6. STEAM TURBINES

6.1 Turbine Types

Large steam turbines are all of the axial-flow type (Fig. 6.1). They may use single flow, double flow or reversed flow (Fig. 6.2, where blades are not shown). Double flow avoids excessively long blades and can reduce axial thrust. Steam enters and leaves cylinder radially, so design must leave space for flow to turn to axial direction with minimum losses.

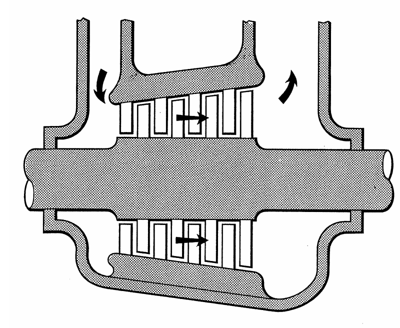


Fig.6.1 Axial-flow turbine (from MPSP).

The limit of a single-cylinder turbine is about 100 MW. Multi-cylinder designs are used in large plant, e.g. one high pressure (HP) turbine, one intermediate pressure (IP) turbine and two low pressure (LP) turbines (Figs 6.3 and 6.4 show various multi-cylinder turbine arrangements). The IP and LP turbines are usually double flow.

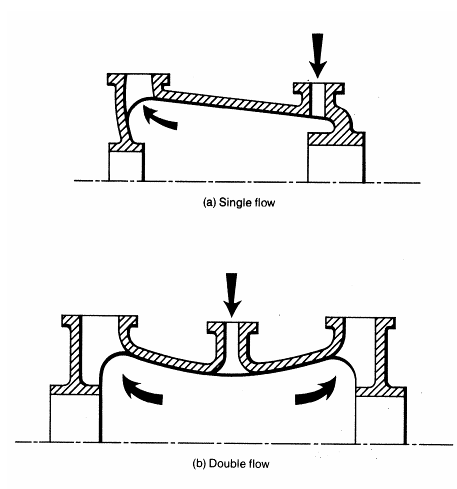


Fig.6.2 Direction of flow in turbines (from MPSP).

Cross compound machines avoid long shafts and can enable fewer LP turbines if LP turbine shafts are run at different speeds. Mainly used with 60Hz grid frequency.

6.2 Speed of Rotation

Speed of shaft rotation is $f = pn$

f = grid frequency (Hz)

p = number of generator pole pairs

n = rotational speed (Hz)

Machine type	Rotational speed (rpm)
Two-pole (full-speed)	3000 (50Hz)
Four-pole (half-speed)	1500 (25Hz)

Turbines to drive boiler feed pumps operate at variable speeds, as high as 8500 rpm, to accommodate the operational range of the driven machine.

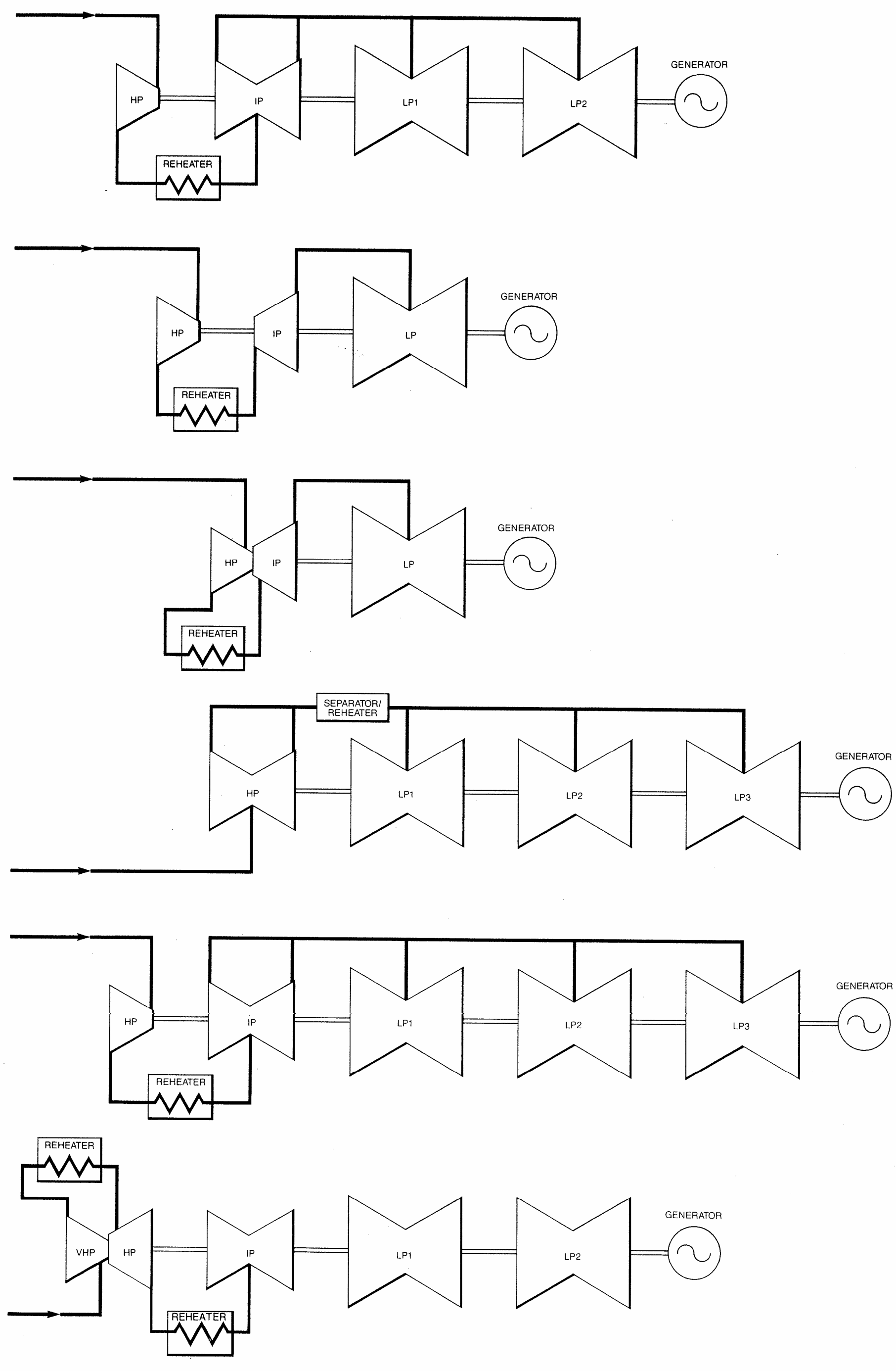


Fig. 6.3 Multi-cylinder turbine arrangements (from MPSP).

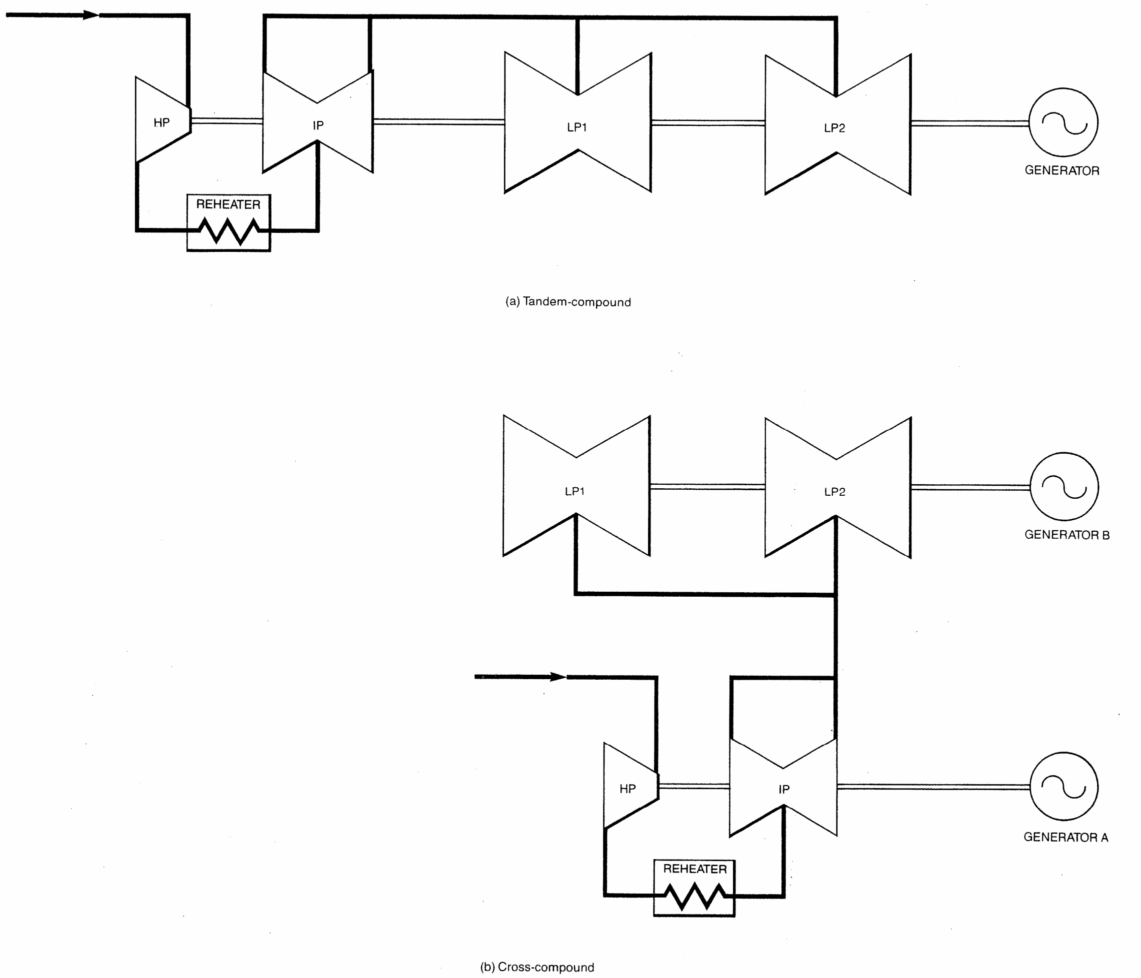


Fig. 6.4 Tandem-compound and cross-compound machines (from MPSP).

6.3 Turbine Stages

An impulse stage consists of stationary blades forming nozzles through which the steam expands, increasing velocity as a result of decreasing pressure. The steam then strikes the rotating blades and performs work on them, which in turn decreases the velocity (kinetic energy) of the steam. The stream then passes through another set of stationary blades which turn it back to the original direction and increases the velocity again though nozzle action.

Ideal reaction stages would consist of rotating nozzles with stationary blades (buckets) to redirect the flow for the next set of rotating nozzles. The expansion in the rotating blades causes a pressure force (reaction) on them that drives them. However, it is impractical to admit steam to rotating nozzles. The expansion of steam in the stationary nozzles of a practical reaction turbine is an impulse action. Therefore, the reaction stage in actual turbine actions is a combination if impulse and reaction principles.

A reaction stage has a higher blade aerodynamic efficiency than an impulse stage, but tip leakage losses are higher because of the pressure drop across the rotating stage. This is significant for short blades (HP) but becomes insignificant for long blades (LP).

Modern turbines are neither purely impulse nor purely reaction. They are a combination of both, with a highly twisted profile so that the inlet and outlet angles conform to the three-dimensional flow characteristics at all blade heights, e.g. Fig. 6.5.

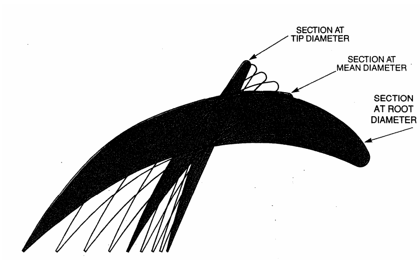


Fig. 6.5 LP last stage moving blade (from MPSP).

Blade efficiencies are not ideal. Profile loss is due to formation of a boundary layer on the blade surface. Secondary loss is due to friction on the casing wall and on the blade root. It is a boundary layer effect. Tip leakage is due to steam passing through the necessary small clearance between the moving blade tip and the casing, or between the end of the fixed blades and the rotating shaft.

A shroud band extends around the entire circumference of the moving blades, joining the tips. The shroud is sealed against the casing by several knife edges (Fig. 6.6).

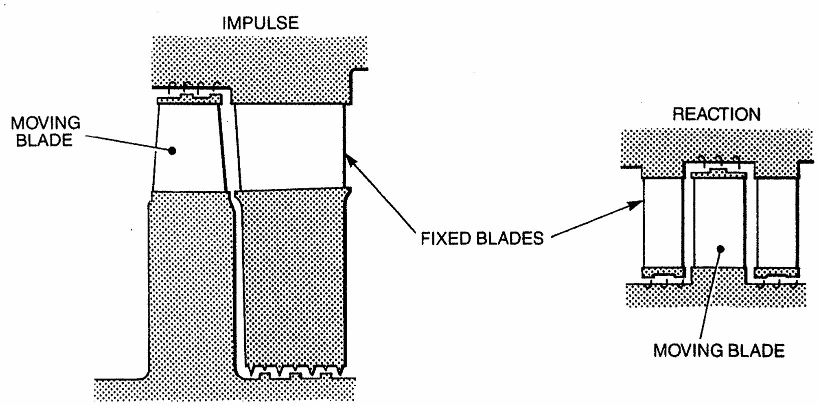


Fig. 6.6 Reaction and impulse turbine interstage sealing (from MPSP).

The long fixed blades of an LP cylinder are stiffened towards their tips with <u>lacing wires</u>. These damp vibrations and raise the resonant frequency of the blade so that it does not coincide with any exciting frequencies. Lacing wires cause some aerodynamic losses.

<u>Wetness loss</u> is associated with moisture droplets entrained in the low pressure steam near the exit plane of the LP turbines. Droplets absorb energy and can erode the leading edges of moving blades.

6.4 Turbine Blading

6.4.1 Impulse Stages

In impulse stage, most of the heat drop occurs in the stationary blading (acting as nozzles to increase velocity and kinetic energy). Driving force arises from change in momentum (direction) across the moving blades.

<u>6.4.1.1 Moving Blades</u>

Moving blades are subject to turbulent disturbances from the nozzle wakes formed by the fixed blades, so resonances must be avoided. Also, momentum change across moving blades is high, so moving

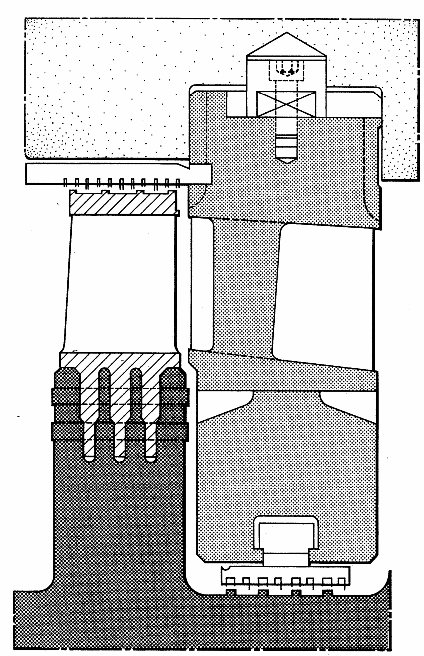


Fig.6.7 Modern impulse stage with labyrinth sealing (from MPSP).

blades must be robust, and tend to be heavy. Moving blades are manufactured individually and attached to a <u>wheel disc</u> which is part of the rotating shaft. Moving blade tips are riveted to a <u>coverband</u> (or shroud band) which acts as a labyrinth seal and braces the moving blades to reduce vibration (Fig. 6.7).

<u>6.4.1.2 Fixed Blades</u>

Fixed blades (Fig. 6.7) are manufactured by:

- Machining from a solid disc;
- Casting steel blades in to a cast iron diaphragm, and then machining; or
- Electrochemical machining from solid.

Both the diaphragm and the blades need to be robust to withstand the large stage pressure drag. Good labyrinth sealing is needed between the diaphragm and the rotor, again because of the large pressure drop.

6.4.2 Reaction Stages

These stages employ equal degrees of impulse and reaction, so fixed and moving blades are of identical section. This leads to economies of manufacture. Driving force arises from the reaction force of the steam as it accelerates through the moving blades. Nozzle wakes from the fixed blades are not strong so there is lower risk of fatigue failure due to vibration in the moving blades. The pressure drop across fixed blades is small so a diaphragm is not needed. Modern practice is to braze together short groups of blades before machining the circumferential serrations (fir-trees) on the sides of the roots (Figs 6.8 and 6.9).

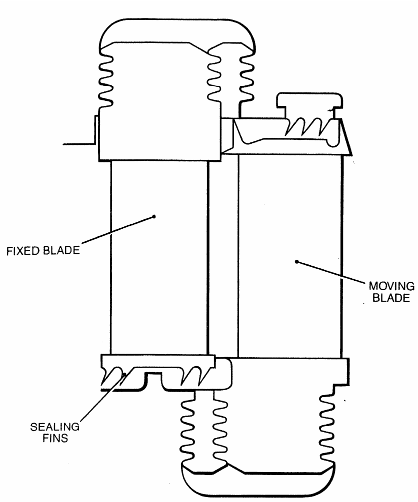


Fig. 6.8 Section through a reaction stage (from MPSP).

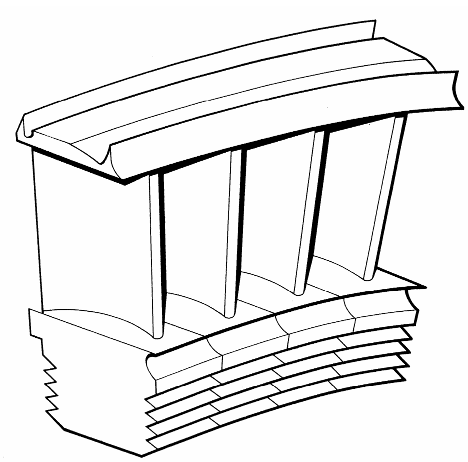


Fig. 6.9 Brazed reaction blade group (from MPSP).

Low Pressure Stages

Blades of up to one metre long can be used. A coverband or lacing wire must behave as a beam spanning the blade pitch in resisting centrifugal loading, and must accommodate the substantial circumferential strains due to elastic extension of the blades and the tendency of the blades to untwist at speed. When lacing wires are used, they are usually of the 'loose' type with circumferential restraint on only one blade in each group, and are free to move circumferentially in adjacent blades, centrifugal forces providing the necessary damping through friction.

A coverband of conventional design is not feasible for slim sections and where the peripheral speed might be approaching Mach 2, but a continuous ring of stiffening devices of sufficient elasticity may be used to accommodate the circumferential strains. The elastic arch banding shown in Fig. 6.10 braces the blade tips and provides some resistance to blade untwist as well as permitting circumferential strain.

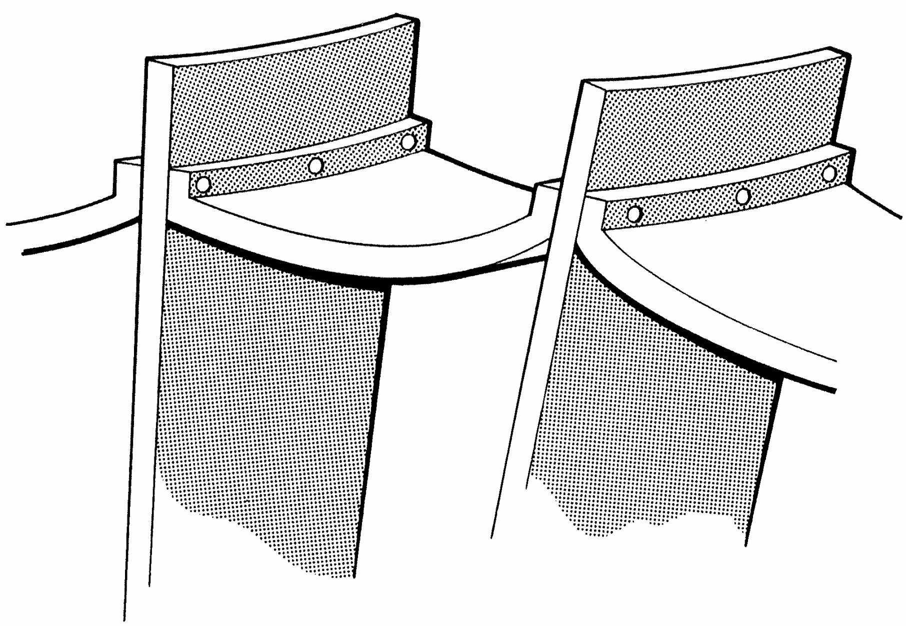


Fig. 6.10 Arch coverbands (from MPSP).

Zigzag spool rods are sometimes used at the tips of last-stage LP blades (Fig. 6.11). They provide no restraint against circumferential expansion or centrifugal untwist, but the reduced sections at the ends of the rods are forced against the holes in the blades by centrifugal action and the sliding friction provides effective damping, minimising blade vibration and high frequency flutter at the blade tip.

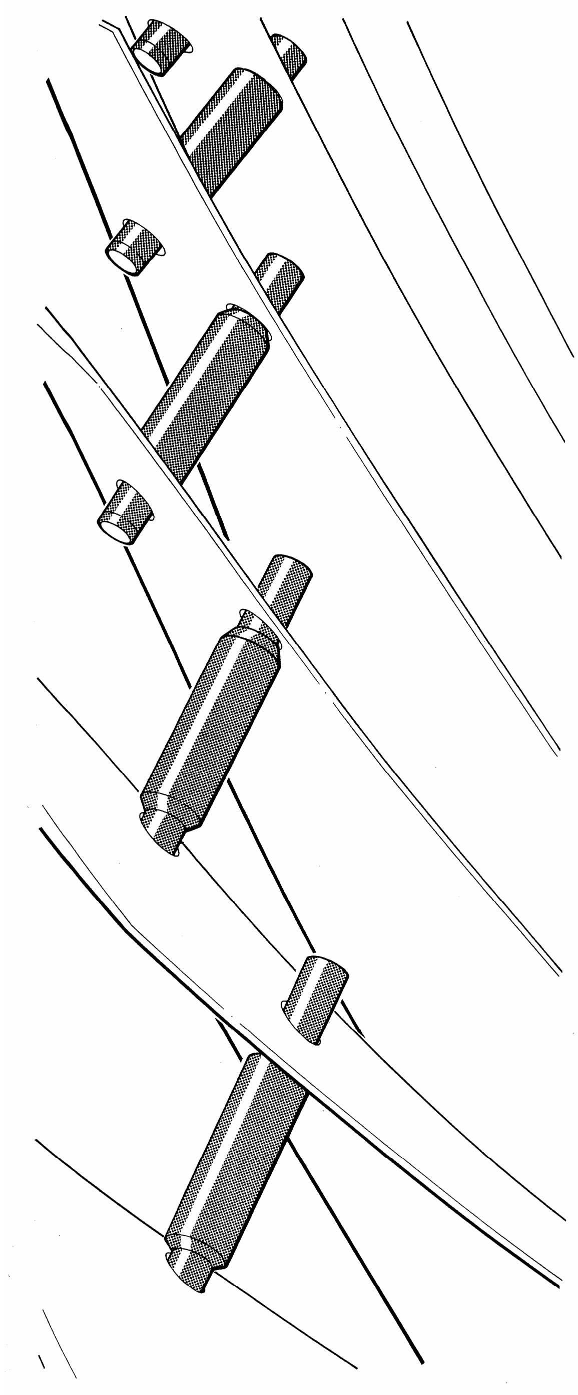


Fig. 6.11 Zigzag spool rod tip-ties (from MPSP).

6.4.4 Moving Blade Root Attachments

Last stage blades develop centrifugal forces of hundreds of tonnes when running. Strong methods of attachment are needed. Fir-tree roots are widely used (Fig. 6.12). There is some looseness in fir-tree roots for assembly but this becomes rigid when the blades rotate. However, it is not possible to measure the zero-speed vibration characteristics of the blades. Pinned roots overcome this but blade replacement is not easy.

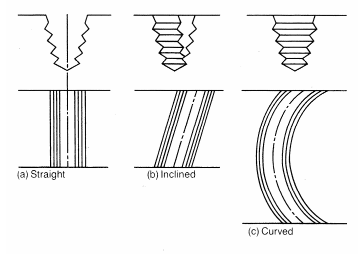


Fig. 6.12 Types of fir-tree roots (from MPSP).

6.4.5 Blade Materials

Blade material must have some or all of the following properties, depending on the position and role.

- corrosion resistance (especially in the wet LP stage)
- tensile strength (to resist centrifugal and bending stresses)
- ductility (to accommodate stress peaks and stress concentrations)
- impact strength (to resist water slugs)
- material damping (to reduce vibration stresses)
- creep resistance

12% Cr stainless steels are a widely used material. Their weakness is at very high temperatures (> 480C). A typical high temperature steel is 12% Cr alloyed with molybdenum and vanadium (to 650C).

Titanium has some attractions but it is expensive and material damping is low. It has poor vibration characteristics. Because of its high strength/weight ratio, titanium is used in lacing wire and for coverbands and shrouding.

6.4.6 Blade Vibration Control

Blade vibration characteristics under operating conditions are very complex and difficult to predict by calculation such as finite element analysis because:

- an individual blade has a very complex geometry

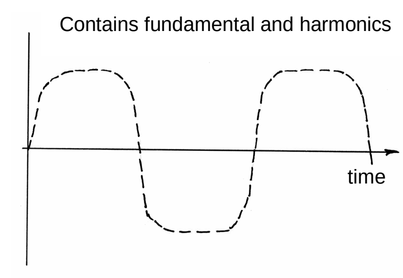
- there are vibration interactions among the blades through the blade disc, diaphragms, coverbands and lacing wires.

The vibration of a fully-bladed disc is much more complicated than is suggested by the characteristics of a single cantilever blade. There is a multiplicity of modes of vibration in the turbine working frequency range. For a single blade, there are only two or three.

Sources of vibration excitation are:

- Non-uniform flow caused by:
  - steam entering over only a portion of the circumference
  - complex axial to radial flow behaviour (which is minimised with good design)
  - flow distortion caused by steam extraction passages for feedheater tappings
- Periodic effects due to manufacturing constraints, e.g.
  - Inexact matching at fir-tree roots
  - Eccentricity of diaphragms
  - Ellipticity of stationary parts
  - Non-uniformity of manufacturing thicknesses
  - Moisture removal slots

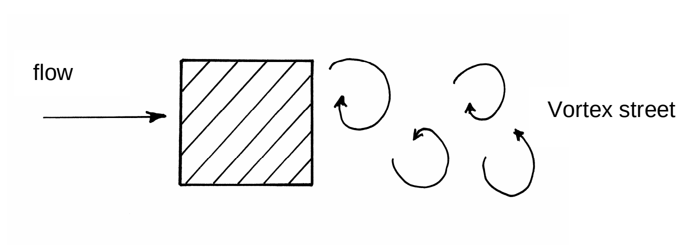
All of the above sources cause excitation at the rotation frequency or low multiples (harmonics) of that frequency. Recall the Fourier content of a non-sinusoidal periodic wave.



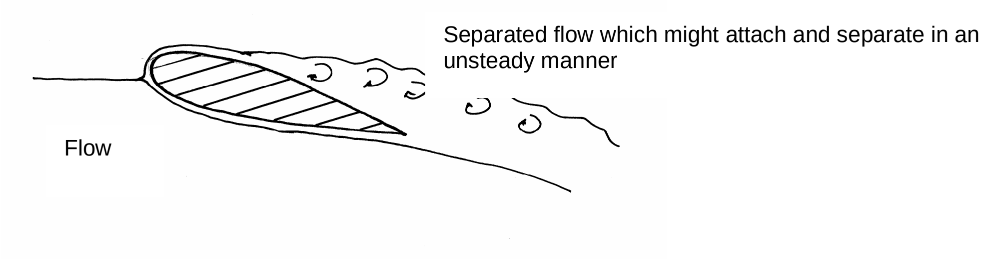
- Nozzle wake excitation as a rotating blade passes a stationary blade.
  Excitation frequency = rotational frequency × number of stationary blades and its multiples.

Some sources can cause excitation at frequencies that are unrelated to rotational frequency,

- acoustic resonances in inlet passages, extraction lines and other cavities
- vortex shedding from bluff bodies



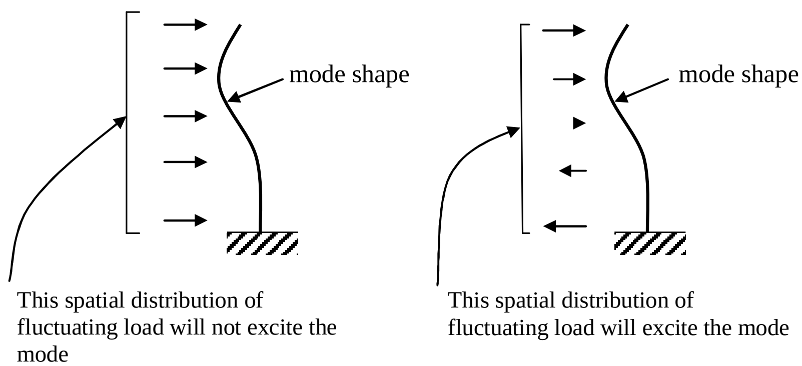
- unsteady flow separation from stationary blades.



- unsteady shock waves in blade passages
- surface pressure fluctuations from impingement of turbulent flow.

A mode will only be excited if:

- the excitation frequency coincides with the resonant frequency of the mode; and
- the loading has the necessary component of spatial distribution.



6.4.7 Vibration Testing of Blades

Predictions from a finite element analysis (resonant frequencies and mode shapes) are verified against measurements with piezoelectric accelerometers on a single cantilevered blade. If agreement is good, predictions for a fully bladed disc are compared with experiments.

For short blade discs (all but the last LP stage) static testing is good enough to verify that all natural frequencies are above the 8th engine order, i.e. 8×50 Hz = 400 Hz. This allows for up to the 8th harmonic in the excitation.

For the larger LP blades, natural frequencies are lower and may coincide with harmonics of the rotation frequency below the 8th order. Testing must be at running speed because centrifugal effects can change the stiffness of long blades. Testing is conducted at speed in a vacuum wheel chamber. The presence of air would necessitate a huge amount of power to a large electric motor to drive the disc. Windage near blade tips would cause overheating and make results difficult to interpret.

The disc is run up to 115% of synchronous speed and blade vibration is detected with strain gauges of piezoelectric crystals. A Campbell Diagram can be developed (Fig. 6.13).

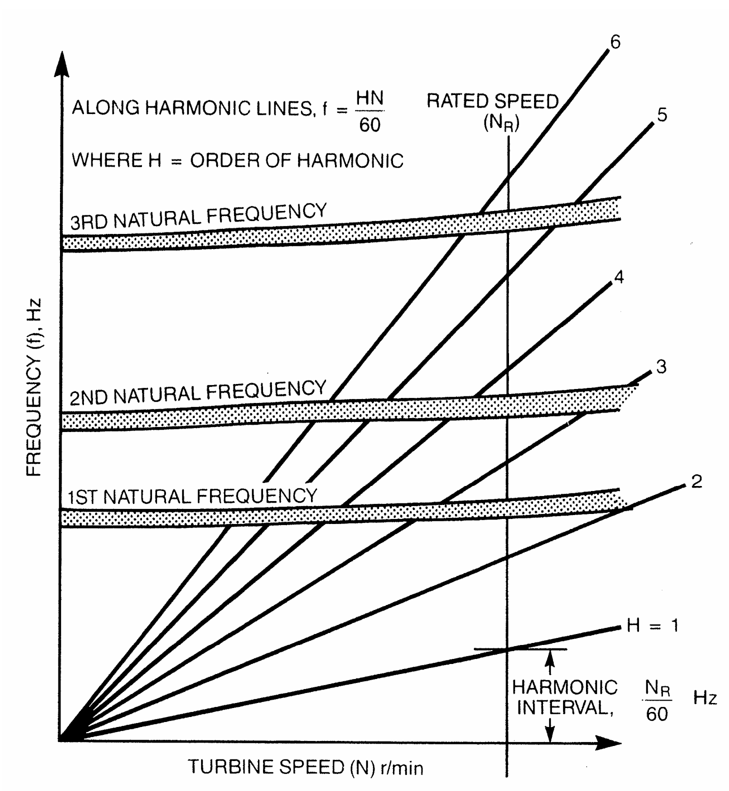


Fig. 6.13 Campbell diagram (from MPSP).

Where lines cross there is a prospect of resonance in service. It is usual to confine attention to $\pm$ 6% of synchronous speed (2820 to 3180 rpm). A common specification is that in this range there be no resonances up to 8th order.

Problem modes can be 'tuned out' by adjusting the blade mass near the tip, or by adding or removing mass to or from the shrouding.

6.4.8 Blade Erosion

Water droplets in the last stages of a turbine can cause erosion at the leading edge of moving blades, and cracks can form. Leading edges can be protected by surface hardening or by welding a shield of hard material such as tungsten chromium tool steel or satellite (an alloy of cobalt and chromium). Shields will probably need to be replaced once during the lifetime of the turbine.

6.5 Turbine Casings

A turbine casing (cylinder) is a high pressure vessel with its weight supported at each end on the horizontal centreline. It is designed to withstand hoop stresses in the transverse plane and to be stiff in the longitudinal direction to maintain accurate clearances between the stationary and rotating parts.

Casings are split along the horizontal centreline to allow internal access and insertion of the rotor as a complete assembly.

High pressures necessitate very thick flanges and bolting. The temperature of these changes more slowly than the rest of the casing during start-up so a flange warming system is used.

HP and IP casings are cast. LP casings can contain some fabrication. Casings are tested to 150% of highest working pressure.

6.5.1 High Pressure Casings

Cross-sections of a single-flow HP casing are shown in Figs 6.14 and 6.15. HP casings are usually of a double shell design. The space between the shells is filled with steam at exhaust conditions. Then each casing can be designed for smaller temperature and pressure differentials. Some exhaust steam leaks past a baffle to fill the space between the shells. The rotor is protected from high pressure steam at the inlet by a deflector ring. Steam leaking past the gland at the HP end is piped to exhaust connections, so there is only a gentle flow between the casings.

Triple casings are used in some machines to further reduce temperature and pressure differentials.

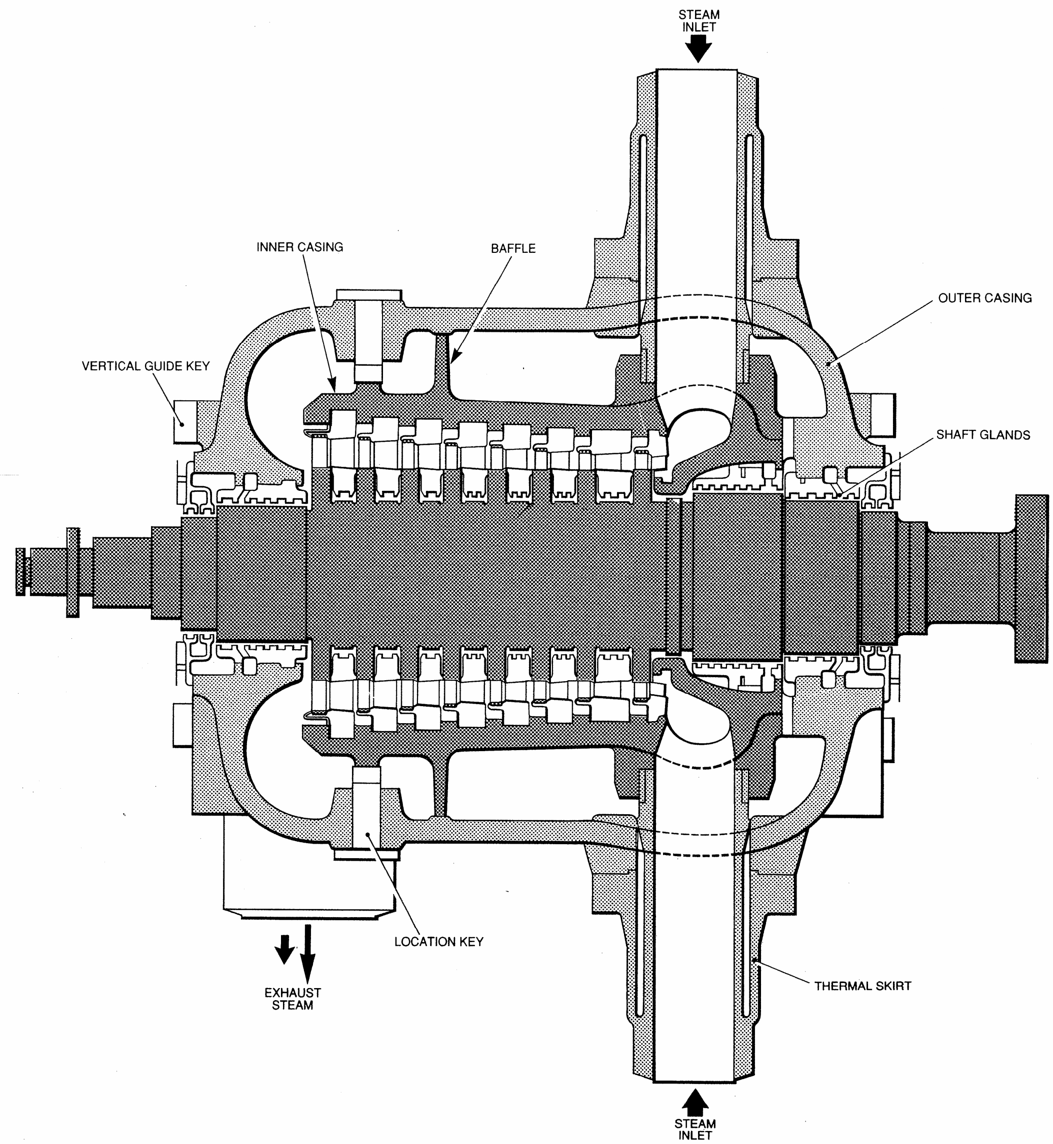


Fig. 6.14 Axial section of HP cylinder (from MPSP).

6.5.2 Intermediate Pressure Casings

IP casings usually have a partial double-casing covering the first few stages. Subsequent stages are supported by carrier rings (Fig. 6.16).

Both the inner casing and the carrier rings reduce pressure and temperature loading on the outer casing. IP cylinders are usually double flow. Cooling flow through the outer casing can be achieved by slight differences in the blading in the two flows to produce different pressures at each end. Fig. 6.17 is a photo of typical HP and IP cylinders.

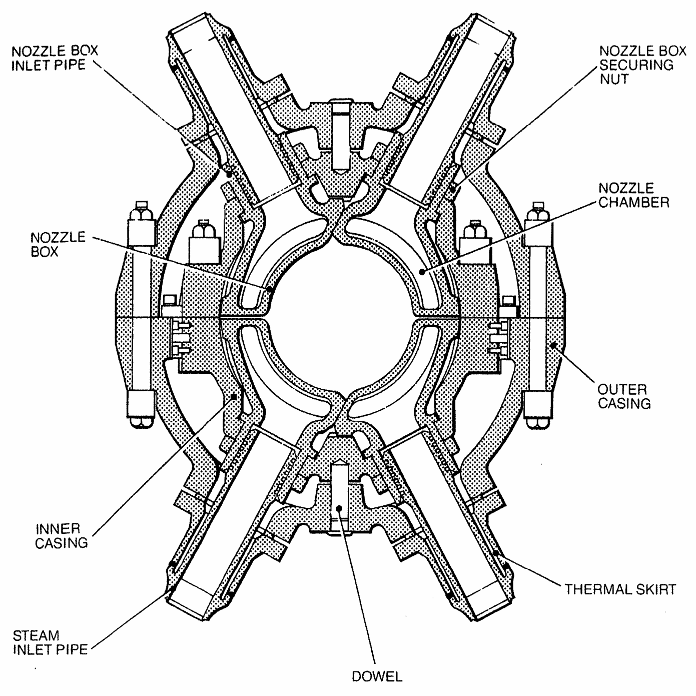


Fig. 6.15 Cross-section of HP cylinder (from MPSP).

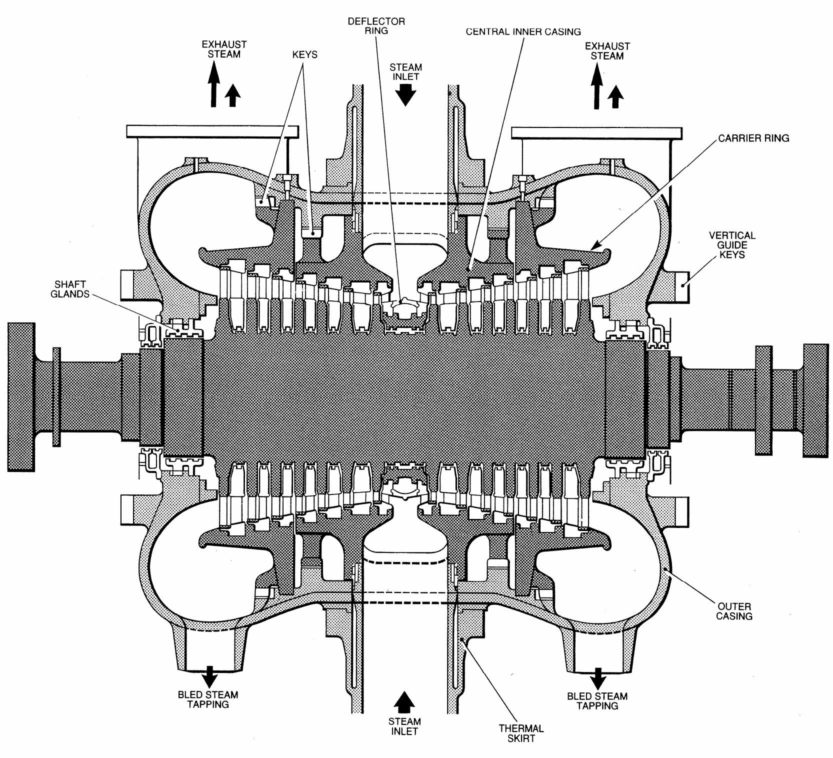


Fig. 6.16 Axial section of IP turbine casing (from MPSP).

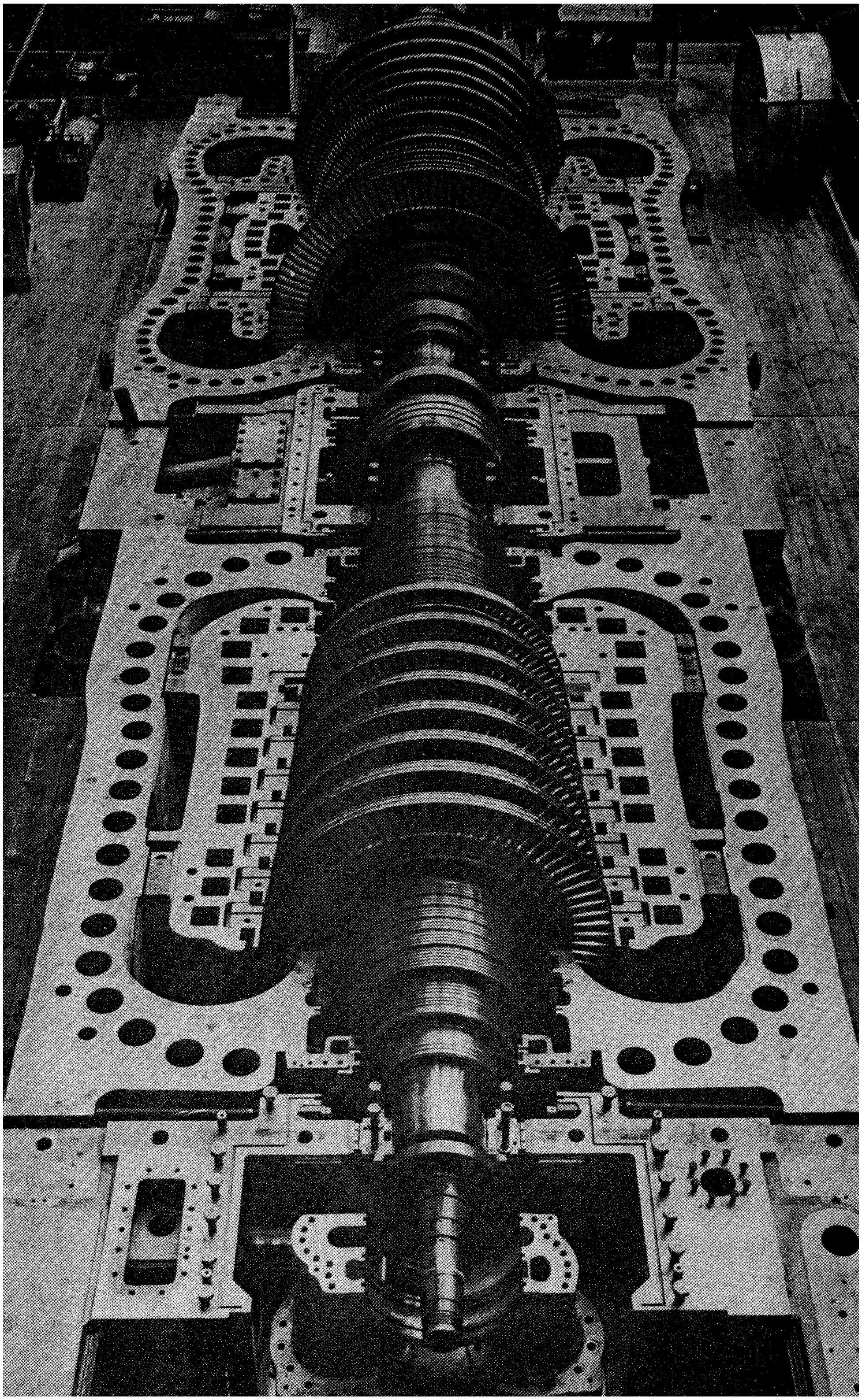


Fig. 6.17 HP and IP casings (from MPSP).

6.5.3 Low Pressure Casings

LP cylinders often have double casings and are usually double-flow. Fig. 6.18 shows a typical cross-section.

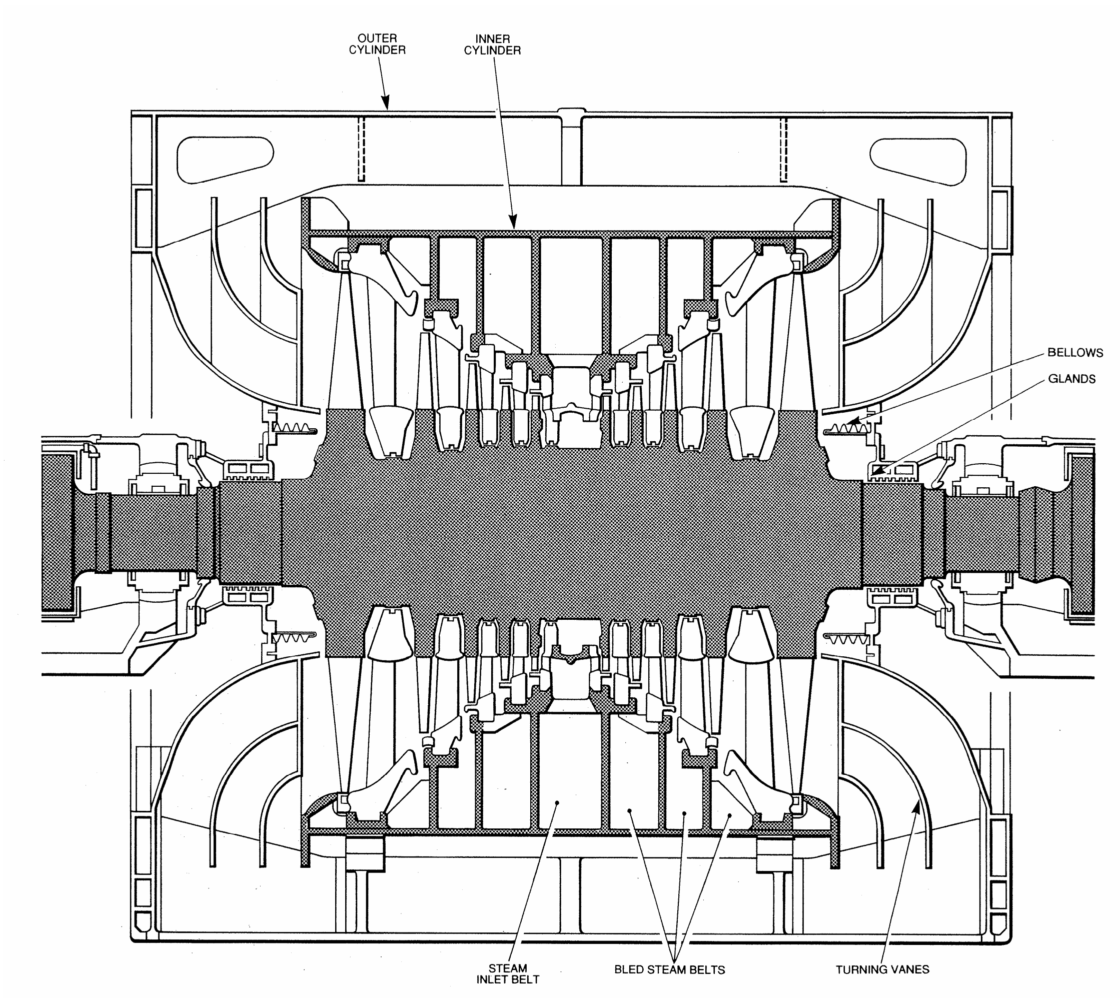


Fig. 6.18 Axial section of LP casing (from MPSP).

6.6 Couplings

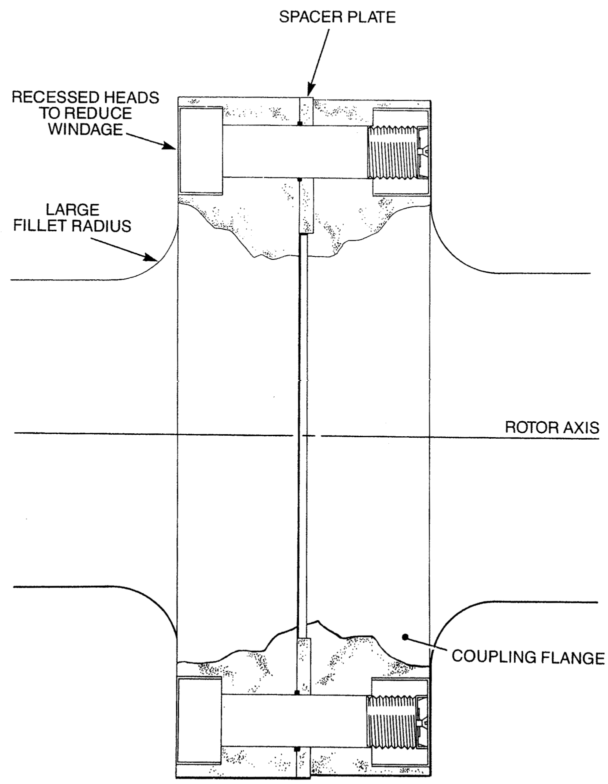
Shaft couplings are needed between the various stages of a turbine/generator set. Ideally, couplings should:

- Transmit torque
- Allow angular misalignment
- Transmit axial thrust

- Ensure axial location or allow relative axial movement
- Provide torsional resilience

Flexible or semi-flexible couplings can provide this but they are impracticable on large turbines because of the high torque to be transmitted.

Rigid couplings are used in large turbines so that the joined shafts can behave as one continuous rotor. They are either integral with the shaft forging (Fig 6.19) or shrunk on to the shaft (Fig. 6.20). In the latter case, high pressure oil can be injected into annular grooves to ensure correct seating during assembly, or to aid removal.



6.19 Rigid forged coupling (from MPSP).

Couplings are designed to withstand a three-phase fault or out-of-phase synchronising without damage (4-5 times full load torque).

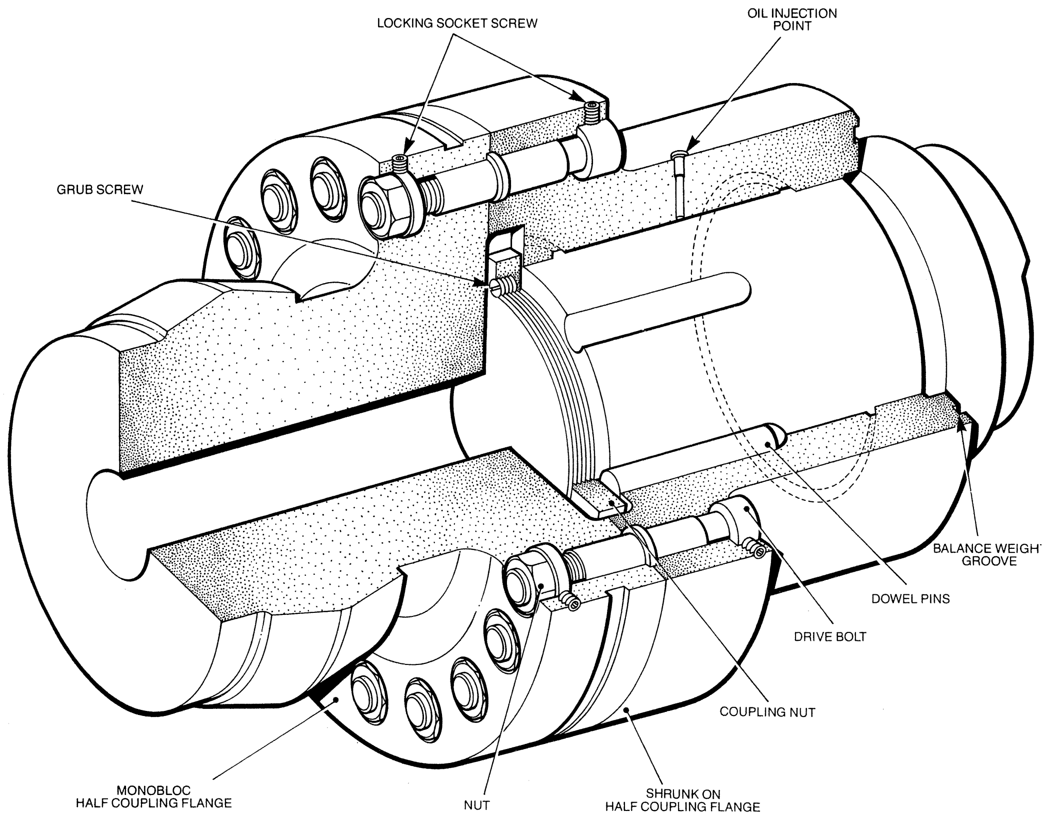


Fig. 6.20 Shrunk-on coupling (from MPSP).

6.6.1 Rotor Alignment

Excessive misalignment of a multi-bearing shaft line can affect the vibration behaviour.

- It causes bending moments at couplings which act like a rotating out-of-balance.
- It can cause bearing unloading which alters shaft vibration behaviour.

A long shaft bends naturally under its own weight to form a catenary (Fig. 6.21) and revolve around a curved centreline. The shape of the catenary depends on the masses and stiffnesses of the rotors.

The aim of alignment is to ensure insignificant bending moments and shear at the couplings. Bearing heights are adjusted so that coupling faces are square to each other, with centrelines coincident and with the same slope where the faces meet. This is done by slightly separating the coupling and turning the rotor to different positions. Bearings are adjusted to get uniform gap and concentricity (measured with a dial gauge).

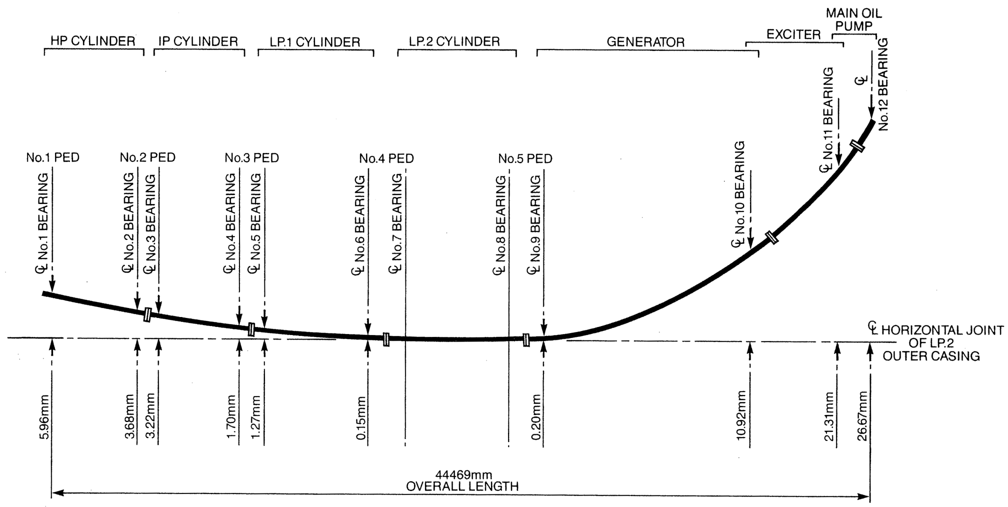


Fig. 6.21 Shaft catenary for a large turbine-generator (from MPSP).

Bending moment cyclic variations can be measured with strain gauges and optical techniques. Lasers are used to set the catenary up initially, prior to adjusting it.

Outer bearings may be 25 mm above the level of central bearings. Changes in service to pedestal bearings are monitored on-line with a manometric system.

6.7 Journal Bearings

Bearings on the shaft line of a large turbine/generator set are invariably white-metalled journal bearings because of their:

- high load capacity
- reliability
- absence of wear through use of hydrodynamically generated films of lubricating oil (no metal-to-metal contact)

Turbine bearings have diameters up to 550 mm, with length/diameter (L/D) ratios of 0.5 to 0.7. Generator bearings have L/D ratios of 0.6 to 1.0 because of the weight of the generator rotor. They are split in halves for assembly of the rotor, with bolts and local dowels (Fig.

6.22). White metal is either cast into a mild steel liner or cast into the bearing body. The bearing body is spherically seated into the pedestal for angular alignment. Shims are available for vertical and horizontal alignment.

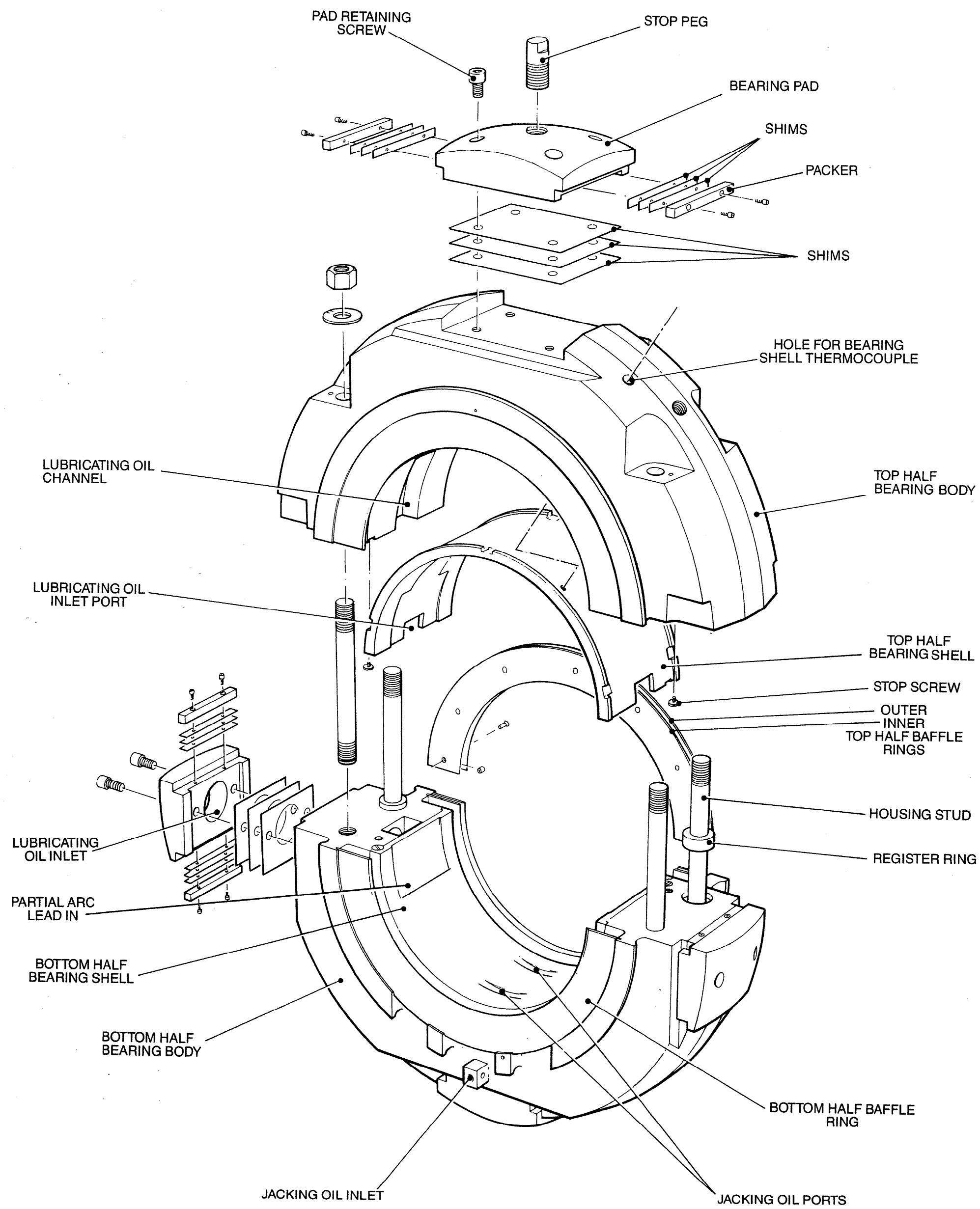


Fig. 6.22 Journal bearing (from MPSP).

White metal (or babbit) is usually composed of 80 to 90 % tin to which is added about 3 to 8% copper and 4 to 14% antimony. These alloys have very little tendency to cause wear to their steel journals because of their ability to embed dirt. They are easily bonded, cast and shaped, and can have good load-carrying and fatigue properties.

The bores of journal bearings are usually elliptical to provide the geometry for hydrodynamic lubrication. A circular bore is machined with shims in the horizontal split. The shims are removed in assembly to give typical diametrical clearance/diameter ratios of 0.001 vertically and 0.00015 horizontally. Oil is fed into the bearing via lead-in ports at two diametrically opposite points on the horizontal centreline. This is to cool and lubricate the bearings and comes from the main turbine lubricating-oil pump.

Each bearing also has a high pressure jacking oil supply at the bottom. This lifts the shaft when starting from rest, until speed is high enough for hydrodynamic lubrication to start-up.

Instrumentation at each bearing normally gives:

- white metal temperature
- lubricating oil outlet temperature and inlet pressure
- jacking oil pressure
- vertical and horizontal vibration

6.7.1 Hydrodynamic Lubrication

(This section is taken from Williams)

Hydrodynamic bearings depend on the presence of a converging, wedge-shaped gap into which a viscous fluid is dragged by the relative motion of two surfaces. A pressure is generated which tends to push the surfaces apart. This balances the load on the bearing.

Large rotating machinery utilises hydrodynamic journal and thrust bearings. An analytical solution of their behaviour is complicated but the elements of behaviour can be understood by studying the simple two-dimensional pad bearing in Fig. 6.23.

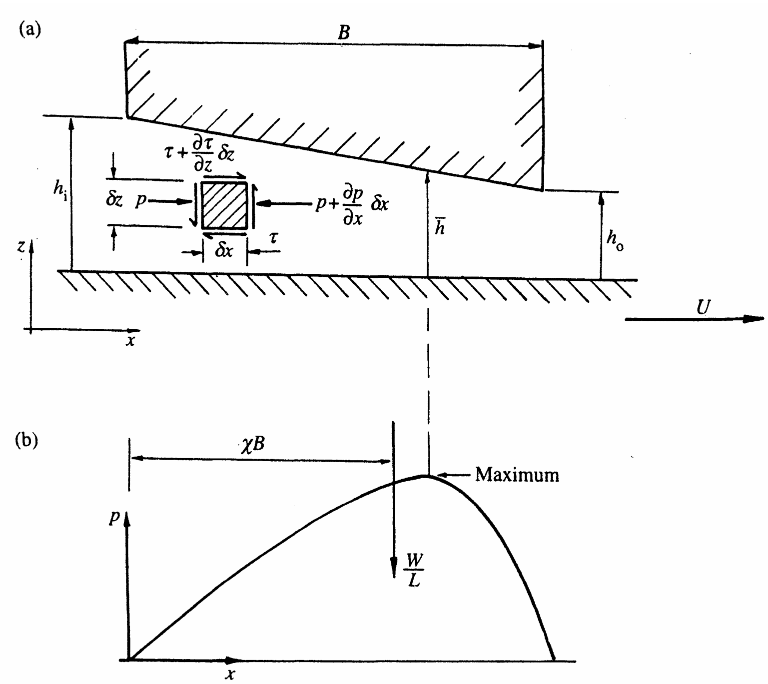


Fig. 6.23 Two-dimensional bearing pad (from Williams).

The bearing is long in the y-direction, so that there is no fluid flow normal to the plane of the paper. The upper, inclined fixed member is of length *B*, while the lower flat slider moves from left to right with velocity *U*. Fig. 6.23 also shows the pressure distribution in the viscous fluid. The integral of this pressure distribution supports *W/L*, the load per unit length into the page.

The angle of the wedge is greatly exaggerated in Fig. 6.23. It is typically only a quarter of a degree.

Consider the equilibrium of the small element of fluid within the gap in Fig. 6.23. The local film thickness is *h* and it varies in a known way from h_i at the entry to h_o at the exit. Gravity and inertia (acceleration) forces can be neglected.

Then

$$\frac{\partial p}{\partial x}\delta x\delta z=\frac{\partial \tau}{\partial z}\delta z\delta x$$

where

$p =$ pressure and $\tau =$ shear stress.

Hence $\dfrac{\partial p}{\partial x} = \dfrac{\partial \tau}{\partial z}$.

But $\tau = \eta \dfrac{\partial u}{\partial z}$

where $\eta =$ Newtonian viscosity

$u =$ local fluid velocity in x-direction.

So $\dfrac{\partial p}{\partial x} = \eta \dfrac{\partial^2 u}{\partial z^2}$

But $h \ll B$, so we can take p = constant across the film thickness, and p is a function of x only.

Therefore $\dfrac{dp}{dx} = \eta \dfrac{\partial^2 u}{\partial z^2}$.

This can be integrated twice to get

$u = \dfrac{dp}{dx}\dfrac{z^2}{2} + Az + C$ where A and C are constants of integration.

Setting boundary conditions, $u=U$ at $z=0$ and $u=0$ at $z=h$ enables evaluation of the constants of integration and leads to:

$$u = \frac{1}{2\eta}\frac{dp}{dx} z\left(z-h\right) + \left(1-\frac{z}{h}\right)U \tag{6.1}$$

The volumetric flow rate q through a unit width (in the Oz-direction) can be obtained by integration of (6.1) across the film.

$$q = \int_0^h u dz = \frac{-h^3}{12\eta}\frac{dp}{dx} + \frac{Uh}{2} \tag{6.2}$$

This is independent of x for an incompressible fluid.

Let $h=\bar{h}$ where $\dfrac{dp}{dx}=0$.

$$\text{Then } q = \frac{U\bar{h}}{2} = \bar{U}\bar{h} \quad \text{where } \bar{U} = U/2 \tag{6.3}$$

Combining (6.2) and (6.3) gives

$$\frac{dp}{dx} = 12\eta\bar{U}\frac{h-\bar{h}}{h^3} \tag{6.4}$$

This is the simplest form of Reynolds' equation. If h is known as a function of x, (6.4) can be integrated to get $p(x)$ and the load carrying capability of the bearing. Fig. 6.24 shows the geometry of a journal bearing and the exaggerated 'wedge' of fluid. Analysis of this geometry is complicated by the finite length of a journal bearing and the flow out of its ends.

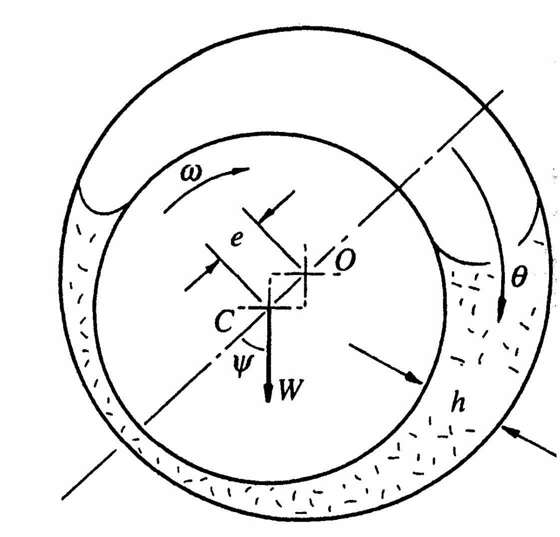


Fig. 6.24 Geometry of journal bearing (from Williams).

Fig. 6.25 also shows the configuration of an ideal steady state hydrodynamic film. If a vibratory load (e.g. due to rotating unbalance) is superimposed on the steady load (weight) the oil thickness can change and move around the circumference. This can lead to whirling of the journal (shaft) and affect the vibratory behaviour of the whole rotor line.

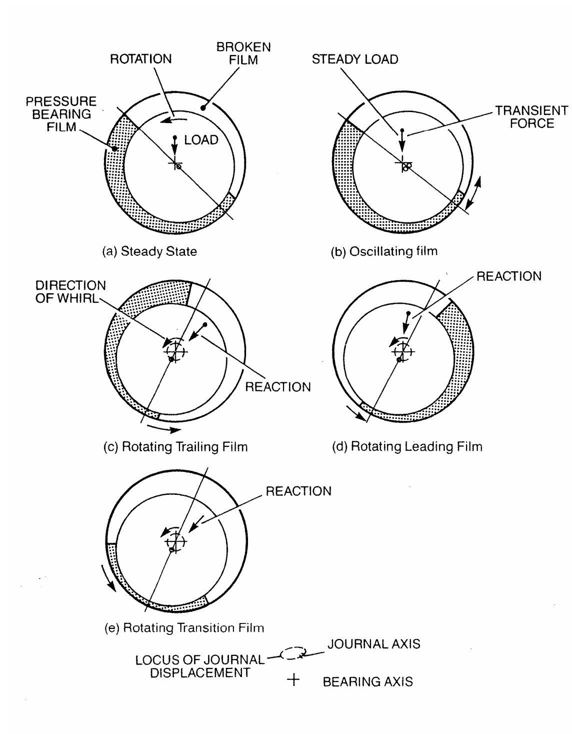


Fig. 6.25 Possible oil fil configurations (from MPSP).

6.7.2 Hydrodynamic Thrust Bearings

By pivoting the 'fixed' part of the bearing in Fig. 6.23, the angle of tilt will vary with load so that it is at the optimum for load-carrying capacity. This was the discovery of the Australian, A.G.M. Michell, and led to his famous thrust bearing design which is used in turbine/generator shaft lines. This is illustrated in Fig. 6.26.

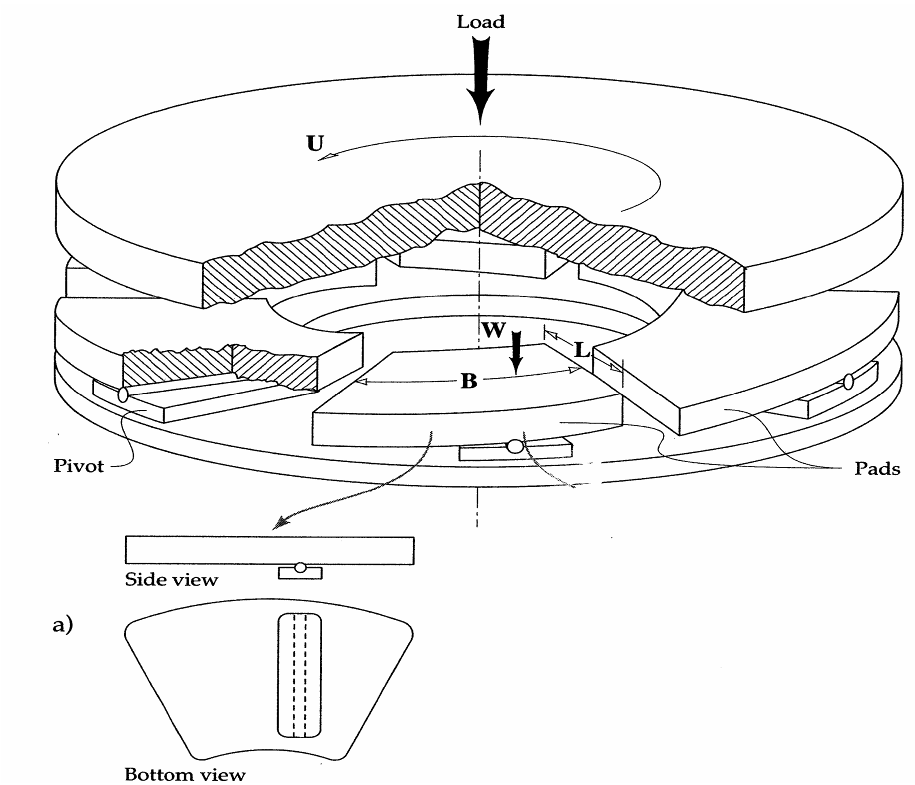


Fig. 2.26 Thrust bearing configuration (from Stachowiak and Batchelor).

A turbine thrust bearing is used to provide axial location for the turbine rotors relative to the cylinders. Because solid couplings are used, only one thrust bearing is used in the shaft line. It is usually located near areas where blade/cylinder clearances are a minimum. It is in two halves for ease of assembly.

Thrust bearings are of the Michell tilting pad design (Fig 6.27).

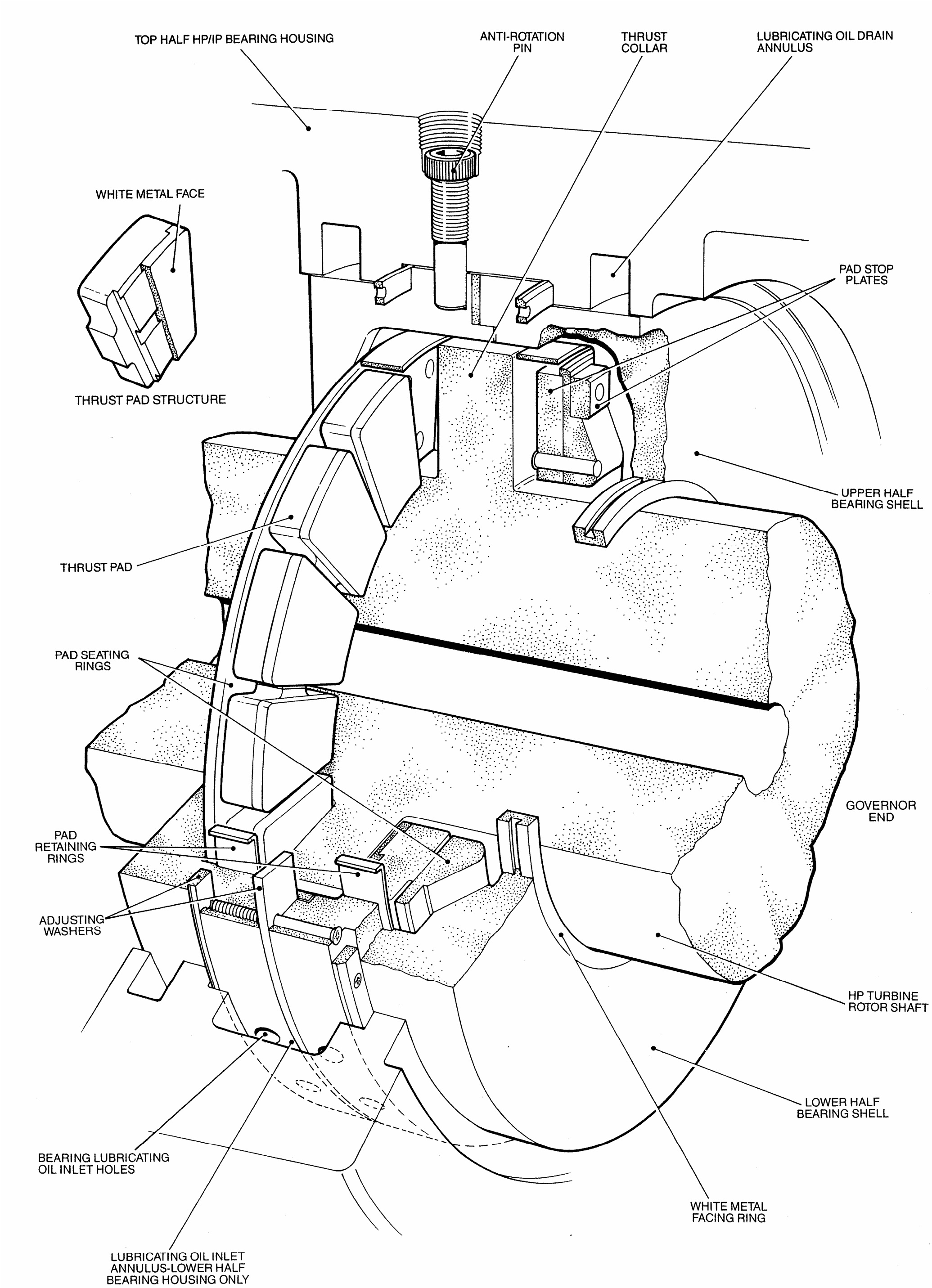


Fig. 6.27 Mitchell thrust bearing (from MPSP).

In normal operating conditions, net thrust is always in the one direction. However, a set of surge pads is incorporated on the opposite side of the shaft collar to take account of transient reversals of thrust which can occur during load reduction and a turbine trip.

6.8 Pedestals

Pedestals support the turbine rotor via the journal bearings in a fixed axial relationship with each cylinder so that gland clearances are maintained. They are usually fabricated in steel and stiffened by ribs and gussets plates.

In the LP area, pedestals are normally bolted and dowelled to the foundations. At the HP end, provision is made for the cylinders to expand by way of sliding points at the top or bottom of the pedestals.

6.9 Turning Gear

Turbine rotors must be turned continuously during warming up and cooling down processes. Otherwise distortion can occur. Usually there are two independent turning gear systems provided:

- A hand barring arrangement. A lever operates on a toothed wheel between two turbine rotors.
- Electric turning gear. An electric motor turns the rotors through a wormshaft and a wormwheel at less than 30 rpm.

6.10 Turbine Rotors

The shaft of each turbine rotor is a single, high quality alloy steel forging, machined to provide the required contours and functioning parts. Each end contains an integral coupling, gland seal area and bearing area. For HP and IP reaction turbines, axial grooves are machined into the rotor for the blades. For impulse HP and IP turbines and for LP turbines, wheels are machined or shrink-fitted onto the rotor with the blades mounted in grooves in the wheels.

The rotors of HP and IP turbines may have a centre bore machined in the shaft to remove impurities formed during the forging, and to allow access for ultrasonic inspection.

Alloy steels are chosen to have good creep resistance and high temperature and high fracture toughness.

6.10.1 Overspeed Testing

A 20% proof overspeed test is specified on all large turbine/generator rotors at the time of manufacture. This tests the forging against spontaneous fast fracture and confirms its balance.

6.10.2 Rotor Balancing

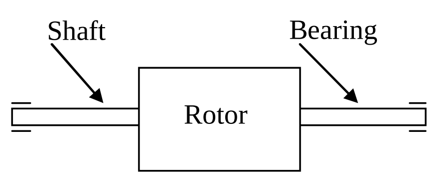
With the blade discs assembled, the rotor is balanced both statically and dynamically. Each blade disc is balanced individually before assembly. Rotors are dynamically balanced at low speed (400 rpm) with weight adjustments made in two planes, one at each end of the rotor. Provision is made to vary screwed plugs in tapped holes, or to add weights. The aim is to get $<25\mu$m amplitude of vibration at the bearing pedestals.

Modal behaviour must be understood for long rotors. These rotors are often balanced at running speeds and critical speeds in a vacuum chamber. When rotor flexibility is important, balancing is done at three or more planes.

On-site vibration testing can be done but it is affected by variations in the stiffness of the bearings, possible shaft misalignment and the coupling of the individually balanced rotors to form the complete shaft system. Access holes are provided in the casing.

6.10.3 Critical Speeds

A stationary shaft and rotor between bearings has a natural frequency of vibration with the shaft in bending. If the speed of rotation coincides with this natural frequency, any small unbalance can cause dangerous vibrations. This is a critical speed.



If the critical speed is below the running speed, the shaft is regarded as flexible. Care is needed to run up through this critical speed quickly. Modern units have rigid shafts, with critical speed above operating speed. With the long shafts in large units, large shaft diameters are needed.

However, each turbine does not act independently of others. There might be up to seven individual rotors in a shaft line. The bearings are hydrodynamic and so have flexibility which might increase with wear. As a result, there are then a number of critical speeds, two or three of which can be below the operating speed. A typical speed-vibration curve from an instrumented bearing housing is shown in Fig. 6.28.

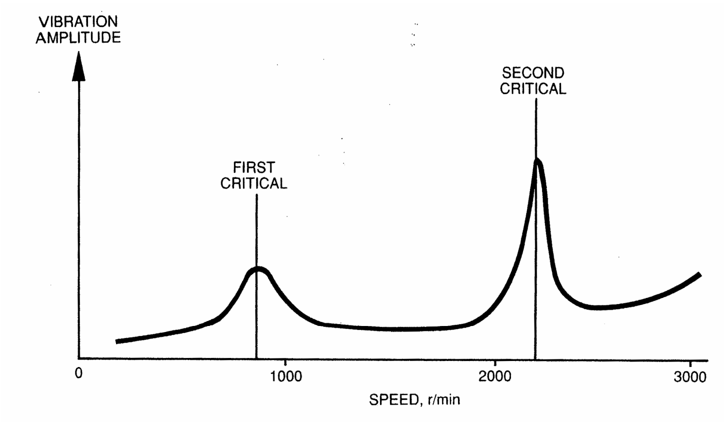


Fig. 6.28 Typical speed-vibration curve at a pedestal (from MPSP).