SOLTEQ®   EQUIPMENT FOR ENGINEERING EDUCATION

EXPERIMENTAL MANUAL

MULTI TURBINE TEST SET

MODEL: FM46

SOLUTION ENGINEERING SDN. BHD.
NO.3, JALAN TPK 2/4, TAMAN PERINDUSTRIAN KINRARA,
47100 PUCHONG, SELANGOR DARUL EHSAN, MALAYSIA.
TEL: 603-80758000 FAX: 603-80755784
E-MAIL: solution@solution.com.my
WEBSITE: www.solution.com.my

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1.0 INTRODUCTION

The SOLTEQ® Multi Turbine Test Set (Model: FM 46) is designed to be a self-contained unit to demonstrate the operation and performance characteristics of three types of turbines, namely Francis (radial flow) turbine, Kaplan (axial) turbine, and Pelton (impulse) turbine. The turbines will be driven by a 3kW pump and come complete with dynamometer. The Multi Turbine Test Set is capable of performing experiments on mechanical power out, hydraulic power in, hydraulic efficiency, and specific speed.

2.0 GENERAL DESCRIPTIONS

2.1 Unit Construction

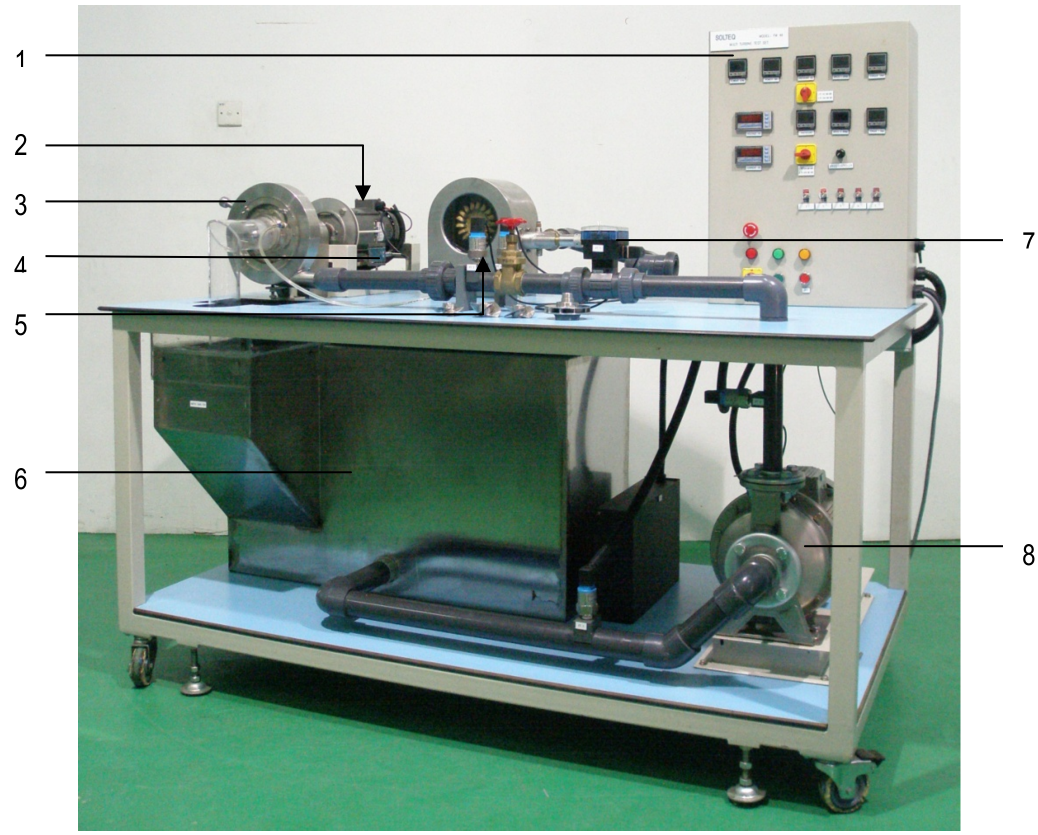


Figure 1: Unit Construction for Multi Turbine Test Set (Model: FM 46)

1. Control Panel	5. Pressure Transmitter
2. DC Generator	6. Water Reservoir Tank
3. Turbine Set	7. Flow-meter
4. Load Cell	8. Centrifugal Pump

2.2 Unit Assembly

The SOLTEQ® Multi Turbine Test Set (Model: FM 46) consists of a centrifugal pump with controlled motor speed, three interchangeable sets of Pelton, Francis, and Kaplan turbine complete with individual casing respectively. The test set is a self contained unit comprising two main assemblies; an instrument/control console and a welded steel base frame, which carries the pump, turbine set, and water reservoir tank. In terms of the functionality, an inverter with speed dial controls the centrifugal pump motor speed in between 0 to approximately 3000 rpm. Power transducer is provided to measure the power consumption of the pump motor. Speed sensors are installed on both pump and DC generator shaft to measure the rotational speed of the motors. Load cells are mounted beneath each motor. The torque arm is directly depressing against the load cell arm, thereby compressing against the motion. With the aid of the torque sensor, we can register and detect the torque or force acting on the motors. A volt meter and an ampere meter are provided to measure volt and current generated by the DC generator. Pressure transmitters are installed on each inlet and outlet of both pump and turbine set to measure the pressure on the respective locations. Five sets of Resistor banks are installed to provide necessary load to the DC generator during experiment.

2.3 Process Flow

Water is pumped by the centrifugal pump from the reservoir tank into the turbine casing and spins the turbine as well as DC generator motor. This will generate power and displayed as voltage and current on the control panel. The water will be re-circulated. The pump motor speed could be varied to perform studies on the centrifugal.

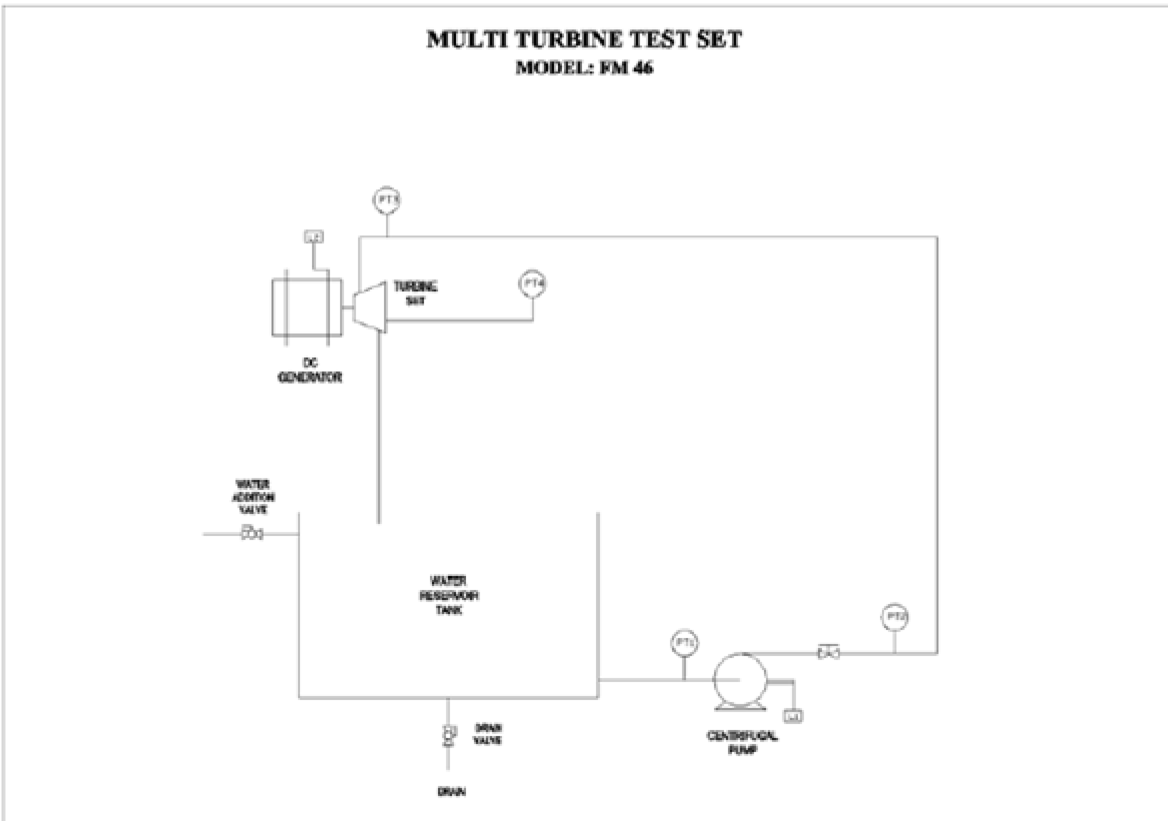


Figure 2: Schematic Diagram for Multi Turbine Test Set (Model: FM 46)

2.4 Description and Specification

2.4.1 Pelton Turbine

1. Description

A Pelton turbine mounts on the 3 kW pump and turbine test set base unit. The case in which the Pelton turbine is mounted is transparent. Water leaves a nozzle and heats the buckets. An adjustable spear valve controls the discharge by varying the diameter of the jet from the nozzle. The centrifugal pump on the base unit supplies water to the nozzle. Water from the turbine discharges back into the base unit tank and recirculates. The turbine shaft connects to the base unit dynamometer. This measures torque and speed.

2. Specification

Dimensions: nett 750 x 600 x 650, packed 0.171 m^3
Weight: nett 14 kg, packed 20 kg
Number of buckets: 16
Nozzle area: variable – 162 mm^2 when fully open
Radius of runner to jet impact position: 56.6 mm
Maximum speed: approximately 1800 rpm
Speed at maximum efficiency (approximately 80%): approx 1200 rpm

2.4.2 Francis Turbine

1. Description

A Francis turbine mounts on the 3 kW pump and turbine test set base unit. The working section of the turbine outlet is transparent. The turbine guide vanes are individually adjustable. Water from the turbine outlet discharges back into the base unit tank and recirculates. A yaw probe indicates the magnitude and direction of the flow from the turbine outlet. The turbine shaft connects to the base unit dynamometer. This measures torque and speed.

2. Specification

Dimensions: nett 700 x 350 x 500 mm, packed 900 x 600 x5000.171 mm
Weight: nett 6 kg, packed 10 kg
Impeller blades: 10
Impeller diameter: 100 mm
Impeller inlet height: 14.5 mm
Impeller outlet diameter: 45 mm
Guide varies: six – adjustable, with positioning guidelines

2.4.3 Kaplan Turbine

1. Description

A Kaplan turbine mounts on the 3 kW pump and turbine test set base unit. The working section of the turbine outlet is transparent. The turbine guide vanes are individually adjustable. The unit includes three interchangeable rotors, each with different blade angles. The centrifugal pump on the base unit powers the turbine. Water from the turbine outlet discharges back into the base unit tank and recirculates. A yaw probe indicates the magnitude and direction of the flow from the turbine outlet. The turbine shaft connects to the base unit dynamometer. This measures torque and speed.

2. Specification

Dimensions: nett 750 x 600 x 650 mm, packed 0.171 m^3
Weight: nett 14 kg, packed 20 kg
Inlet height: 14.5 mm
Outlet diameter: 88.0 mm
Interchangeable rotors: three with blade angles of 10º, 20º and 30º, diameters 87 mm
Guide varies: six equi-spaced, diameter 153.4 mm

2.5 Overall Dimension

Height : 1.85 m

Width : 1.85 m

Depth : 1.25 m

2.6 General Requirements

Electrical : 415VAC/50Hz (3 phase)

Laboratory tap water with hose for water top up

Drainage point

3.0 SUMMARY OF THEORY

3.1 Rotodynamic Machine

A Rotodynamic Machine consists of a rotor having a number of vanes or blades, and there is a transfer of energy between the fluid and the rotor. Whether the fluid does work on the rotor (as in a turbine) or the rotor does work on the fluid (as in a pump), the machine can be classified in the first instance according to the main direction of the fluid's path in the rotor. In a *radial-flow* machine the path is wholly or mainly in the plain of rotation; the fluid enters the rotor at one radius and leaves it at a different radius. Examples of this type of machine are the Francis turbine and the centrifugal pump. If, however, the main flow direction is parallel to the axis of rotation, so that any fluid particle passes through the rotor at a practically constant radius, then the machine is said to be an *axial-flow* machine. The Kaplan turbine and the 'propeller' or axial-flow pump are examples of this type. If the flow is partly radial and partly axial the term *mixed-flow* machine is used.

3.1.1 Types of Turbine

By alternative classification turbines may be placed in one of two general categories: (a) *impulse* and (b) *reaction*. In both types the fluid passes through a runner having blades. The momentum of the fluid in the tangential direction is changed and so a tangential force on the runner is produced. The runner therefore rotates and performs useful work, while the fluid leaves it with reduced energy. The important feature of the impulse machine is that there is no change of static pressure across the runner. In the reaction machine, on the other hand, the static pressure decreases as the fluid passes through the runner. For any turbine the energy held by the fluid is initially in the form of pressure. The impulse turbine has one or more fixed nozzles, in each of which this pressure is converted to the kinetic energy of an unconfined jet. The jet of fluid then impinge on the moving blades of the runner where they lose practically all their kinetic energy and, ideally, the velocity of the fluid at discharge is only just sufficient to enable it to move clear of the runner. In a reaction machine the change from pressure to kinetic energy takes place gradually as the fluid moves through the runner, and for its gradual change of pressure to be possible the runner must be completely enclosed and the passages in it entirely full of the working fluid.

3.1.2 The Pelton Wheel

This is only hydraulic turbine of the impulse type now in common use and is named after Lester A. Pelton (1829-1908), the American engineer. It is an efficient machine, particularly well suited to high head. The rotor consists of a circular disc with several (seldom less than 15) spoon-shaped ‘buckets’ spaced round its periphery. One or more nozzles are mounted so that each directs a jet along a of each bucket is a ‘’splitter’ ridge, which divides the oncoming jet into two equal portions and, after flowing round the smooth inner surface of the bucket, the fluids leaves it with a relative velocity almost opposite in direction to the original jet. The notch in the outer rim of each bucket (see Figure 4) prevents the jet to the preceding bucket being intercepted too soon; it also avoids the deflection of the fluid towards the centre of the wheel as the bucket first meets the jet. The maximum change of momentum of the fluid – and hence the maximum force driving the wheel round - would be obtained if the bucket could deflect the fluid through 180º. In practice, however, the deflection is limited to about 165º if the fluid leaving one bucket is not to strike the back of the following one.

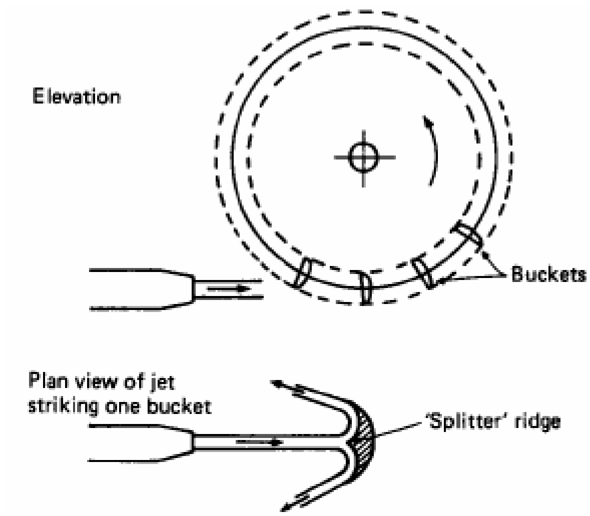


Figure 3: Single-jet Pelton Wheel

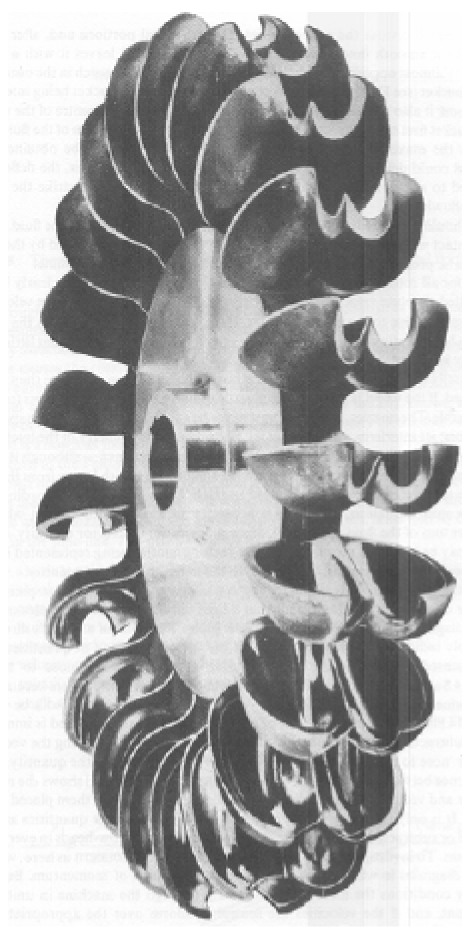


Figure 4: Runner of Pelton Wheel

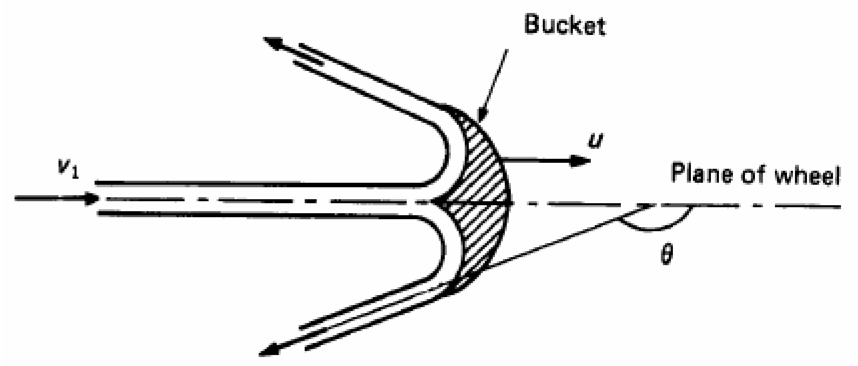


Figure 5: A section through a bucket of Pelton Wheel Runner

Figure 5 shows a section through a bucket that is being acted on by a jet. The plane of section is parallel to the axis of the wheel and contains the axis of the jet. The plane of section is parallel to the axis of the wheel and contains the axis of the jet. The absolute velocity of the jet is determined by the head available at the nozzle, that is, the gross head *Hgr* minus the head loss *hf* due to friction in the pipe –line. The other symbols to be used are these:

v_1 = absolute velocity of jet just before striking bucket
v_2 = absolute velocity of fluid leaving bucket
ω = angular velocity of wheel
r = radius from axis of wheel to axis to jet striking bucket
u = absolute velocity of bucket at this radius = ωr
R_1 = velocity of oncoming jet *relative* to bucket
R_2 = velocity of fluid leaving bucket *relative* to bucket
θ = angle through which fluid is deflected by bucket
C_V = coefficient of velocity for the nozzle – usually between 0.97 and 0.99
Q = volume rate of flow from nozzle (quantity)
ρ = density of fluid

The jet velocity *v1* is given by $C_V\sqrt{2gh}$ where *H* represents the net head $H_V - h_r$. The velocity head of the fluid in the pipe-line is normally negligible compared with *H*.

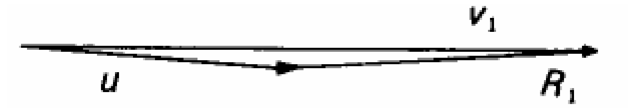
During the time when any one bucket is acting on by the jet the wheel turns through a few degrees and so the direction of motion of the bucket changes slightly. The effect of this change, however, is small, and it is sufficient here to regard the direction of the bucket velocity u as the same as of v_1. Since the radius of the jet is small compared with that of the wheel, all the fluid may be assumed to strike the bucket at radius r. It is also assumed that all the fluid leaves the bucket at radius r and that the velocity of the fluid is steady and uniform over section 1 and 2 where the values v_1 and v_2 are considered.

The relative velocity u_1 at the moment when the fluid meets the bucket is given by $R_1 = v_1 - u$. (Since v_1 and u are collinear, the diagram of velocity vectors is simply a straight line as in Figure 5. The relative velocity R_2 with which the fluid leaves the bucket is somewhat less than the initial relative velocity R_1. There are two reasons for this. First, although the inner surfaces of the buckets are polished so as to minimize frictional losses as the fluid flows over them, such losses cannot be entirely eliminated. Second, some additional loss is inevitable as the fluid strikes the splitter ridge, because the ridge cannot have zero thickness. These losses of mechanical energy reduce the relative velocity between fluid and bucket which is counted as $R_2 = kR_1$ where k is a fraction slightly less than unity.

As the bucket is symmetrical it is sufficient to consider only that part of the flow which traverses one side of it. The diagram of velocity vectors at outlet is therefore the triangle in Figure 6. To obtain the absolute velocity of the fluid at discharge from the bucket, we must add (vectorially) the relative velocity u_2 to the bucket velocity u. The velocity of whirl at outlet, v_{w2}, is the component of v_2 in the direction of the bucket movement. The direction of u being taken as positive, $V_{w2} = u - R_2 \cos(\pi - \theta)$. At inlet the velocity of whirl is v_1 so change in the whirl component is

$$\Delta v_w = v_1 - [u - R_2 \cos(\pi - \theta)] = R_1 + R_2 \cos(\pi - \theta) = R_1(1 - \cos\theta) \qquad (1)$$

(a) Inlet to bucket



(The vectors are actually collinear, but are shown slightly separated here for clarity.)

(b) Outlet from bucket

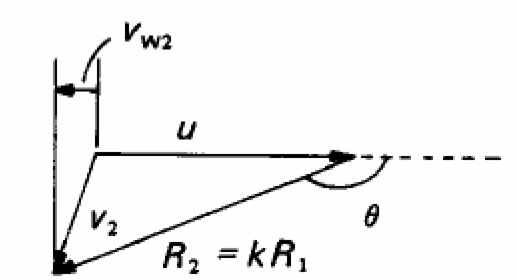


Figure 6: Velocity vectors triangle

The mass flow rate in the jet = $Q\rho$ an so the rate of change of momentum in the whirl direction is $Q\rho\Delta v_w$. This corresponds to the force driving the wheel round. The torque on the wheel is therefore $Q\rho\Delta vwr$ and the power output is $Q\rho\Delta v_w r\omega$ = $Q\rho\Delta v_w u$.

The energy arriving at the wheel is in the form of kinetic energy of the jet, and is given by $Q\rho v_1^2/2$ per unit time. Therefore, the wheel efficiency,

$$\eta_w = \frac{Q\rho\Delta v_w u}{\frac{1}{2}Q\rho v_1^2} = \frac{2u\Delta v_w}{v_1^2}$$

Substituting for Δv_w from Eq. (1) and puttuing $R_1 = V_1 - u$ gives

$$\eta_w = \frac{2u(v_1 - u)(1 - k\cos\theta)}{v_1^2}$$

Which, if k is assumed constant, is maximum at $u/V_1 = 1/2$

The wheel efficiency represents the effectiveness of the wheel in converting the kinetic energy of the jet into mechanical energy of rotation. Not all this energy of rotation is available at the output shaft of the machine, because some is consumed in overcoming friction in the bearing and some in overcoming the 'windage', that is the friction between the wheel and atmosphere in which it rotates. In addition to these losses there is a loss in the nozzle (which is why *CV* is less than unity). The overall efficiency is therefore less than the wheel efficiency. Even so, the overall efficiency is 85 – 90 % may usually be achieved in large machines. Moreover, as the losses due to bearing friction and windage increase rapidly with speed, the peak of overall efficiency occurs when the ratio *u/v1* (often termed the *speed ratio*) is slightly less than the theoretical value of 0.5.

Equation (2) indicates that a graph of efficiency against bucket velocity is parabolic in form, as illustrated in Figure 7.

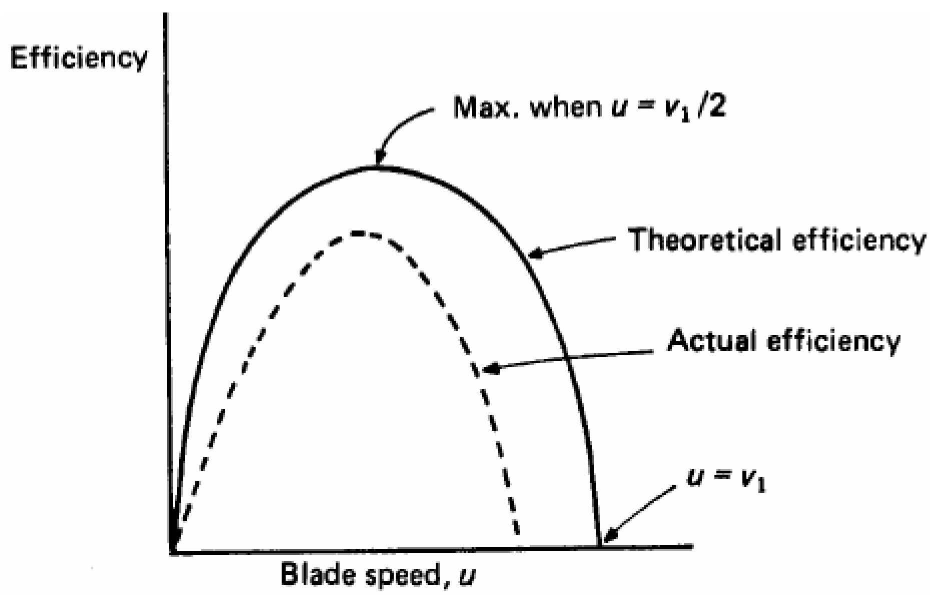


Figure 7: Pelton Wheel Efficiency versus Bucket Velocity

A Pelton wheel is almost invariably used to drive an electrical generator mounted on the same shaft. It is designed to operate at the conditions of maximum efficiency, and the governing of the machine must be such as to allow the efficiency to be maintained even when the power demand at the shaft varies. No variation of the angular velocity, and hence of bucket velocity u, can normally be permitted (for this would alter the frequency of the electrical output). The control must therefore be in the volume of flow Q, and yet there must be no change in the jet velocity because this would alter the speed ratio u/v_1 from its optimum value. Since $Q = Av_1$ it follows that the control must be effected by a variation of the cross sectional area A of the jet. This is usually achieved by a spear valve in the nozzle (Figure 8). Movement of the spear along the nozzle increases or decreases the annular area between the spear and the housing. The spear is so shaped, however, that the fluid coalesces into a circular jet and the effect of the spear movement is to vary the diameter of the jet. Sudden reduction of the rate of flow could result in serious water-hammer problems, and so deflectors (Figure 7) are often used in association with the spear valve. These plates temporarily deflect the jet so not all the fluid reaches the buckets; the spear valve than can be moved slowly to its new position and the rate of flow in the pipe-line reduced gradually. Diffusing plates in the surface of the spears are sometimes used for the same purpose.

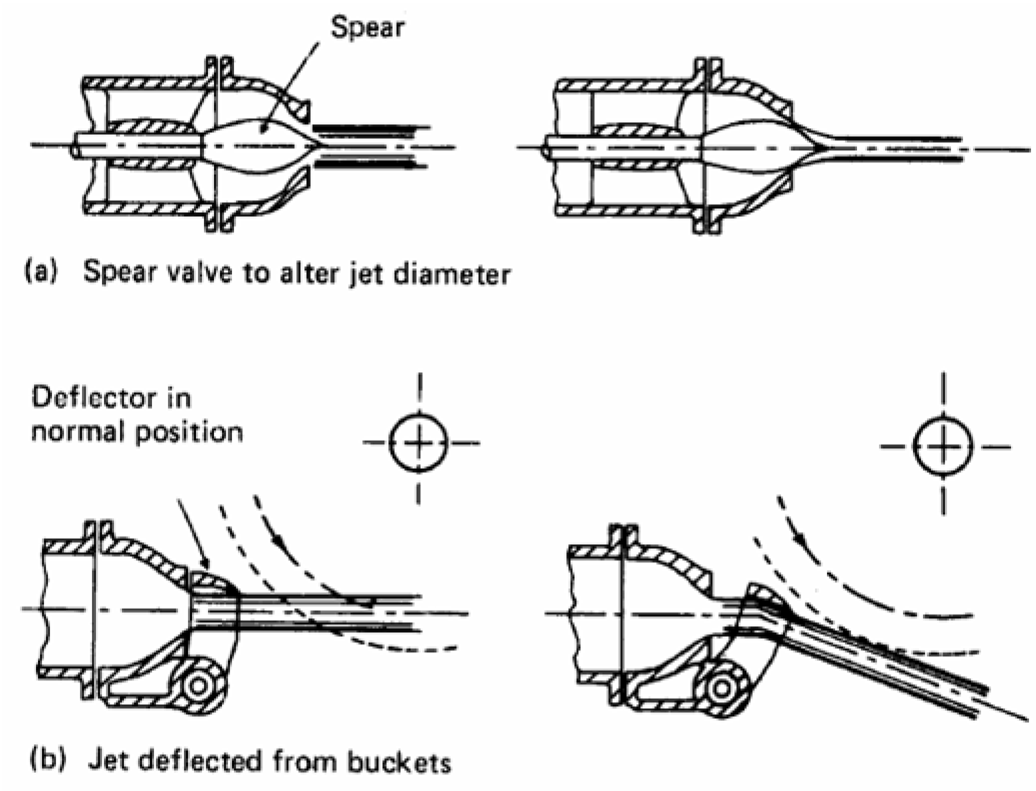


Figure 8: Spear valve in the nozzle

In the design of the Pelton wheel, two parameters are of particular importance: the ratio of the bucket width to the jet diameter, and the ratio of the wheel diameter to the jet diameter. If the bucket width is too small in relation to the jet diameter, the fluid is not smoothly deflected by the buckets, and, consequently, much energy is dissipated in turbulence and the efficiency drops considerably. On the other hand, if the buckets are unduly large, friction on the surfaces is unnecessarily high. The optimum value of the ratio of bucket width to jet diameter has been found to be between 4 and 5. The ratio of wheel diameter to jet diameter has in practice a minimum value of about 10; smaller values involve either too close a spacing of the buckets or too few buckets for the whole jet to be used. There is no upper limit to the ratio, but the larger its value the more bulky is the entire installation; the smaller ratios are usually desirable.

Since $V_1 = C_v\sqrt{2gh}$ the jet velocity is determined by the head available at the nozzle, and the bucket velocity *u* for maximum efficiency is given as approximately 0.46 *v1*. Then, since $u = \omega_r$, the radius of the pitch circle of the buckets may be calculated for a given shaft speed. The required power output *P* determines the volume rate of flow *Q* since $P = Q\rho gH\eta_0$ where η_0 represents the overall efficiency of the turbine. Then, with v_1 already determined, the total cross-sectional area of the jets is given by Q/v_1. It is worth reemphasizing here that *H* represents the head available *at the nozzles*, i.e. the gross head of the reservoir less the head loss to friction in supply pipe. The overall efficiency quoted for the machine always refers to the ability of the machine itself to convert fluid energy to useful mechanical energy: the efficiency thus accounts for losses in the machine but not for losses that occur before the fluid reaches it.

3.1.3 Reaction of Turbines

The principal distinguishing features of a reaction turbine are that only part of the overall head is converted to velocity head before the runner is reached, and that the working fluid, instead of engaging only one or two blades at a time (as in an impulse machine), completely fills all the passages in the runner. Thus the pressure of the fluid changes gradually as it passes through the runner. Figure 9 Francis turbine, a radial-flow machine of the kind developed by the America engineer James B. Francis (1815-92).

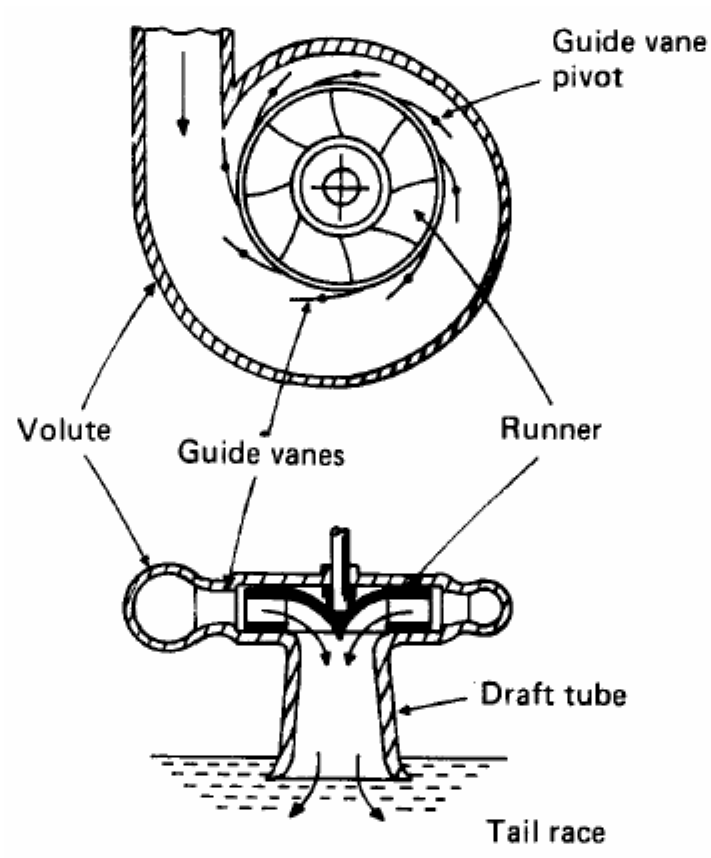


Figure 9: Radial-flow Francis Turbine

Although some smaller machines of this type have horizontal shafts, the majority have vertical shafts as shown in the figure. The fluid enters a spiral casing (called a volute or scroll case) which completely surrounds the runner. The cross-sectional area of the volute decreases along the fluid path in such a way as to keep the fluid velocity constant in magnitude.

From the volute the fluid passes between stationary guide vanes mounted all round the periphery of the runner. The function of these guide vanes is to direct the fluid on the runner at the angle appropriate to the design. Each vane is pivoted and, by a suitable mechanism, all may be turned in synchronism so as to alter the flow rate through machine, and hence the power output, as required by the governing gear. These vanes are also known as *wicket gates*. In its passage through the runner the fluid is deflected by the runner blades so its angular momentum is changed.

From the centre of the runner the fluid is turned into the axial direction and flows to waste via a *draft tube*. The lower end of the draft tube must, under all conditions of operation, be submerged below the level of the water in the *tail race* that is the channel carrying the used water away. Only in the way can it be ensured that a hydraulic turbine is full of water. A carefully designed draft tube is also of value in gradually reducing the velocity of the discharge water so that the kinetic energy lost at the outlet is minimized. The properties of the machine are such that, at the 'design point', the absolute velocity of the fluid leaving the runner has little, if any, whirl component. The Francis turbine is particularly suitable for medium heads (that is, from about 15 to 300 m) and overall efficiencies exceeding 90%, have been achieved for large machines. An inward-flow turbine such as this has a valuable feature of being to some extent self-governing. This is because a 'centrifugal head' like that in a forced vortex is developed in the fluid rotating with the runner. The centrifugal head balances part of the supply head. If for any reason the rotational speed of the runner falls, with the result that a higher rate of flow through the machine is possible and the speed of the runner rise again. The converse action results from an increase of speed. The runner of a reaction turbine I always full of the working fluid, whereas in an impulse machine only a few of the runner blades are in use at any one moment. The reaction turbine is therefore able to deal with a larger quantity of fluid for a given size of runner. For a runner of a given diameter the greatest rate of flow possible is achieved when the flow is parallel to the axis. Such a machine is known as an *axial-flow* reaction turbine or *propeller* turbine. From Fig.8 it will be seen that the arrangement of guide vanes for a propeller turbine is (usually) similar to that for a Francis machine. The function of the vanes is also similar: to give the fluid the initial motion in the direction of whirl. Between the guide vanes and the runner, the fluid in a propeller turbine turns through a right angle into the axial direction and then and closely resembles a ship's propeller. Apart from frictional effects, the flow approaching the runner blades is that of a free vortex (whirl velocity inversely proportional to radius) whereas the velocity of the blades themselves is directly proportional to radius. To cater for the varying relation between the fluid velocity and the blade velocity as the radius increases, the blades are twisted as shown in Figure 11, the angle with the axis being greater at the tip than at the hub. The blade angles may be fixed if the available head and the load are both fairly constant, but where these quantities may vary a runner is used on which the blades may be turned about their own axes while the machine is running. When both guide-vane angle and runner-blade angle may thus be varied, a high efficiency can be maintained over a wide range of operating conditions. Such a turbine is known as a *Kaplan* turbine after its inventor, the Austrian engineer Viktor Kaplan (1876-1934) (see Figure 11).

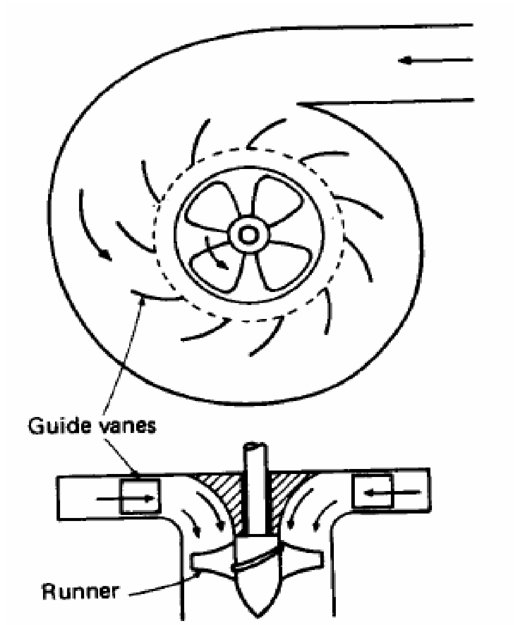


Figure 10: Axial-flow (Propeller) Turbine

In addition to the Francis (radial-flow) turbine and the axial-flow type are so called mixed-flow machines in which the fluid enters the runner in a radial direction and leaves it with a substantial axial component of velocity. In such machines, radial and axial flow is combined in various degrees and there is a continuous transition of runner design from radial-flow only to axial-flow only.

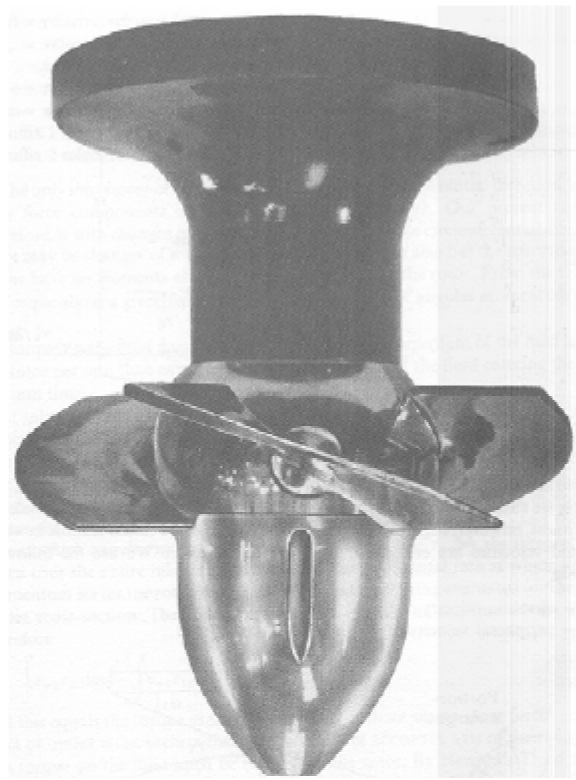


Figure 11: Runner of Kaplan Turbine

Net Head across a Reaction Turbine. The effective head across any turbine is the difference between the head at inlet to the machine and the head at outlet from it. As a reaction turbine must operate 'drowned', that is, completely full of the working fluid, a draft tube is fitted, and so important is the function of the draft tube that it is usually considered as part of the turbine. The kinetic energy of the fluid finally discharged into the tail race is wasted: the draft tube is therefore made divergent so as to reduce the velocity at outlet to a minimum. The angle between the walls of the tube and the axis is limited, however, to about 8° so that the flow does not separate from the walls and thereby defeat the purpose of the increase in cross-sectional area. Flow conditions within the draft tube may be studied by applying the energy equation between any point in the tube and a point in the tail race. The pressure at outlet from the runner is usually less than the atmospheric pressure of the final discharge; the turbine should therefore not be set so high above the tail water that the pressure at outlet from the runner falls to such a low value that cavitation occurs. The net head across the machine corresponds, then, to the total difference in level between the supply reservoir and the tail water, minus the losses external to the machine (that is, those due to pipe friction and the kinetic head at outlet from the draft tube). Figure 12 indicates that the net head across the turbine is

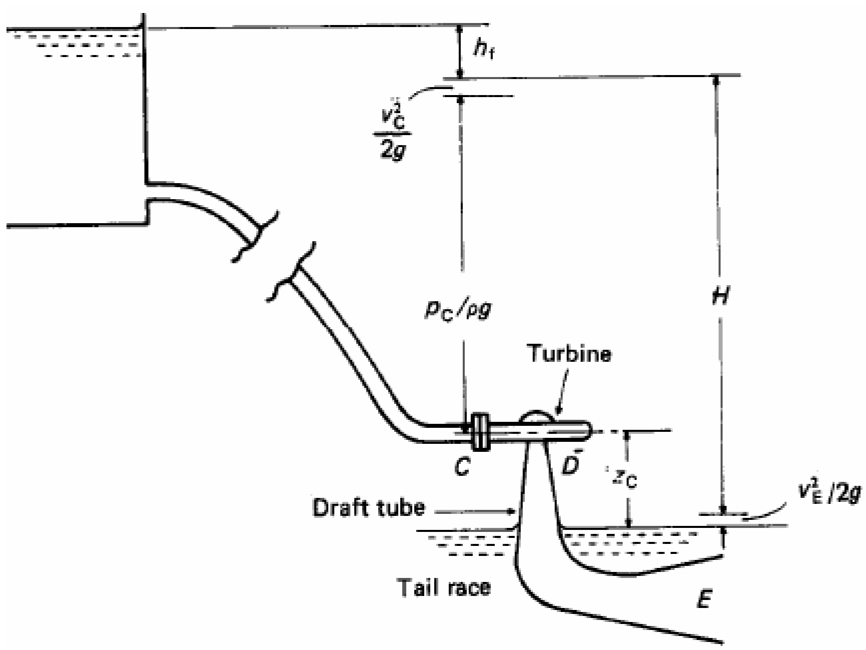


Figure 12: The net head across turbine

H = total head at inlet to machine (C) – total head at discharge to tail race

$$= \frac{p_C}{\rho g} + \frac{v_C^2}{2g} + z_C - \frac{v_E^2}{2g}$$

It will be noticed that, for a given difference between the levels of the supply reservoir and the tail water, the net head across a reaction turbine is greater than that for an impulse machine. The discharge from the runner of an impulse machine is necessarily at atmospheric pressure, and so the portion z_C of the total difference of levels is not then available for use by the turbine. For the high heads normally used by impulse turbines, however, this loss is not of great importance.

3.1.4 Basic Equations for Rotodynamic Machinery

No fundamentally new relations are required to study the flow through rotodynamic machines. The equation of continuity, the momentum equation and the general energy equation are used, but certain special forms of these relations may be developed. In particular an expression is required for the transfer of energy between the fluid and the rotor, and consideration is here restricted to steady conditions. The relation we shall obtain applies in principle to any rotor whatever. For the sake of explicitness, however, Figure 13 represents the runner of a Francis turbine in which the fluid velocities are entirely in the plane of rotation. We use the following symbols:

v = absolute velocity of fluid
u = peripheral velocity of blade at point considered
R = relative velocity between fluid and blade
v_w = velocity of whirl, i.e. component of absolute velocity of fluid in
direction tangential to runner circumference
r = radius from axis of runner
ω = angular velocity of runner
Suffix 1 refers to conditions at inlet to runner
Suffix 2 refers to conditions at outlet from runner

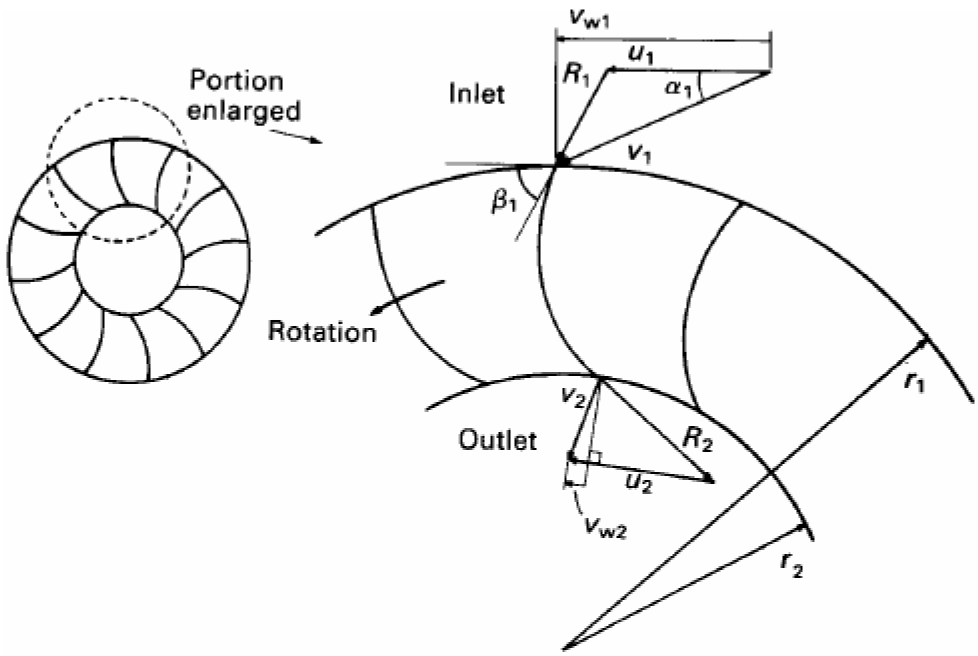


Figure 13: Portion of a Francis Turbine Runner

The only movement of the runner blades is in the circumferential direction, and so only force components in this direction perform work. Our present concern, therefore, is with changes of momentum of the fluid in the circumferential direction: there may be changes of momentum in other directions also but the corresponding forces have no moments about the axis of rotation of the rotor. From the relation

Torque about a given fixed axis = Rate of increase of angular momentum about that axis the torque on the fluid must be equal to the angular momentum of the fluid leaving the rotor per unit time *minus* the angular momentum of the fluid entering the rotor per unit time.

At inlet, any small particle of fluid, of mass δm, has momentum $\delta m v_{w1}$ in the direction tangential to the rotor. Its angular momentum (that is, moment of momentum) is therefore $\delta m v_{w1} r_1$. Suppose that, of the total (constant) mass flow rate $\dot{m}$, a part δm passes through a small element of the inlet cross-section across which values of v_{w1} and r_1 are uniform. Then the rate at which angular momentum passes through that small element of inlet cross-section is $\partial \dot{m} v_{w1} r_1$ and the total rate at which angular momentum enters the rotor is $\int v_{W1} r_1 \, d\dot{m}$, the integral being taken over the entire inlet cross-section. Similarly, the total rate at which angular momentum leaves the rotor is $\int v_{W2} r_2 \, d\dot{m}$, this integral being evaluated for the entire outlet cross-section. The rate of increase of angular momentum of the fluid is therefore

$$\int v_{w2} r_2 d\dot{m} - \int v_{w1} r_1 d\dot{m}$$

and this equals the torque exerted on the fluid. If there are no shear forces at either inlet or outlet cross-section that have a moment about the axis of the rotor, then this torque on the fluid must be exerted by the rotor. By Newton's Third Law, a change of sign gives the torque exerted on the rotor by the fluid:

$$T = \int v_{w1} r_1 d\dot{m} - \int v_{w2} r_2 d\dot{m} \qquad (4)$$

Equation (4) was, in principle, first given by Leonhard Euler (1707-83) and so is sometimes known as Euler's equation. It is a fundamental relation for all forms of rotodynamic machinery -turbines, pumps, fans or compressors. Although the equation has here been developed for a rotor, it applies also to a stationary member (stator) through which the angular momentum of the fluid is changed. A torque equal and opposite to *T* has to be applied to a stator - usually by fixing bolts - to prevent its rotation.

It is worth emphasizing that Eq. (4) is applicable regardless of changes of density or components of velocity in other directions. Moreover, the shape of the path taken by the fluid in moving from inlet to outlet is of no consequence: the expression involves only conditions at inlet and outlet.
In particular, it is independent of losses by turbulence, friction between the fluid and the blades, and changes of temperature. True, these factors may affect the velocity of whirl at outlet: they do not, however, undermine the truth of Eq. (4).

The torque *available* from the shaft of a turbine is somewhat less than that given by Eq.(4) because of friction in bearings and friction between the runner and the fluid outside it.

For a rotor, the shaft work done in unit time interval is

$$T\omega = \int v_{w1}\omega r_1 d\dot{m} - \int v_{w2}\omega r_2 d\dot{m} = \int u_1 v_{w1} d\dot{m} - \int u_2 v_{w2} d\dot{m} \qquad (5)$$

since $u = \omega r$.

The integrals in equations (4) and (5) can in general be evaluated only if it is known in what way the velocity varies over the inlet and outlet cross sections of the rotor. However, a simple result is obtained if the product $v_w r$ is constant at each cross-section concerned. This may be so if there is no significant variation of r at inlet or outlet (as in the rotor illustrated in Figure 13) and *vw* may be assumed uniform at each section. This assumption would be realistic only if the number of vanes guiding the fluid on to the rotor and also the number of blades in the rotor was large so that there would be no significant variation of either inlet or outlet values of v_w with angular position. In such a case, equation (5) becomes

$$T\omega = u_1 v_{w1} \int d\dot{m} - u_2 v_{w2} \int d\dot{m} \qquad (6)$$

Equation (6) is also obtained if the products $v_w r$ are constant both at inlet and outlet, even though v_w and r are not individually constant. Since the relation $v_w r$ = constant is that describing the velocity distribution in a 'free vortex' fluid machines designed according to this condition are frequently termed 'free-vortex machines'. Although this design criterion is widely used for axial-flow machines, it often has to be abandoned because of space limitations, especially close to the hub.

The shaft work done by the fluid per unit mass is obtained by dividing Eq.(5) by the total mass flow rate *m&* . Then work done per unit mass of fluid.

$$= \frac{1}{\dot{m}}\left(\int u_1 v_{w1} d\dot{m} - \int u_2 v_{w2} d\dot{m}\right) = u_1 v_{w1} - u_2 v_{w2} \qquad (7)$$

if the product *uvw* are individually constant.

Dividing this expression by *g* gives work per unit weight, i.e. head, and that form is sometimes known as the 'Euler head' (or 'runner head').

To a turbine the energy available per unit mass of the fluid is *gH* where *H* = the net head.

The hydraulic efficiency of the turbine is thus $\frac{u_1 v_{w1} - u_2 v_{w2}}{gH}$ (if the product uv_w are uniform). This represents the effectiveness with which energy is transferred from the fluid to the runner.

It should be distinguished from the overall efficiency of the machine because, owing to such losses as friction in the bearings and elsewhere, not all the energy received by the runner is available at the output shaft. Similar relations apply to a pump. There the transfer of energy is from rotor to fluid instead of from fluid to rotor and so the expressions (4)-(7) have reversed signs.

Many machines are so constructed that uniformity of conditions at inlet and outlet is impossible to achieve. In an axial-flow machine the blade velocity *u* and the blade angle *β* both vary along the blade; any velocity vector diagram will therefore apply, in general, only to one radius. In ‘mixed flow ‘machines the fluid leaves the rotor at various radii. Even Francis turbines nowadays usually have some ‘mixed flow' at the outlet; moreover, the inlet and outlet edges of the blades are not always parallel to the axis of rotation. Thus the expressions in which the products v_w are assumed uniform do not apply exactly to the flow considered as a whole.

In any case, the assumption that the velocities at inlet and outlet are uniform with respect to angular position is not fulfilled even for a rotor in which all the flow is in the plane of rotation. Individual particles of fluid may have different velocities. Since guide vanes and rotor blades are both limited in number, the direction taken by individual particles may differ appreciably from that indicated by the velocity diagram. Even the average direction of the relative velocity may differ from hat of the blade it is supposed to follow. Thus the velocity diagrams and the expressions based on them should be regarded as only a first approximation to the truth. In spite of these defects, however, the simple theory is useful in indicating ho the performance of a machine varies with changes in the operating conditions, and in what way the design of a machine should be altered in order to modify its characteristics.

With the limitations of the theory in mind we may examine the velocity diagrams further. Figure 13 shows the relative velocity of the fluid at inlet in line with the inlet edge of the blade. This is the ideal condition, in which the fluid enters the rotor smoothly. [A small angle of attack is generally desirable, but this rarely exceeds a few degrees.] If there is an appreciable discrepancy between the direction of R_1 and that of the blade inlet, the fluid is forced to change direction suddenly on entering the rotor. Violent eddies form, a good deal of energy is dissipated as useless heat, and the efficiency of the machine is consequently lowered. For all rotodynamic machines the correct alignment of blades with the velocities relative to them is very important. For the inlet diagram of a turbine (as in Figure 13) the angle α_1 defining the direction of the absolute velocity of the fluid, is determined by the setting of guide vanes. Smooth entry conditions can be achieved for a wide range of blade velocities and rates of flow by adjustment of the guide vanes and therefore of the angle α1. For each value of the angle α1 there is, however, only one shape of inlet velocity diagram that gives the ideal conditions. The angle of R_1 is then determined by the geometry of the vector diagram. At outlet the direction of the relative velocity R_2 is determined by the outlet angle of the blade (β_2) and the geometry of the outlet diagram then determines the magnitude and direction of the absolute velocity v_2.

Not all the energy of the fluid is used by a turbine runner. That remaining unused is principally in the form of kinetic energy and so, for high efficiency, the kinetic energy of the fluid at outlet should be small. For a given rate of flow the minimum value of v_2 occurs when v_2 is perpendicular to u_2 in the outlet vector diagram. The whirl component v_{w2} is then zero, the expression (7) for example becomes simply *u1vw1* and the hydraulic efficiency u_1v_{w1}/gH. Other losses in the machine do not necessarily reach their minimum values at the same conditions, however, and a small whirl component is therefore sometimes allowed in practice. The ideal outlet vector diagram is, in any case, not achieved under all conditions of operation. Nevertheless, a zero, or nearly zero, whirl component at outlet is taken as a basic requirement in turbine design.

It is instructive to derive an alternative form of Eq. (7). Assuming uniform velocities at inlet and outlet and referring again to the inlet vector diagram of Figure 13, we have

$$\Delta u_1 v_{w1} = \frac{1}{2}\left(u_1^2 + v_1^2 - R_1^2\right) \qquad (8)$$

Similarly

$$\Delta u_2 v_{w2} = \frac{1}{2}\left(u_2^2 + v_2^2 - R_2^2\right) \qquad (9)$$

Substituting equations (8) and (9) in equation (7), we obtain work done by the fluid per unit mass

$$\frac{1}{2}\left\{\left(v_1^2 - v_2^2\right) + \left(u_1^2 - u_2^2\right) - \left(R_1^2 - R_2^2\right)\right\} \qquad (10)$$

In an axial-flow machine the fluid does not move radially, and so for aparticular radius u_1 = u_2 and the term $u_1^2 - u_2^2$ is zero. In a radial or mixed-flow machine, however, each of the terms in the expression is effective. For a turbine, that is, a machine in which work is done by the fluid, the expression (10) must be positive. This is most easily achieved by the inward-flow arrangement. Then u_1 > u_2 and, since the flow passages decrease rather than increase in cross-sectional area, R_2 usually exceeds R_1. The contributions of the second and third brackets to the work done by the fluid are thus positive. By appropriate design, however, an outward flow turbine, although seldom desirable, is possible. An inward-flow machine has a number of advantages, an important one being, as already mentioned, that it is to some extent self-governing.

Conversely, in a pump work is done *on* the fluid and so the expression (10) then needs to be negative. Outward flow is thus more suitable for a pump.

3.1.5 Similarity Laws and Specific Speed

Geometric similarity is a prerequisite of dynamic similarity. For fluid machines the geometric similarity must apply to all significant parts of the system: the rotor, the entrance and discharge passages and, in the case of a reaction turbine or a pump, the conditions in the tail race or sump.
Machines that are geometrically similar in these respects form a *homologous series;* members of such a series are therefore simply enlargements or reductions of one another.

The performance of any machine depends not only on its size but on its shape - which is common to all machines in the same homologous series. The shape of machine required in a particular application may therefore be determined by reference to the known characteristics of various homologous series.

Dynamic similarity also requires kinematic similarity, that is, corresponding velocities in a constant ratio. If two machines are to be dynamically similar, then, in particular, those velocities represented in the vector diagram for inlet and outlet of the rotor of one machine must be similar to the corresponding velocities in the other machine. Geometric similarity of the inlet diagrams and of the outlet diagrams is therefore a necessary condition for dynamic similarity. The most succinct statement of the conditions for kinematic similarity (fixed ratio of velocities) and dynamic similarity (fixed ratio of forces) is that certain dimensionless parameters, representing these ratios, are the same for each of the systems being compared. To determine these dimensionless parameters dimensional analysis is being used.

For a machine of a given shape, and handling a homogeneous fluid of constant density, the relevant variables are shown in Table 1.

		Dimensional formula
D	rotor diameter, here chosen as a suitable measure of the size of the machine	[L]
Q	volume rate of flow through the machine	$[L^3T^{-1}]$
N	rotational speed	$[T^{-1}]$
H	difference of head across machine, i.e. energy per unit weight	[L]
g	weight per unit mass	$[LT^{-2}]$
ρ	density of fluid	$[ML^{-3}]$
μ	viscosity of fluid	$[ML^{-1}T^{-1}]$
P	power transferred between fluid and rotor	$[ML^2T^{-3}]$

Table 1 : Relevant Variables in Homogeneous Fluid of Constant Density

If data obtained from tests on a model turbine, for example, are plotted so as to show the variation of the dimensionless parameters Q/ND^3, gH/N^2D^2, $P/\rho N^3D^5$, η with one another, then the graphs are applicable to any machine in the same homologous series. The curves for other homologous series would naturally be different, but one set of curves would be sufficient to describe the performance of all the members of one series.

Particularly useful in showing the characteristics of turbines are results obtained under conditions of constant speed and head. For a particular machine and a particular incompressible fluid, D and p are constant. Then gH/N^2D^2 is constant. From equation $\phi_2\left(\eta, \frac{gH}{N^2D^2}, \frac{P}{\rho N^3D^5}\right) = 0$ η is then simply a function of P, and the results may be presented in the form shown in Figure 14.

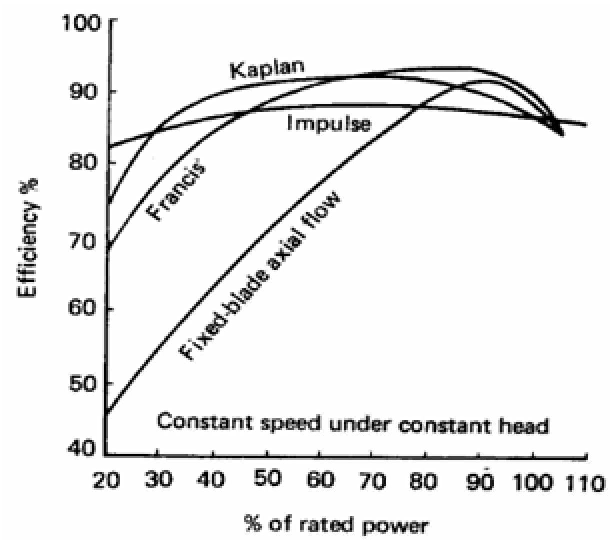


Figure 14: Typical Efficiency Curves

For a turbine using a particular fluid the operating conditions are expressed by values of *N, P* and *H.* It is important to know the range of these conditions which can be covered by a particular design (i.e. shape) of machine. Such information enables us to select the type of machine best suited to a particular application, and serves as a starting point in its design. We require therefore, a parameter characteristic of all the machines of a homologous series and independent of the size represented by *D.* A parameter involving *N, P* and *H* is obtained as

$$\frac{NP^{1/2}}{\rho^{1/2}\left(gH\right)^{1/5}} \qquad (11)$$

For complete similarity of flow in machines of a homologous series, each of the dimensionless parameters must be unchanged throughout the series. Consequently the expression (11) must be unchanged. A particular value of this ratio therefore relates all the combinations of *N, P* and *H* for which the flow conditions are similar in the machines of that homologous series.

The *specific speed* is defined as $N_s = \frac{NP^{\frac{1}{2}}}{H^{\frac{5}{4}}}$ The specific speed is not dimensionless, nor it is a speed, either linear or angular. It is better regarded as a 'shape parameter' which conveniently identifies the homologous series and thus classifies the geometrical shape of the machine. As it is not dimensionless, its numerical value depends on the units with which the constituent magnitudes are expressed.

The dimensionless parameter $K_n = NP^{\frac{1}{2}}\rho^{-\frac{1}{2}}(gH)^{-\frac{5}{4}}$ is often known as the dimensionless specific speed to distinguish it from Ns. However, K_n is truly dimensionless only if *N* involves the radian measure of angle. Yet *N* is invariably measured as number of revolutions per unit time interval and, since it would be pedantic to insist that this figure be multiplied by 2π, K_n is usually calculated with *N* as number of revolutions per unit time interval. Consequently 'rev' remains as the unit of K_n. (Similar remarks of course apply to other 'dimensionless' parameters in which *N* appears.)
When the site of the installation and the output required from a turbine are known, the value of K_n (or of N_s) may be calculated and the type of machine best suite to this condition selected. For the principal types of turbine, experience has shown the values in Table 2 most suitable.

Type of turbine	Approximate range of 'dimensionless specific speed' $K_n = NP^{1/2}\rho^{-1/2}(gH)^{-5/4}$ (rev)
Pelton (if there is more then one jet, *P* is taken as total power ÷ number of jets)	0.015 – 0.024
Francis (not for heads above about 370 m)	0.055 – 0.37
Kaplan (not for heads above about 60 m)	0.3 – 0.8

Table 2: Approximate Range of 'dimensionless specific speed' for different turbines

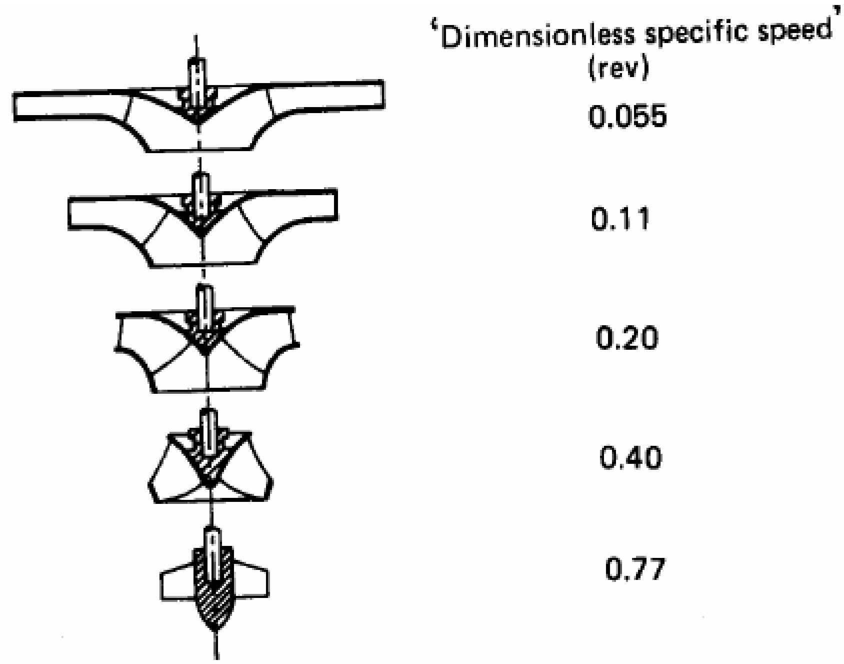


Figure 15: Specific speed for different shape of turbine runner

Figure 15 indicates the variation of specific speed with the shape of the turbine runner. (The values quoted should be regarded as approximate because the shape of other parts of the turbine – for example, the volute, the guide passages and the draft tube – affect the specific speed to some extent.) Forgiven values of H and P, N increases with N_s and K_n. With the same peripheral runner velocity, a larger value of N implies a smaller value of D and so, in general, lowers cost. For this reason, where a choice lies between two machines of different specific speeds, the designer usually prefers that with the higher value. Machines of higher specific speed, however, are limited to low heads because of cavitation.

3.1.6 Cavitation

Non-uniformity of flow in machines may cause the pressure, even in a given cross-section, to vary widely. There may thus be, on the low pressure side of the rotor, regions in which the pressure falls to values considerably below atmospheric. In a liquid, if the pressure at any point falls to the vapour pressure (at the temperature concerned), the liquid there boils and small bubbles of vapour form in large numbers. These bubbles are carried along by the flow, and on reaching a point where the pressure is higher they suddenly collapse as the vapour condenses to liquid again. A cavity results and the surrounding liquid rushes in to fill it. The liquid moving from all directions collides at the centre of the cavity, thus giving rise to very high local pressures (up to 1 GPa). Any solid surface in the vicinity is also subjected to these intense pressures, because, even if the cavities are not actually at the solid surface, the pressures are propagated from the cavities by pressure waves similar to those encountered in water hammer. This alternate formation and collapse of vapour bubbles may be repeated with a frequency of many thousand times a second. The intense pressures, even though acting for only a very brief time over a tiny area can cause severe damage to the surface. The material ultimately fails by fatigue, aided perhaps by corrosion, and so the surface becomes badly scored and pitted. Parts of the surface may even be torn completely away. Associated with cavitating flow there may be considerable vibration and noise; when cavitation occurs in a turbine or pump it may sound as though gravel were passing through the machine.

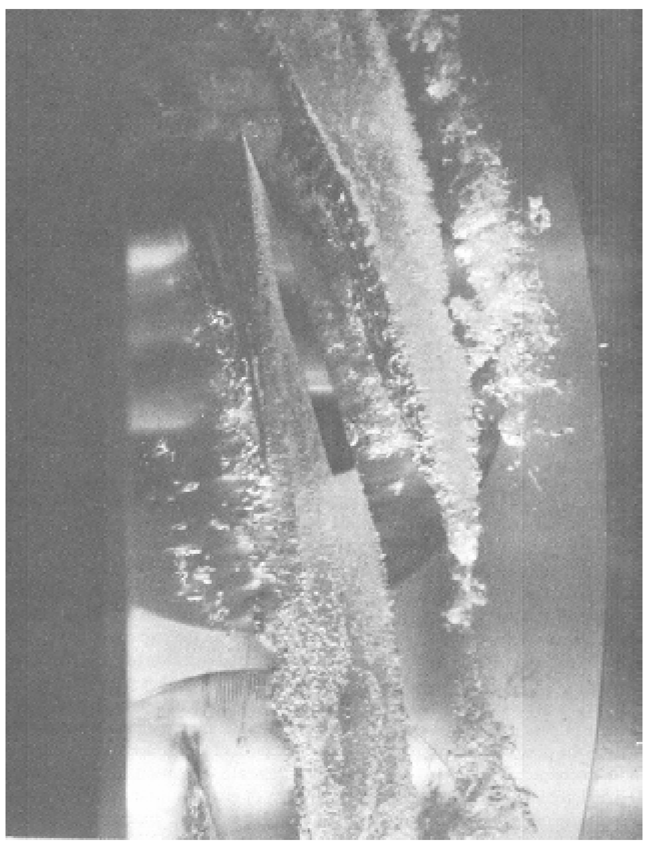


Figure 16: Cavitation

Figure 16 shows cavitation occurring. Everything possible should be done to eliminate cavitation in fluid machinery, that is, to ensure that at every point the pressure of the liquid is above the vapour pressure. When the liquid has air in solution this is released as the pressure falls and so *air cavitation* also occurs. Although air cavitation is less damaging than vapour cavitation to surfaces, it has a similar effect on the efficiency of the machine.

Since cavitation begins when the pressure reaches too low a value, it is likely to occur at points where the velocity or the elevation is high, and particularly at those where high velocity and high elevation are combined. In a reaction turbine the point of minimum pressure is usually at the outlet end of a runner blade, on the leading side. For the flow between such a point and the final discharge into the tail race (where the total head is atmospheric) the energy equation may be written

$$\frac{p_{\min}}{\rho g} + \frac{v^2}{2g} + z - h_r = \frac{p_{atm}}{\rho g} \qquad (12)$$

Here h_r represents the head lost to friction in the draft tube, and the pressures are absolute.

Equation (12) incidentally shows a further reason why the outlet velocity *v* of the fluid from the runner should be as small as possible: the larger the value of *v* the smaller is the value of *pmin* and the more likely is cavitation.

Rearranging the equation gives

$$\frac{v^2}{2g} - h_r = \frac{p_{atm}}{\rho g} - \frac{p_{\min}}{\rho g} - z$$

For a particular design of machine operated under its design conditions, the left-hand side of this relation may be regarded as a particular proportion, say σ0 of the net head *H* across the machine. Then

$$\sigma_c = \frac{p_{atm}/\rho g - p_{min}/\rho g - z}{H}$$

For cavitation not to occur p_{min} must be greater than the vapour pressure of the liquid, p_v , i.e.

$$\sigma > \sigma_c \text{ where } \sigma = \frac{p_{atm}/\rho g - p_v/\rho g - z}{H} \qquad (13)$$

The expression (13) is known as Thoma's cavitation parameter, after the German engineer Dietrich Thoma (1881-1943) who first advocated its use. If either *z* (the height of the turbine runner above the tail water surface) or *H* is increased, *σ* is reduced. To determine whether cavitation is likely in a particular installation, the value of *σ* may be calculated: if it is greater than the tabulated (empirical) value of *σ* for that design of turbine, cavitation should not be experienced.

In practice the expression is used to determine the maximum elevation z_{max} of the turbine above the tail water surface for cavitation to be avoided:

$$z_{max} = p_{atm}/\rho g - p_v/\rho g - \sigma_c H \qquad (14)$$

Equation (14) shows that the greater the net head *H* on which a given turbine operates, the lower it must be placed relative to the tail water level.

Figure 15 shows that turbines of high specific speed have higher values of σ_c *and* so they must be set much lower than those of smaller specific speed. For a high net head *H* it might be necessary to place the turbine *below* the tail water surface, thus adding considerably to the difficulties of construction and maintenance. This consideration restricts the use of propeller turbines to low heads to which, fortunately, they are best suited in other ways.

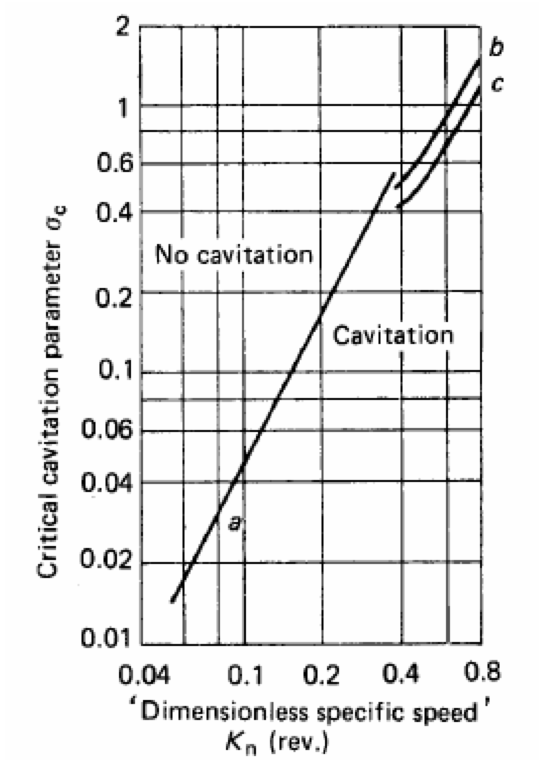


Figure 17: Cavitation limits for reaction turbines (a) Francis, (b) fixed-blade propeller and (c) Kaplan

Figure 17, it should be realized, is no more than a useful general guide; in practice the incidence of cavitation depends very much on details of the design.

The general effect of cavitation on the efficiency of a turbine is indicated by Figure 18.

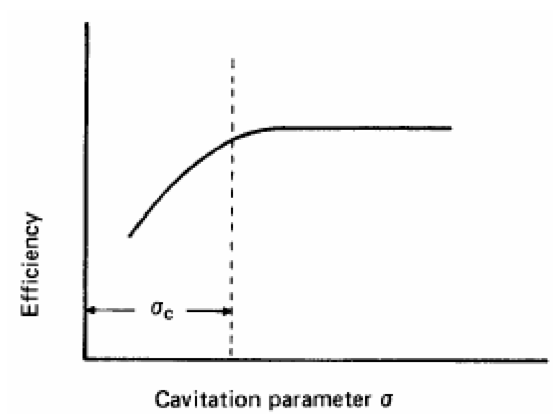


Figure 18: General effect of cavitation on the efficiency of a turbine

Cavitation is a phenomenon by no means confined to turbines. Wherever it exists it is an additional factor to be considered if dynamic similarity is sought between one situation and another. Similarity of cavitation requires the cavitation number $\frac{p - p_v}{\frac{1}{2}\rho v^2}$ to be the same at corresponding points. Experiments suggest, however, that similarity of cavitation is difficult to achieve.

3.1.7 The Performance Characteristics of Turbines

Although desirable, it is not always possible for a turbine to run at its maximum efficiency. Interest therefore attaches to its performance under conditions for which the efficiency is less than the maximum. In testing model machines it is usual for the head to be kept constant (or approximately so) while the load, and consequently the speed, are varied. If the head is constant then for each setting of the guide vane angle (or spear valve for a Pelton wheel) the power output *P*, the efficiency *η* and the flow rate *Q* may be plotted against the speed *N* as the independent variable.

It is more useful, however, to plot dimensionless parameters

$\frac{P}{\rho D^2 (gH)^{3/2}}$, $\frac{Q}{D^2 (gH)^{1/2}}$, $\frac{ND}{(gH)^{1/2}}$ as shown in Figure 18.

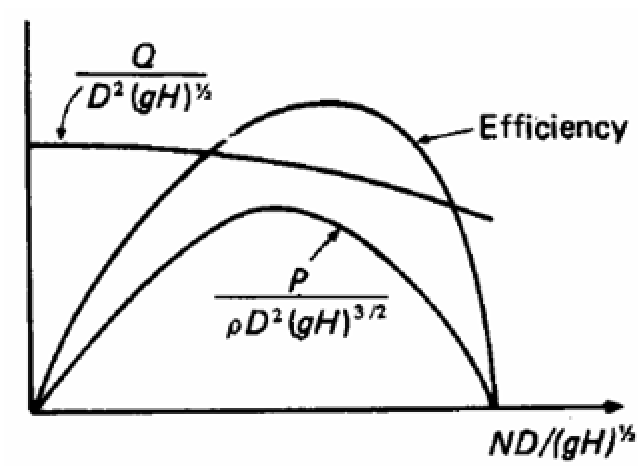


Figure 19: Dimensionless Parameter Curves

Thus one set of curves is applicable not just to the conditions of the test, but to any machine in the same homologous series, operating under any head.

More usually, however, the *p* and *g* terms are dropped from these dimensionless forms. Often the *D* terms are omitted also, and the resulting ratios $P/H^{3/2}$, $Q/H^{1/2}$, $N/H^{1/2}$ are then referred to as 'unit power', 'unit flow' and 'unit speed'. Their *numerical* values correspond respectively to the power, volume flow rate and speed obtainable if the machine could be operated with unchanged efficiency under one unit of head (e.g. 1 m).

Most turbines are required to run at constant speed so that the electrical generators to which they are coupled provide a fixed frequency and voltage. For an impulse machine at constant speed under a given head, the vector diagrams are independent of the rate of flow. In theory, then, the hydraulic efficiency should be unaffected by the load, although in practice there is a small variation of the efficiency. For a reaction turbine, changes of load are dealt with by alteration of the guide vane angle. (True, the power output could be altered by throttling the flow through a partly closed valve in the supply line. This process, however, would wastefully dissipate a large part of the available energy in eddy formation at the valve. Figure 20 shows the general effect of change of guide vane angle for a machine of the Francis type or fixed-blade propeller type. Only at the maximum efficiency point does the direction of the relative velocity at inlet conform to that of the inlet edges of the runner blades. At other conditions these directions do not conform, and so the fluid does not flow smoothly into the passages in the runner. Instead, it strikes either the front or back surfaces of the blades; considerable eddy formation ensues and the consequent dissipation of energy reduces the efficiency of the machine.

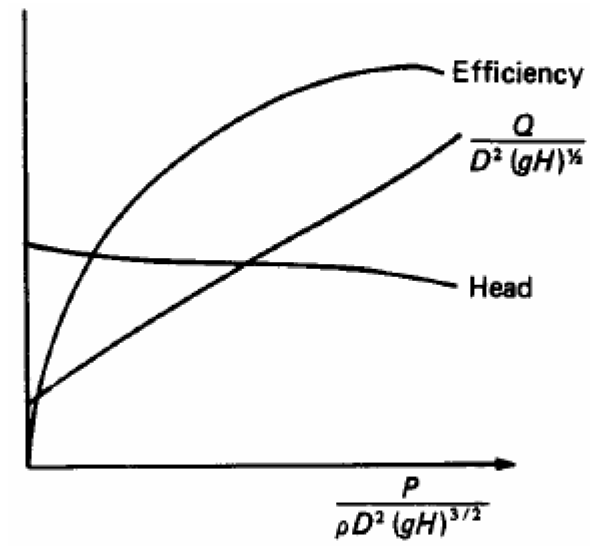


Figure 20: General effect of change of guide vane angle for Francis Turbine

In the Kaplan turbine the runner blade angle may be altered in addition to the guide vane angle. Thus it is possible to match the directions of the relative velocity at inlet and the inlet edges of the runner blades for a wide range of conditions. In consequence, the part-load efficiency of the Kaplan machine is superior to that of other types, as shown in Figure 14.

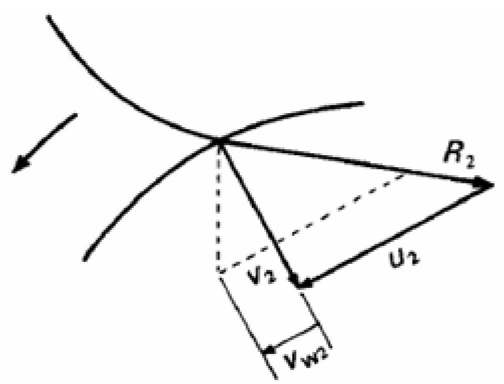


Figure 21: Effect of change of flow rate on outer vector triangle

A change of load also affects the conditions at outlet. A reduction in the rate of flow through the machine results in a decreased value of R_2. Consequently, if the blade velocity u_2 is unaltered, there is a departure from the ideal right-angled vector triangle at outlet (see Figure 21); the resulting whirl component of velocity causes a spiral motion in the draft tube and hence a reduction of the draft-tube efficiency. The possibility of cavitation is also increased.

3.2 Centrifugal Pumps

This type of pump is the converse of the radial-flow (Francis) turbine. Whereas the flow in the turbine is inwards, the flow in the pump is outwards (hence the term 'centrifugal'). The rotor (usually called 'impeller') rotates inside a spiral casing as shown in Fig. 19. The inlet pipe is axial, and fluid enters the 'eye', that is the centre, of the impeller with little, if any, whirl component of velocity. From there it flows outwards in the direction of the blades, and, having received energy from the impeller, is discharged with increased pressure and velocity into the casing. It then has a considerable tangential (whirl) component of velocity which is normally much greater than that required in the discharge pipe. The kinetic energy of the fluid leaving the impeller is largely dissipated in shock losses unless arrangements are made to reduce the velocity gradually. Figure 22 illustrates the simplest sort of pump, the 'volute' type. The volute is a passage of gradually increasing section which serves to reduce the velocity of the fluid and to convert some of the velocity head to static head. In this function the volute is often supplemented by a diverging discharge pipe.

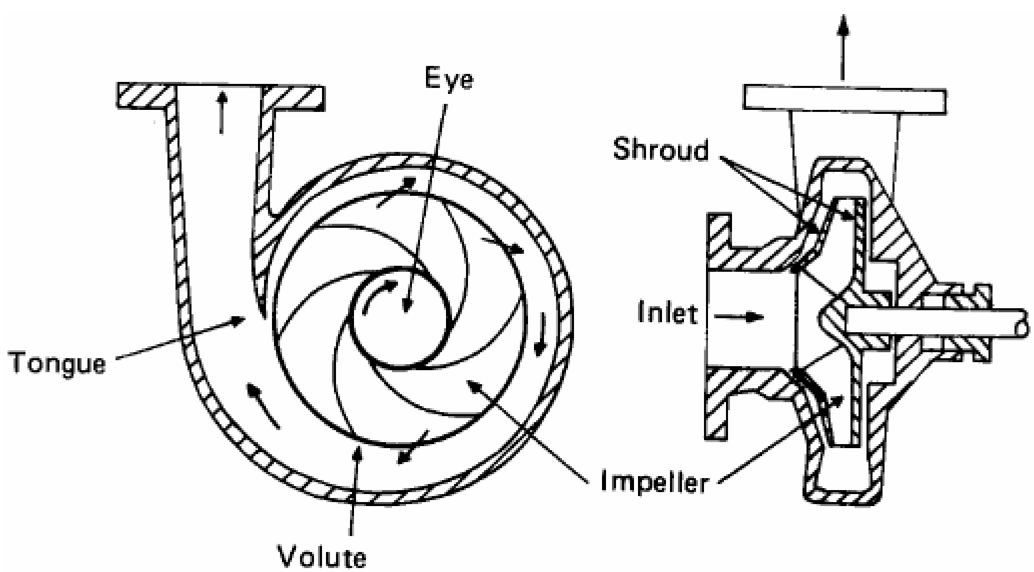


Figure 22: Volute-type centrifugal pump

A higher efficiency may be obtained by fitting a set of fixed guide vanes (a 'diffuser') round the outside of the impeller as shown in Figure 23. These fixed vanes provide more opportunity for the gradual reduction of the velocity of the fluid so that less energy is wasted in eddies. However, unless the absolute velocity of the fluid leaving the impeller is in line with the entry edges of the diffuser blades, shock losses there will offset the gain in efficiency otherwise to be obtained. Thus the diffuser yields an improved efficiency over only a limited range of conditions for which the diffuser blade angles are approximately correct. It is possible to fit diffuser blades having adjustable angles, but the considerable extra complication is not normally warranted. The diffuser, which adds appreciably to the cost and bulk of the pump, is therefore not an unmixed blessing and, except for large machines where running costs are important, the gain in efficiency is not enough to justify its wide use.

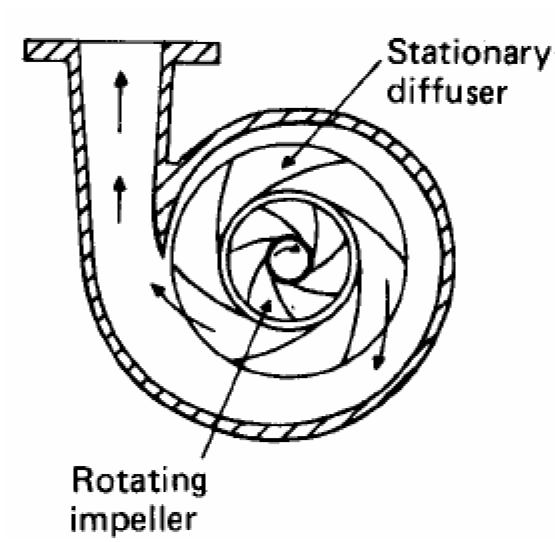


Figure 23: Diffuser-type centrifugal pump

The efficiency of a pump is in any case generally less than that of a turbine. Although the energy losses in the two types of machine are of the same kind, the flow passages of a pump are diverging, whereas those of a turbine are converging. The flow in a pump may therefore readily break away from the boundaries with consequent dissipation of energy in eddies. A modern diffuser pump (sometimes, unfortunately, called a 'turbine' pump because of its superficial resemblance to a reaction turbine) may have a maximum overall efficiency of well over 80%; the usual figure for the simpler volute pump is from 75% to 80%, although somewhat higher values are obtainable for large machines.

3.2.1 The Basic Equations Applied to Centrifugal Pumps

The relations developed in Section A1 are applicable to pumps no less than to turbines. The assumptions on which they are founded are equally important and may be recalled here: we consider steady flow, with velocities at inlet and outlet uniform both in magnitude and in the angle made with the radius. The energy imparted to the fluid by the impeller is given by equation (7) with the sign reversed since work is done *on* the fluid, not *by* it:

$$\text{Work done on fluid per unit mass} = u_2 v_{w2} - u_1 v_{w1} \qquad (15)$$

Suffix 1 again refers to the inlet and suffix 2 to the outlet even though $r_2 > r_1$ for a pump. This expression may be transformed, with the aid of the trigonometric relations for the vector triangles, to the equivalent form (corresponding to Eq.10):

$$\text{Work done on fluid per unit mass} = \frac{1}{2}\left\{\left(v_2^2 - v_1^2\right) + \left(u_2^2 - u_1^2\right) - \left(R_2^2 - R_1^2\right)\right\} \qquad (16)$$

A centrifugal pump rarely has any sort of guide vanes at inlet. The fluid therefore approaches the impeller without appreciable whirl and so the inlet angle of the blades is designed to produce a right-angled vector triangle at inlet (as shown in Figure 24). At conditions other than those for which the impeller was designed-for example, a smaller flow rate at the same shaft speed-the direction of the relative velocity *R1* does not coincide with that of a blade. Consequently the fluid changes direction abruptly on entering the impeller, eddies are formed and energy is dissipated. In addition, the eddies give rise to some back flow into the inlet pipe, thus causing the fluid to have some whirl before entering the impeller. Moreover, particularly if the pump is dealing with a fluid of high viscosity, some pre-whirl may be caused by viscous drag between the impeller and the incoming fluid.

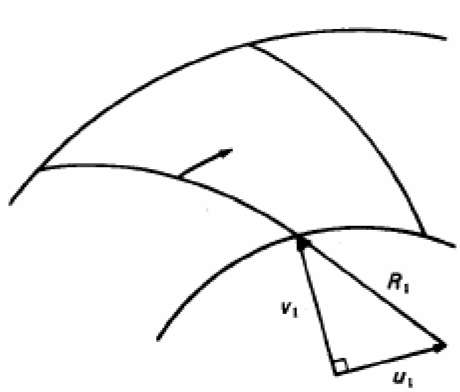


Figure 24: Right-angled vector triangle at inlet

Whatever the immediate cause of such pre-whirl, however, it will have come from the impeller. The initial angular momentum of the fluid may therefore be taken as zero. (This argument is equivalent to considering the inlet boundary of the 'control volume' further upstream of the impeller. Even in the enlarged control volume the impeller is the only thing providing torque.) We may therefore set v_{w1} = 0 in the Euler relation (15) to give

$$\text{Work done on fluid per unit mass} = u_2 v_{w2} \qquad (17)$$

It may be noted that since *u1* no longer enters the expression the work done is independent of the inlet radius. The increase of energy received by the fluid in passing through the pump is most often expressed in terms of head *H,* i.e. energy/weight. It should not be forgotten that this quantity depends on *g;* indeed, a pump in an orbiting space-craft, for example, where the fluid would be weightless, would generate an infinite head! The use of energy/mass is therefore now sometimes advocated, but 'head' still seems to be preferred, if only for the brevity of the term.

The theoretical work done per unit weight is often termed 'the Euler head' and is given by dividing Eq. (17) by *g.* The increase of total head across the pump is less than this, however, because of the energy dissipated in eddies and in friction. The gain in piezometric head, *p*/pg,* across the pump is known as the *manometric head Hm*: it is the difference of head that would be recorded by a manometer connected between the inlet and outlet flanges of the pump. (Accurate readings of inlet pressure, however, may be difficult to obtain if there is any swirling motion in the inlet pipe.)

The ratio of the manometric head to the Euler head is gH_m/u_2v_{w2} (i.e.$\Delta p^*/\rho u_2v_{w2}$) and is known as the *manometric efficiency:* it represents the effectiveness of the pump in producing pressure from the energy given to the fluid by the impeller.

Except for fans, the velocity heads at inlet and outlet are usually similar and small compared with the manometric head; thus the manometric head and the gain of total head are not greatly different. However, the overall efficiency - that is the ratio of the power given to the fluid *(QρgH,* where *H* represents the difference of total head across the pump) to the shaft power - is appreciably lower than the manometric efficiency. This is because additional energy has to be supplied by the shaft to overcome friction in the bearings, and in the fluid in the small clearances surrounding the impeller. The energy required at the shaft (per unit mass of fluid) thus exceeds u_2v_{w2}.

The performance of a pump depends (among other things) on the outlet angle of the impeller blades. The blade outlet may face the direction of rotation (i.e. forwards), be radial, or face backwards. The outlet blade angle 2 φ is usually defined as shown in Figure 25. Thus for forward-facing blades 2 φ > 90° and for backward-facing blades 2 φ < 90°

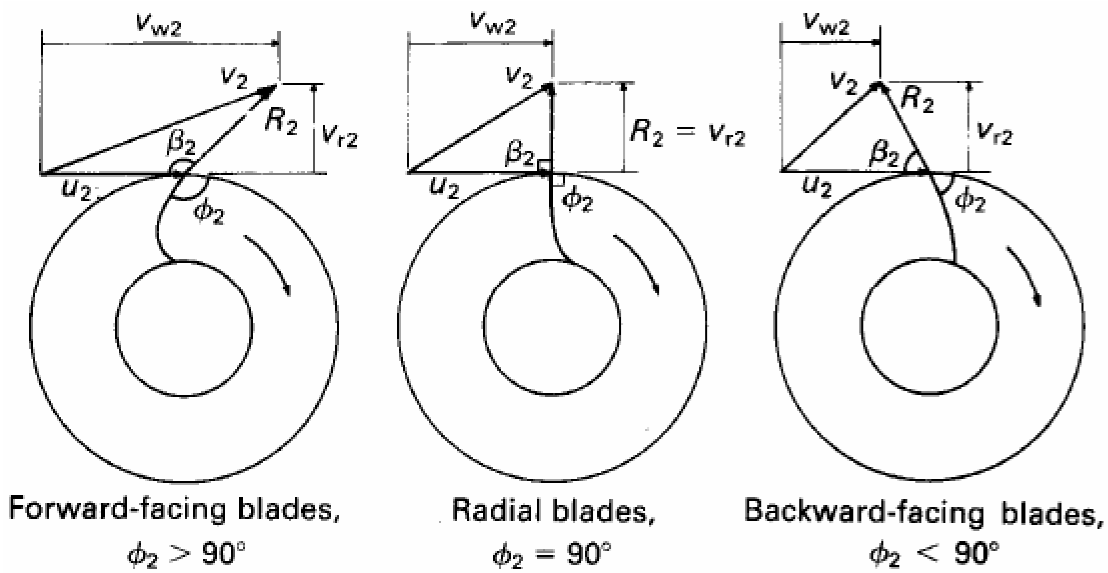


Figure 25: Differences in the outlet vector diagrams of the three types of impeller

We assume for the moment that there is no discrepancy between the direction of the relative velocity R_2 and the outlet edge of a blade. Consequently the blade outlet angle 2 φ is identical with the angle β_2 in the diagram. Figure 25 shows the differences in the outlet vector diagrams of the three types of impeller for the same blade velocity *u2.*

Ideally, that is if the fluid were frictionless, the increase of total energy across the pump would be $u_2 v_{w2}$ per unit mass. Now, from Figure 25, $v_{w2} = u_2 - v_{r2} \cot \beta$ where v_r represents the radial component of the fluid velocity (sometimes termed 'velocity of flow'). If *Q* represents the volume rate of flow through the pump and *A2* the outlet area perpendicular to v_{r2} (that is, the peripheral area of the impeller less the small amount occupied by the blades themselves) then for uniform conditions $v_{r2} = Q/A2$. The ideal increase of energy per unit mass therefore equals

$$u_2 v_{w2} = u_2 (u_2 - v_{r2} \cot \beta_2) = u_2 \left(1 - \frac{Q}{A_2} \cot \beta_2 \right) \qquad (18)$$

The blade velocity u_2 is proportional to the rotational speed N and so the ideal energy increase gH equals $C_1N^2 - C_2NQ$ where C_1 and C_2 are constants. Thus for a fixed speed N the variation of H linear as shown in Figure 26.

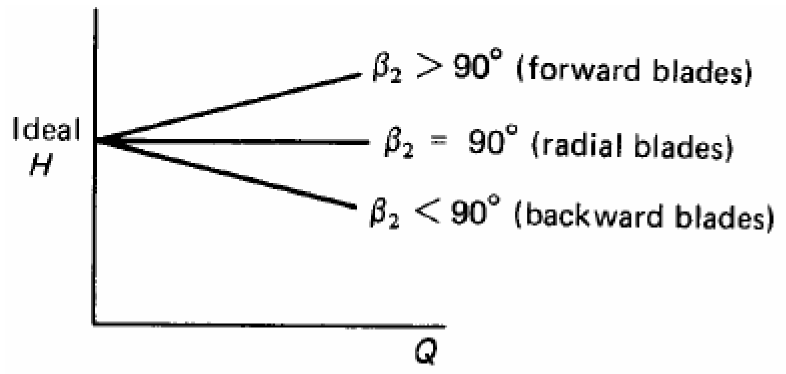


Figure 26: Variation of H Linear for a fixed speed N

In practice, however, energy losses occur and some of the assumptions on which Eq. (18) rests are not fulfilled. Consider the flow in the volute. Apart from frictional effects, no torque is applied to a fluid particle once it has left the impeller. The angular momentum of the particle is therefore constant, that is it follows a path along which $v_w r$ = constant. Ideally, the radial velocity from the impeller does not vary round the circumference. The combination of uniform radial velocity with the free vortex ($v_w r$ =constant) gives a pattern of spiral streamlines which should be matched by the shape of the volute. The latter is thus an important feature of the design of the pump. At maximum efficiency about 10% of the energy increase produced by the impeller is commonly lost in the volute. Even a perfectly designed volute, however, can conform to the ideal streamline pattern at the design conditions only. At rates of flow greater or less than the optimum there are increased shock losses in the volute (Figure 27); in addition there are variations of pressure and therefore of radial velocity v_{r2} round the impeller.

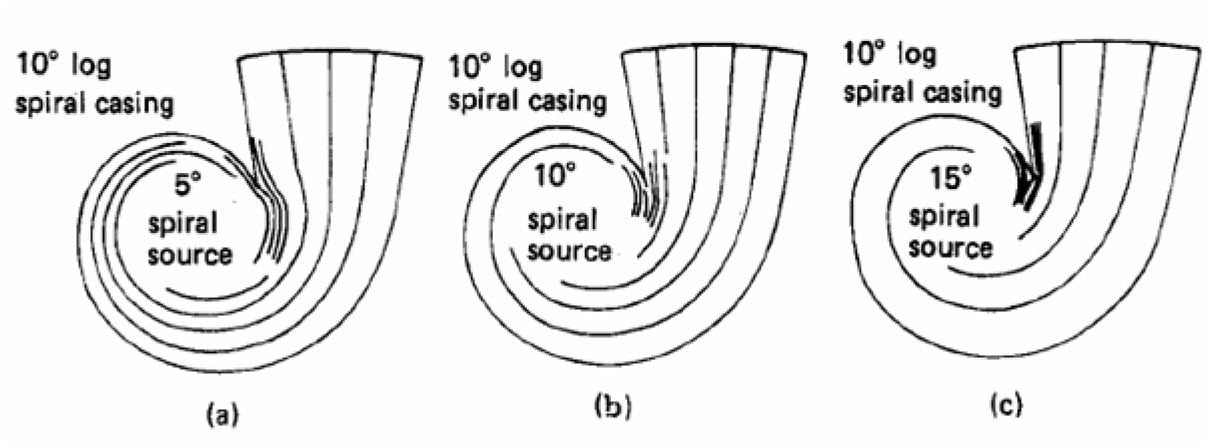


Figure 27: Increased shock losses in the volute

Shock losses are also possible at the inlet to the impeller. At conditions different from those of the design the fluid does not approach the blades in the designed direction. Consequently energy losses arise from the turbulence generated by the impact of the fluid against the blades. The energy (per unit mass) lost in this way may be shown to be approximately proportional to $(Q - Q_{ideal})^2$ for a given rotational speed *N.* Other, smaller, losses arise from friction between the fluid and the boundaries; there is also the recirculation of a small quantity of the fluid after leakage through the clearance spaces outside the impeller (Figure 28).

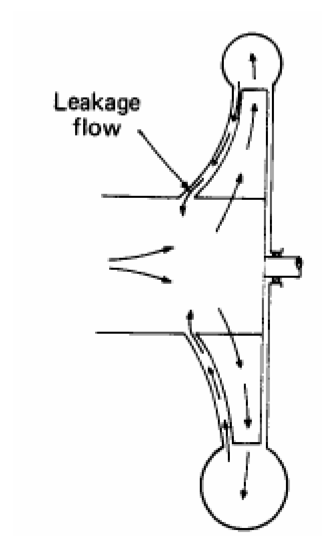


Figure 28: Recirculation of a small quantity of the fluid after leakage

All these effects modify the 'ideal' relation between head and discharge illustrated in Figure 28, and the 'characteristic curves' for the pump take on the shapes shown in Figure 29.

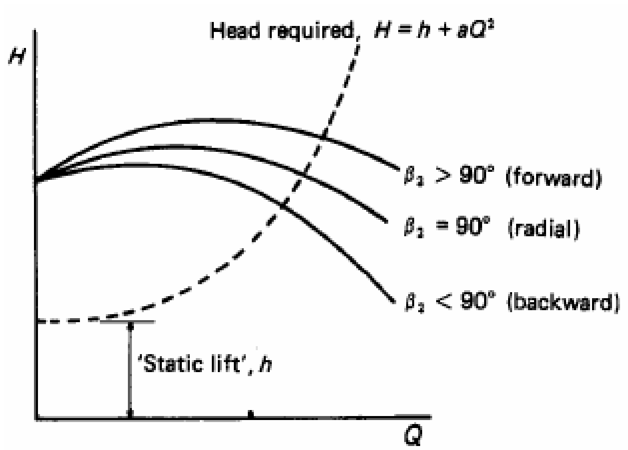


Figure 29: Characteristic curves for the pump take on the shapes

The pump, however, cannot be considered in isolation. The operating conditions at any moment are represented by a certain point on the characteristic curve. This point is determined by the conditions in the external system (that is the pipework, valves, and so on) to which the pump is connected. For example, a pump may be used to lift a liquid from a sump to a higher tank, the vertical distance involved being *h.* In addition to supplying this 'static lift' *h,* the pump must also provide sufficient head to overcome pipe friction and other losses (such as those at valves). As the flow is normally highly turbulent all these losses are proportional to the square of the velocity and thus to Q^2 (say aQ^2). The head required is therefore represented by the line $H = h + aQ^2$ in Fig. 26. The point of intersection of this line with the characteristic curve gives the values of *H* and *Q* at which the pump operates under these conditions. In other words, the pump offers one relation between *H* and *Q*, the external system offers another; these two relations must be satisfied simultaneously.

Impellers with backward-facing blades are often preferred although they give a smaller head for a given size and rate of flow. As shown in Figure 25, this type of impeller gives the fluid a smaller absolute velocity v_2; thus the dissipation of energy in the volute is less and the efficiency correspondingly higher. Such impellers also have the advantage of a negative slope to the characteristic curve for nearly all values of *Q*: this enables two machines to be used in parallel without instability.

3.2.2 The Performance Characteristics of Pumps

Pumps are normally run at constant speed. Interest therefore attaches to the variation of head *H* with discharge *Q* and also to the variation of the efficiency and power required with *Q*. The result of the particular test may be made available for a different speed-or for a homologous pump of different diameter-by plotting the results in dimensionless form that is, using the dimensionless parameters *Q/ND3*, gH/N^2D^2 and P/pN^3D^3 in place of *Q, H* and *P* respectively. Typical 'characteristic curves' for pumps are shown in Figure 30.

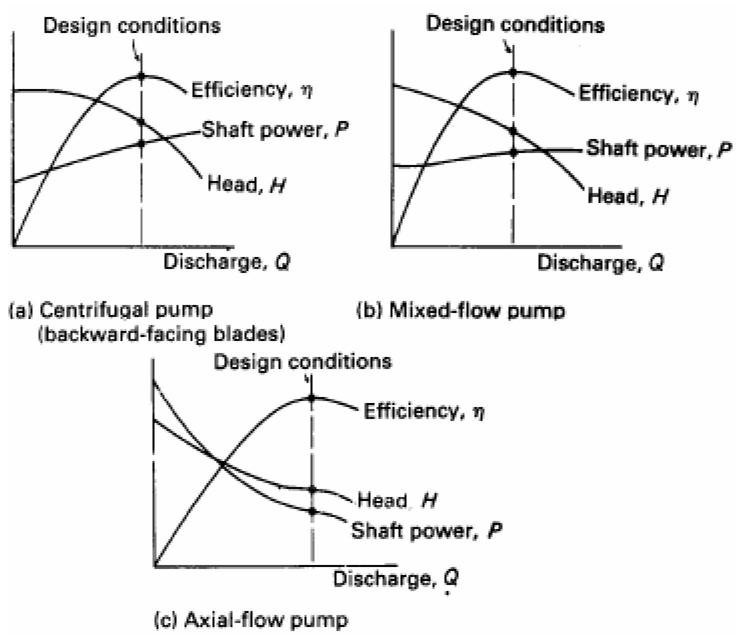


Figure 30 : Typical characteristic curves for pumps

A curve of *H* against *Q* which has a peak is termed an unstable characteristic. This is because the slope of the *H* - *Q* curve for an external system is normally always positive (see the dashed line in Figure 30). If part of the pump characteristic also has a positive slope there is thus the possibility that, at the point of intersection, the pump characteristic could have a greater slope than the other curve. Any slight increase of *Q*, for example, would then result in the pump head rising more than the system head, and the excess head at the pump would cause *Q* to increase still further. Moreover, the two curves could intersect again at a second, higher, value of *Q*; if the two intersections were fairly close together the pump would tend to 'hunt' (or 'surge') from one to the other. Such instability is undesirable and so an unstable part of the characteristic should be outside the normal operating range of the pump.

4.0 EXPERIMENTAL PROCEDURES

4.1 General Start-Up Procedures

Before conducting any experiment, it is necessary to do the following checking to avoid any misused or malfunction equipment.

1. Please make sure that the water tank is filled with water up to at least 50% of the full tank.
2. Switch on the main power supply located on the control panel. The instruments should light up.
3. For the Francis turbine experiment, limit the centrifugal pump speed to 2500 RPM.
4. For the Pelton turbine experiment, limit the centrifugal pump speed to 2500 RPM.

4.2 Experiment to Study the Operating and Performance Characteristics of a Pelton Turbine

4.2.1 Installation and Assembly Instructions

Please refer to Appendix C.

4.2.2 Operating Instructions

Please refer to the general start-up procedure before conducting this experiment.

4.2.3 Experiment 1: Effect of fixed spear valve settings under varied load Conditions

Objective: To study the effect of the spear valve settings under varied load conditions

Procedures:

1. Turn on the centrifugal pump by pushing the green button.
2. Set the pump speed at 2500 RPM.
3. Open the bleeding valve to ensure no air contained in the tube that is connected to the input spear valve pressure transmitter. Once the air is released, closed the valve immediately.
4. Set the spear valve to be fully opened by turning the knob away from the water jet direction.
5. Turn on the load switch L1 by flicking up the switch while the rest of the load switch remains off (L2-L5 switch pointing downward). Observe the changes and record down all the parameters shown on the panel.
6. Repeat step 5 by varying the load switch combination in at least 8 different combinations.

7. Repeat step 4-6 by varying the spear valve opening in at least 3 different opening positions.

Assignments:

i. Calculate the following:
   a. Power generated
   b. Turbine input power
   c. Turbine efficiency
ii. Plot a graph for the power generated versus turbine speed.
iii. Plot a graph for the turbine efficiency versus turbine speed.

4.3 Experiment to Study the Operating and Performance Characteristics of a Francis Turbine

4.3.1 Installation and Assembly Instructions

Please refer to Appendix C.

4.3.2 Operating Instructions

Please refer to the general start-up procedure before conducting this experiment.

4.3.3 Experiment 2: Effect of fixed guide vane opening settings under varied load Conditions

Objective: To study the effect of the guide vane opening under varied load conditions

Procedures:

1. Turn on the centrifugal pump by pushing the green button.
2. Set the pump speed at 2500 RPM.
3. Open the bleeding valve to ensure no air contained in the tube that is connected to the input spear valve pressure transmitter. Once the air is released, closed the valve immediately.
4. Set the guide vane to be fully opened by turning the handle against the water inlet direction. It may be necessary to slacken the retaining ring screws by a small amount. Fix back the ring screw once the vane position is fixed.
5. Turn on the load switch L1 by flicking up the switch while the rest of the load switch remains off (L2-L5 switch pointing downward).
6. Observe the changes and record down all the parameters shown on the panel.
7. Repeat step 5 by varying the load switch combination in at least 8 different combinations.

8. Repeat step 4-6 by varying the guide vane opening in at least 3 different opening positions.

Assignments:

i. Calculate the following:
   a. Power generated
   b. Turbine input power
   c. Turbine efficiency
ii. Plot a graph for the power generated versus turbine speed.
iii. Plot a graph for the turbine efficiency versus turbine speed.

4.4 Operating and Performance Characteristics of a Kaplan Turbine

4.4.1 Installation and Assembly Instructions

Please refer to Appendix C.

4.4.2 Operating Instructions

Please refer to the general start-up procedure before conducting this experiment.

4.4.3 Experiment 3: Effect of guide blade opening angles under varied load Conditions

Objective: To study the effect of the guide blade opening angles under varied load conditions

Procedures:

1. Turn on the centrifugal pump by pushing the green button.
2. Set the pump speed to the maximum.
3. Open the bleeding valve to ensure no air contained in the tube that is connected to the input spear valve pressure transmitter. Once the air is released, closed the valve immediately.
4. Monitor the PT4 indicator reading and the set the guide vane until the pressure reading is approximately indicating "2.23" by turning the handle against the water inlet direction. It may be necessary to slacken the retaining ring screws by a small amount. Fix back the ring screw once the vane position is fixed.
5. Turn on the load switch L1 by flicking up the switch while the rest of the load switch remains off (L2-L5 switch pointing downward). Observe the changes and record down all the parameters shown on the panel.

6. Change the turbine blade to a different angle by unscrewing the perspex cover. Secure back the Perspex once the turbine blade is switched.
7. Repeat step 5 by varying the load switch combination in at least 8 different combinations.
8. Repeat step 4-7 by changing the turbine blade angle.

Assignments:

i. Calculate the following:
   a. Power generated
   b. Turbine input power
   c. Turbine efficiency
ii. Plot a graph for the power generated versus turbine speed for each turbine blade.
iii. Plot a graph for the turbine efficiency versus turbine speed for each turbine blade.

4.5 Operation and Performance Characteristics of a Centrifugal Pump

4.5.1 Operating Instructions

Please refer to the general start-up procedure before conducting this experiment.

4.5.2 Experiment 4: Characteristic Study of a Centrifugal Pump

Objective: To study the characteristic curves of the Centrifugal Pump

Procedures:

1. Turn on the centrifugal pump by pushing the green button.
2. Set the pump speed to the maximum.
3. Set the guide vane to be fully opened. It may be necessary to slacken the retaining ring screws by a small amount. Fix back the ring screw once the vane position is fixed.
4. Reduce to speed to 100 RPM lower than the current speed. Observe the changes and record down all the parameters shown on the panel.
5. Repeat step 4 for at least 8 different speeds.
6. Set back the pump speed to the maximum.
7. Turn the turbine inlet valve knob 180o in closing direction. Observe the changes and record down all the parameters shown on the panel.
8. Repeat step 7 for at least 8 different valve opening positions.

Assignments:

i. Calculate the following:

   a. Pump output power

   b. Total head

   c. Overall efficiency

ii. Plot a graph for the output power versus pump speed.

iii. Plot a graph for the total head versus water flowrate.

iv. Plot a graph for the overall efficiency versus water flowrate.

5.0 REFERENCES

1. Massey B.S. Mechanics of Fluids. Sixth Edition. Van Nostrand Reinhold, 1989.
2. Hamill L. Understanding Hydraulics, Macmillan Press LTD, 1995.

Appendix A
Experimental Data Sheet

Experiment 1: Effect of fixed spear valve settings under varied load conditions

Experiment Data Set No:			1	2	3	4	5	6	7	8
Local Combinations			L1	L2	L1+L2	L3	L1+L3	L1+L2+L3	L4	L5
Pump Inlet Pressure	PT1	Bar Abs								
Pump Outlet Pressure	PT2	Bar Abs								
Turbine Stage 1 Pressure	PT3	Bar Abs								
Turbine Stage 2 Pressure	PT4	Bar Abs								
Water Flowrate	FT	Liter/min								
Pump Speed	SP1	Rpm								
Pump Torque	TQ1	Nm								
Pump Power	PWR1	W								
Turbine Speed	SP2	Rpm								
Turbine Torque	TQ2	Nm								
Turbine Voltage	V1	V								
Turbine Current	I1	A								

Experiment 2: Effect of fixed guide vane opening settings under varied load conditions

Experiment Data Set No:			1	2	3	4	5	6	7	8
Local Combinations			L1	L2	L1+L2	L3	L1+L3	L1+L2+L3	L4	L5
Pump Inlet Pressure	PT1	Bar Abs								
Pump Outlet Pressure	PT2	Bar Abs								
Turbine Stage 1 Pressure	PT3	Bar Abs								
Turbine Stage 2 Pressure	PT4	Bar Abs								
Water Flowrate	FT	Liter/min								
Pump Speed	SP1	Rpm								
Pump Torque	TQ1	Nm								
Pump Power	PWR1	W								
Turbine Speed	SP2	Rpm								
Turbine Torque	TQ2	Nm								
Turbine Voltage	V1	V								
Turbine Current	I1	A								

Experiment 3: Effect of guide blade opening angles under varied load conditions

Blade angle:

Experiment Data Set No:			1	2	3	4	5	6	7	8
Local Combinations			L1	L2	L1+L2	L3	L1+L3	L1+L2+L3	L4	L5
Pump Inlet Pressure	PT1	Bar Abs								
Pump Outlet Pressure	PT2	Bar Abs								
Turbine Stage 1 Pressure	PT3	Bar Abs								
Turbine Stage 2 Pressure	PT4	Bar Abs								
Water Flowrate	FT	Liter/min								
Pump Speed	SP1	Rpm								
Pump Torque	TQ1	Nm								
Pump Power	PWR1	W								
Turbine Speed	SP2	Rpm								
Turbine Torque	TQ2	Nm								
Turbine Voltage	V1	V								
Turbine Current	I1	A								

Experiment 4: Characteristic Study of a Centrifugal Pump

Experiment Data Set No:			1	2	3	4	5	6	7	8
Indicative Pump Speed (RPM)										
Pump Inlet Pressure	PT1	Bar Abs								
Pump Outlet Pressure	PT2	Bar Abs								
Water Flowrate	FT	Liter/min								
Pump Speed	SP1	Rpm								
Pump Torque	TQ1	Nm								
Pump Power	PWR1	W								

Experiment Data Set No:			1	2	3	4	5	6	7	8
Indicative Valve Opening (no. of turns)			0	0.5	1	2	3	4	4.5	5
Pump Inlet Pressure	PT1	Bar Abs								
Pump Outlet Pressure	PT2	Bar Abs								
Water Flowrate	FT	Liter/min								
Pump Speed	SP1	Rpm								
Pump Torque	TQ1	Nm								
Pump Power	PWR1	W								

Appendix B
Sample Experiment Results & Sample Calculations

Experiment 1: Effect of fixed spear valve settings under varied load conditions

Experiment Data Set No:			1	2	3	4	5	6	7	8
Local Combinations			L1	L2	L1+L2	L3	L1+L3	L1+L2+L3	L4	L5
Pump Inlet Pressure	PT1	Bar Abs	1.01	1.01	1.01	1.01	1.01	1.01	1.01	1.01
Pump Outlet Pressure	PT2	Bar Abs	3.50	3.51	3.51	3.50	3.50	3.50	3.50	3.50
Turbine Stage 1 Pressure	PT3	Bar Abs	3.50	3.50	3.50	3.50	3.50	3.50	3.50	3.50
Turbine Stage 2 Pressure	PT4	Bar Abs	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38
Water Flowrate	FT	Liter/min	174.4	172.6	175.5	173.0	167.9	173.4	176.3	176.4
Pump Speed	SP1	Rpm	2425	2423	2425	2422	2426	2426	2420	2424
Pump Torque	TQ1	Nm	6.7	6.6	6.6	6.6	6.6	6.6	6.6	6.6
Pump Power	PWR1	W	2150	2152	2135	2154	2147	2162	2096	2075
Turbine Speed	SP2	Rpm	2845	2761	2697	1774	1746	1687	1268	608
Turbine Torque	TQ2	Nm	0.6	0.8	1.0	2.7	2.6	2.8	3.4	4.1
Turbine Voltage	V1	V	148.7	143.2	138.9	82.8	80.9	77.6	52.8	15.5
Turbine Current	I1	A	0.40	0.73	1.08	4.55	4.67	4.87	6.17	7.60
Power Generated (W)			59.48	104.54	150.01	376.74	377.80	377.91	325.78	117.80
Turbine Input Power (W)			687.9	680.8	692.3	682.4	662.3	684.0	695.4	695.8
Turbine Efficiency (%)			8.65	15.35	21.67	55.21	57.04	55.25	46.85	16.93

Power Generated

$$P = I \times V$$
$$= 148.7\ V \times 0.4\ A$$
$$= 59.48\ W$$

Turbine Input Power

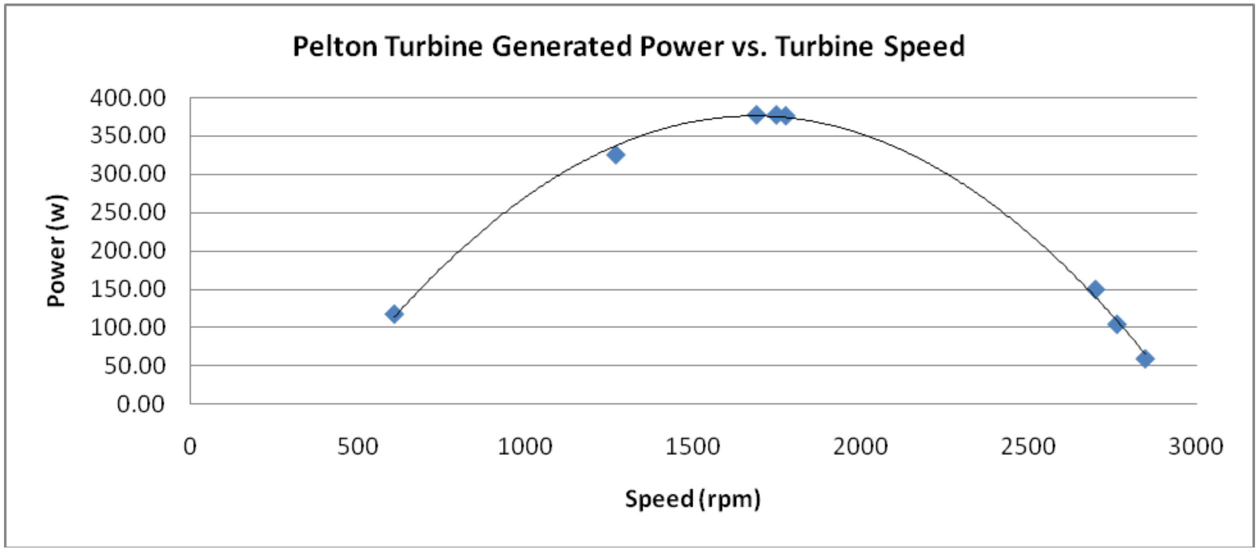
$P_{input} = P_{gage} \bullet Q$
$= (PT4 - P_{atm}) \bullet Q$
$=$
$= 687.9\ W$

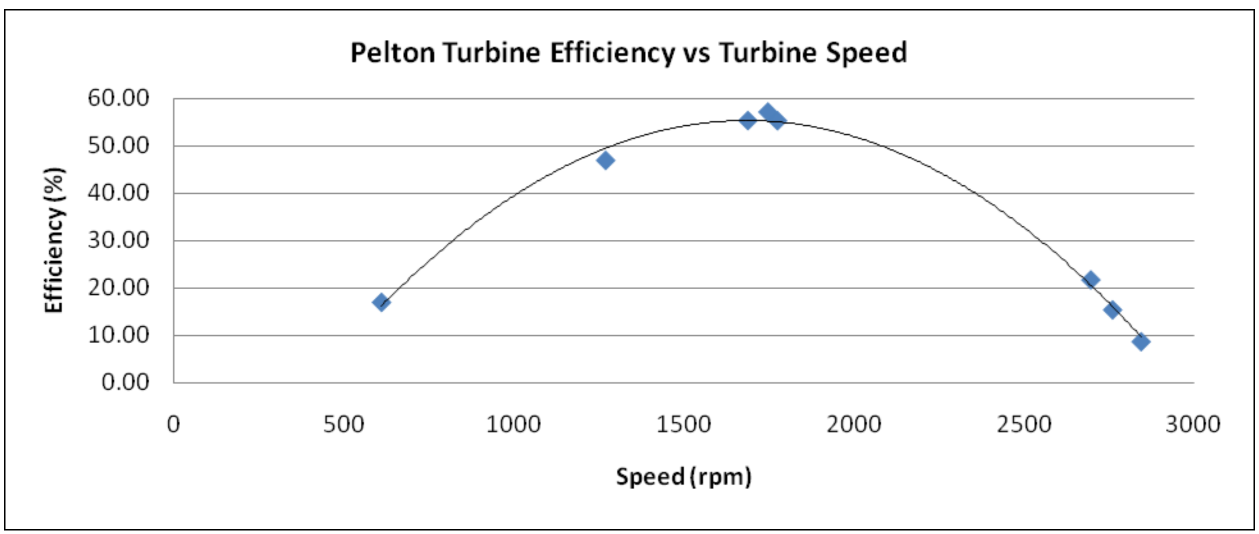
Turbine Efficiency

$$Eff_{turbine} = \frac{P_{generated}}{P_{input}}$$

$$= \frac{59.48\ W}{687.9\ W} \times 100\%$$

$$= 8.65\ \%$$


Pelton Turbine Generated Power vs. Turbine Speed
400.00
350.00
300.00
250.00
200.00
150.00
100.00
50.00
0.00
Power (w)
0
500
1000
1500
2000
2500
3000
Speed (rpm)


Pelton Turbine Efficiency vs Turbine Speed
60.00
50.00
40.00
30.00
20.00
10.00
0.00
Efficiency (%)
0
500
1000
1500
2000
2500
3000
Speed (rpm)

Experiment 2: Effect of fixed guide vane opening settings under varied load conditions

Experiment Data Set No:			1	2	3	4	5	6	7	8
Local Combinations			L1	L2	L1+L2	L3	L1+L3	L1+L2+L3	L4	L5
Pump Inlet Pressure	PT1	Bar Abs	0.95	0.95	0.94	0.92	0.92	0.93	0.92	0.93
Pump Outlet Pressure	PT2	Bar Abs	2.94	2.90	2.86	2.57	2.59	2.59	2.54	2.55
Turbine Stage 1 Pressure	PT3	Bar Abs	2.84	2.80	2.76	2.45	2.45	2.46	2.41	2.45
Turbine Stage 2 Pressure	PT4	Bar Abs	2.15	2.09	2.03	1.51	1.53	1.50	1.45	1.44
Water Flowrate	FT	Liter/min	282.9	290.0	293.0	327.9	323.6	322.1	325.8	329.8
Pump Speed	SP1	Rpm	2512	2512	2515	2512	2513	2515	2513	2514
Pump Torque	TQ1	Nm	8.1	8.2	8.2	8.4	8.3	8.4	8.4	8.3
Pump Power	PWR1	W	2782	2752	2758	2773	2746	2729	2863	2797
Turbine Speed	SP2	Rpm	2664	2559	2487	1419	1392	1324	930	426
Turbine Torque	TQ2	Nm	0.5	0.7	0.8	2.0	2.1	2.2	2.5	2.9
Turbine Voltage	V1	V	139.4	133.5	128.0	66.1	64.1	60.6	38.4	10.7
Turbine Current	I1	A	0.37	0.68	1.00	3.63	3.70	3.83	4.47	5.22
Power Generated (W)			51.58	90.78	128.00	239.94	237.17	232.10	171.65	55.85
Turbine Input Power (W)			325.3	343.2	356.5	513.7	496.2	515.4	521.3	555.2
Turbine Efficiency (%)			15.85	26.45	35.91	46.71	47.80	45.04	32.93	10.06

Power Generated

$$P = V \times I$$
$$= 139.4\ V \times 0.37\ A$$
$$= 51.58\ W$$

Turbine Input Power

$P_{input} = P_{gage} \bullet Q$
$= (PT3 - PT4) \bullet Q$

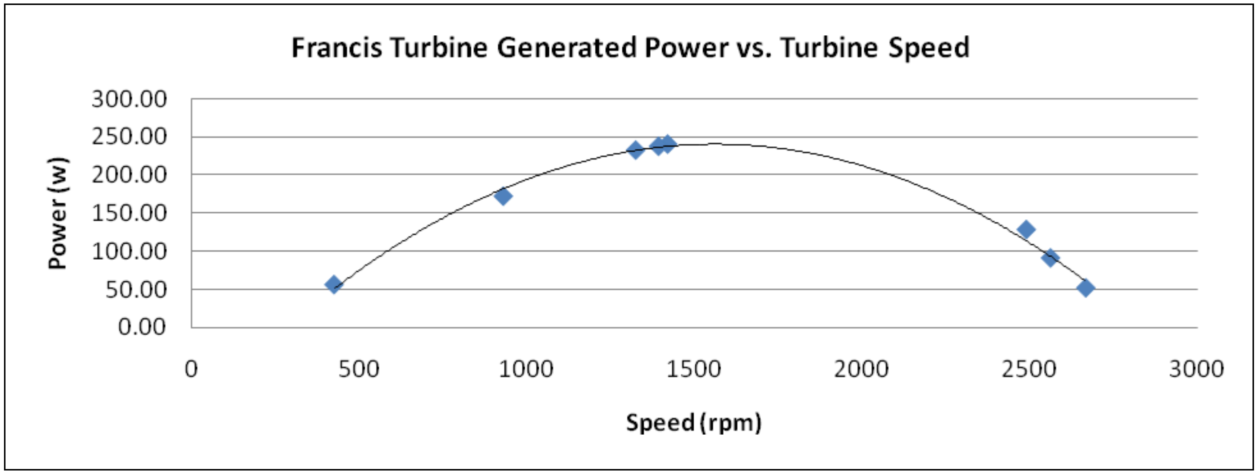
=
= 325.3 W

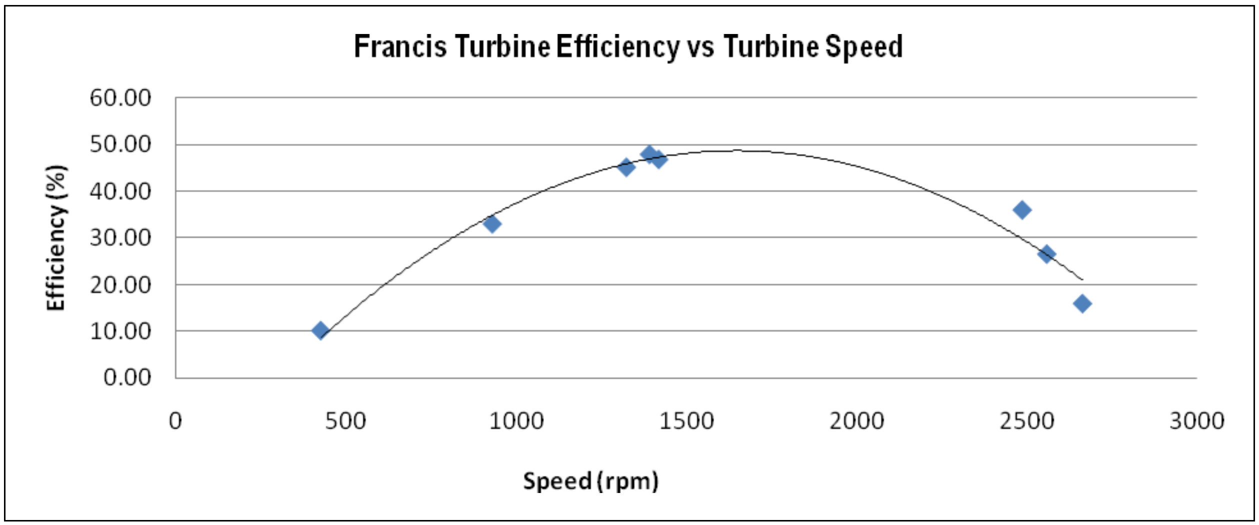
Turbine Efficiency

$$Eff_{turbine} = \frac{P_{generated}}{P_{input}}$$

$$= \frac{51.58\ W}{325}.3W \times 100\%$$

$$= 15.85\ \%$$


Francis Turbine Generated Power vs. Turbine Speed
Power (w)
300.00
250.00
200.00
150.00
100.00
50.00
0.00
0
500
1000
1500
2000
2500
3000
Speed (rpm)


Francis Turbine Efficiency vs Turbine Speed
Efficiency (%)
60.00
50.00
40.00
30.00
20.00
10.00
0.00
0
500
1000
1500
2000
2500
3000
Speed (rpm)

Experiment 3: Effect of guide blade opening angles under varied load conditions

Blade angle: 30^0

Experiment Data Set No:			1	2	3	4	5	6	7	8
Local Combinations			L1	L2	L1+L2	L3	L1+L3	L1+L2+L3	L4	L5
Pump Inlet Pressure	PT1	Bar Abs	0.90	0.90	0.90	0.91	0.91	0.90	0.90	0.91
Pump Outlet Pressure	PT2	Bar Abs	3.06	3.05	3.05	3.08	3.08	3.08	3.10	3.11
Turbine Stage 1 Pressure	PT3	Bar Abs	2.89	2.89	2.90	2.93	2.94	2.96	2.96	2.95
Turbine Stage 2 Pressure	PT4	Bar Abs	2.11	2.11	2.12	2.16	2.15	2.16	2.19	2.20
Water Flowrate	FT	Liter/min	344.1	357.7	348.0	344.8	352.0	353.8	356.9	346.9
Pump Speed	SP1	Rpm	2799	2798	2803	2804	2796	2796	2794	2799
Pump Torque	TQ1	Nm	10.0	10.0	10.1	10.0	10.0	10.0	10.0	10.0
Pump Power	PWR1	W	3393	3397	3401	3401	3393	3384	3375	3381
Turbine Speed	SP2	Rpm	2464	2317	2112	880	846	802	531	244
Turbine Torque	TQ2	Nm	0.5	0.6	0.8	1.4	1.4	1.4	1.5	1.6
Turbine Voltage	V1	V	127.9	118.8	107.5	39.8	37.9	35.6	21.3	5.6
Turbine Current	I1	A	0.33	0.62	0.83	2.20	2.20	2.23	2.48	2.77
Power Generated (W)			42.21	73.66	89.23	87.56	83.38	79.39	52.82	15.51
Turbine Input Power (W)			447.3	465.0	452.4	442.5	463.5	471.7	458.0	433.6
Turbine Efficiency (%)			9.44	15.84	19.72	19.79	17.99	16.83	11.53	3.58

Power Generated

$$P = V \times I$$
$$= 127.9\ V \times 0.33\ A$$
$$= 42.21\ W$$

Turbine Input Power

$P_{input} = P_{gage} \cdot Q$
$= (PT3 – PT4) \cdot Q$

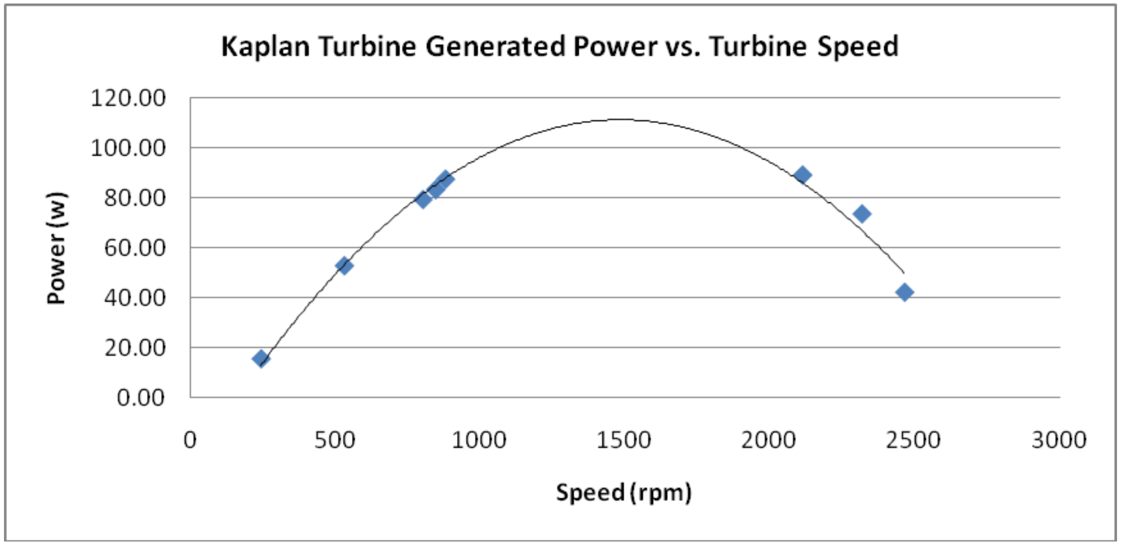
=
= 447.3 W

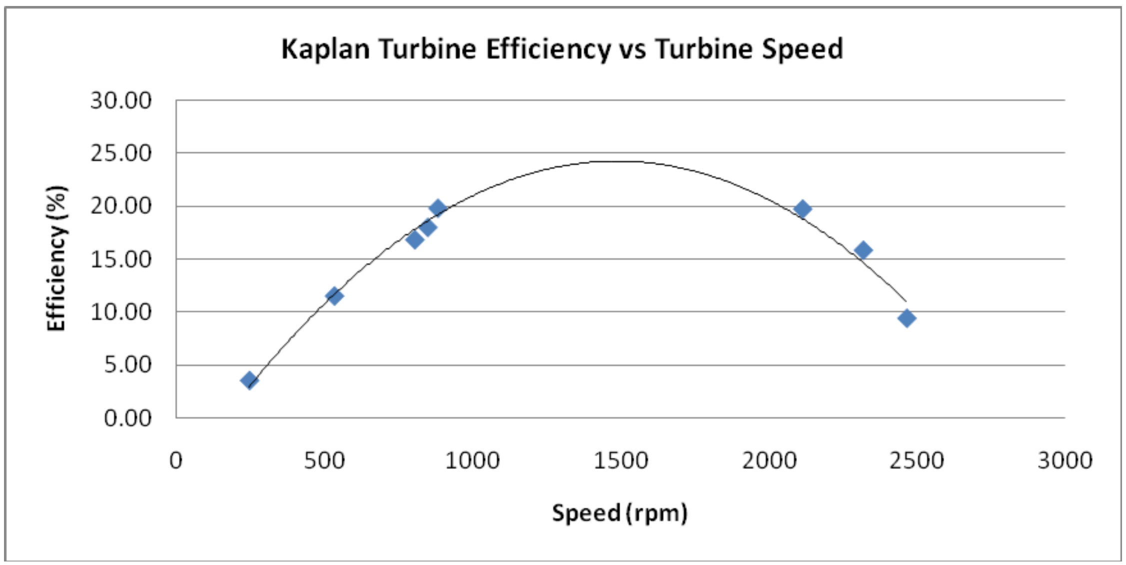
Turbine Efficiency

$$Eff_{turbine} = \frac{P_{generated}}{P_{input}}$$

$$= \frac{42.21\ W}{447.3\ W} \times 100\ \%$$

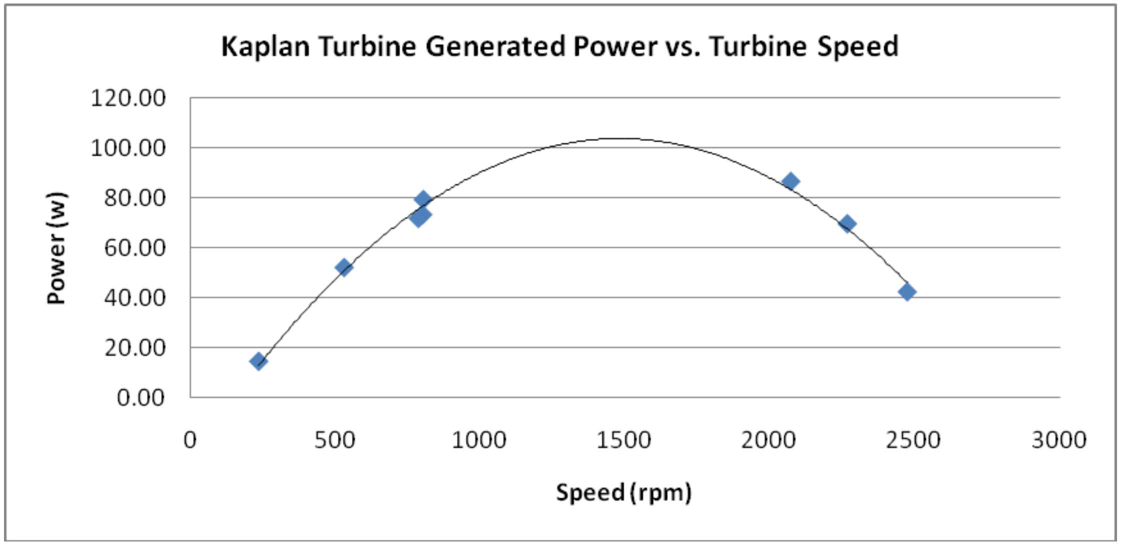
$$= 9.44\ \%$$

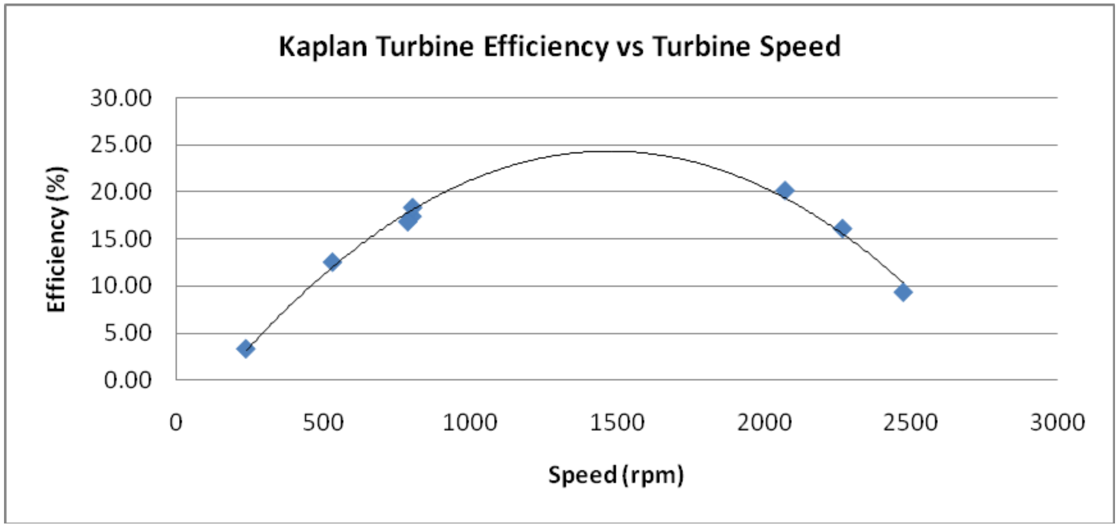

Kaplan Turbine Generated Power vs. Turbine Speed
120.00
100.00
80.00
60.00
40.00
20.00
0.00
Power (w)
0
500
1000
1500
2000
2500
3000
Speed (rpm)


Kaplan Turbine Efficiency vs Turbine Speed
30.00
25.00
20.00
15.00
10.00
5.00
0.00
Efficiency (%)
0
500
1000
1500
2000
2500
3000
Speed (rpm)

Blade angle: 20^0

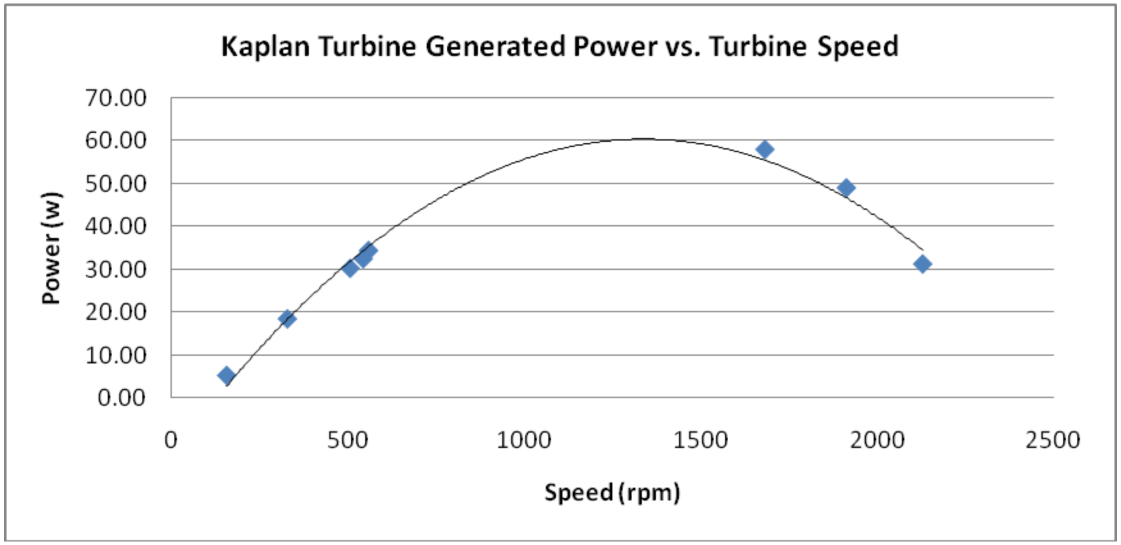
Experiment Data Set No:			1	2	3	4	5	6	7	8
Local Combinations			L1	L2	L1+L2	L3	L1+L3	L1+L2+L3	L4	L5
Pump Inlet Pressure	PT1	Bar Abs	0.91	0.91	0.91	0.91	0.91	0.91	0.91	0.91
Pump Outlet Pressure	PT2	Bar Abs	3.05	3.07	3.08	3.14	3.12	3.13	3.14	3.15
Turbine Stage 1 Pressure	PT3	Bar Abs	2.92	2.92	2.93	2.98	2.99	3.00	3.00	3.01
Turbine Stage 2 Pressure	PT4	Bar Abs	2.14	2.16	2.17	2.23	2.24	2.23	2.25	2.26
Water Flowrate	FT	Liter/min	346.7	340.5	338.9	336.2	340.6	336.4	331.2	341.6
Pump Speed	SP1	Rpm	2774	2772	2775	2773	2774	2773	2773	2774
Pump Torque	TQ1	Nm	9.9	9.9	9.9	9.8	9.8	9.8	9.7	9.7
Pump Power	PWR1	W	2776	3247	3237	3241	3223	3255	3230	3238
Turbine Speed	SP2	Rpm	2474	2267	2072	802	787	804	531	236
Turbine Torque	TQ2	Nm	0.5	0.6	0.7	1.3	1.3	1.4	1.4	1.5
Turbine Voltage	V1	V	128.4	116.1	105.7	36.3	35.4	35.6	21.1	5.4
Turbine Current	I1	A	0.33	0.60	0.82	2.02	2.03	2.23	2.47	2.68
Power Generated (W)			42.37	69.66	86.67	73.33	71.86	79.39	52.12	14.47
Turbine Input Power (W)			450.7	431.3	429.3	420.3	425.8	431.7	414.0	427.0
Turbine Efficiency (%)			9.40	16.15	20.19	17.45	16.88	18.39	12.59	3.39

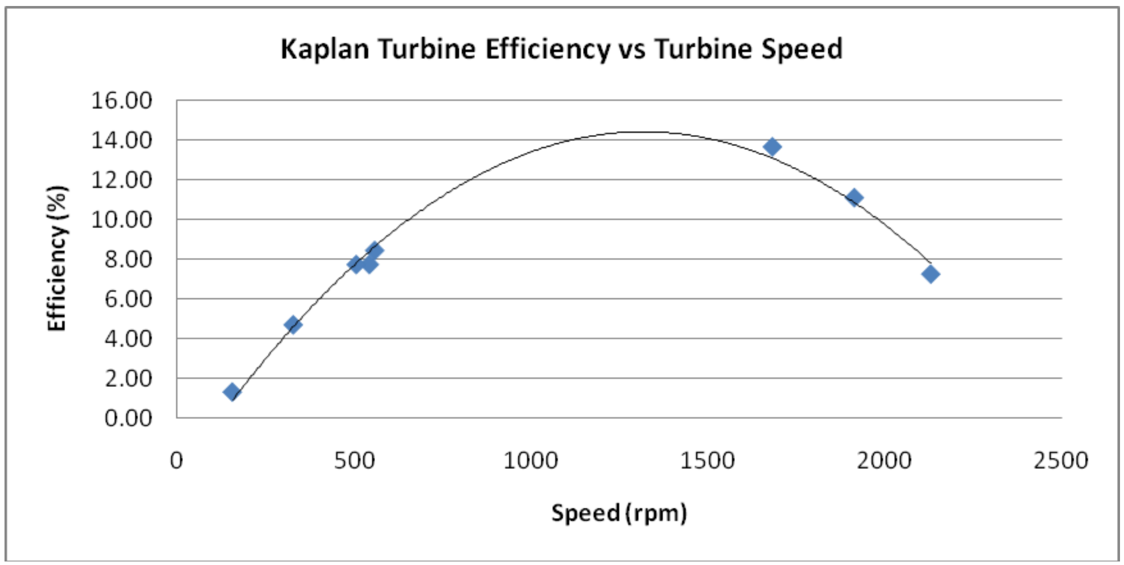

Kaplan Turbine Generated Power vs. Turbine Speed
Power (w)
120.00
100.00
80.00
60.00
40.00
20.00
0.00
0
500
1000
1500
2000
2500
3000
Speed (rpm)


Kaplan Turbine Efficiency vs Turbine Speed
Efficiency (%)
30.00
25.00
20.00
15.00
10.00
5.00
0.00
0
500
1000
1500
2000
2500
3000
Speed (rpm)

Blade angle: 10^0

Experiment Data Set No:			1	2	3	4	5	6	7	8
Local Combinations			L1	L2	L1+L2	L3	L1+L3	L1+L2+L3	L4	L5
Pump Inlet Pressure	PT1	Bar Abs	0.90	0.90	0.90	0.91	0.91	0.91	0.91	0.91
Pump Outlet Pressure	PT2	Bar Abs	3.28	3.28	3.29	3.35	3.35	3.36	3.37	3.40
Turbine Stage 1 Pressure	PT3	Bar Abs	3.13	3.14	3.15	3.22	3.23	3.21	3.23	3.26
Turbine Stage 2 Pressure	PT4	Bar Abs	2.39	2.40	2.42	2.50	2.51	2.52	2.54	2.55
Water Flowrate	FT	Liter/min	348.3	357.0	348.0	338.8	348.9	338.9	340.6	336.0
Pump Speed	SP1	Rpm	2867	2866	2866	2865	2865	2864	2864	2865
Pump Torque	TQ1	Nm	10.5	10.5	10.3	10.5	10.5	10.5	10.2	10.5
Pump Power	PWR1	W	3546	3550	3534	3543	3541	3532	3498	3515
Turbine Speed	SP2	Rpm	2128	1912	1681	558	543	506	328	156
Turbine Torque	TQ2	Nm	0.5	0.6	0.7	0.9	0.9	0.9	1.0	1.0
Turbine Voltage	V1	V	111.6	97.9	86.4	25.1	23.7	21.9	12.5	3.2
Turbine Current	I1	A	0.28	0.50	0.67	1.37	1.37	1.38	1.48	1.67
Power Generated (W)			31.25	48.95	57.89	34.39	32.47	30.22	18.50	5.34
Turbine Input Power (W)			429.6	440.3	423.4	406.6	418.7	389.7	391.7	397.6
Turbine Efficiency (%)			7.27	11.12	13.67	8.46	7.76	7.75	4.72	1.34


Kaplan Turbine Generated Power vs. Turbine Speed
Power (w)
70.00
60.00
50.00
40.00
30.00
20.00
10.00
0.00
0
500
1000
1500
2000
2500
Speed (rpm)


Kaplan Turbine Efficiency vs Turbine Speed
Efficiency (%)
16.00
14.00
12.00
10.00
8.00
6.00
4.00
2.00
0.00
0
500
1000
1500
2000
2500
Speed (rpm)

Experiment 4: Characteristic Study of a Centrifugal Pump

Experiment Data Set No:			1	2	3	4	5	6	7	8
Indicative Pump Speed (RPM)			2367.6	2268.7	2068.4	1965.3	1876.6	1768.7	1673.3	1563.9
Pump Inlet Pressure	PT1	Bar Abs	0.94	0.95	0.95	0.96	0.97	0.97	0.97	0.97
Pump Outlet Pressure	PT2	Bar Abs	3.20	3.01	2.64	2.47	2.33	2.18	2.05	1.90
Water Flowrate	FT	Liter/min	188.0	179.6	165.6	160.1	150.3	143.0	133.4	123.4
Pump Speed	SP1	Rpm	2368	2269	2068	1965	1877	1769	1673	1564
Pump Torque	TQ1	Nm	7.0	6.0	5.5	5.1	4.6	4.1	3.7	3.3
Pump Power	PWR1	W	2091	1912	1563	1362	1210	1052	925	832
Pump Output Power			1723.37	1418.46	1191.41	1053.86	898.20	759.49	648.43	543.79
Total Head			22.6	20.6	16.9	15.1	13.6	12.1	10.8	9.3
Overall Efficiency			82.42	74.19	76.23	77.38	74.23	72.20	70.10	65.36

Pump Output Power

$$P = \frac{2\Pi NT}{60}$$

$$= \frac{2 \times 3.142 \times 2368 \text{ rpm} \times 7.0 \text{ Nm}}{60}$$

$$= 1723.37 \text{ W}$$

Total Head

$$P_{total} = P_{outlet} - P_{inlet}$$

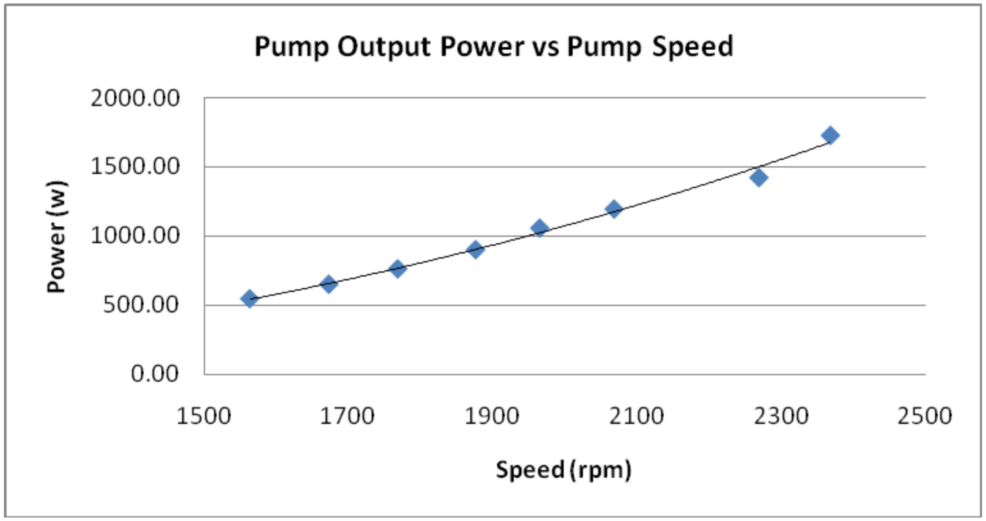
$$= 3.20 \text{ bar} - 0.94 \text{ bar}$$

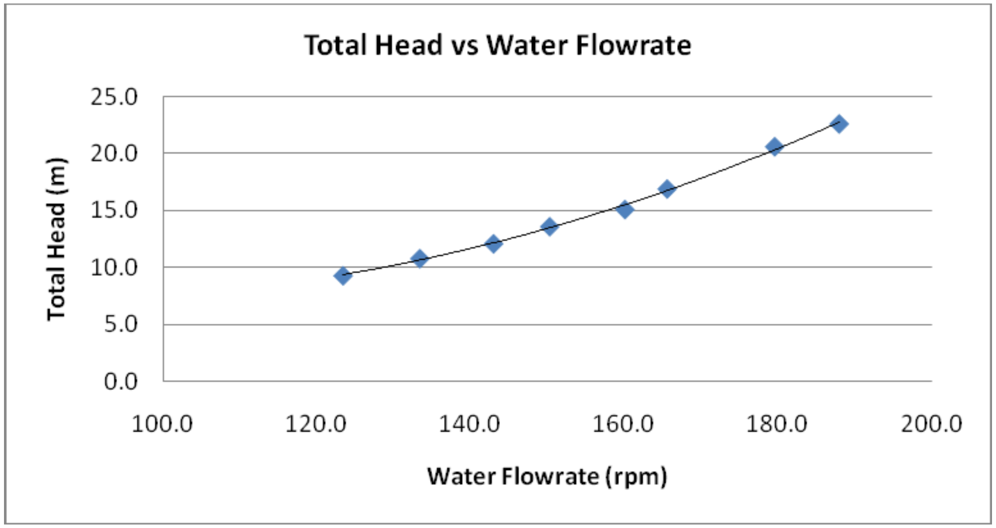
$$= 2.26 \text{ bar}$$

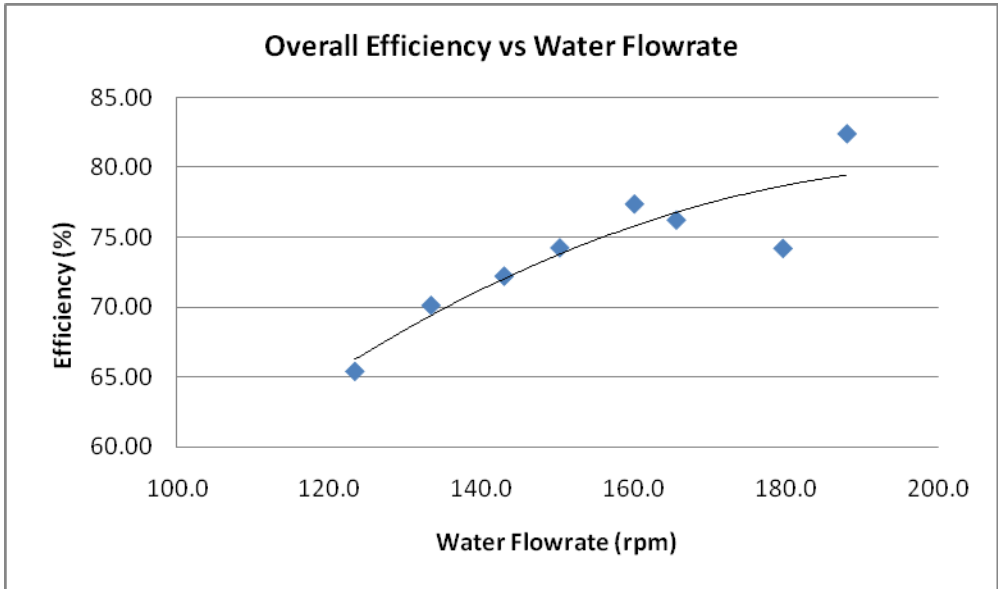
$$= 22.6 \text{ m}$$

Overall Efficiency

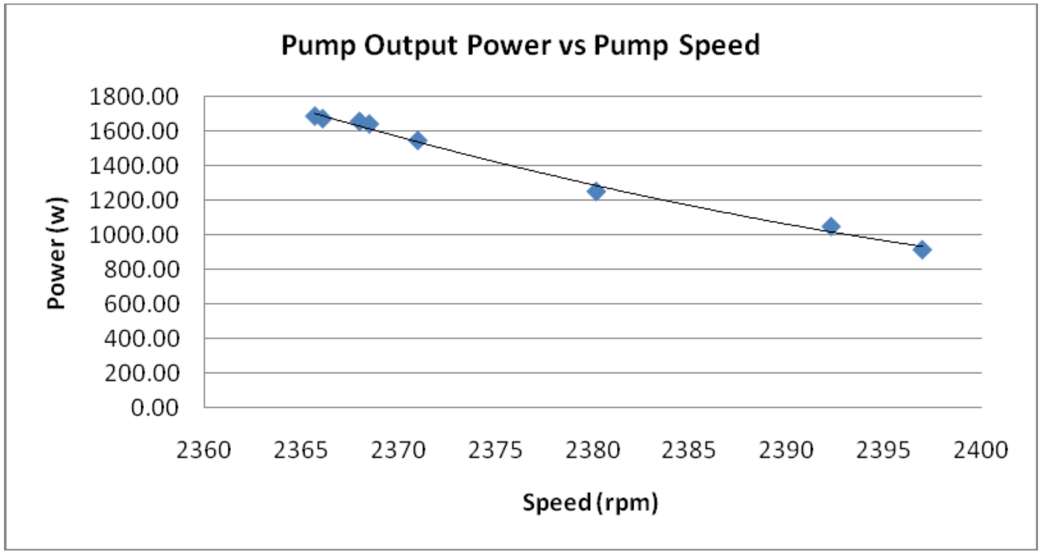
$$Eff_{overall} = \frac{P_{output}}{P_{input}} \times 100\% = \frac{1723.37 \text{ W}}{2091 \text{ W}} \times 100 \% = 82.42 \%$$


Pump Output Power vs Pump Speed
2000.00
1500.00
1000.00
500.00
0.00
Power (w)
1500
1700
1900
2100
2300
2500
Speed (rpm)

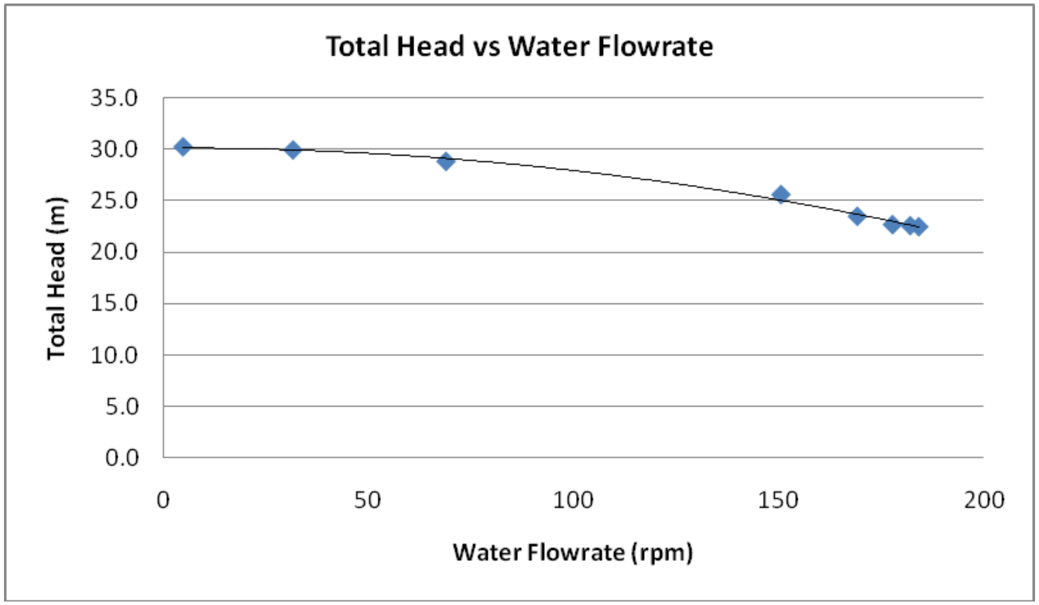

Total Head vs Water Flowrate
25.0
20.0
15.0
10.0
5.0
0.0
Total Head (m)
100.0
120.0
140.0
160.0
180.0
200.0
Water Flowrate (rpm)

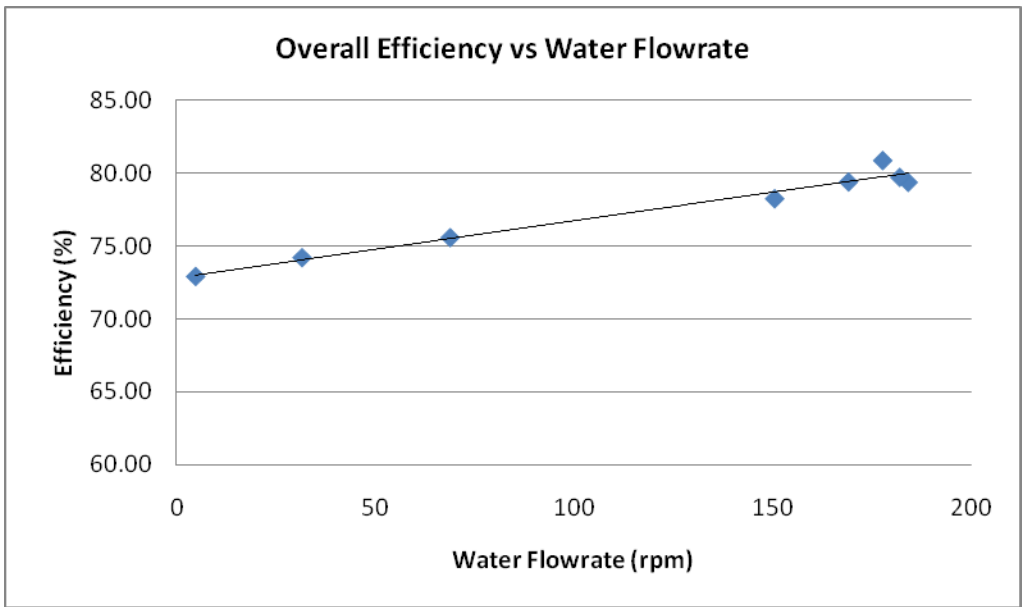

Overall Efficiency vs Water Flowrate
85.00
80.00
75.00
70.00
65.00
60.00
Efficiency (%)
100.0
120.0
140.0
160.0
180.0
200.0
Water Flowrate (rpm)

Experiment Data Set No:			1	2	3	4	5	6	7	8
Indicative Valve Opening (no. of turns)			0	0.5	1	2	3	4	4.5	5
Pump Inlet Pressure	PT1	Bar Abs	0.95	0.95	0.95	0.95	0.96	0.98	0.98	0.99
Pump Outlet Pressure	PT2	Bar Abs	3.20	3.21	3.22	3.30	3.52	3.86	3.97	4.01
Water Flowrate	FT	Liter/min	184.2	182.1	177.8	169.2	150.6	68.9	31.6	4.8
Pump Speed	SP1	Rpm	2368	2366	2366	2369	2371	2380	2392	2397
Pump Torque	TQ1	Nm	6.7	6.7	6.8	6.6	6.2	5.0	4.2	3.6
Pump Power	PWR1	W	2081	2092	2080	2062	1971	1653	1412	1254
Pump Output Power			1651.74	1667.76	1682.34	1637.20	1542.08	1248.92	1047.31	913.81
Total Head			22.5	22.6	22.7	23.5	25.6	28.8	29.9	30.2
Overall Efficiency			79.37	79.72	80.88	79.40	78.24	75.56	74.17	72.87



Pump Output Power vs Pump Speed
Power (w)
1800.00
1600.00
1400.00
1200.00
1000.00
800.00
600.00
400.00
200.00
0.00
2360
2365
2370
2375
2380
2385
2390
2395
2400
Speed (rpm)


Total Head vs Water Flowrate
35.0
30.0
25.0
20.0
15.0
10.0
5.0
0.0
Total Head (m)
0
50
100
150
200
Water Flowrate (rpm)


Overall Efficiency vs Water Flowrate
85.00
80.00
75.00
70.00
65.00
60.00
Efficiency (%)
0
50
100
150
200
Water Flowrate (rpm)

Appendix C
Turbine Installation Guide

Pelton Turbine Installation Guide:

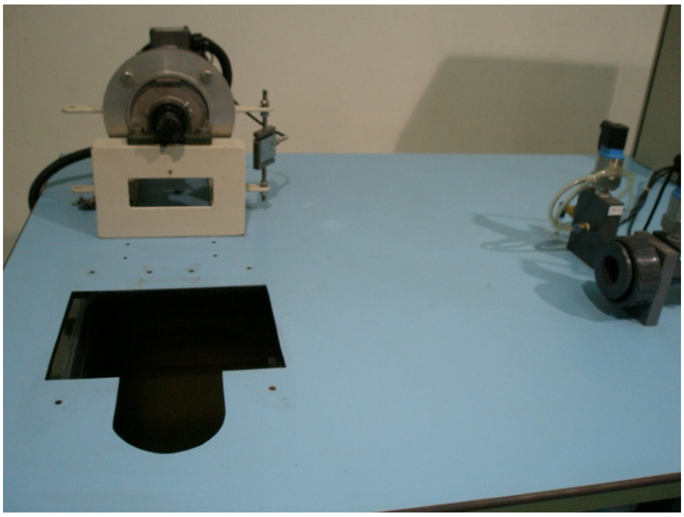


Figure 1

1. The bench shall be ready as shown at Figure 1.

Figure 2

2. Check the O-ring is placed as shown at Figure 2.

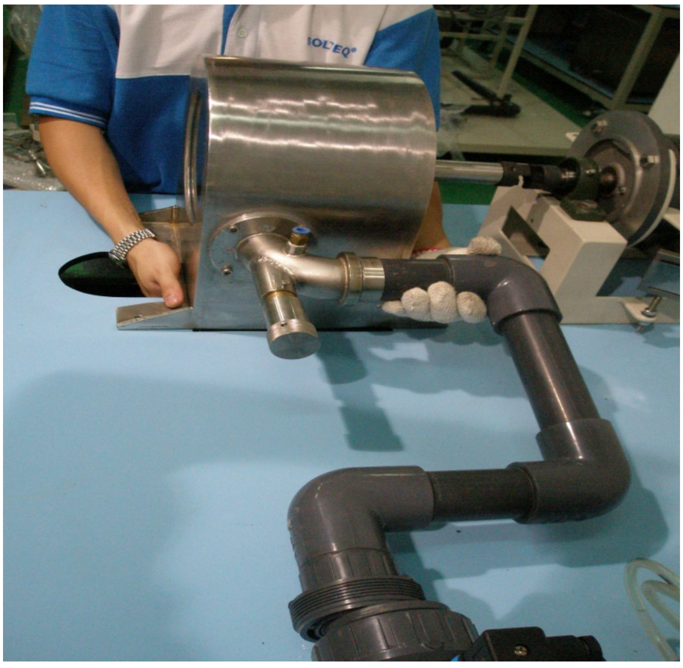


Figure 3

3. Place the turbine housing as shown in the Figure 3.

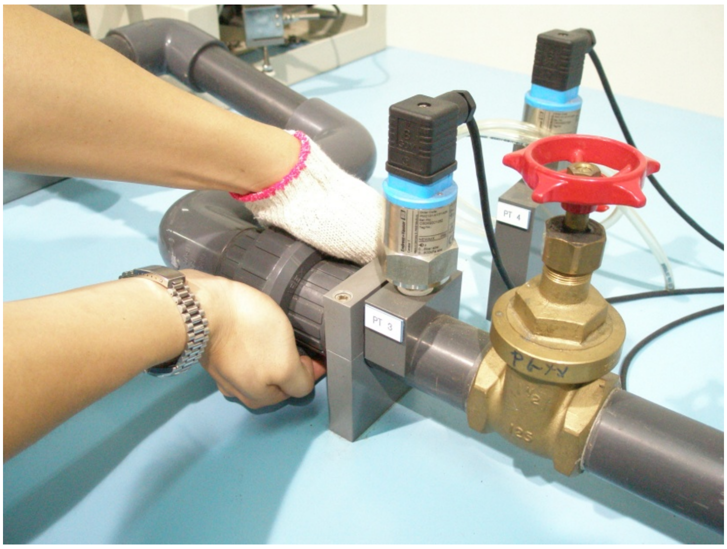


Figure 4

4. Place the union but not to tighten it. (Check the alignment of the shaft coupling to the DC motor before tighten it)

Figure 5

5. Tighten the turbine housing to the bench.

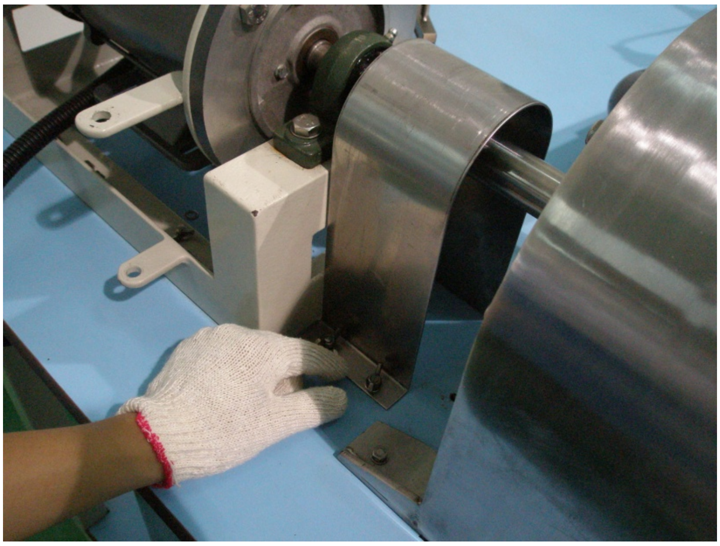


Figure 6

6. Place the mechanical guard to cover the coupling.

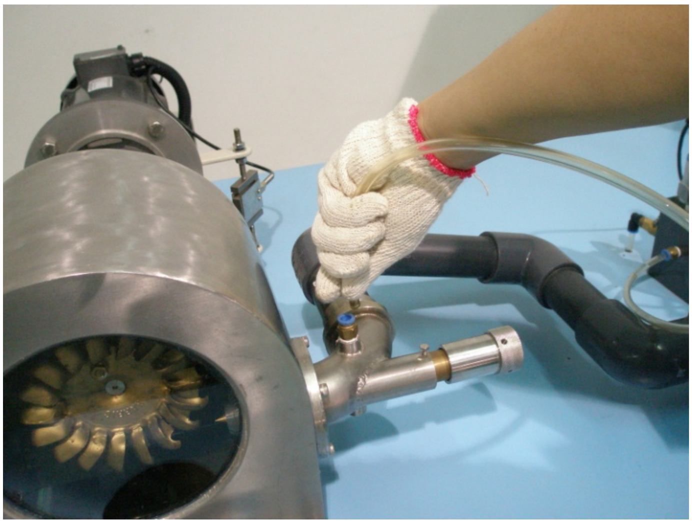


Figure 7

7. Lastly, plug the tube to the pneumatic plug and the Pelton Turbine is ready for testing.

Uninstall Pelton Turbine:

1. Remove the mechanical guard cover.
2. Unplug the tubing.
3. Loosen the union.
4. Remove the turbine housing with care.

(Not to hold the PVC pipe as this will misalign the shafts to the DC motor and union connection.)

Turbine Housing Installation Guide:

1. Place the housing on the bench as shown in Figure 8. Make sure the O-ring is placed in between the PVC union. (Note: NOT to hold the PVC pipe and housing bracket as this will misalign the shaft to the DC motor and PVC union coneection)

Figure 8

2. Rotate clockwise to fix the union position. (Do not fasten the union before we locate all screws and nuts)

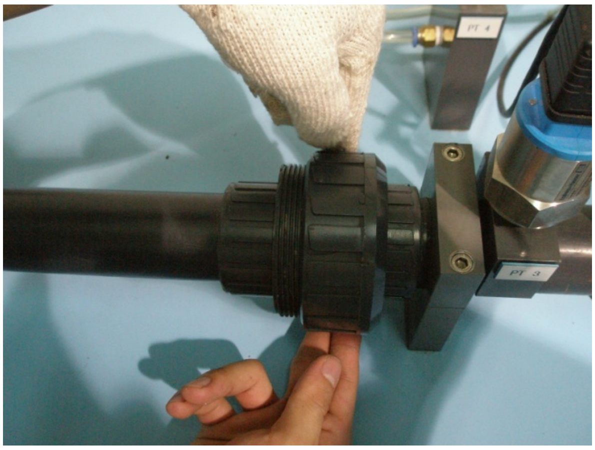


Figure 9

3. Locate the ring nut as shown in Figure 10.

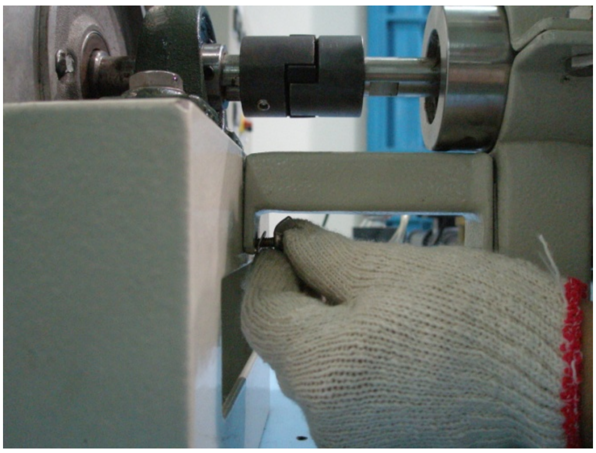


Figure 10

Figure 11

4. Locate and fasten the screws and nuts as shown in Figure 11. After that, we can fasten the union and the ring nuts as in Figure 9 and Figure 10.

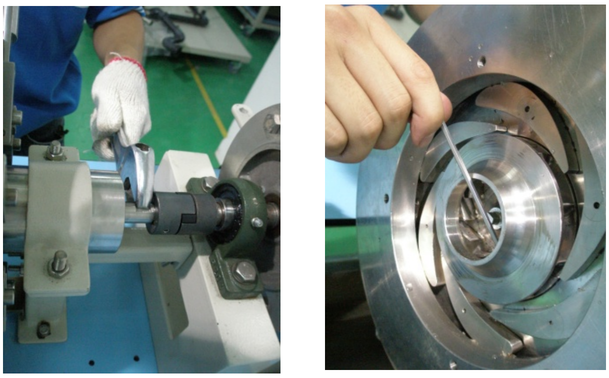


Figure 12

Francis Turbine Installation Guide:

1. Locate the Francis Turbine to the housing and turn clockwise to fasten the turbine as shown in Figure 12.

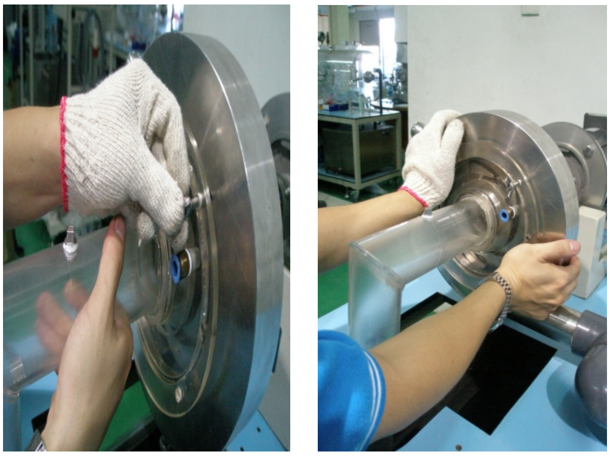
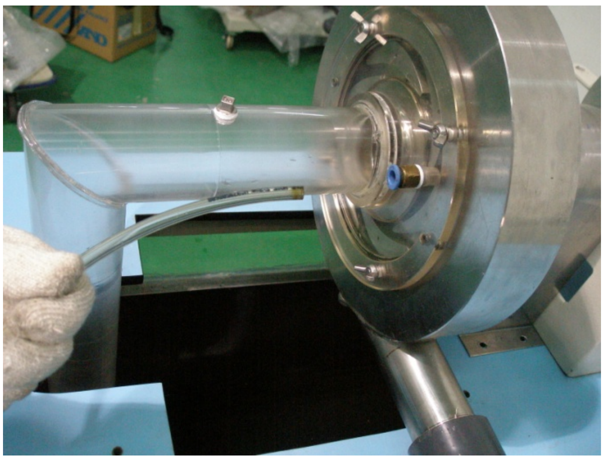


Figure 13

2. Locate the water outlet (smaller diameter) to the housing. Apply some water on the O-ring so that we can install it easier.
3. Then, locate 2 ring nuts (longer) to either 2 holes (cross each other). (They are used to drive the acrylic when we push in the acrylic simultaneously as shown in Figure 13)



4. Remove the 2 long ring nuts and substitute with shorter ring nuts. Fasten the ring nuts cross each other.
5. Lastly, plug in the tube into the pneumatic plug.
6. The Francis Turbine is ready for experiments.

Figure 14

Uninstall Francis Turbine:

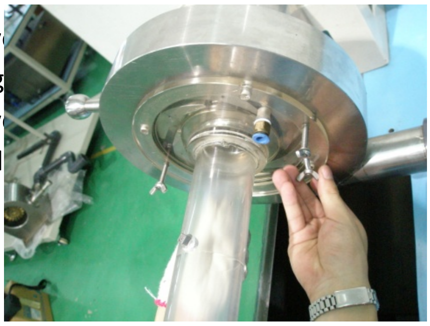


Figure 15

1. Remov
2. Unplug
3. Remov
4. Fix and simultaneouly to the 2 holes which are used to pull out the acrlyic as shown in Figure 15. Failure of doing that will cause acrylic to break.
5. Remove the Francis turbine by using adjustable spanar.

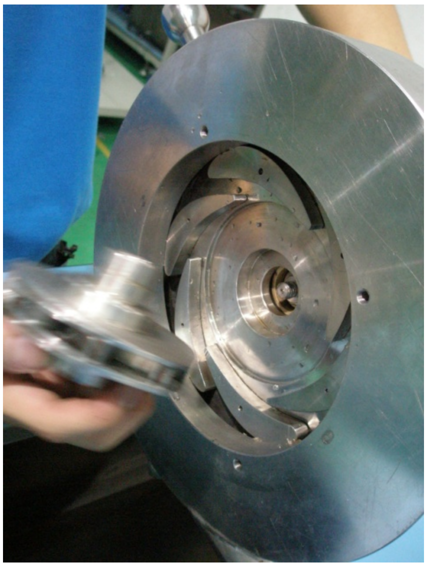


Figure 16

Kaplan Turbine Installation Guide:

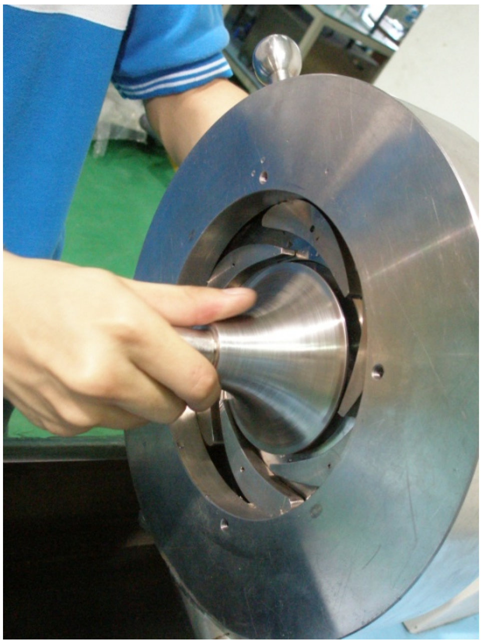


Figure 17

1. Install the Kaplan as shown in Figure 17.

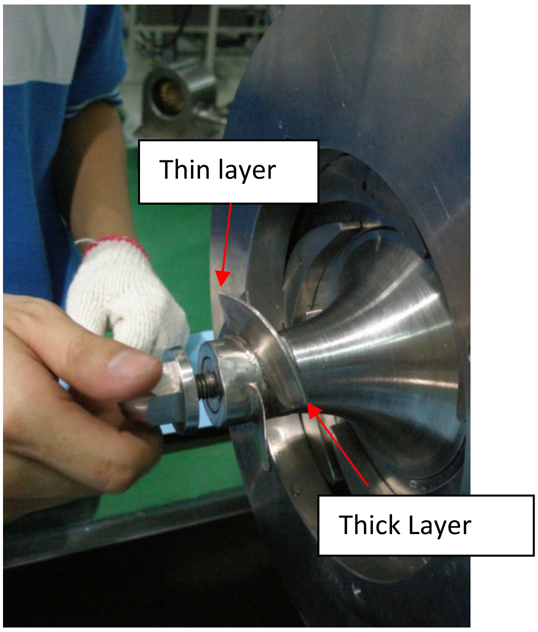
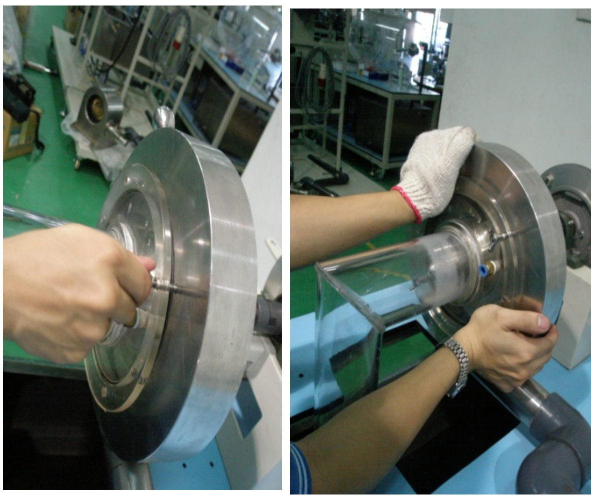


Figure 18

2. Install the blade as shown in Figure 18.



3. Fix 2 ring nuts (longer one) at either 2 holes (cross each other). Then push the acrylic towards the housing as shown ni Figure 19. Remove the 2 ring nuts after the acrylic fixed.

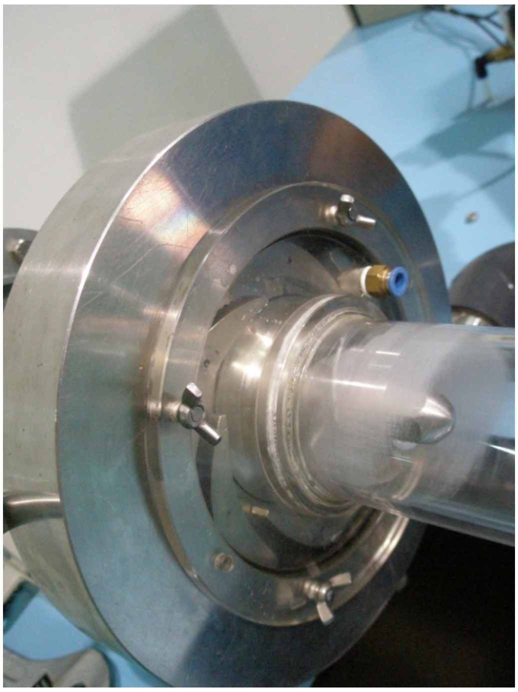


Figure 20

4. Fasten four ring nuts (shorter one) as shown in Figure 20. Plug in the tube into the pneumatic plug.
5. The Kaplan turbine is ready for experiments.

Uninstall Kaplan Turbine

1. Remove the mechanical guard.
2. Remove the pneumatic tube.
3. Remove the ring nuts.
4. Locate and fix 2 ring nuts simultaneo to pull out acrylic as shown in Figure 21.
5. Remove/ Install the Kaplan turbine w blade angle.

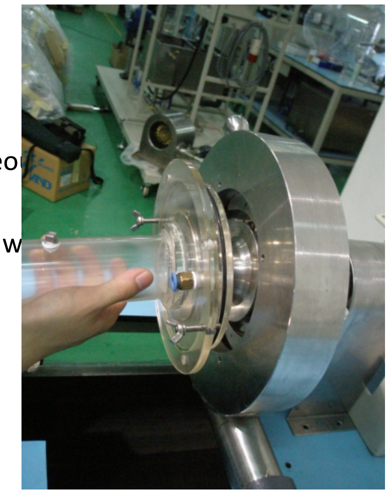


Figure 21

Uninstall Turbine Housing:

1. Remo
2. Remo
3. Remo

   the housing.
4. Loose
5. Loose
6. Loose

   23.
7. Unins

   24.

   (Please do NOT hold turbine shaft, housing bracket

   And PVC pipe as this will misalign the connections.)

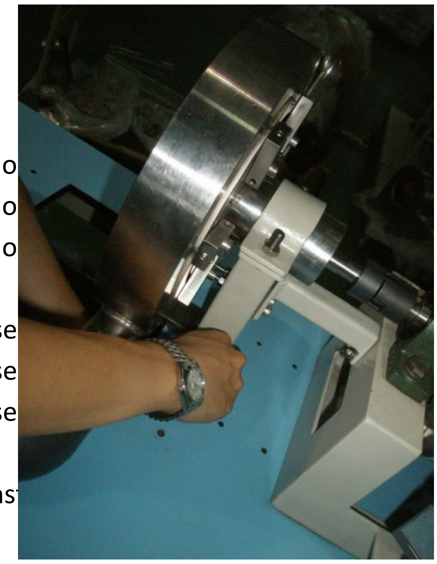


Figure 22

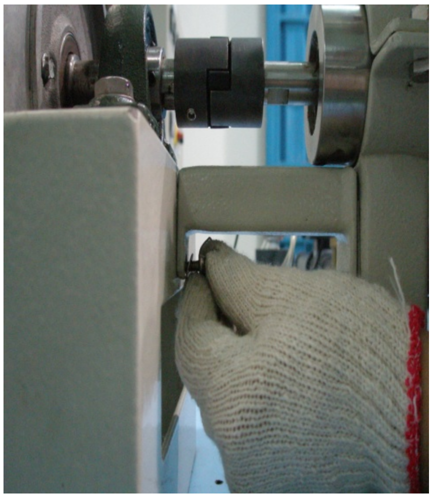


Figure 23

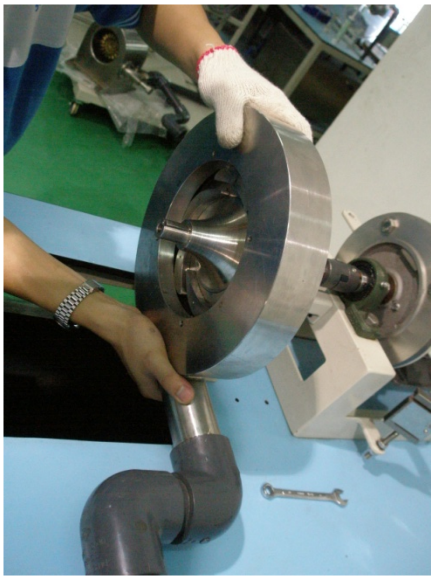


Figure 24