8 Efficient transmission of loads

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The power from the motor shaft to the driven shaft can be transmitted in many ways, depending upon the power and load application. In this chapter we deal with the most common types of driving systems, their influence on the starting characteristics of the motor and their effect on bearings.

8.1 Direct or rigid couplings

These are suitable for smaller ratings only because of their weight, which also includes the weight of the load which falls directly onto the motor and its shaft. The driven equipment, such as a mono block pump is mounted directly on the motor shaft (Figure 8.1). Such couplings offer the advantage of design simplicity but do not allow for any misalignment. They are used for low-speed applications or where only marginal misalignment is anticipated.

8.2 Flexible couplings

The load is now transmitted through ordinary couplings, one half mounted on the motor shaft and the other half on the driven shaft. They are bolted together with a rubber pad between the two. The motor and the driven machine are mounted on a common bed (Figure 8.2). They are now able to provide margin for misalignment of the two shaft ends and thus extend more flexibility.

Note

For efficient load transfer one may opt for direct or flexible couplings as far as possible than belts.

8.3 Delayed action couplings

In the above two types of couplings the transmission of torque is linear, but now the driving torque rises as the square of the input speed \( T \propto N^2 \). These couplings facilitate engagement of the motor shaft with the driven shaft after a pause, by when the motor nearly picks up its rated speed. They are of two types:

1. Fluid or hydraulic couplings
   These are available in two designs:
   - Constant filled or traction type and
   - Variable filling or scoop control type

2. Magnetic couplings

Delayed action clutching helps the load in the following ways:

- It enables the motor to have a soft mechanical start. The motor picks-up lightly and quickly and reduces the pressure on the supply source.
- The motor itself undergoes less stress.
- This is a type of a mechanical clutch that enables the motor start almost at no-load, irrespective of the type of load.
- These couplings are adjustable and therefore can also facilitate speed control.
- Since they work like a mechanical clutch, and engage only slowly and gradually, these couplings enable the motor to have a light start, on the one hand, and a smoother engagement of the load shaft, on the other. It avoids a jerk on the load and an excessive torque demand on the motor. In normal couplings, a high inertia load results in a longer start and demands a high starting torque, and requires either a higher rating or a specially designed high starting torque motor. With the application of delayed action couplings, it is possible to use a lower rating and a normal torque motor. Proper application of these couplings can thus result in low initial cost of electrical accessories (motor, cables, control gear and capacitors etc.) and a substantial saving on energy by
  - Selecting a normal low h.p. motor, even for stringent load requirement and adopting to a DOL switching. Squirrel cage motors are now more appropriate than slip-ring motors for all kinds of applications.
  - Minimizing the starting losses and strain on the feeding lines, as the motor now starts up lightly and accelerates quickly.
  - Lesser wear and tear, low maintenance and consequent longevity of all electrical and mechanical components connected to the motor and the drive also result in substantial saving in the long run.
Eliminating damper, vane or throttle controls, which are a constant source of energy drain for drives having variable load demands. This is possible by achieving automatic speed regulation of the drive through variable-speed couplings. For automatic control of the couplings, sensing devices may be placed in the flow circuit of pumps, fans or any process line that has variable parameters. (See Figure 6.38.) The couplings can thus control the flow of air or fluid by automatically regulating the speed of the drive and using only that amount of power that is actually needed by the load, resulting in substantial energy saving. See Section 6.15.2, which analyses the amount of energy saved.

Here we have attempted to provide only an introduction to the application of fluid couplings for soft starting and energy saving. For more details one may consult the manufacturer. It is however expected that this brief presentation of the subject would suffice a reader or the user to make a better choice between static soft drives, variable drives, and the fluid couplings, to fulfil the operational requirements and save on energy and overall cost. In Table 6.5 we have drawn a broad comparison between a.c. drives and fluid couplings for one to make a better choice.

**Environment-friendly couplings**

These couplings are totally enclosed and are suitable for any environment prone to fire hazard, corrosion, dust or any other pollutants. The controls can be provided remotely in a safer room and the pushbutton stations, which can be easily made suitable for such environments, located with the drive.

### 8.4 Construction and principle of operation

#### 8.4.1 Fluid or hydraulic couplings

These couplings are made in two parts, the impeller as the inner part and the runner (rotor) as the outer. Both are enclosed in a casing. The impeller and runner are bowl shaped and have large number of radial vanes as shown in Figure 8.4. They are separated by an air gap and have no mechanical inter-connection. The impeller is mounted rigidly on the motor shaft while the runner rotates over the impeller, with a uniform air gap and is connected to the load through a flexible coupling or belt drive. It is advisable to mount the drive on the load side and impeller connected to the motor through a flexible coupling to save motor from extra burden. The impeller is filled with fluid (mineral oil) from a filling plug. A fusible plug is provided on the impeller, which blows off and drains out oil from the coupling in the event of sustained over-loading, a locked rotor or stalling.

As the motor picks-up speed, it builds up a centrifugal force in the impeller, which causes the fluid to spill out and fill the air gap between the two couplings. As the air gap is filled only gradually, the runner engages with the impeller gradually. The impeller converts the mechanical power into hydraulic power and the runner converts this back to mechanical power. The transmission of power from the runner to the load may be through flexible couplings or belts as noted above. The torque transmitted during the start is proportional to the square of the motor speed \(T \propto N^2\). There may be small speed variations at the secondary side, which will be due to slip between the impeller and the runner (generally of

![Figure 8.3](image-url) Smoother and quicker acceleration of heavy-duty loads with the use of a delayed-action coupling

\[ N_1 = \text{Coupling speed (Indicates coupling slip)} \]

\[ T_r = \text{Load starting torque} \]
Efficient transmission of loads

the order of 2–5%) at light to heavy loads. Thus the clutching at full speed is rigid.

These couplings are produced in three basic designs,
- Constant filled or traction type (constant-speed couplings)
- Delayed action or delayed filling chamber fluid drives
- Variable filling or scoop controlled fluid couplings.

1. Constant filled or traction type fluid couplings (constant-speed couplings)

These are pre-filled fluid couplings and provide only fixed torque characteristics. The quantity of oil filled in the coupling cannot be varied during running, hence the name. The quantity of fluid can, however, be changed by the manufacturer depending upon the starting or the load requirement. The couplings have a safety device in the form of a fusible plug. The plug blows off and drains out oil from the coupling to provide protection to the coupling and the motor against excessive temperature rise, which may occur due to sustained over-load or under stalled condition. Couplings are usually designed for an operating temperature of 85°C unless designed for higher temperatures. Continuous overheating may damage the couplings. Figures 8.4 and 8.5 illustrate these couplings and Figure 8.6 shows a general coupling arrangement. The operating slip is of the order of 2–5% as noted above. Since they are pre-filled, they are constant-speed couplings.

In view of their easy adaptability, they are manufactured in standard sizes suitable for common types of load applications. They are therefore available on short deliveries, in smaller ranges, say 1 kW to 200 kW and are manufactured up to 2500 kW or so by different manufacturers. They can be selected for the required service conditions (as a motor is selected), to suit a particular load requirement from the available sizes of couplings. These are generally used for conveyor types of loads.

Applications in different industries
- **Automobile Industry**: tanks, cars, tractors, forklift-truck
- **Cement Industry**: fans, aerial ropeways, conveyors, crushers, rotary kiln, coal pulverisers
2. Delayed action or delayed filling chamber fluid drives

A delayed feature is incorporated in the impeller of the constant filled couplings by way of retaining a specific quantity of oil in its periphery during start-up and transferring same to the working circuit via ports provided in the impeller when the coupling has come to speed. Oil quantity can be adjusted according to the speed-torque requirement of the load. The drive now exerts least jerk on the load or the drive itself. The runner clutches with the load when the motor has run up to almost its rated speed as a no-load start, relieving it of all starting stresses and so also stresses on the connected switchgears and cables. Usually the clutching is adjusted so as to happen after the $T_{po}$ region of the motor-torque curve as far as possible to enable the load pick-up up to the rated speed as smooth as possible. Figure 8.3 illustrates this where the clutching point ‘A’ can be adjusted at any point on the motor torque curve according to the torque demand of the load. In Figure 8.3 we have considered this corresponding to $T_{po}$ region of the motor...
Sequence of clutthing in a delayed action drive

Consider a heavy inertia load such as a conveyor, having characteristics according to Figure 2.13. Choose a standard motor having the characteristics shown in Figure 8.3. The fluid coupling will have centrifugal characteristics during start ($T \propto N^2$) as shown. The load-transmitting characteristic of the coupling is preset (by varying the quantity of oil in it) so that the load will clutch at point A in the graph (Figure 8.3) by which time the motor will reach about 80–90% of its rated speed with a torque somewhere in the $T_{po}$ region. By now the motor’s initial inrush current will also droop and motor will be operating almost at its normal running condition. The motor will thus pick-up lightly and smoothly with an accelerating torque, $T_{a1}$, with no strain on it or the supply system. After point A the coupling will gradually clutch with the load with an accelerating torque $T_{a3}$. $T_{a3}$ being high, will accelerate the load quickly once again, without any strain on the motor or the supply system.

3. Variable filling or scoop controlled fluid couplings (variable-speed couplings) (Figures 8.7 (a) and (b))

Through such couplings the output speed of the runner can be changed by varying the volume of oil in the working circuit through the scoop operating lever, as shown in Figure 8.7(b). When the oil volume is full, slip is at minimum and the output speed is maximum. As the oil circuit is emptied, the slip gradually increases. A constant-speed motor can thus be used to provide a stepless variable-speed drive. Speed variation is possible up to 15–20% of $N_r$ for centrifugal loads such as fans and pumps.

Figures 8.7(a) and (b) illustrate variable-speed couplings which can provide a stepless speed variation over a wide
The impeller and runner of the couplings are housed in a stationary housing with a built-in oil sump. Oil is continuously introduced into the working circuit through a separately driven oil pump. A scoop tube is provided to regulate the volume of oil in the working circuit and hence control the speed and the transmitting torque. The position of the sliding scoop tube can be governed by a servo-actuator or manually. The coupling can be attached with the motor on one side and load on the other through flexible couplings as shown in Figure 8.7(a).

Regulation of the quantity of oil in the working circuit between the impeller and the runner is effected by an adjustable scoop tube, which slides in the stationary housing (Figure 8.8). When the fluid coupling is at rest, the oil level is below the opening in the casing. With the scoop tube retracted radially inwards, on starting up the set the oil forms a rotating ring in the reservoir casing due to centrifugal force. The casing is of sufficient capacity to contain the full quantity of oil, clear of the tip of the scoop tube, so that the working circuit remains empty, when the drive is disconnected (Figure 8.8(a)).

By sliding the scoop tube radially outwards its tip enters the rotating ring of oil and a quantity of oil is picked up by the scoop tube and transferred to the working circuit. With the working circuit partially filled (Figure 8.8(b)) the output shaft is driven at a reduced speed or the torque capacity of the drive is restricted. With the scoop fully extended (Figure 8.8(c)) all the oil is transferred to the working circuit, bringing the coupling output shaft up to full...
Efficient transmission of loads

speed. While the coupling is running, oil escapes from the working circuit into the reservoir casing through small leak-off holes, and is returned to the circuit by the scoop. The radial position of the scoop tube determines the depth of the oil ring in the reservoir casing and thereby the volume of oil in the working circuit.

Fluid couplings have a proven record of reliability and ease of operation. They thus have found wide applications wherever a light load start and capacity control of drives is required as in fans and pumps.

The range is from 90 kW to 30 000 kW and applications are as follows:

- **Steel works:** conveyors, crushers, wagon tipplers, skid gears, furnace charges, bogie drives in bogie hearth furnaces, furnace winch drives, cranes, pumps, compressors and fans.
- **Powerhouse auxiliaries:** ID fans, FD fans, primary and secondary air fans, boiler feed pumps and slurry pumps, coal mill drives, conveyors, pulverizers and wagon tipplers.
- **Docks and harbours:** Material handling, mining, railway traction, process and chemical plants.

Most of these applications are heavy inertia loads and require a light load start. The use of such variable drives save large energy.

**Advantages of variable-speed fluid couplings**

1. It is possible to switch the motor at no-load by selecting a variable-speed fluid coupling. The oil need not be pre-filled which can be varied at site during operation as required.

2. The couplings can be designed to develop torques even higher than the $T_{po}$ of the motor. They can thus be made compatible for transmitting loads up to the optimum capacity of the motor and sustaining abnormal load conditions without stalling or damage.

3. Basically a variable drive fluid coupling is a tailor-made clutch to suit a specific load duty for more accurate applications. Speed variation up to 5:1 at variable torques (constant HP) and 3:1 at constant torque is easily possible.

4. They are suitable for stepless speed variation and can be controlled through speed, torque, temperature or flow of a process. They can also be programmed for any sequence of operation.

5. Conventional throttling or damper control of flow is waste of energy. The use of such couplings can vary speed through oil control, relieve strain on the system and save on the energy consumption. The power transferred by the drive is proportional to the cube of the rated speed. There is therefore large energy saving at lower speeds. These drives therefore fall under energy efficient devices.

*Note*

Since these are energy saving devices, they are entitled for state subsidies/incentives like any other energy saving device discussed elsewhere in the book (see also Section 1.19). They constitute ideal energy saving devices even for retrofitting the old installations to turn them to efficient and energy efficient systems.
Analysis of heavy duty loads for application of fluid drives

In the following brief discussion we have attempted to analyze a few typical but stringent load demands and the basic criteria to choose the right type of drive for a smoother transfer of power from motor to the load. These examples can form broad guidelines for a user to make a better choice between the types of drives available in the market, for his kind of load.

Basic criteria

– Soft start
– Speed control where necessary, and
– Energy saving

To choose the conventional methods of starts like DOL, Y/A or A/T switchings and transferring power through rigid or flexible couplings is too rudimentary an approach in today’s scenario. In fact a thorough load study is essential even for normal duty and small loads, as there may be enough scope to save large energy by incorporating an energy saving device, in the system. Such as using a static drive or a fluid coupling. Below we discuss a few large and stringent load demands for heavy starting and running duties for the selection of an appropriate soft starting and load transfer device.

1 Crushers

As used at thermal power plants, cement plants and mines. The main features of such loads are,

– High inertia with intermittent shock loadings because of larger lumps of say, coal or dolomite or choking of hopper during crushing, calling for high starting torque \( T_{st} \) (see Figures 2.12 and 2.13), prolonged starting time \( t_s \) and consequently high thermal withstand capability of the motor.
– Meaning thereby more strain on motor and the associated electricals (switchgears and cables).
– High risk factor of the failure of motor (particularly rotor bars during start) and reduced lives of all associated electricals.
– Higher stresses on the crusher’s various parts and mechanical system associated with it.

Solution

The load calls for no speed variation except a soft start. A delayed action fluid coupling will suit the demand, being economical compared to a static soft drive and possessing all features that can handle such a load. Figure 7.22 illustrates a smooth start using a fluid coupling. A smooth start will facilitate,

– Selection of a standard motor
– A no-load start shall relieve all stresses on motor, electricals, drive and mechanical devices such as gears, belts and the crusher itself. Meaning thereby lesser damages, lower maintenance and enhanced longevity.
– A quick start would mean energy saving. So also during the lean periods as the drive will automatically
adjust its speed and so the power consumption as per
the load requirement.
- Variation in voltage during pick-up shall be immaterial
  as the motor is soft start.
- The payback period is short irrespective of only
  moderate energy saving as this too amounts to a
  sizeable saving.

2 Slurry Pumps
As used at thermal power stations, cement, fertilizers,
metal and steel plants and ore slurry transportation. The
main features of such loads are,
- regular variation in flow of slurry – requiring a vane
  control or throttling or a variable speed drive.

Solution
The load does not call for an accurate speed control.
The best solution would be to go in for a variable speed
scoop control fluid coupling with automatic, semi-
automatic or manual control feature whichever is more
appropriate for the load conditions, to relieve the stresses
on the pump and the pipelines and facilitate large and
recurring energy saving. The payback period is very
short (usually a few months) because of continuous nature
of such plants.

3 Rolling mills
Non-continuous rolling and re-rolling mills essentially
employ a high inertia flywheel to release its stored energy
when the rolling load comes,
- Many mills employ slip-ring (wound) motors to meet
  high starting torque $T_{st}$ and prolonged starting time
  $t_s$ requirements.
- The conventional belt drives act as slipping element
during start and when the load comes. Belts are kept
  loose deliberately to allow them slip during start and
during heavy loads as a safety measure. This causes
  heavy wear of belts and frequent replacements.
- There is enormous power waste because of slippage.
- Prolonged starting time $t_s$ is required to accelerate
  the flywheel.
- There is a shock-loading feature too due to flywheel.
- While performing its rolling duty the motor slows
down and gives current over-shoots.

Solution
Such loads require no speed variation except for a smooth
start of the motor. A delayed action fluid coupling should
provide the best answer by facilitating a soft start. On
load the slip of the coupling may rise up to 8–12% or so
giving an opportunity to the flywheel to release most of
its stored energy to the rolling load. In conventional
drives (belt-drives) the fall in speed may remain in the
range of 3–5%. While this may suggest a good speed–torque characteristic of the motor, it is not desirable for
the flywheel to perform its optimum duty. It is estimated
that the flywheel may give out about 3–6 times more
energy at higher speed variations with slips at 8–12%
and this is gainfully utilized for rolling operations. The
energy released by the flywheel is proportional to the
square of difference in the two velocities (Equation
(3.14)). Higher the difference in speed, higher is the
energy released and can roll higher sections in the same
mill. See Figure 8.8(d).
- Light load start saves wear and tear of belts, gearboxes,
foundation of mill and other mechanical devices and
transmission elements associated with it.
- Reduces maintenance, breakages and enhances
  longevity.
- No need to keep the belts loose.
- All this results in a large energy saving during start,
  rolling operations and lean periods.
- No need to go in for a wound motor.

Retrofitting
An ideal case of retrofitting of all old installations for
energy conservation, guaranteeing a very short payback
period.

4 Large conveyor drives
Where there is material handling there is conveyor system
to transport raw material from one head to another like
from a mine head to a kiln.
Load characteristics:
- The motor may be switched when the conveyors are
  fully loaded calling for a high starting torque $T_{st}$ and
  prolonged starting time $t_s$.
- In conventional drives, therefore, the rating of the
  motor is based on starting characteristics of the
  conveyors rather than the running load.
- So also are chosen other electricals for the system.
- There might be many conveyors running in tandem.
  Stoppage of one would result in spilling over of all
  material on the upper conveyors on to the stopped
  conveyor. To avoid this all the conveyors are stopped.
  It calls for continuous de-clutching of the conveyor
  system, and long down-time.
- Every switching stresses the belts besides the motor
  and the electricals and causes frequent breakages –
calling for higher section of conveyor belts, sturdier

\[ \text{Figure 8.8(d) Release of flywheel energy with the use of delayed action fluid coupling (Courtesy: Fluidomat)} \]
foundations and mechanical system associated with
the conveyors.
– Large systems usually employ MV motors and it is
prohibitive to switch them frequently leading to
unwarranted long down-times. (After a trip on load
the motor calls for a cooling time of a few minutes
before a re-start.)

Solution
Since there is no specific speed control, a variable speed
scoop control fluid coupling would be an ideal choice
extending facility of continuous de-clutching requirements.
In case of a problem the belts may be stopped while the
motor is still running at light load facilitating,
– Selection of a standard motor and so also electricals
associated with the motor.
– A soft start will relieve all stresses on the motor,
electricals and so also the drive, conveyors and
mechanical system-gearboxes, shafts, belts and
foundations associated with the conveying system.
– Lesser breakages and maintenance, thereby enhancing
longevity of the system.
– \( T_{st} \) and so also starting time \( t_s \) can be adjusted as
desired through the scoop control.
– There is now flexibility of operation, facilitating
emergency stops during overflowing or mismatch of
drive without jerks. It also takes care of pull chord
and sway switch operations.
– All this means a large and recurring energy saving.

Retrofitting
An ideal case of retrofitting of all old installations for
energy conservation, guaranteeing a very short payback
period.

5 Forced draught (FD) fans
These are usually large fans and switched with damper
closed or partially opened to meet the process flow and
system requirements, resulting in vibrations and enormous
loss of energy. Besides this, the damper control may not
be able to provide the desired accuracy to the process.

Solution
The FD fan essentially is a variable speed drive to control
the flow of air as required by the process but does not
call for an accurate speed control and therefore a variable
speed scoop control fluid coupling \( (N_1 - 4:1) \) will be
ideal for this application, facilitating,
– Selection of a standard motor and the electricals.
– There is an enormous and recurring energy saving as
there is no need to provide dampers. The flow control
can be achieved by varying the speed through the fluid
drive, loading the motor only as much as is necessary.

Example 8.0
Consider the following FD fan
Fan – 1125 kW 980 r.p.m.
Motor – 1250 kW 990 r.p.m.

Energy saving – For ease of calculations, if we consider an
average speed of the fan as 700 r.p.m., i.e. 700/990 or
about 70% of \( N_r \), then approximate capacity utilization of
the motor = \( 1250 \times 0.61 \) (0.61 being the derating factor at
the lower speed, Table 5.3)

or 762.5 kW

Considering the process as continuous be it a power station,
a fertilizer plant or any other process line and operating for
minimum 300 days a year and 24 hours a day, then energy
saving per year

\[ = (1250 - 762.5) \times 24 \times 300 \text{ kWh/year} \]

\[ = 3.51 \text{ million units}. \]

This is when we have assumed very conservative
operating parameters, actual saving would be much larger
than this. This multiplied by the local tariff gives an
idea of the enormous savings the drive will result in
in terms of money. One may notice that the payback period
will be just a few months and huge recurring savings
thereafter.

Retrofitting
This too is an ideal case of retrofitting of all old
installations for enormous savings in terms of energy
and money.

6 Primary Air (PA) Fans
They are usually employed at thermal power plants at
the boiler house. The main feature of such loads is high
inertia and other features being the same as for the
 crushers discussed in item 1, except that there is no
shock loading now. As it calls for no speed variation a
delayed action fluid coupling shall provide the best
answer.

The above are a few typical applications where delayed
action or variable speed scoop drive couplings will
provide an ideal solution to the operational constraints,
heavy initial cost of electricals, high recurring cost and
large risk factor of failure of motors and electricals,
higher maintenance and breakages of belts and
mechanical systems. They also facilitate large recurring
energy savings. Static soft drives or variable speed drives
are indeed not essential for all such applications unless
the user likes to specifically go in for static controls as
he may have already installed many more static drives
for his other plant requirements and extra cost of a few
more static drives is of no relevance to him. Now one
can save even the fluid coupling losses because of its
own slip. No doubt even this additional cost on a static
drive shall also be recovered within a short period.

Necessity of retrofitting
Energy saving is an essential requirement at all costs
and at all installations. Gradually energy saving may
become mandatory for the user and the consultan to
identify and suitably address equipment and installations
incurring any kind of energy waste that can be saved or
minimized.
Fluid drive surely is a handy energy saving device to address such situations at most installations and ratings is no bar. Fluid drives together with energy efficient belts can make enormous savings at an installation. All such costs are recoverable within a very short period, usually just a few months, depending upon size, operating requirements and running hours. For comparison between fluid drives and static drives see Table 6.5.

8.4.2 Magnetic couplings and eddy current couplings

The principle of operation of these couplings is almost the same as of fluid couplings except the air gap between the impeller and the runner of the coupling, is now filled with iron granules or iron powder instead of fluid. The iron powder condenses into a solid mass when magnetized through an external exciter. The exciter is mounted on the same coupling, and clutches the runner with the impeller. The power of transmission as well as the speed of the runner can be controlled by varying the excitation.

Eddy current couplings have become more common. In this case the impeller of the coupling is a ferromagnetic drum, coupled to the induction motor and housed in an outer shell (Figure 8.9(a)). The shell is fitted with a magnetic yoke and an excitation coil. Within this drum, a multi-pole inductor, made from special alloy steel, rotates freely, maintaining a close and constant air gap. This forms the driven part or the runner of the coupling. When the excitation coil is energized, it develops a magnetic field in the coupling’s runner and impeller. The relative motion between them causes the magnetic field to concentrate on the surface of the drum. This results in a flow of eddy currents in the metallic drum which causes a torque transmission from the prime mover to the runner. The transfer of load from impeller to runner is performed without any mechanical connection. The torque transmitted is a function of the d.c. excitation. By varying the d.c. excitation, the output speed of the drive can be varied smoothly even up to 7–10% of \( N_r \). Similarly, the output speed can be maintained constant, even on a change of load, by sensing the output speed and monitoring the excitation level through a closed-loop speed feedback control system. The speed can be sensed through a tacho-generator mounted integrally on the load shaft. Very accurate feedback control systems are also possible through microprocessor-based analogue and digital controls, as discussed in Sections 6.6 and 13.2.3 to achieve total automation of speed, torque and power or any other process parameter such as, flow of liquid, gas or temperature etc.

Figure 8.9(b) illustrates a general scheme to achieve speed control through such couplings. These couplings, up to 90 kW, are easily available.

Applications

This is as for fluid couplings but at lower ratings. Generally, they are used in the cement, rubber and chemical, paper, chemical fibre, electric wire making and mining industries, as well as in material handling, conveyors and thermal power plants.

8.5 Belt drives

Belt drives are employed to transmit load from the driving shaft of the motor to the driven shaft of the load when they are separately located. With continued R&D over the years the belt drives too have undergone a sea change
improving enormously their power handling capability, stability and efficiency of power transmission. The usual types and sections of belts being produced by leading manufacturers worldwide are noted in Table 8.01.

The flat belt and all types of V belt drives are friction drives transmitting load through the friction between the belt and the pulley, while the synchronous belt is a positive drive having no slip whatsoever. Until about a decade ago for transmission of smaller loads, flat belts and for small and medium sized loads, V belt drives were universally employed.

Flat belts with normal tension have larger slips and with higher tensions cause more strain to the belts and the bearings, and are redundant in today’s scenario. But for general reference and historical significance as also till the existing installations with flat belts get retrofitted with the state-of-the-art energy efficient and space saver V belt drives as noted next, we have retained the section on flat belts also.

8.5.1 Flat belts

These are long centre drives with small slips. The slack side of the belt is kept on the top side to increase the angle of contact with the pulleys by sag on the top side. This is essential for an efficient transfer of load. The recommended maximum power that can be transmitted by one belt of different cross-sectional areas is provided by the belt manufacturer and some ratings are given in Table 8.1. When selecting these drives, the following parameters should be borne in mind:

1. The ratio of the diameter of the pulleys should not exceed 6:1 unless a jockey (idler) pulley or similar arrangement is made to press the top side to indirectly increase the arc of contact.

2. Similarly, for shorter centre distances between the drive and the driven pulleys, the arc of contact will decrease. To ensure a good arc of contact, the centre distance \( C \) (Figure 8.10), should be kept as much as possible, otherwise the provision of a jockey pulley, as noted above, will also be necessary. A higher arc of contact will ensure a better grip of the belt on the pulley and hence a smaller slip during transmission of the load. A smaller slip would mean a higher transmission of load and vice versa.

3. Vertical and right-angle drives should be avoided.

4. Belts should not be tightened more than necessary, otherwise the drive and the driven shafts will come under torsion and excessive bending moment. The bearings would also be subjected to excessive stresses.

### Specification of flat belts

These belts are made of cotton duck (a cotton fabric used in making canvass and tents) with different mixes to provide them with a degree of hardness, as noted in Table 8.2.

The duck is glued in thin layers (plies) with a rubber compound and vulcanized (cured). The number of plies used to make a belt and their quality defines the strength of the belt (e.g. 3 ply \( \times 32 \) or 3 ply \( \times 34 \), etc.).

### Selection of flat belts for transmission of load

The selection of flat belts is made along similar lines to that for V-belts (discussed later in more detail). The load-transmission capacity of a flat belt can be defined by

\[
W = P \cdot \frac{SF}{\text{Correction for arc of contact}}
\]

where

- \( P \) = load to be transmitted in kW
- \( W \) = maximum load-transmission capacity of the belt.

By convention, this is provided by the belt manufacturer per 25 mm of belt width, at different speeds of the faster shaft (smaller pulley) for

### Table 8.01 Usual types and sections of belts

<table>
<thead>
<tr>
<th>Belt types</th>
<th>Belt sections</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Flat belts (see Table 8.1)</td>
<td>Power ratings for different sizes and speeds of belts are mentioned in Table 8.1</td>
<td></td>
</tr>
<tr>
<td>2. Wrapped V belts</td>
<td>Z, A, B, C, D, E</td>
<td>(a) For sectional dimensions see Table 8.3</td>
</tr>
<tr>
<td>i. Classical belts</td>
<td>SPZ, SPA, SPB, SPC</td>
<td>(b) Belts other than classical belts are usually manufactured in limited sections from economy point of view.</td>
</tr>
<tr>
<td>ii. Space saver or wedge belts</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3. Cogged V belts</td>
<td>AX, BX, CX</td>
<td></td>
</tr>
<tr>
<td>i. Classical-cogged belts</td>
<td>SPZX, SPAX, SPBX, SPCX</td>
<td></td>
</tr>
<tr>
<td>ii. Space saver-cogged belts</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5. Synchronous belts (also known as timing belts or gear belt drives)</td>
<td>L, H, XH, XM, XL etc.</td>
<td></td>
</tr>
</tbody>
</table>

**Note**

A manufacturer can adapt to a variety of designs in these belts also to extend more flexibility to cater to different kinds of site requirements.
Table 8.1  Power ratings (kW/25 mm) for different widths of flat belts with 180° arc of contact on a smaller pulley

### Type I

<table>
<thead>
<tr>
<th>Belt type</th>
<th>Speed of faster shaft (r.p.m.)</th>
<th>Smaller pulley diameter (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>100</td>
<td>125</td>
</tr>
<tr>
<td></td>
<td>720</td>
<td></td>
</tr>
<tr>
<td></td>
<td>960</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1440</td>
<td></td>
</tr>
<tr>
<td>4 x 28</td>
<td>100</td>
<td>0.09</td>
</tr>
<tr>
<td>and 4 x 31</td>
<td>200</td>
<td>0.16</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.23</td>
</tr>
<tr>
<td></td>
<td>400</td>
<td>0.29</td>
</tr>
<tr>
<td></td>
<td>500</td>
<td>0.35</td>
</tr>
<tr>
<td></td>
<td>600</td>
<td>0.41</td>
</tr>
<tr>
<td>3 x 28</td>
<td>700</td>
<td>0.47</td>
</tr>
<tr>
<td>and 3 x 31</td>
<td>800</td>
<td>0.53</td>
</tr>
<tr>
<td></td>
<td>900</td>
<td>0.59</td>
</tr>
<tr>
<td></td>
<td>1000</td>
<td>0.65</td>
</tr>
<tr>
<td></td>
<td>1200</td>
<td>0.75</td>
</tr>
<tr>
<td></td>
<td>1400</td>
<td>0.85</td>
</tr>
<tr>
<td></td>
<td>1600</td>
<td>0.95</td>
</tr>
<tr>
<td></td>
<td>1800</td>
<td>1.04</td>
</tr>
</tbody>
</table>

### Type II

<table>
<thead>
<tr>
<th>Belt type</th>
<th>Speed of faster shaft (r.p.m.)</th>
<th>Smaller pulley diameter (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>80</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>720</td>
<td>0.44</td>
</tr>
<tr>
<td></td>
<td>960</td>
<td>0.56</td>
</tr>
<tr>
<td></td>
<td>1440</td>
<td>0.80</td>
</tr>
<tr>
<td></td>
<td>2880</td>
<td>1.18</td>
</tr>
<tr>
<td>3 x 32</td>
<td>100</td>
<td>0.08</td>
</tr>
<tr>
<td>and 3 x 34</td>
<td>200</td>
<td>0.14</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.20</td>
</tr>
<tr>
<td></td>
<td>400</td>
<td>0.26</td>
</tr>
<tr>
<td></td>
<td>500</td>
<td>0.32</td>
</tr>
<tr>
<td></td>
<td>600</td>
<td>0.37</td>
</tr>
<tr>
<td></td>
<td>700</td>
<td>0.43</td>
</tr>
<tr>
<td></td>
<td>800</td>
<td>0.48</td>
</tr>
<tr>
<td></td>
<td>900</td>
<td>0.53</td>
</tr>
<tr>
<td></td>
<td>1000</td>
<td>0.58</td>
</tr>
<tr>
<td></td>
<td>1200</td>
<td>0.68</td>
</tr>
<tr>
<td></td>
<td>1400</td>
<td>0.78</td>
</tr>
<tr>
<td></td>
<td>1600</td>
<td>0.87</td>
</tr>
<tr>
<td></td>
<td>1800</td>
<td>1.32</td>
</tr>
<tr>
<td></td>
<td>2000</td>
<td>1.44</td>
</tr>
<tr>
<td></td>
<td>2200</td>
<td>1.55</td>
</tr>
<tr>
<td></td>
<td>2400</td>
<td>1.65</td>
</tr>
<tr>
<td></td>
<td>2600</td>
<td>1.75</td>
</tr>
<tr>
<td></td>
<td>2800</td>
<td>1.85</td>
</tr>
<tr>
<td></td>
<td>3000</td>
<td>2.58</td>
</tr>
<tr>
<td></td>
<td>3200</td>
<td>2.67</td>
</tr>
</tbody>
</table>

Source: IS 1370. See also ISO 22
different recommended widths of belts, number of plies and type and grade of duck, etc. We show Type I and Type II in Table 8.1 for a general illustration. These ratings may vary marginally from one manufacturer to another, depending upon their product mix and quality of curing.

SF (Service factor) – as in Table 8.5.
Correction for arc of contact – as in Table 8.7.

In the following example we illustrate a brief procedure to select a flat belt for transmission of a load.

**Example 8.1**

Determine the width of a heavy-duty double-leather flat belt for transmitting a load of 18.5 kW at 1480 r.p.m. from a squirrel cage motor to be switched directly on line. Diameters of pulleys are 250 mm on the motor side and 200 mm on the load side. The centre distance between the pulley may be considered as 800 mm.

\[ P = 18.5 \, \text{kW} \]

\[ SF = 1.4 \text{ for heavy-duty Class 3, as shown in Table 8.5, presuming operations for 10 hours/day.} \]

Correction for arc of contact:

\[ \frac{D - d}{C} = \frac{250 - 200}{800} = 0.0625 \]  

(Figure 8.10)

\[ \therefore W = \frac{18.5 \times 1.4}{0.99} = 26.17 \, \text{kW} \]

The belt must be rated at least for this load.

Speed of the faster pulley = \[ \frac{1480 \times 250}{200} \]

= 1850 r.p.m. \hspace{1cm} (a)

or \[ \frac{\pi \times 200 \times 1850}{60 \times 1000} = 19.36 \, \text{m/s, which is quite low} \]  

(< 30 \, \text{m/s}) hence acceptable

Consider a belt width of 200 mm to transmit this load.

Therefore minimum belt rating per 25 mm = \[ \frac{25}{200} \times 26.17 \]

= 3.27 kW \hspace{1cm} (b)

For parameters (a) and (b) the possible belt sizes can be

1. Four-ply 28 soft duck or four-ply 31 hard duck, according to Table 8.1 Type I, having a rating of 3.25 kW at 1800 r.p.m. and 3.45 kW at 2000 r.p.m., or
2. Three-ply 32 soft duck or three-ply 34 hard duck, according to Table 8.1 Type II, having a rating of 3.32 kW at 1800 r.p.m.

For a more judicious selection of belts, keeping economics in mind, it is advisable to seek an opinion from the manufacturer.

### 8.5.2 Energy efficient and space saver belts

These are short centre drives unlike flat belt drives. The belt slip in such drives is negligible. They are an efficient means of power transfer and have added one more area of space saving and energy conservation in industrial applications. Most belts mentioned in Table 8.01 are energy efficient and can save space. But special compact space-saver belts are specially devised to save larger spaces. All these belts have been in practice for more than a decade now. One can use these belts at all new installations and also undertake retrofitting the old ones employing conventional flat or V belts. For one to make a better choice we briefly discuss below the main features of these belts.

#### 1. Wrapped V belts

(i) **Classical V belts**

These are frictional drives as noted before and are an improvisation over conventional V belts. The modern V belts using different product mix possess high power transfer capability with low slips. They are oil and heat resistant and have better traction, flex (bending ability) and low wear and tear. This is achieved by using polyester-cotton blended fabric jacket impregnated with polychloroprene synthetic rubber (Figure 8.10(a)). Their most important feature is low stretch and hence better stability. They do not call for frequent tensioning. This feature alone improves the power transmission efficiency of a multi-belt drive system substantially. Now the load is equally shared by all the belts unlike earlier V belts. It may be noted that in a multi-belt drive system, slackening, which may be due to under-bolting at the time of installation or unequal elongations of one or more belts during operation, would mean uneven sharing of load. Slackened belts sharing the least and remaining

---

**Table 8.2** Specification of flat belts

<table>
<thead>
<tr>
<th>Quality</th>
<th>Type of duck$^a$</th>
<th>Nominal weight (g/m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Soft</td>
<td>28</td>
<td>845</td>
</tr>
<tr>
<td></td>
<td>32</td>
<td>950</td>
</tr>
<tr>
<td>Hard</td>
<td>31</td>
<td>910</td>
</tr>
<tr>
<td></td>
<td>34</td>
<td>970</td>
</tr>
</tbody>
</table>

$^a$As in IS 5996

*Source:* IS 1370. See also ISO 22
Efficient transmission of loads

8

belts sharing the most, getting over-stressed, resulting in higher slippages and poor efficiency, besides getting over-heated and severely affecting their longevity. Table 8.2(a) illustrates the influence of slackening of a few belts on the longevity of the remaining belts.

In earlier V belts even same length belts from the same manufacturer and same batch may have some difference in characteristics, one being marginally longer or shorter than the other and during operation causing excessive tensioning to the shorter belts as shown in the table.

Modern V belts in one size remain the same irrespective of manufacturer, batch and period of manufacture. Therefore, matching of belts is no longer necessary in today’s high-tech manufacturing techniques for uniform loading and longevity of belts. As the belts retain their length, so is retained their power transfer capability and lower remains the slip, improving overall efficiency of the drive system. The procedure of selection of belts remains the same as for conventional V belts discussed in Section 8.5.3. For belt ratings one may consult the manufacturer. The normal cross-section of classical V belts in practice are given in Table 8.3.

(ii) Space saver V belts

By increasing the depth (T) of the classical belts as shown in (Table 8.3) the load transfer capacity of the belts can be raised and so can be reduced the number of belts and the space required for the same load transfer, making them yet more space savers. These belts can carry up to 200% and more power than the classical belts in the same sections. Pulley grooves are now required to be deeper than usual. Narrower pulleys reduce the over-hang on shaft and hence the torsion (bending moment), improving longevity of drive and driven bearings. For load charts one may consult the manufacturer.

2. Cogged V belts

(i) Classical-cogged belts

They have notches and slots as shown in Figure 8.10(b). These notches run perpendicular to the belt length. The slots relax the stiffness of the belt and provide it the desired flexibility. Now they require lesser power and can be used on smaller sheaves than conventional wrapped belts to save on the initial cost of pulleys, belts and subsequent spares. Having same dimensions as the classical V belts, cogged V belts too can be used on the same pulleys as for the classical V belts. They are however, not suitable for serpentine drive, multi-point drive or shaft rotation reversal drives as this can over-stretch the cogs and crack the belt. Following features however, make them energy efficient and an ideal choice for simple belt drives.

**Table 8.3** Nominal cross-sections of various types of V belts and their code numbers

<table>
<thead>
<tr>
<th>Wrapped V belts</th>
<th>Cogged or notched V belts</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Belt Section</strong></td>
<td><strong>Belt Section</strong></td>
</tr>
<tr>
<td>Classical V belts</td>
<td>Classical cogged V belts</td>
</tr>
<tr>
<td>Z</td>
<td>–</td>
</tr>
<tr>
<td>A</td>
<td>AX</td>
</tr>
<tr>
<td>B</td>
<td>BX</td>
</tr>
<tr>
<td>C</td>
<td>CX</td>
</tr>
<tr>
<td>D</td>
<td>–</td>
</tr>
<tr>
<td>E</td>
<td>–</td>
</tr>
</tbody>
</table>

Based on ISO 4184, IS 2494-1
– Being cogged on the pulley side the belt has a better grip with the pulley.
– Better grip and higher coefficient of friction reduces slippage to near negligible and greatly enhances operating efficiency.
– They are precision belts and do not call for frequent re-tensioning.
– Cogs on the inner surface of the belt increase air flow and facilitate cool running.
– Very high flexibility due to moulded cogs allows the use of smaller sized pulleys to save on the initial cost.
– High drive ratios up to 1:12 are easily possible avoiding multi-stage drives.
– Their flat side sitting snugly into the high sided-walled pulleys (Figure 8.10(b)) locks the belt in position and prevents it from turn-over or climbing the pulley walls. This feature also provides the belt better stability and grip.
– These belts therefore are a better choice for vibration damping and shock loadings.
– They have about 30% more power handling capability (per unit width) than the classical V belts.
– Energy saving is estimated to be up to 6.0% over the classical V belts.

The procedure for selection of belts remains the same as discussed in Section 8.5.3. For exact belt ratings one may consult the manufacturer.

(ii) Space saver cogged belts
The space-saver belts can also be cogged for better efficiency and power handling capability. They too have about 30% more power handling capability than the space-saver V belts. Other features remain the same as for classical-cogged V belts. For exact ratings of belts one may consult the manufacturer.

3. Multi-pull ribbed V belts
The individual V belts are now moulded together to turn them into a wider single belt of any size (Figure 8.10(c)). Figure 8.10(d) shows the view of a small drive using these belts. They eliminate using multiple set of belts unlike for classical V belts and ensure equal loading of all belts, a feature not practical in case of other belts (except synchronous belts) due to even the slightest variations in belt lengths however close the tolerances are. Shorter (tight) belts carrying higher loads than the slack (longer) belts as shown in Table 8.2(a). Multi-pull V belts being equally loaded have better longevity. The main features by and large remain the same as for classical V and cogged belts noted above in addition to the following,
– They provide a yet more compact drive
- Being very thin and highly flexible can employ small diameter pulleys saving on space and cost.
- They can be reverse bent round the pulley and used for serpentine drives similar to Figure 8.10(h). The flat side of the belt can also be used to drive another load.
- The higher belt widths facilitate maximum wedge contact and still better grip, negligible slip and silent running.
- They provide a better choice for vibration damping and shock loadings unlike synchronous belts. Shock loadings cause abrupt change in torque and a synchronous belt is unsuitable for such loads as shock loadings may shear-off their moulded teeth.
- The drives can be run at higher speeds (up to 40 m/sec or so). Normal speed for V belts is about 30 m/sec.
- The belts being in one piece cause no variation in length or tension in each V-section and have extended longevity.
- The above features enable transfer of power with relatively low belt tension enhancing longevity of bearings also.
- They have about 40% more power handling capability (per unit width) than the classical V belts.
- Energy saving is about 6–8% over classical V belts.

The procedure for selection of belts remains the same as discussed in Section 8.5.3. For belt sizes and ratings one may consult the manufacturer.

4. Synchronous Belts
(also known as timing belts or gear belt drives)

They are high torque drive belts, toothed and demand for precise mating between the drive and the driven sprocket wheels. Made from polychloroprene and fibre-glass cord tensile members, these belts possess excellent flex and resistance to elongation. Figure 8.10(e) shows the cross-sectional view of these belts. They are available from FHP to 500 kW and more and can be used for speeds up to 20 000 r.p.m. Presently these belts are produced in one or more of the following configurations (Figure 8.10(f)).

(i) One-sided synchronous belts
(ii) Double-sided symmetrical teeth synchronous belts, and
(iii) Double-sided staggered teeth synchronous belts

Figure 8.10(g) shows some views of synchronous belts and timing pulleys. Following features make them highly energy efficient

- These are positive belt drives and have no slip. Efficiency is therefore very high, of the order of 99% compared to about 93% of the other belts barring multi-pull belts discussed above.
- Efficiency remains unaltered even with wide variations in load demands, while the most V belts discussed above bar multi-pull belts may have sharp reduction in efficiency at higher loads (high torques) due to higher slippages.
- Synchronous belts are however not suitable for shock-loading applications as that may shear-off their moulded teeth.
- They possess high flexibility and allow the use of very small pulleys to reduce the initial cost. They can easily adapt to serpentine, multi-point or shaft rotation reversal drives as illustrated in Figure 8.10(h). To meet such requirements they are also manufactured in double-sided synchronous belts as shown in Figure 8.10(i).
- As there being no slip they call for low belt tension and facilitate high power transfer and longer bearing lives
- Low maintenance and long re-tensioning period also enhance their longevity

Note
They may make whirring noise not desirable at many installations. Where a silent operation is mandatory, one may select other energy efficient belts discussed above.

These belts provide the most ideal solution for loads demanding high starting torque but have no shock-loading feature. These are high precision, compact and positive drive belts and are total energy saver. However, they are a costly proposition and are usually employed for special drives only calling for very high precision such as for,
Common type configurations

Multiple shaft drive configuration

Rock and carriage drive configuration

**Figure 8.10(h) A few applications of synchronous belts (Source: Gates)**
Efficient transmission of loads

- printing presses
- paper mills
- packaging industries
- multi-drive process lines
- refineries
- textile and spinning mills etc.

Selection of belts

For sizes and ratings of belts one may consult the manufacturer. The procedure for selection of belts by and large remains the same as for conventional V belts discussed in Section 8.5.3. ISO-5295 also suggests the method of calculating the ratings of synchronous belts, the manufacturer providing the supporting data. For brevity we are not providing the sample calculations for these belts.

Retrofitting:

These belts too are a good device for retrofitting at old installations calling for high precision and total energy conservation and longevity. Retrofitting will call for change of pulleys and belts but will ensure precise and efficient operation and large energy savings. The payback period still can be very low of the order of a few weeks depending upon loading and operating hours.

8.5.3 Selection of V-Belts

The normal cross-sections of classical V belts in practice are given in Table 8.3. The cross-section depends upon the power to be transmitted and its speed. To select the appropriate section of the belt for the required transfer of load refer to Figure 8.11 provided by the manufacturer.

Figure 8.10(i) View of a double-sided synchronous belt (Source: Dura-Belt)

Figure 8.11 Selection of cross-section for V-belts

Note: For other types of belts consult the manufacturer.
Similar charts are available for caged and space-saver V-belts also. It is recommended that the pulleys be used with maximum possible diameters and distance between their centres be maintained as more than the diameter of the larger pulley.

- The load-transmitting capacities of a single V-belt, at 180° arc of contact, are provided by the belt manufacturer as standard selection data for the user for different areas of belt cross-sections and speed of the faster shaft. We have provided this data for a leading manufacturer, for belts of section \( D \), in Table 8.4, to illustrate the selection of conventional V-belts for the drive of Example 8.2.
- Figure 8.12 shows the arc of contact, i.e. the contact angle, the belt would make with the pulley.
- As for a motor a belt too is subject to unfavourable operating conditions that require deratings depending upon the working conditions, arc of contact and the length of the belt selected as noted below:

1. **Service factor (SF)**
   This will depend upon the type of drive, the torque requirement and operations in hour per day etc. The subsequent deratings are the same for all manufacturers and we provide these in Table 8.5.

2. **Correction for length of belt**
   The longer the belt, the larger the load it can transmit and vice versa. These factors are also the same for all manufacturers, as shown in Table 8.6.

3. **Correction for arc of contact**
   The standard ratings of belts are provided at 180° arc of contact. The smaller the diameter or the higher the speed of the smaller pulley, the smaller will be this angle and vice versa. The load-transmitting capacity of the belt diminishes with reduction of this angle and we show this factor in Table 8.7. A contact angle less than 120° would exert more centrifugal forces on the motor shaft and the driving-end (DE) bearing. If this is the case the shaft and the DE bearing, may need reinforcement, the provision of a jack shaft, or an additional support at the far end (DE) bearing. If this is the case the shaft and the DE bearing, may need reinforcement, the provision of a jack shaft, or an additional support at the far end (DE) bearing. Figures 8.15 and 8.16 show the arrangement of a jack shaft and an additional support at the far end of the shaft respectively. Such a situation must, however, be avoided, as far as possible.

   In Example 8.2 we illustrate a step-by-step procedure to select the most appropriate size and length of belts and pulley sizes to transmit a particular load.

**Example 8.2**
Consider a reciprocating compressor operating in a process plant and using a motor of 110 kW, 980 r.p.m. The compressor is required to operate at 825 r.p.m. through V-belts. We have provided this data for a leading manufacturer, for belts of section \( D \), in Table 8.4, to illustrate the selection of conventional V-belts for the drive of Example 8.2.

- The centre distance will now be less than 1 m and can be calculated by

\[
C = A + \sqrt{A^2 - B}
\]

where

\[
A = \frac{L}{4} - \frac{\pi}{8} (D + d)
\]

and

\[
B = \frac{(D - d)^2}{8}
\]

We can adopt the following procedure to select the recommended belt sizes:

1. **Determine the design power of transmission**
   Design power = motor rating \( \times \) SF
   Assuming the motor to be switched DOL and operating for an average 14 hours/day, then the SF according to Table 8.5 = 1.3

\[
\text{and design power} = 110 \times 1.3 = 143 \text{ kW}
\]

2. **Calculate the speed ratio of the faster shaft to the slower shaft**

\[
\frac{980}{825} = 1.19
\]

This is the same as

\[
\text{Pitch dia. of larger pulley} \div \text{Pitch dia. of smaller pulley}
\]

3. **Select the belt cross-section**
   Use the selection curves provided by the manufacturer (which are almost the same for all manufacturers) (see Figure 8.11). For a design power of 143 kW to be transmitted at 980 r.p.m., the recommended cross-section of belts according to these curves is identified as \( D \).

4. **Select the pulley diameter**
   Refer to the manufacturer’s catalogue for the recommended pulley pitch and outside diameters. These data are provided in Table 8.8 for a leading manufacturer.

   Consider small pulley of \( d = 400 \text{ mm} \)

   Then the larger pulley \( D = 400 \times 1.19 = 476 \text{ mm} \)

   The nearest standard size available is 475 mm which is acceptable.

5. **Determine the belt pitch length** (see also BS 3790)
   The pitch length of a belt is its circumferential length at the pitch width of the belt. The pitch width of a belt is shown in Figure 8.12, and is provided by manufacturers as standard data. It can be calculated by

\[
L = 2C + 1.57(D + d) + \frac{(D - d)^2}{4C} \tag{8.2}
\]

where

- \( L \) = belt pitch length (mm)
- \( C \) = centre distance between the two pulleys (mm).
- \( D \) = pitch diameter of the larger pulley (mm)
- \( d \) = pitch diameter of the smaller (faster) pulley (mm)

\[
\therefore \quad L = 2 \times 1000 + 1.57(475 + 400) + \frac{(475 - 400)^2}{4 \times 1000}
\]

\[
= 2000 + 1373.75 + 1.406
\]

\[
= 3375.16 \text{ mm.}
\]

We show the nominal inside length for various standard sections of belts in Table 8.9. According to the table the nearest standard belt available has a pitch length of 3330 or 3736 mm. For a more closer length of belt, either change the pulley size or select another make of belt or have it made to order. But a length of 3330 seems reasonable, hence it is accepted.

\[
\therefore \quad L = 3330 \text{ mm}
\]

The centre distance will now be less than 1 m and can be calculated by

\[
C = A + \sqrt{A^2 - B}
\]

where

\[
A = \frac{L}{4} - \frac{\pi}{8} (D + d)
\]

and

\[
B = \frac{(D - d)^2}{8}
\]
Table 8.4 Power ratings for section D V-belts, with 180° arc of contact with the smaller pulley

<table>
<thead>
<tr>
<th>Speed of faster shaft (r.p.m.)</th>
<th>Power rating* for smaller pulley, with preferred pitch diameter of</th>
<th>Additional power per belt, for speed ratio of</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>mm</td>
<td>kW</td>
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<tr>
<td>1600</td>
<td></td>
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</tr>
</tbody>
</table>

*These ratings are for conventional V-belts. For classical V-belts consult the manufacturer.
### Table 8.5  Service Factors for flat and V-belts

<table>
<thead>
<tr>
<th>Types of loads</th>
<th>Types of driving units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Class</td>
<td>Examples</td>
</tr>
<tr>
<td></td>
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<td></td>
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<tr>
<td></td>
<td></td>
</tr>
<tr>
<td>Class 1</td>
<td>(Light duty)</td>
</tr>
<tr>
<td></td>
<td>Agitators (uniform density)</td>
</tr>
<tr>
<td></td>
<td>Blowers, exhausts and fans (up to 75 kW)</td>
</tr>
<tr>
<td></td>
<td>Centrifugal compressors and pumps</td>
</tr>
<tr>
<td></td>
<td>Belt conveyors (uniformly loaded)</td>
</tr>
<tr>
<td>Class 2</td>
<td>(Medium duty)</td>
</tr>
<tr>
<td></td>
<td>Agitators and mixers (variable density)</td>
</tr>
<tr>
<td></td>
<td>Blowers, exhausts and fans (over 75 kW)</td>
</tr>
<tr>
<td></td>
<td>Rotary compressors and pumps (other than centrifugal)</td>
</tr>
<tr>
<td></td>
<td>Belt conveyors (not uniformly loaded)</td>
</tr>
<tr>
<td></td>
<td>Generators and exciters</td>
</tr>
<tr>
<td></td>
<td>Laundry machinery</td>
</tr>
<tr>
<td></td>
<td>Line shafts</td>
</tr>
<tr>
<td></td>
<td>Machine tools</td>
</tr>
<tr>
<td></td>
<td>Printing machinery</td>
</tr>
<tr>
<td></td>
<td>Sawmill and woodworking machinery</td>
</tr>
<tr>
<td></td>
<td>Screens (rotary)</td>
</tr>
<tr>
<td>Class 3</td>
<td>(Heavy duty)</td>
</tr>
<tr>
<td></td>
<td>Brick machinery</td>
</tr>
<tr>
<td></td>
<td>Bucket elevators</td>
</tr>
<tr>
<td></td>
<td>Compressors and pumps (reciprocating)</td>
</tr>
<tr>
<td></td>
<td>Conveyors (heavy duty)</td>
</tr>
<tr>
<td></td>
<td>Hoists</td>
</tr>
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<td></td>
<td>Mills (hammer)</td>
</tr>
<tr>
<td></td>
<td>Pulverizers</td>
</tr>
<tr>
<td></td>
<td>Punches, presses, shears</td>
</tr>
<tr>
<td></td>
<td>Quarry plant</td>
</tr>
<tr>
<td></td>
<td>Rubber machinery</td>
</tr>
<tr>
<td></td>
<td>Screens (vibrating)</td>
</tr>
<tr>
<td></td>
<td>Textile machinery</td>
</tr>
<tr>
<td>Class 4</td>
<td>(Extra heavy duty)</td>
</tr>
<tr>
<td>Crushers</td>
<td></td>
</tr>
</tbody>
</table>

For speed-increasing drives of speed ratio 1.00 to 1.24: multiply service factor by 1.00
speed ratio 1.25 to 1.74: multiply service factor by 1.05
speed ratio 1.75 to 2.49: multiply service factor by 1.11
speed ratio 2.50 to 3.49: multiply service factor by 1.18
speed ratio 3.50 and over: multiply service factor by 1.25

**Note**
1 The service factors do not apply to light duty drives.
2 The use of a jockey (idler) pulley on the outside of the belt is not recommended.

**Special conditions**
For reversing drives, except where high torque is not present on starting, add 20% to these factors.
Corresponding to 1000 r.p.m. and \( d = 400 \text{ mm} \).

Power rating of belt having section \( D = 23.88 \text{ kW/belt} \), with an additional power-transmitting capacity for a speed ratio of \( 1.19 = 1.74 \text{ kW/belt} \).

\[ \text{Total load-transmitting capacity} = 25.62 \text{ kW/belt} \]

7 Determine correction factors

(i) For the arc of contact shown in Table 8.7, which is the same for all manufacturers. Corresponding to an arc of contact of \( \frac{D-d}{C} \) on the smaller pulley

\[ \text{i.e. for } \frac{475 - 400}{977.40} \text{ or } 0.077 \]

The correction factor is not less than 0.99.

(ii) Determine the belt pitch length correction factor from (Table 8.6) which is same for all manufacturers.

Corresponding to \( L = 3330 \text{ mm} \) for the belt of section \( D \), it is 0.87.

8 Determine the number of belts for the total load to be transmitted.

Corrected power capacity of each belt of section \( D = 25.62 \times 0.99 \times 0.87 \)

\[ = 22.06 \text{ kW} \]

\[ \therefore \text{Number of belts required} = \frac{143}{22.06} \]

\[ = 6.48 \]

or 7 belts

9 Counter-check the selection for the speed of the smaller pulley.

\[ \text{Speed} = \pi \cdot d \cdot N_f \cdot \text{mm/min} \]

where \( N_f = \text{speed of the faster shaft} = 980 \text{ r.p.m.} \)
\[ d = \text{pitch dia. of the faster (smaller) pulley} = 400 \text{ mm.} \]

\[ \therefore \text{Speed of faster pulley} = \frac{\pi \times 400 \times 980}{60 \times 1000} \]

\[ = 20.51 \text{ m/s} \]

which is less than 30 m/s, hence acceptable. Had it not been so, advice from the manufacturer on the selection of belts would have been necessary.

Thus the specification of the belts and the pulleys will be

- **Belts**
  - cross-section \( D \)
  - pitch length \( L \) \( = 3330 \text{ mm} \)
  - number of belts \( = 7 \)

**Note**

1. To avoid uneven distribution of load on the belts when more than one belt is used the pitch length of the belts must be identical. For this purpose, it is advisable to use all belts of one make only. As standard practice all belts are marked on their surfaces with their length and permissible tolerance for easy identification.

2. With the improvised product mix, latest manufacturing practices and close tolerances maintained by the different manufacturers it is now possible to maintain the same dimensions for the same section and size of belts manufactured by different manufacturers. One can even buy replacement belts of a particular size and

---

**Figure 8.12** Determining the belt details in a V-belt drive

(These formulae are available in the product catalogues of all leading manufacturers.)

\[ A = \frac{3330}{4} - \frac{\pi}{8} (475 + 400) \]

\[ = 832.5 - 343.44 \]

\[ = 489.06 \]

and \[ B = \frac{(475 - 400)^2}{8} \]

\[ = 703.12 \]

\[ C = 489.06 + \sqrt{489.06^2 - 703.12} \]

\[ = 489.06 + 488.34 \]

\[ = 977.40 \text{ mm}, \text{which is acceptable} \]

**6 Determine the power rating** per belt from the technical data provided by the manufacturer. This will vary from one manufacturer to another, depending upon the quality of rubber used. We have considered this data, as shown in Table 8.4, of a leading manufacturer.
length of any manufacturer without matching the length as discussed in Section 8.5.2.

- **Pulleys**
  (a) Smaller pulley on motor shaft
  Pitch dia. = 400 mm
  Number of grooves suitable for belts of section $D = 7$
  (b) Larger pulley on the compressor shaft
  Pitch dia. = 475 mm
  Number of grooves suitable for belts of section $D = 7$

**Note**
The above example is just for illustration for a conventional V belt. The procedure for selection of state-of-the-art energy efficient belts shall remain the same with the data substituted for the type of belt selected.

### 8.5.4 Energy conservation

Where there is a motor there is a belt drive, except where rigid or flexible couplings or gear-systems are employed. Belts and pulleys are therefore an essential feature of an industry to transfer power from driving machine to the driven machine day in and day out round the year. Although belts constitute only a fractional cost compared to the main plant and machinery, they are necessarily the only means of power transfer from motor to the load, barring loads driven by other means than pulleys. It is irrespective of the type of switching adapted or energy saving means (static drives or fluid couplings) incorporated in the system.
One can thus visualise the enormous amount of power being transferred through belts round the clock. A small energy saving during transfer of mechanical power can facilitate enormous savings in terms of energy saved and running cost. It is therefore imperative that one opts for the best belt drive system of the various state-of-the-art belts discussed above.

**Retrofitting:**
Retrofitting of old installations with energy efficient belts is an essential requirement in the present scenario having maximum impetus on energy and space saving. Also providing longevity to the entire mechanical system associated with the drive. Since all energy efficient belts are compact and require compact pulleys, space is not a limiting factor while retrofitting. The payback period is extremely short, of the order of a few days or weeks only, depending upon loading and operating hours, while the recurring savings are enormous. Selection of type of belt would depend on

- type of starting – soft start or heavy duty
- type of loading – constant load, variable load or shock loading

From the various features of different types of belts discussed already (except synchronous belts – that are special duty for precision applications) we can derive the following recommendations while selecting the belts, the user himself being the best judge in consultation with the manufacturer.

- For small loads one can be content with classical V or retrofit with cogged V belts which will fit into the same pulleys. Since they are capable of transferring about 30% more power they will now run under-

---

**Table 8.7**  Arc of contact correction factors [flat and all types of V-belts]

<table>
<thead>
<tr>
<th>$\frac{D - d}{C}$</th>
<th>Arc of contact on smaller pulley (degrees)</th>
<th>Correction factor, i.e. proportion of $180^\circ$ rating</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00</td>
<td>180</td>
<td>1.00</td>
</tr>
<tr>
<td>0.05</td>
<td>177</td>
<td>0.99</td>
</tr>
<tr>
<td>0.10</td>
<td>174</td>
<td>0.99</td>
</tr>
<tr>
<td>0.15</td>
<td>171</td>
<td>0.98</td>
</tr>
<tr>
<td>0.20</td>
<td>169</td>
<td>0.97</td>
</tr>
<tr>
<td>0.25</td>
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<td>0.97</td>
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<tr>
<td>0.30</td>
<td>163</td>
<td>0.96</td>
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<tr>
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<td>160</td>
<td>0.95</td>
</tr>
<tr>
<td>0.40</td>
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<tr>
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</tr>
<tr>
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<td>151</td>
<td>0.93</td>
</tr>
<tr>
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<tr>
<td>0.60</td>
<td>145</td>
<td>0.91</td>
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<tr>
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<tr>
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<tr>
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<td>123</td>
<td>0.83</td>
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<tr>
<td>1.00</td>
<td>120</td>
<td>0.82</td>
</tr>
</tbody>
</table>

**Note**

Arc of contact with smaller pulley = $180^\circ \times \frac{57.3(D - d)}{C}$

where

- $C$ is the centre distance (mm)
- $D$ is the pitch diameter of larger pulley (mm)
- $d$ is the pitch diameter of smaller pulley (mm)

---

**Table 8.8**  Recommended standard pulley and outside diameters

<table>
<thead>
<tr>
<th>Pitch diametera (mm)</th>
<th>Outside diameter</th>
</tr>
</thead>
<tbody>
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<tr>
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</tbody>
</table>

**Note**

For space saver and cogged belts consult the manufacturer.

*The limits of tolerances on the pitch diameter will be ± 0.8%.

**Courtesy:** Goodyear. See also IS 3142 and ISO 4183
Table 8.9 Nominal inside lengths, pitch lengths $L$ for all standard sizes of multiple V-belts

<table>
<thead>
<tr>
<th>Nominal inside length</th>
<th>Cross-section A</th>
<th>Cross-section B</th>
<th>Cross-section C</th>
<th>Cross-section D</th>
<th>Cross-section E</th>
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</table>

_Courtesy:_ Goodyear

_Note:_

1. All dimensions are in millimetres.
2. Because of variations in pitch lengths in space saver and cogged belts their inside lengths may also vary slightly. For details consult the manufacturer.
loaded and have about three times longevity than the classical V belts due to the same size of belts.

- Space-saver cogged and multi-pull belts are ideal for medium and large loads. One can choose one of them depending on size of load. Both belts being ideal for variable loads and shock loadings. These belts would require change of pulleys but are recommended in view of enormous energy and recurring cost saving besides space.
- Since these belts are flexible – small diameter and width pulleys can be used saving on over-all cost of replacement,
  - Where reverse or serpentine drives are required multi-pull belts are most ideal.
  - All these belts provide efficient power transmission, cause lesser strain on mechanical system associated with the drive, less wear and tear and lower maintenance. Consequently enhance longevity of the entire mechanical system besides the belts.

Belt drives and the associated pulleys are relatively very low cost items in the whole system and are easily available. One may therefore endeavour for a total retrofitting of an old installation for large energy savings besides savings on maintenance and replacements. The cost of retrofitting is indeed little compared to large savings.

Calculations for energy saving:

Using the same Equation (1.14) as for motors, energy saving per year can be determined as,

\[ = \text{kW} \cdot \frac{1}{\eta_1 - \frac{1}{\eta_2}} \cdot x \cdot y \cdot z \]

(1.14)

where,
- Load transmitted = kW
- Efficiency of conventional V belts = \( \eta_1\% \)
- Efficiency of energy saver belts = \( \eta_2\% \)
- Operating hours/day = \( x \)
- Working days/year = \( y \)
- Local tariff per kWh = \( z \)

8.5.5 Installation and maintenance

Belt tensioning

For an efficient power transmission it is essential to tighten the belts adequately while installing, and maintaining the same tensioning during running by periodic check-ups. Low tensioned belts than recommended transmit less power, so also high tensioned belts, besides subjecting the high tensioned belts, pulleys, shafts and bearings at drive and driven ends to high torsion and consequential excessive heating of bearings, belts and bending moment on shafts. All this may result in premature failure of bearings and belts. It is therefore essential to tighten the belts to the recommended tensioning force. The established belt tensioning forces are provided in Table 8.9(a) for classical V, space-saver V and multi-pull ribbed belts. For cogged belts the force may be raised by roughly

<table>
<thead>
<tr>
<th>Belt</th>
<th>Force required to deflect belt by 16 mm per metre of span</th>
</tr>
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<tbody>
<tr>
<td></td>
<td>Small pulley diameter (mm)</td>
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<td>Classical V belts</td>
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<tr>
<td>A</td>
<td>80 to 140</td>
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<tr>
<td>B</td>
<td>125 to 200</td>
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<tr>
<td>C</td>
<td>200 to 400</td>
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<td>D</td>
<td>355 to 600</td>
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<td>E</td>
<td>500 &amp; above</td>
</tr>
</tbody>
</table>

Note

Belt tensioning remains the same for cogged belts also.

| Space saver V belts | | |
| SPZ | 67 to 95 | 10 to 15 | 1.0 to 1.5 |
| | 100 to 140 | 15 to 20 | 1.5 to 2.0 |
| SPA | 100 to 132 | 20 to 27 | 2.0 to 2.7 |
| | 140 to 200 | 27 to 35 | 2.8 to 3.6 |
| SPB | 160 to 224 | 35 to 50 | 3.6 to 5.1 |
| | 236 to 315 | 50 to 65 | 5.1 to 6.6 |
| SPC | 224 to 355 | 60 to 90 | 6.1 to 9.2 |
| | 375 to 560 | 90 to 120 | 9.2 to 12.2 |

Note

Belt tensioning remains the same for cogged belts also.

| Multi-pull ribbed V belts | | |
| PJ | Below 45 | 1.6 to 3.0 | 0.16 to 0.30 |
| | 45–66 | 3.0 to 5.0 | 0.30 to 0.50 |
| | 67–125 | 4.0 to 7.0 | 0.40 to 0.70 |
| PL | Below 160 | 10 to 15 | 1.0 to 1.5 |
| | 160–224 | 12 to 20 | 1.2 to 2.0 |
| PM | Below 355 | 30 to 45 | 3.0 to 4.5 |
| | 355–560 | 35 to 60 | 3.5 to 6.0 |

Courtesy: Fenner

30% over the wrapped belts. For exact loading consult the manufacturer.

Procedure of belt tensioning

(i) Avoid use of crowbars or screwdrivers while placing belts on the pulleys. Leave enough gap between the drive and driven pulleys to place and tighten belts. After placing the belts move away the motor pulley (usually) from driven pulley until the belts acquire the required tension.

(ii) Belt tension indicators are available to check the belt tension after installation. It measures tension in terms of belt deflection as shown in Figure 8.12(a). When the indicator is placed at right angles at the centre of the belt span and pressed to deflect the belt it records the belt tension. The indicator is set to read the required force as per Table 8.9(a) for a deflection of 16 mm per metre span. To achieve the recommended belt tension the belts are so adjusted that they read the required force at a deflection of 16 mm/m. For example for a span of 2 m the belts are so adjusted (by adjusting the drive or driven equipment whichever is more convenient) so as to
measure the required force for 2 m span for a deflection of 32 mm.

In case of multi-pull belts, sometimes it may be difficult to measure the deflection by this method as the force required may be large to be applied manually. In that case elongation method can be adapted as shown in Table 8.9(b).

After the belts are adjusted, the drive may be run for some time, say half an hour to one hour or so depending upon the type of load to allow the belts to seat in grooves properly and also adjust their elongation (if any). The belt tension be checked again and adjusted as necessary. The belts may be tensioned periodically for most efficient power transfer for all times. For this purpose a logbook can be maintained. With the passage of time the maintenance staff can arrive at the most appropriate service interval for maintenance. The period of maintenance would depend upon the type of load and period of running.

### 8.6 Checking the suitability of bearings

#### 8.6.1 Forces acting on the DE motor bearing

When transmitting the load to the driven equipment, the motor bearing at the driving end (DE) is normally subject to two types of forces, radial and axial. The axial force in a horizontal drive is normally zero. If it is not zero this may be due to eccentricity in the transmitting line or any such reason that may subject the driving end bearing to an axial load in addition to a radial load. These forces become severe when the load from the motor is being transmitted through belts. In such cases it is of utmost importance to check the suitability of the bearing and the motor shaft to transmit the required load under such conditions.

We provide below a few guidelines for checking the suitability of the bearing and the motor shaft, when selecting a motor for these drives:

- **Radial forces**
  The radial forces acting on the bearing can be determined by the belt pull and is expressed by

  \[
  P_t = \frac{K \cdot 973 \cdot kW}{N_r \cdot \frac{D}{2}} \pm W \text{kg} \quad (8.3)
  \]

  where
  \[
  K = \text{belt factor, 2 to 2.5 for V-belts and 2.5 to 3 for flat belts. Sometimes it may be higher (up to 4 to 5). (The higher values must be considered when the distance between shafts is short or belt tension is high).}
  \]
  \[
  D = \text{diameter of pulley on the motor (m)}
  \]
  \[
  l = \text{half of the pulley width (load distance) (m)}
  \]
  \[
  W = \text{weight of complete rotor and driven masses on the motor shaft, such as the pulley and the belts (kg)}
  \]

  ‘+’ applicable when the belt pull is downwards [vertical drives, Figure 8.13(a)] and

  ‘−’ applicable when the belt pull is upwards [vertical drives, Figure 8.13(b)].

  ‘0’ when the shaft is horizontal or the drive and the driven equipment are in the same horizontal plane and in line with each other.

  The allowable pulley loads for standard motors are given in Table 8.10. These are recommended by a few manufacturers for their motors, but they are generally true for other motors also.

- **Bending moment**
  The bending moment at the weakest section of the shaft, i.e. at the collar of the DE shaft (Figure 8.14).
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\[ B.M. = P_t \cdot l = \frac{K \cdot 973 \cdot kW}{N_t \cdot \frac{D}{2}} \cdot l \text{ mkg} \]  (8.4)

- **Axial forces or thrust load**

When these exist they can be expressed by

\[ P_a = \text{axial force of pump, turbine or any other load (kg)} \]

\[ P_a = P_a \text{ (for vertical motors)} = \text{axial force as above + weight of complete rotor and driven masses on the motor shaft such as the pulley and the belts (kg)} \]

**Note**

Cylindrical roller bearings should normally not be used for such applications.

- When a bearing is subject to both radial and axial thrusts the equivalent dynamic bearing loading can be expressed by

\[ P = X \cdot F_r + Y \cdot F_a \]

where

\[ X = \text{radial load factor} \]
\[ Y = \text{axial load factor} \]

The values of these factors are provided by the bearing manufacturers in their catalogues, based on the ratios \( F_a / C_o \) and \( F_r / F_t \), where \( C_o \) is the static load rating in kg or N (based on a contact stress of 4200 MPa for ball bearings and 4000 MPa for roller bearings, also provided in these catalogues. MPa is the unit of stress in Mega-Pascals).

**Note**

To assess the total force a bearing is likely to encounter more precisely, reference may be made to the bearing manufacturers' product catalogues. In severe load conditions, with an excessive bending moment on the shaft, the shaft stiffness and its suitability should also be checked. As shown in Table 8.10, motors for normal industrial use employ bearings with almost 20 000 hours as safe running life. For more stringent duties or continuous drives as in the pulp and paper industry, cement plant, refinery and petrochemical projects, the chemical industry and powerhouses which are 24-hour services, special bearings must be used with a safe running life of 50 000 to 100 000 hours.

### 8.6.2 Load-carrying capacity and life expectancy of bearings

The life of bearing would depend upon various factors such as:

- Type of drive
- Method of load transmission
- Accuracy of alignment
- Environment of installation
- The amount of radial and axial forces acting on the bearings or
- Any other factors discussed earlier.

No rotating mass can be balanced up to 0 micron, for obvious reasons. While rotating they are out of balance,
which gives rise to rotating forces and adds to the radial load on the bearing. This indirectly affects the running life of the bearing, in addition to other factors noted above. The selection of type and size of bearing, is thus governed by the speed of the machine, the type and weight of loads. The required life of the bearing can be expressed by

$$L_b = \frac{10^6}{60 \cdot N_t} \left( \frac{C}{P} \right)^p \text{ hours}$$

(8.5)

where

- $L_b$ = Rated life of bearings at 90% reliability, i.e. $90\%$ of bearings produced by a manufacturer will exceed this life
- $C$ = basic dynamic load rating in kg or $N$ (provided by the bearing manufacturer). It is the load which will give a life of 1 million revolutions
- $P$ = equivalent dynamic bearing load in kg or $N$
- $p$ = exponent of the life equation, which depends upon the type of contact between the races and the rolling elements. It is recommended as $3$ for ball bearings and $10/3$ for roller bearings
- $N_t$ = speed of the machine in r.p.m.

With this equation the life of the bearing can be determined for different load conditions and is predetermined for the type of drive and service requirements. To select a proper bearing, therefore, the type of application and the loading ratio ($C/P$) should be carefully selected to ensure the required minimum life. Bearing manufacturers’ product catalogues provide the working life of bearings for different load factors and may be referred to for data on $C$, $C_o$ and other parameters.

### 8.7 Suitability of rotors for pulley drives

In belt drives particularly, it is advisable that reference be made to the motor manufacturer to determine the suitability of the rotor shaft and the driving end (DE) bearing for transmission of the required load. A typical format of a questionnaire is also given below for providing the manufacturer for load, belt and pulley data. The format is suitable for all drives that may be subjected to excessive forces on the shaft. These data will enable the manufacturer to determine the following parameters to check the suitability of bearing and shaft strength and make suitable changes if warranted:

- Load acting on the motor bearings
- Bending moment at the motor shaft due to pulley and load
- Possible deflection of the shaft

**Note**

The shaft deflection should not be more than $11\%$ of the air gap between the stator and the rotor. For loads that exert more force and torsional stress on the motor shaft and bearings than is permissible, due to the larger width of pulleys which may shift the load farther away from the shaft collar, it is recommended that the pulley be mounted on a separate jack shaft, supported on two pedestals as illustrated in Figure 8.15. Provided that the motor shaft can be made longer, to support such a pulley and have sufficient strength, to take that load, one pedestal may also be adequate to support the free end of the motor shaft as shown in Figure 8.16. The pulley is now mounted between the shaft collar and the pedestal. A jack shaft or additional pedestal to support the motor shaft may also be necessary when the ratio of pulley diameters exceeds 6:1.

In some cases, reinforcement of the shaft by increasing the shaft diameter, employing a better grade of steel and using a superior grade of bearings, may also meet the load requirement. The bearing bore, however, may pose a limitation in increasing the shaft diameter beyond a certain point, say, beyond the diameter of the shaft collar. If a larger shaft diameter is required either a larger frame size of the motor may be employed, which may be uneconomical, or a jack shaft or pedestal may be used as noted above. Replacing standard bearings with larger bore bearings to use a shaft of greater diameter may not be possible in the same frame, due to pre-sized end shields and bearing housings which, for motors up to 250 kW, are normally cast and have fixed dimensions/moulding patterns.

**Questionnaire to determine the suitability of the motor shaft and the bearing for the required belt drive**

For critical loads and belt drives particularly, the user is advised to seek the opinion of the motor manufacturer to determine the mechanical suitability of the motor selected, its shaft and the DE bearing for the load to be transmitted, according to the drive system being adopted. The following are some important parameters that may help to determine the required suitability and should be sent to the manufacturer for their opinion:

![Figure 8.15 Arrangement of a jack shaft](image)

![Figure 8.16 Arrangement of a long shaft](image)
1. (a) Type of driven equipment
   (b) Type of bearings provided in the driven equipment
   (c) Whether any thrust or radial load is falling on the motor bearings from the driven equipment.

2. Details of the belt drive (Figures 8.17 and 8.18)
   (a) Type of belt: flat or V-belts and their width or numbers
   (b) Diameter of pulley on motor shaft \( D_1 \) (mm)
   (c) Width of pulley on motor shaft \( W_1 \) (mm)
   (d) Diameter of pulley on load side \( D_2 \) (mm)
   (e) Width of pulley on load side \( W_2 \) (mm)
   (f) Does the centre of the hub coincide with the centre of the rim of the pulley? If not, what is the eccentricity, \( E \), in mm?
   (g) Distance between centres of pulleys, \( C \) (mm)
   (h) Weight of pulley on motor shaft (kg)
   (i) Magnitude of pulley force \( P \) (kgf or N)
   (j) Inclination \( \beta \) of the pulley system to the horizontal plane as shown in Figure 8.18.

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**Relevant Standards**

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<th>IEC</th>
<th>Title</th>
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<td>Specification for cotton belting ducks.</td>
<td>5996/2003</td>
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<td>Belt drives – V-belts and V-ribbed belts – calculation of power ratings.</td>
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<td>Synchronous belts – calculation of power rating and drive centre distance.</td>
<td>–</td>
<td>–</td>
<td>5295/1987</td>
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</tbody>
</table>

**Notes**

1. In the table of relevant Standards while the latest editions of the Standards are provided, it is possible that revised editions have become available or some of them are even withdrawn. With the advances in technology and/or its application, the upgrading of Standards is a continuous process by different Standards organizations. It is therefore advisable that for more authentic references, one may consult the relevant organizations for the latest version of a Standard.

2. Some of the BS or IS Standards mentioned against IEC may not be identical.

3. The year noted against each Standard may also refer to the year it was last reaffirmed and not necessarily the year of publication.
List of formulae used

Selection of flat belts

\[ W = P \cdot \frac{SF}{\text{Correction for arc of contact}} \]  \hspace{1cm} (8.1)

- \( P \) = load to be transmitted (kW)
- \( W \) = maximum load transmitting capacity of the belt
- \( SF \) = service factor

To determine the pitch length of V-belts

\[ L = 2C + 1.57(D + d) + \frac{(D - d)^2}{4C} \]  \hspace{1cm} (8.2)

- \( L \) = belt pitch length (mm)
- \( C \) = centre distance between the two pulleys (mm)
- \( D \) = pitch diameter of the larger pulley (mm)
- \( d \) = pitch diameter of the smaller (faster) pulley (mm)

Radial forces on DE bearings

\[ P_r = \frac{K \cdot 973 \cdot \text{kW}}{N_t \cdot \frac{D}{2}} \pm W \text{ kg} \]  \hspace{1cm} (8.3)

Bending moment at the shaft

\[ P_{t, \ell} = \frac{K \cdot 973 \cdot \text{kW}}{N_t \cdot \frac{D}{2}} \cdot l \text{ mkg} \]  \hspace{1cm} (8.4)

- \( K \) = belt factor
- \( D \) = diameter of pulley (m)
- \( W \) = Weight of complete pulley and driven masses on the motor shaft.
- \( l \) = half of the pulley width (m)

Life of bearings

\[ L_b = \frac{10^6}{60 \cdot N_t} \left( \frac{C}{P} \right)^p \text{ hours} \]  \hspace{1cm} (8.5)

- \( L_b \) = normal life of bearing in working hours
- \( C \) = basic dynamic load rating (kg or N)
- \( P \) = equivalent dynamic bearing load (kg or N)
- \( p \) = exponent of the life equation
- \( N_t \) = speed of the machine (r.p.m.)