Table 6.4 Preferred values for the module *m*

m	0.5	0.8	1	1.25	1.5	2	2.5	3	4	5
	6	8	10	12	16	20	25	32	40	50

Table 6.5 Preferred standard gear teeth numbers

12	13	14	15	16	18	20	22	24	25	28	30
32	34	38	40	45	50	54	60	64	70	72	75
80	84	90	96	100	120	140	150	180	200	220	250

h

Introduction to gear stresses 6.5.1

Gears experience two principle types of stress, hetbending stress at the root of the teeth due to nkatransmitted load and contact stresses on the fl deof the teeth due to repeated impact, or sustain Acontact, of one tooth surface against another. ssimple method of calculating bending stresses i presented in Section 6.5.2 and this is utilized within a gear selection procedure given in Section 6.6. However, this methodology, whilst suitable for determining the geometry for a gear train at the conceptual phase is inadequate for designing a gear train for highly stressed applications or where the life of the components is important. For the determination of life-critical parts, national and international standards for gears are available such as the AGMA, BS and ISO standards for gears. These are considered in Chapter 7 and the AGMA equations recommended for bending and contact stress for spur gears are introduced.

6.5.2 Bending stresses

The calculation of bending stress in gear teeth can be based on the Lewis formula

$$\sigma = \frac{W_t}{FmY} \tag{6.19}$$

where W_t is transmitted load (N); F is face width (m or mm); m is module (m or mm); and Y is the Lewis form factor and can be found from Table 6.6.

When teeth mesh, the load is delivered to the teeth with some degree of impact. The velocity factor is used to account for this and is given, in the case of cut or milled profile gears, by the Barth equation, 6.20.

$$K_v = \frac{6.1}{6.1 + V} \tag{6.20}$$

Table 6.6 Values for the Lewis form factor Y defined for two different tooth standards (Mitchener and Mabie, 1982)

N, number of teeth	Y ($\phi = 20°$; $a = 0.8\,m$; $b = m$)	Y ($\phi = 20°$; $a = m$; $b = 1.25\,m$)
12	0.33512	0.22960
13	0.34827	0.24317
14	0.35985	0.25530
15	0.37013	0.26622
16	0.37931	0.27610
17	0.38757	0.28508
18	0.39502	0.29327
19	0.40179	0.30078
20	0.40797	0.30769
21	0.41363	0.31406
22	0.41883	0.31997
24	0.42806	0.33056
26	0.43601	0.33979
28	0.44294	0.34790
30	0.44902	0.35510
34	0.45920	0.36731
38	0.46740	0.37727
45	0.47846	0.39093
50	0.48458	0.39860
60	0.49391	0.41047
75	0.50345	0.42283
100	0.51321	0.43574
150	0.52321	0.44930
300	0.53348	0.46364
Rack	0.54406	0.47897

a, addendum; b, dedendum; ϕ, pressure angle; m, module.

where V is the pitch line velocity which is given by

$$V = \frac{d}{2} \times 10^{-3} n \frac{2\pi}{60} \qquad (6.21)$$

where d is in mm and n in rpm.

Introducing the velocity factor into the Lewis equation gives:

$$\sigma = \frac{W_t}{K_v F m Y} \qquad (6.22)$$

Equation 6.22 forms the basis of a simple approach to the calculation of bending stresses in gears.

Example 6.5

A 20° full depth spur pinion is to transmit 1.25 kW at 850 rpm. The pinion has 18 teeth. Determine the Lewis bending stress if the module is 2 and the face width is 25 mm.

Solution

Calculating the pinion pitch diameter

$$d_P = mN_P = 2 \times 18 = 36 \text{ mm}$$

Calculating the pitch line velocity

$$V = \frac{d_P}{2} \times 10^{-3} \times n \frac{2\pi}{60}$$
$$= \frac{0.036}{2} \times 850 \times 0.1047 = 1.602 \text{ m/s}$$

Calculating the velocity factor

$$K_v = \frac{6.1}{6.1 + V} = \frac{6.1}{6.1 + 1.602} = 0.7920$$

Calculating the transmitted load

$$W_t = \frac{\text{Power}}{V} = \frac{1250}{1.602} = 780.2 \text{ N}$$

From Table 6.6, for $N_P = 18$, the Lewis form factor $Y = 0.29327$.

The Lewis Equation for bending stress gives:

$$\sigma = \frac{W_t}{K_v F m Y}$$
$$= \frac{780.2}{0.792 \times 0.025 \times 0.002 \times 0.29327}$$
$$= 67.18 \times 10^6 \text{ N/m}^2 = 67.18 \text{ MPa.}$$

6.6 Simple gear selection procedure

The Lewis formula in the form given by Eq. 6.22 ($\sigma = W_t/(K_v F m Y)$) can be used in a provisional spur gear selection procedure for a given transmission power, input and output speeds. The procedure is outlined below.

1. Select the number of teeth for the pinion and the gear to give the required gear ratio (observe the guidelines presented in Table 6.2 for

Table 6.7 Spur gears: 1.0 module, heavy duty steel 817M40, 655M13, face width 15 mm

Part number	Teeth	PCD (mm)	Outer diameter (mm)	Boss diameter (mm)	Bore diameter (mm)	Part number	Teeth	PCD (mm)	Outer diameter (mm)	Boss diameter (mm)	Bore diameter (mm)
SG1-9	9	10.00	12.00	12	6	SG1-46	46	46.00	48.00	30	8
SG1-10	10	11.00	13.00	13	6	SG1-47	47	47.00	49.00	30	8
SG1-11	11	12.00	14.00	14	6	SG1-48	48	48.00	50.00	30	8
SG1-12	12	12.00	14.00	14	6	SG1-49	49	49.00	51.00	30	8
SG1-13	13	13.00	15.00	15	6	SG1-50	50	50.00	52.00	35	10
SG1-14	14	14.00	16.00	16	6	SG1-51	51	51.00	53.00	35	10
SG1-15	15	15.00	17.00	17	6	SG1-52	52	52.00	54.00	35	10
SG1-16	16	16.00	18.00	18	8	SG1-53	53	53.00	55.00	35	10
SG1-17	17	17.00	19.00	18	8	SG1-54	54	54.00	56.00	35	10
SG1-18	18	18.00	20.00	18	8	SG1-55	55	55.00	57.00	35	10
SG1-19	19	19.00	21.00	18	8	SG1-56	56	56.00	58.00	35	10
SG1-20	20	20.00	22.00	20	8	SG1-57	57	57.00	59.00	35	10
SG1-21	21	21.00	23.00	20	8	SG1-58	58	58.00	60.00	35	10
SG1-22	22	22.00	24.00	20	8	SG1-59	59	59.00	61.00	35	10
SG1-23	23	23.00	25.00	20	8	SG1-60	60	60.00	62.00	35	10
SG1-24	24	24.00	26.00	20	8	SG1-61	61	61.00	63.00	35	10
SG1-25	25	25.00	27.00	20	8	SG1-62	62	62.00	64.00	35	10
SG1-26	26	26.00	28.00	25	8	SG1-63	63	63.00	65.00	35	10
SG1-27	27	27.00	29.00	25	8	SG1-64	64	64.00	66.00	35	10
SG1-28	28	28.00	30.00	25	8	SG1-65	65	65.00	67.00	35	10
SG1-29	29	29.00	31.00	25	8	SG1-66	66	66.00	68.00	35	10
SG1-30	30	30.00	32.00	25	8	SG1-68	68	68.00	70.00	35	10
SG1-31	31	31.00	33.00	25	8	SG1-70	70	70.00	72.00	35	10
SG1-32	32	32.00	34.00	25	8	SG1-72	72	72.00	74.00	35	10
SG1-33	33	33.00	35.00	25	8	SG1-74	74	74.00	76.00	45	10
SG1-34	34	34.00	36.00	25	8	SG1-76	76	76.00	78.00	45	10
SG1-35	35	35.00	37.00	25	8	SG1-78	78	78.00	80.00	45	10
SG1-36	36	36.00	38.00	30	8	SG1-80	80	80.00	82.00	45	10
SG1-37	37	37.00	39.00	30	8	SG1-84	84	84.00	86.00	45	10
SG1-38	38	38.00	40.00	30	8	SG1-88	88	88.00	90.00	45	10
SG1-39	39	39.00	41.00	30	8	SG1-90	90	90.00	92.00	45	10
SG1-40	40	40.00	42.00	30	8	SG1-96	96	96.00	98.00	45	10
SG1-41	41	41.00	43.00	30	8	SG1-100	100	100.00	102.00	45	10
SG1-42	42	42.00	44.00	30	8	SG1-112	112	112.00	114.00	45	10
SG1-43	43	43.00	45.00	30	8	SG1-120	120	120.00	122.00	45	10
SG1-44	44	44.00	46.00	30	8	SG1-130	130	130.00	132.00	45	10
SG1-45	45	45.00	47.00	30	8	SG1-150	150	150.00	152.00	45	10

maximum gear ratios). Note that the minimum number of teeth permissible when using a pressure angle of 20° is 18 (Table 6.3). Use either the standard teeth numbers as listed in Table 6.5 or as listed in a stock gear catalogue.

2. Select a material. This will be limited to those listed in the stock gear catalogues.
3. Select a module, m from Table 6.4 or as listed in a stock gear catalogue (see Tables 6.7 to 6.10 which give examples of a selection of stock gears available).
4. Calculate the pitch diameter, $d = mN$.
5. Calculate the pitch line velocity, $V = (d/2) \times n \times (2\pi/60)$. Ensure this does not exceed the guidelines given in Table 6.2.
6. Calculate the dynamic factor, $K_v = 6/(6 + V)$.
7. Calculate the transmitted load, $W_t = \text{Power}/V$.
8. Calculate an acceptable face width using the Lewis formula in the form,

$$F = \frac{W_t}{K_v m Y \sigma_p} \tag{6.23}$$

Table 6.8 Spur gears: 1.5 module, heavy duty steel 817M40, 655M13, face width 20 mm

Part number	Teeth	PCD (mm)	Outer diameter (mm)	Boss diameter (mm)	Bore diameter (mm)	Part number	Teeth	PCD (mm)	Outer diameter (mm)	Boss diameter (mm)	Bore diameter (mm)
SG1.5-9	9	15.00	18.00	18	8	SG1.5-47	47	70.50	73.50	50	15
SG1.5-10	10	16.50	19.50	19.5	8	SG1.5-48	48	72.00	75.00	50	15
SG1.5-11	11	18.00	21.00	21	8	SG1.5-49	49	73.50	76.50	50	15
SG1.5-12	12	18.00	21.00	21	8	SG1.5-50	50	75.00	78.00	50	15
SG1.5-13	13	19.50	22.50	20	8	SG1.5-51	51	76.50	79.50	50	15
SG1.5-14	14	21.00	24.00	20	8	SG1.5-52	52	78.00	81.00	50	15
SG1.5-15	15	22.50	25.50	20	8	SG1.5-53	53	79.50	82.50	60	15
SG1.5-16	16	24.00	27.00	25	10	SG1.5-54	54	81.00	84.00	60	15
SG1.5-17	17	25.50	28.50	25	10	SG1.5-55	55	82.50	85.50	60	15
SG1.5-18	18	27.00	30.00	25	10	SG1.5-56	56	84.00	87.00	60	15
SG1.5-19	19	28.50	31.50	25	10	SG1.5-57	57	85.50	88.50	60	15
SG1.5-20	20	30.00	33.00	25	10	SG1.5-58	58	87.00	90.00	60	15
SG1.5-21	21	31.50	34.50	25	10	SG1.5-59	59	88.50	91.50	60	15
SG1.5-22	22	33.00	36.00	30	10	SG1.5-60	60	90.00	93.00	60	15
SG1.5-23	23	34.50	37.50	30	10	SG1.5-62	62	93.00	96.00	60	15
SG1.5-24	24	36.00	39.00	30	10	SG1.5-64	64	96.00	99.00	60	15
SG1.5-25	25	37.50	40.50	30	10	SG1.5-65	65	97.50	100.50	60	15
SG1.5-26	26	39.00	42.00	30	10	SG1.5-66	66	99.00	102.00	60	15
SG1.5-27	27	40.50	43.50	30	10	SG1.5-68	68	102.00	105.00	60	15
SG1.5-28	28	42.00	45.00	30	10	SG1.5-70	70	105.00	108.00	60	15
SG1.5-29	29	43.50	46.50	30	10	SG1.5-71	71	106.50	109.50	60	15
SG1.5-30	30	45.00	48.00	30	10	SG1.5-72	72	108.00	111.00	60	15
SG1.5-31	31	46.50	49.50	30	10	SG1.5-73	73	109.50	112.50	60	15
SG1.5-32	32	48.00	51.00	30	10	SG1.5-74	74	111.00	114.00	60	15
SG1.5-33	33	49.50	52.50	30	10	SG1.5-75	75	112.50	115.50	60	15
SG1.5-34	34	51.00	54.00	30	10	SG1.5-76	76	114.00	117.00	60	15
SG1.5-35	35	52.50	55.50	50	15	SG1.5-78	78	117.00	120.00	75	15
SG1.5-36	36	54.00	57.00	50	15	SG1.5-80	80	120.00	123.00	75	15
SG1.5-37	37	55.50	58.50	50	15	SG1.5-86	86	129.00	132.00	75	15
SG1.5-38	38	57.00	60.00	50	15	SG1.5-90	90	135.00	138.00	75	15
SG1.5-39	39	58.50	61.50	50	15	SG1.5-96	96	144.00	147.00	75	15
SG1.5-40	40	60.00	63.00	50	15	SG1.5-98	98	147.00	150.00	75	15
SG1.5-41	41	61.50	64.50	50	15	SG1.5-100	100	150.00	153.00	75	15
SG1.5-42	42	63.00	66.00	50	15	SG1.5-105	105	157.50	160.50	75	15
SG1.5-43	43	64.50	67.50	50	15	SG1.5-110	110	165.00	168.00	75	15
SG1.5-44	44	66.00	69.00	50	15	SG1.5-115	115	175.50	178.50	75	15
SG1.5-45	45	67.50	70.50	50	15	SG1.5-120	120	180.00	183.00	75	15
SG1.5-46	46	69.00	72.00	50	15						

The Lewis form factor, Y, can be obtained from Table 6.6.

The permissible bending stress, σ_p, can be taken as σ_{uts}/factor of safety, where the factor of safety is set by experience, but may range from 2 to 5. Alternatively use values of σ_p as listed for the appropriate material in a stock gear catalogue. Certain plastics are suitable for use as gear materials in application where low weight, low friction, high corrosion resistance, low wear and quiet operation are beneficial. The strength of plastic is usually significantly lower than that of metals. Plastics are often formed using a filler to improve strength, wear, impact resistance, temperature performance, as well as other properties and it is therefore difficult to regulate or standardize properties for plastics and these need to be obtained instead from the manufacturer or a

Table 6.9 Spur gears: 2.0 module, heavy duty steel 817M40, 655M13, face width 25 mm

Part number	Teeth	PCD (mm)	Outer diameter (mm)	Boss diameter (mm)	Bore diameter (mm)	Part number	Teeth	PCD (mm)	Outer diameter (mm)	Boss diameter (mm)	Bore diameter (mm)
SG2-9	9	20.00	24.00	24	12	SG2-47	47	94.00	98.00	60	20
SG2-10	10	22.00	26.00	26	12	SG2-48	48	96.00	100.00	60	20
SG2-11	11	24.00	28.00	28	12	SG2-49	49	98.00	102.00	60	20
SG2-12	12	24.00	28.00	28	12	SG2-50	50	100.00	104.00	60	20
SG2-13	13	26.00	30.00	30	12	SG2-51	51	102.00	106.00	60	20
SG2-14	14	28.00	32.00	30	12	SG2-52	52	104.00	108.00	60	20
SG2-15	15	30.00	34.00	30	12	SG2-53	53	106.00	110.00	60	20
SG2-16	16	32.00	36.00	30	12	SG2-54	54	108.00	112.00	60	20
SG2-17	17	34.00	38.00	35	15	SG2-55	55	110.00	114.00	60	20
SG2-18	18	36.00	40.00	35	15	SG2-56	56	112.00	116.00	60	20
SG2-19	19	38.00	42.00	35	15	SG2-57	57	114.00	118.00	60	20
SG2-20	20	40.00	44.00	35	15	SG2-58	58	116.00	120.00	60	20
SG2-21	21	42.00	46.00	35	15	SG2-59	59	118.00	122.00	60	20
SG2-22	22	44.00	48.00	35	15	SG2-60	60	120.00	124.00	60	20
SG2-23	23	46.00	50.00	35	15	SG2-62	62	124.00	128.00	75	20
SG2-24	24	48.00	52.00	35	15	SG2-64	64	128.00	132.00	75	20
SG2-25	25	50.00	54.00	35	15	SG2-65	65	130.00	134.00	75	20
SG2-26	26	52.00	56.00	35	15	SG2-66	66	132.00	136.00	75	20
SG2-27	27	54.00	58.00	50	20	SG2-68	68	136.00	140.00	75	20
SG2-28	28	56.00	60.00	50	20	SG2-70	70	140.00	144.00	75	20
SG2-29	29	58.00	62.00	50	20	SG2-71	71	142.00	146.00	75	20
SG2-30	30	60.00	64.00	50	20	SG2-72	72	144.00	148.00	75	20
SG2-31	31	62.00	66.00	50	20	SG2-73	73	146.00	150.00	75	20
SG2-32	32	64.00	68.00	50	20	SG2-74	74	148.00	152.00	75	20
SG2-33	33	66.00	70.00	50	20	SG2-75	75	150.00	154.00	75	20
SG2-34	34	68.00	72.00	50	20	SG2-76	76	152.00	156.00	75	20
SG2-35	35	70.00	74.00	50	20	SG2-78	78	156.00	160.00	100	20
SG2-36	36	72.00	76.00	50	20	SG2-80	80	160.00	164.00	100	20
SG2-37	37	74.00	78.00	50	20	SG2-86	86	172.00	176.00	100	20
SG2-38	38	76.00	80.00	50	20	SG2-90	90	180.00	184.00	100	20
SG2-39	39	78.00	82.00	50	20	SG2-96	96	192.00	196.00	100	20
SG2-40	40	80.00	84.00	50	20	SG2-98	98	196.00	200.00	100	20
SG2-41	41	82.00	86.00	50	20	SG2-100	100	200.00	204.00	100	20
SG2-42	42	84.00	88.00	50	20	SG2-105	105	210.00	214.00	100	20
SG2-43	43	86.00	90.00	60	20	SG2-110	110	220.00	224.00	100	20
SG2-44	44	88.00	92.00	60	20	SG2-115	115	230.00	234.00	100	20
SG2-45	45	90.00	94.00	60	20	SG2-120	120	240.00	244.00	100	20
SG2-46	46	92.00	96.00	60	20						

traceable testing laboratory. Plastics used for gears include acrylonitrile-butadiene-styrene (ABS), acetal, nylon, polycarbonate, polyester, polyurethane and styrene-acrylonitrile (SAN). Values of permissible bending stress for a few gear materials are listed in Table 6.11.

The design procedure consists of proposing teeth numbers for the gear and pinion, selecting a suitable material, selecting a module, calculating the various parameters as listed resulting in a value for the face width. If the face width is greater than those available in the stock gear catalogue or if the pitch line velocity is too high, repeat the process for a different module. If this does not provide a sensible solution try a different material, etc. The process can be optimized taking cost and other performance criteria into account if necessary and can be programmed.

Table 6.10 Spur gears: 3.0 module, heavy duty steel 817M40, 655M13, face width 35 mm

Part number	Teeth	PCD (mm)	Outer diameter (mm)	Boss diameter (mm)	Bore diameter (mm)	Part number	Teeth	PCD (mm)	Outer diameter (mm)	Boss diameter (mm)	Bore diameter (mm)
SG3-9	9	30.00	36.00	36	15	SG3-45	45	135.00	141.00	100	25
SG3-10	10	33.00	39.00	39	15	SG3-46	46	138.00	144.00	100	25
SG3-11	11	36.00	42.00	42	15	SG3-47	47	141.00	147.00	100	25
SG3-12	12	36.00	42.00	42	20	SG3-48	48	144.00	150.00	100	25
SG3-13	13	39.00	45.00	45	20	SG3-49	49	147.00	153.00	100	25
SG3-14	14	42.00	48.00	45	20	SG3-50	50	150.00	156.00	100	25
SG3-15	15	45.00	51.00	45	20	SG3-51	51	153.00	159.00	100	25
SG3-16	16	48.00	54.00	45	20	SG3-52	52	156.00	162.00	100	25
SG3-17	17	51.00	57.00	45	20	SG3-53	53	159.00	165.00	100	25
SG3-18	18	54.00	60.00	45	20	SG3-54	54	162.00	168.00	100	25
SG3-19	19	57.00	63.00	45	20	SG3-55	55	165.00	171.00	100	25
SG3-20	20	60.00	66.00	60	25	SG3-56	56	168.00	174.00	100	25
SG3-21	21	63.00	69.00	60	25	SG3-57	57	171.00	177.00	127	25
SG3-22	22	66.00	72.00	60	25	SG3-58	58	174.00	180.00	127	25
SG3-23	23	69.00	75.00	60	25	SG3-59	59	177.00	183.00	127	25
SG3-24	24	72.00	78.00	60	25	SG3-60	60	180.00	186.00	127	25
SG3-25	25	75.00	81.00	60	25	SG3-62	62	186.00	192.00	127	25
SG3-26	26	78.00	84.00	60	25	SG3-63	63	189.00	195.00	127	25
SG3-27	27	81.00	87.00	60	25	SG3-64	64	192.00	198.00	127	25
SG3-28	28	84.00	90.00	60	25	SG3-65	65	195.00	201.00	127	25
SG3-29	29	87.00	93.00	60	25	SG3-68	68	204.00	210.00	127	25
SG3-30	30	90.00	96.00	60	25	SG3-70	70	210.00	216.00	150	30
SG3-31	31	93.00	99.00	60	25	SG3-72	72	216.00	222.00	150	30
SG3-32	32	96.00	102.00	60	25	SG3-75	75	225.00	231.00	150	30
SG3-33	33	99.00	105.00	60	25	SG3-76	76	228.00	234.00	150	30
SG3-34	34	102.00	108.00	75	25	SG3-78	78	234.00	240.00	150	30
SG3-35	35	105.00	111.00	75	25	SG3-80	80	240.00	246.00	150	30
SG3-36	36	108.00	114.00	75	25	SG3-82	82	246.00	252.00	150	30
SG3-37	37	111.00	117.00	75	25	SG3-84	84	252.00	258.00	150	30
SG3-38	38	114.00	120.00	75	25	SG3-86	86	258.00	264.00	150	30
SG3-39	39	117.00	123.00	75	25	SG3-90	90	270.00	276.00	150	30
SG3-40	40	120.00	126.00	75	25	SG3-92	92	276.00	282.00	150	30
SG3-41	41	123.00	129.00	75	25	SG3-94	94	282.00	288.00	150	30
SG3-42	42	126.00	132.00	75	25	SG3-95	95	285.00	291.00	150	30
SG3-43	43	129.00	135.00	75	25	SG3-96	96	288.00	294.00	150	30
SG3-44	44	132.00	138.00	100	25						

Data adapted from HPC Gears Ltd.

Example 6.6

A gearbox is required to transmit 18 kW from a shaft rotating at 2650 rpm. The desired output speed is approximately 12 000 rpm. For space limitation and standardization reasons a double step-up gearbox is requested with equal ratios. Using the limited selection of gears presented in Tables 6.7 to 6.10, select suitable gears for the gear wheels and pinions.

Solution

$$\text{Overall ratio} = 12\,000/2650 = 4.528.$$

$$\text{First stage ratio} = \sqrt{4.528} = 2.128.$$

This could be achieved using a gear with 38 teeth and pinion with 18 teeth (ratio = 38/18 = 2.11).

The gear materials listed in Tables 6.7 to 6.10 are 817M40 and 655M13 steels. From

Table 6.11 Permissible bending stresses for various commonly used gear materials

Material	Treatment	σ_{uts} (MPa)	Permissible bending stress σ_P (MPa)
Nylon		65 (20°C)	27
Tufnol		110	31
080M40		540	131
080M40	Induction hardened	540	117
817M40		772	221
817M40	Induction hardened	772	183
045M10		494	117
045M10	Case hardened	494	276
655M13	Case hardened	849	345

Table 6.11, the 655M13 is the stronger steel and this is selected for this example prior to a more detailed consideration. For 655M13 case hardened steel gears, the permissible stress σ_p = 345 MPa.

Calculations for gear 1: $Y_{38} = 0.37727$, $n = 2650$ rpm.

m	1.5	2.0
$d = mN$ (mm)	57	76
$V = \frac{d}{2} \times 10^{-3} \times n\frac{2\pi}{60}$ (m/s)	7.9	10.5
$W_t = \frac{\text{Power}}{V}$ (N)	2276	1707
$K_v = \frac{6.1}{6.1 + V}$	0.4357	0.3675
$F = \frac{W_t}{K_v m Y \sigma_p}$ (m)	0.027	0.018

$m = 1.5$ gives a face width value greater than the catalogue value of 20 mm, so try $m = 2$. $m = 2$ gives a face width value less than the catalogue value of 25 mm, so design is OK.

Calculations for pinion 1: $Y_{18} = 0.29327$, $n = 5594$ rpm. (No need to do calculations for $m = 1.5$, because it has now been rejected.)

m	2.0
d (mm)	36
V (m/s)	10.5
W_t (N)	1707
K_v	0.3676
F (m)	0.023

$m = 1.5$ gives a face width value greater than the catalogue value of 20 mm, so try $m = 2$. $m = 2$ gives a face width value less than the catalogue value of 25 mm, so design is OK.

Calculations for gear 2: $Y_{38} = 0.37727$, $n = 5594$ rpm.

m	2.0
d (mm)	76
V (m/s)	22.26
W_t (N)	808.6
K_v	0.215
F (m)	0.0144

$m = 2$ gives a face width value lower than catalogue specification, so the design is OK.

Calculations for pinion 2: $Y_{18} = 0.29327$, $n = 11\,810$ rpm.

m	2.0
d (mm)	36
V (m/s)	22.26
W_t (N)	808.6
K_v	0.215
F (m)	0.0186

$m = 2$ gives a face width value lower than catalogue specification, so the design is OK. A general arrangement for the solution is shown in Figure 6.27.

Example 6.7

A gearbox for a cordless hand tool, Figure 6.28, is required to transmit 5 W from an electric motor running at 3000 rpm to an output drive running at approximately 75 rpm. The specification for the gearbox is that it should be reliable, efficient, maintenance free for the life of the device, light and fit within an enclosure of 50 mm diameter. The length is not critical.

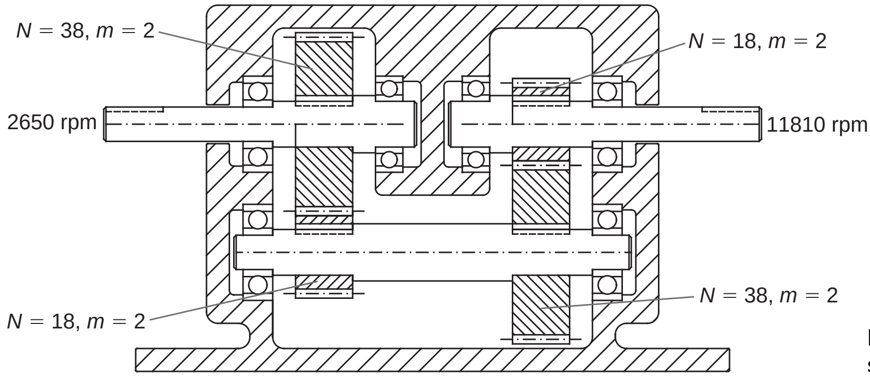

Figure 6.27 Step up gearbox solution.

Worksheet

6.1 For the gear train illustrated in Figure 6.30, determine the output speed and direction of rotation if the input shaft rotates at 1490 rpm clockwise. Gears A to D have a module of 1.5 and gears E to H a module of 2.

6.2 What type of gear could be selected for the following applications.
   (a) To transmit power between two shafts with intersecting axes at 90° to each other.
   (b) To transmit power between two shafts with non-intersecting axes at 90° to each other.

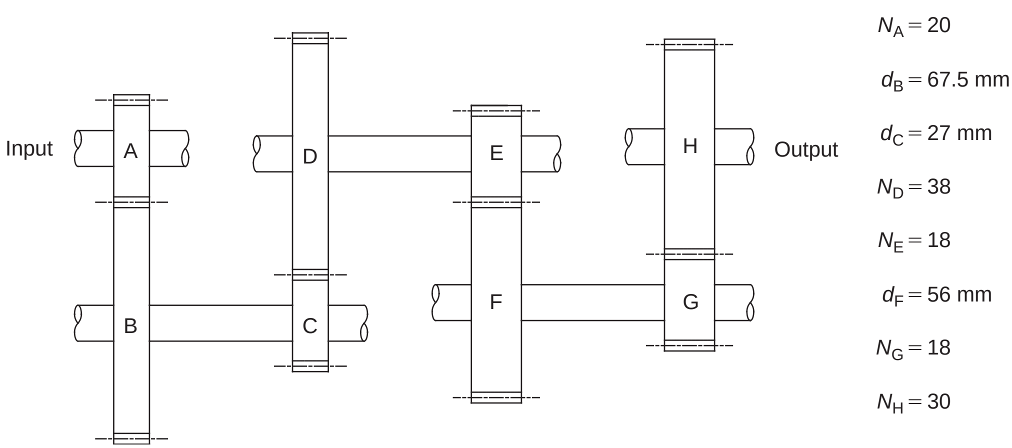

Figure 6.30 Gear train.

(c) To transmit 30 kW between parallel axis shafts. What material might be specified for these gears?
(d) To transmit a low power load in an epicyclic gearbox. What material might be specified for this application?

6.3 Determine the ratio, defined as the sun/arm speed ratio, and the speed and direction of rotation of the planet carrier for the epicyclic gearbox illustrated in Figure 6.31. The sun, planets and annulus gears have 40, 20 and 80 teeth, respectively. The sun gear is the input and rotates at 3000 rpm anticlockwise. The planet carrier is the output. The annulus gear is held stationary.

6.4 Determine the ratio, defined as the sun/arm speed ratio, and the speed and direction of the planet carrier for the epicyclic gear box illustrated in Figure 6.32. The sun, planets and annulus gears have 24, 27 and 78 teeth, respectively. The sun gear is rotating at 5000 rpm clockwise. The sun gear is the input and the planet carrier the output. The annulus gear is held stationary.

6.5 A 20° full depth spur pinion is to transmit 1.75 kW at 1200 rpm. If the pinion has 18 teeth with a module of 2, determine a suitable value for the face width, based on the Lewis formula, if the bending stress should not exceed 75 MPa.

6.6 A 20° full depth spur pinion is required to transmit 1.8 kW at a speed of 1100 rpm. If the pinion has 18 teeth and is manufactured from heavy duty 817M40 steel select a suitable gear from the limited choice given in Tables 6.7–6.10, specifying the module and face width based on the Lewis formula.

6.7 A gearbox is required to transmit 6 kW from a shaft rotating at 2200 rpm. The desired output speed is approximately 300 rpm. Using the Lewis formula to determine gear strength based upon bending stresses select suitable gears from the limited selection presented in Tables 6.7–6.10.

6.8 A 20° full depth spur pinion is required to transmit 1.5 kW at a speed of 1230 rpm. If the pinion has 21 teeth and is manufactured from heavy duty 817M40 steel select a suitable gear from the limited choice given in Tables 6.7–6.10, specifying the module and face width based on the Lewis formula.

6.9 A gearbox is required to transmit 5.5 kW from a shaft rotating at 2000 rpm. The desired output speed is approximately 350 rpm. The output must rotate in the same direction as the input and along the same axis. Select suitable gears.

6.10 A double reduction gearbox is required to transmit 15 kW from a shaft rotating at 1500 rpm. The desired output speed is approximately 200 rpm. Using the Lewis formula to determine gear strength based upon bending stresses select suitable gears from the limited selection presented in Tables 6.7–6.10.

6.11 A gearbox is required to transmit 8 kW from a shaft rotating at 1470 rpm. The desired output speed is approximately 10 000 rpm. The output must rotate in the same direction as the input and along the same axis. Select and specify appropriate gears for the gearbox.

6.12 An epicyclic gearbox is required to transmit 12 kW from a shaft rotating at 3000 rpm. The desired output speed is approximately 500 rpm. Using a module 3 hardened steel gear, determine the number of teeth in each gear for a single stage reduction gearbox.

Answers

6.1 121 rpm, clockwise.
6.2 (a) Bevel; (b) crossed axis helical gears; (c) helical gears, case hardened steel; (d) spur gears, Tufnol
6.3 3, 1000 rpm, anti-clockwise
6.4 4.25, 1176 rpm, clockwise
6.5 25 mm
6.6 $m = 2$, $F = 25$ mm would be acceptable.
6.7 No unique solution. $m = 2$, $F = 25$ mm, $N_1 = 18$, $N_2 = 48$, $N_3 = 18$, $N_4 = 48$, case hardened 655M13 steel would be acceptable.
6.8 No unique solution. $m = 2$, $N = 21$, $F = 25$ mm would be acceptable.
6.9 No unique solution.
6.10 No unique solution.
6.11 No unique solution.
6.12 No unique solution. $N_S = 18$, $N_P = 36$, $N_R = 90$ would be acceptable.