STEAM TURBINES — THEIR CONSTRUCTION, SELECTION AND OPERATION

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Introduction

The first steam engine built by James Watt in the year 1769 was the advent in substituting the low energy rates produced by wind, water, man and beast for the higher mechanical power produced by a machine. A further milestone was in the year 1866 when Werner von Siemens invented the principle of producing electricity from a rotating machine, the so-called electro-magnetic principle. Coupled to the steam engine this had the advantage of producing power centrally and making it available at a large number of points.

The steam engine and also to a latter extent the diesel engine had a limited capacity in producing power, due to the inherent disadvantages of reciprocating machinery. This led to the introduction of rotating machinery to produce the steadily increasing needs for electricity; already well known in the harnessing of water energy the principle of blading was adopted in the steam and at a later date in the gas turbines.

I. Theory of Steam Turbines

The steam turbine obtains its motive power from the change of momentum of a jet of steam flowing over a curved vane. The steam jet, in moving over the curved surface of the blade, exerts a pressure on the blade owing to its centrifugal force. This centrifugal pressure is exerted normal to the blade surface and acts along the whole length of the blade. This fundamental is shown diagrammatically in Fig. 1. The resultant of these centrifugal pressures, plus the effect of change of velocity, is the motive force on the blade.

This introductory paragraph regarding the transfer of energy from the fluid to the blade can also be shown mathematically.

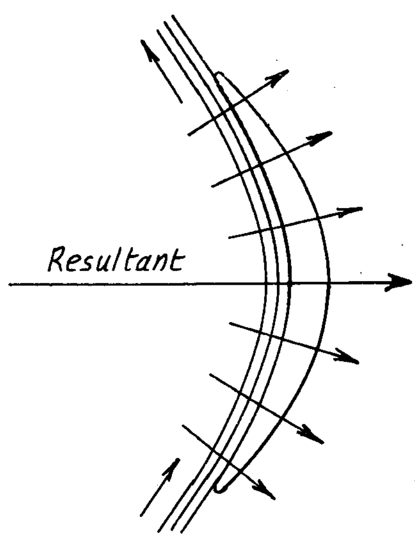


FIGURE 1: Pressure on blade

Fig. 2 shows the general view of a rotor mounted on a shaft, Figs. 2a and 2b the side and end elevations. V_1 is the absolute uniform velocity of the fluid entering the rotor passage, while V_2 is the absolute uniform velocity leaving the rotor passage. Components of V_1 and V_2 are conveniently considered in 3 directions, i.e. axial, radial and tangential.

Torque $$T = \frac{G}{g}(V_{1t} \cdot R_1 - V_{2t} \cdot R_2) \quad \ldots\ldots\ldots\ldots\ldots 1$$

Energy $$E = T \times w = \frac{G}{g}(V_{1t} \cdot R_1 \cdot w - V_{2t} \cdot R_2 \cdot w)$$

$$= \frac{G}{g}(V_{1t} \cdot u_1 - V_{2t} \cdot u_2) \quad \ldots\ldots\ldots\ldots\ldots 2$$

This leads to the so-called Euler equation for turbo machinery.

If $E > 0$ i.e. $V_{1t} \cdot u_1 - V_{2t} \cdot u_2 > 0$ work is done by the fluid as is the case for a turbine

If $E < 0$ i.e. $V_{1t} \cdot u_1 - V_{2t} \cdot u_2 < 0$ work is done on the fluid as is the case for compressor and fans.

Equation 2 may be transformed in terms of blade absolute and relative velocities determined from the physical conditions, i.e. flow area, flow volume and blade angles. We shall consider a 2-dimensional flow as occurs in practically all turbo machinery.

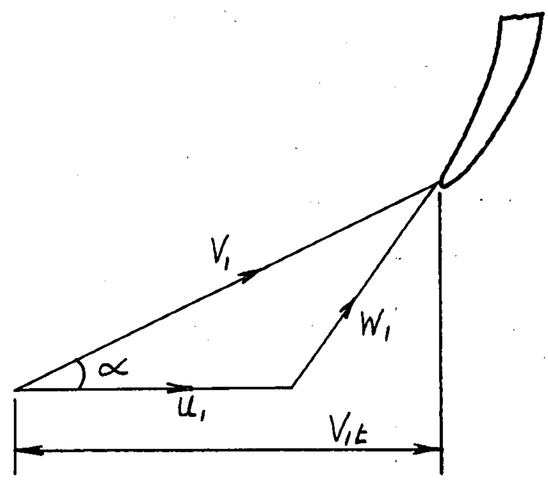


FIGURE 3: 2-dimensional flow

From Fig. 3——

$$W_1^2 = V_1^2 + U_1^2 - 2U_1V_1 \cos\alpha$$

but $V_1 \cos\alpha = V_{1t}$

thus $U_1 V_{1t} = \frac{1}{2}(V_1^2 + U_1^2 - W_1^2)$

A similar expression can be derived for the rotor blade exit.

thus $U_2 V_{2t} = \frac{1}{2}(V_2^2 + U_2^2 + W_2^2)$

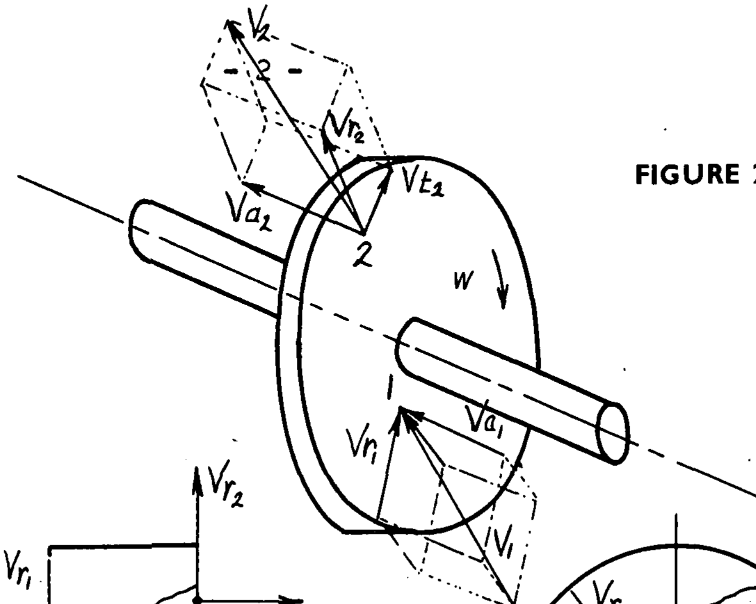


FIGURE 2: Rotor mounted on shaft

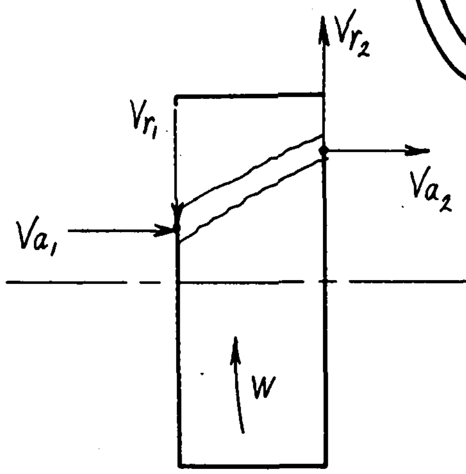


FIGURE 2(a): Side elevation

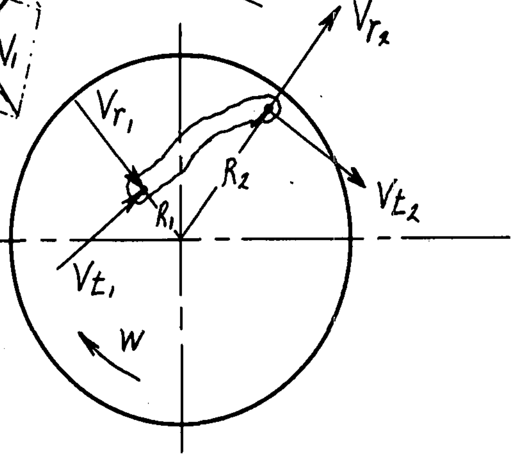


FIGURE 2(b): End Elevation

Substituting in Equation 2:

$$E = \frac{G}{2g}\left[(V_1^2 - V_2^2) + (U_1^2 - U_2^2) + (W_2^2 - W_1^2) \right] \ldots\ldots 3.$$

This ultimate form of the fundamental equation is broken down into 3 components:

$$a)\ \frac{G}{2g}\ (V_1^2 - V_2^2).$$

This represents the absolute kinetic energy change in the fluid as it passes through the rotor (velocity head).

$$b)\ \frac{G}{2g}\ (U_1^2 - U_2^2).$$

This represents the change in static head due to the centrifugal effect (axial flow $U_1 = U_2$).

$$c)\ \frac{G}{2g}\ (W_2^2 - W_1^2).$$

Change in static head due to diffusion or expansion process in the flow passages, i.e. area increasing in the direction of flow, therefore, relative velocity decreases and static pressure increases.

Types of Turbines

There are two types of steam turbines; impulse and reaction. There is a distinct difference in the working of these two types, and manufacturers of steam turbines usually specialise in the production of one of these types only. The main distinction is the manner in which the steam is expanded as it passes through the turbine. In the impulse turbine, the steam is expanded in nozzles and remains at constant pressure when passing over the blades. In the reaction turbine, the steam is continually expanding as it flows over the blades.

The original steam turbine, the De Laval, was an impulse turbine having a single-blade wheel. Other impulse turbines are known as Curtis, Zoëlly and Rateau.

The reaction turbine was invented by Sir Charles Parsons and is known as the Parsons turbine.

In all turbines the blade velocity is proportional to the steam velocity passing over the blade. If the steam is expanded from the boiler pressure to the exhaust pressure in a single stage, its velocity is extremely high. If this high velocity is used up on a single-blade ring, it produces a rotor speed of about 30,000 r.p.m. which is too high for practical purposes. There are several methods of overcoming this high rotor speed, all of which utilise several blade rings. The following are the four principal methods used:

(*a*) *Compounding for velocity.* Rings of moving blades separated by rings of fixed blades, are keyed in series on the turbine shaft, Fig. 4. The steam is expanded through nozzles from the boiler—to the back-pressure, to a high velocity, and is then passd over the first ring of moving blades. Only a portion of the high velocity is absorbed, the remainder being exhausted on to the next ring of fixed blades, which change the steam direction without appreciably altering the velocity. The jet then passes on to the next ring of moving blades, the process repeating itself until practically all the velocity of the jet has been absorbed. It will be noticed that, due to the pressure remaining constant as the steam passes over the blades, the turbine is an impulse turbine.

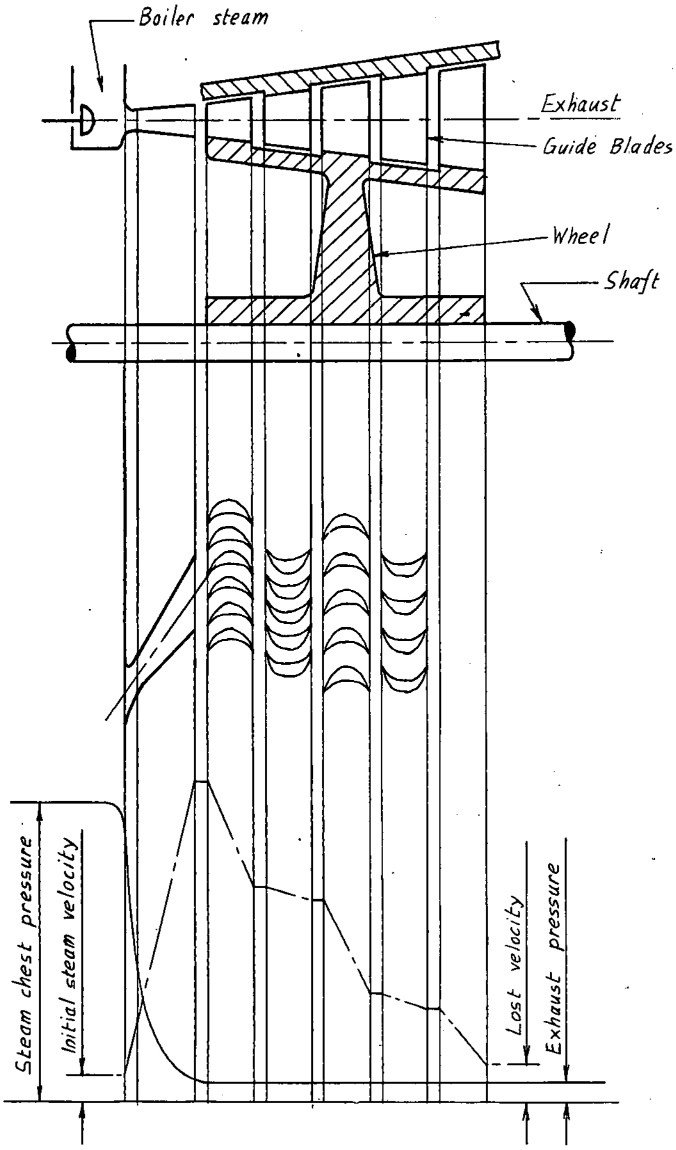


FIGURE 4: Compounding for velocity

(*b*) *Compounding for pressure.* In this type, the total pressure drop of the steam does not take place in the first nozzle ring, but is divided up between all the nozzle rings. The steam from the boiler is passed through the first nozzle ring in which it is only partially expanded. It then passes over the first moving blade ring where nearly all of its velocity is absorbed. From this ring it exhausts into the next nozzle ring and is again partially expanded; this absorbs a further

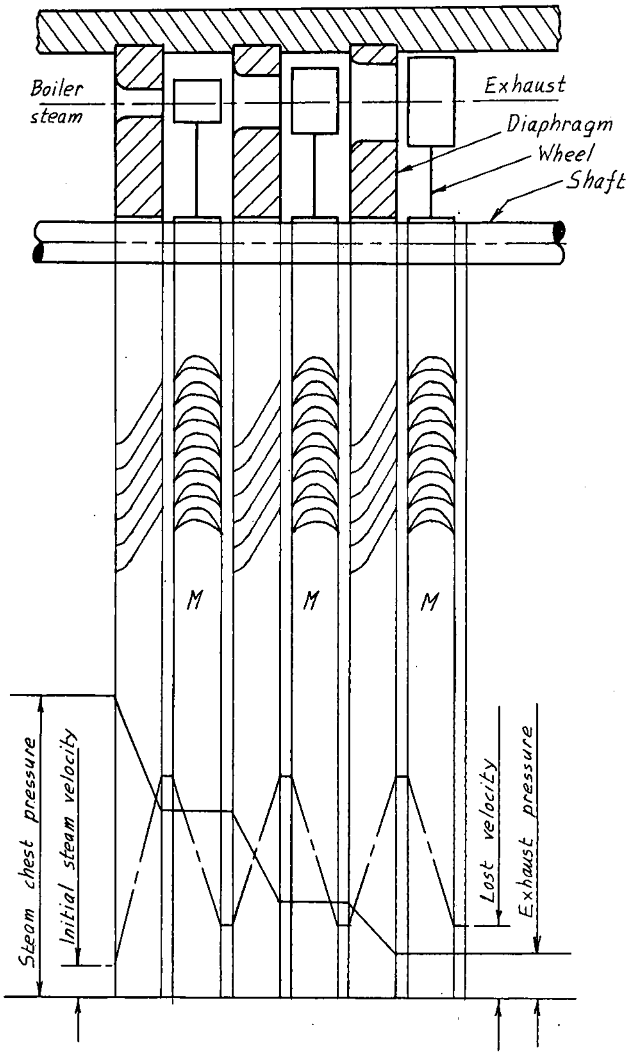


FIGURE 5: Compounding for pressure

portion of its total pressure drop. It then passes over the second ring of moving blades, the process thereby repeating itself. As the pressure remains constant during the flow over the moving blades, the turbine is an impulse turbine. This method of pressure compounding is used in Rateau and Zoëlly turbines.

(*c*) *Pressure-Velocity Compounding.* In this type of turbine, both of the previous two methods are utilised. This has the advantage of allowing a bigger pressure drop in each stage and, consequently, less stages are necessary, resulting in a shorter turbine for a given pressure drop. It may be seen that the pressure is constant during each stage; the turbine is, therefore, an impulse turbine. The method of pressure-velocity compounding is used in the Curtis turbine.

(*d*) *Reaction turbine.* In this type there is no sudden pressure drop; the pressure drop is gradual and takes place continuously over the moving and fixed blades. The fixed blades correspond to nozzles; they change the direction of the steam and, at the same time, allow it to

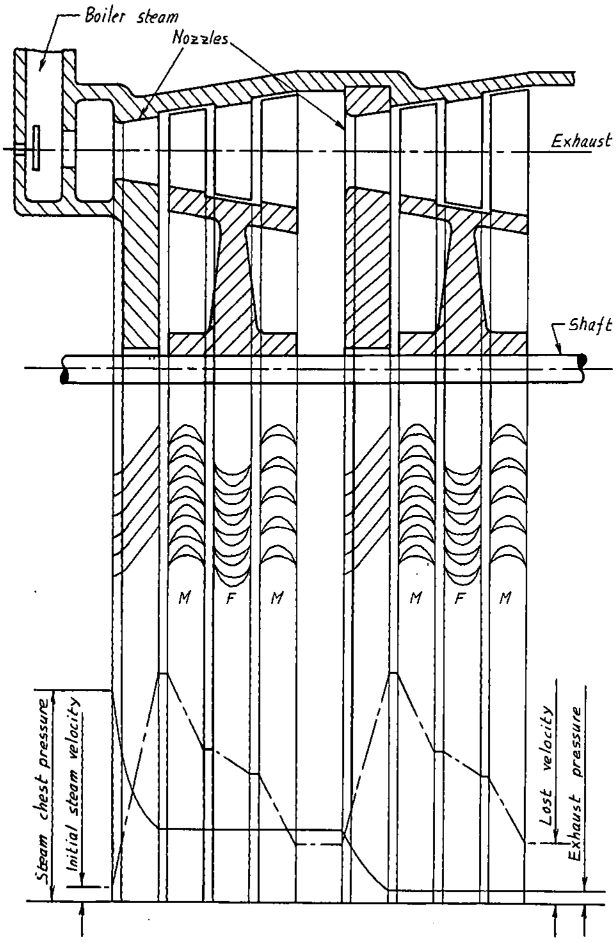


FIGURE 6: Pressure-Velocity Compounding

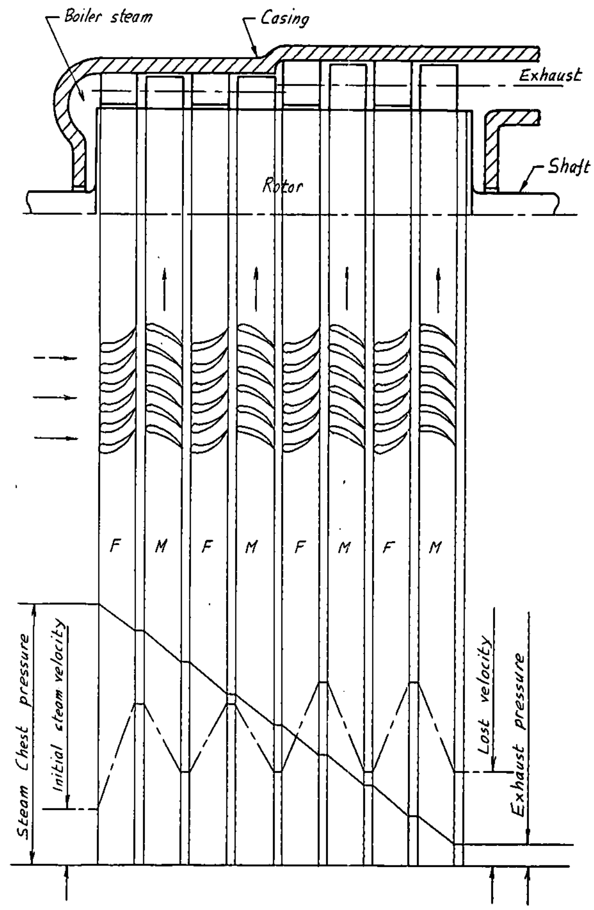


FIGURE 7: Reaction Turbine

expand to a higher velocity. The pressure of the steam falls as it passes over the moving blades; the turbine is, therefore, a reaction turbine.

The Thermo-dynamics of the Steam Turbine Elements

Nozzles:

The steam nozzle is a passage of varying cross-section by means of which the heat energy of steam is converted into kinetic energy. The nozzle is so shaped that it will perform this conversion of energy with the minimum loss. The flow of steam through a nozzle may be regarded, in its simplest form, as being an adiabatic expansion. The steam enters the nozzle with a relatively small velocity and a high initial pressure; the initial velocity is so small compared with the final velocity that it may be neglected.

Let I_{s1} = total heat of steam entering nozzle.

I_{s2} = total heat of steam at any section considered.

v = velocity of steam at section considered in ft. per sec.

u = heat drop during expansion.

$= I_{s1} - I_{s2}$

Then, assuming a frictionless adiabatic flow and considering 1 lb. of steam,

Gain of kinetic energy = heat drop.

$$\frac{v^2}{2g \times J} = u$$

$$v = \sqrt{2g.\ J.u.}$$

$$= \sqrt{2 \times 32.2 \times 778 \times u}$$

$$= 224\sqrt{u} \quad \text{(ft./sec.} \quad \ldots\ldots\ldots\ldots \quad 4$$

In practice, there is a loss in the nozzle, due to friction, of about 10 to 15% of the total heat drop; the effect of this is to reduce the value of u in equation 4.

$$\text{Thus:} \quad v = 224\sqrt{k.u.} \quad \ldots\ldots\ldots\ldots \quad 5$$

The effect of the friction of the nozzle is to reduce the velocity of the steam, and to increase its final dryness or super heat.

Weight of Discharge through Nozzle:

The adiabatic flow of the steam through the nozzle may be approximately represented by the equation

pv^n = constant

where n = 1.135 for saturated steam.

= 1.3 for super-heated steam.

Now, the work done during the cycle will be given by

$$\frac{n}{n-1}\{p_1 v_1 - p_2 v_2\}$$

Then, gain of kinetic energy=work done during the cycle

$$\frac{V^2}{2g} = \frac{n}{n-1}\left\{p_1 v_1 - p_2 v_2\right\}$$

$$= \frac{n}{n-1} \cdot p_1 v_1 \left\{1 - \frac{p_2 v_2}{p_1 v_1}\right\} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 6$$

But: $p_1 v_1^{\,n} = p_2 v_2^{\,n}$

Hence: $\left(\frac{v_2}{v_1}\right)^n = \frac{p_1}{p_2}$ or $\frac{v_2}{v_1} = \left(\frac{p_2}{p_1}\right)^{-\frac{1}{n}}$ ……… 7

Substituting this value in 6:

$$\frac{V^2}{2g} = \frac{n}{n-1} \cdot p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

Hence: $$V = \sqrt{2g\left(\frac{n}{n-1}\right) p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]} \quad \ldots\ldots\ldots \quad 8$$

Let A = area of cross section of nozzle.

v_2 = volume of 1 lb. of steam at absolute pressure p_2

W = weight of steam discharged through nozzle per sec.

Then, $$W = \frac{\text{volume of flow per sec.}}{\text{volume of 1 lb. of steam}} = \frac{AV}{v_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 9$$

From Equation 7:

$$v_2 = v_1 \left(\frac{p_2}{p_1}\right)^{-\frac{1}{n}}$$

Substituting the values of v_2 and V in 9:

$$W = \frac{A}{v_1\left(\frac{p_2}{p_1}\right)^{-\frac{1}{n}}} \sqrt{2g\left(\frac{n}{n-1}\right) \cdot p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

$$W = \frac{A}{v_1}\sqrt{2g\left(\frac{n}{n-1}\right) \cdot p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]} \quad \ldots\ldots \quad 10$$

The equation 10 can be used to obtain the flow of steam through the turbine because:

- A = cross-sectional area of nozzle.
- n = either 1.135 or 1.3.
- p_1 = initial or live steam pressure.
- v_1 = initial or live steam volume per lb.
- p_2 = discharge or wheel-chamber pressure.
- v_2 = discharge or wheel-chamber volume per lb.

Further relationships as regards nozzles are obtainable, i.e. throat pressure for maximum discharge, but these are nozzle details with which we are not directly concerned.

Theory of Blading:

The most important turbine elements are the blades. The following thermo-dynamic approach is meant to briefly show the different efficiencies obtainable with certain blade combinations, and the importance of the so-called speed ratio.

(*a*) *The Simple Impulse Stage:*

The velocity diagram indicates, as shown in Fig. 8, the velocities involved.

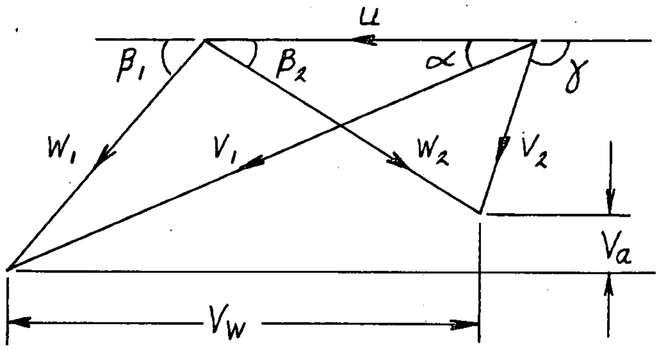


FIGURE 8: Impulse stage velocity diagram

Notations used:

- v_1 = nozzle jet velocity.
- w_1 = relative steam velocity at inlet.
- v_2 = absolute steam velocity at exit.
- w_2 = relative steam velocity at exit.
- u = mean peripheral velocity of blades.
- α = nozzle jet angle.
- β_1 = inlet angle of blade.
- β_2 = outlet angle of blade.
- γ = absolute angle of steam leaving blade.

The useful tangential propelling force in the direction of the blade motion:

$$F_t = \frac{G}{g} \cdot v_w \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 11$$

Expression 11 can be expressed in function of

v_1, α, ρ, β_1 and β_2.

Now: $$v_w = w_1 \cos\beta_1 + w_2 \cos\beta_2$$

$$= w_1 \cos\beta_1 \left(1 + \frac{w_2 \cos\beta_2}{w_1 \cos\beta_1}\right)$$

Let $\frac{w_2}{w_1} = k_b$, $\frac{\cos\beta_2}{\cos\beta_1} = c$

Thus: $v_w = w_1 \cos\beta_1 (1 + k_b \cdot c)$

But: $w_1 \cos\beta_1 = v_1 \cos\alpha - u$

Thus: $$v_w = (v_1 \cos\alpha - u)(1 + k_b \cdot c)$$

$$= v_1 \left(\cos\alpha - \frac{u}{v_1}\right)(1 + k_b \cdot c)$$

Were: $\rho = \frac{u}{v_1}$

thus substituting ρ and v_w in Expression 11:

$$F_t = \frac{G}{g} v_1 (\cos\alpha - \rho)(1 + k_b \cdot c) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 12$$

To obtain the efficiency of the blading the energy E (see Equation 2) may be divided by the kinetic energy of the jet issuing from the nozzle.

$$\text{Efficiency}\ \eta_b = \frac{E}{\frac{G}{g} \cdot \frac{v_1^2}{2}} = \frac{\frac{G}{g} \cdot u \cdot (v_1 t - v_2 t)}{\frac{G}{g} \cdot \frac{V_1^2}{2}} = \frac{2uv_w}{v_1^2}$$

Substituting the value of v_w obtained above:

$$\eta_b = \frac{2 . u . v_1 \left(\cos\alpha - \frac{u}{v_1} \right) (1 + k_b c)}{v_1^2}$$

$$= 2 (\rho \cos\alpha - \rho^2) (1 + k_b c) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 13$$

For maximum η_b differentiate Equation 13:

$$\frac{d\eta_b}{d\rho} = 2 \cos\alpha - 4\rho = 0$$

$$\rho_{opt.} = \frac{\cos\alpha}{2}$$

Substituting in Equation 13:

$$\eta_{b\,max.} = 2 \left(\frac{\cos^2\alpha}{2} - \frac{\cos^2\alpha}{4} \right) (1 + k_b . c)$$

$$= \frac{\cos^2\alpha}{2} (1 + k_b . c)$$

Assuming symmetric blades ($c = 1$) and frictionless flow ($k_b = 1$)

$$\eta_{b\,max.} = \frac{\cos^2\alpha}{2}$$

i.e. for a simple impulse stage or turbine having one impulse row of blades, the maximum blade efficiency can be determined at a corresponding speed ratio.

Impulse Turbine Staging or Compounding:

The simple impulse turbine is limited in its application when a large pressure drop is necessary, because the high nozzle velocities resulting implies high blade speed with related blade and disc stress problems.

Four already discussed methods are available to cope with large pressure drops at reasonable blade speeds. These are:

(*a*) Compounding for velocity.
(*b*) Compounding for pressure.
(*c*) Pressure-velocity compounding.
(*d*) Reaction turbine.

As determined above for the simple impulse stage, ρ *opt.* can also be theoretically determined for the blade configuration mentioned under (*a*) to (*d*). As the principle is the same, only the results shall be stated. The results are based on the assumption of frictionless flow.

For a) $\rho_{opt.} = \frac{1}{m_v} \cdot \frac{\cos\alpha}{2}$

b) $\rho_{opt.} = \frac{1}{\sqrt{m_p}} \cdot \frac{\cos\alpha}{2}$

c) $\rho_{opt.} = \frac{1}{m_v \sqrt{m_p}} \cdot \frac{\cos\alpha}{2}$

d) $\rho_{opt.} = \frac{1}{\sqrt{m_r}} \cdot \cos\alpha$

A comparison of the above results shall help to explain certain features of a definite design.

Comparing velocity and pressure staging, i.e. (*a*) and (*b*) shows the following:

Velocity Compound:

To reduce u to $\frac{1}{mu}$ we require m stages.

Efficiency lower than pressure staging because of higher residual kinetic energy and friction losses.

Advantage of taking large pressure drop in one step and thus reducing the pressure and temperature at which the blades run.

Cheaper (shorter) construction.

Pressure Compound:

To reduce u to $\frac{1}{mu}$ we require m^2 stages.

Lack of simplicity and construction but advantage of uniform distribution of work among stages, K.E. loss $= \frac{1}{m}$ of velocity compound turbine.

More expensive construction, thus higher initial cost, due to longer turbine (more stages required).

To correct the defects of the Curtis and add the qualities of the Rateau, (*c*) may be used, but in practice is very seldom found due to the complicated design and a relatively low efficiency.

A more common arrangement is to provide on the high pressure side one or more Curtis stages, followed by Rateau or Reaction staging. The Curtis stages reduce the pressure and temperature of the fluid to a

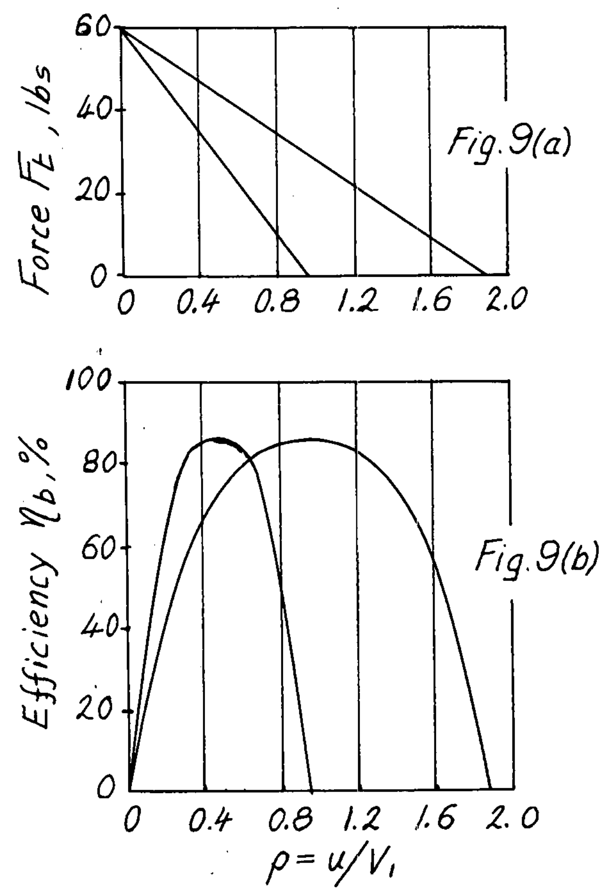


FIGURE 9: Impulse-Reaction Comparison

moderate level with a high proportion of work per stage, and then the more efficient Rateau or Reaction staging absorb the balance of the energy available.

Comparing frictionless conditions, the maximum efficiencies for the simple impulse, Curtis and Reaction blading are equal; however, when friction is taken into account, the reaction stage is found to be the most efficient, followed by Rateau and Curtis in that order. The reason that friction losses are less significant in the reaction stage lies in the fact that the flow velocities are lower.

Fig. 9(a) shows how F_t varies with ρ for both an impulse and reaction turbine. Both types develop the same force at $\rho = 0$, the force dropping to zero at twice ρ for the reaction turbine.

Fig. 9(b) compares blade efficiency for impulse and reaction turbines for the given conditions. Maximum efficiency developed at slightly less than $\rho = 1$ for the reaction turbine, just twice that for the impulse turbine.

Developed forces for both turbines are equal at their maximum blade efficiencies.

II. Selection of Steam Turbines

In the section, Selection of Steam Turbines, the different types of steam turbine shall be discussed on a more practical level so as to be of use to the works and planning engineer.

The Differences between the two most widely used Industrial Steam Turbine types

The two most widely used steam turbine types are the reaction turbine and the pressure-compounded or Rateau turbine. The high pressure part for both types consists of a simple or two-stage velocity-wheel, depending on the live steam conditions at the turbine entry. The impulse stage, known as the governing stage, is generally applicable since it alone permits partial steam admission to the moving wheel.

Both turbine types have inherent features which may be termed an advantage or disadvantage, as the case may be.

Characteristics of the Two Types

Rateau

The rotor is built as a disc rotor with a slender shaft. Between the individual discs, inner labyrinths or seals are arranged on a shaft of small diameter.

Due to the slender rotor construction, the lack of rigidity causes the critical speed or speeds of the rotor to be below the rated speed. In starting up or stopping, the rotor must pass through the critical speed ranges. Although, when running up to speed the critical speed ranges can be rapidly passed by quick opening of the throttle valves, there is no way to accelerate the rate of speed decrease when shutting down. The measure against excessive vibration at the critical speed is to provide adequate clearance between diaphragm and shaft.

The stationary blades and the shaft seals are inserted into half split diaphragms which are, in turn, attached to the casing.

Stationary and moving blades have different profiles. They impose a considerable change in direction of the steam flow.

Since the steam, as it passes through the turbine, is subjected to a heavy rotational motion around the rotor, the steam path at the root and tip of the blades is not uniformly filled with steam.

The friction heat generated at the shaft seals, due to rubbing, is carried off by the leakage steam only.

Reaction

The rotor is of the so-called solid type. The pressure difference between the inlet side and outlet side of the stationary blade is half that of an impulse stage. The sealing thus presents less difficulty.

The rotor of the reaction turbine is rigid, the critical speeds lying above the rated speed. Starting and shutdown require no special precautionary measures against excessive vibrations.

The stationary blades are directly inserted into the casing.

Stationary and moving blades have similar profiles. They impose a smaller change in the direction of the steam flow.

The steam path is uniformly filled with steam, since the steam particles pass mainly in axial direction through the turbine, the rotational motion around the rotor being less.

The friction heat generated at the seals is carried off by the whole working steam quantity.

A few of the above differences between the two types of turbine should not propagate any one design, but should point out inherent differences in construction of both types.

Steam Turbine Selection

In the previous sub-section, the internal characteristics of the two most widely applied turbine types or turbine constructions, were briefly discussed. In this sub-section, the type of turbine refers not to the internal construction, but to the turbine as a whole unit.

Four basic types of turbine are available:

1. Back-pressure turbines expand the live steam supplied by the boiler to the pressure at which the steam is required for the process. The overall plant efficiency of a back-pressure turbine exhausting to a process is high, due to the considerable heat losses through the condenser being eliminated. The electric power generated by the back-pressure turbine is directly proportional to the amount of process steam required. To avoid the direct relationship between back-pressure steam and power, the alternator would have to be connected to the grid, or a by-pass valve installed.

2. Extraction back-pressure turbines are employed where the process steam is used at two different pressure levels. The operating of this type of turbine is similar to the back-pressure machine, a high overall plant efficiency being obtainable. Difficulties exist in the control of such a turbine type, two by-pass valves being necessary to maintain the process pressures if the alternator is not coupled to the grid.
3. The condensing turbine is used where process steam is not necessary. This turbine has a lower overall efficiency due to the loss of heat in the steam condenser, and it is difficult to obtain cheaper station cost per kilowatt-hour than buying electric power, due to the inherent disadvantages of a relatively small condensing turbo-alternator set.
4. The extraction-condensing turbine is used where the power required is in excess of the process steam. The efficiency of this type is lower than a back-pressure turbine due to the partial loss of heat in the condenser. However, this turbine has the advantage that the power generated is not proportional to the process steam required and, furthermore, power can be obtained whilst the process in the factory is shut down.

Using this short introduction, the next logical step is to describe the different turbine types in a thermodynamic language using the Mollier diagram, this showing the relationship between the enthalpy drop of steam and the power obtainable. The turbine shall also be integrated into a steam cycle, showing the employment of reduction valves and related plant material.

1. Back-Pressure Turbine

Fig. 10(a) shows the back-pressure turbine incorporated into the steam cycle. Points 1 and 2 signify the live steam and the back-pressure steam condition respectively. Due to various losses in the turbine, the actual work done h_u will be less than the isentropic or theoretical work h_s. The relationship is called the internal efficiency and may be written thus:

$$\eta_i = \frac{h_u}{h_s}$$

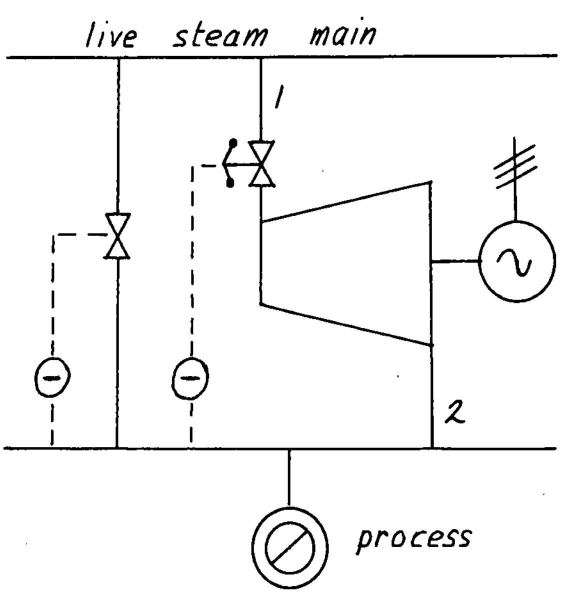
The heat equivalent of the actual work per lb. of steam may be calculated as follows, knowing that 1 kW-h = 3,416 B.T.U.

$$h_u = \frac{kW \times 3416}{G} \quad \text{(B.T.U./lb.)}$$

$$N = \frac{G \cdot h_s \cdot \eta_i}{3416} \quad \text{(kW)} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 14$$

For a turbine installation, h_s and η_i can for any particular load be considered constant, thus N is proportional to G, the quantity of process steam used. Knowing G and assuming η_i it is, therefore, possible to calculate the power obtainable for a specific live steam condition. If N obtainable is lower than N required, the balance can be obtained by coupling the alternator to the grid; vice versa, the excess can be supplied to the grid.

When coupled to the grid, the frequency, i.e. the turbine speed, is maintained by the grid, the back-pressure being maintained by the governor which opens or closes the nozzle valves in the live steam main admitting more or less steam. The steam quantity is thus always adjusted to the requirements of the



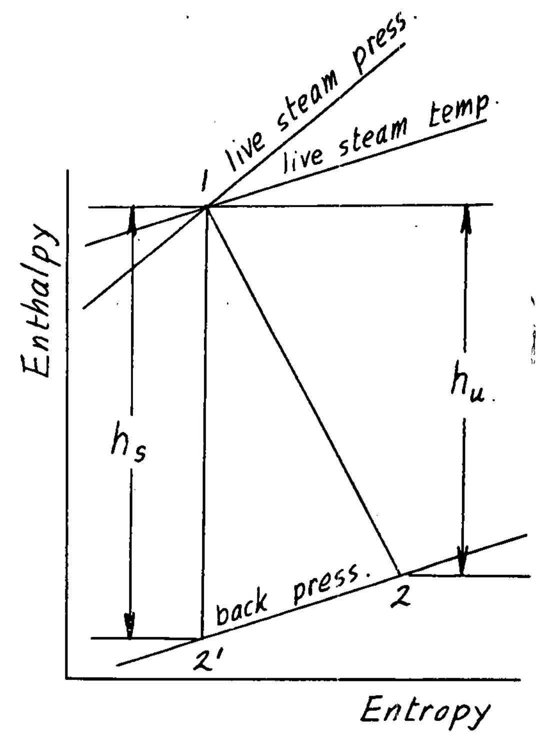


FIGURE 10(a): Steam cycle **FIGURE 10(b)**: Mollier diagram
for the back-pressure turbine

process, the power being balanced due to the grid connection.

In most cases, the turbine is not coupled to the grid, in which case, the governor has to maintain the speed, i.e. frequency of the turbine, at a constant level by adjusting the steam quantity to the power requirements. Here, the opposite is true, the power being the dominating factor, in which case the pressure of the back-pressure steam is maintained at a constant value by installing a by-pass and a blow-off valve.

2. Extraction Back-pressure Turbine

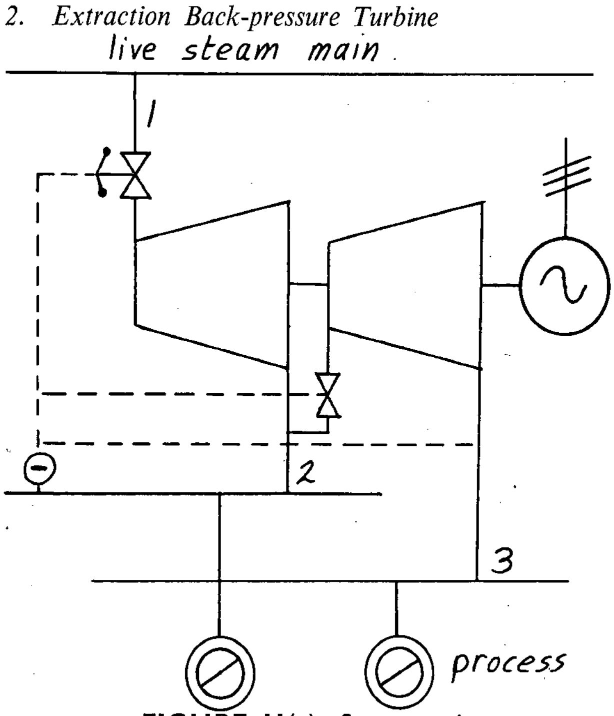


FIGURE 11(a): Steam cycle

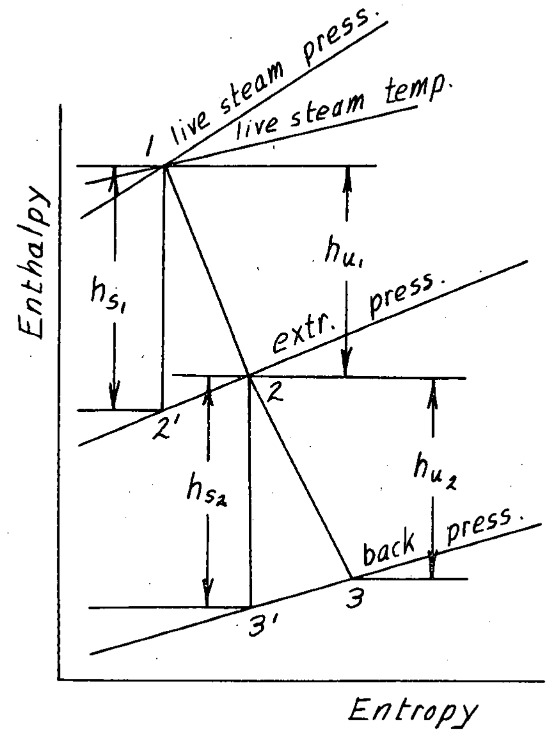


FIGURE 11(b): Mollier diagram for the extraction back-pressure turbine

Fig. 11(a) shows an extraction back-pressure turbine, Fig. 11(b) the steam expansion in the Mollier diagram. Let η_{i1} and η_{i2} be the internal efficiencies of the high and low-pressure parts respectively, and G_1 and G_2 the steam flows; then

$$N = \frac{G_1 \cdot h_{s_1} \cdot \eta_{i1}}{3416} + \frac{G_2 \cdot h_{s_2} \cdot \eta_{i2}}{3416} \text{ (kW)} \quad \ldots\ldots\ldots\ldots\ldots \quad 15$$

Generally, the amount of steam to be supplied to the process is known, let these amounts be termed E_1 and E_2.

Thus: $G_2 = E_2$

$G_1 = E_1 + E_2$

Substituting in Equation 15:

$$N = \frac{(E_1 + E_2) \cdot h_{s_1} \cdot \eta_{i1}}{3416} + \frac{E_2 \cdot h_{s_2} \cdot \eta_{i2}}{3416}$$

$$= \frac{E_2}{3416} \left(h_{s_1} \cdot \eta_{i1} + h_{s_2} \cdot \eta_{i2} \right) + \frac{E_1}{3416} \cdot h_{s_1} \cdot \eta_{i1}$$

The above equation shows that, for definite pressure drops and efficiencies, the amount of power obtainable is proportional to the process steam required.

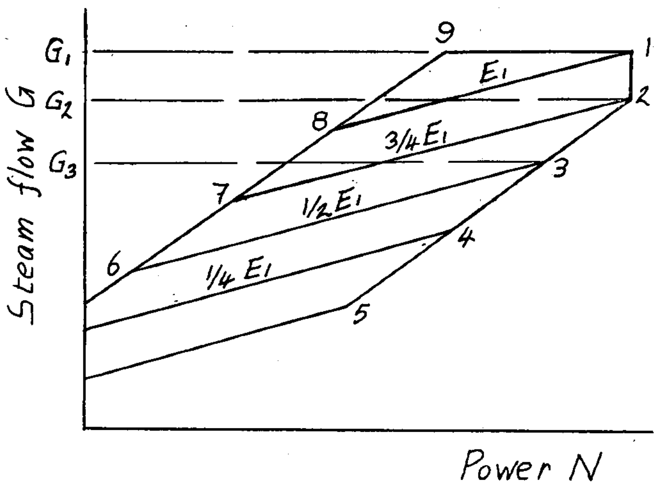
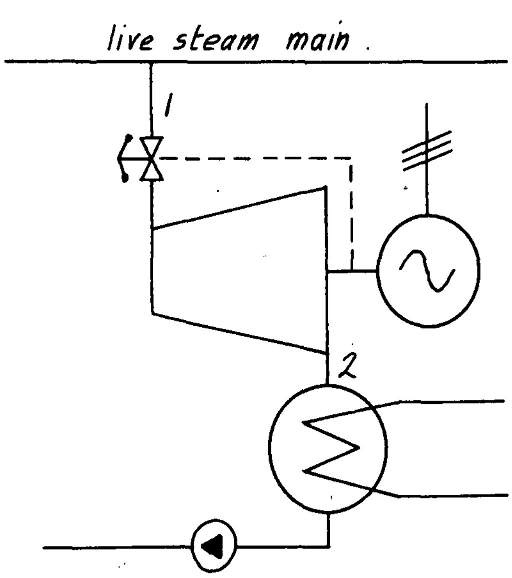


FIGURE 12: Steam consumption diagram

The relationship between the power obtainable and the steam quantities flowing, is shown in Fig. 12. This is a strongly simplified diagram, but may be used to explain the principles involved. Take, for instance, point 1. This point indicates that maximum power is obtainable by passing the steam quantity G_1 through the high-pressure part, and the quantity G_1–E_1 through the low-pressure part. At point 2, the maximum power is still obtainable, the high-pressure steam flow dropping to G_2, the low-pressure steam flow increasing to $(G_2-\frac{3}{4} E_1)$; the reduction in live steam is, therefore, balanced by the increased steam quantity flowing through the low-pressure part.

The process pressures are maintained, when the alternator is coupled to the grid, by the turbine governor receiving impulses from both the extraction and the back-pressure line. When the alternator is not



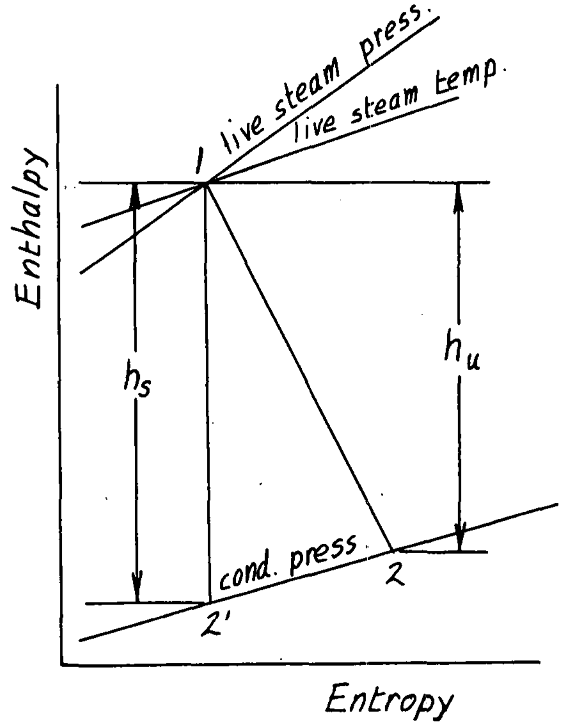


FIGURE 13(a): Steam cycle **FIGURE 13(b):** Mollier diagram
for the condensing turbine

coupled to the grid, the turbine governor has to maintain a constant turbine speed, the extraction and back-pressures being maintained at the desired values by a system of by-pass and blow-off valves.

3. Condensing Turbine

Condensing turbines are seldom used in industry due to the lower efficiency as compared with large power stations. Large power stations use an elaborate system of regenerative feed heating which is not possible with a small industrial condensing steam turbine. The steam is expanded to condenser pressure, the pressure of the exhaust steam being dependent on several factors, as shown in the basic equations given below:

Cooling surface A =

$$\frac{Q}{K} \log_e \frac{t_s - t_i}{t_s - t_o} \quad . \; . \; . \; 16$$

Where A = cooling surface in ft^2.
Q = cooling water quantity in lbs./hr.
t_s = steam exhaust temperature in ° F.
t_i = cooling water inlet temperature in °F.
t_o = cooling water outlet temperature in °F.
K = heat transfer rate in B.T.U./$ft.^2$ °F. hr.

Outlet cooling water temperature

$$t_o = t_i + \frac{WH}{Q} \quad . \; . \; . \; 17$$

Where W = Steam flow into condenser in lb/hr.
H = Heat rejected by steam in B.T.U./lb.

Generally, in calculating the condenser pressure, i.e. the value of H, the cooling water inlet temperature t_1 is known; the relationship of $\frac{Q}{W}$ may be assumed to be approximately 70 and t_o approximately 10° F greater than the inlet temperature of the cooling water. These results would give a value of H and, knowing the approximate expansion line of the steam, the exhaust point can be plotted on the Mollier diagram. With a certain amount of trial and error, the correct expansion end point is obtained. The condenser cooling surface can then be calculated, the value of K obtainable from suitable heat transfer curves.

To calculate the power N, the same basic formula as derived for the back-pressure turbine can be used:

$$\text{i.e. } N = \frac{G.\, h_s.\, \eta_i}{3416}$$

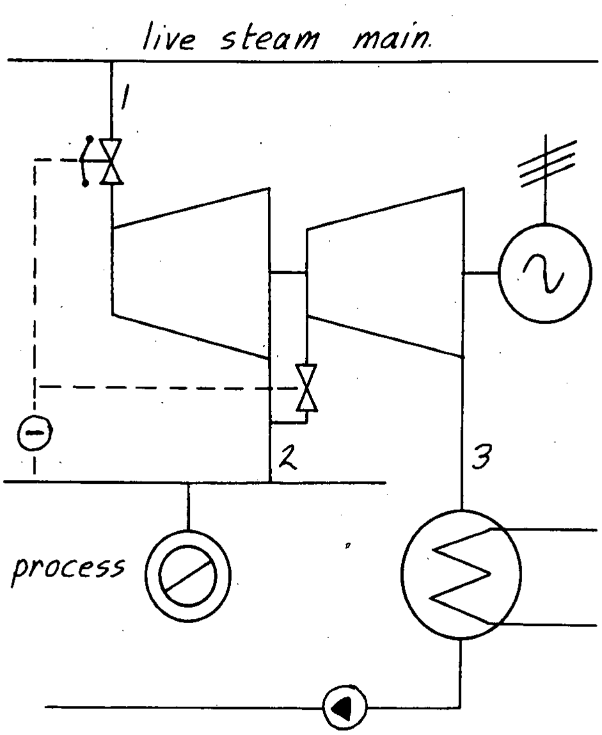
As against back-pressure turbines where the enthalpy drop remains constant at all loads, due to the back-pressure being regulated, the enthalpy drop for a condensing turbine changes with load. This may be explained by considering Formula 17. Assume t_o, t_i, and Q remain substantially constant, then, as W decreases with decreasing load, H would increase, i.e. the condenser pressure rises as the load decreases.

The regulation of condensing turbines presents no problem. If coupled to the grid, the governor can be used to keep the live steam pressure constant; if operated independently, the governor maintains the system frequency.

4. Extraction-condensing Turbine

The extraction-condensing turbine is a combination of the back-pressure and condensing types, the formulas obtained for the latter types may be employed on the extraction-condensing type.

$$\text{Thus power } N = \frac{G_1.\, h_{e_1}.\, \eta_{i_1}}{3416} + \frac{G_2.\, h_{e_2}.\, \eta_{i_2}}{3416} \quad \ldots\ldots\ldots\ldots \; 18$$



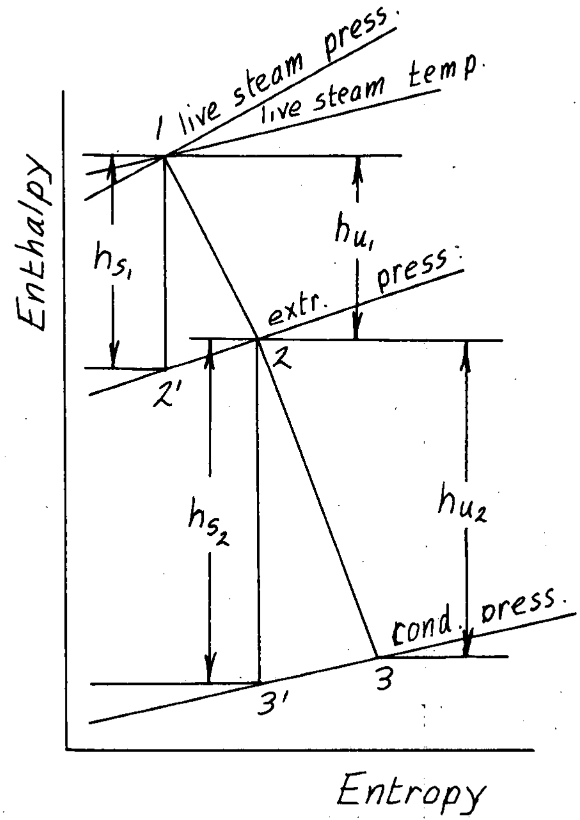


FIGURE 14(a): Steam cycle **FIGURE 14(b):** Mollier diagram
for the extraction condensing turbine

The extraction pressure is fixed by the pressure of the process steam required, the exhaust pressure determined as for the condensing turbine.

The extraction-condensing turbine is widely used, the principal advantage being that due to the condensing part, the power obtainable is not proportional to the process steam quantity required. In a factory where a considerable secondary load exists, or where power is required when the process part of the factory is shut down, the condensing part of the extraction-condensing turbine is able to supply the load. A typical example is a factory with a large irrigation load, the latter being secondary. The irrigation load generally has to be supplied throughout the year.

The relationship between power and steam is basically identical to that shown in Fig. 12 and shall not be repeated. The regulation would also be a repetition of previous explanations and need not be commented on.

Turbine Reduction Gears

The turbine ratings for industrial power plants lie between 1,000 and 10,000 kW, the lower and upper limit depending on the steam conditions. In the theoretical part of this paper, it has been mentioned that the large pressure drop in a single-blade ring would produce a rotor speed of about 30,000 r.p.m. To decrease this speed, compounding, i.e. employing a larger number of stages, is necessary.

The universal speed at which large two-pole alternators operate is 3,000 r.p.m. For industrial turbines, an economic solution has been found in an intermediate speed, generally ranging between 5,000 and 12,000 r.p.m., the latter speed being for turbines of small output. The reasons are obvious, but shall be enumerated as follows:

1. High speeds with the same stage-heat drops result in longer blades of smaller diameter.
2. The flow ducts have favourable dimensions.
3. The peripheral losses are reduced.

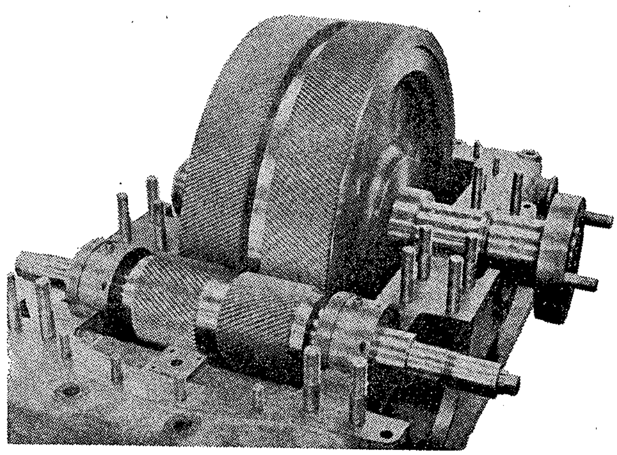


FIGURE 15: Helical spur reduction gear

4. The peripheral clearance is small and, therefore, the clearance losses are correspondingly low.
5. The use of lower speed alternators, generally of 1,500 r.p.m., which are cheaper than the two-pole type.

The reduction gears are generally of the double-helical spur gear type. The turbine and the reduction gear are connected by a flexible coupling, the reduction gear and the generator by a rigid coupling. Planetary gearing is also used; the advantage of co-axial shafts must be weighed against the complications introduced in connection with inspection work.

Turbine Standardisation

No manufacturer who hopes to offer an economically competitive and, at the same time, reliable turbine, can afford to disregard the value of standardisation. For all types of turbines, a definite range of standard sizes are designed. Each standard size represents a casing size, the size or standard casing used depends on the steam conditions and the quantity of steam flowing. The power output, for instance, may vary considerably for any one standard casing size.

The advantages are obvious; due to the completed designs and available drawings, the delivery time may be reduced considerably. The standard parts have proved themselves, so that a higher reliability may be expected. The problem of obtaining spare parts at short notice is non-existent.

Taking a standard turbine, it is possibly to classify the turbine parts into three groups:

1. Parts which have to be adapted to the thermodynamic conditions:

   nozzle valves; nozzles and blades of the impulse wheel; and reaction or Rateau blading.

2. Parts which are mass produced and may be used on several standard sizes:

   operating cylinder for valve gear; labyrinth-glands; thrust bearing; bearing pedestal; journal bearings; main oil pump, shaft driven; emergency stop valve; steam strainer; governing system; and long end-blades for condensing turbines.

3. Parts which are only usable for the standard size considered:

   turbine casing with integral steam inlet chamber; emergency stop valve casing incorporating the live steam inlet flange; exhaust casing incorporating the exhaust flange; turbine rotor; and coupling on alternator or reduction gear end.

The above parts list is by no means complete but it indicates the method of producing a standard turbine.

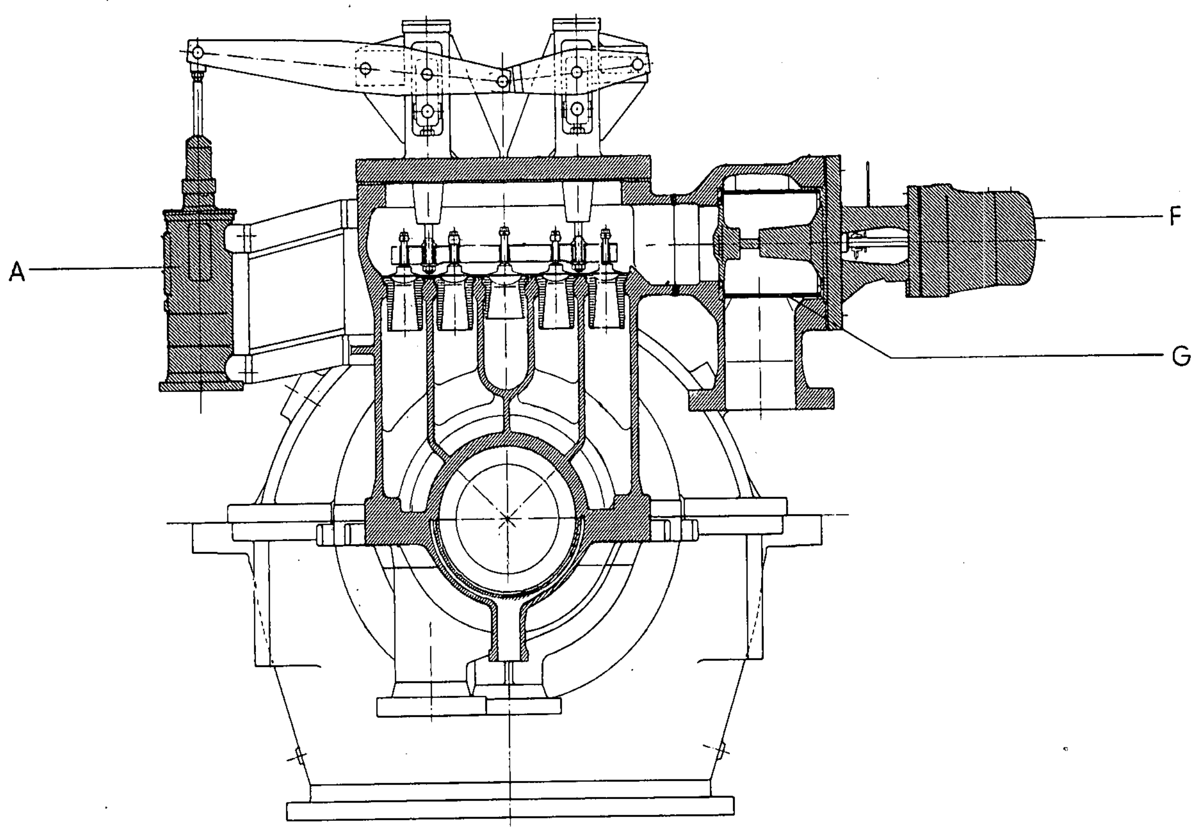
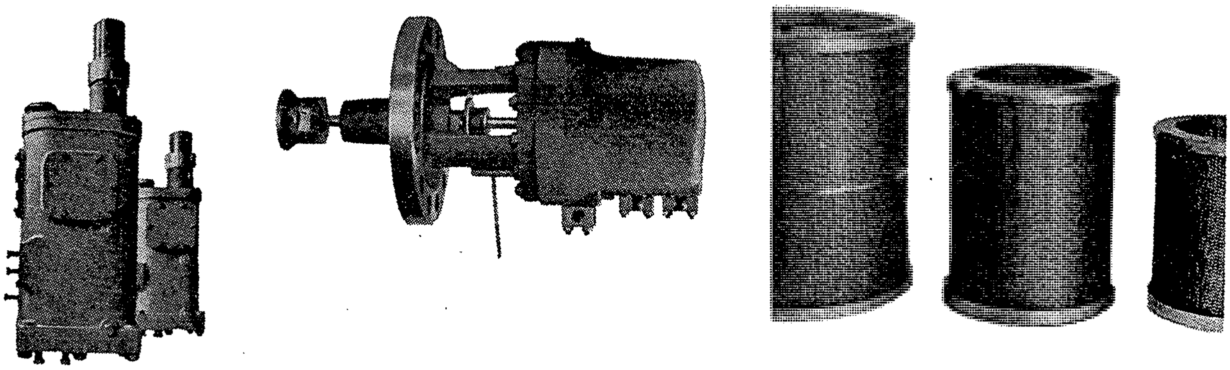


FIGURE 16(a): Turbine cross-section



A: Servo-motor **F:** Emergency stop-valve **G:** Steam strainer

Parts adapted to thermodynamic conditions

Mass produced parts

Parts for standard size considered

FIGURE 16: Standardisation of a condensing turbine

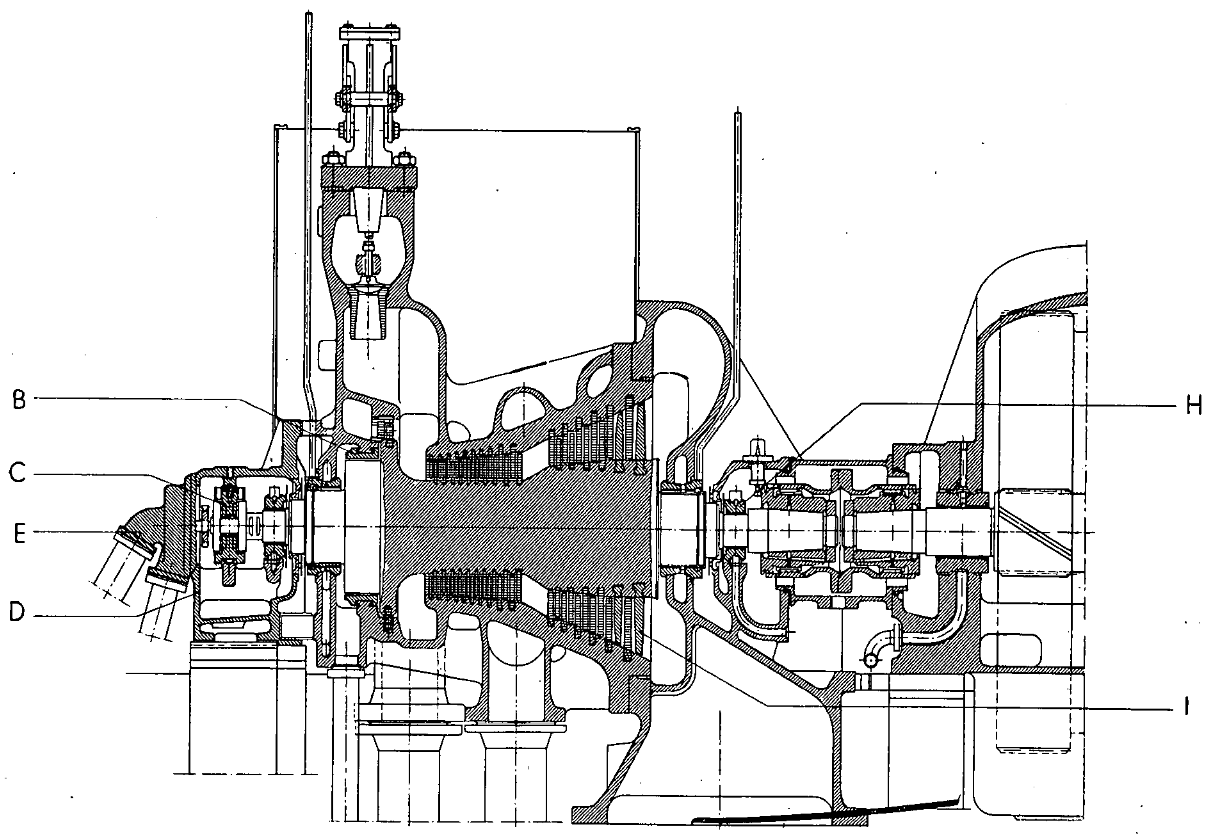
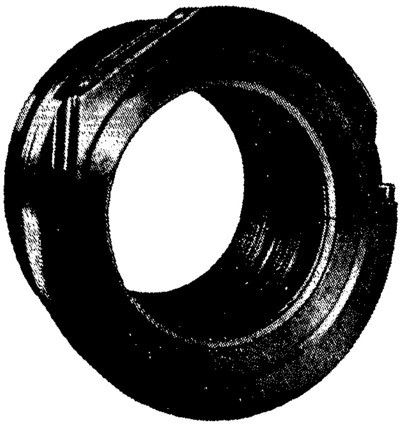
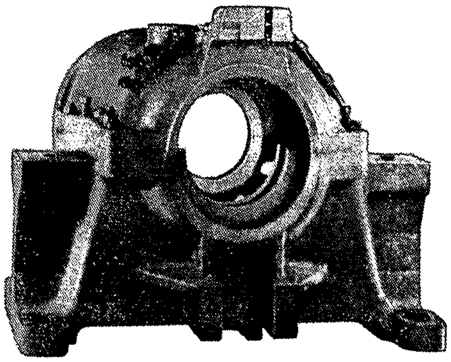


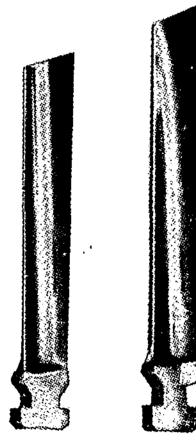
FIGURE 16(b): Turbine longitudinal cross-section



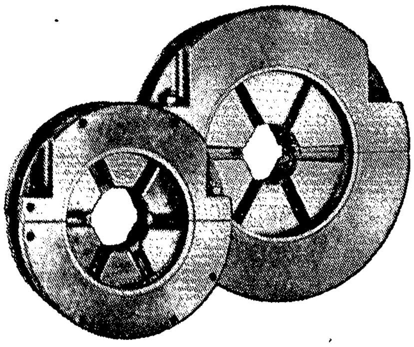
B: Labyrinth glands



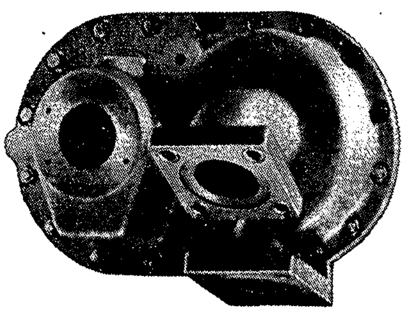
D: Bearing pedestal



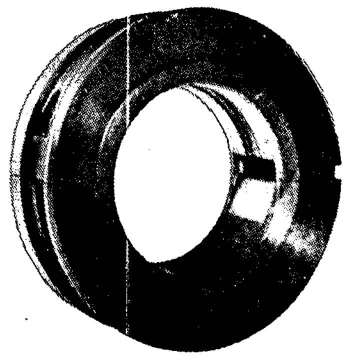
I: End blades



C: Thrust bearing



E: Oil-pump casing



H: Journal bearing

III. Steam Turbine Operation

As in all sections, the wide field of steam turbines has been limited to the most essential features in order to grasp certain basic features and ideas which may lead the way to a more intensive study, if necessary. The same principle has been adhered to in this section.

Turbine Performance at varying loads

The performance of a turbine when operating at loads different from the designed or economic load, depends on the particular method employed for controlling the supply of steam to the turbine, so that the speed of rotation will remain sensibly constant, irrespective of the load. The principal methods of governing are:

1. Throttle governing.
2. Nozzle governing.
3. By-pass governing.
4. Combination of 2 and 3.

1. *Throttle Governing*

The primary aim is to reduce the mass rate of flow. Incidental to reducing the mass rate of flow, the steam experiences an increasing pressure drop across the governing valve and, consequently, a throttling or constant enthalpy process, with an increasing entropy and a corresponding decrease in available energy. This condition is shown in Fig. 17:

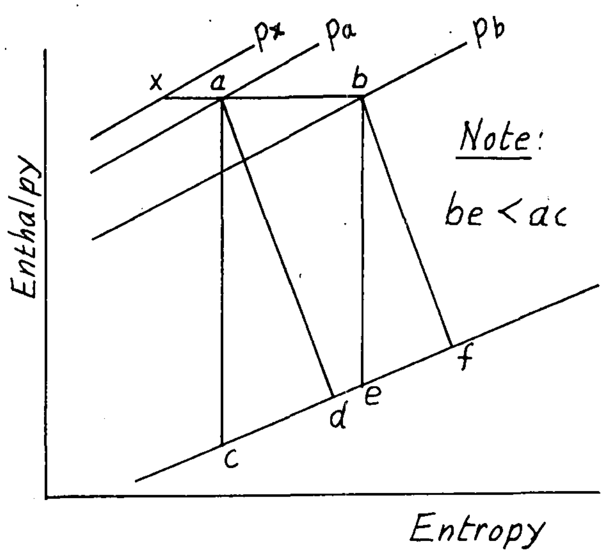


FIGURE 17: Throttling governing

The relationship between load and steam consumption for a turbine governed by throttling is given by the well known "Willans Line", which is a straight line between no load and most economic load, as shown in Fig. 18:

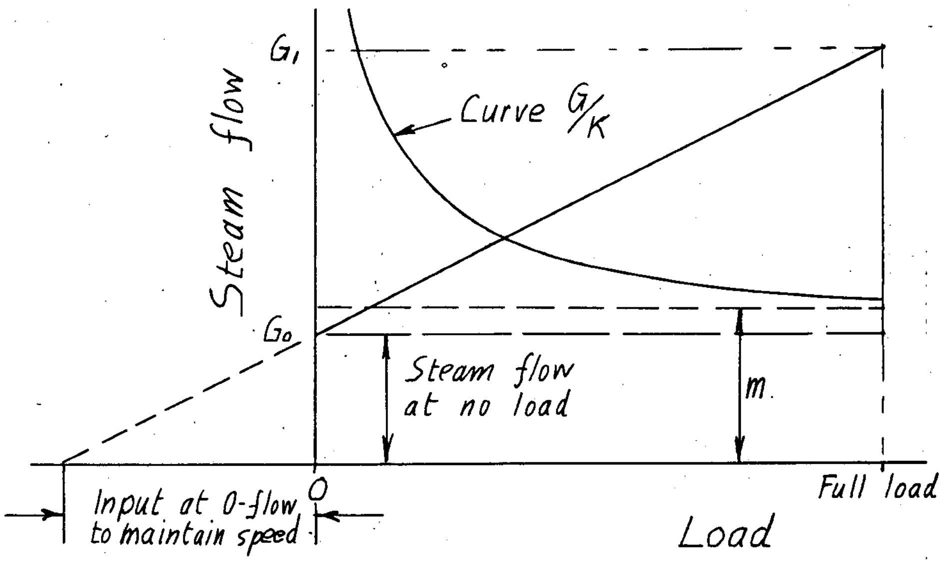


FIGURE 18: Willans line diagram

Equation of a straight line $G = G_o + m.K$. The specific steam consumption is found by dividing by K:

$$\text{i.e.} \quad \frac{G}{K} = \frac{G_o}{K} + m$$

2. *Nozzle Governing*

Ideal governing would be obtained if all nozzles in each and every stage in a turbine could be controlled. The Willians Line for such a turbine would be a straight line, as indicated for the throttle governing, however, with a considerably better specific steam consumption at part loads. In an actual turbine, nozzle governing must be restricted to the first stage nozzles for construction reasons, and even here, groups of nozzles are governed, rather than each nozzle. With nozzle governing, the pressure and temperature entering the first stage nozzles are constant with load. Fig. 19 and Fig. 20 indicate the better specific steam consumption obtainable with nozzle governing.

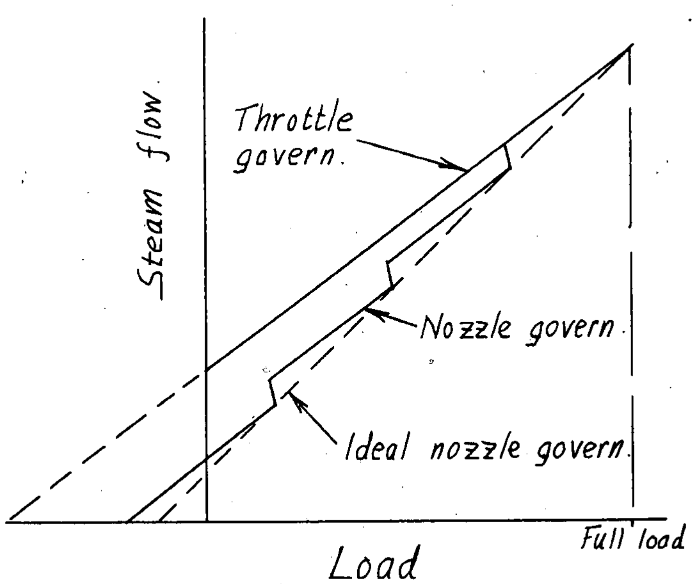


FIGURE 19: Steam flow

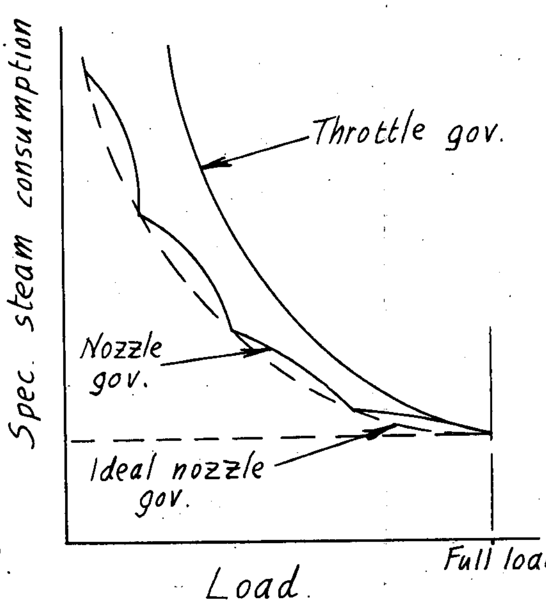


FIGURE 20: steam consumption

Fig. 21 indicates the lines of expansion:

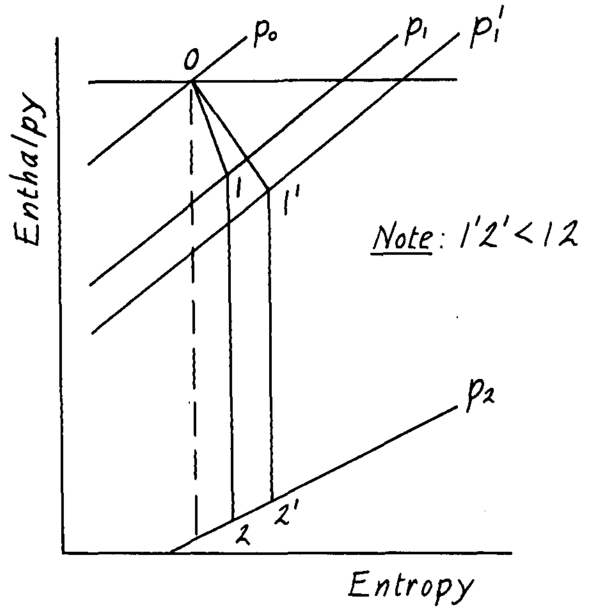


FIGURE 21: Steam expansion for nozzle governing

It is important to note that the absolute pressure of the steam entering the second stage nozzles is in direct proportion to the mass flow through the turbine.

The significant feature of nozzle governing is that considerably less throttling of steam occurs than if a single valve were used. Fig. 22 shows a typical arrangement of nozzle valves:

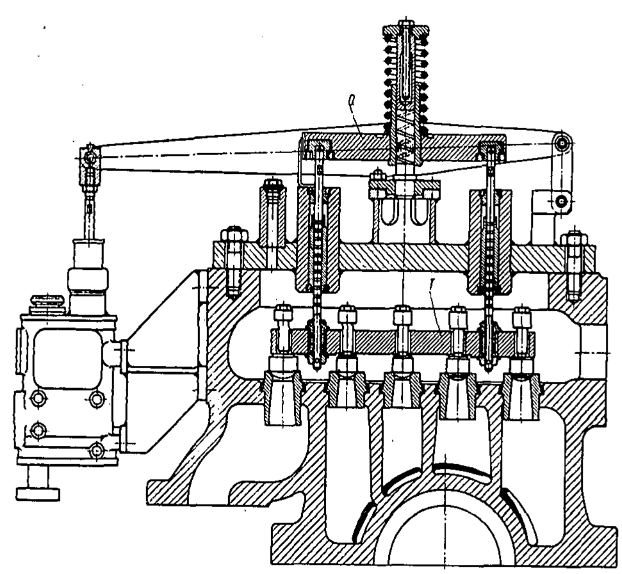


FIGURE 22: Nozzle governing with by-pass

3. By-Pass Governing

This is generally used for the overload valve which passes the steam directly to the steam chest, thereby by-passing the impulse wheel. The normal amount of nozzle valves up to economical load are used.

4. The combination of 2 and 3 has already been outlined under 3 and may also be seen in Fig. 22.

Lubrication

The problem of lubrication resolves itself into three parts which may be enumerated thus:

1. Maintenance of a sufficient supply of oil to the bearings.
2. Cooling of oil to remove the heat generated in the bearings and limit the temperature reached by the oil.
3. Maintenance of the physical properties of the oil at a sufficiently high standard to ensure safe and satisfactory lubrication.

Fig. 23 indicates a typical lubrication system for a turbo-generator.

In all cases, the main oil pump, which is driven directly from an extension to the turbine shaft, performs a dual function; it supplies oil both for lubrication of the bearings and for operating the governor relays. For the latter, the oil is uncooled. The pressure in the relay system is maintained by valve 5, and that in the lubrication system by valve 11. The auxiliary oil pump is arranged to supply the full quantity of oil and to start automatically when the pressure from the main oil pump falls below a predetermined value. A further flushing oil pump is provided for flooding the bearings when starting and stopping. The oil purifier equipment is generally of the centrifugal type and is operated continuously

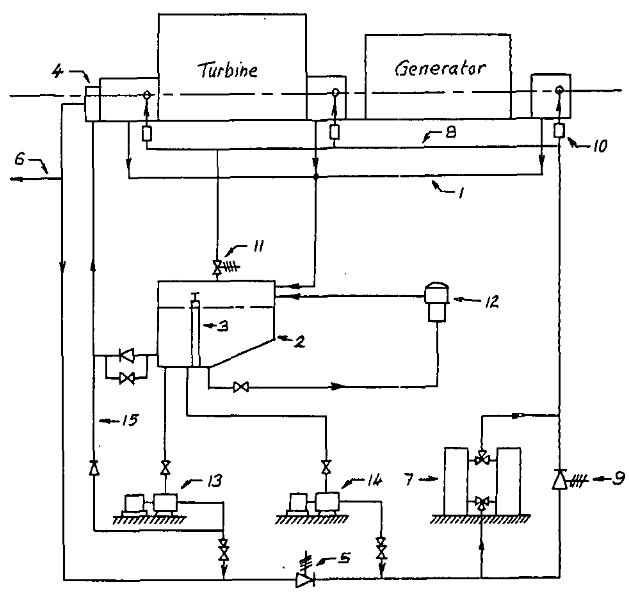


FIGURE 23: Turbine lubrication system

1. Drain from bearings
2. Oil tank
3. Oil strainer
4. Main oil pump
5. Pressure sustaining valve
6. Oil to relay
7. Oil coolers
8. Oil to bearings
9. Spring-loaded by-pass valve
10. Fine strainer
11. L.P. relief valve
12. Oil purifier
13. Auxiliary oil pump
14. Flushing oil pump
15. Priming connexion

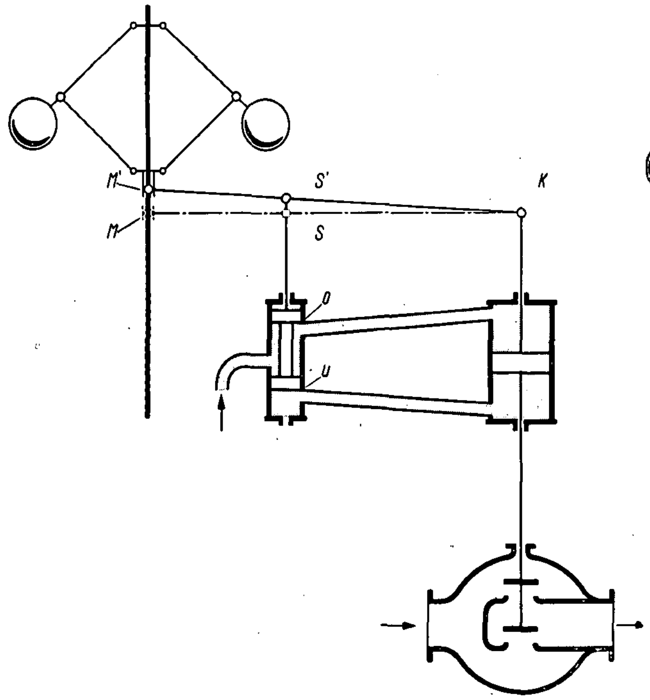


FIGURE 24(a): Linkage un-loading

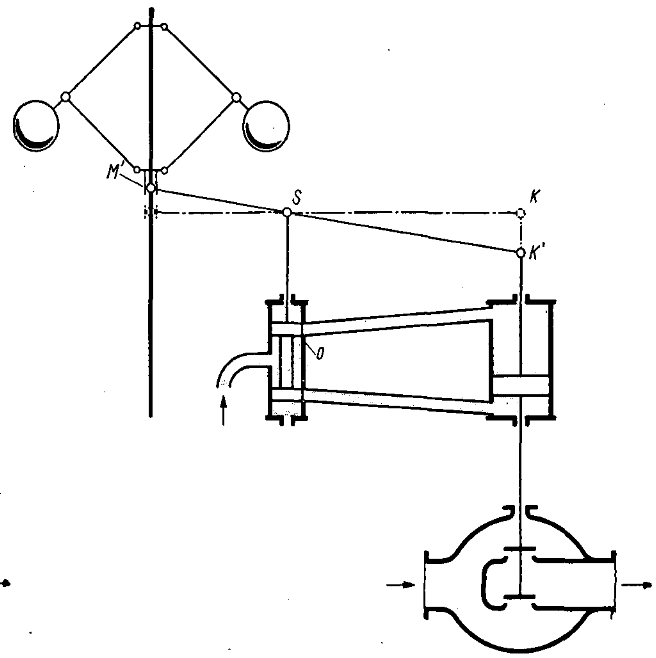


FIGURE 24(b): Linkage un-loaded

while the turbine is running, and should have a capacity not less than one-tenth of the total quantity of oil in the turbine lubrication system.

Governors and Governor Gear

As a flow-pressure type machine operating at high speed, the steam turbine is fitted with a rotor having a very low moment of inertia. This means that its speed responds immediately to any changes in load. The turbine governor whose duty it is to maintain constant speed must, therefore, be sensitive and operate very rapidly and reliably.

Fig. 24(a) shows a ball-type governor acting through means of a pilot valve and servomotor on the steam valve. When the load decreases, the speed of the machine increases; the flyballs lift the sleeve M to M^1. Point K on the connecting rod linking the power piston remains at rest, forming the fulcrum of the motion which forces the pilot valve into position S.[1] The pilot valve now allows high-pressure oil to flow above the power piston, which moves downwards and closes the regulating valve.

Fig. 24(b) shows the position of the governor linkage on completion of unloading. Since M^1 is now the fulcrum, the pilot valve returns to its original position as K moves to K^1, thereby shutting off the high-pressure oil.

The difference in speed between no load and full load referred to the rated speed, is designated as the proportional range or static regulation of the governor. It is determined by the characteristic of the speeder spring.

A turbine with a normal speed of 3,000 r.p.m., which operates as an isolated unit independent of other prime movers, and which is fitted with a governor having 4% static regulation runs, for in-

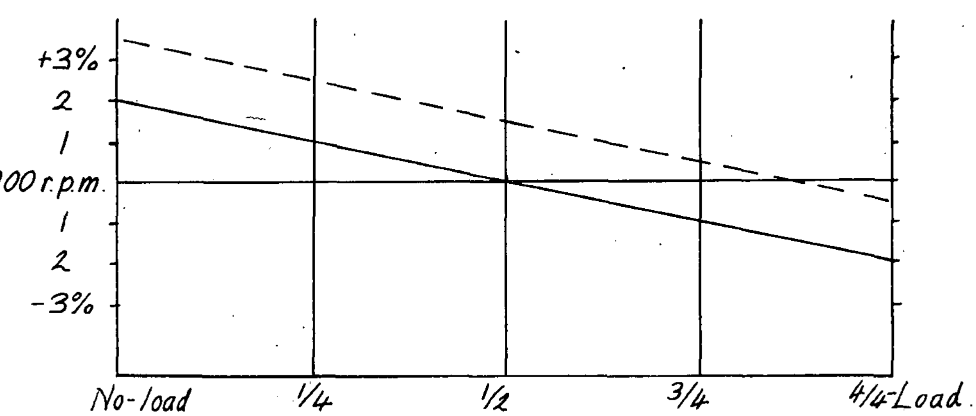


FIGURE 25: Static regulation of speed governor

stance, at 3,060 r.p.m. at no load and 2,940 r.p.m. at full load in accordance with the continuous line in Fig. 25. If the governor sleeve is additionally loaded, e.g. by further compression of the speeder spring, the flyweights must produce a greater force in order to maintain the sleeve in its position. This can take place only with rising speed, as shown by the dotted line in Fig. 25.

Whereas, with an isolated unit, operation of the speed adjusting device changes the speed at constant load, with the alternator coupled to the grid, it is the load that changes, the speed remaining constant. Assuming 4% steady state regulation, on adjustment of the speed changer by an amount which would correspond to a speed increase of 60 r.p.m. (2%) with an isolated unit, the machine which operates at point C, see Fig. 26, with 25% of full load, now takes on additional load and operates at point C[1] with 75% of full load.

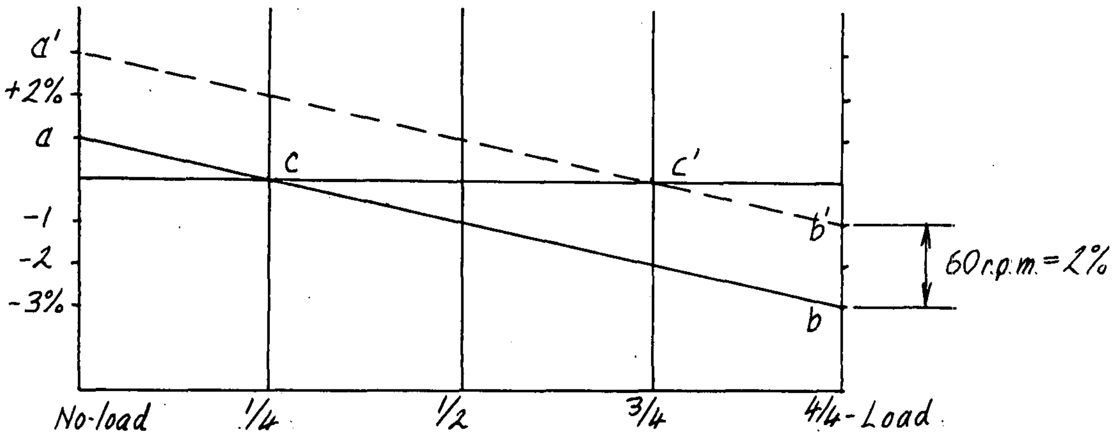


FIGURE 26: Loading of the alternator

Many types of governing systems have been employed. A more modern development is the hydraulic speed governor which eliminates the centrifugal balls and their associated linkages. This type is shown in Fig. 27:

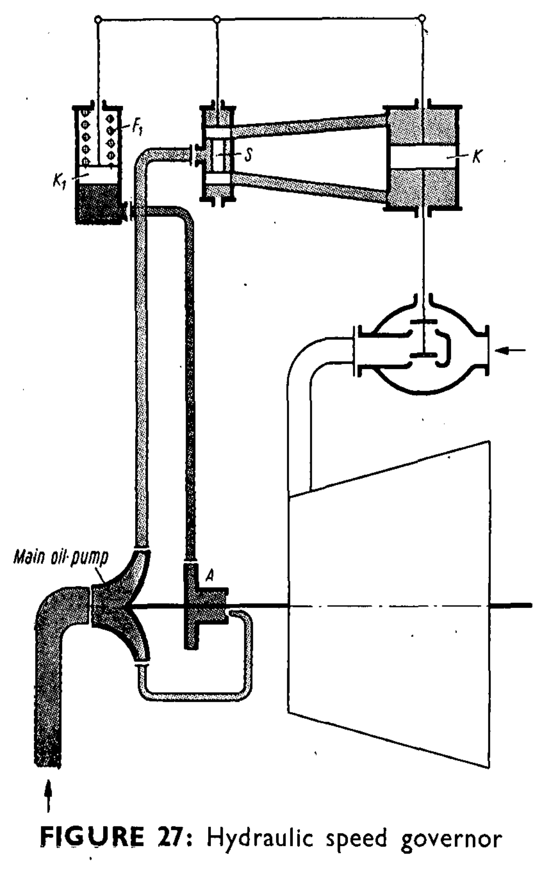


FIGURE 27: Hydraulic speed governor

The hydraulic governor utilises as regulating impulses the pressure difference produced by a small oil gyro upon a change in the turbine speed. The main oil pump itself is not used as impulse transmitter because of the non-uniform high pressure oil requirements of the governing system.

Supervision and Instrumentation

Thermometers for local indication are fitted to all bearings and to the oil coolers. If necessary, a contact thermometer for alarm signals can be fitted behind the oil coolers. All water inlets and outlets at the oil coolers and condensers are also fitted with thermometers.

A thermometer at the steam admission section serves to measure the live steam temperature. The pressure gauges for steam pressures ahead of and within the turbine, as well as vacuum gauges for the condensers, pressure gauges for pressure oil, trip oil, governor oil and bearing oil, are conveniently arranged in a pressure gauge pillar. Such a pillar for nine pressure gauges in shown in Fig. 28:

Measuring devices for monitoring the axial shaft position are fitted to the bearing pedestals. The measuring device at the front bearing pedestal serves to supervise the thrust bearing in order to detect any wear in good time. The measuring device at the rear bearing pedestal indicates the axial differential expansion of shaft and casing. Both measuring devices are provided with vernier scales for adequate measuring accuracy.

A condensate drain controller maintains the condensate at a constant level. This can be read off on a level gauge.

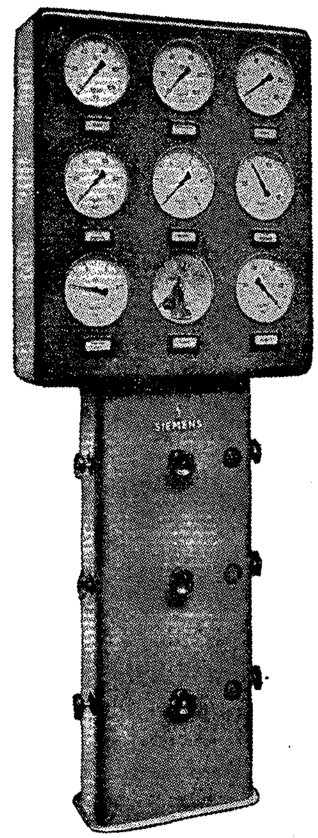


FIGURE 28: Pressure gauge console

Turbine Troubles

The causes of failure naturally divide themselves into two groups, namely, those inherent in the design or in the material used in the construction of the turbine, and those which are related to the operating conditions. Some of the former are inter-related to some of the latter; for instance, blade erosion might be due to unsuitable material or to bad steam conditions.

The causes of failure inherent in design and materials of construction may be classified as follows:

(*a*) Shaft vibration.
(*b*) Disc vibration.
(*c*) Blade vibration.
(*d*) Faults in machining.
(*e*) Incorrect design of casing, faulty arrangement of steam pipes, causing distortion of the casing.
(*f*) Materials of construction.

To discuss each of the above points would be far beyond the scope of this paper.

IV. Summary

The installation of the industrial turbine in a factory where both process steam and electrical power are required in the right proportions, can only be of economical advantage.

References

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Kearton, W. J. (1964). Steam Turbine Operation.