Appendix (A)

Examinations Questions of Previous Academic Years

University of Technology
Department of Machines and Equipment Engineering
Final Year Examination 2013/2014

Subject: DesignII
Division: General
Examiner(s): Design group

Year: Fourth year
Exam Time: 3 Hrs.
Date: 29/5/ 2014

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

Q1. Figure.1, shows the portable concrete mixture, 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 no.(1), give at least one another idea with a simple, clear sketch for each sub-system of this machine. **(15 marks)**
3- Draw the system scheme by using your new ideas in no.(2) above. **(10 marks)**

Figure. 1

Q2. Figure.2, shows a heavily loaded coal conveyor to be driven by a gasoline engine through a chain drive. The input speed will be 900 rpm, and the desired output speed is 230 to 240 rpm. The conveyor requires 11.2 kW.

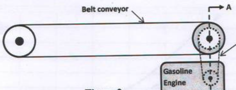


Figure.2

Requirements:

1- Design the chain drive to satisfy the given operating conditions. **(20 marks)**
2- Draw section (A-A) showing all fixation 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	
Couplings and connectors-axial	
Loading and unloading mechanism	

(10 marks)

University of Technology
Department of Machines and Equipment Engineering
Final Year Examination 2013/2014

Subject: DesignII
Division: General
Examiner(s): Design group

Year: Fourth year
Exam Time: 3 Hrs.
Date: 29/5/2014

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

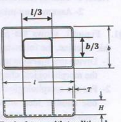
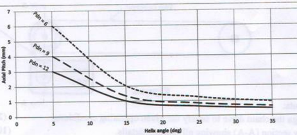


Figure. 3 Typical tray with traditional hollow rectangular shape.

(25 marks)

(B) The helical gear will be designed within a 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:

$(16^o > \Psi > 9^o)$

$(12 > P_{dn} > 9)$

$(4 > P_X > 2)$

(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)

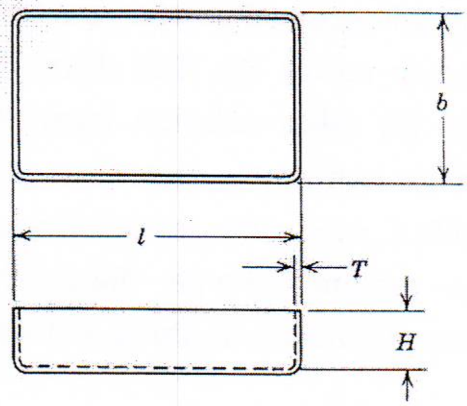


Figure. (1a) Typical tray with traditional rectangular shape.

(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:

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

$500\ mm \leq L \leq 750\ mm$

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

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

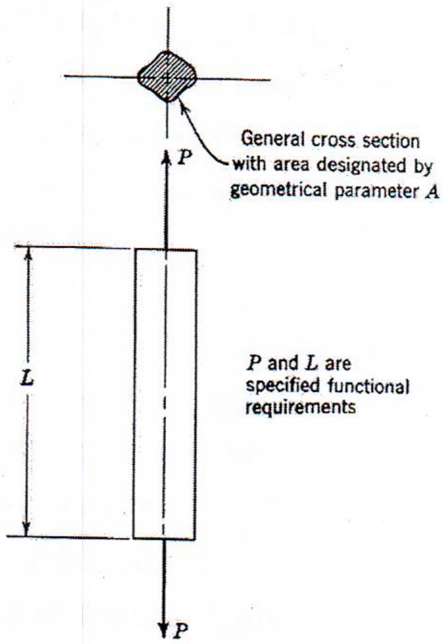


Figure.(1b) simple tensile bar with uniformly distributed specified axial load (P)

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)

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

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

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.

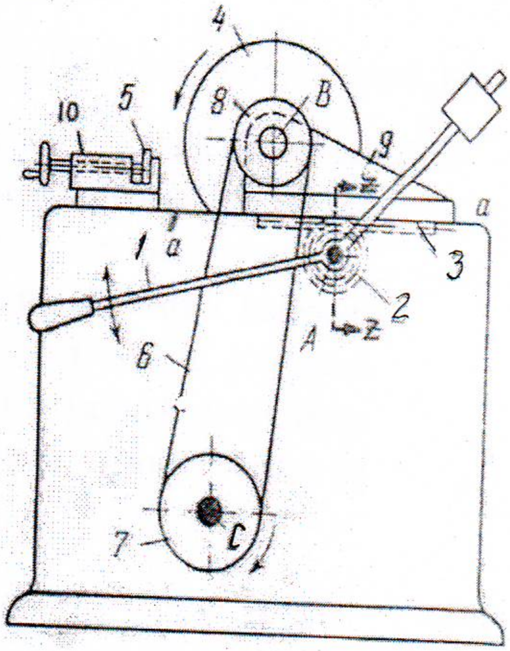


Figure.2 system scheme for the industrial saw

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)

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**

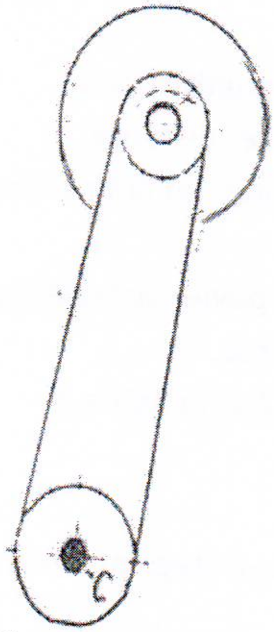
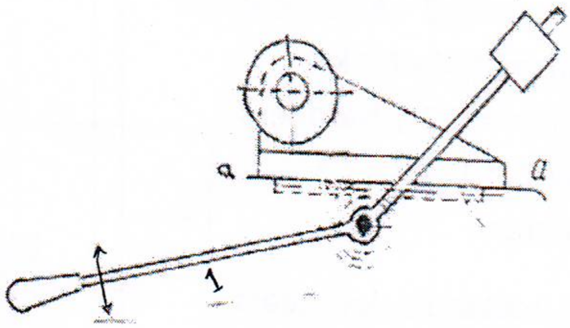
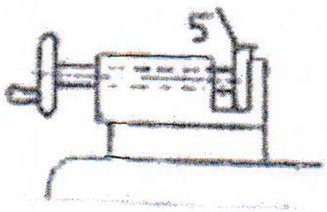
Alternatives / Sub-systems	1	2
Power transmission from motor to saw No.4	A1 	
Mechanisms for moving the saw toward the specimen	B1 	B2
Mechanisms for clamping specimen	C1 	C2

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

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.

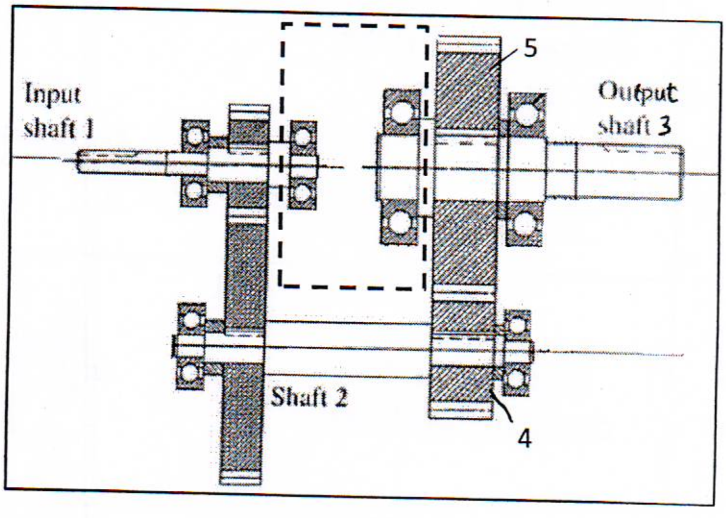


Figure.3

(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)

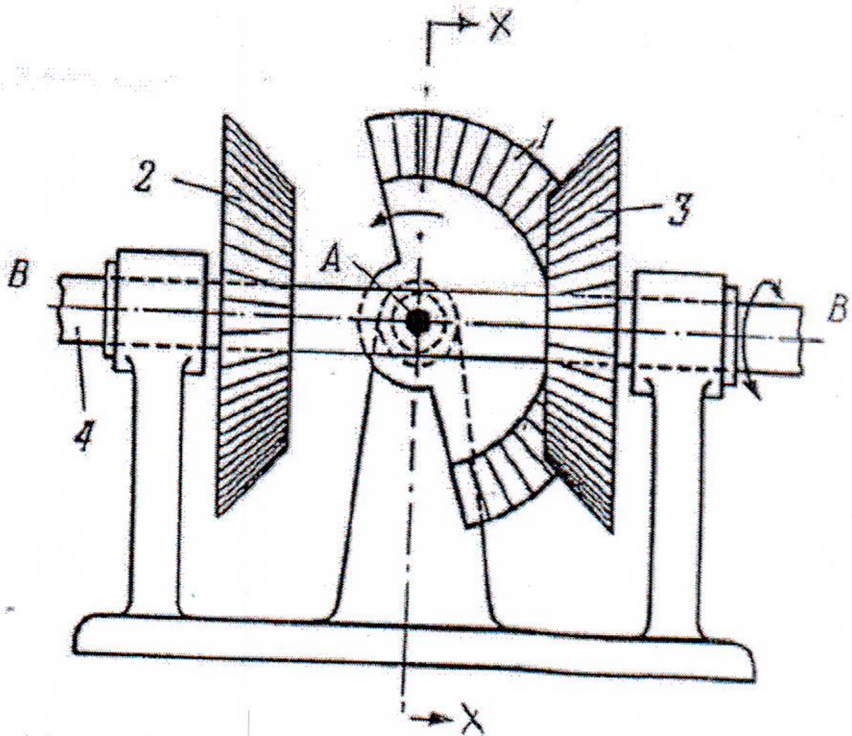


Figure.4

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

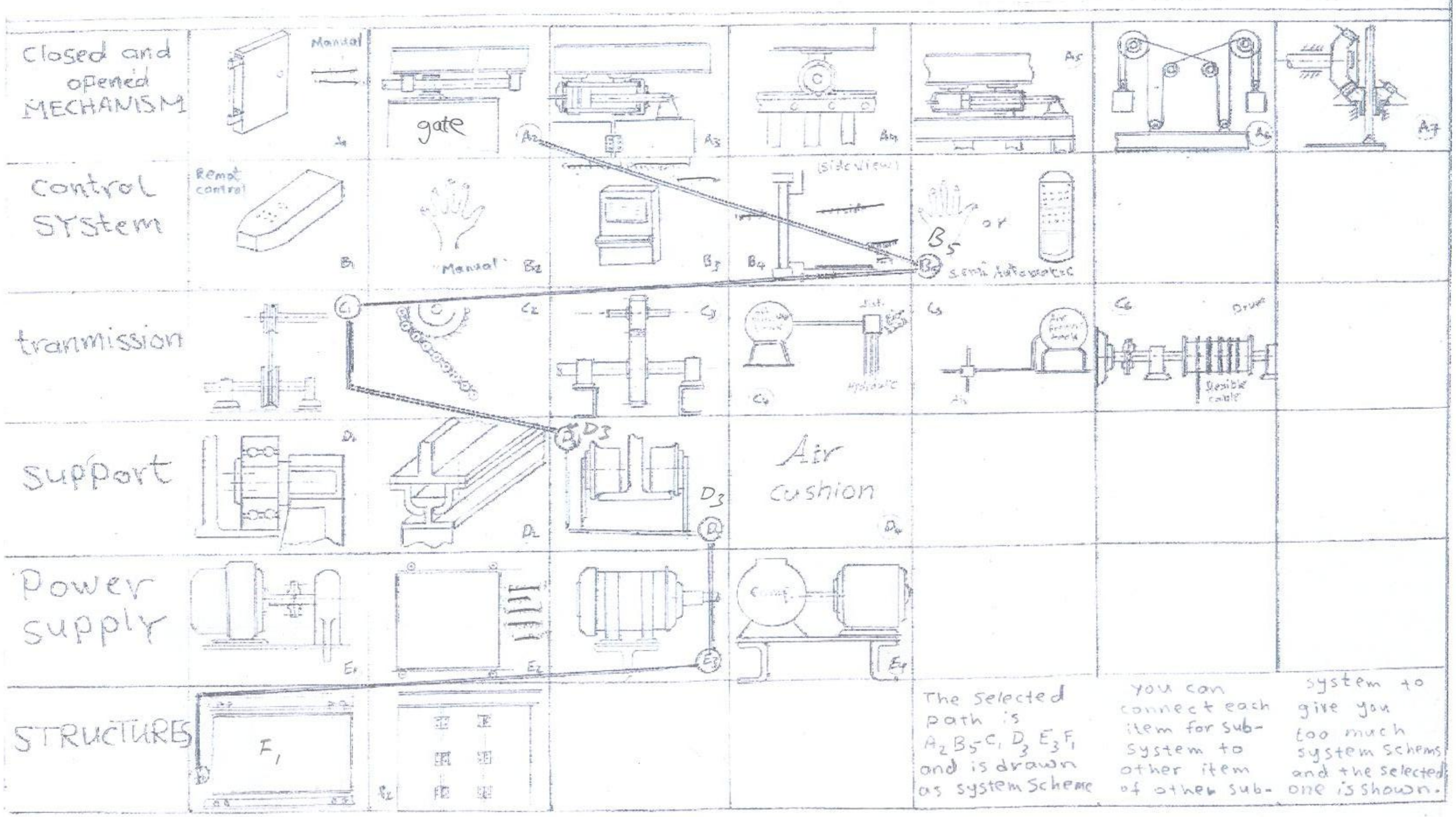


Fig.1.

الشكل يمثل احد الاختيارات للنظام من قبل احد الطلاب

ملاحظة : في حالة عدم وضوح اي شكل من الممكن ان تقوم بتصميم مناسب

لكي تصل الى الاختيار النهائي حسب ماتراه .

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	=		mm²
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.

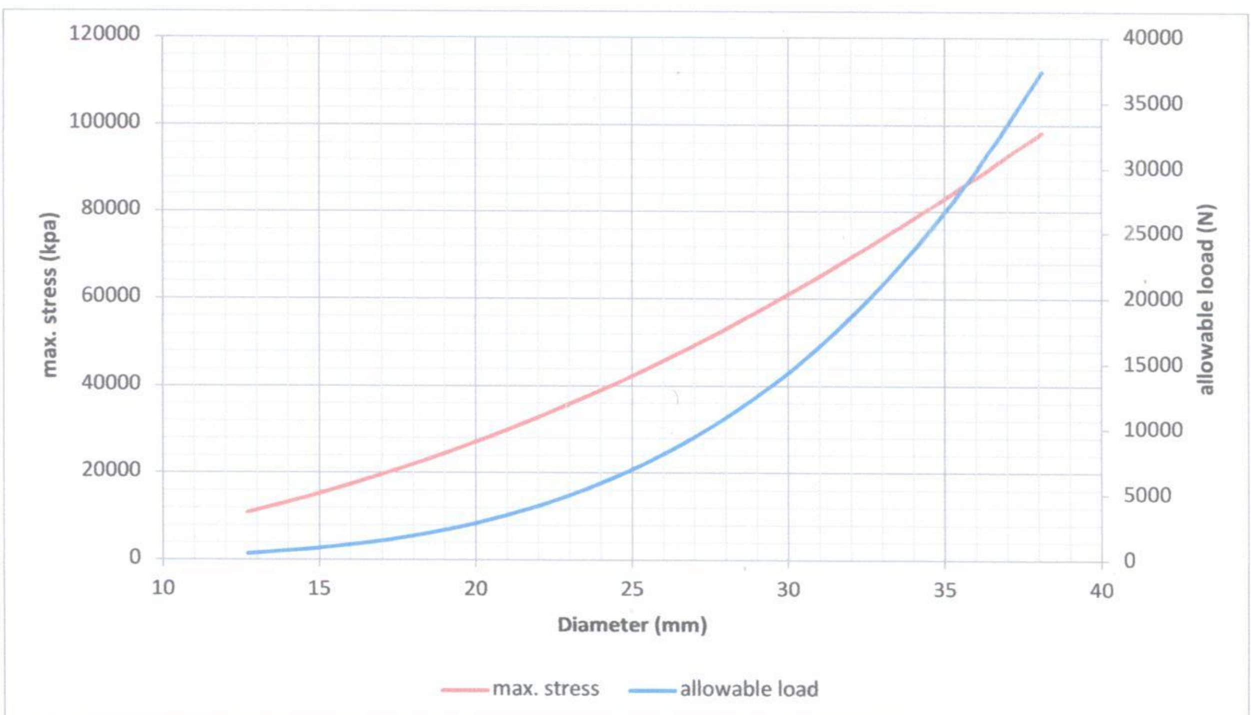
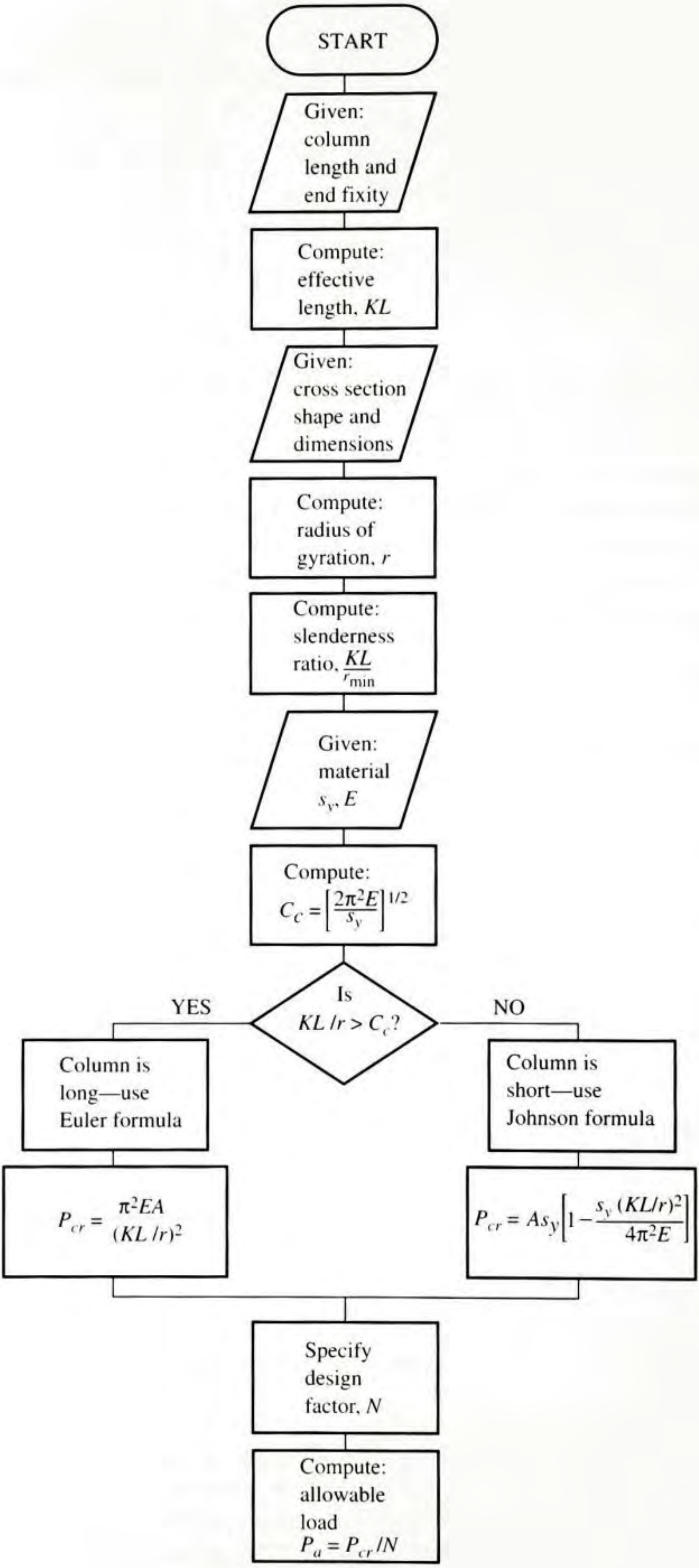


Fig. 2.


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 $P_a = P_{cr}/N$

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).

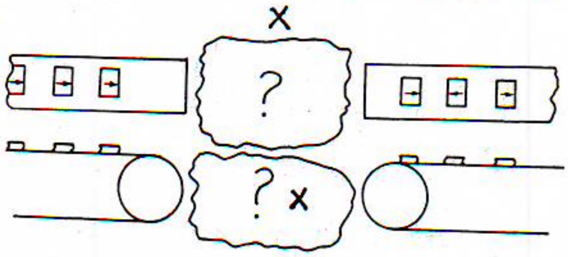


Fig (1)

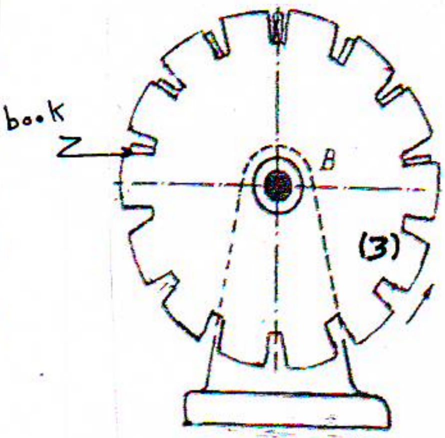


Fig (2)

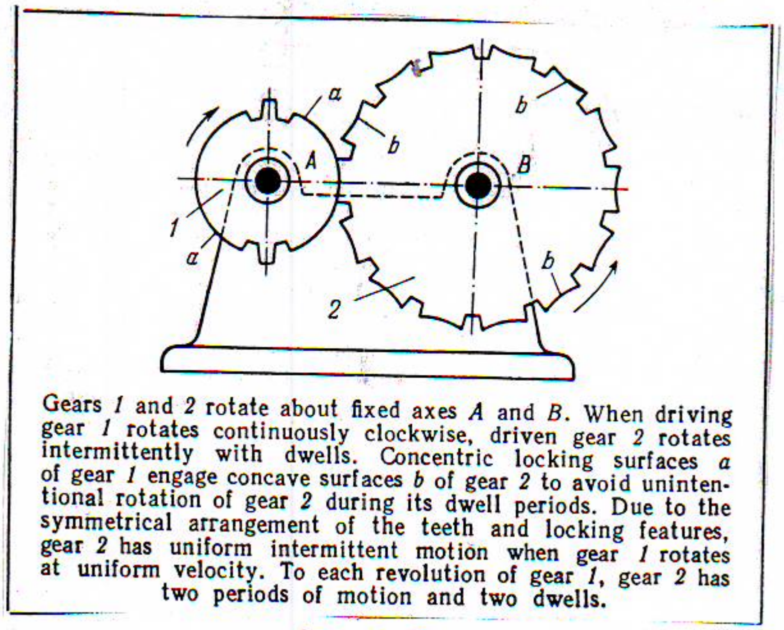


Fig (3)

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

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

$Y_f \cong 1.4$ $q_{\varepsilon 1} \cong q_{\varepsilon 2} \cong 3.4$ $\varepsilon = \varepsilon_w \approx \varepsilon_n \approx 1.6$

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

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

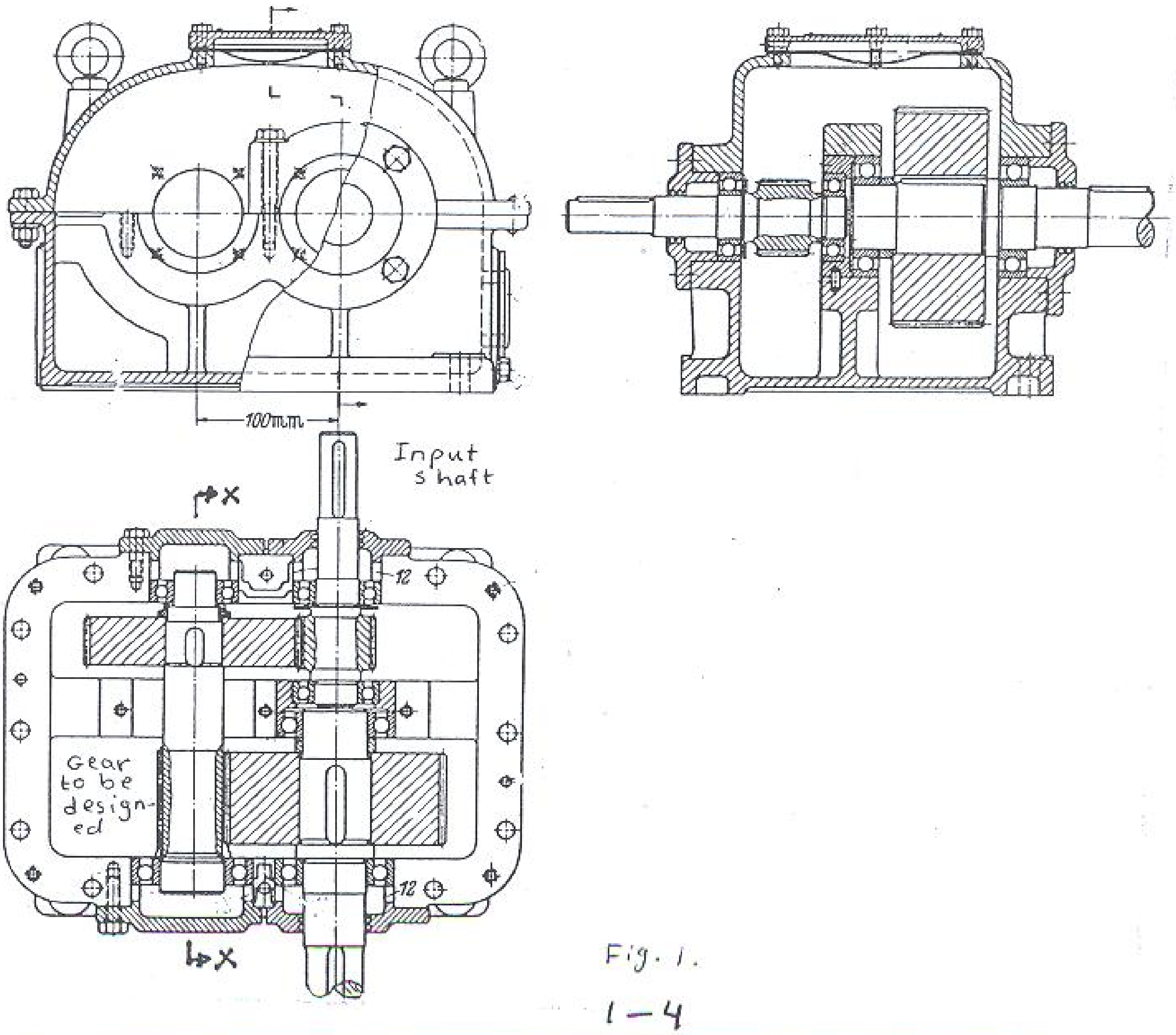


Fig. 1.

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).

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

$C_S . C_D . C_T \approx 1.75$

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)

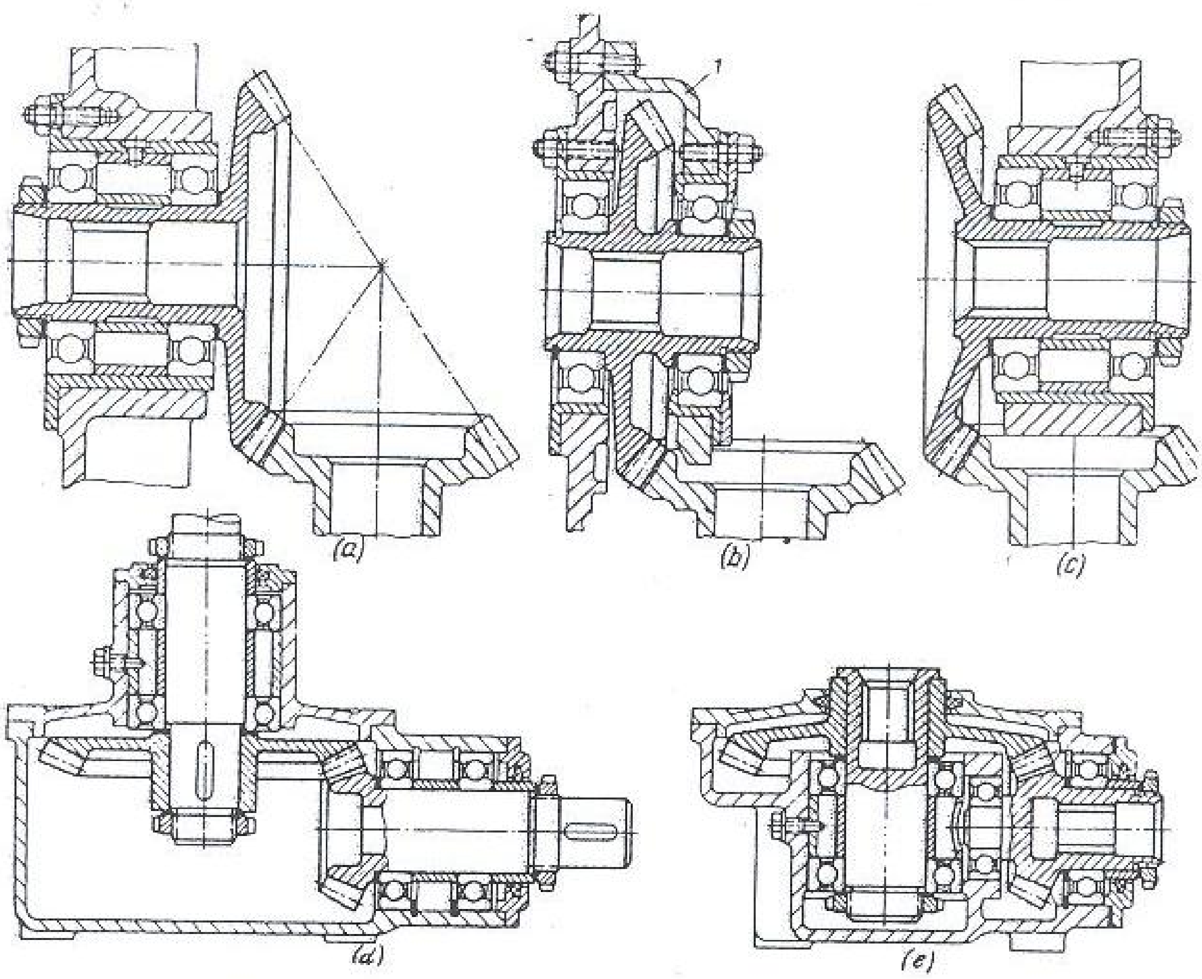


Fig 2 Reduction of overall dimensions of bevel gear drives

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)

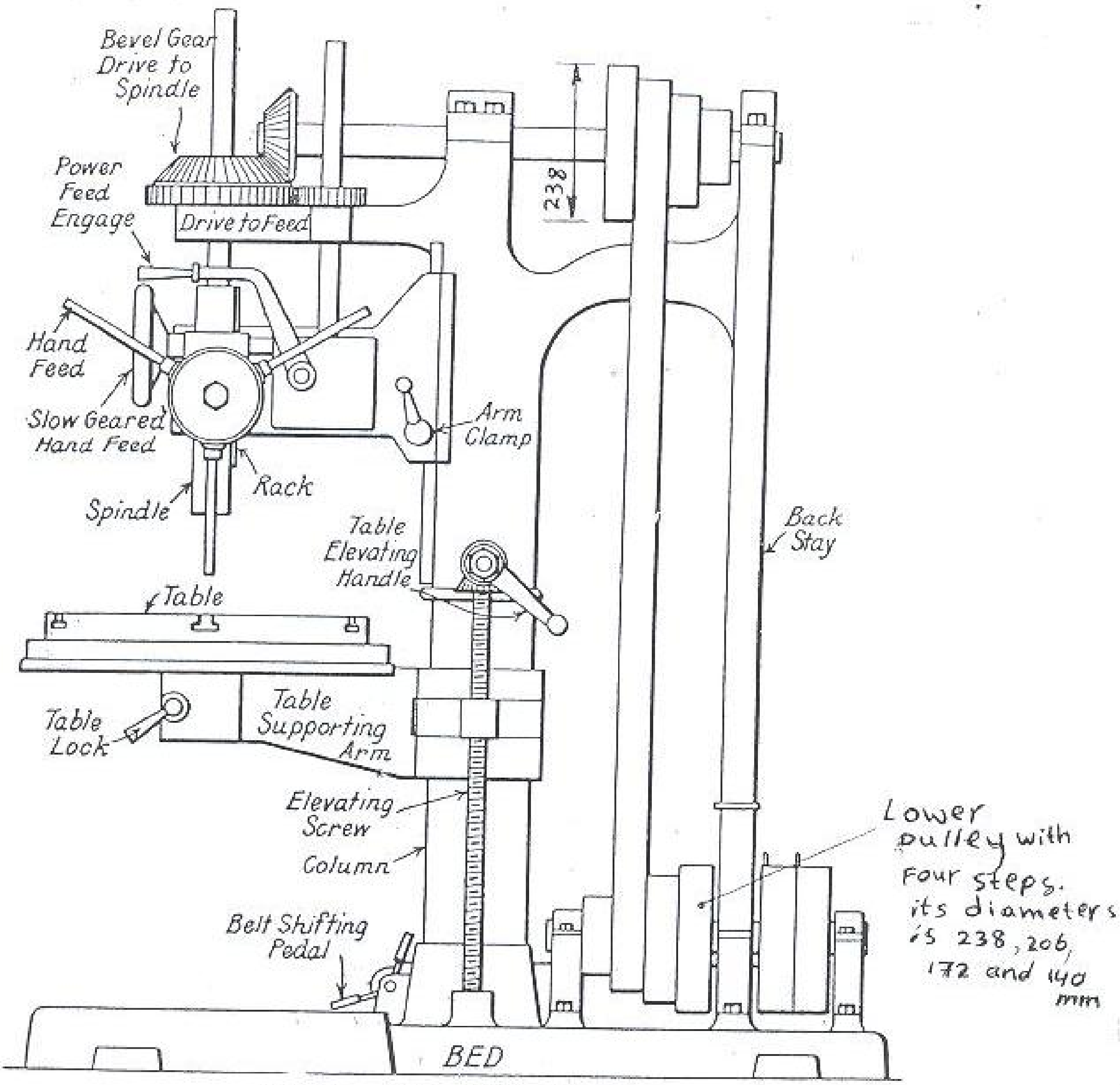


Fig. 3. THE POWER DRILL

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)

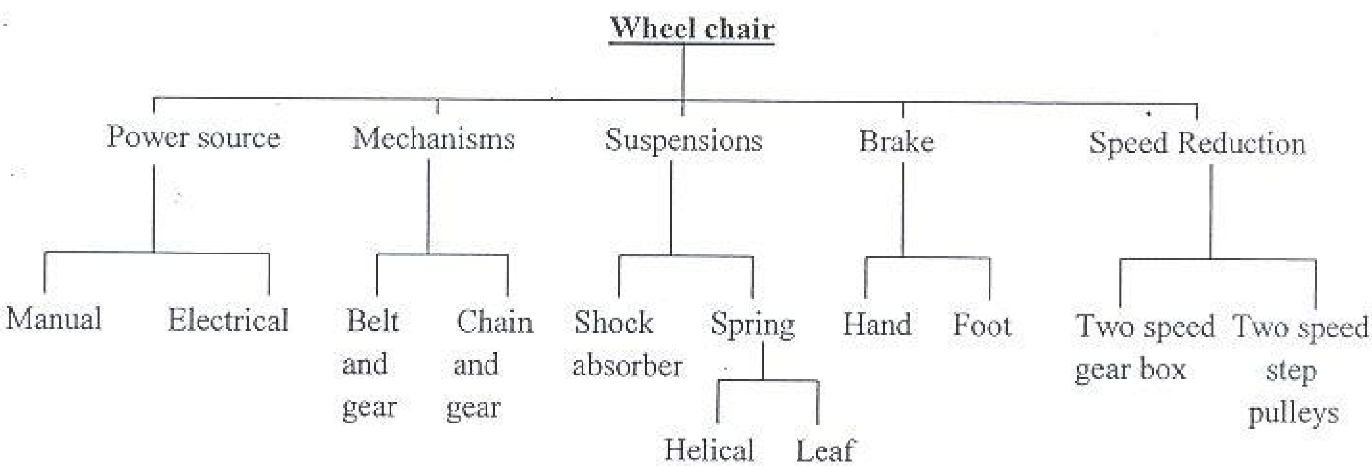


Fig. 4.

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
قسم هندسة المكائن والمعدات
الصف : الرابع عام
المادة : تصميم مكائن II

العام الدراسي : 2007-2008
امتحان الفصل الأول
31 \1\ 2008
الزمن ساعة

الملاحظات
1- يسمح باستخدام الكتب والمحاضرات
2- تمنع الاعارة لطفا
3- افرض القيم التي تراها مناسبة
4- الفروع كافة تحمل درجات متساوية
5- أذكر رأيك بالنتائج

س1 : شكل رقم (1) يمثل صندوق تروس مائلة ((Helical gear box)) ذو ثلاثة سرع .
أن القابض رقم (3) ينزلق على المقطع المربع للعمود رقم (I) و القابض رقم (4)
ينزلق على المقطع المربع للعمود المجوف رقم (II) . التروس رقم (8) و(9) و(10)
تدور مع العمود رقم (2) باستخدام الخوابير والمتعشقة مع التروس (5) و(6) و(7)
ويتحركون بحرية حول المحاور (I) و(II).
عندما يدور العمود رقم (I) تنتقل الحركة الى العمود رقم (2) بثلاث سرع كما موضحة بالشكل
وحسب اقطار التروس المتعشقة مع بعض .

المعلومات

مقدار القدرة المنقولة عن طريق العمود رقم (1) = **24 hp**

السرعة الدورانية للعمود رقم (1) = **2100 rpm**

عدد الاسنان للتروس : $(z_5 = 36)$ $(z_6 = 31)$ $(z_7 = 26)$ $(z_8 = 14)$ $(z_9 = 19)$ $(z_{10} = 24)$

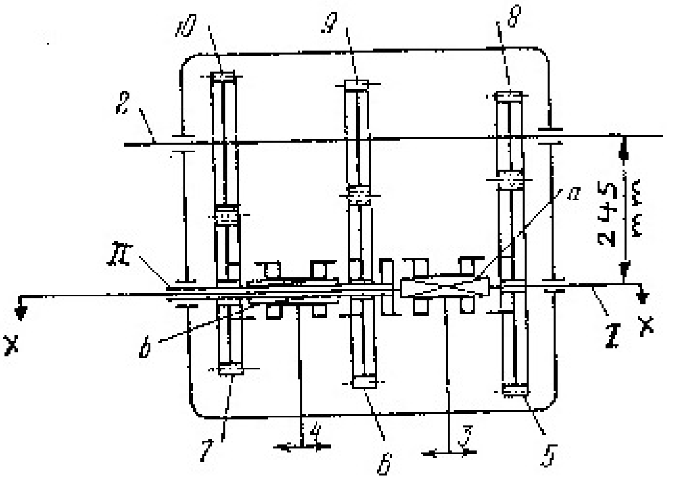
زاوية ميلان السن لجميع التروس $(\beta_o) = 20^\circ$

مقدار $\dfrac{b}{d_{b1}} = 0.3$

المسافة المركزية بين التروس (5,8) و(9,6) و(7,10) = **245 mm**

المطلوب :-

1- اوجد الابعاد الاساسية للترسين (5) و (8) **فقط**
2- اجب عن احد الفرعين :
أ- نسبة التلامس الفعالة بين الترسين (5) و(8)
Effect contact ratio between gear (5) and (8)
ب- ارسم المقطع x -x موضحا كافة التثبيتات لكل الاجزاء.



Clutch *3* slides along square guide *a* of shaft *1*. Clutch *4* slides along square guide *b* of hollow shaft *11*. Gears *8*, *9* and *10* are keyed to shaft *2* and are in constant engagement with gears *5*, *6* and *7* which rotate freely about shafts *1* and *11*. When driving shaft *1* rotates, driven shaft *2* can have any one of three speeds: for the first, clutch *4* is engaged to gear *7* and clutch *3* to hollow shaft *11*; for the second, clutch *4* is engaged to gear *6* and clutch *3* to hollow shaft *11*; and for the third, clutch *4* is in the neutral position and clutch *3* is engaged to gear *5*.

شكل رقم (1)

س2 : الشكل رقم (2) يمثل مقترح لآليه تقوم بعملية رفع بوابة (Gate) لاحد السدود.
هذه الآلية تتالف من محرك (Motor) وتروس دودية (Worm gears) وتروس مخروطية (Bevel gears) ولولب نقل قدرة (Power screw) وصامولة (Nut) . هذا وان الصامولة مثبتة على الترس المخروطي الكبير كما في الشكل رقم (3)

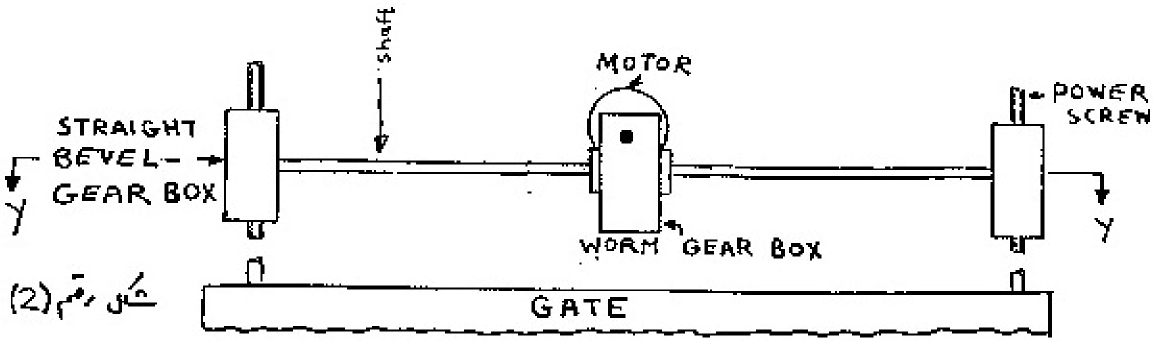
المعلومات :-

قدرة المحرك – **5 kw**
السرعة الدورانية للمحرك = **3000 rpm**
نسبة التخفيض للتروس الدودية = **$(i_1) = 20$**
المسافة المركزية بين الترس الدودي والمسنن الكبير = **220 mm**
المعدن المستخدم للمسنن الكبير (worm wheel) – ((phosphor bronze))
نسبة التخفيض للتروس المخروطية = **$i_2 = 2$**
عرض التروس المخروطية (b) = **33.6 mm**
المعدن المستخدم للترسين المخروطيين – **GG18**
استخدم المعلومات التالية لتسهيل الحل للتروس المخروطية :

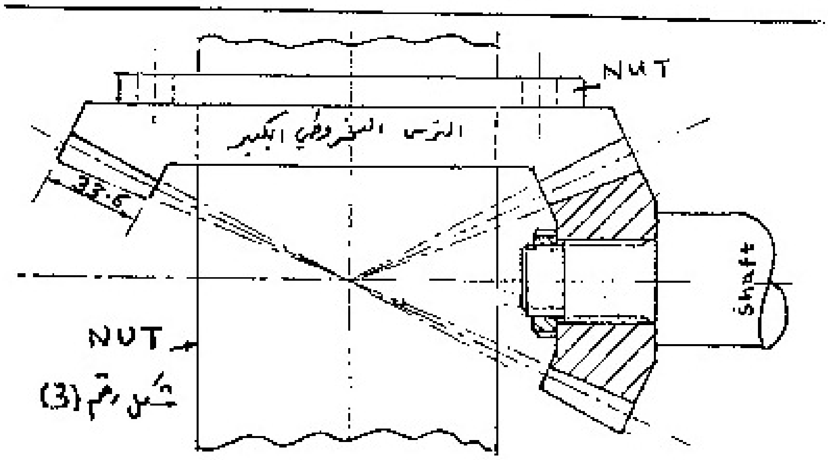
$q_{w1} \approx q_{w2} \approx 1.85$ $\quad$ $y_{w1} \approx y_{w2} \approx 3.11$ $\quad$ $y_f \approx 0.4$ $\quad$ $C_S \approx C_D \approx C_1 \approx 1$

المطلوب :-

ملاحظة : اجب عن فرعين فقط .
1-عوامل الأمان كافة للتروس الدودية ((عامل الامان ضد الكسر وضد التآكل وغيرها ....))
2-عوامل الأمان كافة للتروس المخروطية ((عامل الأمان ضد الكسر وضد التنقر وضد التخدش))
3- ارسم المقطع (y y) موضحا كافة تفاصيل التثبيت لكل الاجزاء



شكل رقم (2)



شكل رقم (3)

God with you

الجامعة التكنولوجية	العام الدراسي : 2007-2008	الملاحظات
قسم هندسة المكائن والمعدات	امتحان الدور الأول	1- يسمح باستخدام 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

س 1 :

الشكل رقم (I) 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

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

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
للمسافة المركزية للأحزمة = 350 mm
نسبة التخفيض للبكرات = 4
للأحزمة أفرض أن $C_1 = C_2 = C_3 = C_4 = C_5 = C_6 = 1$ $C_7 = 1.25$

المطلوب :

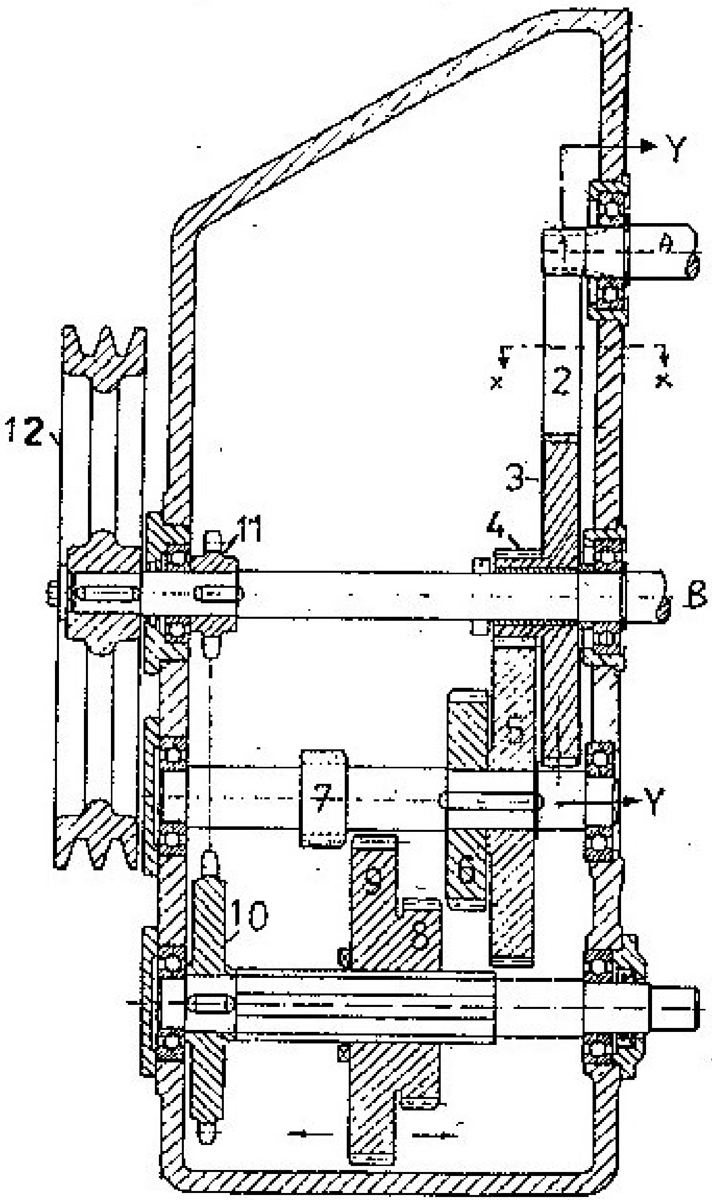
1- ايجاد مقدار التصحيح (X_2) و المسافة الطرفية (Addendum) للمسنن رقم (2) h_{K2}
والمسافة المركزية (a) , (centre distance) بين المسنن (2) و المسنن رقم (3)
(12 درجة)

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)

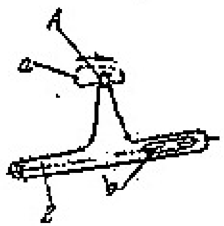
3- شكل رقم (2) يوضح ((system scheme)) لاحد الحلول و التي تعطي فكرة واضحة للحركة حيث ان القارب يتحرك باستخدام المجذاف المثبت بيد الشخص المجذف و تاتي الحركة عن طريق محرك يقوم بتدوير عمود المرفق رقم (1) و الذي بدوره يحرك ذراع التوصيل رقم (2) الذي يحتوي الشق (b) الذي ينزلق على المسمار رقم (3) المثبت بالقارب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

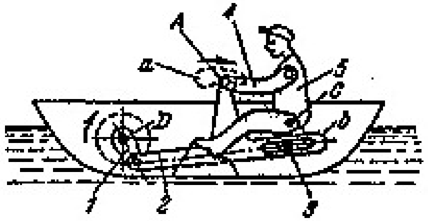
النقطة (A) في ذراع التوصيل رقم (2) تعمل المنحنى (a) لتحرك ذراع الشخص رقم (4) و التي تحمل المجذاف (oars) و كذلك تحرك جسم الشخص المُجذف (oarsman) رقم (5) الذي يتذبذب حول المحور (c) ليمنح الحركة المطلوبة .

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

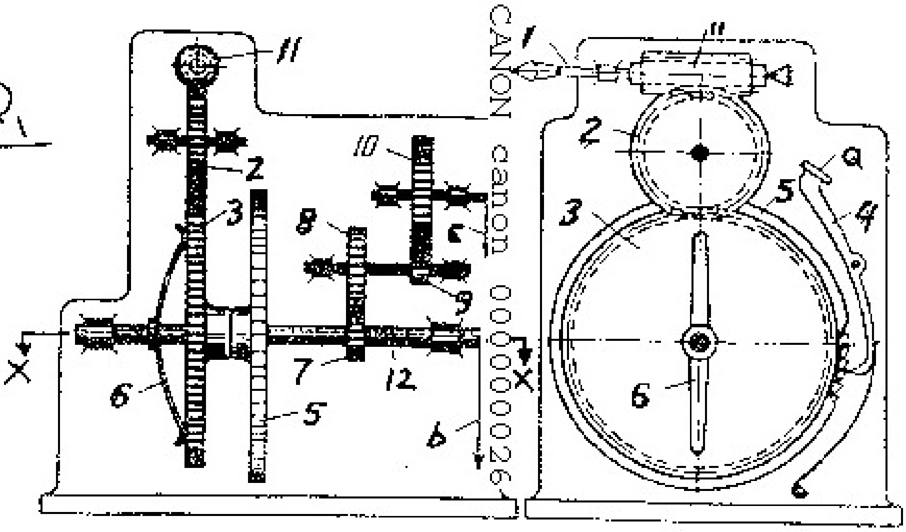
Alternatives → البدائل / ↓ الانظمة الفرعية Sub- systems	I	2
A مصدر الطاقة	A1 استخدم محرك كهربائي ذو بطاريتين (وحسب ما معطى بالسؤال)	....
B شكل القارب والاجزاء المثبتة عليه لتساعد على الحصول على الحركة المطلوبة	B1 	B2
C الية توصيل الحركة بين مصدر الطاقة والشخص	C1 	C2
D حركة الشخص المُجذف لتعطي متعة للطفل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

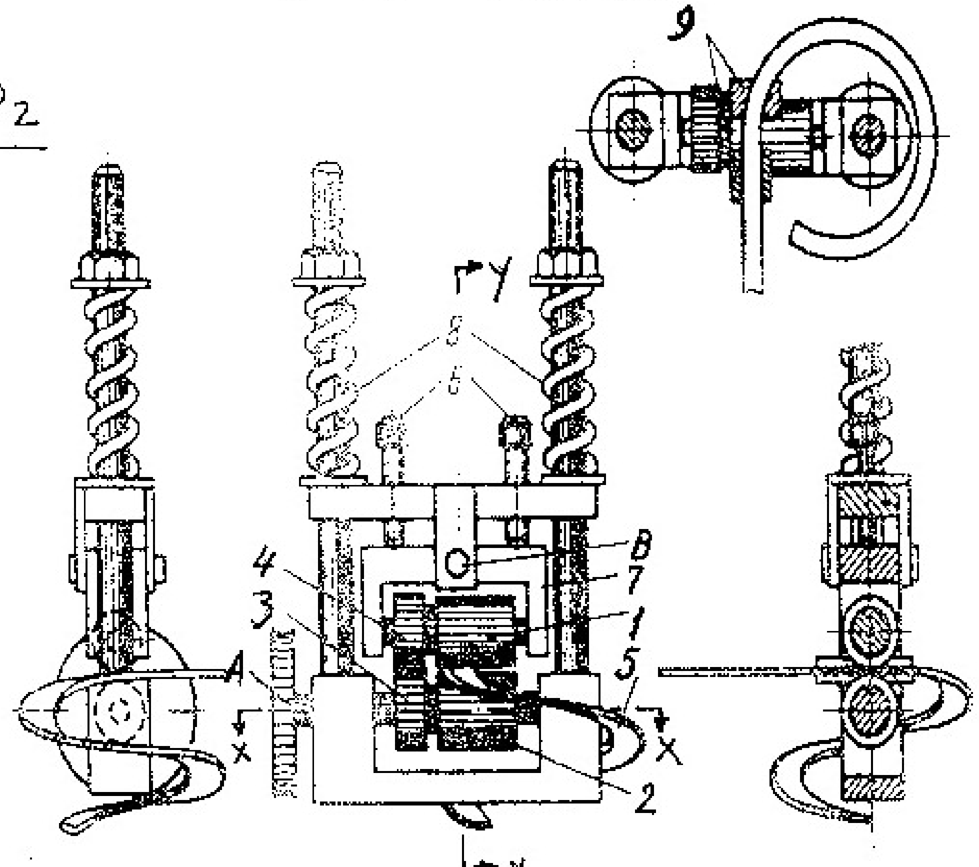
Q1



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

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 رايك بالنتائج
4- تمنع الاعارة لطفا

ملاحظة : اجب عن ثلاثة اسئلة فقط

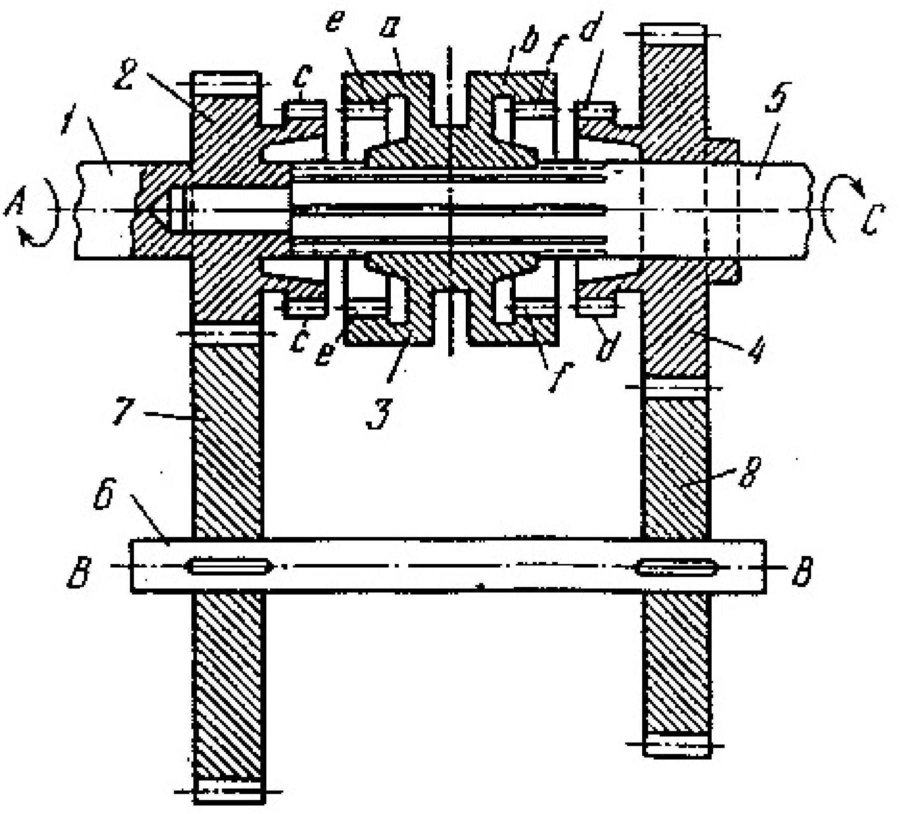
س1- الالية في شكل رقم (1) تمثل مسننات لتغير السرع و بسرعتين مختلفتين . ان الحركة تنتقل عن طريق المسنن رقم (2) الذي هو جزء من العمود رقم (1) الى العمود رقم (5) عن طريق القابض المسنن رقم (3) المتعشق مع العمود رقم (5) عن طريق التحززات (splines) بين العمود والقابض .
ان المسنن رقم (4) يدور بحرية حول العمود 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) $\beta_o = 22°$
مقدار التضمين Mn للتروس الحلزونية = 3 mm

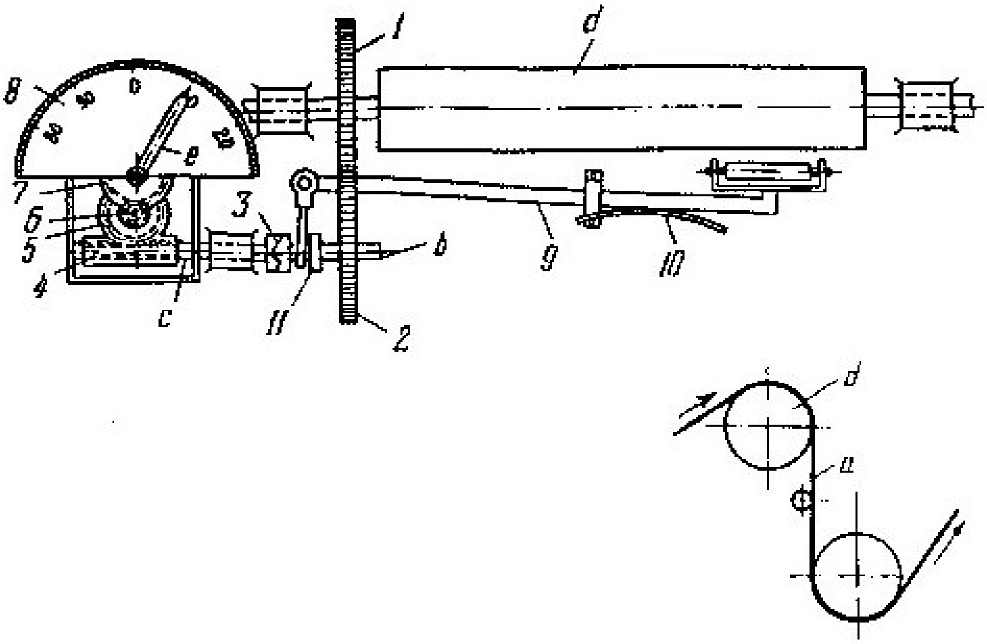
المطلوب

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



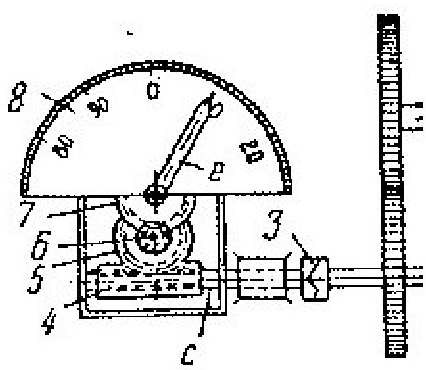
شكل رقم (1)

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 لطفا

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

ملاحظة: اجب عن سؤالين **فقط**

السؤال الأول

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 والترس المستقيم رقم (3) تنتقل الحركة الى العمود (12) وهذا 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) والمؤشر (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

الدرفيل (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 الصمولتين 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 سرع رقم (9) و الذي يدار من خلال القرص الدودي (10) و (11) .

المعلومات

التضمين m = 4 mm

قطر الدودة (worm) = 50 mm

قطر القرص الدودي (worm wheel) = 200 mm

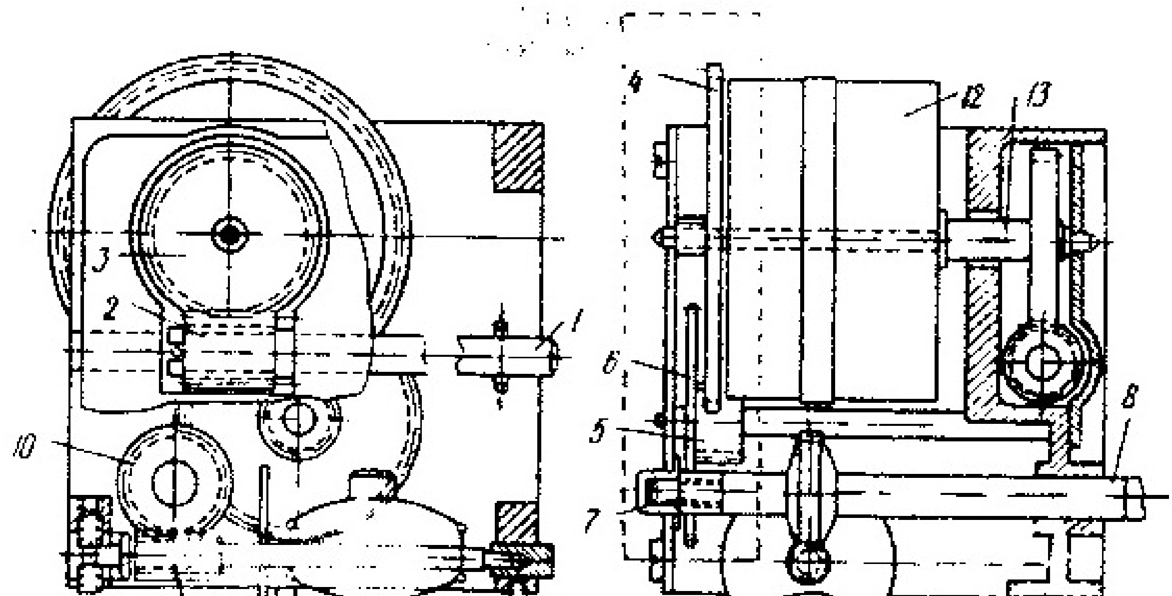
نسبة التخفيض للتروس الدودية = i = 12.5

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 مبدأ 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*.

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الصف الرابع عام   الزمن : ساعة ونصف .

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 التداخل 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) = 300mm

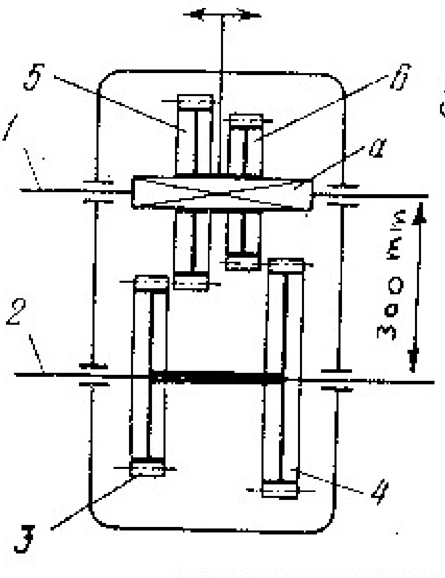
نسبة التخفيض بين الترسين (5)و(3) = $i_1 = 1$

" " " (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 تعرية السطح للترس رقم (4) فقط .

٣- ارسم مقطع متكامل مع كافة التفاصيل لصندوق التروس .

أفرض مايلي لتسهيل الحل

$B_{all} \simeq B_w$

$\varepsilon = \varepsilon_n \simeq \varepsilon_w$

$b/d_{b_1} \simeq 0.2$

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

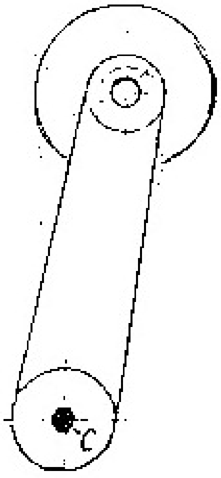
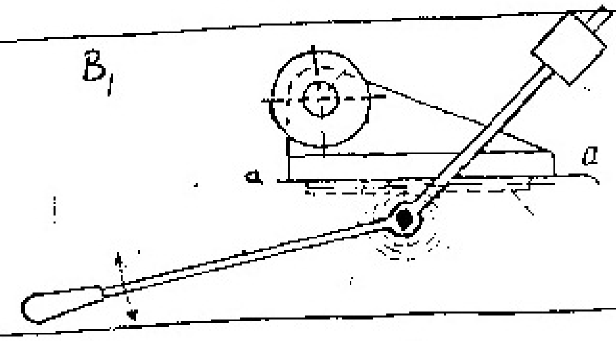
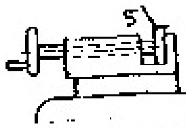
الجدول رقم (1) ادناه يمثل «morphological chart» لهذا النظام بعد تقسيمه الى انظمة فرعية «sub-systems».

المطلوب

١- ارسم فقط وبدون شرح مختصر او مفصل 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 بإستبدال A_1. ------ «11 درجات»

٢- ارسم «system scheme» الذي يمثل التصميم الجديد والذي يتبع المسار (A_2 ، B_2 ، C_2) رسماً 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 آليه 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 (1) يمثل تقسيم النظام الى انظمة فرعية وايجاد أفكار بديلة.

س٣: المنشار الدائري رقم (4) يقوم بقص القطعة 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) فعلياً وعلى المسار (a-a) وعند حركة الذراع رقم (1) مع عقرب الساعة يبتعد قرص المنشار عن القطعة رقم (5).

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

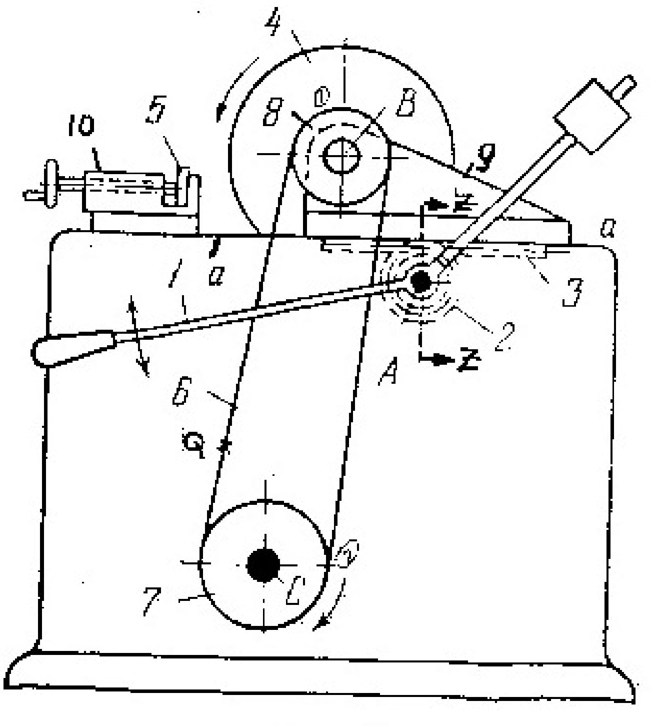
نوع الحزام المستخدم: « HG Leather belt ».

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 بإستخدام 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) .

المعلومات :

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

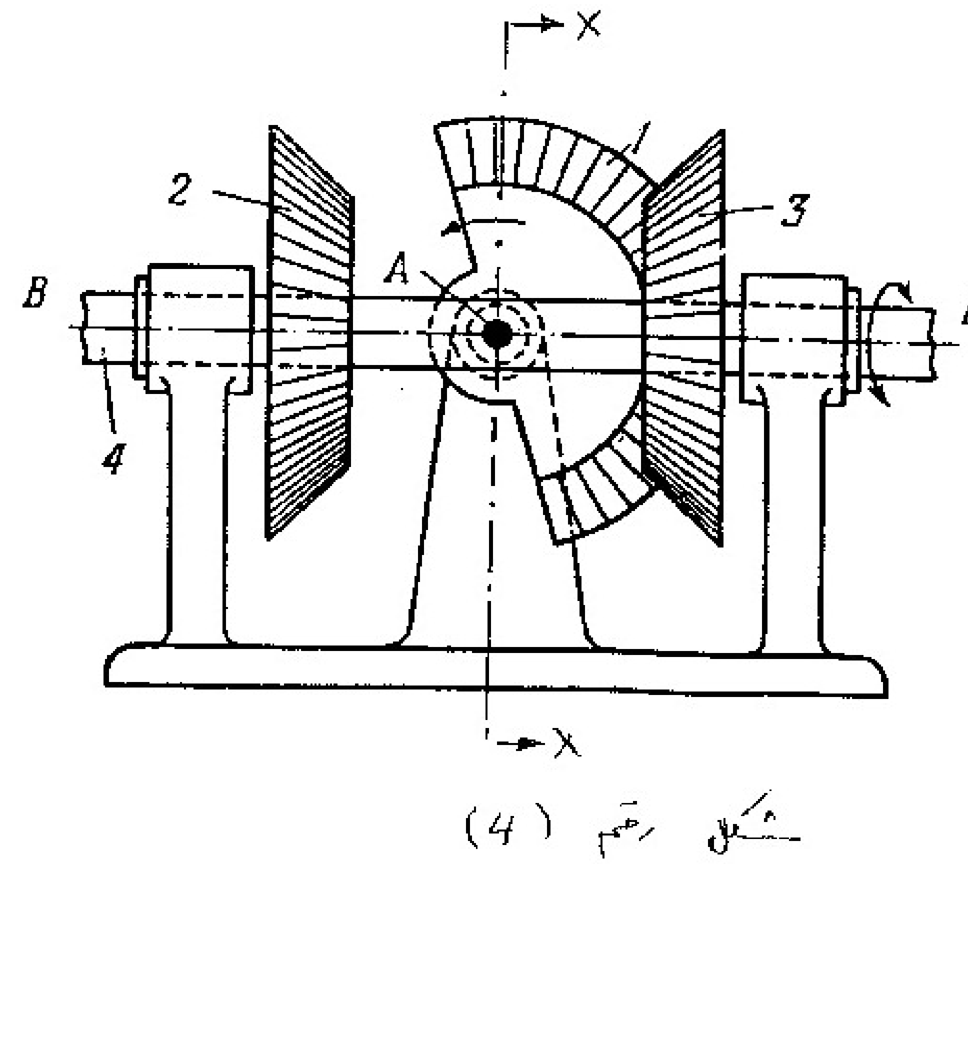
قطر دائرة الخطوة للمسنن رقم (1) = قطر دائرة الخطوة للمسنن رقم (2) = قطر دائرة الخطوة للمسنن رقم (3) = 90 mm

افرض ما يلي لتسهيل الحل ($y_\varepsilon = 0.85$) . ($1.25 = \varepsilon_w$) . ($1.5 = \varepsilon_n$) . ($1.75 = C_S, C_D, C_T$)

المطلوب :

1- اوجد كافة عوامل الامان 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 الية 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.

المطلوب :

1- ارسم فقط 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 الية 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) .

المعلومات :

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

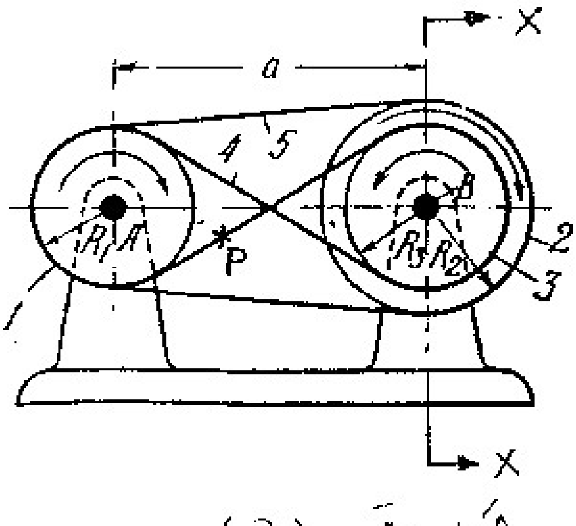
لتسهيل الحل افرض C= C1.C2.C3. C4. C5 = 1

المطلوب :

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 الخلف .

University of Technology
Department of Machines and Equipment Engineering
First Mid-Examination 2013/2014

Subject: Design II | Year: fourth
Division: General Mech. | Exam Time: 1:30 Hrs.
Examiners: Design Group | Date: 2/12/2013

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.

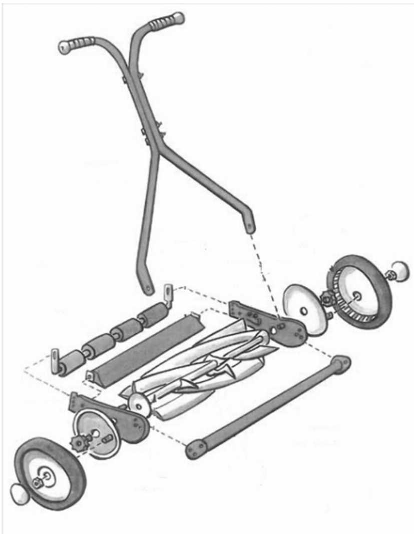


Figure.2: The exploded drawing for the regular manual push reel mower.

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).

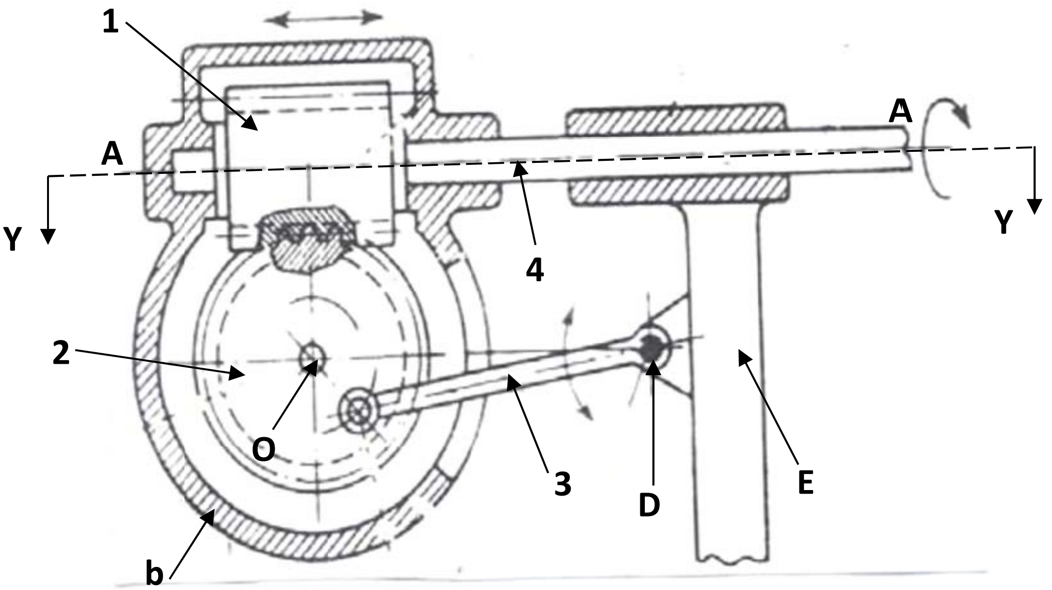


Figure (1)

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?

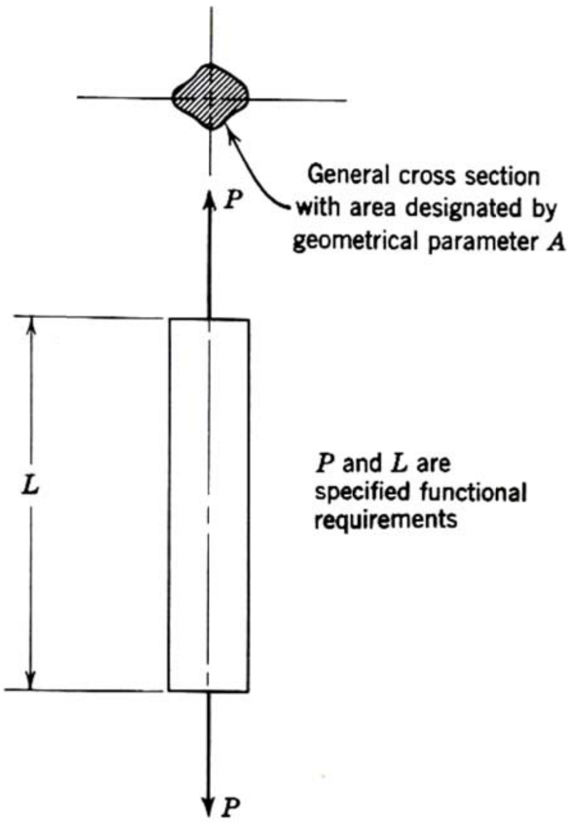


Figure (2)

(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)

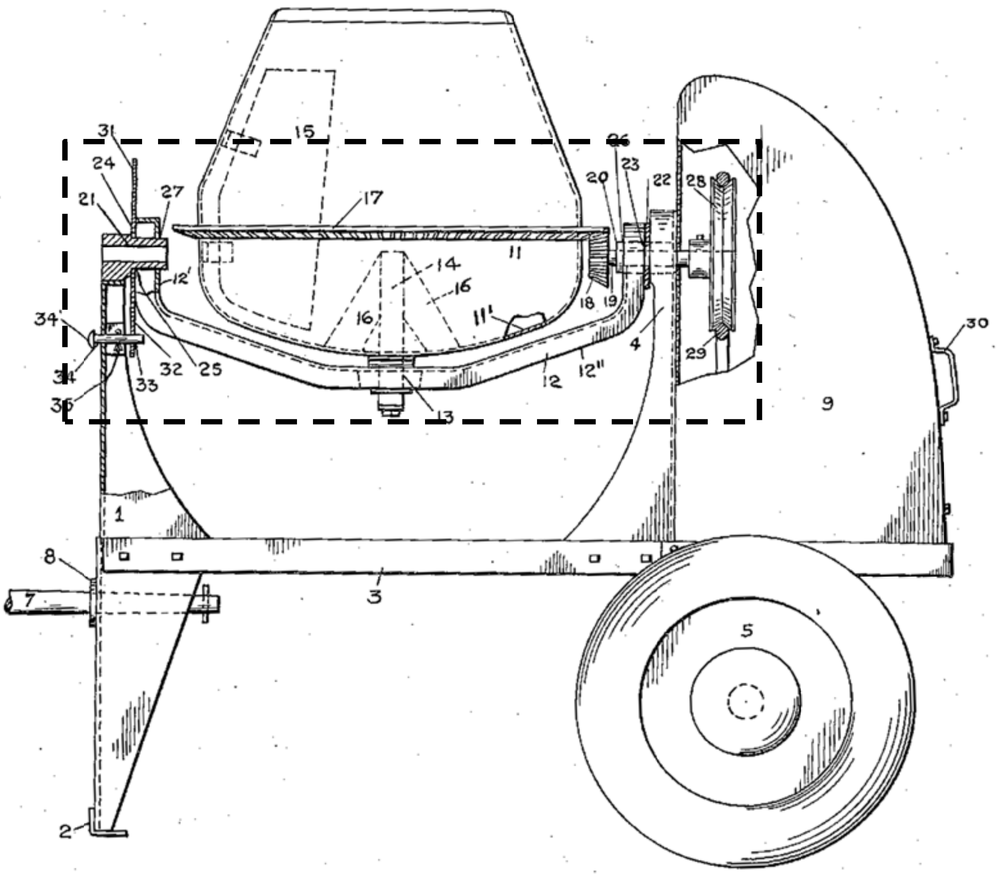

Figure (1)

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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?

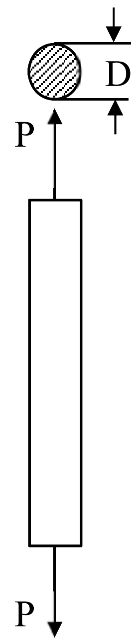
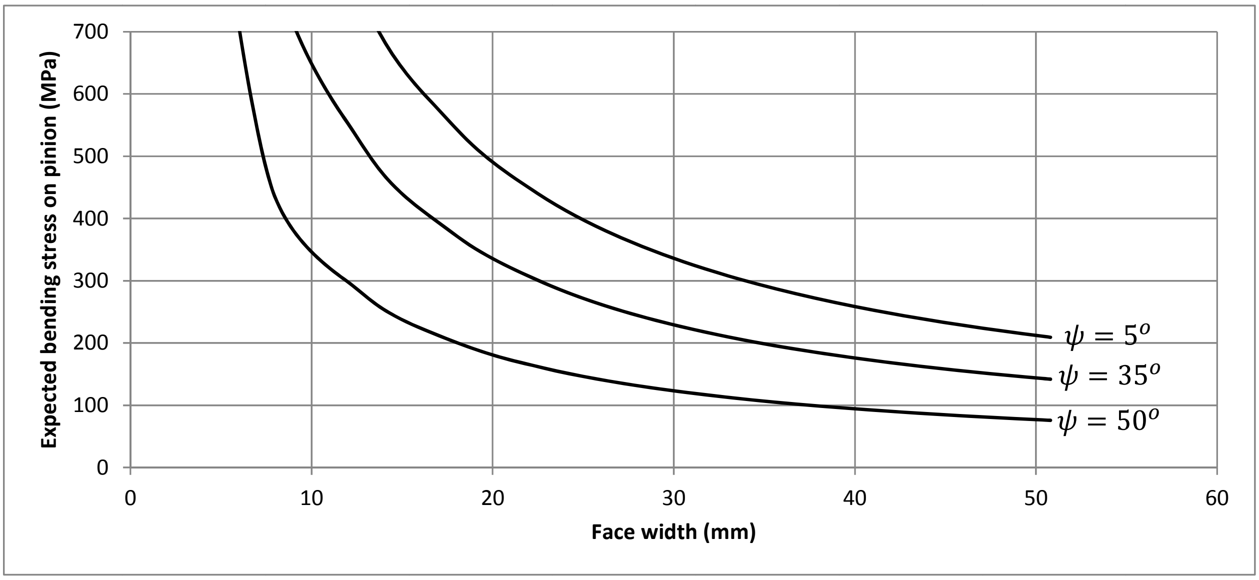


Figure.(2) simple tensile bar with uniformly distributed specified axial load (P)

(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)

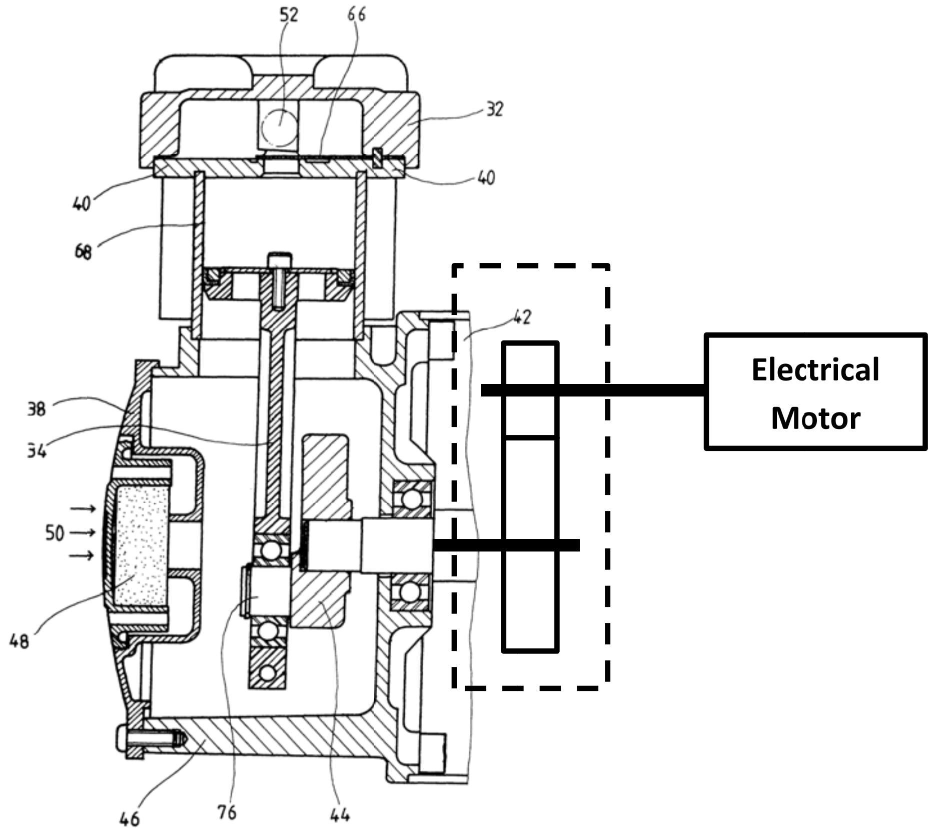


Figure (1)

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:

$10\text{ mm} \leq D \leq 20\text{ mm}$

$S_y = 300\ \text{N}/\text{mm}^2$

Safety factor ≥ 3 , c = unit volume cost of shaft = 2500 \$/m3

P=1000 N

Find minimum cost and at what length and area?

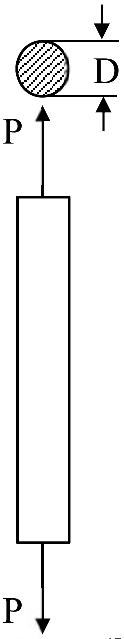
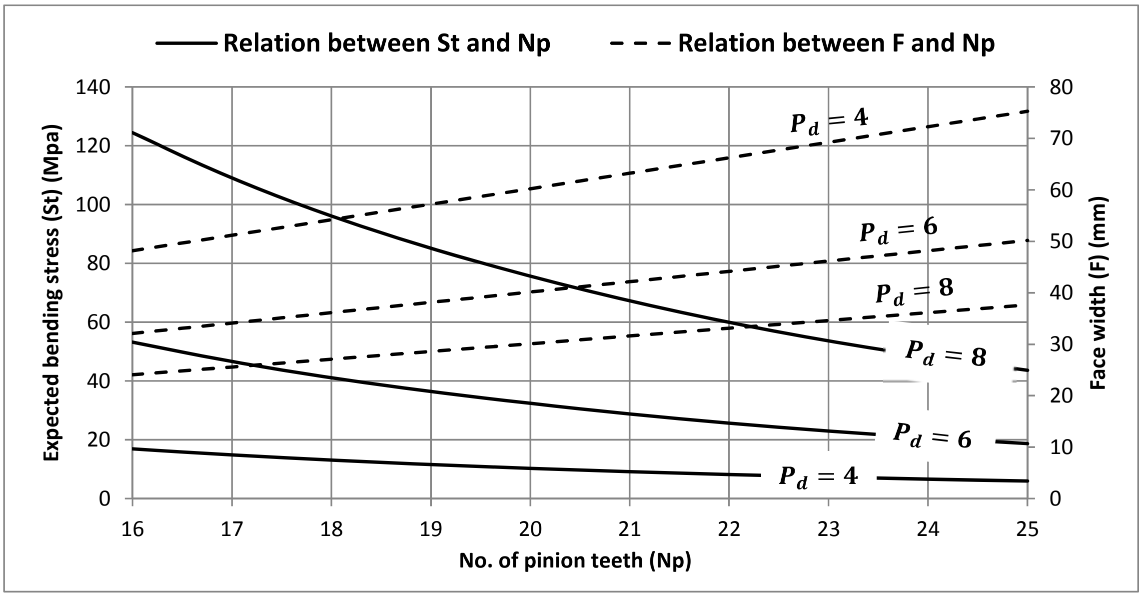


Figure.(2) simple tensile bar with uniformly distributed specified axial load (P)

(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).

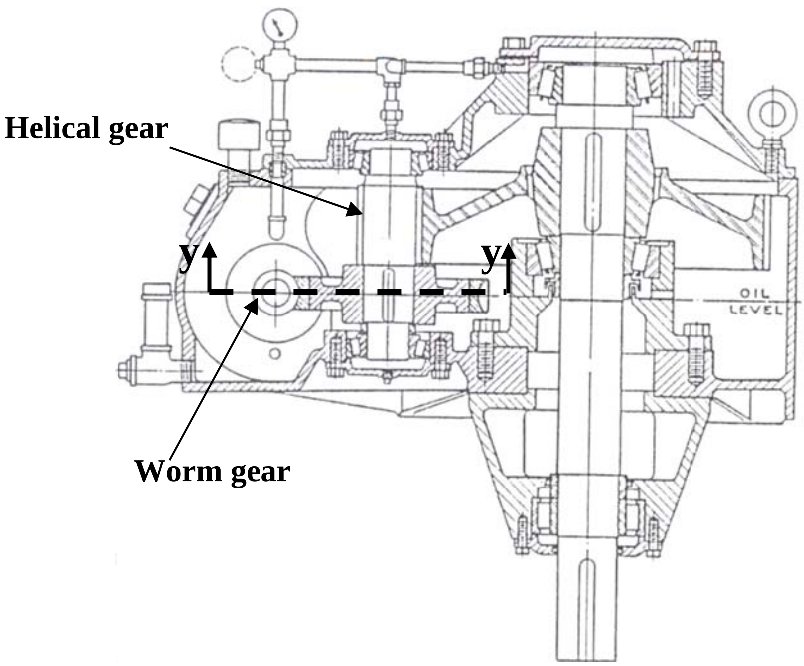


Figure (1)

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^o
Normal pressure angle = 20^o
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)

University of Technology
Department of Machines and Equipment Engineering
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:

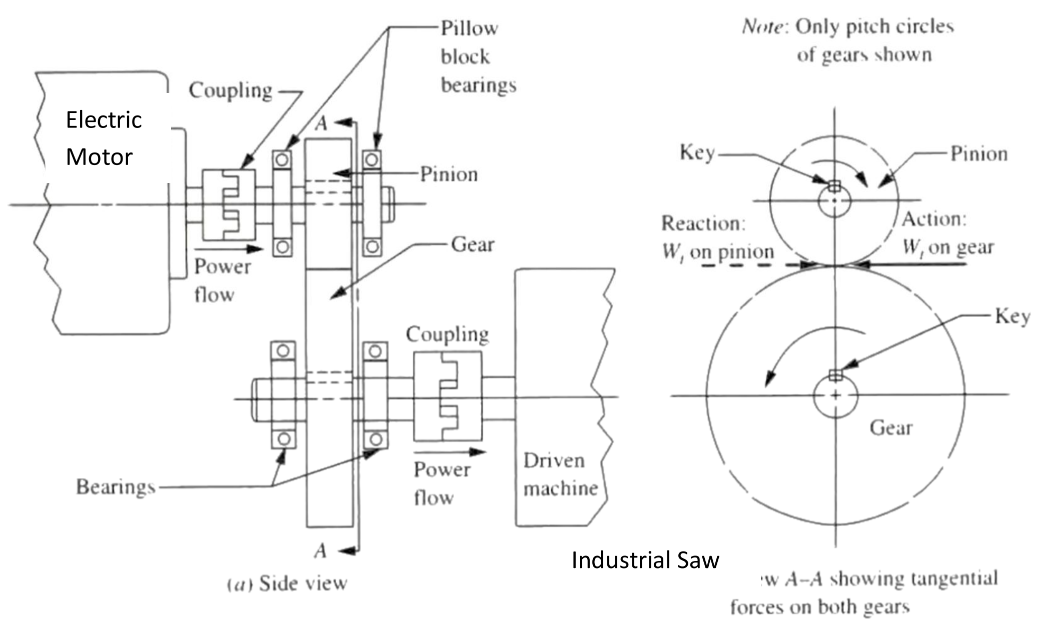


Table.1 shows a trial for solving the spur gear above (Fig.1):

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.

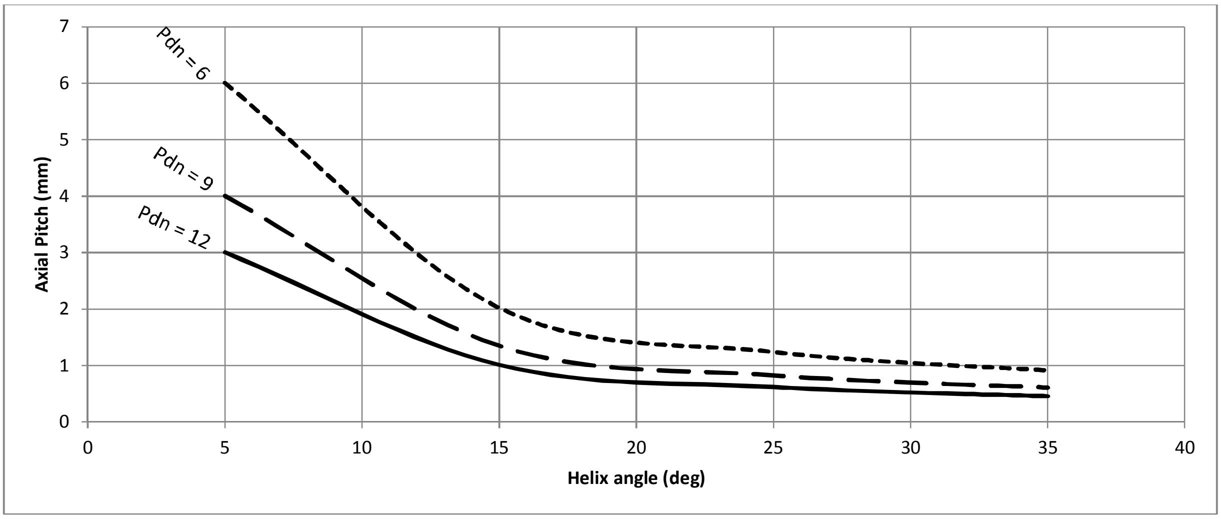
(35 mark)

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

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

Year: fourth
Exam Time: 1:30 Hrs.
Date: 22/4/2013

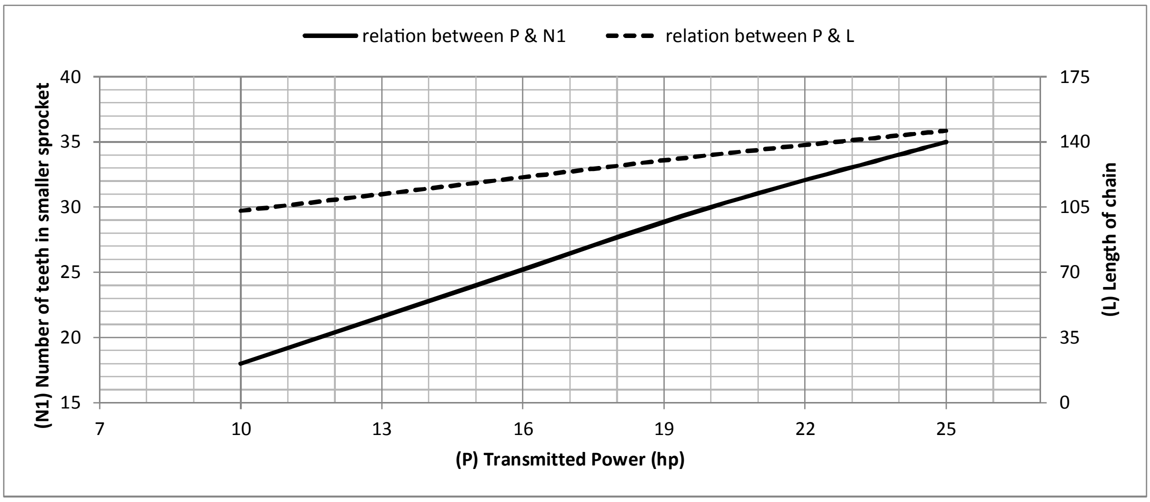
Q2: A: Use helical gear instead of spur gear for question one, with helix angle ($\Psi = 5^o - 35^o$) and (P_{dn}= 6 – 12 teeth/in), the following figure is the results between axial pitch (P_x) and (Ψ):



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

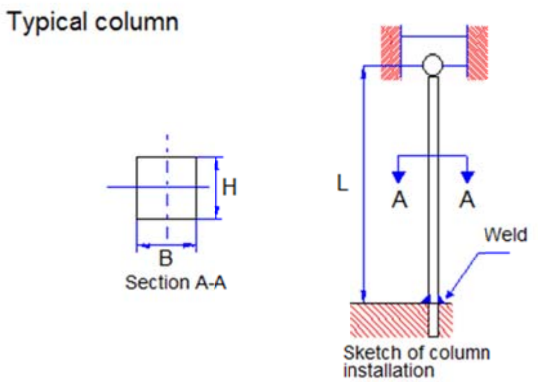
$$P_a = P_y / N \quad \text{or} \quad P_y = NP_a$$

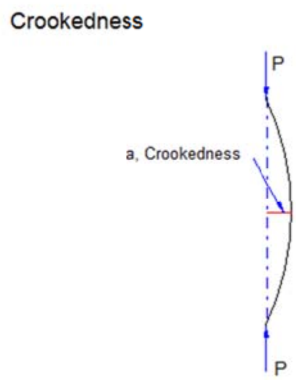
this equation becomes

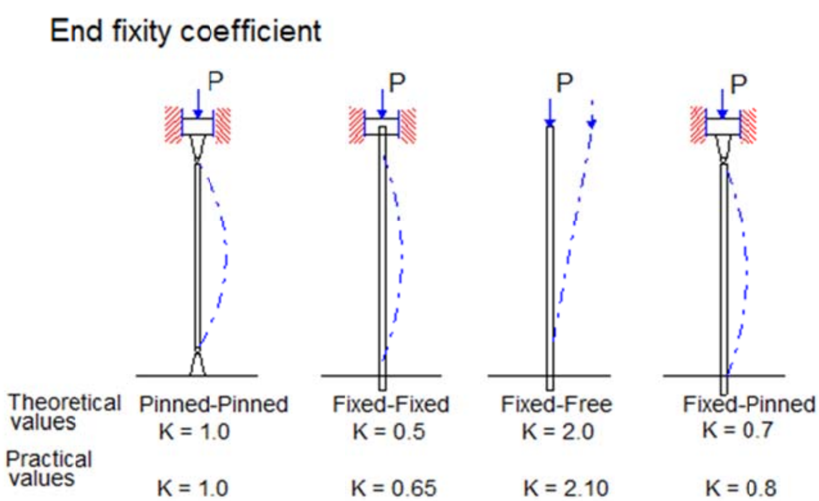
$$\text{Required} \qquad S_y = \frac{NP_a}{A}\left[1 + \frac{ec}{r^2}\sec\left(\frac{KL}{2r}\sqrt{\frac{NP_a}{AE}}\right)\right]$$

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:

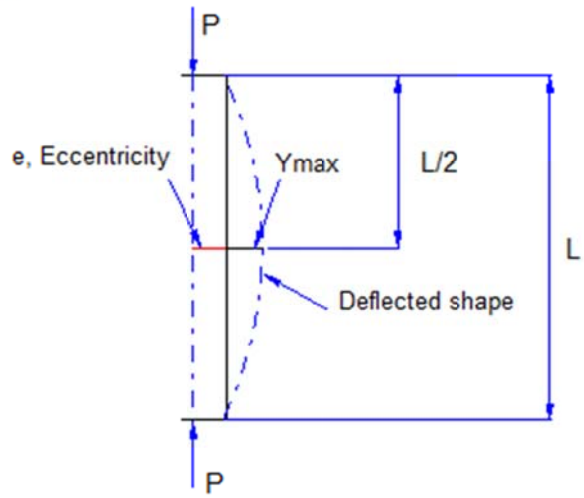
$$Y_{\max} = e\left[\sec\left(\frac{KL}{2r}\sqrt{\frac{P}{AE}}\right) - 1\right]$$



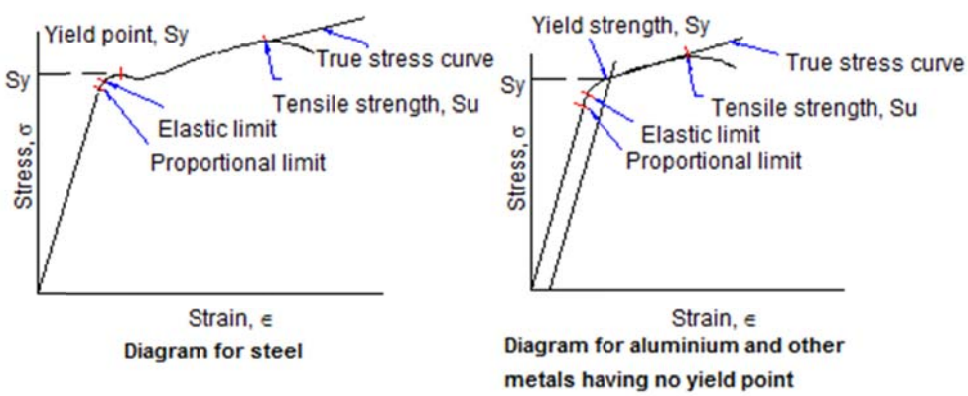




Eccentricity



Typical stress-strain diagram



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:

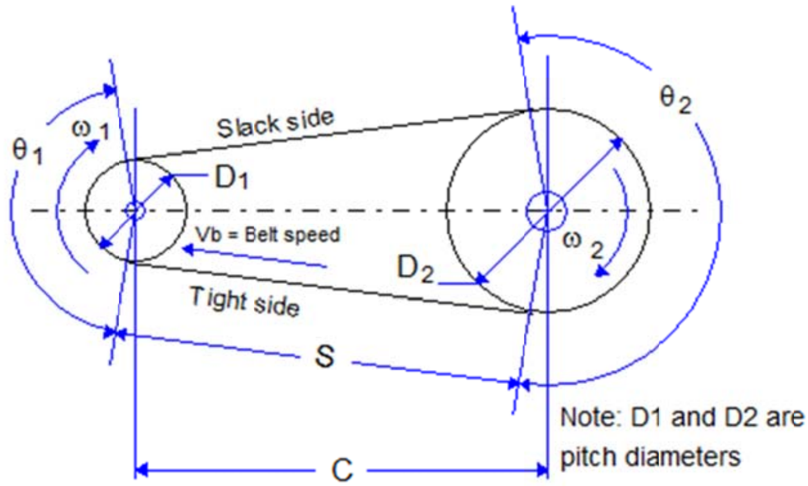
- Adjustment for the center distance must be provided in both directions from the nominal value. The center distance must be shortened at the time of installation to enable the belt to be placed in the grooves of the sheaves without force. Provision for increasing the center distance must be made to 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

$$D_2 < C < 3(D_2 + D_1)$$

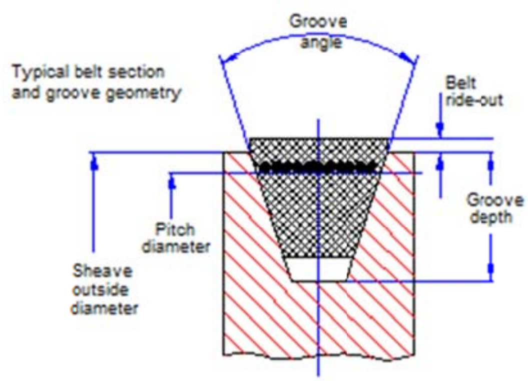
- The angle of wrap on the smaller sheave should be greater than 120°.
- Most commercially available sheaves are cast iron, which should be limited to 6 500-ft/min belt speed.
- Consider an alternative type of drive, such as a gear type or chain, if the belt speed is less than 1 000 ft/min.
- Avoid elevated 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.

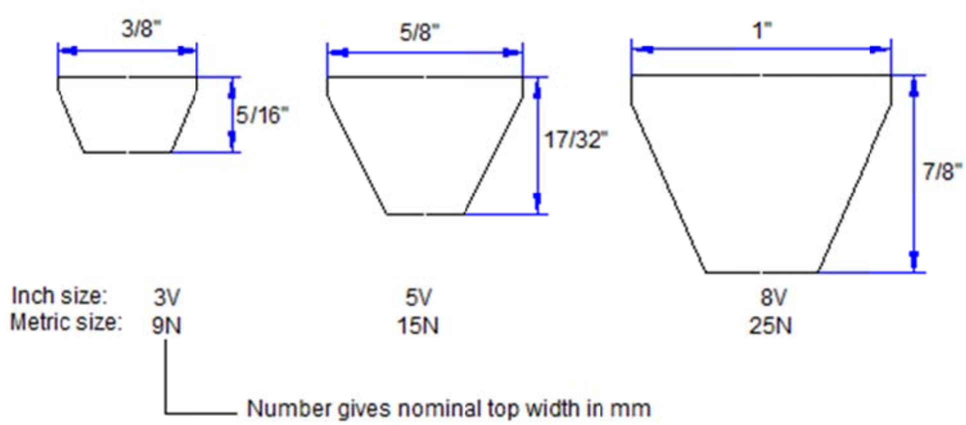
Basic belt drive geometry



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.

$$V_c = \frac{\pi \cdot D \cdot n}{12}$$

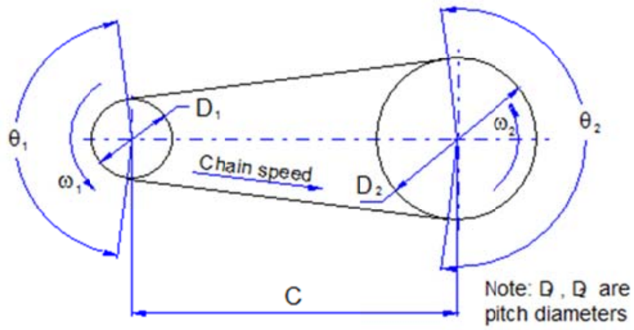
where D = pitch diameter of sprocket;
n = rotational speed of sprocket

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

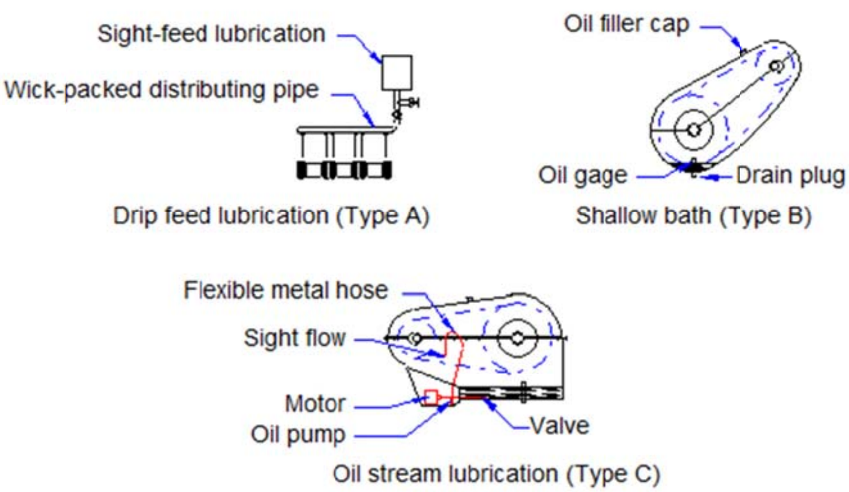


Roller chain styles

Standard roller chain,
single strand

Standard roller chain,
two-strand (also available
with three and four strands)

Lubrication methods



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 $F = \dfrac{12}{P_d}$

Notice that $\dfrac{F}{D_p} < 2.00$ 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.

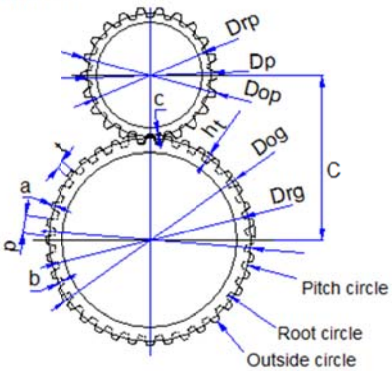
For m_B >1.2, the rim is sufficiently strong and stiff to support the tooth, and K_B =1.0.

For m_B <1.2, rim thickness factor determined:

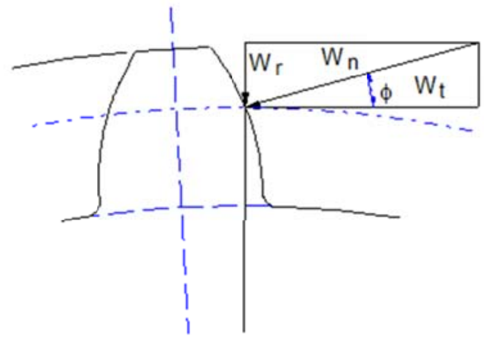
$$K_B = 1.6 \ln\left(\frac{2.242}{m_B}\right)$$

When a solid gear blank is used, input a large value (say tR > 1.0 inch) for rim thickness. The resulting value for $K_B = 1$.

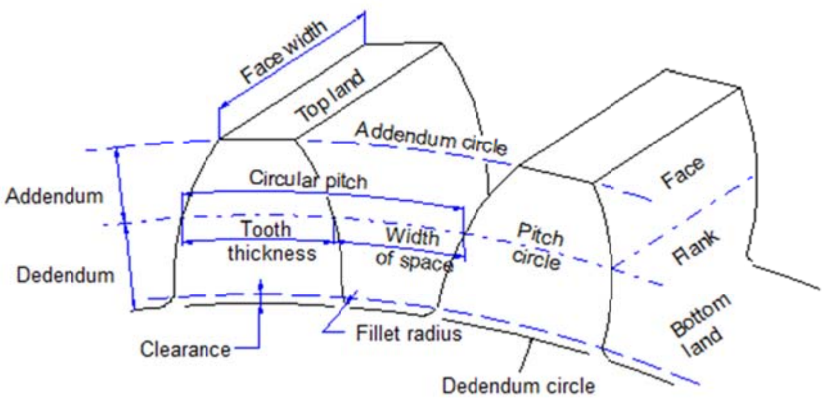
Gear pair features



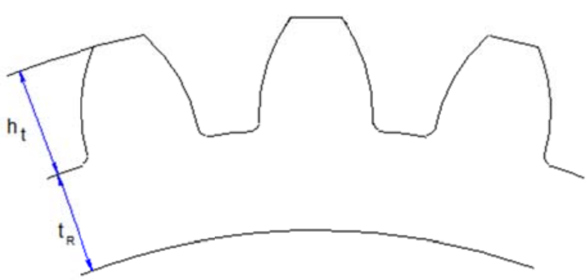
Forces on the spur gear tooth



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

$$t = \frac{\pi}{2P_{dn}}$$

- 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

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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

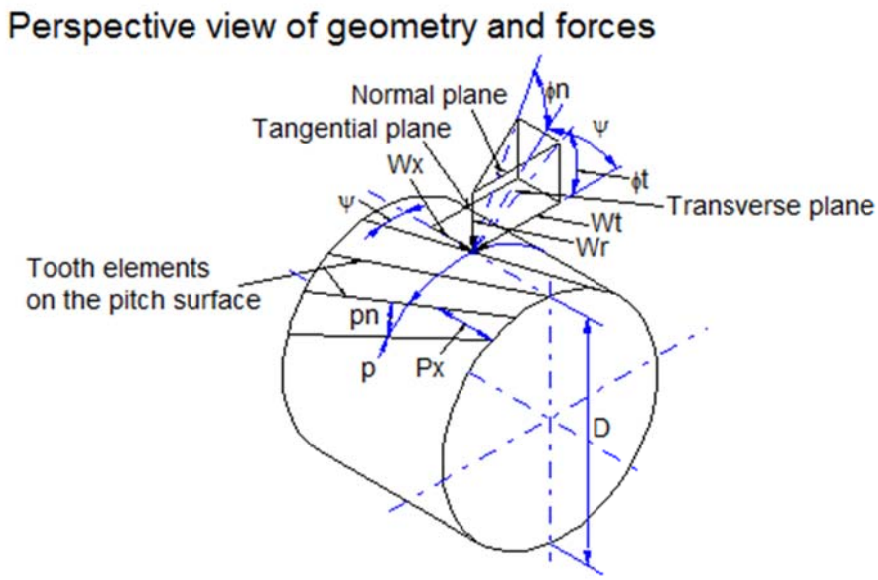
Nominal face width

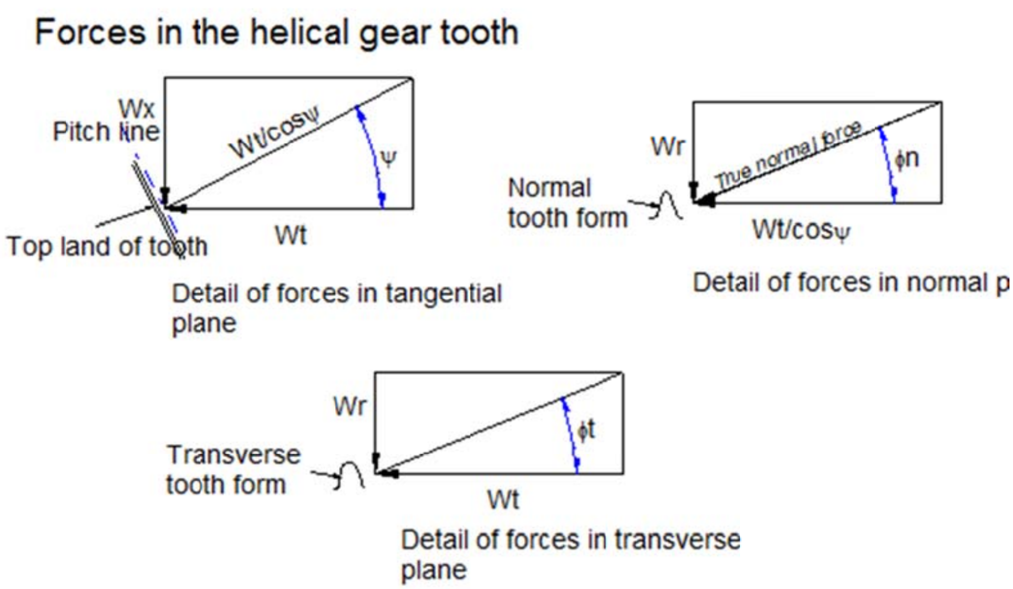
$$F \geq 2 \cdot (P_x)$$

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.





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

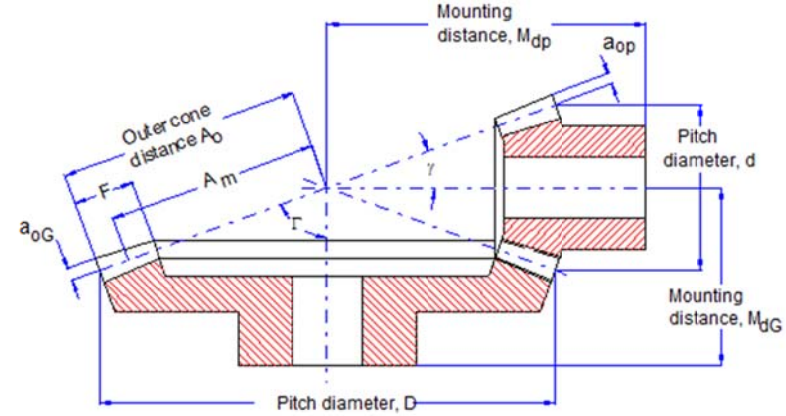
For certain combinations of number of teeth in a gear pair, there is interference between the tip of the teeth on the pinion and the fillet root of the teeth on the gear. Obviously this cannot be tolerated because the gears simply will not mesh.

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.

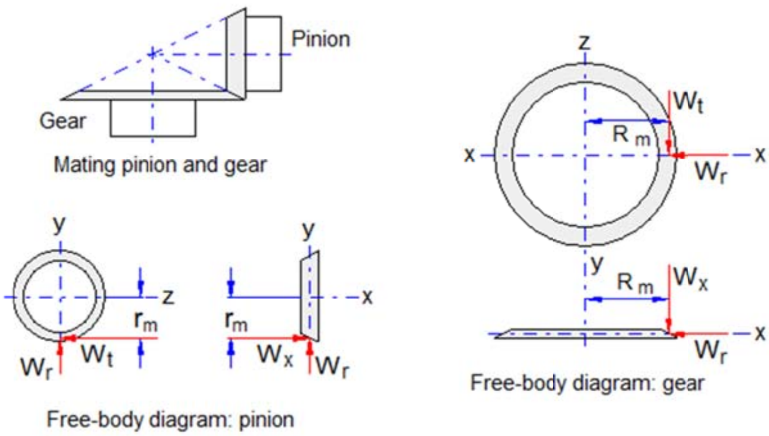
The minimum number of teeth for straight bevel gears is typically 13. The Gleason Works of Rochester, N.Y., has done an excellent job of standardizing the designs of these kinds of gears. The various Gleason systems have the amount of addendum for the gear and the pinion worked out so as to avoid undercut 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

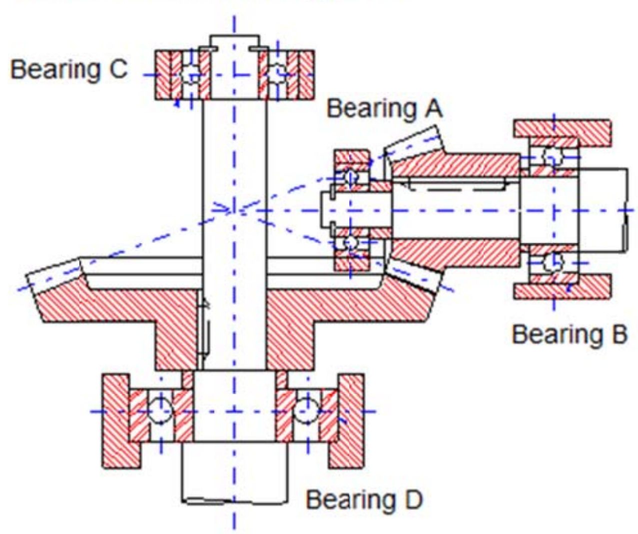
Key dimensions of straight bevel gear pair



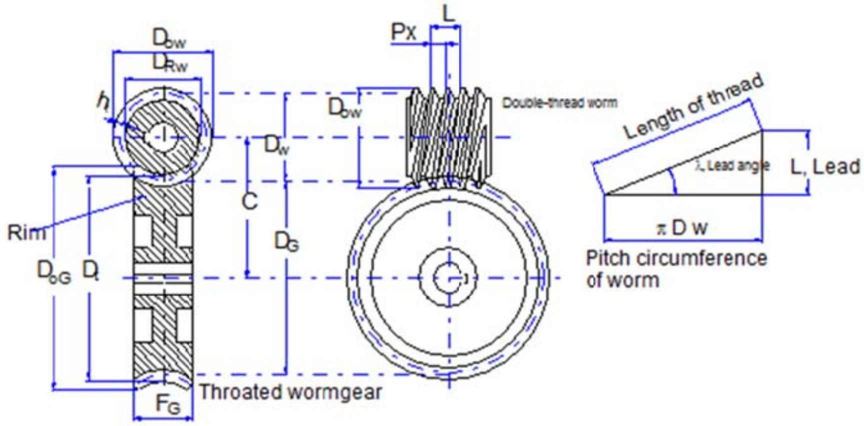
Forces on bevel gears



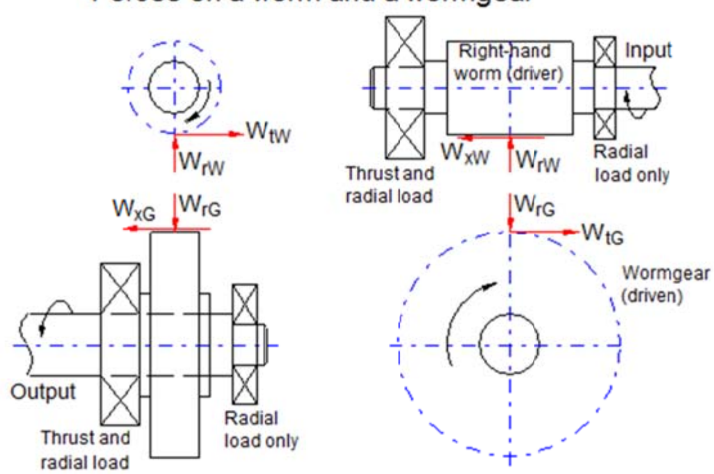
Straddle mounted gears



Wormgearing Design



Forces on a worm and a wormgear



$$L = N_W P_X$$

$$\tan\lambda = \frac{L}{\pi \cdot D_W}$$

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.

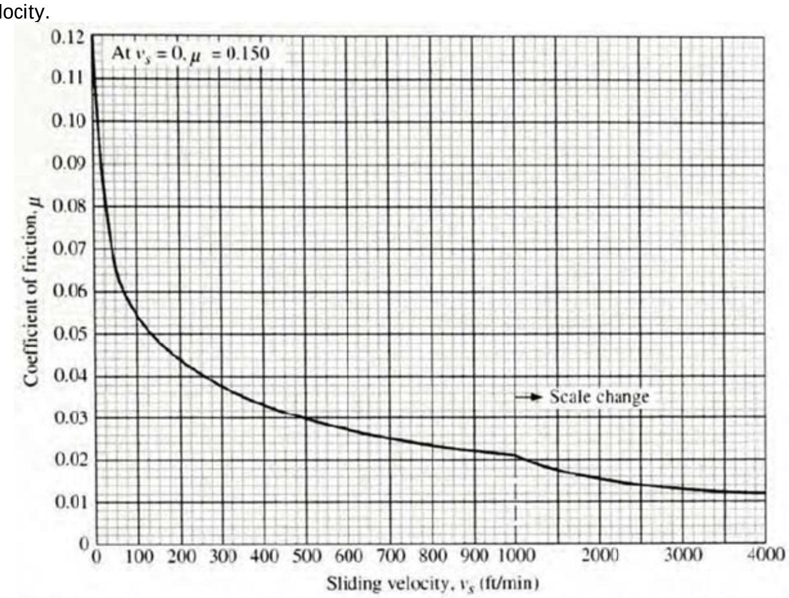
the sliding velocity is

$$V_s = \frac{V_{tG}}{\sin \lambda}$$

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;

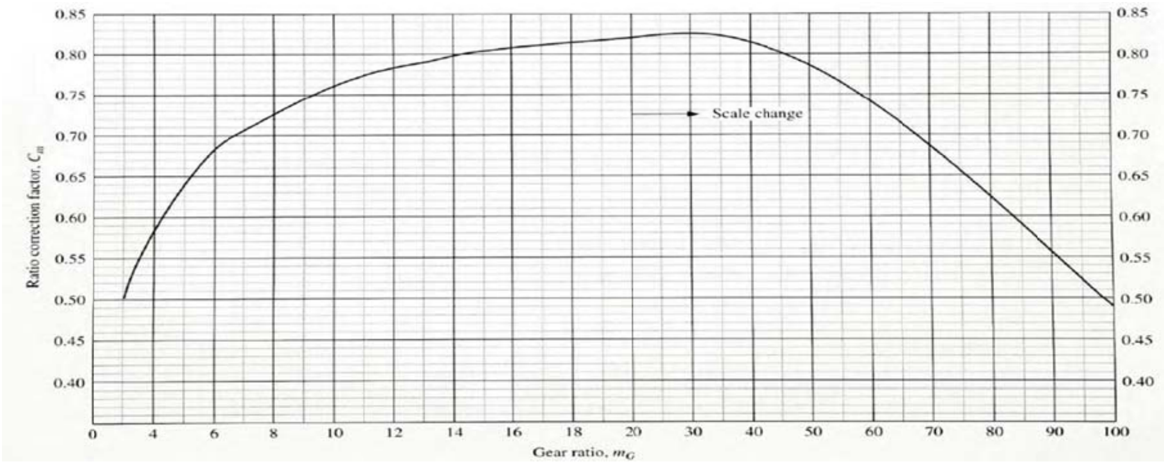
F_e = effective face width, in inches. Use the actual face width of the wormgear up to a maximum of $0.67 \cdot D_W$;

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

$$C_V = 13.31 \cdot e^{(-0.571)}$$

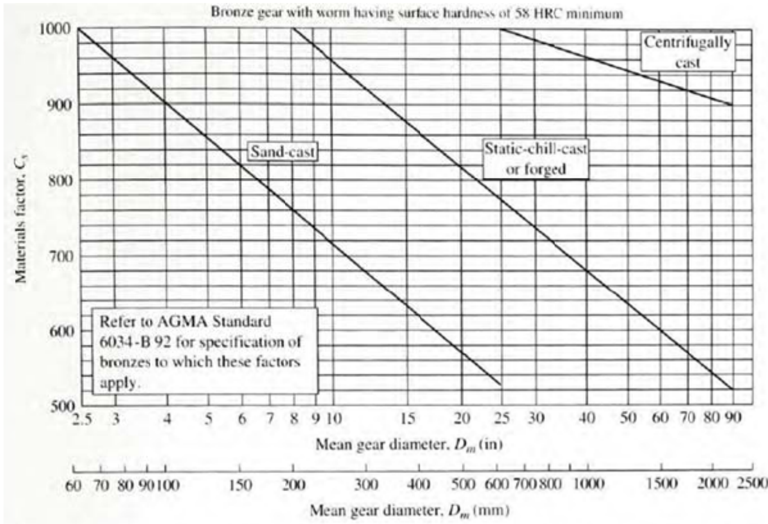
- For V_S > 3000 ft/min

$$C_V = 65.52 \cdot e^{(-0.774)}$$

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

$$\text{Torque} = \text{power/rotational speed} = \frac{P}{n}$$

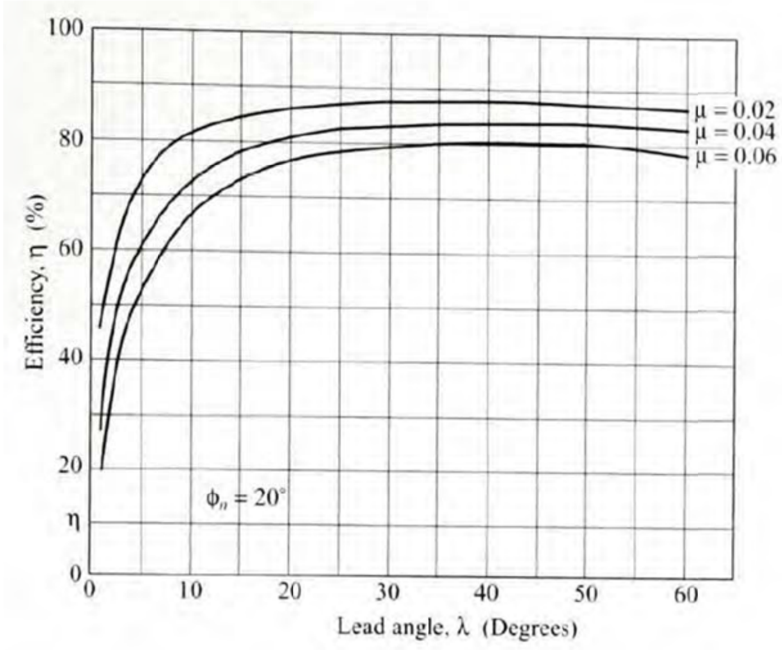
$$P_L = \frac{V_s \cdot W_f}{33000}$$

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°