applied at the desired temperature.**
- **Although each control should be considered ergonomically to ensure easy operation it should be remembered that this equipment is intended for professional cleaners.**

enable immediate restart.

Construction:

- **The cleaner should be of robust construction to suit both its usage and handling.**
- **Any component part which is likely to require removal or replacement during the normal working life of the machine should be easily accessible.**

Maintenance and Reliability:

Reliability must be regarded as an important feature of the design. It is expected that servicing will be carried out twice a year by Columbus Dixon Service Engineers.

Size and Weight:

Although no specific limit is imposed on either size or weight of the cleaner it should be as compact and light as is reasonably possible . Since the mobility is to be a special feature of the machine.

Appearance and Finish:

The equipment is to be styled in such a way as to make it attractive to the potential customer. Adequate protection from corrosion shall be given and external surfaces shall be finished in company colors where appropriate.

Safety:

The associated standards referred to earlier must be complied with in order to ensure safe operation of the equipment. Any moving parts that could provide a hazard shall be adequately covered.

Life:

In keeping with company policy the useful life of this product shall be a minimum of ten years, subject to reasonable use.

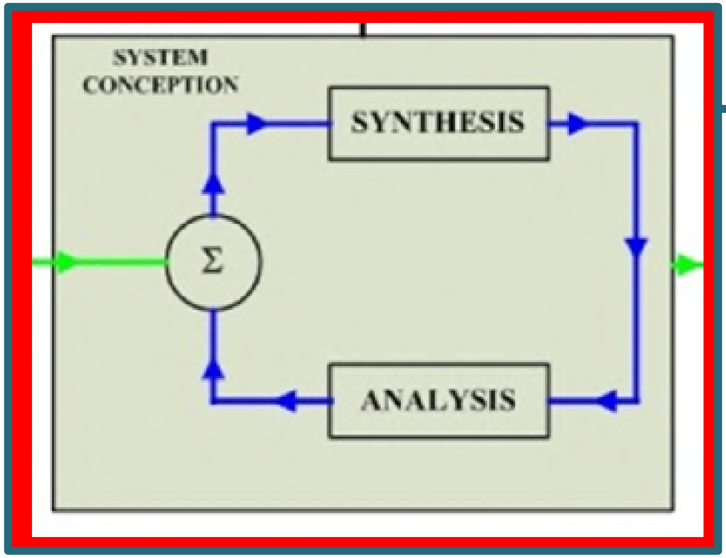
Price:

It is intended to market this product in either September 2005 or September 2006 , dependent on the development work necessary, at a selling price not exceeding ($) at present day values.

Mechanical Engineering Design II

Sixth, Seventh & Eighth Lectures

System Conception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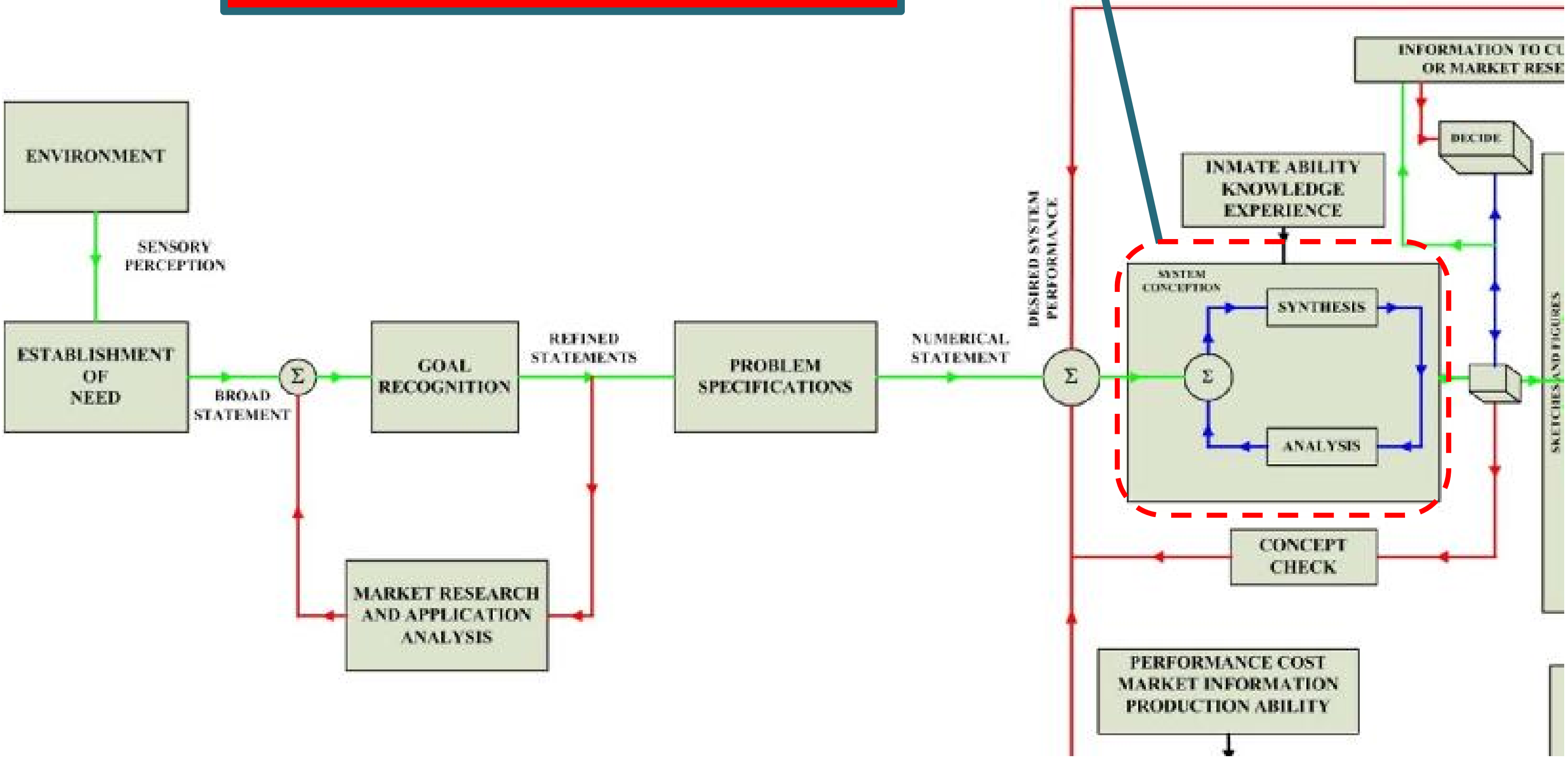
ANALYSIS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PROBLEM
SPECIFICATIONS
NUMERICAL
STATEMENT
MARKET RESEARCH
AND APPLICATION
ANALYSIS
DESIRED SYSTEM
PERFORMANCE
INMATE ABILITY
KNOWLEDGE
EXPERIENCE
SYSTEM
CONCEPTION
SYNTHESIS
ANALYSIS
CONCEPT
CHECK
PERFORMANCE COST
MARKET INFORMATION
PRODUCTION ABILITY
DECIDE

Synthesis and Analysis:

The processes of synthesis and analysis appear repeatedly in the System Engineering Flow Chart. For this reason Synthesis will be considered in much greater detail. Analysis is the subject of study in most undergraduate courses and will, therefore, not require such a detailed study here.

Synthesis

The process of synthesis in design is the bringing together of information which, when taken in combination, satisfies an initial requirement. The process of checking that the outcome satisfies the initial requirements is the analysis process. It has been mentioned previously that systematic methods can assist in the processes of design, the use of methods in Synthesis is by no means an exception. However, at this point in our knowledge of the design process. We, as lecturers, consider that it is best to know of a number of methods to improve synthesis, the application of any one or more of these methods in any particular circumstance being left to the discretion of the designer. Systematic methods of synthesis are termed Heuristics.

Creativity:

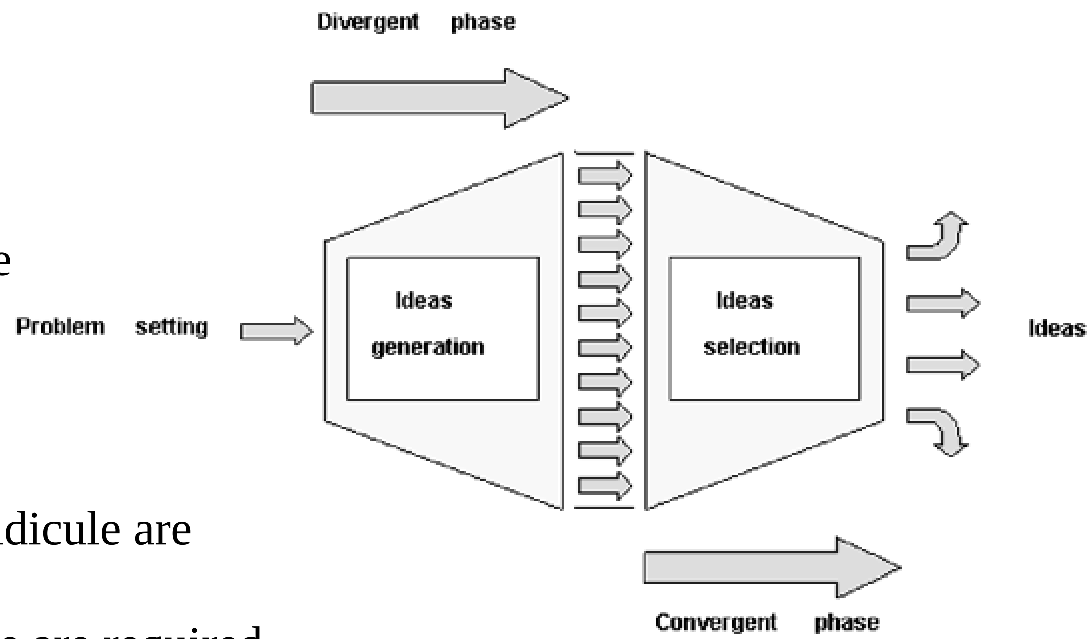
- **Creativity, synthesis and applied imagination are all terms used to denote the production of alternative solutions to problems.**
- **However, for design purposes, we must add the additional constraint that the alternative solutions must eventually lead to a useful conclusion which satisfies initial requirements.**
- **The starting point in the creative process should always be to verify the facts which initiate the creative process. Thus the goal towards which we are heading becomes clearly defined, this reduces time wastage.**
- **An aid to this verification process is to attempt to rewrite the problem in different words, carefully analyzing and verifying each point in turn. Features which need clarification or additional information should be noted on one side, but since we are eager to produce something, let us temporarily ignore these sideline points if this possible. These sideline points can be attended to and included in the synthesis at suitable times.**

- Now, if we stop the shaking after the production of some short chains we can take these chains from the box and use various patterns of combination to join them together. We could, say, lay them onto a table to form letters, then these letters could be combined to form words, and so on. Note that we would be using rules from subjects other than those which could be devised for the ‘art’ of paperclip joining.
- In design synthesis we can use rules from mathematics, art, business management, language; in fact anything that will help.

Heuristics

Brainstorming

The brainstorming technique was created by Alex Osborn in 1938.

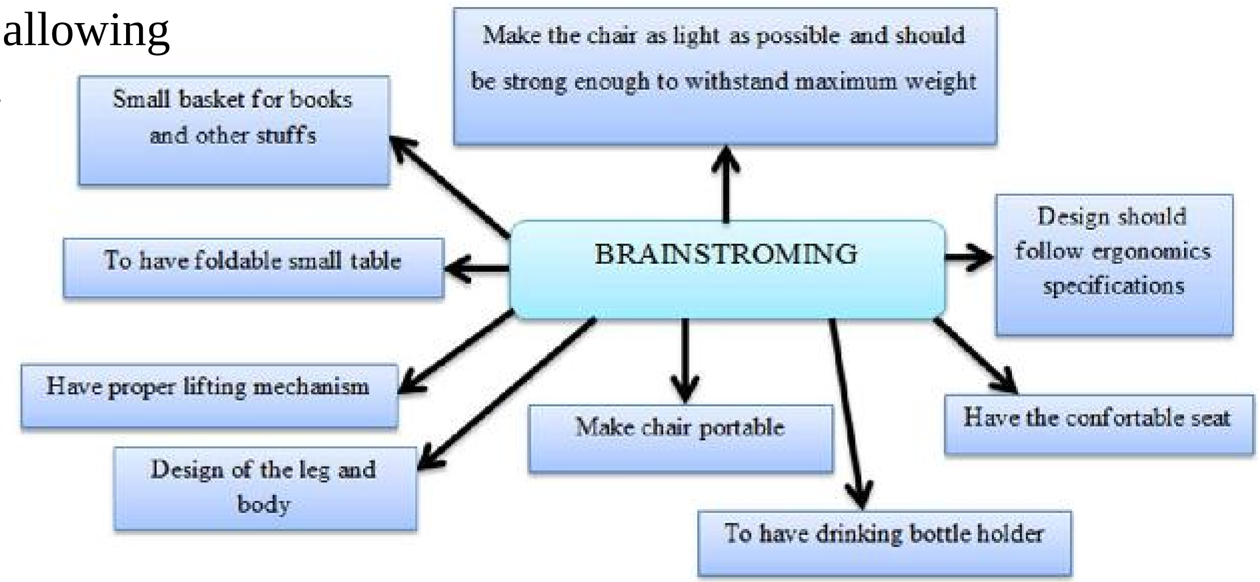


Rules of Brainstorming:

1-Criticism, judgment and ridicule are eliminated completely.

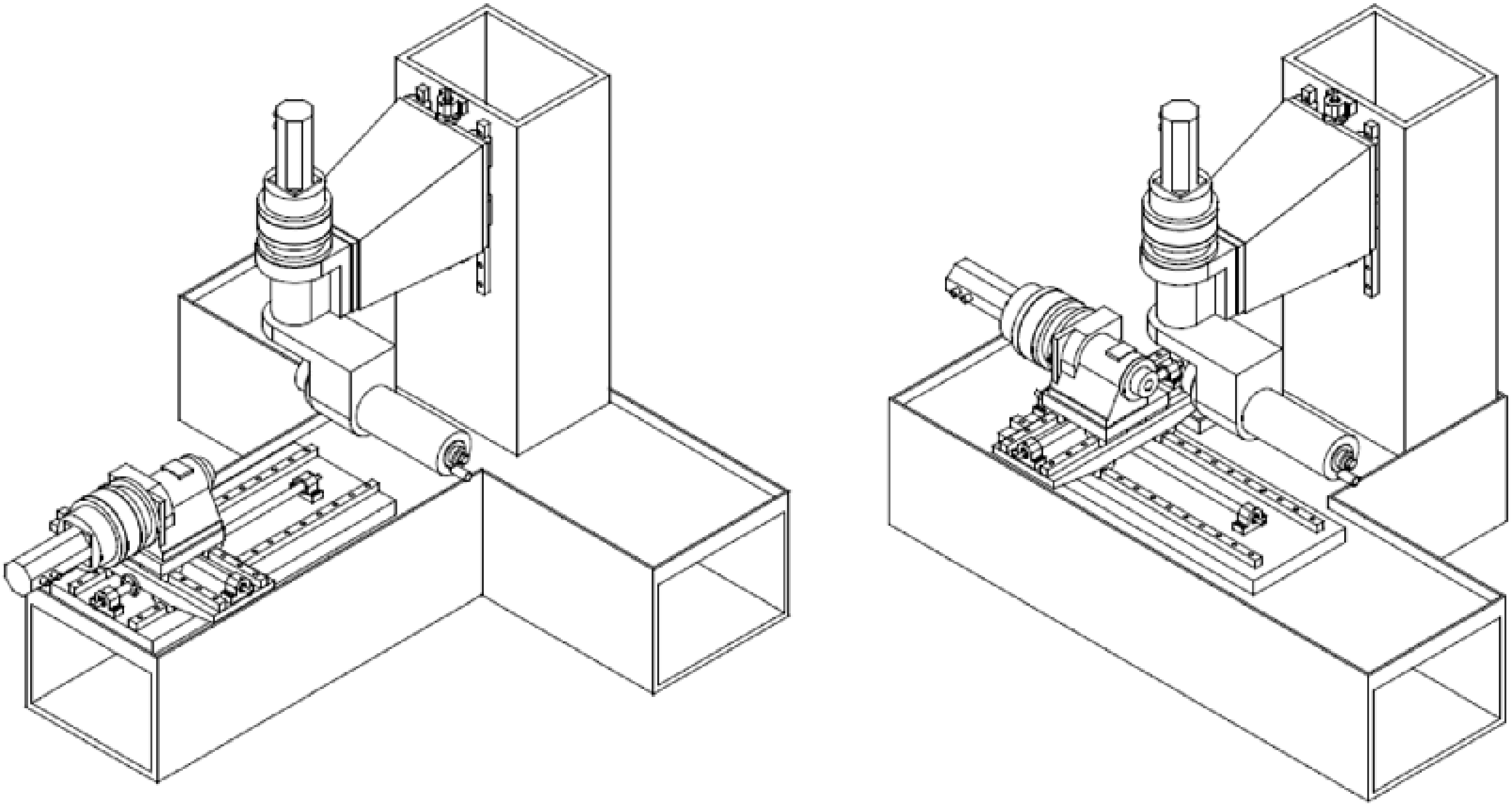
2- Copious ideas, of any type are required.

3- Think wild, don't allow criticism in your own mind before allowing the ideas to erupt.



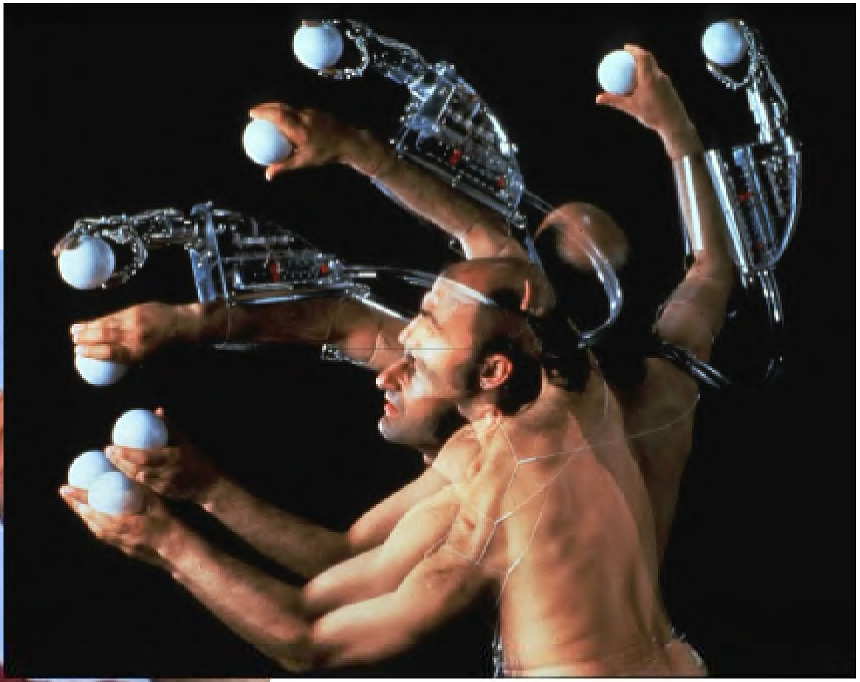
Inversion

Turn things inside-out, upside down, stop moving parts, start stationary parts, …



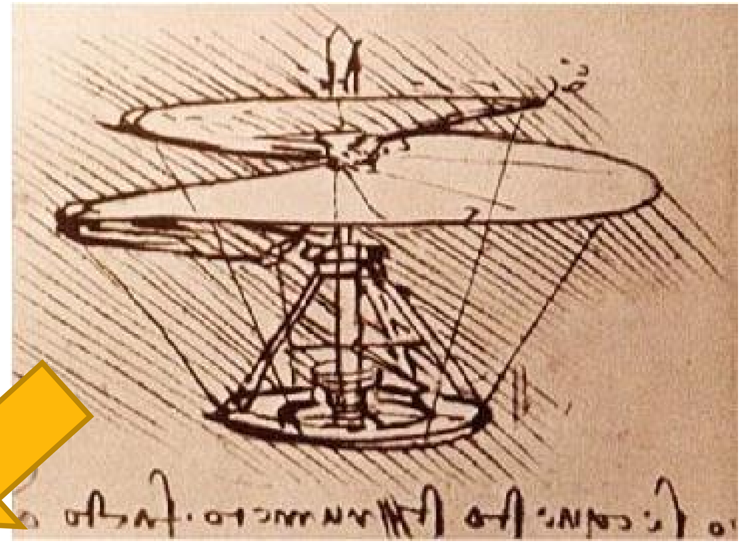
Empathy

- Putting one's self in another's place.
- Identifying physically and personally with the part, product, or process that is to be created:
  - Body/mind must actually perform the function(s).
  - State how it feels and what we would need or do if we were to do the task.



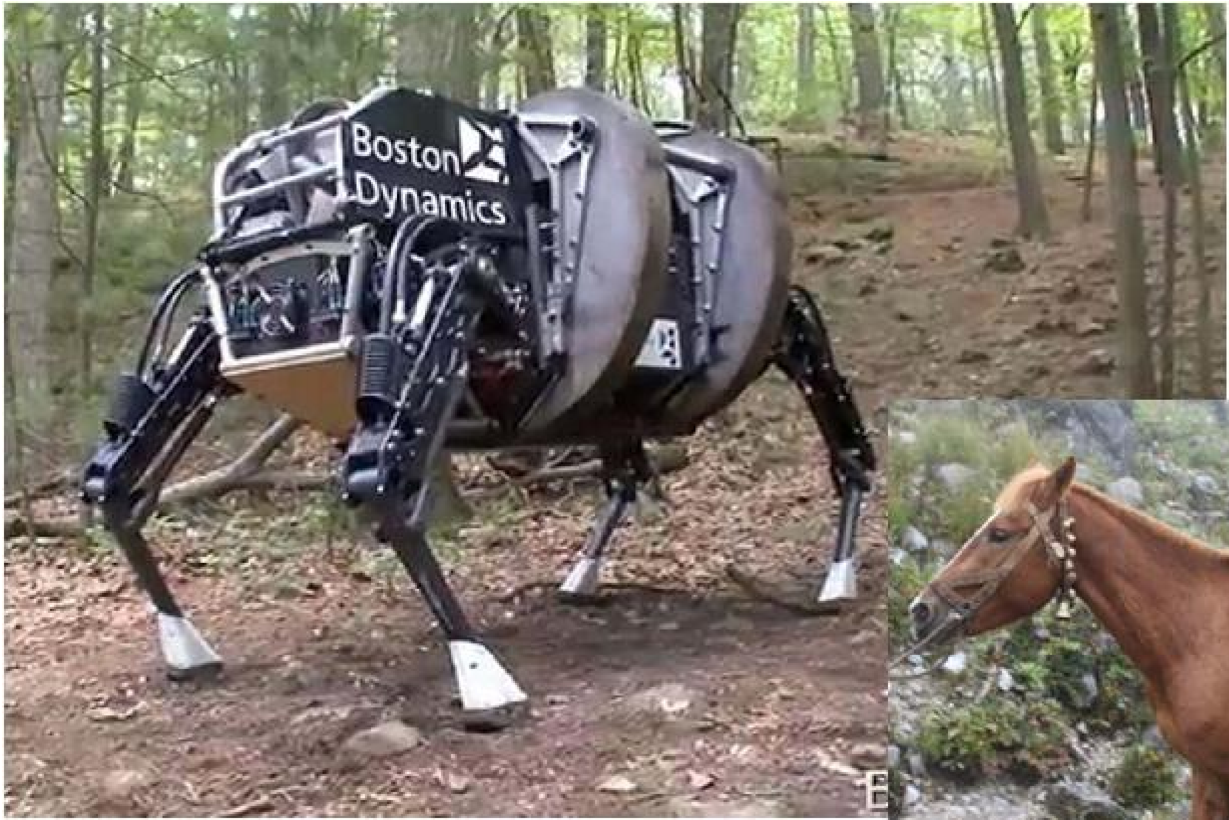
Fantasy

- Imagining or wishing that something is possible
- Sometimes impossible, but more importantly – sometimes possible and sometimes the ideas can be modified to be possible



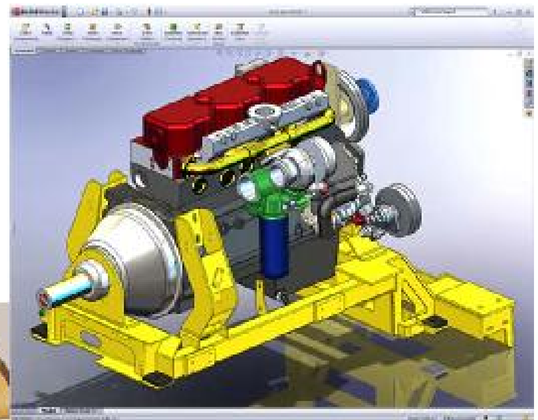
Analogy

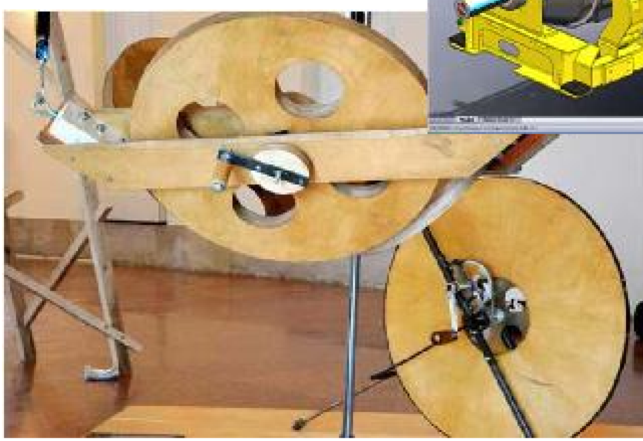
- Analogy to other physical phenomena.
- Think about the problem in general enough terms so that the characteristics that it has with other disciplines/situations become apparent.
- Remember to think about the natural world.

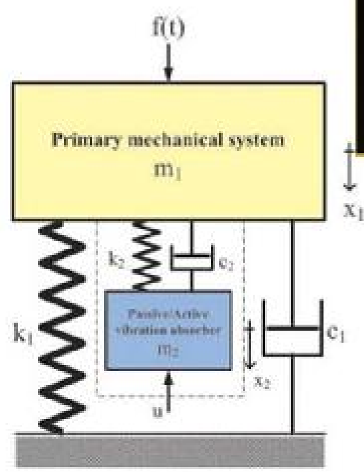


Models

- The use of model can refer to material models, which is the interpretation of the layman. The material model can be iconic model, that is model with only a change in scale.
- Other material models can have changes in material.
- Analogue models are of the type where one property is used to represent a different kind of property.
- Symbolic models cover the mathematical models.
- Digital models are a special type of model for use on digital computer.
- Graphic models such as drawings, sketches, etc.

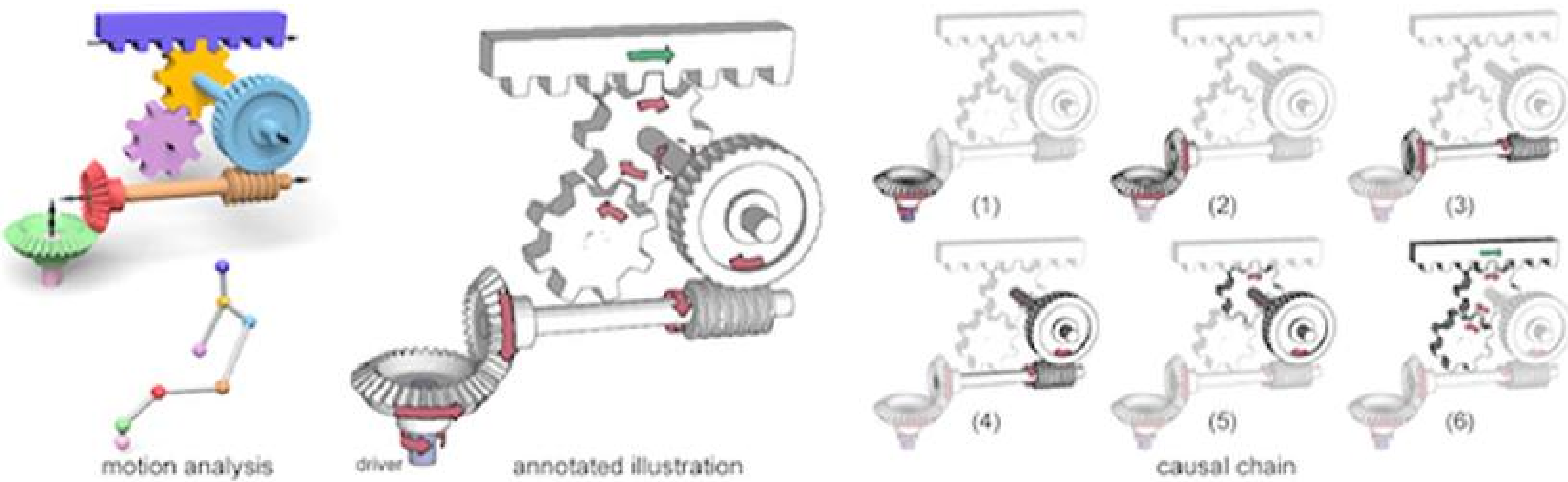






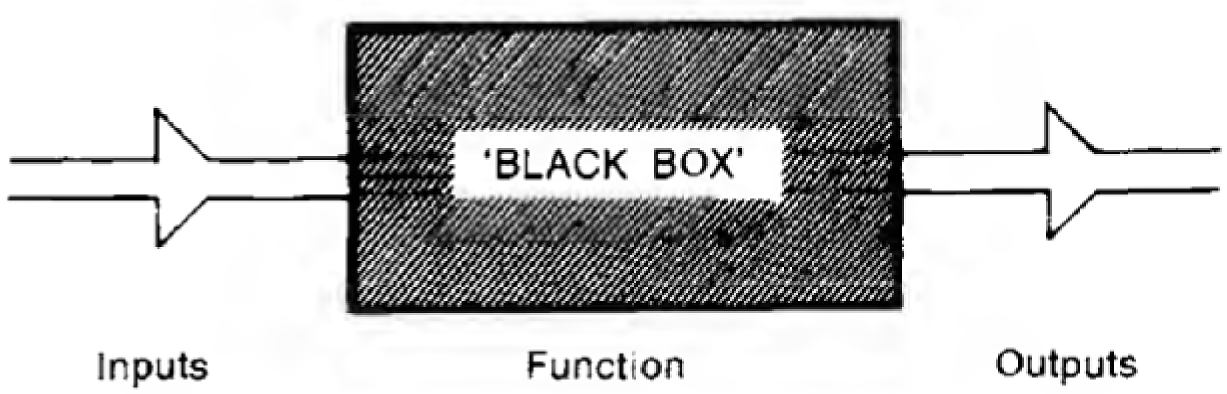
Visualization

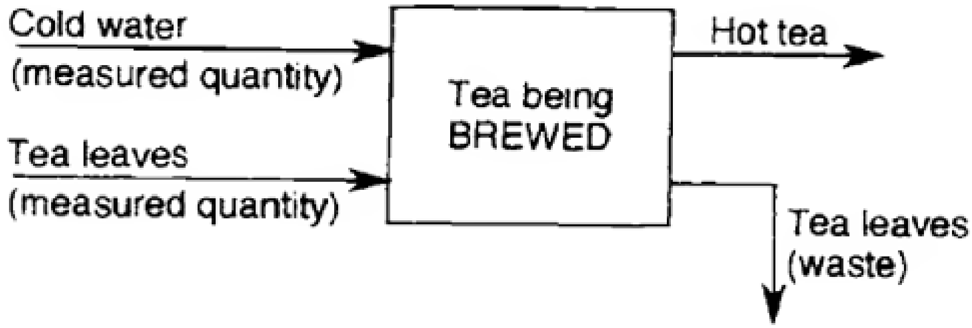
This is the use of certain types of model to enable those carrying out the synthesis process to be able to “see in their minds” a number of facts simultaneously.



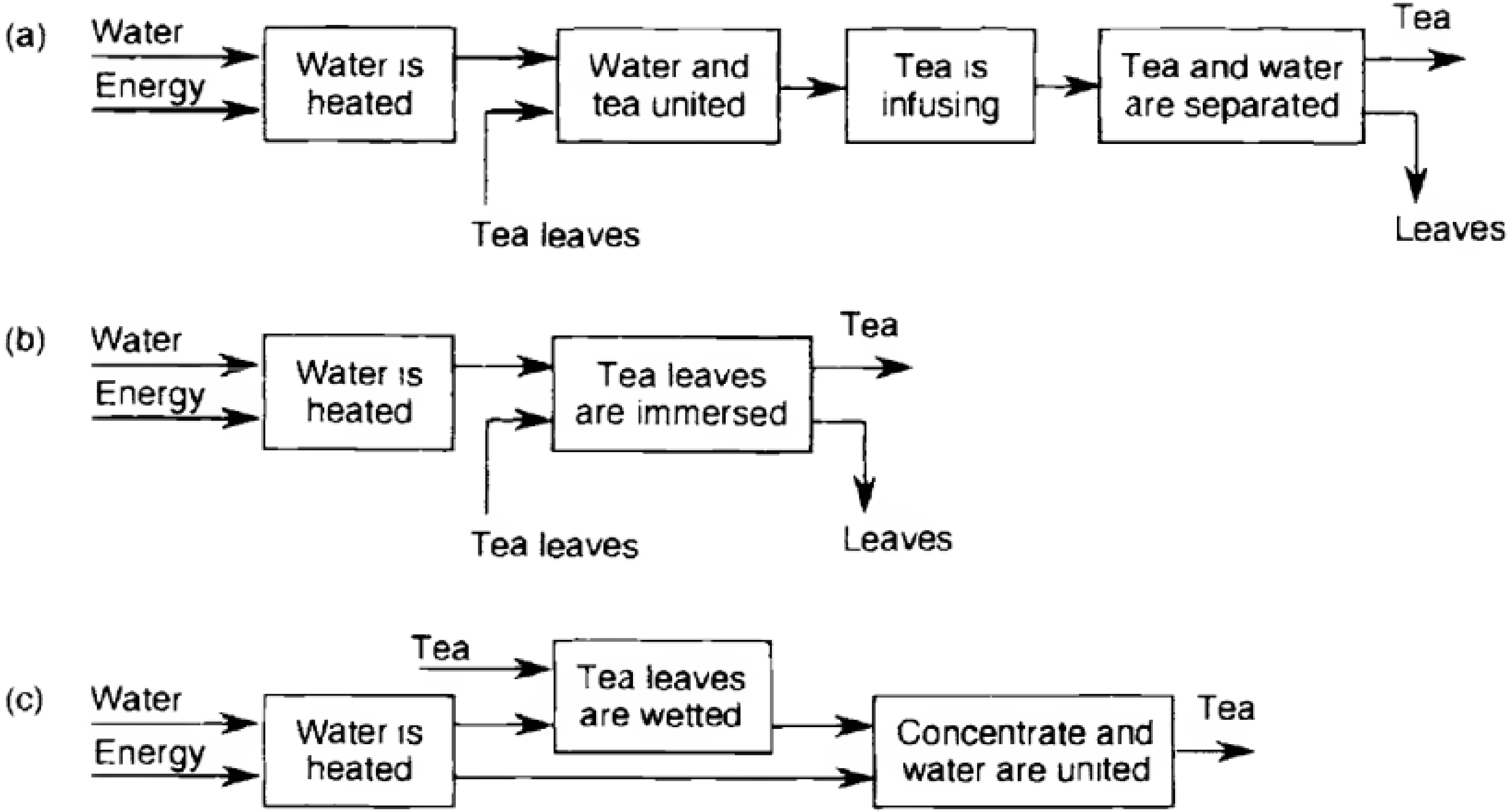
Black box concept

- This consist of a system of great complexity (in a relative manner) whose internals are impossible to characterize completely. Alternatively, there may be no desire to characterize the internals completely at a certain point in the design process.
- The main features about the black-box can be learned from observing the inputs and outputs. The output can then be expressed in a number of cases as some function of the input, i.e. the transfer function of the black box is equal to the ratio of the output to the input.
- In general there are three kinds of problem relating to the black-box. If any two of the features: input, output and transfer function are known then the third feature can be found.





Three alternative process models for tea brewing



Morphological Analysis

- This can be described in general terms as:- The analysis or those features that may be required in the system, and how alternatives of these features way be combined.
- This is a well-established creative process originated in one form by Zwicky. The process is to initially describe the problem in the broadest possible manner so that the type of solution is in no way defined.
- Next, the various groups of features are listed so that all the alternatives of each feature appear together, combinations of these alternatives of various features which can be combined in a large number of ways to trigger off the actual creative process which results in a solution.
- The method can be extended by using a number of cube boxes in which one axis is common to all the boxes, the other two axis being different on all boxes and totaling up to give the required number of features.
- An alternative method is to put the features as columns of alternatives with liberal spacing between columns.
- The next stage is to select combinations of alternatives from the massive array of possibilities, either by labelling compartments of the cubes, or by joining alternatives of the columns of features.
- These ideas are then evaluated by subjecting each idea to a previously prepared “specification” or comparison “check-list”.

A general morphological analysis table

Subsystem	Means			
1	Method 1 of fulfilling subsystem 1	Method 2 of fulfilling subsystem 1	Method 3 of fulfilling subsystem 1	Method *n* of fulfilling subsystem 1
2	Method 1 of fulfilling subsystem 2	Method 2 of fulfilling subsystem 2	Method 3 of fulfilling subsystem 2	Method *n* of fulfilling subsystem 2
3	Method 1 of fulfilling subsystem 3	Method 2 of fulfilling subsystem 3	Method 3 of fulfilling subsystem 3	Method *n* of fulfilling subsystem 3
4	Method 1 of fulfilling subsystem 4	Method 2 of fulfilling subsystem 4	Method 3 of fulfilling subsystem 4	Method *n* of fulfilling subsystem 4
5	Method 1 of fulfilling subsystem 5	Method 2 of fulfilling subsystem 5	Method 3 of fulfilling subsystem 5	Method *n* of fulfilling subsystem 5

Morphological chart for a pallet moving device with choices identified

Feature	Means				
Support	Track	**Wheels**	Air cushion	Slides	Pedipulators
Propulsion	**Driven wheels**	Air thrust	Moving cable	Linear induction	
Power	Electric	Diesel	**Petrol**	Bottled gas	Steam
Transmission	Belts	Chains	**Gears and shafts**	Hydraulics	Flexible cable
Steering	**Turning wheels**	Air thrust	Rails	Magnetism	
Stopping	**Brakes**	Reverse thrust	Ratchet	Magnetism	Anchor
Lifting	Hydraulic ram	Rack and pinion	Screw	Chain or rope hoist	**Linkage**
Operator	Standing	Walking	Seated at front	**Seated at rear**	Remote control

Design Trees

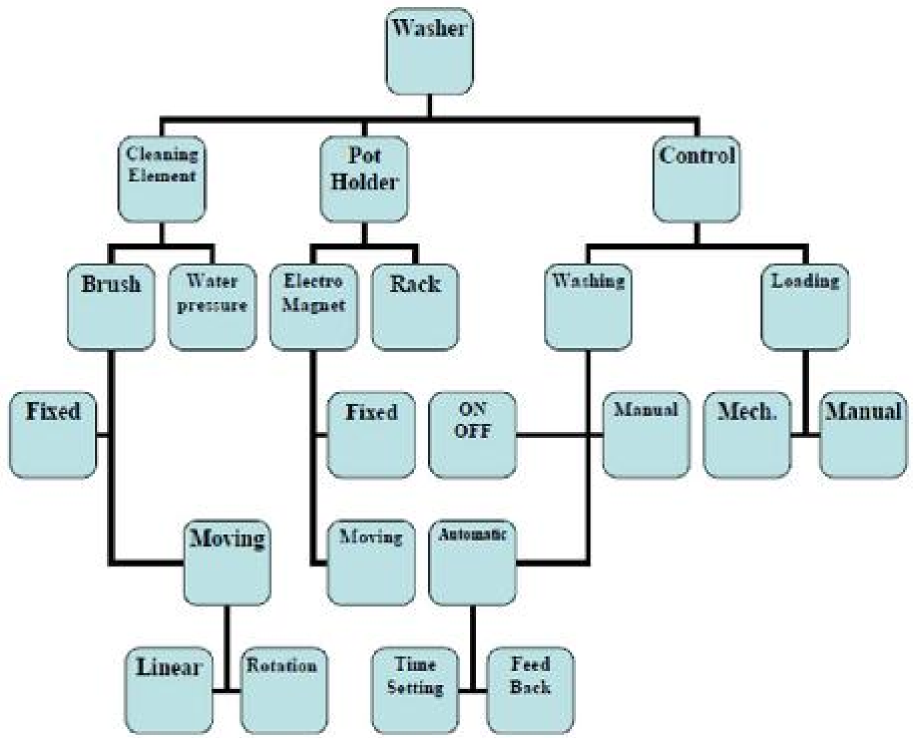
1. This is a generalized model of the design process in the form of a tree.
2. Any design can be regarded as the outcome of a sequence of problems and solutions.

The rules of results from path trees are:

1. When numbers of alternative solutions to a single problem are presented, any one may be accepted and the rest ignored.
2. All the problems dependent on the choice of a particular alternative solution must, however, be solved.
3. A particular branch of the tree must be followed until a solution is reached.

Example:

This is a Design of a cleaning device for pots and pans.



Useful Mechanisms:

Snap-action mechanisms

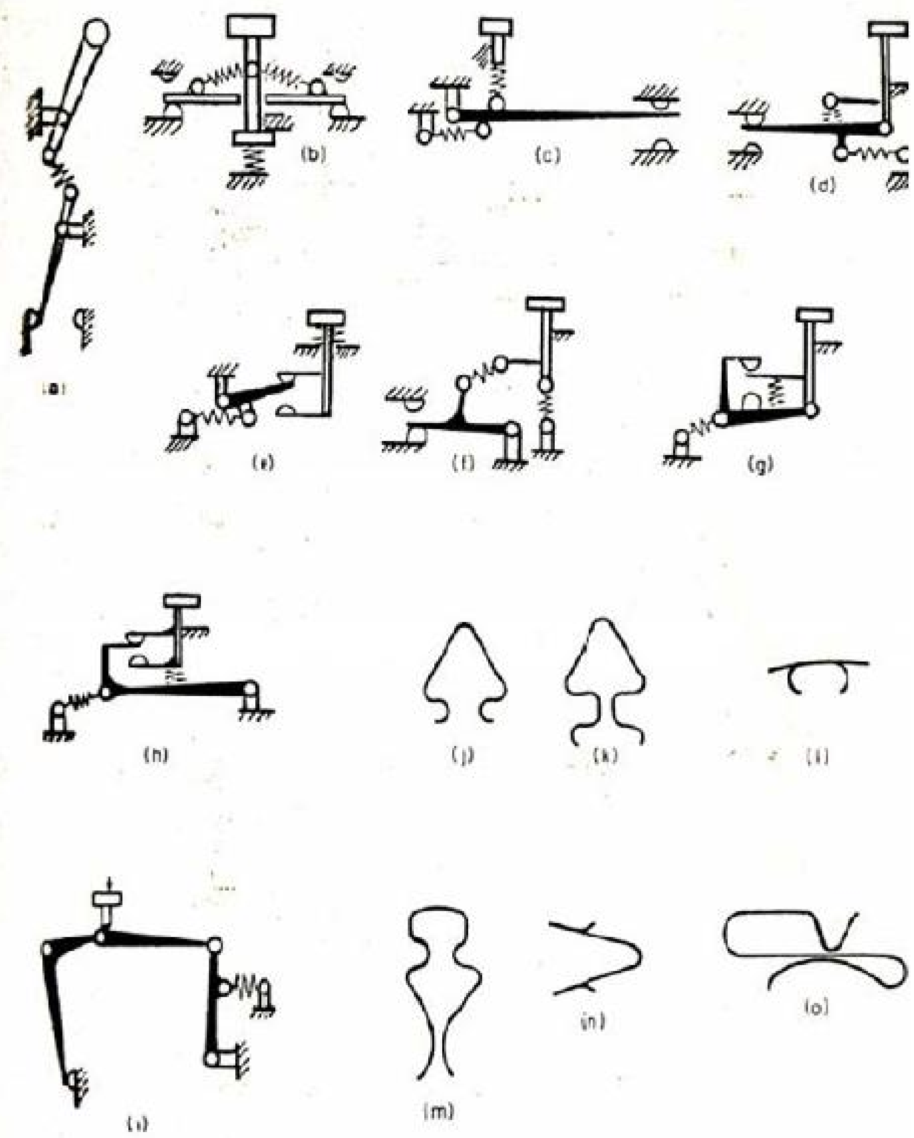


FIG. 39-1 Snap-action mechanisms. These mechanisms are bistable elements in machi… They are used in switches to quickly make and break electric circuits and for fastening ite… (a) Snap-action toggle switch; (b) to (h) seven variations of snap-action switches; (i) cir… breaker; (j) to (o), spring clips.

Robots mechanisms

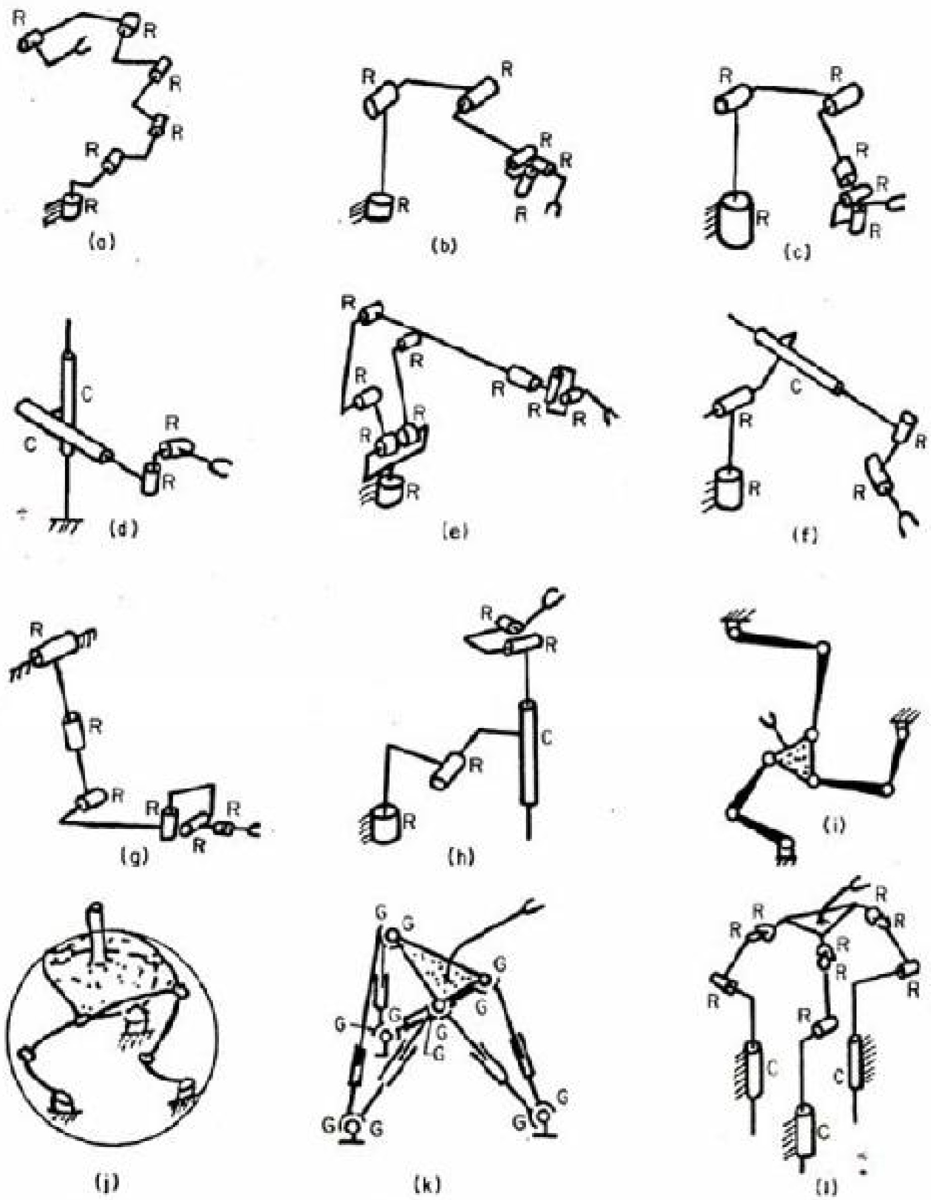


FIG. 39-28 Robots. These are multidegree-of-freedom devices used for positioning or assembly of items. They usually have some degree of machine intelligence and work under computer control. (*a*) A general 6R robot; (*b*) to (*h*) some forms of existing robots; (*i*) parallel actuation of a planar 3-degrees-of-freedom robot; (*j*) Stewart platform which uses the 3-degrees-of-freedom principle; (*k*) Florida shoulder with parallel actuation; (*l*) general robot with parallel actuation.

Clamping mechanisms

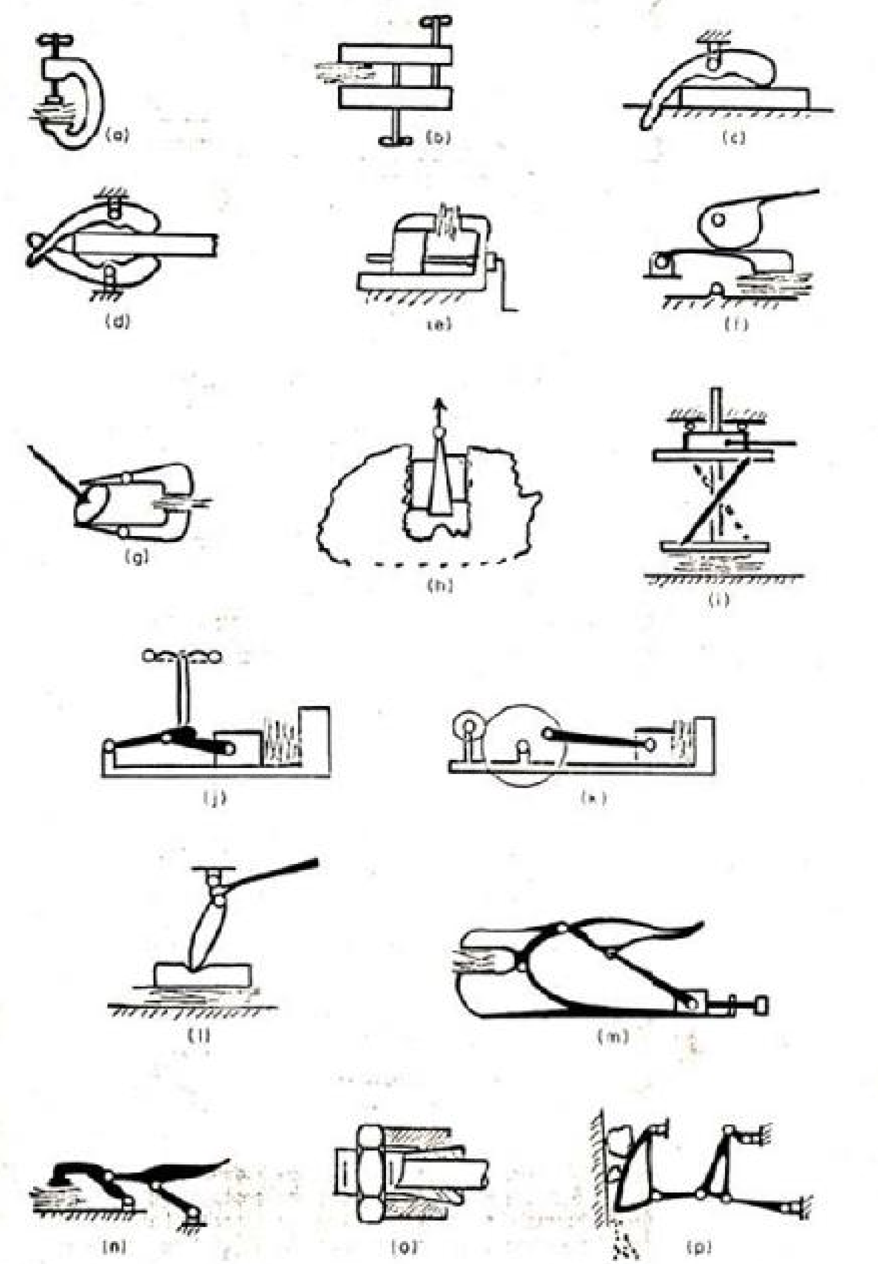


FIG. 39-5 Clamping mechanisms. These devices are used to hold items for machining operations or to exert great forces for embossing or printing. (*a*) C clamp; (*b*) screw clamp; (*c*) cam clamp; (*d*) double cam clamp; (*e*) vise; (*f*) cam-operated clamp; (*g*) double cam-actuated clamp; (*h*) double wedge; (*i*) to (*l*) toggle press; (*m*) vise grips; (*n*) toggle clamp; (*o*) collet; (*p*) rock

Fine adjustments mechanisms

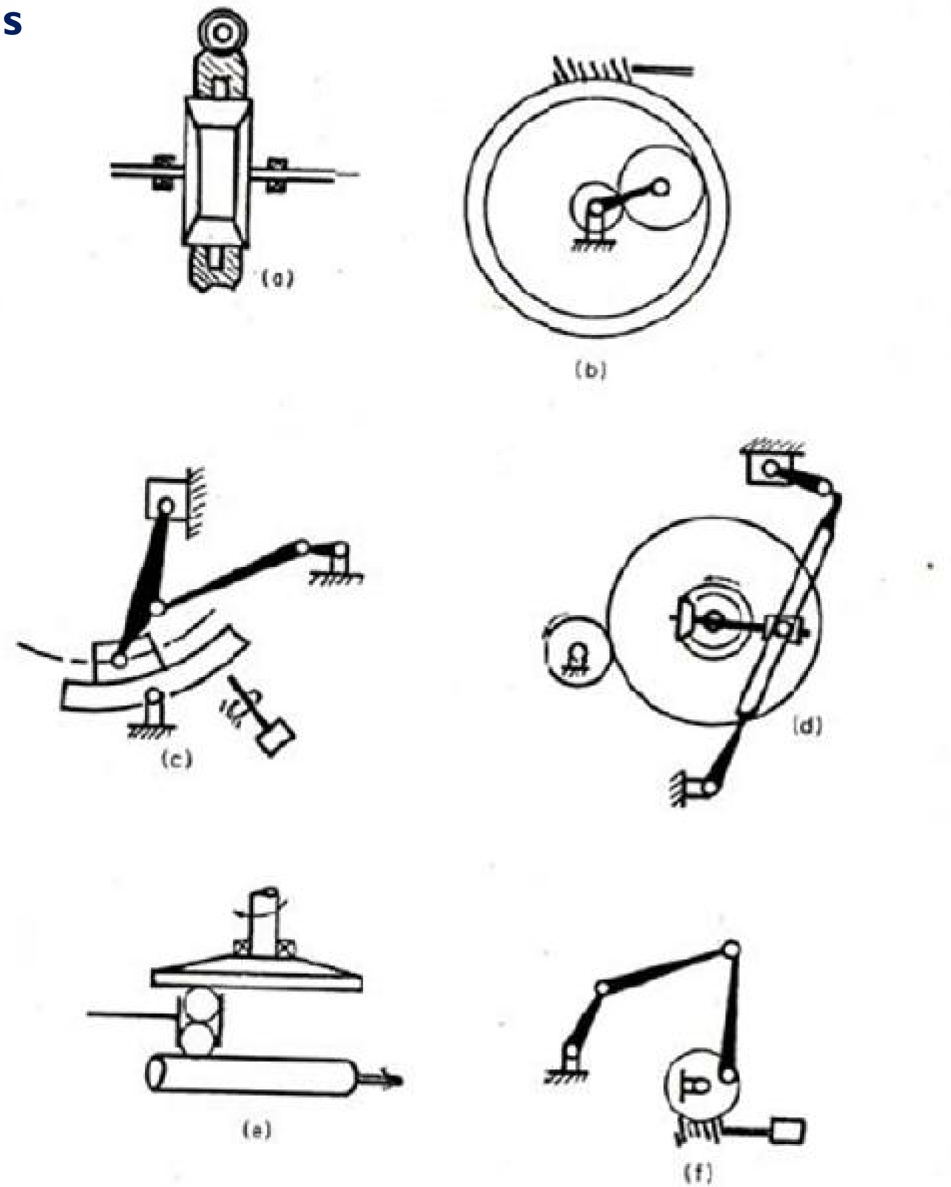


FIG. 39-4 Fine adjustments II. Fine adjustments for moving mechanisms are adjusting devices which control the motion of linkages such as stroke, etc., while the mechanism is in motion. (*a*), (*b*) Differential gear adjustment; (*c*) adjustable-stroke engine; (*d*) adjustable stroke of shaper mechanism; (*e*) ball and disk speed changer; (*f*) adjusting fixed center of linkage for changing motion properties.

Escarpments mechanisms

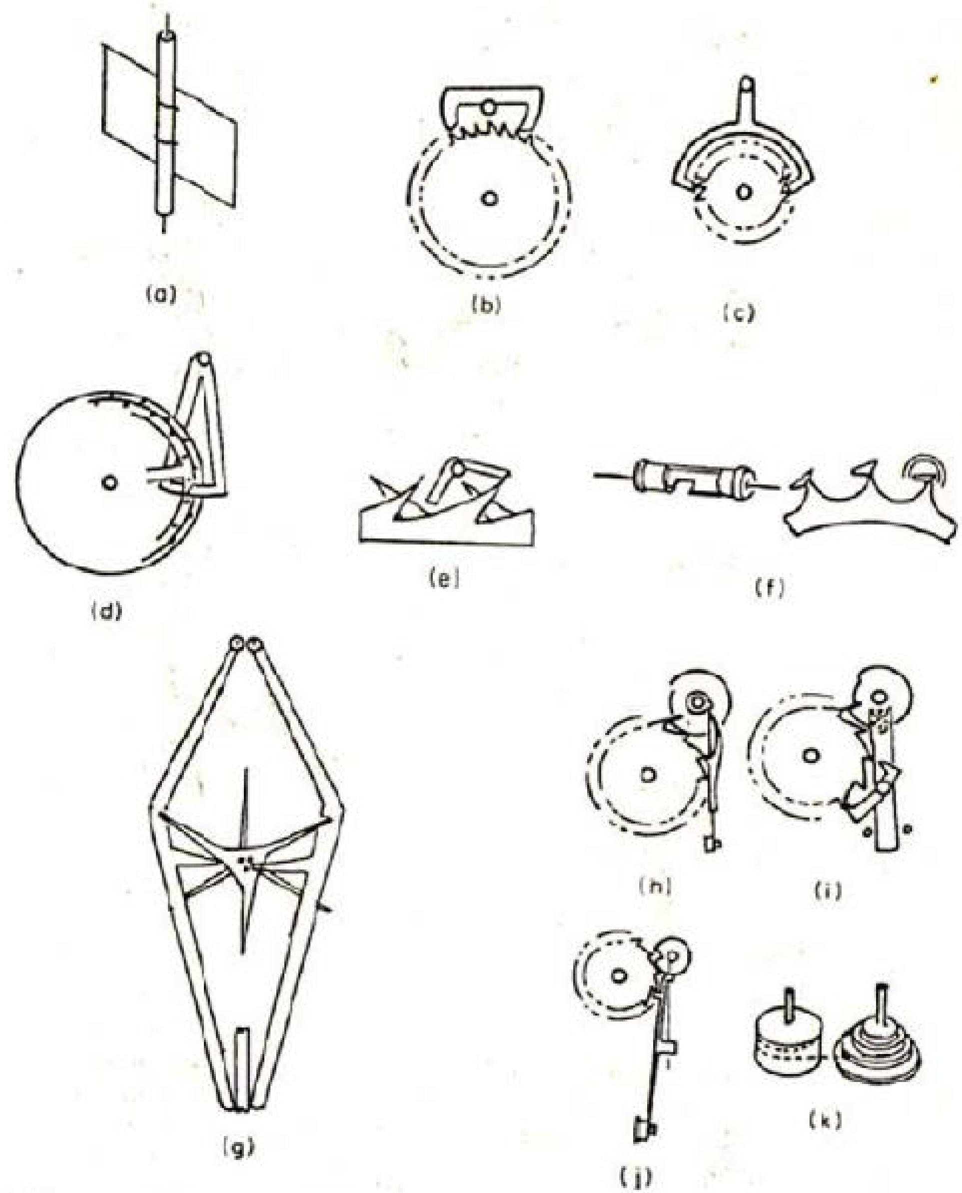


FIG. 39-7 Escapements. These devices slowly release the potential energy stored in a spring to control devices such as clocks. (*a*) Paddle wheel; (*b*) recoil escapement; (*c*) dead-beat escapement; (*d*) stud escapement; (*e*) early anchor escapement; (*f*) cylinder escapement; (*g*) double three-legged escapement for tower clocks; (*h*) to (*j*) chronometer escapements; (*k*) fuse used to give uniform torque at escapement as the spring unwinds.

Locating mechanisms

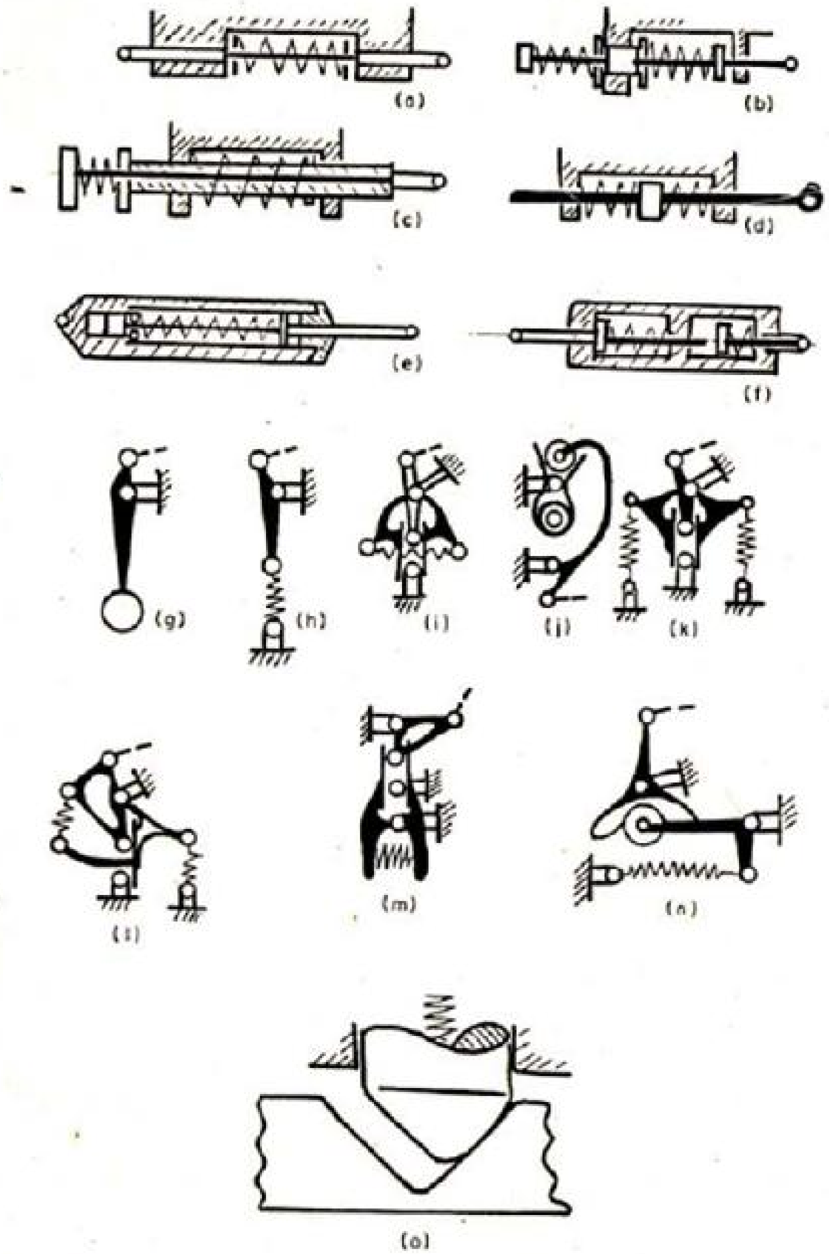


FIG. 39-6 Locating mechanisms. These are devices which properly position a linkage member when the load is removed. (*a*) to (*f*) Self-centering linear devices; (*g*) to (*n*) self-centering angular devices; (*o*) detent.

Oscillating mechanisms I

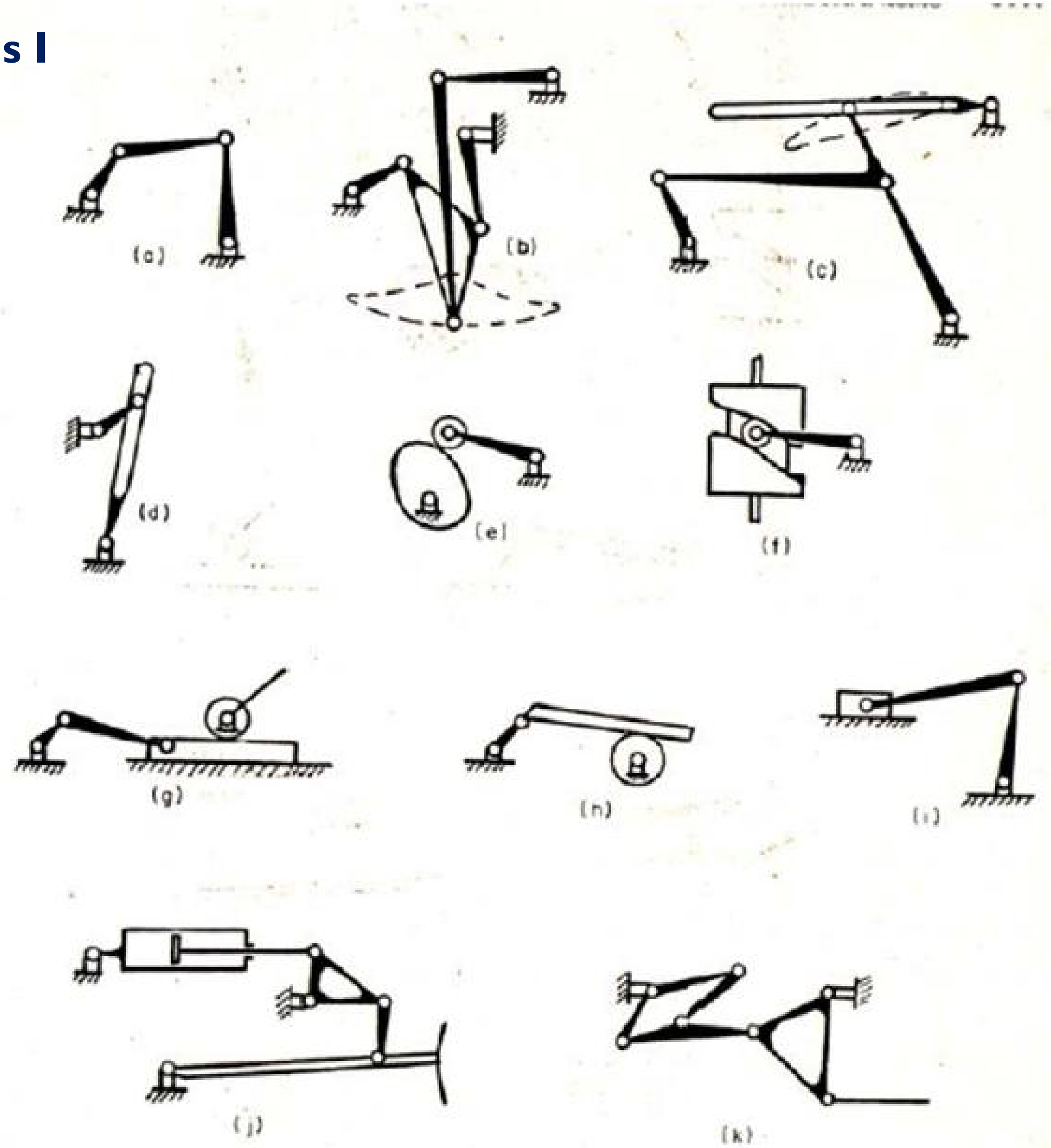


FIG. 39-9 Oscillating mechanisms I. These mechanisms cause an output to repeatedly swing through a preset angle. (*a*) Four-bar linkage; (*b*) six-bar linkage; (*c*) six-bar linkage with pin in slot; (*d*) inverted slide-crank quick-return linkages; (*e*) radial cam and follower; (*f*) cylindrical cam; (*g*) geared slider crank; (*h*) geared inverted slider crank; (*i*) slider-driven crank; (*j*) bulldozer lift mechanism; (*k*) oscillator of the Corliss valve gear.

Oscillating mechanisms II

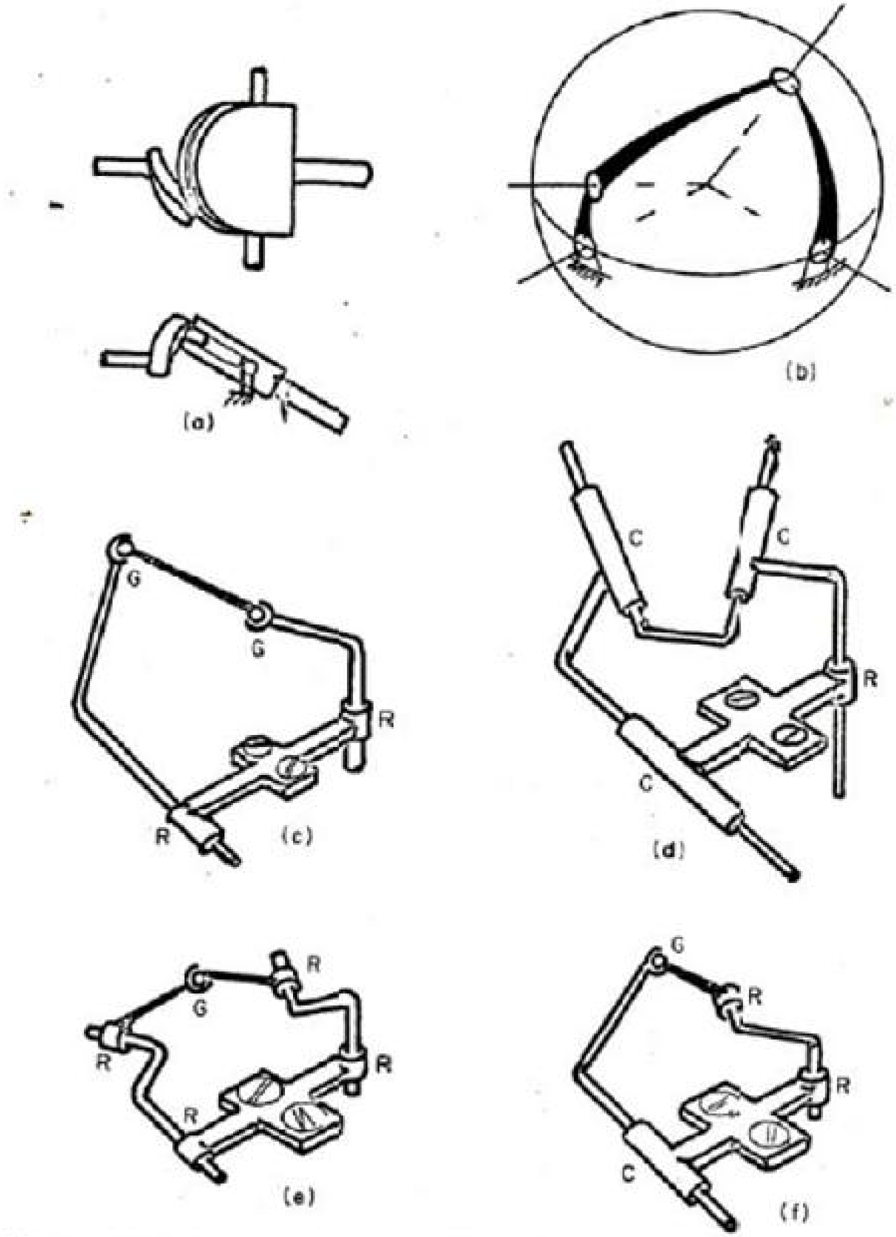


FIG. 39-10 Oscillating mechanisms II. These all use spatial linkages. (*a*) Spatial pin and yoke; (*b*) spherical four-bar linkage; (*c*) spatial RGGR linkage; (*d*) spatial RCCC; (*e*) spatial RRGRR; (*f*) spatial RRGC.

Indexing mechanisms

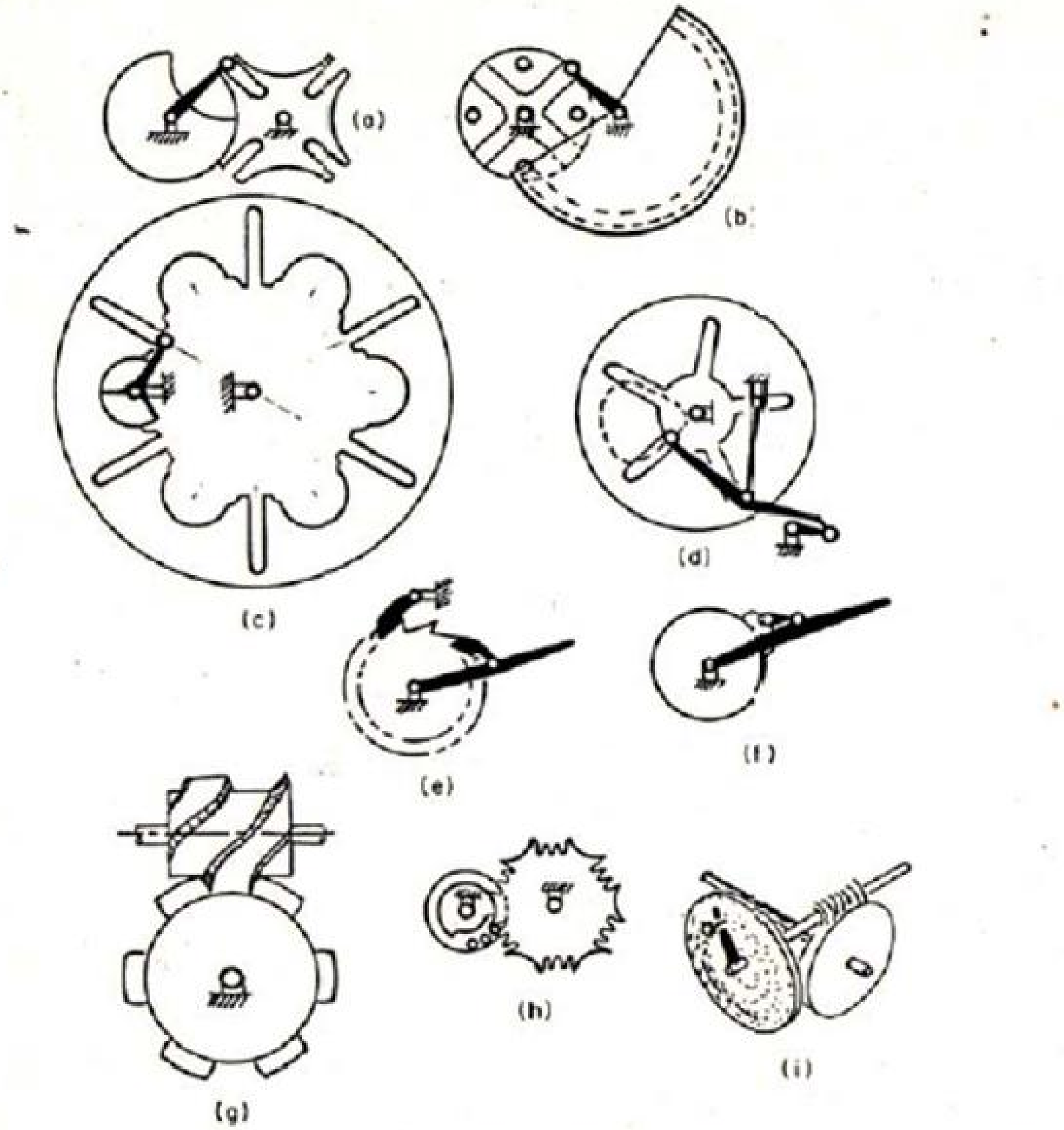


FIG. 39-8 Indexing mechanisms. These mechanical devices advance a body to a specific position, hold it there for a period, and then advance it again. (*a*) to (*c*) Geneva stops; (*d*) four-bar links used to reduce jerk; (*e*) ratchet mechanism; (*f*) friction ratchet; (*g*) cylindrical cam-stop mechanism; (*h*) pin gearing used in indexing; (*i*) dividing head.

Ratchets and latches mechanisms

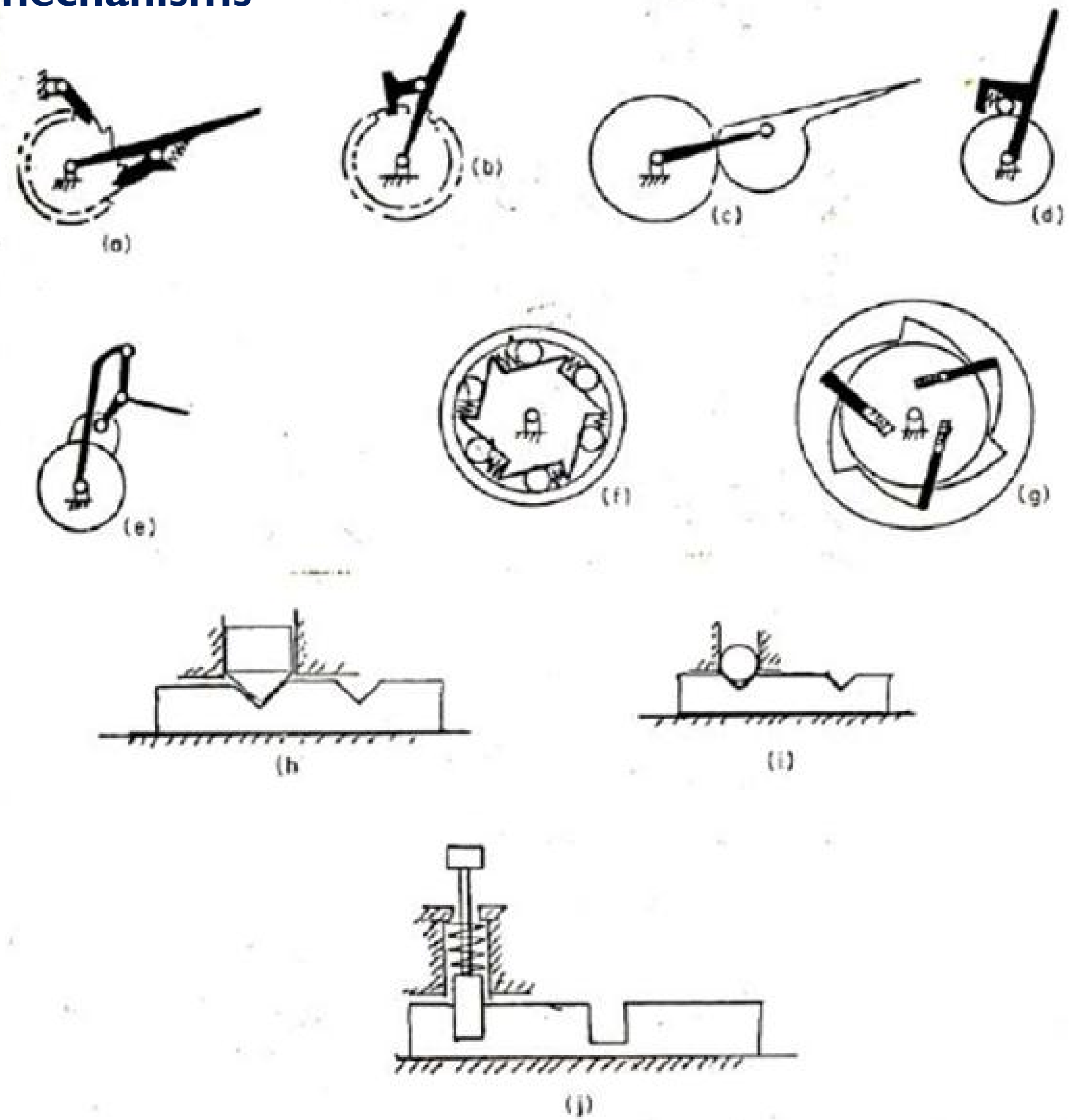


FIG. 39-11 Ratchets and latches. These are mechanisms that advance or hold a machine member. (*a*) Ratchet and pawl; (*b*) reversible ratchet; (*c*) cam-lock ratchet; (*d*) ball-lock ratchet; (*e*) toggle ratchet; (*f*) overrunning clutch; (*g*) high-torque ratchet; (*h*), (*i*) detents; (*j*) locking bolts.

Reciprocating mechanisms I

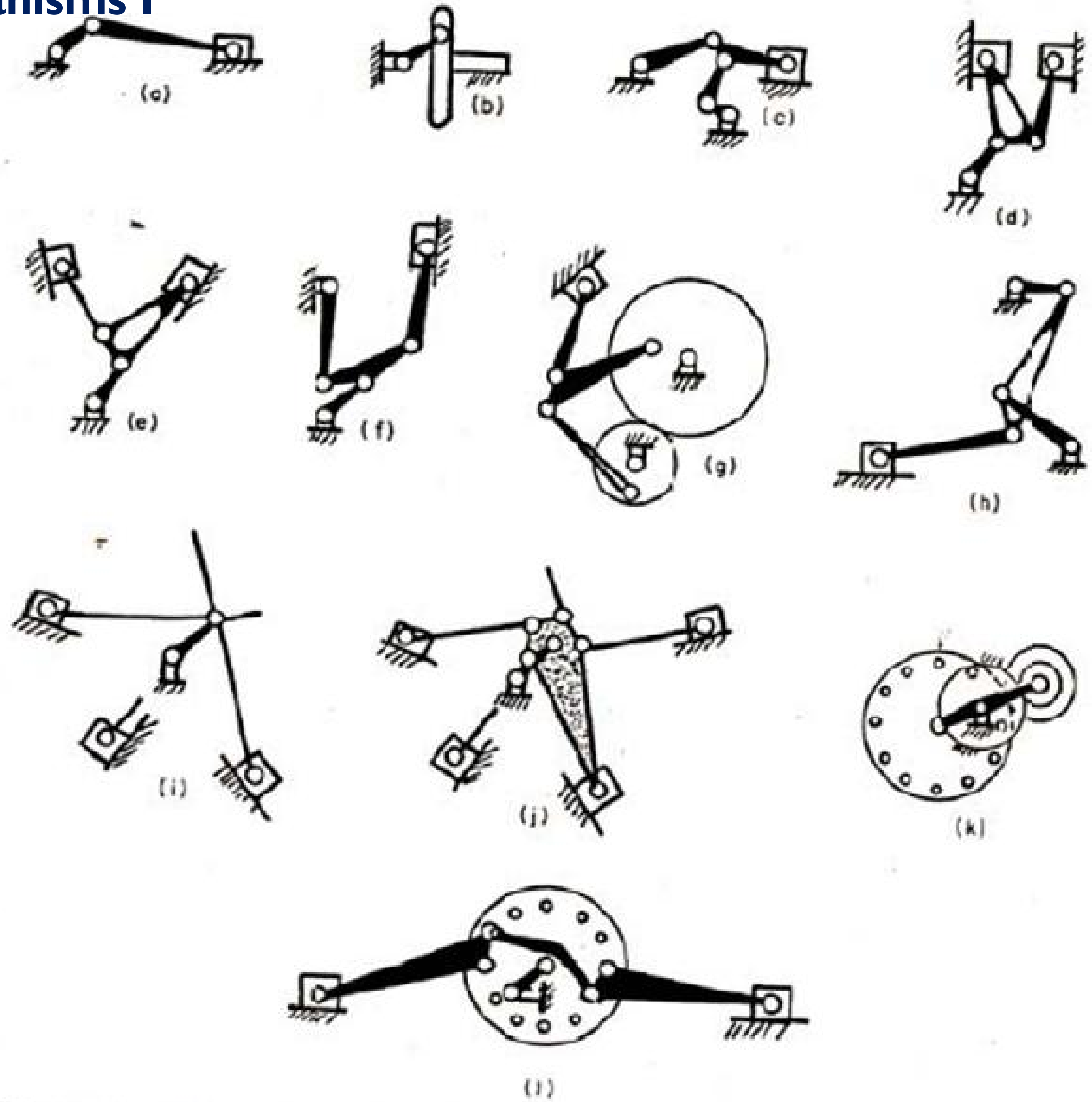


FIG. 39-12 Reciprocating mechanisms I. These mechanical devices cause a member to translate on a straight line. (*a*) Slider crank; (*b*) Scotch yoke; (*c*) toggle mechanism; (*d*) Zoller engine; (*e*) V Engine; (*f*) double-stroke engine; (*g*) geared engine; (*h*) Atkinson gas engine; (*i*) ideal radial engine; (*j*) practical radial engine; (*k*) geared Nordberg radial engine; (*l*) linked Nordberg radial engine.

Reciprocating mechanisms II

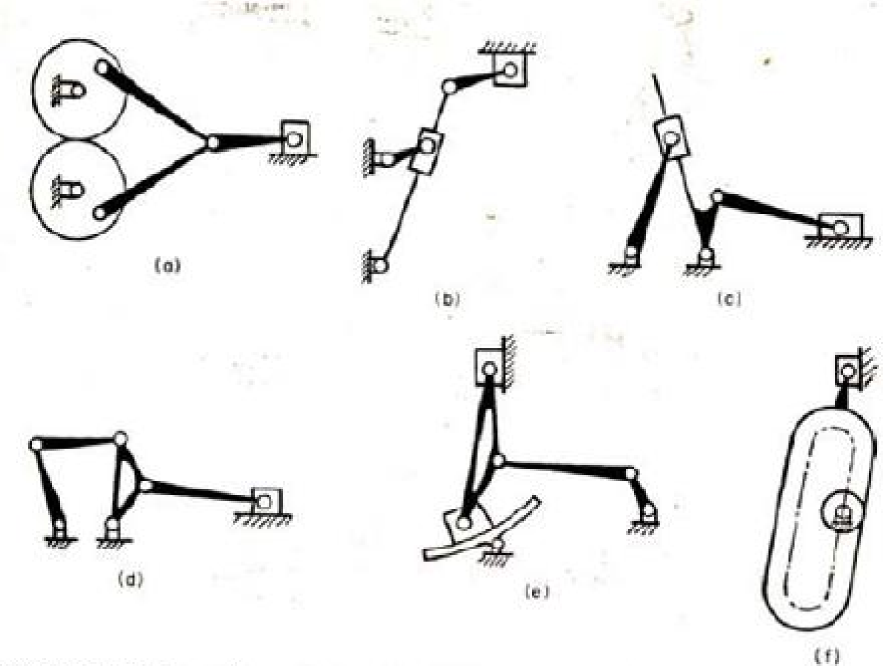


FIG. 39-13 Reciprocating mechanisms II. (*a*) Geared cranks; (*b*) shaper mechanism; (*c*) slider or Whitworth quick-return mechanisms; (*d*) slider on drag-link mechanism; (*e*) variable-stroke engine; (*f*) gear-driven slider.

Reversing mechanisms

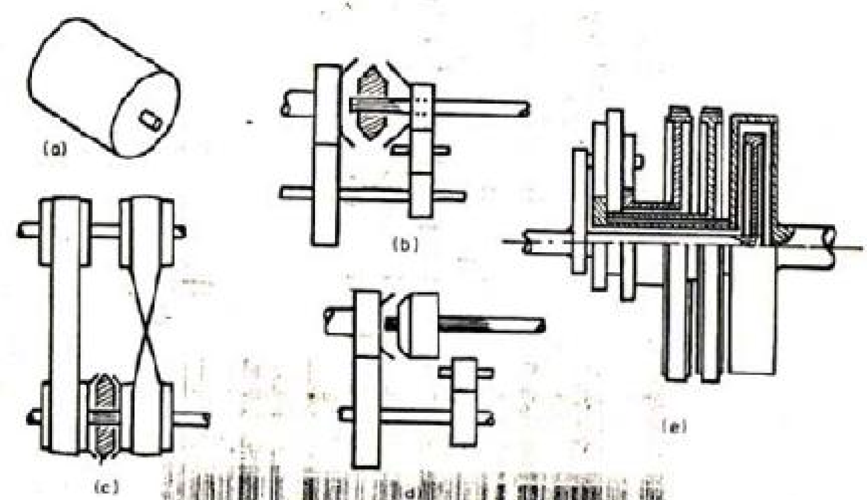


FIG. 39-14 Reversing mechanisms. These mechanical devices change the direction of rotation of the output. (*a*) Reversible prime movers; (*b*) reversing gears; (*c*) reversing belts; (*d*) transmission; (*e*) epicyclic gears as in Model T Ford.

Coupling and connectors -parallel shafts mechanisms

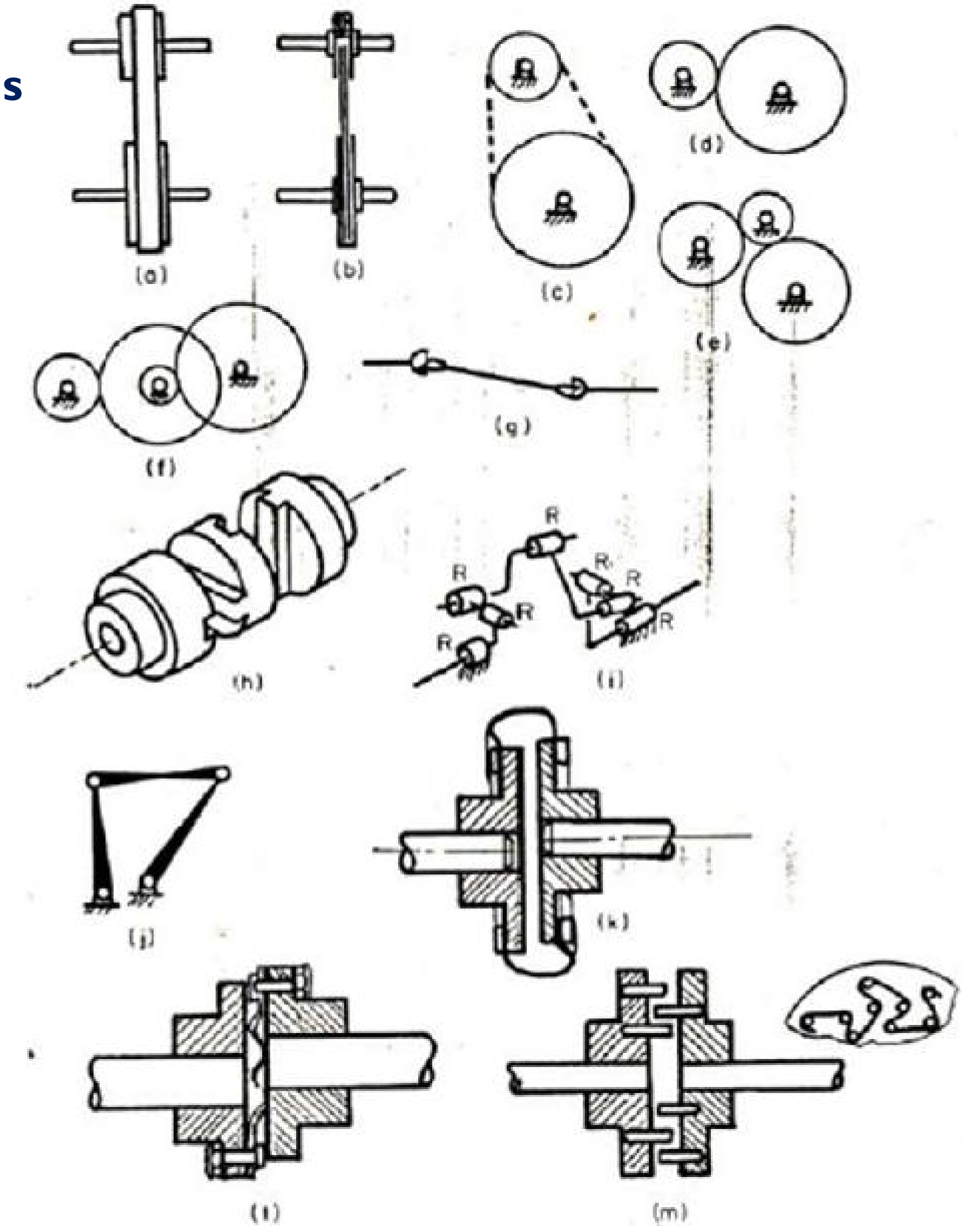


FIG. 39-16 Couplings and connectors—parallel shafts. (*a*) Flat belt; (*b*) V belt; (*c*) chain; (*d*) to (*f*) gears; (*g*) Hooke joints; (*h*) Oldham coupling; (*i*) Hunt's constant-velocity coupling; (*j*) drag link; (*k*) to (*m*) flexible coupling.

Coupling and connectors
-axial mechanisms

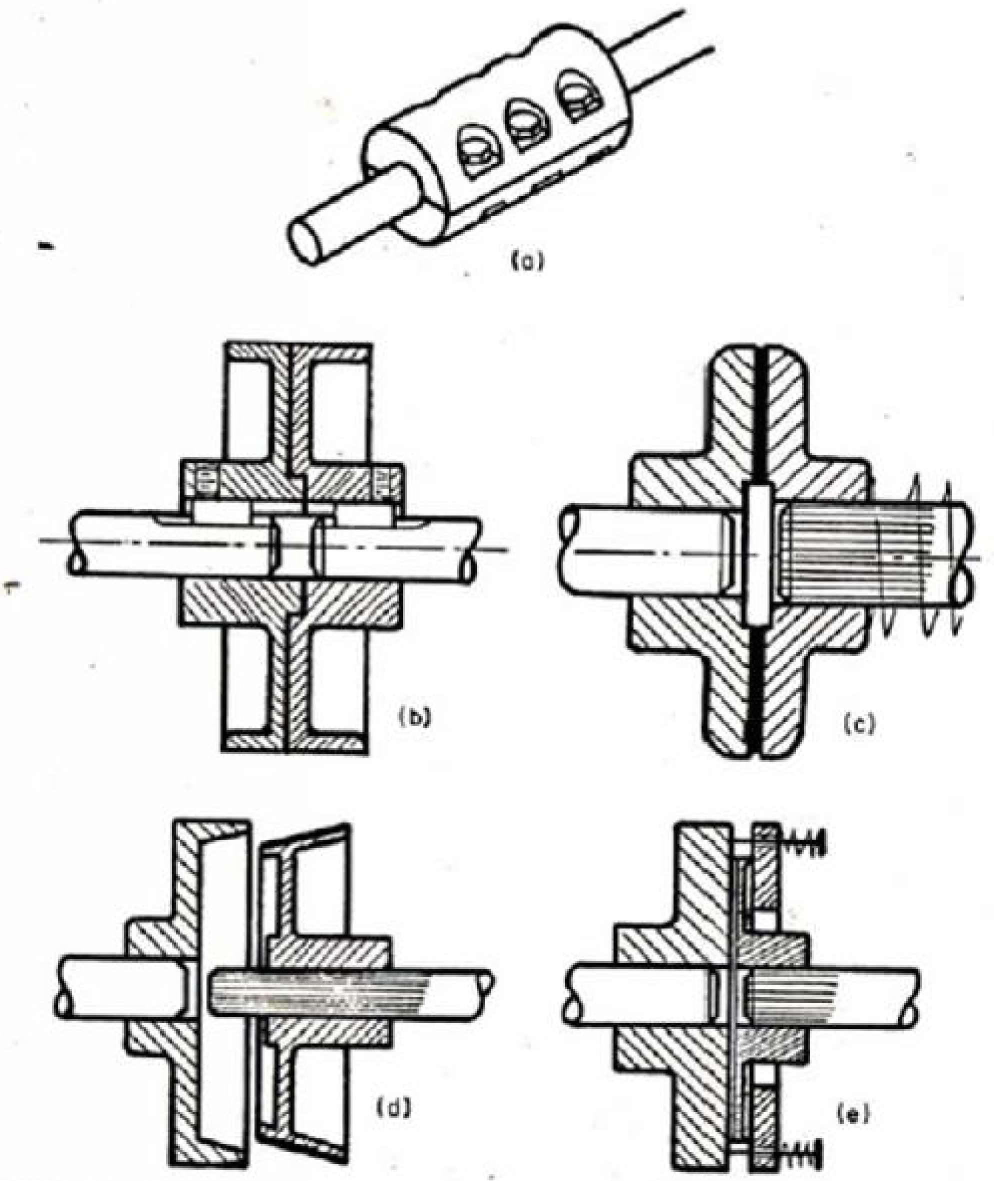


FIG. 39-15 **Couplings and connectors—axial.** These are used to connect coaxial shafts. (*a*) Rigid coupling; (*b*) flanged coupling; (*c*) disk clutch; (*d*) cone clutch; (*e*) plate clutch.

Coupling and connectors
-skew shafts mechanisms

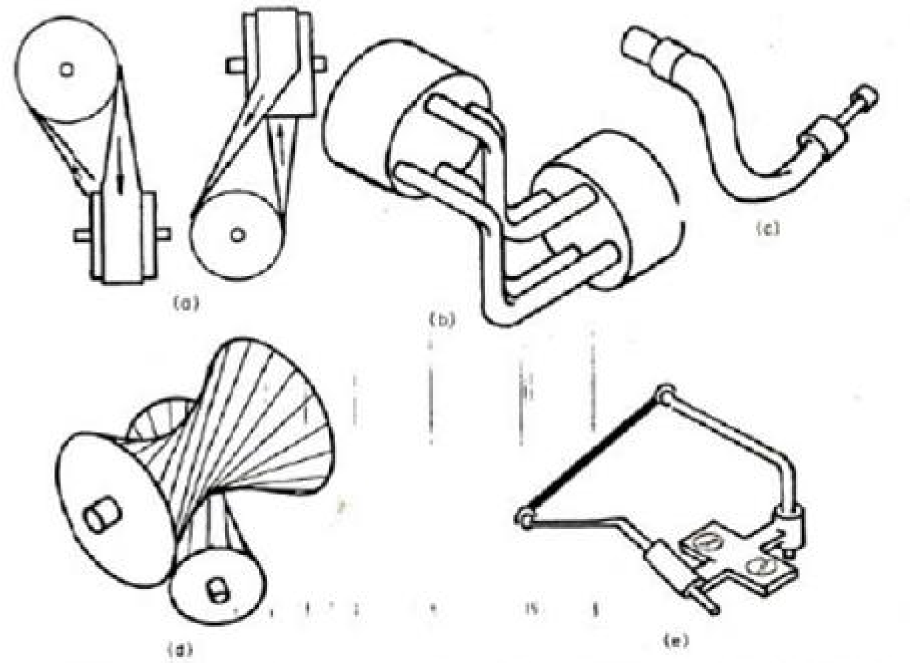


FIG. 39-18 Couplings and connectors—skew shafts. (*a*) Flat belts; (*b*) spatial RCCR; (*c*) flexible shaft; (*d*) hypoid gears; (*e*) spatial RGGR.

Slider connectors mechanisms

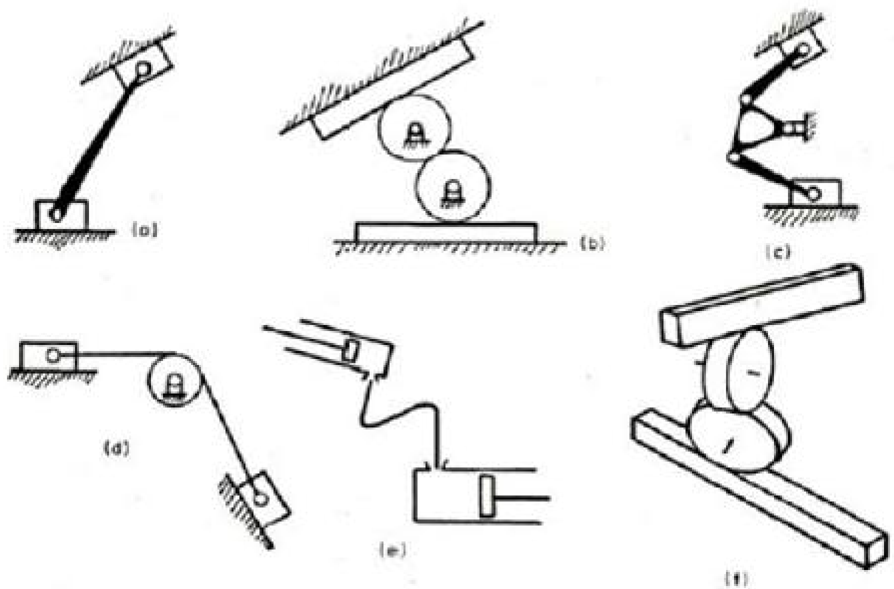


FIG. 39-19 Slider connectors. These devices connect two or more reciprocating devices. (*a*) Elliptic trammel; (*b*) gears; (*c*) slider-crank-slider; (*d*) cable; (*e*) hydraulic; (*f*) helical gearing.

Coupling and connectors -interacting shafts mechanisms

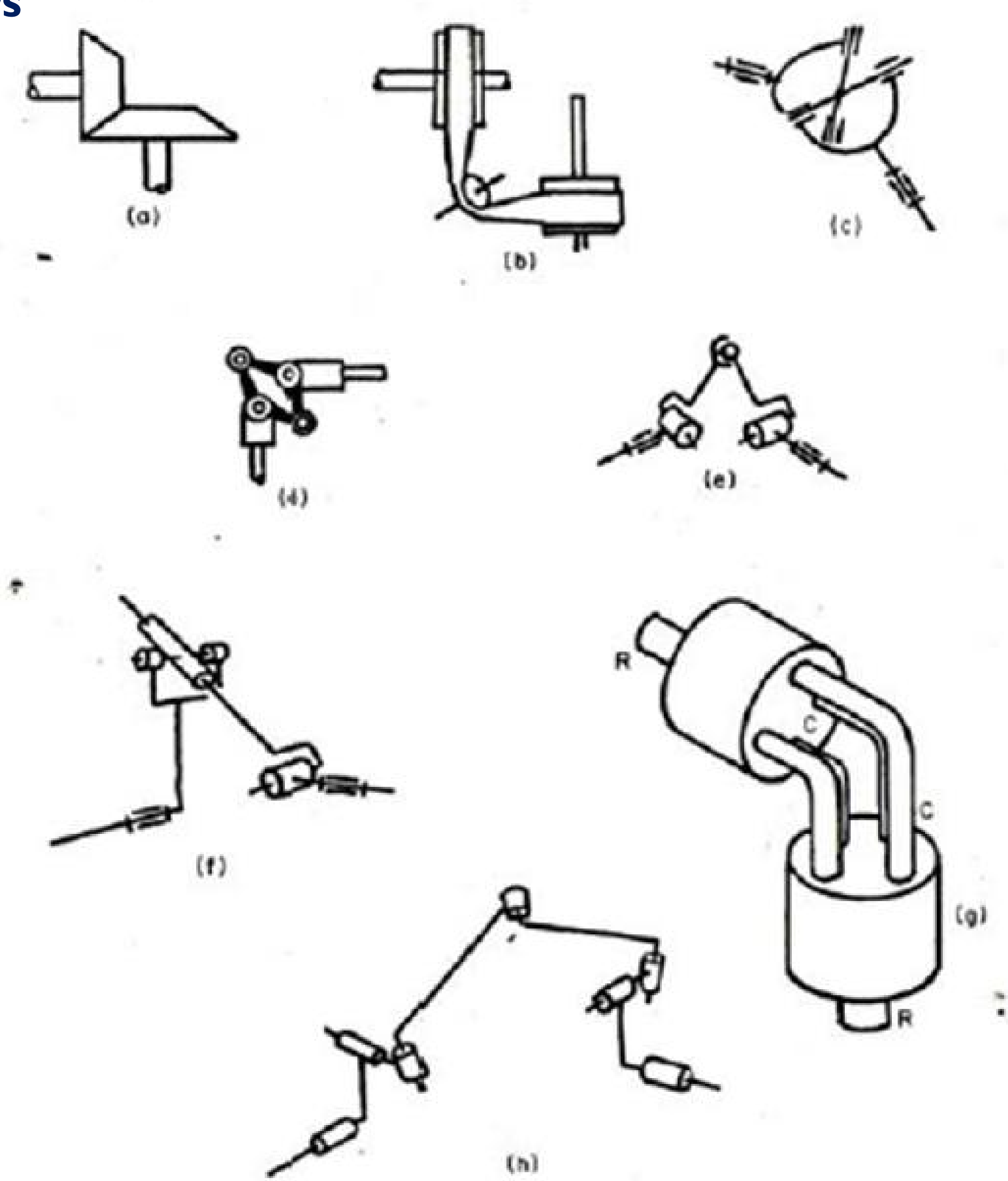


FIG. 39-17 Couplings and connectors—intersecting shafts. (*a*) Bevel gears; (*b*) flat belts with idlers; (*c*) Hooke joint; (*d*) Hooke's coupling; (*e*) Clemens coupling; (*f*) Rouleaux coupling; (*g*) spatial RCCR; (*h*) Hunt's constant-velocity coupling.

Transportation mechanisms

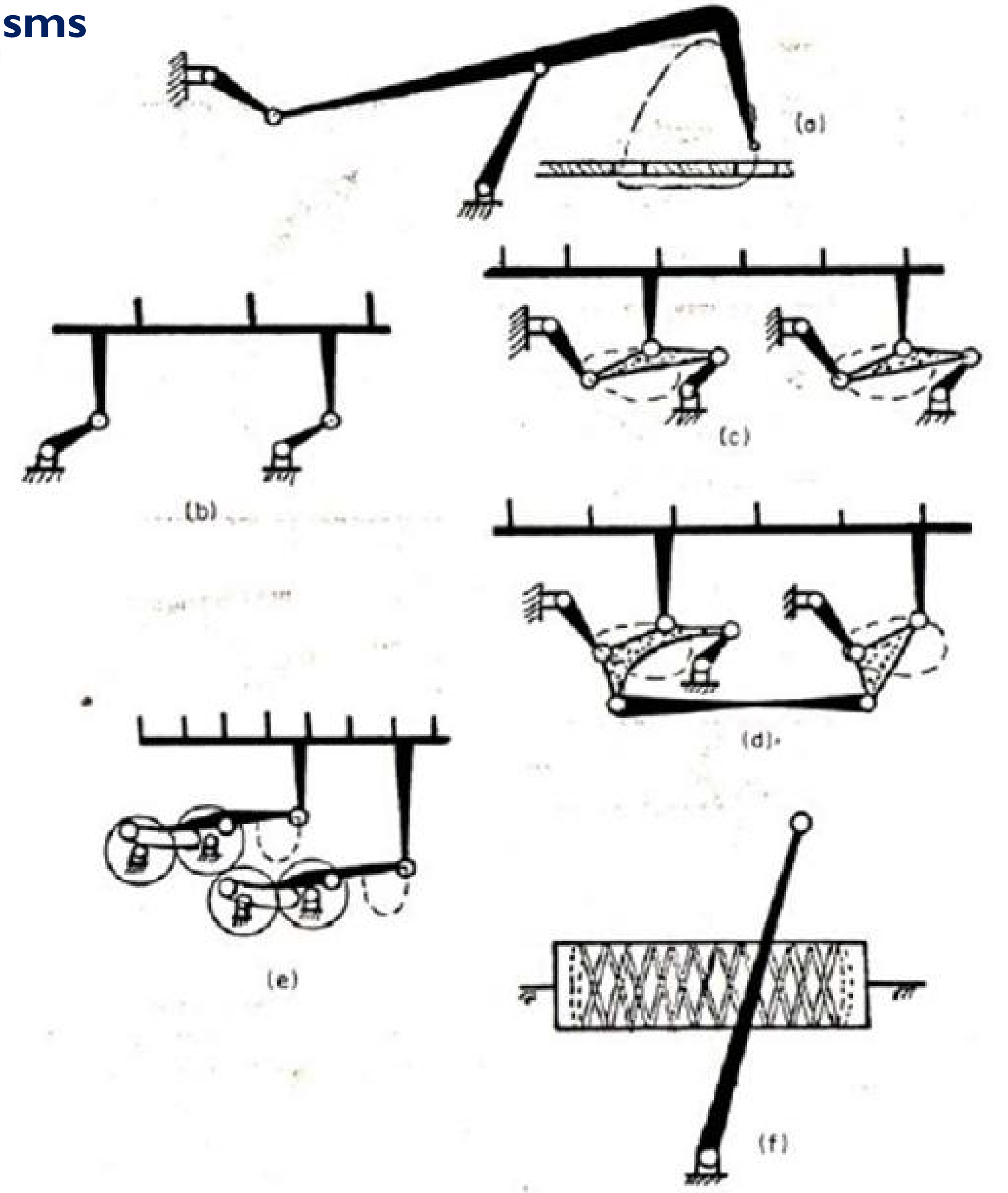


FIG. 39-21 Transportation devices. These mechanisms move one or more objects a discrete distance in stepped motion. (*a*) Four-bar film advance; (*b*) circular-motion transport; (*c*), (*d*) coupler-curve transport; (*e*) geared linkage transport; (*f*) fishing-reel feed.

Loading and unloading mechanisms I

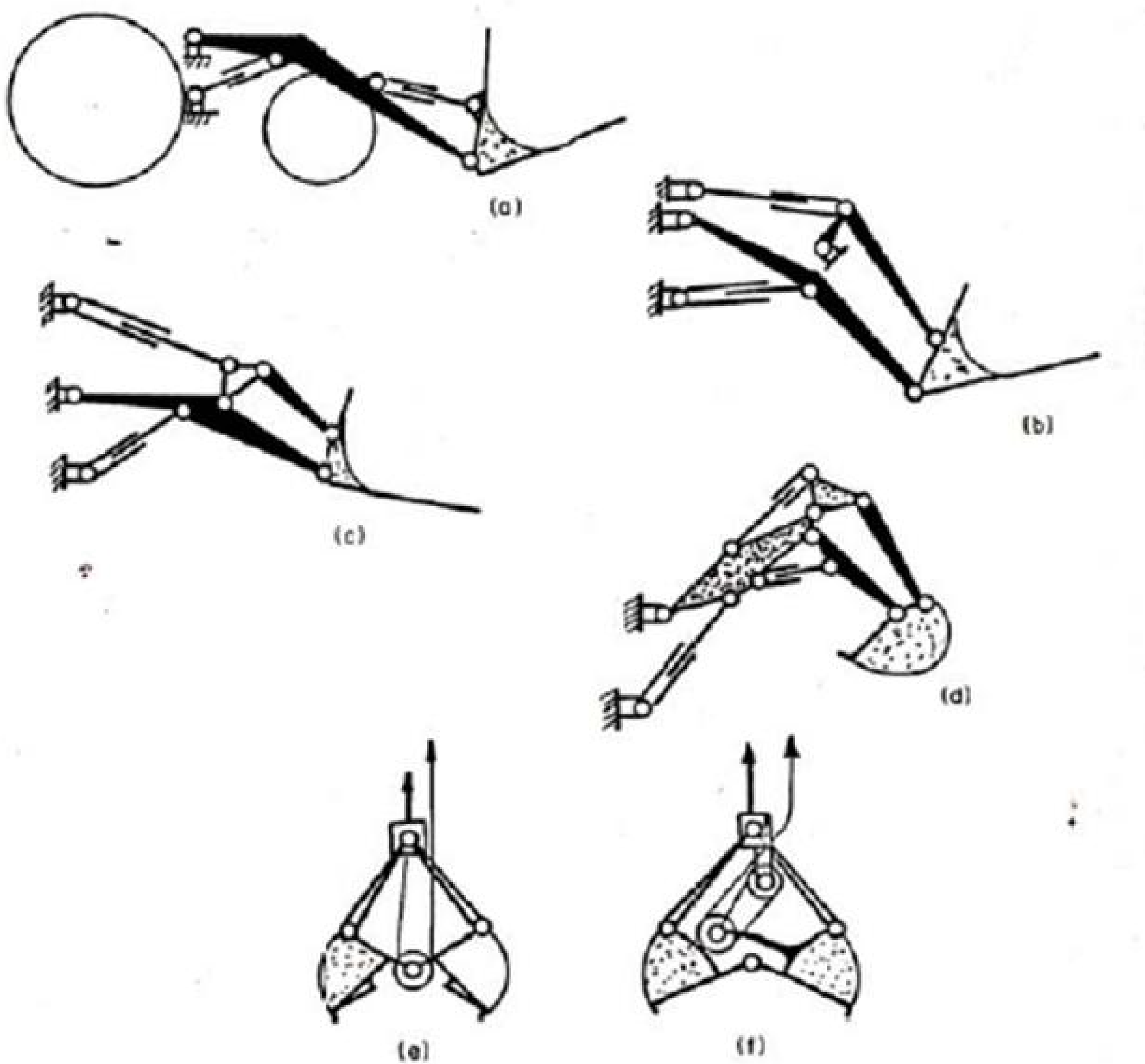


FIG. 39-22 Loading and unloading mechanisms I. These mechanisms pick up material and transport it to another location. (*a*) to (*c*) Front-end loaders; (*d*) back hoe; (*e*), (*f*) clamshell loaders.

Loading and unloading mechanisms II

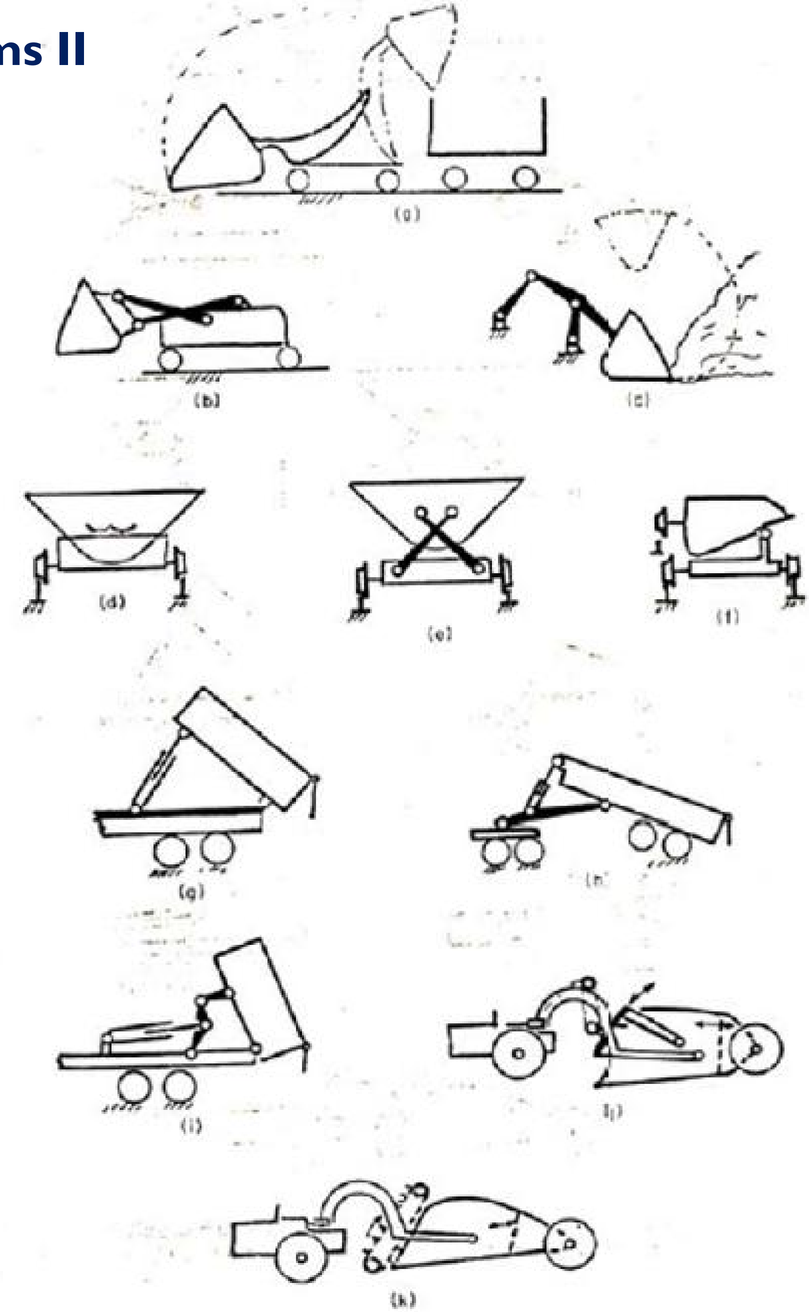


FIG. 39-23 Loading and unloading mechanisms II. (*a*), (*b*) Mucking machines; (*c*) scooping mechanism; (*d*) to (*f*) dumping mine cars; (*g*) to (*i*) dump trucks; (*j*) motor scraper; (*k*) elevat-

Mechanical Engineering Design II

Ninth & Tenth Lectures

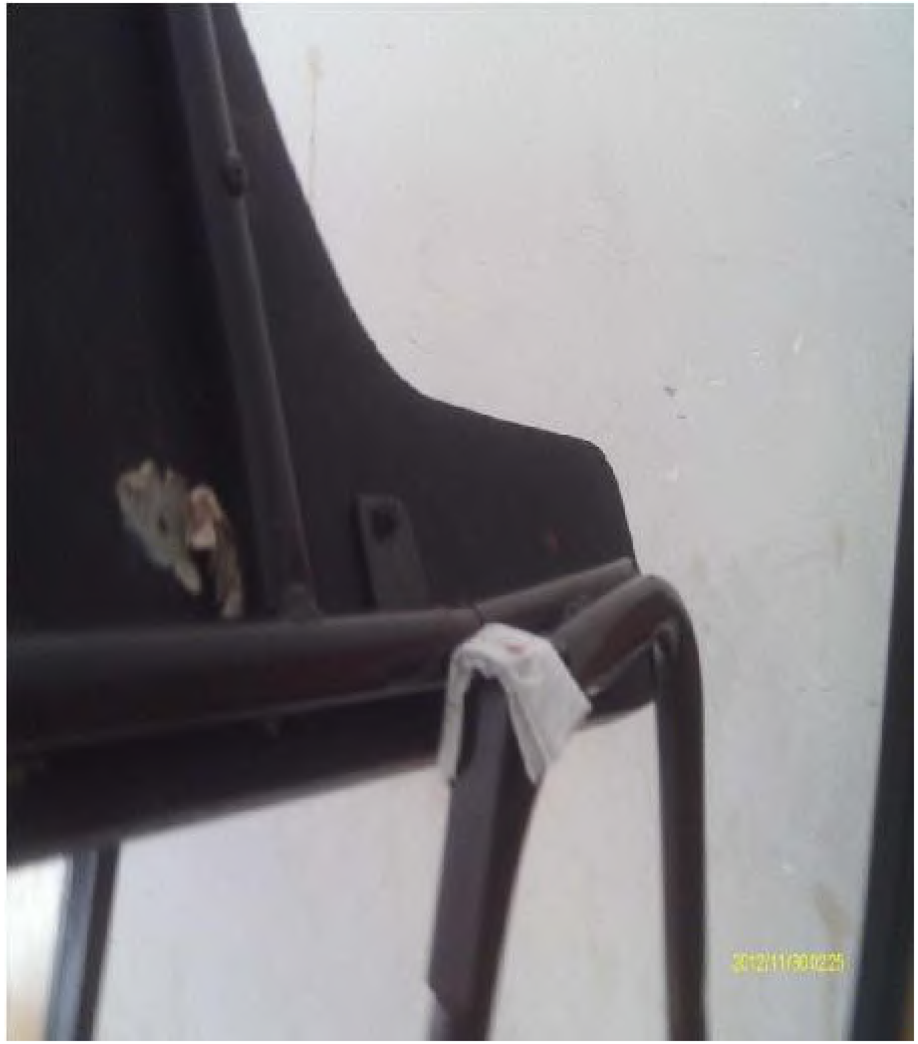
Decision Making

Portable Chair

Problems of Existing Design:

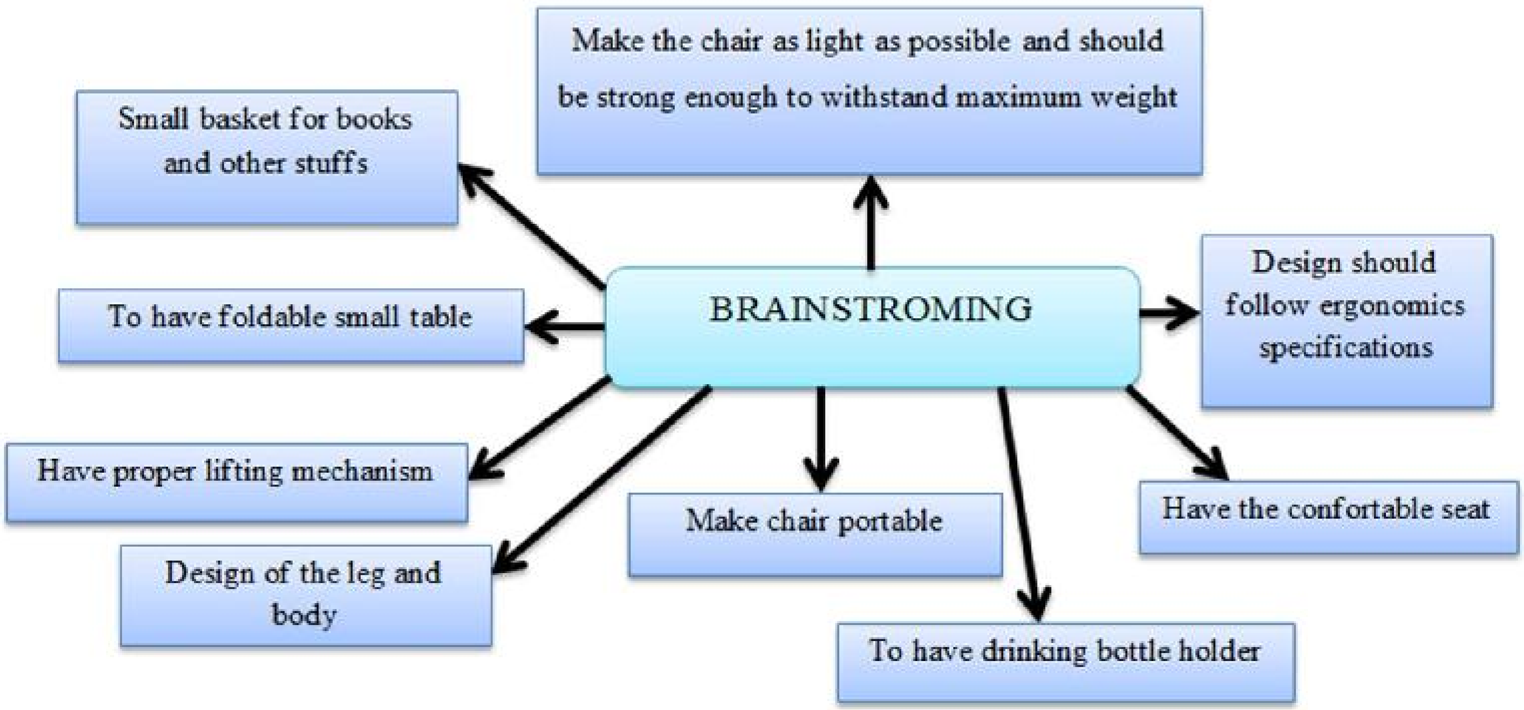
Bending of holding frame

Broken small table

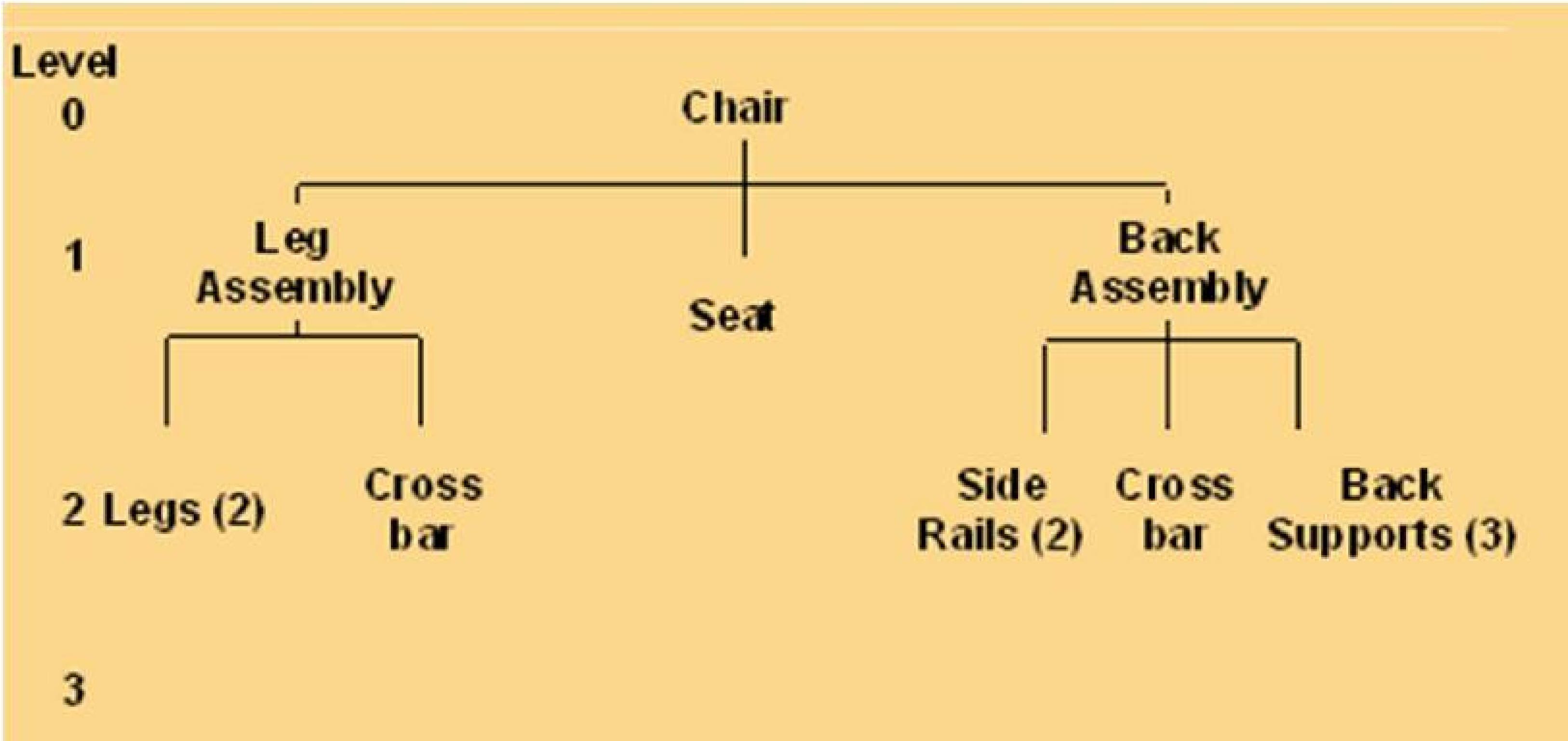


Paper is used to eliminate the shaking of the small table

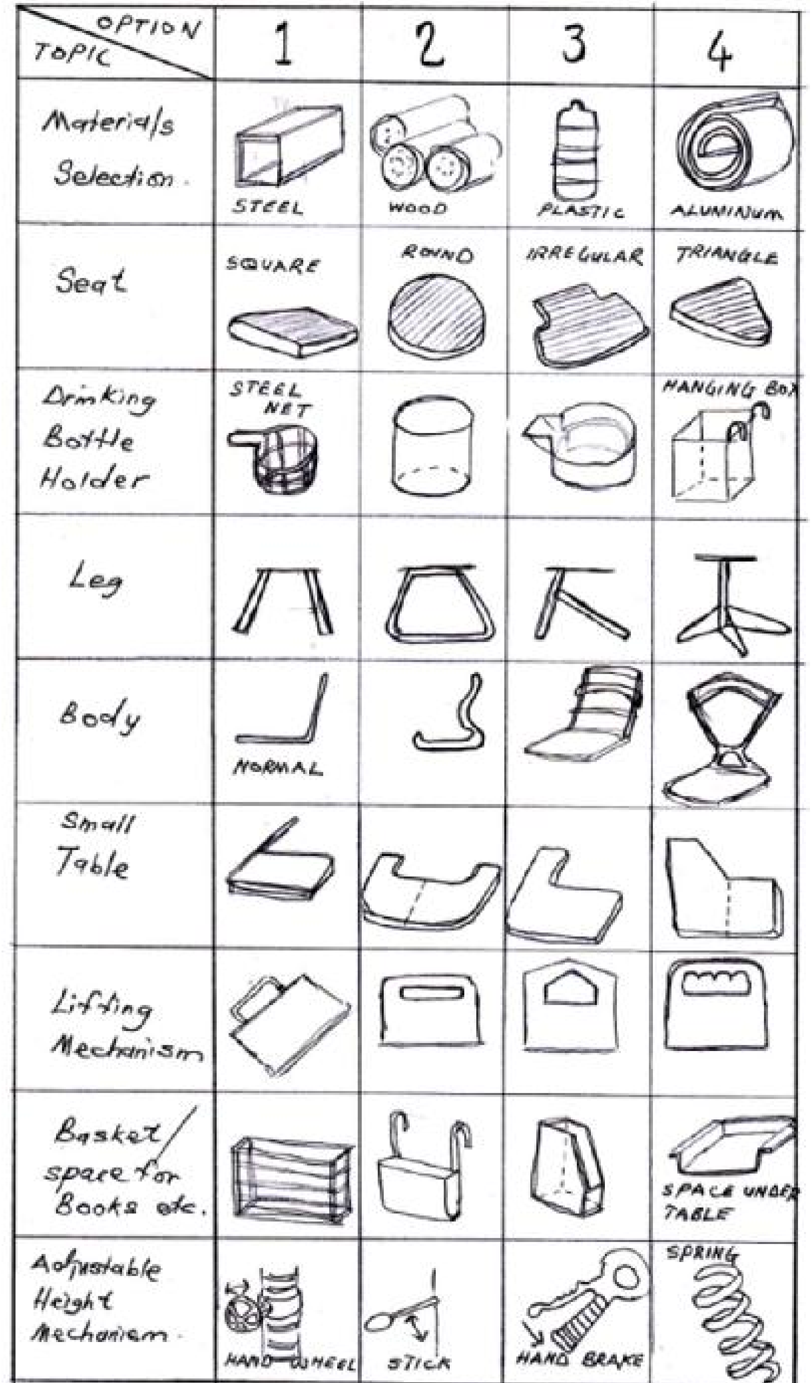
We used brainstorming method to generate as much idea as possible. It is the most common method used for generating ideas.



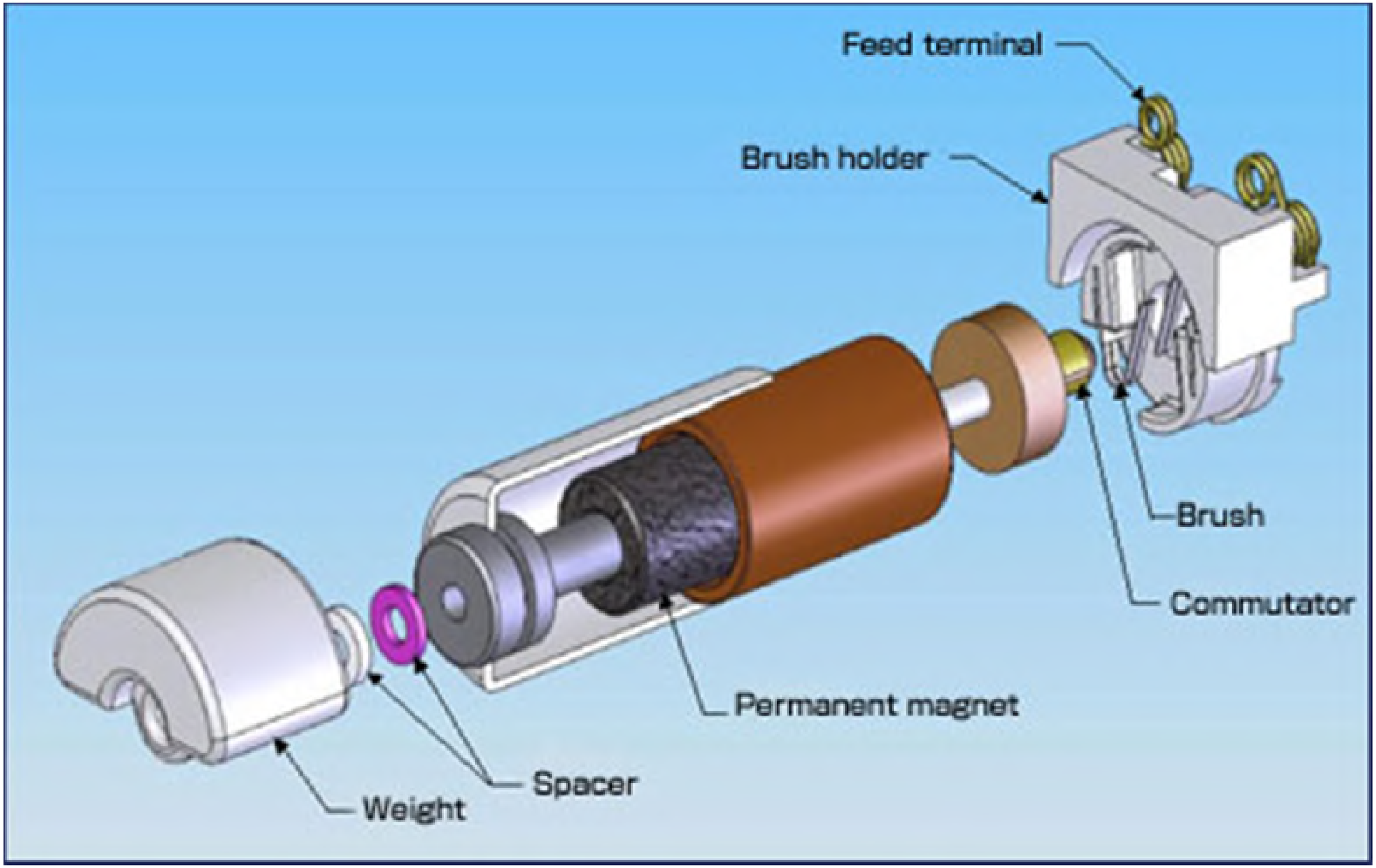
Design Tree

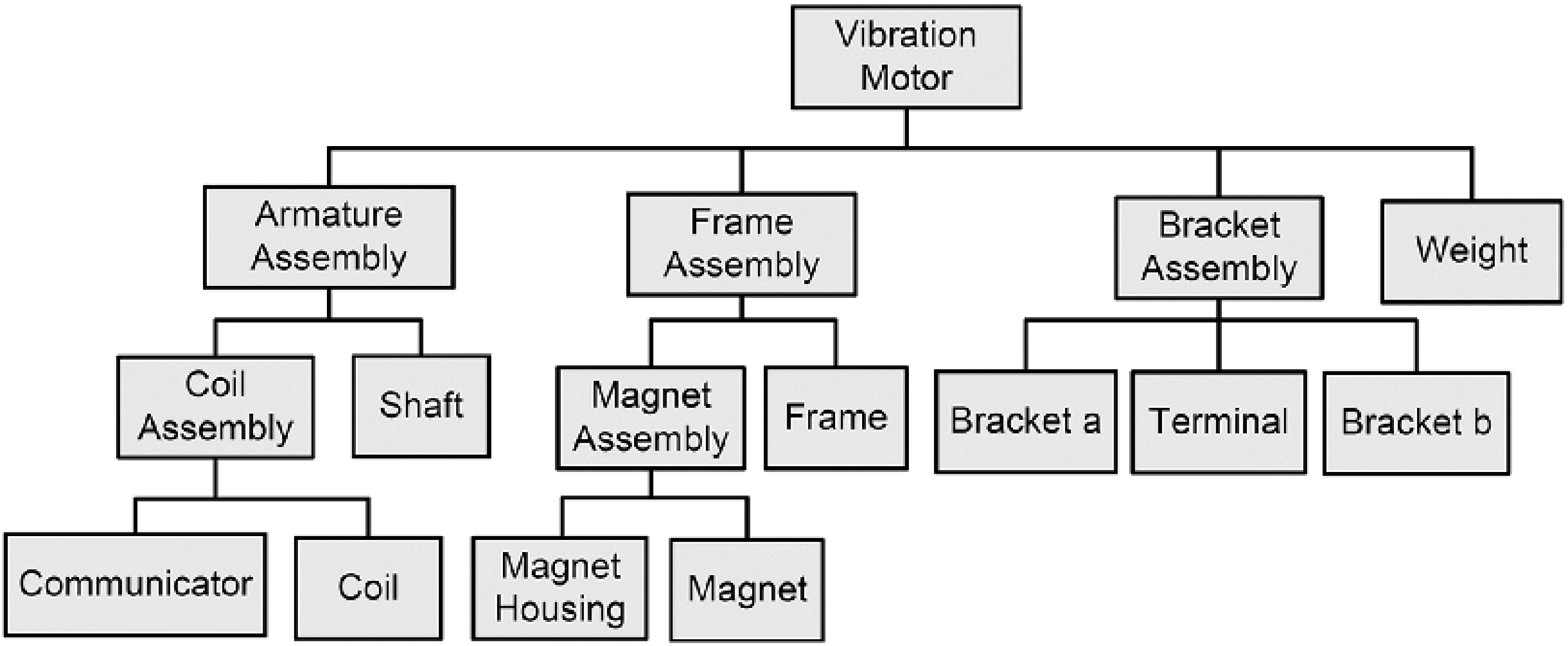

Level
0
Chair
1
Leg Assembly
Seat
Back Assembly
2
Legs (2)
Cross bar
Side Rails (2)
Cross bar
Back Supports (3)
3

Morphological Chart



Example1. Vibration Motor

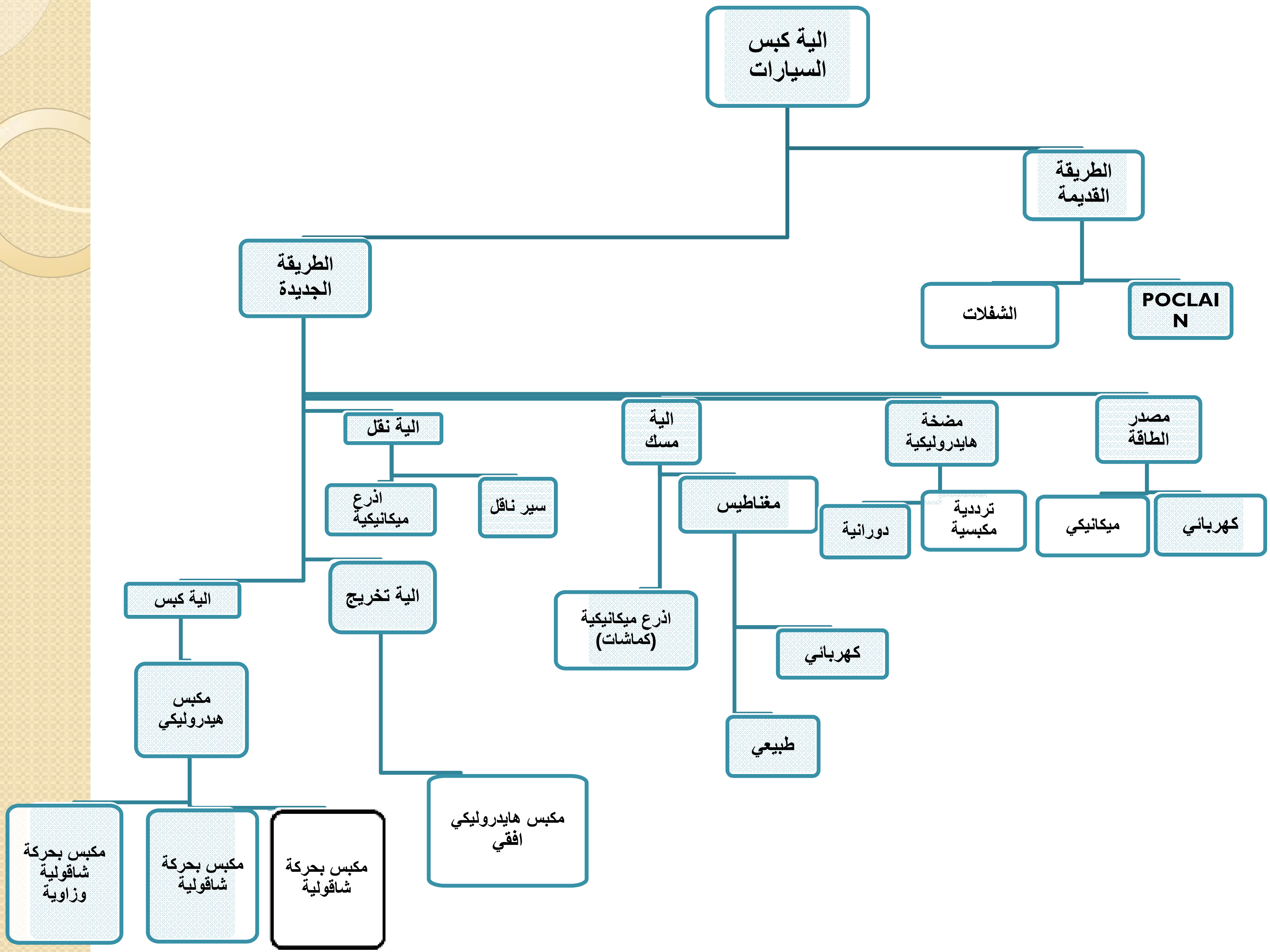




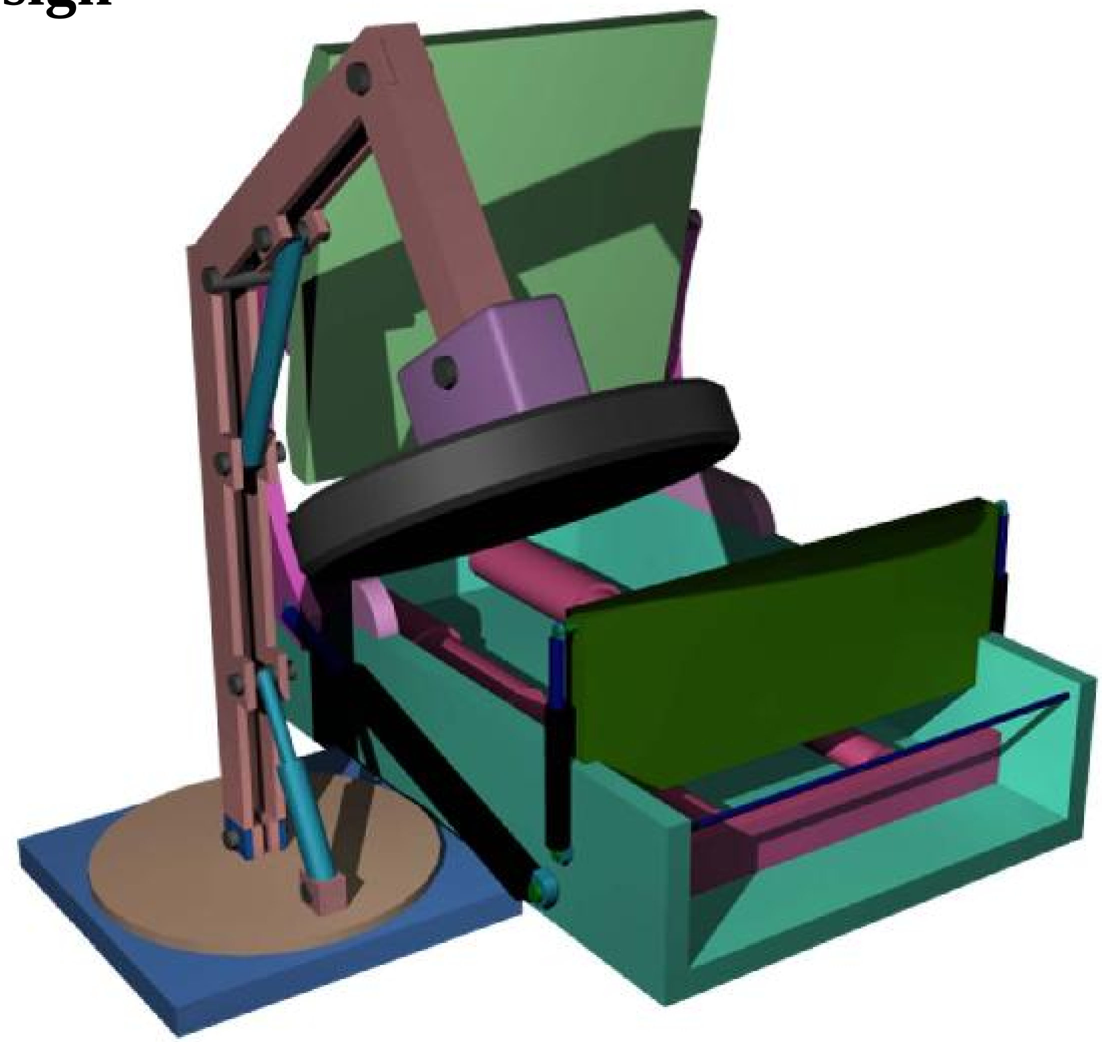
Vibration Motor
Armature Assembly
Frame Assembly
Bracket Assembly
Weight
Coil Assembly
Shaft
Magnet Assembly
Frame
Bracket a
Terminal
Bracket b
Communicator
Coil
Magnet Housing
Magnet

Example2.

Current solution


الية كبس السيارات
الطريقة القديمة
الشفلات
POCLAIN
الطريقة الجديدة
مصدر الطاقة
كهربائي
ميكانيكي
مضخة هايدروليكية
ترددية مكبسية
دورانية
الية مسك
مغناطيس
كهربائي
طبيعي
اذرع ميكانيكية (كماشات)
الية نقل
سير ناقل
اذرع ميكانيكية
الية تخريج
مكبس هايدروليكي افقي
الية كبس
مكبس هيدروليكي
مكبس بحركة شاقولية
مكبس بحركة شاقولية
مكبس بحركة شاقولية وزاوية

Final Design

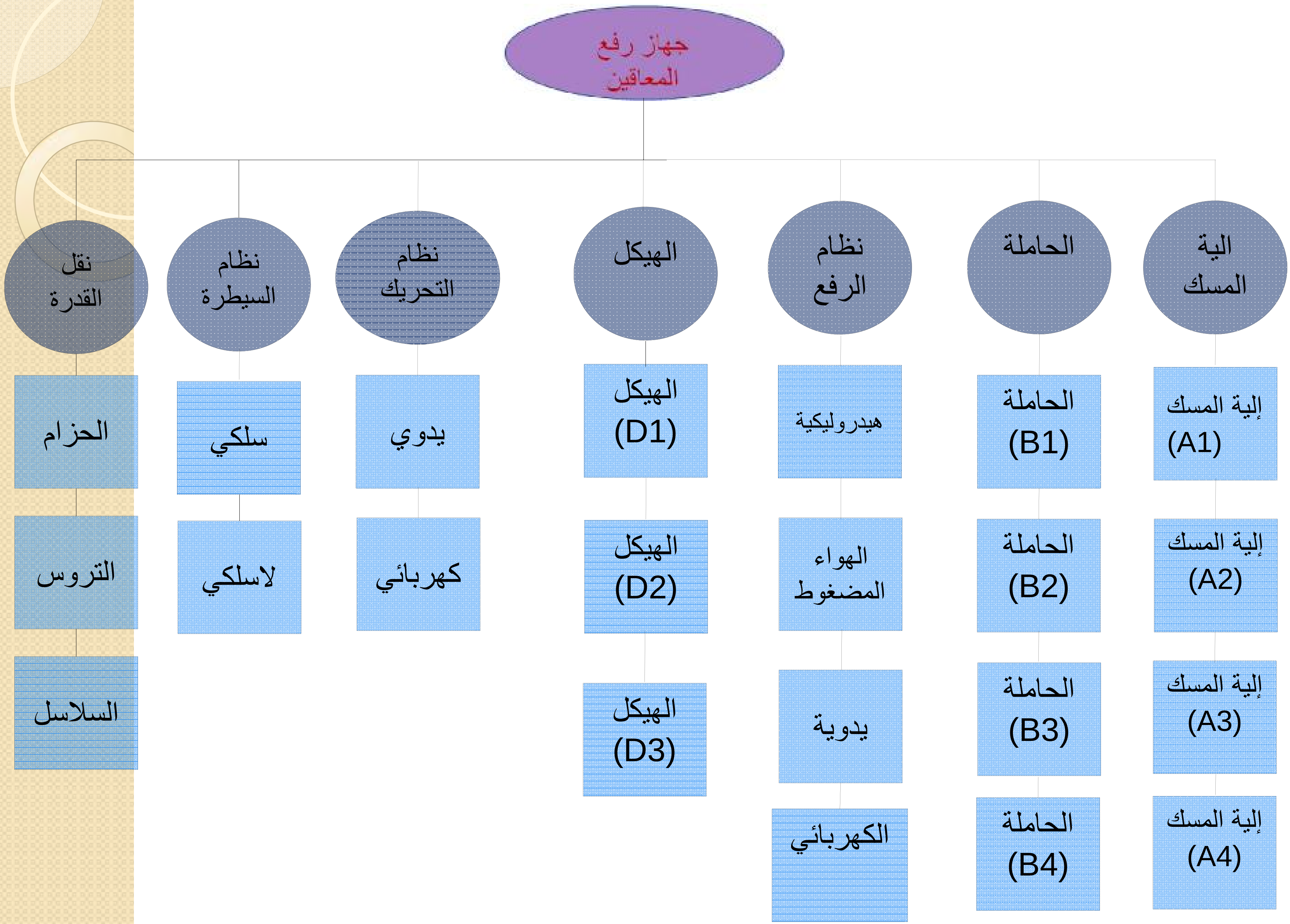


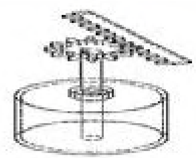
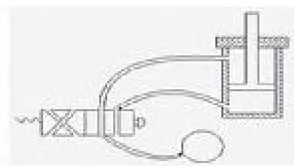
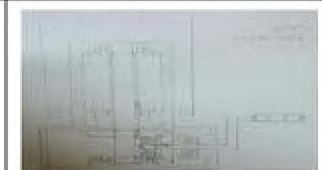
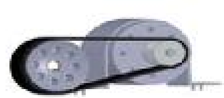
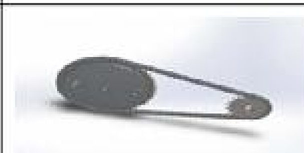
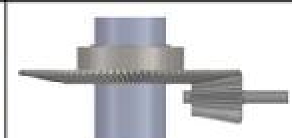

CE
SGS
JEWEL
PACKING

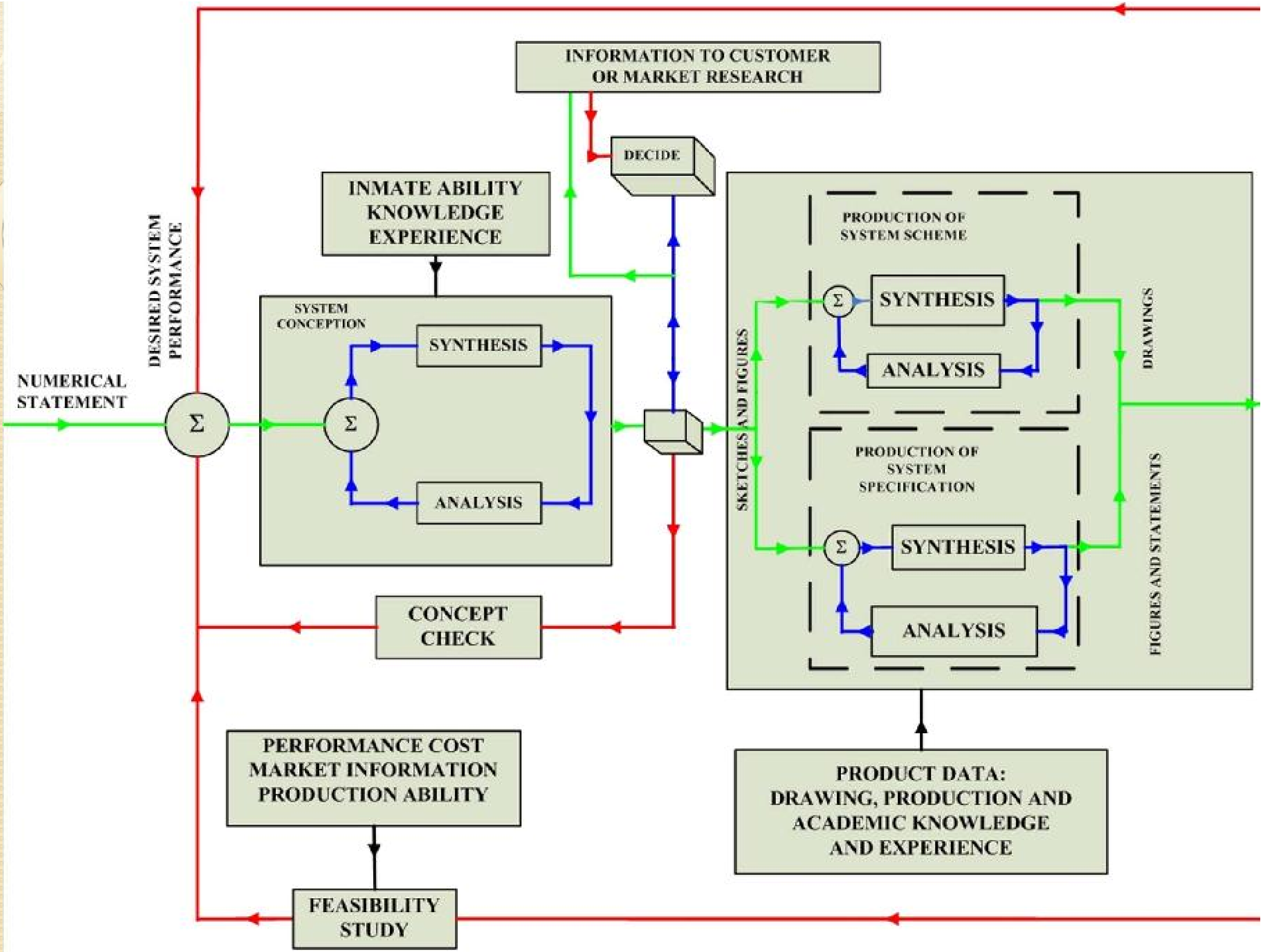

Audited Supplier

Example3.

Current solutions


جهاز رفع المعاقين
الية المسك
إلية المسك (A1)
إلية المسك (A2)
إلية المسك (A3)
إلية المسك (A4)
الحاملة
الحاملة (B1)
الحاملة (B2)
الحاملة (B3)
الحاملة (B4)
نظام الرفع
هيدروليكية
الهواء المضغوط
يدوية
الكهربائي
الهيكل
الهيكل (D1)
الهيكل (D2)
الهيكل (D3)
نظام التحريك
يدوي
كهربائي
نظام السيطرة
سلكي
لاسلكي
نقل القدرة
الحزام
التروس
السلاسل

Alternatives / Main features	Alternative 1	Alternative 2	Alternative 3	Alternative 4
ألية ألمسك				
الحامل				
ألية الرفع				
ألهيكل				
ألية ألتحريك				
التحكم				
نقل القدرة				



INFORMATION TO CUSTOMER OR MARKET RESEARCH
DECIDE
INMATE ABILITY KNOWLEDGE EXPERIENCE
DESIRED SYSTEM PERFORMANCE
NUMERICAL STATEMENT
Σ
SYSTEM CONCEPTION
SYNTHESIS
ANALYSIS
Σ
SKETCHES AND FIGURES
PRODUCTION OF SYSTEM SCHEME
SYNTHESIS
ANALYSIS
DRAWINGS
PRODUCTION OF SYSTEM SPECIFICATION
SYNTHESIS
ANALYSIS
FIGURES AND STATEMENTS
CONCEPT CHECK
PERFORMANCE COST MARKET INFORMATION PRODUCTION ABILITY
PRODUCT DATA: DRAWING, PRODUCTION AND ACADEMIC KNOWLEDGE AND EXPERIENCE
FEASIBILITY STUDY

Decision Making

Simple comparison

one is better or worse than the other.

Weighting Chart or Emphasis Curve

By weighting

put them in order of performance .

Σ GW Method

The Decision Network

By mathematical analysis and probability

its very rare in design.

Weighting Chart or Emphasis Curve

1. The features, (such as, weights, cost, power, wear), are given letters, A, B, C, D, etc, for easier compilation of a table.
2. These features are used for heading both the rows and the columns.
3. A diagonal is placed across the table, because (A) cannot be compound with (A), and so on of (B, C, and D).
4. Then (A) compared with (B, C, and D)
5. (B) is compared with (C and D)
6. (C) is compared with (D).
7. When the top half of the table completed, the number of times that letter (A), appears in the table marked at the right hand edge of the table against row (A), and this repeated for the others. (These values termed the performance values).
8. Now the weighting considered to take the best choice.
9. This method is simple and the designer decides the measure of importance by using whatever Aid he may find useful.
10. This method used when there is not detail information.

Alternatives / Alternatives	A	B	D	C
A		A	A	A
B			D	C
D				D
C				
Order of preference	1st	4th	2nd	3rd

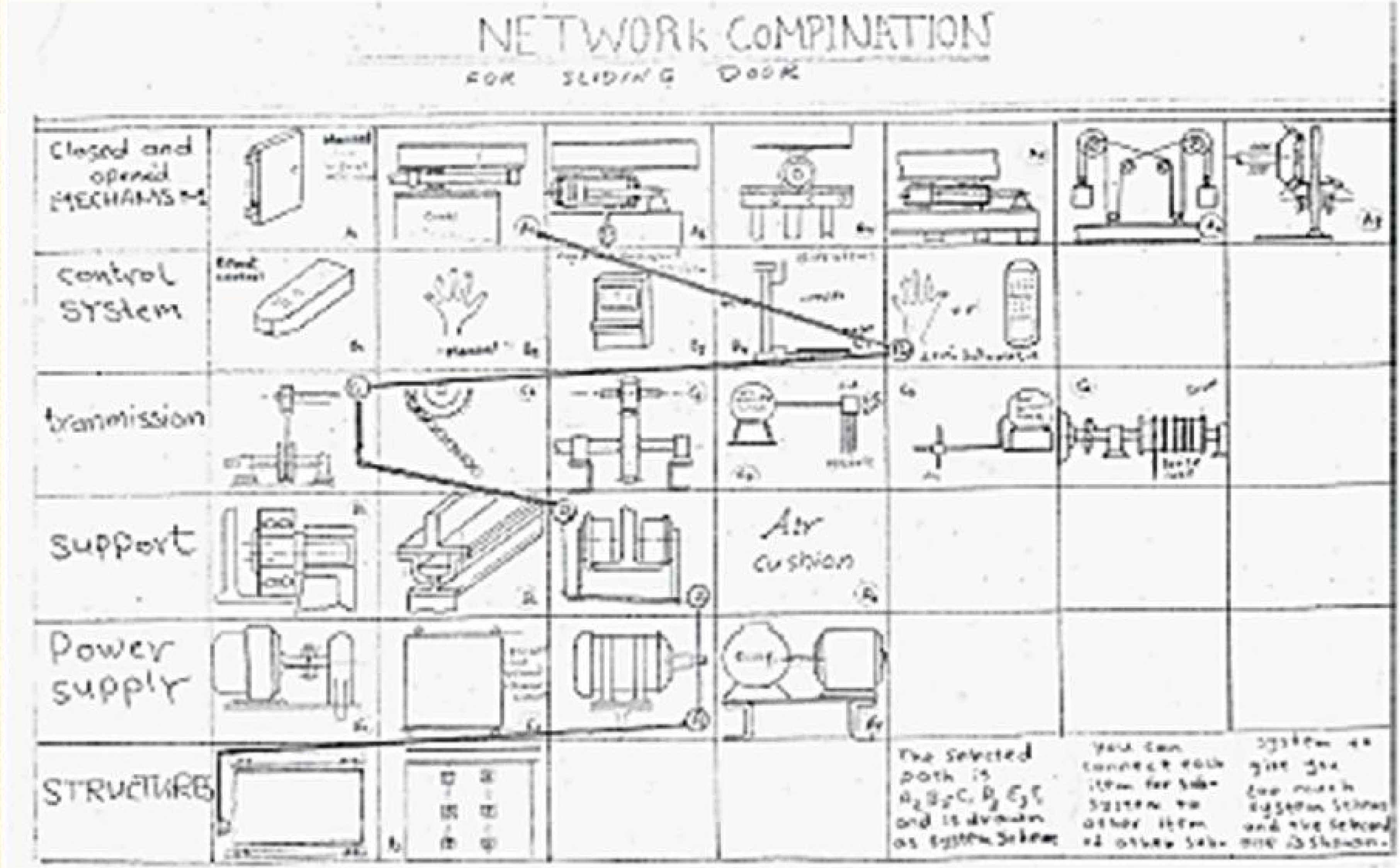
Σ GW Method

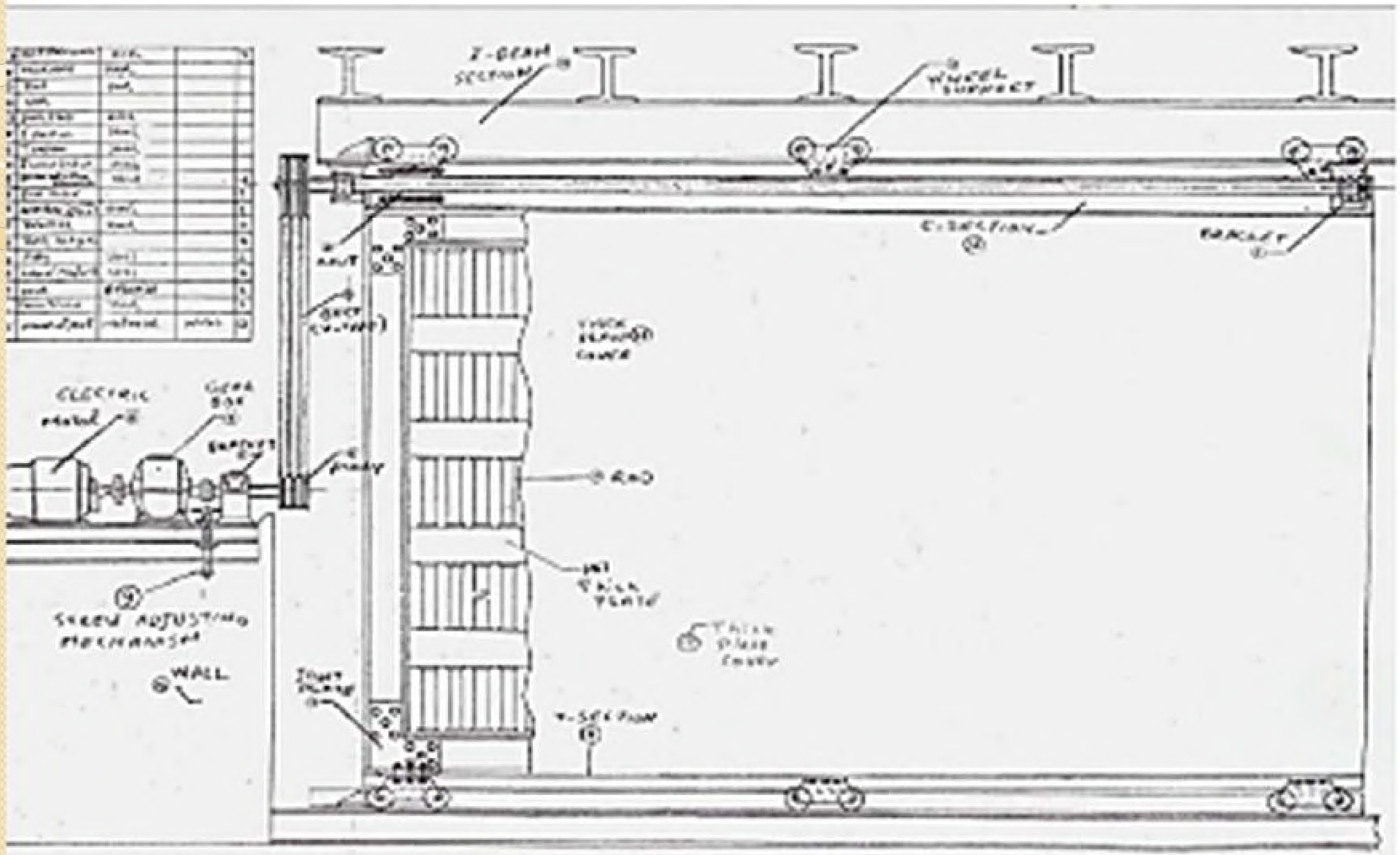
1. At the bottom of the table, we now have a measure of performance of the different designs when considered in relation to the chosen feature.
2. When the values at the bottom of the table differs by about 10%, then they should be considered to be equal at this stage, that's mean, the accurate of this method is about 90%.
3. The disadvantages of this method lie in difficulty in making the initial weighting decisions.
4. The other difficulty lies in the choice of features that will be considered in order to obtain a measure of performance.

Main features \ Alternatives	W	C1		C2		C3		C4	
		G	GW	G	GW	G	GW	G	GW
safety	20	14	280	16	320	17	340	11	220
weight	15	9	135	8	120	11	165	12	180
life	12	7	84	9	108	10	120	5	60
control	13	6	78	7	91	11	143	5	65
noise	15	5	75	6	90	12	180	13	195
Appearance	14	9	126	7	98	10	140	8	112
Total cost	10	5	50	6	60	8	80	9	90
size	11	4	44	5	55	7	77	8	88
performance	18	10	180	11	198	14	252	9	162
$\sum GW$		1052		1140		1497		1172	
Order of preference		4_{th}		3_{rd}		1_{st}		2_{nd}	

The Decision Network

1. This method involves a little from each of the methods above.
2. It is difficult to use, but it worth the effort.
3. The example below clarifies this method.
4. After doing the example, trace the parts from top to the bottom of the decision network.
5. Find the measure of performance.


NETWORK COMPINATION
FOR SLIDING DOOR
Closed and opened MECHANISM
control system
transmission
support
Air cushion
Power supply
STRUCTURE
The selected path is

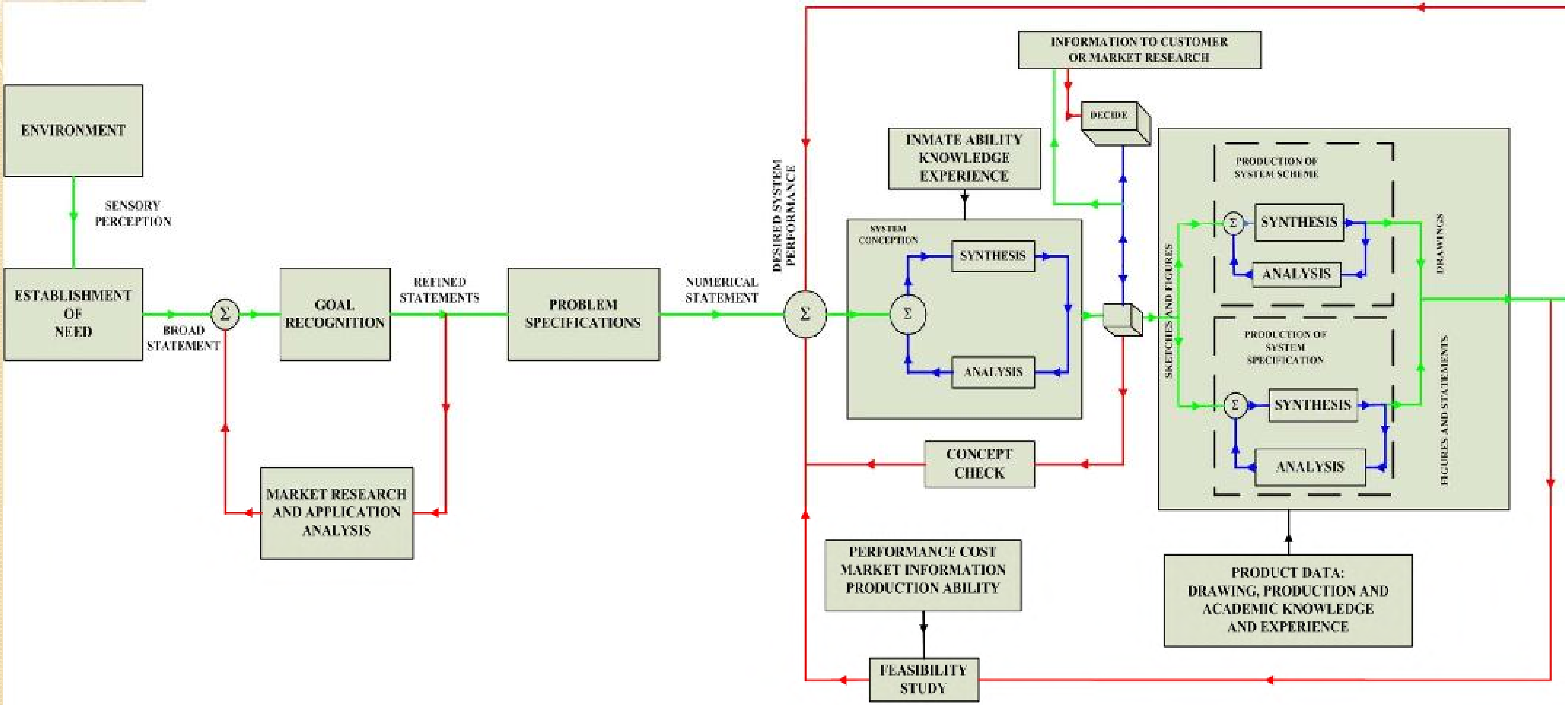


SYSTEM SCHEME FOR SELECTED PATH $A_1B_2C_1D_2E_1F_1$

Mechanical Engineering Design II

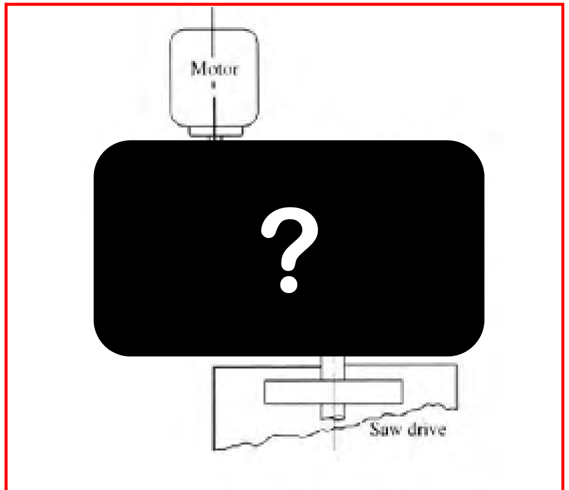
Eleventh & Twelfth Lectures

- **Production of system specification**
- **Production of system scheme**
- **Feasibility Study**



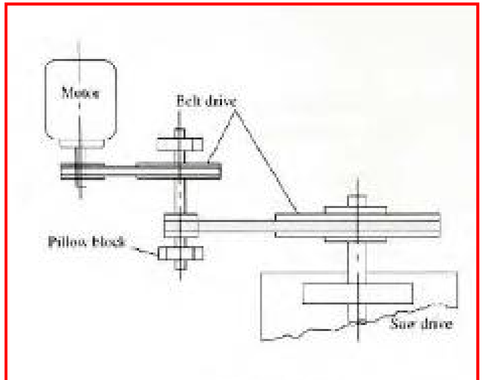
ENVIRONMENT
SENSORY PERCEPTION
ESTABLISHMENT OF NEED
BROAD STATEMENT
GOAL RECOGNITION
REFINED STATEMENTS
PROBLEM SPECIFICATIONS
NUMERICAL STATEMENT
MARKET RESEARCH AND APPLICATION ANALYSIS
DESIRED SYSTEM PERFORMANCE
INFORMATION TO CUSTOMER OR MARKET RESEARCH
DECIDE
INMATE ABILITY KNOWLEDGE EXPERIENCE
SYSTEM CONCEPTION
SYNTHESIS
ANALYSIS
CONCEPT CHECK
PRODUCTION OF SYSTEM SCHEME
SYNTHESIS
ANALYSIS
PRODUCTION OF SYSTEM SPECIFICATION
SYNTHESIS
ANALYSIS
SKETCHES AND FIGURES
DRAWINGS
FIGURES AND STATEMENTS
PERFORMANCE COST MARKET INFORMATION PRODUCTION ABILITY
FEASIBILITY STUDY
PRODUCT DATA: DRAWING, PRODUCTION AND ACADEMIC KNOWLEDGE AND EXPERIENCE

Power Transmission Problem

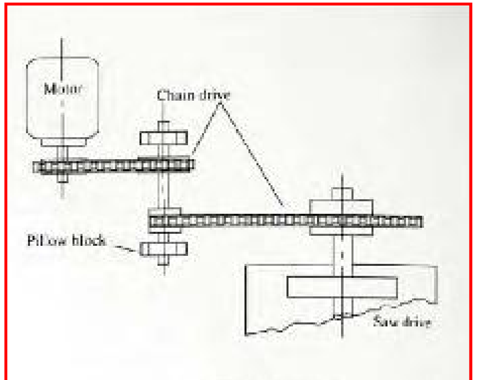


(c) Single-stage gear-reducer option

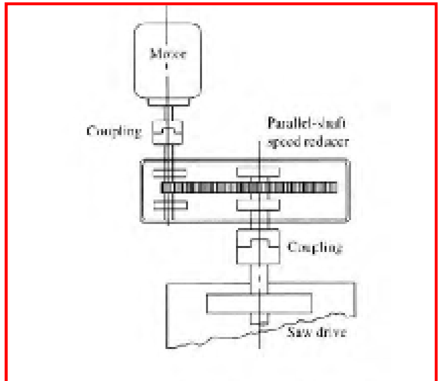
Alternative solutions



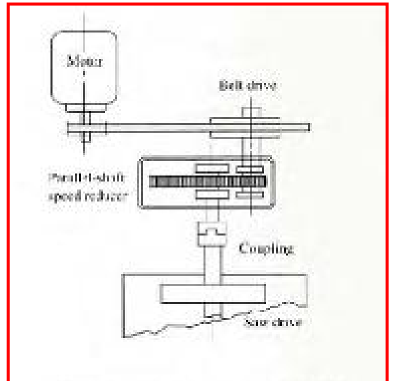
(a) Belt drive option



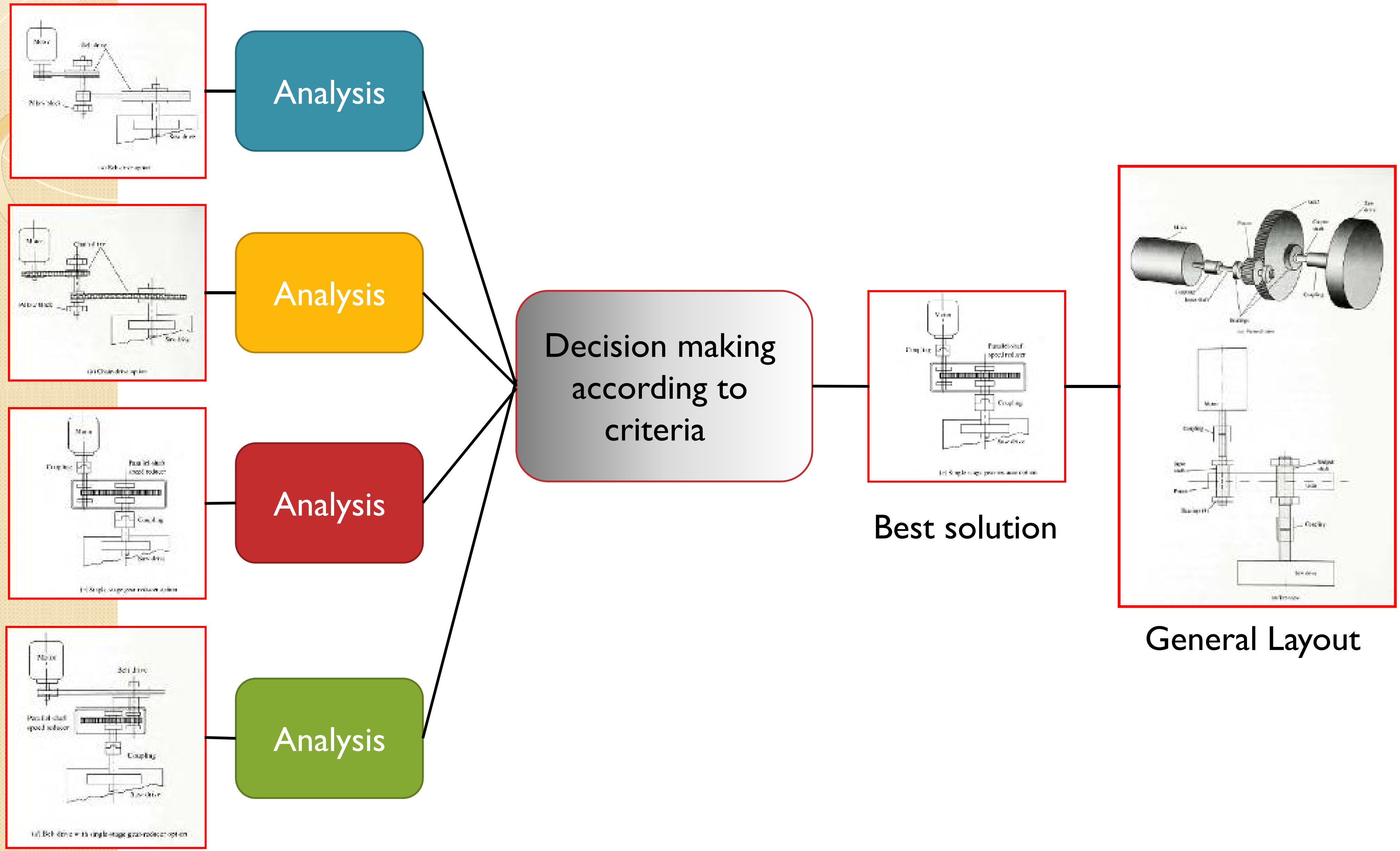
(b) Chain drive option

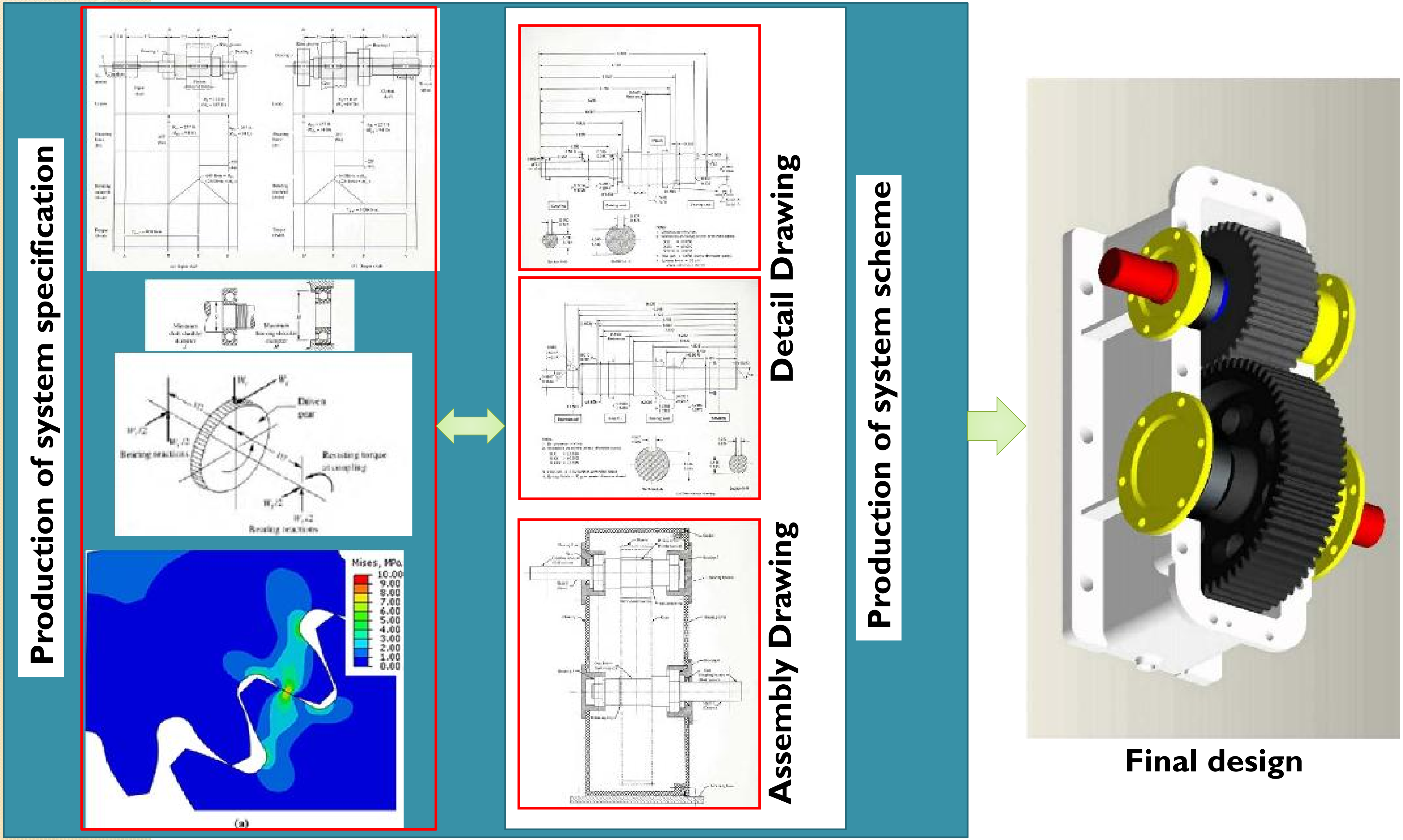


(c) Single-stage gear-reducer option

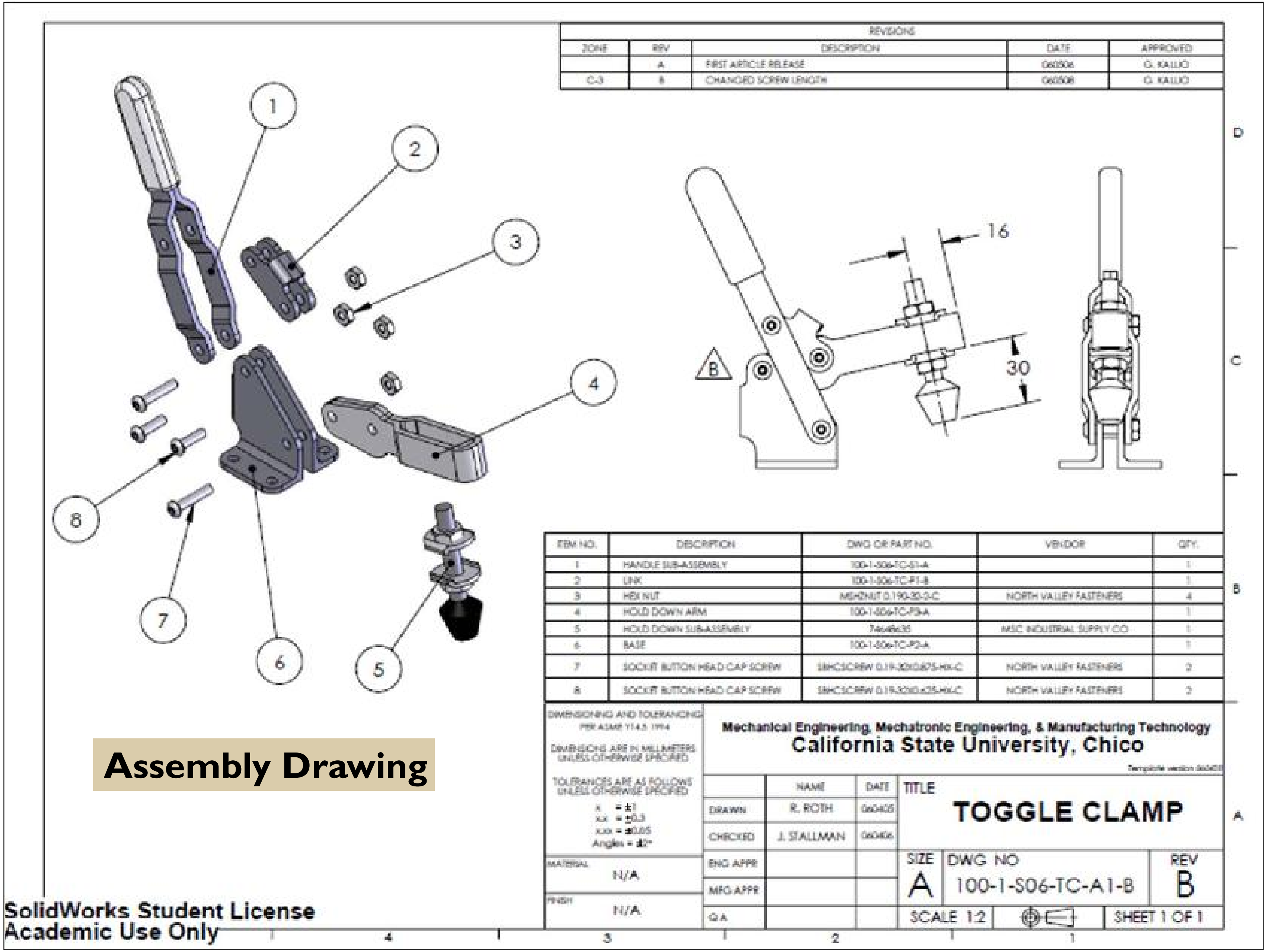


(d) Belt drive with single-stage gear-reducer option


Analysis
Analysis
Analysis
Analysis
Decision making according to criteria
Best solution
General Layout



Production of system specification
Driven gear
Bearing reactions
Resisting torque at coupling
Bearing reactions
Mises, MPa
10.00
9.00
8.00
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
(a)
Detail Drawing
Assembly Drawing
Production of system scheme
Final design

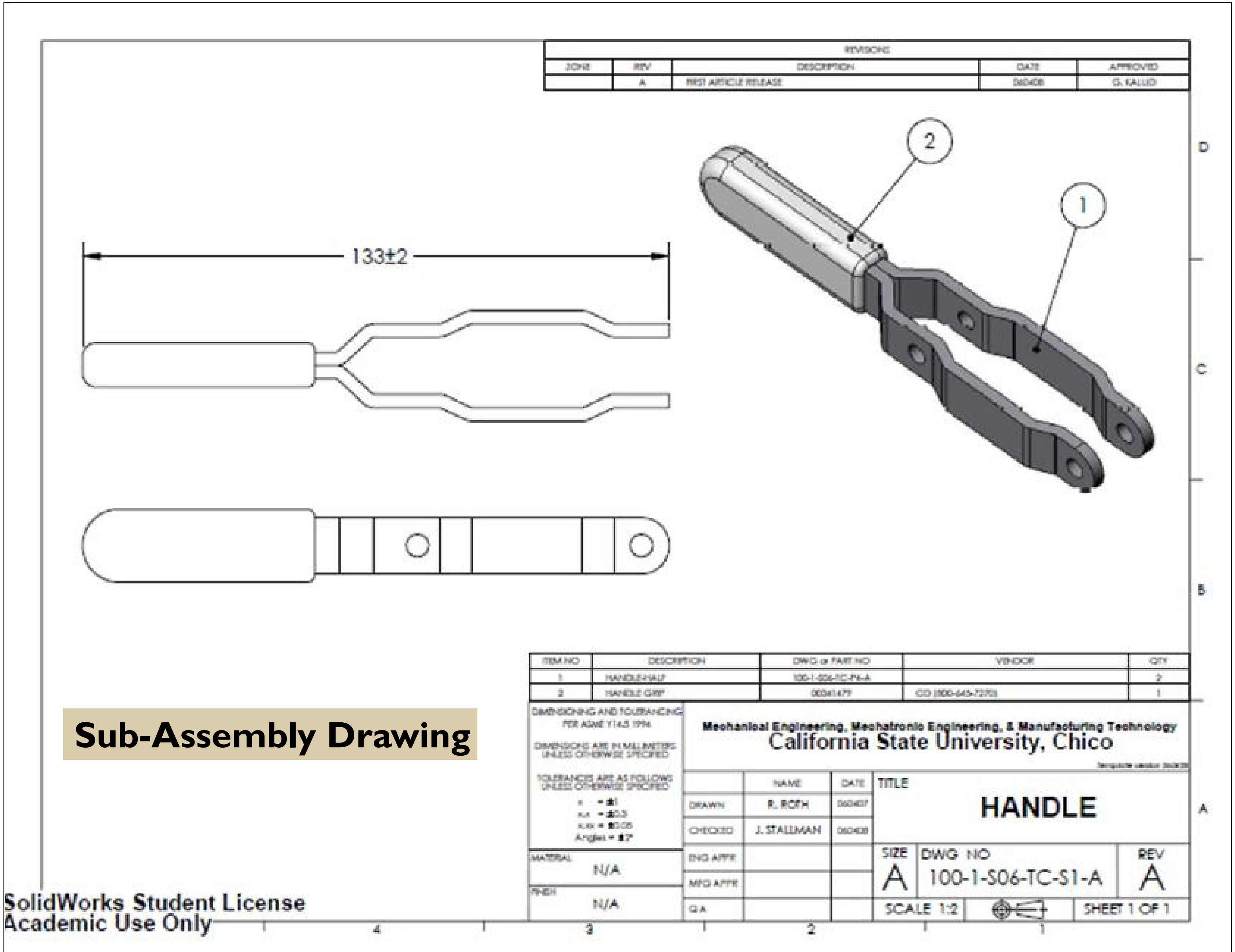


REVISIONS				
ZONE	REV	DESCRIPTION	DATE	APPROVED
	A	FIRST ARTICLE RELEASE	060506	G. KALLIO
C-3	B	CHANGED SCREW LENGTH	060508	G. KALLIO

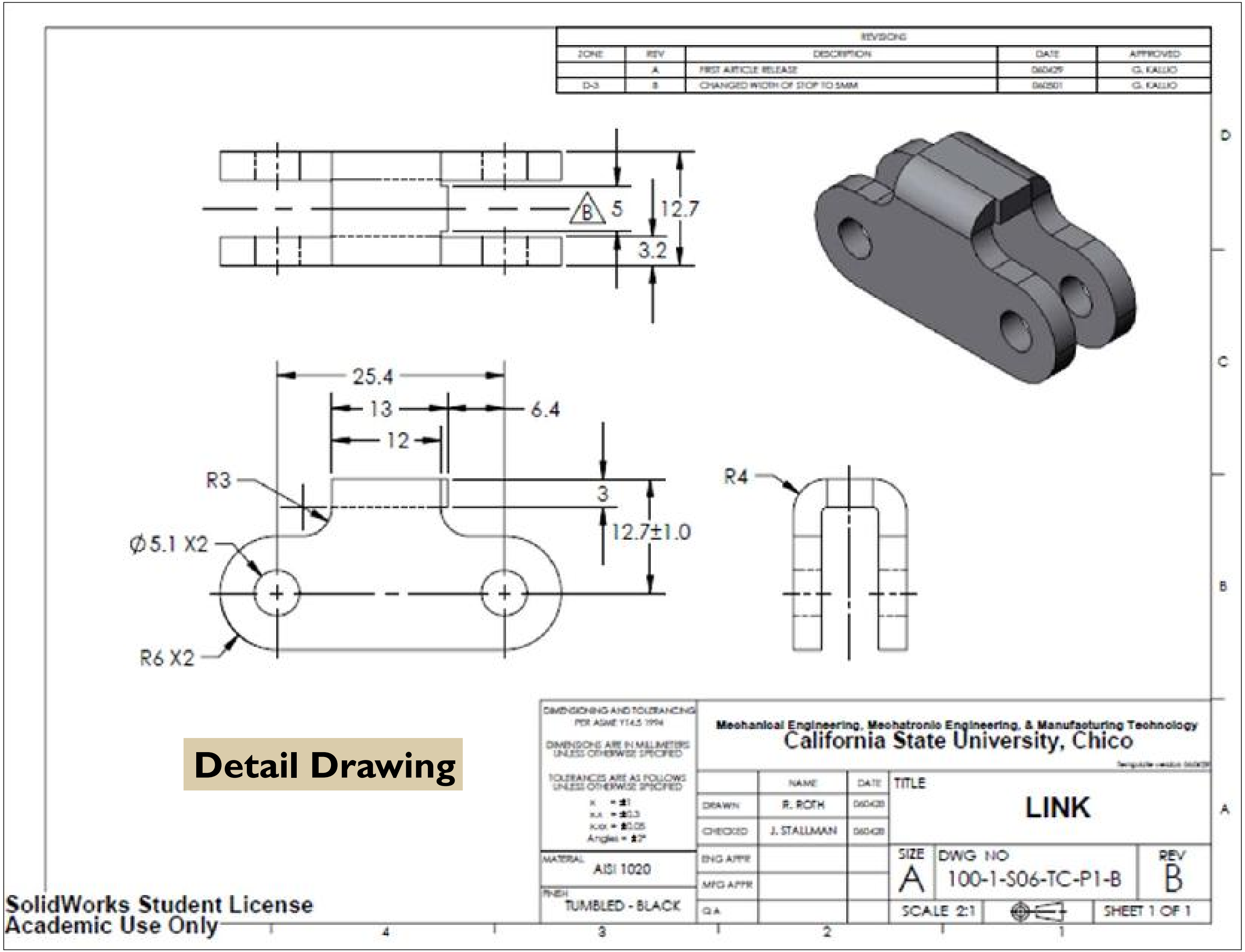
ITEM NO.	DESCRIPTION	DWG OR PART NO.	VENDOR	QTY.
1	HANDLE SUB-ASSEMBLY	100-1-S06-TC-S1-A		1
2	LINK	100-1-S06-TC-P1-B		1
3	HEX NUT	MSHZNUT 0.190-32-0-C	NORTH VALLEY FASTENERS	4
4	HOLD DOWN ARM	100-1-S06-TC-P3-A		1
5	HOLD DOWN SUB-ASSEMBLY	74648635	MSC INDUSTRIAL SUPPLY CO	1
6	BASE	100-1-S06-TC-P2-A		1
7	SOCKET BUTTON HEAD CAP SCREW	SBHCSCREW 0.19-32X0.875-HX-C	NORTH VALLEY FASTENERS	2
8	SOCKET BUTTON HEAD CAP SCREW	SBHCSCREW 0.19-32X0.625-HX-C	NORTH VALLEY FASTENERS	2

Assembly Drawing

Sub-Assembly Drawing



Detail Drawing



The Feasibility Study

(Design Acceptance or Design Practicality)

- The words “the feasibility study” are used to describe the analysis of the design which attempts to take into account all relevant facts. The outcome of the feasibility study is a major factor in the decision to continue with the design.
- It is not suggested that feasibility checking cannot begin until the system scheme and system specification are produced however, since this is the first point in the design process where the indicate the final form of the system, it is the most logical point at which to check that the system can be made economically and will perform reasonably well in the desired manner. (Any study carried out at an earlier stage could be termed a Plausibility Study).

During the feasibility study, the following points should be checked:

1. **Is the system compatible with the basic physical laws? (Mainly conservation laws).**
2. **Is the system compatible as whole with its environment? In those cases where compatibility is relevant, are all parts of the system compatible with each other, and with adjacent parts of the environment?**
3. **Is the system economically feasible from the manufacturer's points of view?**
4. **Is the system feasible as regards its market?**
5. **Are all the properties of the system acceptable with the terms of the original design specification? Consider the follow-up of properties not mentioned in the design specification.**
6. **Is this particular design no better than previous existing designs? Compare and contrast. Consider the cost of covering features which are additional to those on existing equipment.**
7. **will the time taken to manufacture – test – develop and transport be reasonable? Critical path analysis can be used to highlight difficult regions.**
8. **Does the design demand new manufacturing techniques within the manufacturer's organization, or does the design demand completely new manufacturing techniques to make it economically feasible?**
9. **Is the system considered to be reliable enough for successful operation, or should special provisions be made to improve reliability.**
10. **Can the system be readily maintained, or is the design such that the minimum of maintenance is required?**
11. **Is the system legally saleable? Have any patent rights been violated?**

Mechanical Engineering Design II

Thirteenth Lecture

Design-Reliability and Failure

Design-Reliability and Failure

Failure of a design when in service costs money for replacement, and often the failure itself causes other financial looses. On the other hand, to design so that the chance of failure is very remote also costs money.

It can therefore be seen that the designer should consider failure of the design. The optimum design should be a compromise, that is, it is optimum only for the particular criteria relevant to that design.

It is known that a considerable amount of mechanical design is associated with very small quantities, and in a number of cases only one system is manufactured. In most designs the only items of equipment in frequent use are nuts and bolts. For this reason statistical methods which are often used to analyze failure and improve reliability are only of use in specialized mechanical engineering design, such as mass-produced car design.

The mechanical engineer has devised a number of methods to overcome the problem of lack of numbers, some of which will be mentioned here. These methods are also used by other designers, often in addition to statistical methods.

1. To make systems more reliable, use accurate methods of design analysis coupled with realistic assessment of loads.

e.g. (a) Use adequate checking procedures.

(b) Use more accurate thick cylinder theory rather than the simpler and quicker theory in relevant cases.

(c) Use fatigue theory rather than arbitrary factors when considering cyclic loads.

(d) Assess loads realistically by dynamic analysis and careful thought rather than by arbitrary load factors.

2. If difficulty is encountered in putting note.1 into practice, then use generous reserve factors to take account of ignorance. Make use of rerating, (i.e. working at much reduced stress levels).

3. Remember that evolutionary design is in general more reliable than revolutionary design.

4. Make the design as simple as possible. The less parts there are in a piece of equipment, then the less chances of failure there are.

5. Use standard parts and materials of known and proven reliability.
6. Use British Standards. Also, use any other engineering standards and Codes of Practice which are known to improve reliability.
7. Build in redundancy (here interpreted in the same way as redundancy in beam analysis) and analyze as though the redundancy were missing. This is very similar to note.2 above, but it can be dangerous to use this method without considerable experience.
8. Design so that if one very important part should fail and give a catastrophic result, then another part or system takes over temporarily. It is better if the second system operates on a different principle to the first. Repair is executed as soon as possible.
9. Design to take overloads on the system by designing so that they cause visible distortion, but not catastrophic failure. Repair is executed as soon as possible.
10. If further functioning would be likely to cause damage to the system or its surroundings, incorporate devices or design features into the system so that failure causes the system to stop functioning. Take care that the sudden cessation of functioning does not itself cause troubles. Repair is executed as soon as possible.
11. Use composite structures so that failure of any one part, or a few parts, does not cause complete failure. Repair is executed as soon as possible.
12. Include devices that relieve the overload, but allow normal use to continue.

13. Incorporate devices or design features so that incorrect assembly and use is preferably impossible. If incorrect assembly is possible, then there should be features which prevent normal operation.
14. Incorporate warning devices so that malfunctioning is obvious. Make warning devices so that the malfunctioning of the warning device itself is obvious.
15. Prepare adequate test specifications to ensure desired reliability.
16. Although it is included under the heading of note.6, ensure that electrochemical corrosion possibilities are adequately studied.
17. Consider special features relating to human safety if failure should occur.
18. Supply adequate information to all persons directly or remotely concerned with the equipment.
19. Set up adequate lines of communications and records to record and collate all information relating to failures.

Certain of the notes above also relate to (failsafe design), that is the design of equipment that will revert to a safe condition if a failure should occur.

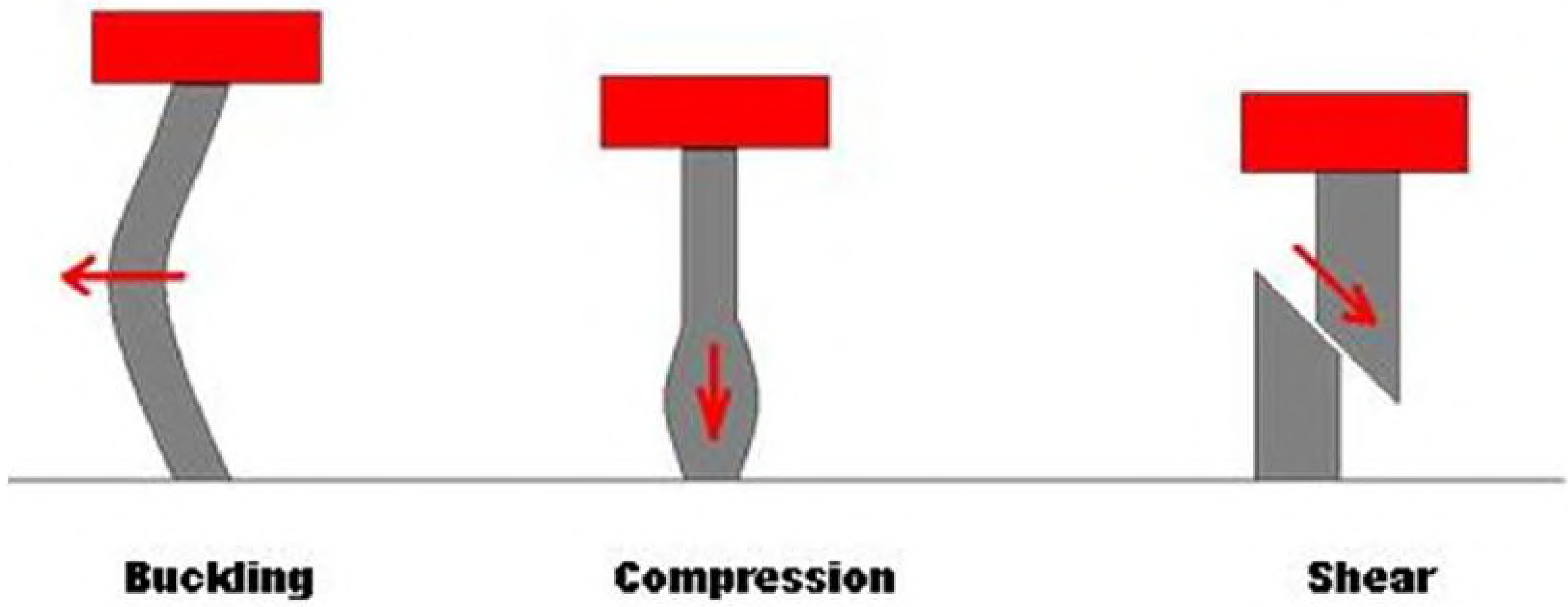
Mechanical Engineering Design II

Fourteenth & Fifteenth Lectures

Columns Design

Columns Design

A column is a long, slender member that carries an axial compressive load and that fails due to buckling rather than due to failure of the material of the column.



Failure due to buckling



Analysis of Columns

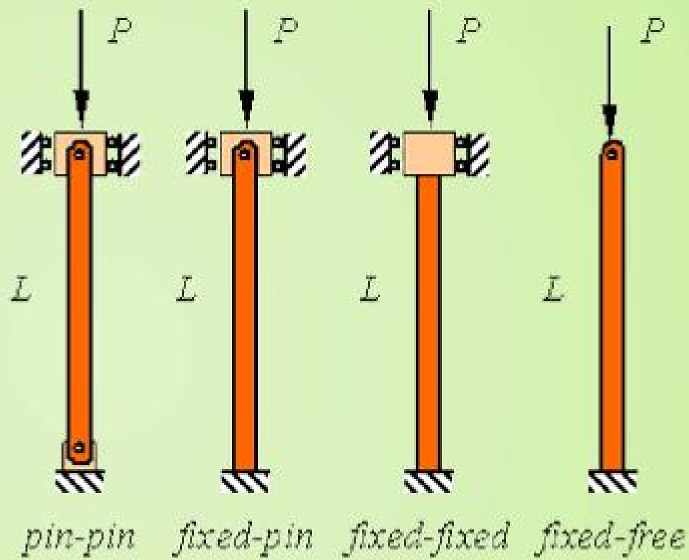
Properties of C.S

1. Cross Sectional Area (A)
2. Moment of Inertia (I)
3. Radius of gyration $r = \sqrt{\frac{I}{A}}$

Materials

1. Modulus of elasticity (E)
2. Yield strength (S_y)

Connections



Theoretical value	K=1	K=0.7	K=0.5	K=2
Practical value	K=1	K=0.8	K=0.65	K=2.1

Column type

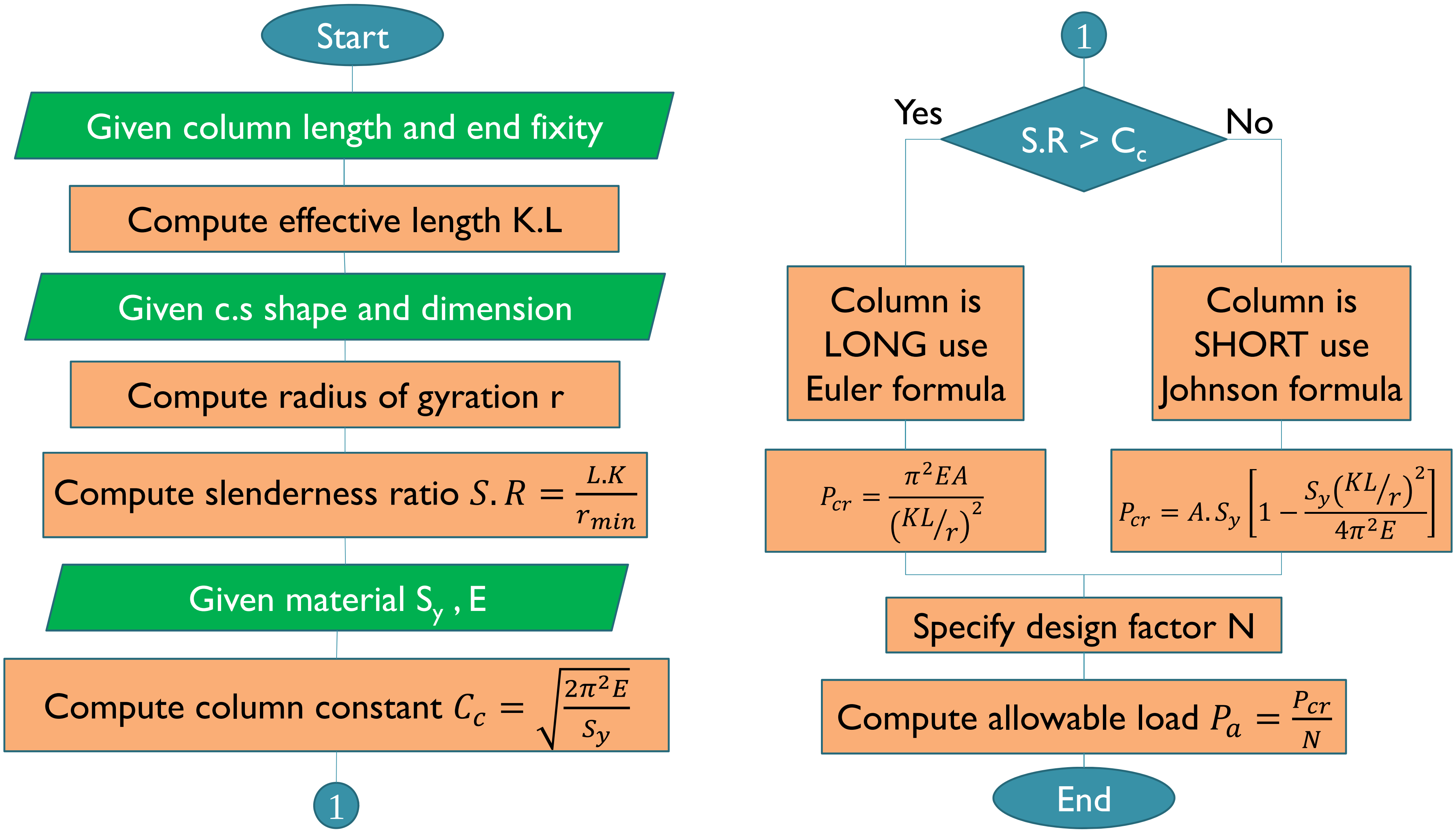
1. Effective length $L_e = K.L$
2. Slenderness ratio $S.R = \frac{L_e}{r_{min}} = \frac{L.K}{r_{min}}$
3. Column constant $C_c = \sqrt{\frac{2\pi^2 E}{S_y}}$

   (Transition slenderness ratio)

If S.R > C_c then column is LONG

If S.R < C_c then column is SHORT

Flowchart for analyzing of straight centrally loaded column:



Example.1 : A column has a solid circular cross section, 31.75 mm in diameter; it has a length of 1.3716 m and is pinned at both ends. If it is made from AISI 1020 cold-drawn steel, what would be a safe column loading?

Solution: use the flowchart above.

Results :

Step.1: $K = 1 \qquad KL = 1 \times 1.3716 = 1.3716mm$

Step.2: for solid round section $r = \frac{D}{4} = \frac{31.75}{4} = 7.9375mm$

Step.3: compute slenderness ratio $\frac{KL}{r} = \frac{1.3716}{7.9375} = 173$

Step.4: compute the column constant $C_c = \sqrt{\frac{2\pi^2 E}{S_y}} = \sqrt{\frac{2\pi^2(207Gpa)}{350Mpa}} = 108$

For material AISI 1020 cold-drawn steel:
E=207 Gpa
S_y=350 Mpa

Step.5: because $\frac{KL}{r} > C_c$ column is long , Eulers formula should be used :

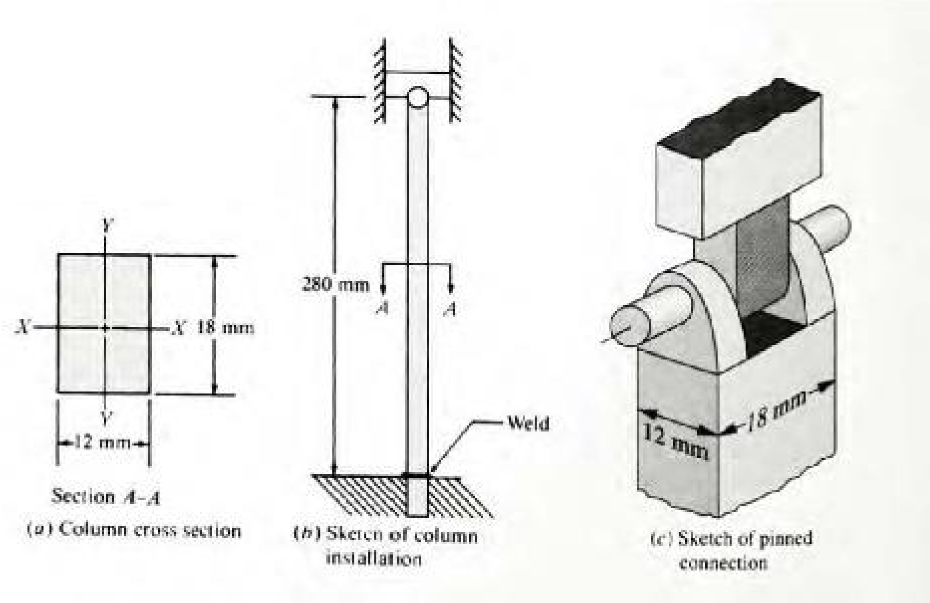
$A = \frac{\pi D^2}{4} = \frac{\pi(31.75)^2}{4} = 793.596\ mm^2$

The critical load is $P_{cr} = \frac{\pi^2 EA}{\left({KL}/{r}\right)^2} = \frac{\pi^2(207Gpa)(793.596\ mm^2)}{(173)^2} = 54265.6\ N$

At this load the column just begin to buckle, now let N=3

Allowable load $P_a = \frac{P_{cr}}{N} = \frac{54265.6}{3} = 18090.016\ N\ \ (the\ safe\ load\ on\ the\ column)$

Example .2: Determine the critical load for:
Mat. AISI 1040 hot rolled steel
lower end is welded and upper was pinned



Solution: use the flowchart above.

Results :

Step.1: $K = 0.8 \quad KL = 1 \times 1.3716 = 1.3716mm$

Step.2: for solid rectangular section $r = \frac{B}{\sqrt{12}}$

$$r = \frac{12}{\sqrt{12}} = 3.46\ mm$$

Step.3: compute slenderness ratio $\frac{KL}{r} = \frac{0.8\times280}{3.46} = 64.7$

Step.4: compute the column constant $C_c = \sqrt{\frac{2\pi^2 E}{S_y}} = \sqrt{\frac{2\pi^2(207Gpa)}{290Mpa}} = 119$

For material AISI 1040 hot rolled steel:
E=207 Gpa
S_y=290 Mpa

Step.5: because $\frac{KL}{r} < C_c$ column is short, J.B.Johnson formula should be used :

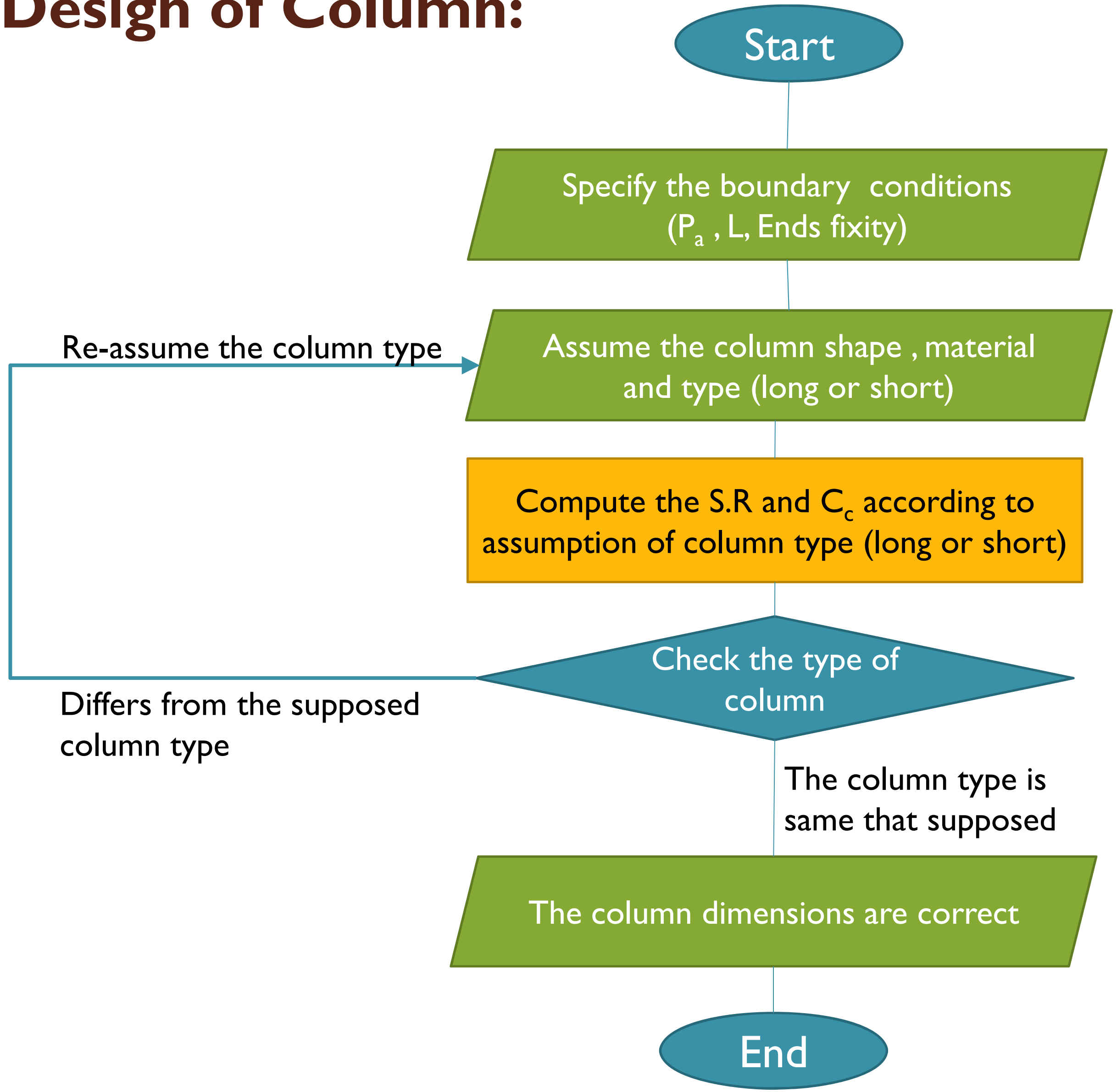
$A = 12 \times 18 = 216\ mm^2$

The critical load is $P_{cr} = A.S_y\left[1 - \frac{S_y(KL/r)^2}{4\pi^2 E}\right] = 53.3\ kN$

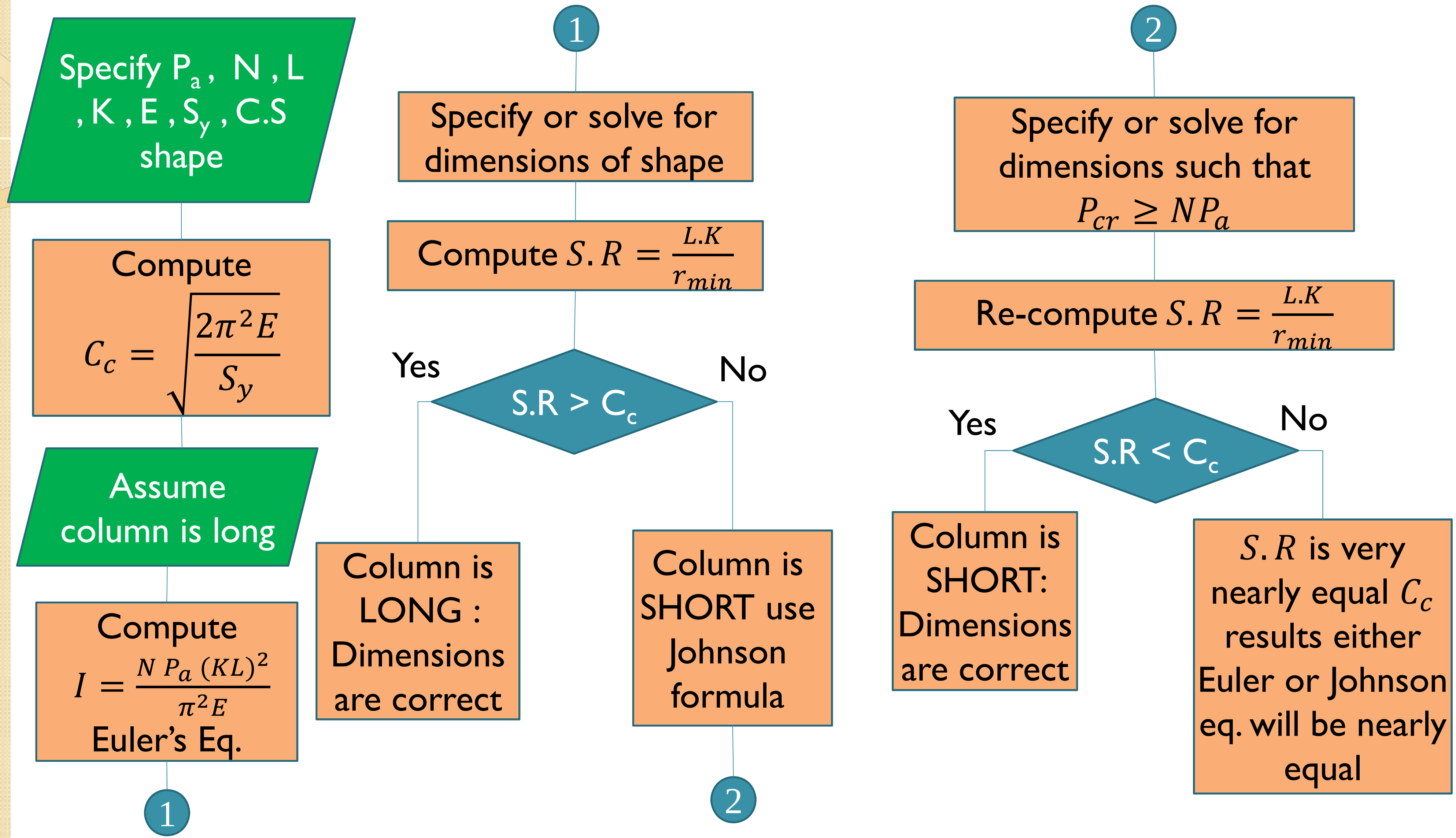
At this load the column just begin to buckle, now let N=3

Allowable load $P_a = \frac{P_{cr}}{N} = \frac{53.3}{3} = 17.8\ kN\ \ (the\ safe\ load\ on\ the\ column)$

The Design of Column:



Flowchart for designing of straight centrally loaded column:



Design assuming a long column (Euler's Eq.)

$$I = \frac{P_{cr}\,(KL)^2}{\pi^2 E} = \frac{N\,P_a\,(KL)^2}{\pi^2 E}$$

For solid circular section $I = \frac{\pi D^4}{64}$

$$\therefore\ D = \left[\frac{64\,N\,P_a (KL)^2}{\pi^2 E}\right]^{1/4}$$

Design assuming a short column (Johnson's Eq.)

$$P_{cr} = A.S_y\left[1 - \frac{S_y\left({KL}/{r}\right)^2}{4\pi^2 E}\right]$$

For solid circular section $\mathrm{A} = \frac{\pi D^2}{4}$, $r = \frac{D}{4}$

$$\therefore\ D = \left[\frac{4NP_a}{\pi S_y} + \frac{4\,S_y (KL)^2}{\pi^2 E}\right]^{1/2}$$

Example.3 : specify a suitable dia. of solid, round cross section for a machine link if it carry 43590.4 N of axial compressive load. L=635mm , ends will be pinned, N=3 , Mat. AISI 1020 hot-rolled steel.

Analysis: * use the flowchart shown before.
* Assume the column is LONG.

Results: $D = \left[\frac{64\ N\ P_a (KL)^2}{\pi^2 E}\right]^{1/4} = \left[\frac{64 \times 3 \times 43590.4\ (635mm)^2}{\pi^2 (207 \times 10^9)}\right]^{1/4} = 26.924mm$

$\therefore r = \frac{D}{4} = 6.731mm\ ,\ S.R = \frac{L.K}{r_{min}} = \frac{635 \times 1}{6.731} = 94.3$ and C_s=138 $\therefore S.R < C_s$

Column is redesign as short column by using Johnson's eq.:

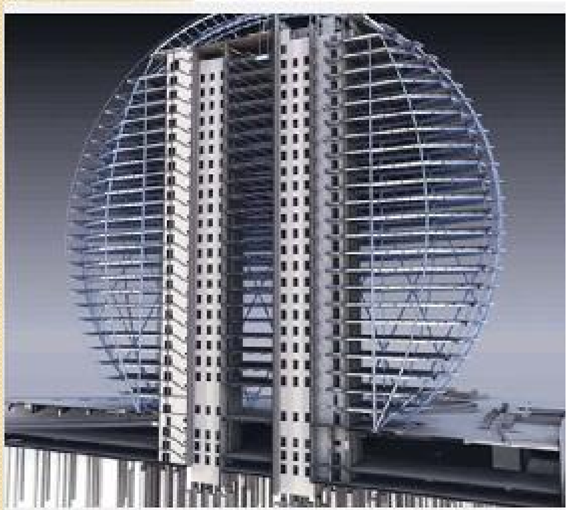
$$D = \left[\frac{4NP_a}{\pi S_y} + \frac{4\ S_y (KL)^2}{\pi^2 E}\right]^{1/2} = \left[\frac{4(3)(43590.4)}{\pi(206.85MPa)} + \frac{4\ (206.85MPa)(635mm)^2}{\pi^2 (207 \times 10^9)}\right]^{1/2} = 31.242mm$$

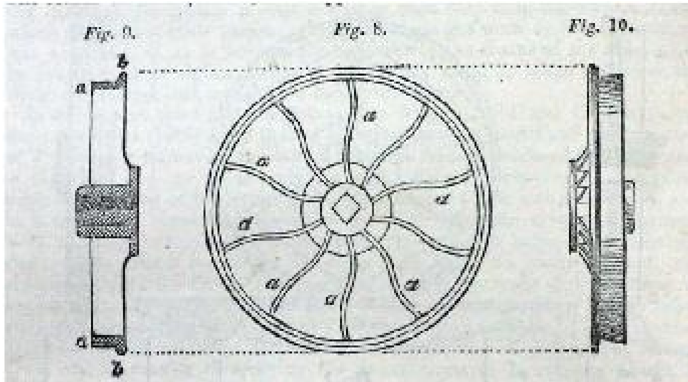
Checking the S.R again, we have : $S.R = \frac{L.K}{r_{min}} = \frac{635 \times 1}{31.242/4} = 81.3$

Comments: this is still lass than C_s , then our analyzing is acceptable .

Crooked Column:

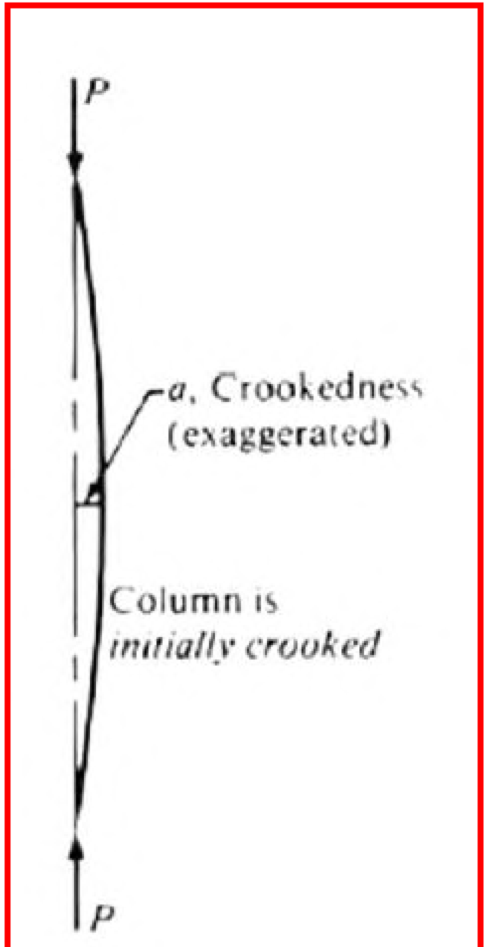
The columns shown in figures below are crooked, bending occurs in addition to the column action.





The Euler and Johnson formulas assume that the column is straight and that the load acts in line with the centroid of the cross section of the column. So the crooked column has special formula:

$$P_a^{\,2} - \frac{1}{N}\left[S_y A + \left(1 + \frac{ac}{r^2}\right) P_{cr}\right] P_a + \frac{S_y . A . P_{cr}}{N^2} = 0$$



Where a = initial crookedness.
c = distance from the neutral axis of C.S about which bending occurs to its outer edge.

The equation can be written as :

$$P_a^{\,2} + C_1 P_a + C_2 = 0 \qquad and \qquad P_a = 0.5\left[-C_1 - \sqrt{C_1^{\,2} - 4C_2}\right]$$

The smaller of the two possible solutions is selected.

Example.4 : A column has both ends pinned and has a length of 812.8mm. It has a circular cross section with a diameter of 19.05mm and an initial crookedness of 3.175mm. The material is AISI 1040 hot-rolled steel. Compute the allowable load for a design factor of 3.

Analysis: use equation above to evaluate C_1 and C_2 then find P_a

Results: $S_y = 289590\ KPa$

$$A = \frac{\pi D^2}{4} = 285.1784mm^2$$

$$r = \frac{D}{4} = 4.7752mm \quad \text{and} \quad c = \frac{D}{2} = 9.525mm$$

$$\frac{KL}{r} = 171 \quad \text{and} \quad P_{cr} = \frac{\pi^2 EA}{\left({}^{KL}/_{r}\right)^2} = \frac{\pi^2 (207Gpa)(285.1784\ mm^2)}{(171)^2} = 19909.25N$$

$$C_1 = -\frac{1}{N}\left[S_y A + \left(1 + \frac{ac}{r^2}\right) P_{cr}\right] = -42969 \qquad C_2 = \frac{S_y.A.P_{cr}}{N^2} = 182.7 \times 10^6$$

$$\therefore The\ eq.\ is: \quad {P_a}^2 - 42969P_a + 182.7 \times 10^6 = 0$$

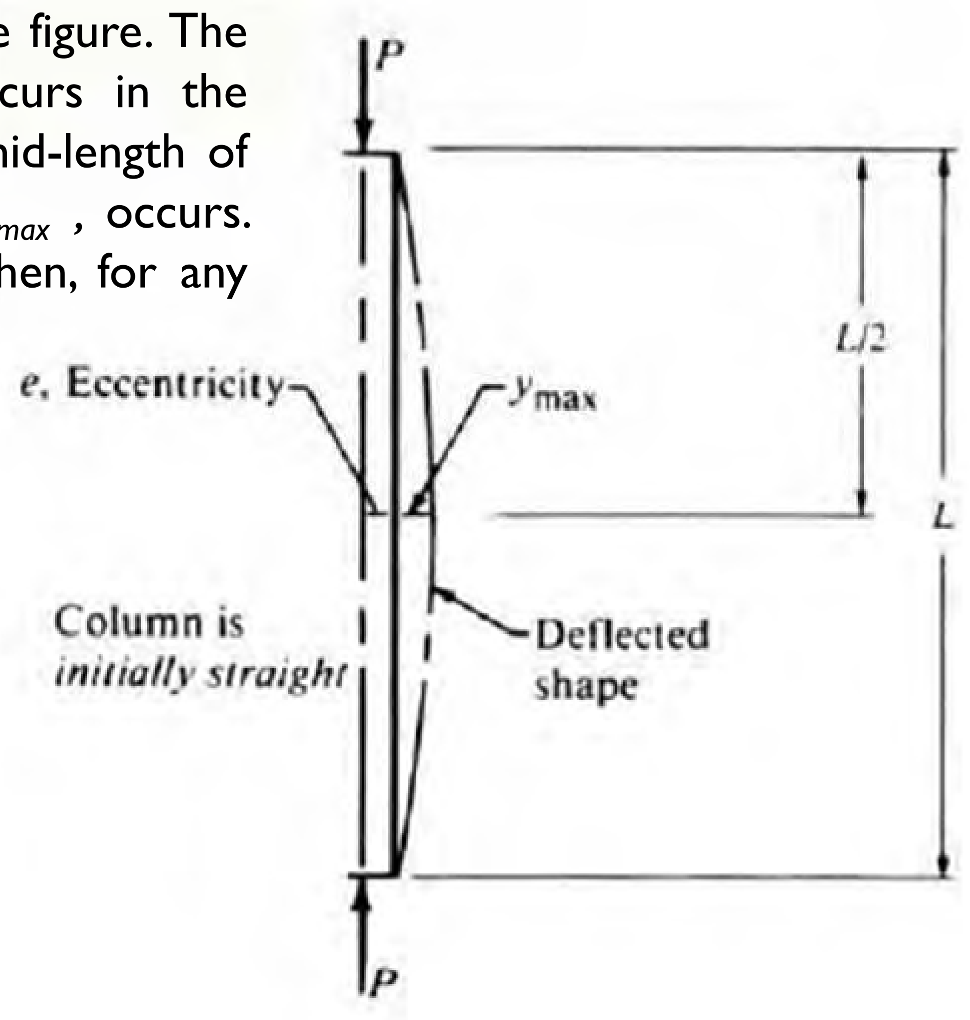
$$\therefore P_a = 4784.7\ N\ \ is\ the\ allowable\ load.$$

Note: This solution process is most accurate for long column.

Eccentrically Loaded Columns:

An eccentric load is one that applied away from the centroidal axis of the c.s of column as shown in the following figures.

Such a load exerts bending in addition to the column action that results in the deflected shape shown in the figure. The maximum stress in the deflected column occurs in the outermost fibers of the cross section at the mid-length of the column where the maximum deflection, y_{max} , occurs. Let's denote the stress at this point as $\sigma_{L/2}$ Then, for any applied load, *P*,



$$\sigma_{L/2} = \frac{P}{A}\left[1 + \frac{ec}{r^2}.\sec\left(\frac{KL}{2r}\sqrt{\frac{P}{AE}}\right)\right]$$

When max. stress is equal yield stress S_y

$$\therefore\ S_y = \frac{P_y}{A}\left[1 + \frac{ec}{r^2}.\sec\left(\frac{KL}{2r}\sqrt{\frac{P_y}{AE}}\right)\right]$$

But $P_a = \frac{P_y}{N}$

$$\therefore Required \;\; S_y = \frac{NP_a}{A}\left[1 + \frac{ec}{r^2}.\sec\left(\frac{KL}{2r}\sqrt{\frac{NP_a}{AE}}\right)\right]$$

* The above eq. cannot be solved for either A or P_a , therefore an iterative solution is required as will be demonstrated in the example.6 .

* Another critical factor may be amount of deflection of the axis of the column due to the eccentric load.

$$y_{max} = e\left[\sec\left(\frac{KL}{2r}\sqrt{\frac{P}{AE}}\right) - 1\right]$$

Example.5 : For the column of Example.4, compute the maximum stress and deflection if a load of 4781.6N is applied with an eccentricity of 19.05mm. The column is initially straight.

Given: e=19.05mm , D=19.05mm , L=812.8mm , both ends pinned, KL=812.8mm
r= 4.7752mm , c=D/2=9.525mm , Mat. AISI 1040 hot-rolled steel

Results: $$\sigma_{L/2} = \frac{4781.6}{272.25}\left[1 + \frac{19.05\times 9.525}{(4.7752)^2}.\sec\left(\frac{812.8}{2(4.7752)}\sqrt{\frac{4781.6}{285.1784\times 206.85\times 10^9}}\right)\right]$$

=202023.5 Kpa

$$y_{max} = 19.05\left[\sec\left(\frac{812.8}{2(4.7752)}\sqrt{\frac{4781.6}{285.1784\times 206.85\times 10^6}}\right) - 1\right]$$

=7.4022mm

Example.6 : The stress in the column found in Example.5 seems high for the AISI 1040 hot-rolled .steel. Redesign the column to achieve a design factor of at least 3.

Results: * S_y for AISI 1040 HR to be 289590 Kpa

* If we choose to retain same material. The c.s of the column must increased to decrease stress.

* The objective is to find suitable value for A , c , and r such that P_a=4781.6N N=3 & L_e=8128mm & e=19.05mm and $S < S_y$.

* The original design D=19.05mm, Let us try D=25.4mm then:

$$A = \frac{\pi D^2}{4} = 506.482mm^2\,, r = \frac{D}{4} = 6.35mm\,, r^2 = 40.325mm^2 \;\&\; c = \frac{D}{2} = 12.7mm$$

$$S = \frac{3 \times 4781.6}{506.45}\left[1 + \frac{19.05 \times 12.7}{40.325}.\sec\left(\frac{812.8}{2(6.35)}\sqrt{\frac{3 \times 4781.6}{506.482 \times 207 \times 10^9}}\right)\right]$$

= 260217.3 Kpa < Required value for S_y then we have satisfactory results

$$y_{max} = 19.05\left[\sec\left(\frac{812.8}{2(6.35)}\sqrt{\frac{4781.6}{506.482 \times 207 \times 10^9}}\right) - 1\right] = 1.9304\ mm$$

$\therefore$ *dia assumed is satisfactory*

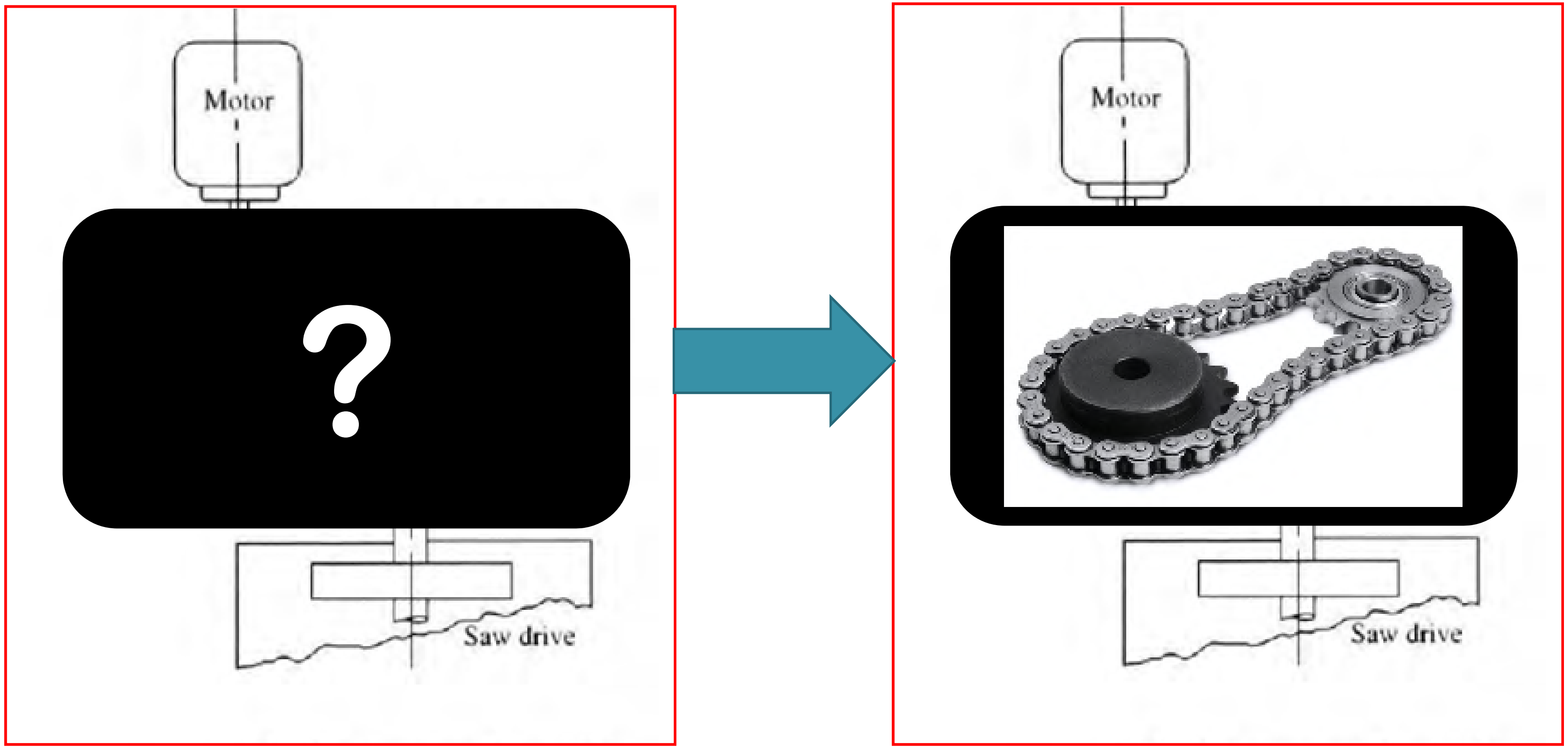
Mechanical Engineering Design II

Sixteenth Lecture

Design of Chain Drive

Power Transmission Problem

Proposed solution (Chain Drive)



Transmitted Power is Known
Input and output range speed is Known

Flowchart for designing a chain drive:

Transmitted Power , Input and Output range speed

Specify a service factor from Table 7-8 page.290 (306Pdf)

TABLE 7–8 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; light, uniformly loaded conveyors)	1.0	1.0	1.2
Moderate shock (machine tools, cranes, heavy conveyors, food mixers and grinders)	1.2	1.3	1.4
Heavy shock (punch presses, hammer mills, reciprocating conveyors, rolling mill drive)	1.4	1.5	1.7

1

1

COMPUTE THE DESIGN POWER

Design Power=service factor x transmitted power

COMPUTE THE DESIRED SPEED RATIO

Ratio=Input Speed / Middle Output Speed

Specify the standard chain size from Table 7-4 page.284 (300 Pdf)

TABLE 7–4 Roller chain sizes

Chain number	Pitch (in)	Roller diameter	Roller width	Link plate thickness	Average tensile strength (lb)
25	1/4	None	–	0.030	925
35	3/8	None	–	0.050	2100
41	1/2	0.306	0.250	0.050	2000
40	1/2	0.312	0.312	0.060	3700
50	5/8	0.400	0.375	0.080	6100
60	3/4	0.469	0.500	0.094	8500
80	1	0.626	0.625	0.125	14 500
100	$1\frac{1}{4}$	0.750	0.750	0.156	24 000
120	$1\frac{1}{2}$	0.875	1.000	0.187	34 000
140	$1\frac{3}{4}$	1.000	1.000	0.219	46 000
160	2	1.125	1.250	0.250	58 000
180	$2\frac{1}{4}$	1.406	1.406	0.281	80 000
200	$2\frac{1}{2}$	1.562	1.500	0.312	95 000
240	3	1.875	1.875	0.375	130 000

2

Select the chain pitch from Tables 7-5, 7-6, and 7-7 page(287-289) (Pdf 303-305)

TABLE 7–5 Horsepower ratings—single strand roller chain no. 40

0.500 inch pitch

No. of teeth	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
	Rotational speed of small sprocket, rev/min																								
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	13.68	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	6.08	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

3

TABLE 7–6 Horsepower ratings—single strand roller chain no. 60

No. of teeth	0.750 inch 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

4

TABLE 7–7 Horsepower ratings—single strand roller chain no. 80

1.000 inch pitch

Rotational speed of small sprocket, rev/min

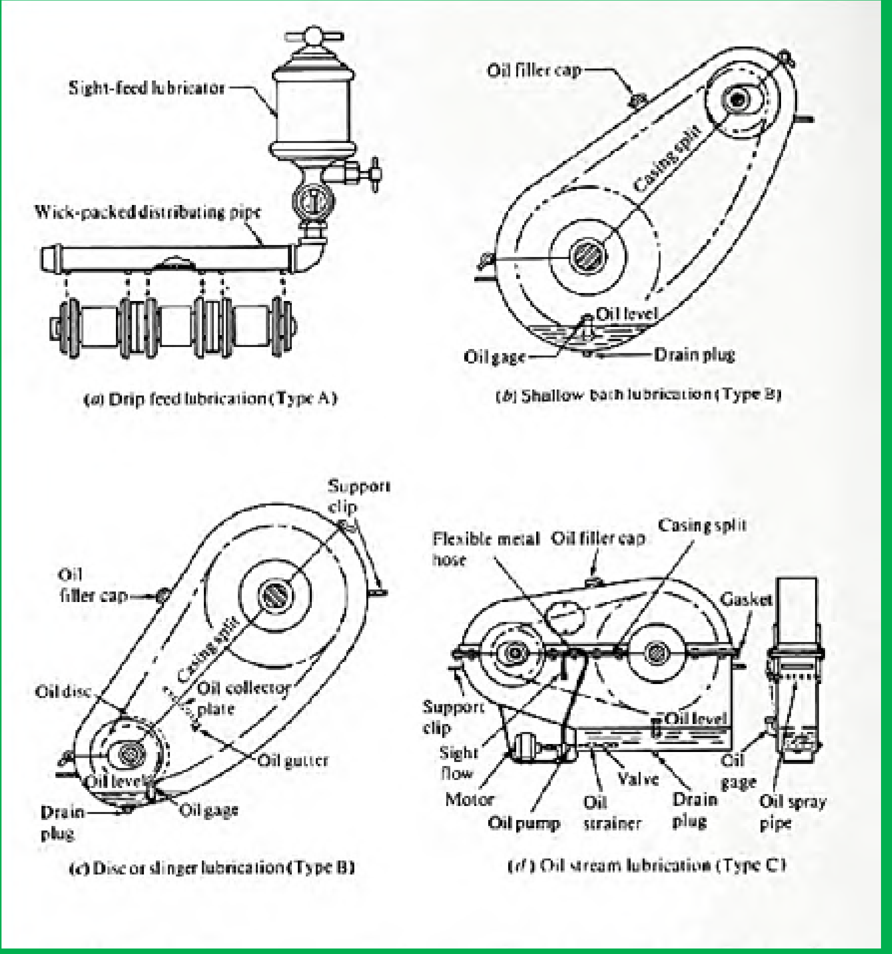
No. of teeth	10	25	50	75	88	100	200	300	400	500	600	700	800	900	1000	1200	1400	1600	1800	2000	2500	3000	3500	4000	4500
11	0.44	1.06	2.07	3.05	3.56	4.03	7.83	11.56	15.23	18.87	22.48	26.07	27.41	22.97	19.61	14.92	11.84	9.69	8.12	6.83	4.96	3.77	3.00	2.45	0.00
12	0.48	1.16	2.26	3.33	3.88	4.39	8.54	12.61	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

4

5

Select the lubrication type at the rotational speed of smaller sprocket from Tables 7-5, 7-6, and 7-7 page(287-289) (Pdf 303-305)



6

COMPUTE THE REQUIRED NUMBER OF TEETH ON THE LARGE SPROCKET:

$N_2 = N_1 \; X$ **Ratio**

COMPUTE THE ACTUAL EXPECTED OUTPUT SPEED:

$n_2 = n_2\,(N_1/N_2)$

COMPUTE THE PITCH DIAMETERS OF THE SPROCKETS

$$D_1 = \frac{P_1}{\sin\left({}^{180}/_{N_1}\right)} \quad \text{and} \quad D_2 = \frac{P_2}{\sin\left({}^{180}/_{N_2}\right)}$$

ASSUME THE CENTER DISTANCE BETWEEN SPROCKETS

$$C \cong 30 \; to \; 50 \; (pitches)$$

COMPUTE THE REQUIRED CHAIN LENGTH IN PITCHES

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

7

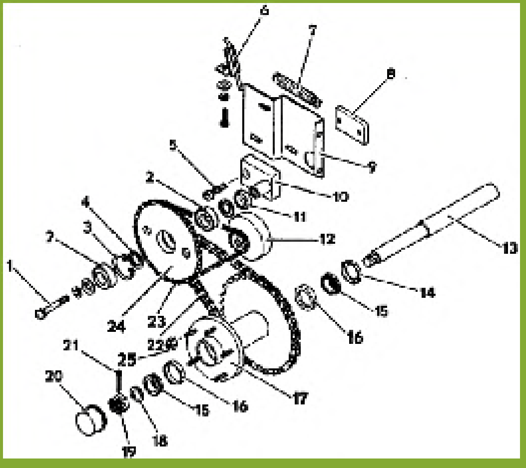
SPECIFY AN INTEGRAL & EVEN NUMBER OF PITCHES FOR THE CHAIN LENGTH, AND COMPUTE THE ACTUAL THEORETICAL CENTER DISTANCE

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

COMPUTE THE ANGLE OF WRAP OF THE CHAIN FOR EACH SPROCKET

$$\theta_1 = 180^o - 2\sin^{-1}\left[\frac{D_2 - D_1}{2C}\right] \ (\textit{must be larger than } 120\ ^o)$$

$$\theta_2 = 180^o + 2\sin^{-1}\left[\frac{D_2 - D_1}{2C}\right]$$

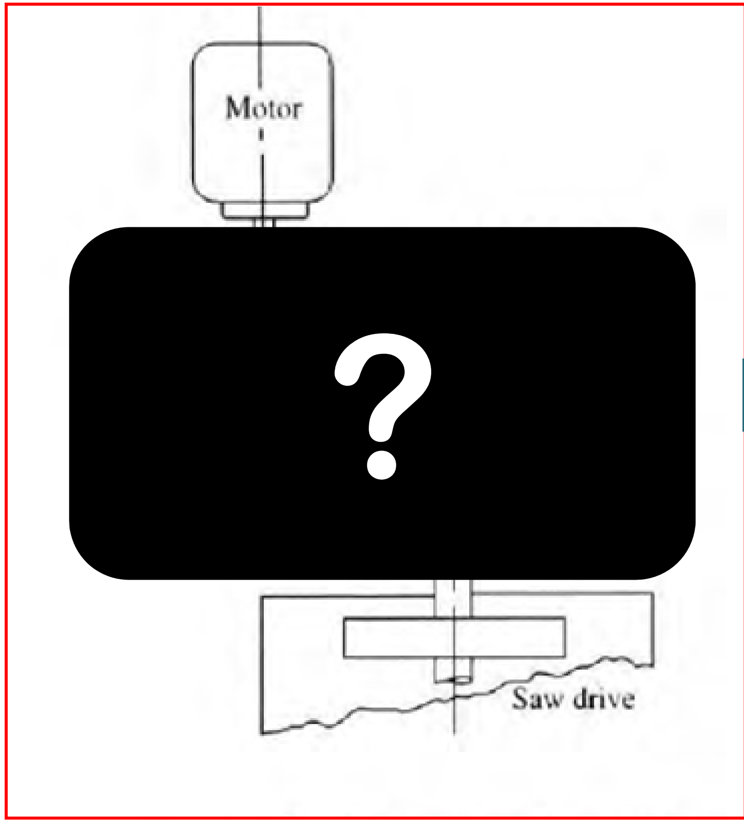


Mechanical Engineering Design II

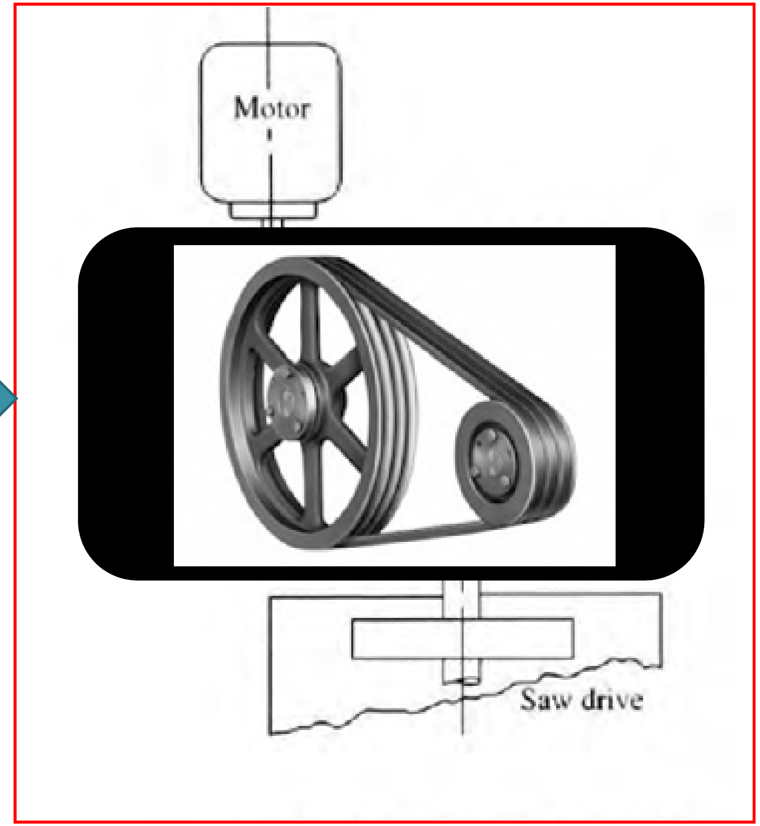
Seventeenth Lecture

Design of Belt Derive

Power Transmission Problem



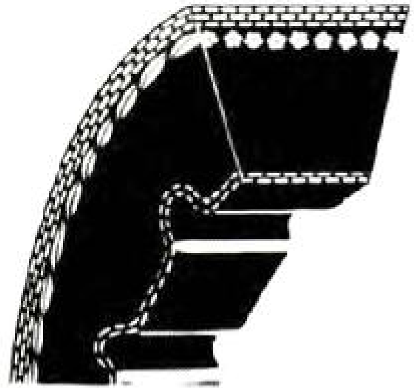
Proposed solution (Belt Drive)



Transmitted Power is Known
Input and output range speed is Known

Types of Belt Drives

(a) Wrapped construction



(b) Die cut, cog type

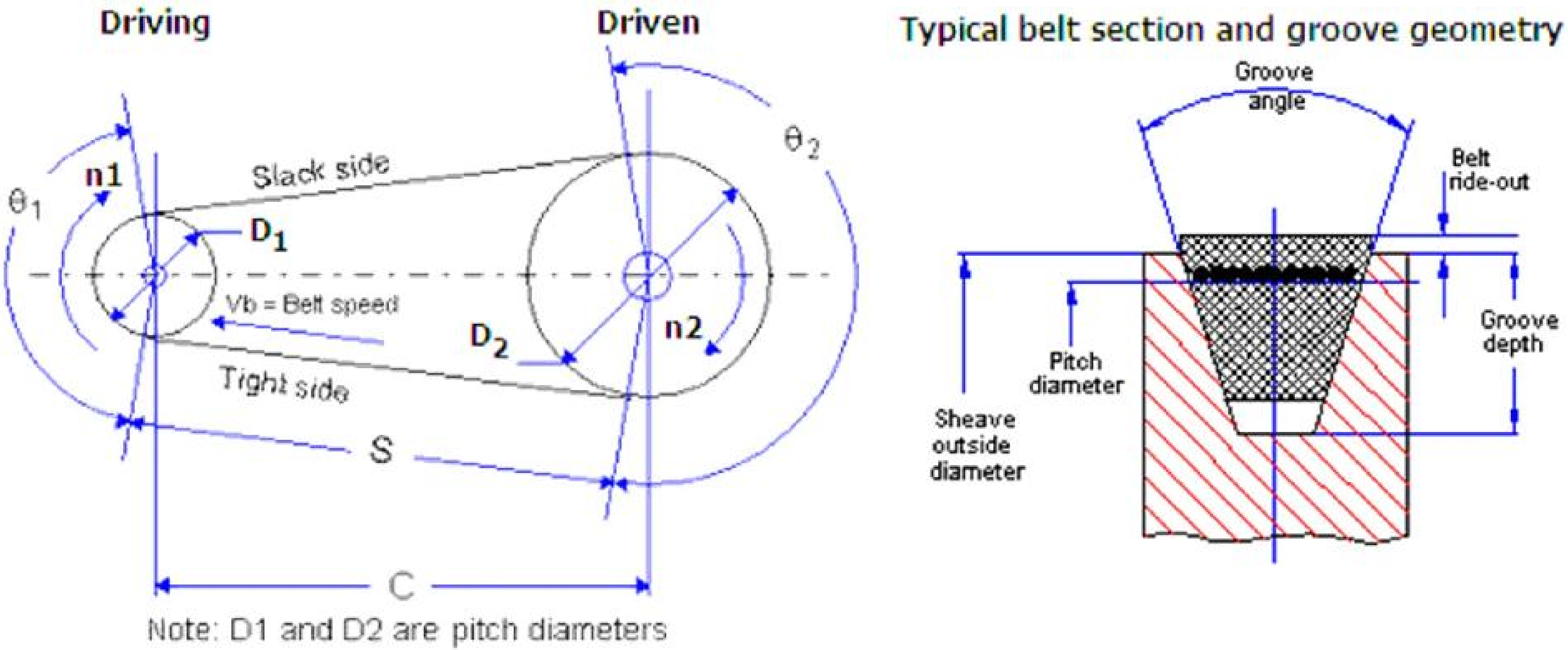
(c) Synchronous belt

(d) Poly-rib belt

(e) Vee-band

(f) Double angle V-belt

Basic Belt Drive Geometry



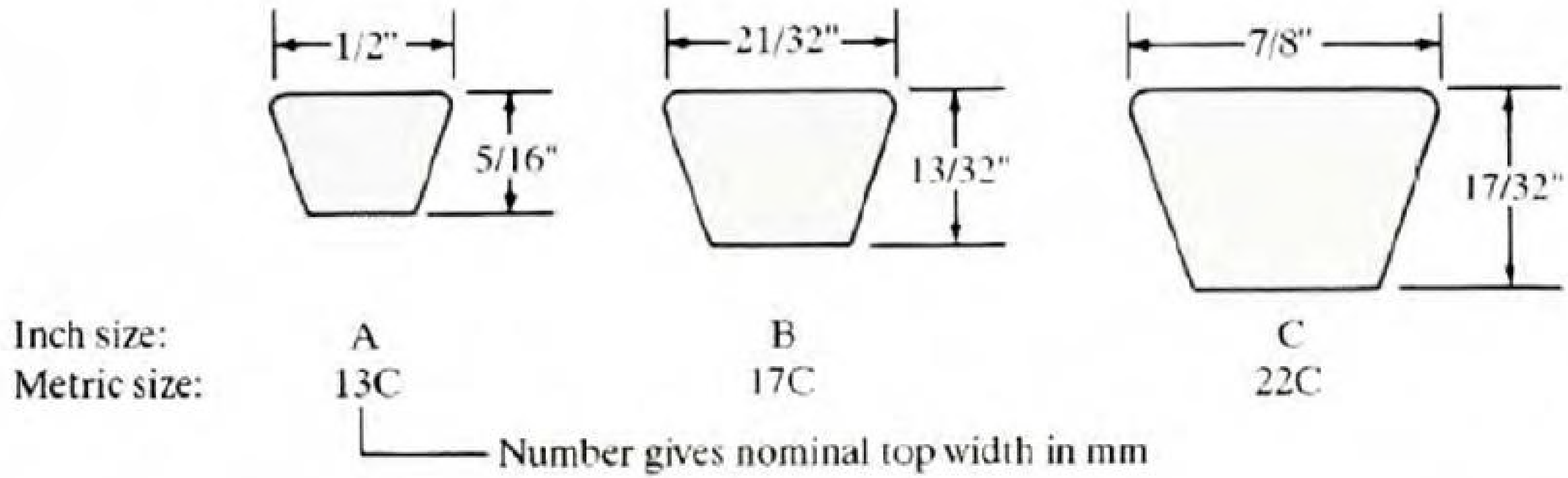
θ_1, θ_2 are the angles of wrap for small and big sheaves respectively

n_1, n_2 are the angular velocity for small and big sheaves respectively

C the center distance

S the span distance

Standard Belt Cross Sections



SAE Standard J636: V-belts and pulleys

SAE Standard J637: Automotive V-belt drives

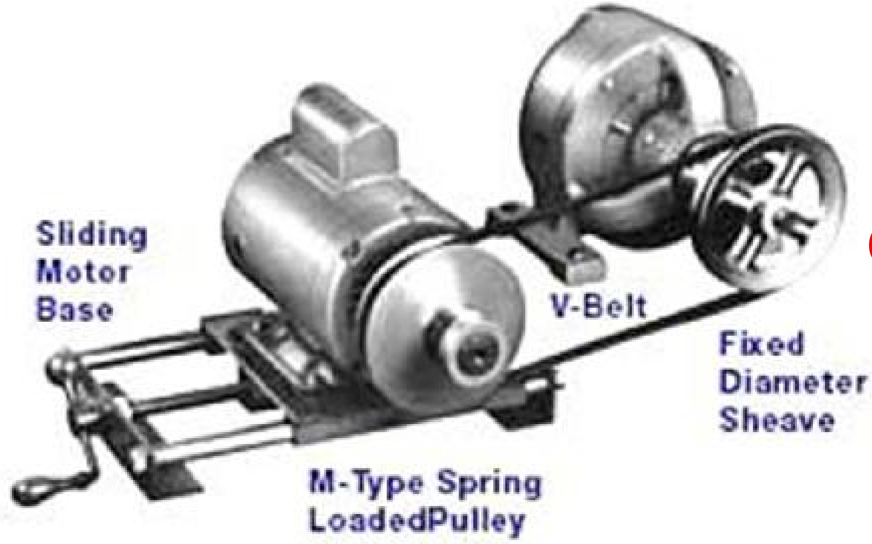
SAE Standard J1278: SI (metric) synchronous belts and pulleys

SAE Standard J1313: Automotive synchronous belt drives

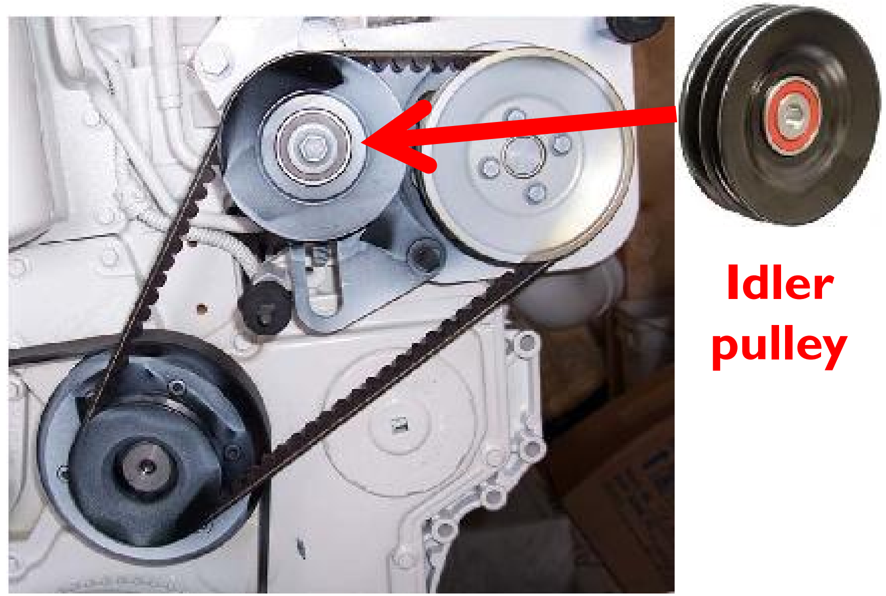
SAE Standard J1459: V-ribbed belts and pulleys

Design Conditions for V-belt

- The center distance must be adjustable in both directions, or if it was fixed, idler pulleys should be used.



or



- The nominal range of center distance should be: $D_2 < C < 3(D_2 + D_1)$
- The angle of wrap on smaller sheave (θ_1)should be $> 120^o$
- Most commercially available sheaves are cast iron, which should be limited to ***(1981 m/min = 33 m/sec)***
- Consider an alternative type of drive, such as a gear type or chain, ***if the belt speed is less than (304.8 m/min = 5 m/sec)***

V-Belt Drive Design

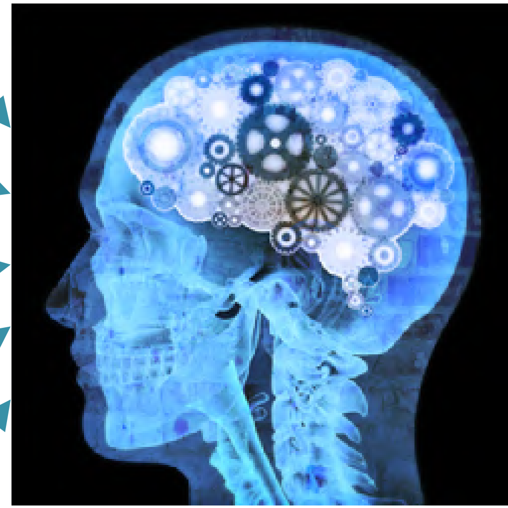
The rated power of the driving motor or other prime mover

Type of driver and driven load

The center distance

The size of the driving and driven sheaves

Speed of the smaller sheave



Designer

The service factor

The power rating for one belt

The belt length

The correction factor for belt length

The correction factor for the angle of wrap on the smaller sheave

The number of belts

The initial tension on the belt

Flowchart for designing a Belt drive:

Transmitted Power , Input and Output speed, Type of driver and driven load

Specify a service factor from Table 7-1 page.274 (290 Pdf)

TABLE 7–1 V-belt service factors

Driven machine type	AC motors: Normal torque[a] DC motors: Shunt-wound Engines: Multiple-cylinder <6 h per day	6–15 h per day	>15 h per day	AC motors: High torque[b] DC motors: Series-wound, compound-wound Engines: 4-cylinder or less <6 h per day	6–15 h per day	>15 h per day
Agitators, blowers, fans, centrifugal pumps, light conveyors	1.0	1.1	1.2	1.1	1.2	1.3
Generators, machine tools, mixers, gravel conveyors	1.1	1.2	1.3	1.2	1.3	1.4
Bucket elevators, textile machines, hammer mills, heavy conveyors	1.2	1.3	1.4	1.4	1.5	1.6
Crushers, ball mills, hoists, rubber extruders	1.3	1.4	1.5	1.5	1.6	1.8
Any machine that can choke	2.0	2.0	2.0	2.0	2.0	2.0

[a]Synchronous, split-phase, three-phase with starting torque or breakdown torque less than 175% of full-load torque.
[b]Single-phase, three-phase with starting torque or breakdown torque greater than 175% of full-load torque.

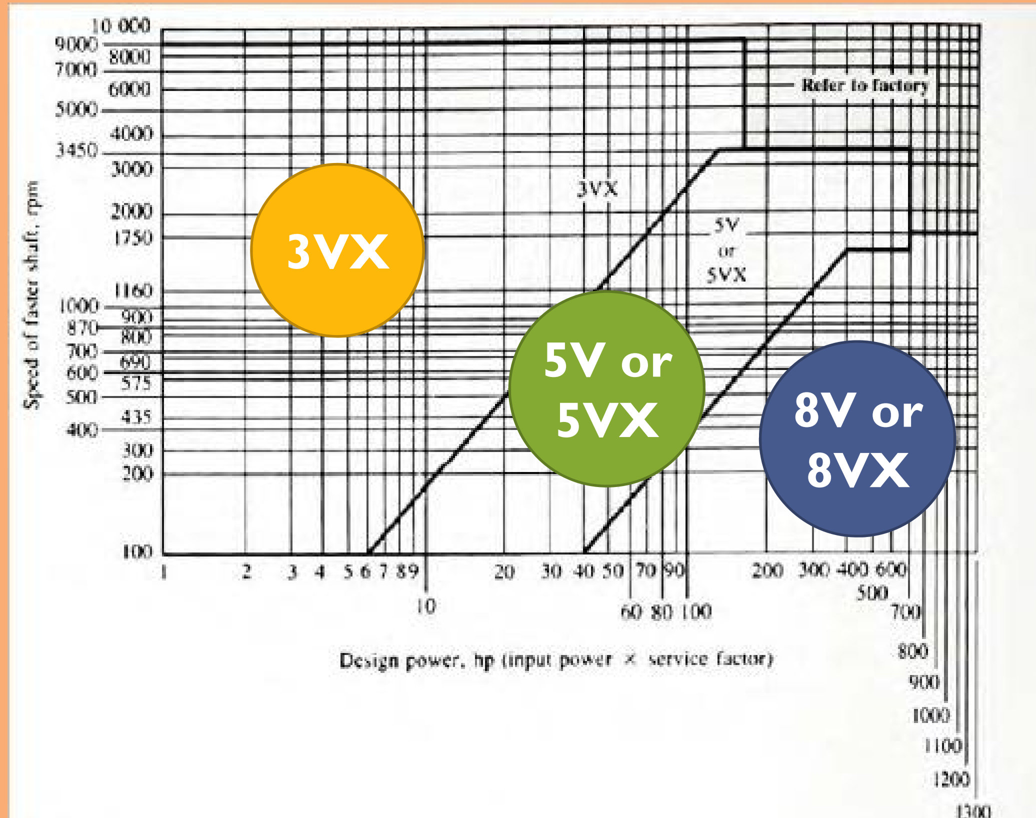
1

1

COMPUTE THE DESIGN POWER

Design Power=service factor x transmitted power

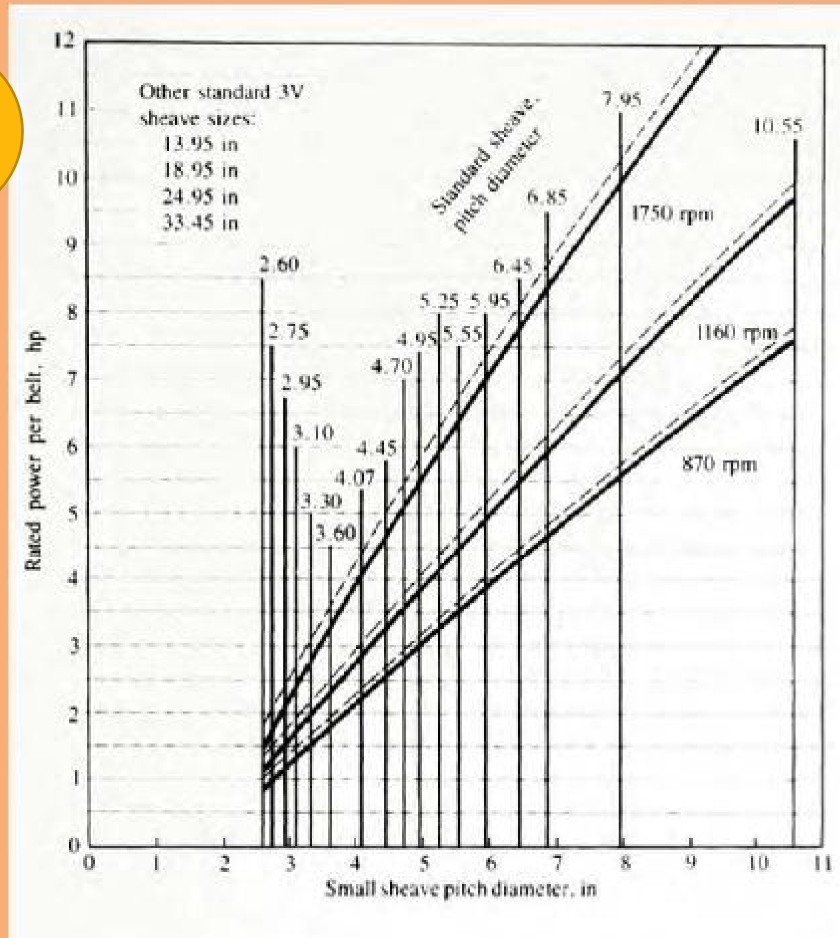
Specify the standard belt size from Figure 7-9 page.284 (300 Pdf)



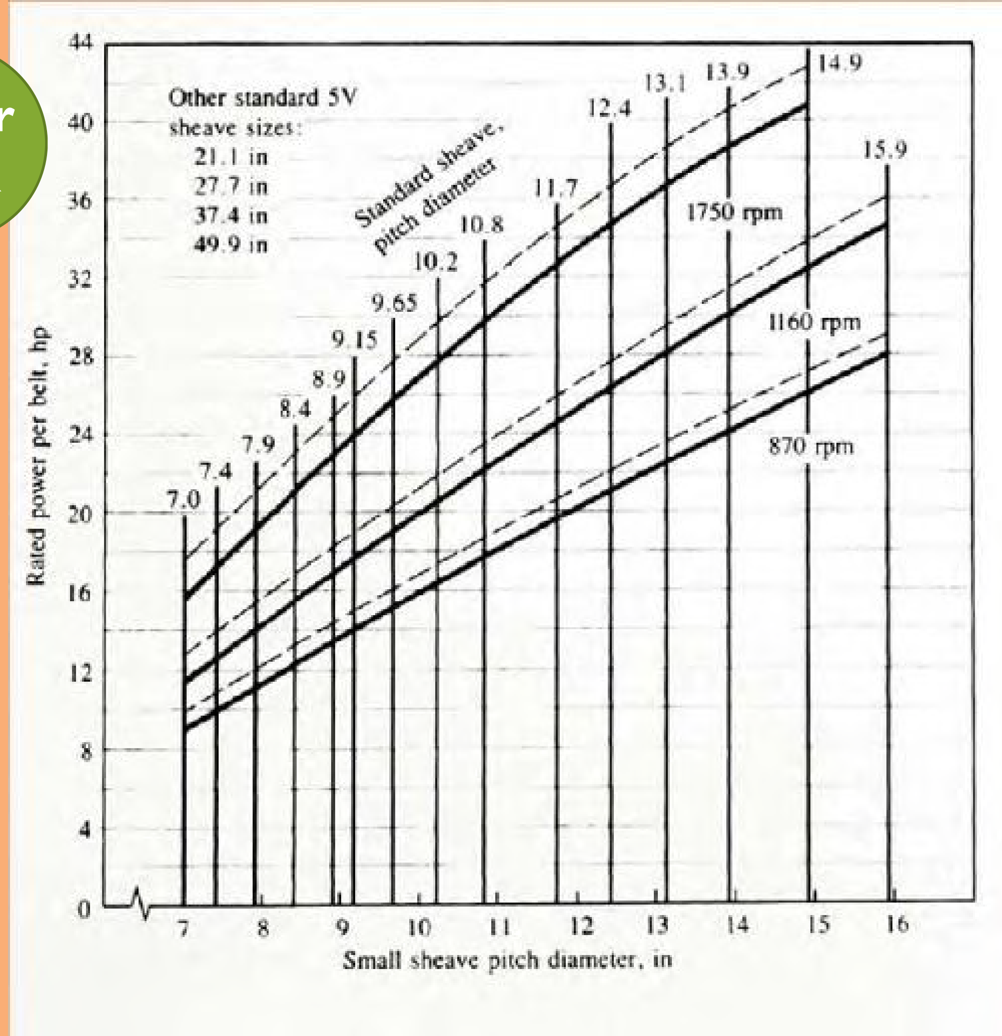
2

State the power rating for one belt from Figures (7-10, 7-11, and 7-12) page(275-276) (Pdf 291-292)

3VX



5V or 5VX

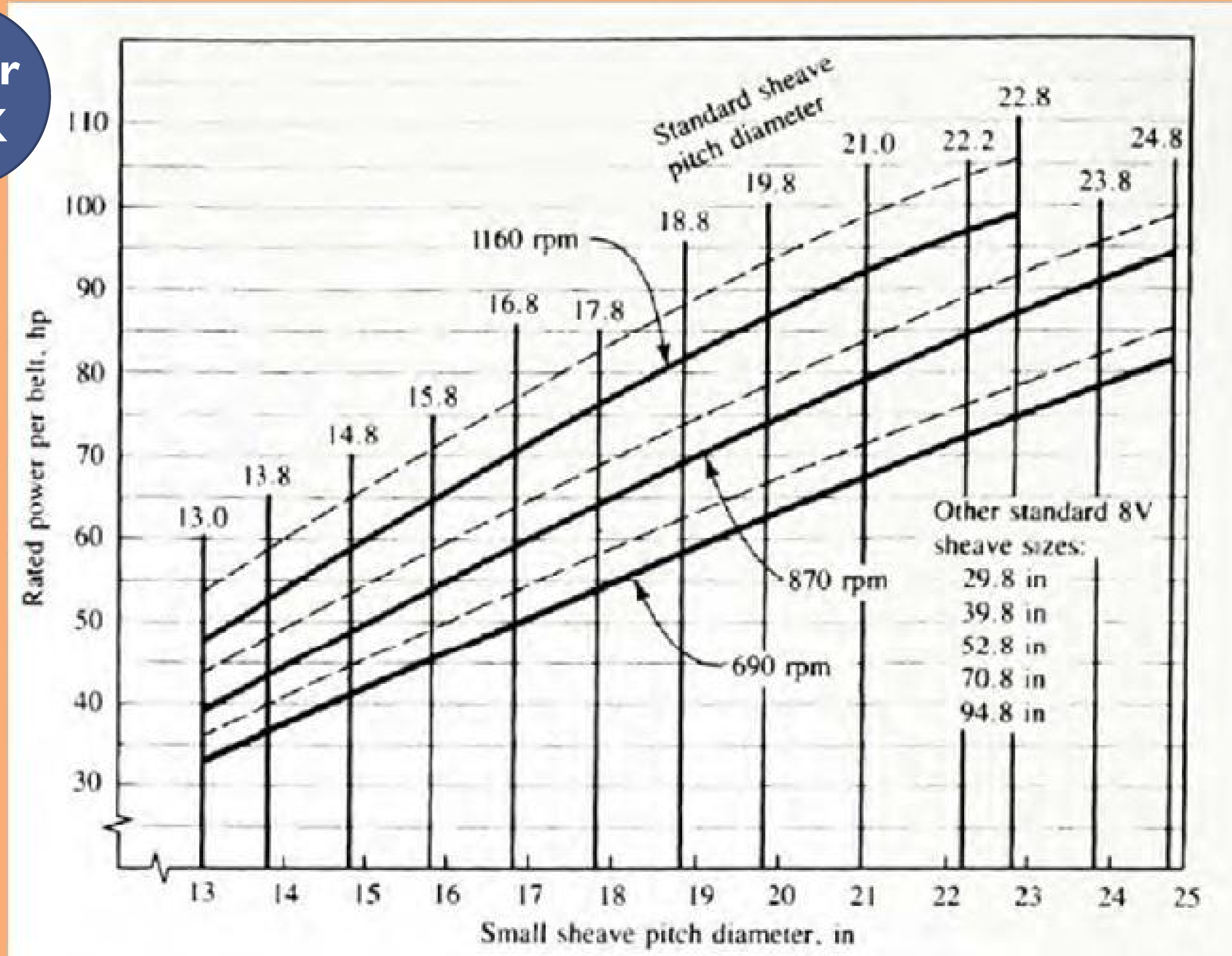

Other standard 5V sheave sizes:
21.1 in
27.7 in
37.4 in
49.9 in
Standard sheave, pitch diameter
7.0
7.4
7.9
8.4
8.9
9.15
9.65
10.2
10.8
11.7
12.4
13.1
13.9
14.9
15.9
1750 rpm
1160 rpm
870 rpm
Rated power per belt, hp
0
4
8
12
16
20
24
28
32
36
40
44
7
8
9
10
11
12
13
14
15
16
Small sheave pitch diameter, in

3

4

4

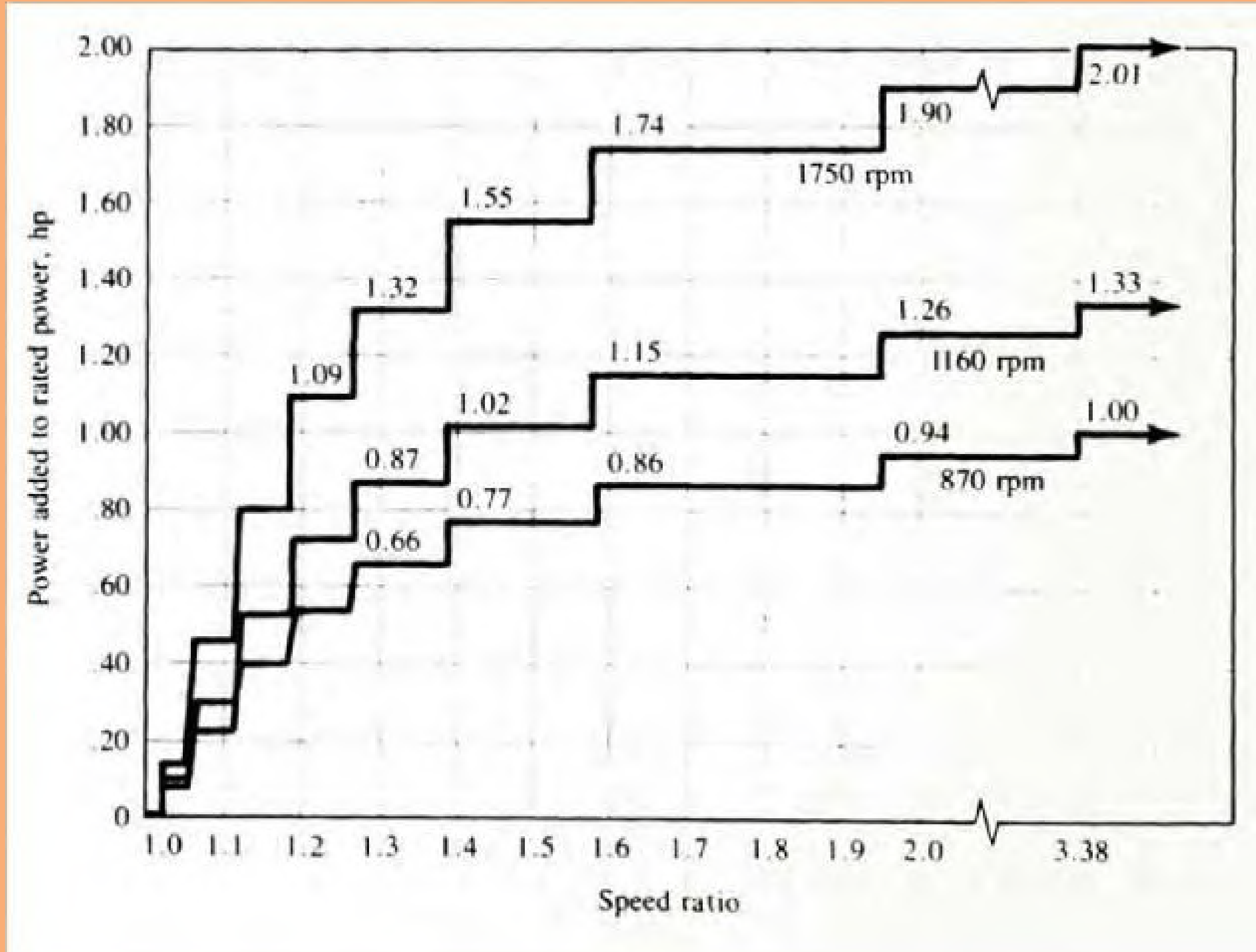
8V or 8VX



Standard sheave pitch diameter
1160 rpm
870 rpm
690 rpm
13.0
13.8
14.8
15.8
16.8
17.8
18.8
19.8
21.0
22.2
22.8
23.8
24.8
Other standard 8V sheave sizes:
29.8 in
39.8 in
52.8 in
70.8 in
94.8 in
Rated power per belt, hp
Small sheave pitch diameter, in

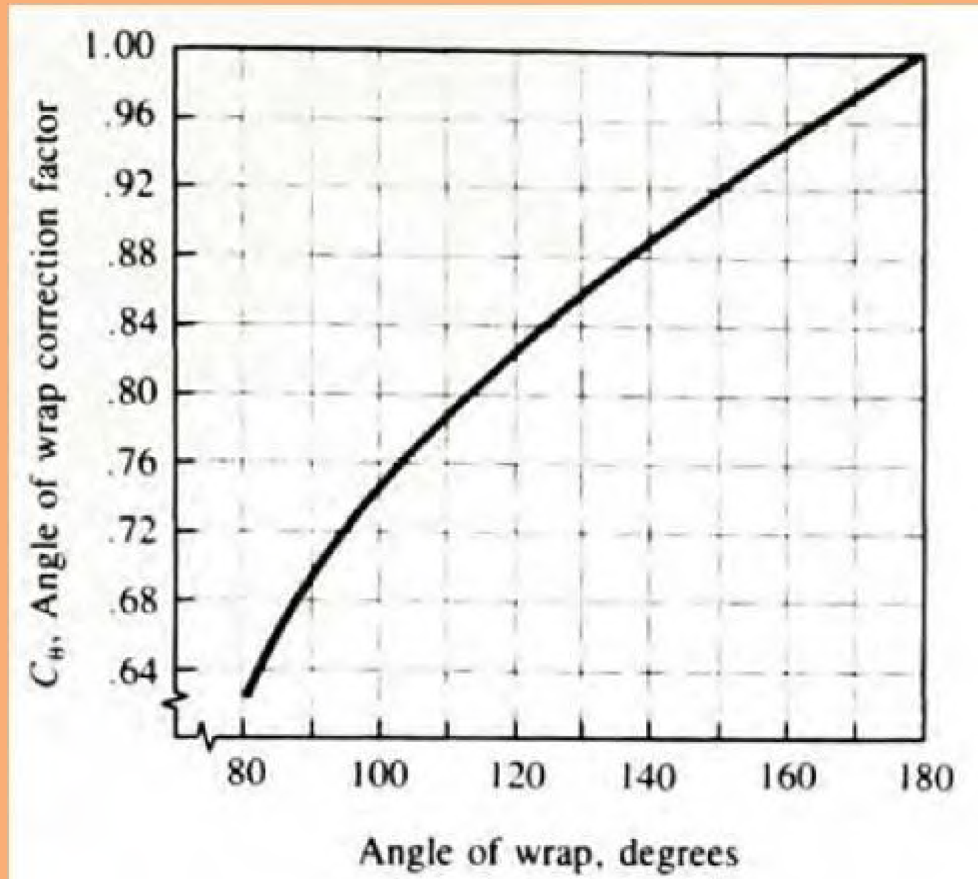
5

State the power added to rated power from Figure (7-13) page(276) (Pdf 292)



Compute the angle of wrap: $\theta_1 = 180^O - 2\sin^{-1}\left[\frac{D_2 - D_1}{2C}\right]$

State the correction factor of wrap angle C_θ from figure (7-14) page (277) (293 pdf):



7

Specify a trial center distance: $D_2 < C < 3(D_2 + D_1)$

Compute the required belt length: $L = 2C + 1.57(D_2 + D_1) + \frac{(D_2+D_1)^2}{4C}$

Select the nearest standard belt length from Table (7-2) page (277) (293 pdf)

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	

8

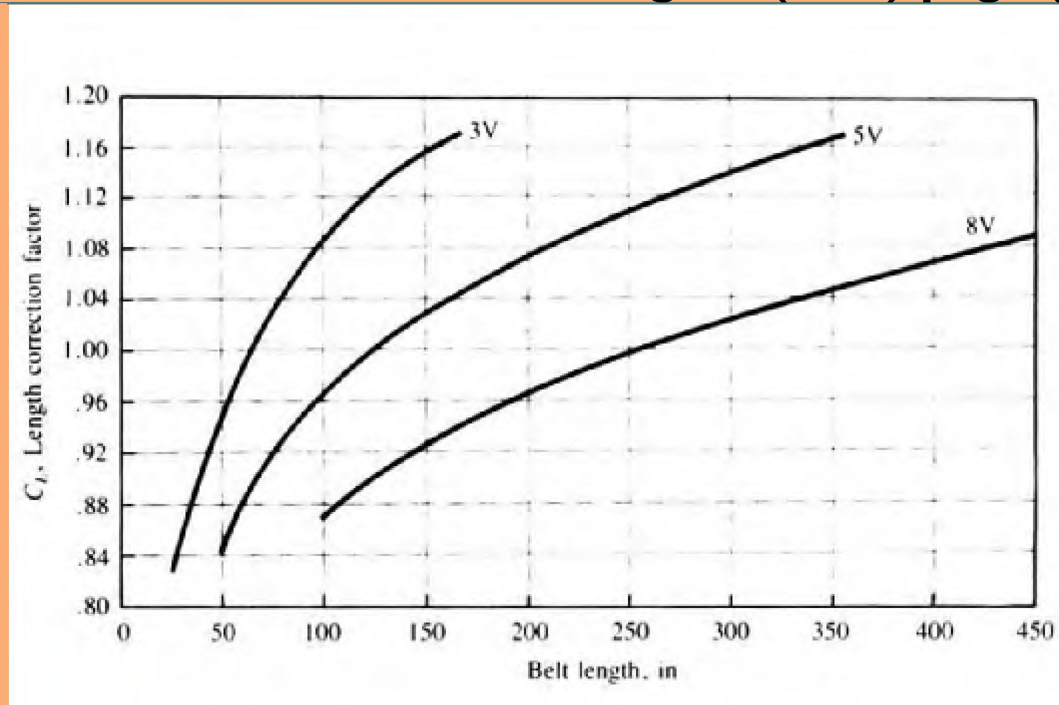
8

Re-calculate the center distance and angle of wrap:

$$C = \frac{B+\sqrt{B^2-32(D_2-D_1)^2}}{16} \text{ where } B = 4L - 6.28(D_2 + D_1)$$

$$\theta_1 = 180^O - 2\sin^{-1}\left[\frac{D_2 - D_1}{2C}\right]$$

State the length correction factor from Figure (7-15) page (277) (293 pdf)



9

Compute the corrected rated power per belt and the number of belts required to carry the design power:

$$\text{Corrected rated power} = C_\theta C_L P$$

where

$$P = (\text{actual rated power}) = \boxed{\begin{array}{c}\text{rated power}\\ (fig.\,7-10, 11\ \&12)\end{array}} + \boxed{\begin{array}{c}\text{added power}\\ (fig.\,7-13)\end{array}}$$

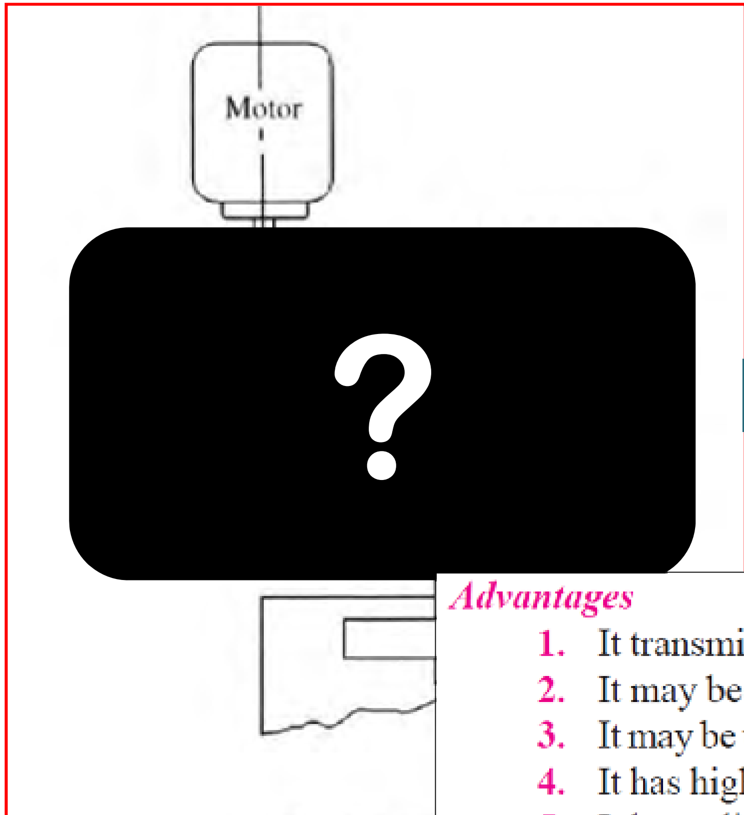
$$\text{Number of belts} = \frac{\text{Design Power}}{\text{Corrected Rated Power}}$$

Mechanical Engineering Design II

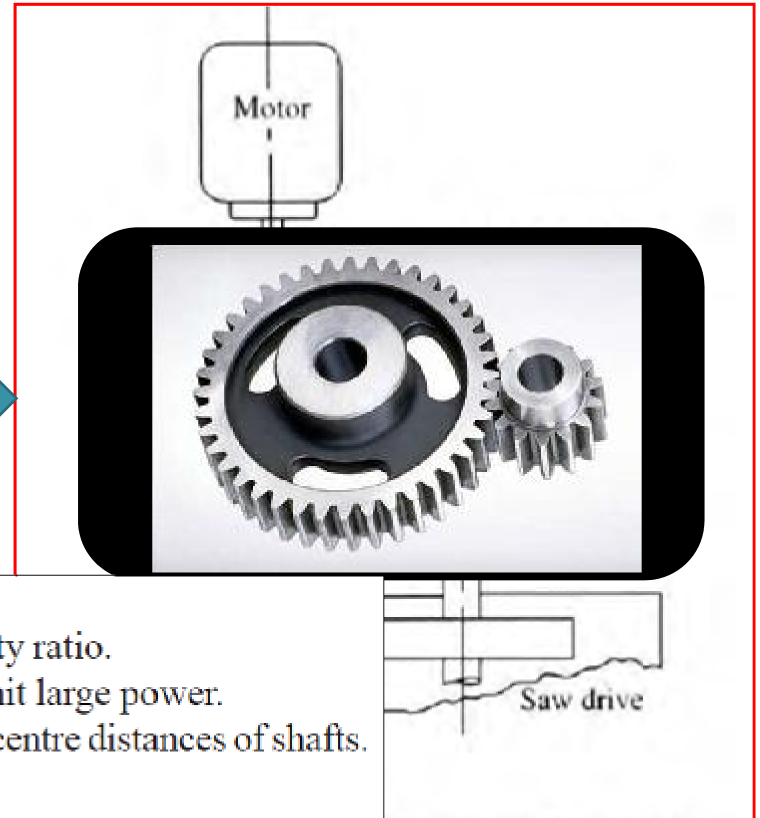
Eighteenth & Nineteenth Lectures

Design of Spur Gear

Power Transmission Problem



Proposed solution (Spur Gear)



Advantages

1. It transmits exact velocity ratio.
2. It may be used to transmit large power.
3. It may be used for small centre distances of shafts.
4. It has high efficiency.
5. It has reliable service.
6. It has compact layout.

Disadvantages

1. Since the manufacture of gears require special tools and equipment, therefore it is costlier than other drives.

Gear Manufacturing

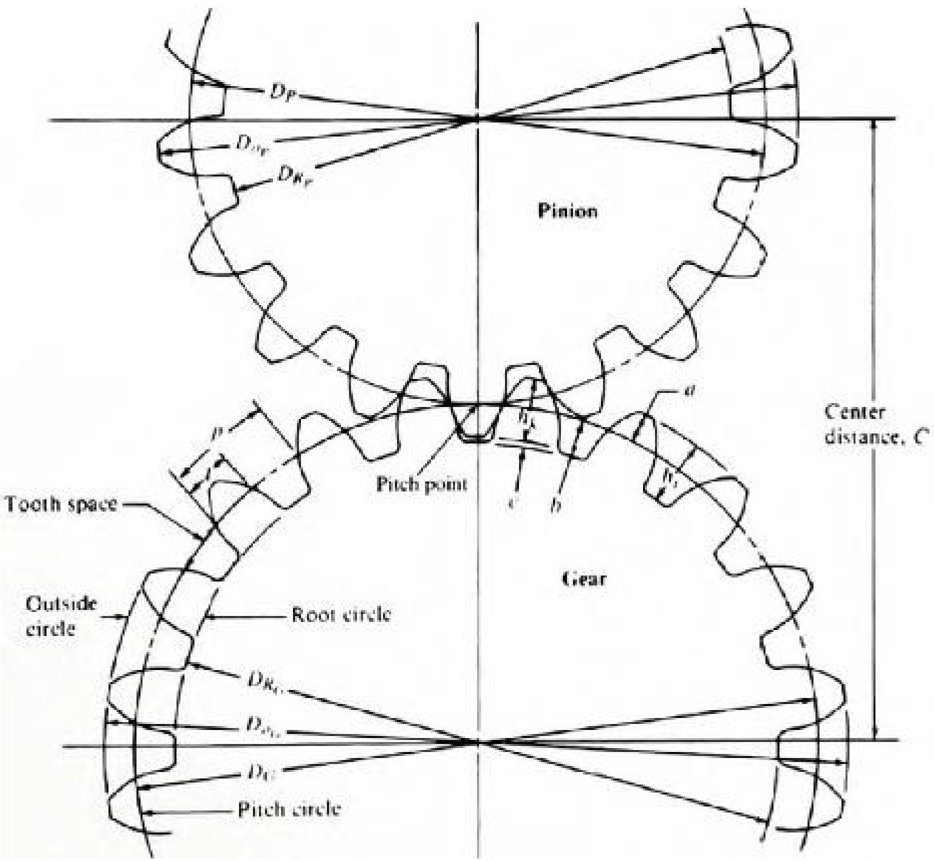
(a) Form milling cutter

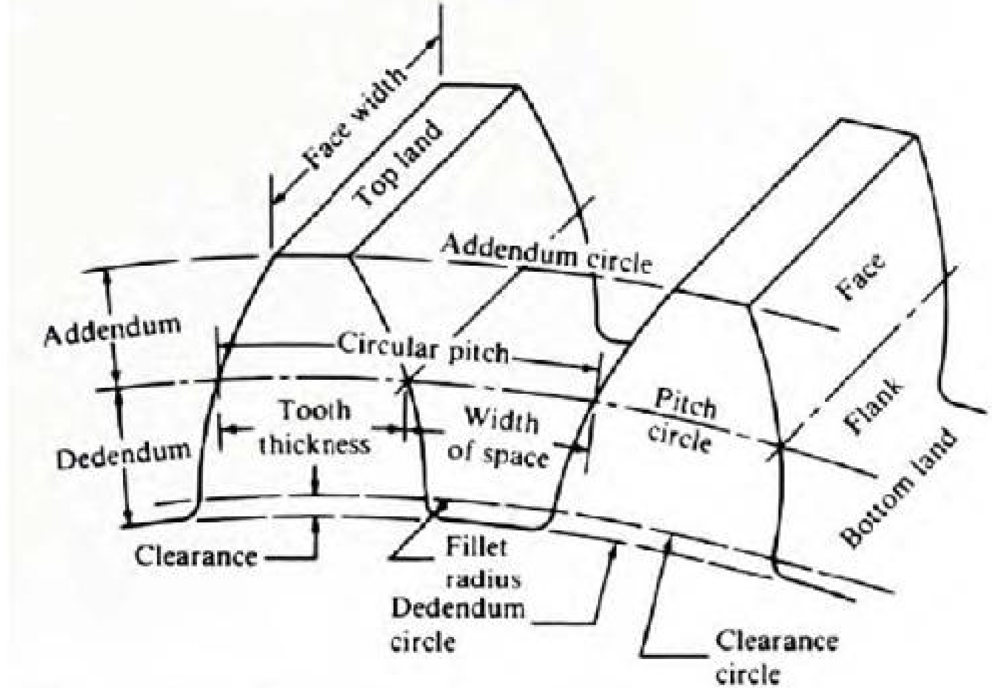
(b) Spur gear shaper cutter

(c) Hob for small pitch gears having large teeth

(d) Hob for high pitch gears having small teeth

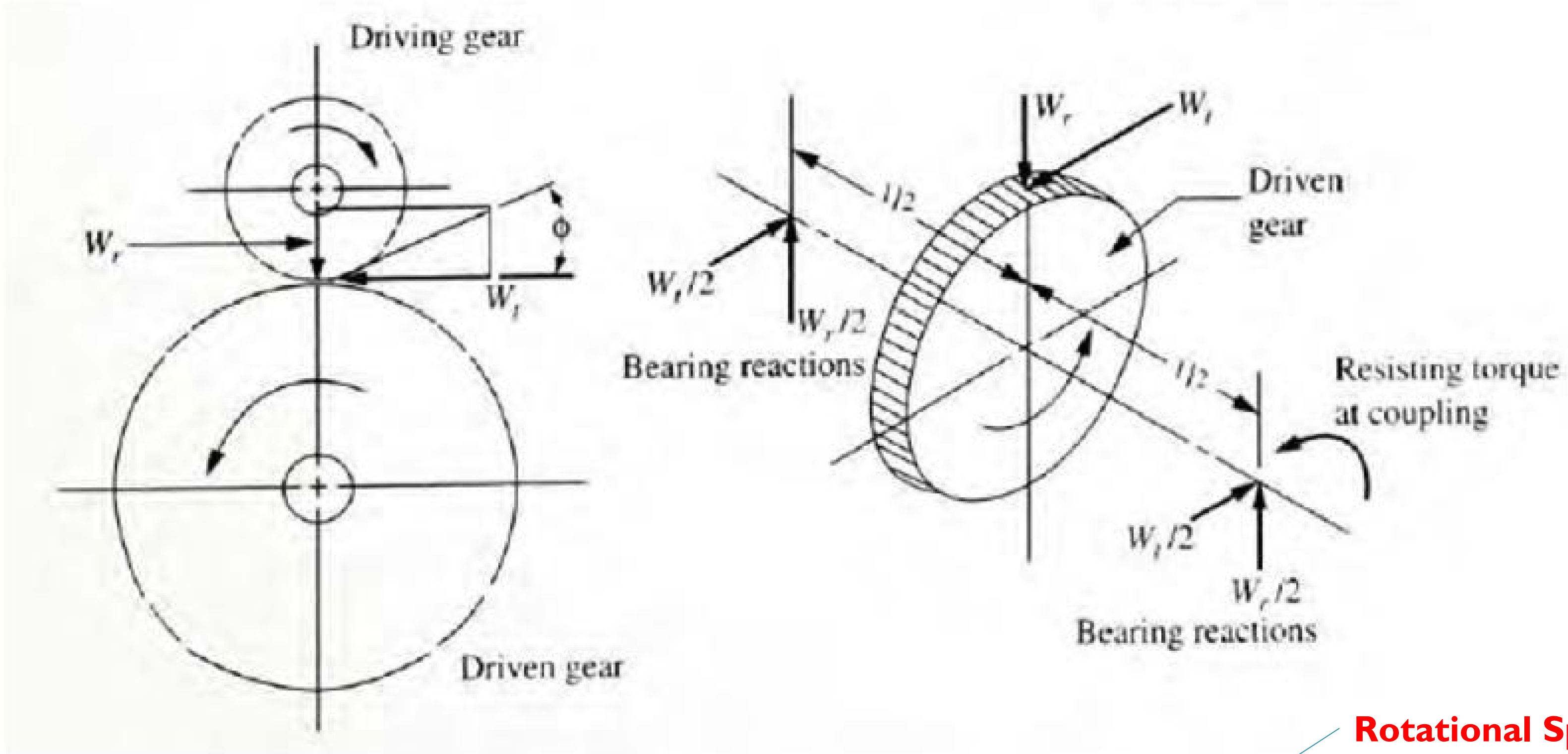
Basic Spur Gear Geometry





Chapter.8 page. 309 pdf

Kinematics of Spur Gear



Torque (N.m)

Rotational Speed (rad/sec)

$$T = W_t(R) = W_t(D/2) = P/n$$

Transmitted Power (watt)

Tangential Force (N)

$$W_t = \frac{2P}{Dn}$$

Pitch Diameter (m)

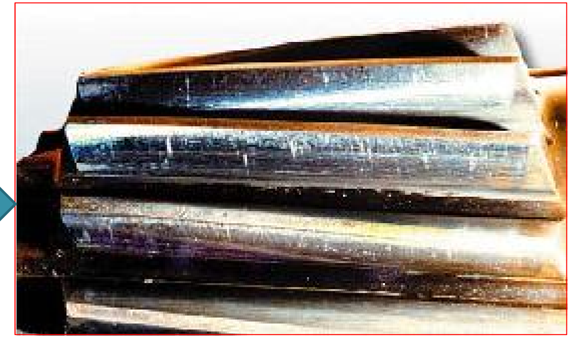
Modes of Gear Tooth Failure

1. Bending failure.

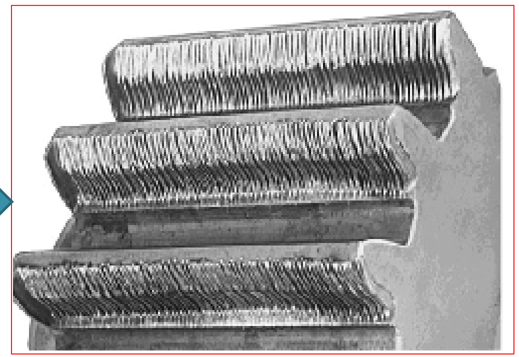
Root failure

2. Pitting.

3. Scoring.



4. Abrasive wear.



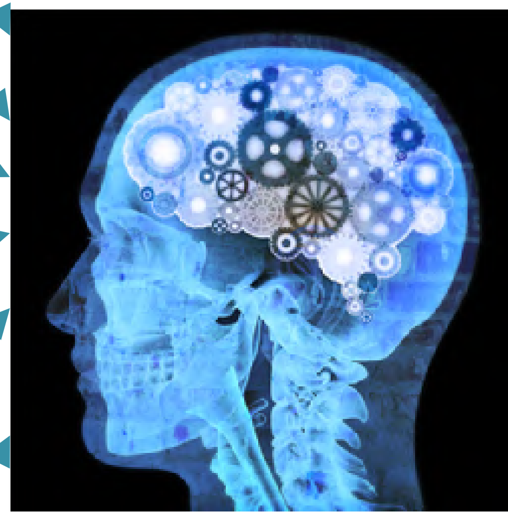
5. Corrosive wear.

Surface failure

Dangerous Bar

Spur Gear Design

- The power to be transmitted
- Type of driver and driven load
- The speed of the driving gear
- The center distance
- The speed of the driven gear or the velocity ratio
- Other information related to problem specification



Designer

- The gear teeth should not fail under static loading or dynamic loading during normal running conditions.
- The gear teeth should have wear characteristics so that their life is satisfactory.
- The use of space and material should be economical.
- The alignment of the gears and deflections of the shafts must be considered.
- The lubrication of the gears must be satisfactory.

Flowchart for spur gear designing process:

Transmitted Power , Input and Output speed, Center distance, Type of driver and driven load

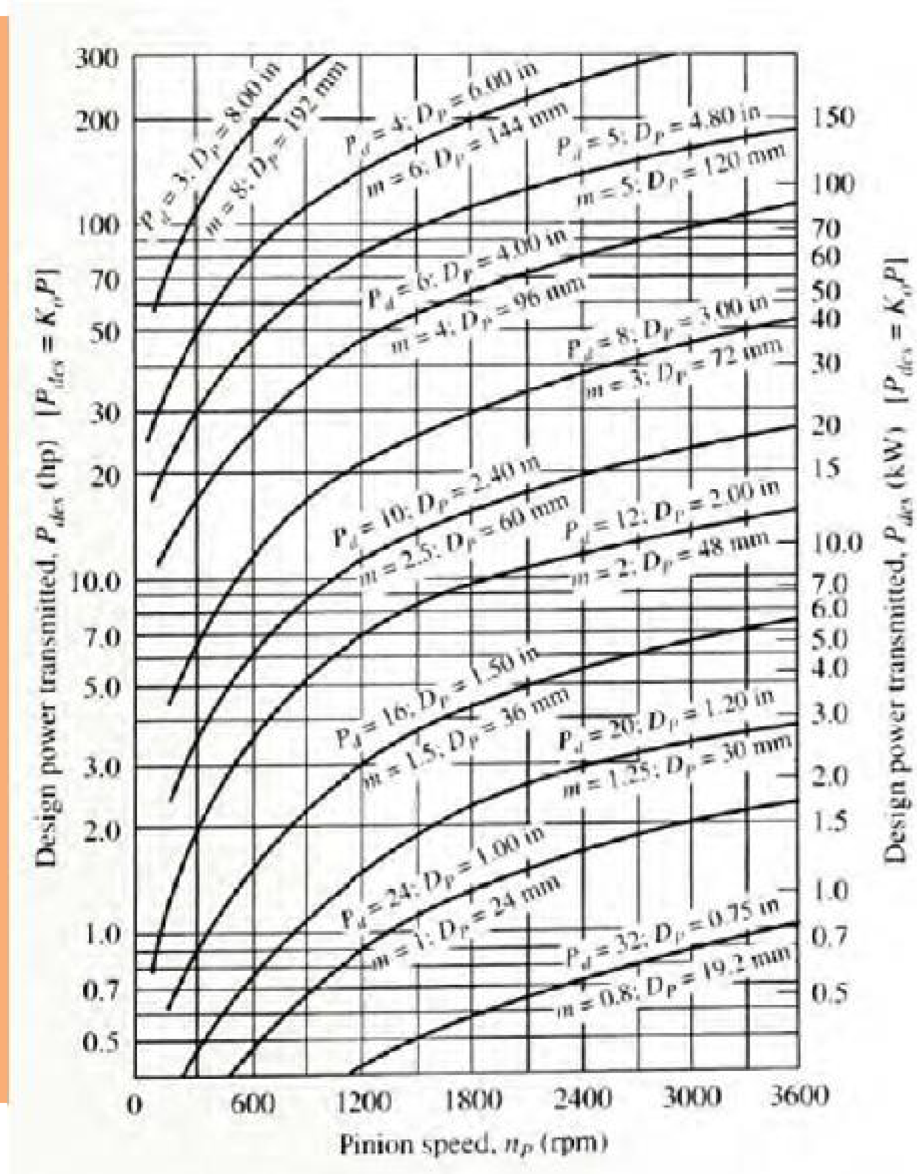
Choose the over load factor (K_o) from Table (9-5) page(389) (405pdf)

	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

Compute the design power
Design Power=K_o x transmitted power

1

Find the trial value for Diametral pitch (P_d) or Module (m) and Pitch Diameter (D_p) from Figure 9-27 page.409 (425 Pdf)



2

Specify the no. of teeth for Pinion N_P (from 17 to 20)

Compute the nominal velocity ratio $VR = \frac{n_P}{n_G}$

Compute the approximate no. of teeth for Gear
$N_G = N_P \times VR$

Compute the actual velocity ratio $VR = \frac{N_G}{N_P}$

Compute the actual output velocity $n_G = n_P \frac{N_P}{N_G}$

Compute the pitch diameters $D_p = \frac{N_p}{P_d}$, $D_G = \frac{N_G}{P_d}$, center distance $c = \frac{N_P + N_G}{2P_d}$,
pitch line speed $v_t = \frac{\pi D_p n_P}{60}$ and tangential force, $W_t = \frac{2P}{D_P n_p}$

3

Specify the face width within the following recommended range for general machine drive gears:

$^{8}/_{P_d} < F < {}^{16}/_{P}$ **Nominal value of** $F = {}^{12}/_{P}$

Specify the quality number Q_v , from Table (9-2) page (378) (394 pdf)

Application	Quality number	Application	Quality number
Cement mixer drum drive	3–5	Small power drill	7–9
Cement kiln	5–6	Clothes washing machine	8–10
Steel mill drives	5–6	Printing press	9–11
Grain harvester	5–7	Computing mechanism	10–11
Cranes	5–7	Automotive transmission	10–11
Punch press	5–7	Radar antenna drive	10–12
Mining conveyor	5–7	Marine propulsion drive	10–12
Paper-box-making machine	6–8	Aircraft engine drive	10–13
Gas meter mechanism	7–9	Gyroscope	12–14

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

Pitch line speed (fpm)	Quality number	Pitch line speed (m/s)
0–800	6–8	0–4
800–2000	8–10	4–11
2000–4000	10–12	11–22
Over 4000	12–14	Over 22

4

Analyzing of gear tooth failure mode

Root (Bending) Failure Mode

Surface (Pitting, Scoring,…) Failure Mode

Bending Stress Number

$$S_t = \frac{W_t P_d}{F\,J} K_o K_s K_m K_B K_v < S_{at} \frac{Y_N}{K_R (S.F)}$$

Contact Stress Number

$$S_c = C_p \sqrt{\frac{W_t K_o K_s K_m K_v}{F D_p I}} < S_{ac} \frac{Z_N C_H}{K_R (S.F)}$$

Find the values of factors ($J, I, K_s, K_m, K_B, K_v, C_p, Y_N, Z_N, C_H, K_R, S.F$) as in the following steps

5

5

Specify the type of material for the gears to find the Elastic Coefficient C_p from Table (9-9) page(400) (Pdf 416)

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

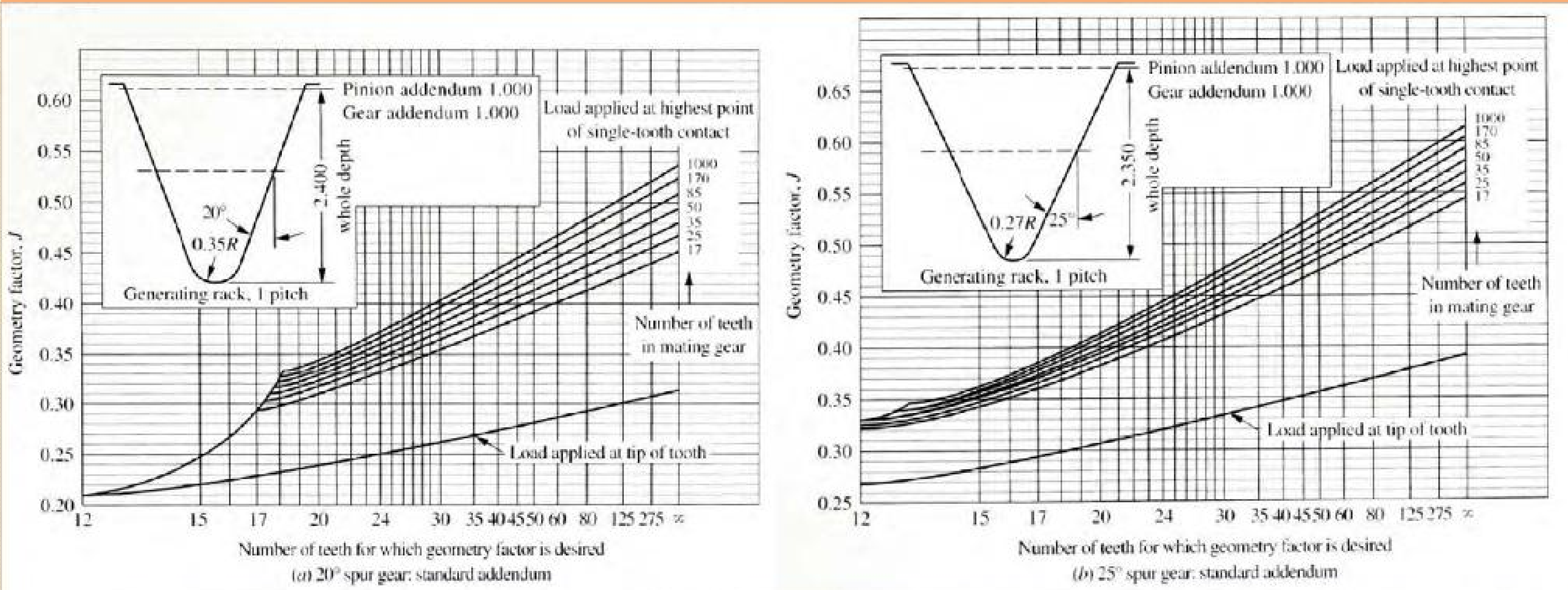
Source: Extracted from AGMA Standard 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, VA 22314.

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

6

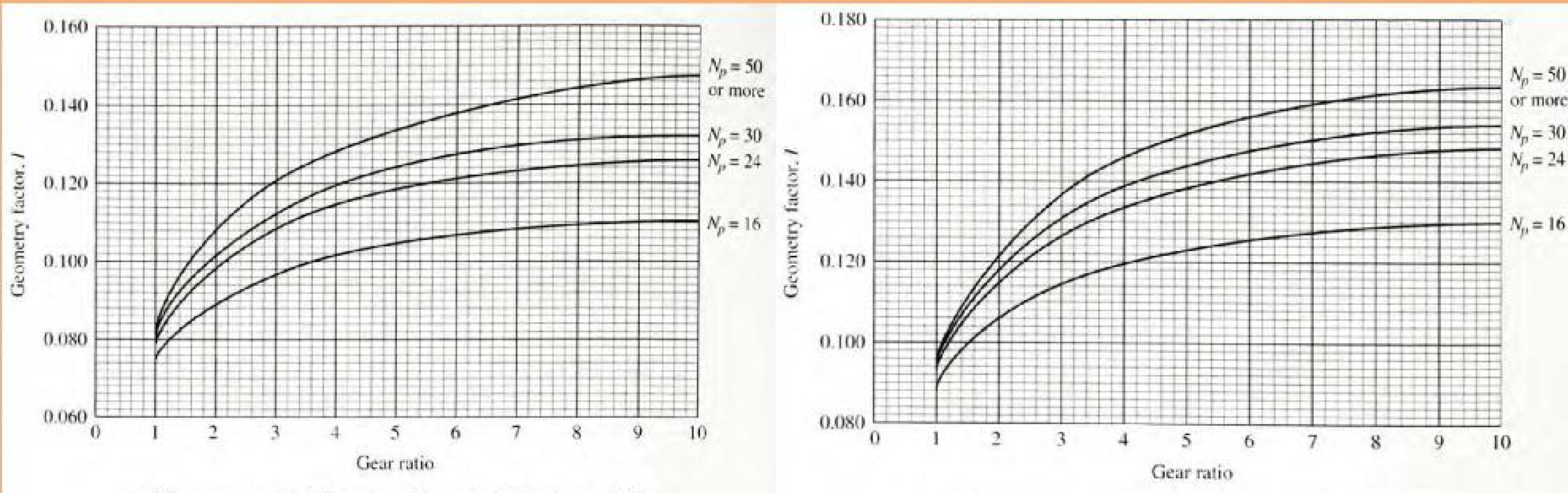
6

Specify the bending geometry factor (J) from figure (9-17) page (387) (403pdf):



7

Specify the pitting geometry factor (I) from figure (9-23) page (402) (418 pdf):



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

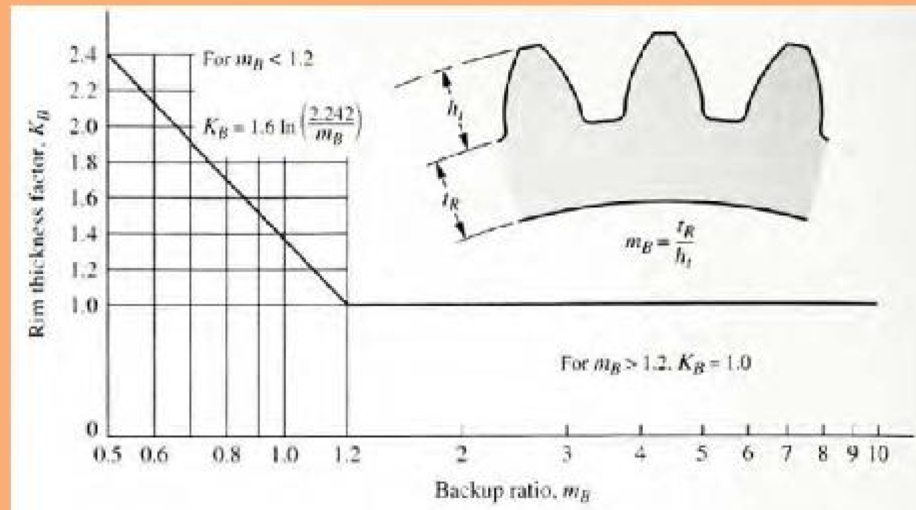
(b) 25° pressure angle, full-depth teeth (standard addendum = $1/P_d$)

8

Specify the size factor (K_S) from Figure (9-6) page (389) (293 pdf)

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

Specify the rim thickness factor (K_B) from Figure (9-20) page (392) (408 pdf)

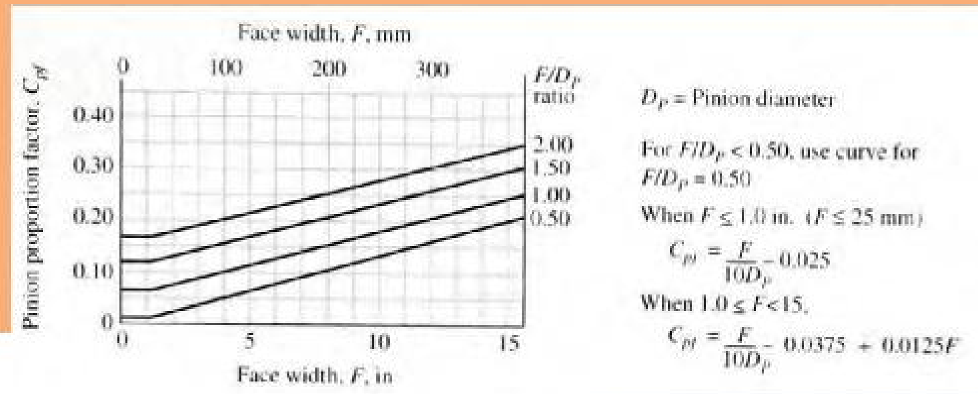


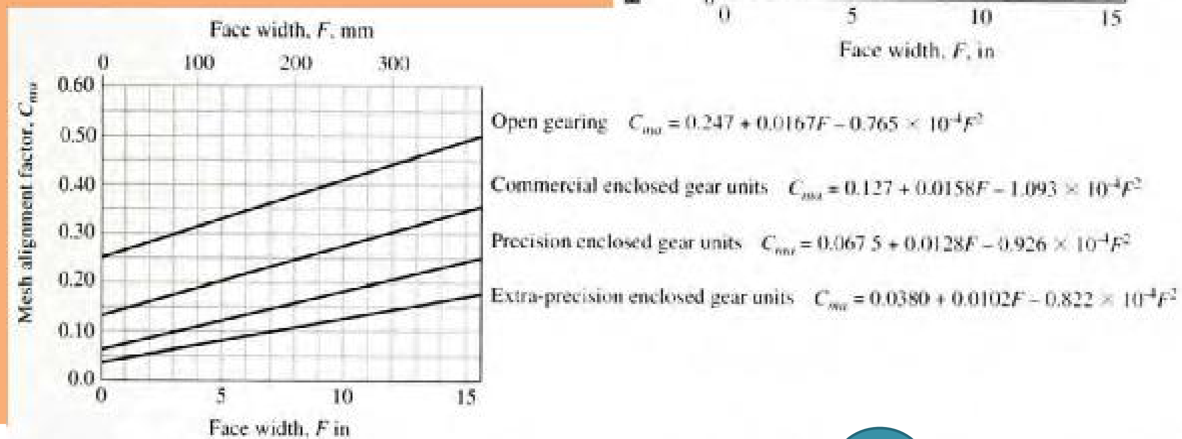
9

Determine the load distribution factor (K_m) : $K_m = 1.0 + C_{pf} + C_{ma}$

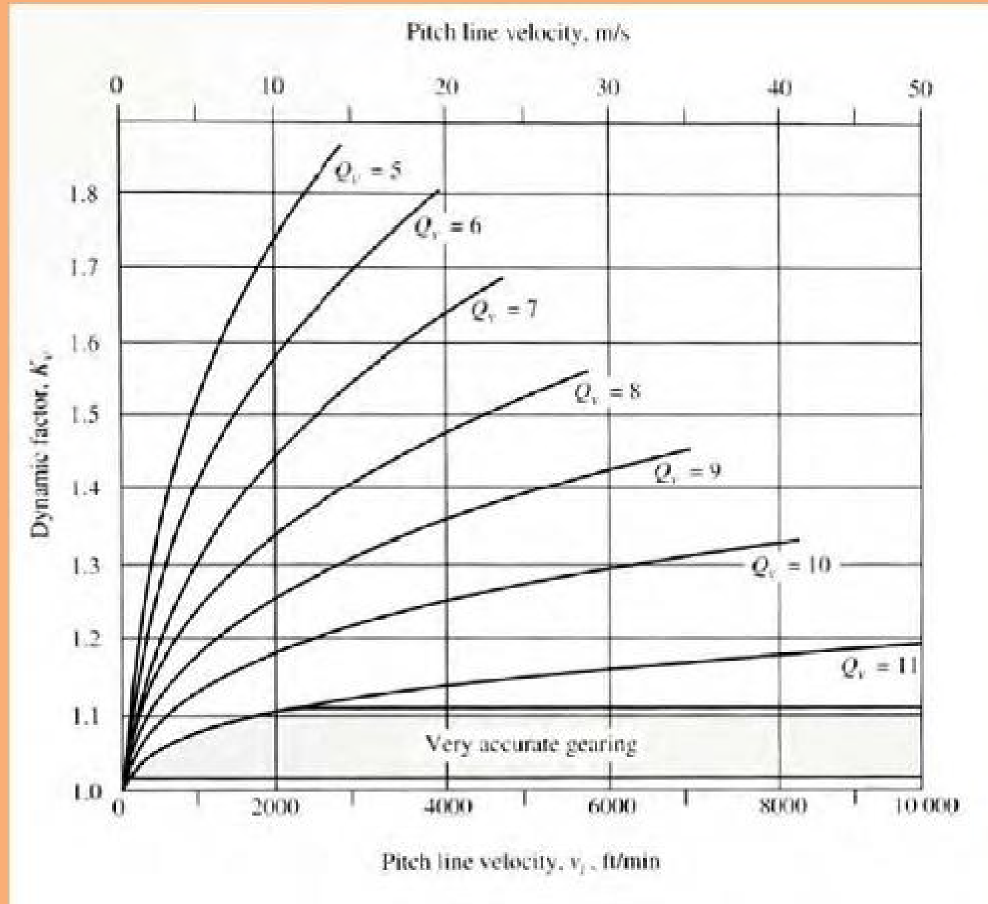
Where C_{pf}***= pinion proportion factor from figure (9-18) page(391) (407 pdf)***

C_{ma}***= mesh alignment factor from figure (9-19) page(391) (407 pdf)***





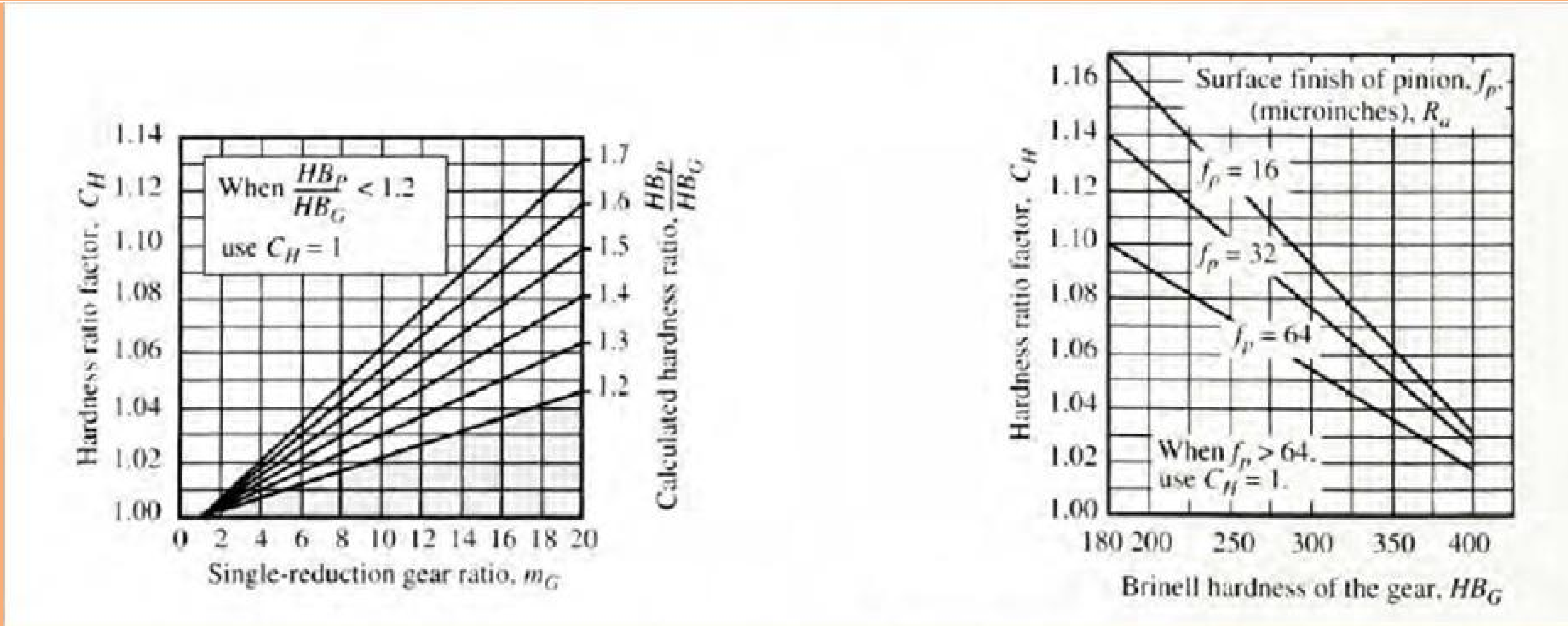
Specify the dynamic factor (K_v) from figure (9-21) page (393) (409 pdf):



11

Specify the safety factor (S.F) typically from 1 to 1.5

Specify the hardness ratio factor from Figure (9-25 & 26) page (404) (420 pdf)



12

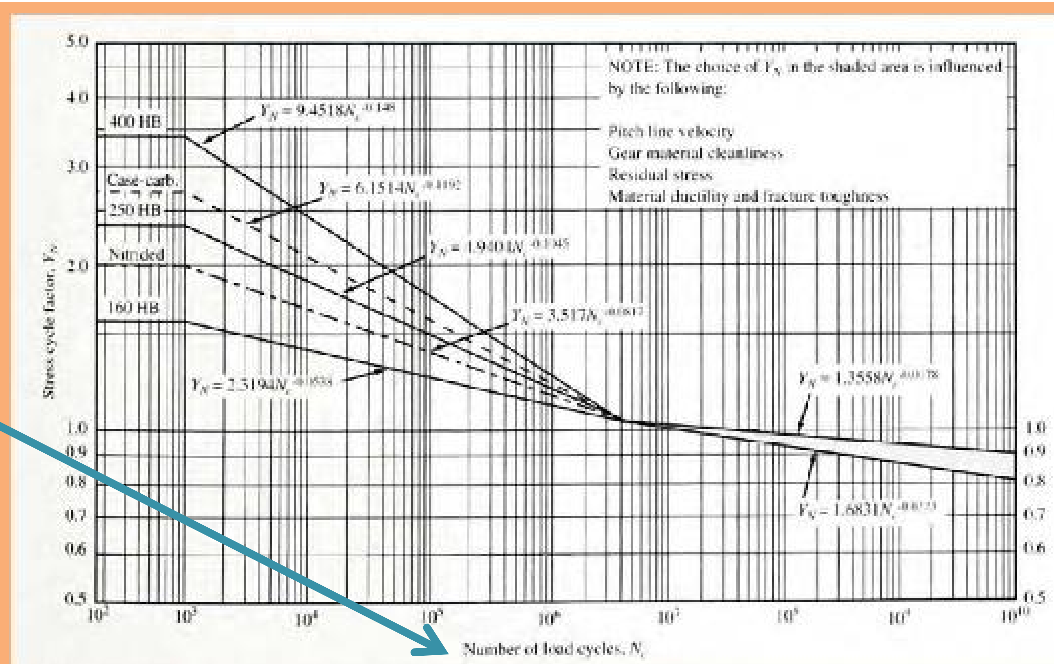
Specify the reliability factor (K_R) from Table (9-8) page (396) (412 pdf):

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

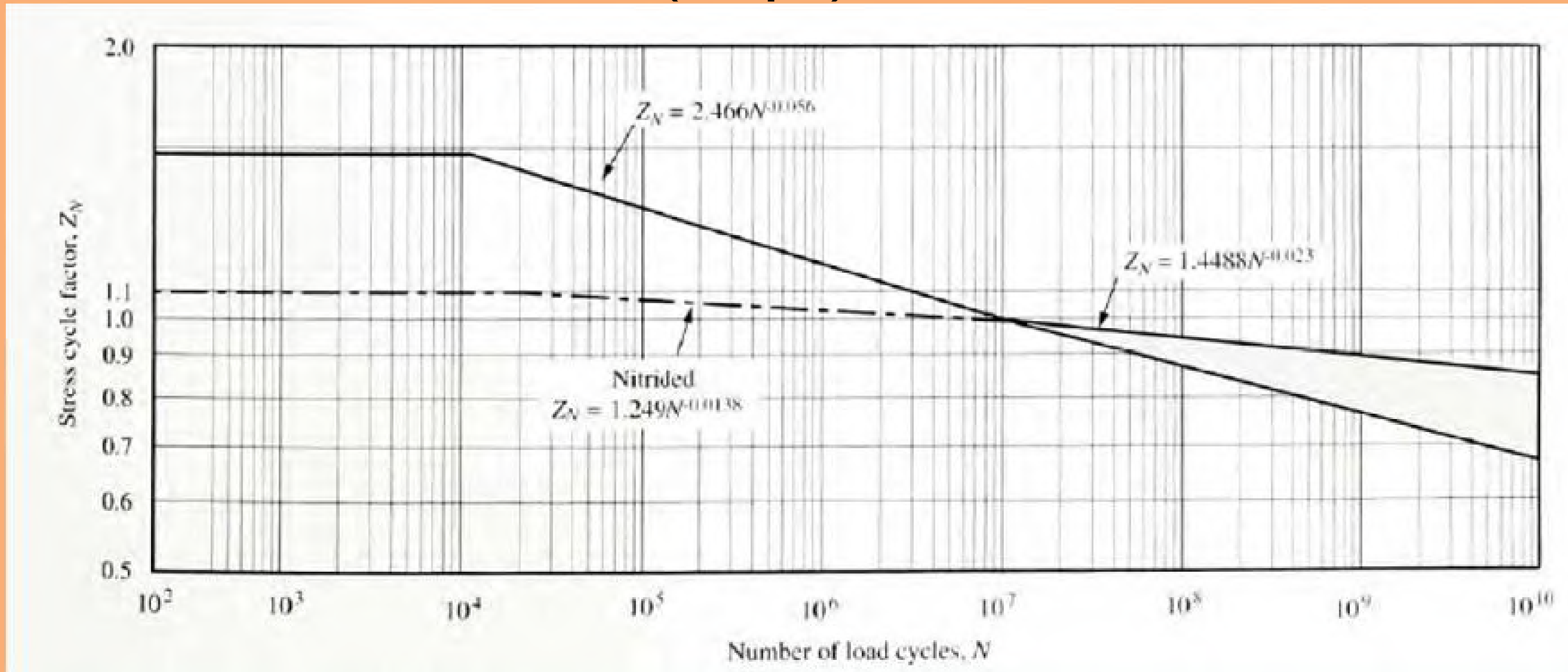
Specify the stress cycle life (Y_N) from Figure (9-8) page (395) (411 pdf):

TABLE 9–7 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



Specify the pitting resistance stress cycle factor (Z_N) from figure (9-24) page (403) (419 pdf):



Choose material for pinion and gear or (S_{ac} , S_{at}) from figures [(9-10) page (379) (395pdf) , (9-11) page (380) (396pdf)] with tables [(9-3) page(381) (397pdf) , (9-4) page (385) (401pdf)] and see also Appendix 3 to 5 [p(A-6) to (A-11)].

Check if the selected material satisfy the following design conditions:

$$S_t \frac{K_R(S.F)}{Y_N} < S_{at}$$

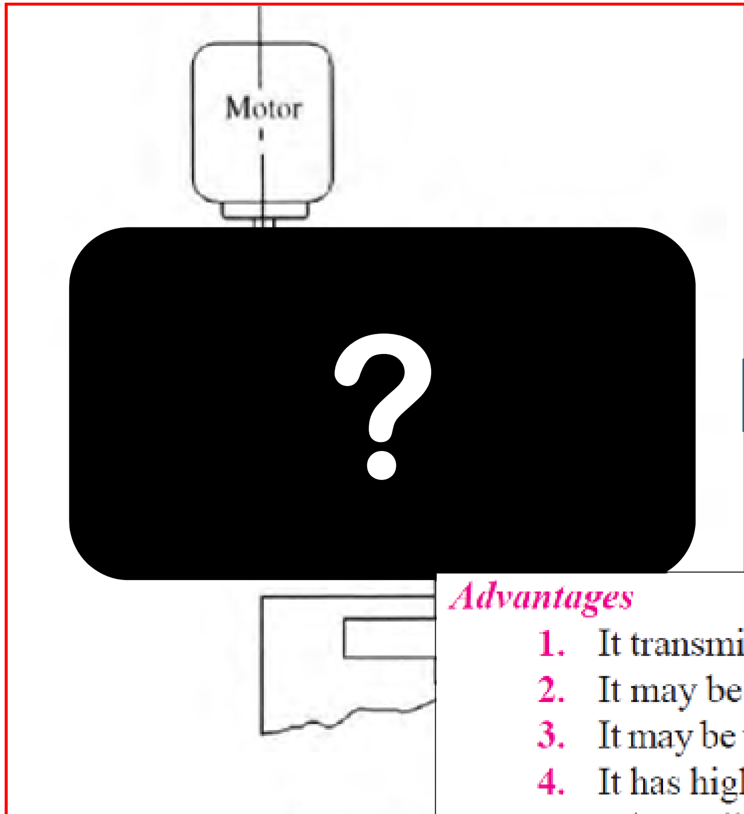
$$S_c \frac{K_R(S.F)}{Z_N C_H} < S_{ac}$$

Mechanical Engineering Design II

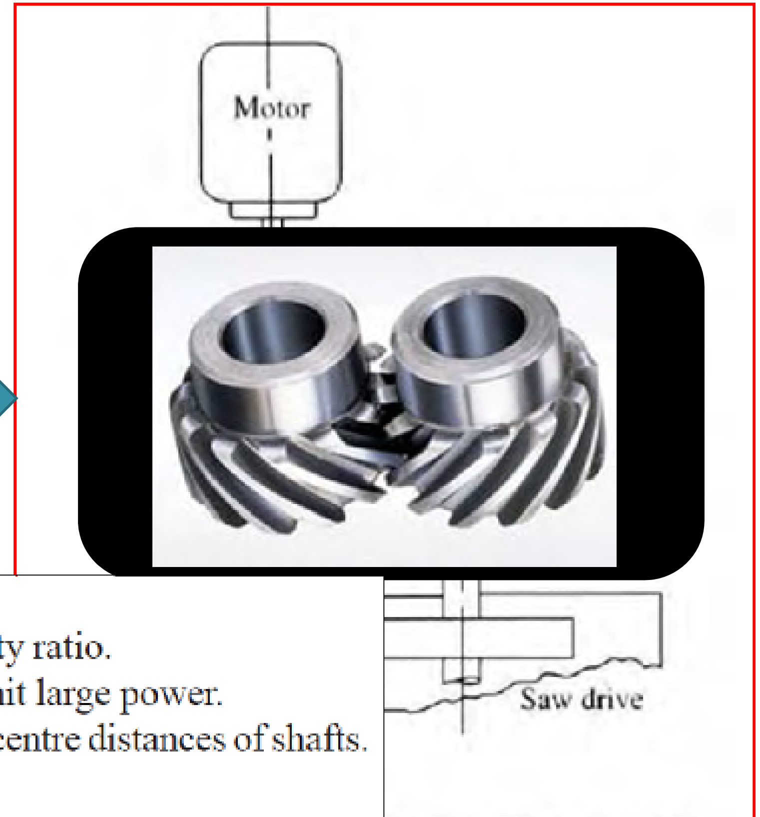
Twentieth Lecture

Design of Helical Gear

Power Transmission Problem



Proposed solution (Helical Gear)



Advantages

1. It transmits exact velocity ratio.
2. It may be used to transmit large power.
3. It may be used for small centre distances of shafts.
4. It has high efficiency.
5. It has reliable service.
6. It has compact layout.

Disadvantages

1. Since the manufacture of gears require special tools and equipment, therefore it is costlier than other drives.

Gear Manufacturing

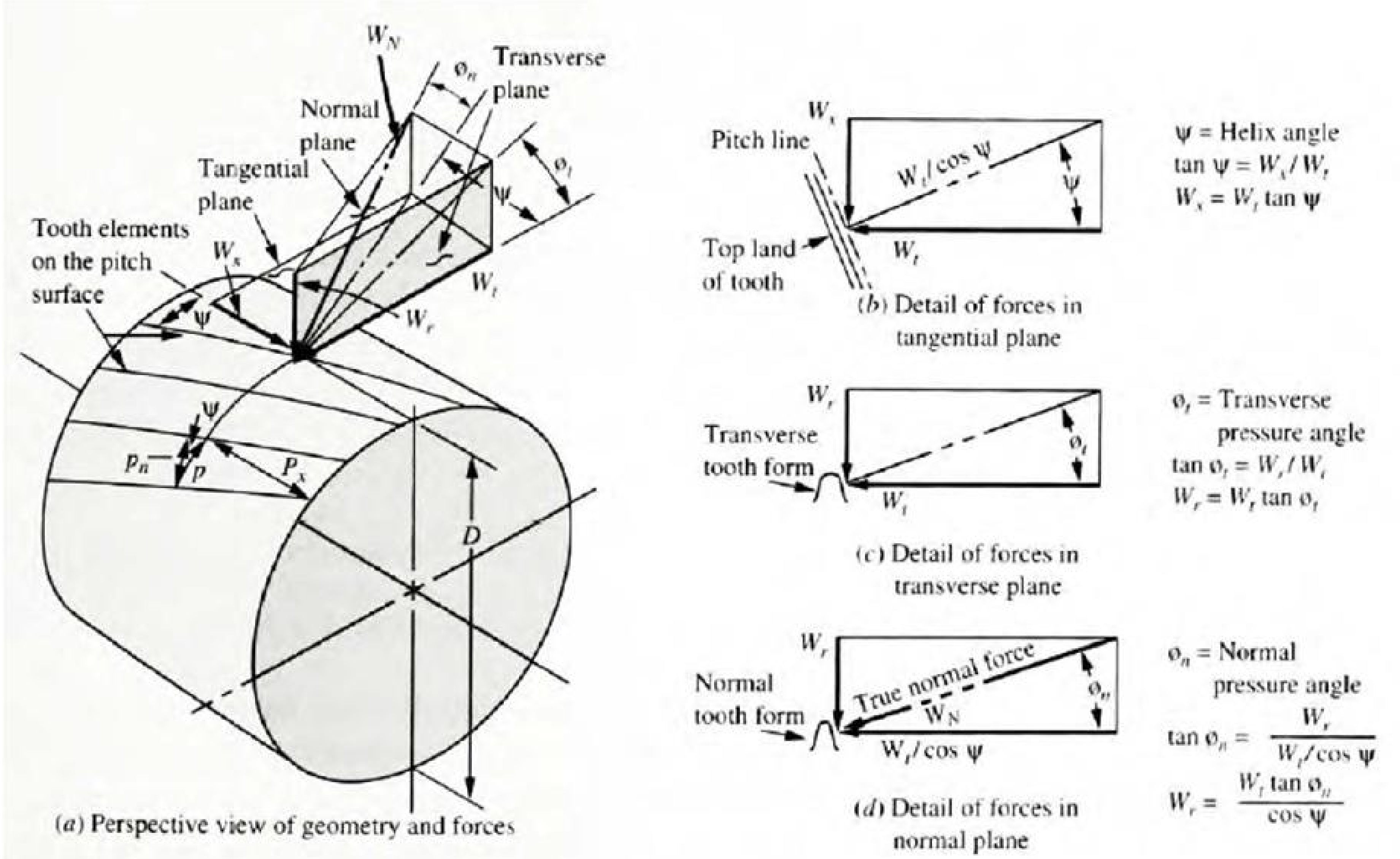
(a) Form milling cutter

(b) Spur gear shaper cutter

(c) Hob for small pitch gears having large teeth

(d) Hob for high pitch gears having small teeth

Basic Helical Gear Geometry and force Kinematics



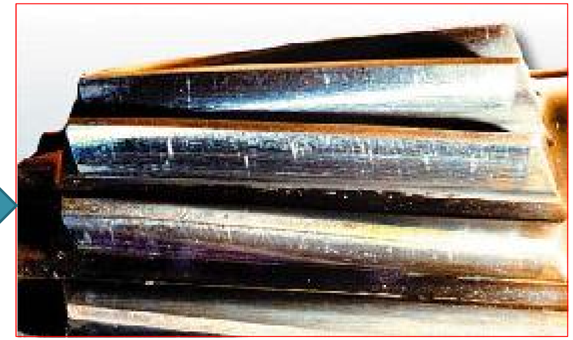
Modes of Gear Tooth Failure

1. Bending failure.

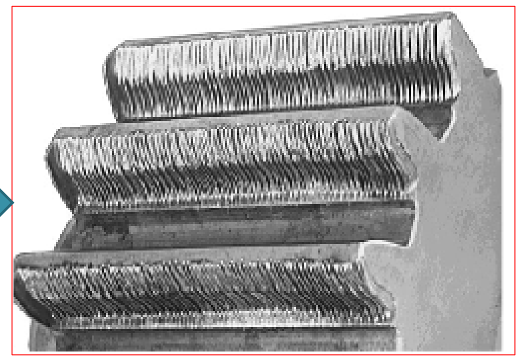
Root failure

2. Pitting.

3. Scoring.



4. Abrasive wear.



5. Corrosive wear.

Surface failure

Dangerous Bar

Helical Gear Design

The power to be transmitted

Type of driver and driven load

The speed of the driving gear

The center distance

The speed of the driven gear or the velocity ratio

Other information related to problem specification

Designer

The gear teeth should not fail under static loading or dynamic loading during normal running conditions.

The gear teeth should have wear characteristics so that their life is satisfactory.

The use of space and material should be economical.

The alignment of the gears and deflections of the shafts must be considered.

The lubrication of the gears must be satisfactory.

Flowchart for spur gear designing process:

Transmitted Power , Input and Output speed, Center distance, Type of driver and driven load

Choose the over load factor (K_o) from Table (9-5) page(389) (405pdf)

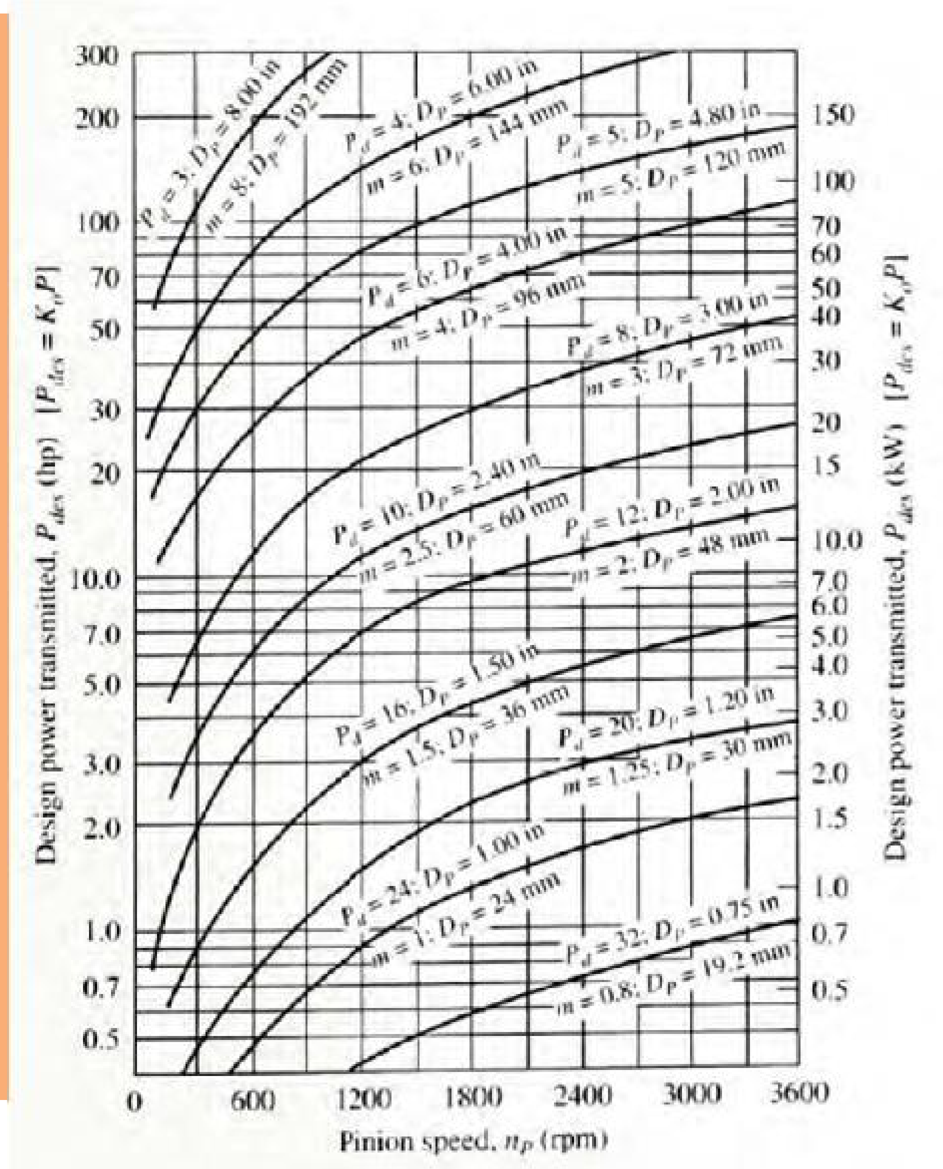
	Driven Machine			
Power source	Uniform	Light shock	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.50	1.75
Light shock	1.20	1.40	1.75	2.25
Moderate shock	1.30	1.70	2.00	2.75

Compute the design power
Design Power=K_o x transmitted power

1

1

Find the trial value for Diametral pitch (P_{dn}) or Module (m) and Pitch Diameter (D_p) from Figure 9-27 page.409 (425 Pdf)



2

2

Specify the no. of teeth for Pinion N_P (from 17 to 20)

Compute the nominal velocity ratio $VR = \frac{n_P}{n_G}$

Compute the approximate no. of teeth for Gear $N_G = N_P \times VR$

Compute the actual velocity ratio $VR = \frac{N_G}{N_P}$

Compute the actual output velocity $n_G = n_P \frac{N_P}{N_G}$

Compute the pitch diameters $D_p = \frac{N_p}{P_d}$, $D_G = \frac{N_G}{P_d}$, center distance $c = \frac{N_P+N_G}{2P_d}$, pitch line speed $v_t = \frac{\pi D_p n_P}{60}$, axial pitch $P_x = \frac{\pi}{P_d \tan\psi}$, $P_d = P_{dn} \cos\psi$ and tangential force, $W_t = \frac{60P}{\pi D_P n_p}$

3

3

Compute the face width : $F = 2\,P_x$

Specify the quality number Q_v , from Table (9-2) page (378) (394 pdf)

Application	Quality number	Application	Quality number
Cement mixer drum drive	3–5	Small power drill	7–9
Cement kiln	5–6	Clothes washing machine	8–10
Steel mill drives	5–6	Printing press	9–11
Grain harvester	5–7	Computing mechanism	10–11
Cranes	5–7	Automotive transmission	10–11
Punch press	5–7	Radar antenna drive	10–12
Mining conveyor	5–7	Marine propulsion drive	10–12
Paper-box-making machine	6–8	Aircraft engine drive	10–13
Gas meter mechanism	7–9	Gyroscope	12–14

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

Pitch line speed (fpm)	Quality number	Pitch line speed (m/s)
0–800	6–8	0–4
800–2000	8–10	4–11
2000–4000	10–12	11–22
Over 4000	12–14	Over 22

4

4

Analyzing of gear tooth failure mode

Root (Bending) Failure Mode

Bending Stress Number

$$S_t = \frac{W_t P_d}{F\,J} K_o K_s K_m K_B K_v < S_{at} \frac{Y_N}{K_R (S.F)}$$

Surface (Pitting, Scoring,…) Failure Mode

Contact Stress Number

$$S_c = C_p \sqrt{\frac{W_t K_o K_s K_m K_v}{F D_p I}} < S_{ac} \frac{Z_N C_H}{K_R (S.F)}$$

Find the values of factors ($J, I, K_s, K_m, K_B, K_v, C_p, Y_N, Z_N, C_H, K_R, S.F$) as in the following steps

5

5

Specify the type of material for the gears to find the Elastic Coefficient C_p from Table (9-9) page(400) (Pdf 416)

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

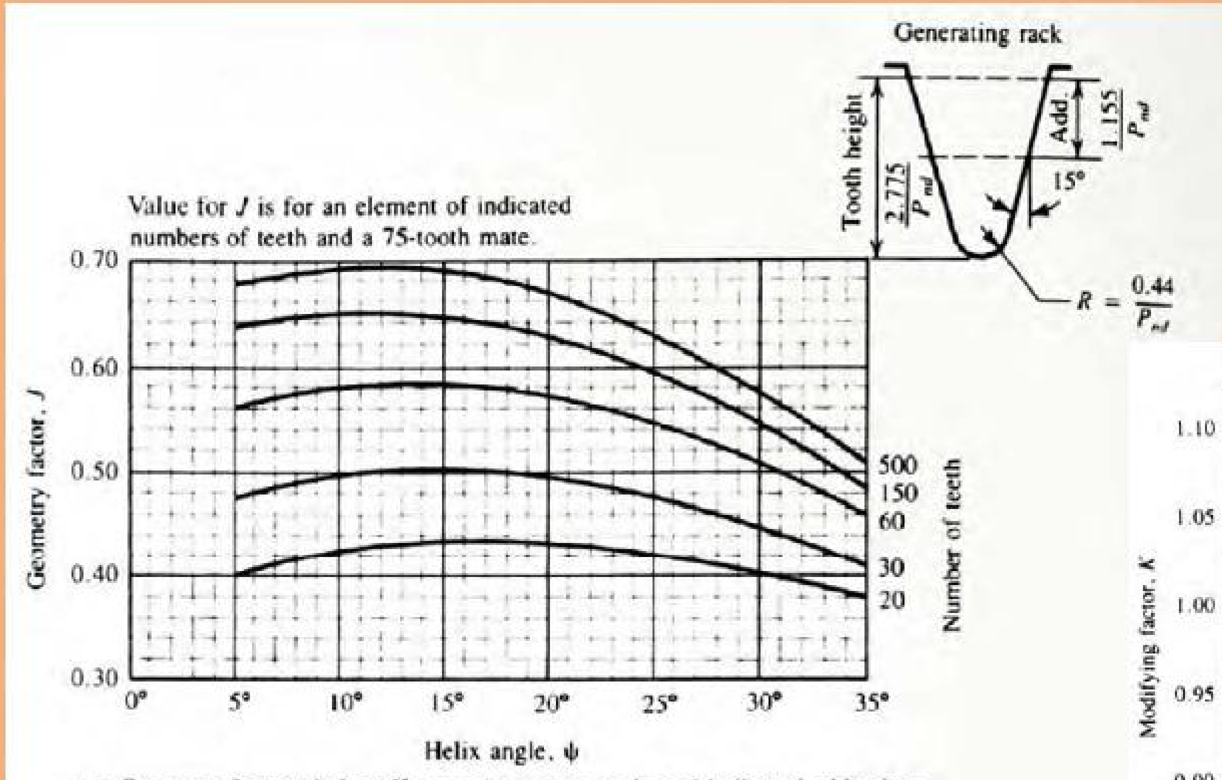
Source: Extracted from AGMA Standard 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, VA 22314.

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

6

6

Specify the bending geometry factor (J) for 15º normal pressure angle from figure (10-5) page (456) (472pdf):

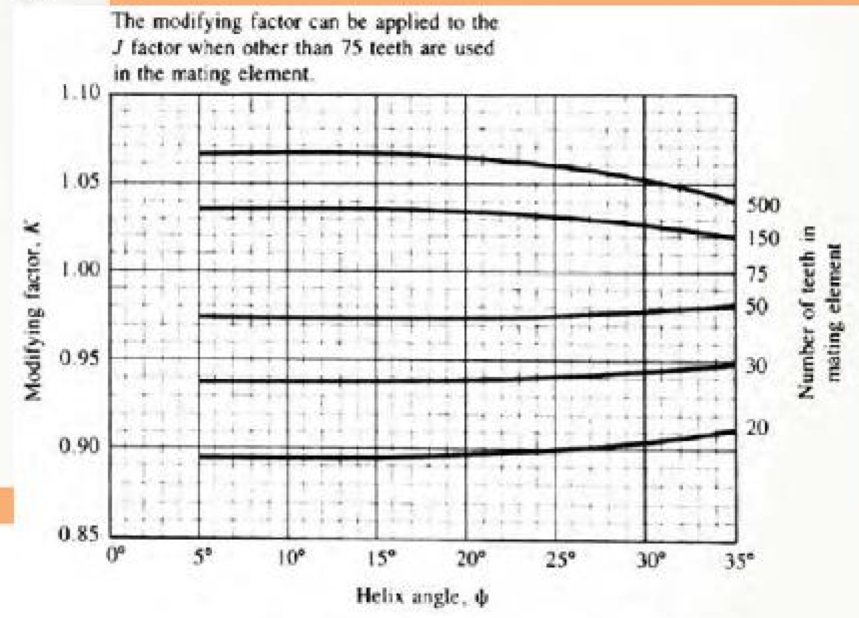


(a) Geometry factor (J) for 15° normal pressure angle and indicated addendum

$$J = \frac{Y}{K_t}$$

$$Y = Lewis\ factor$$

$$K_t = stress\ concentration\ factor$$

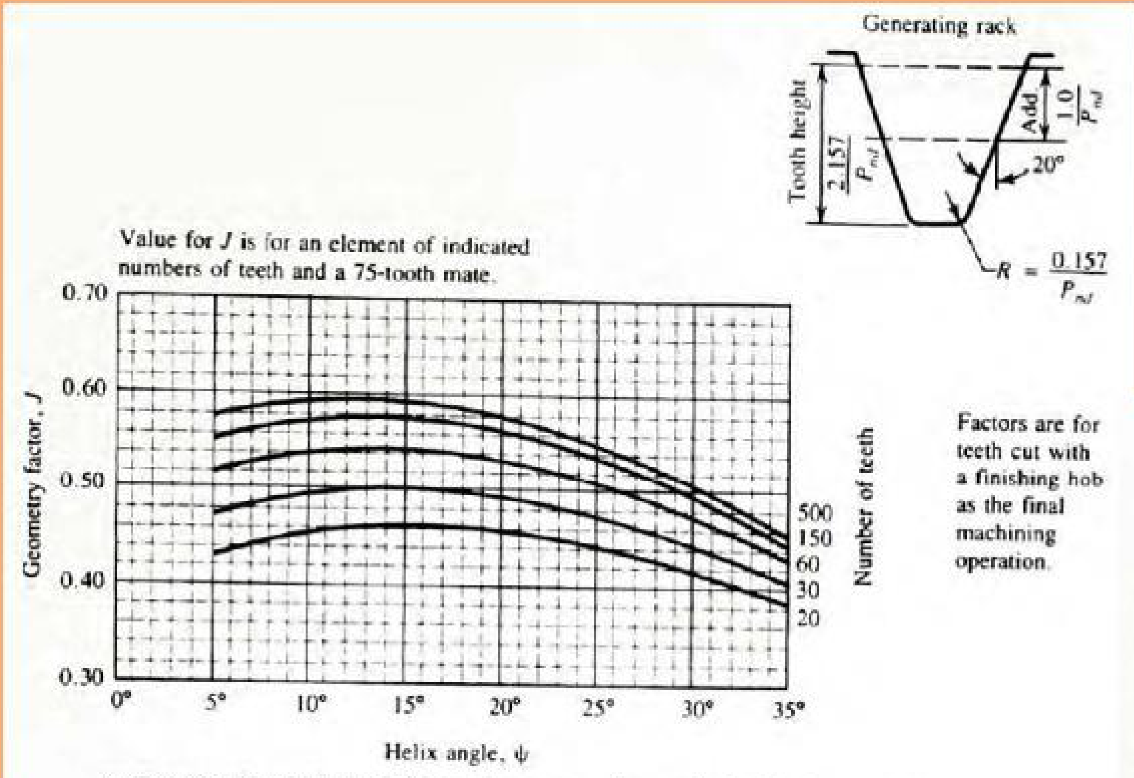


(b) J factor multipliers

7

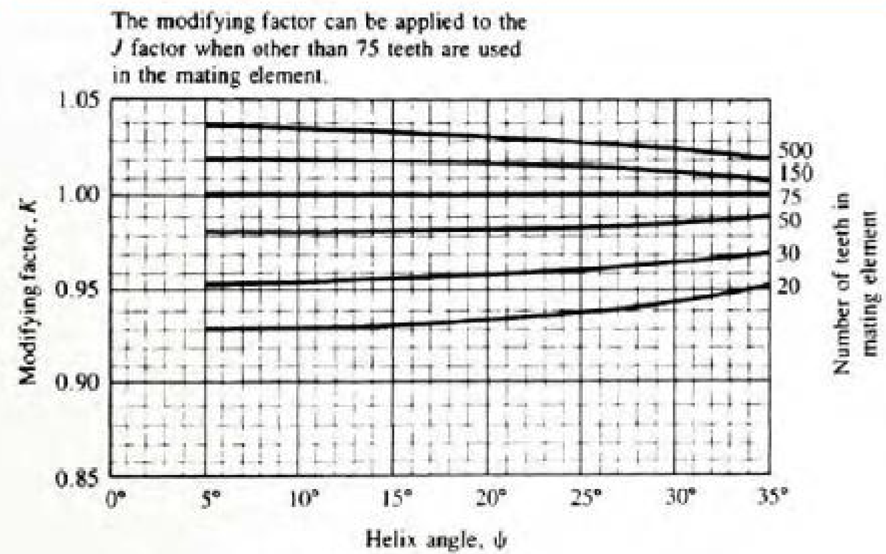
7

Specify the bending geometry factor (J) for 20º normal pressure angle from figure (10-6) page (457) (473pdf):



$\frac{2.157}{P_{nd}}$ $\frac{1.0}{P_{nd}}$ $R = \frac{0.157}{P_{nd}}$

(a) Geometry factor (J) for 20° normal pressure angle, standard addendum, and finishing hob

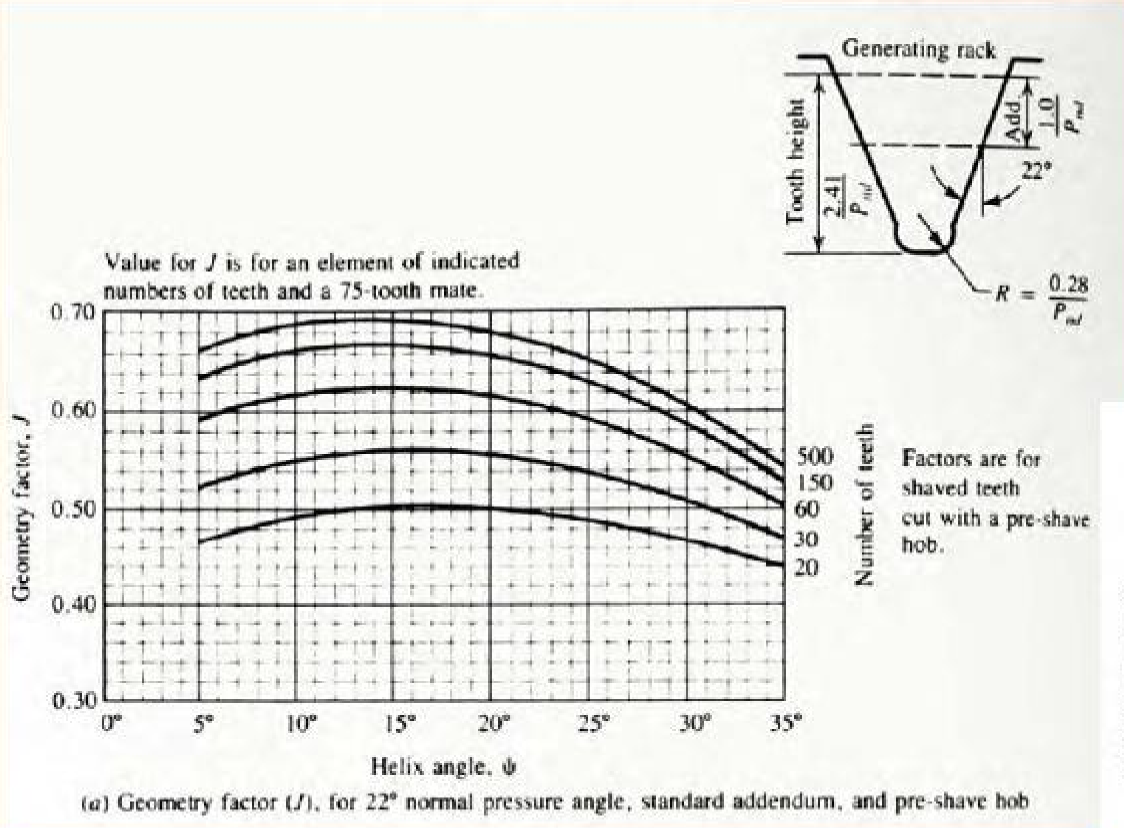


(b) J factor multipliers

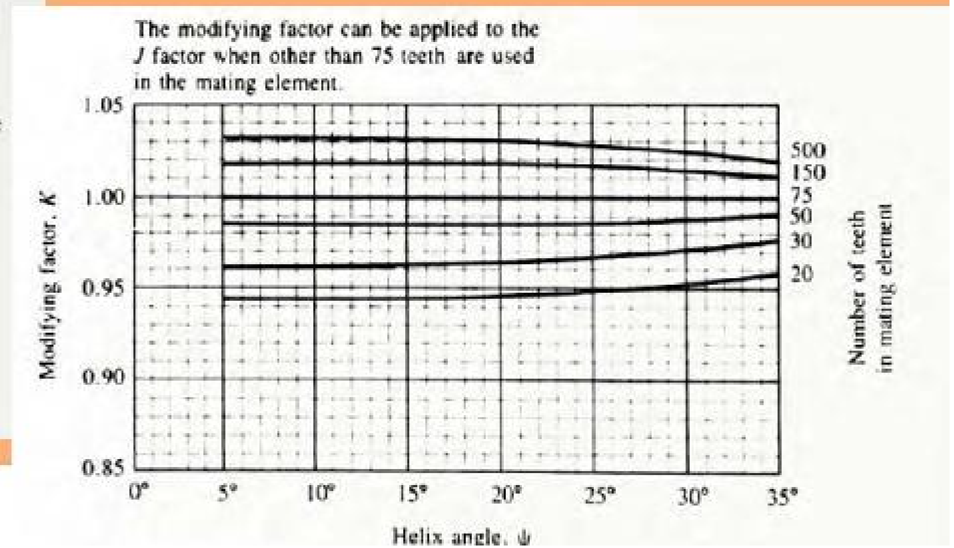
8

8

Specify the bending geometry factor (J) for 22° normal pressure angle from figure (10-7) page (458) (474pdf):



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



9

Specify the pitting geometry factor (I) with 20º normal pressure angle from Table(10-1) page (459) (475pdf):

A. Helix angle $\psi = 15.0°$

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 $\psi = 25.0°$

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

Specify the pitting geometry factor (I) with 20° normal pressure angle from Table(10-2) page (460) (476pdf):

A. Helix angle $\psi = 15.0°$

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

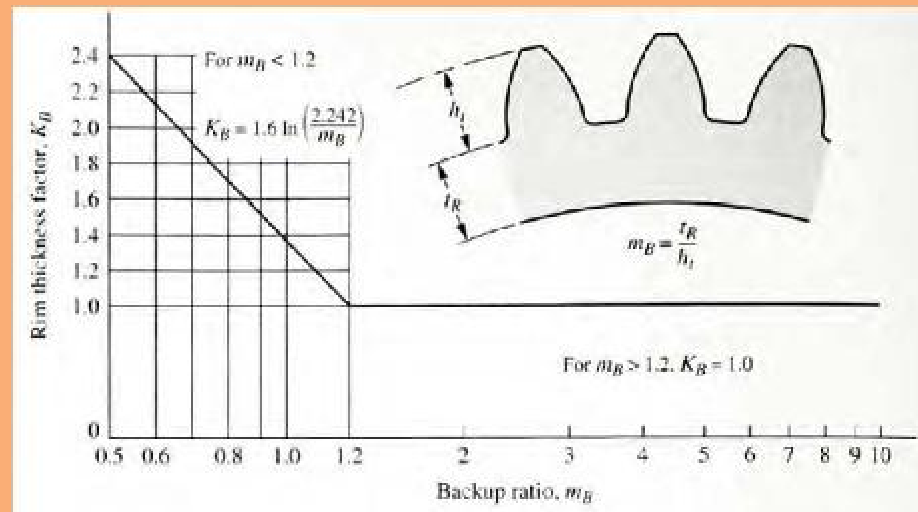
B. Helix angle $\psi = 25.0°$

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

Specify the size factor (K_S) from Table(9-6) page (389) (293 pdf)

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

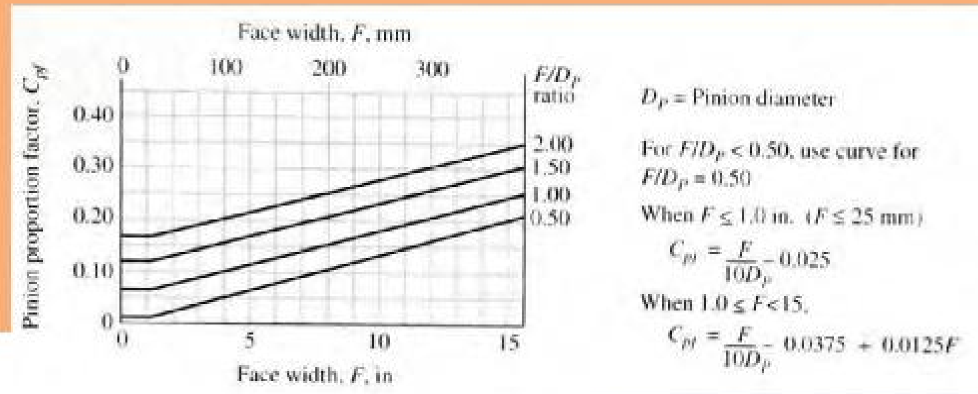
Specify the rim thickness factor (K_B) from Figure (9-20) page (392) (408 pdf)

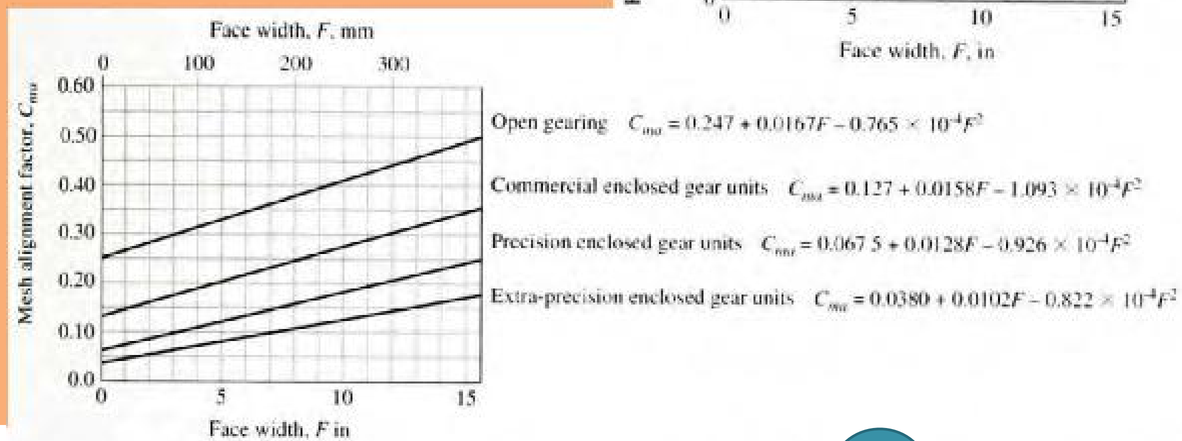


Determine the load distribution factor (K_m) : $K_m = 1.0 + C_{pf} + C_{ma}$

Where C_{pf}***= pinion proportion factor from figure (9-18) page(391) (407 pdf)***

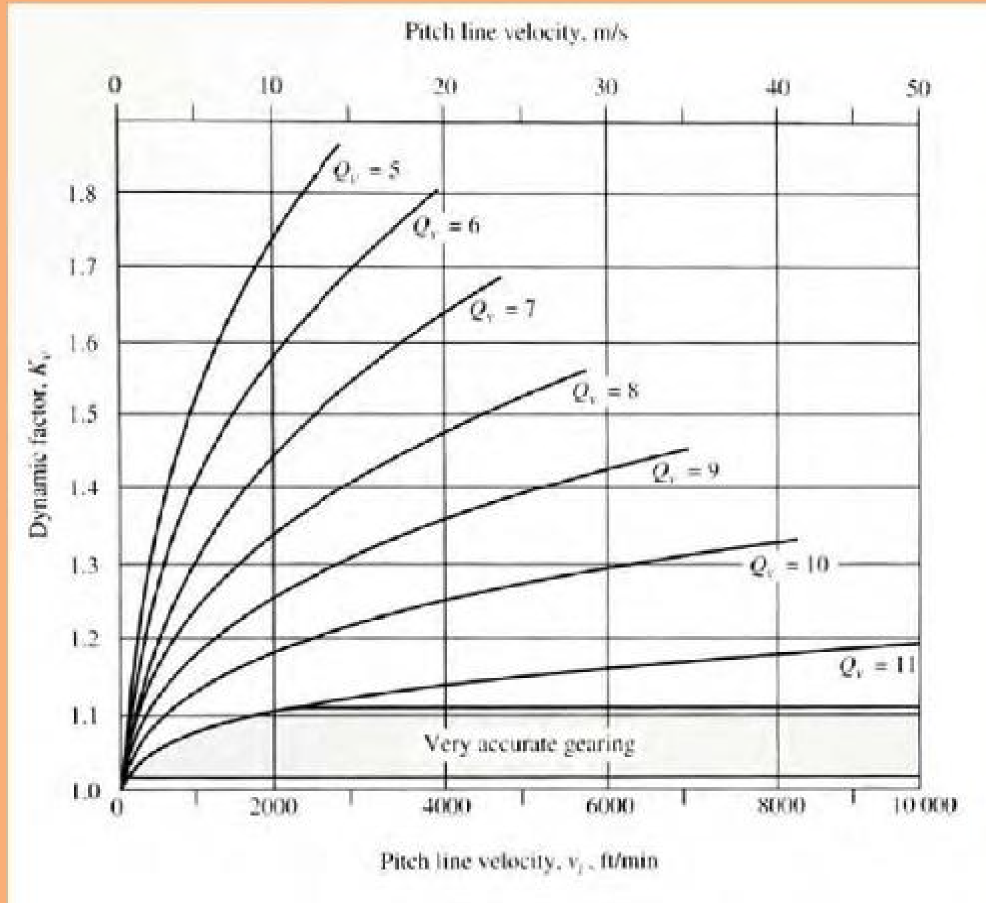
C_{ma}***= mesh alignment factor from figure (9-19) page(391) (407 pdf)***





14

Specify the dynamic factor (K_v) from figure (9-21) page (393) (409 pdf):

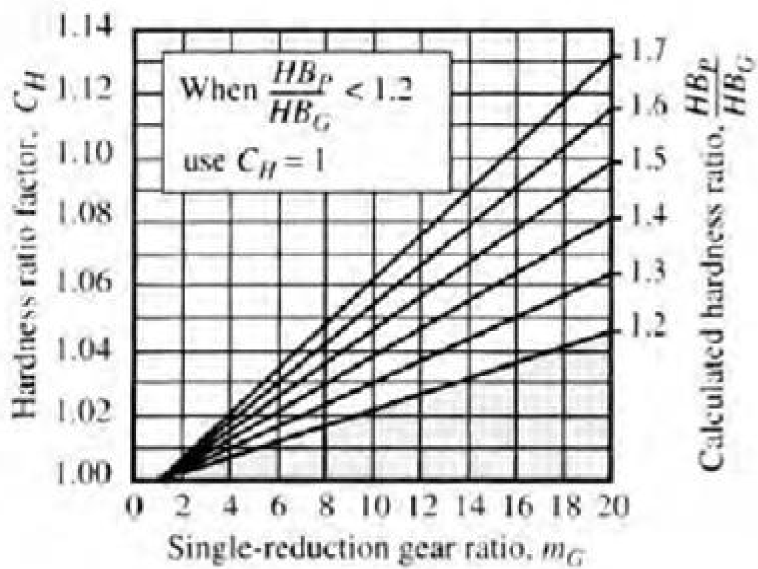


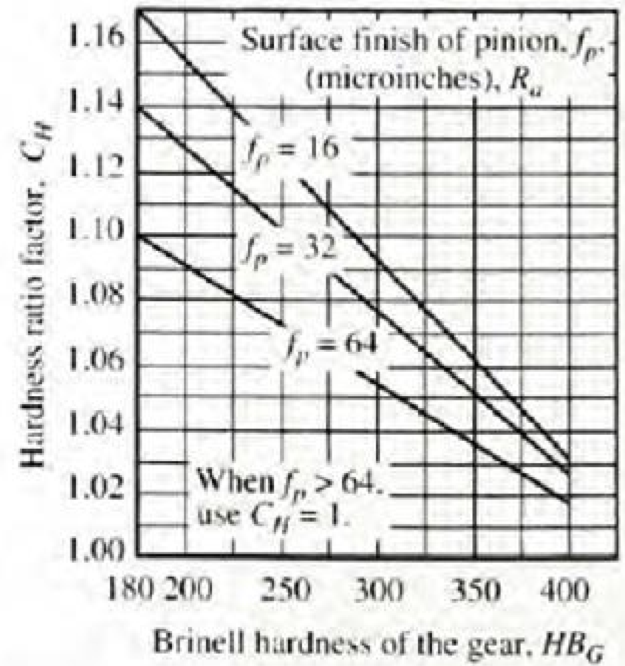
15

15

Specify the safety factor (S.F) typically from 1 to 1.5

Specify the hardness ratio factor (C_H) from Figure (9-25 & 26) page (404) (420 pdf)





16

Specify the reliability factor (K_R) from Table (9-8) page (396) (412 pdf):

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

Specify the stress cycle life (Y_N) from Figure (9-8) page (395) (411 pdf):

TABLE 9–7 Recommended design life

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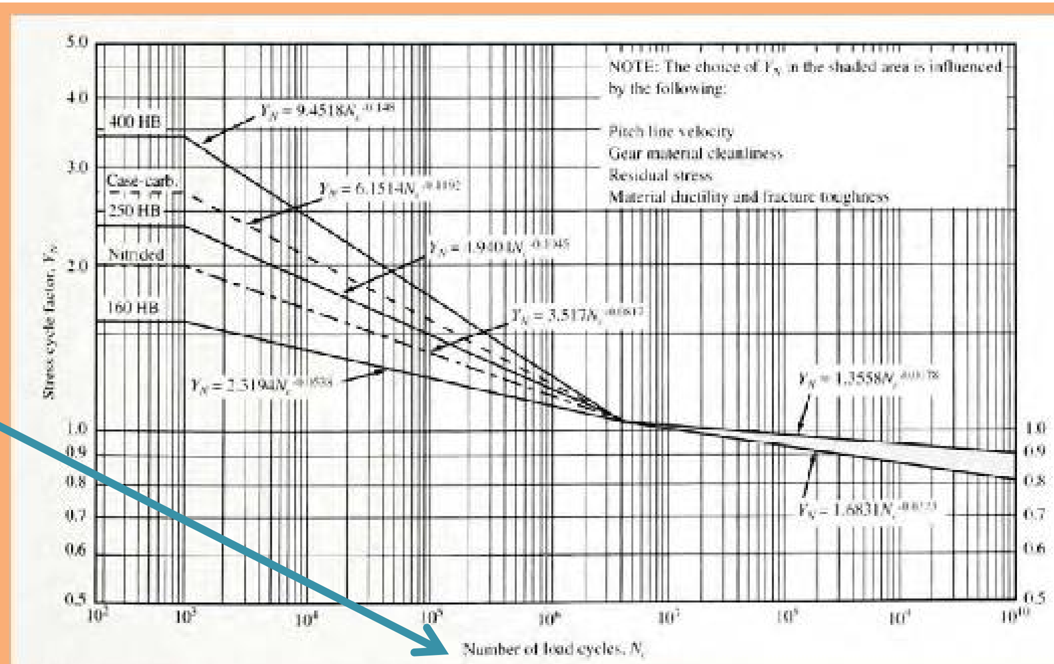
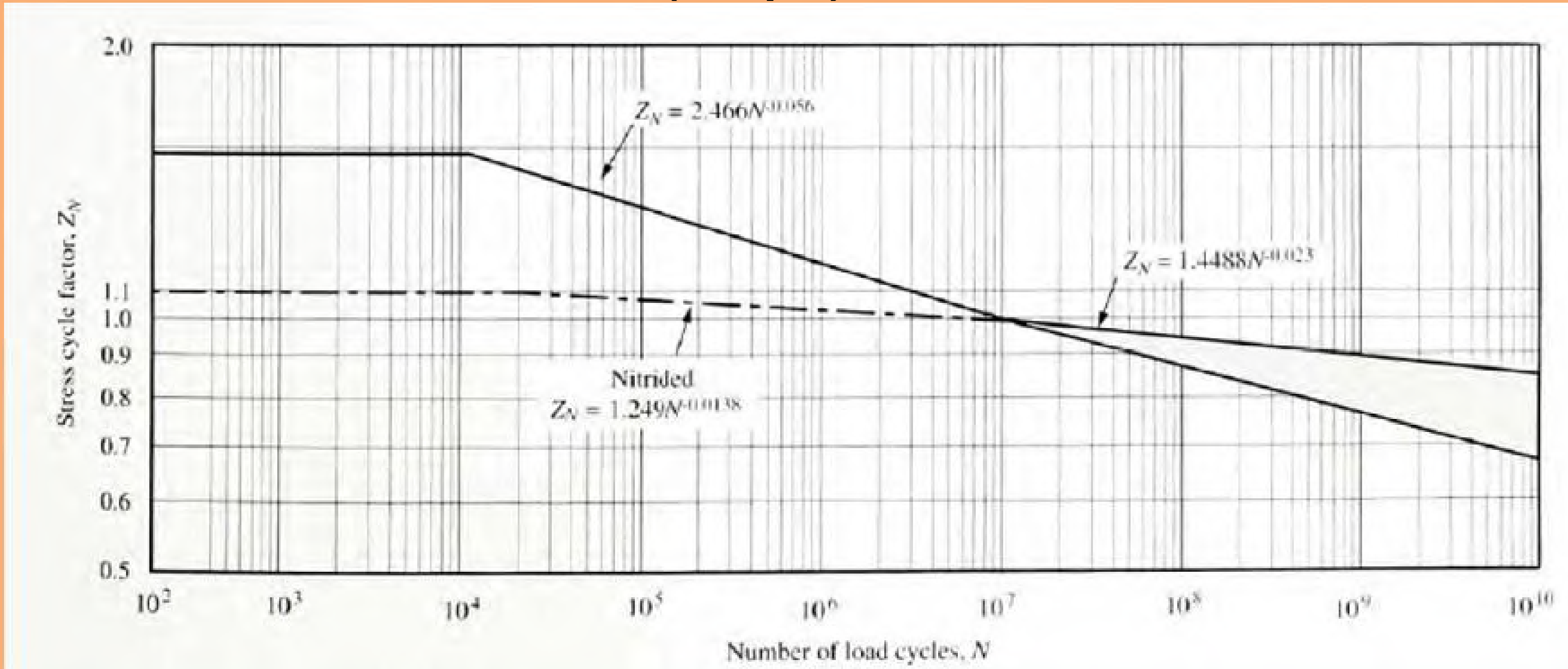
Elevators, industrial fans, multipurpose gearing	8000–15 000
Electric motors, industrial blowers, general industrial machines	20 000–30 000
Pumps and compressors	40 000–60 000
Critical equipment in continuous 24-h operation	100 000–200 000



Specify the pitting resistance stress cycle factor (Z_N) from figure (9-24) page (403) (419 pdf):



Choose material for pinion and gear or (S_{ac} , S_{at}) from figures [(9-10) page (379) (395pdf) , (9-11) page (380) (396pdf)] with tables [(9-3) page(381) (397pdf) , (9-4) page (385) (401pdf)] and see also Appendix 3 to 5 [p(A-6) to (A-11)].

Check if the selected material satisfy the following design conditions:

$$S_t \frac{K_R(S.F)}{Y_N} < S_{at}$$

$$S_c \frac{K_R(S.F)}{Z_N C_H} < S_{ac}$$

Example (10-2) p.461(477pdf):

A pair of helical gears for a milling machine drive is to transmit 48.47 kW (65 hp) with a pinion speed of 3450 rpm and a gear speed of 1100 rpm. The power is from an electric motor. Design the gears.

Solution:

Given data:

P = transmitted power = 48.47 kW (65 hp)

Power source = electric motor , driven machine = milling machine

$n_p = 3450\ rpm$, $n_G = 1100\ rpm$

Initial assumptions:

$$P_{d_n} = 12 \,, N_p = 24 \,, \psi = 15^o \,, \phi = 20^o \,, Q = 8$$

Basic dimensions computations:

$$P_d = P_{dn}\cos\psi = 12\cos(15^o) = 11.59\ mm$$

$$P_x = \frac{\pi}{P_d\tan\psi} = \frac{\pi}{11.59\tan(15^o)} = 25.7\ mm$$

$$\phi_t = \tan^{-1}(\tan\phi_n/\cos\psi) = tan^{-1}\left[tan\left(20^0\right)/cos(15^o)\right] = 20.65^o$$

$$D_p = {}^{N_p}\!/_{P_d} = {}^{24}\!/_{11.59} = 2.07\ in = 52.6\ mm$$

$$F = 2P_x = 2(25.7) = 51.41\ mm = 2.25\ in$$

Gear kinematics computations:

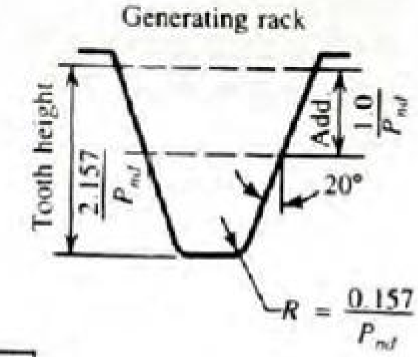
$$v_t = \frac{\pi D_p n_P}{60} = \frac{\pi\left(52.6\times 10^{-3}\right)(3450)}{60} = 9.5\frac{m}{s}$$

$$W_t = \frac{60P}{\pi D_P n_p} = \frac{60\left(48.47\times 10^3\right)}{\pi(52.6\times 10^{-3})(3450)} = 5.1\ kN = 1146\ lb$$

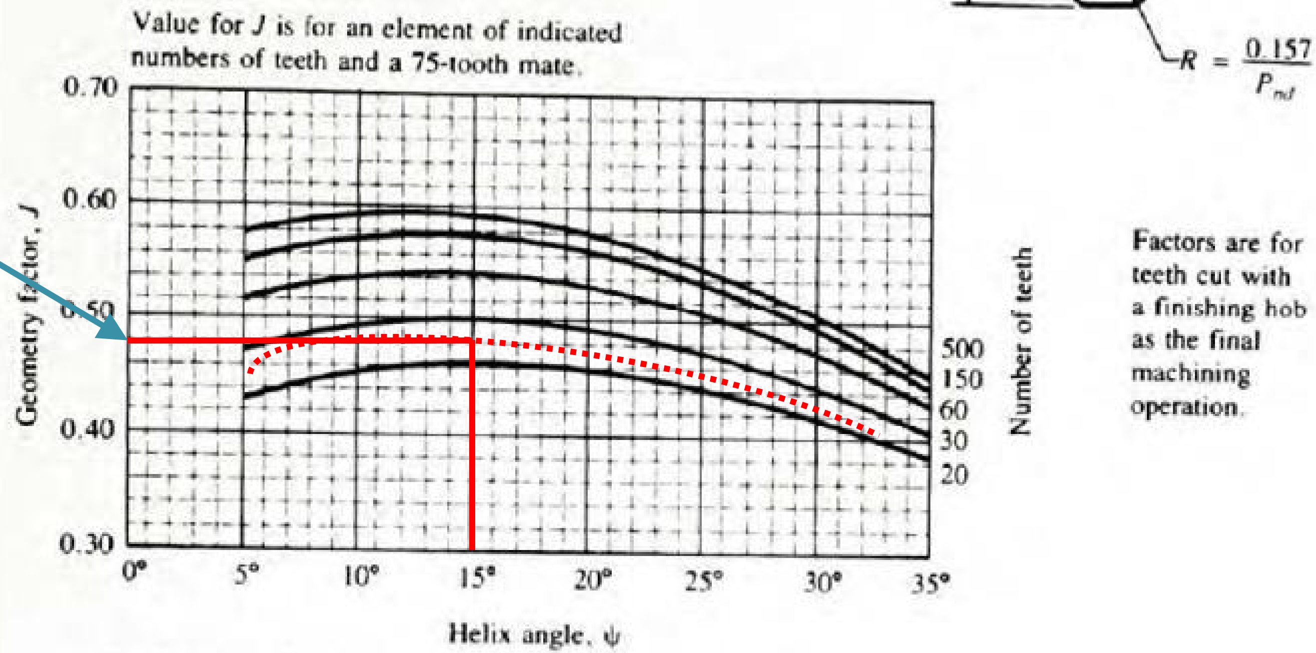
$$VR = {}^{N_G}\!/_{N_P} = {}^{n_P}\!/_{n_G} = {}^{3450}\!/_{1100} = 3.14\,,$$

$$N_G = N_P(VR) = 24(3.14) = 75.3 \cong 75\ teeth$$

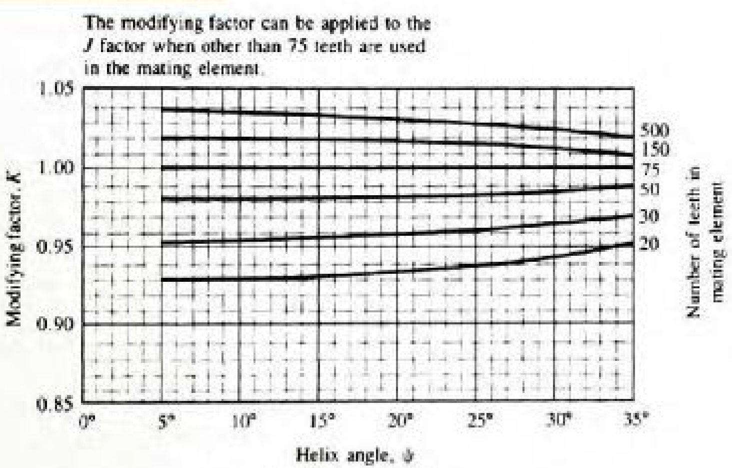
J_p from N_p=24 and N_G=75 (figure 10-6):



J_p=0.48



(a) Geometry factor (J) for 20° normal pressure angle, standard addendum, and finishing hob



(b) J factor multipliers

K_o from power source (electric motor) and driven machine (milling machine)(table 9-5):

Power source	Driven Machine: 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

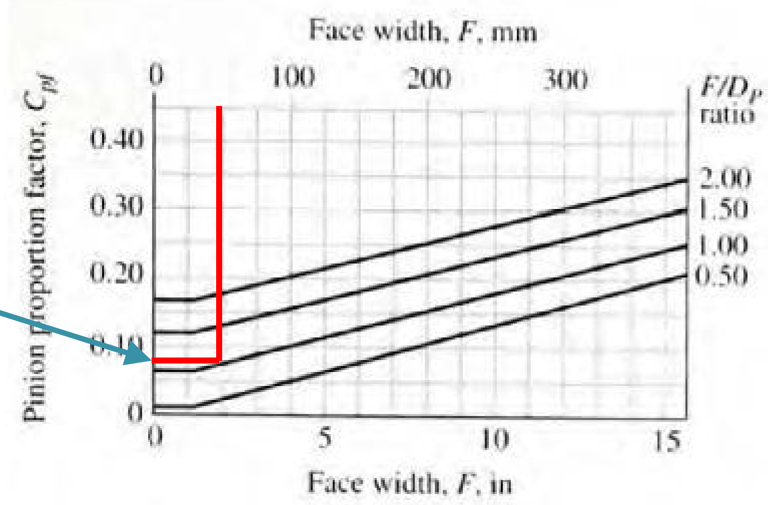
K_s from $P_d = 11.59 > 5$ (table 9-6):

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

K_m from F= 51.41 mm & D_p =52.6 mm , F/D_p =0.977 (figure 9-18 & 19):

$$K_m = 1.0 + C_{pf} + C_{ma}$$
$$=1+0.09+0.17=1.26$$

C_{pf}=0.09



D_p = Pinion diameter

For $F/D_p < 0.50$, use curve for $F/D_p = 0.50$

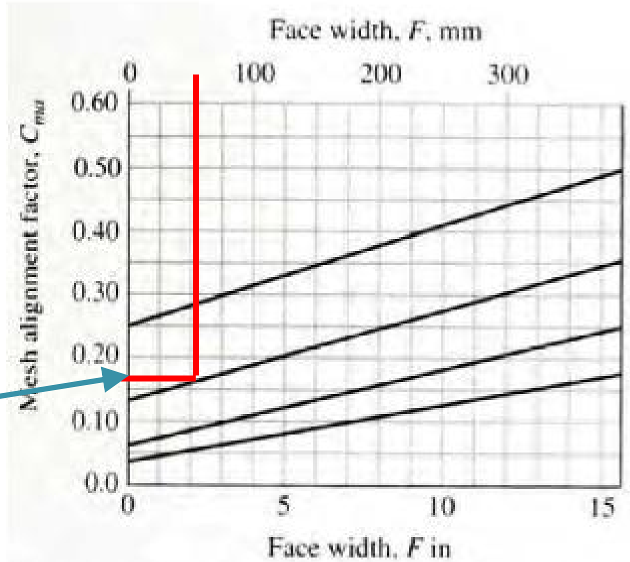
When $F \le 1.0$ in. ($F \le 25$ mm)

$$C_{pf} = \frac{F}{10D_p} - 0.025$$

When $1.0 \le F < 15$,

$$C_{pf} = \frac{F}{10D_p} - 0.0375 + 0.0125F$$

C_{mf}=0.17



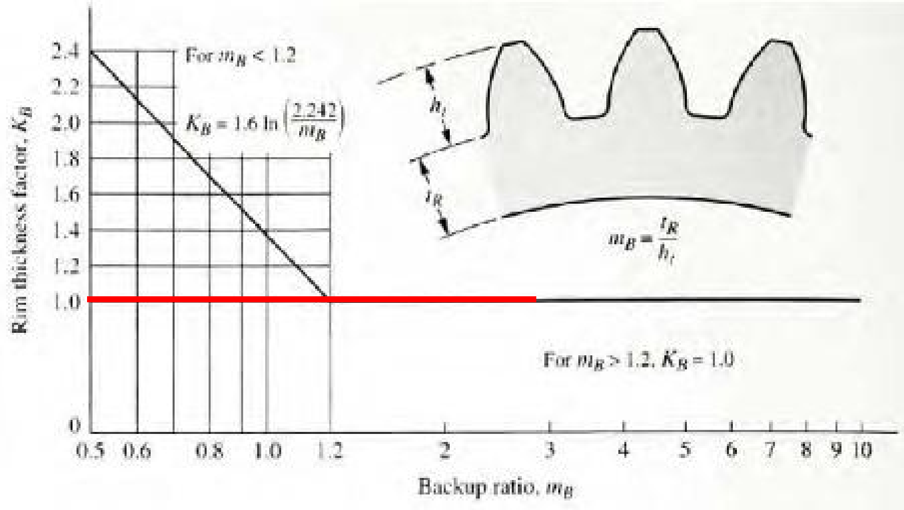
Open gearing $C_{ma} = 0.247 + 0.0167F - 0.765 \times 10^{-4}F^2$

Commercial enclosed gear units $C_{ma} = 0.127 + 0.0158F - 1.093 \times 10^{-4}F^2$

Precision enclosed gear units $C_{ma} = 0.0675 + 0.0128F - 0.926 \times 10^{-4}F^2$

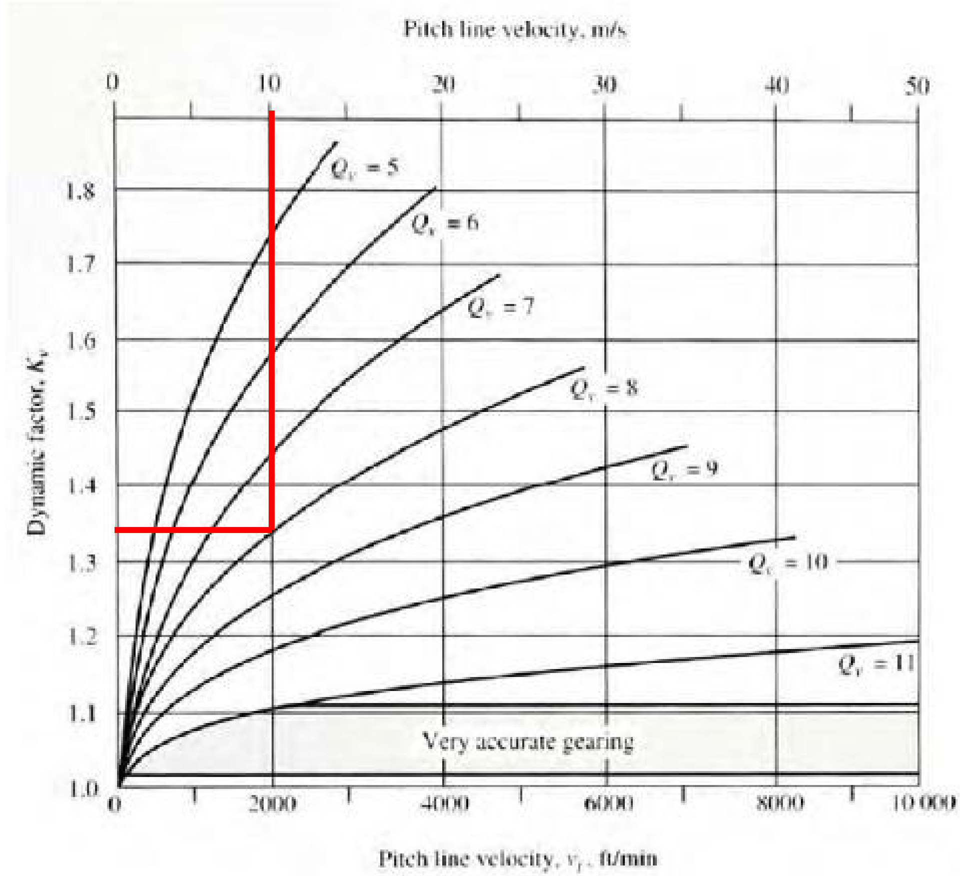
Extra-precision enclosed gear units $C_{ma} = 0.0380 + 0.0102F - 0.822 \times 10^{-4}F^2$

$K_B = 1$ *from (solid gear) (figure 9-20):*



$K_v = 1.35$

from $Q_v = 8$ & $v_t = 9.5\ m/s$ *(figure 9-21):*



Design for reliability of 0.999 (less than one failure in 1000): K_R=1.25

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

No unusual conditions seem to exist in this application beyond those already considered in the various K factors. Therefore we use a service factor S.F of 1.0

Design life: Let's design for 10 000 h of life as suggested in Table 9-7 for multipurpose gearing. Then, using Equation (9-18), we can compute the number of cycles of loading. For the pinion rotating at 3450 rpm with one cycle of loading per revolution,

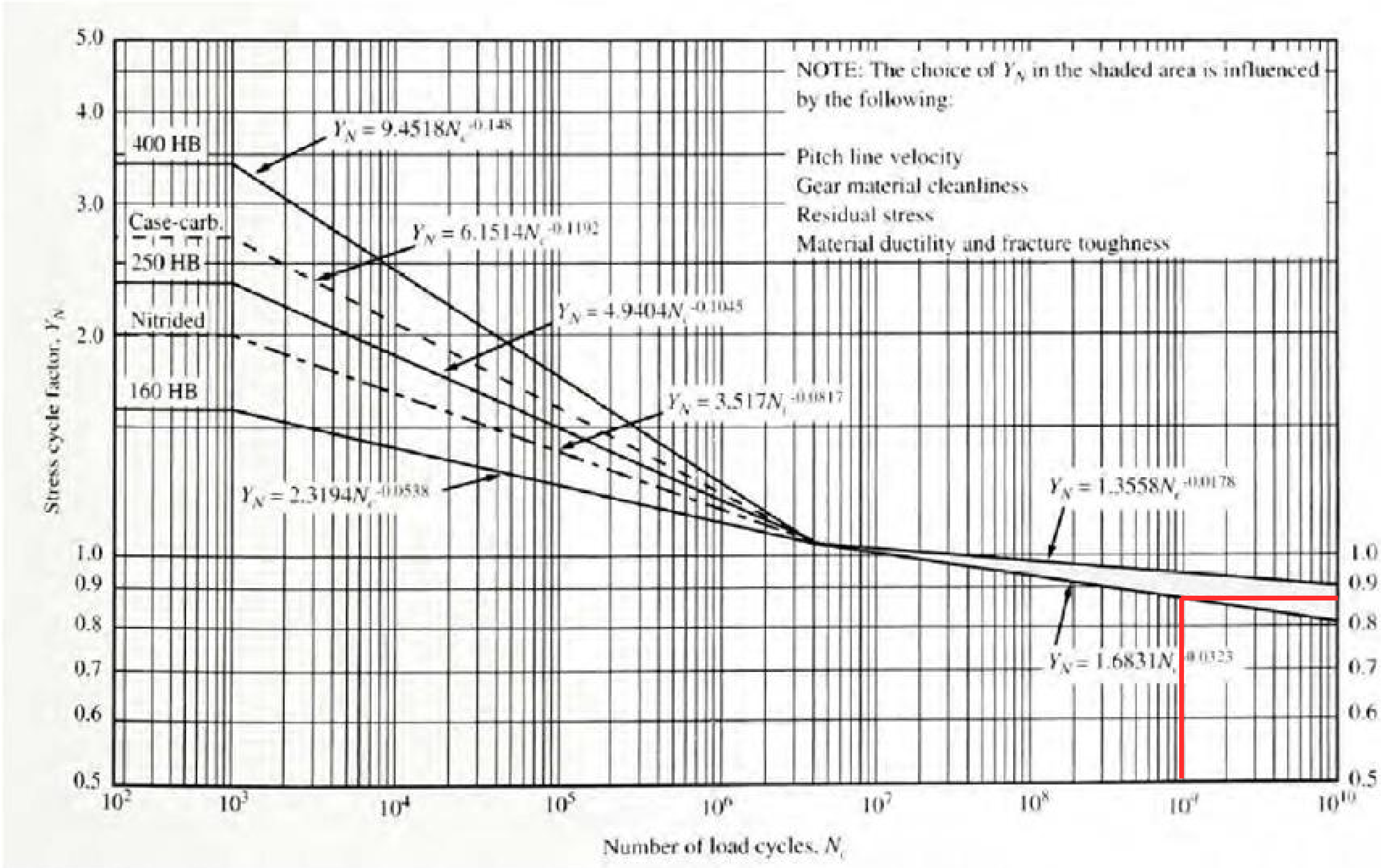
$$N_{c_p} = (60)(L)(n_p)(q)$$
$$= (60)(10000)(3450)(1) = 2.1 \times 10^9 cycles$$

(q) = number of load applications per revolution

TABLE 9–7 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

Specify the stress cycle life (Y_N) from Figure (9-8) page (395) (411 pdf):
Y_N=0.85



The bending stress in the pinion can now be computed:

$$S_{tp} = \frac{W_t P_d}{F\, J_p} K_o K_s K_m K_B K_v \left(\frac{K_R \times S.F}{Y_N}\right)$$

$$= \frac{(1146)(11.59)}{(2.25)(0.48)}(1.5)(1)(1.26)(1)(1.35)\left(\frac{1.25\times 1}{0.85}\right) = 46.145\,ksi = 318.16\ MPa$$

Specify the type of material for the gears to find the Elastic Coefficient C_p from Table (9-9) page(400) (Pdf 416)

		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 (2×10^5)	2300 (191)	2180 (181)	2160 (179)	2100 (174)	1950 (162)	1900 (158)
Mall. iron	25×10^6 (1.7×10^5)	2180 (181)	2090 (174)	2070 (172)	2020 (168)	1900 (158)	1850 (154)
Nod. iron	24×10^6 (1.7×10^5)	2160 (179)	2070 (172)	2050 (170)	2000 (166)	1880 (156)	1830 (152)
Cast iron	22×10^6 (1.5×10^5)	2100 (174)	2020 (168)	2000 (166)	1960 (163)	1850 (154)	1800 (149)
Al. bronze	17.5×10^6 (1.2×10^5)	1950 (162)	1900 (158)	1880 (156)	1850 (154)	1750 (145)	1700 (141)
Tin bronze	16×10^6 (1.1×10^5)	1900 (158)	1850 (154)	1830 (152)	1800 (149)	1700 (141)	1650 (137)

Source: Extracted from AGMA Standard 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, VA 22314.
Note: Poisson's ratio = 0.30; units for C_p are $(\text{lb/in}^2)^{0.5}$ or $(\text{MPa})^{0.5}$.

Specify the pitting geometry factor (I) with 20° normal pressure angle from Table(10-1) page (459) (475pdf):

A. Helix angle $\psi = 15.0°$

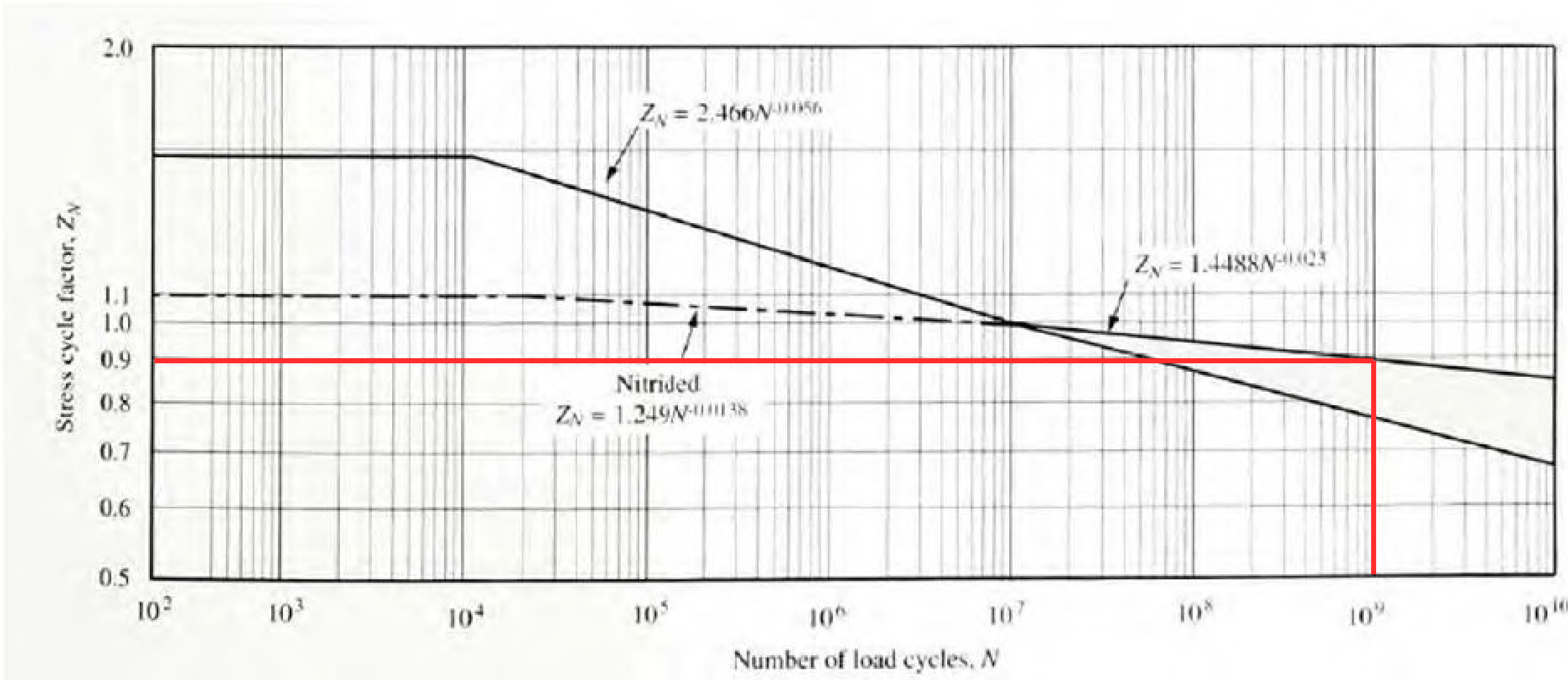
Gear teeth	Pinion teeth 17	21	26	35	55
17	0.124				
21	0.139	0.128			
26	0.154	0.143	0.132		
35	0.175	0.165	0.154	0.137	
55	0.204	0.196	0.187	0.171	0.143
135	0.244	0.241	0.237	0.229	0.209

24 75 **0.202**

B. Helix angle $\psi = 25.0°$

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

Specify the pitting resistance stress cycle factor (Z_N) from figure (9-24) page (403) (419 pdf): Z_N=0.89



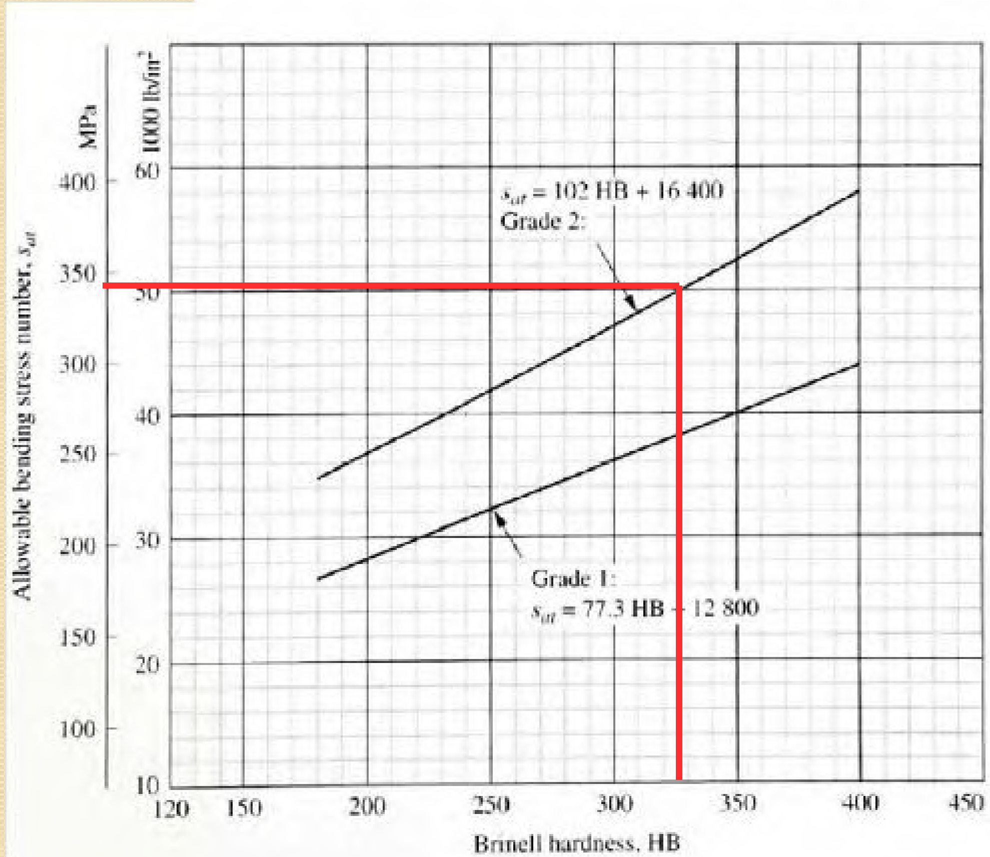
$$for\ \frac{HB_P}{HB_G} = 1 \quad use\ C_H = 1$$

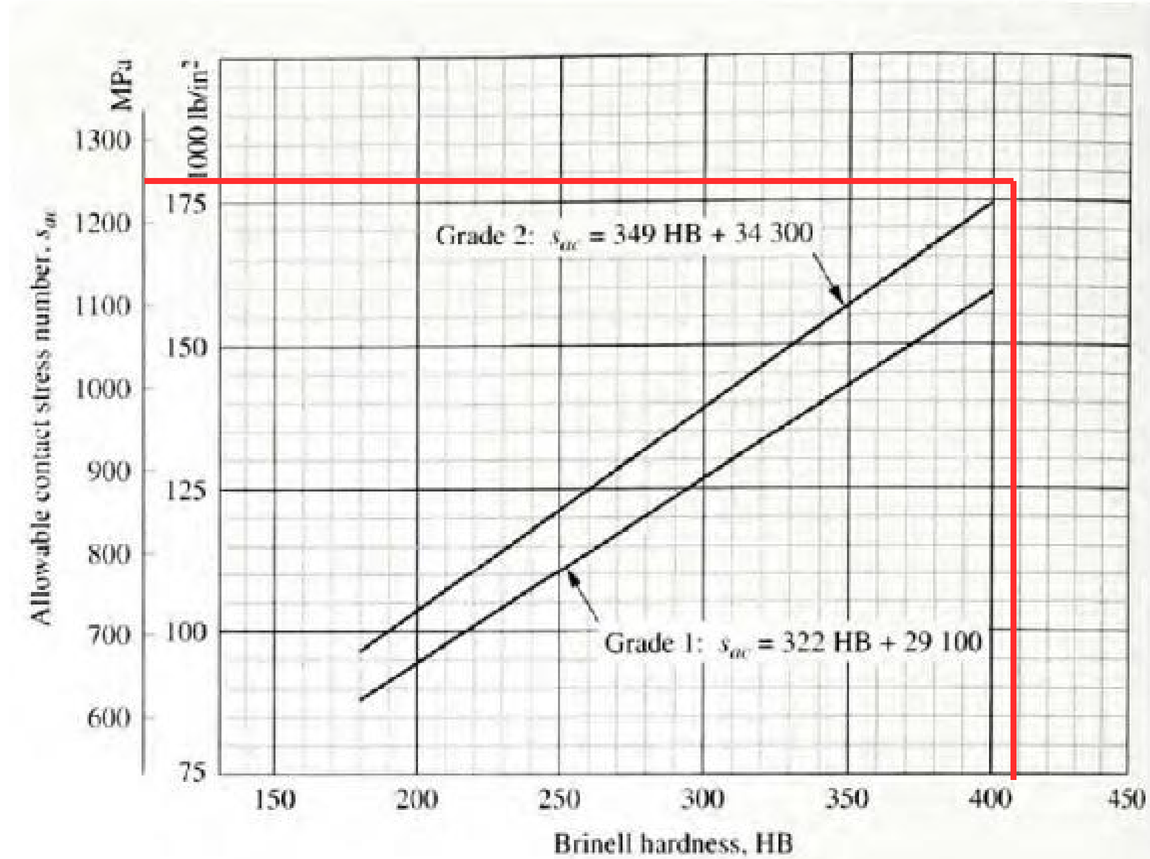
The pitting stress number in the pinion can now be computed:

$$S_c = C_p\sqrt{\frac{W_t K_o K_s K_m K_v}{F D_p I}}\left(\frac{K_R(S.F)}{Z_N C_H}\right)$$

$$= (2300)\sqrt{\frac{(1146)(1.5)(1)(1.26)(1.35)}{(2.25)(2.071)(0.202)}}\left(\frac{1.25\times 1}{0.89\times 1}\right)$$

$$= 180\ ksi = 1241\ MPa$$



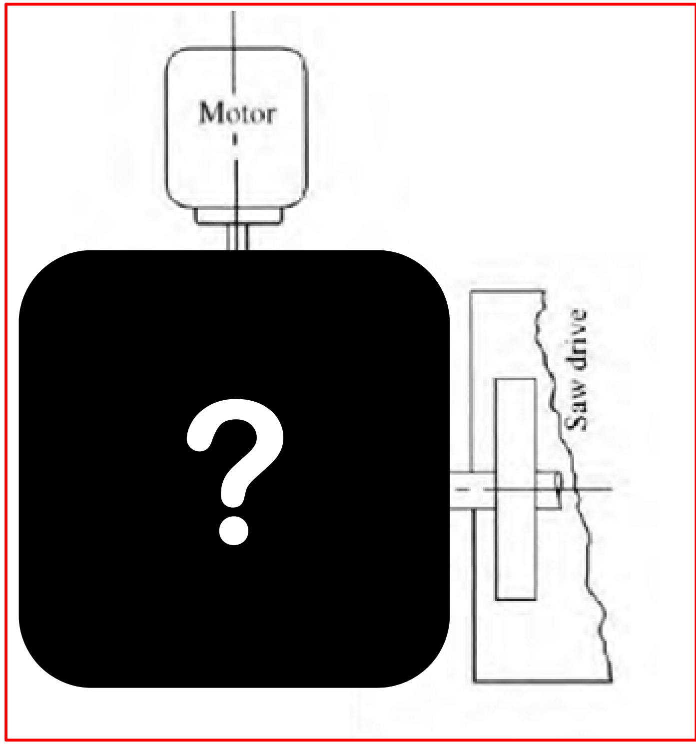


Mechanical Engineering Design II

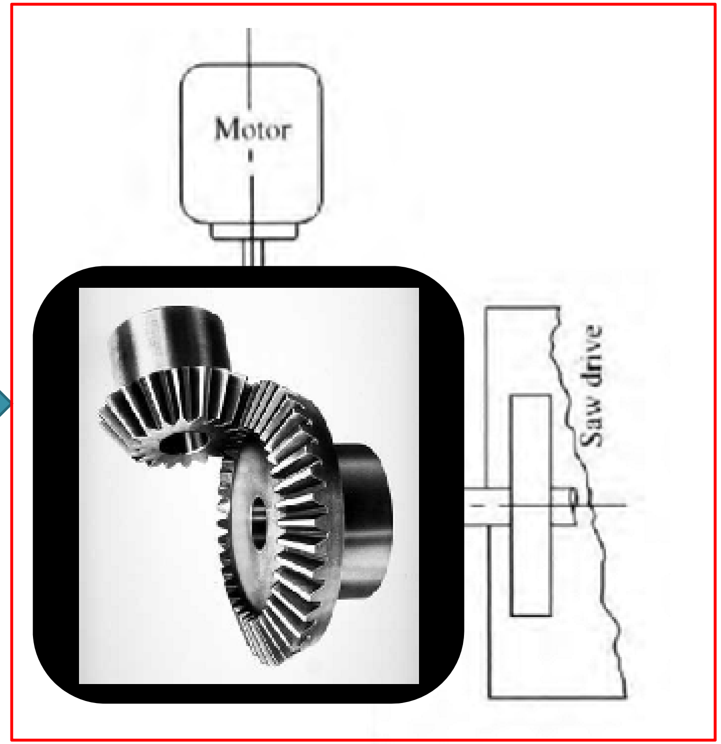
Twenty-one Lecture

Design of Bevel Gear

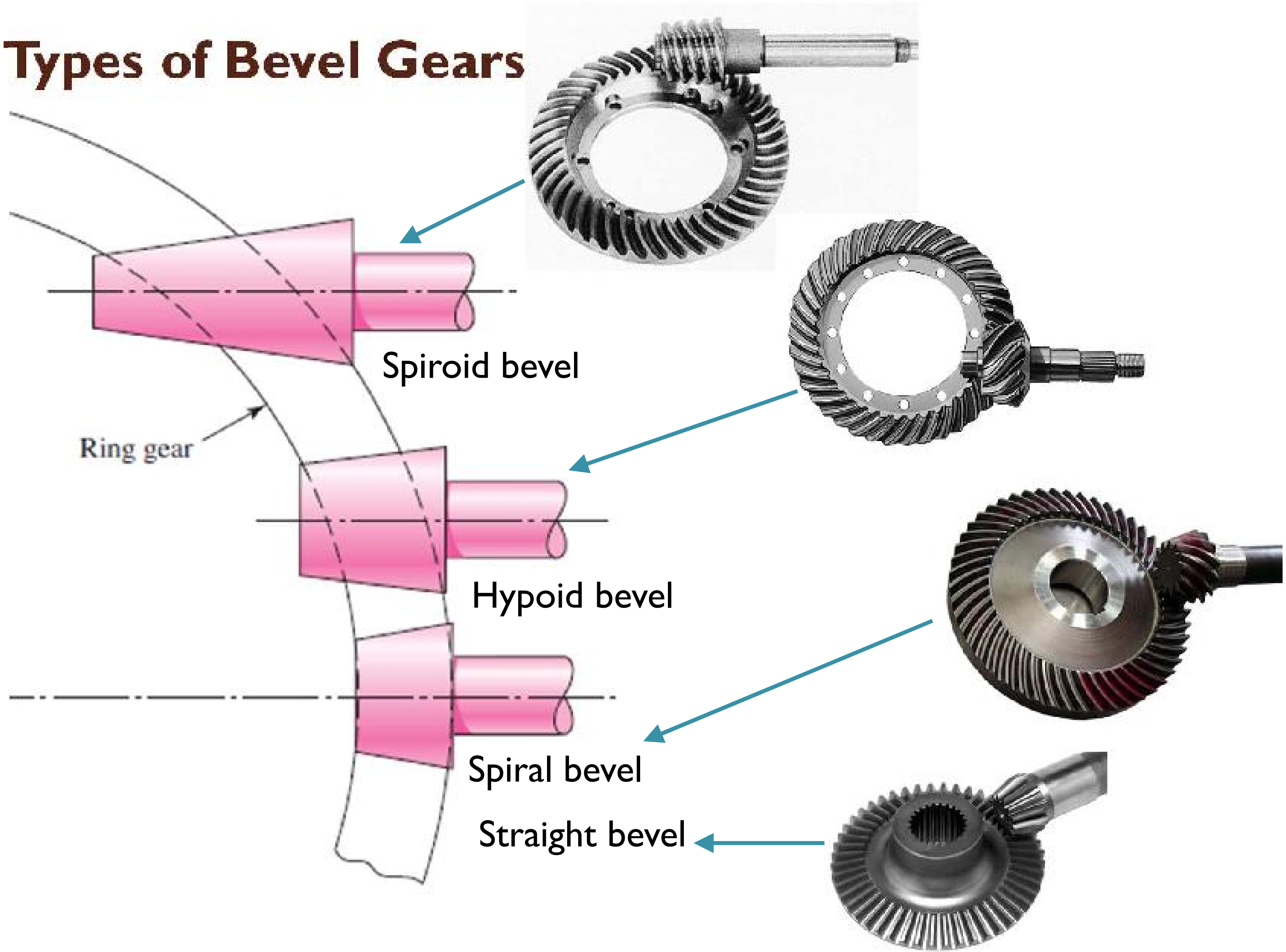
Power Transmission Problem



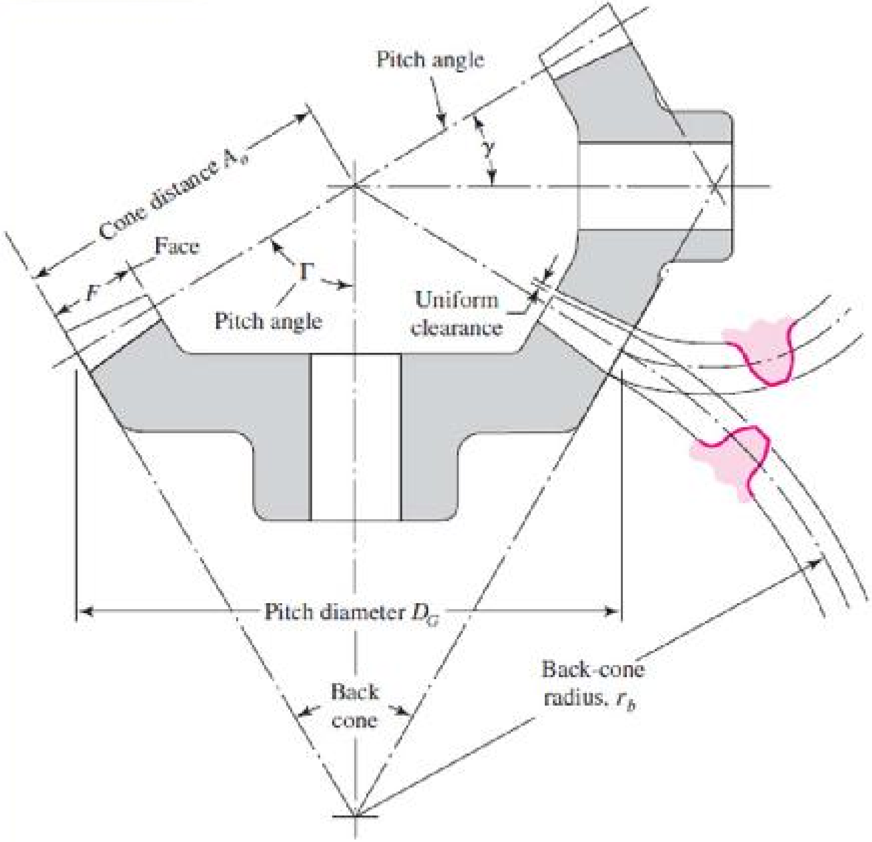
Proposed solution (Bevel Gear)



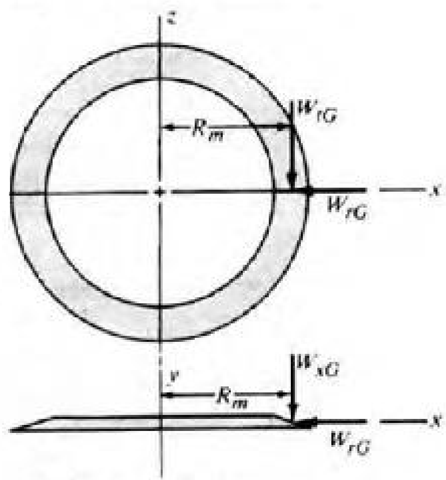
Types of Bevel Gears



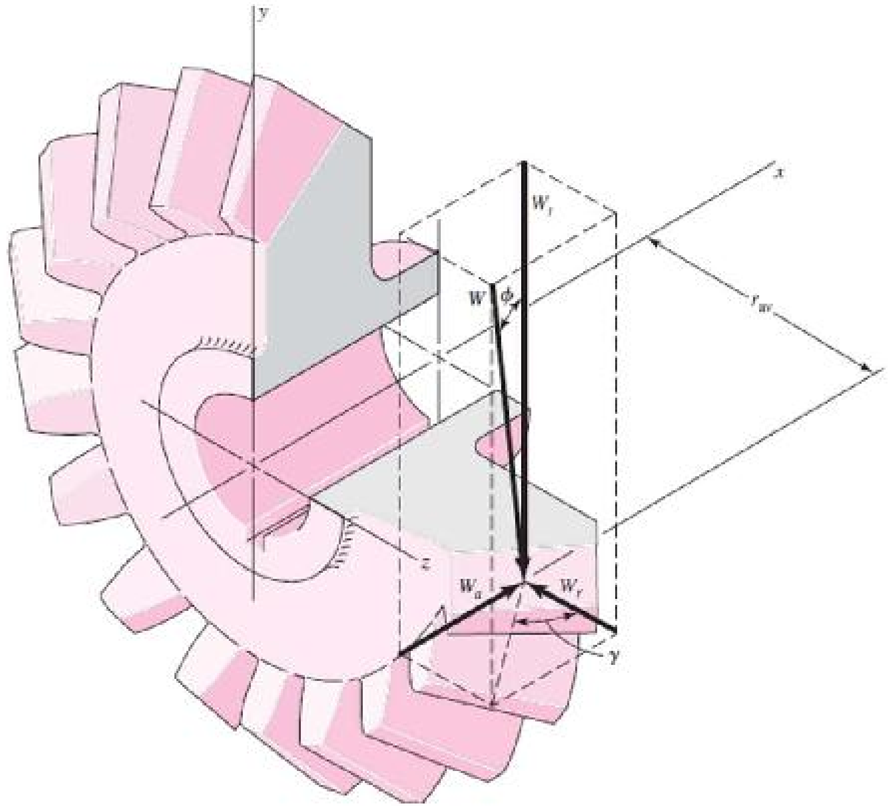
Basic bevel Gear Geometry and force Kinematics

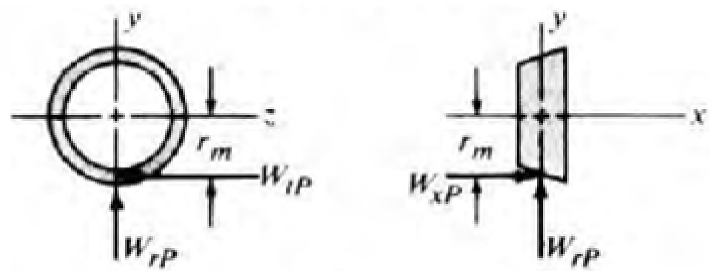


Notes: Shaded area is pitch cone surface.
Considering magnitudes:
$W_{tP} = W_{tG}$
$W_{xP} = W_{rG}$
$W_{rP} = W_{xG}$



(c) Free-body diagram: gear





(b) Free-body diagram: pinion

Modes of Gear Tooth Failure

1. Bending failure.

Root failure

2. Pitting.

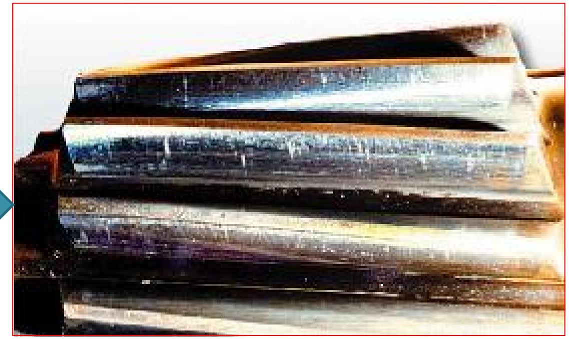
3. Scoring.

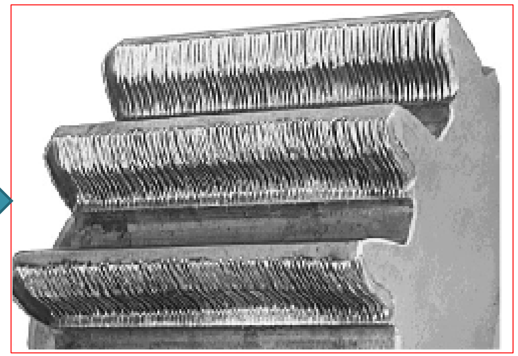
4. Abrasive wear.

5. Corrosive wear.

Surface failure

Dangerous Bar





Bevel Gear Design

The power to be transmitted

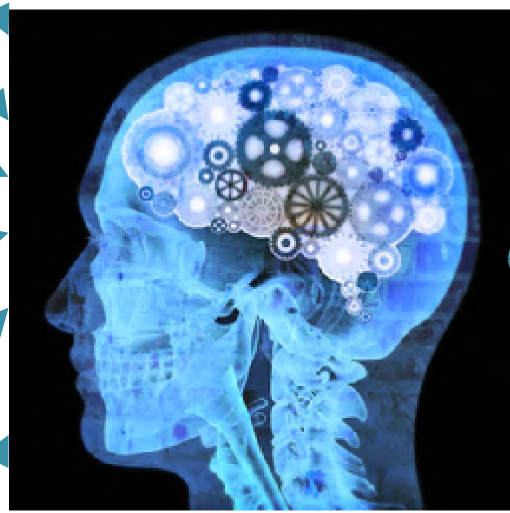
Type of driver and driven load

The speed of the driving gear

The center distance

The speed of the driven gear or the velocity ratio

Other information related to problem specification



Designer

The gear teeth should not fail under static loading or dynamic loading during normal running conditions.

The gear teeth should have wear characteristics so that their life is satisfactory.

The use of space and material should be economical.

The alignment of the gears and deflections of the shafts must be considered.

The lubrication of the gears must be satisfactory.

Flowchart for bevel gear designing process:

Transmitted Power , Input and Output speed, Center distance, Type of driver and driven load

Choose the over load factor (K_o) from Table (9-5) page(389) (405pdf)

	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

Compute the design power

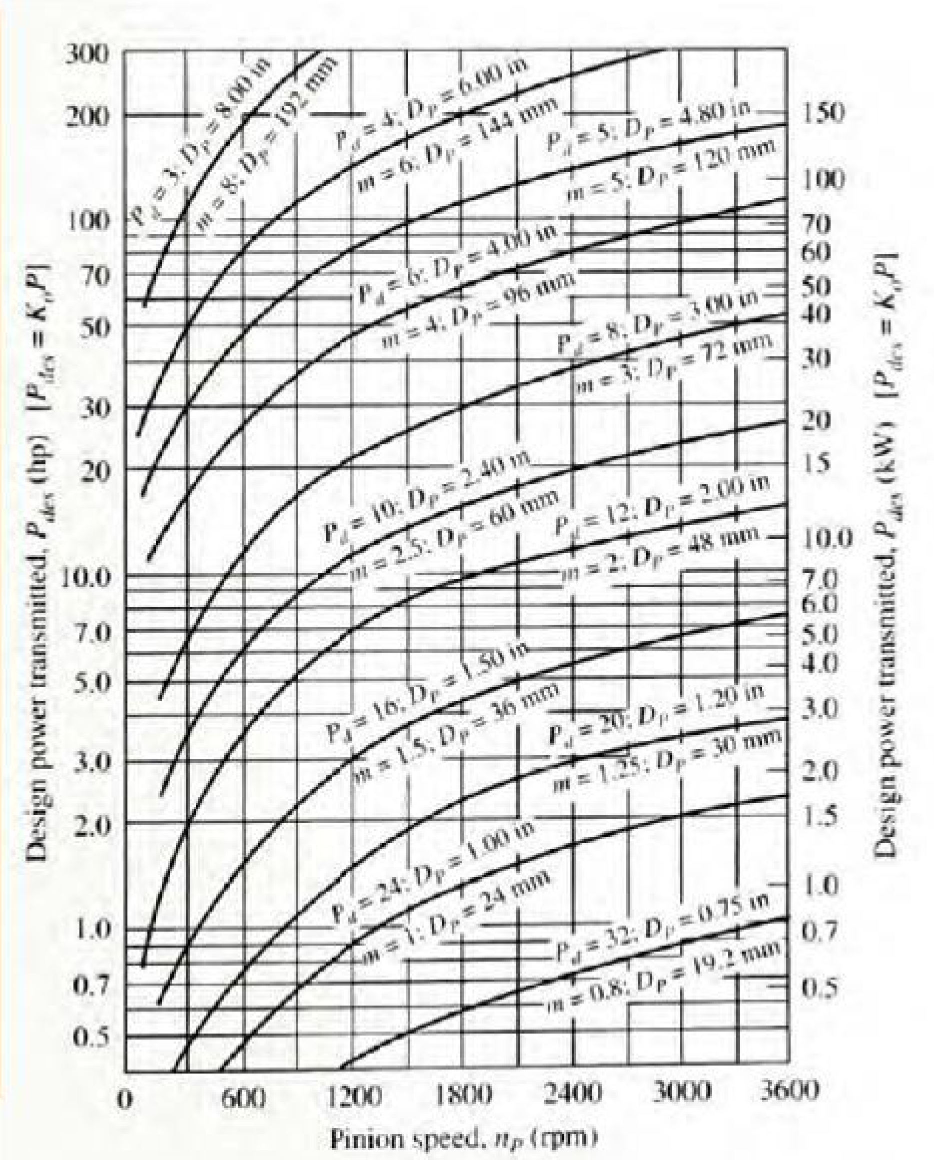
Design Power=K_o x transmitted power

1

1

Find the trial value for Diametral pitch (P_d) or Module (m) from Figure 9-27 page.409 (425 Pdf)

Standard_diametral_pitches_(teeth/in)
1.25
1.5
1.75
2
2.5
3
3.5
4
5
6
7
8
9
10
11
12
13
14
15
16
18
20
24
32
48
64
72
80
96
120
200



2

2

Specify the no. of teeth for Pinion N_P (from 17 to 20)

Compute the nominal velocity ratio $VR = \frac{n_P}{n_G}$

Compute the approximate no. of teeth for Gear $N_G = N_P \times VR$

Compute the actual velocity ratio $VR = \frac{N_G}{N_P}$

Compute the actual output velocity $n_G = n_P \frac{N_P}{N_G}$

Compute the pitch diameters $D_p = \frac{N_p}{P_d}$, $D_G = \frac{N_G}{P_d}$,

face width $F = min\left(0.3A_o, {}^{10}/_{P_d}\right)$, $A_o = \frac{D_P}{2\sin\gamma} = \frac{D_G}{2\sin\Gamma}$

pinion mean radius $r_m = {}^{D_p}/_2 - (F/2)\sin\gamma$ where $\gamma = \tan^{-1}\left(N_p/N_G\right)$,

gear mean radius $R_m = {}^{D_G}/_2 - (F/2)\sin\Gamma$ where $\Gamma = \tan^{-1}(N_G/N_P)$,

pitch line speed $v_t = \frac{2\pi r_m n_P}{60}$, and tangential force, $W_t = \frac{60P}{2\pi r_m n_p} = \frac{T}{r_m}$

3

3

Analyzing of gear tooth failure mode

Root (Bending) Failure Mode

Bending Stress Number

$$S_t = \frac{W_t P_d}{F\,J}\frac{K_o K_s K_m}{K_v} < S_{at}\frac{Y_N}{K_R(S.F)}$$

Surface (Pitting, Scoring,…) Failure Mode

Contact Stress Number

$$S_c = C_p C_b \sqrt{\frac{W_t}{F D_p I}\frac{C_o C_m}{C_v}} < S_{ac}\frac{Z_N C_H}{K_R(S.F)}$$

Find the values of factors

$(J, I, K_s, K_m, K_v, C_p, C_b, C_o, C_m, C_v, Y_N, Z_N, C_H, K_R, S.F)$

as in the following steps

4

4

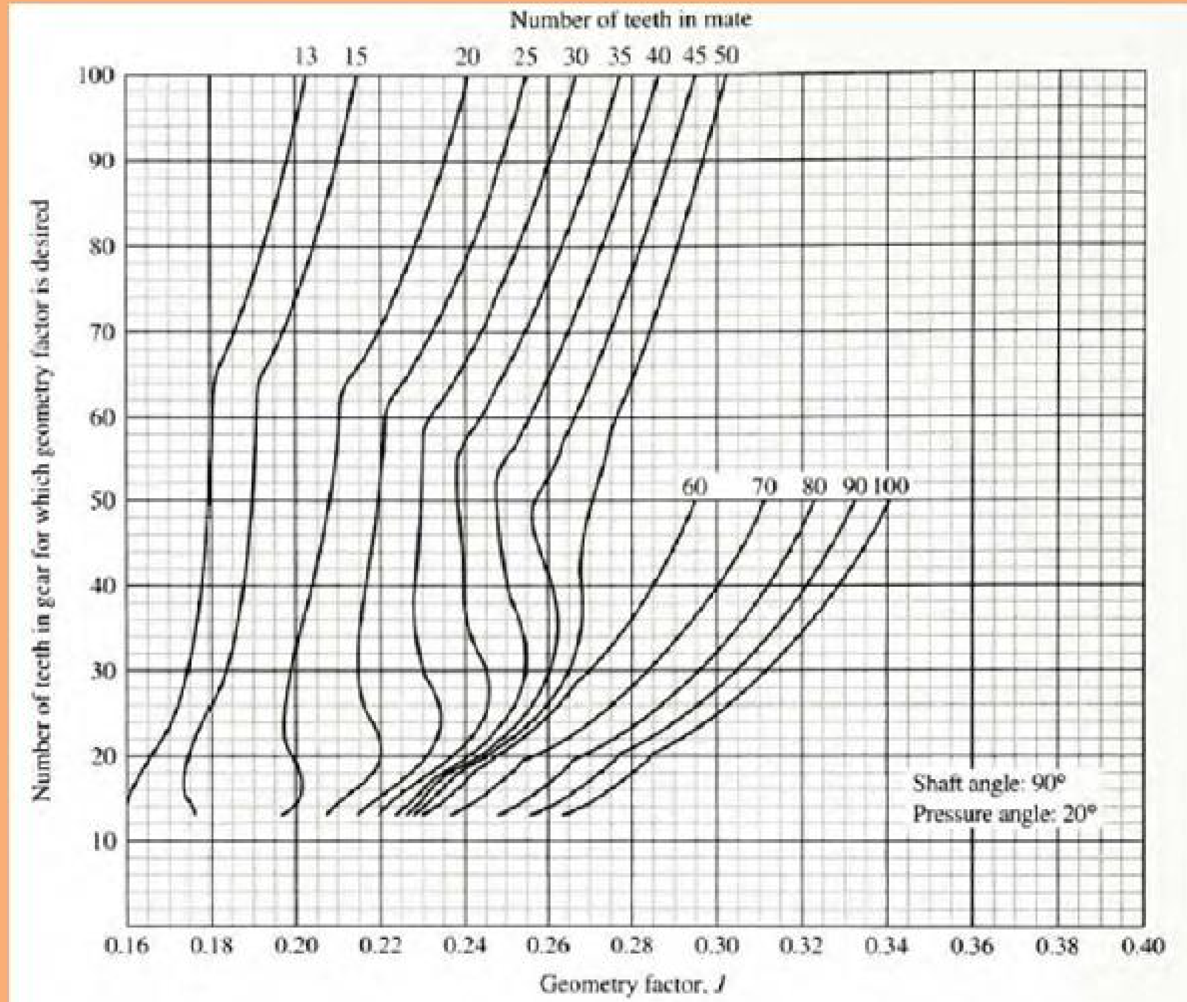
Specify the type of material for the gears to find the Elastic Coefficient C_p from Table (9-9) page(400) (Pdf 416)

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

Source: Extracted from AGMA Standard 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*, with the permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, VA 22314.
Note: Poisson's ratio = 0.30; units for C_p are $(\text{lb/in}^2)^{0.5}$ or $(\text{MPa})^{0.5}$.

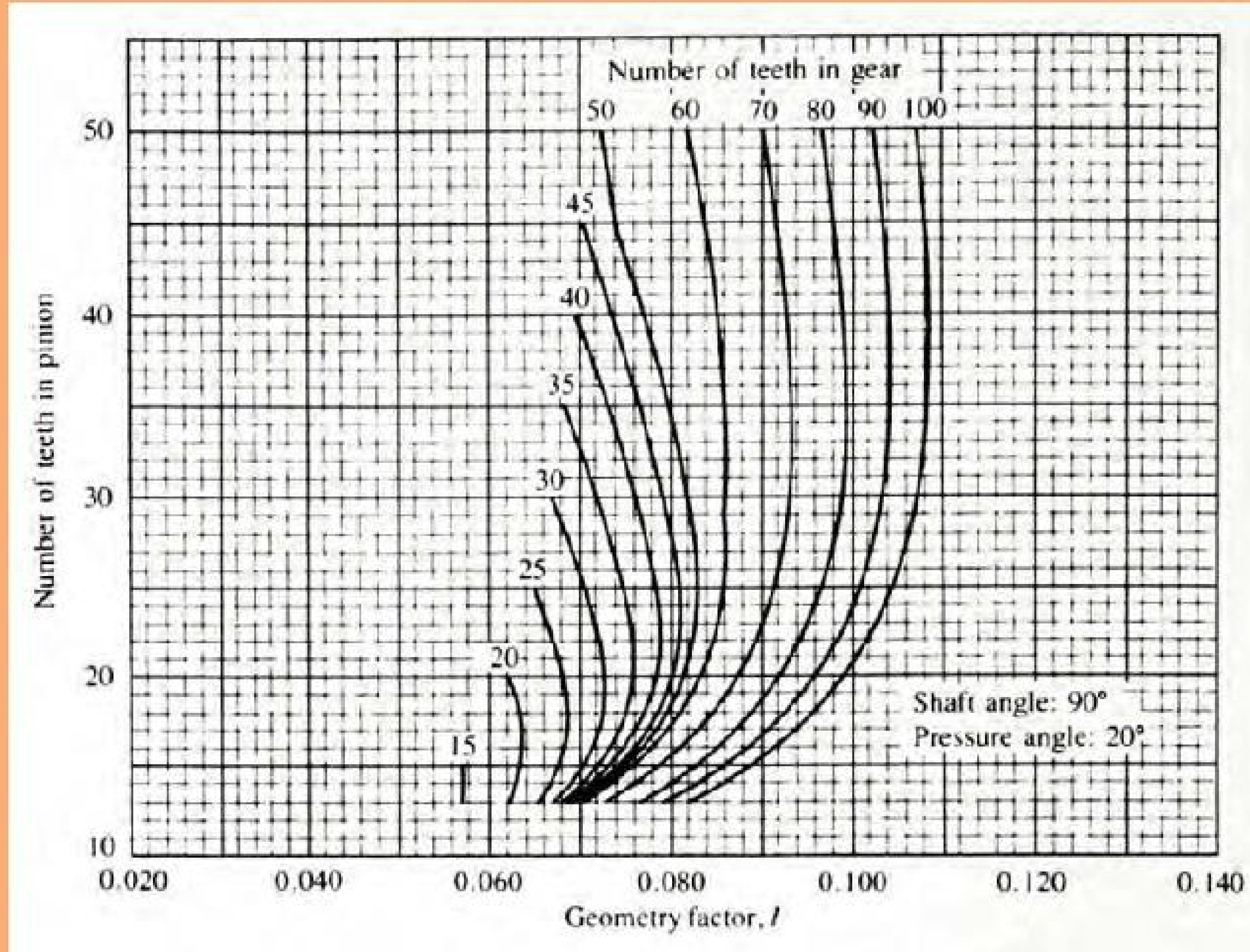
5

Specify the bending geometry factor (J) for 20º pressure angle and 90º shaft angle from figure (10-13) page (472) (488pdf):



6

Specify the pitting geometry factor (I) with 20º normal pressure angle and 90º shaft angle from Figure (10-14) page (474) (490pdf):



7

7

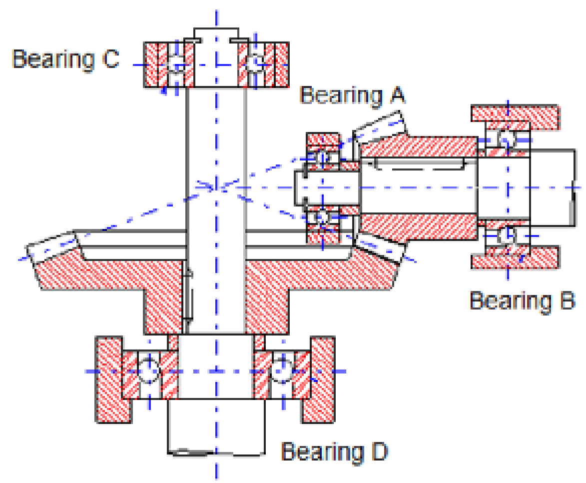
Specify the size factor (K_s) from Table(9-6) page (389) (293 pdf)

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

Specify the load distribution factor (K_m) from Table(10-3) page (471) (487 pdf)

Type of gearing	Both gears straddle-mounted	One gear straddle-mounted	Neither gear straddle-mounted
General commercial-quality	1.44	1.58	1.80
High-quality, commercial gearing	1.20	1.32	1.50

Straddle mounted gears



8

8

Specify the quality number Q_v , from Table (9-2) page (378) (394 pdf)

Application	Quality number	Application	Quality number
Cement mixer drum drive	3–5	Small power drill	7–9
Cement kiln	5–6	Clothes washing machine	8–10
Steel mill drives	5–6	Printing press	9–11
Grain harvester	5–7	Computing mechanism	10–11
Cranes	5–7	Automotive transmission	10–11
Punch press	5–7	Radar antenna drive	10–12
Mining conveyor	5–7	Marine propulsion drive	10–12
Paper-box-making machine	6–8	Aircraft engine drive	10–13
Gas meter mechanism	7–9	Gyroscope	12–14

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

Pitch line speed (fpm)	Quality number	Pitch line speed (m/s)
0–800	6–8	0–4
800–2000	8–10	4–11
2000–4000	10–12	11–22
Over 4000	12–14	Over 22

9

9

Choose material for pinion and gear or (S_{ac} , S_{at}) from figures [(9-10) page (379) (395pdf) , (9-11) page (380) (396pdf)] with tables [(9-3) page(381) (397pdf) , (9-4) page (385) (401pdf)] and see also Appendix 3 to 5 [p(A-6) to (A-11)].

Determine the dynamic factor (K_v) from the following equation:

$$K_v = \left[\frac{K_z}{K_z + \sqrt{v_t}}\right]^u$$

$$u = \frac{8}{(2)^{0.5Q}} - S_{at}\left[\frac{125}{E_P + E_G}\right] \quad and \quad K_z = 85 - 10(u)$$

$$if\ u = negative\ value\ \ then\ use \quad u = 0.0$$

For checking K_v must be greater than ($K_{v_{min}} = \frac{2}{\pi}\tan^{-1}({}^{v_t}/_{333})$), if its not achieved then a higher quality number should be specified

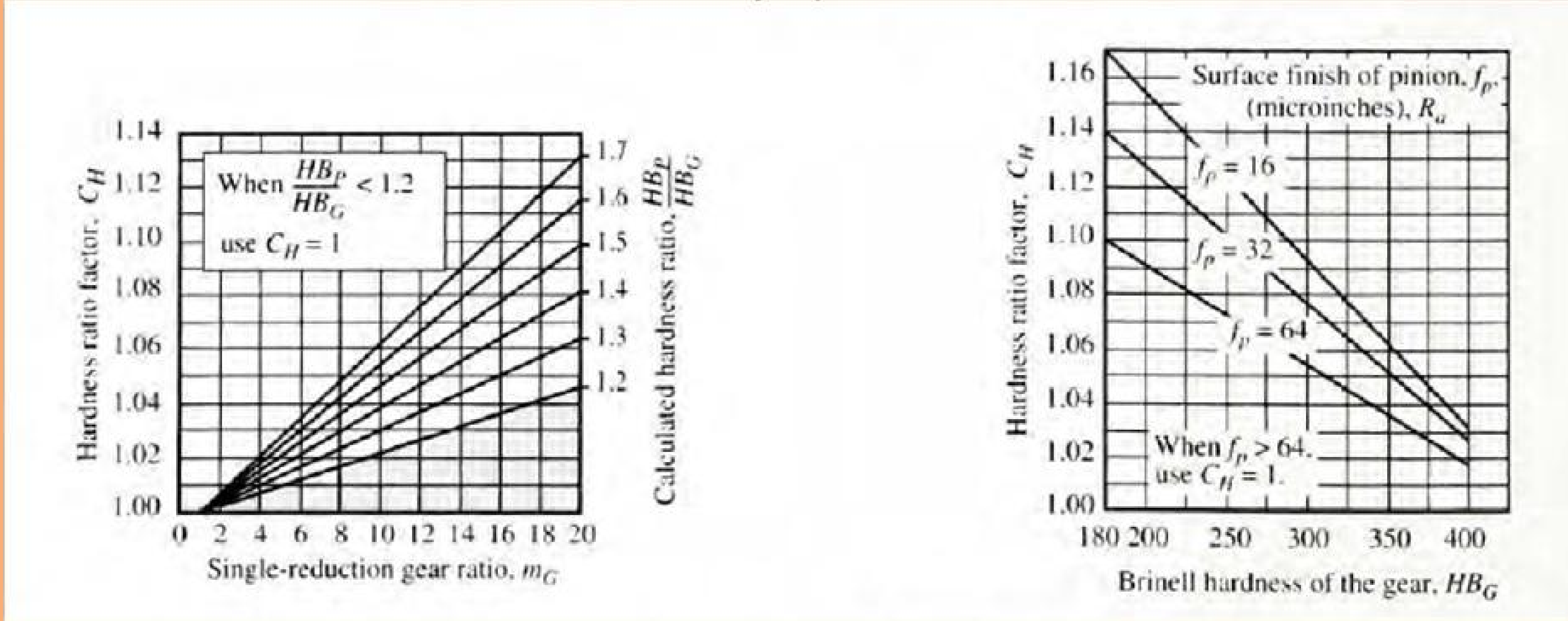
Note : the calculation of inverse tangent must be in radians

10

10

Specify the safety factor (S.F) typically from 1 to 1.5

Specify the hardness ratio factor (C_H) from Figure (9-25 & 26) page (404) (420 pdf)



11

11

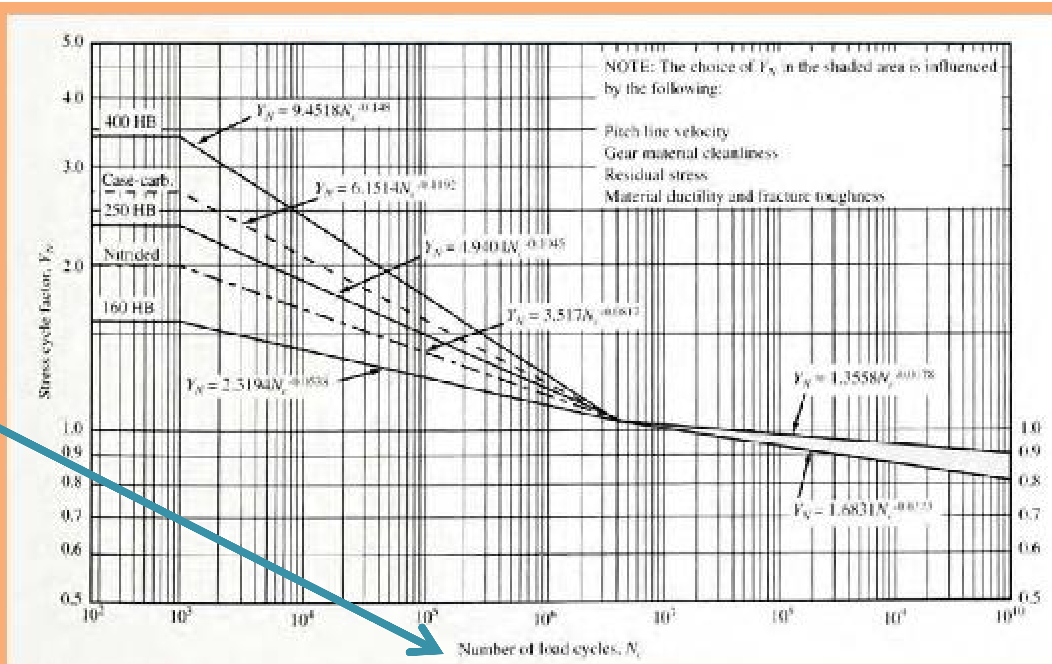
Specify the reliability factor (K_R) from Table (9-8) page (396) (412 pdf):

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

Specify the stress cycle life (Y_N) from Figure (9-8) page (395) (411 pdf):

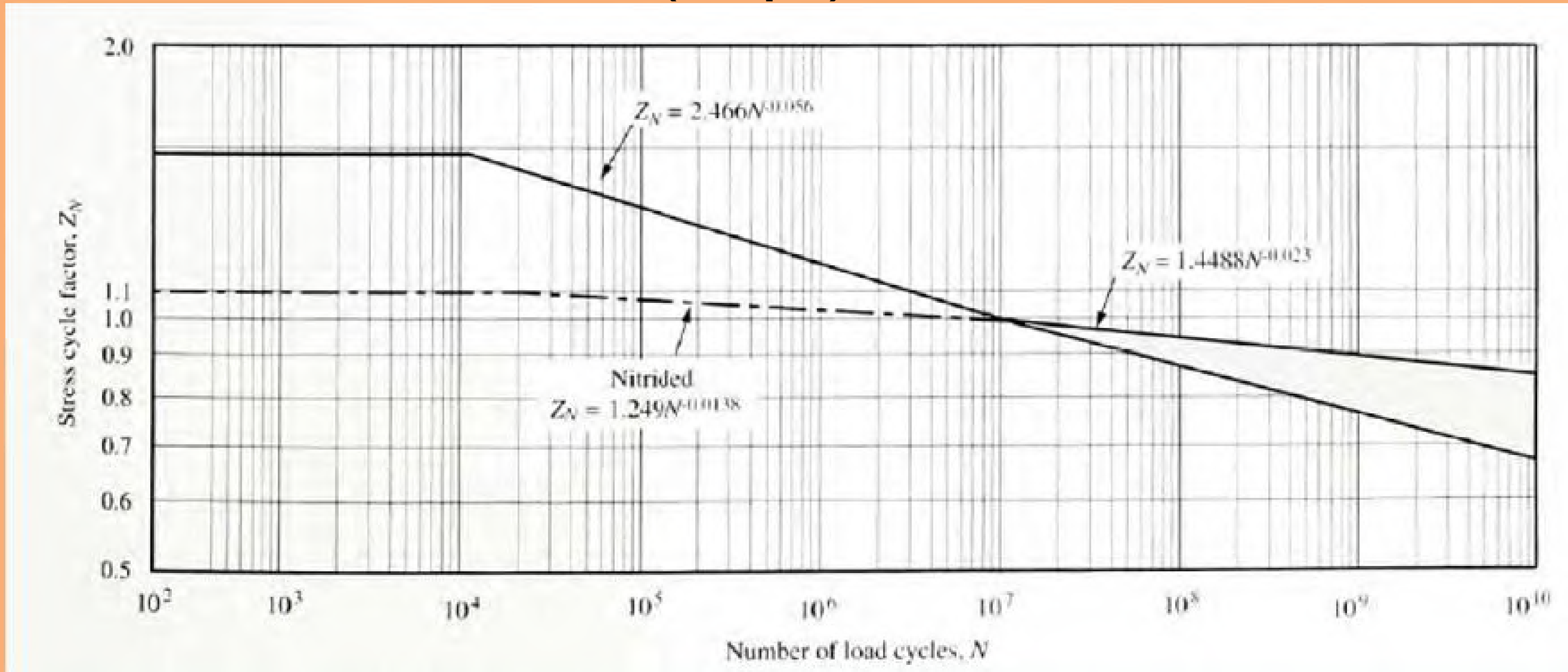
TABLE 9–7 Recommended design life

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



12

Specify the pitting resistance stress cycle factor (Z_N) from figure (9-24) page (403) (419 pdf):



The factors C_o , C_v and C_m are the same as K_o , K_v and K_m

Using $C_b = 0.634$ allows the use of the same allowable contact stress as for spur and helical gear

Check if the selected material satisfy the following design conditions:

$$S_t \frac{K_R(S.F)}{Y_N} < S_{at}$$

$$S_c \frac{K_R(S.F)}{Z_N C_H} < S_{ac}$$

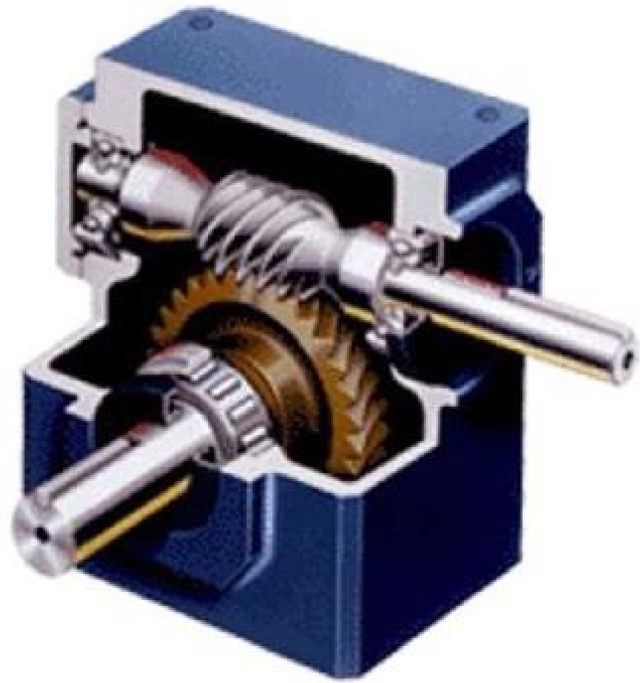
Mechanical Engineering Design II

Twenty-two Lecture

Design of Worm Gear

Power Transmission Problem

Proposed solution (Worm Gear)



Design Requirements

- ✓ high velocity ratios in a single step in a minimum of space
- ✓ non-intersecting shafts at right angles

Specifications of Worm Gears

Advantages:

(1) A large speed ratio;

(2) Silent and smooth operation;

(3) Small drive size

(4) Better load distribution;

(5) Self-locking action.

Disadvantages:

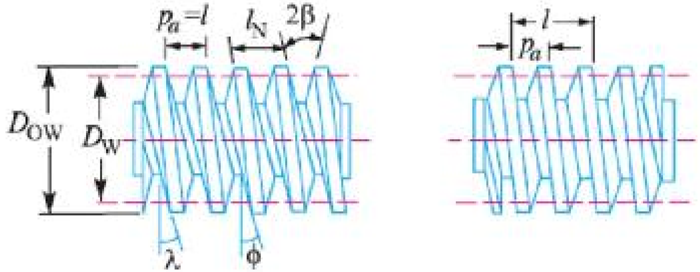
(1) Low efficiency;

(2) Expensive antifriction materials;

(3) Considerable sliding speed;

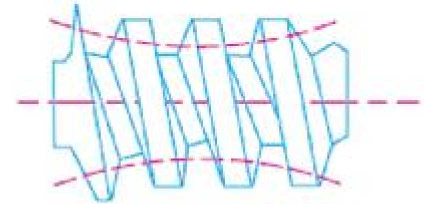
(4) Considerable heat generated.

Types of Worm Gears

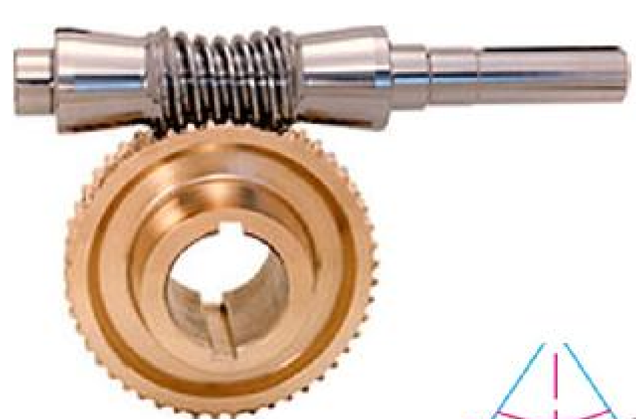


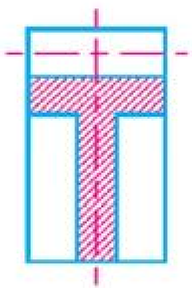
Single threaded. Double threaded.

(*a*) Cylindrical or straight worm.

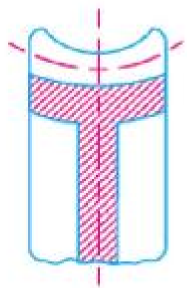


(*b*) Cone or double enveloping worm.

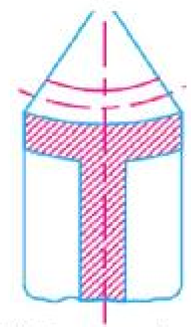




(*a*) Straight face.

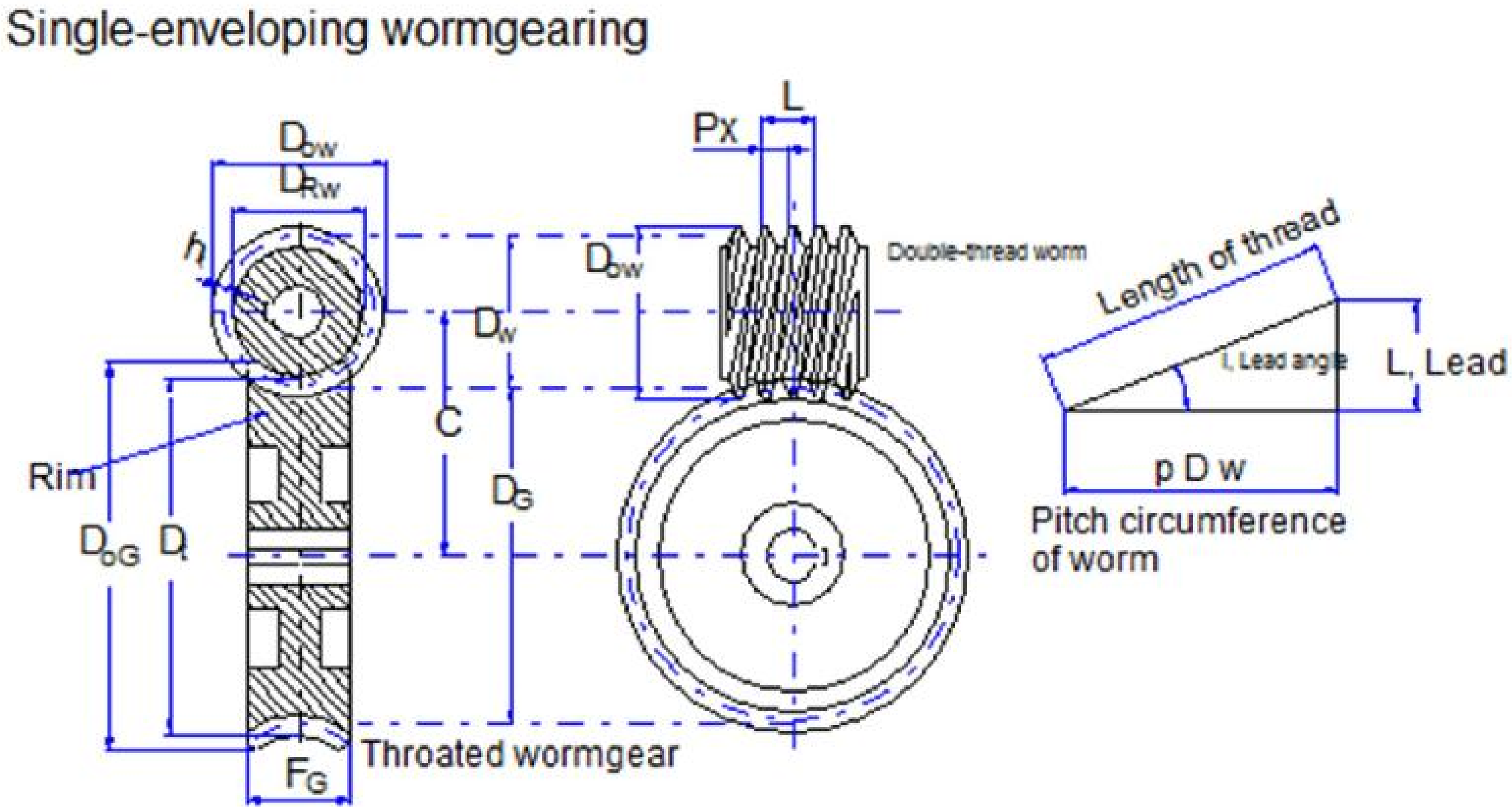


(*b*) Hobbed straight face.



(*c*) Concave face.

Basic Worm Gear Geometry



D_G = pitch diameter of the gear

D_W = pitch diameter of the worm

N_G = number of teeth in the gear

N_G = number of teeth in the worm

C = center distance

L = lead

$\lambda =$ lead angle

P_x = axial pitch

F_G = Face width of gear

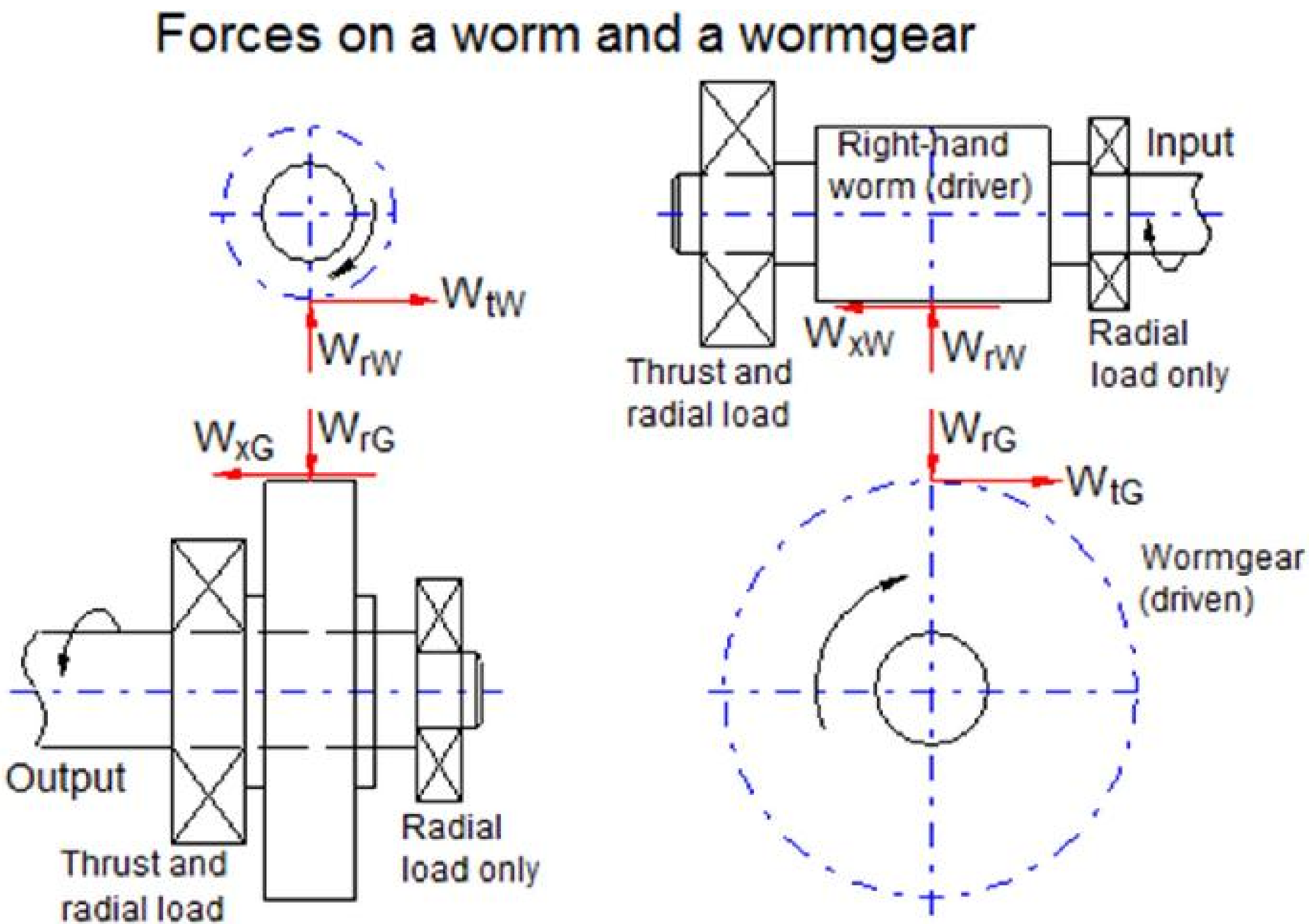
F_w = Face length of the worm

Forces on Worm Gear

$$W_{tG} = W_{xW}$$

$$W_{xG} = W_{tW}$$

$$W_{rG} = W_{rW}$$



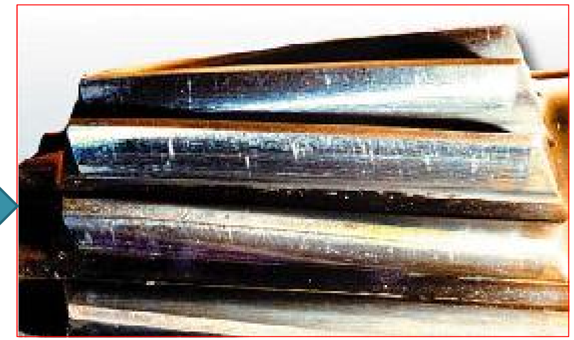
Modes of Gear Tooth Failure

1. Bending failure.

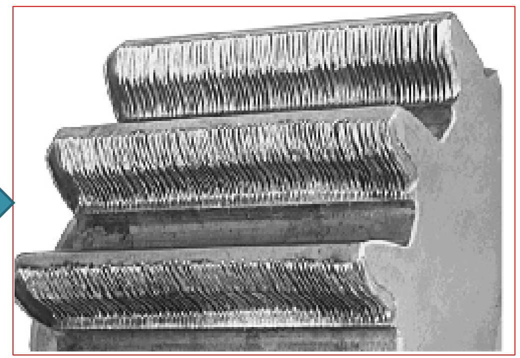
Root failure

2. Pitting.

3. Scoring.



4. Abrasive wear.



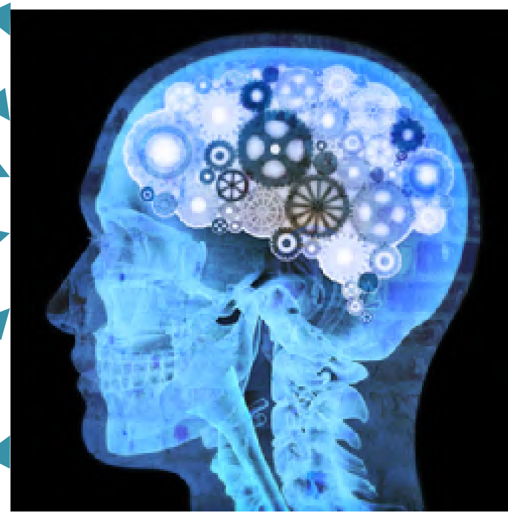
5. Corrosive wear.

Surface failure

Dangerous Bar

Worm Gear Design

- The power to be transmitted
- Type of driver and driven load
- The speed of the driving gear
- The center distance
- The speed of the driven gear or the velocity ratio
- Other information related to problem specification



Designer

- The gear teeth should not fail under static loading or dynamic loading during normal running conditions.
- The gear teeth should have wear characteristics so that their life is satisfactory.
- The use of space and material should be economical.
- The alignment of the gears and deflections of the shafts must be considered.
- The lubrication of the gears must be satisfactory.

Flowchart for worm gear designing process:

Transmitted Power , Input and Output speed, Center distance, Type of driver and driven load

Specify the no. of threads for Worm N_W (from 2 to 8 or more)

Specify the diametral pitch P_d (3 , 4 , 5 , 6 , 8 , 10 , 12 , 16 , 24 , 32 , 48)

Specify the Pressure angle ϕ_n (14.5° , 20° , 25° , 30°)

Compute the nominal velocity ratio $VR = \frac{n_w}{n_G} = \frac{N_G}{N_P}$

Compute the circular pitch for Gear $p = \pi/P_d$ = axial pitch for worm P_x , normal circular pitch $p_n = p\cos\lambda$, and the face width of gear $F = 2p$

Compute the lead $L = N_W \times P_x$

Compute the pitch diameter of the worm within $\frac{C^{0.875}}{1.6} > D_W > \frac{C^{0.875}}{3}$

1

1

Compute the lead angle $\lambda = \tan^{-1}(L/\pi D_W)$

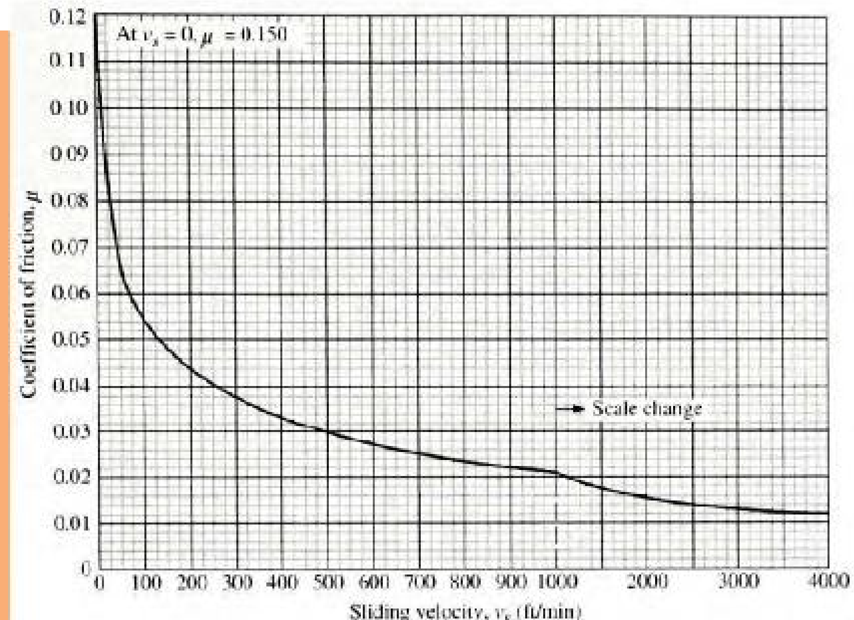
Compute the pitch diameter of gear $D_G = N_G/P_d$

Compute the pitch line speed of the worm and gear

$$v_{tw} = \frac{2\pi D_W n_W}{60} \quad , v_{tG} = \frac{2\pi D_G n_G}{60}$$

Compute the sliding velocity $v_s = v_{tG}/\sin\lambda = v_{tw}/\cos\lambda$

Find the coefficient of friction μ **from figure (10-18) page(477) (493pdf)**



choice of formula depends on the sliding velocity. *Note:* v_s must be in ft/min in the formulas; 1.0 ft/min = 0.0051 m/s.

2

Compute the output torque $T_o = \frac{P_o}{\omega_G} = W_{tG}(\frac{D_G}{2})$,

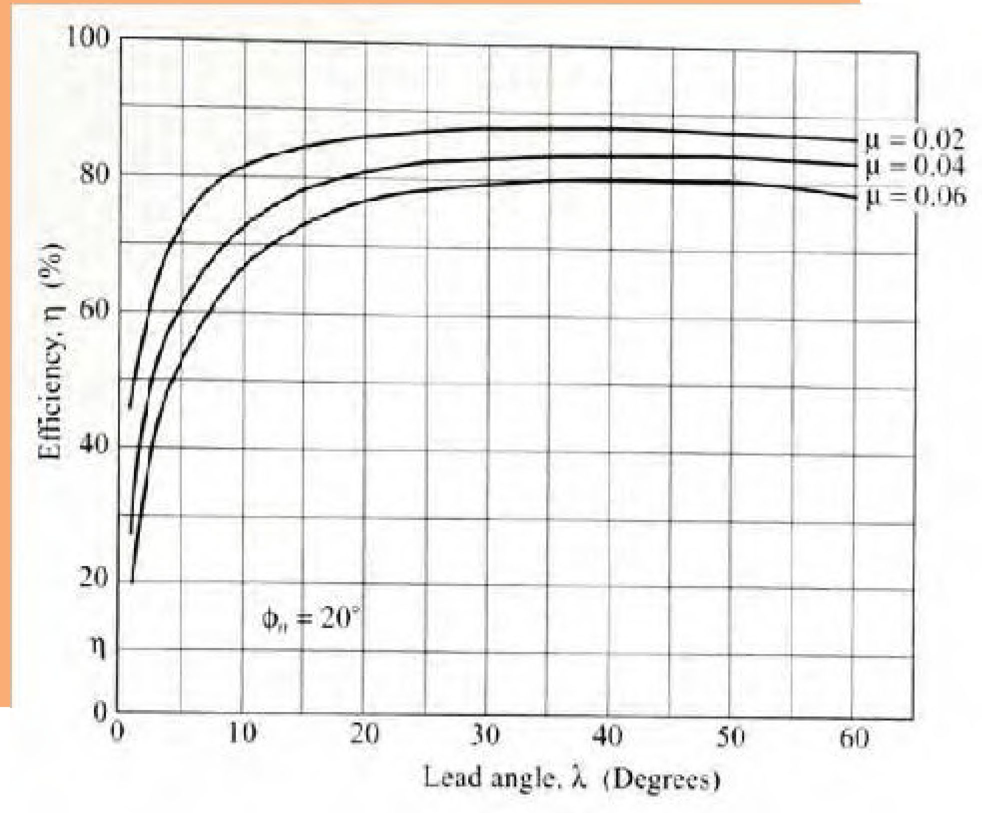
$$W_{xG} = W_{tG}\frac{\cos\phi_n \sin\lambda + \mu\cos\lambda}{\cos\phi_n \cos\lambda - \mu\sin\lambda} \quad , \quad W_{rG} = W_{tG}\frac{\sin\phi_n}{\cos\phi_n \cos\lambda - \mu\sin\lambda}$$

Friction force $W_f = \frac{\mu\, W_{tG}}{\cos\phi_n \cos\lambda - \mu\sin\lambda}$, Power loss $P_L = v_s W_f/60$,

Input Power $P_i = P_o + P_L$,

Efficiency $\eta = \frac{P_o}{P_i} = \frac{\cos\phi_n - \mu\tan\lambda}{\cos\phi_n - \mu\tan\lambda}$

or from figure (10-19) page (479) (495pdf)



3

Analyzing of gear tooth failure mode

Root (Bending) Failure Mode

Surface (Pitting, Scoring,…) Failure Mode

Bending Stress

$$\sigma = \frac{W_d}{yF_G p_n} < S_{at}$$

Surface Durability (Rated Tangential Load)

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

Find the values of factors (y, K_v, C_s, C_m, C_v) as in the following steps

4

4

Specify the Lewis form factor for worm gear teeth from Table (10-4) page(482) (Pdf 498)

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

Compute the dynamic load factor $K_v = 1200/(1200 + v_{tG})$ (v_{tG} in ft/min)

Compute the dynamic load $W_d = W_{tG}/K_v$

5

Specify the material factor (C_s) from figure (10-20) page (483) (499pdf) or from the following equations:

Sand-Casting Bronzes:

$for\ D_G > 63.5mm$

$$C_s = 1189.636 - 476.545\log_{10}(D_G)$$

$for\ D_G < 63.5mm$

$$C_s = 1000$$

Static-Chill-Cast or Forged Bronzes:

$for\ D_G > 203.2mm$

$$C_s = 1411.651 - 455.825\log_{10}(D_G)$$

$for\ D_G < 203.2mm$

$$C_s = 1000$$

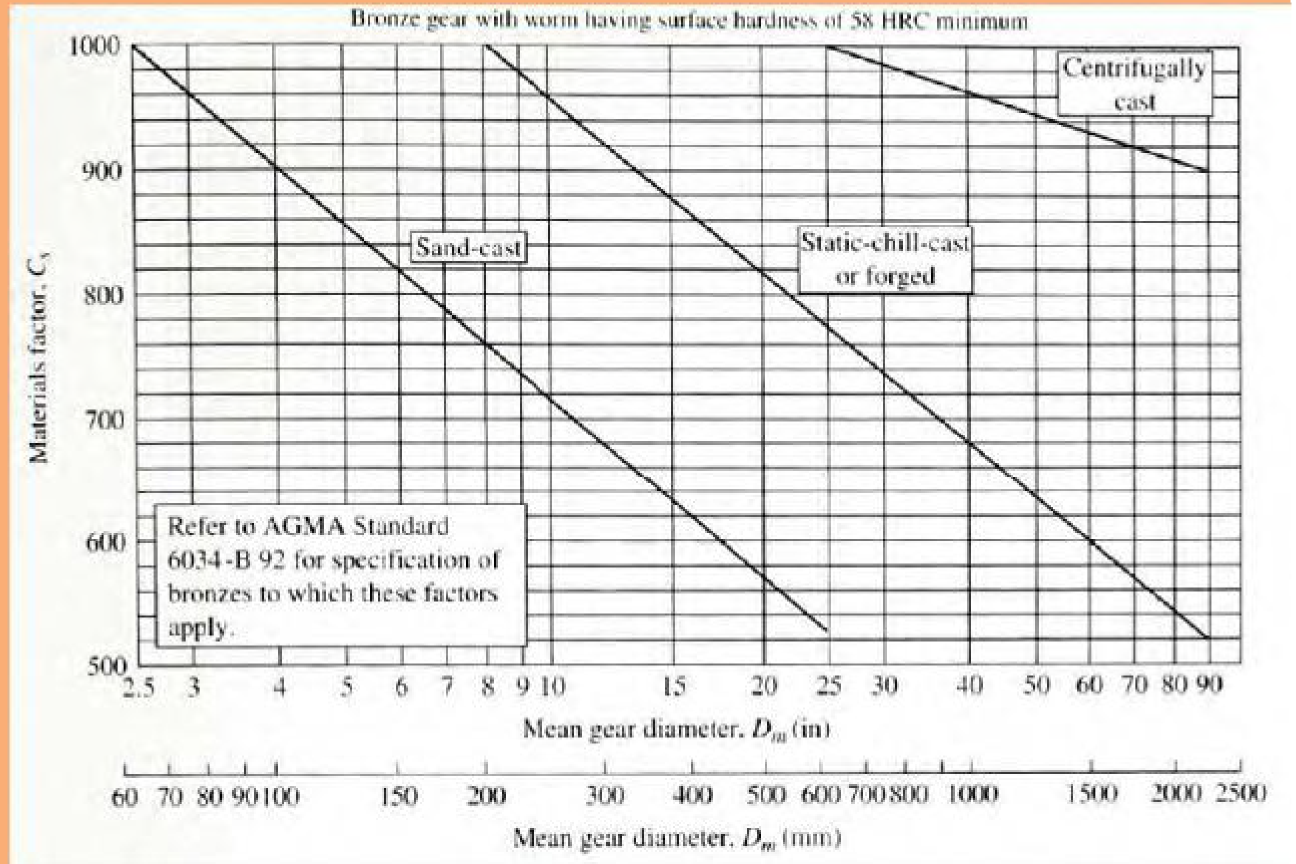
Centrifugally Cast Bronzes:

$for\ D_G > 635\ mm$

$$C_s = 1251.291 - 179.750\log_{10}(D_G)$$

$for\ D_G < 635\ mm$

$$C_s = 1000$$



Note: if standard addendum gears are used, $D_m = D_G$

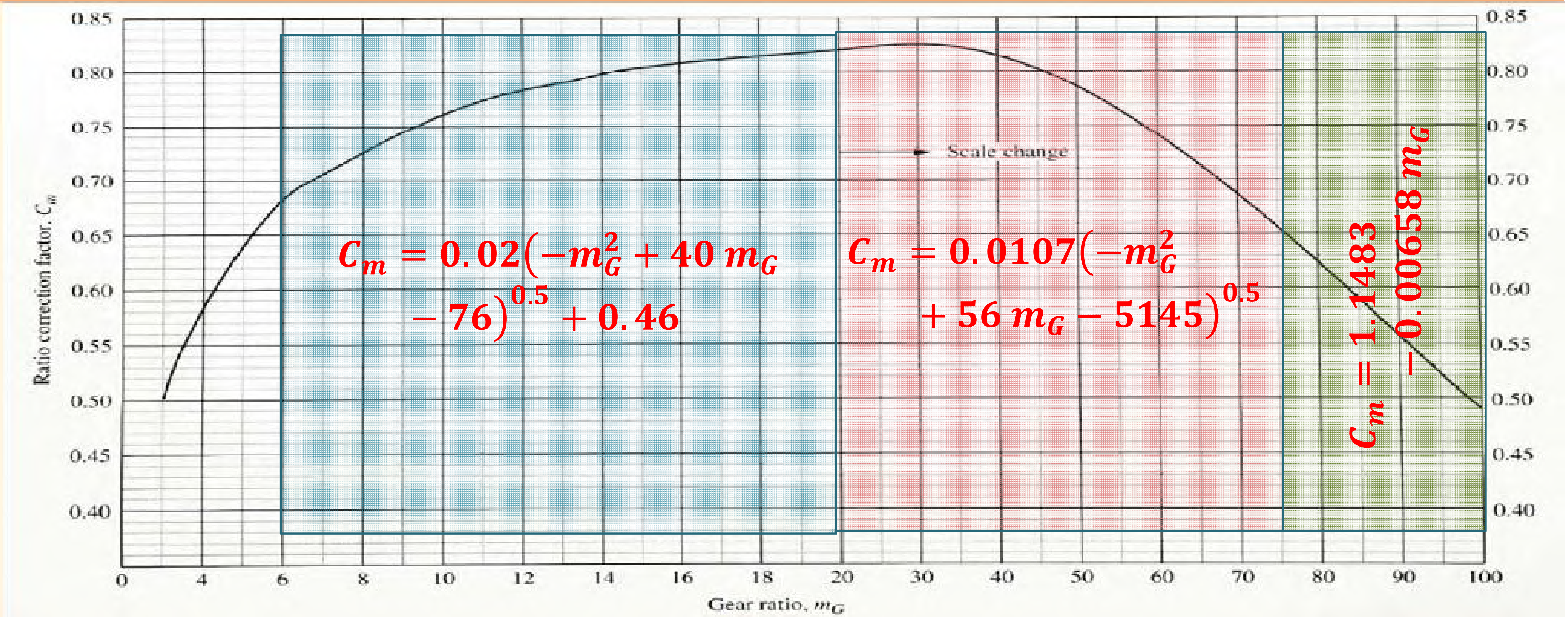
6

Compute the effective face width in inches

$$\text{if } F_G < 0.667\,(D_w) \text{ then } F_e = F_G$$

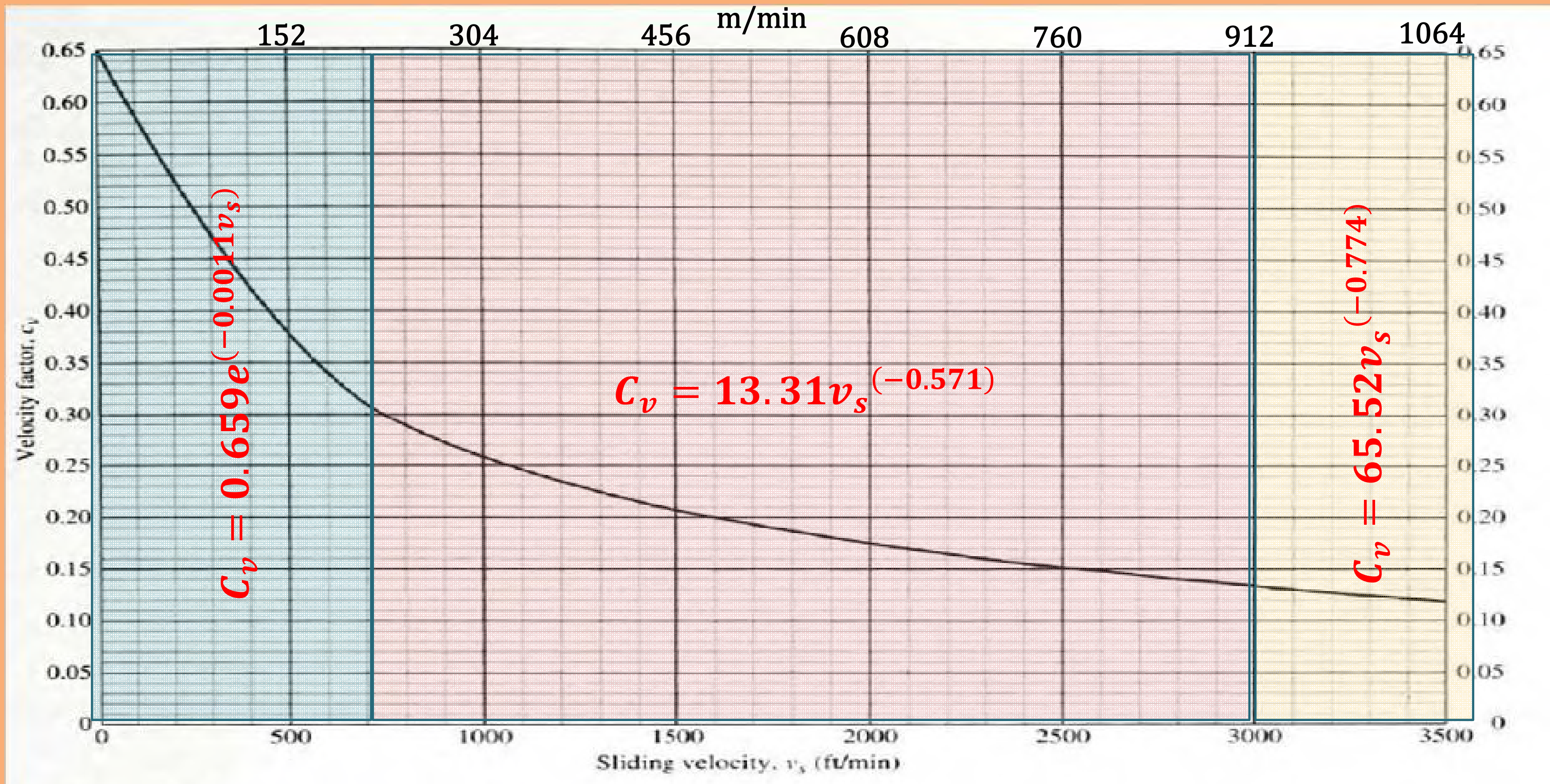
$$\text{else if } F_G > 0.667\,(D_w) \text{ then } F_e = 0.67\,(D_w)$$

Specify the Ratio correction factor C_m from figure (10-21) page(484) (500pdf)



7

Specify the velocity factor (C_v) from figure(10-22) page (485) (501 pdf)



Check if the selected dimensions satisfy the following design conditions:

First condition

$\sigma < S_{at}$ ($\mathbf{117\ MPa}$ **for manganese gear bronze**

, $\mathbf{165.5\ MPa}$ **for phosphor gear bronze ,**

and for cast iron use approximately $0.35\,\sigma_u$

Second condition

$$W_{tR} > W_{tG}$$

Third condition

$$maximum\ deflection\ of\ worm\ < 0.005\sqrt{P_x}$$

Example Problem 10–8 Is the wormgear set described in Example Problem 8–7 satisfactory with regard to strength and wear when operating under the conditions of Example Problem 10–7? The wormgear has a face width of 1.25 in.

Solution From previous problems and solutions.

$$W_{tG} = 962 \text{ lb} \qquad VR = m_G = 17.33$$
$$v_{tG} = 229 \text{ ft/min} \qquad v_s = 944 \text{ ft/min}$$
$$D_G = 8.667 \text{ in} \qquad D_W = 2.000 \text{ in}$$

Assume 58 HRC minimum for the steel worm. Assume that the bronze gear is sand-cast.

Stress

$$K_v = 1200/(1200 + v_{tG}) = 1200/(1200 + 229) = 0.84$$
$$W_d = W_{tG}/K_v = 962/0.84 = 1145 \text{ lb}$$
$$F = 1.25 \text{ in}$$
$$y = 0.125 \text{ (from Table 10–4)}$$
$$p_n = p \cos\lambda = (0.5236)\cos\ 14.04° = 0.508 \text{ in}$$

Then

$$\sigma = \frac{W_d}{yFp_n} = \frac{1145}{(0.125)(1.25)(0.508)} = 14\,430 \text{ psi}$$

The guidelines in Section 10–12 indicate that this stress level would be adequate for either manganese or phosphor gear bronze.

Surface Durability: Use Equation (10–36):

$$W_{tR} = C_s D_G^{0.8} F_e C_m C_v \tag{10–36}$$

C Factors: The values for the C factors can be found from Figures 10–20, 10–21, and 10–22. We find

$$C_s = 740 \text{ for sand-cast bronze} \quad \text{and} \quad D_G = 8.667 \text{ in}$$
$$C_m = 0.184 \quad \text{for } m_G = 17.33$$
$$C_v = 0.265 \quad \text{for } v_s = 944 \text{ ft/min}$$

We can use $F_e = F = 1.25$ in if this value is not greater than 0.67 times the worm diameter. For $D_W = 2.000$ in,

$$0.67D_W = (0.67)(2.00 \text{ in}) = 1.333 \text{ in}$$

Therefore, use $F_e = 1.25$ in. Then the rated tangential load is

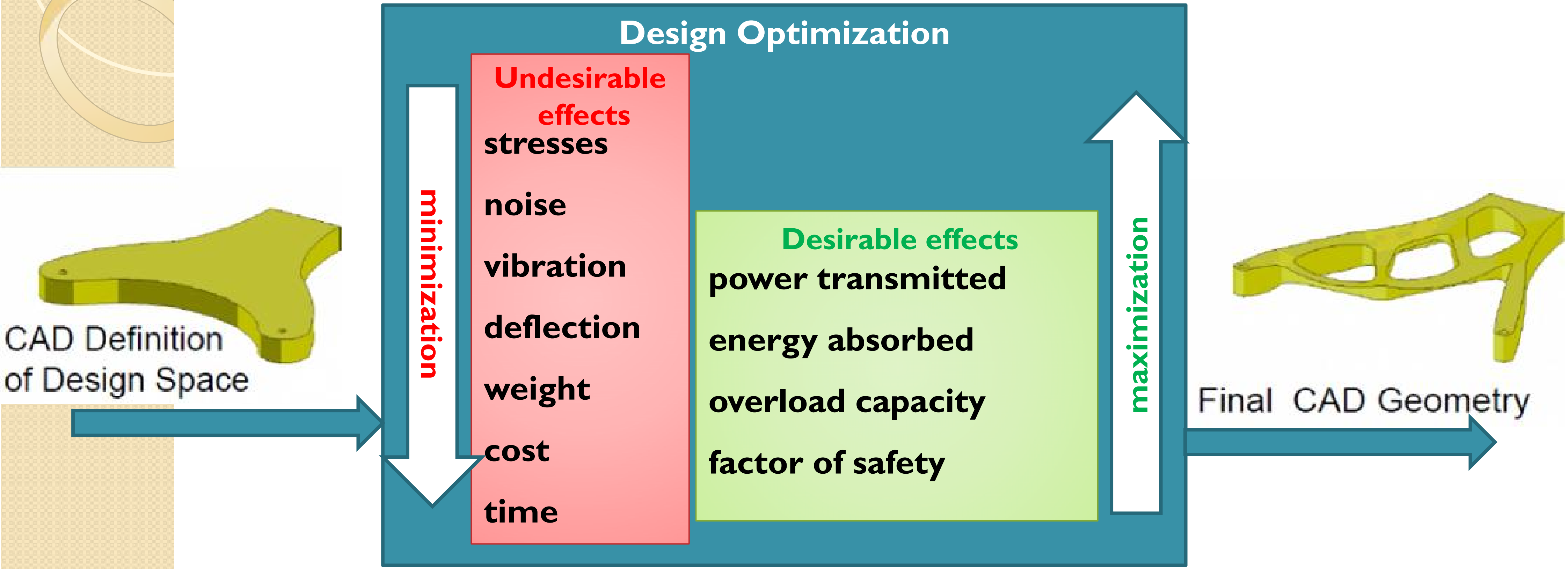
$$W_{tR} = (740)(8.667)^{0.8}(1.25)(0.814)(0.265) = 1123 \text{ lb}$$

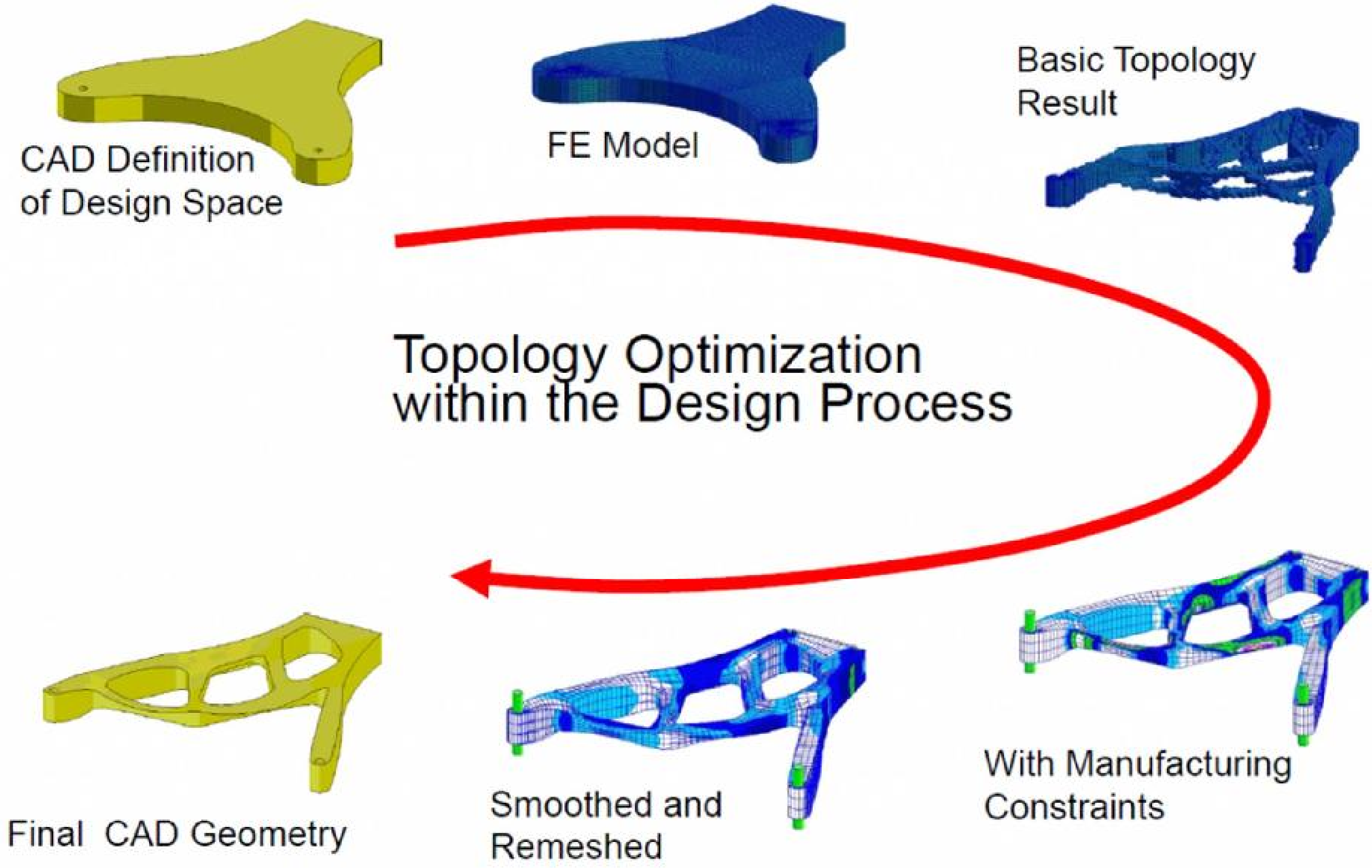
Because this value is greater than the actual tangential load of 962 lb, the design should be satisfactory, provided that the conditions defined for the application of Equation (10–36) are met.

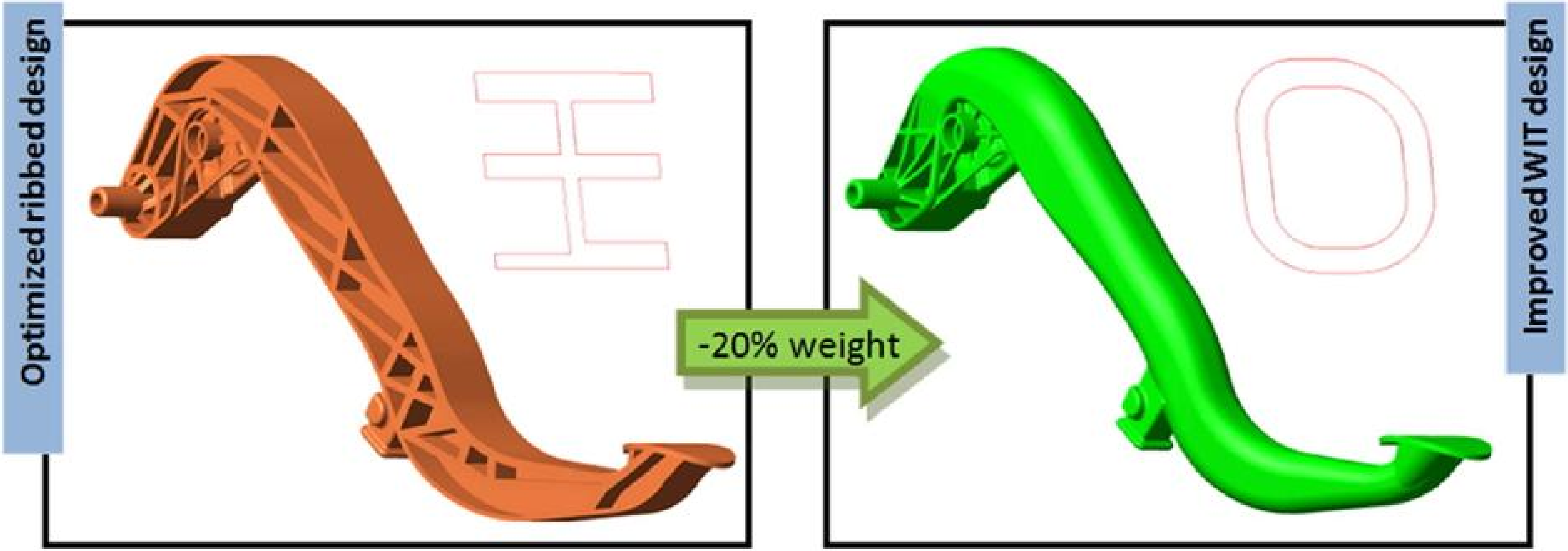
Mechanical Engineering Design II

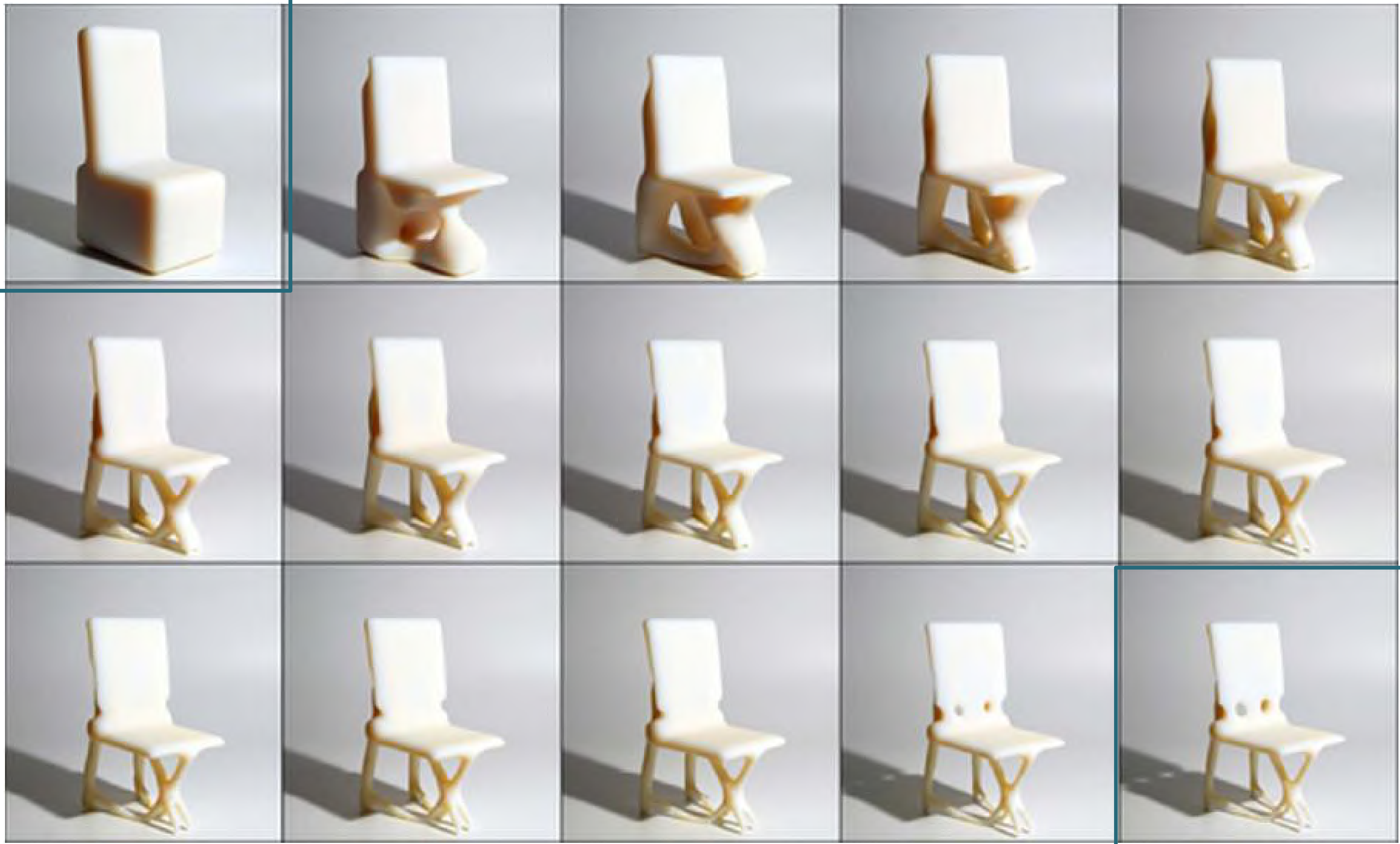
Twenty-three Lecture

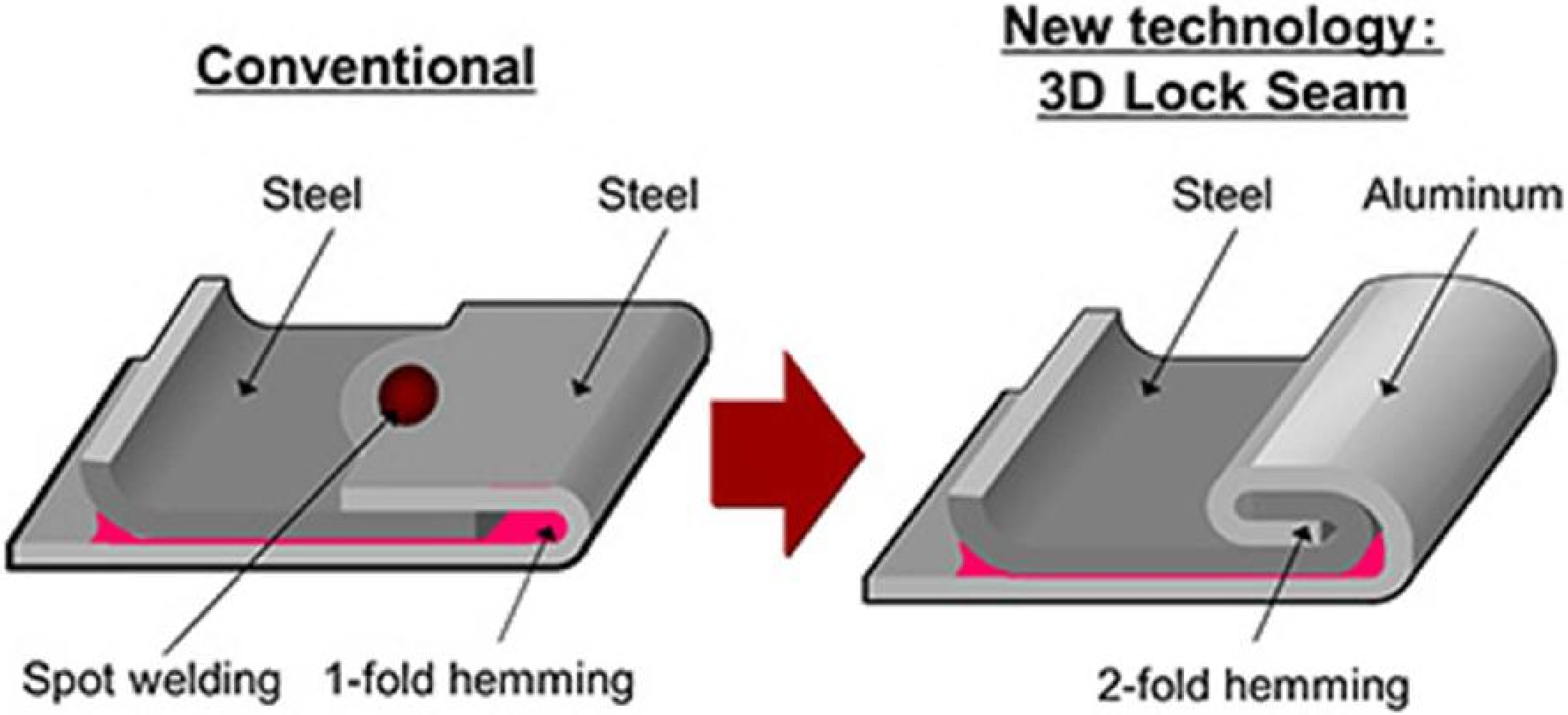
Introduction to Optimum Design

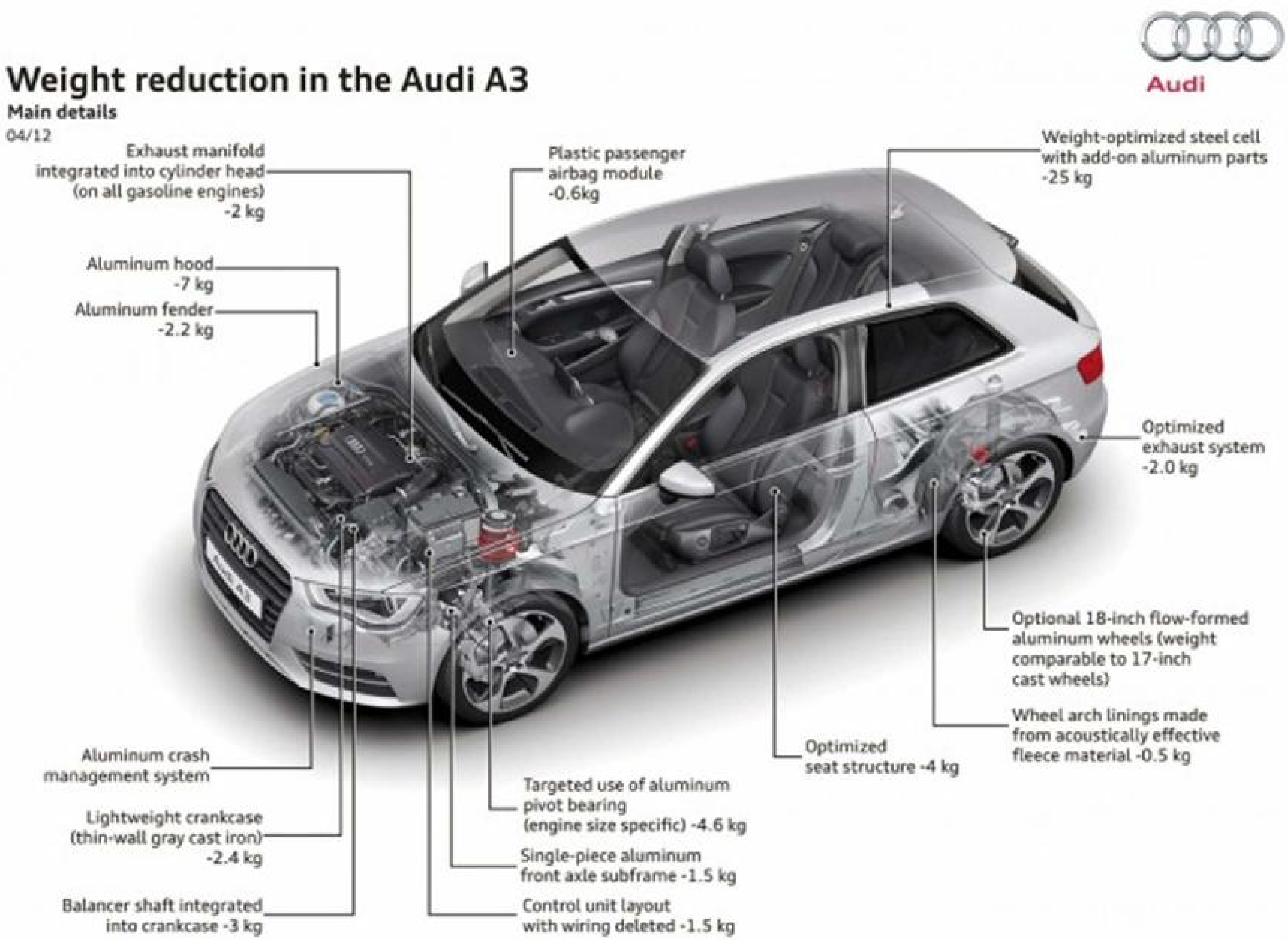

Design Optimization
Undesirable effects
stresses
noise
vibration
deflection
weight
cost
time
minimization
Desirable effects
power transmitted
energy absorbed
overload capacity
factor of safety
maximization
CAD Definition of Design Space
Final CAD Geometry


CAD Definition
of Design Space
FE Model
Basic Topology
Result
Topology Optimization
within the Design Process
Final CAD Geometry
Smoothed and
Remeshed
With Manufacturing
Constraints


Optimized ribbed design
-20% weight
Improved WIT design


Initial design
Final design


Conventional
Steel
Steel
Spot welding
1-fold hemming
New technology:
3D Lock Seam
Steel
Aluminum
2-fold hemming



Weight reduction in the Audi A3
Main details
04/12
Audi
Exhaust manifold integrated into cylinder head (on all gasoline engines) -2 kg
Plastic passenger airbag module -0.6kg
Weight-optimized steel cell with add-on aluminum parts -25 kg
Aluminum hood -7 kg
Aluminum fender -2.2 kg
Optimized exhaust system -2.0 kg
Optional 18-inch flow-formed aluminum wheels (weight comparable to 17-inch cast wheels)
Wheel arch linings made from acoustically effective fleece material -0.5 kg
Optimized seat structure -4 kg
Aluminum crash management system
Lightweight crankcase (thin-wall gray cast iron) -2.4 kg
Balancer shaft integrated into crankcase -3 kg
Targeted use of aluminum pivot bearing (engine size specific) -4.6 kg
Single-piece aluminum front axle subframe -1.5 kg
Control unit layout with wiring deleted -1.5 kg

Method of Optimum Design, MOD

Sketching a model of the item to be designed

↓

Reviewing the Boundary Conditions

Functional Requirements

Undesirable Effects

↓

Summarizing the Specifications and Constraints

↓

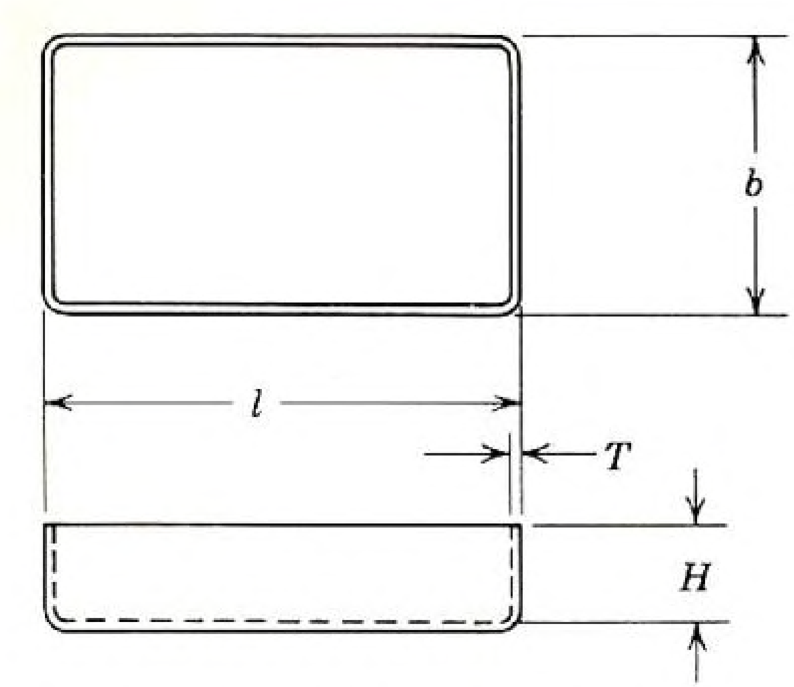
Deriving the equations which tie the design variables together mathematically

↓

Deriving the Primary Design Equation (P. D. E.)

The Basic Design Problem:

Design a *plastic tray capable of holding a specified volume of liquid, (V), such that the liquid has a specified depth of (H), and the wall thickness of the tray is to be a specified thickness, (T). The tray is to be in large quantities.*



Adequate Design Solution:

$$V = b * l * H \ldots\ldots\ldots\ldots\ldots\ldots\ldots. (1)$$

1. **It is possible to know the value of (*l*) by choosing a value for (*b*).**
2. **It is possible to choose a certain type of material for the tray.**
3. **It is possible to choose an appropriate manufacturing method.**

Optimum Design Solution:

Since the tray is manufactured in large quantity, then:

1. **The most significant undesirable effect for this problem is *COST*.**
2. **The objective of this design is to minimize *COST*.**

The design should be the best one with respect to the following:

- **Geometry**
- **Material**
- **Manufacturing method**

The cost (C) of a tray may be written as:

$$C = C_o + C_t + C_l + C_m \text{..................................(2)}$$

, which is called the *Primary Design Equation* (*P. D. E.*), where:

C = total cost
C_o = overhead cost
C_t = tooling cost
C_l = labour cost
C_m = plastic material cost
Assume Vacuum-forming techniques will be our available manufacturing method. Hence, (*Co, Ct,* and *Cl*) are independent of reasonable geometrical shapes and feasible plastic materials.

$$C_m = c.\left((b.l) + (2.b.H) + (2.l.H)\right).T \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3a)$$

where: c = a unit volume of tray material, (ID / m3).

From equation (1) and equation (3a):

$$C_m = c.\left(\left(\frac{V}{H}\right) + (2.b.\mathrm{H}) + \left(\frac{2.V}{b}\right)\right).\mathrm{T} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3b)$$

$$\frac{\partial C_m}{\partial b} = c.T.\left((2.H) - \left(2.\frac{V}{b^2}\right)\right) = 2.c.T.\left(H - \frac{V}{b^2}\right) = 0$$

$$\therefore H = \frac{V}{b^2}$$

$$\therefore b_{opt.} = \sqrt{\frac{V}{H}}$$

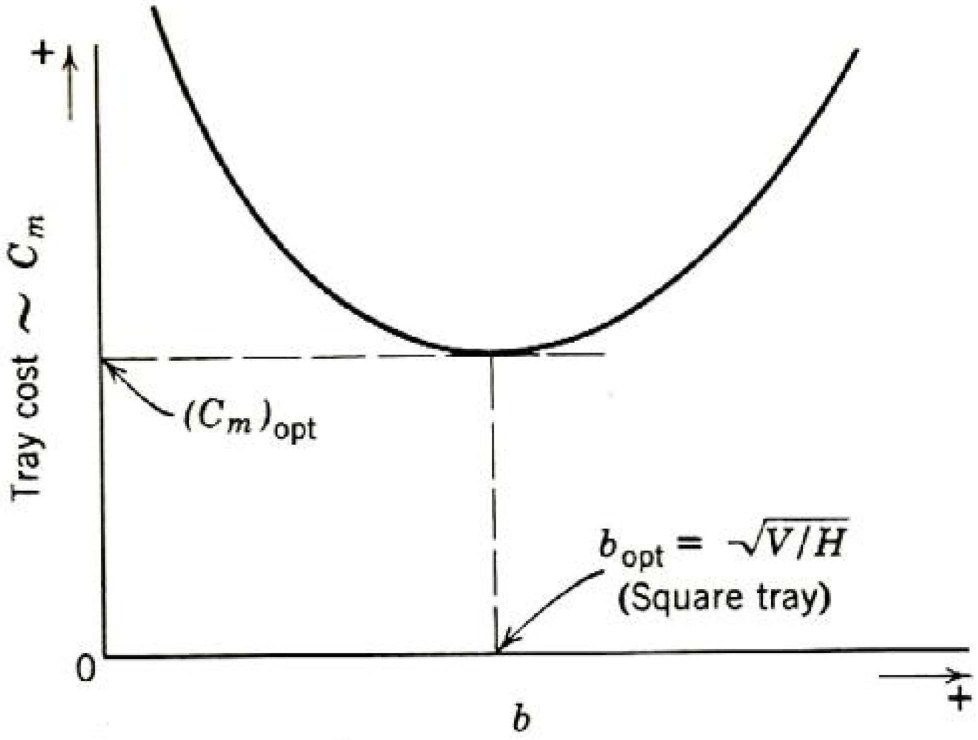
Now, $V = b * l * H$

$$\therefore l = \frac{V}{H.b} = \frac{V}{H.\sqrt{\frac{V}{H}}} = \sqrt{\frac{V}{H}}$$

$$\therefore l_{opt.} = b_{opt.} = \sqrt{\frac{V}{H}}$$

$$C_{m\,(opt)} = c.\left(\left(\frac{V}{H}\right) + \left(4.\sqrt{V.H}\right)\right).T$$

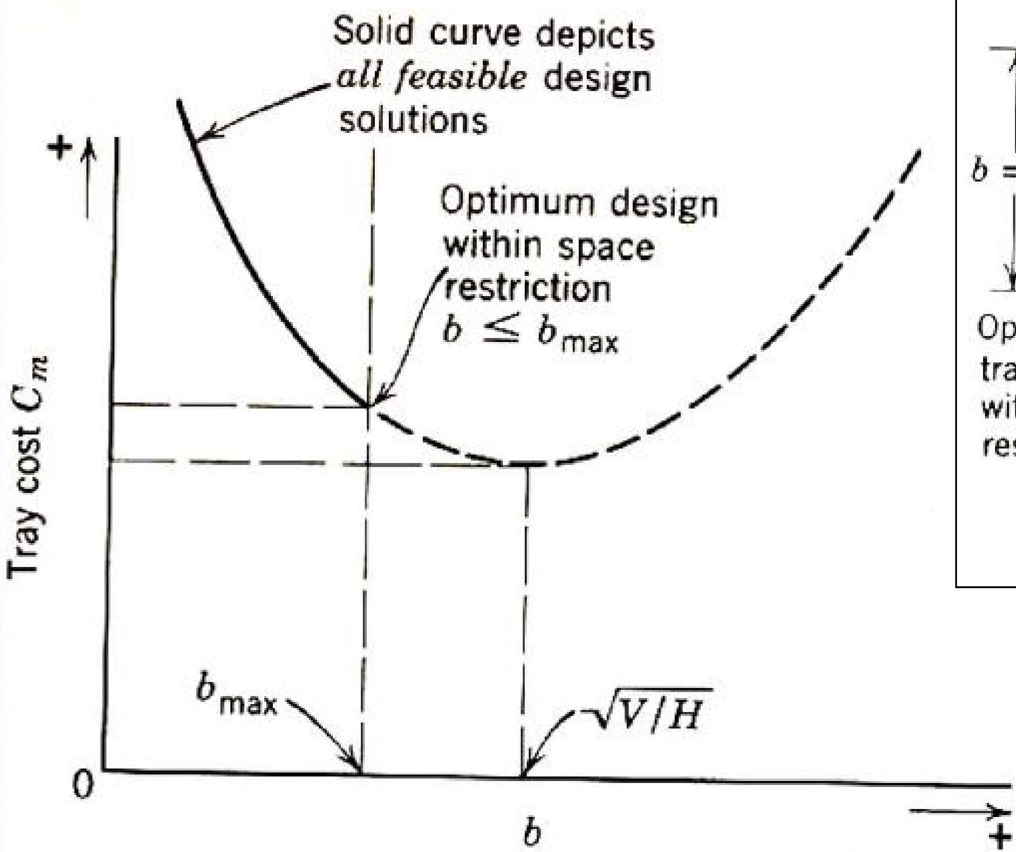
$$\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..(3c)$$

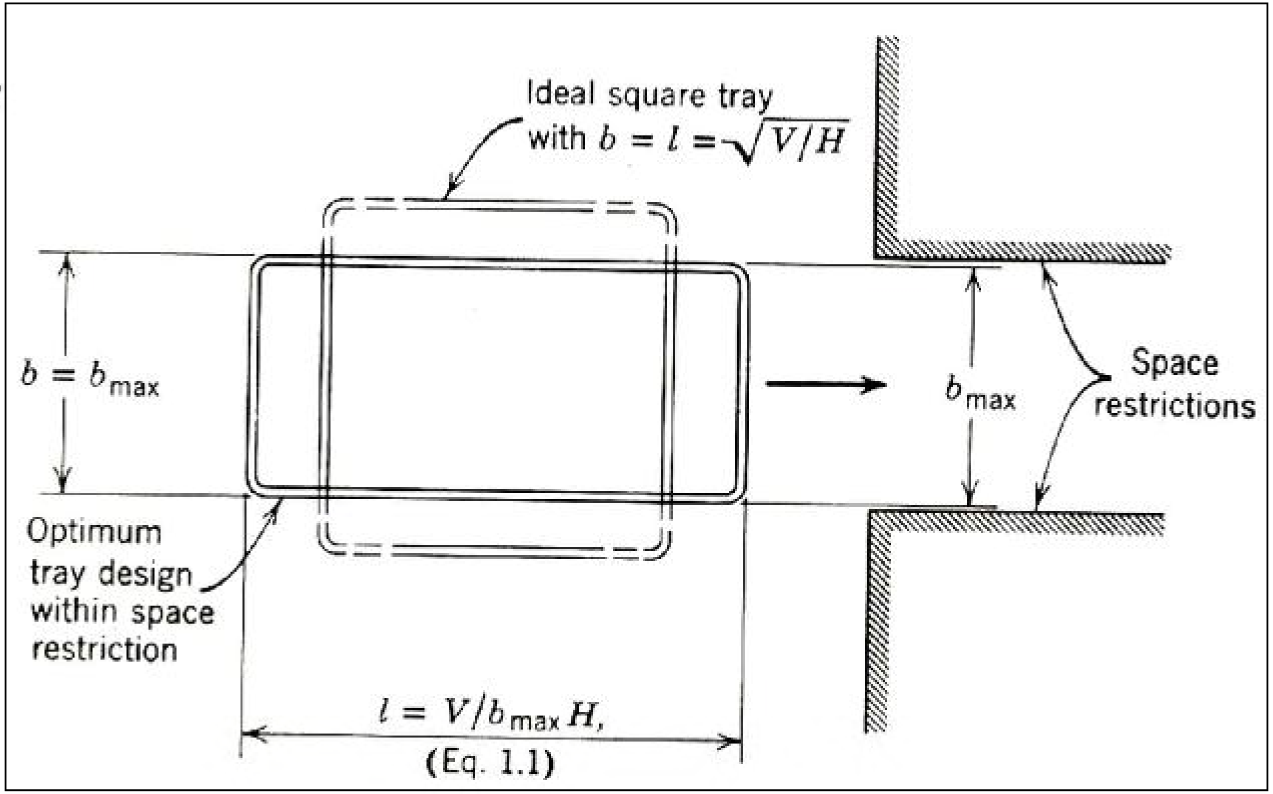


Optimum Design for Existing Restrictions:

Let:

$$b \le b_{max} \quad \text{and} \quad l \le l_{max}$$





Mechanical Engineering Design II

Twenty-four & Twenty-five Lectures

Summary of Design Equations in Optimum Design

Typical Schematic Representation of Optimum Design:

$$\overbrace{(\text{P.D.E.})}^{\substack{\text{Quantity to be}\\\text{maximized or to}\\\text{be minimized by}\\\text{optimum design}}} = f\left[\overbrace{\begin{pmatrix}\text{Functional}\\\text{requirement}\\\text{parameters}\end{pmatrix}, \overbrace{\begin{pmatrix}\text{Material}\\\text{parameters}\end{pmatrix}, \quad \begin{pmatrix}\text{Geometrical}\\\text{parameters}\end{pmatrix}}^{\substack{\text{Design parameters}\\\text{(uniquely define element)}}}}^{\text{Three } \textit{independent} \text{ groups}}\right]$$

Individually specified or limited parameters

Individual parameters dependent on each other because of physics and chemistry of the material

Individually independent parameters

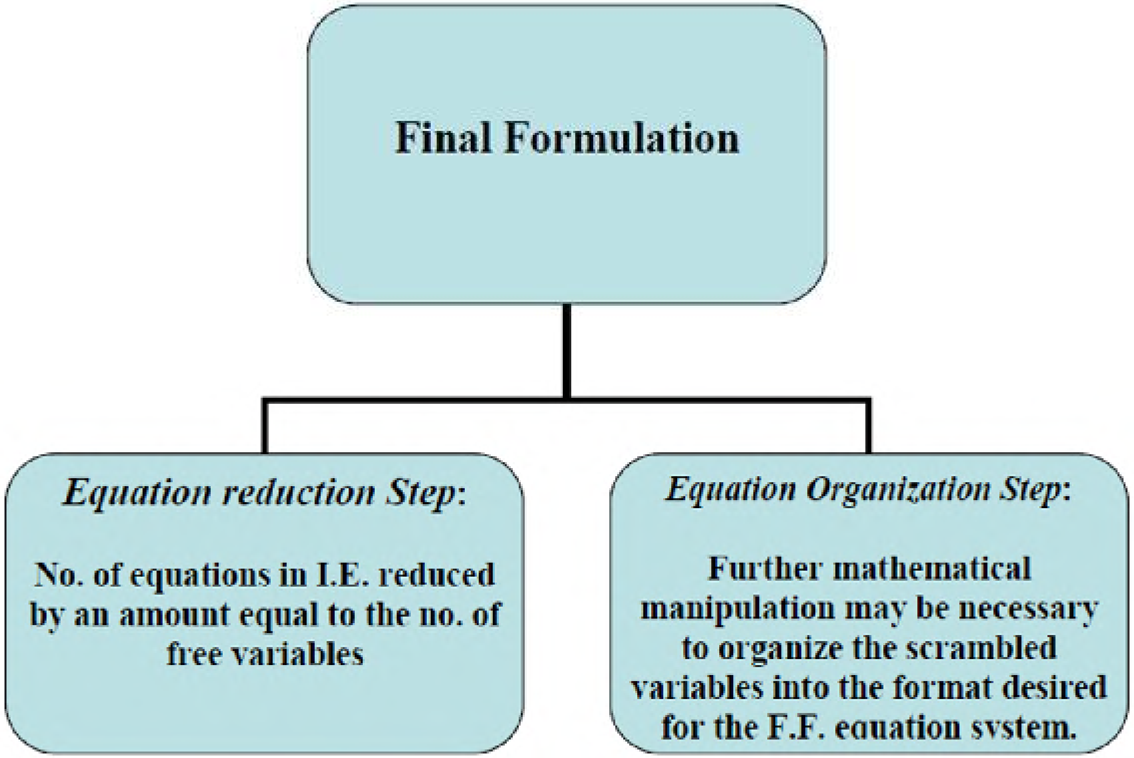
$$\underbrace{(\text{S.D.E.})}_{\substack{\text{Specified or}\\\text{limited quantity}}} = g\left[\underbrace{\begin{pmatrix}\text{Functional}\\\text{requirement}\\\text{parameters}\end{pmatrix}}_{\substack{\text{Independent}\\\text{group}}}, \underbrace{\begin{pmatrix}\text{Material}\\\text{parameters}\end{pmatrix}, \quad \begin{pmatrix}\text{Geometrical}\\\text{parameters}\end{pmatrix}}_{\substack{\textit{Groups} \text{ are dependent on specified or}\\\text{limited subsidiary design quantities } \textit{and}\\\text{on optimum design of element}}}\right]$$

$$\underbrace{h_1}_{\substack{\text{Lower limit,}\\\text{depending on}\\\text{factors external}\\\text{to element}}} \leq \underbrace{(P)}_{\substack{\text{Limited design}\\\text{parameter or}\\\text{limited (S.D.E.)}\\\text{quantity}}} \leq \underbrace{h_2}_{\substack{\text{Upper limit,}\\\text{depending on}\\\text{factors external}\\\text{to element}}}$$

Basic Procedural Steps for M.O.D.:

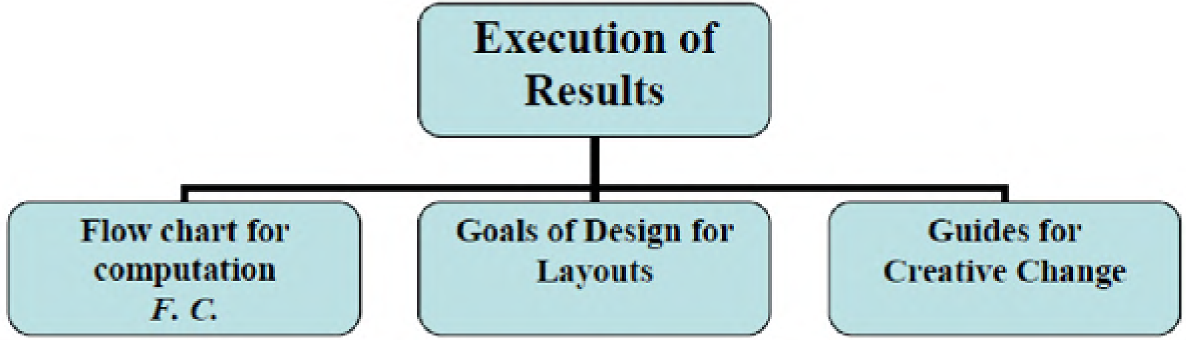
Successive steps of a systematic plan lead to the specification of the optimum design, which is summarized as follows:

1. **Initial Formulation, (*I.F.*): it is the summary of the initial system equations including (*P.D, E.*), (*S.D.E.*), and (*L.E.*)'s.**
2. **Final Formulation, (*F.F.*): it is a suitable transformation of (*I.F.*) for the use in the variation study step.**



3. Variation Study, (*V.S.*): in this step, the (*F.F.*) equation system is considered simultaneously along with the Constraints for general determination of the point's potential for Optimum Design. The sketching of Variation Diagram facilitates this step.

4. Execution of Results:



5. Evaluation of Optimum Design: here, the design is analyzed to determine what has been achieved numerically for the optimization quantity in order to confirm our acceptance of the design.

Types of Variables in I.F.:

- **Constraints parameters in number, (n_c), defined as the ones having either regional constraints or discrete value, and directly imposed by (*L.E.*)'s in the (*I.F.*).**

- **Free variables in number, (n_f), defined as, the one with no constraints directly imposed on them through (*L.E.*)'s in the (*I.F.*).**

- **Total number of the variables: $n_v = n_c + n_f$**

Types of Problems in M.O.D.:

There are basically, three classes of problems encountered in application of (*M.O.D.*). They are summarized briefly below:

1. *Case of Normal Specifications, N.S.*:

P.D.E. **is single, often an independent material selection factor, (*M.S.F.*), is recognized.**

N.S. types of problems is: [$n_f >= N_s$], where *Ns* = the number of (*S.D.E.*)'s.

2. *Case of Redundant Specifications, R.S.*:

- **Ignoring some of constraints on selected parameters that are called Eliminated Parameters.**
- **The *P.D.E.* designated as equation (I).**
- **The Eliminated Parameters must be expressed by what is called as the relating equations and designated as (II, III, IV, and so on), in (*F.F.*).**
- **The test for the (*R.S.*) type problem is [$n_f < N_s$].**

3. *Case of Incompatible Specifications*:

This is, in reality, nothing more than a special form of (*R.S.*).

There is merely no design solution that satisfy all constraints, and speed.

If boundary values changes, the domain of feasible design is opened and the optimum design can be determined by the (*M.O.D.*).

General Planning (I.F.) to (F.F.) in M.O.D.:

One of the most difficult details of execution lies in the transformation of the (*I.F.*) to (*F.F.*). Three general items are helpful in this respect. They will be outlined briefly below for the (R.S.) type of problem:

1. *Exploratory Calculations*:

$D_{vs} = n_v - N_s + 1$, where:

D_{vs} =number of dimensions required for a (*V.S.*) , where: $A_t = \frac{n_c!}{[n_e!(n_c - n_e)!]}$

A_t = number of different approaches

n_c = number of constraints variables

n_e = number of eliminated parameters

$n_e = N_s - n_f$

$n_r = n_c - n_e$, where: n_r = number of related parameters.

2. *Choosing the Approach*:

- **Choice of the particular approach to take for derivation for the specific (*F.F.*), should be made after thought.**
- **Choose the simplest approach.**

3. *General Format of (F.F.)*:

Summarizing the (F.F.) equations can be very helpful.

Example (1): Case of Normal Specifications

Simple tensile bar, (mass production manufacturing)

L = specified length
P = constant force
Optimum design required to minimizing cost

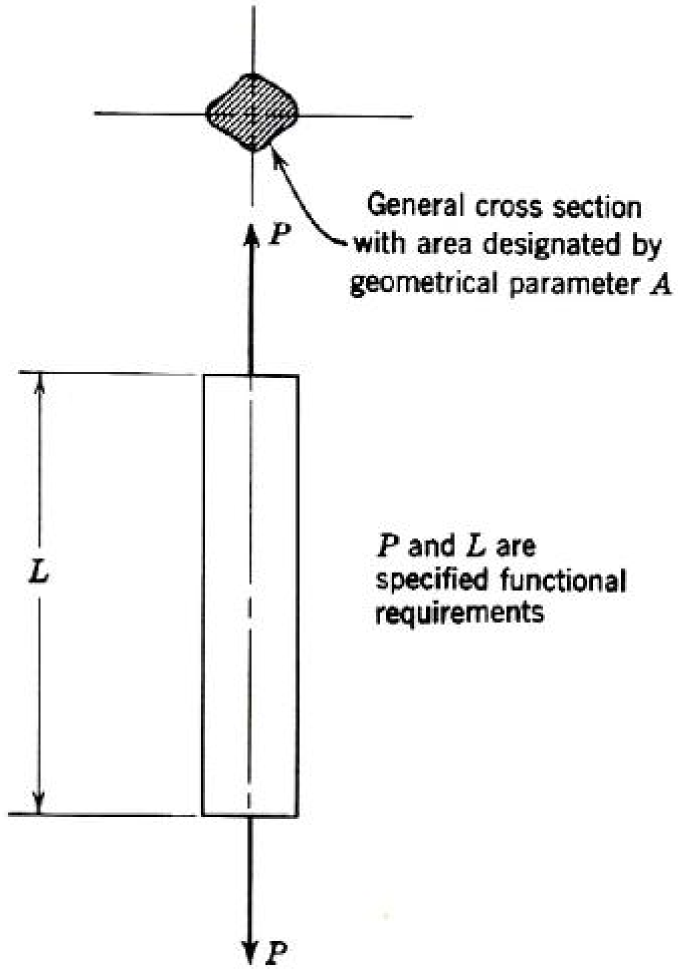
- Now, the (P.D.E.) is:

Cm = material cost =*c.V* = *c.(A.L)*, where:
Cm = cost of the bar material,
c = unit volume material cost

- The undesirable effect are the stress =σ=*P/A*

, which is the (*S.D.E.*).

- The limit equations are: $\sigma \leq \frac{S_y}{N_y}$



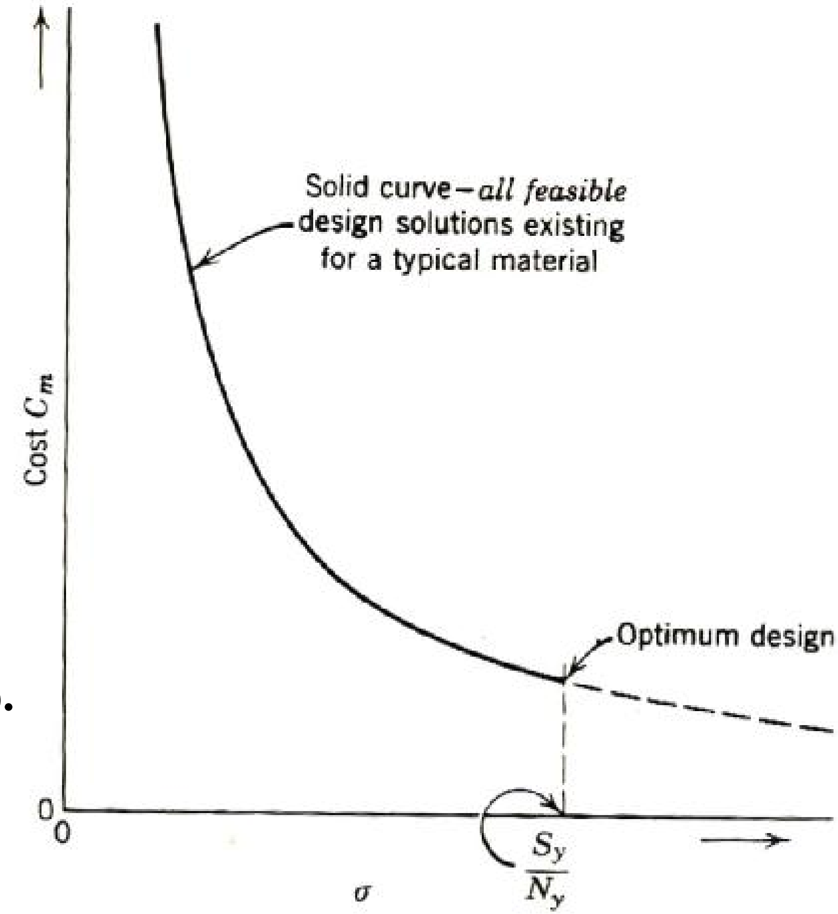
- Now, *P*, *L* and *Ny* are the Functional Requirements,
- *c* & *Sy* are the Material Parameters,
- *A* is the Geometrical Parameter,
- σ &*Cm* are the Undesirable Parameters.

- **Now, number of free variables = 1, which is (*A*), number of (S.D.E.) = 1, therefore $n_f = N_s$**

- **Now combine the (*S.D.E.*) with the (*P.D.E.*), eliminating the unlimited and unspecified geometrical parameter (*A*). thus:**

$$c_m = c \cdot L \cdot \left(\frac{P}{\sigma}\right),$$

$$c_m = c \cdot L \cdot \left(\frac{P}{S_y / N_y}\right) = (P \cdot L \cdot N_y) \cdot \left(\frac{C}{S_y}\right)$$ **which is the (*P.D.E.*).**



***P*, *L* and *Ny* are specified functional requirement, which is probably, cannot be changed.**

$\left(\frac{C}{S_y}\right)$, **is the (M.S.F.) (Material Selection Factor) which is the independent group.**

- **Finally, for determining the optimum value of (*C.S.A.*), (*A*), we would calculate very simply the optimum value of the area (*A*).**

Example (2): Case of Redundant Specifications

Simple tensile bar, (mass production manufacturing), with: $A \geq A_{min}$

Now, the (*I.F.*) is:

$C_m = C \cdot L \cdot A$ **(*P.D.E.*)**

$\sigma = \dfrac{P}{A}$ **(*S.D.E.*)**

$\sigma \leq \dfrac{S_y}{N_y}$ **(*L.E.*)**

$A \geq A_{\min}$ **(*L.E.*)**

- **In this case, it is impossible to combine (*S.D.E.*) with (*P.D.E.*). Now:**

$n_f = 0$ and $N_s = 1$

Therefore, the case is a Redundant Specification.

- **There are two approaches:**

1. **Ignore the (*S.D.E.*)**
2. **Ignore the (*L.E.*), on stress or in (*A*).**

Exploratory calculations:

There are two approaches possible for (_F.F._)'s.
Our (_V.S._) will be two dimensional in character.

- **Say the simplest approach:**
  σ is the eliminated parameter
  ***A* is the related parameter.**

$$n_f = o \,\&\, N_s = 1 \,\&\, n_v = n_c + n_f = 2$$

$$n_e = N_s - n_f = 1 - 0 = 1 \,\&\, n_r = n_c - n_e = 2 - 1 = 1$$

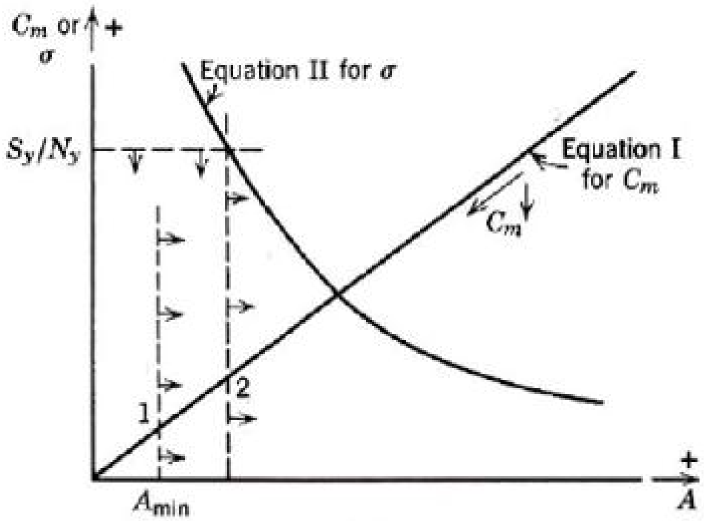
$$D_{vs} = n_v - N_s + 1 = 2 - 1 + 1 = 2$$

$$A_t = \frac{n_c\cdot!}{[n_e!\cdot(n_c - n_e)!]} = \frac{2\cdot!}{1!\cdot(2-1)!} = 2$$

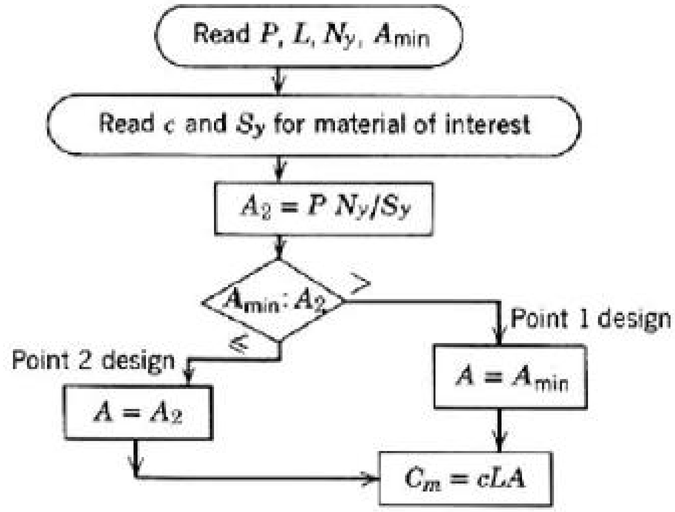
The (F.F.) is the same as the (I.F.)

$$C_m = C \cdot L \cdot A \qquad (I)$$

$$\sigma = \frac{P}{A} \qquad (II)$$



(a)



(b)

Example (3): Case of Redundant Specifications

Simple tensile bar, (mass production manufacturing), with:

A ≥ Amin,
Δ ≥ Δmax,
Lmin ≤ L ≤ Lmax

The (*I.F.*) is:

$$C_m = cLA \qquad \text{(P.D.E.)}$$

$$\sigma = \frac{P}{A} \qquad \text{(S.D.E.)}$$

$$\Delta = \frac{PL}{(AE)} \qquad \text{(S.D.E.)}$$

$$\sigma \leq \frac{S_y}{N_y} \qquad \text{(L.E.)}$$

$$A \geq A_{\min} \qquad \text{(L.E.)}$$

$$\Delta \leq \Delta_{\max} \qquad \text{(L.E.)}$$

$$L_{\min} \leq L \leq L_{\max} \qquad \text{(L.E.)}$$

The (*V.S.*) are:

$$n_f = o < N_s = 2$$

$$n_v = n_c = 4$$

$$n_e = 2 - 0 = 2$$

$$n_r = 4 - 2 = 2$$

$$D_{vs} = 4 - 2 + 1 = 3$$

$$A_t = \frac{n_c \cdot !}{[n_e ! \cdot (n_c - n_e)!]} = \frac{4 \cdot !}{2 \cdot ! \cdot (4-1)!} = 6$$

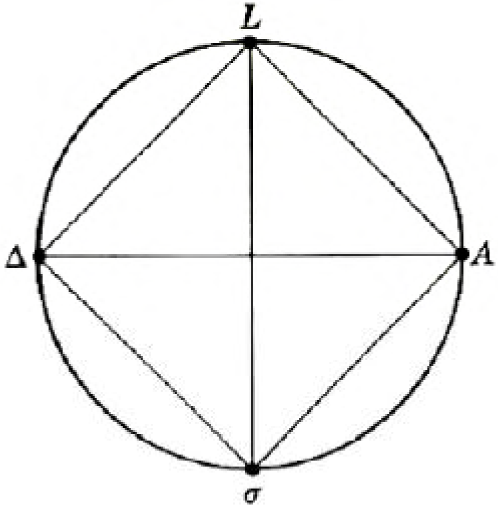


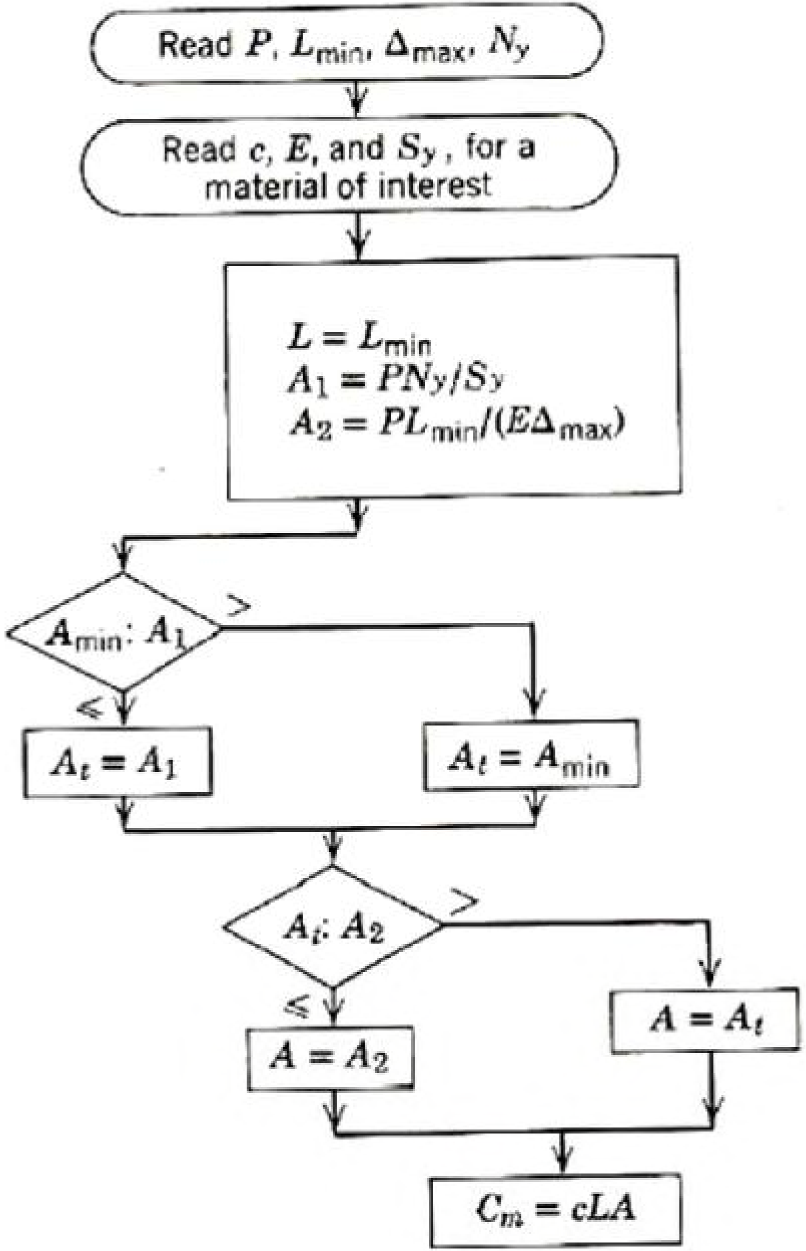
Figure 6.11 Circle diagram for example 6.4.

Therefore, the (*F.F.*) are:

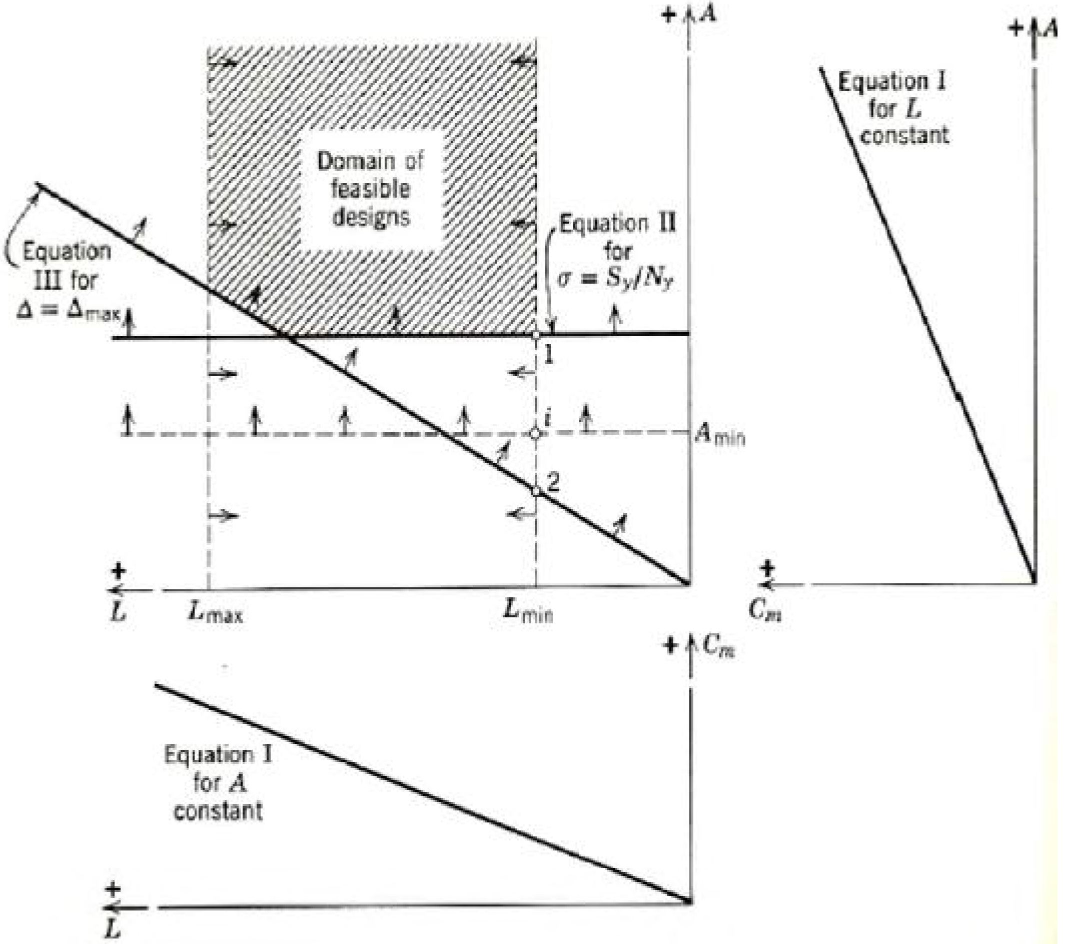
$$C_m = C \cdot L \cdot A \qquad (I)$$

$$\sigma = \frac{P}{A} \qquad (II)$$

$$\Delta = \frac{P \cdot L}{A \cdot E} \qquad (III)$$



Read P, L_{min}, Δ_{max}, N_y
Read c, E, and S_y, for a material of interest
$L = L_{min}$
$A_1 = PN_y/S_y$
$A_2 = PL_{min}/(E\Delta_{max})$
$A_{min} : A_1$
$>$
$\leqslant$
$A_t = A_1$
$A_t = A_{min}$
$A_t : A_2$
$>$
$\leqslant$
$A = A_2$
$A = A_t$
$C_m = cLA$



$+ A$
Domain of feasible designs
Equation III for $\Delta = \Delta_{max}$
Equation II for $\sigma = S_y/N_y$
1
i
2
A_{min}
$+ L$
L_{max}
L_{min}
Equation I for L constant
$+ C_m$
Equation I for A constant

Appendix (A)

Examinations Questions of Previous Academic Years

University of Technology
Department of Machines and Equipment Engineering
Final Year Examination [illegible]

Subject: Design III
Division: General
Examiner: Design group

Year: Fourth Year
Exam Time: 3 Hrs.
Date: [illegible]

Note: 1- Tables, charts and design formulas are allowed.
2- Assume any missing data.

Q1. Figure 1, shows the portable concrete mixer. For this machine, the following are required:

1- Divide this machine to multi-sub-systems by using the morphological chart method. (10 marks)
2- For the morphological chart in (1), give at least one another item with a simple clear sketch for each sub-system of this machine. (15 marks)
3- Describe the new scheme by using your own ideas in (2) [illegible]. (10 marks)

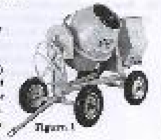


Figure 1

Q2. Figure 2, shows a heavily loaded belt conveyor to be driven by a gear reducer through a chain drive. The input speed will be 900 rpm, and the desired output speed is 250 to 300 rpm. The conveyor requires 11.2 kW.

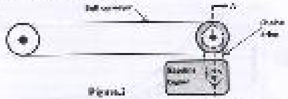
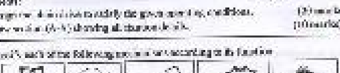


Figure 2

Requirements:

1- Design the chain drive to satisfy the given operating conditions. (20 marks)
2- Draw section (A-A) showing all chain drive details. (10 marks)

Q3. Classify each of the following mechanisms according to its function.



(a) (b) (c) (d) (e)

Function of mechanism	Mechanism
Reciprocating mechanism	
Clamping mechanism	
Indexing mechanism	
Coupling and connecting mechanism	
Locking and release [illegible]	

(10 marks)

Q4. Note: Answer branch (A) or (B).

(A) Design a plastic tray capable of holding a specified volume of liquid (V). State that the liquid has a specified depth (H) and the wall thickness of the tray is to be a specified thickness (T). The tray is to be manufactured by injection molding.

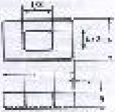
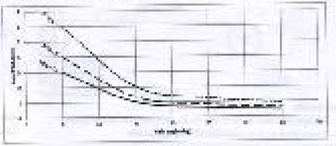


Figure 3: Typical tray with a defined hollow rectangular shape.

(25 marks)

(B) The spur gear will be designed with a helix angle ([illegible] = 5° – 25°) and (Pd = 6 – 12 teeth/in), the following figure shows the results for normal pitch (Pn) and [illegible]



Find the feasible point that gives the smallest size of the gear set that has the following limitations:

(10° < [illegible] < 25°)

(12 < Pn < 99)

([illegible] < Pg < 22)

(25 marks)

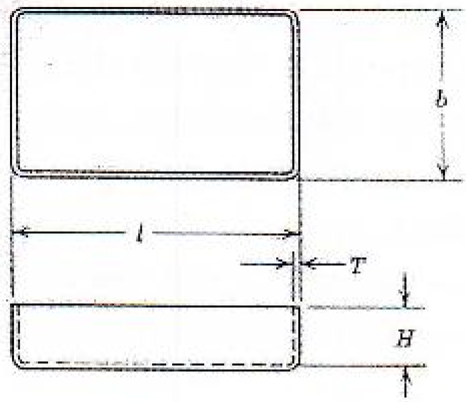
University of Technology
Department of Machines and Equipment Engineering
Final Examination 2012/2013

Subject: Design II | Year: fourth
Division: General Mech. | Exam Time: 3 Hrs.
Examiners: Design Group | Date: 27/5/2013

Answer (four) Questions Only

Q1: Note: answer branch (A) or branch (B).

(A) Design a plastic tray capable of holding a specified volume of liquid (V). Such that the liquid has a specified depth (H) and the wall thickness of the tray is to be a specified thickness (T). The tray is to be manufactured in large quantities.



(25 mark) **Figure. (1a) Typical tray with traditional rectangular shape.**

(B) The simple tensile bar which must transmit a specified constant magnitude force (P) as shown in figure (1b). Assume that the bar will be manufactured in large quantities, thus a logical objective for optimum design would be minimization of cost. For an acceptable design, cross-sectional area (A), elongation (Δ), length (L), and nominal stress (σ) must satisfy the following constraints:

$87.5\ mm^2 \le A \le 314\ mm^2$

$500\ mm \le L \le 750\ mm$

$0.0077mm \le \left(\Delta = \frac{P.L}{EA}\right) \le 0.02mm$

$\sigma_{all} \le 100\ N/_{mm^2}$

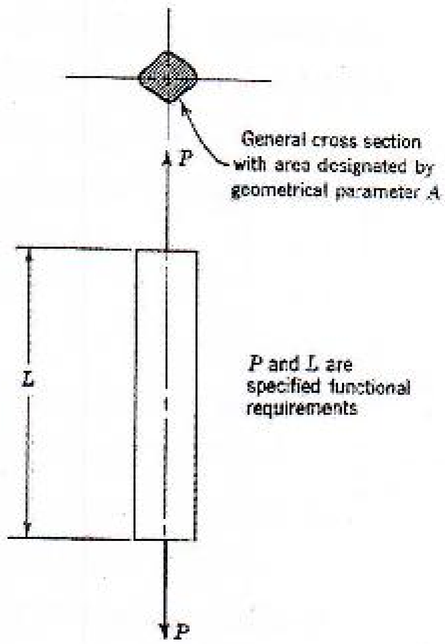


Figure.(1b) simple tensile bar with uniformly distributed specified axial load (P)

Safety factor ≥ 3 , c = unite volume cost of shaft = 2500 $\$/m^3$
E= 207 Gpa & P=1000 N

Find minimum cost and at what length and area? (25 mark)

Q2: Figure.2 shows industrial saw No.4 that will be used to cut the specimen No.5. The saw will received power from the shaft of an electric motor (C). The drive shaft for the saw (B) takes rotation from shaft (C), by using v-belt No.6.

Table.1 shows the morphological chart for the system scheme (figure.2), after the system divided into three sub systems.

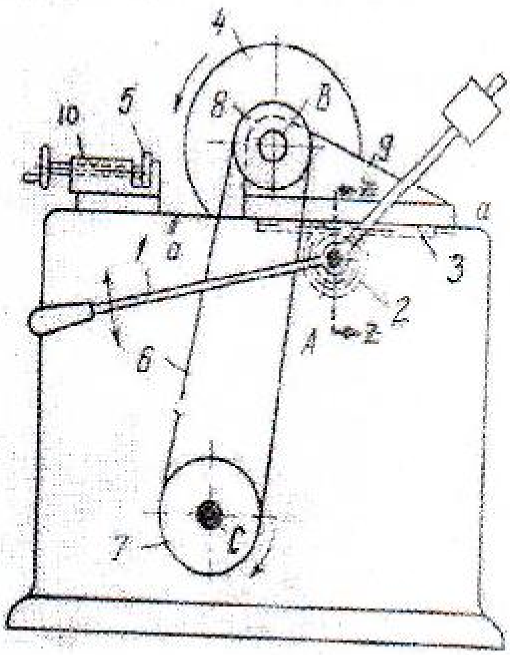


Figure.2 system scheme for the industrial saw

المنشار الدائري رقم (4) يقوم بقص القطعة رقم (5) المثبتة على الماسكة رقم (10) . ان المنشار يدار عن طريق آلية من ضمنها البكرات رقم (8) ورقم (7) والحزام رقم (6). ولتقريب المنشار على القطعة رقم (5) يتم تحريك الذراع رقم (1) عكس عقرب الساعة عندها يدور الترس رقم (2) الذي يقوم بتحريك الجريدة المسننة رقم (3) المثبتة على القطعة رقم (9) خطياً وعلى المسار (a-a) وعند حركة الذراع رقم (1) مع عقرب الساعة يبتعد قرص المنشار عن القطعة رقم (5).

Requirements:

1- Draw only section showing all details for other ideas for mechanism B2 and C2, Completely different than B1 and C1.
2- Draw the new system scheme that follow the path (A1,B2,C2) showing all necessary details to clarify the new system.
3- Apply the method of inversion to find a new idea for any sub-system you are choosing from figure.2 .

(25 mark)

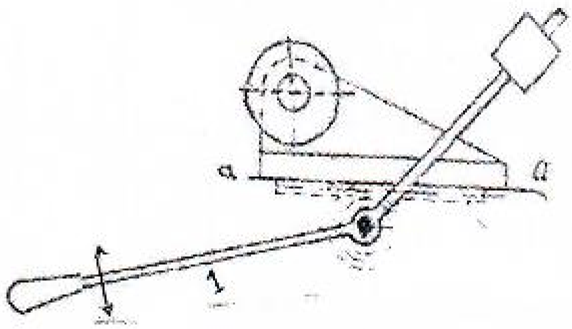
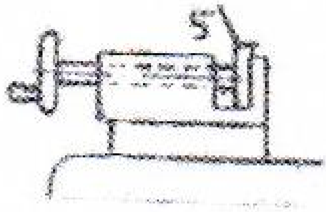
Sub-systems \ Alternatives	1	2
Power transmission from motor to saw No.4	A1 	
Mechanisms for moving the saw toward the specimen	B1 	B2
Mechanisms for clamping specimen	C1 	C2

Table.1 The system will be divided into sub-systems

University of Technology
Department of Machines and Equipment Engineering
Final Examination 2012/2013

Subject: Design II
Division: General Mech.
Examiners: Design Group

Year: fourth
Exam Time: 3 Hrs.
Date: 27/5/2013

Q3: If the type of belt No.6 used in figure.2. is 5V belt that will be applied to two sheaves No.7 and No.8 with pitch diameters (703.58 mm) and (213.36 mm) respectively, with center distance of no more than (1524 mm).

Requirements:

1- Find standard length of belt.
2- Find actual center distance.
3- Find angle of wrap on both of the sheaves after finding the actual center distance.
4- Find the rated power considering corrections for speed ratio, belt length and angle of wrap.
5- Draw section (Z-Z) in figure.2

(25 mark)

Q4: Figure.3 shows two step spur gear reducer,

Speed of shaft No.1 = 183.22 rad/sec
Speed of shaft No.3 = 30.57 rad/sec
Speed of shaft No.2 = 61.14 rad/sec
Power transmitted =2.2 kw
Assume: Km = 1.3 , Hardness ratio factor C_H =1 , Kv = 1.3
Requirements:

1- Specify materials for gears No.4 and No.5.
2- Draw the free hand sketch for doted area showing how the outer races of the bearings were fixed in the housing of the gearbox.

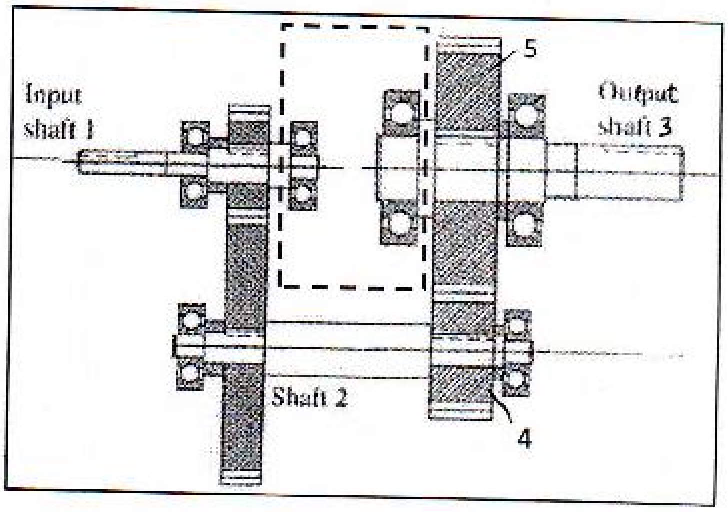


Figure.3

(25 mark)

Q5: Figure (4) shows a straight bevel gear pair has the following data:

Number of teeth $N_1 = N_2 = N_3 = 25$

$P_d = 10$ (m=2.54)

Gear speed = 1250 r.p.m

Power transmitted =2.61 kw

Assume: Kv=0.823 , Km=1.44

Requirements:

1- Specify a suitable material and heat treatment for all gears.
2- Draw section (x-x) .

(25 mark)

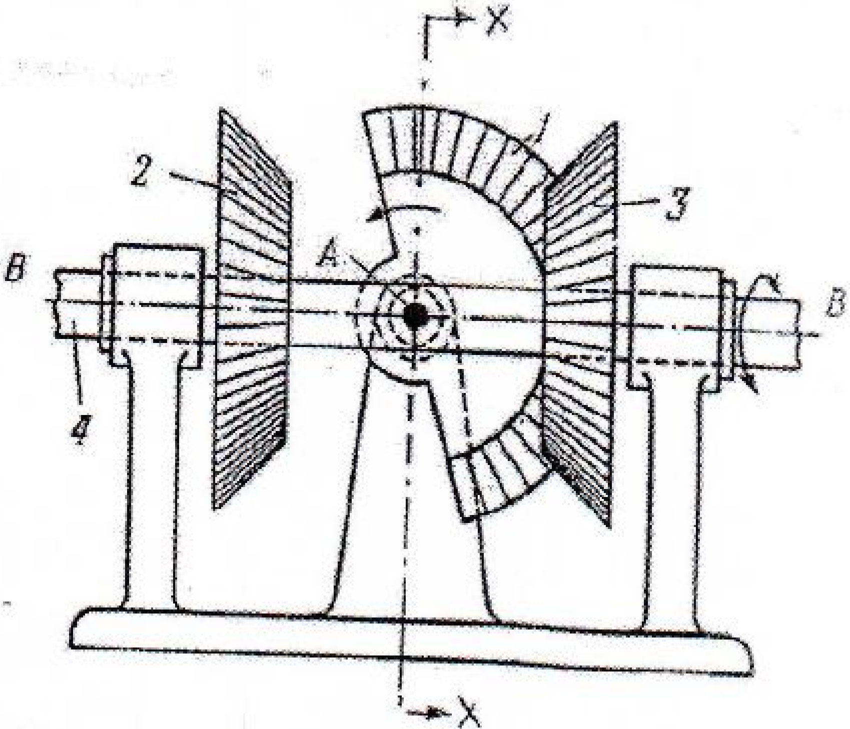


Figure.4

الشكل (4) يمثل الية نقل الحركة باستخدام التروس المخروطية حيث ان المسنن رقم (1) هو نصف او جزء من مسنن مخروطي يدور حول المحور A. الترسين المخروطين (2) و (3) متعشقان مع العمود رقم (4).

University of Technology
Department of Machines and Equipment Engineering
First Term Postponed Examination 2012/2013

Subject: Design II | **Year: fourth**
Division: General Mech. | **Exam Time: 1:30 Hrs.**
Examiners: Design Group | **Date: 28/2/2013**

Answer (two) Questions Only

Q1: Sketch the system design flowchart showing all details. Then discuss the market research and application analysis for ((Design Equipment to Convert Waste)). Write also the refined statement for the system.

(50 mark)

Q2: Fig.1 shows the network combination for sliding door done by a student. He choose path (A_2 B_5 C_1 D_3 E_3 F_1). You can connect or add each item from subsystem to another item from other subsystem. The requirements are:

1- Draw complete system scheme in detail.
2- Discuss how to improve reliability of the system.

(50 mark)

Q3: A link in a mechanism is (1371.6mm) the input data and result was shown with missing information. The relationship between diameter, maximum stress and allowable load as shown in fig.2. The flowchart for the column is shown in fig.3. choose suitable stress to find all missing information in datasheet.(Note: safety factor = N = 6).

(50 mark)

NETWORK COMPINATION
FOR SLIDING DOOR

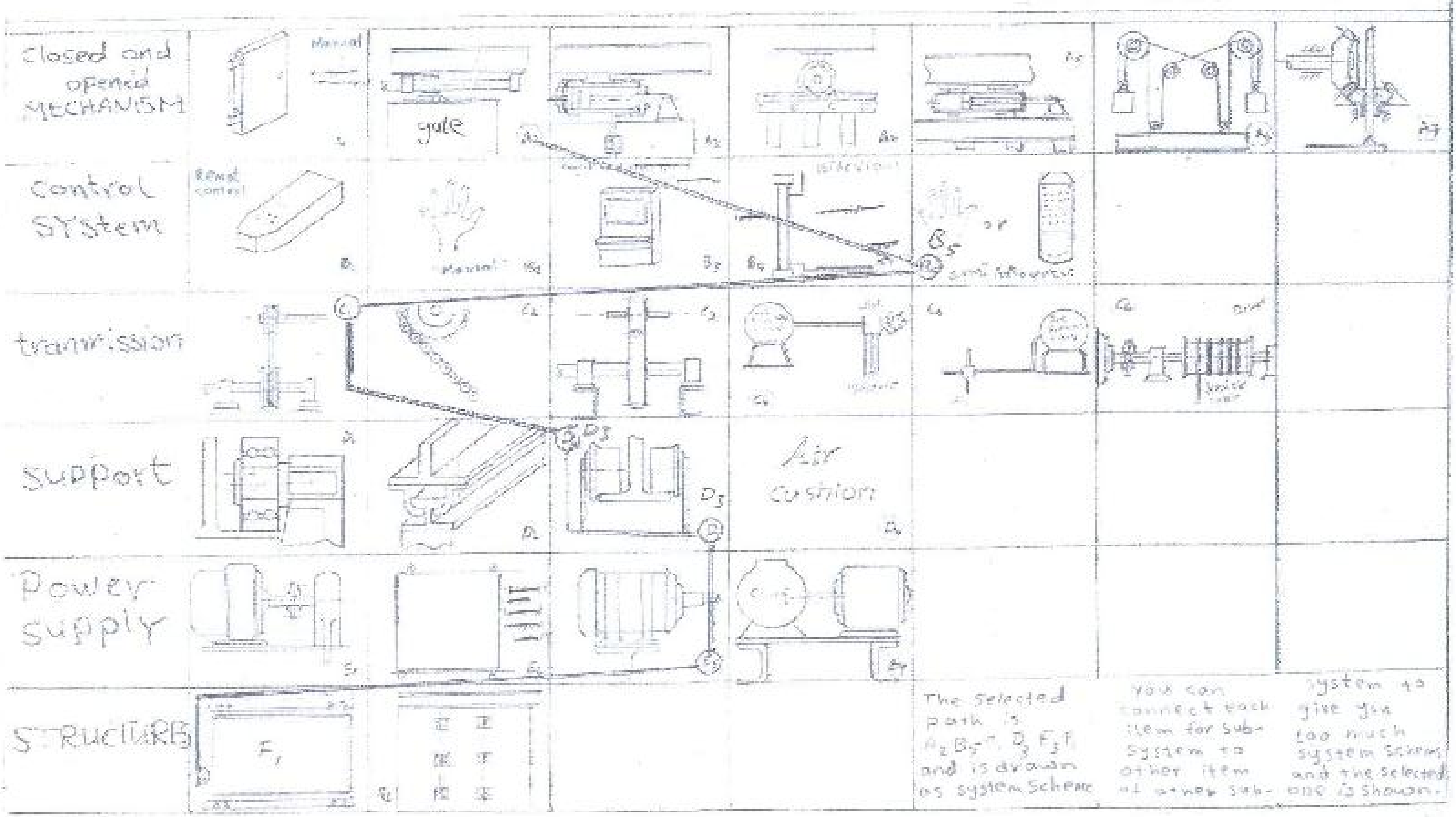


Fig. 1.

الشكل يمثل احد الاختيارات للنظام من قبل احد الطلاب

ملاحظة: في حالة عدم وضوح اي شكل من الاشكال اعلاه يتقدم بتصميم مناسب
لكي يمثل لك الاختيار النهائي حسب ما تراه.

Input data:

Column Analysis

	Length and End Fixity		
Column length	L =	1371.6	mm
End fixity coefficient	K =	1	
Initial crookedness	a =	0	mm
Eccentricity	e =	0	mm
Applied load	P =		N
	Material Properties		
Yield strength	Sy =	351632.76	kPa
Modulus of elasticity	E =	206842800	kPa
	Cross Section Properties		
Type of the column cross-section	Circle		
Diameter	D =		mm
	Design Factor		
Design factor on load	N =	6	

Results

The column is Long, straight

Area	A	=		mm^2
Neutral axis to outside	c	=		mm
Effective length	KL	=		mm
Radius of gyration	r	=		mm
Slenderness ratio	KL/r	=		
Column constant	Cc	=		
Critical buckling load	Pcr	=		N
Allowable load	Pa	=		N
Maximum stress	σ	=		kPa

No relevant formula at this moment to calculate Ymax.

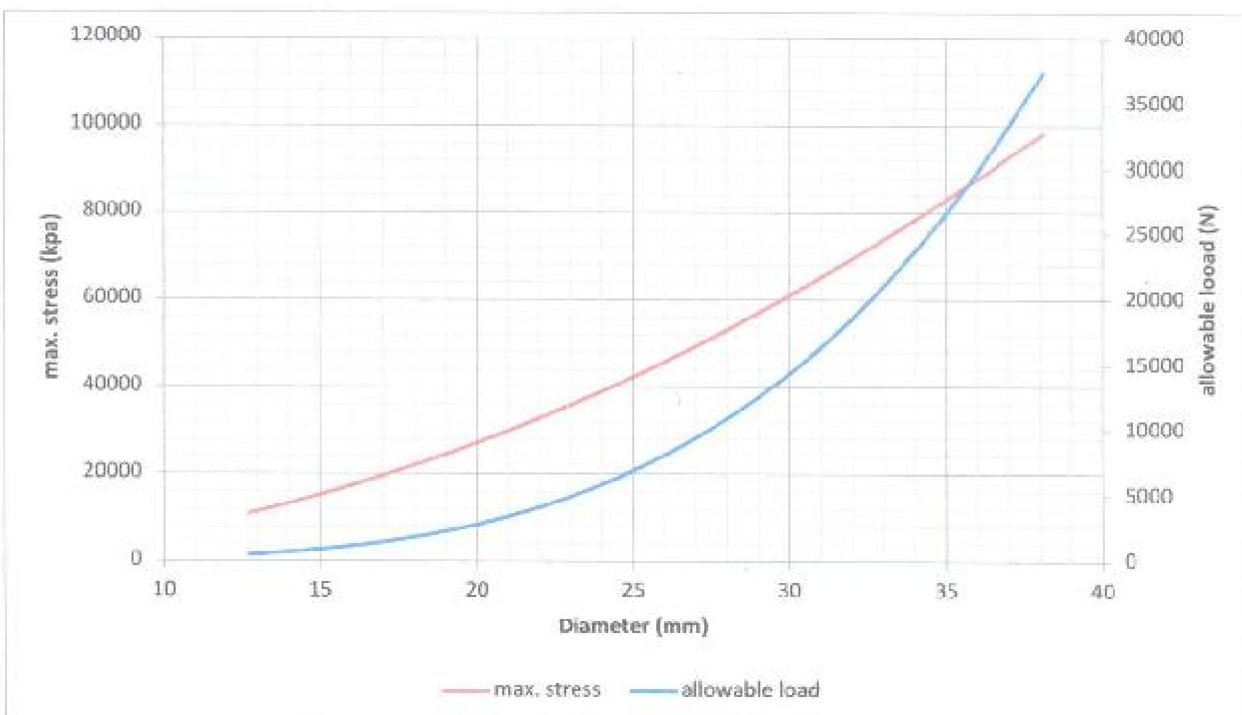
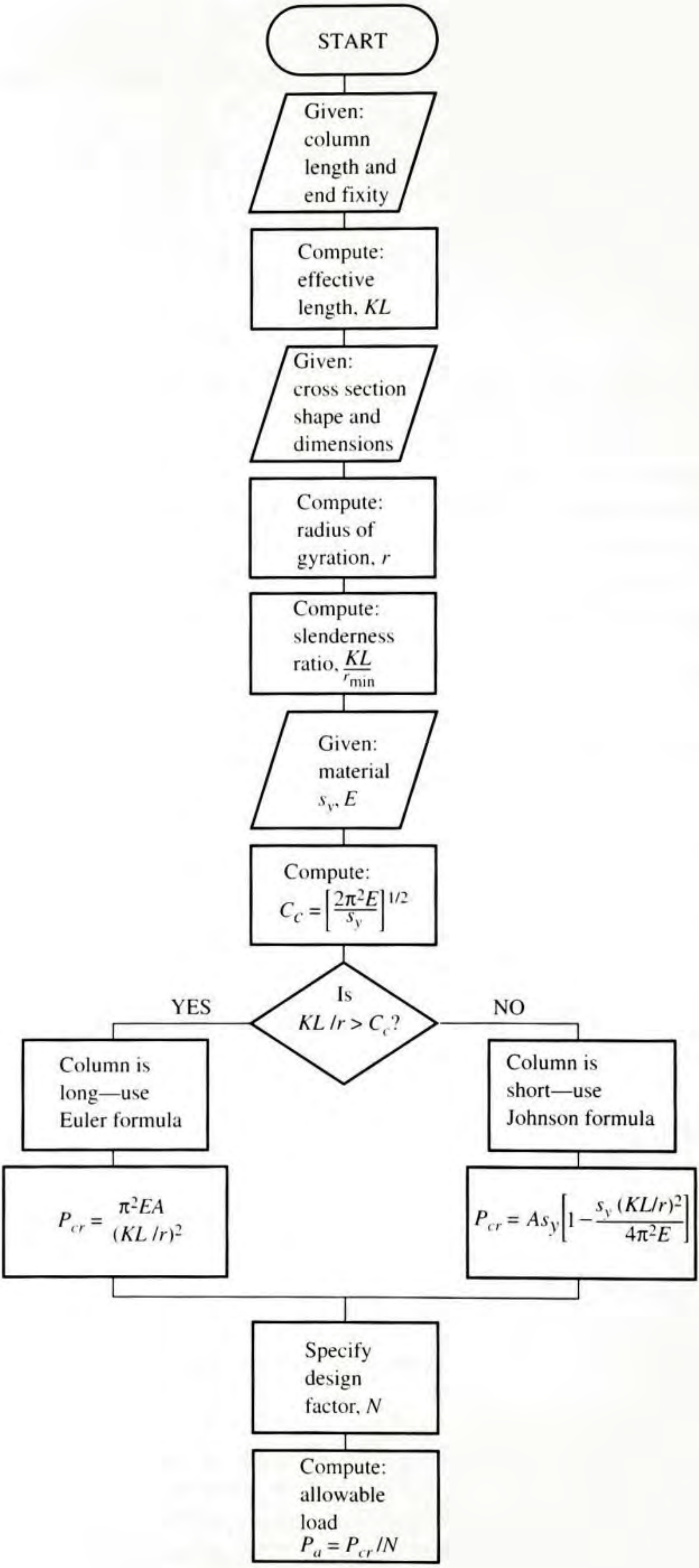


Fig. 2.


START
Given: column length and end fixity
Compute: effective length, KL
Given: cross section shape and dimensions
Compute: radius of gyration, r
Compute: slenderness ratio, $\frac{KL}{r_{min}}$
Given: material s_y, E
Compute: $C_c = \left[\frac{2\pi^2 E}{s_y}\right]^{1/2}$
Is $KL/r > C_c$?
YES
NO
Column is long—use Euler formula
$P_{cr} = \frac{\pi^2 EA}{(KL/r)^2}$
Column is short—use Johnson formula
$P_{cr} = As_y\left[1 - \frac{s_y (KL/r)^2}{4\pi^2 E}\right]$
Specify design factor, N
Compute: allowable load $P_a = P_{cr}/N$

University of Technology
Department of Machines and Equipment Engineering
First Term Examination 2012/2013

Subject: Design II
Division: General Mech.
Examiner(s): Group Design

Year: forth
Exam Time:1:30 Hrs.
Date: 22 / 1 / 2013

Answer (two) Questions Only

Q1: A link in a mechanism is 2000 mm long and has a circular cross section .It carries a compressive load of (2000 N) with an eccentricity of (20 mm). The following spread sheet of MDESIGN analysis of eccentric columns shown in table (1) by using different diameters of column. The maximum deflection (Y_{max}) should not exceeds (20mm). Find the exact suitable value of diameter then find the suitable stress and deflection for that value of diameter.

L(mm)	2000	2000	2000	2000	2000	2000
K	1	1	1	1	1	1
e (mm)	20	20	20	20	20	20
P(N)	2000	2000	2000	2000	2000	2000
S_Y(N/mm^2)	≈300	≈300	≈300	≈300	≈300	≈300
E (N/mm^2)	206843	206843	206843	206843	206843	206843
D(mm)	18	19	20	21	22	23
N	3	3	3	3	3	3
A (mm^2)	251.4	283.46	314	346.275	380.038	415.375
c (mm)	9	9.5	10	10.5	11	11.5
KL(mm)	2000	2000	2000	2000	2000	2000
r	4.5	4.75	5	5.25	5.5	5.75
KL/r	444.44	421.05	400	380.95	363.636	347.826
C_c	117.35	117.35	117.35	117.35	117.35	117.35
Max. stress σ(N/mm^2)	360	185	120	90	70	55
Max. deflection Y_{max}(mm)	80	40	25	18	13	10

Table(1)

Note: Do not make calculation .sketch different relationships and give your opinion.

$$Y_{max.} = e\ \{\sec(KL/r\ .\sqrt{P/AE})-1\}$$

$$S_Y = NP_a /A\ \{1+ec/r^2 \sec(KL/2r\ .\sqrt{NP_a/AE})\}$$

(50 marks)

Q2: ((Book Alternator))

Book came from the binding operation, passes through a wrapping machine, and proceeds to the packing section. Before the books can be properly packed for shipment, it is necessary that every other book be rotated through 180 degrees. Fig.(1) shows the books before and after rotation. 60 books a minute is the rate of production.

Space between books=(1.5) times book length.

Requirement:

Apply the system design flow-chart to do the following requirement s:

1- **Apply black-box concept to find ideas for region (x).** (10 marks)

2- **Apply Morphological –chart to find ideas for region (x).** (10 marks)

3- **Make a decision making to find the best solution.** (10 marks)

4- **Draw system scheme showing a section for all details of the complete construction.** (20 marks)

Note: To give you an idea about region (x) see figures (1),(2)and (3).

الشكل رقم (١) يمثل مخطط لحزام ناقل يقوم بنقل الكتب بعد تجليدها وتغليفها الى المنطقة (x) والتي تقوم بعملية قلب الكتاب ١٨٠ درجة وذلك للتأكد من جودة التغليف قبل رزمها للتسويق .ان المسافة بين كتاب واخر على الحزام الناقل =١,٥ من طول الكتاب .

ملاحظة:

لكي يتم اعطاء فكرة عن المنطقة (x)، ان جزء من احد الحلول الواجب عدم استخدامها عند حل السؤال ولكن للتوضيح فقط .فقد قام احد الاشخاص بوضع دولاب رقم (3) فيه عدد من الأخاديد على محيطه وعلى شكل شبه منحرف وكما موضح في الشكل رقم (2) ويدور هذا الدولاب عن طريق العمود (B).ان الدولاب يدور فترة ويتوقف فترة اخرى ليتيح الفرصة للكتاب بالدخول الى الحز المخصص له ويدور الكتاب مع الدولاب ويخرج الى الحزام الناقل من الجهة الاخرى وقد تم قلب الكتاب ١٨٠ درجة .

ان عملية دوران وتوقف الدولاب تتم عن طريق الالية الموضحة في شكل رقم (3) حيث ان الدولاب رقم 3 يتحرك عن طريق العمود (BB) المربوط على القرص رقم (2) الذي يتحرك فترة ويتوقف فترة عن طريق القرص رقم (1) والذي يدار عن طريق العمود (A) المربوط بالمحرك .ان تحدب الجزء (a) وتقعر الجزء (b)يجعل القرص (2) ثابت الى ان يتعشق السن في قرص (1) مع الفراغ في قرص (2) لكي يدور الدولاب (3) . هذا ويجب ان يكون هناك تزامن بين حركة الحزام الناقل وحركة وتوقف الدولاب رقم (3).

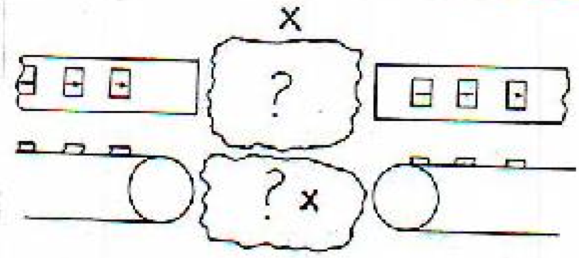


Fig (1)

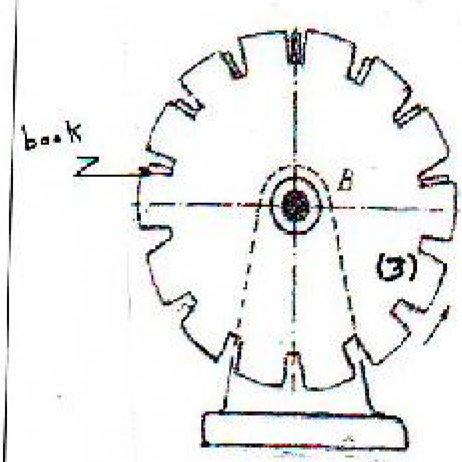
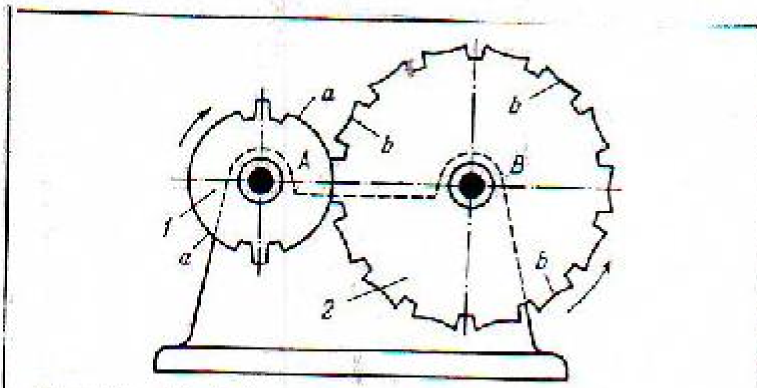


Fig (2)



Gears 1 and 2 rotate about fixed axes A and B. When driving gear 1 rotates continuously clockwise, driven gear 2 rotates intermittently with dwells. Concentric locking surfaces a of gear 1 engage concave surfaces b of gear 2 to avoid unintentional rotation of gear 2 during its dwell periods. Due to the symmetrical arrangement of the teeth and locking features, gear 2 has uniform intermittent motion when gear 1 rotates at uniform velocity. To each revolution of gear 1, gear 2 has two periods of motion and two dwells.

Fig (3)

Q3: **Sketch the system Design flow chart showing all details. Then discuss five various methods in system conception to generate the different ideas. You can use your own design project or any design projects, to give example on each method that you discussed.**

(50 marks)

University of Technology
Mechanical Engineering Department
Final Examination 2011/2012

Subject: Machine Design II
Branch: General Mech. Eng.
Examiner(s):

Glass: 4^{th} year
Time: 3 Hours
Date: 20/5/2012

Attempt Three questions only

Note: 1. **Make any change you seen advisable.**
2. Assume Missing data. **3. Open book exam.**

Q1: Fig. (1) Shows two steps spur gearing:

Data
i_1= (Reduction ratio for first step) = 1.5
i_2= (Reduction ratio for second step) = 3
Power transmitted through the gear box =4.5 kw
Rotational speed of pinion on input shaft =2000 rpm.
Center distance between two shafts = 100 mm.

Assume following data to save time

$\frac{b}{d_{b1}} \cong 0.25$, $F_e \approx 65\ \mu m$ $F_R \approx 20\ \mu m$ $F_{RW} \approx 12\ \mu m$

$Y_f \cong 1.4$ $q_{\varepsilon 1} \cong q_{\varepsilon 2} \cong 3.4$ $\mathcal{E} = \mathcal{E}_w \approx \mathcal{E}_n \approx 1.6$

$Y_\varepsilon \approx 0.7$ $\sigma_D = \sigma_o$ & $k_D \approx k_o$.

Requirements

1. Find all safety factors for pinion in second step reduction. (13 Marks)
2. Draw section x-x showing all details. (4 Marks)

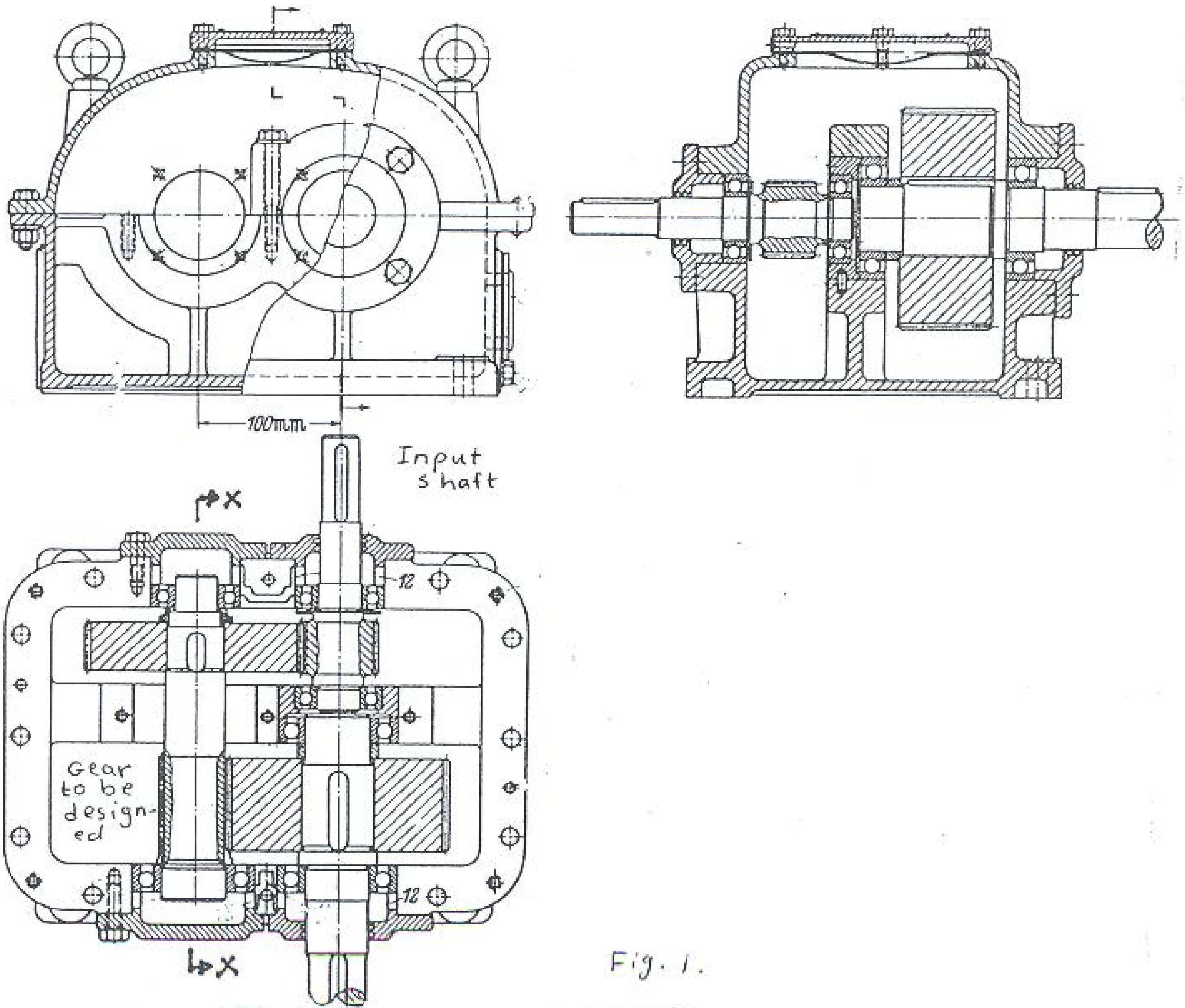


Fig. 1.

Q2: Fig(2) shows the reduction of overall dimensions of bevel gear drivers .

Data

Power transmitted by bevel gears = 10 kw.

Rotational speed of pinion = 1500 rpm.

Speed reduction =1.5

Assume following data to save time

Material of gears is (St. 60.11).

$q_{\varepsilon 1} \approx q_{\varepsilon 2} \approx 1$ $Y_{\omega 1} \approx Y_{\omega 2} \approx 3.11$

$C_S . C_D . C_T \approx 1.75$

Requirements

1. From the fig. you have five designs, you are asked to choose the best design for, good supporting of bevel gear, decreasing the size of gears, good stability, simplicity and ease of maintenance. (6 Marks)
2. Find factor of safety against breakage for pinion and wheel. (11 Marks)

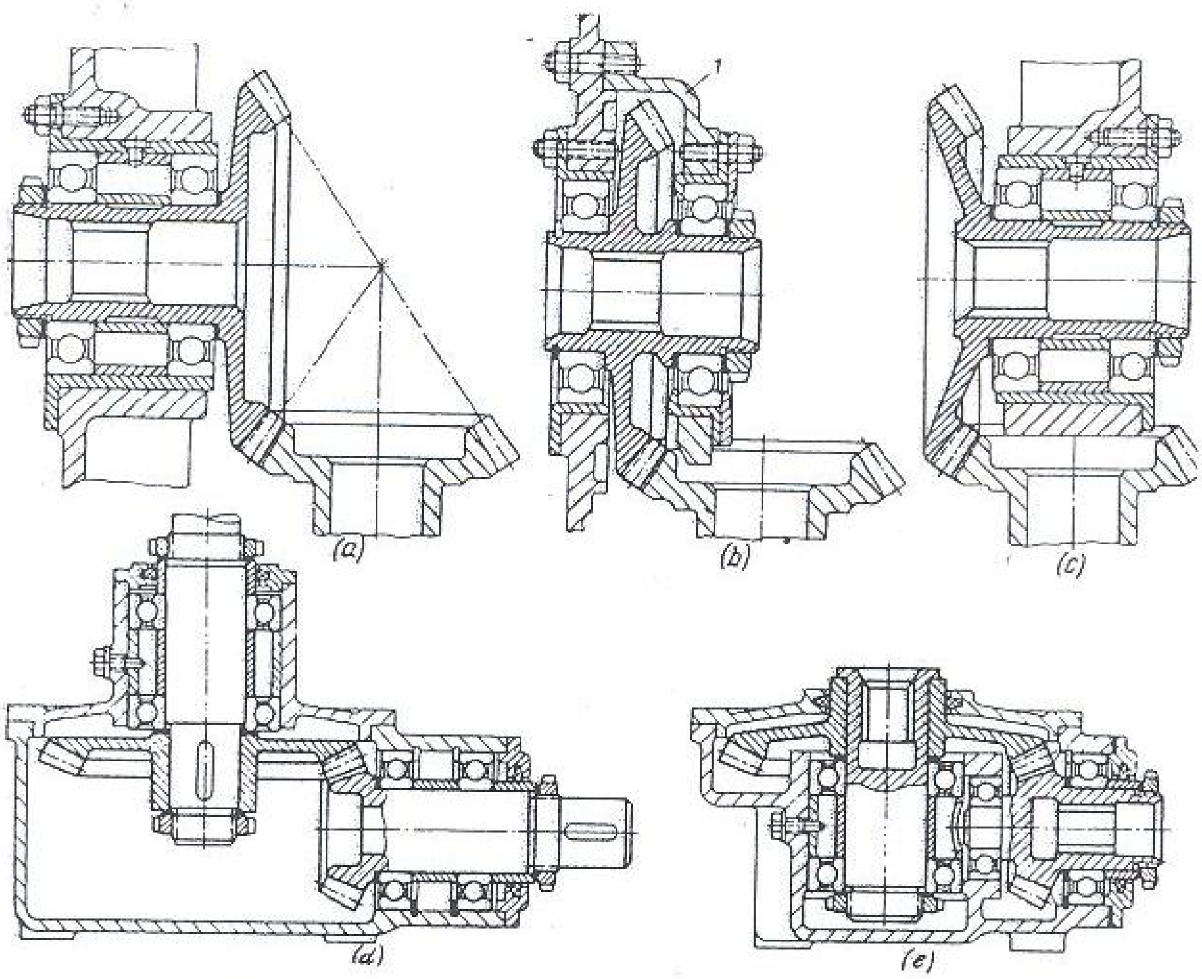


Fig 2 Reduction of overall dimensions of bevel gear drives

Q3: Fig (3) shows very old power drill. You are asked to improve the mechanism of raising and lowering the table only, which is raised and lowered by means of elevating screw.

Data : Power transmitted through flat belt = 10 kw.

Rotational speed of motor that drive the lower pulley = 1500 rpm.

Diameters of pulleys are equals 140, 172, 206 and 238 mm.

The reduction ratio may be one of these values ($\frac{238}{140}$, $\frac{206}{172}$, $\frac{172}{206}$, $\frac{140}{238}$)

The center distance between two pulleys = 2000 mm.

Type of belt is leather flexible.

Requirements

1. Find width of flat belt (b) only. (8 Marks)

2. a) Draw sectional view showing how the elevating screw will raise and lower the table.

   b) Sketch (without description) two other different ideas than you draw in branch (a) above showing all important parts. (6 Marks)

   c) Make a decision making to choose the best one of three ideas you draw in branch (a) and (b) above. (3 Marks)

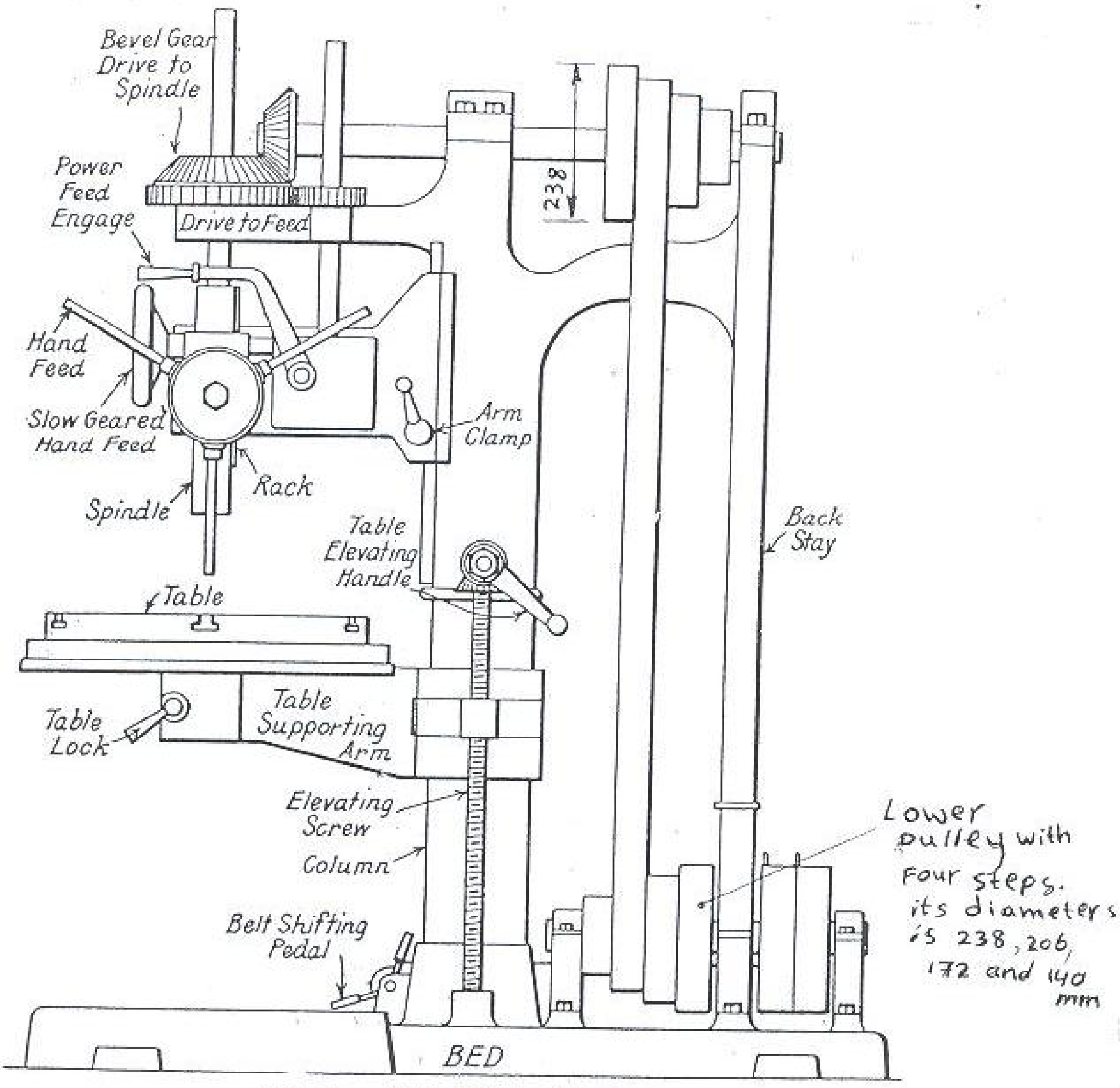


Fig. 3. THE POWER DRILL

Q4:Fig(4)shows a design tree for wheel chair for disabled person done by a student in previous years.

Use this design tree (you can add or reject any idea you seen advisable),to make the following requirements:

1.Draw a system scheme showing all parts as you think that is the best solution for the problem. (11 Marks)

2.Choose only two of the following requirements:

a. Write four items from problem specification which you depends on selecting the best scheme. (3 Marks)

b. Show how you can apply inversion (on system conception) for one idea on sub-systems by drawing in details without description. (3 Marks)

c. Write four points from feasibility study which depends only on the scheme you select. (3 Marks)

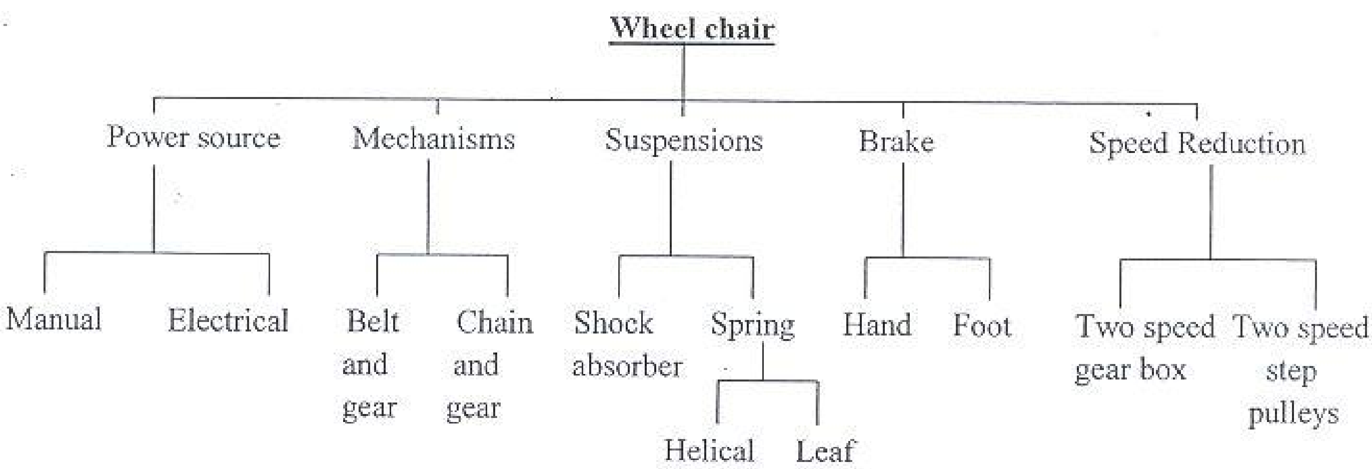


Fig. 4.

Note: you can add sub-systems such as frames, control ....etc. then you can add branches for sub-system if you seen advisable.

٢٠١٢/٥/١٧

الجامعة التكنولوجية	العام الدراسي : 2007-2008	الملاحظات
قسم هندسة المكائن والمعدات	أمتحان الدور الأول	1- يسمح بإستخدام الكتب والمحاضرات
الصف : الرابع عام	4 \6\2008	2- تمنع الاعارة لطفا
المادة : تصميم مكائن II	الزمن ثلاث ساعات	3- افرض القيم التي تراها مناسبة

س1 :

الشكل رقم (1) يمثل جزء من منظومة تخفيض السرع
الحركة تنتقل من المحرك الى العمود (A) ومنه الى المسننات (1) و (2) و (3)
أن المسنن (3) و المسنن (4) هما قطعة واحدة و تنتقل الحركة الى المسنن (5) و من ثم الى مسنن (6) و (7) .
أن المسنن (8) و المسنن (9) هما ايضا مصنوعان من قطعة واحدة و عندما يتحركان الى اليمين يتعشق مسنن (6) مع مسنن (8) و عندما يتحركان الى اليسار يتعشق مسنن (7) مع مسنن (9).
وبعدها تنتقل الحركة الى العجلة المسننة الكبيرة رقم (10) و بسرعتين مختلفتين حسب تعشيق مسنن (8) أو مسنن (9) و منها الى العجلة المسننة الصغيرة رقم (11) ومنها الى البكرة الكبيرة رقم (12) ثم الى البكرة الصغيرة التي لم تظهر بالرسم .

المعلومات

عزم الألتواء على العمود (A) = 1 Kgf. m
السرعة الدورانية على العمود (A) = 2900 rpm

$Z_1 = 7 \quad Z_2 = 42 \quad Z_3 = 56 \quad Z_4 = 10 \quad Z_5 = 36 \quad Z_6 = 24$
$Z_7 = 10 \quad Z_8 = 24 \quad Z_9 = 40 \quad Z_{10} = 40 \quad Z_{11} = 25$

المسافة المركزية بين المسنن رقم (1) و المسنن رقم (2) = 123.5 mm
مقدار التصحيح للمسنن رقم (1) = $X_1 = +0.6$
مقدار التصحيح للمسنن رقم (3) = $X_3 = -0.3$

للمسنن رقم (7) أفرض أن $b / d_{b1} \approx 0.5$
للمسنن رقم (7) أفرض أن $B_W \approx B_{all}$
للمسنن رقم (7) أفرض أن $y_\varepsilon \approx 1$
للمسنن رقم (7) أفرض أن $X_7 = 0$
أفرض أن معدن المسنن رقم (7) ((bath - nitrided steel)) 42 Cr Mo 4
المسافة المركزية للسلسلة = 235 mm
المسافة المركزية للأحزمة = 350 mm
نسبة التخفيض للبكرات = 4
للأحزمة أفرض أن $C_1 = C_2 = C_3 = C_4 = C_5 = C_6 = 1 \qquad C_7 = 1.25$

المطلوب :

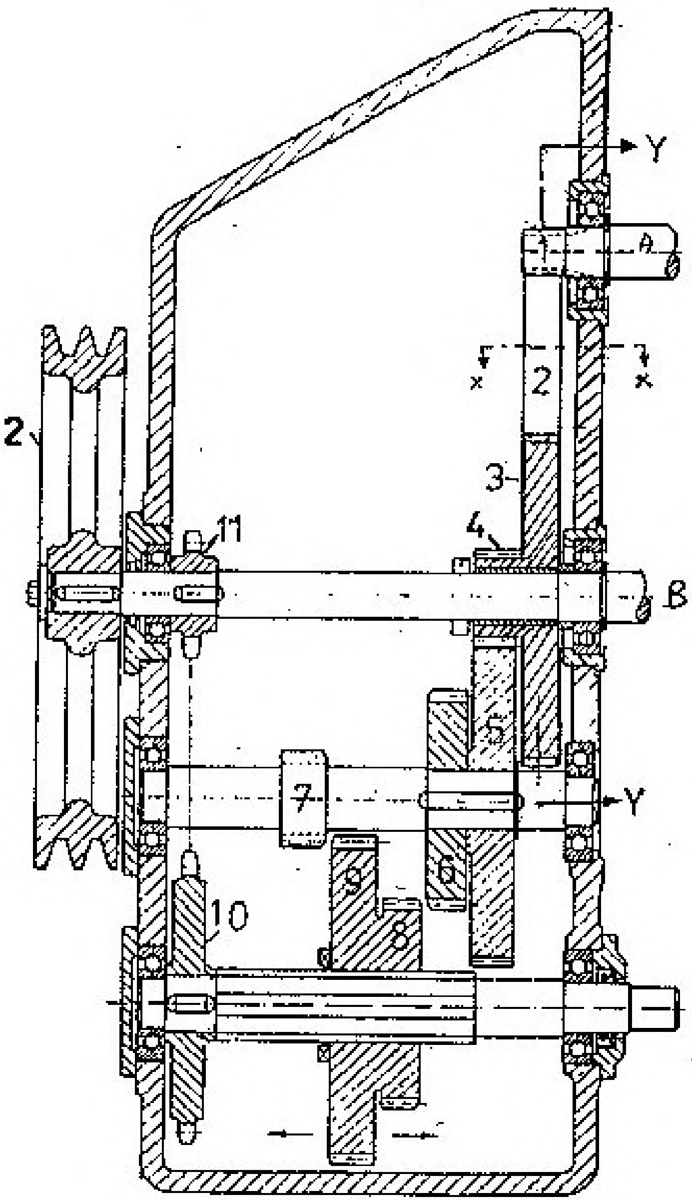
1- ايجاد مقدار التصحيح (X_2) و المسافة الطرفية (Addendum) للمسنن رقم (2) h_{K2}
والمسافة المركزية (a) , (centre distance) بين المسنن (2) و المسنن رقم (3)
(12 درجة)

2-1) ما مقدار السرعة الدورانية و عزم الالتواء على العمود (B) (3 درجة)
ب) ارسم المقطع (X---X) موضحا بالتفصيل كيفية تثبيت المسنن الوسيط رقم (2) على العمود (ليس موضح بالرسم) و كيفية تثبيت هذا العمود في صندوق التروس (6 درجات)
ج) ارسم بالتفصيل المقطع (Y --- Y) لتوضيح كيفية تعشيق المسننات الثلاثة (1 , 2 , 3)
(3 درجات)

3- احسب عمر المسنن رقم (7) بالساعات من حيث تنقر السن فقط
Full load life in case of pitting for gear no. 7 (12 درجة)

4- إذا علمت إن على السلسلة يوجد أكثر من صف أوجد عدد الصفوف (j) number of strands
واحسب عامل الأمان للسلسلة (12 درجة)

5- إذا علمت إن على البكرة رقم (12) يوجد أكثر من حزامين , أوجد عدد هذه الأحزمة و أوجد اكبر إجهاد يتعرض له الحزام الواحد بعد اختيارك إبعاد الحزام المناسب (12 درجة)



الشكل رقم (1)

اذكر رايك بالنتائج

س2:

شركة لصناعة لعب الاطفال لديها المشكلة التالية

في النية تصميم لعبة اطفال على شكل قارب يتحرك في حوض ماء لا يزيد عرضه عن 80 سم و طوله لا يزيد عن 80 سم . و اعطاء متعة للطفل من خلال حركة القارب و حركة الشخص الموجود في القارب لتعطي صورة مشابهة للحالة الحقيقية .

المطلوب تصميم هذا القارب بابعاد مناسبة باستخدام محرك كهربائي يدار باستخدام بطاريتين صغيرتين . هناك اعتبارات تصميمية متعددة من الممكن اخذها بنظر الاعتبار مثل الاداء , الكلفة , سهولة التصميم , حجم المكان و غيرها .

استخدم موضوع تصميم المنظومات (System Design) لحل المشكلة و طبق ما يلي :

ملاحظة : ((اجب عن فرعين على ان يكون الفرع الثاني من ضمنها))

1- اكتب المواصفات الابتدائية (problem specification or initial specification) لهذه المنظومة مع تحديد قيمة هذه الصفات حسب وجهة نظرك .

ملاحظة : ((اذكر اربع صفات مهمة فقط على ان لا تتجاوز سطرين لكل صفة)) (6 درجات)

2- في موضوع مفهوم النظام ((system conception)) هناك طرق مختلفة لايجاد افكار مختلفة . احد هذه الطرق ((Morphological chart)) و عند تطبيق هذه الطريقة تم تقسيم النظام الى انظمة فرعية و تم ايجاد احد الافكار و حسب رأي احد منتسبي الشركة و كما موضح في جدول رقم (1) .

المطلوب رسم فقط و بدون شرح مختصر او مفصل افكار توضح تفاصيل و تثبيتات لافكار مناسبة مثلا (D_2 , C_2 , B_2) تكون بديلة عن الافكار (D_1 , C_1 , B_1) ومن الممكن تغيير او حذف او اضافة انظمة فرعية او افكارها و ان جدول رقم (1) هو تقريب لتصور الفكرة الأساسية فقط .

(8 درجات)

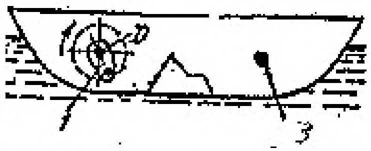
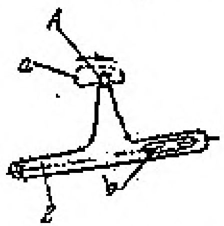
3- شكل رقم (2) يوضح ((system scheme)) لاحد الحلول و التي تعطي فكرة واضحة للحركة حيث ان القارب يتحرك باستخدام المجذاف المثبت بيد الشخص المجذف و تاتي الحركة عن طريق محرك يقوم بتدوير عمود المرفق رقم (1) و الذي بدوره يحرك ذراع التوصيل رقم (2) الذي يحتوي الشق (b) الذي ينزلق على المسمار رقم (3) المثبت بالقارب .

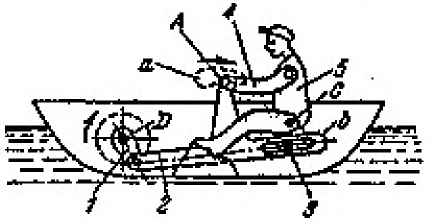
النقطة (A) في ذراع التوصيل رقم (2) تعمل المنحنى (a) لتحرك ذراع الشخص رقم (4) و التي تحمل المجذاف (oars) و كذلك تحرك جسم الشخص المُجذف (oarsman) رقم (5) الذي يتذبذب حول المحور (c) ليمنح الحركة المطلوبة .

المطلوب رسم بالتفصيل (system scheme) للفكرة الجديدة و التي قد تتبع المسار (D_2 , C_2 , B_2 , A_1) او اي مسار تراه مناسبا موضحا كافة توصيلات و تثبيتات الاجزاء ببعضها و من الممكن رسم اكثر من شكل لتوضيح الافكار المختلفة (6 درجات)

God with you

جدول رقم (1) يمثل تقسيم النظام الى انظمة فرعية وايجاد افكار بديلة

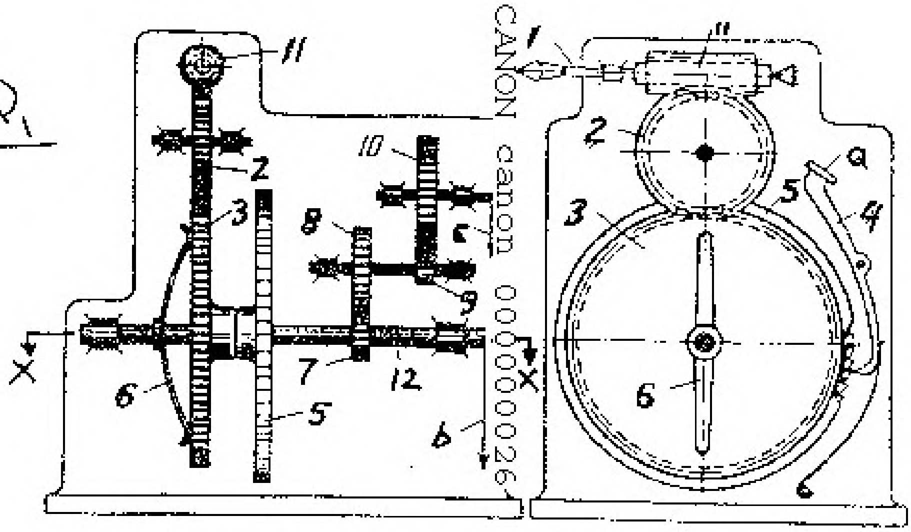
Alternatives → البدائل / ↓ الانظمة الفرعية Sub- systems	I	2
A مصدر الطاقة	A1 استخدم محرك كهربائي ذو بطاريتن (وحسب ما معطى بالسؤال)	----------
B شكل القارب والاجزاء المثبتة عليه لتساعد على الحصول على الحركة المطلوبة	B1 	B2
C الية توصيل الحركة بين مصدر الطاقة والشخص	C1 	C2
D حركة الشخص المُجذ ف لتعطي متعة للطفل وتحقيق حركة القارب	D1 	D2



Crank *1* rotates about axis *D* fixed in the boat. When crank *1* rotates, connecting rod *2*, having slot *b*, slides along pin *3* fixed in the boat. At this, point *A* of connecting rod *2* describes connecting-rod curve *a* as a result of which arms *4* holding the oars and body *5* of the oarsman, oscillating about axis *C*, are imparted the required motions.

الشكل رقم (2)

Q1

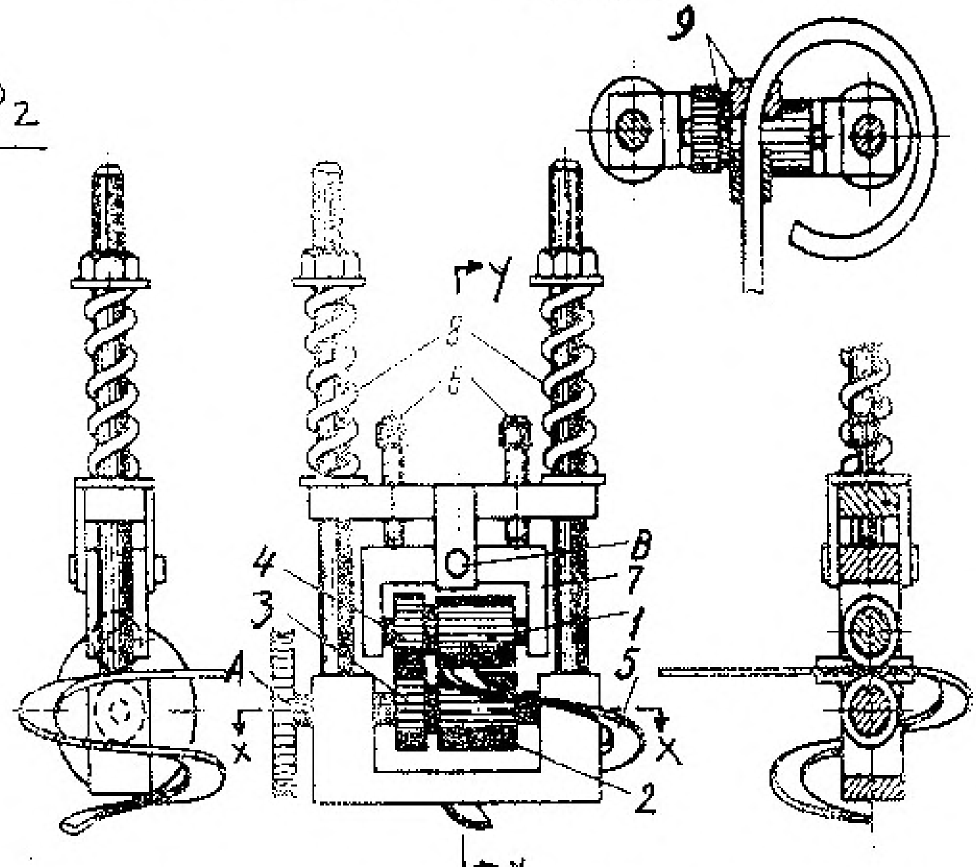


CANON canon 00000026

When spindle *1* is connected to the shaft whose revolutions are to be counted, worm *11*, worm wheel *2* and gear *8* begin to rotate. When button *a* is pressed, pawl *4* is disengaged from ratchet wheel *5* which is keyed to a shaft with flat spring *6* and begins to rotate due to friction between the spring and gear *8*. This leads to rotation of gears *7*, *8*, *9* and *10* which transmit rotation to hands *b* and *c*. The numbers of teeth of the gears are selected so that hand *b* makes 10 revolutions to 1000 revolutions of spindle *1*, and hand *c* makes only one revolution.

شكل رقم (١) تابع للسؤال الأول

Q2



Rolls *1* and *2*, whose axes make a small angle, are driven from shaft *A* through meshing gears *3* and *4*. The strip of stock *5* is subject to varying pressure as it passes through the rolls. The left edge is subject to more strain than the right edge. This forms the straight strip into a helical ribbon. The angle between the rolls is adjusted by screws *6* which turn yoke *7* about fixed axis *B*. Pressure is applied to the top roll by springs *8* whose tension can be varied by nuts. The diameter of the helix is maintained constant by guide *9*.

شكل رقم (2) تابع للسؤال الثاني

الجامعة التكنولوجية

قسم هندسة المكائن و المعدات
امتحان الدور الأول 2005-2006

CANON canon 000000000

المرحلة : الربع مكائن عام
المادة : تصميم مكائن II
التاريخ : 2006\6\12
الزمن : ثلاث ساعات

الملاحظات 1- يسمح باستخدام الكتب والمحاضرات
2- افرض القيم التي تحتاجها
3- اذكر وفيك بالنتائج
4- تمنع الاعارة لطفا

ملاحظة : اجب عن ثلاثة اسئلة فقط

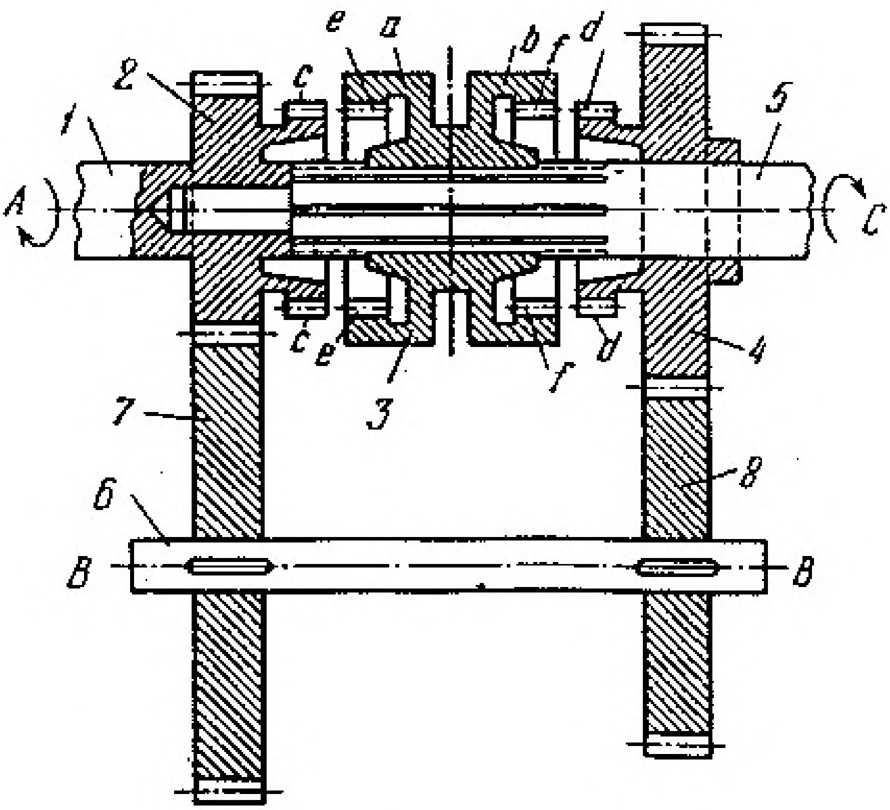
س1- الالية في شكل رقم (1) تمثل مسننات لتغير السرع و بسرعتين مختلفتين . ان الحركة تنتقل عن طريق المسنن رقم (2) الذي هو جزء من العمود رقم (1) الى العمود رقم (5) عن طريق القابض المسنن رقم (3) المتعشق مع العمود رقم (5) عن طريق التحززات (splines) بين العمود والقابض .
ان المسنن رقم (4) يدور بحرية حول العمود رقم (5) . وان المسننين (2) و (4) متعشقان بشكل مستمر مع المسننين (7) و (8) . ان السرعة الاولى تتحقق عن عن طريق حركة القابض (3) الى اليسار حيث يتعشق المسنن (c) مع المسنن (c) وعند تحقيق السرعة الثانية يتحرك القابض رقم (3) الى اليمين .

المعلومات

قطر دائرة التدحرج للمسنن رقم (2) = 68 mm
قطر دائرة التدحرج للمسنن رقم (7) = 122 mm
قطر دائرة التدحرج للمسنن رقم (4) = قطر دائرة التدحرج للمسنن رقم (8) = 95 mm
نوع التروس حلزونية (Helical Gears) $\beta_o = 22^o$
مقدار التضمين Mn للتروس الحلزونية = 3 mm

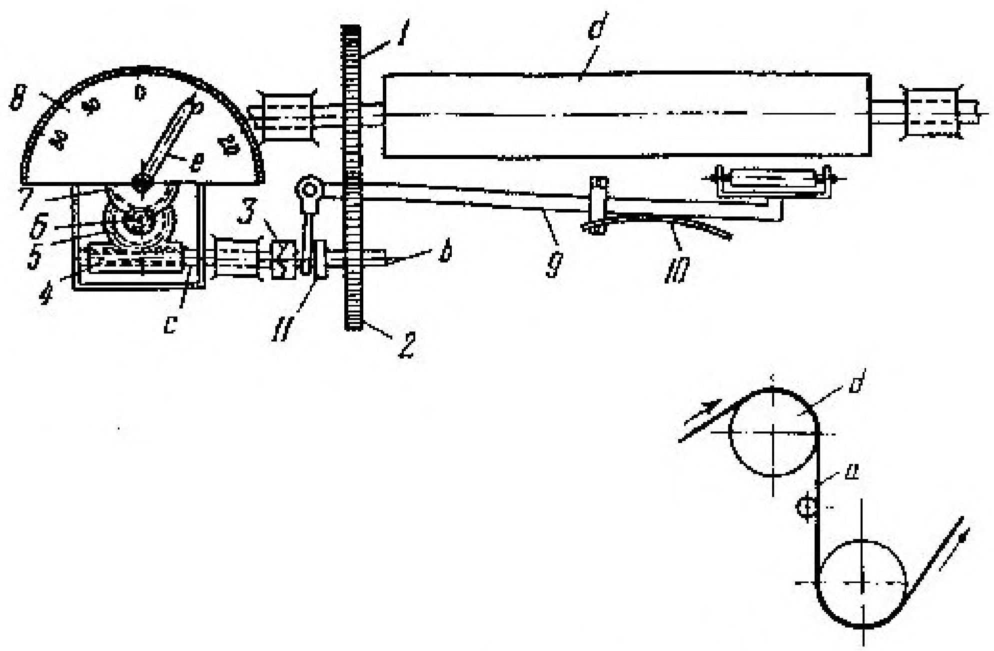
المطلوب

1- اوجد مقدار hk (addendum) للترس رقم (4) و (8) (8) درجات
2- اوجد العامل y_ε (Gear Geometry factor) للترسين (4) و (8) (5) درجات
3- ضع ركيزتين متدحرجتين مناسبتين بين الترسين رقم (4) ورقم (5) موضحا كيفية تثبيت الاطواق الداخلية و الخارجية للركيزتين
ملاحظة :لا ترسم شكل رقم (1) بالكامل و لكن ارسم فقط الترس رقم (4) والعمود رقم (5) والركائز مع كافة تفاصيل تثبيتهما (4) درجات



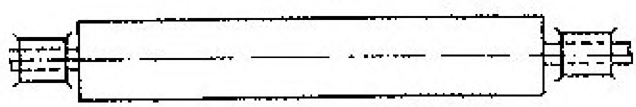
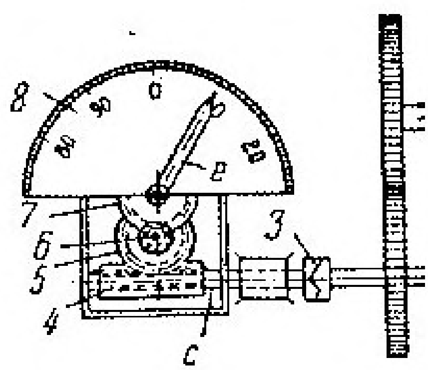
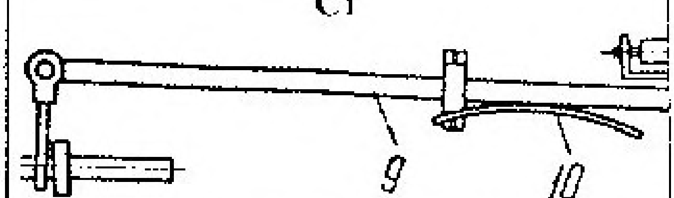
شكل رقم (1)

يتبع لطفاً



شكل رقم (2)

الجدول رقم (1) أدناه يمثل تقسيم مبسط لاجزاء النظام وهذا الجدول يسمى (morphological chart) .

(Alternatives) البدائل → / **Sub Systems** ↓	1	2
A الاليات التي تمر عليها شريحة الورق	**A1** 	——
B اليات نقل الحركة الى مؤشر القياس	 **B1**	**B2**
C آليات السيطرة في حالة حدوث تمزق لشريحة الورق	**C1** 	**C2**

يتبع لطفاً

الجامعة التكنولوجية
قسم هندسة المكائن والمعدات
الصف الرابع مكائن عام
تصميم المكائن

امتحان الفصل الأول
2006/5/
الزمن ساعتان

ملاحظات: 1- افرض القيم المناسبة
2-يسمح باستخدام الكتب و المحاضرات
3-اعمل اي تغيير تراه مناسبا
4- تمنع الاعارة رجاءا

CANON 0000000042

ملاحظة :اجب عن سؤالين **فقط**

السؤال الاول

الشكل رقم (1) يمثل الية لحساب عدد الدورات لعمود يُربط على العمود رقم (1) من خلال هذا الربط تنتقل الحركة الى التروس الدودية رقم (11) ورقم (2) وبعدها ينتقل الدوران الى الترس رقم (3). ان الحركة الدورانية لا تنتقل الى العمود رقم (12) الا بعد ان يتم الضغط على الذراع (a) بحيث تنفصل القطعة رقم (4) عن القرص المحزز رقم (5) والمتعشق مع العمود رقم (12) وعن طريق النابض الورقي رقم (6) . وبالاحتكاك الموجود بين النابض و الترس المستقيم رقم (3) تنتقل الحركة الى العمود (12) و هذا يؤدي الى دوران التروس (7)،(8)،(9)،(10) وبعدها ينتقل الدوران الى المؤشر (b) ، (c) . ان عدد الأسنان للتروس يجب ان يتم اختيارها بحيث المؤشر (b) يكون (10) دورات لكل (1000) دورة للمحور رقم (1) و المؤشر (c) يدور دورة واحدة .

المعلومات :

التضمين لكل التروس = m = 3 mm

زاوية ميلان الترس الدودي (Lead Angle θ) = 19.62°

المسافة المركزية بين الترسين المستقيمين (7) ، (8) = 120 mm

المطلوب

1- ايجاد الأقطار و عدد الأسنان للتروس 10,9,8,7,3,2,11 (23 درجة)
2- لماذا تم وضع مؤشرين لقياس عدد الدورات فسر ذلك باختصار (2 درجة)
3- ارسم بالتفصيل المقطع (x-x) موضحا طريقة نقل الحركة (25 درجة)

السؤال الثاني

الدرفيلن (1) و (2) يعملان زاوية صغيرة مع محوريهما .ان الدرفيلين يداران عن طريق العمود (A) من خلال تعشيق الترسين رقم (3) و (4) .ان الشريط المعدني رقم (5) يتعرض الى ضغط متغير عندما يمر من خلال الدرفيلين .ان الجهة اليمنى تتعرض الى ضغط يختلف عن الجهة اليسرى لكي يعطي للشريط الشكل الحلزوني الموضح بالرسم .ان الزاوية بين الدرفيلين من الممكن التحكم بها عن طريق اللوالب رقم (6) والتي تقوم بتحريك المفصل رقم (7) حول المحور (B) .ان الضغط على الدرفيل العلوي يُسلط عن طريق النابض رقم (8). وان الصامولتين اعلى النابض تتحكم بمقدار ضغط النابض .ان قطر الشكل الحلزوني للشريط يتكون عن طريق الدليل رقم (9).

المعلومات

السرعة الدورانية للعمود (A) = 100 rpm
التضمين للتروس رقم (3) ورقم (4) = mn = 4 mm
زاوية الحلزون β = 14°
مقدار التصحيح للتروس رقم (3) ورقم (4) = x1 = x2 = 0.5
مقدار شدة الحمل الفعالة Bw = Ball= 0.5 Kgf/ mm²
عرض المسنن رقم (3) و (4) = b = 25 mm

المطلوب

1- ايجاد عامل الأمان ضد كسر السن للترس رقم (3) فقط (20 درجة)
2- ارسم المقطع (X-X) موضحا كافة التفاصيل (25 درجة)
3- ارسم المقطع (Y-Y) موضحا كافة التفاصيل (5 درجات)

السؤال الثالث

ان الحركة تنتقل من العمود رقم (1) و عن طريق التروس الدودية رقم (2) و (3) تنتقل الحركة الى العمود رقم (13) و بالتالي سوف يتم لف النابض الحلزوني الموجود داخل الأسطوانة رقم (12). ان احدى نهايات النابض مثبتة على العمود رقم (13) والنهاية الأخرى مثبتة على جسم الأسطوانة رقم (12). ان الطاقة المخزونة في النابض تنقل خلال الترس رقم (4) و الترس (5) و الترس (6) و الترس (7) و من ثم الى العمود المساق رقم (8). ان سرعة العمود رقم (8) سيطر عليها باستخدام منظم سريع رقم (9) و الذي يدار من خلال القرص الدودي (10) و (11).

المعلومات

التخمين m = 4 mm

قطر الدودة (worm) = 50 mm

قطر القرص الدودي (worm wheel) = 200 mm

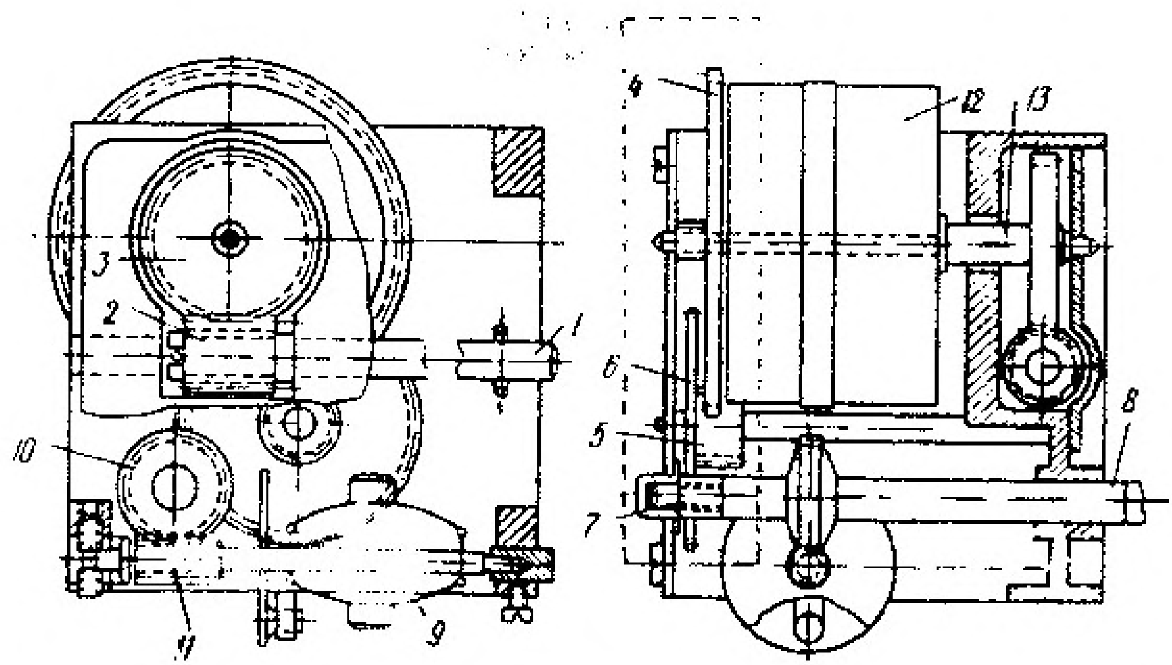
نسبة التخفيض للتروس الدودية = i = 12.5

السرعة الدورانية للعمود رقم (1) 500 rpm

المطلوب

1- اكبر قدرة ممكن ان ينقلها الترس الدودي رقم (2) (25 درجة)

2- ارسم بالتفصيل المنطقة المنقطة لتوضيح مبدا عمل المنظومة (25 درجة)



Rotation is transmitted from shaft *1* through worm *2* and worm wheel *3* to shaft *13*. This winds up a flat spiral power spring enclosed in casing *12*. One end of the spring is secured to shaft *13* and the other to casing *12*. The energy stored in the spiral spring is transmitted through gear *4*, rigidly attached to casing *12*, and gears *5*, *6* and *7* to driven shaft *8*. The speed of shaft *8* is controlled by spring-type centrifugal governor *9* which is driven through worm wheel *10* and worm *11*.

3

الجامعة التكنولوجية - قسم هندسة المكائن والمعدات - امتحان تصميم آلات
الصف الرابع عام   الزمن: ساعة ونصف.

CANON canon 00000021

المخطط يمثل صندوق تروس ذو مرحلتين. عندما يتحرك الترس رقم (6) الى اليمين نحصل على السرعة الاولى وعندما يتحرك الترس رقم (5) الى اليسار نحصل على السرعة الثانية عن طريق الدفاع الموجود بين الترسين (5) و (6).
ان الترسين (5) و (6) يكونان مربوطان معاً او منفصلين يتحركان على المقطع المربع (a) على العمود (1).

المعلومات

المسافة المركزية بين الترسين (5)و(3) او (6)و(4) = $300 mm$

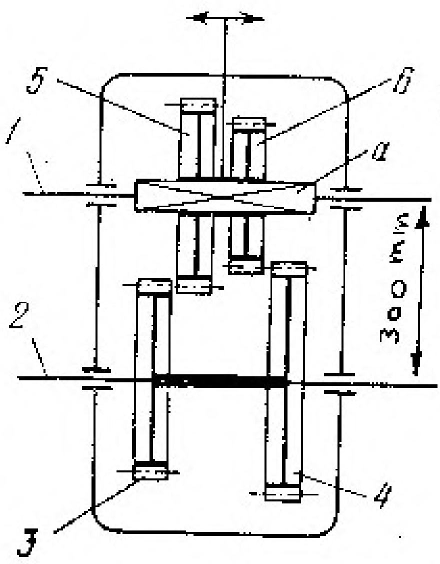
نسبة التخفيض بين الترسين (5)و(3) = $i_1 = 1$

" " " (6)و(4) = $i_2 = \frac{13}{7}$

السرعة الدورانية للعمود رقم (1) = $n_1 = 600\ rpm$

القدرة المنقولة عن طريق العمود رقم (1) = $N_1 = 4\ hp$

المطلوب



١- الابعاد الاساسية للترسين (6) و (4) بشرط ان Z_6 اقل من 50 سن
٢- عامل الامان ضد تقعر سطح الترس رقم (4) فقط.
٣- ارسم مقطع متكامل مع كافة التفاصيل لصندوق التروس.

افرض ما يلي لتسهيل الحل.

$B_{all} \simeq B_w$

$\varepsilon = \varepsilon_n \simeq \varepsilon_w$

$b/d_{b1} \simeq 0.2$

س٤ : شكل رقم (4) في السؤال الثالث يمثل «system scheme» لآلية مقص القطعة رقم (5) باستخدام المنشار الدائري رقم (4).

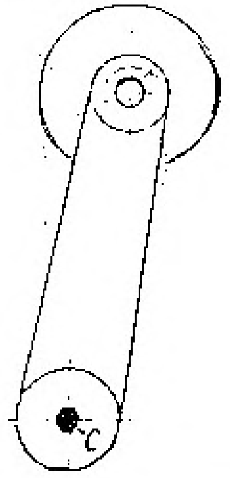
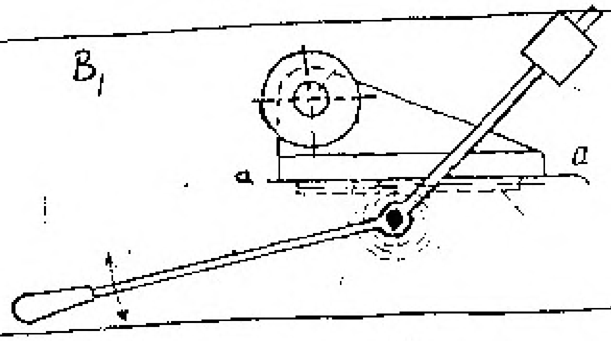
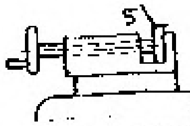
الجدول رقم (١) ادناه يمثل «morphological chart» لهذا النظام بعد تقسيمه الى انظمة فرعية «sub-systems».

المطلوب

١- ارسم فقط وبدون شرح مختصر او منفصل مقطع يوضح التفاصيل والتثبيتات لفكرة مناسبة لـ A_2 تكون بديلة ومختلفة عن الفكرة A_1. وكذلك ارسم افكار بديلة لـ B_1 و C_1 كما عملت بأستبدال A_1. ———— «4 درجات»

٢- ارسم «system scheme» الذي يمثل التصميم الجديد والذي يتبع المسار (A_2 ، B_2 ، C_2) راسماً مقطع يوضح كافة التثبيتات لتعطي صورة واضحة عن الفكرة الجديدة التي تمت بتصميمها. ———— «8 درجات»

٣- طبعه طريقة شجرة التصميم «Design tree» للنظام. ---- «3 درجات»

Alternatives البدائل → / ↓ Sub-systems	1	2
A آلية نقل الحركة الدورانية من المحرك الى المنشار	A_1 	A_2 chain
B آلية الحركة الخطية لتقريب المنشار على العينة المراد قصها	B_1 	B_2
C آلية مسك العينة	C_1 	C_2

جدول رقم (١) يمثل تقسيم النظام الى انظمة فرعية وايجاد افكار بديلة.

س٣: المنشار الدائري رقم (4) يقوم بقص القائمة رقم (5) المثبتة على الماسكة رقم (10) كما موضح بالشكل رقم (4). ان المنشار يدار عن طريق آلية من ضمنها البكرات رقم (8) ورقم (7) والحزام المسطح رقم (6). لتقريب المنشار على القطعة رقم (5) يتم حركة الذراع رقم (1) عكس عقرب الساعة عندما يدور الترس رقم (2) الذي يقوم بتحريك الجريدة المسننة رقم (3) المثبتة على القطعة رقم (9) قطعياً وعلى المسار (a-a) وعند حركة الذراع رقم (1) مع عقرب الساعة يبتعد قرص المنشار عن القطعة رقم (5).

CANON canon 00000022

المعلومات

قطر البكرة رقم (7) $d_7 = 180\ mm$

قطر البكرة رقم (8) $d_8 = 140\ mm$

المسافة المركزية بين البكرة (7) والبكرة (8) = 575 mm

السرعة الدورانية للبكرة (7) = 1500 rpm

قدرة المحرك الذي يدور البكرة (7) = 5 hp

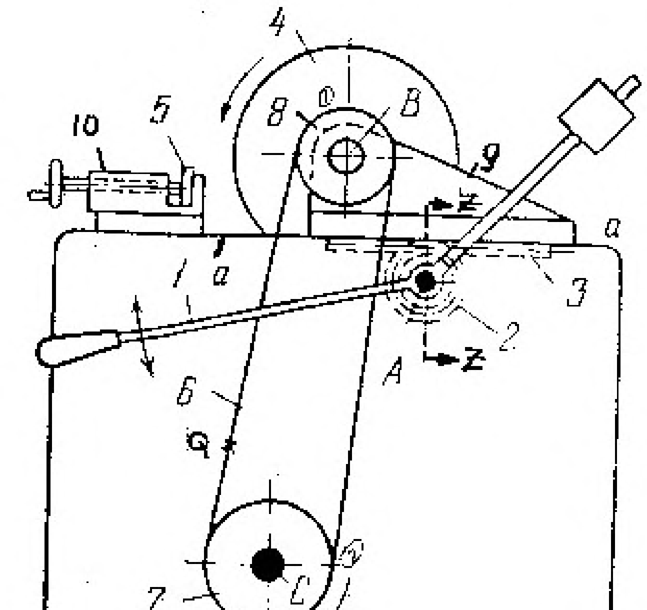
نوع الحزام المستخدم: « HG Leather belt ».

المطلوب

١- عرض الحزام المسطح ... (6 درجات)

٢- الاجهاد في النقطة Q ... (8 درجات)

٣- ارسم المقطع Z-Z ... (6 درجات)



شكل رقم -4-

س4: الشكل رقم (4) يمثل الية نقل الحركة باستخدام التروس المخروطية حيث ان المسنن رقم (1) هو نصف او جزء من مسنن مخروطي يدور حول المحور A. الترسين المخروطيين (2) و (3) متعشقان مع العمود رقم (4) . عندما يدور المسنن رقم (1) بشكل مستمر عكس عقرب الساعة يقوم هذا المسنن بتدوير المسننين (3) و (2) بالتعاقب وبذلك يتغير اتجاه حركة العمود رقم (4) من اتجاه عكس عقرب الساعة الى اتجاه مع عقرب الساعة بكل دورة من دورات المسنن رقم (1) .

<u>المعلومات :</u>

القدرة المنقولة عن طريق المسنن رقم (1) = 8 hp

السرعة الدورانية رقم (1) = 300 rpm

المعدن المستخدم لكافة التروس C 60

التضمين لكافة التروس 3 mm

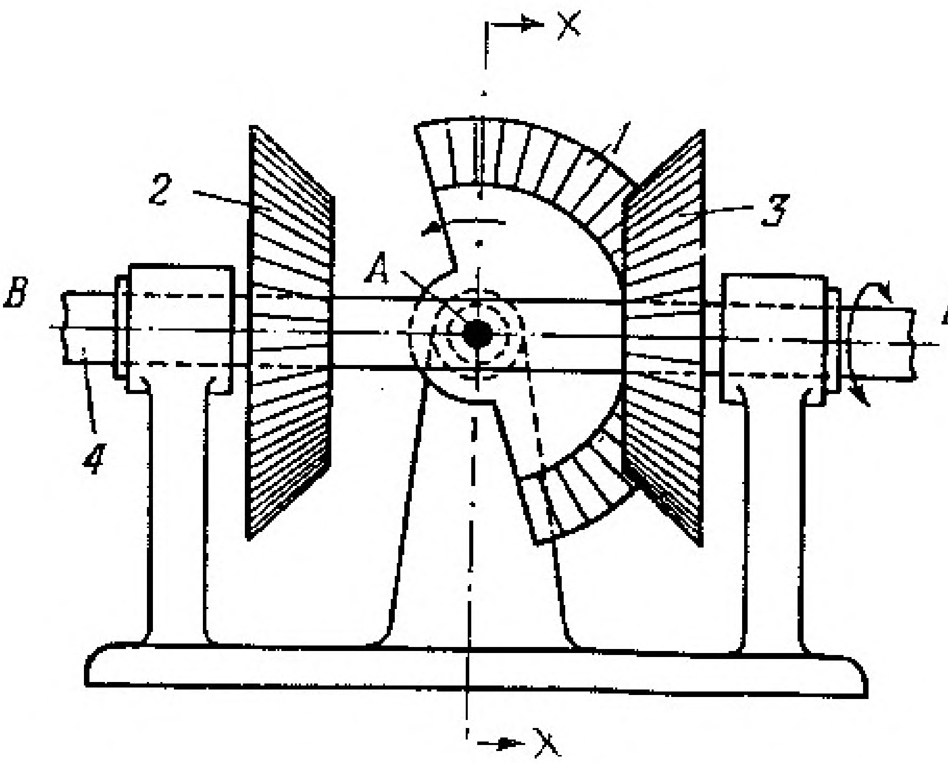
قطر دائرة الخطوة للمسنن رقم (1) = قطر دائرة الخطوة للمسنن رقم (2) = قطر دائرة الخطوة للمسنن رقم (3) = 90 mm

افرض ما يلي لتسهيل الحل $(Y_\varepsilon = 0.85) \cdot (1.25 = \varepsilon_w) \cdot (1.5 = \varepsilon_n) \cdot (1.75 = CS, CD, CT)$

<u>المطلوب :</u>

1- اوجد كافة عوامل الامن لكافة التروس (10 درجات)

2- ارسم المقطع X—X موضحا كافة التثبيتات وكافة التفاصيل . (7 درجات)



شكل رقم (4)

س2 : الشكل رقم (2) يمثل (system scheme) لآلية تقوم بقياس طول شريحة الورق التي تمر على الاسطوانة (d) . ان الحركة تنتقل من الاسطوانة (d) الى التروس المستقيمة (1) و (2) ويتم بعد ذلك دوران العمود (b) . ان الحركة تنتقل من العمود (b) الى العمود (c) عن طريق تعشيق القابض رقم (3) . وبعدها تنتقل الحركة الى التروس الدودية رقم (4) و (5) ومنها الى التروس المستقيمة (6) و (7) . ومن الترس رقم (7) تنتقل الحركة الى المؤشر (e) . ان آلية السيطرة بتوقيف عمل المؤشر (e) تتم عندما تتمزق شريحة الورق التي تمر على الاسطوانة (d) حيث يرتفع الذراع رقم (9) عند عملية التمزق بفعل القابض رقم (10) وبالتالي يتم فصل القابض رقم (3) وتنفصل حركة العمود (b) عن العمود (c) وبالتالي تتوقف عملية قياس طول الشريحة من الورق.

<u>المطلوب :</u>

1- ارسم <u>فقط</u> وبدون شرح مختصر او مفصل مقطع يوضح كافة التوصيل والتثبيتات لفكرة مناسبة في (B2) بديلة ومختلفة عن الفكرة (B1) وكذلك فكرة مناسبة في (C2) بديلة ومختلفة عن الفكرة (C1) , انظر الجدول رقم (1) . (8 درجات)

2- ارسم بالتفصيل (system scheme) الذي يمثل التصميم الجديد والذي يتبع المسار (C2*B2*A1) راسما مقطع يوضح كافة التثبيتات لتعطي صورة واضحة عن الفكرة الجديدة التي قمت بتصميمها (5 درجات)

3- طبق طريقة (Black –box concept) لنظام قياس طول الورق (2 درجة)

4- طبق طريقة (Bridgining and Terminal trees) لنظام قياس طول الورق (2 درجة)

س3 : الشكل رقم (3) يمثل آلية نقل الحركة عن طريق الاحزمة .
ان البكرة رقم (1) تدور حول العمود (A) وان البكرتان رقم (2) و (3) تدوران بشكل منفصل حول المحور (B) . الحزام المسطح (Flat Belt) رقم (5) يدور حول البكرتين (1) و (2) اما الحزام المسطح المتقاطع (Crossed Flat Belt) فيدور حول البكرتين (1) و (3) .

<u>المعلومات :</u>

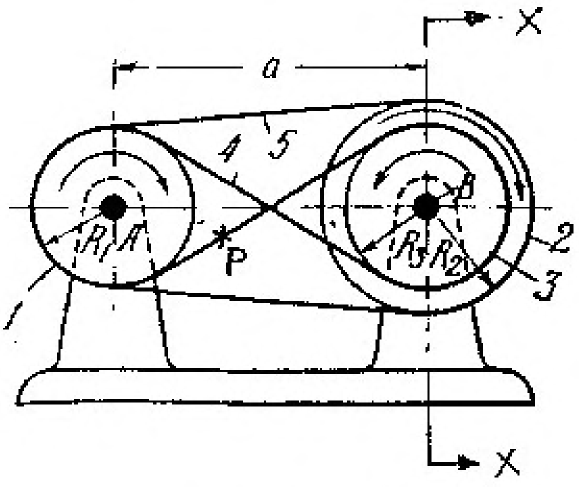
السرعة الدورانية للبكرة رقم (1) = 500 rpm
عرض الحزام رقم (4) b = 30 mm
نوع الحزام رقم (4) (Extremultus Belt (plastic compound belt 1B .
قطر البكرة رقم (1) = 100 mm
قطر البكرة رقم (2) = 140 mm
قطر البكرة رقم (3) = 100 mm
المسافة المركزية بين البكرة رقم (1) و (3) = 200 mm
لتسهيل الحل افرض C= C1.C2.C3. C4. C5 = 1

<u>المطلوب :</u>

1- اوجد اكبر قدرة ممكن ان ينقلها الحزام رقم (4) (2 درجة)

2- اوجد الاجهاد الذي يتعرض له الحزام رقم (4) في النقطة (P) (8 درجات)

3- ارسم المقطع X — X موضحا كيفية تثبيت البكرات والركائز والاعمدة وكافة الاجزاء الاخرى . لاحظ ان هناك سهمين متعاكسان على البكرة رقم (2) ورقم (3) وهذا يدل على ان هناك عمودين خارجين من المنظومة ، وكل عمود يدور عكس الاخر . وضح المقطع بالرسم فقط وبدون ان تقوم بشرح مختصر او مفصل . (7 درجات)



شكل رقم (3)

يتبع القفا .

Q: Figure.1 shows the novel design for reel lawn mower (جزازة العشب- الثيل) which is powered electrically (Battery). The main design feature is the re-location of the wheels, compared to a regular manual push reel mower. This new design features four wheels. This was done in order to eliminate (الغاء) the effects of larger wheels and weight flattening (تسطيح) the grass before it has had a chance to be cut. Therefore, this design will minimize the imprint (الاثر) of the wheels on the grass (الثيل اوالعشب). The smaller front wheels of the mower are not intended to support a large amount of weight; however, they are in place to provide balance. The height of the front wheels will also be adjustable allowing the cutting height to be modified quickly by the operator.

(a) (b)

Figure.1 : (a) The novel design of the reel lawn mower, (b) the regular manual push reel mower.

Requirements:

A: Divide the main system which described above to multi-sub-systems by using the design tree method.

B: Table.1 involves the basic requirements to ensure that the novel design can compete with other products on the market. From this table, write the initial specifications and measure of value.

Requirement	Description	Justification
B-1	The blade reel shall not exceed 20 inches (51 cm) in width.	This width allows for an even cut over uneven terrain.
B-2	The overall width of the reel mower shall not exceed 30 inches (76 cm).	Less than 30 inches allows the mower to be pushed through a standard door frame.
B-3	The mower shall weigh less than 50 lbs (22.7 kg).	One person may operate and lift the mower.
B-4	One battery pack shall be used. The *concept of two packs will be* demonstrated.	Two battery packs allows for increased run *time by providing the ability to swap* batteries.
B-5	The battery packs shall last at least 30 minutes on one full charge when used to perform regular lawn maintenance.	A 30 minute run time is the industry standard. This will allow adequate run time for average lawns while maintaining an appropriate weight.
B-6	The battery packs shall charge in 30 minutes or less.	A charge time that is equal to the run time will allow for efficient battery swapping when mowing large lawns.

Table.1 : The basic requirements.

C: According to the system design flowchart, state the suitable blocks in the flowchart to involve the following paragraph:

Manually propelled reel mowers do not generate enough power to cut through thick grass, weeds (الاعشاب الضارة او الدغل) and other small debris (الطمى والحصى الصغيرة).

D: Use the exploded drawing shown in figure.2 to build the morphological chart with giving more ideas for each sub-system. For this task, the black box, inversion and analogy concepts should be used and specified.

E: Use network combination used in decision making to find the best system and draw its system scheme.

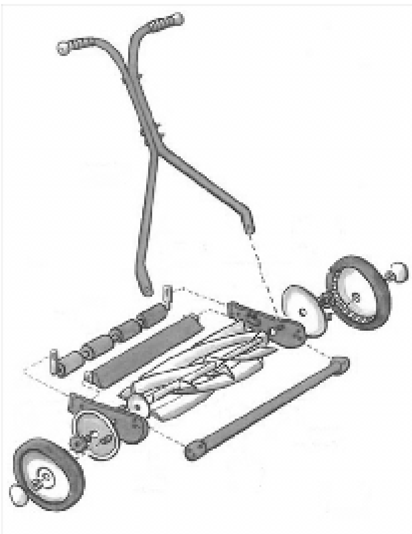


Figure.2: The exploded drawing for the regular manual push reel mower.

Q1: A wormgear shown in figure (1) has a single-thread worm with a pitch diameter of (31.75 mm), a diametral pitch of 10 (module m=2.54), and a normal pressure angle of 14.5°. If the worm meshes with a wormgear having 40 teeth and a face width of (15.87mm), the wormgear is transmitting (104 N.m) of torque at its output shaft, which is rotating at (30 rpm).

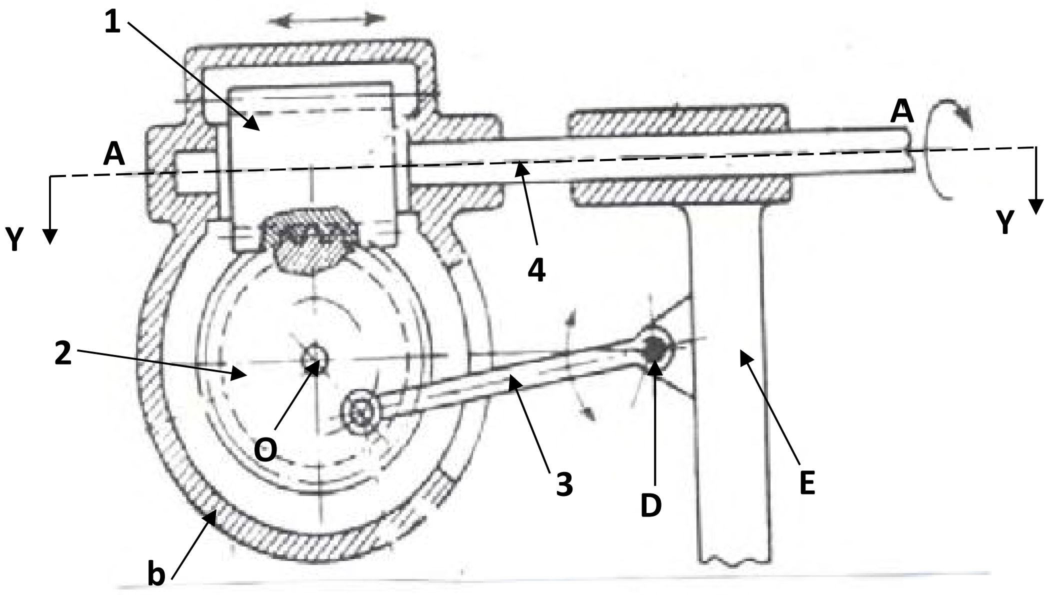


Figure (1)

Requirements:

1- **Choose the suitable material from the following, based on the stress on the gear teeth. (comment on your answer)**
(manganese gear bronze S_{at}=117 MPa)
(phosphor gear bronze S_{at}=165.5 MPa)
(25 mark)

2- **Evaluate the rated load and determine whether the design is satisfactory for pitting resistance.**
(25 mark)

3- **Draw the section Y-Y showing how each part fixed.**
(25 mark)

(الترس الدودي رقم (1) يدور حول المحور (A-A) والذي يتحرك حركة خطية ايضاً. ان الترس الدودي رقم (1) متعشق مع الترس الكبير رقم (2) والذي يدور حول المحور (O) داخل صندوق التروس (b) ويتحركان معا حركة خطية داخل التجويف في الهيكل (E). ان الترس الكبير مربوط بالذراع رقم (3) وهذا الذراع يتحرك حول المسمار (D) المثبت على الهيكل وعند دوران العمود رقم (4) يتحرك الترس الدودي حركة دورانية بالاضافة الى الحركة الخطية وكذلك بالنسبة للترس الكبير رقم (2)) .

Q2: The simple tensile bar which must transmit a specified constant magnitude force (P) as shown in figure (2). Assume that the bar will be manufactured in large quantities, thus a logical objective for optimum design would be minimization of cost. For an acceptable design, cross-sectional area (A), and nominal stress (σ) must satisfy the following constraints:

$A \geq 87.5\ mm^2$

$500\ mm \leq L \leq 750\ mm$

$\left(\Delta = \frac{P.L}{EA}\right) \geq 0.0077mm$

$\sigma_{all} \leq 100\ {}^{N}/_{mm^2}$

c = unite volume cost of shaft = 2500 \$/m^3

P=1000 N

Find minimum cost and at what area?

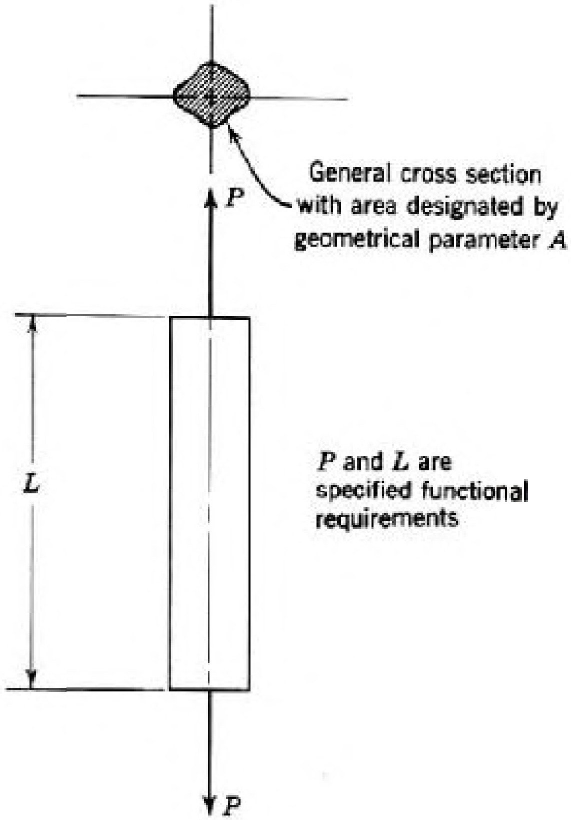


Figure (2)

(25 mark)

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Department of Machines and Equipment Engineering
Second Term Examination 2013/2014

Subject: Design II | **Year: fourth**
Division: General Mech. | **Exam Time: 1.5 Hrs.**
Examiners: Design Group | **Date: 6/5/2014**

Q1: A straight bevel gear pair as shown in Figure (1), has the following data:

N_p= 15

N_G= 45

P_d= 6 (m=4.23); pressure angle=20°.

Transmitted Power=2.23 kW

The pinion speed = 300 rpm.

The face width = 31.75mm.

The gears are driven by a gasoline engine, and the load is a concrete mixer providing moderate shock. Assume that neither gear is straddle-mounted (K_m=1.8). Also assume Kv=1.

Requirements:

1- Compute the bending stress and the contact stress for the teeth. (50 marks)

2- Draw the front section for the dotted area in figure (1) with showing all the fixations details. (20 marks)

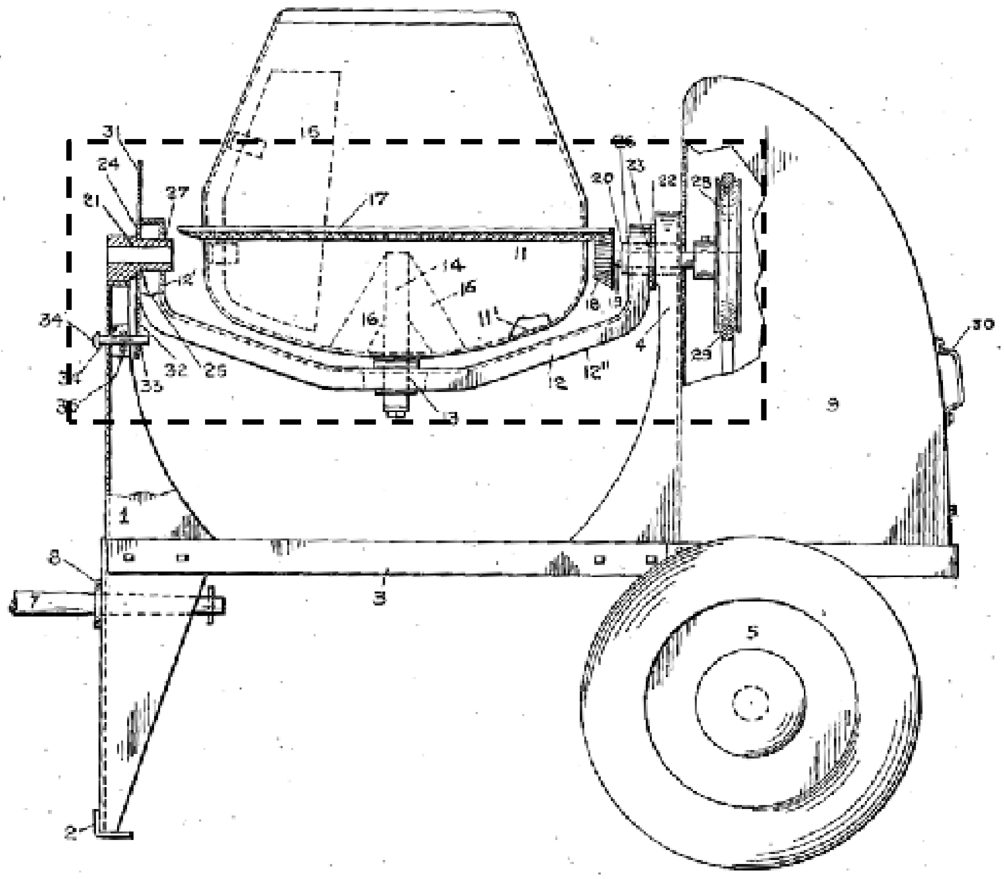

Figure (1)

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Subject: Design II | **Year: fourth**
Division: General Mech. | **Exam Time: 1.5 Hrs.**
Examiners: Design Group | **Date: 6/5/2014**

Q2: Note: Answer branch (A) or branch (B)

(A) The simple tensile bar which must transmit a specified constant magnitude force (P) as shown in figure (2). Assume that the bar will be manufactured in large quantities, thus a logical objective for optimum design would be minimization of cost. For an acceptable design, cross-sectional diameter (D), elongation (Δ), length (L), and nominal stress (σ) must satisfy the following constraints:

$10 \text{ mm} \leq D \leq 20 \text{ mm}$

$500 \text{ mm} \leq L \leq 750 \text{ mm}$

$0.0077\text{mm} \leq \left(\Delta = \frac{P.L}{EA}\right) \leq 0.02\text{mm}$

$\sigma_{all} \leq 100 \ {}^{N}/_{mm^2}$

Safety factor ≥ 3 , c = unit volume cost of shaft = 2500 \$/m3
E= 207 Gpa & P=1000 N
Find minimum cost and at what length and area?

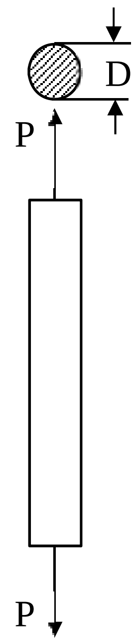
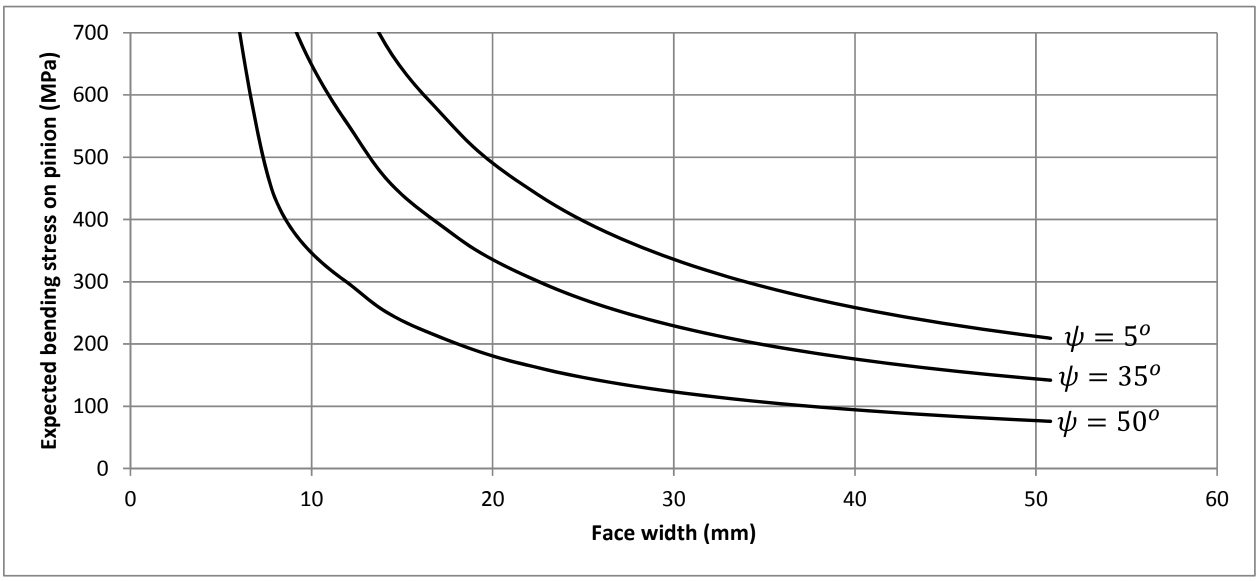


Figure.(2) simple tensile bar with uniformly distributed specified axial load (P)

(B) A helical gear is made from the material has allowable bending stress (S_{at}=447 MPa), with a safety factor ($1.5 \leq S.F \leq 3$) and the face width ($F \leq 20$mm). Assume the bending stress cycle factor (Y_n=0.914) and the reliability factor (K_R=1.25). Find the optimum helix angle and face width from the following graph to satisfy the weight minimization for this gear.



(30 marks)

University of Technology
Department of Machines and Equipment Engineering
Postpond 2nd Term Examination 2013/2014

Subject: Design II **Year: fourth**
Division: General Mech. **Exam Time: 1.5 Hrs.**
Examiners: Design Group **Date: 12/5/2014**

Q1: A helical gear has a transverse diametral pitch of (8), a transverse pressure angle of (14.5°), (45 teeth), a face width of (50mm), and a helix angle of (30°). The gear transmits (4kW) at an input speed of (1250 rpm), and it operates with a pinion having (15 teeth). The power comes from an electric motor, and the drive is to a reciprocating pump.

Requirements:

1- Compute the expected bending stress and the contact stress on the pinion teeth. (50 marks)

2- Complete the section of the reciprocating pump (shown in figure.1) by adding the details of the gearbox which surrounded by dotted line. (20 marks)

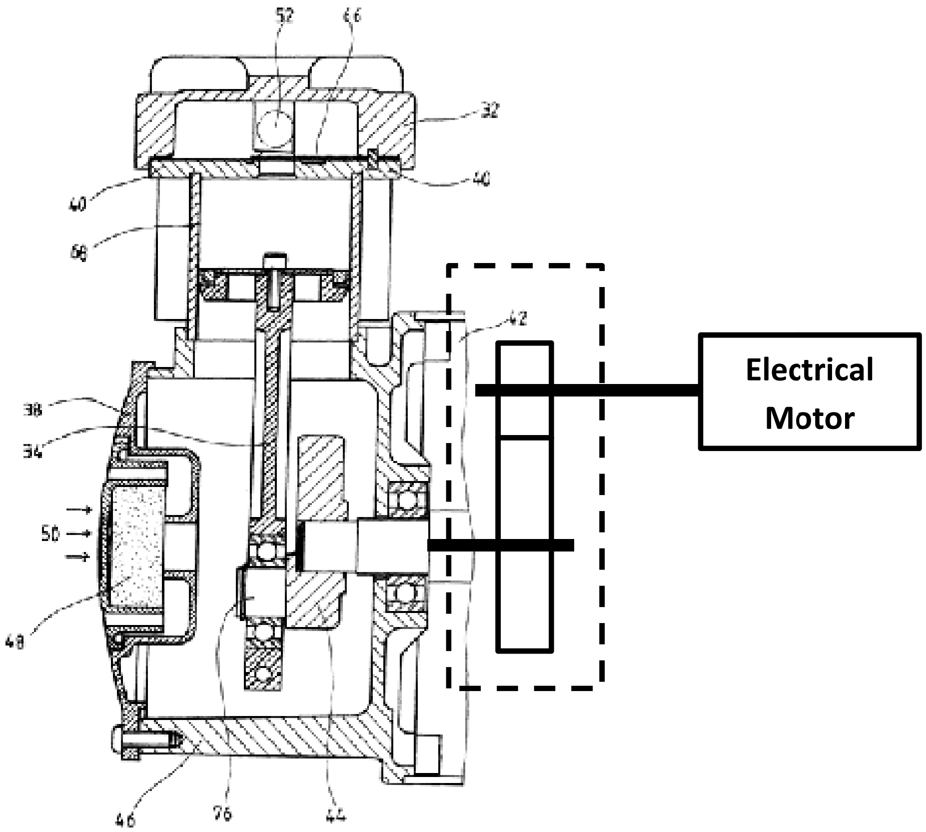


Figure (1)

University of Technology
Department of Machines and Equipment Engineering
Postpond 2nd Term Examination 2013/2014

Subject: Design II **Year: fourth**
Division: General Mech. **Exam Time: 1.5 Hrs.**
Examiners: Design Group **Date: 12/5/2014**

Q2: Note: Answer branch (A) or branch (B)

(A) The simple tensile bar which must transmit a specified constant magnitude force (P) as shown in figure (2). Assume that the bar will be manufactured in large quantities, thus a logical objective for optimum design would be minimization of cost. For an acceptable design, cross-sectional diameter (D), length (L), and nominal stress (σ) must satisfy the following constraints:

$10 \text{ mm} \leq D \leq 20 \text{ mm}$

$S_y = 300 \ ^{N}/_{mm^2}$

Safety factor ≥ 3 , c = unit volume cost of shaft = 2500 \$/m3

P=1000 N

Find minimum cost and at what length and area?

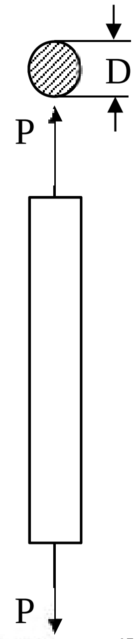
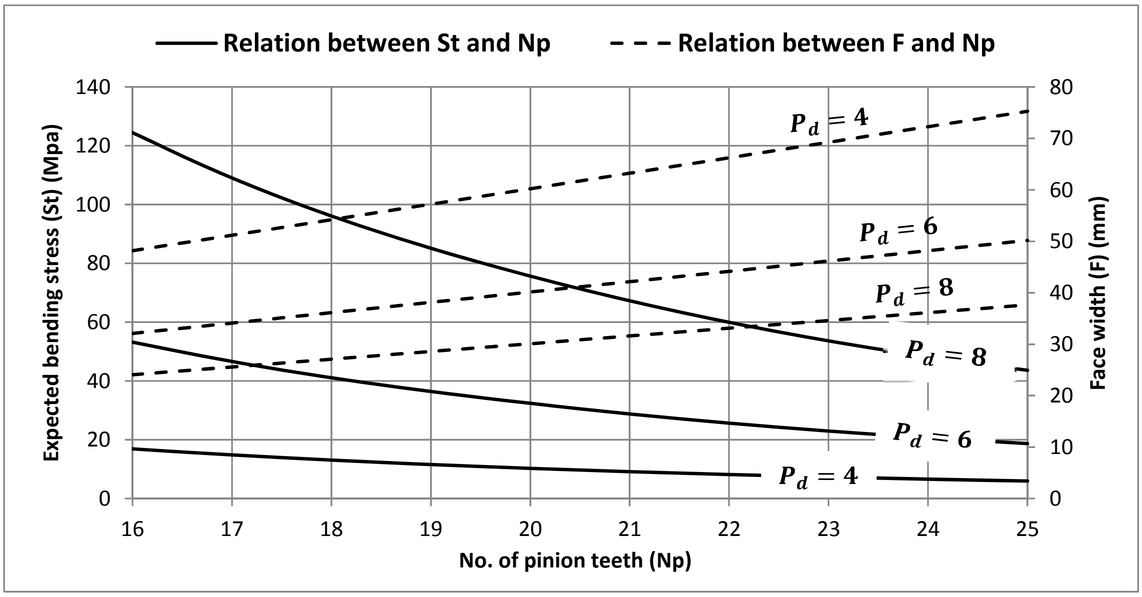


Figure.(2) simple tensile bar with uniformly distributed specified axial load (P)

(B) A bevel gear is made from the material has allowable bending stress (S_{at}=154 MPa), with a safety factor (S.F = 1.5). Assume the bending stress cycle factor (Y_n=0.955) and the reliability factor (K_R=1). Find the feasible no. of pinion teeth, face width and diametral pitch from the following graph.



(30 marks)

Q1: A wormgear shown in figure (1) has a single-thread worm with a pitch diameter of (31.75 mm), a diametral pitch of 10 (module m=2.54), and a normal pressure angle of 14.5°. If the worm meshes with a wormgear having 40 teeth and a face width of (15.87mm), the wormgear is transmitting (104 N.m) of torque at its output shaft, which is rotating at (100 rpm).

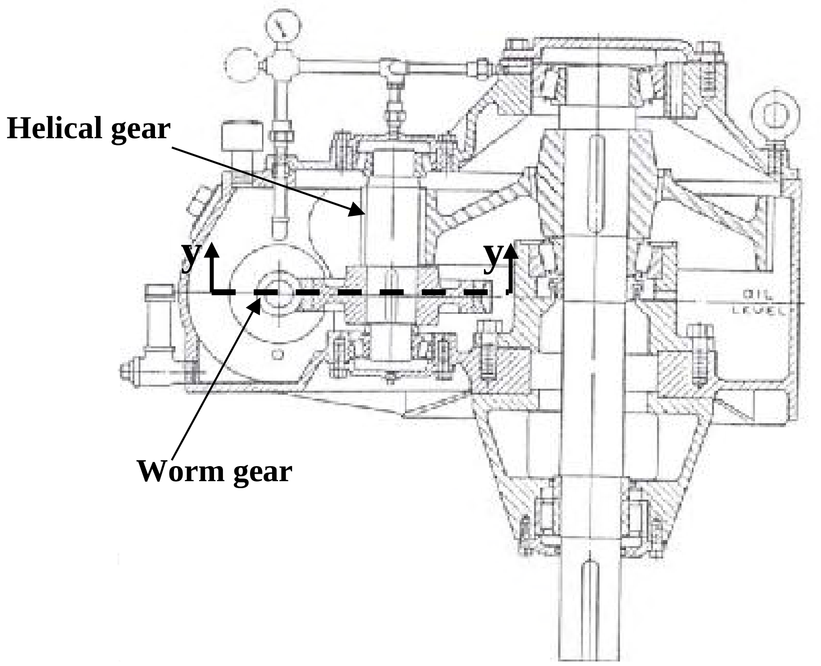


Figure (1)

Requirements:

1- **Evaluate the rated load and determine whether the design is satisfactory for pitting resistance.** **(30 mark)**

2- **Assume the following data for helical gear:**

Reduction ratio =4
Normal diametral pitch = 12
No. of teeth of pinion = 24
Helix angle = 15°
Normal pressure angle = 20°
A quality number = 8
Dynamic factor (kv) =1.35
Load distribution factor (km) =1.26
Hardness ratio factor (C_H) =1

(40 mark)

3- **Draw the section Y-Y showing how each part fixed.**

(30 mark)

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Second Term Examination 2012/2013

Subject: Design II | Year: fourth
Division: General Mech. | Exam Time: 1:30 Hrs.
Examiners: Design Group | Date: 22/4/2013

Answer (Three) Questions Only

(Assume missing data)

Q1: Fig.1 shows a power flow through a gear pair:

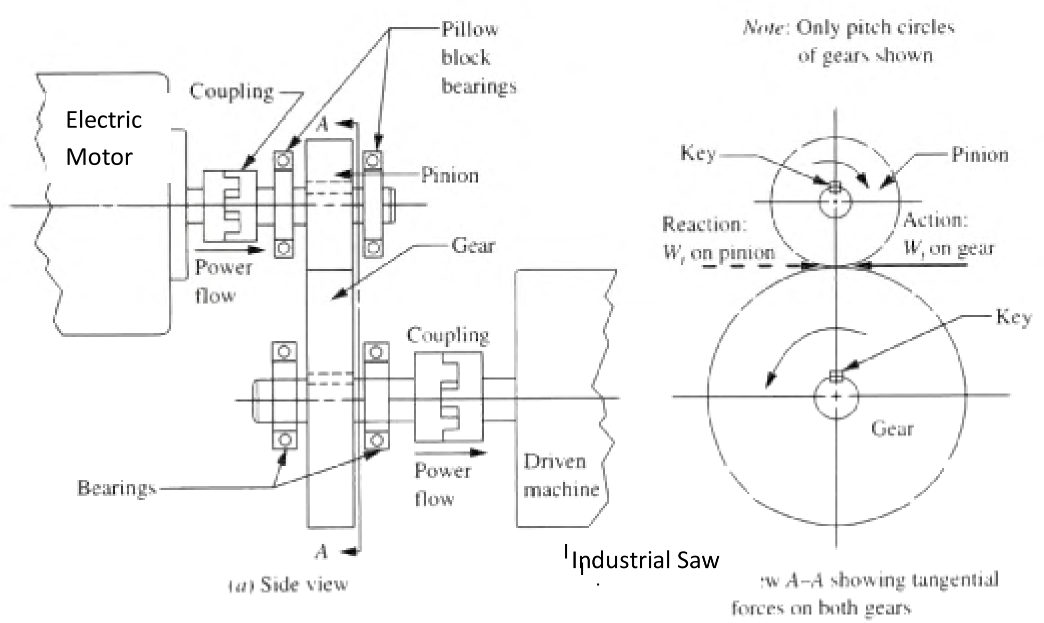

Table.1 shows a trial for solving the spur gear above (Fig.1):

P	18.65 kw = 25 hp
n_p	1750 rpm
n_g	500 rpm
N_p	20
N_g	70
P_d	8
D_p	63.5 mm
C	142.8 mm
W_t	3.2 kN
F	38.1 mm
Q_v	6
K_v	1.45
K_m	1.2

Find suitable material for this case. Then give your comment.

(35 mark)

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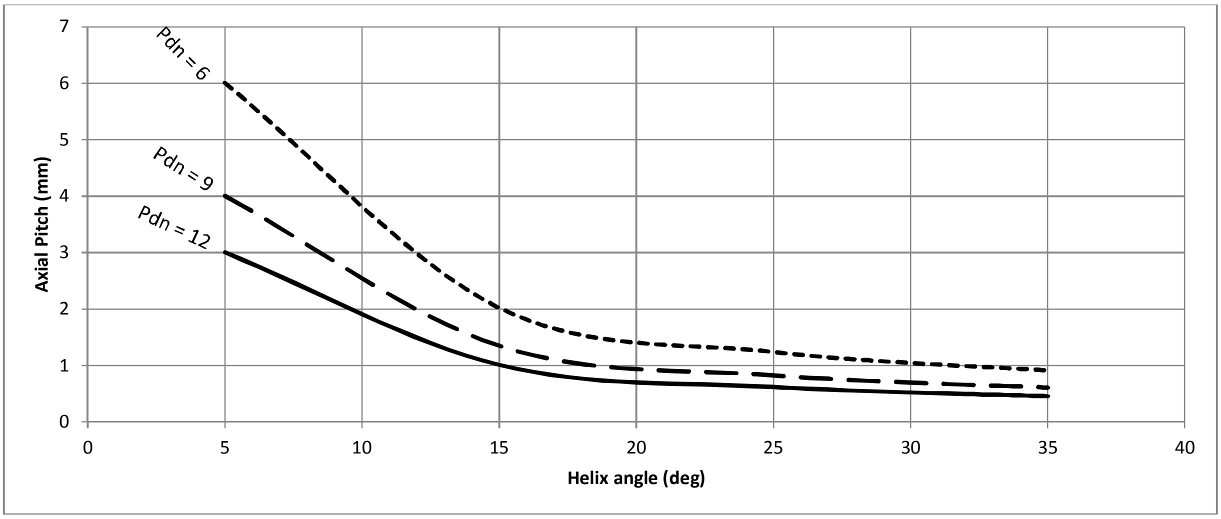
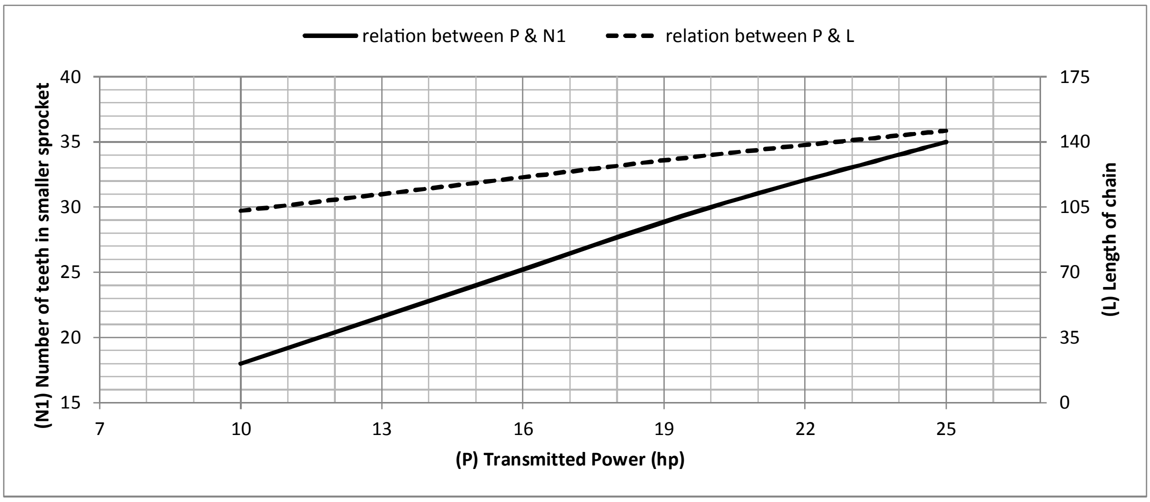
Q2: A: Use helical gear instead of spur gear for question one, with helix angle ($\Psi = 5^o - 35^o$) and ($P_{dn} = 6 - 12$ teeth/in), the following figure is the results between axial pitch (P_x) and (Ψ):



Find the feasible point that gives the smaller size of the gears for the following information: ($16^o > \Psi > 9^o$), ($12 > P_{dn} > 9$) & ($4 > P_x > 2$)

(15 mark)

B: If a chain will be used: $n_1 = 1750$ rpm , chain type 40, number of chain=1, The following figure is the results between power, number of teeth in small sprocket (N1) and length of chain (L).



Find max. power that satisfy the following limits:
(26 < N1 < 22)

(112 < L < 133)

And then find the exact values for (N1) & (L) at the selected power.

(20 mark)

Q3: If the gear will be changed to bevel gears with following information: $P=25hp$, $P_d=6$, $N_p=16$, $n_g=500$ rpm, $n_p=1750$ rpm, $Q_v=9$. Find type of material that can be used in this case.

(35 mark)

Q4: If a v-belt are used with information: $P=25hp$, $n_1=1750$ rpm, $n_2 \approx 500$ rpm, Find type of belts, no. of belts, length of belts, diameter of belts, angle of contacts, actual output speed (give your opinion about the results).

(35 mark)

Q5: Draw sectional view showing how each part fixed especially fixation of outer and inner races using for example step shaft, snap rings, covers… etc on shaft and housing. for any one of the previous questions that you solved above.

(35 mark)

Appendix (B)

Allowable Formulas for Mechanical Design Open Book Examination

Column analysis

The procedure for analyzing straight, centrally loaded columns:

1. For the given column, compute its actual slenderness ratio.
2. Compute the value of C_c .
3. Compare C_c with *KL/r*. Because C_c represents the value of the slenderness ratio that separates a long column from a short one, the result of the comparison indicates which type of analysis should be used.
4. If the actual *KL/r* is greater than Cc the column is *long*. Use Euler's equation:

$$P_{cr} = \frac{\pi^2 EA}{(KL\ /\ r)^2}$$

The equation gives the critical load, *Pcr*, at which the column would begin to buckle.
An alternative form of the Euler formula is often desirable. Note that:

$$P_{cr} = \frac{\pi^2 EA}{(KL / r)^2} = \frac{\pi^2 EA}{(KL)^2 / r^2} = \frac{\pi^2 EAr^2}{(KL)^2}$$

But, from the definition of the radius of gyration, r,

$$r = \sqrt{I / A}$$

$$r^2 = I / A$$

Then

$$P_{cr} = \frac{\pi^2 EA}{(KL)^2} \frac{I}{A} = \frac{\pi^2 EI}{(KL)^2}$$

This form of the Euler equation aids in a design problem in which the objective is to specify a size and a shape of a column cross section to carry a certain load.

Notice that the buckling load is dependent only on the geometry (length and cross section) of the column and the stiffness of the material represented by the modulus of elasticity. The strength of the material is not involved at all. For these reasons, it is often of no benefit to specify a high-strength material in a long column application. A lower-strength material having the same stiffness, E, would perform as well.

5. If KL/r is less than Cc, the column is *short*. Use the J. B. Johnson formula:

Use of the Euler formula in this range would predict a critical load greater than it really is. The J. B. Johnson formula is written as follows:

$$P_{cr} = AS_y \left[1 - \frac{S_y (KL / r)^2}{4\pi^2 E}\right]$$

The critical load for a short column is affected by the strength of the material in addition to its stiffness, E. As shown in the preceding section, strength is not a factor for a long column when the Euler formula is used.

Eccentrically Loaded Columns

An eccentric load is one that is applied away from the centroidal axis of the cross section of the column, as shown in the graphic help entitled “Eccentric column”. Such a load exerts bending in addition to the column action that results in the deflected shape shown in the figure. The maximum stress in the deflected column occurs in the outermost fibers of the cross section at the midlength of the column where the maximum deflection, y_{max} occurs. Let's denote the stress at this point as $\sigma L/2$. Then, for any applied load, P,

$$\sigma_{L/2} = \frac{P}{A}\left[1 + \frac{ec}{r^2}\sec\left(\frac{KL}{2r}\sqrt{\frac{P}{AE}}\right)\right]$$

Note that this stress is not directly proportional to the load. When evaluating the secant in this formula, note that its argument in the parentheses is in radians. Also, because most calculators do not have the secant function, recall that the secant is equal to 1/cosine.

For design purposes, we would like to specify a design factor, N, that can be applied to the failure load similar to that defined for straight, centrally loaded columns. However, in this case, failure is predicted when the maximum stress in the column exceeds the yield strength of the material. Let's now define a new term, Py, to be the load applied to the eccentrically loaded column when the maximum stress is equal to the yield strength. The equation then becomes

$$S_y = \frac{P_y}{A}\left[1+\frac{ec}{r^2}\sec\left(\frac{KL}{2r}\sqrt{\frac{P_y}{AE}}\right)\right]$$

Now, if we define the allowable load to be

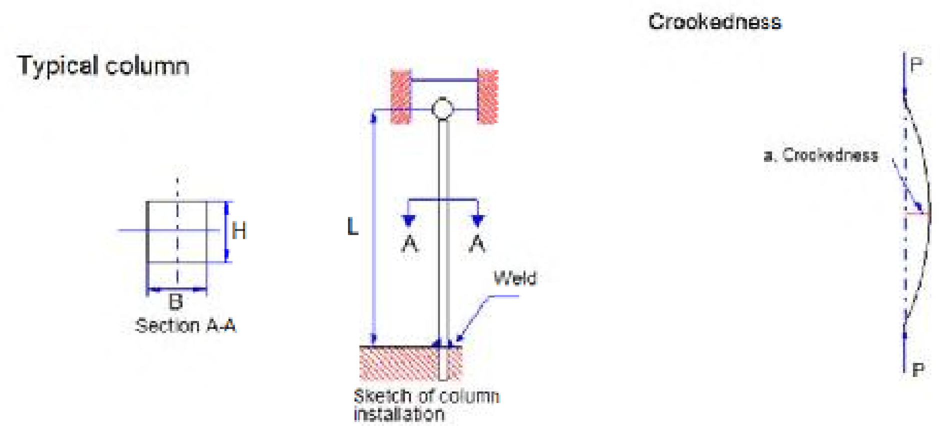
$$P_a = P_y / N \text{ or } P_y = NP_a$$

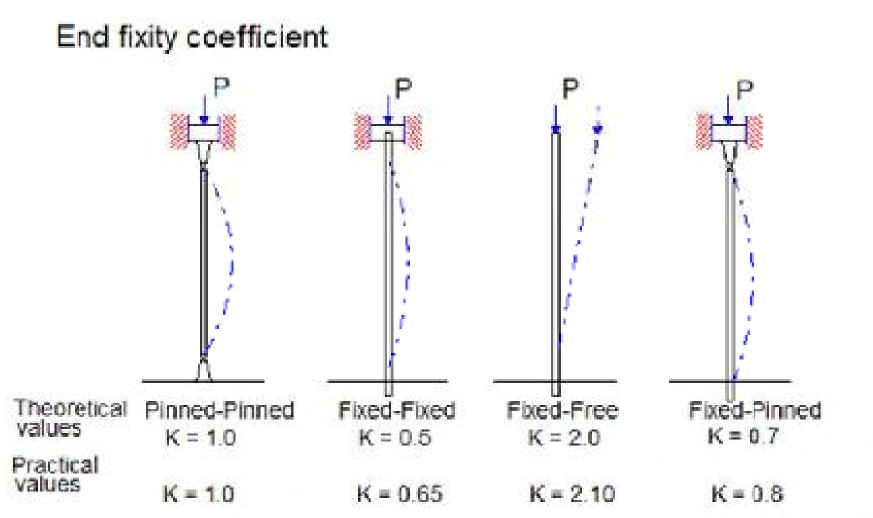
this equation becomes

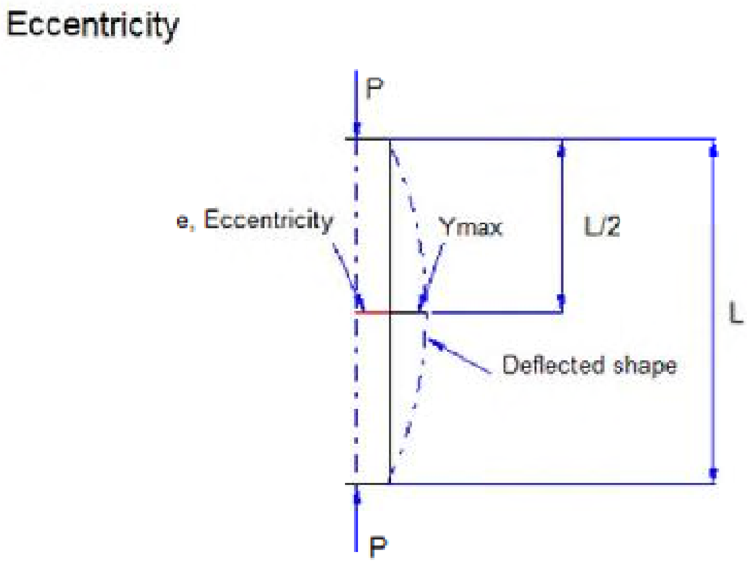
$$\text{Required} \qquad S_y = \frac{NP_a}{A}\left[1+\frac{ec}{r^2}\sec\left(\frac{KL}{2r}\sqrt{\frac{NP_a}{AE}}\right)\right]$$

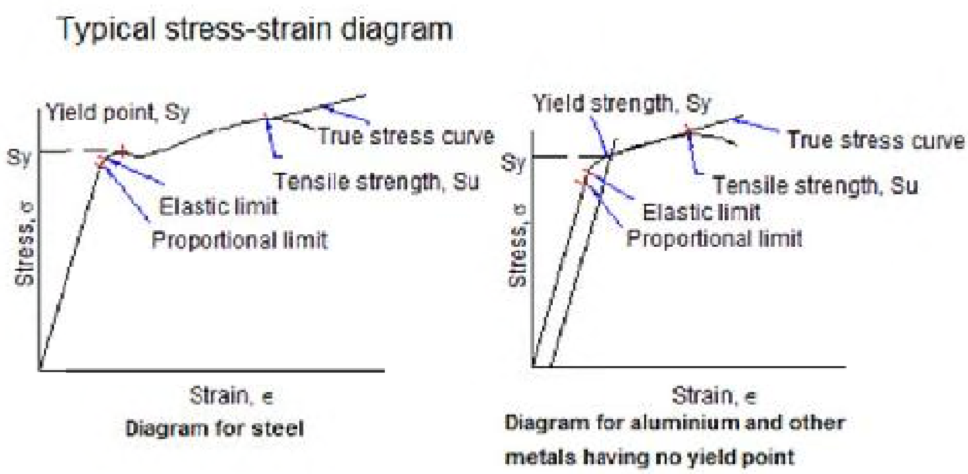
This equation cannot be solved for either A or Pa, so an iterative solution is required.
Another critical factor may be the amount of deflection of the axis of the column due to the eccentric load:

$$Y_{max} = e\left[\sec\left(\frac{KL}{2r}\sqrt{\frac{P}{AE}}\right)-1\right]$$









V-Belt Drive Design

A belt is a flexible power transmission element that seats tightly on a set of pulleys or sheaves. When the belt is used for speed reduction, the typical case, the smaller sheave is mounted on the high-speed shaft, such as the shaft of an electric motor. The larger sheave is mounted on the driven machine. The belt is designed to ride around the two sheaves without slipping.

The belt is installed by placing it around the two sheaves while the center distance between them is reduced. Then the sheaves are moved apart, placing the belt in a rather high initial tension. When the belt is transmitting power, friction causes the belt to grip the driving sheave, increasing the tension in one side, called the "tight side," of the drive. The tensile force in the belt exerts a tangential force on the driven sheave, and thus a torque is applied to the driven shaft. The opposite side of the belt is still under tension, but at a smaller value. Thus, it is called the "slack side."

The most widely used type of belt, particularly in industrial drives and vehicular applications, is the V-belt drive. The V-shape causes the belt to wedge tightly into the groove, increasing friction and allowing high torques to be transmitted before slipping occurs. Most belts have high-strength cords positioned at the pitch diameter of the belt cross section to increase the tensile strength of the belt. The cords, made from natural fibers, synthetic strands, or steel, are embedded in a firm rubber compound to provide the flexibility needed to allow the belt to pass around the sheave. Often an outer fabric cover is added to give the belt good durability. The data given in this program are for the narrow-section belts: 3V, 5V and 8V.

The pulley, with a circumferential groove carrying the belt, is called a sheave (usually pronounced "shiv").

The size of a sheave is indicated by its pitch diameter, slightly smaller than the outside diameter of the sheave.

The speed ratio between the driving and the driven sheaves is inversely proportional to the ratio of the sheave pitch diameters. This follows from the observation that there is no slipping (under normal loads). Thus, the linear speed of the pitch line of both sheaves is the same as and equal to the belt speed, v_b. Then

$$V_b = R_1 \cdot \omega_1 = R_2 \cdot \omega_2$$

Since $R_1 = D_1 / 2$ and $R_2 = D_2 / 2$, then

$$V_b = \frac{D_1 \cdot \omega_1}{2} = \frac{D_2 \cdot \omega_2}{2}$$

The angular velocity ratio is

$$\frac{\omega_1}{\omega_2} = \frac{D_2}{D_1}$$

The relationships between pitch length, L, center distance, C, and the sheave diameters are

$$L = 2C + 1.57(D_2 + D_1) + \frac{(D_2 - D_1)^2}{4C}$$

$$C = \frac{B + \sqrt{B^2 - 32(D_2 - D_1)^2}}{16}$$

Where: $B = 4 \cdot L - 6.28 \cdot (D_2 + D_1)$

The angle of contact of the belt on each sheave is

$$\theta_1 = 180° - 2\sin^{-1}\left(\frac{D_2 - D_1}{2 \cdot C}\right)$$

$$\theta_2 = 180° + 2\sin^{-1}\left(\frac{D_2 - D_1}{2 \cdot C}\right)$$

These angles are important because commercially available belts are rated with an assumed contact angle of 180°. This will occur only if the drive ratio is 1 (no speed change). The angle of contact on the smaller of the two sheaves will always be less than 180°, requiring a lower power rating. Note: the angle of wrap on the smaller sheave should be greater then 120°.

The length of the span between the two sheaves, over which the belt is unsupported, is

$$S = \sqrt{C^2 - \left(\frac{D_2 - D_1}{2}\right)^2}$$

This is important for two reasons: You can check the proper belt tension by measuring the amount of force required to deflect the belt at the middle of the span by a given amount. Also, the tendency for the belt to vibrate or whip is dependent on this length.

The contributors to the stress in the belt are as follows:

1. The tensile force in the belt, maximum on the tight side of the belt.
2. The bending of the belt around the sheaves, maximum as the tight side of the belt bends around the smaller sheave.
3. Centrifugal forces created as the belt moves around the sheaves.

The maximum total stress occurs where the belt enters the smaller sheave, and the bending stress is a major part. Thus, there are recommended minimum sheave diameters for standard belts. Using smaller sheaves drastically reduces belt life. The design value of the ratio of the tight side tension to the slack side tension is 5.0 for V-belt drives. The actual value may range as high as 10.0.

The factors involved in selection of a V-belt and the driving and driven sheaves and proper installation of the drive are summarized in this section. Abbreviated examples of the data available from suppliers are given for illustration. Catalogs contain extensive data, and step-by-step instructions are given for their use. The basic data required for drive selection are the following:

- The rated power of the driving motor or other prime mover
- The service factor based on the type of driver and driven load
- The center distance
- The power rating for one belt as a function of the size and speed of the smaller sheave
- The belt length
- The size of the driving and driven sheaves-- As a guide this software suggests selecting a standard input driving sheave that produces a belt speed of 4000 ft/min.
- The correction factor for belt length
- The correction factor for the angle of wrap on the smaller sheave
- The number of belts
- The initial tension on the belt

Many design decisions depend on the application and on space limitations. A few guidelines are given here:

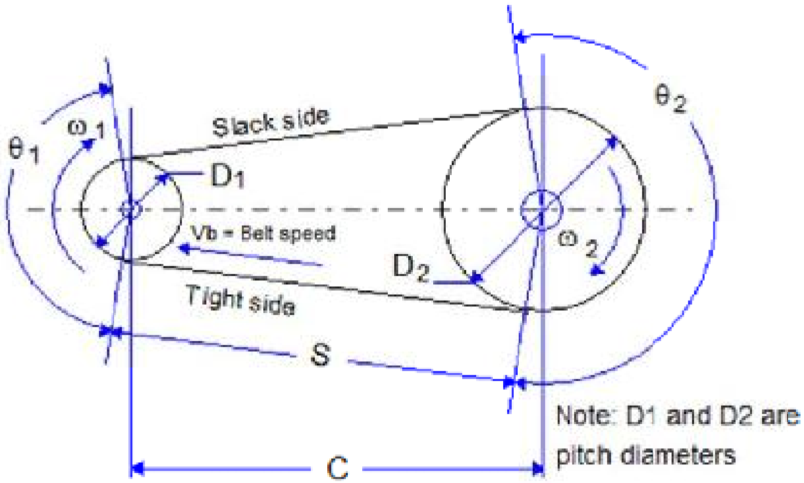
- Adjustment for the center distance must be provided in both directions from the nominal value. The center distance must be shortened at the time of installation to enable the belt to be placed in the grooves of the sheaves without force. Provision for increasing the center distance must be made to permit the initial tensioning of the drive and to take up for belt stretch. Manufacturers' catalogs give the data. One convenient way to accomplish the adjustment is the use of a take-up unit.
- If fixed centers are required, idler pulleys should be used. It is best to use a grooved idler on the inside of the belt, close to the large sheave. Adjustable tensioners are commercially available to carry the idler.
- The nominal range of center distances should be

$$D_2 < C < 3(D_2 + D_1)$$

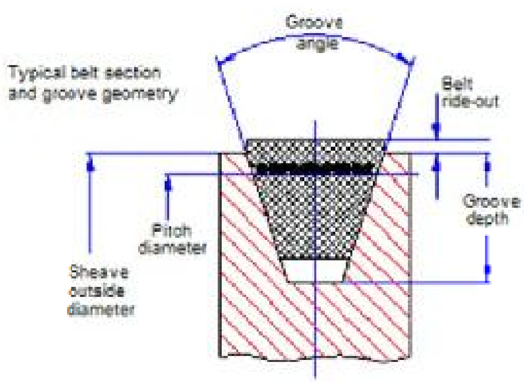
- The angle of wrap on the smaller sheave should be greater than 120°.
- Most commercially available sheaves are cast iron, which should be limited to 6 500-ft/min belt speed.
- Consider an alternative type of drive, such as a gear type or chain, if the belt speed is less than 1 000 ft/min.
- Avoid elevated temperatures around belts.
- Ensure that the shafts carrying mating sheaves are parallel and that the sheaves are in alignment so that the belts track smoothly into the grooves.
- In multibelt installations, matched belts are required. Match numbers are printed on industrial belts, with 50 indicating a belt length very close to nominal. Longer belts carry match numbers above 50; shorter belts below 50.
- Belts must be installed with the initial tension recommended by the manufacturer. Tension should be checked after the first few hours of operation because seating and initial stretch occur.

Most manufacturers offer two kinds of belts in each cross section. The ones with the "X" are cog belts, and if there is no "X", it is of plain construction. Both types have the same cross sectional dimensions and will therefore fit in the same sheave.

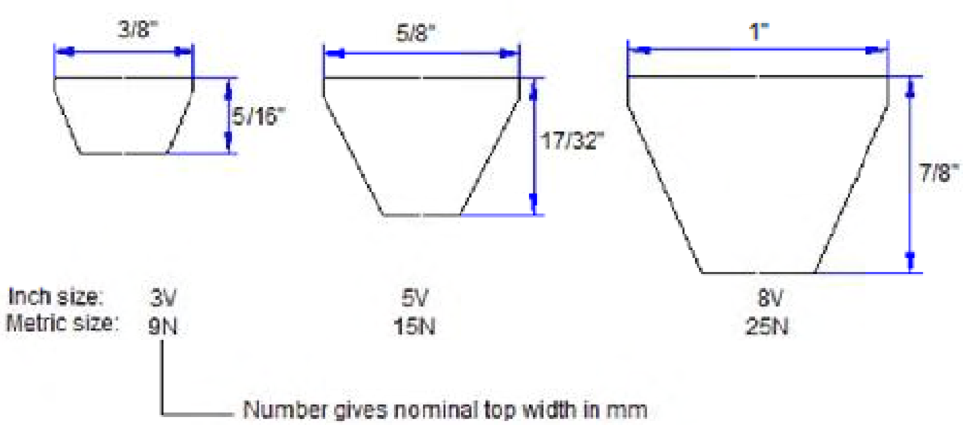
Basic belt drive geometry



Cross section of V-belt and sheave groove



Industrial narrow-section V-belts



Standard roller chain drive design

A chain is a power transmission element made as a series of pin-connected links. The design provides for flexibility while enabling the chain to transmit large tensile forces. When transmitting power between rotating shafts, the chain engages mating toothed wheels, called *sprockets*.

The most common type of chain is the *roller chain*, in which the roller on each pin provides exceptionally low friction between the chain and the sprockets. Other types include a variety of extended link designs used mostly in conveyor applications.

Roller chain is classified by its pitch, the distance between corresponding parts of adjacent links. The pitch is usually illustrated as the distance between the centers of adjacent pins. Standard roller chain carries a size designation from 40 to 240. The digits (other than the final zero) indicate the pitch of the chain in eighths of an inch. For example, the no. 100 chain has a pitch of 10/8 or $1\frac{1}{4}$ in. A series of heavy-duty sizes, with the suffix H on the designation (60H-240H), has the same basic dimensions as the standard chain of the same number except for thicker side plates. In addition, there are the smaller and lighter sizes: 25, 35, and 41.

Manufacturers supply the average tensile strengths of the various chain sizes. These data can be used for very low speed drives or for applications in which the function of the chain is to apply a tensile force or to support a load. It is recommended that only 10% of the average tensile strength be used in such applications. For power transmission, the rating of a given chain size as a function of the speed of rotation must be determined, as explained later.

A variety of attachments are available to facilitate the application of roller chain to conveying or other material handling uses. Usually in the form of extended plates or tabs with holes provided, the attachments make it easy to connect rods, buckets, parts pushers, part support devices, or conveyor slats to the chain.

The rating of chain for its power transmission capacity considers three modes of failure:

1. Fatigue of the link plates due to the repeated application of the tension in the tight side of the chain
2. Impact of the rollers as they engage the sprocket teeth
3. Galling between the pins of each link and the bushings on the pins.

The ratings are based on empirical data with a smooth driver and a smooth load (service factor = 1.0) and with a rated life of approximately 15 000 h. The important variables are the pitch of the chain and the size and rotational speed of the smaller sprocket. Lubrication is critical to the satisfactory operation of a chain drive. Manufacturers recommend the type of lubrication method for given combinations of chain size, sprocket size, and speed.

The standard sizes of chain are: no. 25 (1/4 in), no. 35 (0.375 in), no. 40 (1/2 in), no. 41 (1/2 in), no. 50 (0.625 in), no. 60 (3/4 in), no. 80 (1.00 in), no. 100 (1.25 in), no. 120 (1.5 in), no. 140 (1.75 in), no. 160 (2 in), no. 180 (2.25 in), no. 200 (2.5 in), no. 240 (3 in),. These are typical of the types of data available for all chain sizes in manufacturers' catalogs. Notice these features of the data:

The ratings are based on the speed of the smaller sprocket.

For a given speed, the power capacity increases with the number of teeth on the sprocket. Of course, the larger the number of teeth, the larger the diameter of the sprocket. Note that the use of a chain with a small pitch on a large sprocket produces the quieter drive.

For a given sprocket size (a given number of teeth), the power capacity increases with increasing speed up to a point; then it decreases. Fatigue due to the tension in the chain governs at the low to moderate speeds; impact on the sprockets governs at the higher speeds. Each sprocket size has an absolute upper-limit speed due to the onset of galling between the pins and the bushings of the chain. This explains the abrupt drop in power capacity to zero at the limiting speed.

The manufacturers' ratings are for a single strand of chain. Although multiple strands do increase the power capacity, they do not provide a direct multiple of the single-strand capacity. The capacity for 2, 3, and 4 strand systems are 1.7, 2.5 and 3.3 respectively.

The manufacturers' ratings are for a service factor of 1.0. The designer must specify a service factor for a given application based on the type of driver and load for that system.

The following are general recommendations for designing chain drives:

The minimum number of teeth in a sprocket should be 17 unless the drive is operating at a very low speed, under 100 rpm.

The maximum speed ratio should be 7.0, although higher ratios are feasible. Two or more stages of reduction can be used to achieve higher ratios.

The center distance between the sprocket axes should be approximately 30 to 50 pitches (30 to 50 times the pitch of the chain).

The arc of contact of the chain on the smaller sprocket should be no smaller than 120°.

The larger sprocket should normally have no more than 120 teeth.

The preferred arrangement for a chain drive is with the centerline of the sprockets horizontal and with the tight side on top.

The chain length must be an integral multiple of the pitch, and an even number of pitches is recommended. The center distance should be made adjustable to accommodate the chain length and to take up for tolerances and wear. Excessive sag on the slack side should be avoided, especially on drives that are not horizontal. A convenient relation between center distance (C), chain length (L), number of teeth in the small sprocket (N_1), and number of teeth in the large sprocket (N_2), expressed in pitches, is

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

The exact theoretical center distance for a given chain length, again in pitches, is

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

The theoretical center distance assumes no sag in either the tight or the slack side of the chain, and thus it is a *maximum.* Negative tolerances or adjustment must be provided.

The pitch diameter of a sprocket with N teeth for a chain with a pitch of p is

$$D = \frac{p}{\sin(180^\circ / N)}$$

The minimum sprocket diameter and therefore the minimum number of teeth in a sprocket are often limited by the size of the shaft on which it is mounted. Check the sprocket catalog.

Rotational speeds and lubrication methods

Chains are typically used in lower speed, higher torque conditions than are belts.

$$V_c = \frac{\pi \cdot D \cdot n}{12}$$

where D = pitch diameter of sprocket;
n = rotational speed of sprocket

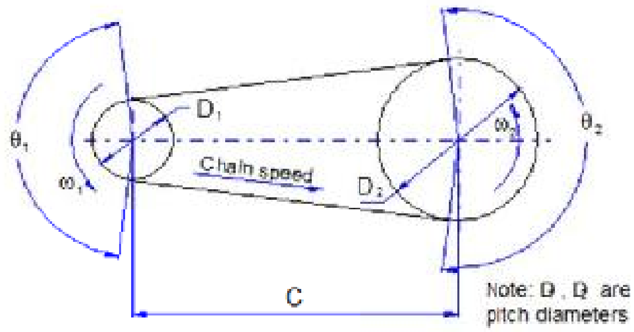
A constant supply of clean oil is essential to smooth operation and satisfactory life of the chain drive. Chain manufacturers recommend three different methods of applying lubrication, depending on the linear speed of the chain V_c. Although there may be modest differences between manufacturers, the following are the general guidelines for the limits of speed. Refer to the graphic help for illustrations of the methods.

Type A (170 to 650 ft/min). Manual or drip lubrication: For manual lubrication, oil is applied with a brush or a spout can, preferably at least once every 8 h of operation. For drip feed lubrication, oil is fed directly onto the link plates of each chain strand.

Type B (650 to 1 500 ft/min). Bath or disc lubrication: The chain cover provides a sump of oil into which the chain dips continuously. Alternatively, a disc or a slinger can be attached to one of the shafts to lift oil to a trough above the lower strand of chain. The trough then delivers a stream of oil to the chain. The chain itself, then, does not need to dip into the oil.

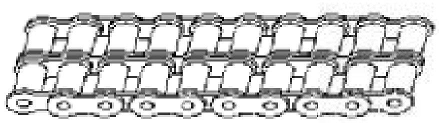
Type C (above 1 500 ft/min). Oil stream lubrication: An oil pump delivers a continuous stream of oil on the lower part of the chain.

Basic chain drive geometry



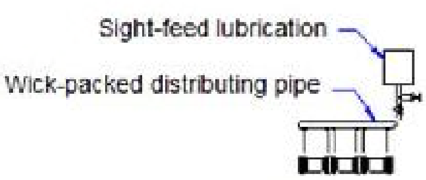
Roller chain styles

Standard roller chain, single strand

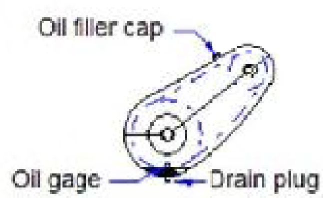


Standard roller chain, two-strand (also available with three and four strands)

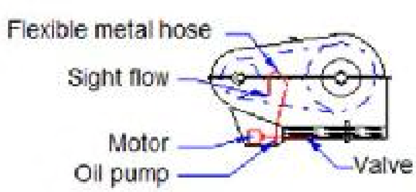
Lubrication methods



Drip feed lubrication (Type A)



Shallow bath (Type B)



Oil stream lubrication (Type C)

Spur gears

Spur gears have teeth that are straight and arranged parallel to the axis of the shaft that carries the gear. The curved shape of the faces of the spur gear teeth has a special geometry called an involute curve. This shape makes it possible for two gears to operate together with smooth, positive transmission of power. The shafts carrying gears are parallel.

Spur gear design

- Actual output speed (gear)

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

n_P = rotational speed of the pinion

VR = gear ratio

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

N_G, N_P = number of gear, pinion teeth.

The spreadsheet computes the approximate number of gear teeth to produce the desired speed from $N_G = N_P \frac{n_{Gd}}{n_P}$ (n_{Gd} = desired output speed). But, of course, the number of teeth on any gear must be an integer, and the actual value of N_G is selected by the designer.

Spur gear geometry For full depth involute teeth in the diametral pitch system

- Pitch diameter

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

- Diametral Pitch

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

- Outside diameter

$$D_o = \frac{N + 2}{P_d}$$

- Addendum

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

- Dedendum

  if $P_d < 20$

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

  if $P_d \geq 20$

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

- Clearance

  if $P_d < 20$

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

if $P_d \geq 20$

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

- Root diameter

$$D_R = D - 2b$$

- Base circle diameter

$$D_b = D \cos \phi$$

- Circular pitch

$$p = \frac{\pi D}{N}$$

- Whole depth

$$h_t = a + b$$

- Working depth

$$h_k = 2a$$

- Tooth thickness

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

- Center distance

$$C = \frac{D_G + D_P}{2}$$

Bending geometry factor, J, is dependent on the number of teeth of gear for which geometry factor is desired and on the number of teeth in mating gear. Values can be found from AGMA 908-B89(R1995).

Pitting geometry factor, I, is dependent on the tooth geometry and on gear ratio. Values can be found from AGMA Standard 218.01.

Force and speed factors

- Pitch line speed

$$V_t = \frac{\pi D_P n_P}{12}$$

- Tangential force

$$W_t = \frac{33000 \cdot (P)}{V_t}$$

or

$$W_t = \frac{126000 \cdot (P)}{nD}$$

where:

P = transmitted power

- Radial force

$$W_r = W_t \tan\phi$$

- Normal force

$$W_n = \frac{W_t}{\cos\phi}$$

- Expected bending stress

$$S_t = \frac{W_t P_d}{F \cdot J} K_o K_s K_m K_B K_v$$

where:

J = bending geometry factor

K_o = overload factor

K_s = size factor

K_m = load-distribution factor

K_B = rim thickness factor

K_v = dynamic factor.

The AGMA indicates that the size factor can be taken to be 1.00 for most gears. But for gears with large-size teeth or large face widths, a value greater than 1.00 recommended. The program computes the size factor automatically.

The determination of load-distribution factor is based on many variables in the design of the gears themselves as well as in the shafts, bearings, housings, and the structure in which the gear drive is installed. Therefore, it is one of the most difficult factors to specify. Much analytical and experimental work is continuing on the determination of values for K_m. We will use the following equation for computing the value of the load-distribution factor:

$$K_m = 1.0 + C_{pf} + C_{ma}$$

where:

C_{pf} = pinion proportion factor is dependent on face width and pitch diameter

C_{ma} = mesh alignment factor.

The dynamic factor, K_v, accounts for the fact that the load is assumed by a tooth with some degree of impact and that the actual load subjected to the tooth is higher than the transmitted load alone. The value of K_v depends on the accuracy of tooth profile, the elastic properties of tooth, and the speed with which the teeth come into contact. AGMA Standard 2001-C95 gives recommended values for K_v based on the AGMA quality number, Q_v, and the pitch line velocity. Gears in typical machine design would have AGMA quality ratings of 5 through 7, which are for gears made by hobbing or shaping with average to good tooling. If the teeth are finish-ground or shaved to improve the accuracy of the tooth profile and spacing, quality numbers in the 8 - 11 range should be used. Under very special conditions where teeth of high precision are used in applications where there is little chance of developing external dynamic loads, higher quality numbers can be used. If the teeth are cut by form milling, factors lower than those found from Q_v = 5 should be used. Note that the quality 5 gears should not be used at pitch line speed above 2500 ft/min. Note that the dynamic factors are approximate.

Expected contact stress

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

where:

C_P = elastic coefficient that depends on the material of both the pinion and the gear. C_P = 2300 for two steel gears. The program automatically selects the appropriate value after the user specifies the materials.

Procedure for selecting materials for bending stress

$$\frac{K_R(SF)}{Y_N} s_t < s_{at}$$

where:

K_R = reliability factor

SF = factor of safety

Y_N = stress cycle factor for bending.

AGMA Standard 2001-C95 allows the determination of the life adjustment factor, Y_N, if the teeth of the gear being analyzed are expected to experience a number of cycles of loading much different from 10^7. Note that the general type of material is a factor for the lower number of cycles. For the higher number of cycles, a range is indicated by a shaded area.

Expected number of cycles of loading

$$N_c = (60)(L)(n)(q)$$

where:

L = design life in hours

n = rotational speed in rpm

q = number of load applications per revolution.

Procedure for selecting materials for contact stress

$$\frac{K_R(SF)}{Z_N} s_c < s_{ac}$$

where:

Z_N = pitting resistance stress cycle factor.

AGMA Standard 2001-C95 specifies the determination of the stress cycle factor, Z_N. If the teeth of the gear being analyzed are expected to experience a number of cycles of loading much different from 10^7, a factor should be used. The user specifies the desired life for the system in hours and the program computes the values for Y_N and Z_N.

After computing the values for allowable bending stress number, s_{at}, and for allowable contact stress number, s_{ac}, you should go to the data in AGMA Standard 2001-C95, to select a suitable material. Consider first whether the material should be steel, cast iron, bronze, or plastic. Then consult the related tables of data.

Diametral pitch

The most common pitch system used today is the diametral pitch system, the number of teeth per inch of pitch diameter. Its basic definition is

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

N_p, N_G = number of teeth of the pinion and the gear;

D_p, D_G = pitch diameter of the pinion and the gear.

Face width

The face width can be specified once the diametral pitch is chosen. Although a wide range of face widths is possible, the following limits are used for general machine drive gears:

$$\frac{8}{P_d} < F < \frac{16}{P_d}$$

Nominal value of $F = \frac{12}{P_d}$

Notice that $\frac{F}{D_p} < 2.00$ is recommended.

Rim thickness

The rim thickness factor, K_B, accounts for a rim that may be too thin. The basic analysis used to develop the Lewis equation ssumes that the gear tooth behaves as a cantilever attached to a perfectly rigid support structure at its base. If the rim of the gear is too thin, it can deform and cause the point of maximum stress to shift from the area of the gear-tooth fillet to a point within the rim.

The key geometry parameter is called the *backup ratio*, m_B, where

$$m_B = \frac{t_R}{h_t}$$

t_R = rim thickness;

h_t = whole depth of the gear tooth.

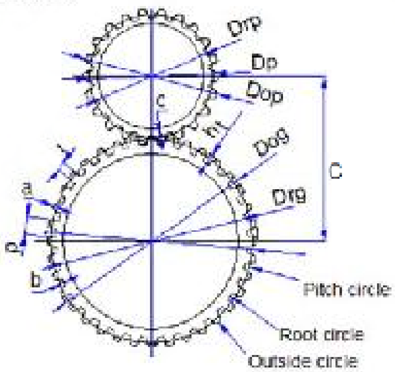
For m_B >1.2, the rim is sufficiently strong and stiff to support the tooth, and K_B =1.0.

For m_B <1.2, rim thickness factor determined:

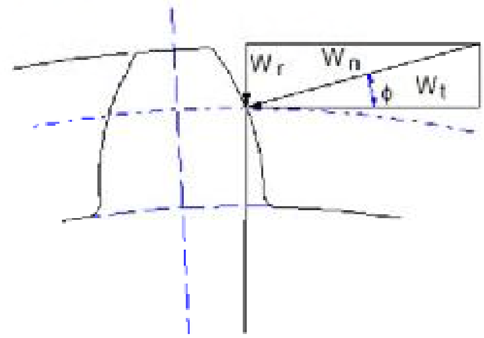
$$K_B = 1.6 \ln\left(\frac{2.242}{m_B}\right)$$

When a solid gear blank is used, input a large value (say tR > 1.0 inch) for rim thickness. The resulting value for $K_B = 1$.

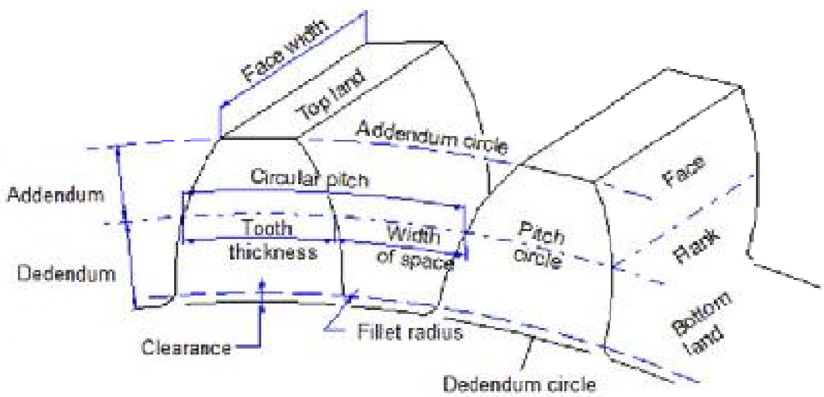
Gear pair features



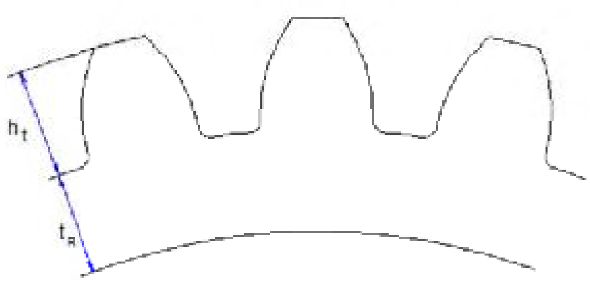
Forces on the spur gear tooth



Spur gear teeth features



Rim thickness and whole depth of the gear tooth



Helical gears

The teeth on helical gears are inclined at an angle with the axis, that angle being called the helix angle. If the gear were very wide, it would appear that the teeth wind around the gear blank in a continuous, helical path. However, practical considerations limit the width of the gears so that the teeth normally appear to be merely inclined with respect to the axis of the shaft.

Helical gear design

- Actual output speed (gear)

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

n_P = rotational speed of the pinion

VR= *Velocity Ratio* ; VR= mg = *gear ratio* for speed reducers

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

N_G, N_P = number of teeth on the gear, pinion

The spreadsheet computes the approximate number of gear teeth to produce the desired speed from $N_G = N_P \frac{n_{Gd}}{n_P}$ (n_{Gd} = desired output speed). But, of course, the number of teeth in any gear must be integer, and the actual value of N_G is selected by the designer.

Helical gear geometry

- Pitch diameter

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

- Outside diameter

$$D_o = \frac{N+2}{P_d}$$

- Addendum

$$a = \frac{1}{P_{dn}}$$

- Dedendum

$$b = \frac{1.25}{P_{dn}}$$

- Clearance

$$c = \frac{0.25}{P_{dn}}$$

- Root diameter

$$D_R = D - 2b$$

- Base circle diameter

$$D_b = D \cos \phi_t$$

where:

ϕ_t = transverse pressure angle

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

- Circular pitch

$$p = \frac{\pi D}{N}$$

- Normal circular pitch

$$p_n = p \cdot \cos \psi$$

- Diametral pitch

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

- Normal Diametral Pitch

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

- Axial pitch

$$P_x = \frac{p}{\tan \psi}$$

- Whole depth

$$h_t = a + b$$

- Working depth

$$h_k = a + a$$

- Tooth thickness

$$t = \frac{\pi}{2 P_{dn}}$$

- Center distance

$$C = \frac{D_G + D_P}{2}$$

Bending geometry factor, J, is dependent on the number of teeth on the gear and helix angle for which the geometry factor is desired and on the number of teeth in the mating gear. Values can be found from AGMA Standard 908-B89(R1995).

Pitting geometry factor, I, is dependent on the number of teeth of gear and helix angle for which geometry factor is desired and on the number of teeth in mating gear. Values can be found from AGMA Standard 908-B89(R1995)AGMA Standard 218.01.

Force and speed factors

- Pitch line speed

$$V_t = \frac{\pi D_P n_P}{12}$$

- Tangential force

$$W_t = \frac{33000 \cdot (P)}{V_t}$$

or

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

where:

P = transmitted power

- Radial force

$$W_r = W_t \tan \phi_t$$

- Normal force

$$W_n = \frac{W_t}{\cos \psi \cos \phi_n}$$

- Axial force

$$W_x = W_t \tan \psi$$

- Expected bending stress

$$S_t = \frac{W_t P_d}{F\ J} K_o K_s K_m K_B K_V$$

where:

K_o = overload factor

K_s = size factor

K_m = load-distribution factor

K_B = rim thickness factor

K_V = dynamic factor.

The AGMA indicates that the size factor can be taken to be 1.00 for most gears. But for gears with large-size teeth or large face width F, a value greater than 1.00 recommended. The program computes the size factor automatically.

The determination of load-distribution factor is based on many variables in the design of the gears themselves as well as in the shafts, bearings, housings, and the structure in which the gear drive is installed. Therefore, it is one of the most difficult factors to specify. Much analytical and experimental work is continuing of values for K_m. We will use the following equation for computing the value of the load-distribution factor:

$$K_m = 1.0 + C_{pf} + C_{ma}$$

where:

C_{pf} = pinion proportion factor is dependent on face width and pitch diameter

C_{ma} = mesh alignment factor.

The dynamic factor, K_V, accounts for the fact that the load is assumed by a tooth with some degree of impact and that the actual load subjected to the tooth is higher than the transmitted load alone. The value of K_V depends on the accuracy of tooth profile, the elastic properties of tooth, and the speed with which the teeth come into contact. AGMA Standard 2001-C95 gives recommended values for K_V based on the AGMA quality number, Q_V, and the pitch line velocity. Gears in typical machine design would have AGMA quality ratings of 5 through 7, which are for gears made by hobbing or shaping with average to good tooling. If the teeth are finish-ground or shaved to improve the accuracy of the tooth profile and spacing, quality numbers in the 8 - 11 range should be used. Under very special conditions where teeth of high precision are used in applications where there is little chance of developing external dynamic loads, higher quality numbers can be used. If the teeth are cut by form milling, factors lower than those found from Q_V = 5 should be used. Note that the quality 5 gears should not be used at pitch line speed above 2500 ft/min. Note that the dynamic factors are approximate.

Expected contact stress

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

where:

C_P = elastic coefficient that depends on the material of both the pinion and the gear. C_P = 2300 for two steel gears. The program automatically selects the appropriate value after the user specifies the materials.

Procedure for selecting materials for bending stress

$$\frac{K_R(SF)}{Y_N} S_t < S_{at}$$

where:

K_R = reliability factor

SF = factor of safety

Y_N = stress cycle factor for bending.

AGMA Standard 2001-C95 allows the determination of the life adjustment factor, Y_N, if the teeth of the gear being analyzed are expected to experience a number of cycles of loading much different from 10^7. Note that the general type of material is a factor for the lower number of cycles. For the higher number of cycles, a range is indicated by a shaded area.

Expected number of cycles of loading

$$N_c = (60)(L)(n)(q)$$

where:

L = design life in hours

n = rotational speed in rpm

q = number of load applications per revolution.

Procedure for selecting materials for contact stress

$$\frac{K_R(SF)}{Z_N} S_c < S_{ac}$$

where:

Z_N = pitting resistance stress cycle factor.

AGMA Standard 2001-C95 specifies the determination of the stress cycle factor, Z_N. If the teeth of the gear being analyzed are expected to experience a number of cycles of loading much different from 10^7, a factor should be used. The user specifies the desired life for the system in hours and the program computes the values for Y_N and Z_N.

After computing the values for allowable bending stress number, S_{at}, and for allowable contact stress number, S_{ac}, you should go to the data in AGMA Standard 2001-C95, to select a suitable material. Consider first whether the material should be steel, cast iron, bronze, or plastic. Then consult the related tables of data.

Normal diametral pitch

The most common pitch system used today is the diametral pitch system. Normal diametral pitch is the equivalent diametral pitch in the plane normal to the teeth:

$$P_{dn} = \frac{P_d}{\cos\psi}$$

where:

P_d = diametral pitch

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

N_p, N_G = number of teeth on the pinion and the gear;

D_p, D_G = pitch diameter of the pinion and the gear.

Face width

Nominal face width

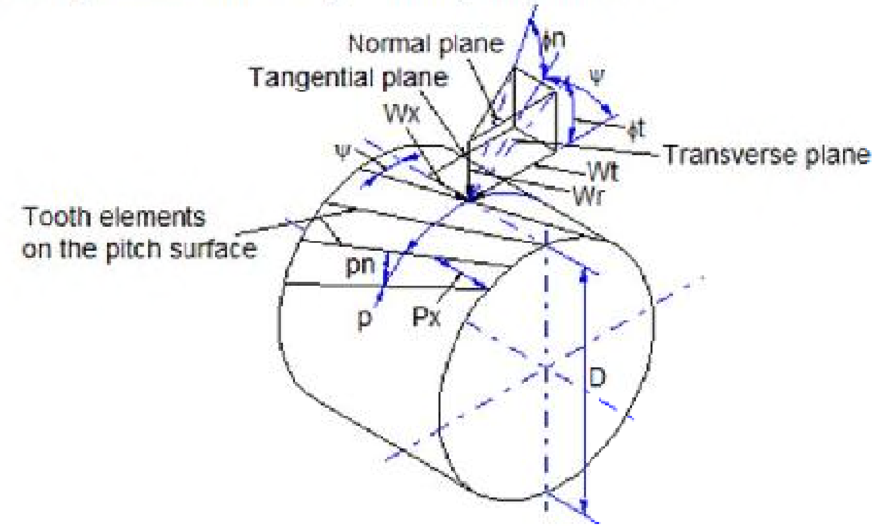
$$F \geq 2 \cdot (P_x)$$

where:

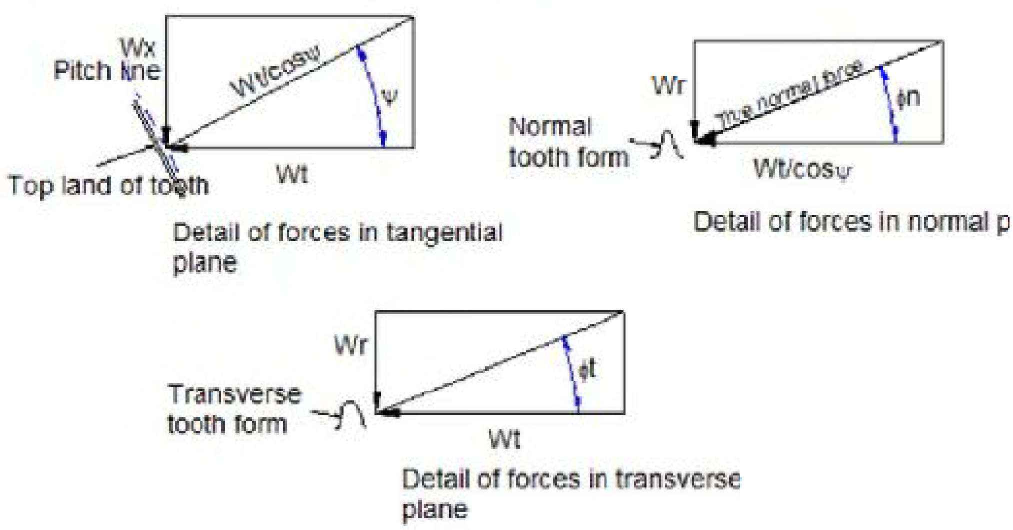
P_x = axial pitch.

If the number of axial pitches in the face width is less than 2.0 there won't be full helical action. The program computes a suggested value of F = 2.0(Px) and calls for a user-supplied value. A convenient size greater than the suggested value should be specified.

Perspective view of geometry and forces



Forces in the helical gear tooth



Straight Bevel Gearing Design

Geometrical features of straight bevel gears:

- Gear ratio

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

- Pitch diameters

pinion

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

gear

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

- Pitch cone angles

pinion

$$\gamma = \tan^{-1}\left(\frac{N_p}{N_G}\right)$$

gear

$$\Gamma = \tan^{-1}\left(\frac{N_G}{Np}\right)$$

- Outer cone distance

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

- Nominal face width

$$F_{nom} = 0.3 \cdot A_o$$

- Maximum face width

$$F_{max} = \frac{A_o}{3} \quad \text{or} \quad F_{max} = \frac{10}{P_d} \quad \text{(whichever is less)}$$

- Mean cone distance

$$A_m = A_o - 0.5 \cdot F$$

Note: A_m is defined for the gear, also called A_{mG} .

- Mean circular pitch

$$p_m = \frac{\pi \cdot A_m}{P_d \cdot A_o}$$

- Mean working depth

$$h = \frac{2 \cdot A_m}{P_d \cdot A_o}$$

- Clearance

$$c = 0.125 \cdot h$$

- Mean whole depth

$$h_m = h + c$$

- Mean addendum factor

$$c_1 = 0.21 + \frac{0.29}{(m_G)^2}$$

- Gear mean addendum

$$a_G = c_1 \cdot h$$

- Pinion mean addendum

$$a_p = h - a_G$$

- Gear mean dedendum

$$b_G = h_m - a_G$$

- Pinion mean addendum

$$b_p = h_m - a_p$$

- Gear dedendum angle

$$\delta_G = \tan^{-1}\left(\frac{b_G}{A_{mG}}\right)$$

- Pinion dedendum angle

$$\delta_p = \tan^{-1}\left(\frac{b_p}{A_{mG}}\right)$$

- Gear outer addendum

$$a_{oG} = a_G + 0.5 \cdot F \cdot \tan \delta_p$$

- Pinion outer addendum

$$a_{op} = a_p + 0.5 \cdot F \cdot \tan \delta_G$$

- Gear outside diameter

$$D_o = D + 2 \cdot a_{oG} \cdot \cos \Gamma$$

- Pinion outside diameter

$$d_o = d + 2 \cdot a_{op} \cdot \cos \gamma$$

Because of the conical shape of bevel gears and because of the involute-tooth form, a three-component set of forces acts on bevel gear teeth. Using notation similar to that for helical gears, we will compute the tangential force, W_t ; radial force, W_r ; and axial force, W_x . It is assumed that the three forces act concurrently at the midface of the teeth and on the pitch cone. Also the actual of the resultant force is a little displaced from the middle, no serious error results.

The tangential force acts tangential to the pitch cone and is the force that produces the torque on the pinion and the gear. The torque can be computed from the known power transmitted and the rotational speed:

$$T = \frac{63000 \cdot P}{n}$$

Then, using the pinion, for example, the transmitted load is

$$W_t = \frac{T}{r_m}$$

where:

r_m = mean radius of the pinion

$$r_m = \frac{d}{2} - \frac{F \cdot \sin \gamma}{2}$$

Remember that the pitch diameter, d, is measured to the pitch line of the tooth at its large end.

The radial load acts towards the center of pinion, perpendicular to its axis, causing bending of the pinion shaft. Thus,

$$W_{rp} = W_t \cdot \tan \phi \cos \gamma$$

The axial load acts parallel to the axis of the pinion, tending to push it away from the mating. It causes a thrust load on the shaft bearings. It also produces a bending moment on the shaft because it acts at the distance from the axis equal to the mean radius of the gear. Thus,

$$W_{xp} = W_t \cdot \tan \phi \sin \gamma$$

The stress analysis for bevel gear teeth is similar to that already presented for spur and helical gear teeth. The maximum bending stress occurs at the root of the tooth in the fillet. This stress can be computed

$$S_t = \frac{W_t \cdot P_d}{F \cdot J} \cdot \frac{K_o \cdot K_s \cdot K_m}{K_v}$$

where:

K_o = overload factor;

K_s = size factor

K_m = load-distribution factor

K_v = dynamic factor.

Factors affecting the dynamic factor include the accuracy of manufacture of gear teeth (quality number Q); the pitch line velocity, V_t ; the tooth load; and the stiffness of teeth. AGMA Standard 2003-A86 recommends the following procedure for computing K_v for bending strength calculation

$$K_v = \left[\frac{K_z}{K_z + \sqrt{V_t}}\right]^u$$

where:

$$u = \frac{8}{2^{0.5Q}} - S_{at}\left[\frac{125}{E_p + E_G}\right]$$

$$K_z = 85 - 10 \cdot u$$

Usually as a design decision, use two Grade 1 steel gears that are through-hardened at 300 HB with 36000 psi. The modulus of elasticity for both gears is 30×10^6 psi.

Bending geometry factor, J, is dependent on the number of teeth of gear for which geometry factor is desired and on the number of teeth in mating gear. Values can be found from AGMA Standard 6010-E88.

The approach to design of bevel gears for pitting resistance is similar to that for spur gears. The failure mode is fatigue of the surface of the teeth under the influence of the contact stress between the mating gears.

The contact stress, called the Hertz stress, S_c , can be computed from

$$S_c = C_p C_b \sqrt{\frac{W_t}{F \cdot d \cdot I} \cdot \frac{K_o \cdot K_m}{K_v}}$$

where:

C_p = elastic coefficient;

Using C_b = 0.634 allows the use of the same allowable contact stress as for spur and helical gears.

Pitting geometry factor, I , is dependent on the number of teeth of gear and helix angle for which geometry factor is desired and on the number of teeth in mating gear. Values can be found from AGMA Standard 2003-A86.

Procedure for selecting materials for bending stress

$$\frac{K_R(SF)}{Y_N} S_t < S_{at}$$

where:

K_R = reliability factor

SF = factor of safety

Y_N = stress cycle factor.

AGMA Standard 2001-C95 allows the determinations of the life adjustment factor, Y_N , if the teeth of the gear being analyzed are expected to experience a number of cycles of loading much different from 10^7 . Note that the general type of material is a factor for the lower number of cycles. For the higher number of cycles, a range is indicated by a shaded area.

Expected number of cycles of loading

$$N_c = (60)(L)(n)(q)$$

where:

L = design life in hours

n = rotational speed in rpm

q = number of load applications per revolution.

Procedure for selecting materials for contact stress

$$\frac{K_R(SF)}{Z_N}s_c < s_{ac}$$

where:

Z_N = pitting resistance stress cycle factor.

AGMA Standard 2001-C95 allows the determinations of the life adjustment factor, Z_N, if the teeth of the gear being analyzed are expected to experience a number of cycles of loading much different from 10^7. Note that the general type of material is a factor for the lower number of cycles. For the higher number of cycles, a range is indicated by a shaded area.

After computing the values for allowable bending stress number, s_{at}, and for allowable contact stress number, s_{ac}, you should go to the data in AGMA Standard 2001-C95, to select a suitable material. Consider first whether the material should be steel, cast iron, bronze, or plastic. Then consult the related tables of data.

Diametral pitch

The most common pitch system used today is the diametral pitch system, the number of teeth per inch of pitch diameter. Its basic definition is

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

N_p, N_G = number of teeth of the pinion and the gear;

D_p, D_G = pitch diameter of the pinion and the gear.

Number of pinion teeth

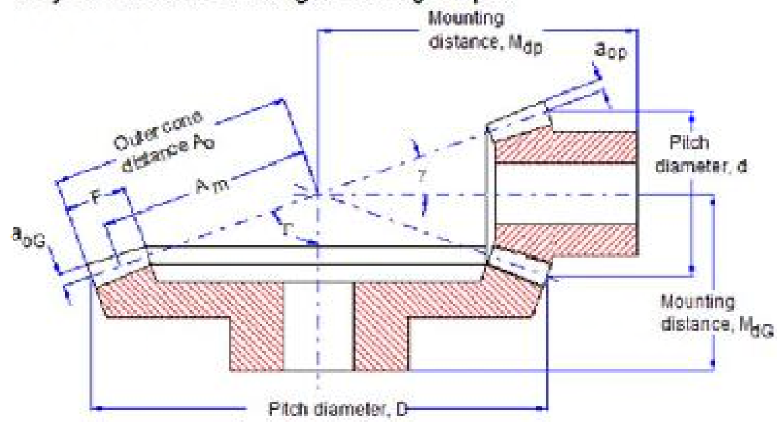
For certain combinations of number of teeth in a gear pair, there is interference between the tip of the teeth on the pinion and the fillet root of the teeth on the gear. Obviously this cannot be tolerated because the gears simply will not mesh.

It is the designer's responsibility to ensure that interference does not occur in given application. The surest way to do this is to control the minimum number of teeth in the pinion.

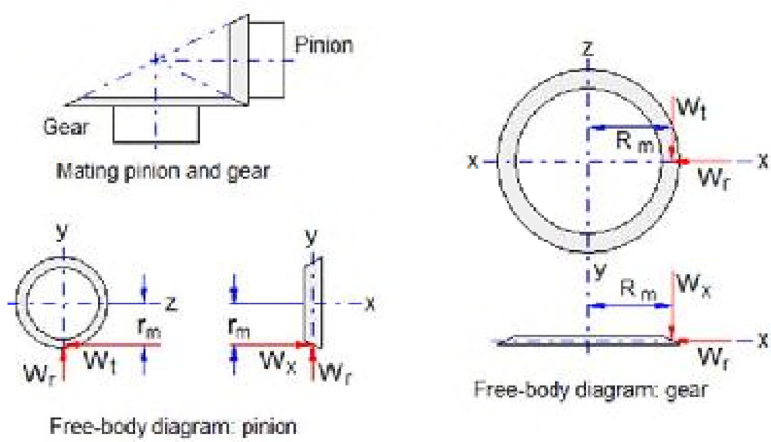
The minimum number of teeth for straight bevel gears is typically 13. The Gleason Works of Rochester, N.Y., has done an excellent job of standardizing the designs of these kinds of gears. The various Gleason systems have the amount of addendum for the gear and the pinion worked out so as to avoid undercut with low numbers of teeth and balance the strength of gear and pinion teeth. In each case, though, there is a limit to how far the system will go. Use the following values for the minimum number of gear teeth (for pressure angle $20°$).

Number of pinion teeth	Min. number of gear teeth
13	31
14	20
15	17
16	16

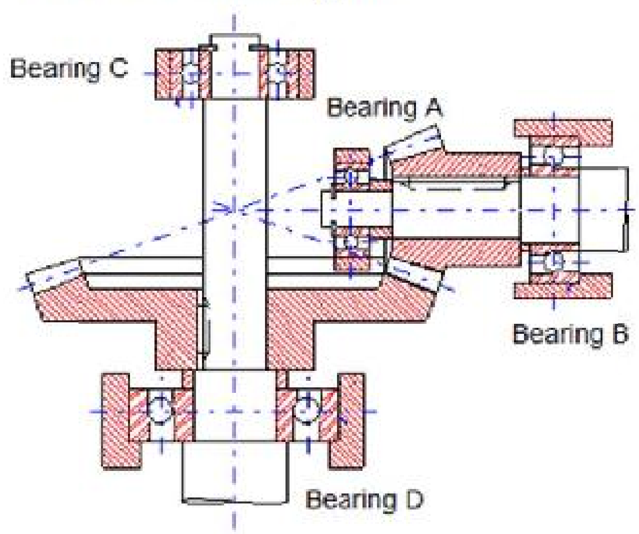
Key dimensions of straight bevel gear pair



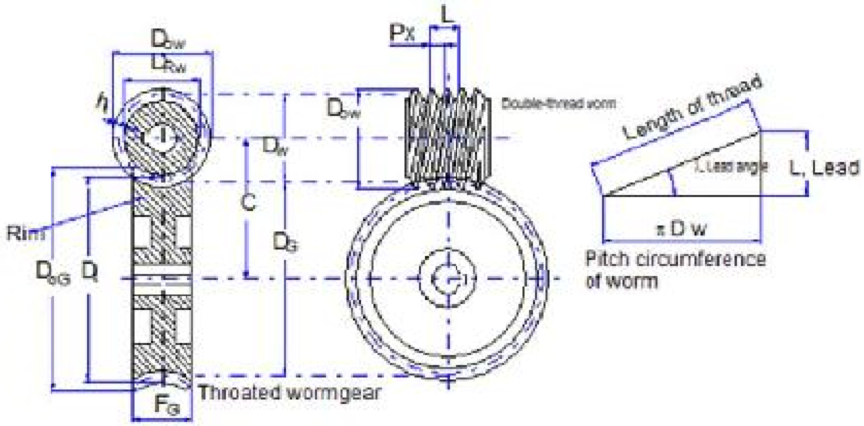
Forces on bevel gears

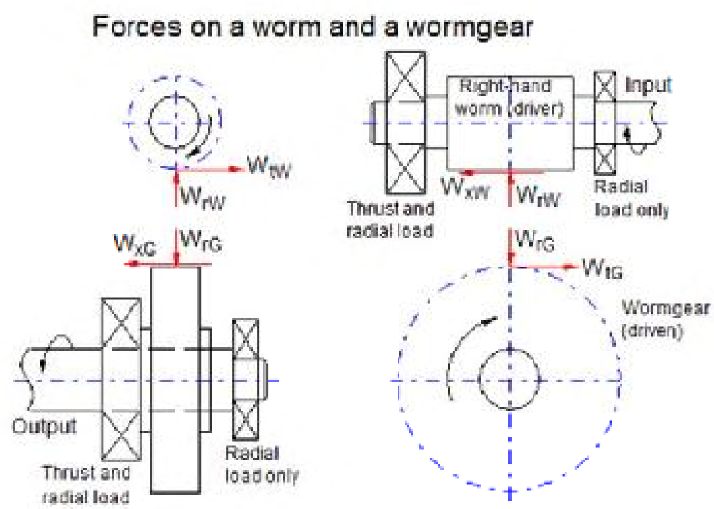


Straddle mounted gears



Wormgearing Design





$$L = N_W P_X$$

$$\tan\lambda = \frac{L}{\pi \cdot D_W}$$

the pitch line speed is the linear velocity of a point on the pitch line for the worm or the wormgear. For the worm having a pitch diameter D_W in, rotating at n_W rpm,

$$V_{tw} = \frac{\pi \cdot D_W \cdot n_W}{12} \text{ ft/min}$$

For the wormgear having a pitch diameter D_G in, rotating at n_G rpm,

$$V_{tG} = \frac{\pi \cdot D_G \cdot n_G}{12} \text{ ft/min}$$

Note that these two values for pitch line speed are not equal.

It is most convenient to calculate the velocity ratio of a worm and wormgear set from the ratio of the input rotational speed to the output rotational speed:

$$VR = \frac{\text{speed of worm}}{\text{speed of gear}} = \frac{n_W}{n_G} = \frac{N_G}{N_W}$$

The diameter of the worm affects the lead angle, which in turn affects the efficiency of the set. For this reason, small diameters are desirable. But for practical reasons and proper proportion with respect to the wormgear, it is recommended that the worm diameter be approximately $C^{0.875}/2.2$, where C is the center distance between the worm and the wormgear. Variation of about 30% is allowed. Thus, the worm diameter should fall in the range

$$1.6 < \frac{C^{0.875}}{D_W} < 3.0$$

- Addendum

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

- Whole depth

$$h_t = \frac{2.157}{P_d}$$

- Working depth

$$h_k = \frac{2}{P_d}$$

- Dedendum

$$b = \frac{1.157}{P_d}$$

- Root diameter of worm

$$D_{rW} = D_W - 2b$$

- Outside diameter of worm

$$D_{oW} = D_W + 2a$$

- Root diameter of gear

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

- Throat diameter of gear

$$D_t = D_G + 2a$$

- The recommended face width for the wormgear is

$$F_G = \left(D_{oW}^2 - D_W^2\right)^{1/2},$$

For maximum load sharing, the worm face length should extend to at least the point where the outside diameter of the worm intersects the throat diameter of the wormgear. This length is

$$F_W = 2\left[\left(\frac{D_t}{2}\right)^2 - \left(\frac{D_G}{2} - a\right)^2\right]^{1/2}$$

In most design problems for wormgear drives, the output torque and the rotating speed of the output shaft will be known from the requirements of the driven machine. Torque and speed are related to the output power by

$$T_o = \frac{63000 \cdot P_o}{n_G}$$

Tangential force on a wormgear

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

Axial force on a wormgear

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

where:

μ = coefficient of friction.

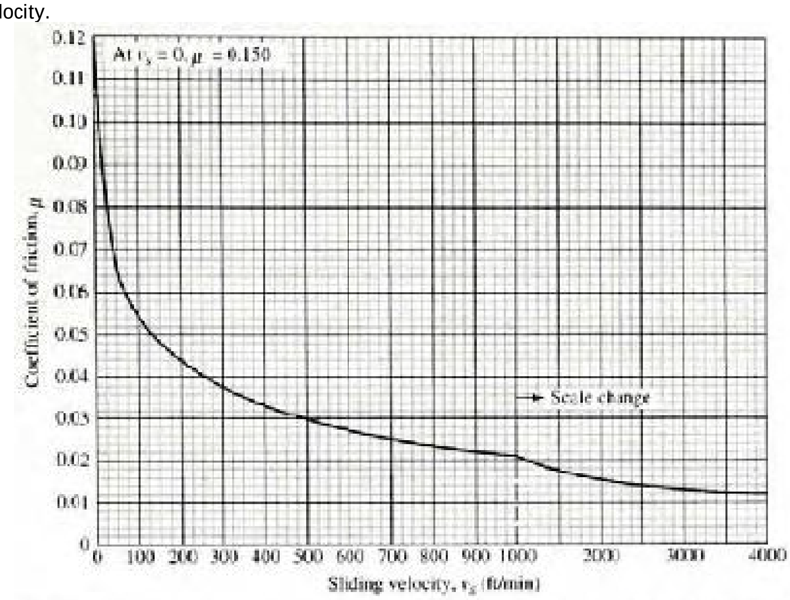
the sliding velocity is

$$V_s = \frac{V_{tG}}{\sin \lambda}$$

Based on the pitch line speed of the worm,

$$V_s = \frac{V_{tw}}{\cos \lambda}$$

The AGMA recommends the following formulas to estimate the coefficient of friction for a hardened steel worm (58 HRC minimum), smoothly ground, or polished, or rolled, or with an equivalent finish, operating on a bronze wormgear. The choice of formula depends on the sliding velocity.



choice of formula depends on the sliding velocity. *Note:* v_s must be in ft/min in the formulas: 1.0 ft/min = 0.0051 m/s.

- Static Condition, $V_s = 0$

$$\mu = 0.150$$

- Low Speed, $V_s < 10$ ft/min

$$\mu = 0.124 e^{\left(-0.07 V_s^{0.645}\right)}$$

- Higher Speed, $V_s > 10$ ft/min

$$\mu = 0.103 e^{\left(-0.11 V_s^{0.45}\right)} + 0.012$$

Radial force on a wormgear

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

Forces on a worm

- Tangential force on a worm

$$W_{tw} = W_{xG}$$

- Axial force on a wormgear

$$W_{xw} = W_{tG}$$

- Radial force on a wormgear

$$W_{rw} = W_{rG}$$

The friction force, W_f , acts parallel to the face of the worm threads and the gear teeth and depends on the tangential force on the gear, the coefficient of friction, and the geometry of the teeth:

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

The AGMA, in its Standard 6034-A87, does not include a method of analyzing wormgears for strength. Only the wormgear teeth are analyzed because the worm threads are inherently stronger and are typically made from a stronger material.
The stress in the gear teeth can be computed from

$$\sigma = \frac{W_d}{y \cdot F_G \cdot p_n}$$

where:

W_d = dynamic load on the gear teeth

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

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

y = Lewis form factor

Only one value is given for the Lewis form factor for a given pressure angle because the actual value is very difficult to calculate precisely and does not vary much with the number of teeth. The actual face width should be used, up to the limit of two-thirds of the pitch diameter of the worm.

ϕ_n	y
14.5	0.100
20	0.125
25	0.150
30	0.175

p_n = normal circular pitch

$$p = \frac{\pi \cdot \cos\lambda}{P_d}$$

The computed value of tooth bending stress from Equation (10-25) can be compared with the fatigue strength of the material of the gear. For manganese gear bronze, use a fatigue strength of 17 000 psi; for phosphor gear bronze, use 24 000 psi. For cast iron, use approximately 0.35 times the ultimate strength, unless specific data are available for fatigue strength.

AGMA Standard 6034-A87 gives a method for rating the surface durability of hardened steel worms operating with bronze gears. The ratings are based on the ability of the gears to operate without significant damage from pitting or wear.
The procedure calls for the calculation of a rated tangential load, W_{tR} , from

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

where:

C_s = materials factor;

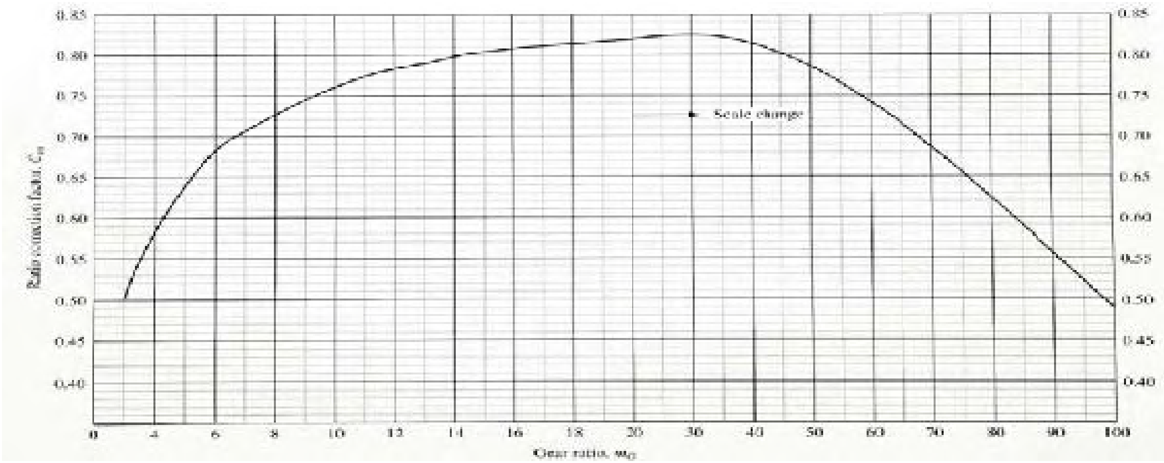
F_e = effective face width, in inches. Use the actual face width of the wormgear up to a maximum of $0.67 \cdot D_W$;

C_m = ratio correction factor;

C_V = velocity factor.

Use the actual face width, F, of the wormgear as F_e if $F < 0.667 \cdot (D_W)$. For larger face widths, use $F_e = 0.667 \cdot (D_W)$, because the excess width is not effective.

The ratio correction factor, C_m, can be computed from the following figure and formulas.



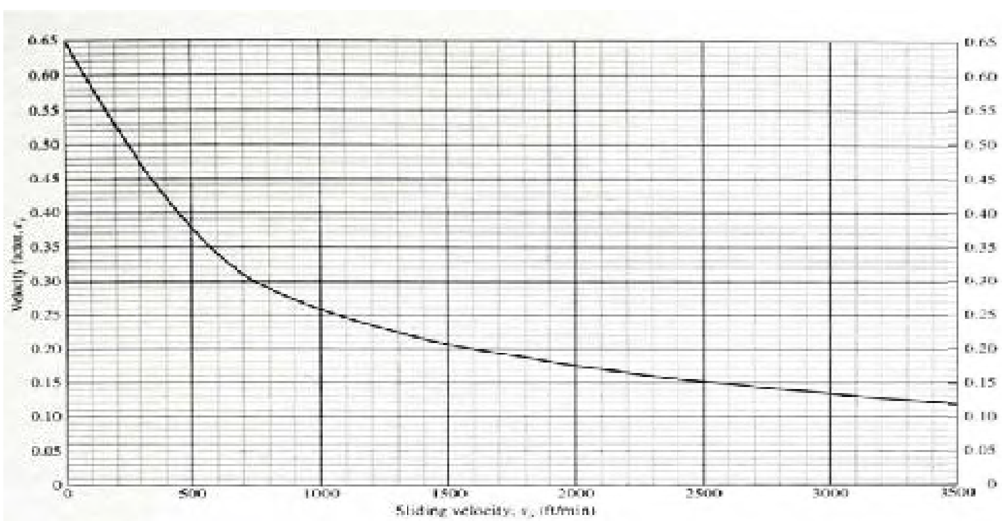
- For Gear Ratios, m_G, from 6 to 20

$$C_m = 0.020\left(-m_G^2 + 40 \cdot m_G - 76\right)^{0.5} + 0.46$$

- For Gear Ratios, m_G, from 20 to 76

$$C_m = 0.0107\left(-m_G^2 + 56 \cdot m_G + 5145\right)^{0.5}$$

The velocity factor depends on the sliding velocity, V_s. Values for C_V can be computed from the following figure and formulas.



- For V_s from 0 to 700 ft/min

$$C_V = 0.659 \cdot e^{(-0.001 \cdot V_s)}$$

- For V_s from 700 to 3000 ft/min

$$C_V = 13.31 \cdot e^{(-0.571)}$$

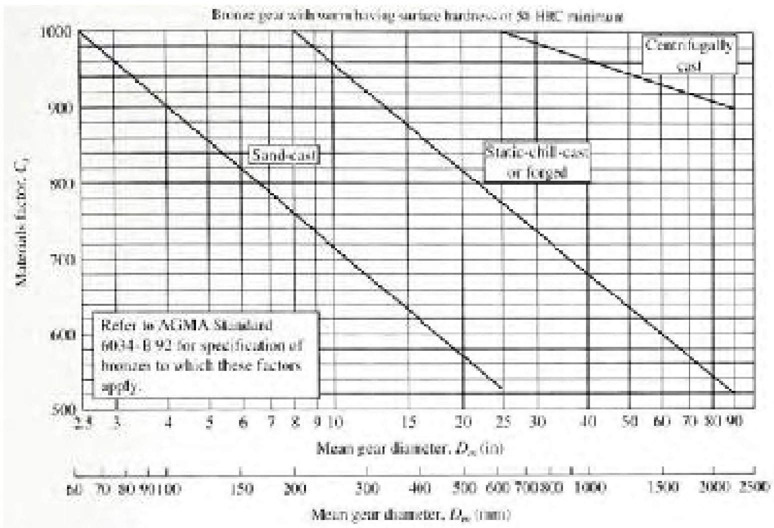
- For V_S > 3000 ft/min

$$C_V = 65.52 \cdot e^{(-0.774)}$$

When you are analyzing a given wormgear set, the value of the rated tangential load, W_{tR}, must be greater than the actual tangential load, W_{tG} for satisfactory life.

Method of casting the bronze

AGMA provides a procedure for rating the surface durability of wormgear drives. The analysis is valid only for a hardened steel worm (58 HRC minimum) operating with gear bronzes specified in AGMA Standard 6034-A87. The classes of bronzes typically used are tin bronze, phosphor bronze, manganese bronze, and aluminum bronze. The materials factor, C_S, is dependent on the method of casting the bronze. The values for C_S can be computed from the following formulas.



- Sand-cast Bronzes:
  For D_G > 2.5 in,

$$C_S = 1189.636 - 476.545 \cdot \log_{10}(D_G)$$

  For D_G < 2.5 in,

$$C_S = 1000.$$

- Static-Chill-cast or Forged Bronzes:
  For D_G > 8.0 in,

$$C_S = 1411.651 - 455.825 \cdot \log_{10}(D_G)$$

  For D_G < 8.0 in,

$$C_S = 1000.$$

- Centrifugally Cast Bronzes:
  For D_G > 25 in,

$$C_S = 1251.291 - 179.750 \cdot \log_{10}(D_G)$$

  For D_G < 8.0 in,

$$C_S = 1000.$$

Normal pressure angle

Most commercially available wormgears are made with pressure angles of $14\frac{1}{2}^\circ$, 20°, 25° or 30°.

$$\tan\phi_n = \tan\phi_t \cdot \cos\lambda$$

Diametral pitch

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

where:

D_G = pitch diameter of the gear

N_G = number of teeth on the gear.

Some wormgears are made according to the circular pitch convention. But, as noted with spur gears, commercially available wormgear sets are usually made to a diametral pitch convention with the following pitches readily available: 48, 32, 24, 16, 12, 10, 8, 6, 5, 4, and 3. The diametral pitch is defined for the gear as

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

The conversion from diametral pitch to circular pitch can be made from the following equation:

$$P_d \cdot p = \pi$$

Output power

$$\text{Torque} = \text{power/rotational speed} = \frac{P}{n}$$

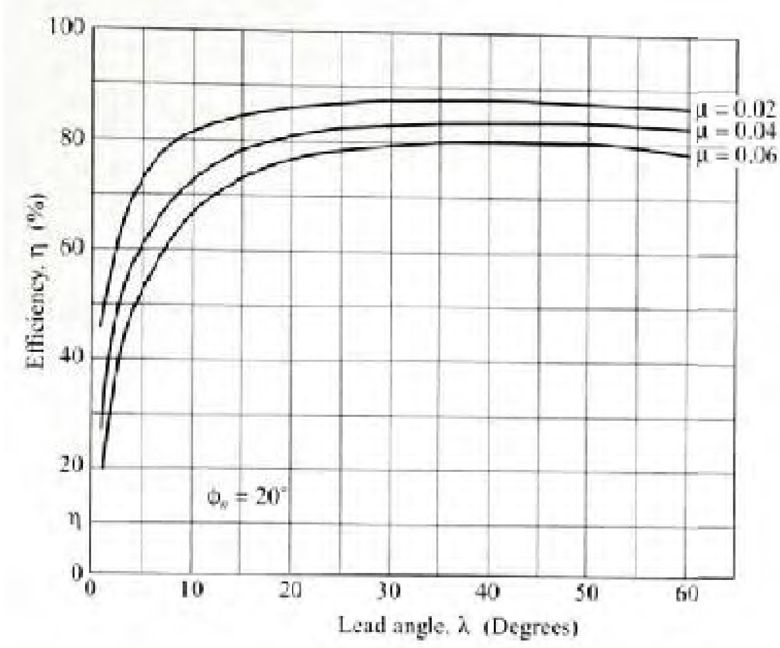
$$P_L = \frac{V_s \cdot W_f}{33000}$$

The input power is the sum of the output power and the power loss due to friction:

$$P_i = P_o + P_L$$

Efficiency is defined as the ratio of the output power to the input power:

$$\eta = \frac{P_o}{P_i}$$


100
80
60
40
20
η
0
0
10
20
30
40
50
60
Efficiency, η (%)
Lead angle, λ (Degrees)
μ = 0.02
μ = 0.04
μ = 0.06
φn = 20°