HMIC
INSTITUTE OF TECHNOLOGY

INDUSTRIAL MAINTENANCE

MECHANICAL SYSTEMS

FOREWORD

Mechanical systems provide the most fundamental platform for all heavy industrial plant and is the absolute requirement for all systems requiring movement today.

MIC has produced this book for use in its Industrial Maintenance Mechanical Systems Evening Programme and it is specifically designed to introduce the basics of mechanical fitting.

This book is intended for use as a course text to be supplemented by notes and explanations and does not stand alone.

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THIS BOOK IS NOT FOR SALE

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INTORDUCTION

The primary purpose of mechanical fitting is to transmit forces across parts of a system with as little loss as possible and with minimum of wear. The better the fits the more efficient the system. The primary units required to be fitted are gears, clutches, couplings, belt and chain drives and bearings.

To produce these forces there are four main units: pumps, compressors, turbines and levers. The major aspects of these devices will be discussed in relation to proper maintenance procedures, fault-finding methods and fitting techniques. The information given can be applied in almost every instance of maintenance fitting and will provide a spring board for acquiring more advanced techniques and knowledge in the areas outlined. Where specific areas have not been covered the methods and information given can be interpolated to fit the circumstances at the time.

MAINTENANCE AND SAFETY

1. Ensure all safety devices are operational; test for faults

2. Keep all guards in place

3. KEEP IT CLEAN at all times during and after maintenance

4. Ensure all instruments are accurate; test regularly

5. Bench lights must be adequate and functional

6. Maintain the integrity of all tools

7. Ensure proper and correct lubrication at all times

8. Use the correct filters (size and type) and ensure that they are properly installed

9. Ensure all locks and fixtures are functional

10. Always use the correct tool for the job

11. Monitor wear and tear on all moving parts e.g. chains, belts bearings, shafts etc.

12. Ensure all connections are tight, clean and correctly identified where applicable

13. Ensure all switches operate correctly

14. Ensure there are no conductive particles where there are electrical conductors

15. Check that all indicators are working

16. Check for damaged insulation material

17. Ensure that the equipment is protected against hostile environments

18. Always be a TEAM MEMBER; one man bands are prone to error

19. Always use the correct and prescribed protection devices

20. Use logical priorities not emotion; e.g., (1) safety, (2) schedule (3) age, (4) opportunity

21. NEVER wear jewellery or long open hair

BRAKES
&
CLUTCHES

BRAKES AND CLUTCHES

1. Operation

Brakes resist torque; clutches transmit it.

Brakes are used in machinery to create tension, to limit or reduce the speed of rotating parts, or to hold them stopped. Clutches transmit a driving force to make a machine move. Most brakes and clutches handle rotational or twisting force, called torque.


Figure 6: Ratchet
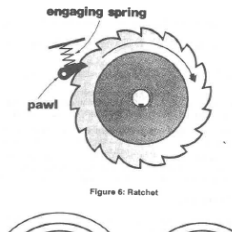
Brakes

Many common brakes resist torque by friction. A rotating part is pushed against a stationary part, and there is rubbing or sliding friction between them. If the rotating part stops, static friction holds it stopped.

A brake with high friction can stop a big load fast, and hold it stopped firmly. It can provide more tension in a winding machine.

Sliding friction produces heat.

In most brakes, the mechanical energy of the rotating part is converted into heat energy by friction. The faster the rotating part turns, and the greater the torque the brake is resisting the quicker the brake will heat up.


fig.2
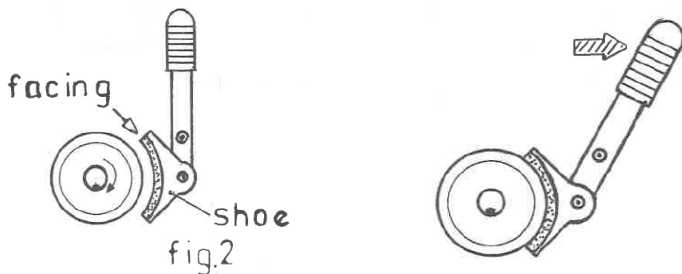
Clutches

Clutches connect or disconnect input and output.

The input and output of a clutch can be a shaft, pulley, sheave, sprocket, gear, or any other rotating part. In many common clutches, input and output parts are squeezed together; friction transmits torque and the output moves.

If input and output parts are pressed together hard enough so that they rotate at the same speed without slipping, the clutch is fully engaged, or locked up. Torque is transmitted by static friction.

If the output is being pressed against the input, but is not rotating at the same speed, the clutch is slipping. Sliding friction is transmitting the torque in this case. Like a brake, a slipping clutch will heat up from the friction that converts rotating power into heat.

There are many ways that brakes resist torque, and many ways clutches transmit it. There are also a variety of ways that brakes and clutches can be operated. Effective maintenance and troubleshooting require familiarity with how specific units work and what they are used for.

Figure 3: Friction Clutch
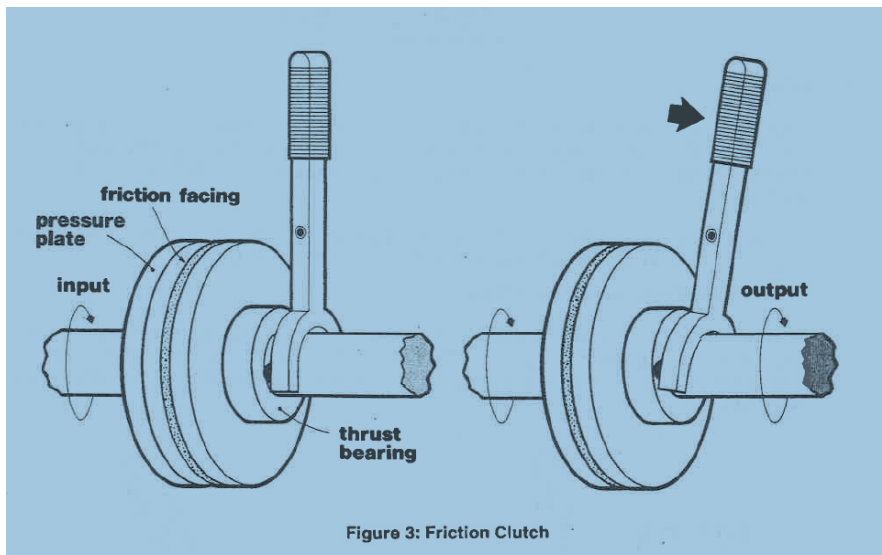
Positive Engagement Units

Positive engagement units do not allow slip.

Positive engagement brakes either hold something stopped or let it turn freely. Similarly, positive engagement clutches are either locked up or free. There is no sliding friction in positive engagement units, and no heat generation.

There are many kinds of positive engagement brakes and clutches. Sane engage or disengage themselves automatically. Others are actuated by something external, like a manual lever or air pressure. Some will resist or transmit torque in both directions, others will not.

Bi-Directional Units

Bi-directional units work in both directions.

Bi-directional brakes and clutches will resist or transmit torque whether parts are turning forward or backward.

Brakes

A stationary pawl or dog that can engage a cogged wheel is one kind of positive engagement brake. Since it cannot slip, it can only hold something stopped; it cannot limit or reduce speed like most brakes.
If the input is turning when the brake is engaged there will be clashing and shockloads.
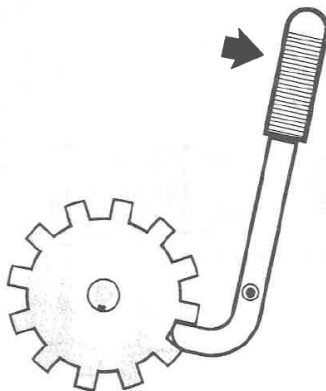

Figure 4: Positive Engagement Brake

Applications. Positive engagement brakes are used for:

- . Accurate and repeatable positioning of parts being machined
- . Locking of spindles, chucks, and face plates for tool changing
- . Parking locks for movable equipment and vehicles.

Without slip, there will be no friction, heat or wear.

Troubleshooting. There is not much that can go wrong with this kind of device if it is always engaged when the input is already fully stopped. Forced engagement when the input is turning will wear or break the teeth and pawl.

Sometimes the pawl or engaging tooth is rounded and engaged by a spring. This allows it to bump along over the wheel teeth if the input it turning. Too much bumping may wear the teeth so that the pawl jumps out of engagement underload.

Clutches

Dog, jaw, or tooth clutches connect or disconnect parts that are stopped or rotating at the same speed. Clashing or shockloads will result if they are engaged when input and output speeds are not the same.
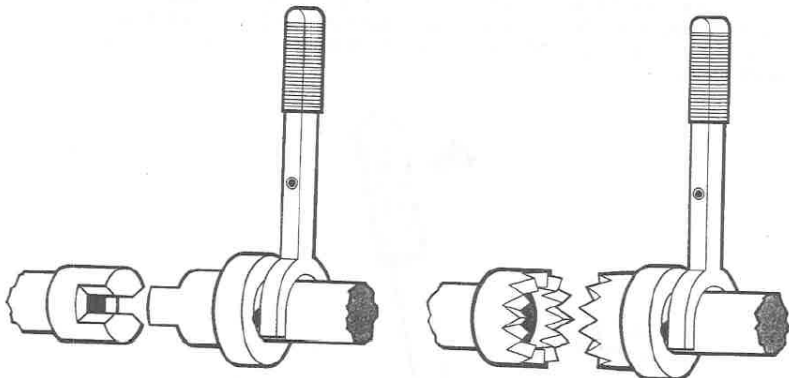

Figure 5: Jaw Clutch and Tooth Clutch

Positive engagement clutches are like quick disconnect couplings.

Applications. The following list describes some of the uses for positive engagement clutches:

. Selective connection of motors or shafts to gears, pulleys, or other shafts.

. Gear changes or reversing in light duty, slow speed equipment, where rotating mass is small and slight clashing can be tolerated.

. Synchromesh mechanisms, where a tooth clutch locks input to output only after they are brought to the same speed by some kind of friction clutch.

. Overload Protection. Some units are intended to jump out of engagement automatically when too much torque goes through them. Some require a manual reset. Others automatically lock up again after one revolution.

Engagement problems are common.

Troubleshooting. If a non-overload unit does not go into mesh easily, or jumps out of mesh, it is probably due to:

. Tooth or jaw wear from too much clashing
. Misalignment of input and output
. A worn shift collar, yoke, or bearing. Worn or misadjusted operating linkage can also be responsible.
. Defective overcenter springs or detent parts. Many units include these devices which are intended to keep them either engaged or disengaged.
If an overload unit jumps out of mesh, it probably means the load is too large. Do not just tighten up the torque adjustment without checking the specification.

One-Way Units

One-way units either engage or disengage when their rotation is reversed.

One-way positive engagement brakes prevent rotation in one direction and allow free rotation the other way. One-way positive engagement clutches lock up when torque is applied in one direction and disengage when torque is applied in the other direction. There are various mechanisms that provide one-way operation.

Ratchets

A ratchet holds or releases, depending on which way the wheel turns.

Ratchets are used in low-speed machinery. They are likely to be noisy and to wear quickly during overruning, as the ratchet pawl rides over the wheel teeth. The wheel may need to reverse direction as much as a full tooth before lockup, so precision is limited.

Ratchets should be checked for broken or rounded teeth and a broken, worn, loose, or rounded pawl. A broken pawl engaging spring may cause erratic engagement.
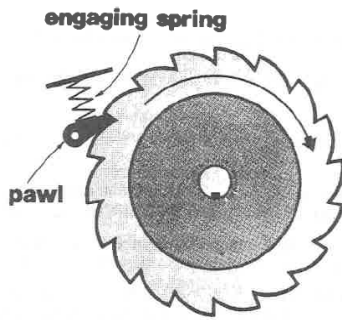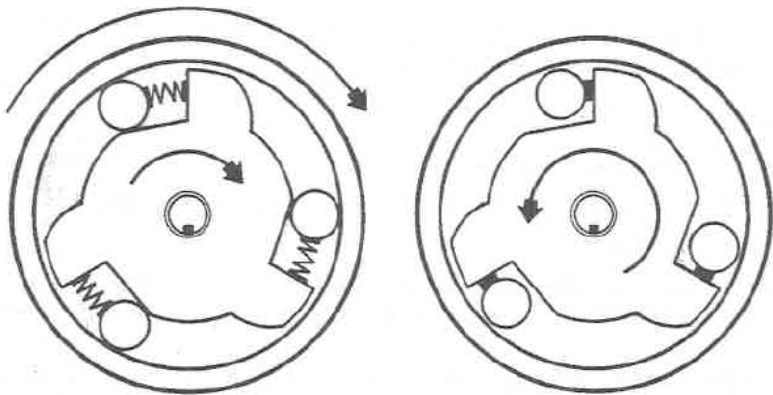
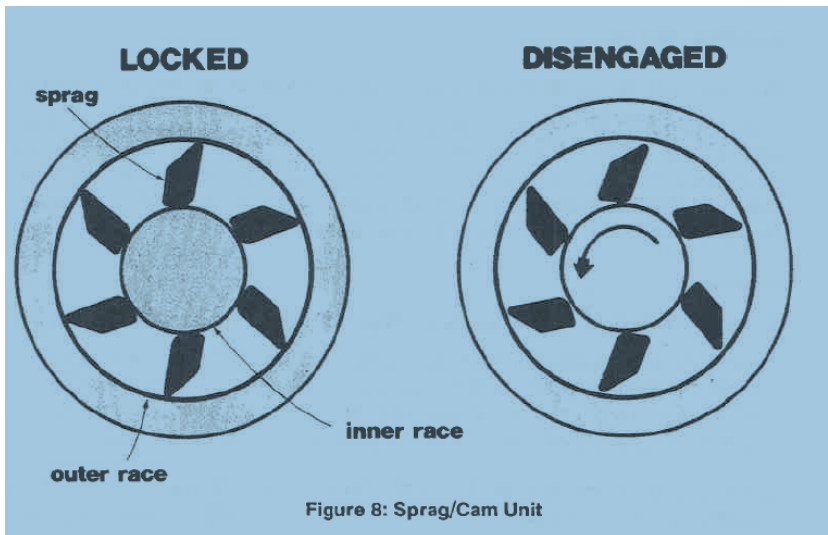Figure 6: Ratchet

Figure 7: Roller Unit

Figure 8: Sprag/Cam Unit

Roller and Sprag/Cam Units

A roller or sprag/cam unit can be used either as a brake or a clutch. It will lock up with almost no lost motion, and will handle high torque and speed. Overrunning friction, wear, and noise are low.

Rollers or sprag/cams can wedge anywhere instantly.

The principle of operation in both roller and sprag/cam units is the same. If the input and output elements try to turn so the rollers or sprag/cams jam between them, the unit will lock up. If the input and output elements turn so the rollers or sprag/cams push out of the way, the unit will disengage.

Applications

One-way positive engagement clutches and brakes are used in many kinds of equipment:

. As a backstop or hold back brake on an elevator or conveyor to prevent downward motion in case power fails.

. As an overrunning or freewheeling clutch to allow the output to turn faster than the input. A bicycle drive is an example; the rider can stop pedalling while the bike coasts.

. As an auxiliary or inspection drive that allows two or more power sources to turn the same equipment independently. If one motor stops, its clutch freewheels while the load is driven by another power source.

Maintenance of Sprag/Cam and Roller Units

Use the right amount of the right oil or grease.

Many sprag/cam and roller clutches are lubricated and sealed for life, and require no maintenance other than periodic inspection for leakage.

Large, heavy-duty units, however, sometimes have oil reservoirs built in. Follow the manufacturer's recommendations, and do not overfill with oil. Usually half full is enough. Too much oil will get in the way and increase overrunning friction and heat build-up. Oil with extreme pressure additives may cause the sprag/cams or rollers to slide rather than grab.

Some units are grease lubricated or at least there may be grease fittings on the integral bearings. Be sure to use the correct grease.

Troubleshooting

Wear is usually slight but stresses are high and parts can break.

Roller and sprag/cam units should never clash or jerk, since the change from engaged to disengaged occurs only at the instant when the input and output are turning at the same speed.

If the unit jerks, slips, fails to lock up, or locks up in both directions, it is probably due to:

. A cracked outer race
. Flat spots worn on rollers or sprag/cams
. Broken springs or retainers
. Lubrication failure.

Misalignment can cause problems. Some units rely on the input shaft, or both input and output shafts, for alignment of internal parts.
Most units include integral bearings. If overloaded or misaligned, the bearings will fail, and the resulting looseness will interfere with proper operation.
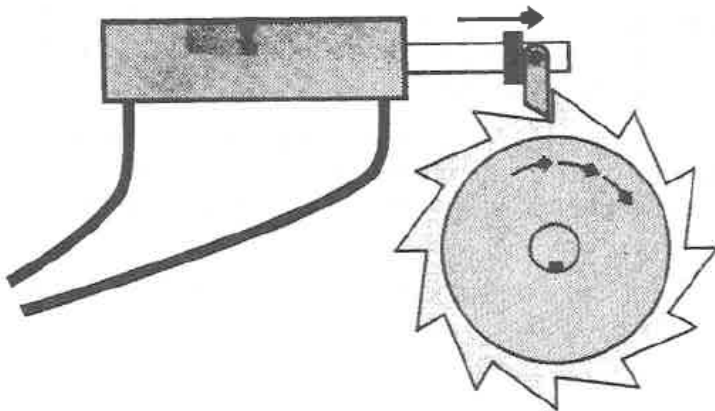

Figure 9: Ratchet Indexing Mechanism

Friction Units

Friction units can slip, allowing them to start or stop loads smoothly.

There are two basic types of friction brakes and clutches. In dry friction brakes and clutches, a steel surface and friction facings or linings are in direct contact. Wet brakes and clutches are similar except that there is an oil film between the surfaces whenever they are slipping. Oil clinging to both surfaces couples them together by viscous friction.

Contacting Parts

The contacting parts are a steel surface and friction facings or linings. The steel surface may be a drum, a disk, or disks.

Drums can be internal when the friction facing shoes push out on the inside of the drum, or external when the friction facing bands or shoes clamp around the outside.
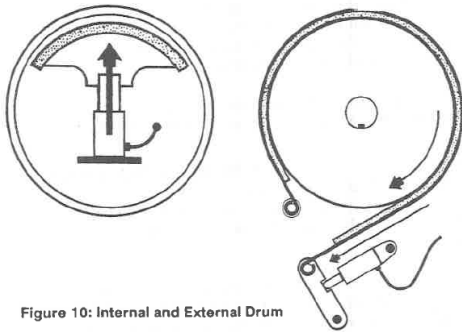

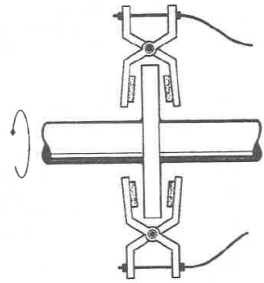Figure 10: Internal and External Drum
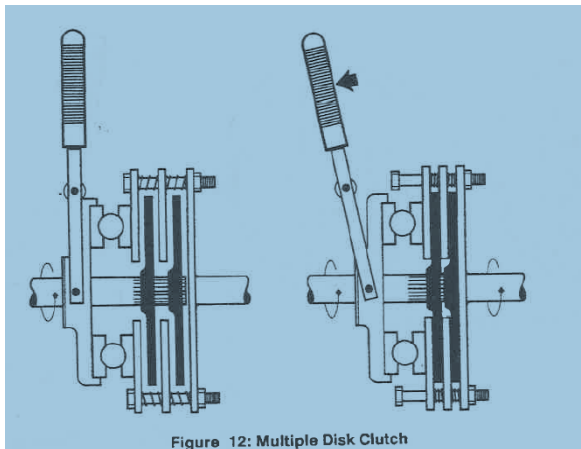
Figure 11: Caliper Disk Brake

Figure 12: Multiple Disk Clutch

Disks sometimes are squeezed on both sides by friction facings as in a caliper disk brake.

In many units, the full surface of a disk is contacted by friction facings. There may be several steel disks in a clutch or brake with disks of friction facing material between each of them. This allows more contact area, and thus more torque, in a compact space.

Actuation

Brakes are simple to release or apply, since one part is stationary.

Friction brakes and clutches require some way to squeeze the friction facings against the steel surface. Or they may need sane way to hold the facings away from the steel surface, if a spring in the unit normally squeezes them together. In brakes, this is not complicated, because one part does not turn. It can easily be pushed against the other part, or pulled away from it.

A common Motor Brake, for example, is applied by spring pressure. The unit is usually mounted directly on the motor. A solenoid coil wired to the motor releases the brake whenever the motor is switched on. When the motor is shut off, the spring applies the brake.

In a clutch, however, both friction facings and the steel surface turn. This is no problem when the actuating mechanism turns with the clutch.

A simple Overload Clutch is an example. A spring inside the clutch always pushes the friction facings against the steel surface. There is no need to separate the friction surfaces. The unit will slip and protect the drive motor and machinery from overloads. When the overload is removed, the clutch stops slipping.

A Centrifugal Clutch likewise does not need anything external to actuate it. Weighted shoes turning inside a drum are flung outward into contact with the drum, and friction transmits the torque. At higher speeds there is more friction and less slip.
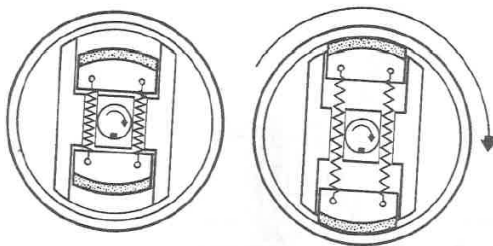
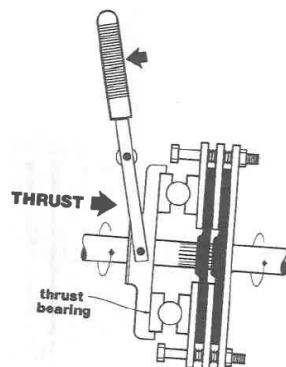Figure 13: Centrifugal Clutch

THRUST

thrust bearing

Figure 14: Thrust Bearing

Frequently clutches are engaged or disengaged by some non-turning mechanism outside the clutch. In this case, a thrust bearing must be used.

Clutches often require a thrust bearing to apply the operating force.

A typical automotive clutch, for example, is engaged by the pressure of springs which turn with it. But it is disengaged by force applied to a pedal, and delivered to the clutch through linkage and a thrust bearing.

Sometimes, however, it is more convenient to build them into the clutch and allow them to rotate with the unit. In this case, air pressure or oil pressure must be sent to them through some kind of rotating union.
The input pipe to a rotating union is stationary; the output pipe can turn with the clutch.

Some clutches use an electromagnet to squeeze the facings together or separate them against spring pressure. If the electramagnet.is part of the clutch and rotates with it, current can be sent to it through stationery brushes contacting slip rings which turn with the clutch.

Often the electromagnet itself is stationary, however, the electromagnetic force is passed across a gap between the electromagnet and the rotating clutch parts. It pulls the friction facings in by magnetism.
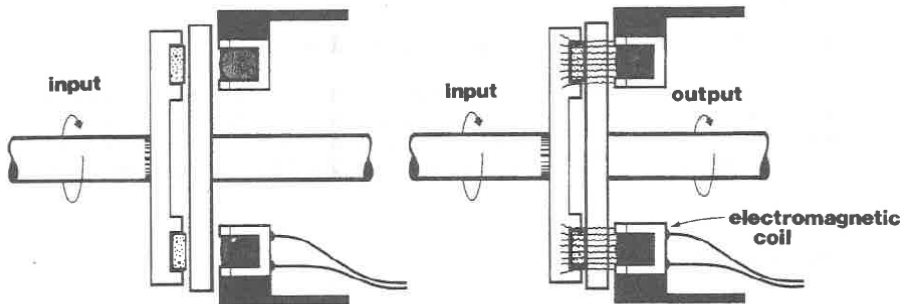

Figure 15: Electromagnetic Clutch

Self-energizing units need less applying force.

Self-Energizing Brakes and Clutches may be actuated by any of the above methods. But the unit is designed so that the torque it is transmitting or resisting aids the applying force. Automotive drum brakes, for example, use the friction between the drum and the shoes to wedge the shoes tighter to the drum.
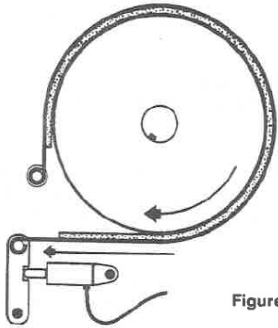
Figure 16: Self-Energizing Band Brake

Brake bands can be arranged to work the same way, wrapping tighter as the friction increases. Self-energizing brakes and clutches will hold better in one direction than the other. In fact they may tend to grab one way, and not hold at all the other, like a sprag/cam or roller unit.

When the brake is actuated, friction between the band and the drum pulls on the bid the same way the actuator does, increasing the applying force.

Dry Friction Units

Dry friction units do not slip until overload

Dry friction units will hold without any slip until torque exceeds a certain value. Once it does, slip starts immediately. The unit seems to break away, and transmits (or resists) less torque while it is slipping. Slip stops only when torque drops below the original value.

Applications

Dry friction units are, Simple, reliable, and easy to work on, so they are more widely used than any other type. Dry friction brakes are used to:

•   Bring coasting machinery to a stop quickly in an emergency, or when tools need to be changed with minimum downtime.

- Limits and regulate rotating speed to prevent runaway loads.
- Provide controlled tension on winding and wrapping machinery. Dry friction clutch applications include:
- Providing overload protection: when machinery jams, or is overloaded in any manner, a torque clutch, slip clutch, or overload clutch will slip aid prevent damage to the equipment, the drive motor, 'or the load itself.

Dry friction clutches control and protect rotating machinery.

- Allowing an electric motor to start up without a load for reduced current draw.
- Disconnecting an internal combustion engine from its load so the engine can be started and allowed to run while the load is stopped.
- Bringing a load up to speed gradually or at a controlled acceleration rate without jerks.
- Allowing one power source to drive several different loads independently.
- Routing driving power selectively to provide reversing and speed changes in transmissions.

Maintenance

Slip causes wear of friction facings.

The most common maintenance is checking, adjusting, or replacing friction facings.

Like any powered machinery, clutches and brakes can be dangerous to work on.

It is important to observe the following precautions:

1. Follow standard lockout and tagout procedures when working on any clutch or brake.
2. Make sure no tools or loose parts are left on, in, or partially attached to clutches and brakes.
3. All shafts, pulleys, sheaves, belts, chains, sprockets, and couplings must be properly realigned and tightened.
4. Turn the equipment over by hand at least a few turns before starting the motor.
5. Replace all guards before starting things up, and do not stand directly in line with the clutch or brake. Grease and oil spatters from high speed rotating machinery can be flung outward with considerable force, as can screws or bolts that may not have been tightened.

Adjusting friction facings. Friction facing wear does not affect the operation of some units. Sane are self-adjusting for wear. But others need periodic adjustment to compensate for friction facing wear. Be careful not to adjust the unit so that it drags when released.

Check the thickness of the friction facings regularly. Worn friction facings cause:

- **More slip**: If the facings are worn, applying springs will not be compressed as far as normal, and they may not press facings and steel together tightly enough. Releasing springs will be compressed too far and will prevent the friction surfaces from being pressed together tightly enough. More slip produces heat and accelerates wear.
- **Damage to the steel surface**: Some friction facings are held in place by screws or rivets. Sane are bonded with a high-temperature cement. If the facings wear too thin, the screws or rivets, or the shoe or plate behind the facing, will contact the steel surface and score it.

Replacing friction facings. Replacing friction facings may be simple as in the case of most caliper disk brakes and may drum clutches and brakes. Single-disk dry friction units sometimes have segmented friction facings which can be replaced without taking the unit apart. Multiple disk units will probably require complete disassembly.

Steel surfaces contacting friction facings must be smooth.

If the steel surface contacting the friction facings is grooved, scored, or roughened in any way, it must be resurfaced or replaced. New friction facings rubbing on a rough or grooved surface will not grip as well and will wear quickly. Abrasive dirt in the unit, as well as worn linings, can roughen the steel surface.

> **Dry friction linings are sometimes made with asbestos. This is a hazardous material. There may be asbestos dust in dry friction brakes and clutches. Be careful not to breathe it or blow it around. Do not create asbestos dust by grinding, filing, sawing, or sanding friction facing material.**

Dirt, oil or grease on the friction facings will ruin the unit.

When working on dry friction units, or when lubricating nearby equipment, be sure the friction facings are not contaminated with any oil, grease, dirt or fluid. Operation may be seriously affected.

Clutch and brake friction facings often need gentle treatment during their break-in period. Before the facings have worn enough to conform fully to the steel surfaces, high spots on them will overheat. Slip characteristics may be different, and fade is more likely.

Troubleshooting

Hot operation is a common problem.

Overheating. Same tensioning drag brakes are expected to slip continuously, but the speed and torque are low, and heat build-up is limited. But most dry friction units will overheat if they slip too much while resisting or transmitting torque.

Many units are intended for brief periods of slip only. If cycled too frequently, without having a chance to cool, they will run hot. Multiple-disk units are particularly difficult to cool because of their large torque capacity and limited surface area.

Too much heat can damage friction facings and crack or distort steel parts. Drums will expand, either tightening up on the friction facings, if external, or loosening if internal. Disks may wrap and become wavy. Operation will be seriously affected.

Free air circulation is essential

Air-Cooled Units: Most dry friction units are air cooled. They often have fins or air channels built in to encourage air circulation.
These must be kept clean so air flow can carry off heat. Any screens or vents in the clutch or brake enclosure must also be clear. On the other hand, do not leave any inspection or adjustment access hole covers off in a dirty environment.

Fluid-Cooled Units: Some large, heavy-duty applications use water or some other fluid, piped through shafts or housings, for cooling. Be sure the cooling fluid is circulating and is carrying heat away. Fluid leaving should be warmer than fluid entering.

Almost anything which increases slip will cause hot operation.

Causes of Overheating: Increased slip and overheating can be due to:
- Weak springs
- Misadjusted manual operating linkage. 'Check to see if there is adequate free play when no manual force is being applied.
- Misuse by operator.
- Low voltage, air pressure, or hydraulic pressure. Check for bad slip rings, in electro-magnetically operated units. The air pressure in pneumatic units, or the oil pressure in hydraulically operated units may need to be adjusted. Check for leaks at piston seals, hoses, rubber bladders, and rotating unions where the air or oil canes in.

Sometimes brakes will sip, or refuse to hold, when they are wet. This is the only kind of slip which will not cause overheating. As the moisture evaporates, it will absorb and carry off heat until the unit is dry and the problem ends.

Grabbing, chattering, dragging, and failure to disengage are also common problems

Other problems. Oil, grease, or other contamination on the friction facings often causes violent operational symptoms. Loose bolts or nuts, worn keyways and splines, broken damper springs, misalignment of shafts, loose equipment mounts, and many other mechanical conditions can also be responsible.

Dirty or worn slip rings, which do not provide good electrical contact, can cause malfunctions in electromagnetic clutches using them.

Failure to disengage completely is usually due to misadjusted operating linkage or broken release springs.

Wet Friction or Oil Shear Units

Viscous friction in oil transmits the torque

Wet clutches and brakes are made to operate in an oil bath. The oil forms a film on the friction facings and the steel, preventing direct contact. The oil is slippery, and allows the friction facings to slide on the steel. But the oil is also sticky and viscous enough to cling to both surfaces and couple them together.

Torque depends on oil film thickness and rate of slip.

The tighter the facings are squeezed together, the thinner the oil film between them, and the more torque the unit will take. If all that oil is squeezed out, the facings will contact the steel directly and the unit will lock up.

Wet units last longer and are easier to cool

The main advantage of wet brakes and clutches is less wear, since slipping takes place in the oil rather than between rubbing surfaces. Heat is still generated, as it is whenever input power is greater than output power, but the oil carries it away better than air.

Some wet clutches and brakes are much like dry friction units, in fact some dry friction units can be used in oil without any modification, though they will naturally slip easier. Most wet brakes and clutches use a different friction facing material than dry units. Multiple disk wet units are common since heat is less of a problem.

Wet units are frequently operated by hydraulic force. Since they must have oil for lubrication anyway, it is usually simplest to operate them with oil pressure.

Applications

Transmissions frequently include wet clutches and brakes, particularly when gear changes must be made under load. A combined oil system for gear lubrication and clutch/brake operation is convenient. The oil is often sent to an external cooler.

Wet clutches are sometimes used in large winding drives where they are expected to slip most or all of the time. In such applications they are usually referred to as "oil shear variable speed drives."

Maintenance

Wet clutches and brakes must not run dry.

Oil level, quality, pressure, and temperature should be checked frequently.
The exact viscosity and friction characteristics of the oil used are important to proper operation. Some oils have special friction modifier additives matched to the kind of friction facing materials used. Be sure to use only the specified oil.

If the unit is operated by oil pressure, there must be adequate pressure at the right time for proper operation. Often there is a plug in the oil line where a gauge can be installed to monitor the oil pressure.

Overheated oil must be changed

The operating temperature of the oil must not be too high or heat will damage both the oil and the friction facings. Oil that has been repeatedly overheated is likely to be oxidized and no longer suitable.

Make sure oil is circulating to the cooler, and being cooled there. Clogged air passages, bad thermostats, non-working fans or coolant pumps, and plugged coolant lines can all cause hot operation.

A case or oil pan caked with dirt can also raise the oil temperature inside.

Troubleshooting

Excessive slip and dragging are probably due to inadequate oil pressure.

The hydraulic circuitry which controls pressure may be defective. Bad seals around the actuator piston, or in the line to the actuator, may cause internal leaks which bleed off oil that should be applying or releasing a clutch or brake.

Weak release or apply springs can have the same effect.

Actuator pistons occasionally stick.

Friction facing wear is rarely the problem, although sometimes the lining material can become glazed from overheating or bad oil.

MECHANICAL POWER TRANSMISSION

COUPLINGS

A coupling is a device which connects two parts of a mechanical system. A very common use of couplings is on machine shafts for transmitting torque.

The various purposes for which couplings can be used are:

1. To couple two or more shafts where transmission length of shafting is more than the commercially available length of the shafts.
2. To connect the shaft of a driving machine to a separately built unit e.g. a motor and a pump or fan
3. In this function, coupling also provides for disconnection for repairs or alterations of either driver or driven unit.
4. To reduce the transmission of shock loads from one shaft to another
5. To provide for misalignment of shafts or to introduce mechanical flexibility.
6. To alter the vibration characteristics of rotating units
7. Sometimes these are also used to introduce protection against overloads.

The various requirements of a shaft coupling are: - It should be easy to connect or disconnect, transmit full power, have no projective parts and hold the shafts in alignment.

The various types of couplings in common use are rigid couplings, flexible couplings and hydraulic couplings. Each type has got further more varieties which will be discussed. The selection of a particular coupling is based upon the considerations like loading, misalignment and stiffness. In loading one has to see what is the maximum loading, whether the load to be transmitted is of steady or vibratory nature, and what is the torque to be transmitted. In misalignment one has to consider the maximum parallel and angular misalignment and the ability to compensate for axial displacement; in other words capability of taking care of misalignment in any of the three planes. Stiffness is very important in case of flexible couplings; it is the permissible deflection across the coupling due to static and dynamic loads, and the ability to provide damping and detuning effects.

INSTALLATION OF COUPLINGS AND SHAFTING

Types of Coupling

1. Chain Couplings

If the coupling has a cover, it must be packed with grease. Take care to prevent damage to the seals during fitting. When the coupling is in use, the grease must be replenished regularly. If the coupling has no grease cover, the chain must be oiled regularly. See Fig. 17

2. Gear Couplings

The sleeves have to be replaced on the shafts before the hubs are fitted. Take care to avoid damage to the seals in the sleeves during assembly. The coupling must be lubricated according to the manufacturer's instructions after assembly. When in use, the grease must be replenished regularly. See Fig. 18

3. Sleeve Couplings

The shafts must be correctly aligned before the coupling is fitted. Fig. 19

4. Hookes joints

Check the manufacturer's instructions for details of lubrication. On all the following couplings avoid contamination of the rubber components by oil or grease. Fig. 20

5. Disc couplings

When in use the rubber disc must be regularly checked. If it is worn or torn it must be replaced. Fig. 21

6. Pin and rubber bush couplings

Check the manufacturer's instructions to see which half of the coupling is fitted to the driving shaft and which to the driven.

When in use, the rubber bushes should be checked regularly and replaced if worn or stretched. Fig. 22
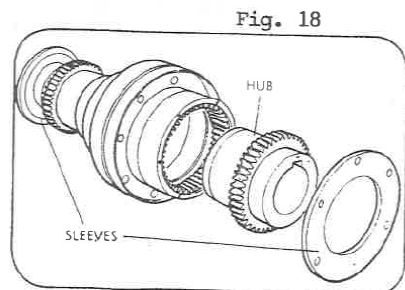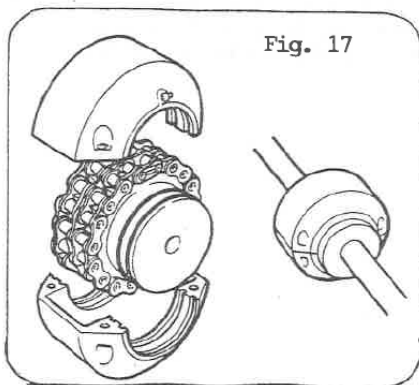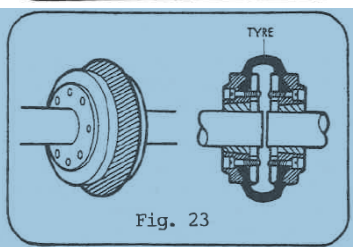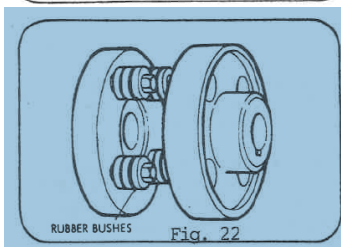
Fig. 17

Fig. 18

HUB

SLEEVES

FIG. 19

PINS
SLEEVE COUPLING

SPLIT SLEEVE COUPLING

PRINCIPLE OF
THE HOOKES
JOINT
Fig. 20

RUBBER DISC
Fig. 21

RUBBER BUSHES   Fig. 22
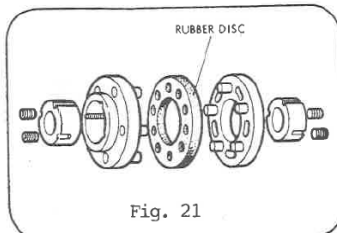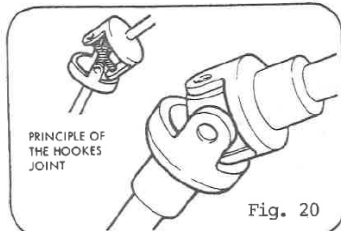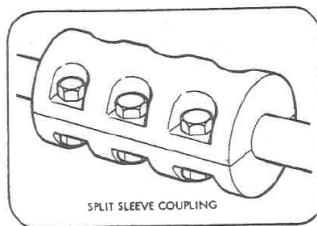
TYRE
Fig. 23

7. Rubber tyre couplings
Safety dogs are sometimes fitted. Check to see if they are required. Check the tyre for wear or stretching, and replace as necessary. Fig 23.

8. Spider couplings
Care must be taken to avoid damage to the spider during fitting. Fig. 24

9. Barrel couplings (Fig. 25)

When in use the flexible tyre must be checked for signs of wear.

SPIDER
Fig. 24

FLEXIBLE TYRE
Fig. 25

Shaft misalignment can be of two types:

(i) Lateral misalignment, in which case the two axes are parallel but at a relatively small distance apart
(ii) Angular misalignment in which case, shafts have intersecting axes
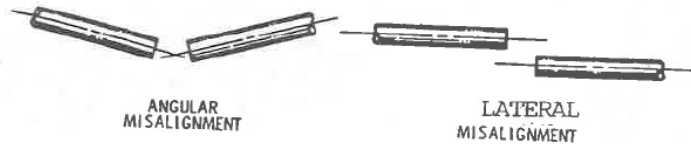
Fig. 26

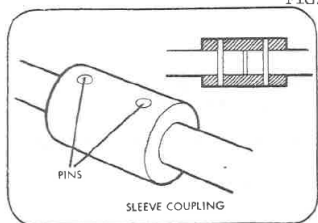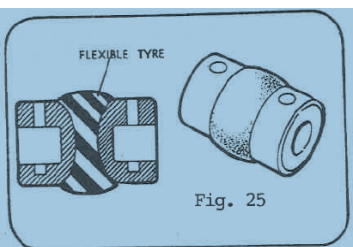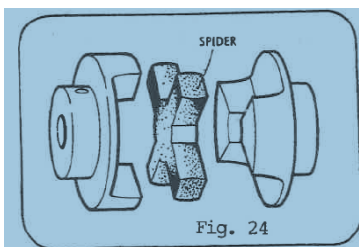Most of the couplings are of two types, namely, rigid and flexible. Rigid couplings have no flexibility or resilience and can be used for shafts in good alignment both laterally and angularly.

1. Flexible Coupling

Flexible couplings allow slight angular misalignment and also allow the axes of the shafts to float or run slightly out of alignment, and in some cases allow the shafts to move end-wise. Flexible couplings are also used to compensate for slight inaccuracy of workman-ship in aligning the two connected machine elements.

2.   Rigid Couplings

Rigid couplings are used to connect shaft ends together rigidly. They provide a fixed union that is equivalent to a shaft extension. They should only be used then true alignment and rigidity are required as with line shafting or where provision must be made to allow parting of a solid shaft. There are two basic rules that should be followed to obtain satisfactory service from rigid couplings:

1. A force fit must be used in assembly of the coupling half to the shaft end
2. After assembly, a run out check of all mating surfaces of the coupling halves must be made and if errors are present the surfaces must be machined true.

Rigid couplings should not be used to connect shafts of independent units that are aligned at installation.

Five types of rigid couplings are commonly used:

(1)   Sleeve type

(2)   Ring compression type

(3)   Flange compression type

(4)   Clamp type

(5)   Flange type

2.1) Sleeve or Collar Coupling and Compression Coupling

A sleeve or collar coupling consists of a cylindrical sleeve pressed over the ends of two collinear shafts, approximately one-half of the sleeve containing each shaft. The radial movement is prevented
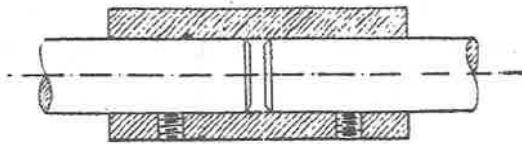
FIG. 27 SLEEVE COUPLING

by means of keys, taper pins, or set screws. These are generally used with small diameter shafts. With their use it becomes difficult to disconnect the coupling.

The disadvantage of sleeve coupling of not being able to disconnect can be overcome by using compression coupling in which it is possible to install and remove the coupling without disturbing the shafts. A ring compression type coupling utilises two split cones which are drawn together by the bolts in order to produce a wedging action with tightens the parts of the coupling and shafts. The torque is transmitted by keys between the shafts and the cones, and by friction between the cones and the outer sleeve.
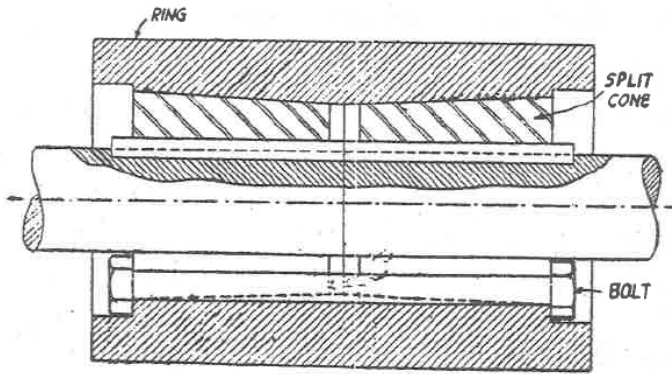
FIG. 28 - Ring Compression Coupling

2.2 Flange Compression Couplings

These may or may not be keyed and have the advantage of simple adjustability and low cost.
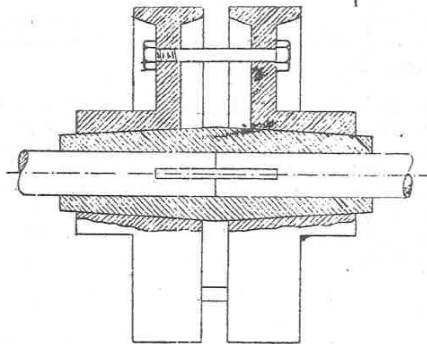
FIG 29- Flange Compression Type Coupling

2.3 Clamp-type Couplings

These are split longitudinally, and both parts of the coupling are machined to fit the shaft and clamped tightly against the surface of the shaft by through bolts. Torque is generally transmitted through a key. Its construction in two halves affords ease in assembly and disassembly. For safety purposes, the bolt heads and nuts are protected by ribs.
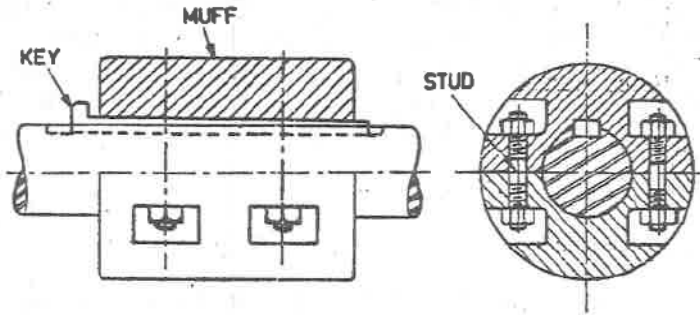
FIG. 30 Clamp-type Coupling

2.4 Flange Coupling

It is the most common coupling due to its simplicity and low cost. Its use is possible only when the connected shafts are accurately aligned in order to prevent severe bending stresses and excessive wear in the bearings. These are generally used for heavy loads and in permanent installations. They have the advantage of being shorter
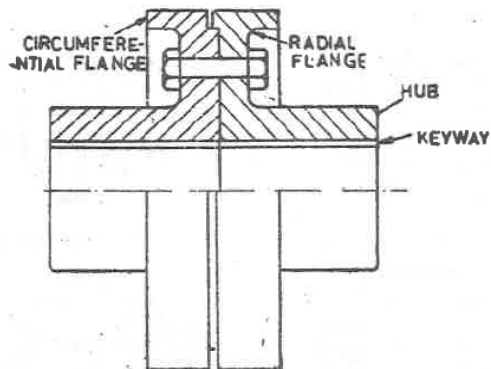
FIG. 31 - Flange Coupling

and lighter than many other heavy duty couplings. It consists of two mating halves which are shown assembled in Fig. 31. This coupling transmits torque by virtue of friction between flange faces and shear stress in the bolts or pins. Where friction is employed, bolts are required to provide the normal force associated with sufficient friction. The coupling bolts should be ground and fitted to holes that are drilled and reamed in the radial flanges after the coupling halves are assembled in order to allow all the bolts to share equal load when torque is transmitted. The circumferential flange is provided to avoid danger of exposed rotating nuts and bolt heads. The length of the hub is determined by the length of key required, and the hub diameter is about twice the bore. The thickness of the flange is determined by the permissible bearing pressure on the bolts. The failure of the radial flange by shearing at the juncture with the hub should also be checked.

3. Flexible Coupling

When coaxial shafts of independent units are to be connected, "flexible" couplings are used. Flexible couplings can accommodate shaft misalignment and end float while they transmit power and positive rotation. This kind of coupling is used because it is impossible to perfectly align the connection of electric motors, internal combustion engines, and similar prime movers to speed reducers, fans, pumps and other equipment with built-in shaft supports. Flexible couplings are intended simply to overcome slight unavoidable errors in alignment, not to connect shafts on different planes or at an angle. In addition to being able to tolerate both angular and parallel misalignment, flexible couplings also allow some axial movement, or float, and may, depending on the type, allow torsional movement as well. Illustrated in Fig.32 are the four types of flexibility that may be provided by flexible couplings.

(A) Angular misalignment. (B) Parallel misalignment.

(C) Axial movement. (D) Torsional movement.
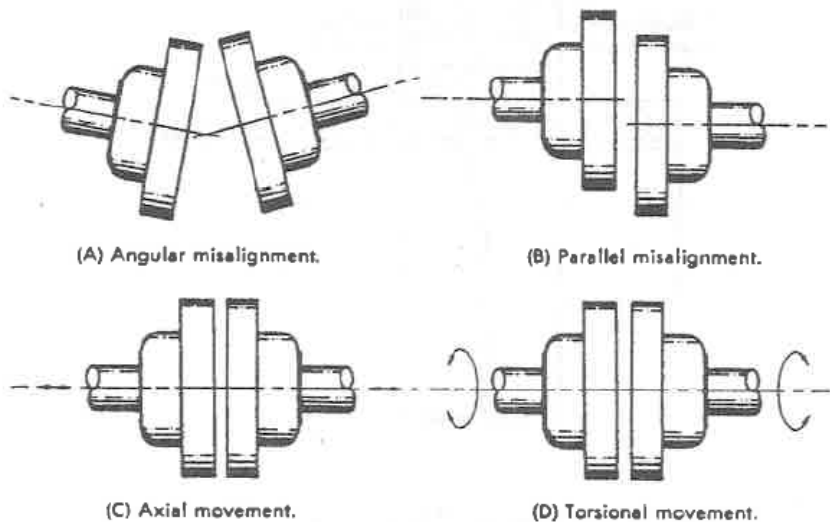
FIG. 32 - Four Types of Coupling Flexibility

3.1) Flexible Couplings Employing Rigid Members
i) Oldham Coupling

Also known as double-slider coupling, it is used for shafts with lateral misalignment amounting to as much as 5% of the shaft diameter with considerable axial play and with relatively small angular mis-alignment (one degree). It consists of three parts - two slotted hubs and a central piece having two tongues, one on each face perpendicular to each other. In the assembled form, the tongues on the central piece engage the grooves on the end pieces which are attached to the shafts to be connected. It is important to lubricate and keep clean the sliding surfaces.
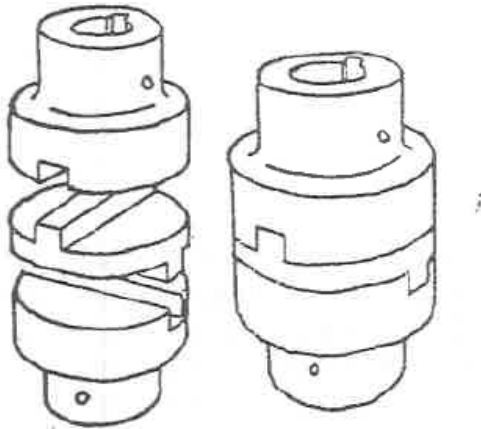
FIG. 33 - Oldham Coupling

3.ii) American Flexible Coupling

It also operates on Oldham principle (Fig34). Central floating member is a block which can ride between the faces of the jaw flanges of the driving and driven members.

In the position shown, the left member can slide up and down, the right member can slide to and fro on the square floating centre member. The combined action, when the parts are assembled, produces a flexible connection that will care for misalignment. The bearing surfaces of the floating block are provided with replaceable non-metallic strips which are lubricated from the grease reservoir in the hollow block.

This coupling may be used with some angular misalignment also in addition to lateral misalignment of shafts if the coupling halves are properly mounted with axial clearances.
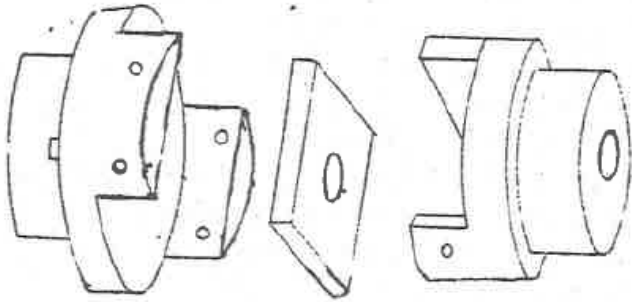
FIG. 34 American Flexible Coupling

iii) Chain Coupling

This coupling consists of two sprockets mounted face to face on the ends of the shafts and coupled and connected by a length of double chain (roller chain or silent chain) wrapped around the sprockets. The drive is thus through the chain. Flexibility is obtained by the lateral play in the fit of the chain over the sprocket teeth (no circumferential play). It is relatively inexpensive type. It should preferably be installed with a casing to contain grease lubricant. Small lateral and angular misalignment can be accommodated.
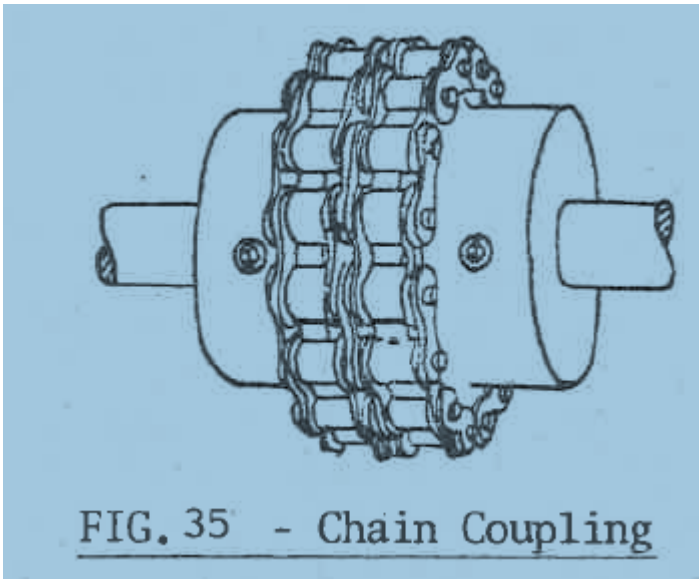
FIG. 35 - Chain Coupling

iv) Gear Teeth Coupling

These are double engagement type and may be used in the case of angular and parallel misalignment. The hubs have external gear teeth that mesh with internal teeth in the casing through 360° as in a splined connection. Gear teeth are of a special design with involute flanks and faces curved in both directions so that they may accommodate angular misalignment by a ball and socket action. Flexibility is obtained by play between the teeth. The use of gears improves the accuracy of the fits and results in smoother and quieter operation. It is necessary that clearance between shaft ends be provided. This coupling has wide application in railway, automotive, air - transportation, marine equipment and industry.
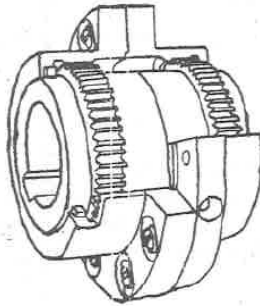
FIG 36 - Gear Teeth Coupling

3.2) Flexible Couplings with Resilient Parts

The ability of a coupling to compensate for fluctuating torsional load depends primarily upon the dynamic characteristics of the resilient member employed.

Bibby (Falk) or Flex Steel Coupling In this type of coupling, each of the hubs is provided with teeth or slots, and both hubs are connected by a continuous steel strip spring lying in slots. The teeth are flared (widened toward the centre) and the stiffness of the coupling spring changes with the deflection or load. These couplings allow axial, traverse, and angular misalignment and have a detuning ability in case of vibratory disturbances. In this case coupling parts can be readily disconnected by removing the spring.
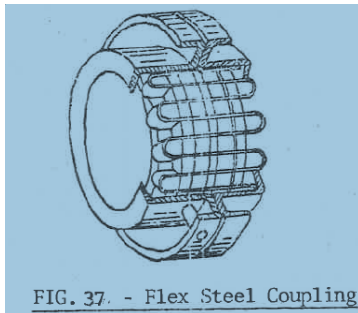
FIG. 37. - Flex Steel Coupling

ii) Peripheral Spring Couplings

These utilise helical springs as a resilient connection between the two halves as shown in Fig.34. The springs are inserted between the hubs of the coupling (against the hardened spring seats). This coupling is used to reduce the transmission of shock loads, and due to certain amount of backlash, it is also capable of detuning action
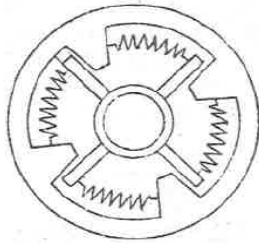
FIG. 38 - Peripheral Spring Coupling

iii) Spoked Couplings
These utilise number of spokes which are usually fixed at one end and may be fixed, pivoted or free at the other end. Spokes are loaded in a similar way as a cantilever beam. Angular displacement changes the effective length of the spoke, which in turn changes the spring constant.

3.3) Resilient material flexible Couplings

These are simple in design and have low cost. The torque is transmitted through resilient elements generally exposed to shear and usually permitting larger misalignment. The application of rubber can be limited to lighter loads only because of its low strength. Owing to the non-linear characteristics of rubber elements, this type of coupling can be used for detuning purposes.

Ajax flexible coupling is an example of this type which consists of two halves connected together by several pins. One set of pins is rigidly attached to one-half of the coupling and the alternate pins attached to the other half. Each steel pin that is attached to one half of the coupling projects into a bronze bushing that is mounted in a rubber sleeve in the other half as shown in Fig. 39. Usually a small amount of axial clearance is kept between the coupling halves.
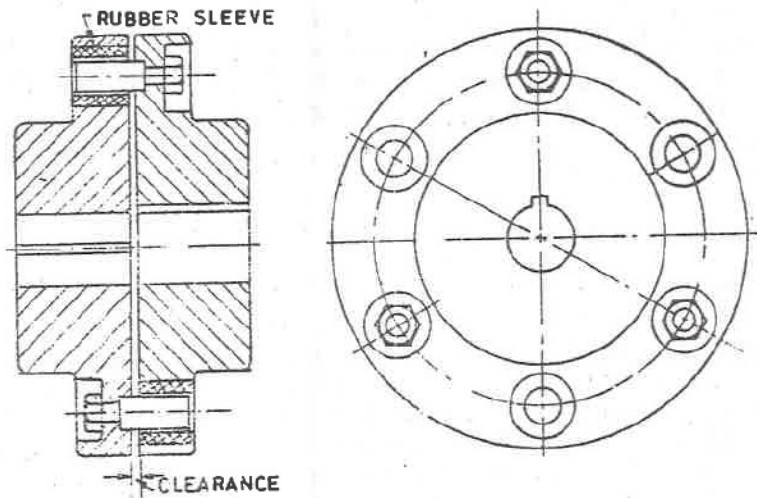
FIG. 39 - Ajax Flexible Coupling

3.4 Slip Coupling

It is used to prevent transmission of over loads on the electric motors. Its constructional details are shown in Fig. 40.
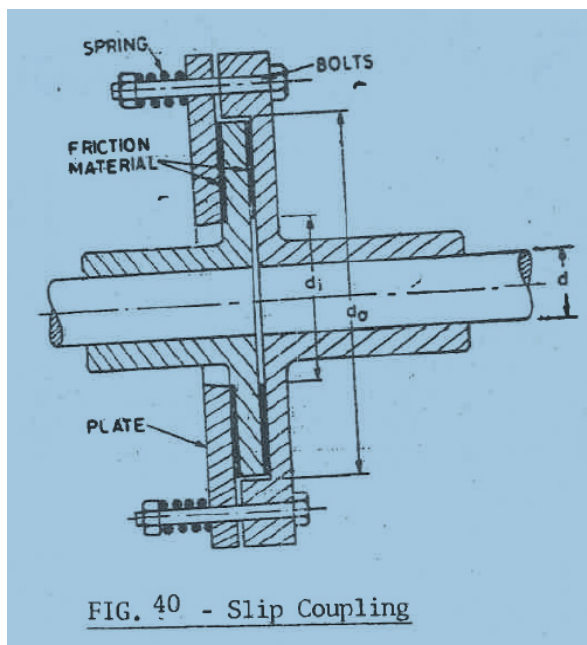
FIG. 40 - Slip Coupling

Two frictional discs having outside diameter d0 and inside diameter d1 are employed. Springs loaded by screwing down the nuts create the necessary pressure on the friction surfaces to ensure a constant maximum moment of friction transmitted by the coupling. When over load occurs i.e. when torque on the shaft exceeds the maximum friction moment of the coupling, the discs start to slip.

4. Alignment of Couplings
1. Introduction

Independent units connected with flexible couplings, such as motors, reducers or pumps, must be aligned when installed. This alignment operation, commonly called "coupling alignment", is actually the alignment of the centre-lines of the coaxial shafts the coupling connects. Alignment operations are performed on the coupling because it provides convenient surfaces at the shaft ends to use for this purpose.

Misalignment may be in any plane through 360 degrees. A practical method of brining centre lines into true alignment is to check and align them in two planes at right angles. The practice commonly followed, because it is convenient, is to align the shaft centre lines in the vertical and horizontal planes as illustrated in Fig. 41 and 42.
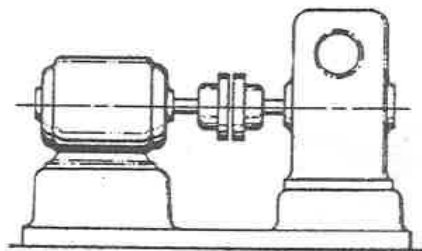
VERTICAL ALIGNMENT
MEASUREMENTS TAKEN IN A VERTICAL
PLANE, AT TOP AND BOTTOM OF COUPLING

HORIZONTAL ALIGNMENT
MEASUREMENTS TAKEN IN A HORIZONTAL
PLANE: PLANE, AT SIDES OF COUPLING

FIG. 41 Vertical Alignment   FIG.42 Horizontal Alignment

Because measurements and adjustments must be made to correct angular and parallel misalignment in both the vertical and horizontal planes, four operations must be performed. Coupling alignment, therefore requires the following four settings: vertical alignment (angular and parallel) and horizontal alignment (angular and parallel).

It was previously stated that alignment operations are performed on the coupling surfaces because they are convenient to use. An extremely important requirement when this is done is that these surfaces, and the shaft, run true. If there is any run-out of the shaft or the coupling surfaces, a proportionate error in alignment will be assembled into the coupling connection. While measurements may indicate true alignment, if the surfaces from which measurements are made are of true, then the alignment will also be out of true. Therefore, prior to any alignment measurements and adjustments, the condition of the shaft and coupling surfaces should be checked and run-out eliminated if present. If run-out correction cannot be made, the alignment of the coupling must be done with an indicator and both shafts must be rotated while making measurements.-This procedure is explained in the "indicator coupling alignment" section.

When preparing to align a coupling, it must be determined which unit is to be adjusted - the driver or the driven unit. Common practice, and the one generally recommended, is to position, level, and secure the driven unit at the required elevation. Then to adjust the driver to align with it. Connections to the driven unit, such as pipe connections to a pump, or output shaft connections on a reducer, should be completed prior to proceeding with coupling alignment. The driven unit must be set with its shaft centre line slightly higher than the driver to allow for alignment shimming. Another important preparatory step is to check the footing of the driver. If there is any rocking motion, the open point must be shimmed, since the driver must rest solidly on the base plate.

2. ALIGNMENT OPERATIONS

Two methods of coupling alignment are widely used - straight-edge thickness gauge method, and indicator method. Regardless of the method used, the four alignment operations previously discussed must be performed in a specific order. Only when performed in the correct order can adjustments be made at each step without disturbing prior settings. The four operations or steps in coupling alignment must therefore be made in the following order.

Step 1 - Vertical Angular (Face) Alignment
The first adjustment is made to correct angular misalignment in the vertical plane. This is accomplished by tipping the driver as required. The gap at the top and bottom of the coupling is measured, and adjustment is made to bring these faces true. Fig.43 shows the relation of the coupling faces and where the measurements are made.
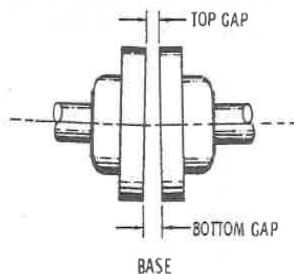
FIG. 43 - Vertical Angular Alignment

Step 2 - Vertical Parallel (Height) Alignment

This adjustment corrects parallel misalignment in the vertical plane. The driver is raised to match the driven unit without changing its angular position. Centre line height difference between driver and driven unit is determined by measuring on the OD of the coupling at top and/or bottom as illustrated in Fig. 44.
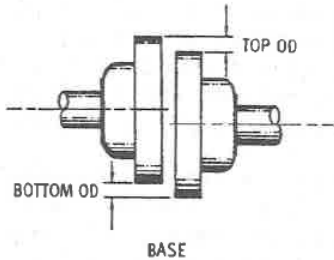
FIG. 44 - Vertical Parallel Alignment

Step 3 and 4 Horizontal Angular and Parallel (Face and OD Alignment)

When the driver has been brought into vertical alignment with the driven unit, shimming is complete. The horizontal alignment operations may then be done simultaneously. The driver is moved as required to obtain a uniform gap and to align the OD surfaces of the coupling. Measurements for these settings are made at the sides of the coupling as shown in Fig. 45.
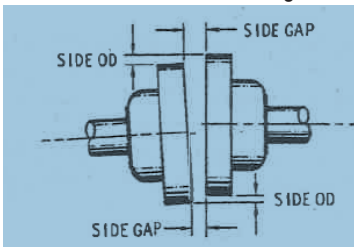
FIG. 45 - Horizontal Angular and Parallel Alignment

The initial alignment step, vertical face alignment (Step 1) can be a time-consuming operation when done in an unorganized manner. Correction of angular misalignment requires tilting one of the units into correct position by using shims. Selection of correct shim thickness is commonly made by trial and error. Much time can be saved, and accuracy gained, by using simple proportion to determine the shim thickness.

The tilt required, or the angle of change needed, at the base is the same as the angle of misalignment at the coupling faces. Because of this angular relationship, the shim thickness is proportional to the misalignment. For example in Fig.46_ the angle of misalignment at the coupling faces is 0.006 inch in 5 inches, therefore, each 5 inches of base must be tilted .006 inch to correct misalignment. As the base length is twice 5 inches, the shim thickness must he twice 0.006 inch or 0.012 inch
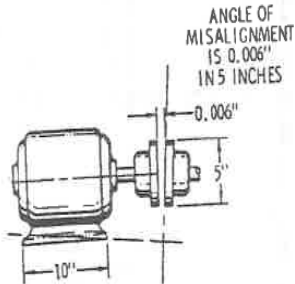
FIG. 46 - Correcting Misalignment by using Shims

A simple rule for shim thickness selection that gives extremely accurate results and enables correction to be made in one setting is: Shim thickness is as many times greater than the misalignment as the base length is greater than the coupling diameter.

4.2.1. Straightedge - Thickness Gauge Method

Practically all flexible couplings on drives operating at average speeds will perform satisfactorily when misaligned as much as 0.005 inch. Some will tolerate much greater misalignment. Alignment well within 0.005 inch is easily and quickly attainable using the straightedge and thickness gauge when correct methods are followed.

For vertical angular (face) alignment, use a thickness gauge to measure the width of gap at the top and bottom between the coupling faces. Using the difference between these two measurements, determine the shim thickness required to correct alignment. (It will be as many times greater than the coupling diameter). Shim under the low end of the driver to tilt it into alignment, with the driven unit.
ff
For example, assume that measured alignment is 0.160 inch minus 0.153 inch equalling 0.008 inch misalignment in 5 inches. See example illustrated in Fig. 22 Base length is about 2 1/2 times coupling diameter; therefore, the shim required is 2 1/2 times 0.008 inch, or 0.020 inch. A 0.020 inch shim placed under the low end of the driver will tilt into approximate angular alignment with the driven unit. (Slight error is well within tolerance of flexible-coupling adjustment).
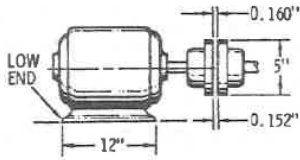
FIG. 47   Step 1 in the Straight
           edge Alignment Process

For vertical parallel (height) alignment use a straightedge and thickness gauge, measure the height difference between the driver and driven unit on the OD surfaces of the coupling. Place shims at all driver support points equal in thickness to the measured height difference. This is illustrated in Fig. 48
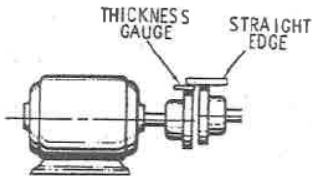
FIG. 48 - Step 2 in the Straight
          edge Alignment Process

For horizontal angular and parallel (face and OD) alignment use a straightedge as shown in Fig.49 Check alignment of OD's at sides of coupling. Using a thickness gauge as in Fig.47 check the gap between coupling faces at the side of the coupling.
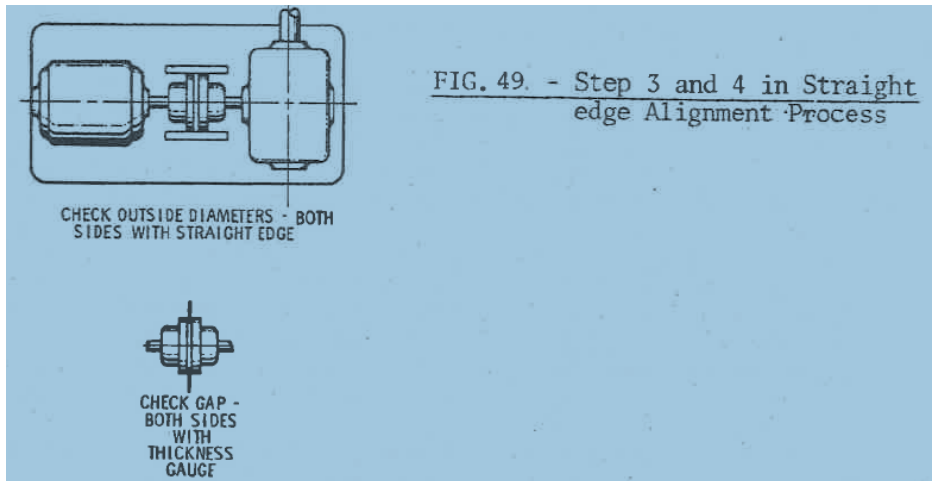
FIG. 49. - Step 3 and 4 in Straight
           edge Alignment Process

Adjust driver as necessary to align the OD's at the side and set the gap between faces equal at the sides. Do not disturb shims.

4.2.2 Measuring Misalignment with an Indicator

When an indicator is used as a measuring instrument to align coaxial shafts, it must be attached to the shaft or coupling of one of the units so it can be rotated. A stationary indicator will only show run-out, when in contact with an object being rotated. When the indicator is rotated, its tip will describe a circle which is concentric with the shaft bearings. The mating unit shaft should also be rotated as the point contacted by the indicator tip will also describe a true circle. When these two circles coincide (zero run-out on the indicator) the shafts will be in alignment in the plane being measured. See Fig.50.
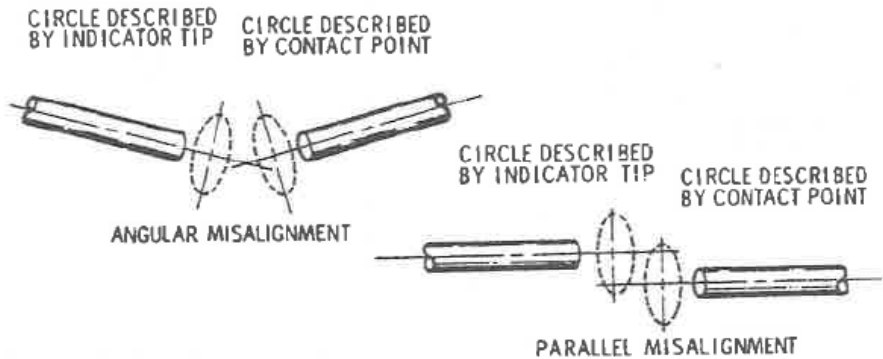
FIG. 50 - Using an Indicator to Measure Misalignment

While the ideal practice is to rotate both units while indicating this is sometimes impossible. In such cases, the indicator is attached to the unit that can be rotated and the mating unit remains stationary. When this is done the surfaces of the coupling half remaining stationary must be perfectly true. Any error in its surfaces will result in a corresponding error in shaft alignment.
If both units can be rotated while measuring, coupling surface run-out does not affect accuracy of alignment. This is an advantage of the indicator alignment method since it is possible to accurately align shafts in spite of coupling surfaces that are not running true.
As with the straightedge-thickness gauge method, the coupling is aligned in two places at right angles - usually the vertical and horizontal planes. Again four settings are made: angular and parallel settings in both the vertical and horizontal planes.

Step 1 - Vertical Angular (Face) Alignment

Attach indicator to driver shaft or coupling half. Place indicator tip in contact with face or the driven unit coupling half. Rotate shafts of both units together. Note indicator readings at top and bottom. Total indicator run-out is a measure of the vertical angular alignment. Place shims under driver at low end, tipping it into alignment with driven unit. The shim thickness will be as many times greater than the coupling face misalignment as the driver base length is greater than the coupling diameter.
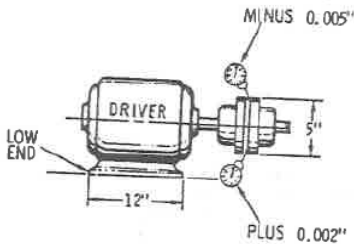
FIG. 51 - Step 1 in using the Indicator Alignment Process

An example is illustrated in Fig. 51. Assume that total indicator run-out is 0.007 in (minus 0.005 in to plus 0.002 in) as shown. The driver length is 2 1/2 times the coupling diameter 12 in to 5 in) so shim thickness should be 2 1/2 times 0.007 inch or 0.017 in.

Step 2 - Vertical Parallel (Height) Alignment

Place indicator tip in contact with outside surface of driven unit coupling half. Rotate shafts of both units together (Fig. 52) Note reading at top and bottom. Height difference is one half total indicator
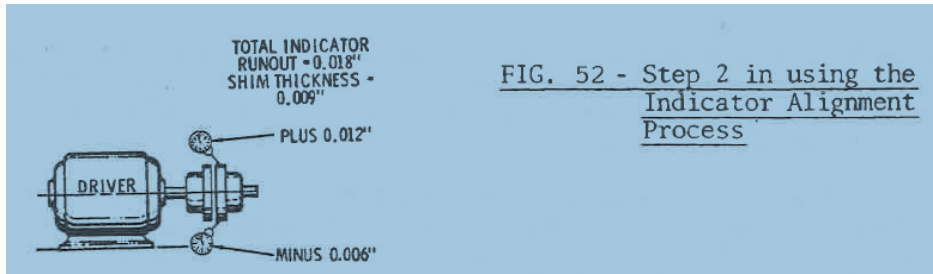
FIG. 52 - Step 2 in using the Indicator Alignment Process

run-out. Place shims at all driver support points equal in thickness to one half total indicator run-out. Example is illustrated in Fig. Assume the total indicator run-out is 0.018 in (plus 0.012 in to minus 0.006 in) so the shim thickness is one half 0.018 in or 0.009 in.F
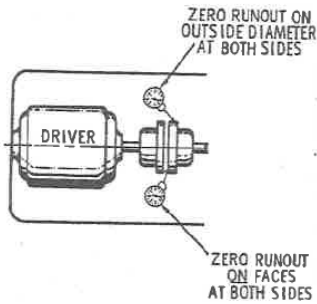
FIG. 53   Step 3 and 4 in using
          the indicator alignment
          Process

Step 3 and 4 - Horizontal Angular and Parallel (Face and OD) Alignment

Place indicator tip in contact with face of driven unit coupling half (Fig 53) Move driver to obtain zero reading on indicator.
Place indicator tip in contact with outside surface of driven unit coupling half. Move driver until zero reading is obtained. Repeat operations as necessary to obtain zero readings on both face and outside surfaces at sides of coupling.

2.3 TEMPERATURE CHANGE COMPENSATION

To compensate for temperature difference between installation conditions and operation conditions, it may be necessary to set one unit high, or low, when aligning. For example, centrifugal pumps handling cold water, and directly connected to electric motors, require a low setting to compensate for the expansion of the motor housing as its temperature rises. If the same units were handling liquids hotter than the motor operating temperature, it might be necessary to set the motor high. Manufacturers' recommendations should be followed for initial setting when compensation for temperature change is made at a cold installation.

The final alignment of equipment with appreciable operating temperature difference should be made after it has been run under actual operating conditions long enough to bring both units to operating temperatures. When these temperatures have been reached, the equipment should be stopped, the coupling immediately disconnected, and alignment checked and adjusted as necessary.

Using the preceding step-by-step methods, a coupling may be aligned in one direction at a time without one operation upsetting another. However, loosening the fasteners, adjusting and retightening, may have some effect on prior settings. In most cases any change would be less than the 0.005 inch tolerance of common flexible coupling drives. In the event a drive may require extreme accuracy, this may be attained by repeating the alignment steps as necessary.

GEARS

GEARS

- Gears are used to transmit power and motion
- used when centre distances are relatively short
- used when transmitting power from parallel, intersecting or angular shafts.

1. Spur Gear

It can be used for connecting shafts whose axes are parallel. The teeth are straight and parallel to the shaft axis.

Kis.
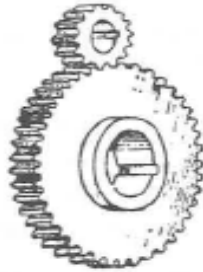
Fig. 54

2. Rack

This is a spur gear of infinite diameter or sometimes called a straight line gear.

3. Helical Gear

This can be used for connecting shafts whose axes are parallel. The teeth are inclined to the axis. The principal advantage is the quiet smooth action that results from the sliding contact of the meshing teeth.

4. Bevel Gear

It transmits motion between-angular or intersecting shafts cone shape.
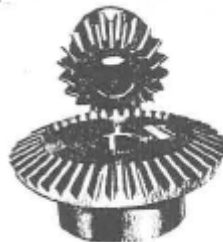
Fig. 56

5. Worm and Worm Wheel

This is used when hi4h ratio speed reduction is required. Motion is transmitted between shafts at right angles. The worm is always the driver and the worm gear the driven member. The ratio of a mating worm and worm gear is found by dividing the number of teeth in worm gear by the number of threads in the worm. The number of threads in a worm is determined by counting the number of starts at the end of the worm.

Gear Calculations

No. of teeth on driver   Speed of driver
                            =
No. of teeth on driven   Speed of driven

Gear Box
- Housing to support bearings, hold lubricant and to protect the components.
- Shafts for mounting the gears
- Bearings to support shafts
- Spacers to maintain distance between the gears and the bearings

In the figure below, speed change is obtained by using sliding gears. Three speeds are possible.
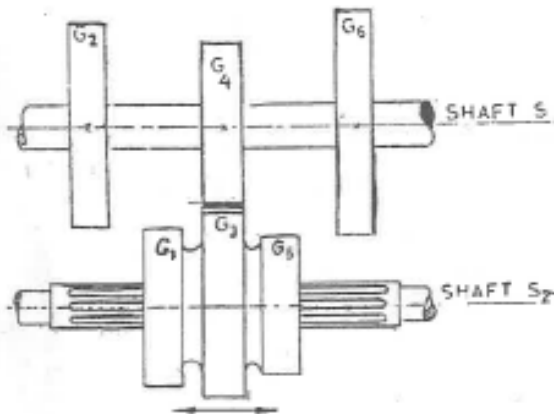
Fig. 58

BEARINGS

BEARINGS

1. Types and Operation

Bearings enable machine parts to move smoothly and accurately.

Bearings are used in machinery to keep moving parts accurately positioned to limit wear, and to allow easy movement with little friction.

Usually the movement is rotational: either a shaft turns inside the bearing, or the bearing rotates around a shaft. In some cases the movement may be linear, with the shaft moving endways through the bearing. Bearings used in a particular application must be able to handle the kind of machine movement involved.

Bearing Loads

Loads can push sideways or endways on a bearing.

The load on a bearing is the total of all the forces pressing on it. There are two kinds of load forces: axial (or thrust) and radial.

If the force is in line with the shaft, it is an axial or thrust load. A boat propeller always puts an axial or thrust load on its bearing equal to how hard it is pushing the boat.
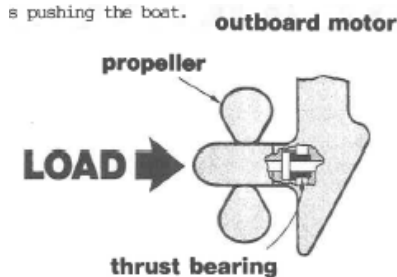
Fig.59 : Axial or Thrust Load

If the force is at right angles to the shaft, it is a radial load. A belt over a pulley puts a purely radial load on the shaft bearing, since there are no endways forces involved. This is the kind of load most bearings are intended to take.

Some bearings have to bear both kinds of load. An automotive wheel bearing, for example, always bears radial loads - the weight of the car, and acceleration and braking forces. When the car goes around a corner the wheel bearings must take an axial load too.

Fig. 60   Radial Load

43
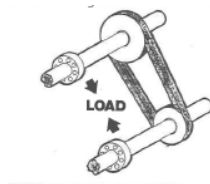
Varieties of Bearings

Plain and rolling element bearings do the same job, but work completely differently.

There are two basic kinds of bearings - plain bearings and rolling element bearings. Both kinds can be used to bear radial loads or thrust loads, and to allow linear or rotational movement. The bearings used in a particular application will depend upon such things as the size of the load and the operating speed of the machine.

2. Avoiding Bearing Failure
Most bearing failure is preventable.

If a designer chooses the right bearing for an application, and if all operating conditions are right for the bearing, it will last a long time, whatever kind it is. The main goal of bearing maintenance is to ensure proper operating conditions.

Storage and Handling

Even before a bearing is installed, correct maintenance procedures are important. Bearings must not be contaminated or damaged when stored, shipped or handled.

Rust
Rusty or dirty bearings will not last.

The slightest moisture, or even the humidity in the air, can rust rolling element bearings. The steel backing shell on some plain bearings will also rust.
. Handle bearings as little as possible before actual installation.
. Keep rolling elements and any other steel parts coated with grease, oil, or a protective film of some kind.
. A protective film makes dirt very likely to stick. Keep all bearings well wrapped.

Fretting Corrosion
Vibration causes fretting corrosion.

If the rolling elements in a bearing are vibrated in one spot on the races, the lubrication film will be pushed aside. The metal-to-metal contact will cause fine particles of steel to be worn off. These rust and grind away the surface as the vibration continues. The eventual result is depressions in the races and flat spots on the rolling elements. To avoid fretting corrosion.
. Lay rolling element bearings flat on their shelves. If they stand up, the weight of the inner race is supported by a few rolling elements at the bottom, which will be more subject to fretting corrosion.

• Do not store rolling element bearings near vibrating machinery, and do not ship them in a way that will vibrate them.

Installation
Bearing damage during installation is common.

Installation often involves applying large forces to bearings. Without careful preparation, and without the right equipment and installation procedures, bearings can easily be ruined during installation.

1. Removing Failed bearings
Do not ruin parts removing the old bearing.

If an old bearing must be removed, remove it carefully. Use the proper pulling or pressing equipment so that the shaft the bearing fits on, and the bore that it fits into, are not damaged. Do not use a hammer, grinder, or torch. When the bearing is free, inspect it for the kind of failure it experienced.

Many machines have several bearings supported a single rigid, rotating part. If one bearing goes bad, it may be necessary to replace them all. One new, tight bearing may take more than its share of the load and wear abnormally.

2. Preliminary Checks

Whatever kind of bearing is being 'installed, the shaft which will go through and the bore it will fit into must be accurately machined or ground to size. Parts must fit together properly.

Bearing races must be held in place firmly.

Rolling element bearing races must fit tightly in their housings and on their shafts. If a race is too loose, it may creep around and gradually wear looser. Replacement bearings will also be loose if the shaft or housing has worn. In sane applications a special cement may help, but replacing or building up and re-machining the shaft or housing is the only permanent solution.

Looseness also allows a race to vibrate in its housing or on its shaft. This can cause fretting corrosion of the races and the shaft or housing. The surfaces will be rusty looking and eroded. If a race is tight enough, vibration will not move it and fretting corrosion will not occur.

Bearings must also fit tightly so that heat will be transferred out of them as quickly as possible.

The life of a plain bearing depends on the shaft inside it.

The condition of the shaft or journal is particularly important to a plain bearing. Any roughness will wear away the bearing metal. But the shaft surface should not be too finely polished, or the lubricant film will not stick to it.

All parts should be cleaned, deburred and lubricated before a bearing is installed. Shafts and bores must not be out-of-round, tapered, worn, scored, rusted or corroded.

3. Installing One-Piece Plain Bearings

- Line up oil or grease feed holes with the corresponding holes in the bore.
- Use the proper driver or press adapter so that the bearing is not damaged by the installing force.
- Press or drive squarely, keeping the bearing aligned with the bore.

4. Installing Half Shell Plain Bearings

- Mark the cap so it can be reinstalled the same way it came off. Some bearing caps will go on turned backwards, but may not fit exactly. They will either distort the bearing or break when torqued down.
- Locate the oil or grease holes correctly and install the bearing shells in the cap and housing base with the locating pin, dowel, screw or tab in the proper position. If the shells are different, be sure the right one goes in the base and cap.
- Do not file or whim the bearing shells or the cap (unless shims under the cap are provided in the original installation).
- Lubricate the bearing; clean and lubricate the shaft or journal.
- nstall the shaft or journal in the bearing.
- Replace the cap in the correct position and torque the bolts to specifications.

5. Installing Rolling Element Bearings

There are various methods that can be used to ensure that the races of a rolling element bearing are fixed tightly to shafts and bores. Whether bearings are fitted by pressing, by heat shrinking or with race locking devices, good technique is essential.

- **Pressing.** Press fitting requires precision and cleanliness
- Use the right press equipment. Using a hammer may dent, chip, nick, or crack the bearing and lead to quick failure.

- Whatever pushes on the bearing during pressing must not leave dirt, chips, burrs or splinters in it. The bearing must be kept completely clean throughout the installation procedure.
- Press equipment must push straight and equally all around on a plain bearing, or on a rolling element bearing race. If a bearing is pressed crooked, it may jam, or be permanently distorted. Clearances will be uneven and loads will be concentrated at certain points.
- Press on the correct race. The press force, a heavy axial load, must not be transferred from one race to the other through the rolling elements. This is the most common cause of brinelling, a condition where the races and rolling elements are permanently dented.
- Like a press, heat is a useful tool, but it can damage a bearing.
- Heat shrinking. Sometimes a bearing must be heated to expand it for installation over a shaft. There are several precautions:
- Heat the bearing evenly in an electric oven, an induction heater, under a heat lamp, or with some other arrangement that will not apply concentrated heat. Do not use a torch.

Figure 61: Misaligned Press

- Do not overheat. The grease and seals may be damaged by high temperatures. Usually 250°F (121°C) or 275°F (135°C) is a hot as the bearing can safely tolerate for a brief period.
- Hold the bearing in place while it cools so that it does not shrink back from the shoulder which locates it on the shaft.

Set screw locking devices requires snug fit between race and shaft.

Race locking devices. Bearing races may be locked to the shaft with set screws and various kinds of collars.
- Under tightening set screws can allow movement and loosening particularly if the fit between race and shaft is not quite tight enough. Overtightening can gouge the shaft or other parts, and make bearing removal difficult. Large set screws should be torqued to specifications.
- Eccentric collars should be turned in the direction of normal shaft rotation before locking the collar in place with its set screw.

Tapered adapter sleeves wedge between the bearing's inner race and the shaft. Tighten the nut on a tapered adapter sleeve to specifications. Overtightening can expand the inner race and preload the bearing by decreasing the clearance between the races.

INCORRECT

press ram

press adapter

bearing

shaft

CORRECT

press adapter

INCORRECT

press ram

press adapter

bearing

pulley
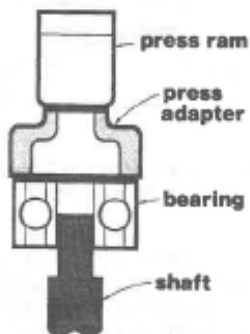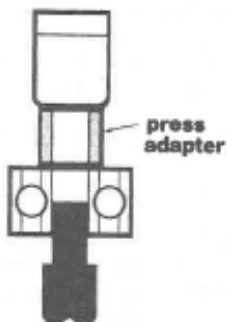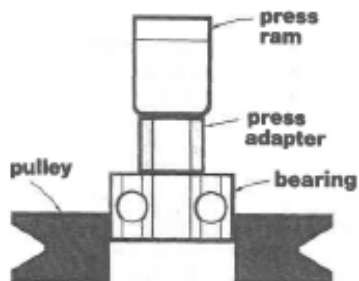
CORRECT
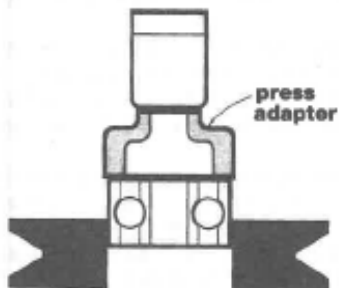
press adapter

Figure 62 Use the Correct Press Adapter to Avoid Pressing on the Wrong Race

Any bearing needs a little internal operating clearance.

6. Checking Operating Clearances

Part of installing any kind of bearing is checking to be sure that both radial and axial clearances of the installed bearing are within specified limits.
At the very least, shafts or bearings should be turned by hand to make sure there is no binding at any point. More accurate checking may be necessary.
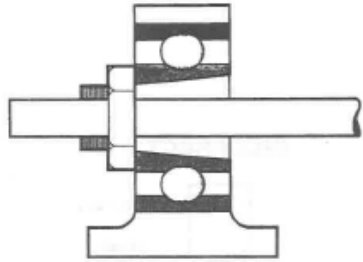
Figure 63: Overtightening a Tapered Adapter Can Distort a Bearing

Axial clearance. Axial clearance can be checked with either a dial indicator or with a feeler gauge:
• If a bearing controls shaft endplay, its axial clearance can often be checked with a dial indicator. Install the indicator with its plunger parallel to the shaft, and resting on a shoulder or flange. Prying the shaft back and forth will give a reading for shaft endplay and bearing axial clearance.
• The axial clearance of a plain bearing can sometimes be checked with a feeler gauge inserted between the bearing thrust flange and the thrust shoulder on the shaft.

Radial clearance. There are a number of ways to check radial clearance:

• The clearance between the rolling elements and the races may be checked directly with feeler gauges in some large rolling element bearings. Simply slide the feeler between the rolling element and the race. Do not roll the bearing over the feeler gauge.
• If there is nothing else besides the bearing holding the shaft from moving radially, a dial indicator will work. Mount it at right angles to the shaft and pry the shaft back and forth towards the plunger and away from it. Turn the shaft and measure the clearance again two or three times in order to check for an out-of-round shaft. If the dial indicator can be mounted in several positions around the shaft, it will detect an out-of-round bearing.
• In a plain bearing the radial clearance is the inside diameter of the bearing minus the shaft outside diameter. These dimensions can be measured directly with micrometers before the shaft is installed in the bearing.
• Radial clearance in a plain journal bearing can also be measured with a lead wire or a string of soft plastic clamped between bearing and journal. Both will squash to the clearance in the bearing. Measuring the thickness of the lead wire with a micrometer, or comparing the width of the plastic string to a chart, will give the bearing clearance.

7. Adjusting Bearing Clearances

In same applications, it is possible to adjust the bearing clearances. Adjustment should be made carefully, however, because overtightening will preload a bearing.

• Journal bearings sometimes have shims under the cap. Removing one or more shim will tighten the bearing across one diameter.
• Tapered roller bearings and angular contact ball bearing pairs are often adjustable with shims, spacers, or a nut. Both axial and radial clearances will be altered by tightening the nut or removing shims.
• Other kinds of bearing pairs require careful axial positioning to make sure they are not thrust loading each other. Before bolting than finally in place, make sure that each bearing is centred in its axial clearance at operating temperature.
• Plain bearings sometimes must be fitted to a shaft or journal by scraping, shimming, burnishing, expanding, reaming or line boring. Whatever the method, the operating clearance in the bearing must be adequate for a lubrication film - usually at least half a thousandth of an inch.
• Some plain bearings, when first put into service, need a "run-in" period for high points on the bearing to wear off. If possible loads should be kept light during this period, or the bearing metal on the high points may be overstressed and melt before it has a chance to wear off normally.

8. Aligning Bearing and Equipment

To avoid preloads, shafts and bearings must be properly aligned.

Some bearings are self-aligning. In spherical roller bearings, and sane ball bearings, the outer race can swivel, or change its orientation with respect to the rolling elements and the inner race.
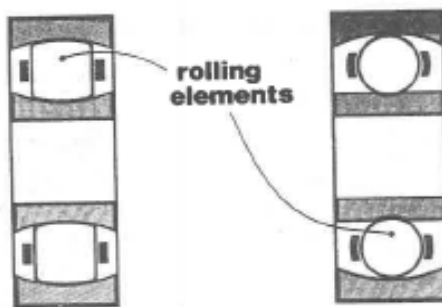
Figure 64: Spherical Roller Bearing and Ball Bearing with Spherical Outer Race

Many pillow block and flange block bearing units are also made to be self-aligning. The bearing itself can swivel in the housing. Since bearings are fairly tight in their housings, however, some tapping on the housing may be necessary to help the unit self-align when it is first installed.

Self-aligning bearings do not automatically solve all alignment problems. Shaft couplings must be properly aligned, or the shaft bearings will be overloaded, whether they are self-aligning or not. If it is necessary to pull a shaft to one side to couple it or to put it through a bearing, something is misaligned and should be corrected.

Individual bearing housings must be solidly supported.

To bring shafts or couplings into alignment, it may be necessary to move a motor or part of a machine to one side or the other, or to shim individual bearing housings or whole machines.

Be sure to support the whole machined surface of the pillow block or flange block housing with a completely flat surface to avoid distorting or breaking the housing.

9. Installing Seals
- To keep dirt out of bearings and grease or oil in, seals must be in good condition. Felt seals and lip seals wear out, and may need to be replaced when a bearing is replaced.
- Remove the old seal carefully. Do not scratch or nick the shaft or housing bore.,
- Lubricate the seal and the shaft surface before installation.
- Be sure the seal faces the right direction.
- Protect the seal from sharp keyways, splines, or shoulders during installation.
- Press or drive the seal squarely with the proper seal driver or press adapter.
- The shaft where the seal rubs must be smooth, without rings or grooves. Use a sleeve if necessary.
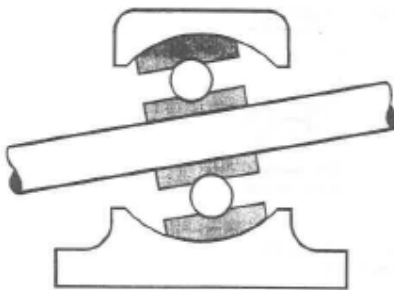
Figure 65 Self-Aligning Pillow Block
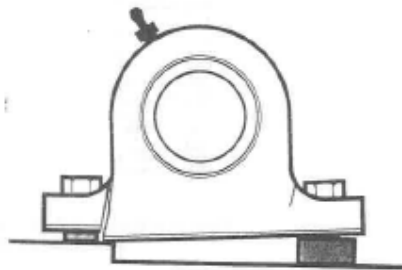
Figure 66 Shimming With Washers Can Damage a Pillow Block

3. TROUBLESHOOTING

The first goal of troubleshooting is to detect failing bearings before they have damaged other parts or caused expensive downtime. The second job of the troubleshooter is to determine why a bearing failed, so that the conditions that caused it can be corrected.

Symptoms of Failing Bearings

If bearing problems are caught while the equipment is still operable, repairs can often be scheduled between shifts or at production line changeovers.

1. Vibration and Noise
Any noise or shake indicates a problem.

A worn plain bearing may allow parts to knock or thump audibly, or vibrate visibly.

Rolling element bearings produce vibrations of many different frequencies from ultrasound though high-pitched squeals to whines, growls, and low-frequency grinding noises. Listening devices, vibration analyzers and ultrasound detectors are all useful in diagnosing bearing condition.

2. Looseness or Roughness

Sometime a bearing can be turned by hand, and bad spots detected by feel. Pushing and pulling on the shaft may reveal looseness, but checking the clearances with a feeler gauge or a dial indicator is more conclusive.

3. Hot Operation

Measuring bearing temperature frequently can spot problems early.

Normal operating temperature may be considerably higher than nearby air temperature. In some special applications bearings may run normally at 300°F (149°C) or even hotter. But normal operating temperatures in most bearings will be under 200°F (93°C). Any rise above normal indicates some problem has increased friction in the bearing.

Failed Bearing Appearance
Close examination of a failed bearing can often reveal the cause of its failure.
1. Spalling
Spalling leaves a very rough surface on races and rollers.
Any loaded rolling element bearing will eventually fail from spelling. The steel breaks out, leaving a very rough surface on races and rolling elements. Premature spelling is the result of overloading.

Spelling occurs first where loads are heaviest.

Once spalling starts, total bearing failure is usually rigid. The steel chunks circulating in the bearing destroy what remains of the smooth finish on races and rolling elements.
- It is often possible to tell how a bearing was overloaded from the way it is spalled:
- Spalling at the edges of the races usually indicates excessive thrust loads.
- Spalling in the bottom of ball bearing races indicates heavy radial loads.
- If a stationary race is equally spalled all the way around, either the race must have been creeping around on the shaft or in the bore, or the load must have been rotating. Possibly something was out of balance.
- If half of a stationary race is spalled on one side, and half is spalled on the other side, misalignment is the probably cause.

The rotating race in a misaligned bearing may show spalling in a wide band all the way around. Spalling also sometimes occurs in plain bearings. The soft bearing metal is forced to deform too many times and eventually breaks loose. Shock loads, or an out-of-round shaft can be the problem.
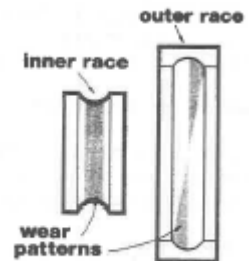
Figure 67. Misalignment Failure

2. Fretting Corrosion

Fretting corrosion is caused by vibration.

Fretting corrosion can occur while bearings are not in operation and while they are running. In either case, it is caused by vibration.

Fretting corrosion of the races and rolling elements, which occurs while the bearings are not operating, leaves small depressions where the races and rolling elements were in contact. The depressions may be part way around the races, or on the side of the races only, depending on the position of the bearing and the stationary load on it.

This type of corrosion can occur in new bearings during shipping or storage before they are ever installed. It may also occur in bearings already in machinery that is not operating.

Shipping the machinery, or subjecting it to vibration from nearby operating equipment, can be responsible. Turning unused equipment by hand periodically can reduce the likelihood of this kind of fretting corrosion.

Fretting corrosion between a race and the shaft in it or between a race and the housing bore around it is caused by vibration that occurs while bearings are running. There may be a rusty-looking powder or paste present, and the surfaces will be eroded.

The vibration which causes this kind of fretting corrosion may be a normal result of the operation of the equipment. Wheel bearings on vehicles, for example, are often subjected to enough vibration stresses to cause it. In other applications, out-of-balance rotating parts may be responsible.

Tight race fits and balanced parts reduce fretting corrosion of races, shafts and housings.

3   Heat Damage
Most failed bearings will show the effects of heat.

Almost all bearing failure involves excessive heat. If a bearing has been running hot for some reason, the lubricant and the metal itself may show damage.

The soft metal of plain bearings often has a fairly low melting temperature. Partial softening or melting usually leaves the bearing metal streaked or smeared looking. In extreme cases, it will completely melt out, allowing the shaft or journal to contact the steel backing or housing steel. Extensive damage results.

The steel in rolling element bearings can take higher temperatures without melting. But races may show dark streaks where the steel has been heated so much that it has lost its heat treatment and hardness.

Both kinds of bearings may have varnish deposits from overheated lubrication, and the grease left in the bearing may be dark and dried out looking.

4.   Brinelling
A bearing not in operation can be damaged by a smaller load than one that is running. Brinelling sometimes looks like fretting corrosion in that there is a regular pattern of depressions in the races, and flat spots on the rolling elements. The races and rolling elements have not worn away on each other, however, but have been pressed together so hard that they have permanently dented each other. It occurs when a heavy load is imposed on a bearing that is not turning.
The position of the dents in the races can sometimes indicate the direction of the load. Dents on the side of a ball bearing race, for example, indicate a heavy thrust load. The most probably cause of this is pressing on the wrong race during installation.
Brinnelled bearings, like bearings with fretting corrosion, may "sing" or whine with a tone that depends on operating speed.

5. Pitting
Electric current damages bearings.
Current passing through a rolling element bearing often leaves pits in the races. These pits can look like the early stages of spalling although sometimes there is a regular pattern to them. Occasionally the pits turn into regular grooves across the races.

Current can come from bad insulation in a motor, poor brush or slip ring contact, static discharge, or arc welding. High current bursts can weld a bearing solid.

6. Wear
Dirt will cause or speed we in all bearings
Rolling element bearings do not normally wear. But dirt or abrasives in the bearing or the lubricant will cause wear. Sometimes the race and rolling element surfaces may still be fairly smooth, but the bearing will be loose. The retainer may also be noticeably loose on the rolling elements.

Metal plain bearings will wear whenever the lubrication film is inadequate to prevent metal-to-metal contact. In extreme cases, the surface layer of bearing material may be worn through, exposing the steel backing. In this case, the shaft or journal will be noticeably roughened. The position of worst wear in a bearing indicates the direction of the load.

Plastic, graphite and graphite composition bearings are normally expected to wear. But unusually rapid wear can be caused by excessive dirt or other contamination in the bearing, as well as high loads or speeds.

<div align="center">

Lubricants, tools and fittings must be clean
Do not force dirt into a bearing when lubricating.
</div>

7. Rust and Corrosion
Bearings must be kept dry, both in storage and in machinery.

Rust on races or rolling elements roughen surfaces and interferes with frictionless rolling. Rust particles, as they wear off, are a particularly abrasive contaminant.

Although the steel backing shell of some plain bearings can rust, the bearing metal itself will not. It is however, subject to other kinds of corrosion. Acids in oxidized lubrication can eat it away.

Rust is due to moisture, which is probably the result of bad seals or an unusually wet operating environment. Moisture may also enter during lubrication.
Moisture will rust the shafts or journals in plain bearings, and greatly speed wear. The turning part in any plain bearing metal or plastic, must be very smooth for long bearing life.
To keep dirt and moisture out of bearings, be certain that all seals are in good condition.

Pitch

A word commonly used in connection with machinery and mechanical operations meaning: The distance from a point to a corresponding point.

Fig. 68

Pitch Diameter and Pitch Circle

The pitch diameter specifies the distance across the center of the pitch circle. Pitch diameter dimensions are specific values even though the pitch circle is imaginary. Rotary power transmission calculations are based on the concept of circles or cylinders in contact. These circles are called pitch circles.
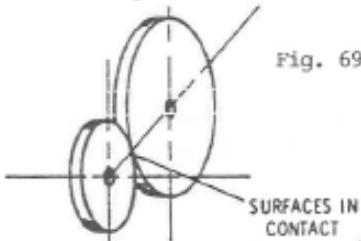
Fig. 69
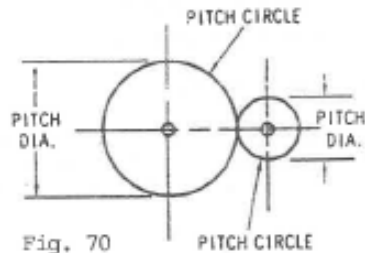
Fig. 70

As the shafts rotate, the surfaces of the pitch circles travel equal distances at equal speeds (assuming no slippage). Shafts then rotate at speeds proportional to the circumference of the pitch circles and therefore proportional to the pitch diameters.

This concept of pitch circles in contact applies to belt and chain drives although the pitch circles are actually separated. This is true because the belt or chain is in effect an extension of the pitch circle surface.

A. As rotation occurs the pitch circle surfaces will travel the same distance at the same surface speed.

B. Because the circumference of a 4" pitch circle is double that of a 2" pitch circle its rotation will be one half as much.

C. The rotation speed therefore of the 4"pitch circle will be one half that of the 2" pitch circle.
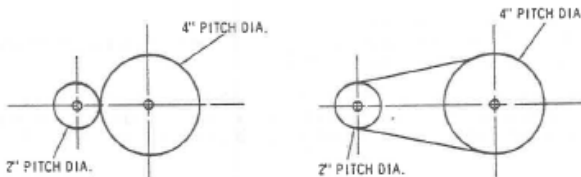
Fig. 71

Calculations

Rotational speed and pitch diameter calculations for belts, chains, and gears are based on the concept of pitch circles in contact. The relationship that results from this concept may be stated as follows:

Shaft speeds are inversely proportional to pitch diameters.
In terms of rotational speeds and pitch diameters this relationship may be expressed in equation form as follows:

$$\frac{\text{Driver Rotational Speed}}{\text{Driven Rotational Speed}} = \frac{\text{Driven Pitch Diameter}}{\text{Driver Pitch Diameter}}$$

To simplify the use of the equation, letters and numbers instead of words, are used to represent the terms.

S1 for Driver Rotational Speed
S2 for Driven Rotational Speed
P1 for Driver Pitch Diameter
P2 for Driven Pitch Diameter

The basic equation then becomes
$$\frac{S1}{S2} = \frac{P2}{P1}$$

The equation may be rearranged into the following forms, one for each of the four values. To find an unknown value the known values are substituted in the appropriate equation.

$$S1 = \frac{P2 \times S2}{P1} \quad S2 = \frac{P1 \times S1}{P2} \quad S2 = \frac{S2 \times P2}{S1} \quad P2 = \frac{S1 \times P1}{S2}$$

The above equations may also be stated as rules. Following these rules is another convenient way to calculate unknown shaft speeds and pitch diameters.

To Find -   Driving Shaft Speed: Multiply driving pitch diameter by speed of driven shaft and divide by driving pitch diameter.
To Find -   Driven Shaft Speed: Multiply driving pitch diameter by speed of driving shaft and divide by speed of driven pitch diameter.
To Find -   Driving Pitch Diameter: Multiply driven pitch diameter by speed of driven shaft and divide by speed of driving shaft.
To Find -   Driven Pitch Diameter: Multiply driving pitch diameter by speed of driving shaft and divide by speed of driven shaft.
In gear and sprocket calculations the number of teeth is used rather than the pitch diameters. The equations and rules hold true because the number of teeth in a gear or sprocket is directly proportional to its pitch diameter.

"V" BELTS

The "V' belt has a tapered cross-sectional shape which causes it to wedge firmly into the sheave groove under load. Its driving action takes place through frictional contact between the sides of the belt and the sheave groove surfaces. While the cross-sectional shape varies slightly with make, type and size, the included angle of most "V" belts is about 42 degrees. There are three general classifications of "V" belts: Frictional Horsepower, Standard Multiple and Wedge.

Fractional Horsepower

Used principally as single belts on fractional horsepower drives. Designed for intermittent and relatively light loads. Manufactured in four standard cross-sectional sizes as shown in Fig. 72.

Standard belt lengths vary by one-inch increments between a minimum length of 10 inches and a maximum length of 100 inches. In addition some fractional horsepower belts are made to fractional inch lengths.

The numbering system used indicates the cross-sectional size and the nominal outside length. The last digit of the belt number indicating tenths of an inch. Because the belt number indicates length along the outside surface, belts are slightly shorter along the pitch line than the nominal size number indicates.

Belt Number	Size	Outside Length
3L 470	3L	47"
4L 425	4L	42½"

FRACTIONAL HORSEPOWER

SIZE	DIMENSIONS
"2L"	1/4 5/32
"3L"	3/8 7/32
"4L"	1/2 5/16
"5L"	21/32 3/8

STANDARD MULTIPLE

SIZE	DIMENSIONS
"A"	1/2 5/16
"B"	21/32 13/32
"C"	7/8 17/32
"D"	1-1/4 3/4
"E"	1-1/2 29/32

WEDGE

SIZE	DIMENSIONS
"3V"	3/8 5/16
"5V"	5/8 17/32
"8V"	1 7/8

LENGTH RANGE OUTSIDE MEASURE

3L150	to	3L750
4L170	to	4L1000
5L230	to	5L1000

LENGTH RANGE INSIDE MEASURE

A26	to	A128
B35	to	B300
C51	to	C420
D120	to	D660
E144	to	E660

LENGTH RANGE PITCH LINE MEASURE

3V250	to	3V1650
5V500	to	5V3550
8V1000	to	8V5000

Standard Multiple

Standard multiple belts are designed for the continuous service usually encountered in industrial applications. As the name indicates, more than one belt provides the required power transmission capacity. Most manufacturers furnish two grades, a standard and a premium quality. The standard belt is suitable for the majority of industrial drives that have normal loads, speeds, center distances, sheave diameters, and operating conditions. The premium quality is made for drives subjected to severe loads, shock, vibration, temperatures etc.

The standard multiple "V" belt is manufactured in five standard cross-sectional sizes designated: "A", "B", "C", "D", "E" as shown in Fig 31B.

The actual pitch length of standard multiple belts may be from one to several inches greater than the nominal length indicated by the belt number. This is because the belt numbers indicate the length of the belt along its inside surface. As belt length calculations are in terms of belt length on the pitch line, a table of pitch line belt lengths is recommended when selecting belts.

Wedge

The wedge belt is an improved design "V" belt which makes possible a reduction in size, weight and cost of "V' belt drives. Utilizing improved materials, these multiple belts have a smaller cross-section per horsepower and use smaller diameter sheaves at shorter center distances than is possible with standard multiple belts. Because of the premium quality heavy-duty construction only three cross-sectional belt sizes are used to cover the duty range of the five sizes of standard multiple belts. The dimensions of the three standard "wedge" belt cross-sectional sizes "3V", "5V", "8V" are shown in Fig. 31 C.

The wedge belt number indicates the number of 1/8 inches of top width of the belt. As shown above the "3V" belt has a top width of 3/8 inches the "5V" a width of 5/8 inches and the "8V" a full one inch of top width.

The belt length indicated by the wedge belt number is the effective pitch line length of the belt. As belt calculations are in terms of pitch line lengths, nominal belt numbers can be used directly when choosing wedge belts.

"V" Belt Matching

Satisfactory operation of multiple belt drives require that each belt carry its share of the load. To accomplish this all belts in a drive must be essentially equal length. Because it is not economically practical to manufacture belts to exact length, most manufacturers follow a practice of code marking to indicate exact length.

"V" Belts

Fig. 73

Table 4. Standard "V" Belt Lengths

A BELTS — BELT NUMBER Standard	Pitch Length	Outside Length
A26	27.3	28.0
A31	32.3	33.0
A35	36.3	37.0
A38	39.3	40.0
A42	43.3	44.0
A46	47.3	48.0
A51	52.3	53.0
A55	56.3	57.0
A60	61.3	62.0
A68	69.3	70.0
A75	76.3	77.0
A80	81.3	82.0
A85	86.3	87.0
A90	91.3	92.0
A96	97.3	98.0
A105	106.3	107.0
A112	113.3	114.0
A120	121.3	122.0
A128	129.3	130.0

B BELTS — BELT NUMBER Standard	Pitch Length	Outside Length
B35	36.8	38.0
B38	39.8	41.0
B42	43.8	45.0
B46	47.8	49.0
B51	52.8	54.0
B55	56.8	58.0
B60	61.8	63.0
B68	69.8	71.0
B75	76.8	78.0
B81	82.8	84.0
B85	86.8	88.0
B90	91.8	93.0
B97	98.8	100.0
B105	106.8	108.0
B112	113.8	115.0
B120	121.8	123.0
B128	129.8	131.0
B136	137.8	139.0
B144	145.8	147.0
B158	159.8	161.0
B173	174.8	176.0
B180	181.8	183.0
B195	196.8	198.0
B210	211.8	213.0
B240	240.3	241.5
B270	270.3	271.5
B300	300.3	301.5

C BELTS — BELT NUMBER Standard	Pitch Length	Outside Length
C51	53.9	55.0
C60	62.9	64.0
C68	70.9	72.0
C75	77.9	79.0
C81	83.9	85.0
C85	87.9	89.0
C90	92.9	94.0
C96	98.9	100.0
C105	107.9	109.0
C112	114.9	116.0
C120	122.9	124.0
C128	130.9	132.0
C136	138.9	140.0
C144	146.9	148.0
C158	160.9	162.0
C162	164.9	166.0
C173	175.9	177.0
C180	182.9	184.0
C195	197.9	199.0
C210	212.9	214.0
C240	240.9	242.0
C270	270.9	272.0
C300	300.9	302.0
C360	360.9	362.0
C390	390.9	392.0
C420	420.9	422.0

Table 4. Standard "V" Belt Lengths (continued)

D BELTS	Pitch Length	Outside Length
D210	213.3	215.0
D240	240.8	242.5
D270	270.8	272.5
D300	300.8	302.5
D330	330.8	332.5
D360	360.8	362.5
D390	390.8	392.5
D420	420.8	422.5
D480	480.8	482.5
D540	540.8	542.5
D600	600.8	602.5

E BELTS	Pitch Length	Outside Length
E180	184.5	187.5
E195	199.5	202.5
E210	214.5	217.5
E240	241.0	244.0
E270	271.0	274.0
E300	301.0	304.0
E330	331.0	334.0
E360	361.0	364.0
E390	391.0	394.0
E420	421.0	424.0
E480	481.0	484.0
E540	541.0	544.0
E600	601.0	604.0

3V (No. / Length)		5V (No. / Length)		8V (No. / Length)	
3V425	42.5	5V850	85.0	8V1700	170.0
3V450	45.0	5V900	90.0	8V1800	180.0
3V475	47.5	5V950	95.0	8V1900	190.0
3V500	50.0	5V1000	100.0	8V2000	200.0
3V530	53.0	5V1060	106.0	8V2120	212.0
3V560	56.0	5V1120	112.0	8V2240	224.0
3V600	60.0	5V1180	118.0	8V2360	236.0
3V630	63.0	5V1250	125.0	8V2500	250.0
3V670	67.0	5V1320	132.0	8V2650	265.0
3V710	71.0	5V1400	140.0	8V2800	280.0
3V750	75.0	5V1500	150.0	8V3000	300.0
3V800	80.0	5V1600	160.0	8V3150	315.0
3V850	85.0	5V1700	170.0	8V3350	335.0
3V900	90.0	5V1800	180.0	8V3550	355.0
3V950	95.0	5V1900	190.0	8V3750	375.0
3V1000	100.0	5V2000	200.0	8V4000	400.0
3V1060	106.0	5V2120	212.0	8V4250	425.0
3V1120	112.0	5V2240	224.0	8V4500	450.0
3V1180	118.0	5V2360	236.0	*8V5000	500.0
3V1250	125.0	5V2500	250.0		
3V1320	132.0	5V2650	265.0		
3V1400	140.0	5V2800	280.0		
		5V3000	300.0		
		5V3150	315.0		
		5V3350	335.0		
		5V3550	355.0		

D BELTS	Pitch Length	Outside Length
D120	123.3	125.0
D128	131.3	133.0
D144	147.3	149.0
D158	161.3	163.0
D162	165.3	167.0
D173	176.3	178.0
D180	183.3	185.0
D195	198.3	200.0

3V BELTS (No. / Length)		5V BELTS (No. / Length)		8V BELTS (No. / Length)	
3V355	35.0	5V530	53.0	8V1000	100.0
3V385	38.5	5V560	56.0	8V1060	106.0
3V380	38.0	5V600	60.0	8V1120	112.0
3V300	30.0	5V630	63.0	8V1180	118.0
3V315	31.5	5V670	67.0	8V1250	125.0
3V335	33.5	5V710	71.0	8V1320	132.0
3V355	35.5	5V750	75.0	8V1400	140.0
3V375	37.5	5V800	80.0	8V1500	150.0
3V400	40.0	5V800	80.0	8V1600	160.0

Each belt is measured under specific tension and marked with a code number to indicate its variation from nominal length. The number 50 is commonly used as the code number to indicate a belt within tolerance of it nominal length. For each 1/10 of an inch over nominal length the number 50 is increased by 1. For each 1/10 of an inch under nominal length. 1 is subtracted from the number 50. Most manufacturers code mark as shown in Fig -73.

TYPICAL CODE MARKING

800 ... MANUFACTURER'S NAME ... 50

NOMINAL SIZE & LENGTH   LENGTH CODE NUMBER
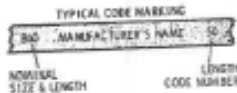
Fig. 74

For example, if the 60 inch "B" section belt shown in Fig. is manufactured 3/10 of an inch long, it will be code marked 53 rather than the 50 shown. Or if made 3/10 of an inch short it will be code marked 47. While both of these belts have the belt number B60 they cannot be used satisfactorily in a set because of the difference in their actual length.

It is possible for the length of belts to change slightly during storage. Under satisfactory conditions, however, changes will not exceed measuring tolerances. Therefore, belts may be combined by matching code numbers. Ideally, sets should be made up of belts having the same code numbers, however, the resiliency of the belts allow some length variation.
Table 5 lists the maximum recommended variations for standard multiple belts when making up matched belt sets.

Table 75

Matching Number Range	Belt Lengths				
	A	B	C	D	E
2	26-180	55-180	51-180		
3		195-315	195-355	120-268	144-240
4			270-360	270-360	270-360
5			390-420	390-660	390-660

"V" Belt Drive Alignment

The life of a "V" belt is dependent on: first the quality of materials and manufacture; and second on installation and maintenance. One of the most important installation factors influencing operating life is belt alignment. In fact, misalignment is probably the most frequent cause of shortened belt life.

While "V" belts because of their inherent flexibility can accommodate themselves to a degree of misalignment not tolerated by other types of power transmission, they still must be held within reasonable limits. Maximum life can be attained only with true alignment and as misalignment increases belt life is proportionally reduced. If misalignment is greater than 1/16 of an inch for each 12 inches of center distance, very rapid wear will result.

Misalignment of belt drives results from shafts being out of angular or parallel alignment, or from the sheave grooves being out of axial alignment. These three types of misalignment are illustrated in Fig. 33.
F
Because the shafts of most "V" belt drives are in a horizontal plane, angular shaft alignment is easily obtained by levelling the shafts. In those cases where shafts are not horizontal, a careful check must be made to ensure the angle of inclination of both shafts is the same.
Before any check is made for parallel-shaft and axial-groove alignment, the shaft and sheaves must be checked for run-out. Any shaft run-out or sheave wobble will cause proportionate inaccuracies in alignment.
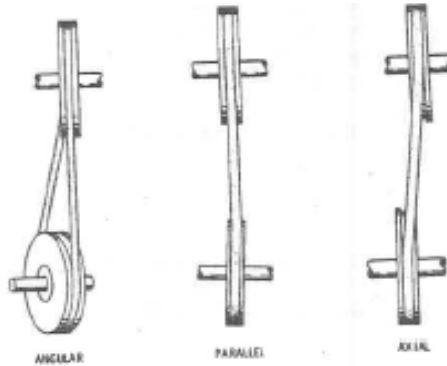
Fig. 76

The checking and adjusting for parallel shaft and axial-groove alignment of most drives can be done simultaneously if the shafts and sheaves are true.

"V" Belt Alignment

The most satisfactory method of checking parallel-shaft and axial-groove alignment is with a straightedge. It may also be done with a taut line, however, when using this method care must be exercised as the line is easily distorted. The straightedge checking method is illustrated in Fig 34, with arrows indicating the four check points. When sheaves are properly aligned no light should be visible at these four points.

Fig. 77

"V" Belt installation

"V" belts should never be 'run-on" to sheaves. To do so places excessive stress on the cords, usually straining or breaking sane of them. A belt damaged in this manner will flop under load and turn over in the sheave groove. The proper installation method is to loosen the adjustable mount, reduce the center distance, and slip the belts loosely into the sheave grooves.

The following six general rules should be followed when installing "V" belts:

1. Reduce centers so belts can be slipped on sheaves
2. Have all belts slack on the same side (top of drive)
3. Tighten belts to approximately correct tension
4. Start unit and allow belts to seat in grooves.
5. Stop - retighten to correct tension
6. Recheck belt tension after 24 to 48 hours of operation.

Checking Belt Tension

Belt tension is a vital factor in operating efficiency and service life. Too low a tension results in slippage and rapid wear of both belts and sheave grooves. Too high a tension stresses the belts excessively and unnecessarily increases bearing loads.

The tensioning of fractional horsepower and standard multiple belts may be done satisfactorily by tightening until the proper "feel" is attained. The proper "feel" is when the belt has a live springy action when struck with the hand. If there is insufficient tension the belt will feel loose or dead when struck. Too much tension will cause the belts to feel taut, as there will be no give to them.

Wedge Belt Tension

The 3V, 5V and 8V wedge belts operate under very high tension since there are fewer belts and/or smaller belts per horsepower. When properly tightened they are taut and have little give, therefore, tightening by "feel" is not dependable. A better method is to use the belt tension measuring tool shown in Fig 78.
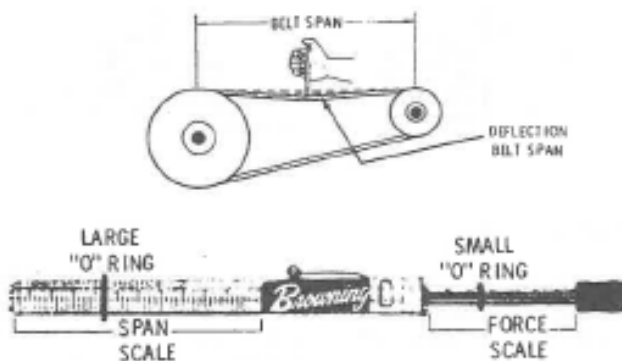
Fig. 78

POSITIVE-DRIVE BELTS

Positive-drive belts, also called timing belts and gear belts, carbine the flexibility of belt drives with the advantages of chain and gear drives. Power is transmitted by positive engagement of belt teeth with pulley grooves, as in chain drives, rather than by friction as in belt drives. This positive engagement of belt teeth with pulley grooves eliminates slippage and speed variations. There is no metal-to-metal contact and no lubrication required. See Fig. 79.

The positive-drive belt is constructed with gear-like teeth which engage mating grooves in the pulley. Unlike most other belts, they do not derive their tensile strength from their thickness. Instead, these belts are built thin, their transmission capacity resulting from the steel-cable tension members and tough molded teeth shown in Fig.80.

High-strength synthetic materials are used in positive-belt construction Fig. 8'1 shows the four basic components: neoprene backing, steel tension members, neoprene teeth with nylon facing. The nylon covering on the teeth is highly wear resistant and after a short "run in"" period becomes highly polished with resultant low friction. Belts are so constructed that tooth strength exceeds the tensile strength of the belt when 6 or more teeth are meshed with a mating pulley.

Exploded view of gear belt.

Fig. 80

Courtesy Browning Mfg. Co.

NEOPRENE TEETH

NYLON FACING

TENSION MEMBER

NEOPRENE BACKING
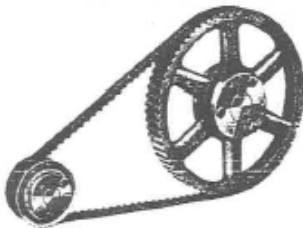
Courtesy Browning Mfg. Co.

.. A positive-drive belt; sometimes called a gear belt.

Fig. 79
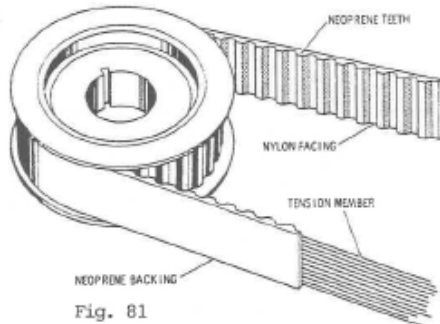
Fig. 81

... The four basic components used in the construction of the gear belt.

With positive-drive belts, as with gear and chain drives, pitch is a fundamental consideration. In this case circular pitch is the distance between tooth centers (measured on the pitch line of the belt or the distance between groove centers) measured on the pitch circle of the pulley as indicated in Fig. 82.

The pitch line of a positive-drive belt is located within the cable tension members. The pitch circle of a positive-drive pulley coincides with the pitch line of the belt mating with it. The pulley pitch diameter is always greater than its face diameter. All positive-drive belts must be run with pulleys of the same pitch. A belt of one pitch cannot be used successfully with pulleys of a different pitch.

Positive-drive belts are made in five (5) stock pitches. The following code system is used to indicate the pitch of positive drive system:

Code	Code Meaning	Pitch
XL	Extra Light	1/5 inch
L	Light	3/8 inch
H	Heavy	1/2 inch
XH	Extra Heavy	7/8 inch
XXH	Double Extra Heavy	1 1/4 inch

The standard positive-drive belt numbering system is made up of three parts: First, the pitch length of the belt, which is the actual pitch length multiplied by ten (10); Second, the code for the pitch of the drive; Third, the belts width multiplied by one hundred (100).

For example: 390L 100
390 - Pitch length - 39.0 inches
L - Pitch cf drive - Light - 3/8 inch pitch
100 - Belt width - 1 inch

PITCH ICIRCULAR PITCHI

BELT PITCH LINE

PITCH DIAMETER

OUTSIDE DIAMETER

PULLEY PITCH CIRCLE

*. Measurement to consider when using a gear belt.
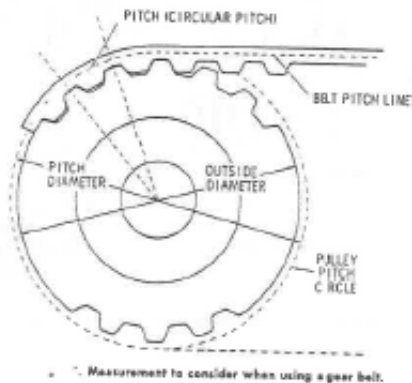
Fig. 82

The following list is the stock sizes of positive-drive belts carried by most manufacturers. In addition, much wider and longer drives can be furnished on special order.

Code	Pitch	Stock Widths	Length Range
XL	1/5"	1/4"-5/16"-3/8"	6" to 26"
L	3/8"	1/2"-3/4"-1"	12" to 60"
H	1/2"	3/4"-1"-1 1/2"-2"-3"	24" to 170"
XH	7/8"	2"-3"-4"	50" to 175"
XXH	1 1/4"	2"-3"-4"-5"	70" to 180"

Because of a slight side thrust of positive-drive belts in motion, at least one pulley in a drive must be flanged. When centre distance between shafts is eight or more times the diameter of the small pulley, or when the drive is operating on vertical shafts, both pulleys should be flanged.

INSTALLATION AND OPERATION OF BELT DRIVES

Although comparatively old in principle today's belt drive is an extremely efficient method of transmitting power between prime mover and machinery.

It owes its present high performance standards to many years of research and development by engineers and technologists. This has led to significant refinements in materials and processes. To derive maximum benefit from such advances it is important that the simple installation and operation procedures set out here are closely followed. Making these routines standard practice will enable you to obtain optimum performance and long trouble-free life from your Fenner belt drives.

INSTALLATION

Pulleys
Before assembling the drive, check the pulley grooves are free from scores or sharp edges, and all dimensions conform to the relevant standard.
The Taper Lock bush is your starting point when it comes to assembly and installation. Drive installation is a straight forward job with Taper Lock - but the stages set out on the installation leaflet provided with every Taper Lock bush should be followed closely.

Alignment

Good alignment of pulleys prior to belt installation is important to avoid belt flank wear. The diagrams show some of the common alignment faults.

Belts

When the pulleys have been correctly positioned on the shafts, the belts can be installed to complete the drive. The drive centre distance should be reduced prior to the installation of the belts so that they may be fitted with the use of force. Under no circumstances must belts be prised into the grooves. Belts and pulley grooves can easily be damaged by using sharp tools to stretch the belts over the pulley rim. The installation allowance given in the table is minimum recommended reduction in centre distance for the various belt section and lengths to allow for correct fitting. The take-up allowance given in the same table should be added on to the calculated centre distance to allow for belt stretch.

Guards

Where guards are necessary it is desirable to use the mesh type to permit adequate ventilation.

Tensioning Pulleys

If tensioning (jockey) pulleys are to be used on Wedge-Belt drives, they must be fitted with a grooved pulley bearing on the inside of the drive, preferably on the slack side. The pulley should be positioned as close as possible to the large pulley. Flat tensioning pulleys, bearing on the outside of the drive are permissible only with V and not with Wedge-belts. They should be positioned within one third of the centre distance from the small pulley.

The tensioning pulley must be at least the same diameter as the small pulley of the drive. Tensioning pulley movement can only be determined by laying out the drive to scale. It must allow for passing the belts over the outside diameter of one of the pulleys on installation and should also allow for belt stretch.

Fig. 83

1	2	3	4
Shafts are not parallel to one another.	Shafts are not in correct alignment although they appear parallel when seen from above.	Shafts are parallel and in alignment but pulleys are not in alignment.	Correct installation both shafts and pulleys are parallel and in alignment.

End view of the above.

The dotted lines emphasise the faults by indicating the correct position.

Belt Pitch Length (mm)		Installation Allowances					Take-up (mm)
		SPZ Z	SPA A	SPB B	SPC C	8V D	
410 to	530						5
530 to	840						10
850 to	1160						15
1170 to	1500						20
1510 to	1830	20					25
1840 to	2170						30
2180 to	2830		25				40
2840 to	3500			30			50
3520 to	4160				50		60
4170 to	5140					65	70
5220 to	6150						85
6180 to	7500						105
7600 to	8500						125
8880 to	10170						145
10600 to	12500						175

INSTALLATION AND TAKE-UP ALLOWANCE

BROWNING BELT TENSION CHECKER
Instructions

To determine the lbs. force required to tension a drive with the BROWNING Belt Tensioner you simply do the following.

1. Measure the Belt Span as shown.
2. Divide belt span by 64 to get belt deflection needed to check tension.
3. Set large "0" ring on span scale at required belt deflection. This scale is in 1/16" increments.
4. Set small "0" ring at zero on the "Force Scale" (plunger).
5. Place the larger end of the tension checker squarely on one belt at the center of the belt span. Apply force on the plunger until the bottom of the large "0" ring is even with the top of the next belt or with the bottom of a straight edge laid across the sheaves.
6. Read the force scale under the small "0" ring to determine the force required to give the needed deflection.
7. Compare the force scale reading with the correct value for the belt style and cross section used as given in table below. The force should be between the minimum and maximum values shown.
8. If there is too little deflection force, the belts should be tightened. If there is too much deflection force the belts should be loosened.

Belt Cross Section	Small P.D. Range	Deflection Force Lbs.	
		Min.	Max.
3V	2.65 - 3.65	3	4½
	4.12 - 6.90	4	6
5V	7.1 - 10.9	8	12
	11.8 - 16.0	10	15
8V	12.5 - 17.0	18	27
	18.0 - 22.4	20	30

Fig. 84

"V" Belt Replacement

When replacing "V" belts care must be exercised that the correct type is selected. Errors in choice might be made since the top width of sane of the sizes in the three types are essentially the same. Also belts from different manufacturers should not be mixed on the same drive because of variations from nominal dimensions.

When determining the length belt required for most drives it is not necessary to be exact. First, because of the adjustment built into most drives and second because belt selection is limited to the standard lengths available. Since the standard lengths vary in steps of several inches, an approximate length calculation is usually adequate. For these reasons the following easy method of belt calculation can be used for most "V" belt drives:

1. Add the pitch diameters of the sheaves and multiply by 1 1/2.
2. To this add twice the distance between centres.
3. Select the nearest longer standard belt.
4.

This method should not be used when centres are fixed or if there are extreme pitch-diameter differences on short centres.

Sheave Groove Wear

All "V' belts and sheaves will wear to some degree with use. As wear occurs the belts will ride lower in the grooves. Generally a new belt should not seat more than 1/6 inch below the top of the groove. While belt wear is usually noticed, sheave-groove wear is often overlooked.

As wear occurs at the contact surfaces on the sides of the grooves, a dished condition develops. This results in reduced wedging action, loss of gripping power, and accelerated wear as slippage occurs. Installing new belts in worn grooves will give temporary improvement in operation but belt wear will be rapid. When changing belts, therefore, sheave-groove wear should be checked with gauges or templates.

Care must be taken when checking grooves that the correct gauge or template in respect to type, size and pitch diameter is used. As sheave grooves are designed to conform to the belt cross-section change as it bends, small diameter sheaves have less angle than larger diameter sheaves. The variation in sheave-groove included angles range from 34 degrees for small diameter sheaves up to 42 degrees for the largest diameter sheaves.

Sheave Grooves

When sheave groove wear becomes excessive, shoulders will develop on the groove side walls. If the sheave is not repaired or replaced these shoulders will quickly chew the bottom corners off new belts and ruin them.
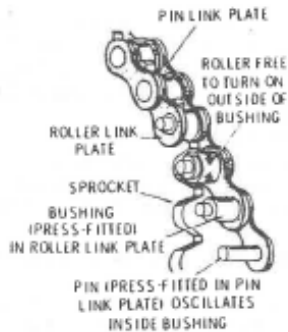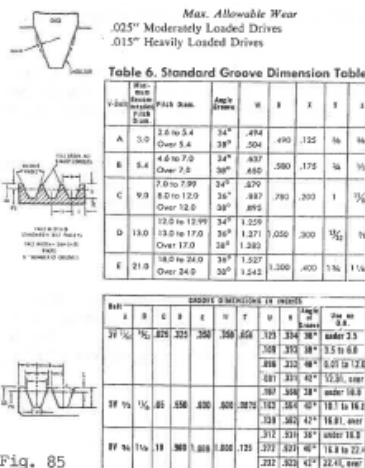

Fig. 85

Fig. 86

The more heavily loaded a drive, the greater the effect of groove wear on its operation. Light to moderately loaded drives may tolerate as much as 1/32" wear, whereas, heavily loaded drives will be adversely affected by .010" to .015" of wear. Wear should be checked with the appropriate gauge at the point illustrated in Fig. 36.

CHAIN DRIVES

A chain drive consists of an endless chain whose links mesh with toothed wheels called sprockets. Chain drives maintain a positive ratio between the driving and driven shafts, as they transmit power without slip or creep. The roller chain is the most widely used of the various styles of power transmission chain.

Roller Chain

Roller Chain is composed of an alternating series of "roller links" and "pin links". The roller links consist of two pin-link plates, two bushings and two rollers. The rollers turn freely on the bushings which are press-fitted into the link plates. The pin links consist of two link plates into which two pins are press-fitted.

In operation the pins move freely inside the bushings while the rollers turn on the outside of the bushings. The relationship of the pins, bushings, rollers and link plates is illustrated in Fig. 37.

Roller Chain Dimensions

The principal roller chain dimensions are "pitch", "chain width" and "roller diameter". These dimensions are standardized and although there are slight differences between manufacturer's products, because of this standardization chains and sprockets of different manufacturers are interchangeable.

"Pitch" is the distance from a point on one link to a corresponding point on an adjacent link. "Chain width" is the minimum distance between link plates of a roller link. "Roller diameter" is the outside diameter of a roller, and is approximately 5/8 of the pitch.

Standard series chains range in pitch from 1/4 inch to 3 inches. There is also a heavy series chain ranging from 3/4 inch to 3 inch pitch. The heavy series dimensions are the same as the standard except that the heavy series has thicker link plates.

Standard Roller Chain Numbers

The standard roller chain numbering system provides complete identification of a chain by number. The right hand digit in the chain number is 0 for chain of the usual proportions, 1 for lightweight chain, and 5 for a roller less bushing chain. The number to the left of the right-hand figure,

denotes the number of 1/8 inches in the pitch. The letter H following the chain number denotes the heavy series. The hyphenated 2 suffixed to the chain number denotes a double strand chain, 3 a triple strand etc.

For example, the number 60 indicates a chain with 6, 1/8th's, or 3/4 inch pitch. The number 41 a narrow lightweight 1/2 inch pitch chain. The number 25 indicates a 1/4 inch pitch roller less chain. The number 120 a chain having 12, 1/8th's or 1 1/2 inches pitch. In multiple strand chains 50-2 designates two strands of 50 chain, 50-3 triple strand etc. General chain dimensions for standard roller chain from 1/4 inch pitch to 3-inch pitch are tabulated in Fig. 87.

Roller Chain Connections
A length of roller chain before it is made endless will normally be made up of an even number of pitches. At either end will be an unconnected roller link with an open bushing. A special type of pin link called a connecting link is used to connect the two ends. The partially assembled connecting link consists of two pins press-fitted and riveted in one link plate. The pin holes in the free link plate are sized for either a slip fit or a light press fit on the exposed pins. The plate is secured in place either by cotter pin as shown in Fig.89 or by a spring clip as shown in Fig. 89.

American Standard Roller Chain Dimensions

USA STANDARD CHAIN NO.		PITCH (P)	MAX ROLLER DIAMETER (Dr)	(WIDTH) (W)	PIN DIAMETER (Dp)	THICKNESS LINK PLATE (LPT)	
STD.	HEAVY					STD.	HEAVY
25*	—	¼	0.130*	⅛	0.0905	0.030	—
35*	—	⅜	0.200*	¾	0.141	0.050	—
41†	—	½	0.306	¼	0.156	0.050	—
40	—	½	⅝	¾	0.156	0.060	—
50	—	⅝	0.400	⅜	0.300	0.080	—
60	60H	¾	¾₂	½	0.234	0.094	.125
80	80H	1	⅝	⅝	0.312	0.125	.156
100	100H	1¼	¾	¾	0.375	0.156	.187
120	120H	1½	¾	1	0.437	0.187	.219
140	140H	1¾	1	1	0.500	0.219	.250
160	160H	2	1⅛	1¼	0.562	0.250	.281
180	180H	2¼	1⅜	1⅜	0.687	0.281	.312
200	200H	2½	1⅜	1½	0.781	0.312	.375
240	240H	3	1¾	1⅞	0.937	0.375	.500

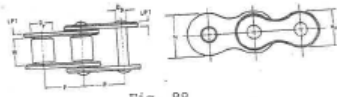* Without rollers.
† Lightweight Chain.

Fig. 87

Fig. 88

Fig. 89

If an odd number of pitches is required an offset link may be substituted for an end roller link. A more stable method of providing an odd number of pitches is by use of the offset section. A standard connecting link is used with the offset section (Fig. 90).

OFFSET SECTION

OFFSET LINK

Fig. 90

72
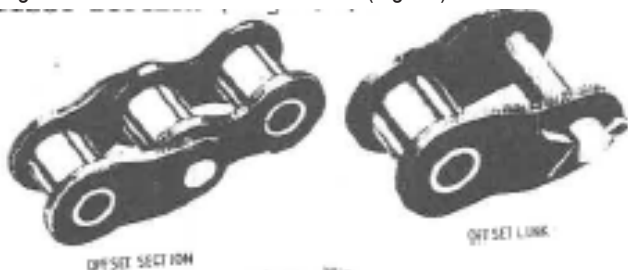
Roller Chain Sprockets

Roller chain sprockets are made to standard dimensions, tolerances and tooth form. The standard includes four types (Fig 91.); type A, a plain sprocket without hubs; type B has a hub on one side; type C has a hub on both sides; type D has a detachable hub.
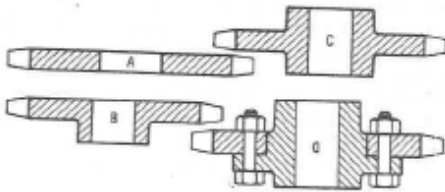
Fig. 91
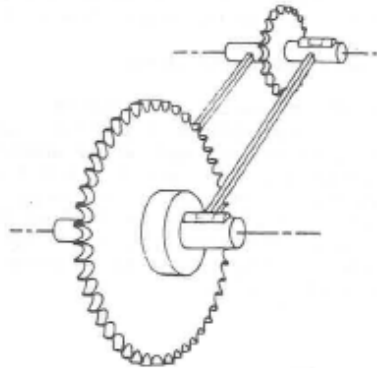
Fig. 91A

Roller Chain Installation

Correct installation of a roller chain drive requires that the shafts and the sprockets be accurately aligned. Shafts must be set level, or if inclined from a level position, both shafts must be at exactly the, same angle. The shafts must also be positioned parallel within very close limits. The sprockets must be in true axial alignment for correct sprocket tooth and chain alignment.

Horizontal shafts may be aligned with the aid of a spirit level. The bubble in the level will tell when they are both in exact horizontal position. Shafts may be adjusted for parallel alignment as shown in Fig. 91A. Any suitable measuring device such as callipers, feeler bars, etc. may be used. The distance between shafts on both sides of the sprockets should be equal. For an adjustable shaft drive make the distance less than final operating distance for easier chain installation. For drives with fixed shafts, the centre distance must be set at the exact dimension specified.

To set axial alignment of the sprockets, apply a straight-edge to the machined side surfaces as shown in Fig. 92. Tighten the set screws in the hubs to hold the sprockets and keys in position. If one of the sprockets is subject to end float, locate the sprocket so that it will be aligned when the shaft is in its normal running position. If the centre distance is too great for the available straightedge, a taut piano wire may be used.

Roller Chain Installation

Fig. 93

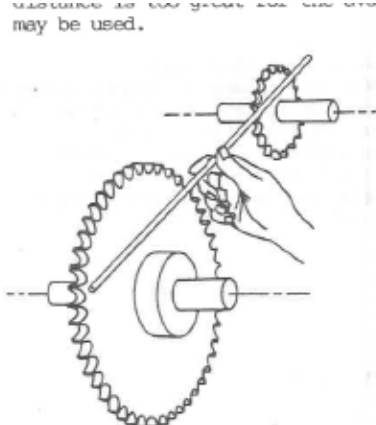| Shaft Centers	Sag
18"	3/8"
24"	1/2"
30"	5/8"
36"	3/4"
42"	7/8"
48"	1"
54"	1 1/8"
60"	1 1/4"
70"	1 1/2"
80"	1 5/8"
90"	1 3/4"
100"	2"
125"	2 1/2"

Fig. 92

Fig. 94

Roller Chain Installation
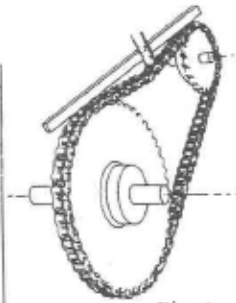
Roller Chain Installation

To install roller chain, fit it on both sprockets, bringing the free chain ends together on one sprocket. Insert the pins of the connecting link in the two end links of the chain as shown in Fig.93; then install the free plate of the connecting link. Fasten the plate with the cotters or spring clip depending on type used. When fastened, tap back the ends of the connecting link pins so the outside of the free plate of the connecting link. Fasten the plate with the cotters or spring clip depending on the type used. When fastened, tap back the ends of the connecting link pins so the outside of the free plate comes snugly against the fastener. This will prevent the connecting link squeezing the sprocket teeth which might interfere with free flexing of the joint and proper lubrication.

Adjustable drives must be positioned to provide proper chain tension. Horizontal and inclined drives should have an initial sag equal to 2% of the shaft centres. Measurements are made as shown in Fig.94. The table shows measurements for various centre distances to obtain approximately the recommended 2% sag.
To measure the amount of sag, pull the bottom side of the chain taut so that all of the excess chain will be in the top span. Pull the top side of the chain down at its centre and measure the sag as shown in Fig. 94 then adjust the centres until the proper amount is obtained. Make sure the shafts are rigidly supported and securely anchored to prevent deflection or movement which would destroy alignment.

Silent Chain

Silent chain, also called inverted-tooth chain, is constructed of leaf links having inverted teeth so designed that they engage cut tooth wheels in a manner similar to the way a rack engages a gear. The chain links are alternately assembled, either with pins or a combination of joint components.

Silent chain and sprockets are manufactured to a standard that is intended primarily to provide for interchangeability between chains and sprockets of different manufacturers. It does not provide for a standardization of joint components and link plate contours, which differ in each manufacturer's design. However, all manufacturer's links are contoured to engage the standard sprocket tooth, so joint centres lie on pitch diameter of the sprocket. The general proportions and designations of a typical silent chain link are shown in Fig.95.
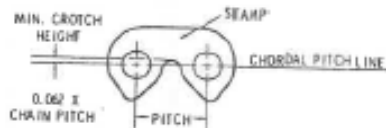
Fig. 95

Silent chain is manufactured in a wide range of pitches and widths in various styles. Chain under 3/4 inch pitch has outside guide links which engage the sides of the sprocket. The most widely used style is the middle-guide designs with one or more rows of guide links that fit guide grooves in the sprockets. (Same manufacturers use the term "wheel" rather than sprocket). Silent chains are designated by a combined letter and number symbol as follows:

1. A two-letter symbol: SC
2. One or two numerical digits indicating the pitch in eighths of inches (Usually stamped on each chain link).
3. Two or three numerical digits indicating the chain width in quarter inches.

For example, the number "SC302" designates a silent chain of 3/8-inch pitch and 1/2 inch width. Or the number "SC1012" designates a silent chain of 1 1/4 inch pitch and 3 inches width.

Chain Replacement

During operation chain pins and bushings slide against each other as the chain engages, wraps and disengages from the sprockets. Even when parts are well lubricated, some metal-to-metal contact does occur, and these parts eventually wear. This progressive joint wear elongates chain pitch, causing the chain to lengthen and ride higher on the sprocket teeth. The number of teeth in the large sprocket determines the amount of joint wear that can be tolerated before the chain jumps or rides over the ends of the sprocket teeth. When this critical degree of elongation is reached, the chain must be replaced.

Chain manufacturers have established tables of maximum elongation to aid in the determination of when wear has reached a critical point and replacement should be made. By placing a certain number of pitches under tension, elongation can be measured. When elongation reaches the limits recommended in the table, the chain should be replaced.

The recommended measuring procedure is to remove the chain and suspend it vertically with a weight attached to the bottom. When the chain must be measured while on sprockets, remove all slack and apply sufficient tension to keep the chain section that is being measured taut.
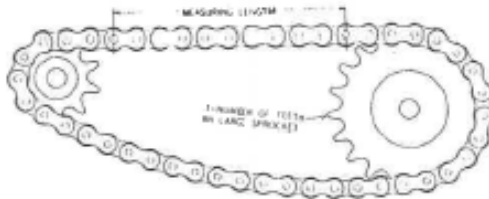
Fig. 96

Chain Elongation Limits

Chain Number	Pitch inches	Measuring length Number of pitches	Measuring length Nominal length inches	Length of chain when replacement is required Number of teeth in largest sprocket (T)							
				Up to 57	60-73	74-81	82-90	91-103	104-118	119-140	141-173
25	1/4	32	12	12 3/4	12 15/16	12 7/8	12 7/8	12 3/4	12 5/16	12 7/8	12 5/8
40	1/2	24	12	12 3/4	12 13/16	12 7/8	12 5/8	12 3/4	12 5/8	12 7/8	12 5/8
50	5/8	20	12 1/2	12 7/8	12 13/16	12 13/16	12 13/16	13 3/4	12 5/8	12 15/16	12 5/8
60	3/4	16	12	12 3/4	12 15/16	12 7/8	13 5/16	12 1/4	13 5/16	12 7/8	12 5/8
80	1	24	24	24 3/4	24 9/16	24 7/8	24 7/8	24 1/2	24 5/8	24 7/8	24 7/8
100	1 1/4	20	25	25 3/4	25 7/16	25 7/8	25 7/8	25 1/2	25 7/8	25 7/8	25 7/8
120	1 1/2	16	24	24 3/4	24 13/16	24 7/8	24 7/8	24 1/2	24 7/8	24 7/8	24 7/8
140	1 3/4	14	24 1/2	25 1/4	25 7/8	25 1/8	25 7/8	25	24 13/16	24 7/8	24 11/16
160	2	12	24	24 3/4	24 15/16	24 7/8	24 7/8	24 1/2	24 7/8	24 7/8	24 7/8
180	2 1/4	11	24 3/4	25 1/2	25 7/8	25 3/4	25 7/8	25 1/4	25 7/8	25 1/2	25 7/8
200	2 1/2	10	25	25 3/4	25 7/16	25 3/4	25 7/8	25 1/2	25 7/8	25 3/4	25 7/8
240	3	8	24	24 3/4	24 7/16	24 7/8	24 7/8	24 1/2	24 7/8	24 3/4	24 7/8

* Valid for drives with adjustable centers or drives employing adjustable idler sprockets.

Fig. 97

CENTRIFUGAL PUMP TROUBLES

Centrifugal Pump Troubles

Symptoms	Possible cause of trouble Each number is denied in the list below
Pump does not deliver water	1, 2, 3, 4, 6, 11, 14, 16, 17, 22, 23
Insufficient capacity delivered	2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 14, 17, 20, 22, 23, 29, 30, 31
Insufficient pressure developed	5, 14, 16, 17, 20, 22, 29, 30, 31
Pump loses prime after starting	2, 3, 5, 6, 7, 8, 11, 12, 13
Pump requires excessive power	15, 16, 17, 18, 19, 20, 23, 24, 26, 27, 29, 33, 34, 37
Stuffing box leaks excessively	13, 24, 26, 32, 33, 34, 35, 36, 38, 39, 40
Packing has short life	12, 13, 24, 26, 28, 32, 33, 34, 35, 36, 37, 38, 39, 40
Pump vibrates or is noisy	2, 3, 4, 9, 10, 11, 21, 23, 24, 25, 26, 27, 28, 30, 35, 36, 41, 42, 43, 44, 45, 46, 47
Bearings have short life	24, 26, 27, 28, 35, 36, 41, 42, 43, 44, 45, 46, 47
Pump overheats and seizes	1, 4, 21, 22, 24, 27, 28, 35, 36, 41

Suction troubles:
1. Pump not primed
2. Pump or suction pipe not completely filled with liquid
3. Suction lift too high
4. Insufficient margin between suction pressure and vapour pressure
5. Excessive amount of air or gas in liquid
6. Air pocket in suction line
7. Air leaks into suction line
8. Air leaks into pump through stuffing boxes
9. Foot valve too small
10. Foot valve partially clogged
11. Inlet of suction pipe insufficiently submerged
12. Water-seal pipe plugged
13. Seal cage improperly located in stuffing box, preventing sealing fluid from entering space to form the seal.

System troubles:
14. Speed too low
15. Speed too high
16. Wrong direction of rotation
17. Total head of system higher than design head of pump
18. Total head of system lower than pump design head

19.   Specific gravity of liquid different from design
20.   Viscosity of liquid different from that for which designed
21.   Operation at very low capacity
22.   Parallel operation of pumps unsuitable for such operation

Mechanical troubles:

23.   Foreign matter in impeller
24.   Misalignment
25.   Foundations not rigid
26.   Shaft bend
27.   Rotating part rubbing on stationary part
28.   Bearings worn
29.   Wearing rings worn
30.   Impeller damaged
31.   Casing gasket defective, permitting internal leakage
32.   Shaft or shaft sleeves worn or scored at the packing
33.   Packing improperly installed
34.   Incorrect type of packing for operating conditions
35.   Shaft running off centre because of worn bearings or misalignment
36.   Rotor out of balance, causing vibration
37.   Gland too tight, resulting in no flow of liquid to lubricate packing
38.   Failure to provide cooling liquid to water-cooled stuffing boxes
39.   Excessive clearance at bottom of stuffing box between shaft and casing, causing packing to be forced into pump interior.
40.   Dirt or grit in sealing liquid, leaking to scoring of shaft or shaft sleeve
41.   Excessive thrust caused by a mechanical failure inside the pump or by the failure of the hydraulic balancing device, if any
42.   Excessive grease or oil in antifriction-bearing housing or lack of cooling, causing excessive bearing temperature.
43.   Lack of lubrication
44.   Improper installation of antifriction bearings (damaged' during assembly, incorrect assembly of stacked bearings, use of unmatched bearings as a pair, etc)
45.   Dirt in bearings
46.   Rusting of bearings from water in housing
47.   Excessive cooling of water-cooled bearing, resulting in condensation of moisture from the atmosphere in the bearing housing.

Rotary Pump Troubles

Symptoms	Possible cause of trouble Each number is denied in the list below
Pump fails to discharge	1, 2, 3, 4, 5, 6, 8, 9, 16
Pump noisy	6, 10, 11, 17, 18, 19
Pump wears rapidly	11, 12, 13, 20, 24
Pump not up to capacity	3, 5, 6, 7, 9, 16, 21, 22
Pump starts, then loses suction	1, 2, 6, 7, 10
Pump takes excessive power	14, 15, 17, 20, 23

Suction troubles:
1. Not properly primed
2. Suction pipe not submerged
3. Strainer clogged
4. Leaking foot valve
5. Suction lift too high
6. Air leaks in suction
7. Suction pipe too small

System problems:
8. Wrong direction of rotation
9. Low speed
10. Insufficient liquid supply
11. Excessive pressure
12. Grit or dirt in liquid
13. Pump runs dry
14. Viscosity higher than specified
15. Obstruction in discharge line

Mechanical troubles:
16. Pump worn
17. Bent drive shaft
18. Coupling out of balance or alignment
19. Relief valve chatter
20. Pipe strain on pump casing
21. Air leak at packing
22. Relief valve improperly seated
23. Packing too tight
24. Corrosion

Symptoms	Possible cause of trouble Each number is denied in the list below
Liquid end noise	1, 2, 7, 8, 9, 10, 14, 15, 16
Power end noise	17, 18, 19, 20
Overheated power end	10, 19, 21, 22, 23, 24
Water in crankcase	25
Oil leak from crankcase	26, 27
Rapid packing or plunger wear	11, 12, 28, 29
Pitted valves and seats	3, 11, 30
Valves hanging up	31, 32
Leak at cylinder-valve hole plugs	10, 13, 33, 34
Loss of prime	1, 4, 5, 6

Suction troubles:
1. Insufficient suction pressure
2. Partial loss of prime
3. Cavitation
4. Lift too high
5. Leaking suction at foot valve
6. Acceleration head requirement too high

System problems:
7. System shocks
8. Poorly supported piping, abrupt turns in piping,
   pipe size too small, piping misaligned.
9. Air in liquid
10. Overpressure or over-speed
11. Dirty liquid
12. Dirty environment
13. Water hammer

Mechanical troubles:
14. Broken or badly worn valves
15. Packing worn
16. Obstruction under valve
17. Loose main bearings
18. Worn bearings
19. Low oil level
20. Plunger loose
21. Tight main bearings
22. Inadequate ventilation

23. Belts too tight
24. Driver misaligned
25. Condensation
26. Worn seals
27. Oil level too high
28. Pump not set level and rigid
29. Loose packing
30. Corrosion
31. Valve binding
32. Broken valve spring
33. Loose cylinder plug
34. Damaged 0-ring seal

SELECTION GUIDE
FITS AND TOLERANCES

For ordinary engineering practice, holes H6, H7, H8 and H11 are satisfactory for most purposes.

Shafts:

a - g	Both limits less than basic size	Clearance fits
h	Nominal size as upper limit	Close running fits
k & n	Limits above and below basic size	Push, light drive
P & s	Limits above basic size	Force, heavy drive fits

Slack running	H9/e9)	
Average running	H8/f7)	Clearance fits
Close running	H7/g6)	

TransitionPush fit		H7/k6)	
Fits	Light drive	H7/n6)	

	Medium drive	H7/P6)	Interference fits
	Force/heavy drive	fH7/S6)	

BRITISH STANDARD

SELECTED ISO FITS—HOLE BASIS

Clearance fits

Diagram to scale for 25 mm. diameter

H11, c11, H9, d10, H9, e9, H8, f7, H7, g6
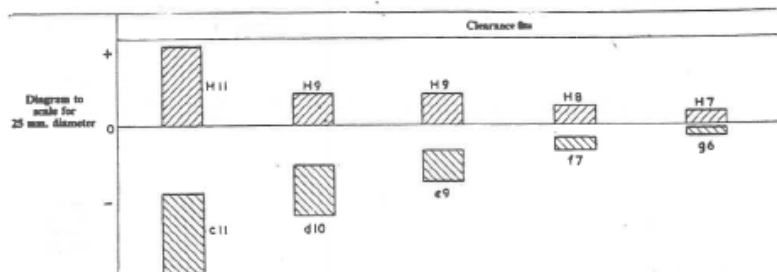
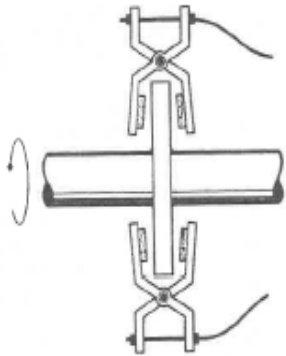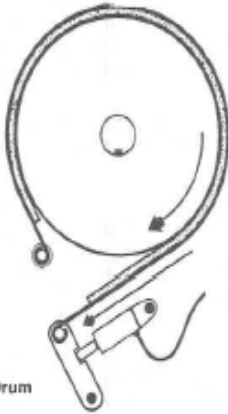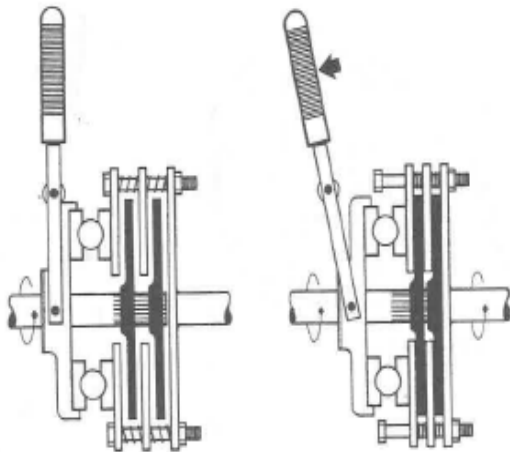| Nominal sizes		Tolerance		Tolerance		Tolerance		Tolerance		Tolerance	
Over	To	H11	c11	H9	d10	H9	e9	H8	f7	H7	g6
mm	mm	0·001 mm	0·001 mm	0·001 mm	0·001 mm	0·001 mm	0·001 mm	0·001 mm	0·001 mm	0·001 mm	0·001 mm
—	3	+ 60 / 0	− 60 / − 120	+ 25 / 0	− 20 / − 60	+ 25 / 0	− 14 / − 39	+ 14 / 0	− 6 / − 16	+ 10 / 0	− 2 / − 8
3	6	+ 75 / 0	− 70 / − 145	+ 30 / 0	− 30 / − 78	+ 30 / 0	− 20 / − 50	+ 18 / 0	− 10 / − 22	+ 12 / 0	− 4 / − 12
6	10	+ 90 / 0	− 80 / − 170	+ 36 / 0	− 40 / − 98	+ 36 / 0	− 25 / − 61	+ 22 / 0	− 13 / − 28	+ 15 / 0	− 5 / − 14
10	18	+ 110 / 0	− 95 / − 205	+ 43 / 0	− 50 / − 120	+ 43 / 0	− 32 / − 75	+ 27 / 0	− 16 / − 34	+ 18 / 0	− 6 / − 17
18	30	+ 130 / 0	− 110 / − 240	+ 52 / 0	− 65 / − 149	+ 52 / 0	− 40 / − 92	+ 33 / 0	− 20 / − 41	+ 21 / 0	− 7 / − 20
30	40	+ 160 / 0	− 120 / − 280	+ 62 / 0	− 80 / − 180	+ 62 / 0	− 50 / − 112	+ 39 / 0	− 25 / − 50	+ 25 / 0	− 9 / − 25
40	50	+ 160 / 0	− 130 / − 290								
50	65	+ 190 / 0	− 140 / − 330	+ 74 / 0	− 100 / − 220	+ 74 / 0	− 60 / − 134	+ 46 / 0	− 30 / − 60	+ 30 / 0	− 10 / − 29
65	80	+ 190 / 0	− 150 / − 340								
80	100	+ 220 / 0	− 170 / − 390	+ 87 / 0	− 120 / − 260	+ 87 / 0	− 72 / − 159	+ 54 / 0	− 36 / − 71	+ 35 / 0	− 12 / − 34
100	120	+ 220 / 0	− 180 / − 400								
120	140	+ 250 / 0	− 200 / − 450	+ 100 / 0	− 145 / − 305	+ 100 / 0	− 84 / − 185	+ 63 / 0	− 43 / − 83	+ 40 / 0	− 14 / − 39
140	160	+ 250 / 0	− 210 / − 460								
160	180	+ 250 / 0	− 230 / − 480								
180	200	+ 290 / 0	− 240 / − 530	+ 115 / 0	− 170 / − 355	+ 115 / 0	− 100 / − 215	+ 72 / 0	− 50 / − 96	+ 46 / 0	− 15 / − 44
200	225	+ 290 / 0	− 260 / − 550								
225	250	+ 290 / 0	− 280 / − 570								
250	280	+ 320 / 0	− 300 / − 620	+ 130 / 0	− 190 / − 400	+ 130 / 0	− 110 / − 240	+ 81 / 0	− 56 / − 108	+ 52 / 0	− 17 / − 49
280	315	+ 320 / 0	− 330 / − 650								
315	355	+ 360 / 0	− 360 / − 720	+ 140 / 0	− 210 / − 440	+ 140 / 0	− 125 / − 265	+ 89 / 0	− 62 / − 119	+ 57 / 0	− 18 / − 54
355	400	+ 360 / 0	− 400 / − 760								
400	450	+ 400 / 0	− 440 / − 840	+ 155 / 0	− 230 / − 480	+ 155 / 0	− 135 / − 290	+ 97 / 0	− 68 / − 131	+ 63 / 0	− 20 / − 60
450	500	+ 400 / 0	− 480 / − 880								

Figure 10: Internal and External Drum

Figure 11: Caliper Disk Brake

Figure 12: Multiple Disk Clutch

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