DRIVE DESIGN & CALCULATION GUIDE

DESIGN

Correct design of the drive is essential to gain the full advantage of a CMW Power Transmission V belt. The key factors which influence the capacity and service of our belts are detailed in this manual The rating specified in this manual are calculated to achieve trouble-free operation and optimum life. Therefore, we specify the mathematical formulae for calculating operating drives with this in mind.

The tension in the belts and the diameters of the pulleys establish the degree of flexing, and the belt speed and length determine the frequency of flexing. Increasing the severity of any of these factors without compensating the severity of some other factor, can be done only with a reduction on belt life and with a penalty of the i low operating costs of the drive.

There are fundamental relations involving tensions, speeds, friction, etc., which apply to all types of belt driving. These basic concepts of the fundamental relationships involving tensions, speeds, and friction are noted and their applications to belt drives are detailed.

DEFINITIONS OF TERMS

Tension in a belt is a force acting along the length of the belt and tending to elongate it. Belt tension is measured in Newtons (N).

Torque is the effectiveness of a force to produce rotation about an axis, and thus involves the size of the force and its moment arm. Torque is the product of a force (or tension) and the length of the arm through which it acts. The units for torque are Newton metres (Nm) and kilo-Newton metres (kNm).

Energy and work are closely related and are expressed in the same units. The units are the Joule (J) and the kilo-Joule (kJ).

Work is the product of a force and the distance through which it acts.

Energy is the capacity for performing work. The units are the Joule (J) and the kilo-Joule (kJ).

The energy of a moving body in Joules is given by:

$\frac{1}{2} mv^2$

where "m" is the mass, in kilograms (kgs.) and "v" is its velocity in metres/second (m/s).

Power is the rate of doing work or transmitting energy. The unit normally used is the kilowatt (kW) which is the work done when a force of 1000 Newtons is displaced through a distance of one metre in one second.

Power exerted for a period of time pr

COEFFICIENT OF FRICTION

A coefficient of friction is a value that shows the relationship between two objects and the normal reaction between the objects that are involved. It is a value that is sometimes used in physics to find an object's normal force or frictional force when other methods are unavailable.

The coefficient of static friction is the friction force between two objects when neither of the objects is moving. The coefficient of dynamic friction is the force between two objects when one object is moving, or if two objects are moving against one another.
The coefficient of friction is dimensionless and it does not have any unit. It is a scalar, meaning the direction of the force does not affect the physical quantity.
The coefficient of friction depends on the objects that are causing friction. The value is usually between 0 and 1 but can be greater than 1. A value of 0 means there is no friction at all between the objects; such is possible with Superfluidity. All objects, otherwise, will have some friction when they touch each other. A value of 1 means the frictional force is equal to the normal force. It is a misconception that the coefficient of friction is limited to values between zero and one. A coefficient of friction that is more than one just means that the frictional force is stronger than the normal force. An object such as silicone rubber, for example, can have a coefficient of friction much greater than one.
The friction force is the force exerted by a surface when an object moves across it - or makes an effort to move across it.

If, as in Fig. 5, a body of mass “m” kg rests on a horizontal plane surface and a force “p” parallel to the surface is just enough to cause the body to be at the point of slipping, the ratio (f) of tangential to normal force is known as the coefficient of friction.

$f = {}^{p}/{mg}$

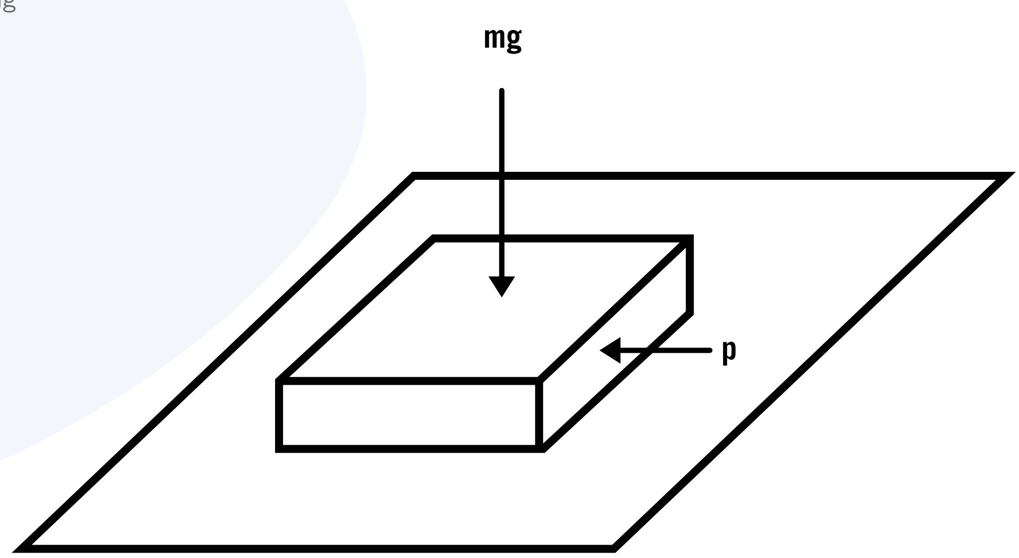

FIG. 5

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TENSION

Consider a rope-or--belt as in Fig. 6 hanging over a pulley which resists rotation.

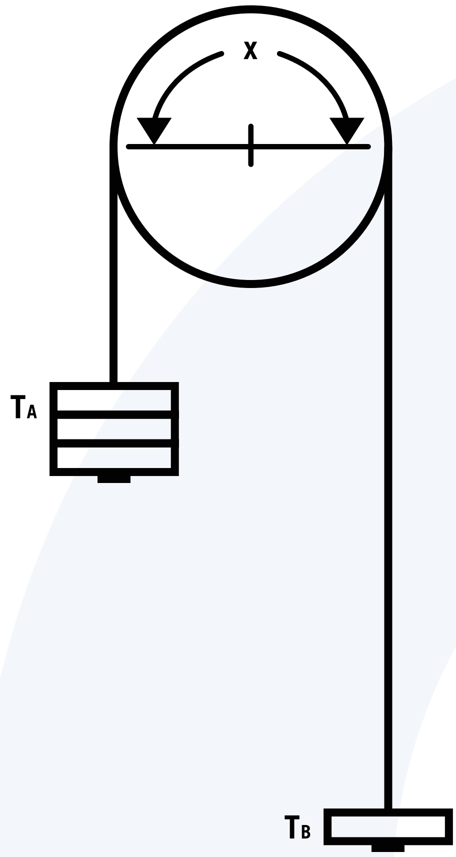


FIG. 6

Tension TA and TB are caused by large and small weights respectively. So, if the coefficient of friction between belt and pulley is large enough, a considerable difference in tension is possible in such a system.

When the arc of contact is reduced (as in Fig. 7 with a freely turning idler), TB must be larger to keep the belt from slipping.

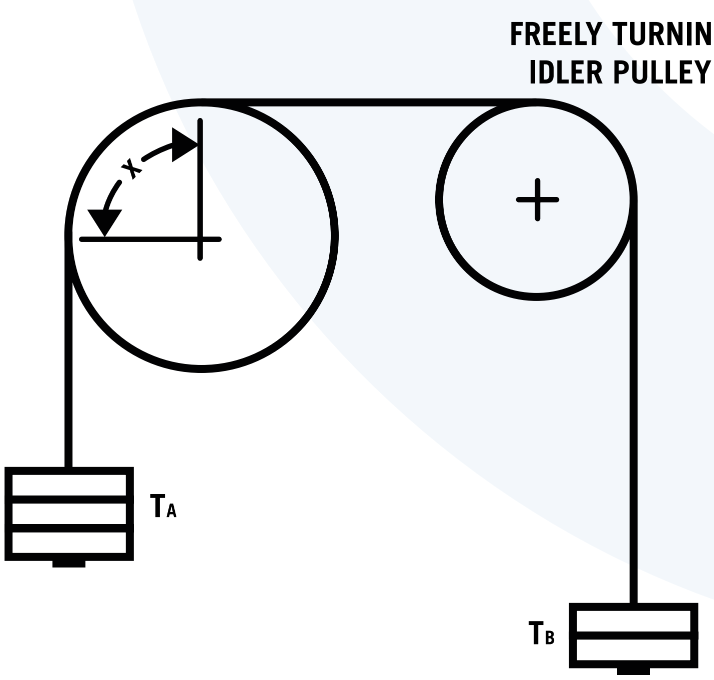


FIG. 7

The essential factors are the tensions, the coefficient of friction and the angle or arc of contact.

If in Fig. 6 or 7 the unbalanced tension (T_A - T_B) is large enough to overcome the resistance, the pulley will turn but the action is limited by the length of the belt.

In Fig. 8, where a joined or endless belt is applied to two pulleys and a turning moment or torque applied at shaft 0_1 causes a torque at shaft 0_2.

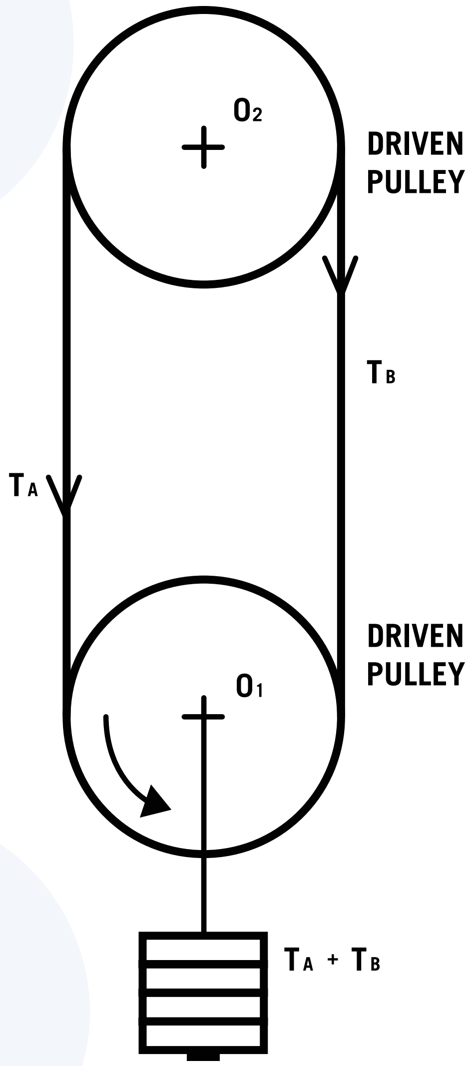


FIG. 8

Thus the action described in Fig. 6 is applicable continuously in a system like Fig. 8, illustrating the fundamental tension relations in belt driving. To find the relation of TA, coefficient of friction (*f*) and the arc of contact (a in radians) refer to Fig. 9 representing a very small element of the belt of Fig.6, 7 or 8.

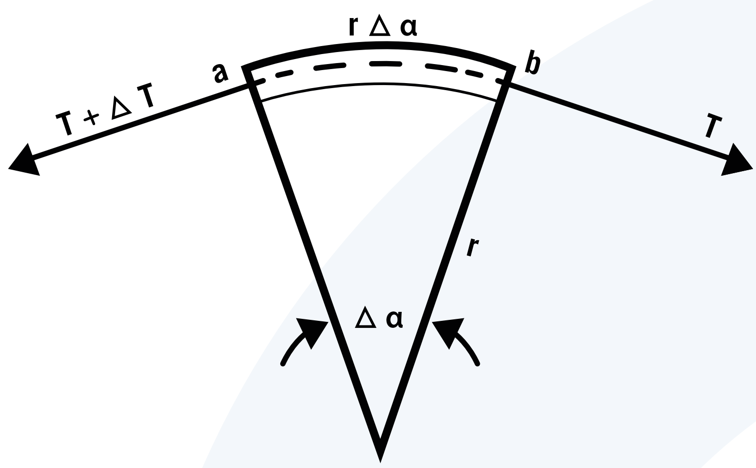


FIG. 9

The tension in the belt at “b” is T, and at “a” is T+ AT due to friction. The element “ab” subtends the very small angle Aa. The forces are more clearly represented in Fig. 10 which shows that the force Fn between this portion of the belt and the pulley is given by:

$F_n = 2T \ sin \ \frac{\Delta\alpha}{2}$ *(here ΔT is negligible)*

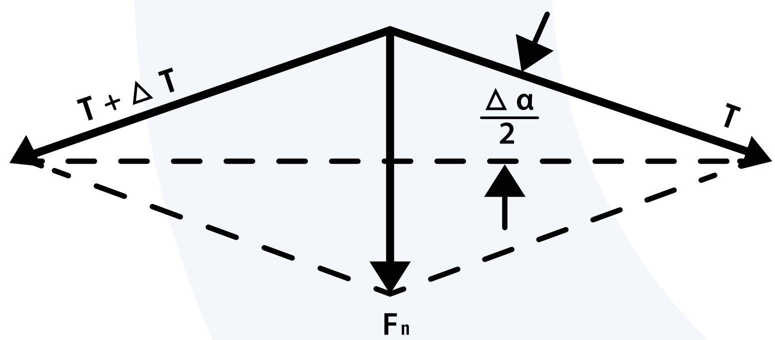


FIG. 10

Equations (4) and (5) below were developed for flat belts.

$\Delta T = f\,F_n$ *(the belt being at the point of slipping)*

Taking limits as ø< approaches zero, —

$\frac{dT}{d\alpha} = fT$

Integrating,

$$\int_{o}^{\alpha} d\alpha = \int_{T_B}^{T_A} \frac{dT}{T}$$

$$f\alpha = \log_e \frac{T_A}{T_B}$$

$$\frac{T_A}{T_B} = e^{f\alpha}$$

$$\frac{T_A}{T_B} = e^{f0,0175fa}$$

—
where "a" is arc of contact in degrees
—

With V and wedge belts the radial force causes a larger total against the faces of the pulley groove due to the wedging effect for which we shall use the symbol "u". The wedging effect is seen by reference to Fig. 11 to be as follows:

$$F_2 + F_3 = \frac{F_2}{\sin \frac{\beta}{2}}$$

$$u = \frac{F_2 + F_3}{F_1} = \frac{1}{\sin \frac{\beta}{2}}$$

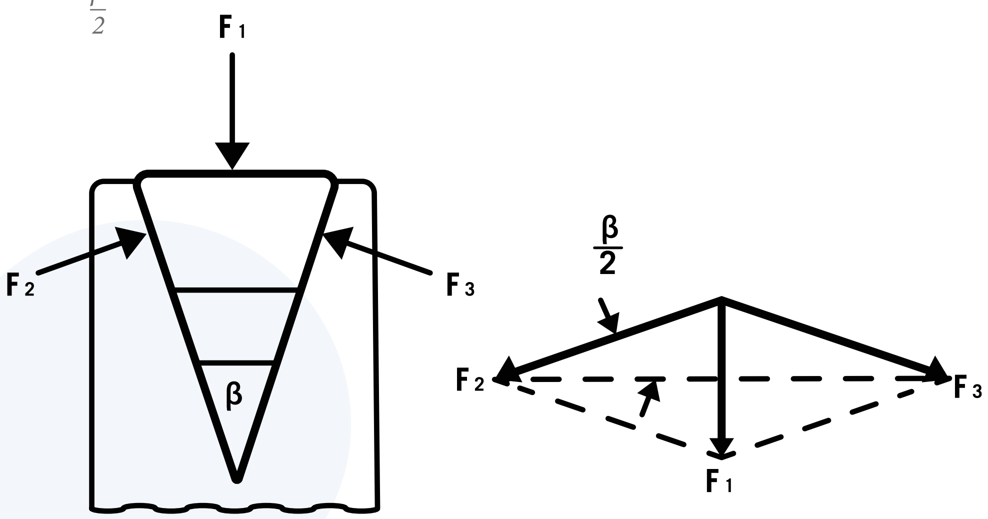


FIG. 11

Then, with V and wedge belts, the force required to cause the infinitesimal section to slip is the radial force multiplied by 'f' and again by "u". Thus, while the coefficient of friction is independent of the shape of the belt, the effectiveness of 'f' is multiplied by "u" in wedge belts and the tension formula then becomes:

$$\frac{T_A}{T_B} = e^{uf\alpha}$$

With V-grooved pulleys of 38° included angle

$$u = \frac{1}{\sin 19^o} = 3$$

CENTRIFUGAL TENSION

At speeds at which V and wedge belts are usually operated, centrifugal tension may be a very important factor. The centrifugal force F, (Newtons) acting on a body of mass "m" kg moving "v" m/s in a curved path of radius "r" metres is given by:

$$F_2 = \frac{mv^2}{r}$$

If, instead of representing the mass of the body, "m" represents kg per metre of length, the centrifugal force for the element in Fig. 9 is given by:

$$F_2 = \frac{mv^2\,\Delta\alpha}{\mathrm{r}} = mv^{2r}\,r\Delta\alpha$$

The centrifugal force acting on the elements of a belt is balanced by centrifugal tension (Ta) in the belt. From a relationship similar to Fig. 10.

$$F_c = 2T_c\sin\frac{\Delta\alpha}{2}$$

From (8) and (9),

$$T_c = \frac{mv^2\,\Delta\alpha/2}{sin\Delta\alpha/2}$$

Taking limits as A a approaches zero.

$$T_c = mv^2$$

Now if $T1 = T_A + T_1$ and $T_2 = T_B + T_C$
Then,

$$R = \frac{T_1 - T_c}{T_2 - T_c} = e^{f\alpha}$$

Where R = ratio of tensions.
Note that equations (4), (5), (6) and (11) are valid only under the condition for which they were derived, i.e. when the belt is at the point of slipping.
Otherwise the existing tension ratio will be less than indicated.

Slack Side Tension
For any given condition, there is a minimum slack side tension below which the drive will not operate. The effective tension is established by the power requirement (expressed in kilowatts) and the belt speed. Since the tight side tension (which is the maximum in the belt) is the sum of the effective tension and the slack side tension, it is advisable to keep the slack side tension as low as possible. This minimum can be best maintained automatically, in which case the actual slack side tension may be very little more than the minimum required.

Drives having a screw adjustment must be set up with some reserve tension to allow for stretch. As the belt gradually stretches, the tension will diminish to a point where the belt must be retightened.

With fixed centre (manually adjusted) drives and with 1800 arc of contact, V and wedge belt drives should be applied with ratio of tensions R = 5.00 and the tension should be restored when tension slacks off to the point where R = 8.00. If automatic tension adjustment is used with V and wedge belts, R may be permanently held at 8.00 (arc of contact 180°).

Creep

In belt practice the change in length that occurs with time as the belt continues in service is not called "creep", as might be supposed from the use of this term in other engineering activities. With belts, this change of dimensions with time is called "stretch" or "growth in length". The term "creep" as applied to belt driving refers to a loss of driven speed as the result of alternate lengthening and shortening of each portion of the belt as it experiences the cycle of tight and slack side tensions.

Whenever a belt passes around a pulley and there is a differencebetween the entering and leaving tensions, there is belt creep. Consider a portion or element of belt approaching a driving pulley. If the tension is high with reference to the torque the belt will travel at the same speed as the pulley face through some part of the arc of contact.

Through the remainder of the arc of contact this portion of belt will be under progressively less tension down to the slack side tension at the exit point. During the slackening process the belt element shortens (recovers from elongation) and consequently moves slower than the pulley face. This relative motion is creep.

If the load is increased, the arc in which creep occurs (the "arc of creep") increases. If the load is sufficiently increased the arc of creep may become as large as the arc of contact, in which case the belt will be at the point of slipping. The remedy, of course, is to provide more slack side tension.

Whether the belt is being driven by a pulley or is itself driving a pulley the arc of creep always starts from the exit point and progresses towards the entry point as the load increases.

Consider the action in the vicinity of the driving pulley. If "E" is the dynamic modulus of elasticity of the belt and "v," and "v2" are the entry and exit velocities respectively.

$$\%\ Creep = 100\ \frac{V_1 - V_2}{V_2} =$$

$$100 \left[\frac{\left(1 + \frac{T_1}{E}\right) - \left(1 + \frac{T_2}{E}\right)}{\left(1 + \frac{T_1}{E}\right)} \right] = 100 \left(\frac{T_1 - T_2}{E + T_1} \right)$$

$$\%\ Creep = \frac{100 T_e}{E + T_1}$$

Since T1 is small compared with E we may write:

$$\%\ Creep = \frac{100 T_e}{E}\ (approximately)$$

Using the belt velocity as it approaches the drive pulley as a base, the belt slows down where it leaves the drive pulley by the amount of the creep percentage. The recovery of this velocity loss occurs where the belt leaves the driven pulley.

While the creep percentage is usually small enough to be neglected, without appreciable error, there are cases where the creep value may be significant.

Torque and power

Where P = Power in kW

v = belt speed (m/s)

Te = effective tension in Newtons

rev/min = revolutions per minuteP= TeV/1,000

$$P = \frac{T_e v}{1{,}000}$$

$$P = Torque\ in\ Nm \times \frac{rev/min}{9550}$$

$$Torque\ in\ Nm = \frac{9500 \times P}{rev/min}$$

CMW Power Transmission Belt Flexing Formula

The results of extensive testing and subsequent confirmation under field conditions are expressed in our Belt Flexing Formula:

$$Service\ line\ in\ Flexing = \frac{k \times d^{5.35} \times L}{v^{0.5} \times T_1{}^{4.12}}$$

Where k = a constant of proportionality

d = pulley diameter

L = length of belt

v = belt speed

T1 = tight side tension

The exponents in this formula show the extremely large effect ofchanges in pulley diameter and belt tension. Similarly, they show how a necessarily severe value of one of the factors can be compensated by appropriate changes in the others.

In considering this formula it should be remembered that it is based on resistance to internal damage only. Obviously, for example, the tension in a belt may be limited by bearing loads, stretch considerations, or other external conditions not included in the formula.

In actual service it is difficult to segregate all factors as was done in our laboratory tests. For this reason the main utility of the formula is not in evaluating "K" to predict absolute values of service, but in the comparisons it makes possible between applications having some degree of similarity. Therefore, we have found the following formula to be very useful for comparing drives using V and wedge belts of the same cross sectional size.

$$\frac{Flexing\ service_x}{Flexing\ service_a} =$$

$$\left(\frac{d_x}{d_a}\right)^{5.35} \times \left(\frac{T_{1a}}{T_{1x}}\right)^{4.12} \times \left(\frac{v_x}{v_a}\right)^{0.5} \times \left(\frac{L_x}{L_{1x}}\right)$$

where “a” is a set of known conditions and 'Y' represents the desired conditions. Note that if any factor is the same in both cases, that term in (18) becomes unity and does not affect the ratio of service.

Very often only one factor is changed and formula (18) spotlights the effect of such a change.

Calculation of Arc of Contact

For ordinary two-pulley drives, the arc of contact can be determined from the following approximate formula:

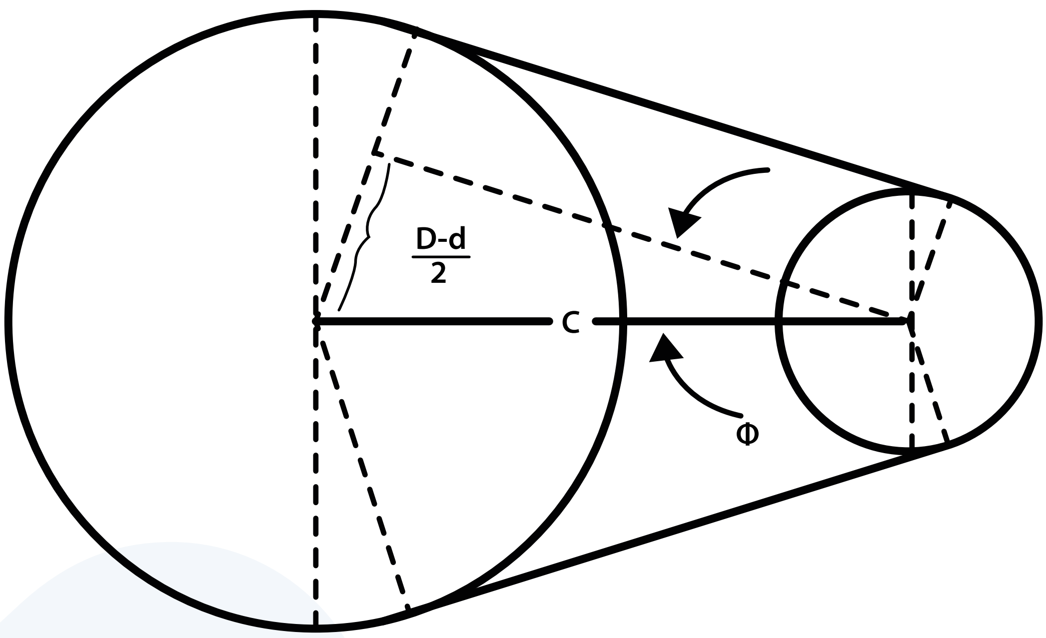
$$Arc\ of\ Contact = 180° - \frac{60\,(D-d)}{C}$$

Where D = large pulley diameter
d = small pulley diameter
C = centre to centre distance
(n.b. units of measurement must be the same).

The approximate formula is a simplification of the theoretical, which states:

$$Arc = \pi - 2\sin^{-1}\left(\frac{D-d}{2C}\right)\ (in\ radians)$$

The formula is apparent from the following sketch.



For small angles, it is assumed that an angle in radians is equal to its sine, and on this assumption,

$$Arc = 180 - 57.3\left(\frac{D-d}{2C}\right)\ (in\ degrees)$$

where 57.3 is the factor for converting radians to degrees. 57.3 is replaced by 60 (in formula 19) in order to compensate somewhat for the slight error introduced by the first assumption.

The approximate formula (19) is within 1° of the theoretical between 180° and 110° it gives arcs 3° too high at 100° and 50° too high at 90°. We recommend that the theoretical formula be used for arcs less than 100°.

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Arc of Contact v Area of Contact

From the basic concept of friction between sliding surfaces, the area of surface does not influence the amount of friction. On the other hand, the friction is solely dependent on the character of the faces and the total pressure normal to the faces. It is possible to increase the area of contact and at the same time actually decrease

the power capacity. Investigation will disclose the fact that when the power capacity is increased, this is accomplished by increasing one of the vital factors, such as the arc of contact, width of belt, or diameters of the pulleys and speed of belt.

Very often area of contact is thus increased incidentally, but it also frequently happens that of two drives the one with the smaller area has the larger capacity.

Length Formula

The precise formula for belt length around two pulleys as in Fig. 12 is as follows:

$$L = 2C \cos \varnothing + \frac{\pi (D+d)}{2} + \frac{\pi \varnothing (D+d)}{180}$$

Where L = belt length
C = centre to centre distance
D = large pulley diameter
d = small pulley diameter

$$\varnothing = \sin^{-1} \frac{(D - d)}{2C} \ (in\ degrees)$$

L, C, D, and d must all be expressed in the same unit of length. The following approximate formula is easier to use and is accurate to within 0.15% with a 7 to 1 pulley ratio and centre distance of 6d, and even more accurate for the average drive:

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

This formula can be solved for centre distance instead of belt length as follows:

$$C = \frac{L - 1.57 (D+d)}{4} + \sqrt{\left\{\frac{L - 1.57 (D+d)}{4}\right\}^2 - \frac{(D - d)^2}{8}}$$

Where D = chosen diameter of large pulley
d = chosen diameter of small pulley
L = length of belt (appertaining to defined diamete
C = centre distance

(n.b. all units of measurement must be the same).

INERTIA

In the foregoing pages, inertia loads have not been considered. Generally speaking, inertia will not be a problem.
There has been little attempt to use quantitative values for the effect of inertia, mostly because of the limited knowledge of polar moments or acceleration values of individual components.
Where inertia is shown or suspected to be a factor on a particular drive and the moments and acceleration are known, an effective tension can be derived from inertia considerations using the following formula:

$$T_e = ø \frac{\pi\, dl_p}{0.015\,(D)^2} \times \frac{\Delta N}{\Delta t} \times \frac{67\pi\, dl_p}{(D)^2} \times \frac{\Delta N}{\Delta t}$$

Where d = diameter of driven pulley (mm)
Ip = moment of inertia of driven unit (kgm2)
D = pulley diameter of driven unit (mm)

$\frac{\Delta N}{\Delta t}$ = change in driven speed (rev/min)in time. T_e is in Newtons and must be added to the calculated T_e at this speed before proceeding to compute the maximum bet tension

The moment of inertia is most easily found by experiment.

CMW POWER TRANSMISSION V-BELT DRIVE CONSIDERATIONS

Note: More information on installation, maintenance and troubleshooting can be found on pages 136.

1. Warped pulleys or rough pulley groove surfaces shorten belt life. For satisfactory service, surface roughness should not exceed 3.0 micrometres.

2. Belt life is affected inversely as the third power of the ambient temperature.

3. Correct belt tension is the key to good belt life. Too little tension will result in slip damage. Too much tension will markedly reduce flex life. The flex life will decrease as the fourth power of the tension.

4. While there is a factor of safety in a properly designed belt drive which will permit, in an emergency, its operation without the full quota of belts, advantage should be taken of this only long enough to secure belts needed for replacement. When, after any considerable period of service, it becomes necessary to replace damaged or worn belts on the drive, it is essential that the entire set be replaced. All belts in service for any length of time are subject to stretch which makes the partial replacement of belts impractical.

5. A good many drives are now in existence, some of which are not designed according to the ratings given in this Manual. Where such drives are found, CMW Power Transmission belts can be used with the knowledge that they will give service equal or superior to any other comparable belts which may have been used on the drive.
CMW Power Transmission belts are of standard dimensions and can be used interchangeably, complete set by complete set, with any other standard belts and as replacements on any drive having standard pulleys and centre distances. If, however, satisfactory service has not been obtained from the drive, consideration should be given to changing it to conform with the ratings shown in this Manual.

6. As a rule, the use of idlers on the back of V and wedge belts is not recommended, since the reverse bend reduces the life of the belts. Idlers have been used with some degree of success, however, and where no other means of take-up is possible, properly designed idlers can be used. Grooved idler pulleys operating on the

inside of the belt, are preferred. Inside idlers reduce the arc contact and this must be allowed for.
7. Where it is possible to provide automatic take-up either by mounting the motor on rails or by pivoting it on the base, the tensions in the belt can be held much more uniform and greater life will result.

8. In assembling a drive, care must be taken to insure proper alignment of pulleys.

The belts should be operated at tensions just sufficient to insure against slippage. If the tensions are too low the belts will slip, with resulting power loss and increased wear on the belt and pulleys; if too high, the life of the belts will be decreased and the bearing pressures and other machine stresses will be higher than need be.

Time spent in insuring correct tensions will be well repaid by decreased operating and maintenance costs.

IDLERS

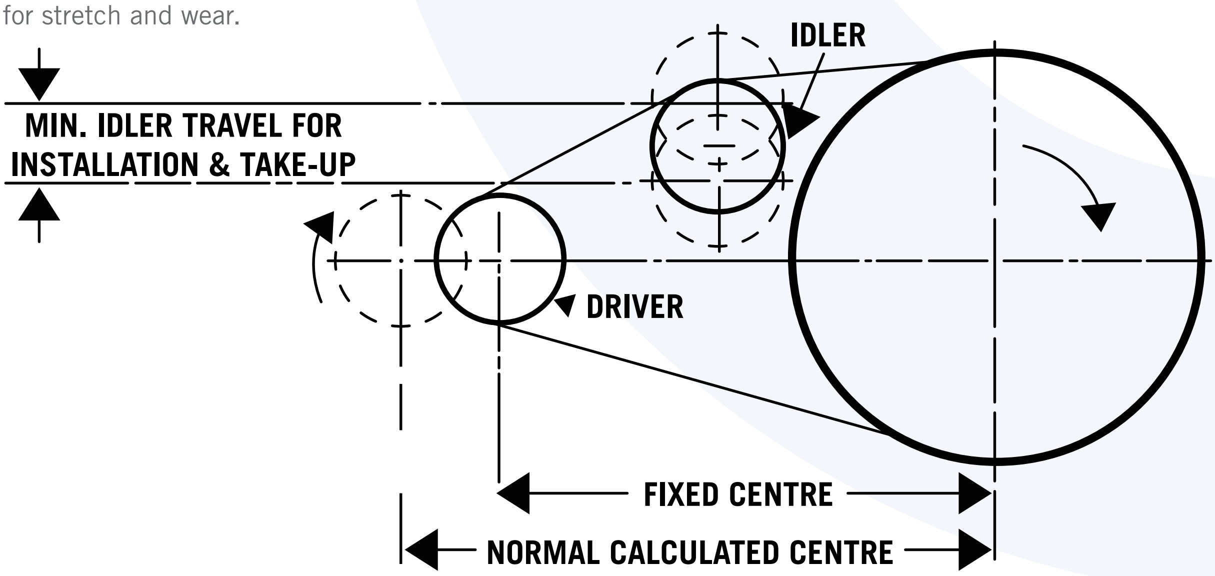
IDLERS FOR FIXED CENTRE V AND WEDGE BELT DRIVE APPLICATIONS

There must be some provision in the installation of a belt drive for "take-up" to compensate for limited belt stretch and to provide for assembly of belts without undue strain. This adjustment is usually taken care of in the motor rails or sliding base, or by some special provisions, such as shims, etc. On fixed centre applications, when neither the DriveN nor the DriveN unit can be moved to provide necessary adjustment, a mechanical belt tightener such as an idler is necessary. When designing a fixed centre installation, centres should be set so that belts do not have to he forced over the pulley grooves. Calculate the exact running centre distance in mm's and then deduct the following amount for each particular belt section to facilitate belt mounting:

Z	13A	17B	22C	32D
9.0	11.1	14.7	19.0	25.4

SPZ	SPA	SPB	SPC
11.5	12.5	19.0	25.0

The initial operating position of the idler should be located at the point where the idler pulley just takes up the belt slack caused by the reduced centres necessary for belt installation.
Final idler position design should allow for take-up equal to twice the recommended minimum allowance for stretch and wear.



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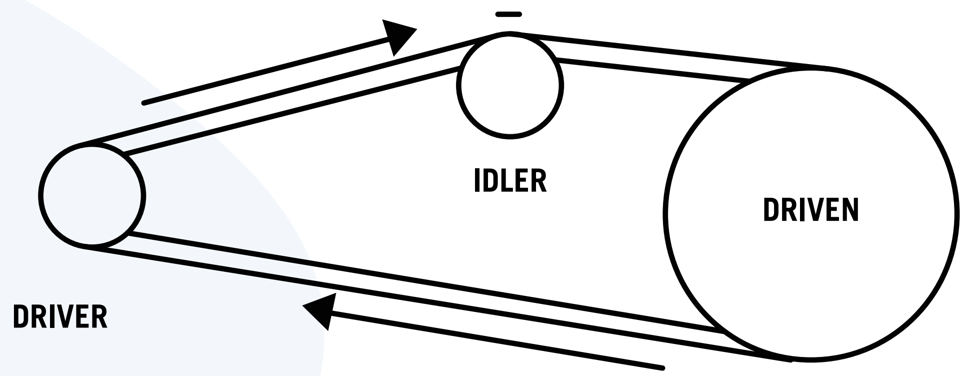
LOCATION AND SIZE OF IDLER

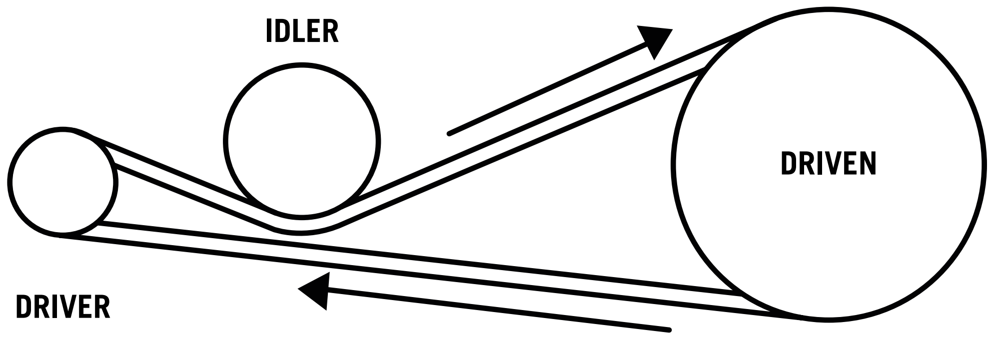
Inside Idler

A grooved idler on the inside of the belts on the slack side of the drive is recommended over a back side idler. Always place the idler as close to the large pulley as possible. The size of the grooved idler pulley should be as large or larger than the small pulley in the drive.



Back Side Idler

Although a back side idler increases the arc of contact on both pulleys, it forces a backward bend in the belts and contributes to premature failure. If such an idler must be used, the diameter of the flat idler pulley should be at least 11/2 times the diameter of the small pulley in the drive located as close to the small pulley as possible.

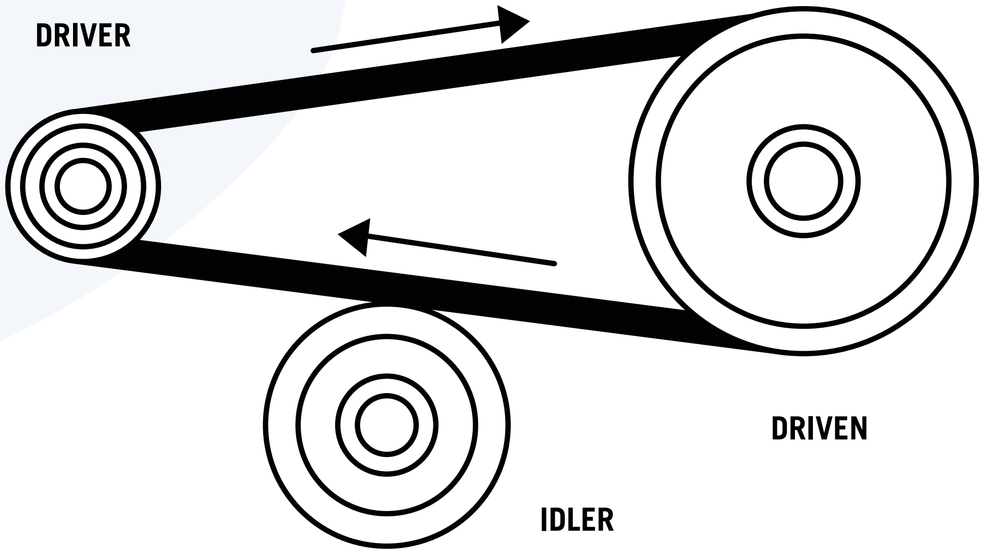


Kiss Idler

A kiss idler differs from the back side idler because it does not penetrate the belt span creating a back bend on the belt and conse- quently does not contribute to premature failure of the belt.

The usage of kiss idlers is not too common, however it provides a method of controlling belt vibration and whip on shock and pulsa- ting load drives. The kiss idler could be used in single belt drives where joined belts are not applicable.

If a kiss idler is used, the diameter of the flat pulley should be at least 11⁄2 times the diameter of the small pulley.



CALCULATION OF V BELT DRIVES

CMW V belt drive guarantees you extremely high efficiency and long lifetime.

In order to get the best performances a correct calculation must be done.

Here below are shown the required equations and factors for calculation as well as the calculations steps.

Required data for correct calculation of a timing belt drive are:
- Driven machine type
- Drive motor type
- Motor power and/or required driving power
- Operating factor
- Rotational speed of the motor shaft
- Rotational speed of the driven shaft
- Center distance

Total service factor Co

The total service factor Co is determined by adding factors C1, C2 and C3:

$$C_0 = C_1 + C_2 + C_3$$

Acceleration factor C1

Transmission ratio Rt	Acceleration factor c3
1,00 - 1,25	-
> 1,25 - 1,75	0,1
> 1,75 - 2,50	0,2
> 2,50 - 3,50	0,3
> 3,50	0,4

$$R_t = Z_1/Z_2$$

Glossary of symbol, unit and definition

Symbol	Unit	Definiton
a	mm	Center distance
c0	-	Predefined total service factor
c0 err	-	Calculated total service factor
c2	-	Load factor
c1	-	Acceleration factor
c3	-	Fatigue factor
c4	-	Length factor
Rt	-	Transmission ratio
Lw	mm	Pitch length of v-belt
n1	min-1	Speed of driving pulley
n2	min-1	Speed of driving pulley
P	kW	Power to be transmitted
PR	kW	Power rating for selected width of belt
Pm	kW	Engine power
d	mm	Pitch diameter of driving pulley
D	mm	Pitch diameter of driven pulley

Fatigue factor c3

If idlers are used, add the following to the service factor.

Idler On Slack Side (Inside)
None

Idler On Slack Side (Outside)
0,1

Idler On Tight Side (Inside)
0,1

Idler On Tight Side (Outside)
0,2

LOAD FACTOR C_2

The correct service factor is determined by:

1. The extent and frequency of peak loads.

2. The number of operating hours per year, broken down into average hours per day of continuous service.

3. The proper service category (intermittent, normal or continuous). Select the one that most closely approximates your application condition.

Intermittent service

a. Light Duty – Not more than 6 hours per day.
b. Never exceeding rated load.

Normal service

a. Daily service 6 to 16 hours per day.
b. Where occasional starting or peak load does not exceed 200% of the full load.

Intermittent service

a. Where starting or peak load is in excess of 200% of the full load or where starting or peak loads and overloads occur frequently.
b. Continuous service 16 to 24 hours per day.

Typical Service Factors

DRIVEN MACHINES TYPES		DRIVER TYPES					
Driven Machine Types noted below are representative samples only. Select a category most closely approximating your application from those listed below		**ELECTRIC MOTORS:** AC Normal Torque Squirrel Cage and Synchronous AC Split Phase DC Shunt Wound **Internal Combustion Engines ov**er 600 rev/min			**ELECTRIC MOTORS:** AC Hi-Torque AC Hi-Slip AC Repulsion-Induction AC Single Phase Series Wound AC Slip Ring DC Compound Wound Series Wound Single Cylinder Engines and Internal Combustion Engines under 600 rev/min, line shafts, brakes, clutches, direct on line starting.		
DRIVEN MACHINE TYPES		**"SOFT"/NORMAL TORQUE STARTS**			**"HEAVY"/HIGH TORQUE STARTS**		
		Intermittent Service	**Normal Service**	**Continuous Service**	**Intermittent Service**	**Normal Service**	**Continuous Service**
Agitator: Liquid	Light Duty	1,0	1,1	1,2	1,1	1,2	1,3
Blowers and Exhausters	Light Duty	1,0	1,1	1,2	1,1	1,2	1,3
Centrifugal pumps and compressor	Light Duty	1,0	1,1	1,2	1,1	1,2	1,3
Fans up to 7,5 kW	Light Duty	1,0	1,1	1,2	1,1	1,2	1,3
Light Duty Conveyors	Light Duty	1,0	1,1	1,2	1,1	1,2	1,3
Belt Conveyors for Sand, Grain, etc.	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Dough Mixers	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Fans over 7,5 kW	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Generators	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Line Shafts	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Laundry Machinery	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Machine Tools	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Punches-Presses-Shears	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Printing Machinery	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Positive Displacement Rotary Pumps	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Revolving and Vibrating Screens	Medium Duty	1,1	1,2	1,3	1,2	1,3	1,4
Brick Machinery	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Bucket Elevators	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Exciters	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Piston Compressors	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Conveyors (Drag-Pan-Screw)	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Hammer Mills	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Paper Mill Beaters	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Piston Pumps	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Positive Displacement Blowers	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Pulverizers	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Saw Mill and Woodworking Machinery	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Textile Machinery	Medium Duty	1,2	1,3	1,4	1,4	1,5	1,6
Crushers (Gyratory-Jaw-Roll)	Extra Heavy Duty	1,3	1,4	1,5	1,5	1,6	1,7
Mill (Ball-Rod-Tube)	Extra Heavy Duty	1,3	1,4	1,5	1,5	1,6	1,7
Hoists	Extra Heavy Duty	1,3	1,4	1,5	1,5	1,6	1,7
Rubber Calenders-Extruders-Mills	Extra Heavy Duty	1,3	1,4	1,5	1,5	1,6	1,7
Chokable Equipment		Minimum Service Factor 2.0					

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BELT CROSS SECTION SYMBOL

SPZ/3V		SPA		SPB/5V		SPC/5V	
Datum Length	L_c Factor	Datum Length	L_c Factor	Datum Length	L_c Factor	Datum Length	L_c Factor
630	0,83	800	0,82	1250	0,85	2000	0,86
710	0,85	900	0,84	1400	0,87	2240	0,88
800	0,87	1000	0,86	1600	0,89	2500	0,90
900	0,89	1120	0,88	1800	0,91	2800	0,91
1000	0,92	1250	0,90	2090	0,92	3150	0,93
1120	0,94	1400	0,92	2240	0,95	3550	0,95
1250	0,96	1600	0,94	2500	0,96	4000	0,97
1400	0,98	1800	0,96	2800	0,98	4500	0,98
1600	1,00	2000	0,98	3150	1,00	5000	1,00
1800	1,02	2240	1,00	3550	1,02	5600	1,02
2000	1,04	2500	1,02	4090	1,04	6300	1,04
2240	1,07	2800	1,04	4500	1,06	7100	1,05
2500	1,09	3150	1,06	5000	1,08	8000	1,07
2800	1,11	3550	1,08	5600	1,09	9000	1,09
3150	1,13	4000	1,10	6300	1,11	10000	1,11
3550	1,15	4500	1,12	7100	1,13	11200	1,12
-	-	-	-	7800	1,15	12500	1,14

Z		13A		17B		22C	
Datum Length	L_c Factor	Datum Length	L_c Factor	Datum Length	L_c Factor	Datum Length	L_c Factor
530	0,92	630	0,80	930	0,81	1560	0,82
625	0,95	700	0,82	1000	0,83	1760	0,84
700	0,98	790	0,84	1100	0,85	1950	0,87
780	1,00	890	0,86	1210	0,87	2190	0,90
920	1,04	990	0,88	1370	0,90	2340	0,91
1080	1,07	1100	0,90	1560	0,92	2490	0,92
-	-	1250	0,93	1760	0,95	2720	0,94
-	-	1430	0,96	1950	0,97	2800	0,95
-	-	1550	0,98	2180	0,99	3080	0,96
-	-	1640	0,99	2300	1,00	3310	0,98
-	-	1750	1,00	2500	1,02	3520	0,99
-	-	1940	1,02	2700	1,04	4060	1,02
-	-	2050	1,04	2850	1,05	4600	1,05
-	-	2200	1,05	3200	1,08	5380	1,08
-	-	2300	1,06	3600	1,10	6100	1,11
-	-	2480	1,08	4060	1,13	6860	1,14
-	-	2570	1,09	4430	1,15	7600	1,16
-	-	2700	1,10	4820	1,16	9100	1,21
-	-	2910	1,12	5000	1,18	10700	1,24
-	-	3080	1,13	5370	1,19	-	-
-	-	3290	1,14	8070	1,20	-	-
-	-	3540	1,16	-	-	-	-