NATIONAL EXAMINATIONS

May 2019

16-MEC-A6 FLUID MACHINERY

Three hours duration

Notes to Candidates

1. This is a **Closed Book** examination.
2. Examination 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).**
4. **Six questions constitute a complete paper.** (Total 60 marks).
5. **All questions are of equal value.** (Each 10 marks).
6. 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.
7. Read the entire question before commencing the calculations and take note of hints or recommendations given.
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 10 to 15. **All pages from which data has been obtained or on which answers have been written are to be returned with the answer booklet to show any working. Candidate's names must be on these sheets.**
11. **Reference formulae and constants** are given on pages 16 to 20.
12. **Drawing Instruments** (scale ruler, protractor and sharp pencil) are required for vector diagrams. While calculation of velocities by trigonometric ratios with reference to a sketch is acceptable it is longer and more time consuming.

SECTION A CALCULATIVE QUESTIONS

Show all steps in the calculations and state the units for all intermediate and final answers.

QUESTION 1 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. (2)

(b) Calculate the overall efficiency of the turbine based on the design parameters. (1)

Prior to construction, a model test is required 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 do the following:

(c) Determine the speed at which the model should run. (2)

(d) Determine the necessary flow through the model. (2)

(e) Determine the ideal (no friction) power developed by the model. (1)

Due to scaling differences, the efficiency of the model and the prototype are not identical. 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. (2)

[10 marks]

QUESTION 2 COMPRESSOR FIRST STAGE

Refer to the Examination Paper Attachments Page 10 **Acacia and Port Rex Power Stations** and Page 11 **Compressor Velocity Diagram**.

Each power station has three units. Each unit has an output of 60 MW and is powered by twin back to back gas turbines driving a common electrical generator. The diagram and specifications on Page 10 are for <u>one</u> gas turbine only.

Consider the first stage of the compressor (N1 rotor) which has the following approximate parameters:

Rotor hub diameter at inlet	D_1 = 480 mm
Blade tip diameter at inlet	D_2 = 1120 mm
Inlet guide vane outlet angle	α_1 = 30°
First stage moving blade outlet angle	β_2 = 40°

Assume that the plant is operating under the following conditions:

Rotational speed	N = 6800 rev/min
Air flow at inlet to compressor	M = 136 kg/s
Air temperature at inlet to compressor	T_1 = 15°C (288°K)

Assume that ideal conditions prevail (no friction losses).

(a) Calculate the <u>mean</u> blade velocity U and air inlet axial velocity C_{X1} as defined on Page 11. (2)

(b) Draw to scale (see note below) the velocity diagrams at the first stage moving blade inlet and outlet and measure the absolute velocities C_1 and C_2 and relative velocities W_1 and W_2 as defined on Page 11. (3)

(c) Determine the work done in kJ/kg and power input to the first stage in kW. (1)

(d) Determine the enthalpy rise and hence temperature rise in the first stage. (3)

(e) Determine the pressure ratio of the first stage assuming isentropic conditions. (1)

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]

QUESTION 3 STEAM TURBINE BLADES

Refer to the Examination Paper Attachments Page 12 **Steam Turbine Velocity Diagram**.

One stage (set of fixed and moving blades) of a steam turbine receives steam at a mass flow rate of 30 kg/s. The fixed blade outlet angle θ is 20°. The moving blade is symmetrical such that the moving blade outlet angle γ is equal to the moving blade inlet angle Φ.

The initial absolute steam velocity V_{SI} leaving the fixed blades is 450 m/s. The blade velocity V_B is 250 m/s. Due to friction in the moving blades, the outlet relative steam velocity V_{R2} from the moving blades is equal to 0.95 of the inlet relative steam velocity V_{R1} to these blades.

Draw to scale a velocity diagram for this turbine stage and determine the following:

(a) Final absolute steam velocity leaving the moving blades and its direction (angle δ). (8)

(b) Power developed in this turbine stage due to change in momentum of the steam. (1)

(c) Blade efficiency related to the energy and flow of the initial steam jet. (1)

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]

QUESTION 4 PUMP OPERATIONAL CHARACTERISTICS

Refer to the Examination Paper Attachments Page 13 **Pump H-Q Curve and Efficiency**.

A ¼ HP Mastercraft submersible sump sump has the following head versus flow specified characteristics:

Head H (m)	Flow Q (L/minute)
5.5	0
4.6	42
3.1	102
1.5	144

(a) On the given axes, plot the head versus flow curve (H-Q curve) for the pump with head in H metres and flow Q in litres/second. (2)

(b) Assuming that the pump draws 186 W (¼ HP) of electrical power, plot the efficiency η of the pump versus flow Q on the given axes. (4)

(c) Determine the best operating point for the pump and show it on the plot. (1)

(d) An actual test of the pump in the configuration shown on Page 13 and an estimate of the pipe friction indicated the following:

Flow rate	1.6 L/s
Head difference	2.10 m
Head loss in pipe	1.03 m

Determine the actual operating point of the pump as well as the efficiency and show the operating point on the plot. (2)

(e) Compare the results of the actual test with the specified pump characteristics and comment on the result. (1)

Note: Return Page 16 with the examination answer booklet with your name on it.

[10 marks]

QUESTION 5 PUMP SIMULATION EQUATION

Refer to the Examination Paper Attachments Page 13 **Pump H-Q Curve and Efficiency**.

In a computer simulation of the performance of a pump and a system it is convenient to express the characteristic head versus flow curve (H-Q curve) of the pump as a quadratic equation (second order polynomial equation) in the following form:

$$H = a - bQ - cQ^2$$

A ¼ HP Mastercraft submersible sump pump has the flowing head versus flow specified characteristics:

Head H (m)	Flow Q (L/second)
5.5	0.0
4.6	0.7
3.1	1.7
1.5	2.4

(a) Using the first three head and flow points, determine the values of the coefficients (a, b, & c) in the quadratic equation so as to simulate the characteristics of the pump. Rewrite the equation accordingly. (5)

(b) Sketch the head versus flow curve (H-Q curve) and explain what the constants (a, b, & c) in the equation represent with respect to the pump characteristic. (2)

(c) For a flow of 2.4 L/s (fourth point in list above), calculate the anticipated head and compare this with the specified head. Show the difference on the diagram as sketched in (b) above. (2)

(e) State how the quadratic equation could be modified to better match the specified characteristics. (1)

[10 marks]

SECTION B DESCRIPTIVE QUESTIONS

Note that a ten mark question requires an answer of approximately two full pages of complete explanations with sketches, if appropriate, to support the explanation.

QUESTION 6 FAN CONTROL

Refer to the Examination Paper Attachments Page 14 **Fan Control Methods**.

The diagrams show (as dashed lines) the system and fan characteristics and operating point for normal design conditions of a typical air flow system.

The volume flow rate through the system can be controlled (reduced) by three methods:

- Dampers in the ducting which may be progressively closed.
- Vanes at the fan inlet which can create increasing pre-whirl.
- Speed of driving motor which can progressively reduce fan speed.

(a) For each of these three methods, show on the diagrams how the system or fan characteristics change to give a new operating point.

(i) Control by duct dampers.
(ii) Control by inlet vanes.
(iii) Control by fan speed.

In each case show the new operating point. (5)

Note: Return Page 14 with the examination answer booklet and with your name on it.

(b) Explain, with reference to the diagrams, why and how the flow is reduced in each case.

(i) Effect of duct dampers.
(ii) Effect of inlet vanes.
(iii) Effect of fan speed. (5)

Hint: Sketches of velocity diagrams may be useful in the explanation for (b) (ii).

[10 marks]

QUESTION 7 TURBINE BLADE FLOW

PART I VARIATION IN VELOCITY AND PRESSURE

Refer to the Examination Paper Attachments Page 15 **Pressure and Velocity Variation.**

Consider conditions at the inlet of a multistage pressure compounded (pressure difference across each stage) impulse steam turbine. Sketch on the attached graph on Page 16 the variation of steam pressure and steam velocity (absolute velocity) as steam passes through the fixed and moving blades of four stages of such a turbine.

Note: Return Page 15 with the examination answer booklet and with your name on it.

(4 marks)

PART II EFFECT OF BLADE LENGTH

Refer to the Examination Paper Attachments Page 12 **Steam Turbine Velocity Diagram.**

In a multistage steam turbine near the exhaust of the turbine the blade length increases substantially necessitating twisted blades where the blade angle changes progressively from the root (base) to the tip (top).

(a) Explain why the length of the blade must increase and why this change in shape of the blade is necessary.

(b) Assuming that the Steam Turbine Velocity Diagram on Page 12 represents the conditions at the root (base) of the blade:

(i) Sketch the velocity diagram at the root (base) of the blade.
(ii) Sketch the velocity diagram at mid-length (half height) of the blade.
(iii) Sketch the velocity diagram at the tip (top) of the blade.

The diagrams must show clearly how the velocity diagrams change so as to determine the shape of the blade.

(6 marks)

[10 marks]

QUESTION 8 PUMP AND TURBINE CAVITATION AND SETTING

PART I PHENOMENON OF CAVITATION

Describe what determines the formation and collapse of vapour bubbles in a liquid. With reference to the mode of collapse, explain the phenomenon of cavitation and the mechanism of damage to the surface of hydraulic machine components. Show the process of collapse and surface damage with the aid of sketches. Clarify with reasons which parts of pumps and turbines could be damaged due to cavitation.

(5 marks)

PART II PUMP AND TURBINE SETTING

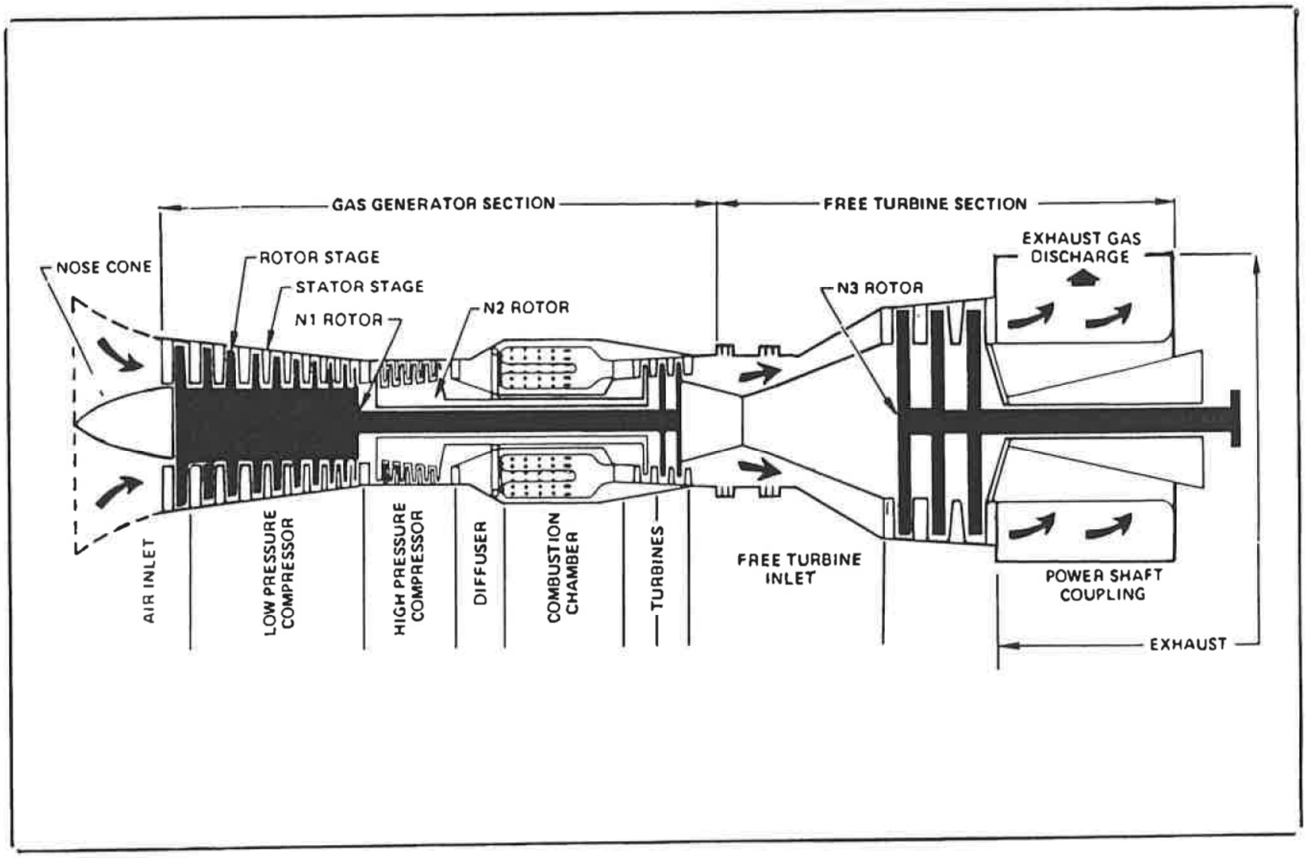
Explain the importance of pump and turbine setting (elevation of machine with respect to lower water surface level). Identify and clarify what parameters need to be taken into account in determining the required setting. Explain how the setting may be different for different types of pumps and turbines. Explain the consequences of incorrect setting.

(5 marks)

[10 marks]

EXAMINATION PAPER ATTACHMENTS

QUESTION 2 ACACIA AND PORT REX POWER STATIONS



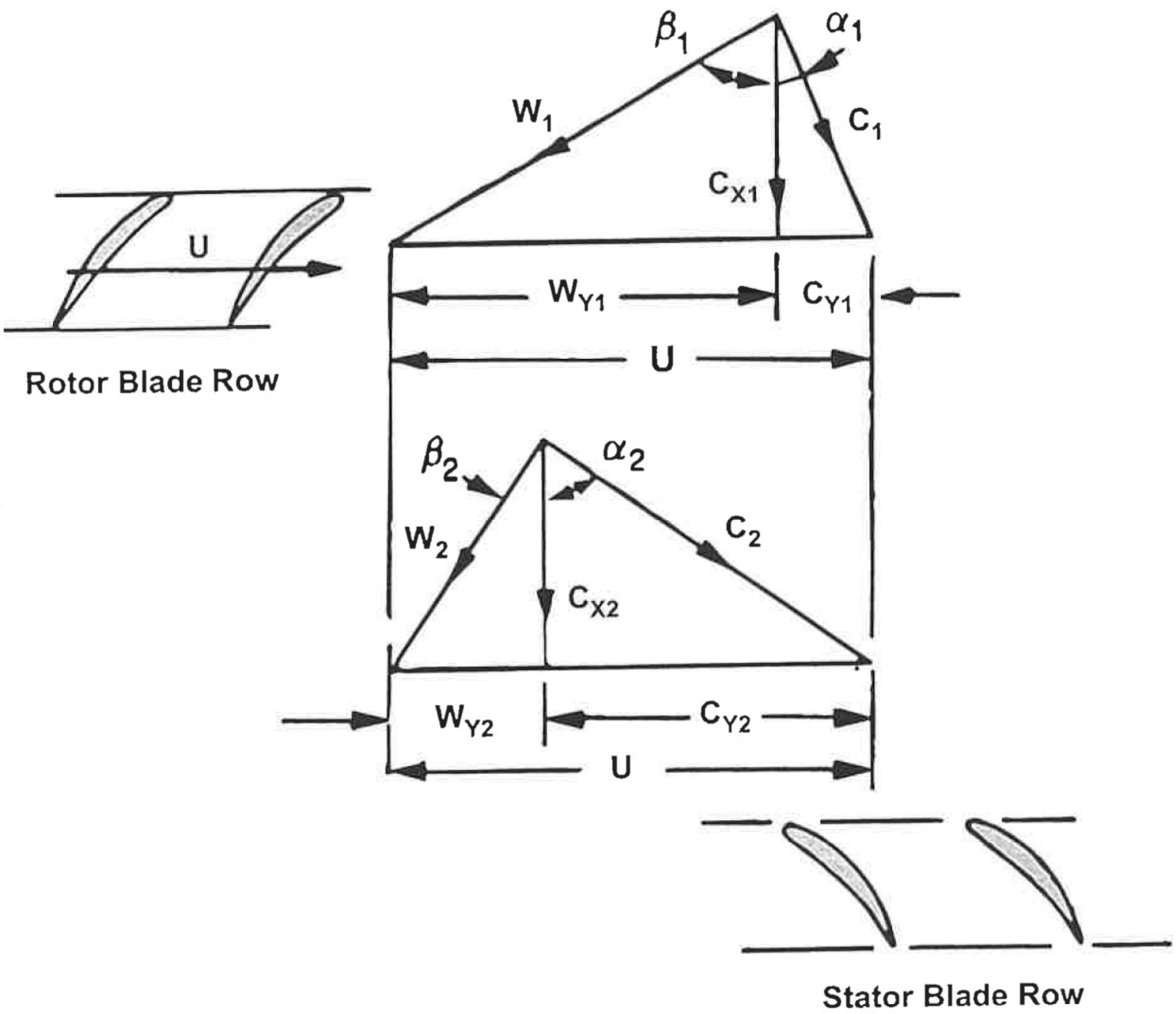
Technical Specifications		Peak Load	Base Load
Net Output	(kW)	60 860	57 100
Heat Rate	(kJ/kWh)	11 791	11 887
Speed - N1 Rotor	(rev/min)	6 805	6 640
Speed - N2 Rotor	(rev/min)	8 395	8 320
Speed - Power Turbine	(rev/min)	3 000	3 000
Temperature - Gas Generator Turbine Inlet	(°C)	1 077	1 043
Temperature - Power Turbine Inlet	(°C)	682	657
Temperature - Power Turbine Exhaust	(°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

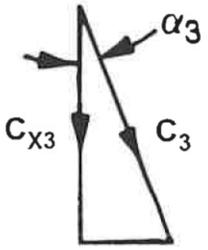
Inlet Air Conditions 15°C

EXAMINATION PAPER ATTACHMENTS

QUESTION 2 COMPRESSOR VELOCITY DIAGRAM



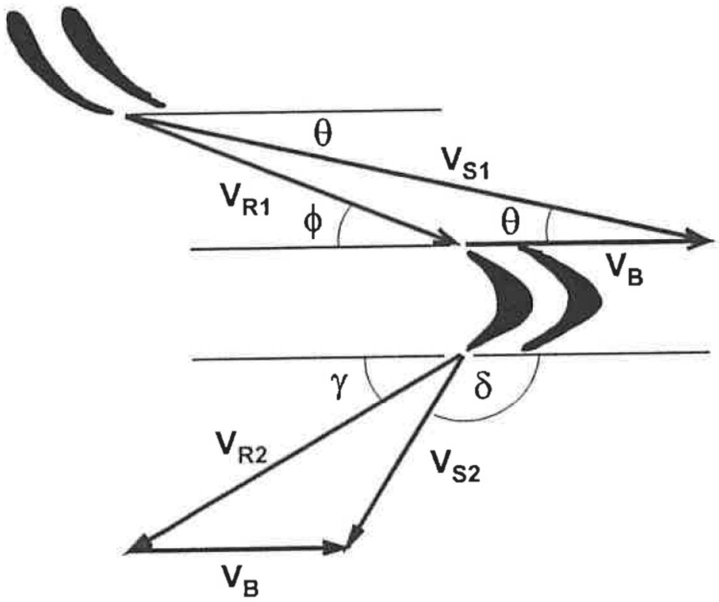
U	Blade velocity
C_1	Rotor blade absolute inlet velocity
W_1	Rotor blade relative inlet velocity
C_2	Rotor blade absolute outlet velocity
W_2	Rotor blade relative outlet velocity
C_3	Stator blade absolute outlet velocity



EXAMINATION PAPER ATTACHMENTS

QUESTION 3 STEAM TURBINE VELOCITY DIAGRAM

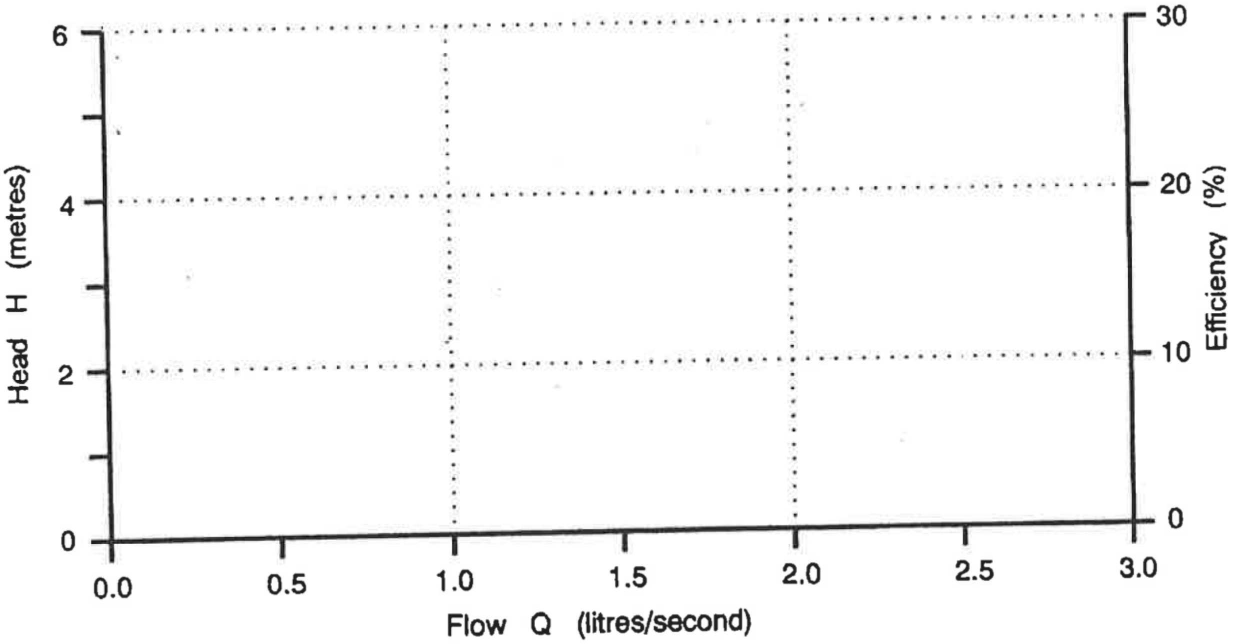
Nomenclature for velocity vectors and angles



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

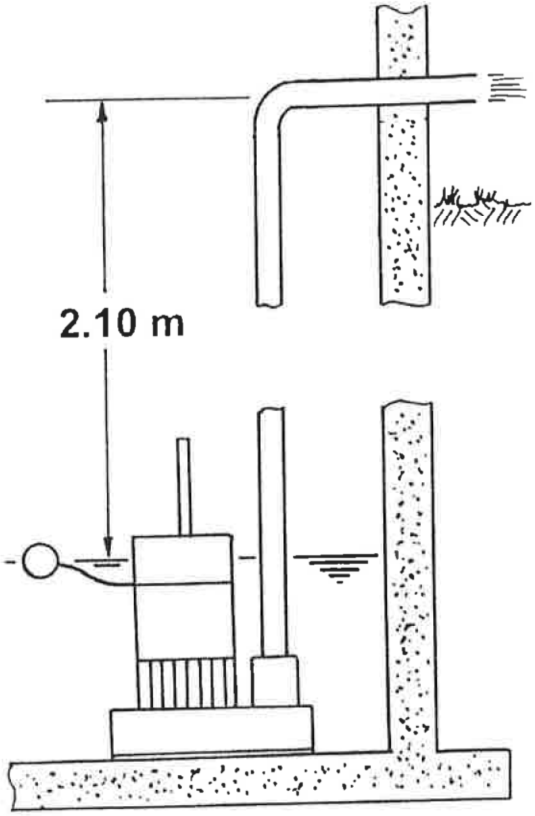
EXAMINATION PAPER ATTACHMENTS

QUESTION 4 PUMP H-Q CURVE AND EFFICIENCY **NAME** ....



Pump Characteristics

Head H (metres)	Flow Q (litres/minute)
5.5	0
4.6	42
3.1	102
1.5	144

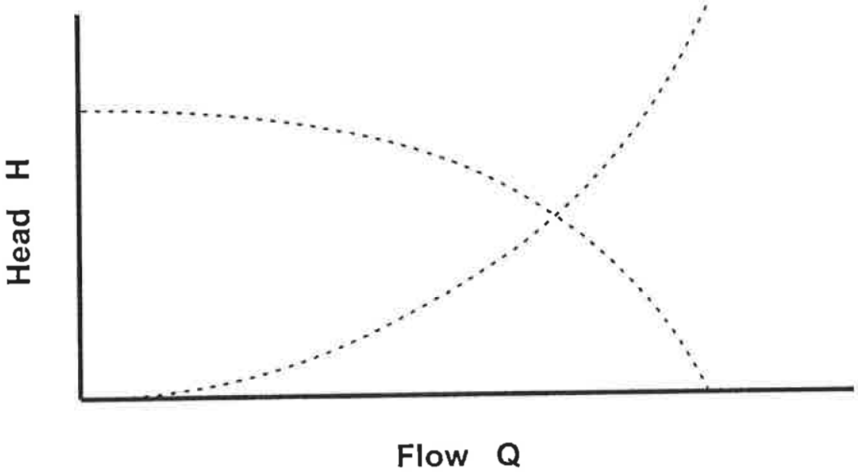


EXAMINATION PAPER ATTACHMENTS

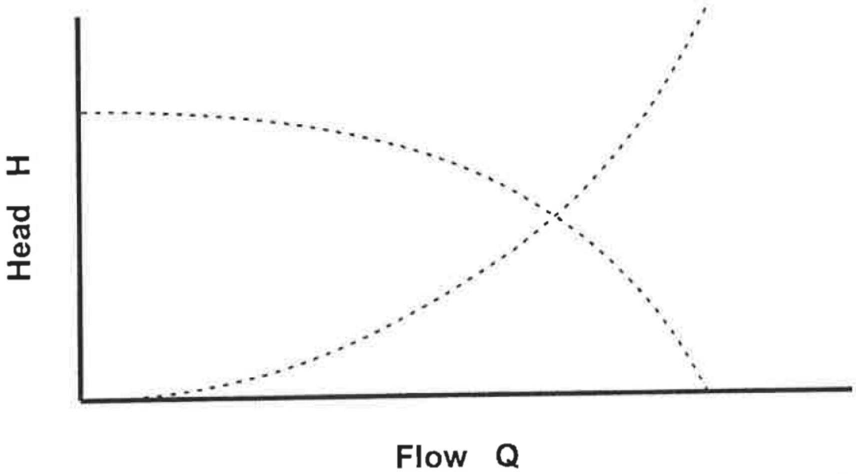
QUESTION 6 FAN CONTROL METHODS

NAME ....

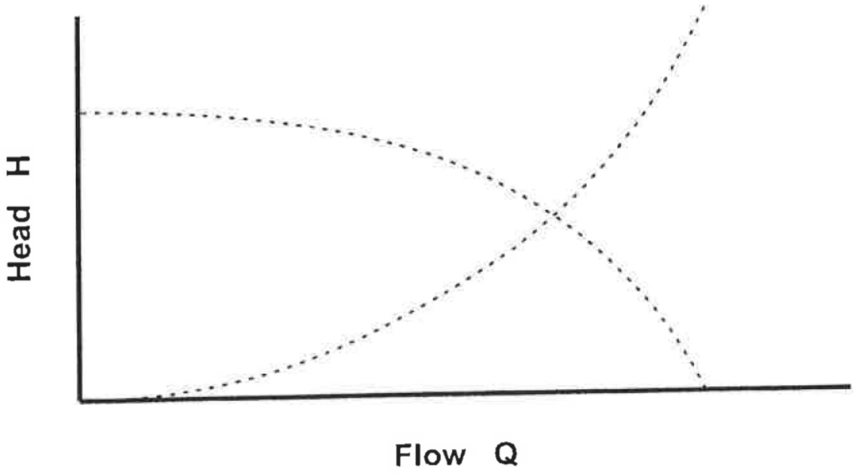
(i) Control by duct dampers



(ii) Control by inlet vanes

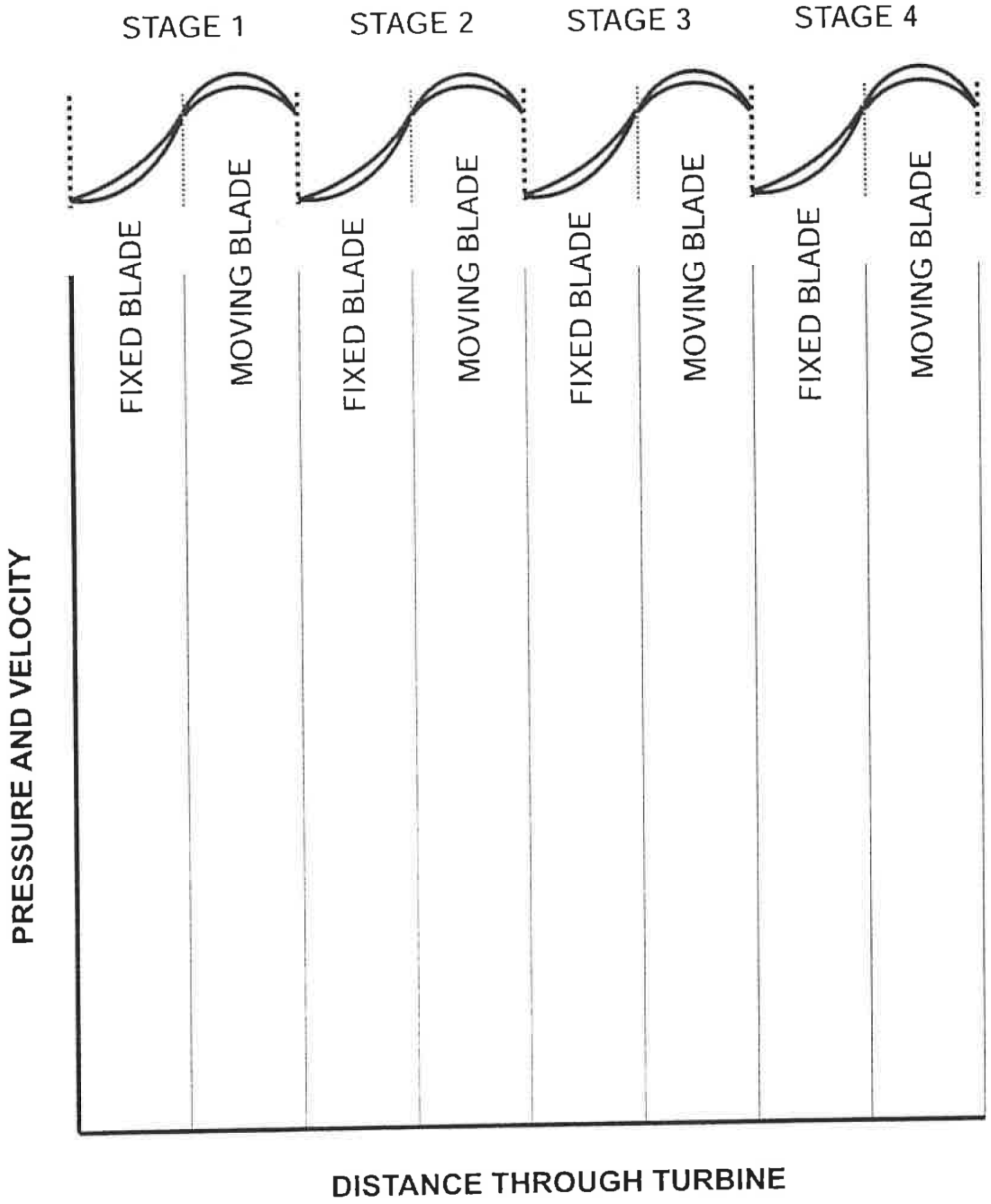


(iii) Control by fan speed



EXAMINATION PAPER ATTACHMENTS

QUESTION 7 PRESSURE AND VELOCITY VARIATION **NAME** ....



On the above diagram show the pressure variation through the first four stages of a multistage pressure compounded impulse turbine.

Show pressure as a solid line ————————
Show velocity as a dashed line ---------------- (absolute velocity relative to turbine casing)

EXAMINATION REFERENCE MATERIAL

NOMENCLATURE FOR REFERENCE EQUATIONS (SI UNITS)

a	Sonic velocity	m/s
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
M_A	Mach number	
N	Rotational speed	rev/min
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
α	Pump blade angle	°
α	Compressor blade angle	°
β	Pump blade angle	°
β	Compressor blade angle	°
γ	Turbine blade angle	°
φ	Turbine blade angle	°
δ	Turbine blade angle	°
η	Efficiency	
θ	Nozzle angle	°
μ	Dynamic viscosity	Ns/m^2
ν	Kinematic viscosity	m^2/s
ρ	Density	kg/m^3
σ_C	Critical cavitation parameter	
τ	Thrust	N
τ	Torque	Nm
φ	Peripheral velocity factor	
ω	Rotational speed	rad/s
Ω	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: $p_{vapour} = 1.71$ kPa (at 15°C)

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)

Energy Equation

Pump and Turbine $p_1/\rho g + z_1 + V_1^2/2g + w_{in}/g = p_2/\rho g + z_2 + V_2^2/2g + w_{out}/g$
Pipe Flow $p_1/\rho g + z_1 + V_1^2/2g = p_2/\rho g + z_2 + V_2^2/2g + h_L$

Compressible Flow

Mach Number $M_A = V/a$
Sonic Velocity $a = [kRT]^{1/2}$

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: $P = wM$
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: $P = M\,[(V_{S1}^2 - V_{S2}^2) + (V_{R2}^2 - V_{R1}^2)] / 2$

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 $h_1 + \frac{1}{2}W_1^2 = h_2 + \frac{1}{2}W_2^2$
Stator Enthalpy Change $h_2 + \frac{1}{2}C_2^2 = h_3 + \frac{1}{2}C_3^2$
Isentropic Equation $(T_3/T_1) = (p_3/p_1)^{(k-1)/k}$

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: $\eta_P = 1 - (1 - \eta_M)\,(D_M/D_P)^{1/4}\,(H_M/H_P)^{1/10}$
Power: $P = \rho g Q H$

Pumps

Hydraulic Torque: $\tau = \rho Q\,(r_2 V_{2T} - r_1 V_{1T})$
Hydraulic Torque: $\tau = \rho Q\,(r_2 V_2 \cos\alpha_2 - r_1 V_1 \cos\alpha_1)$
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: $(1 - \eta_P) / (1 - \eta_M) \approx (D_M/D_P)^{1/5}$

Hydro Turbines

Power $P = \tau\omega$
Power $P = \rho Q\omega\,(r_1 V_1 \cos\alpha_1 - r_2 V_2 \cos\alpha_2)$
Power $P = \rho Q\,(U_1 V_1 \cos\alpha_1 - U_2 V_2 \cos\alpha_2)$
Peripheral Runner Velocity $U_1 = \varphi\,(2gH)^{1/2}$
Absolute Water Velocity $V_1 = C_1\,(2gH)^{1/2}$
Tangential Velocity $U = \omega r$
Angular Velocity $\omega = 2\pi N / 60$

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: $\eta_p = 2\, V_{aircraft} / (V_{jet} + V_{aircraft})$

Wind Turbines

Maximum Ideal Power: $P_{max} = 8\, \rho A V_1^3 / 27$