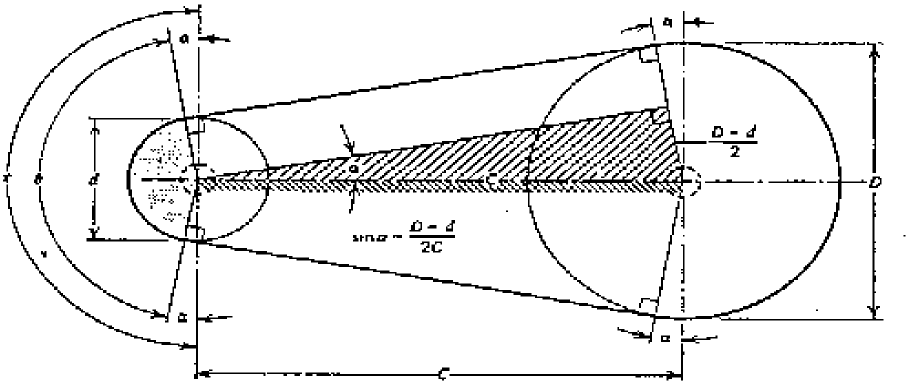
BELT DRIVES

Belt drives are used when large distance between shafts make gears impractical or when the designated speed is too high for chain drives. They can be used only when some variation in speed can be tolerated.

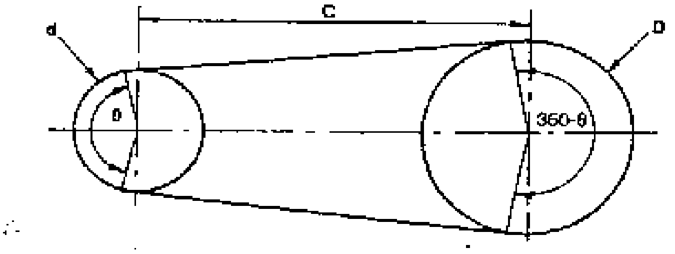


angle of contact θ (in degrees) $= 180° - 2\alpha$

θ (in radians) $\simeq \pi - \frac{D-d}{C}$

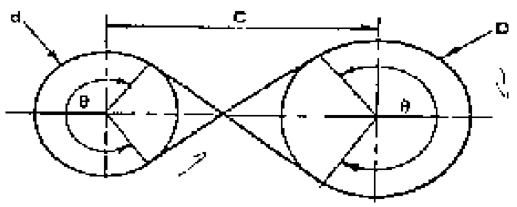
centre distance $C = \begin{cases} D + 1.5d & \text{for } \frac{D}{d} < 3 \\ D & \text{for } \frac{D}{d} > 3 \end{cases}$

OPEN BELT DRIVES



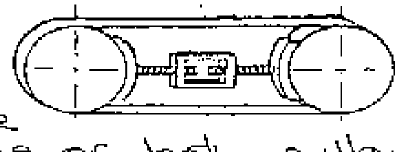
1. both pulleys rotate in the **same** direction
2. $\theta \simeq \pi - \frac{D-d}{C}$
3. length of belt is given by:- $L \simeq \frac{\pi}{2}(D+d) + 2C + \frac{(D-d)^2}{4C}$

CROSSED BELT DRIVES



1. the two pulleys rotate in **opposite** directions
2. $\theta \simeq \pi + \frac{D+d}{C}$
3. length of belt is given by:- $L \simeq \frac{\pi}{2}(D+d) + 2C + \frac{(D+d)^2}{4C}$

To function properly, belt drives must maintain certain tension levels. Too much tension shortens belt life (due to fatigue) as well as the life of other drive components such as shaft bearings. Too little tension allows slip generating heat and wear that also reduce component life. Adequate tension can be achieved by adjusting one or both pulleys.



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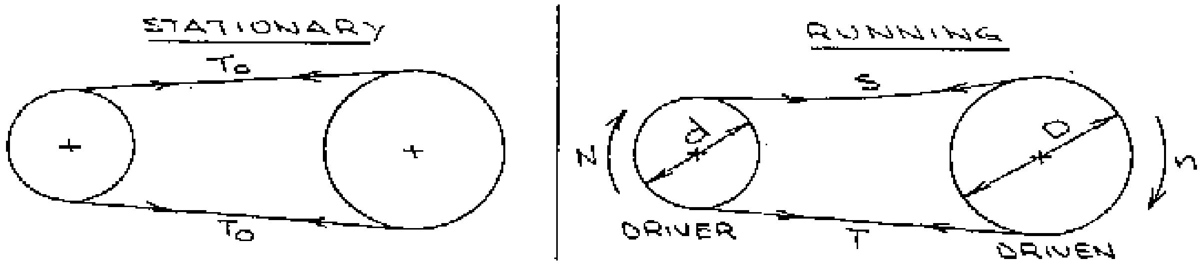


$$\boxed{2T_o = T + S}$$

where T_o = initial tension (N)
T = tight side tension (N)
S = slack side tension (N)

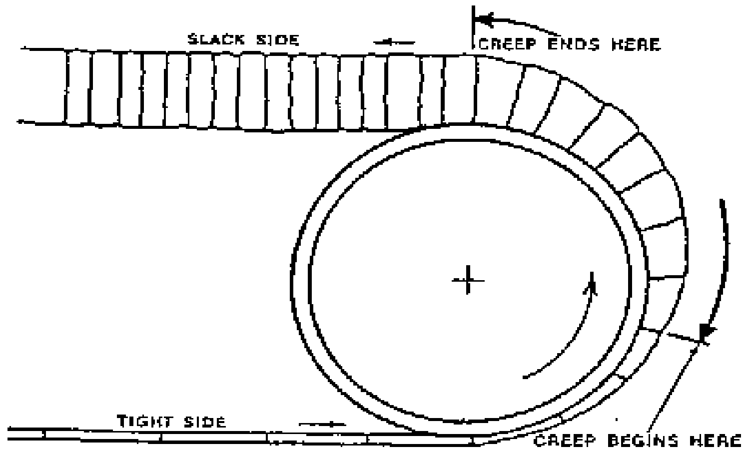
$$\text{speed ratio} = \frac{\text{speed of driver}}{\text{speed of driven}} = \frac{N}{n} = \frac{D}{d}$$

for no slip and creep –

$$n = N \times \frac{d}{D}$$

$$\text{speed of driven} = \text{speed of driver} \times \frac{\text{diam. of driver}}{\text{diam. of driven}}$$

Slip and creep cause a reduction in speed. Slip occurs mainly at the smaller pulley. Creep is a slight relative movement between belt and pulley that causes a loss of driven speed.

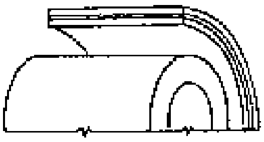
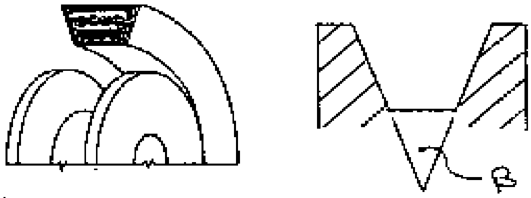


for 2% slip at driver pulley, $n = N \times \frac{d}{D}(1 - 0{,}02)$

for 2% slip at driver pulley and 1% slip at driven pulley $n = N \times \frac{d}{D}(1 - 0{,}02)(1 - 0{,}01)$

Loss of speed due to slip can be compensated for by making a small change in one pulley diameter.

Ratio of tensions

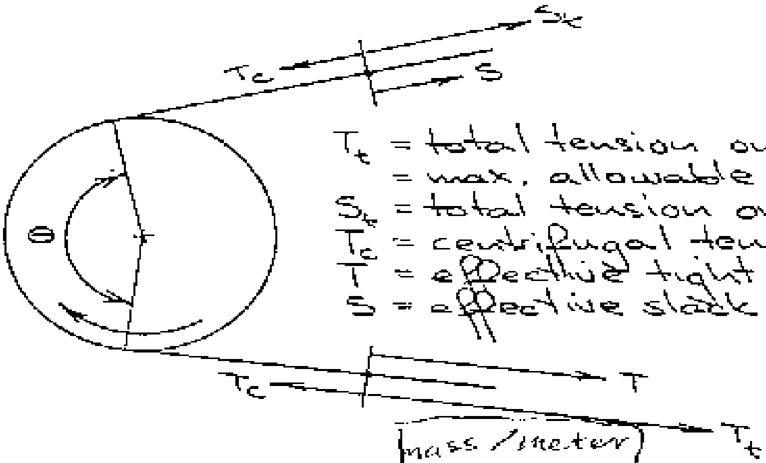
FLAT BELTS	V-BELTS
	

disregarding centrifugal tension

$$\frac{T}{S} = e^{\mu\theta} \qquad\qquad \frac{T}{S} = e^{\frac{\mu\theta}{\sin\beta/2}}$$

allowing for centrifugal tension

$$\frac{T_t - T_c}{S_t - T_c} = e^{\mu\theta} \qquad\qquad \frac{T_t - T_c}{S_t - T_c} = e^{\frac{\mu\theta}{\sin\beta/2}}$$

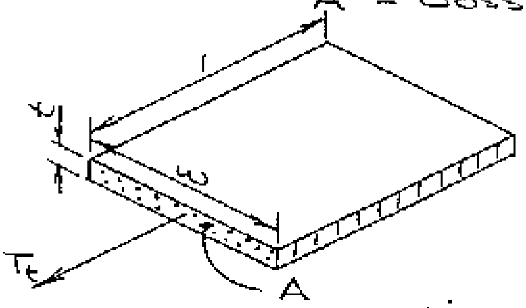


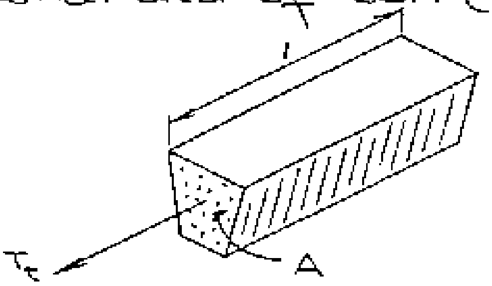
θ = angle of wrap on smaller pulley

T_t = total tension on tight side = max. allowable pull in belt
S_t = total tension on slack side
T_c = centrifugal tension
T = effective tight side tension
S = effective slack side tension

centrifugal tension $T_c = mv^2 = \rho \times A \times l \times v^2$

where m = mass per metre of belt (kg/m)
v = peripheral speed of belt (m/s) $= \frac{\pi d N}{60}$
ρ = density of belt material (kg/m³)
A = cross-sectional area of belt (m²)





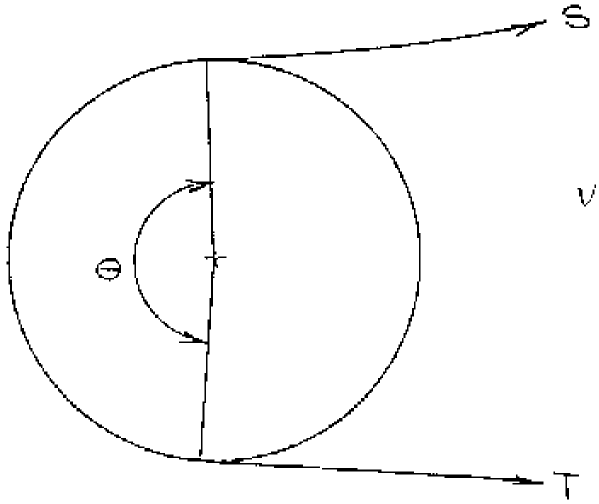
allowable tight side tension = cross-sectional area × allowable stress

$$T_t = A \times \sigma$$

Power transmitted by belts

$$\text{power} \quad P = (T - S)\,v$$

where T = effective tight side tension $= (T_t - T_c)$
S = effective slack side tension $= (S_t - T_c)$
v = peripheral speed of belt (m/s)



$v = \pi$ ([illegible]) N

FLAT BELTS	V-BELTS
$\frac{T}{S} = e^{\mu\theta}$	$\frac{T}{S} = e^{\frac{\mu\theta}{\sin\beta/2}}$
giving $S = \frac{T}{e^{\mu\theta}}$	giving $S = \frac{T}{e^{\frac{\mu\theta}{\sin\beta/2}}}$
$\therefore P = T\left(1 - \frac{1}{e^{\mu\theta}}\right)v$	$\therefore P = T\left(1 - \frac{1}{e^{\frac{\mu\theta}{\sin\beta/2}}}\right)v$
$\boxed{P = (T_t - T_c)\left(1 - \frac{1}{e^{\mu\theta}}\right)v}$	$P = (T_t - T_c)\left(1 - \frac{1}{e^{\frac{\mu\theta}{\sin\beta/2}}}\right)v$

We see then that the power that can be transmitted is directly proportional to the peripheral speed v. Thus a high v is desirable – modern belts do indeed operate at high speeds of 15 to 30 m/s. But v also affects the centrifugal term ($T_c = mv^2$) which is in fact directly proportional to v^2, therefore the mass m should be kept to a minimum.

maximum power occurs when $T_c = \frac{1}{3}T_t$

$\max. P = \frac{2}{3}T_t\left(1 - \frac{1}{e^{\mu\theta}}\right)v$	$\max. P = \frac{2}{3}T_t\left(1 - \frac{1}{e^{\frac{\mu\theta}{\sin\beta/2}}}\right)v$

Data for flat belts

Belt designations (grades) and sizes (in mm)

Ply Symbol	Ply Name	Average thickness	Minimum economic width	Maximum width
MS	Medium single	4	40	200
HS	Heavy single	5	50	200
LD	Light double	7	75	300
MD	Medium double	8	90	300
HD	Heavy double	9	100	300
MT	Medium triple	12	125	600
HT	Heavy triple	14	150	-

Minimum pulley diameters for various belt speeds*

Belt thickness	0–10 m/s		11–20 m/s		21–30 m/s	
MS	100 mm		115 mm		125 mm	
HS	110 mm		120 mm		140 mm	
LD	120 mm		140 mm		200 mm	
MD	125	175	150	200	175	225
HD	200	250	225	275	250	300
MT	400	500	450	550	500	600
HT	500	600	550	650	600	700

*For belt widths 200 mm and over, use second figure in column.

Service factors

Drive	Machines	
Light starting load, uniform speed	electricity generators, light duty textile machinery, light duty evenly loaded conveyors, centrifugal pumps, automatic lathes	1,0
Medium starting load, uneven speed	fans <7,5kW machine tools, rotary compressors, light to medium woodworking machinery, belt conveyors with intermittent loads, live roller conveyors, roller mills for grain, group drives	1,1
Medium starting load, irregular speed, fluctuating loads	piston pumps and compressors with a speed fluctuation of > 1 : 80, centrifuges, fans >7,5kW dough mixers, pulpers, ball mills, tube mills, grinding machines, carding engines, spinning frames, propeller shafts, gang saws	1,3
Heavy starting, pulsating and intermittent loads	piston pumps and compressors with a speed fluctuation of < 1 : 80, vibrators, dredges, pan grinders, calenders, paper mill rolls, brick machines, forging presses, power presses and shears, rolling mills for non-ferrous metals	1,5
Very heavy starting, very uneven speed, unusually severe shock loads	piston pumps and compressors without flywheels, crushers, extrusion presses, cold rolling mills	1,7

Worked Example

A stone-crushing machine is to be driven by means of an open belt drive from a 15 kW electric motor running at 1440 r/min. The drive is to be designed on the basis of maximum belt transmissible power. Take belt density to be 1200 kg/m³. The maximum safe working stress is limited to 2,7 MPa. The speed of the machine pulley is to be ± 750 r/min and the centre distance of the pulleys is to be twice the large pulley diameter. Assume $\mu = 0{,}35$. Determine:

(a) belt designation (grade) and sizes;
(b) pulley diameters;
(c) centre distance;
(d) belt length;
(e) initial belt tension.

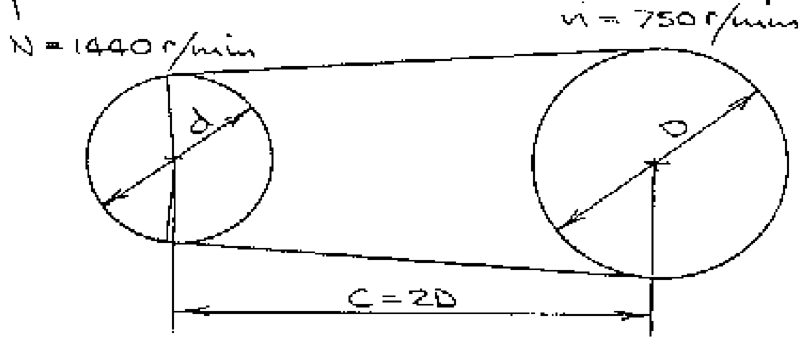
15 kW
SF = 1,7
(see p. 46)

$$\text{design power} = \text{normal power} \times \text{service factor}$$
$$= 15 \times 1{,}7$$
$$= 25{,}5 \text{ kW}$$

P = 25,5 kW
$\mu = 0{,}35$
$\theta = ?$
$v = ?$

$$\max.\ P = \tfrac{2}{3} T_t \left(1 - \frac{1}{e^{\mu\theta}}\right) v$$

to find θ



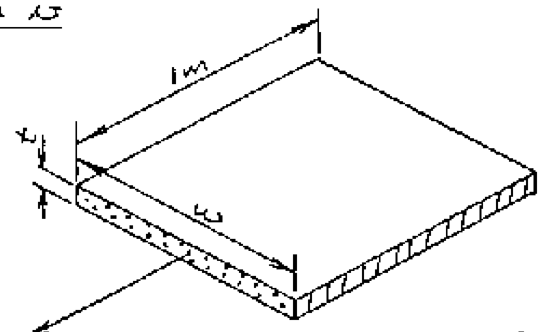
N = 1440 r/min
n = 750 r/min
C = 2D

$$\frac{D}{d} = \frac{N}{n} = \frac{1440}{750} = 1{,}92$$
$$\therefore D = 1{,}92d$$

and $C = 2D = 2 \times 1{,}92d = 3{,}84d$

then from
$$\theta \simeq \pi - \frac{D-d}{C}$$
$$= \pi - \frac{1{,}92d - d}{3{,}84d}$$
$$= \pi - 0{,}2396$$
$$= 2{,}9 \text{ rad}$$
$$= 166^\circ$$

to find v



$T_t = A \times \sigma$ and from $T_c = \frac{T_t}{3}$

$$m v^2 = \frac{A \times \sigma}{3}$$

$$A \times 1 \times \rho \times v^2 = \frac{A \times \sigma}{3}$$

$\sigma = 2{,}7$ MPa
$\rho = 1200$ kg/m³

$$v^2 = \frac{\sigma}{3 \times \rho}$$

$$v = \sqrt{\frac{2{,}7 \times 10^6}{3 \times 1200}}$$

$$= 27{,}39 \text{ m/s}$$

then substituting in –

$$P = \frac{2}{3} T_t \left(1 - \frac{1}{e^{\mu\theta}}\right) v$$

$P = 25{,}5$ kW
$T_t = A \times \sigma$
$= A \times 2{,}7 \times 10^6$
$\mu = 0{,}35$
$\theta = 2{,}9$ rad
$v = 27{,}39$ m/s

$$25{,}5 \times 10^3 = \frac{2}{3} \times A \times 2{,}7 \times 10^6 \times \left(1 - \frac{1}{e^{0{,}35 \times 2{,}9}}\right) \times 27{,}39$$

$$25{,}5 \times 10^3 = \frac{2}{3} \times A \times 2{,}7 \times 10^6 \times 0{,}6376 \times 27{,}39$$

$$A = \frac{25{,}5 \times 10^3 \times 3}{2 \times 2{,}7 \times 10^6 \times 0{,}6376 \times 27{,}39}$$

$$\therefore w \times t = 0{,}000811 \text{ m}^2 = 811 \text{ mm}^2$$

from data on page 46 select a medium double belt of 8 mm thickness – the width is then 100 mm

MD
$t = 8$ mm
$w = 100$ mm

---

allowing for belt thickness ($t = 8$ mm) the peripheral speed v in relation to the smaller pulley is given by –

$v = 27{,}39$ m/s
$t = 8$ mm
$N = 1440$ r/min

$$v = \frac{\pi \times (d + t) \times N}{60}$$

$$27{,}39 = \frac{\pi \times (d + 0{,}008) \times 1440}{60}$$

$$d + 0{,}008 = \frac{27{,}39 \times 60}{\pi \times 1440}$$

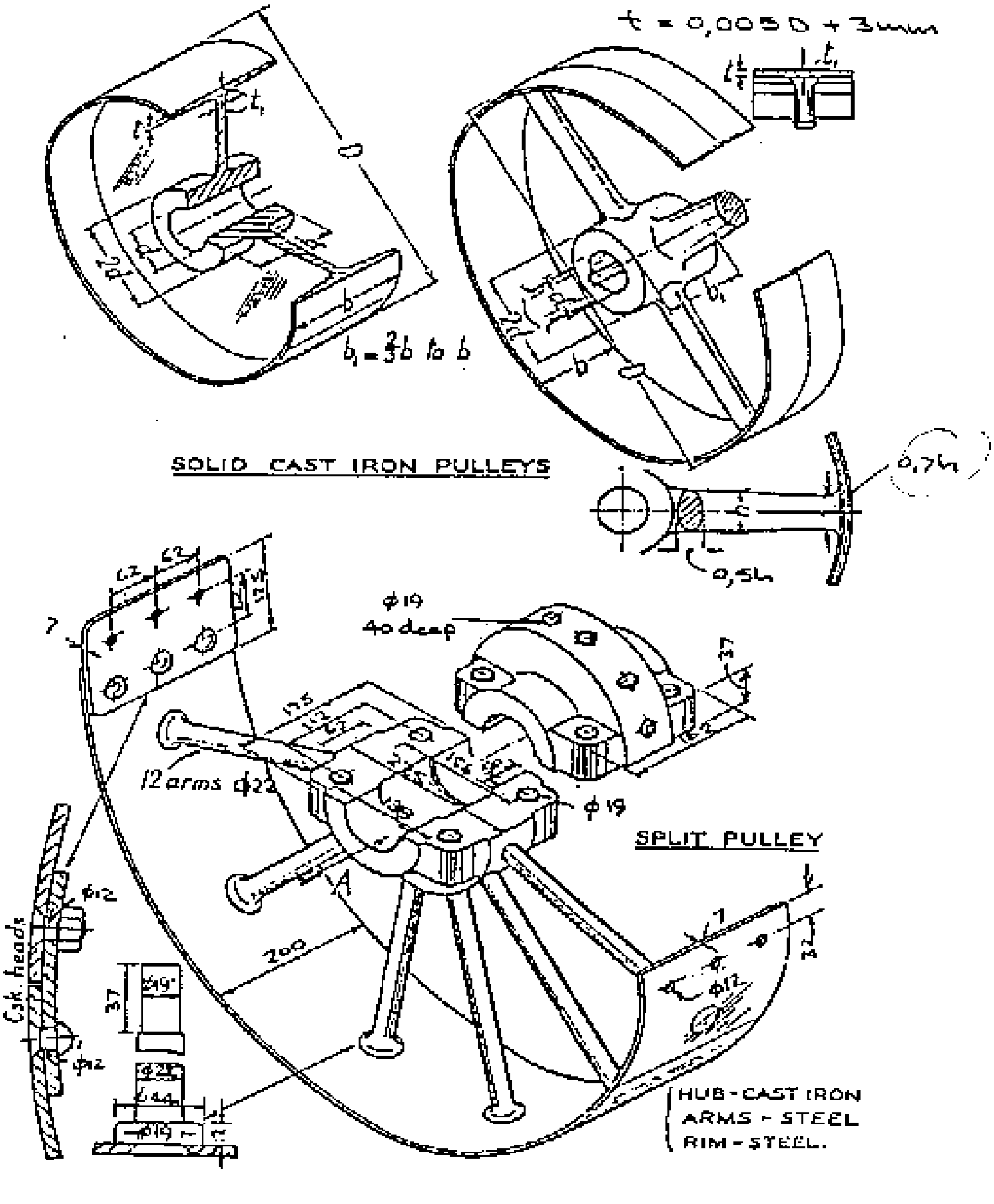
$$d = 0{,}363 - 0{,}008$$

$$= 0{,}355 \text{ m}$$

and from speed ratio $\frac{N}{n} = \frac{D + t}{d + t}$

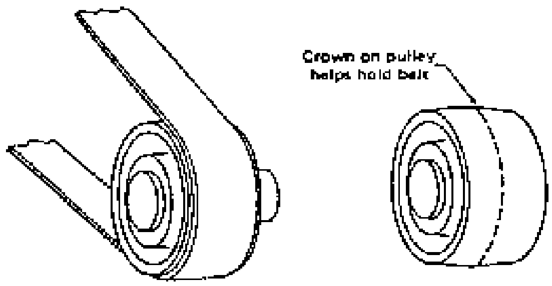
$n = 750$ r/min	$\frac{1440}{750} = \frac{D+t}{0,362}$ $D+t = 1,92 \times 0,363$ $D + 0,008 = 0,697$ $D = 0,697 - 0,008$ $= 0,689$ m	$d = 355$ mm $D = 690$ mm
$D = 690$ mm	$C = 2 \times D = 2 \times 690 = 1380$ mm	$C = 1380$ mm
$D = 690$ mm $d = 355$ mm $C = 1380$ mm	$L \simeq \frac{\pi}{2}(D+d) + 2C + \frac{(D-d)^2}{4C}$ $= \frac{\pi}{2}(690+355) + 2 \times 1380 + \frac{(690-355)^2}{4 \times 1380}$ $= 4422$ mm	$L = 4422$ mm
$w = 100$ mm $t = 8$ mm $\sigma = 2,7$ MPa $\mu = 0,35$ $\theta = 2,9$ rad	initial belt tension $T_o = \frac{T_t + S_t}{2}$ $T_t = w \times t \times \sigma = 0,1 \times 0,008 \times 2,7 \times 10^6 = 2160$ N to find S_t from - $\frac{T_t - T_c}{S_t - T_c} = e^{\mu\theta}$ we remember that for maximum P - $T_c = \frac{T_t}{3} = \frac{2160}{3} = 720$ N $\therefore \frac{2160 - 720}{S_t - 720} = e^{0,35 \times 2,9} = 2,76$ $2,76(S_t - 720) = 1440$ $S_t - 720 = 522$ $S_t = 1242$ N $\therefore T_o = \frac{2160 + 1242}{2} = 1701$ N	$T_o = 1,7$ kN

Pulleys for flat belt drives These are usually made either wholly of cast-iron or with a cast-iron boss and a steel rim. They may be in one piece or in halves to facilitate assembly on the shaft. The rim may be secured to the central boss either by a web or by radial arms.



Pulley width should exceed belt width by $\boxed{\pm 10\%}$

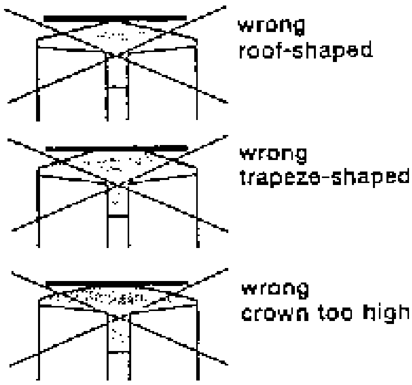
Crowning of pulley rims will make flat belts centre themselves on their pulleys.



The only correct shape for any crown is a regular symmetrical curve.

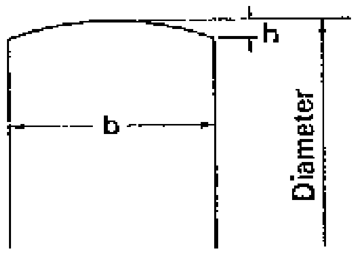
These three pulley crowns are wrong and have a detrimental effect on all belts – even though they are still being used here and there. They subject belts to additional lateral stresses which reduce their working life.

Belts cannot seat correctly on any one of these crownshapes which tends to lower their performance capacity. This could be neutralized to some extent and thereby its tension but this would result in higher shaft loads and their unpleasant consequences.



The best pulley profile: slightly round crowning

Diameter, mm	Crown h, mm	
40 to 112	0·3	
125 and 140	0·4	
160 and 180	0·5	
200 and 224	0·6	Based upon International
250 and 280	0·8	Standards Organisation
315 and 355	1	Recommendation



Width b, mm	⩽ 125	140 160	180 200	224 250	280 315	355	⩾ 400
Diameter, mm	Crown h, mm						
400	1	1·2	1·2	1·2	1·2	1·2	1·2
450	1	1·2	1·2	1·2	1·2	1·2	1·2
500	1	1·5	1·5	1·5	1·5	1·5	1·5
560	1	1·5	1·5	1·5	1·5	1·5	1·5
630	1	1·5	2	2	2	2	2
710	1	1·5	2	2	2	2	2
800	1	1·5	2	2·5	2·5	2·5	2·5
900	1	1·5	2	2·5	2·5	2·5	2·5
1 000	1	1·5	2	2·5	3	3	3
1 120	1·2	1·5	2	2·5	3	3	3·5
1 250	1·2	1·5	2	2·5	3	3·5	4
1 400	1·5	2	2·5	3	3·5	4	4
1 600	1·5	2	2·5	3	3·5	4	5
1 800	2	2·5	3	3·5	4	5	5
2 000	2	2·5	3	3·5	4	5	6

Choice of pulley diameter

Diameters should be as large as possible, since a flat belt works more efficiently at higher speeds. But we must ensure that the hoop stress (circumferential stress) σ_h in the pulley rim is within the safety limits for the rim material.

$$\sigma_h = \rho v^2 = \rho r^2 \omega^2$$

where σ_h = rotational hoop stress in rim (Pa)
ρ = density of rim material (kg/m^3)
v = peripheral speed of rim (m/s) $= \frac{\pi D N}{60}$
r = radius of rim (m)
ω = angular velocity of pulley (rad/s)

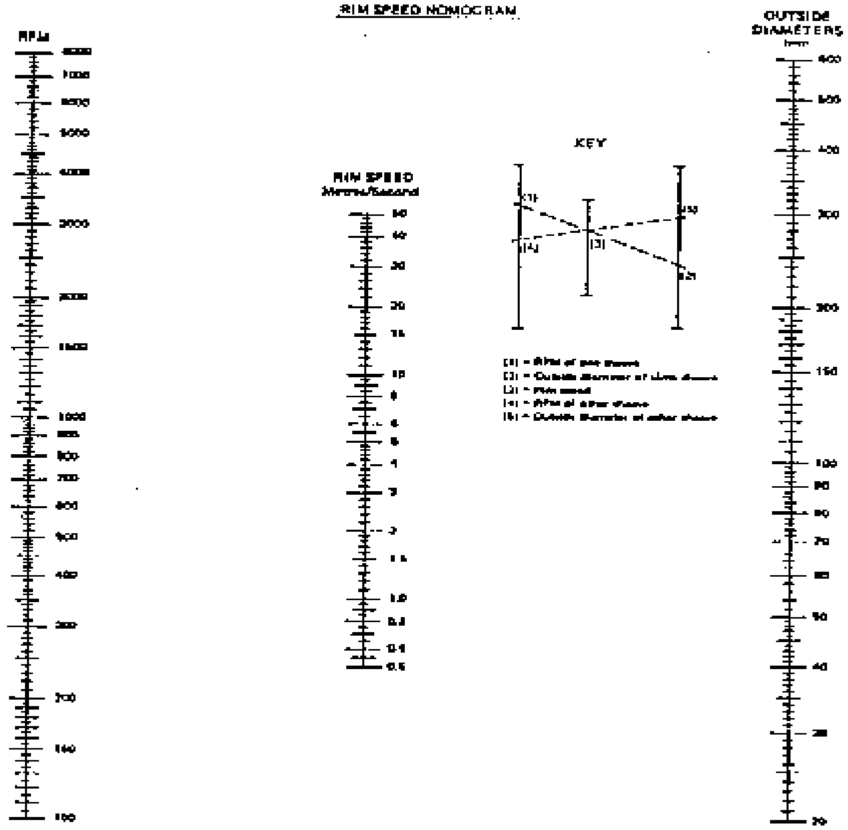
MATERIAL	DENSITY
Cast-iron	7250 kg/m^3
Steel	7800 kg/m^3

A 'rule-of-thumb' used by designers is that a pulley rim should never run faster than a peripheral speed of –

$$v = 30\ \text{m/s}$$

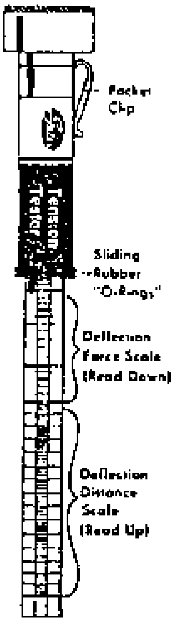
Rim Speed

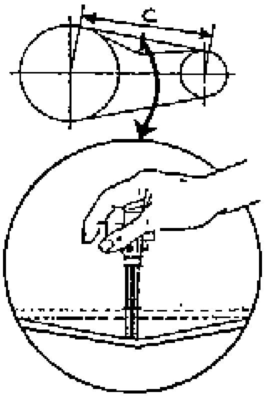
RIM SPEED NOMOGRAM



Installation tension

A drive should be tensioned according to the procedure outlined on page 52 of the BLUE BOOKLET and re-tensioned after between 30 minutes and 4 hours, at full load, to compensate for the small initial belt stretch and bedding into the pulley grooves.





How to use Tension Tester

1. Measure the span length = C mm.
2. Refer to the BLUE BOOKLET page 52 and from the table, for the applicable belt section and small pulley diameter, obtain e.
3. Calculate span deflection $\Delta = \frac{e \times C}{100}$
4. At centre of span apply force, with Tension Tester perpendicular to the span, large enough to deflect the belt by Δ - sight across the top(s) of other belt(s) or, to ensure greater accuracy, lay a straight-edge across the belt.
5. Read magnitude of deflection force on upper scale of Tension Tester - the sliding rubber 'O-ring' slides down scale as Tester compresses and stays down for accurate reading of force.
6. Compare deflection force measured on Tester with force recommended in table on page 52 of BLUE BOOKLET.

Design procedure for classical V-belt drives

Essentially the same as for wedge belts except that different selection chart, belt designations, power ratings, and belt length correction factors are used.

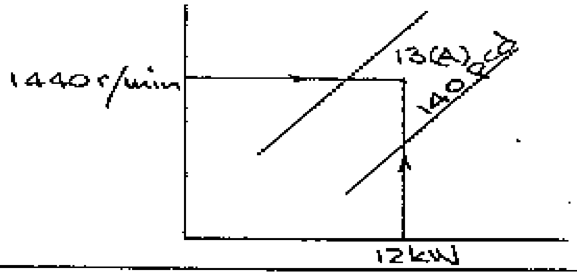
The following worked example will illustrate the procedure.

Worked Example

A V-belt drive is required to drive a fan at 800 r/min from a 10 kW 1440 r/min AC motor for 8 hours a day. The fan shaft is 50 mm and the motor shaft 42 mm. Centres are to be recommended.

Your design should include a recommended belt deflection for setting correct belt tension and the required deflection force.

10 kW page 37	design power = 10×1.2 = 12kW	12kW

page 38		13(A)
	speed ratio $= \frac{1440}{800} = 1{,}8$	1,8 : 1
page 38 page 24	tentative $d_p = 140$ mm then $D_p = 1{,}8 \times d_p = 1{,}8 \times 140 = 252$ mm	$d_p = 140$ mm $D_p = 250$ mm
	$C = 2 \times \sqrt{2 \times D_p \times d_p}$ $= 2 \times \sqrt{2 \times 250 \times 140}$ $= 529{,}15$ mm say 530 mm tentative belt length – $L = \frac{\pi}{2}(D_p + d_p) + 2C + \frac{(D_p - d_p)^2}{4C}$ $= \frac{\pi}{2}(250 + 140) + 2 \times 530 + \frac{(250-140)^2}{4 \times 530}$ $= 1678{,}3$ mm	
page 39	choose 13×8 (A) 1690	L = 1690 mm
page 40 page 41 page 46 page 47	rated power per belt = 3,79 kW additional kW per belt = 0,16 kW belt length correction factor = 1,0 $\frac{D_p - d_p}{C} = \frac{250-140}{530} = 0{,}21$ arc of contact correction factor = 0,97 corrected power rating per belt $= (3{,}79 + 0{,}16) \times 1{,}0 \times 0{,}97$ $= 3{,}8315$ kW/belt required number of belts $= \frac{12}{3{,}8315} = 3{,}13$	n = 4 belts
page 52	from table – F = 25 N and e = 1,5 mm/100 mm span $\therefore$ belt deflection $\Delta = \frac{e \times C}{100}$ $= \frac{1{,}5 \times 530}{100}$ $= 7{,}95$ mm	$\Delta = 7{,}95$ mm F = 25 N

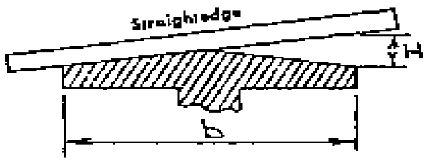
Drive details

Motor pulley - 140 mm x 4 SPA
Bush - 2517 x 42 look at Pulley

The large pulley has a greater contact angle that compensates for the loss of wedging action. Considerable savings can be made by using a flat pulley or fly wheel already on hand.

V-flat drives are feasible only when the arc of contact of the flat driven pulleys exceeds about 210°. This corresponds to $\frac{D-d}{C}$ greater than 0,5. However, the best results are obtained for $0,8 < \frac{D-d}{C} < 0,9$. If $\frac{D-d}{C} < 0,85$ a V-flat drive requires more tension than a V-V drive to keep it from slipping on the flat pulley, but the tension is still less than that for a straight flat drive. For V-flat drives the arc of contact of the grooved driver pulley may be as low as 130° as against the recommended value of 150° for V-belts.

A flat pulley should not be used in a V-flat drive if H exceeds 10mm per metre of face width (b)



$\frac{b}{H}$ must be greater than 100

If you are using a flat pulley on a drive other than the one for which it was originally intended, check its construction for strength.

Design procedure for V-flat drives

Besides the required data for the existing flat pulley (flywheel), only four other things have to be known before starting on a design:

1. kW of the driver
2. r/min of the driver and its shaft diameter
3. r/min of the driven flat pulley (flywheel)
4. approximate centre distance of shafts.

NOTE: To obtain pitch diameter of large flat pulley (flywheel) refer to pages 2 or 36 of BWE BOOKLET for the formula and data giving the amount to add to the outside diameter.

Worked Example

A 20kW squirrel cage motor (speed 720 r/min) is to drive a jaw crusher (speed 220 r/min) which has a 1258 OD x 305mm wide flat flywheel which is to be used as the driven pulley. Operation will be 20 hours per day and there is no limitation on centre distance.

Belt tensioners

These are used for the following reasons:

(i) the centre distance between drive shaft and driven shaft cannot be adjusted;

(ii) to take up stretch in the belt.

(iii) imultaneously for the effective tensions T and S.

1. then power $= (T - S)v$

and total tension on tight side $T_t = T + T_c$

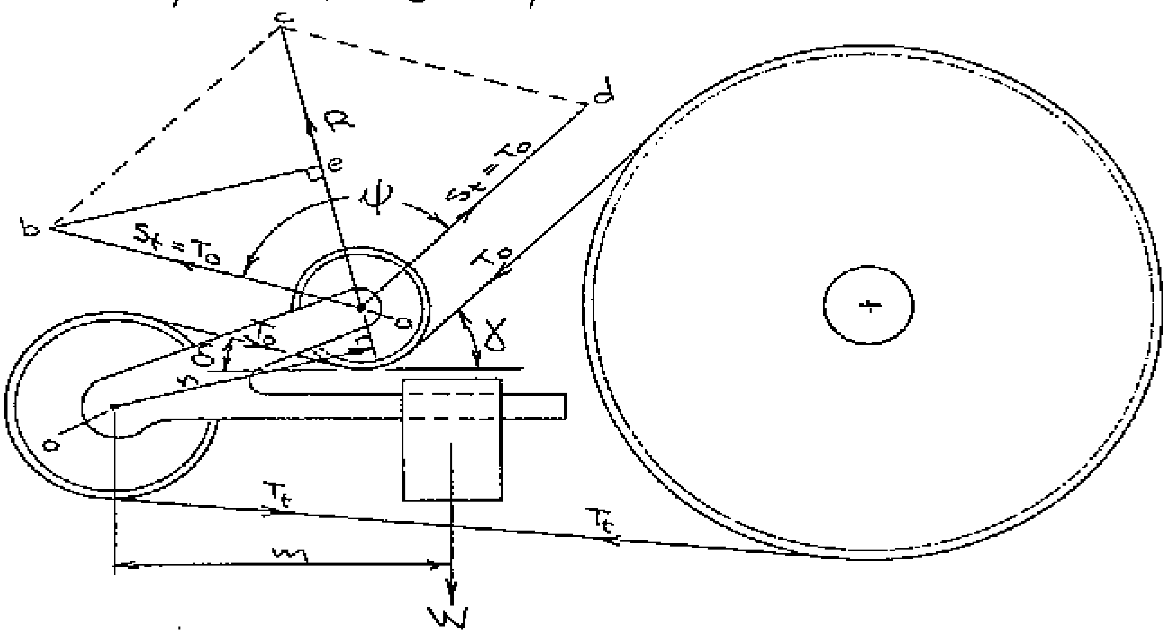
total tension on slack side $S_t = S + T_c$

$$T_c = mv^2$$

where m = mass per metre (kg/m)

v = belt speed (m/sec)

(ii) Jockey pulley (gravity idler pulley)



2.

In this version a deadweight causes a jockey ulley (gravity idler pulley) to press against the slack ide of the belt. This fixes the initial tension o (obtained by taking moments) which, once the rive is running (and transmitting power), remains e same to give S_t which in turn, through e ratio $\frac{T}{S} = e^{\mu\theta}$, allows us to get T or T_t and ence the power that can be transmitted for a iven μ and measured θ.

taking moments about arm pivot o gives –

$$R \times n = W \times m$$

further from the right-angled triangle abe –

$$\cos\frac{\psi}{2} = \frac{ae}{ab} = \frac{R/2}{S_t} \qquad \psi = 180^\circ - (\gamma + \delta)$$

giving

$$R = 2 S_t \cos\frac{\psi}{2}$$

that is $\quad 2 \times S_t \times \cos\frac{\psi}{2} \times n = W \times m$

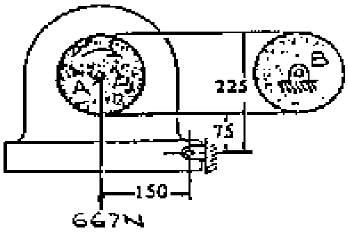
$$S_t = \frac{W \times m}{2 \times n \times \cos\frac{\psi}{2}}$$

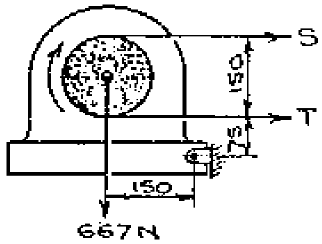
which together with $\frac{T_t - T_c}{S_t - T_c} = e^{\mu\theta}$ allows us to get T_t and then -

$$\text{power} = (T_t - S_t)\,v$$

Worked Example

A pivoted motor drive is shown diagrammatically. The two pulleys are of the same size, 150 mm diameter. The motor runs at 1910 r/min and weighs 667 N. The belt is 100 mm wide and 3,2 mm thick, the coefficient of friction is 0,4 and the belt density is 1,1 Mg/m³. Determine:
(a) the tensions in the belt;
(b) the power transmitted;
(c) the maximum stress in the belt.





taking moments about the pivot -

$$T \times 75 + S \times 225 = 667 \times 150$$

$\mu = 0,4$
$\theta = 180^\circ = \pi$ rad

also $\quad \frac{T}{S} = e^{\mu\theta} = e^{0,4\times\pi} = 3,51$

giving $\quad T = 3,51\,S$

hence $3,51S \times 75 + 225\,S = 667 \times 150$

$$263,25\,S + 225\,S = 667 \times 150$$

$$S = \frac{667 \times 150}{488,25} = 205\,N$$

then $\quad T = 3,51 \times 205\,N = 720\,N$

$w = 100$ mm
$t = 3,2$ mm
$\rho = 1,1$ Mg/m³
$d = 150$ mm
$N = 910$ r/min

further $\quad T_c = mv^2$

$$m = w \times t \times \rho = 0,1 \times 0,0032 \times 1 \times 1,1 \times 10^3 = 0,352\ \text{kg/m}$$

$$v = \frac{\pi d N}{60} = \frac{\pi \times 0,15 \times 910}{60} = 15\ \text{m/sec}$$

	$\therefore T_c = 0{,}352 \times (15)^2 = 79{,}2\ N$ hence $T_t = T + T_c = 720 + 79{,}2 = 799{,}2\ N$ $S_t = S + T_c = 205 + 79{,}2 = 284{,}2\ N$	$T_t = 800N$ $S_t = 284N$
$T = 720N$ $S = 205N$ $v = 15 m/sec$	power transmitted $P = (T - S)v$ $= (720 - 205) \times 15$ $= 7725\ W$	$P = 7{,}73 kW$
$T_t = 800N$ $w = 100mm$ $t = 3{,}2mm$	maximum stress in belt $= \dfrac{\text{max. tension}}{\text{X-sect. area}}$ $= \dfrac{T_t}{w \times t}$ $= \dfrac{800}{0{,}1 \times 0{,}0032}$ $= 2{,}5\ MPa$	$\sigma_t = 2{,}5 MPa$

The Selection of Wedge - belt Drives

Ref. Pg 24 Section 2.16

Before starting on a design, you have to have the following information:

a) KW of driver, i.e. Normal Power
b) rpm of driver and its shaft ϕ
c) type of driven machine
d) rpm of driven machine and its shaft ϕ
e) approximate centre distance of shafts.

* N.B.

Step 1 Speed ratio = $\frac{\text{Faster rpm}}{\text{Slow rpm}}$ i.e. $\frac{\text{Driver speed}}{\text{Driven speed}}$

(2) Service Factor --- Ref. [Table 2.7 on Pg 26]

(3) Design Power = Normal Power x Service Factor

(4) Belt selection .... Ref. [Table 2.9 on Pg 27]

(5) Motor pulley Limitations .... Ref. [Table 2.8 on Pg 26]
(If electrically driven)

(6) Pulley pitch diameter ... Ref. [Chart 1 → 4 on Pg 25]

(7) Belt Length, Centre distances & Correction Factor
Ref. --- [Pg 28 → 39]
If centre distance (c) is not fixed obtain c from:-
$C = 2 \times \sqrt{2D \times d}$
and then a tentative belt length L is :- $L = \frac{\pi}{2}(D+d) + 2C + \frac{(D}{4}$
after which refer to Ref Pages under the selected belt section choose a suitable belt length.

(8) Basic Power per belt ... Ref [Chart SPB belts on Pg 34 & beyond]

(9) Speed ratio power increment ..... Ref [below Pg 34 & beyond]

(10) Corrected power per belt ... Ref [Add (9) + (8)] * (7)

(11) Number of belts required --- Ref. (3) ÷ (10)
Factions must be taken to the next Whole Number
(e.g. 3.15 means take 4 belts)

(12) Bore size --- Ref. [Table 2.9 on Pg 27]

DESIGN A WEDGE BELT DRIVE FROM A 50KW 6 CYL ENGINE WHICH RUNS AT 1050 rpm TO A RECIPROCATING COMPRESSOR RUNNING AT 660rpm. THE CENTRE DIST. IS TO BE 1600 mm AND THE DUTY 24 HRS / DAY. ENGINE SHAFT 70 mm Ø AND COMPRESSOR SHAFT = 80 mm Ø

1 SPEED RATIO = $\frac{DRIVER}{DRIVEN}$ = $\frac{1050}{660}$ = 1.59 : 1

2 Pg 26 TABLE 2.7 SERVICE FACTOR = 1.4
SINCE > 16 HRS AND RECIPROCATING COMPRESS. DRIVEN BY I.C. ENGINE WITH MORE THAN 6 CYL.

3 DESIGN POWER.
ENGINE POWER OUTPUT * SERVICE FACTOR.
= 50 x 1.4
= 70 KW.

4 BELT SECTION.

Pg 27 TABLE 2.9

2 - POSSIBILITIES - CONSIDERING INTERSECTION OF DESIGN POWER AND RPM OF FASTER SHAFT

OPTION 1 SPB PCD 260 5 grooves (P = 50kW instead of 70kW)
OPTION 2 SPC PCD 300 4 grooves.

EITHER OPTION ACCEPTABLE.

SELECTION OPTION 1

5 MOTOR PULLEY LIMITATIONS

TABLE 2.8 DOES NOT APPLY SINCE DRIVER IS NOT AN ELECTRIC MOTOR.

6 Pulley Pitch Ø

REF THE TABLES RELATING TO SPB BELTS W.R.T CENTRE DIST.

DRIVING PULLEY = 315 mm
DRIVEN PULLEY = 500 mm

● BELT LENGTH, CENTRE DIST & CORRECTION FACTOR.

CLOSEST VALUE TO 1600 IN TABLE FOR C. DIST

1637 mm, CORRECTION FACTOR 1.04 AND BELT SIZE AT THE HEAD = 16N SPB 4560

8● Basic Power Per Belt.

315 Pitch Ø = 17.82 kW

by interpolation @ 1050 rpm

9 Speed Ratio Power increments

ALSO BY INTERPOLATION
FOR SPEED RATIO of 1.59 @ 1050 rpm
= 0.77 kW

TABLE 2.B DOES NOT APPLY SINCE DRIVER IS NOT AN ELECTRIC MOTOR.

6 PULLEY PITCH Ø

REF THE TABLES RELATING TO SPB BELTS w r t CENTRE DIST.

DRIVING PULLEY I 315 mm
DRIVEN PULLEY I 500 mm

7 BELT LENGTH, CENTRE DIST & CORRECTION FACTOR.

CLOSEST VALUE TO 1600 IN TABLE FOR C. DIST IS

1627 mm, CORRECTION FACTOR 1.04 AND
BELT SIZE AT THE HEAD = 16N SPB 4560

8 BASIC POWER PER BELT.

315 PITCH Ø = 17.82 kW

by INTERPOLATION @ 1050 rpm

9 SPEED RATIO POWER INCREMENTS

ALSO BY INTERPOLATION
FOR SPEED RATIO of 1.59 @ 1050 rpm
= 0.77 kW

⑨ Thus Corrected Power / Belt

$= (17.82 + 0.77) \times 1.04$

$= 19.33$ kW / BELT

∴ No of Belts Required

$= \frac{\text{Design Power}}{\text{Corrected P / Belt}}$

$= \frac{70}{19.33}$

$= 3.62$ ∴ Choose 4 belts.

10 Bore Size

From Pulley charts

315 mm x 4 SPB ⇒ Bore size 90 mm

which is greater.

Drive Specifications

Engine Pulley

315 x 4 SPB 3535 / 70 mm

Compressor Pulley

500 x 4 SPB 3635 / 80 mm
4 x 16 N SPB · 4560 WEDGE BELTS.