**NATIONAL EXAMINATIONS**

**December 2013**

# 07-MEC-A6-1 FLUID MACHINERY

**Three hours duration**

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## Notes to Candidates

1. This is a **Closed Book** examination.

2. Exam consists of two Sections **Section A is Calculative (5 questions)** and **Section B is Descriptive (3 questions).**

3. **Do four (4) questions (including all parts of each question) from Section A (Calculative) and two (2) questions from Section B (Descriptive).** Note that Question 1 is on <u>two</u> pages,

4. **Six questions constitute a complete paper.** (Total 60 marks).

5. **All questions are of equal value.** (Each 10 marks).

6. Descriptive questions require comprehensive answers with complete explanations and sketches, if appropriate, to support the explanation. See note on Page 8

7. If doubt exists as to the interpretation of any question, the candidate is urged to submit, with the answer paper, a clear statement of any assumptions made.

8. If any initial parts of a multi-part question cannot be solved the remaining parts may be worked by making appropriate assumptions for the first parts from the technical data given.

9. Candidates may use one of the approved **Casio** or **Sharp** calculators.

10. **Reference data** for particular questions are given in the Attachments on pages 11 to 17. **All pages on which answers have been written are to be returned with the answer booklet. Candidate's names must be on these sheets.**

11. **Reference formulae and constants** are given on pages 18 to 22.

12. **Drawing Instruments** (scale ruler, protractor and sharp pencil) are required for vector diagrams.

## SECTION A CALCULATIVE QUESTIONS

### QUESTION 1 HYDRO TURBINES

PART I HYDRO TURBINE PLANT

Refer to the Examination Paper Attachments Page 11 **Hydro Power Plant**.

The figure shows a cross section of a small hydro power plant utilizing a Kaplan turbine in a low head application. The maximum power output of the turbine generator is 4 MW. Assuming that the turbine efficiency is 100% and that there is negligible hydraulic friction in the system:

(a) Determine the flow rate of water when operating at maximum power between normal low water level in the reservoir and full load water level in the tail race.

(b) Determine the velocity of the water in the penstock when operating under the conditions in (a) above. Note that the penstock is circular in cross section and its diameter must be estimated from the information given on the drawing.

(c) Should normal high water level in the reservoir be reached with that in the tailrace unchanged, state with reasons whether the velocity in the penstock will be greater or less than that given in (b) to achieve full power.

(d) Determine the velocity of the water in the penstock when operating under the conditions in (c) above.

*(5 marks)*

***This question is continued on the next page***

## PART II KAPLAN TURBINE EFFICIENCY

Refer to the Examination Paper Attachments Page 12 **Kaplan Turbine** for illustration only.

Hydro turbines of the Kaplan type are installed at Mactaquac on the Saint John River. In order to determine the efficiency of the Mactaquac turbines the following hypothetical measurements are considered:

| | |
|---|---|
| Turbine-generator speed | 112.5 rev/min |
| Generator electrical output | 110 MW |
| Water flow rate | 354 $m^3/s$ |
| Inlet pipe diameter (not in picture) | 6.4 m |
| Outlet pipe diameter (bottom of picture) | 7.0 m |
| Inlet water pressure | 226 kPa gauge |
| Outlet water pressure | -4.5 m $H_2O$ |

The elevation of the outlet pressure measuring point is 5.0 m below that of the inlet pressure measuring point.

Determine the following:

(a) Hydraulic power produced by the water (input to turbine-generator).

(b) Electrical power output.

(c) Efficiency of turbine-generator

*( 5 marks )*

*[10 marks ]*

## QUESTION 2 HYDRO TURBINE MODEL

Technical specifications for the hydro turbines at Vanderkloof Hydro Power Station are as follows:

| | |
|---|---|
| Generator design output | 120 MW (at 0.90 power factor lagging) |
| Speed of machine | 125 rev/min |
| Electrical frequency | 50 Hz |
| Generator voltage | 11 kV |
| Design net head | 65 m |
| Design water flow | 200 $m^3/s$ |
| Maximum water flow | 213 $m^3/s$ |
| Turbine runner diameter | 5.462 m |
| Turbine runner material | stainless cast steel |

(a) Calculate the specific speed of the machine and determine the type of turbine.

(b) Calculate the overall efficiency of the turbine based on the design parameters.

In practice, the actual efficiency is not measured directly on site due to the difficulty and cost in measuring large volume flows with sufficient accuracy. A model test is therefore carried out at the manufacturer's works to prove the performance of the prototype machine. Assume that an homologous (scaled to be geometrically identical) model runner 200 mm in diameter is a available and can be tested in an instrumented hydraulic system under a head of 10 m. Use the turbine affinity laws or similarity rules to answer the following:

(c) Determine the speed at which the model should run.

(d) Determine the necessary flow through the model.

(e) Determine the ideal (no friction) power developed by the model.

The efficiency of the model and the prototype are not identical due to different hydraulic friction values arising due to scaling. The Moody equation allows the hydraulic efficiencies of the model and prototype to be compared.

(f) Assuming that the prototype has an electrical efficiency of 98%, determine the efficiency that should be measured on the model to ensure that the prototype will meet its specified efficiency.

*[ 10 marks ]*

## QUESTION 3 PUMP PERFORMANCE

Refer to the Examination Paper Attachments Page 13 **Pump Velocity Diagram**

The attached diagram clarifies the nomenclature to be used in answering the question.

The picture below the velocity diagram shows a pump impeller of a radial flow centrifugal pump for pumping water. The key dimensions are as follows:

| | |
|---|---|
| Blade inner diameter | $D_1$ = 130 mm |
| Blade outer diameter | $D_2$ = 300 mm |
| Blade inner height | $h_1$ = 20 mm (in axial direction) |
| Blade outer height | $h_2$ = 10 mm (in axial direction) |
| Blade inlet angle | $\beta_1$ = 20° |
| Blade outlet angle | $\beta_2$ = 25° |
| Pump speed | N = 1750 rev/min |
| Water flow rate | Q = 0.030 $m^3/s$ |
| Hydraulic head | H = 35 m |

For the given speed and a flow rate draw to a scale the velocity diagrams at inlet and outlet and determine the following neglecting the vane thickness:

(a) Tangential blade velocities at inlet and outlet

(b) Radial water velocity at inlet and outlet

(c) Tangential water velocity at inlet and outlet

(d) Torque and power required to drive the impeller

(e) Hydraulic power and efficiency of pump

*Note: The scale drawing should be to a large enough scale for accurate measurements (a scale of 10 mm = 2 m/s is suggested). While calculation of velocities by trigonometric ratios is acceptable it is longer and more time consuming.*

*[10 marks ]*

## QUESTION 4 CURTIS TYPE IMPULSE TURBINE

Refer to the Examination Paper Attachments Page 14 **Steam Turbine Velocity Diagram** (one stage only) for nomenclature of velocities and angles

Steam exits the nozzles and enters the first stage moving blades of a velocity compounded two stage (Curtis) impulse turbine at 1411 m/s. The nozzle angle is 20° and the fixed blade exit angle is the same as its inlet angle, that is, the fixed blades are symmetrical. The moving blades of both stages are also symmetrical but with different angles. Assume zero fluid friction in nozzles and blades.

(a) Determine a blade velocity to give optimum work (minimum exit kinetic energy).

(b) Draw to scale the velocity diagrams for the two stages.

(c) Determine all the actual and relative steam velocities and blade angles and show them on the diagrams.

(d) Calculate the work done by each stage, in kJ/kg of steam.

(e) Calculate the total power output for a steam flow of 100 kg/s.

(f) Calculate the blade efficiency

*Note: The scale drawing should be to a large enough scale for accurate measurements (a scale of 1 mm = 10 m/s is suggested). While calculation of velocities by trigonometric ratios is acceptable it is longer and more time consuming.*

*[ 10 marks ]*

## QUESTION 5 GAS TURBINE BLADES

Refer to the Examination Paper Attachments Page 15 **Acacia and Port Rex Power Stations** and Page 16 **Gas Turbine Velocity Diagram**.

Each power station has three units. Each unit has a nominal output of 60 MW and is powered by twin back to back gas turbines driving a common electrical generator. The diagram on Page 13 is for one gas turbine only while the specifications are for both gas turbines combined. The net power output and exhaust gas flow rate as given must therefore be divided by two for the purposes of this question.

Consider conditions on the power turbine (free turbine with N3 rotor) at peak load.

The power turbine has the following approximate blade dimensions for the first stage:

| | | | |
|---|---|---|---|
| Stator blade inlet angle $\alpha_0$ | 30° | Blade tip diameter | 1500 mm |
| Stator blade exit angle $\alpha_1$ | 60° | Blade root (hub) diameter | 1050 mm |
| Rotor blade inlet angle $\beta_1$ | 30° | | |
| Rotor blade exit angle $\beta_2$ | 60° | | |

The exhaust gas has parameters slightly different from those of cold air, so use the following values for its specific heat:

$c_p$ = 1.148 kJ/kg°C $\qquad$ $c_v$ = 0.861 kJ/kg°C

Based on these dimensions and conditions for the first stage and, assuming that the gas flow conditions are the same for the second and third stages of the power turbine, determine the following:

(a) Mean blade velocity (velocity at mid-height of blades).

(b) The velocity diagram for the first stage of the turbine drawn to scale as recommended below.

(c) Relative and absolute gas velocities from the velocity diagram

(d) Power turbine output based on gas velocities and gas mass flow rate.

(e) Power turbine output based on actual temperature change and gas flow rate.

(f) Difference between answers to (d) and (e) above and specified value and comment on any discrepancy.

*Note: The scale drawing should be to a large enough scale for accurate measurements (a scale of 10 m/s = 4 mm is suggested). While calculation of velocities by trigonometric ratios is acceptable it is longer and more time consuming.*

*[ 10 marks ]*

## SECTION B DESCRIPTIVE QUESTIONS

***Note that each five mark part of each question requires a full page answer with complete explanations with sketches or diagrams, if appropriate, to support the explanation. A full page means approximately 250 words unless diagrams take the place of some words.***

***While each part of each question specifies several aspects, more emphasis may be put on one or more aspects and less on others provided an overall comprehensive answer is given as required by the above.***

### QUESTION 6 PUMP AND TURBINE SPECIFIC SPEED

PART I PUMP SPECIFIC SPEED

Explain how specific speed is used to identify the type of pump to be installed for a particular application. Show by means of sketches how impeller shape varies with specific speed. Clarify the direction of flow through the impeller and how this may change with specific speed. Explain how specific speed is related to the head and flow of the pump.

*( 5 marks )*

PART II TURBINE SPECIFIC SPEED

Explain how specific speed is used to identify the type of hydraulic turbine to be installed for a particular application. Show by means of sketches how impeller shape varies with specific speed. Explain how specific speed is related to the head and flow of the turbine. Give the common names of hydraulic turbines and match these with sketches of impeller shape.

*( 5 marks )*

*[ 10 marks ]*

## QUESTION 7 COMPRESSOR AND PUMP CHARACTERISTICS

### PART II STALLING IN COMPRESSORS

Describe stalling in an axial flow compressor as used in a typical gas turbine. Clarify under what conditions stalling can occur. Explain how the phenomenon of stalling affects the design of the compressor especially with regard to the number of stages required.

*( 5 marks )*

### PART I CENTRIFUGAL PUMPS

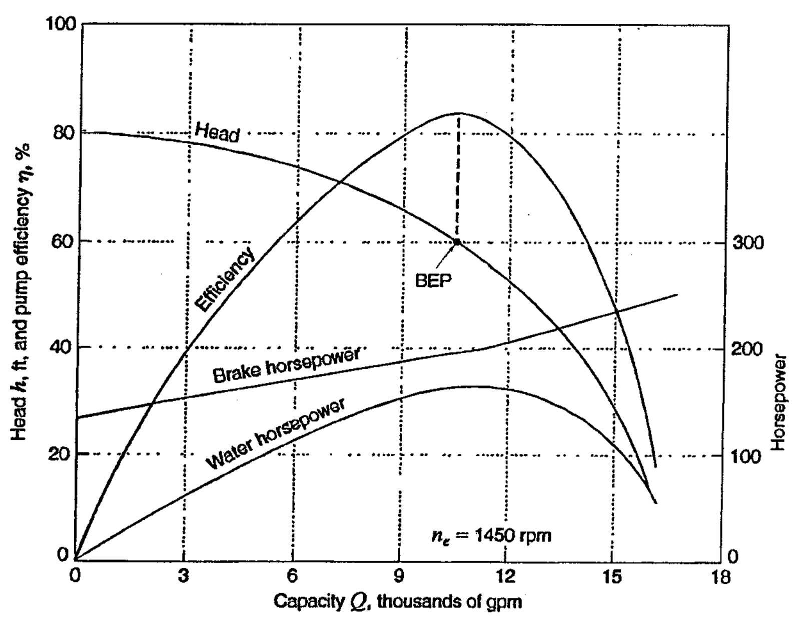
With reference to the figure below explain the following:

(a) Why the hydraulic efficiency (water horsepower) rises from zero to a peak and then declines towards zero

(b) Why the difference between the hydraulic power (water horsepower) and the mechanical power (brake horsepower) decreases to a low value and then increases to a value greater than the initial value.

*( 5 marks )*

*[ 10 marks ]*




**Characteristic curves for a typical mixed-flow centrifugal pump.**

## QUESTION 8 FAN BLADE SHAPE

Refer to the Examination Paper Attachments Page 17 **Fan Characteristics**.

***This page is to be returned with the answer booklet. Write your name on it.***

Radial flow centrifugal fans may have blades that are radial or curved forwards or backwards. Explain what effect the shape of the blades has on the performance of the fan.

(a) Show graphically in sketches on Page 17 how the velocity diagram changes and how the head versus flow curve is different for the three conditions.

(b) Explain the advantages of forward or backward curved blades with respect to the other. Hence clarify with reasons which configuration is more common.

*[ 10 marks ]*

## QUESTION 1 PART I HYDRO POWER PLANT

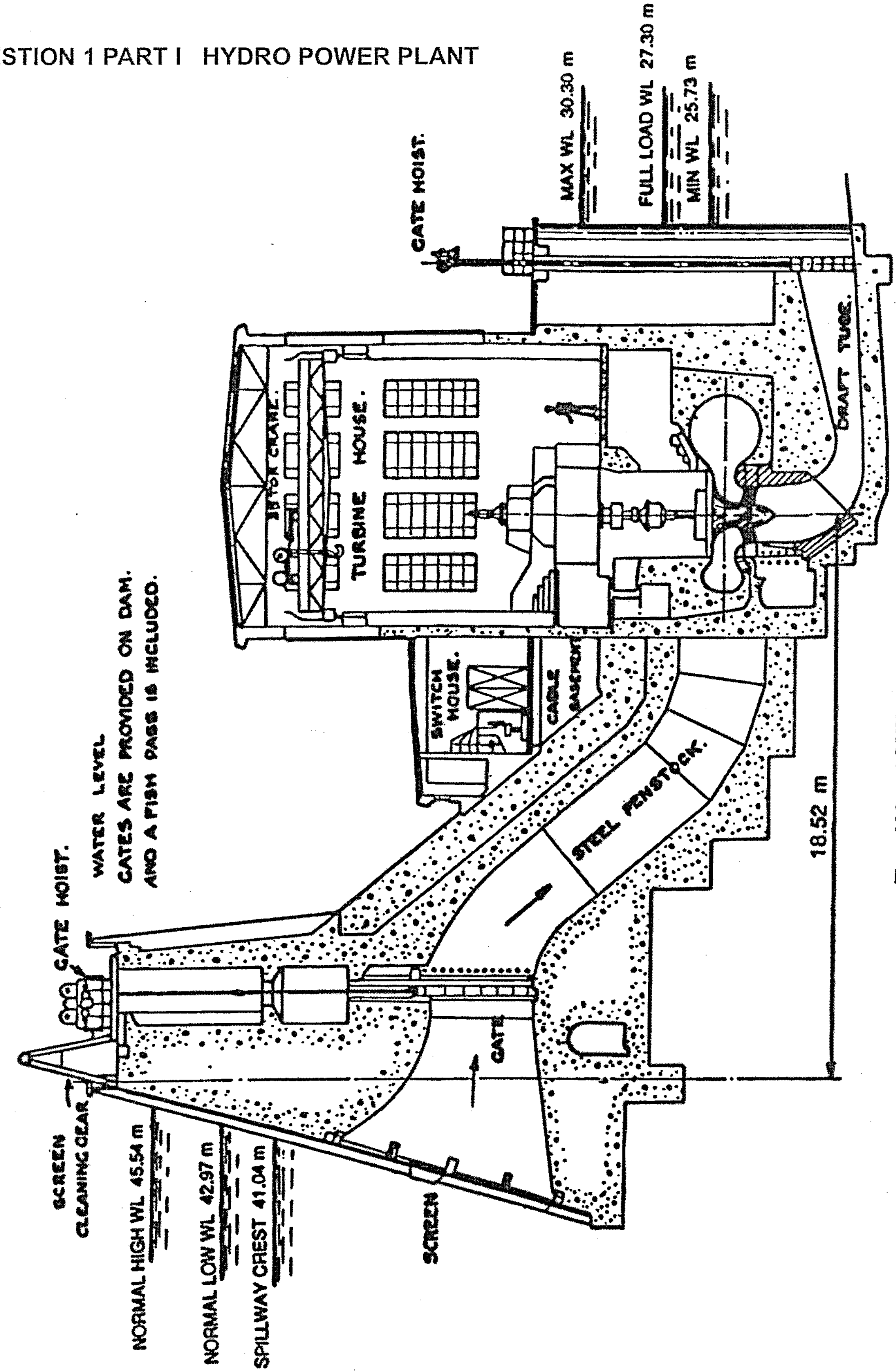



Fig. 602. 4 MW Kaplan Turbine Setting.

## QUESTION 1 II KAPLAN TURBINE

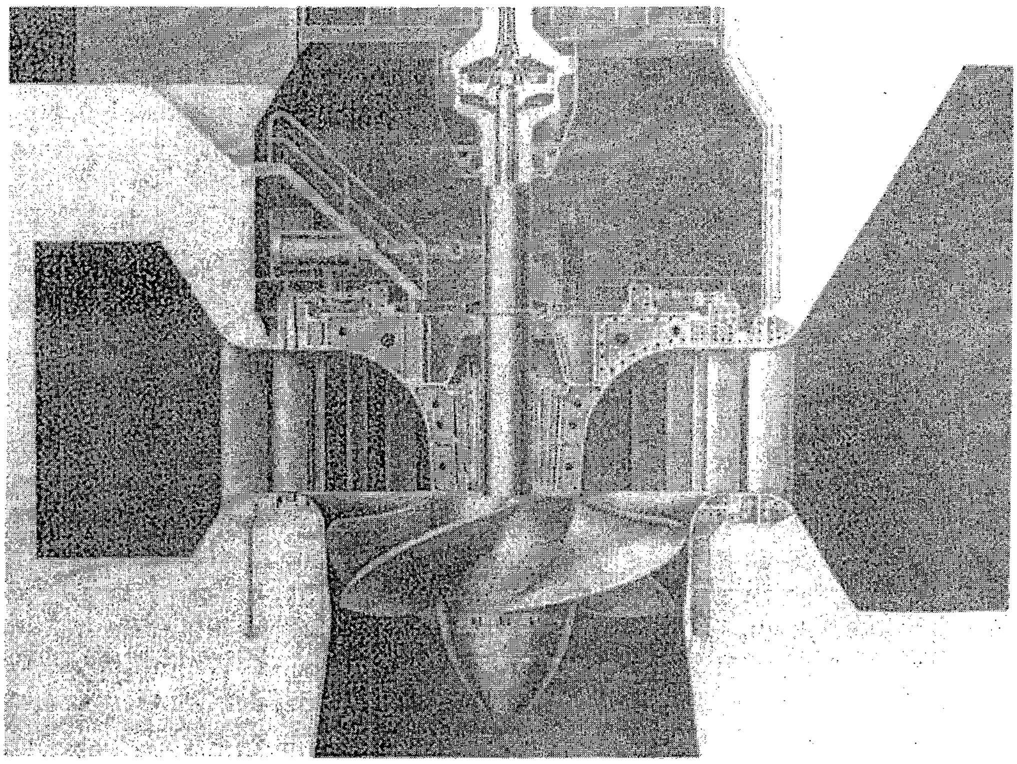
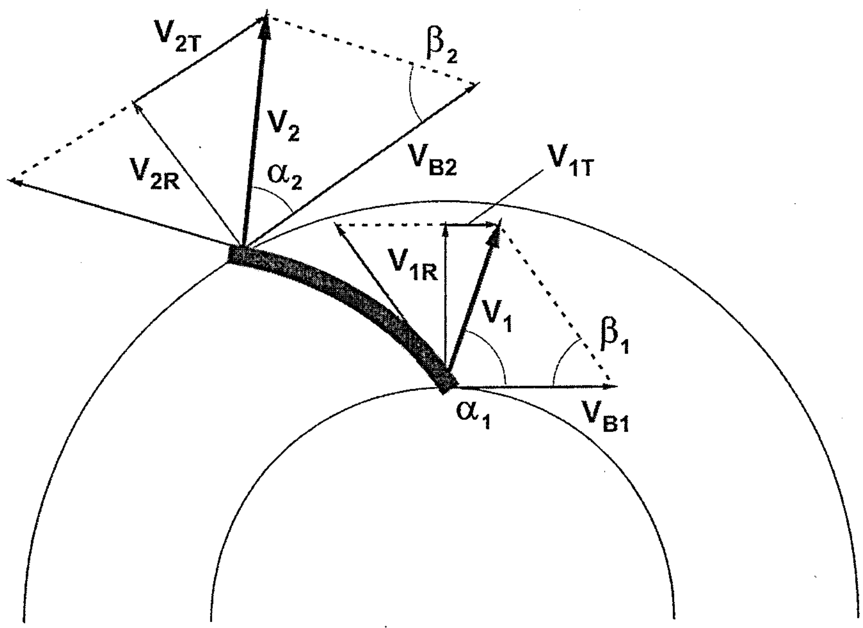
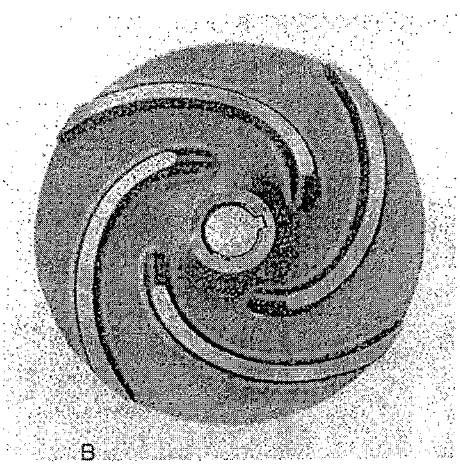


**Figure 16.7** Kaplan turbine at Watts Bar Dam. 42,000 hp at 94.7 rpm under a head of 52 ft.

## QUESTION 3 PUMP VELOCITY DIAGRAM

Nomenclature for velocity vectors and angles

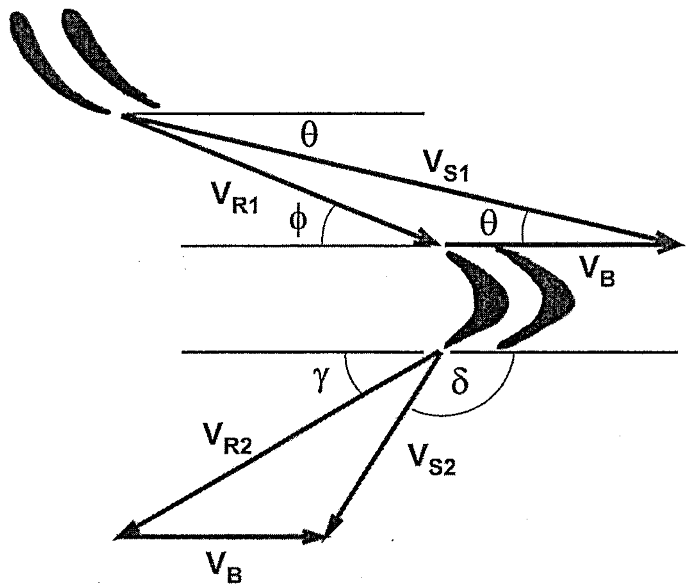



| | |
|---|---|
| $V_1$ | Absolute water velocity at inlet |
| $V_{B1}$ | Blade velocity at inlet |
| $V_{1R}$ | Radial water velocity at inlet |
| $V_{1T}$ | Tangential water velocity at inlet |
| $V_2$ | Absolute water velocity at outlet |
| $V_{B1}$ | Blade velocity at outlet |
| $V_{2R}$ | Radial water velocity at outlet |
| $V_{2T}$ | Tangential water velocity at outlet |

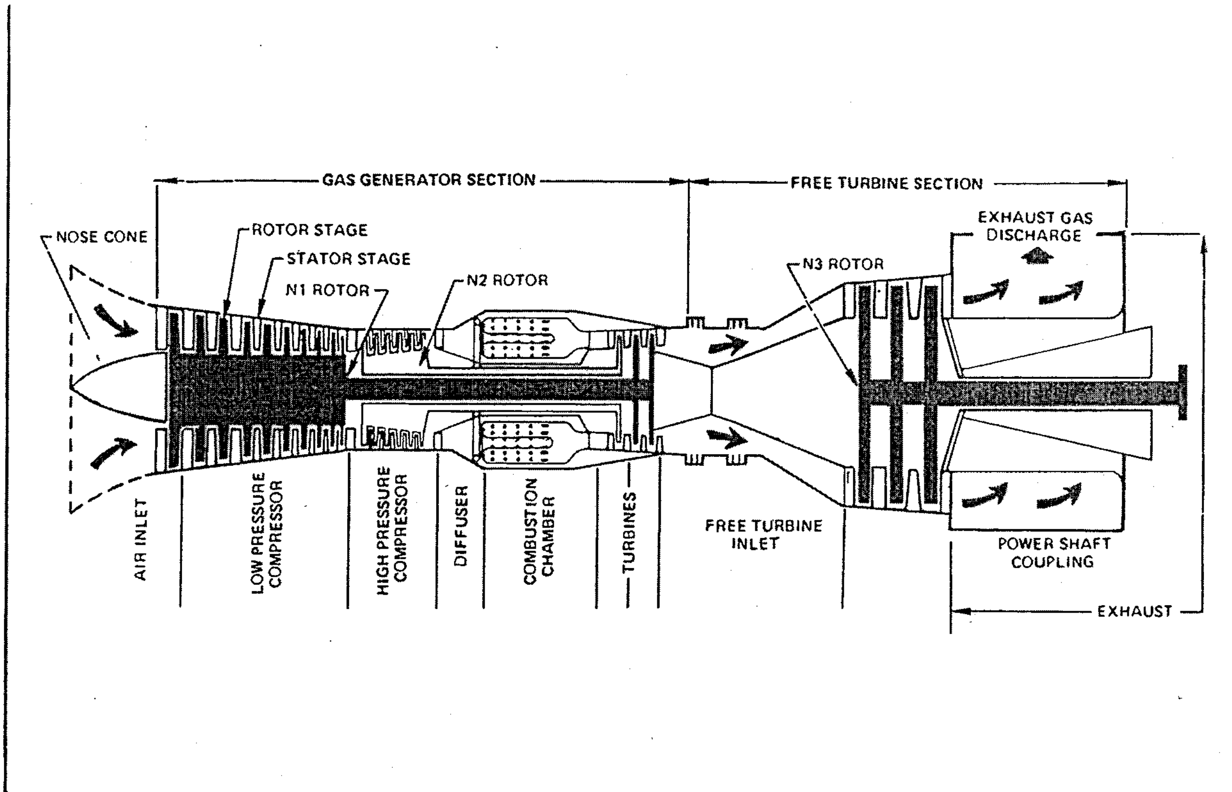
**QUESTION 4 STEAM TURBINE VELOCITY DIAGRAM**

Nomenclature for velocity vectors and angles




| | |
|---|---|
| $V_{S1}$ | Absolute steam velocity entering moving blades |
| $V_{R1}$ | Relative steam velocity entering moving blades |
| $V_B$ | Moving blade velocity |
| $V_{R2}$ | Relative steam velocity leaving moving blades |
| $V_{S2}$ | Absolute steam velocity leaving moving blades |

## QUESTION 5 ACACIA AND PORT REX POWER STATIONS

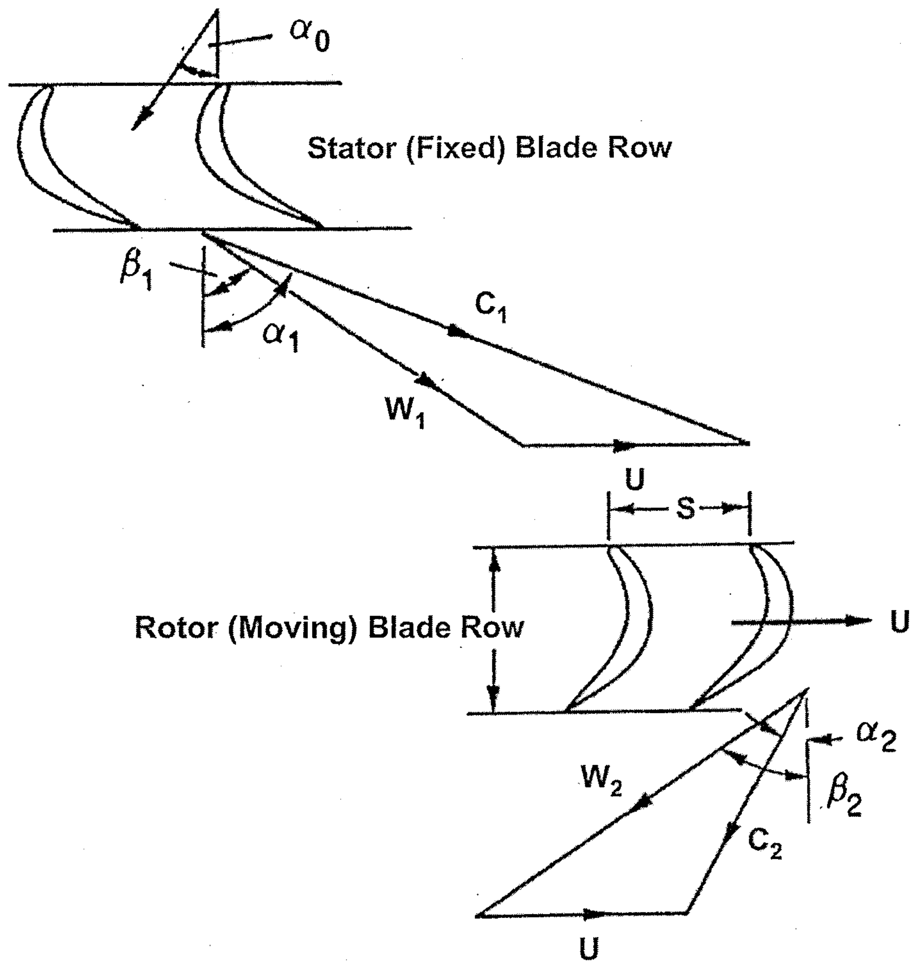



| Technical Specifications | | Peak Load | Base Load |
|---|---|---|---|
| Net Output | (kW) | 60 860 | 57 100 |
| Heat Rate | (kJ/kWh) | 11 791 | 11 887 |
| Speed - N1 Rotor | (rev/min) | 6 805 | 6 640 |
| Speed - N2 Rotor | (rev/min) | 8 395 | 8 320 |
| Speed - Power Turbine | (rev/min) | 3 000 | 3 000 |
| Temperature - Gas Generator Turbine Inlet | ($^{o}$C) | 1 077 | 1 043 |
| Temperature - Power Turbine Inlet | ($^{o}$C) | 682 | 657 |
| Temperature - Power Turbine Exhaust | ($^{o}$C) | 483 | 467 |
| Exhaust Gas Flow Rate | (kg/s) | 278 | 272 |
| Gas Generator Pressure Ratio | | 14.1 | 13.6 |

N1 Low Speed Compressor and Turbine
N2 High Speed Compressor and Turbine

Inlet Air Conditions 15$^{o}$C

**QUESTION 5 GAS TURBINE VELOCITY DIAGRAM**



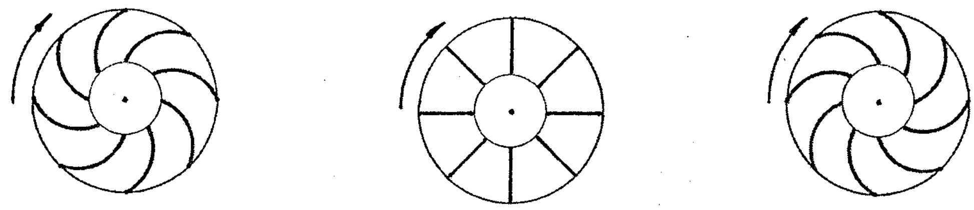

U Blade velocity
$C_1$ Rotor blade absolute inlet velocity
$W_1$ Rotor blade relative inlet velocity
$C_2$ Rotor blade absolute outlet velocity
$W_2$ Rotor blade relative outlet velocity
$C_3$ Stator blade absolute outlet velocity

07-MEC-A6-1 NAME ..................................................

**QUESTION 8 FAN CHARACTERISTICS**

**(a) Velocity Diagrams**

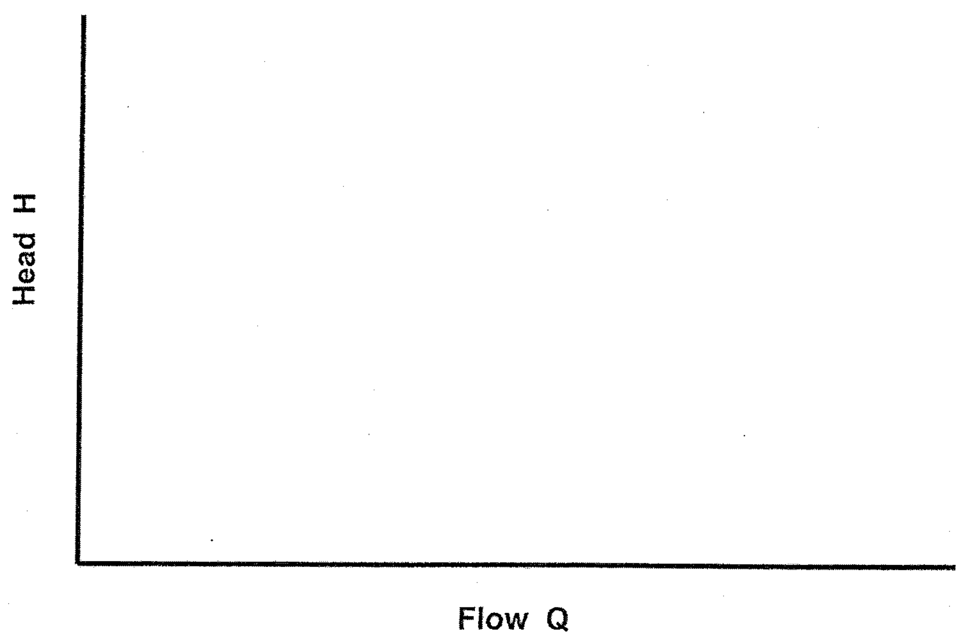
Draw velocity diagrams above each diagram and in line with the topmost blade for the three cases given.



**Forward curved** **Radial** **Backward curved**

**(a) Head versus Flow**

On the axes given draw the head versus flow characteristics for the three cases above. Label the curves accordingly.



## EXAMINATION REFERENCE MATERIAL

## NOMENCLATURE FOR REFERENCE EQUATIONS (SI UNITS)

| | | |
|---|---|---|
| A | Flow area, Surface area | $m^2$ |
| $c_p$ | Specific heat at constant pressure | J/kg°C |
| $c_v$ | Specific heat at constant volume | J/kg°C |
| b | Width | m |
| C | Velocity | m/s |
| D | Diameter | m |
| E | Energy | J |
| F | Force | N |
| g | Gravitational acceleration | $m/s^2$ |
| h | Specific enthalpy | J/kg |
| h | System head | m |
| $h_L$ | Head loss | m |
| H | Pump or turbine head | m |
| k | Ratio of specific heats | |
| L | Length | m |
| m | Mass | kg |
| M | Mass flow rate | kg/s |
| N | Rotational speed | rev/s |
| $N_S$ | Specific Speed | |
| p | Pressure | Pa ($N/m^2$) |
| P | Power | W (J/s) |
| q | Heat transferred | J/kg |
| Q | Heat | J |
| Q | Flow rate | $m^3/s$ |
| r | Radius | m |
| R | Specific gas constant | J/kg K |
| s | Entropy | J/kg K |
| T | Temperature | K |
| u | Specific internal energy | J/kg |
| U | Internal Energy | J |
| U | Velocity | m/s |
| v | Specific volume | $m^3/kg$ |
| V | Velocity | m/s |
| w | Specific work | J/kg |
| W | Work | J |
| W | Velocity | m/s |
| x | Length | m |
| z | Elevation | m |

| | | |
|---|---|---|
| $\alpha$ | Pump blade angle | |
| $\alpha$ | Compressor blade angle | |
| $\beta$ | Pump blade angle | |
| $\beta$ | Compressor blade angle | |
| $\gamma$ | Turbine blade angle | |
| $\varphi$ | Turbine blade angle | |
| $\delta$ | Turbine blade angle | |
| $\eta$ | Efficiency | |
| $\theta$ | Nozzle angle | |
| $\mu$ | Dynamic viscosity | $Ns/m^2$ |
| $v$ | Kinematic viscosity | $m^2/s$ |
| $\rho$ | Density | $kg/m^3$ |
| $\sigma_C$ | Critical cavitation parameter | |
| $\tau$ | Thrust | N |
| $\tau$ | Torque | Nm |
| $\varphi$ | Peripheral velocity factor | |
| $\omega$ | Rotational speed | rad/s |
| $\Omega$ | Heat transfer rate | J/s |

## GENERAL CONSTANTS

***Use unless otherwise specified***

| | | |
|---|---|---|
| Acceleration due to gravity: | $g = 9.81\ m/s^2$ | |
| Atmospheric pressure: | $p_{atm} = 100$ kPa | |
| Water vapour pressure: | $p_{vapour} = 2.34$ kPa | (at 20°C) |
| Density of water: | $\rho_{water} = 1000\ kg/m^3$ | |
| Density of air: | $\rho_{air} = 1.21\ kg/m^3$ | (at 15°C) |
| Density of air: | $\rho_{air} = 1.19\ kg/m^3$ | (at 20°C) |
| Specific heat of air: | $c_p = 1.005$ kJ/kg°C | |
| Specific heat of air: | $c_v = 0.718$ kJ/kg°C | |
| Specific heat of water: | $c_p = 4.19$ kJ/kg°C | |

## GENERAL REFERENCE EQUATIONS

### Basic Thermodynamics

| | |
|---|---|
| First Law: | $dE = \delta Q - \delta W$ |
| Enthalpy: | $h = u + pv$ |
| Continuity: | $\rho VA = \text{constant}$ |
| Potential Energy: | $E_{PE} = mgz$ |
| Kinetic Energy: | $E_{KE} = V^2/2$ |
| Internal Energy: | $E_{IN} = U$ |
| Flow Work: | $w = \Delta(pv)$ |
| Energy Equation: | $zg + V^2/2 + u + pv + \Delta w + \Delta q = \text{constant}$ |

### Ideal Gas Relationships

| | |
|---|---|
| Gas Law: | $pv = RT$ |
| Specific Heat at Constant Pressure: | $c_p = \Delta h/\Delta T$ |
| Specific Heat at Constant Volume: | $c_v = \Delta u/\Delta T$ |
| Specific Gas Constant | $R = c_p - c_v$ |
| Ratio of Specific Heats | $k = c_p / c_v$ |
| Isentropic Relations: | $p_1/p_2 = (v_2/v_1)^k = (T_1/T_2)^{k/(k-1)}$ |

## FLUID MACHINERY REFERENCE EQUATIONS

### Fluid Mechanics

| | |
|---|---|
| Pressure | $p = \rho gh$ |
| Continuity Equation: | $\rho_1 V_1 A_1 = \rho_2 V_2 A_2 = M$ |
| Bernoulli's Equation: | $p_1/\rho g + z_1 + V_1^2/2g = p_2/\rho g + z_2 + V_2^2/2g$ |
| Momentum Equation: | $F = p_1 A_1 - p_2 A_2 - \rho VA(V_2 - V_1)$ (one dimensional) |

### Steam Turbines

| | |
|---|---|
| Nozzle Equation: | $h_1 - h_2 = (V_2^2 - V_1^2) / 2$ |
| Work: | $w = [(V_{1\,absolute}^2 - V_{2\,absolute}^2) + (V_{2\,relative}^2 - V_{1\,relative}^2)] / 2$ |
| Work: | $w = (V_{S1}\cos\theta - V_{S2}\cos\delta)\, V_{blade}$ |
| Power: | $P = wM$ |

## Gas Turbines

| | | |
|---|---|---|
| State Equation: | $pv = RT$ | |
| Isentropic Equation: | $(T_2/T_1) = (p_2/p_1)^{(k-1)/k}$ | |
| Enthalpy Change: | $h_1 - h_2 = c_p(T_1 - T_2)$ | (ideal gas) |
| Nozzle Equation: | $h_1 - h_2 = (V_2^2 - V_1^2) / 2$ | |
| Work: | $w = (C_1 \sin\alpha_1 + C_2 \sin\alpha_2)\, U$ | |
| Work: | $w = [(C_1^2 - C_2^2) + (W_2^2 - W_1^2)] / 2$ | |
| Power: | $P = wM$ | |

## Compressors

| | |
|---|---|
| Work | $W = U(C_{Y2} - C_{Y1})$ |
| Rotor Enthalpy Change | $h_1 + ½W_1^2 = h_2 + ½W_2^2$ |
| Stator Enthalpy Change | $h_2 + ½C_2^2 = h_3 + ½C_3^2$ |
| Isentropic Equation: | $(T_3/T_1) = (p_3/p_1)^{(k-1)/k}$ |

## Jet Propulsion

| | |
|---|---|
| Thrust: | $\tau = M(V_{jet} - V_{aircraft})$ |
| Thrust Power: | $\tau V_{aircraft} = M(V_{jet} - V_{aircraft})V_{aircraft}$ |
| Jet Power: | $P = M(V_{jet}^2 - V_{aircraft}^2) / 2$ |
| Propulsion Efficiency: | $\eta_p = 2V_{aircraft}/(V_{jet} + V_{aircraft})$ |

## Wind Turbine

| | |
|---|---|
| Maximum Ideal Power: | $P_{max} = 8\, \rho\, AV_1^3 / 27$ |

## Energy Equation

| | |
|---|---|
| Pump and Turbine | $p_1/\rho g + z_1 + V_1^2/2g + w_{in}/g = p_2/\rho g + z_2 + V_2^2/2g + w_{out}/g$ |
| With Friction: | $p_1/\rho g + z_1 + V_1^2/2g = p_2/\rho g + z_2 + V_2^2/2g + h_L$ |

## Hydraulic Machines

| | |
|---|---|
| Similarity Equations: | $Q_M/Q_P = (\omega_M/\omega_P)\,(D_M/D_P)^3$ |
| | $H_M/H_P = (\omega_M/\omega_P)^2\,(D_M/D_P)^2$ |
| | $P_M/P_P = (\rho_M/\rho_P)\,(\omega_M/\omega_P)^3\,(D_M/D_P)^5$ |
| Pump Specific Speed: | $N_S = \omega\, Q^{1/2} / (gH)^{3/4}$ |
| Turbine Specific Speed: | $N_S = \omega\, P^{1/2} / [\rho^{1/2} (gH)^{5/4}]$ |
| Critical Cavitation Parameter: | $\sigma = [\{(p_{atmosphere} - p_{vapour}) / \rho g\} - \Delta z] / H$ |
| Moody Efficiency Relationship: | $\eta_P = 1 - (1 - \eta_M)\,(D_M/D_P)^{1/4}\,(H_M/H_P)^{1/10}$ |
| Approximate Moody Efficiency: | $(1 - \eta_M)/(1 - \eta_P) \approx (D_P/D_M)^{1/5}$ |
| Power: | $p = \rho g Q H$ |

**Pumps**

| | |
|---|---|
| Hydraulic Torque: | $\tau = \rho Q\,(r_2V_{2T} - r_1V_{1T})$ |
| Hydraulic Torque: | $\tau = \rho Q\,(r_2V_2\cos\alpha_2 - r_1V_1\cos\alpha_1)$ |
| Power: | $P = 2\pi N \tau$ |
| Net Positive Suction Head: | $NPSH = [(p_{atmosphere} - p_{vapour}) / \rho g] - \Delta z - h_L$ |
| Peripheral Velocity Factor: | $\varphi = V_{B2} / (2gh)^{1/2}$ |
| Critical Cavitation Parameter: | $\sigma_C = NPSH / H$ |
| Approximate Moody Efficiency: | $(1 - \eta_P)/(1 - \eta_M) \approx (D_M/D_P)^{1/5}$ |

**Steam Turbines**

| | |
|---|---|
| Force on Blades: | $F = M\,(V_{S1}\cos\theta - V_{S2}\cos\delta)$ |
| Power to Blades: | $P = M\,(V_{S1}\cos\theta - V_{S2}\cos\delta)\,V_B$ |
| Power to Blades: | $P = M\,[(V_{S1}^2 - V_{S2}^2) + (V_{R2}^2 - V_{R1}^2)] / 2$ |