NATIONAL EXAMINATIONS

May 2016

07-MEC-A6-1 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 11 to 17. **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 18 to 22.
12. **Drawing Instruments** (scale ruler, protractor and sharp pencil) are required for vector diagrams.

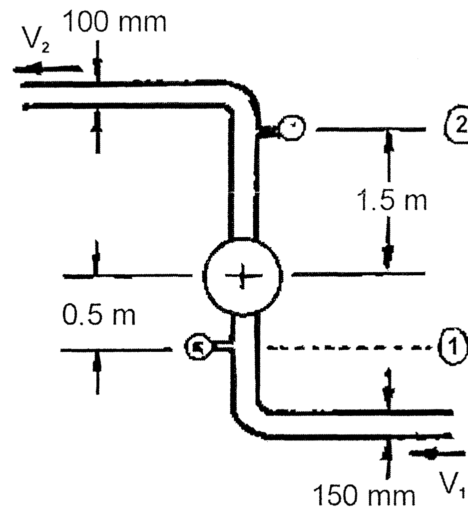
SECTION A CALCULATIVE QUESTIONS

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

QUESTION 1 PUMP POWER AND EFFICIENCY

PART I PUMP POWER REQUIREMENT

The diameters of the suction and discharge pipes of a pump are 150 mm and 100 mm, respectively. The discharge pressure is read by a gauge at a point 1.5 m above the centre line of the pump, and the suction pressure is read by a gauge 0.5 m below the centre line. The pressure gauge reads a pressure of 150 kPa and the suction gauge reads a vacuum of 30 kPa (negative gauge pressure) when gasoline having a specific gravity of 0.75 is pumped at the rate of 0.035 m^3/s. Calculate the electrical power required to pump the fluid if the pump efficiency is 75%.



(5 marks)

PART II HOMOLOGOUS PUMP SCALING

A one tenth (1/10) scale model pump impeller with a diameter of 188 mm is tested in a test facility using water under the following conditions:

Pump Speed	3600 rev/min
Head	39.6 m
Flow	0.085 m^3/s

Under these conditions the model impeller was found to have an efficiency of 84%.

A homologous prototype (with an impeller 10 times the diameter of the model and geometrically identical to it) is to be installed and operated under a head of 110 m while pumping water.

Determine the following:

(a) Speed at which the prototype pump would need to operate to ensure homologous conditions.

(b) Flow rate of water through the pump under homologous conditions.

(c) Power required to operate the pump assuming ideal conditions.

(d) Efficiency anticipated under the above conditions. *(5 marks)*

[10 marks]

QUESTION 2 PUMP APPLICATION

Refer to the Examination Paper Attachments Page 11 **Pump Characteristics** and Page 12 **Cavitation Parameter.** Use the SI units on these charts.

A centrifugal pump is required to pump potable water from a water treatment plant to a reservoir for subsequent distribution. The required flow is 85 L/s and the head 30 m. The pump will be driven by an induction motor operating at 60 Hz and 400 V with a slip of 3% and electrical losses of 4%. To ensure satisfactory operation basic preliminary design information is required.

Assuming that the head loss in the pump inlet piping is 1.0 m and that the vapour pressure under the prevailing conditions is 2 kPa, determine the flowing:

(a) Pump specific speed

(b) Type of pump and sketch of impeller

(c) Diameter of impeller

(d) Critical cavitation parameter

(e) Desired net positive suction head

(f) Maximum elevation of pump relative to supply level

(g) Efficiency of pump

(h) Electric power consumption

<u>Show on the attached diagrams on Page 11 and Page 12 where data has been obtained and return these pages with your answer booklet.</u>

[10 marks]

QUESTION 3 HYDRO TURBINES

PART I PELTON WHEEL

Refer to the Examination Paper Attachments Page 13 **Bridge River Plant**.

Note: Convert the given data to SI units using the conversions below and solve in SI units.

1 HP = 746 watts
1 inch = 25.4 mm

Data:	Gross head	1 226 ft
	Net head	1 118 ft
	Power output	62 000 HP
	Rotational speed	300 rpm
	Pitch diameter	95 in

Determine the following:

(a) Ratio of actual blade velocity to anticipated jet velocity.

(b) Deviation as a percentage of the ratio calculated in (a) above from the ideal ratio and give a possible reason for this deviation.

(c) Volume flow rate required to give the specified output.

(5 marks)

This question is continued on the next page

Question 3 Continued

PART II TURBINE SETTING

Refer to the Examination Paper Attachments Page 14 **Critical Cavitation Parameter**.

Vanderkloof Hydro Power Station has the following technical parameters:

Electrical generator design output	120 MW
Electrical generator voltage	11 kV
Speed of turbine-generator	125 rev/min
Type of hydro turbine	Francis
Design head on turbine	65 m
Maximum water consumption (at lower head)	217 m^3/s
Inlet diameter to spiral casing	7 m
Turbine runner diameter	5462 mm
Turbine runner material	Stainless cast steel

(a) Calculate the specific speed of the turbine.

(b) From the graph determine the Thoma cavitation parameter σ.

(c) Calculate the setting (maximum elevation) of the turbine runner relative to the tailrace water level based on the critical cavitation parameter (Thoma coefficient)

Show on the attached diagram on Page 14 where data has been obtained and return this page with your answer booklet.

(5 marks)

[10 marks]

QUESTION 4 CURTIS TYPE IMPULSE TURBINE

Refer to the Examination Paper Attachments Page 15 **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 BOILER DRAUGHT FANS

Two induced draught (ID) fans are employed in parallel at the exhaust of a large coal fired boiler to extract the combustion gases. Each fan has the following head versus flow characteristic where H is in kPa and Q is in m^3/s:

$$H = K_1 - K_2 Q - K_3 Q^2$$

The complete exhaust system has the following head versus flow characteristic where h is in kPa and Q is in m^3/s

$$h = K_4 Q^2$$

The constants and full load speed of the fans are as follows where rotational speed N is in rev/min:

K_1	=	$4.5 \times 10^{-6} N^2$
K_2	=	0.0
K_3	=	16.0×10^{-6}
K_4	=	5.5×10^{-6}
N	=	1155 rev/min

Sketch the following:

(a) Head versus flow characteristics with one fan in operation and with both fans in operation. In each case identify the operating point of the system.

Calculate the following:

(b) Volume flow rate of exhaust gas with only one fan in operation.

(c) Volume flow rate of exhaust gas with both fans in operation (to give maximum load on the boiler).

(d) Load possible on the boiler with one fan in operation (as a percentage of maximum load as calculated in (c) above).

(e) Speed requirement of both fans to give together the same load as is possible with only one fan in operation (as in (b) above).

[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, if appropriate, to support the explanation.

QUESTION 6 TURBINE BLADE CHARACTERISTICS

PART I IMPULSE AND REACTION

Explain the difference between an impulse turbine and a reaction turbine. In particular refer to the changes in velocity in both the fixed and moving blades. Clarify how the forces developed are created and how they influence the transfer of energy from the fluid to the blades. If appropriate, show the difference between impulse and reaction in velocity diagrams for an axial flow gas or steam turbine.

(5 marks)

PART II OPTIMUM BLADE EFFICIENCY

With respect to a Pelton turbine show graphically in a sketch (efficiency versus blade velocity) how and explain why the efficiency varies with turbine blade velocity (wheel rotational speed) when the water jet velocity remains constant. Consider the whole range of possibilities from a blade velocity of zero to a blade velocity equal to that of the jet. If appropriate, draw velocity diagrams to illustrate the explanation.

(5 marks)

[10 marks]

QUESTION 7 TURBINE BLADE FLOW

PART I VARIATION IN VELOCITY AND PRESSURE

Refer to the Examination Paper Attachments Page 16 **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.

Return Page 16 with the examination answer booklet.

(4 marks)

PART II EFFECT OF BLADE LENGTH

Refer to the Examination Paper Attachments Page 15 **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 15 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 FAN CONTROL

Refer to the Examination Paper Attachments Page 17 **Fan Control Methods**.

The diagrams show (as dotted lines) the system and fan characteristics and operating point for normal design conditions.

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.

Return Page 17 with the examination answer booklet.

(5 marks)

(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.

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

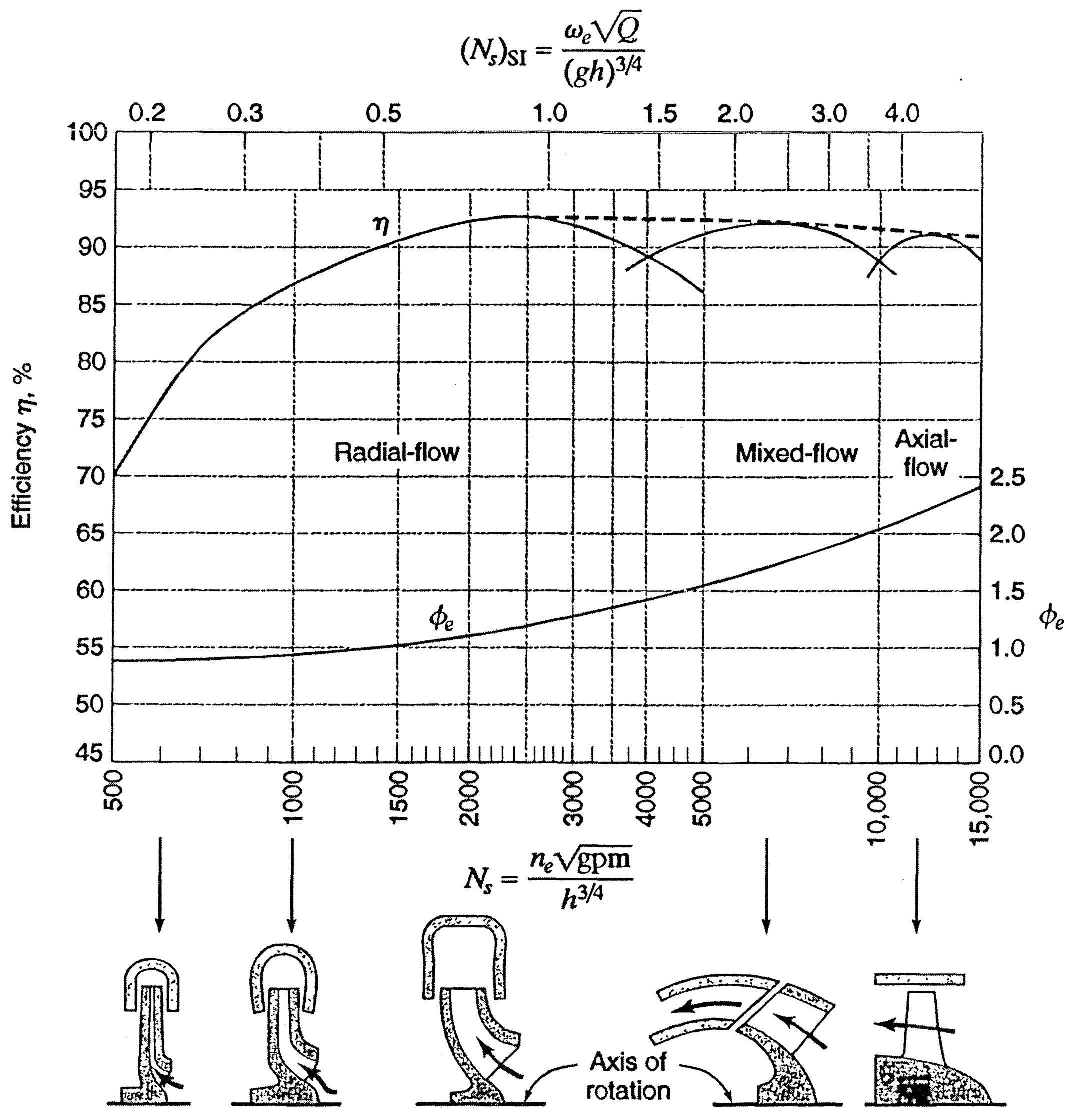
(5 marks)

[10 marks]

EXAMINATION PAPER ATTACHMENTS

NAME ....

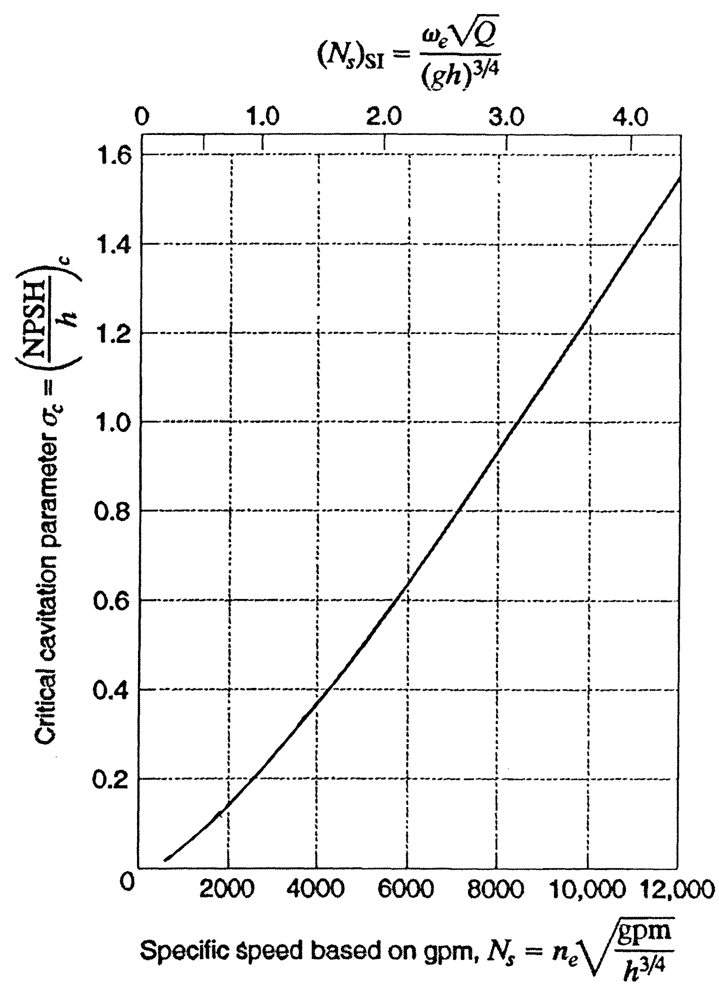
QUESTION 2 PUMP CHARACTERISTICS



Optimum efficiency and typical values of ϕ_e for water pumps as a function of specific speed.

NAME ....

QUESTION 2 CAVITATION PARAMETER



Approximate values of critical cavitation parameter σ_c as a function of specific speed.

QUESTION 3 PART I BRIDGE RIVER PLANT

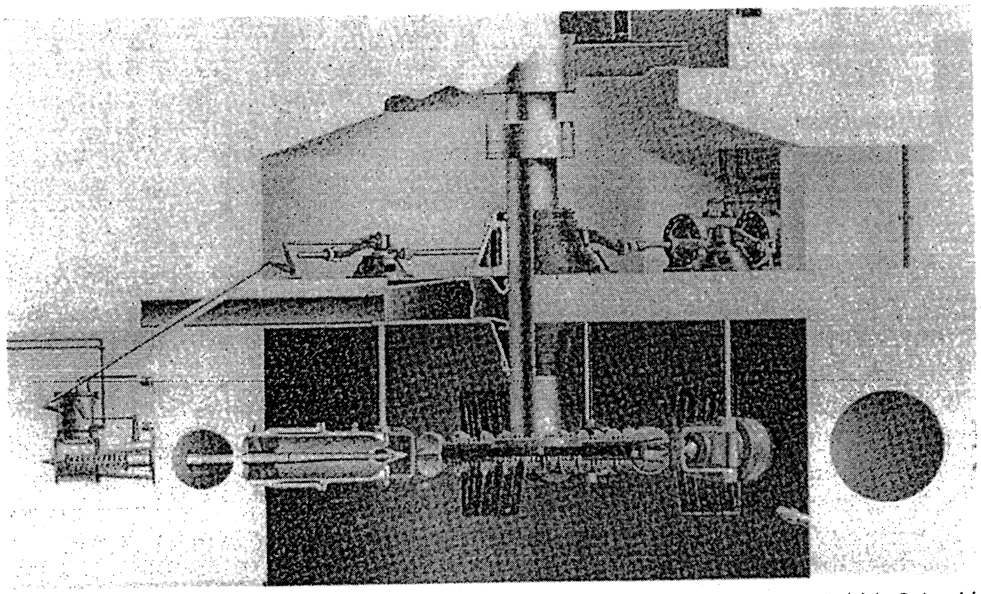
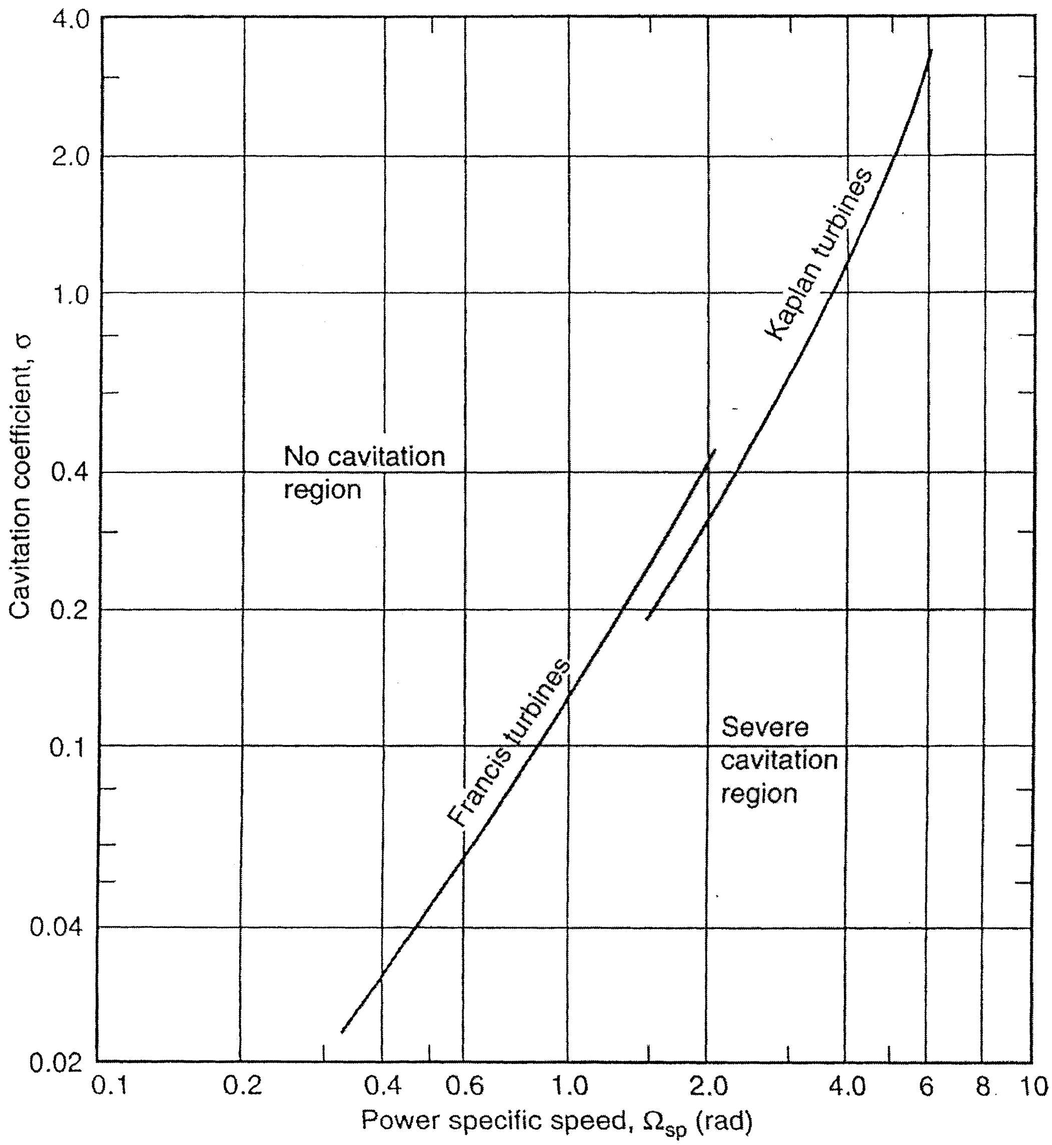


Figure 15.2 Vertical-shaft impulse turbine with six nozzles at Bridge River plant in British Columbia. Gross head = 1,226 ft, net head = 1,118 ft, 62,000 hp, n = 300 rpm, pitch diameter = 95 in.

NAME ....

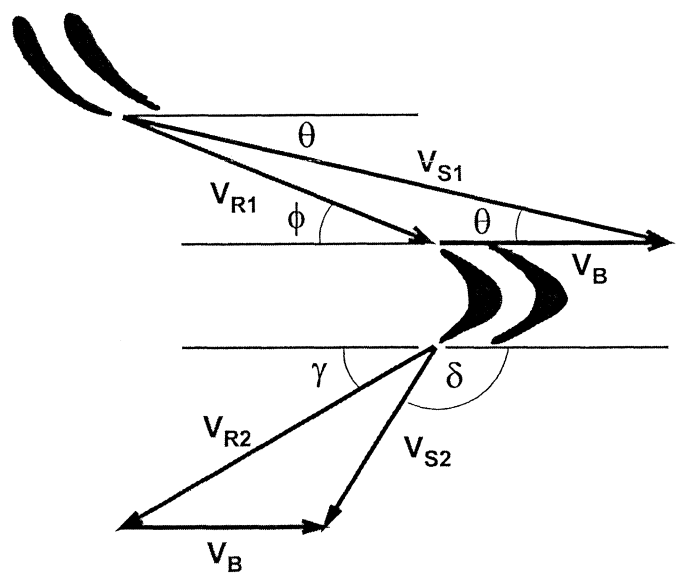
QUESTION 3 PART II CRITICAL CAVITATION PARAMETER



Variation of critical cavitation parameter with non-dimensional specific speed in SI units for Francis and Kaplan turbines

QUESTION 4 & QUESTION 7 PART II STEAM TURBINE VELOCITY DIAGRAM

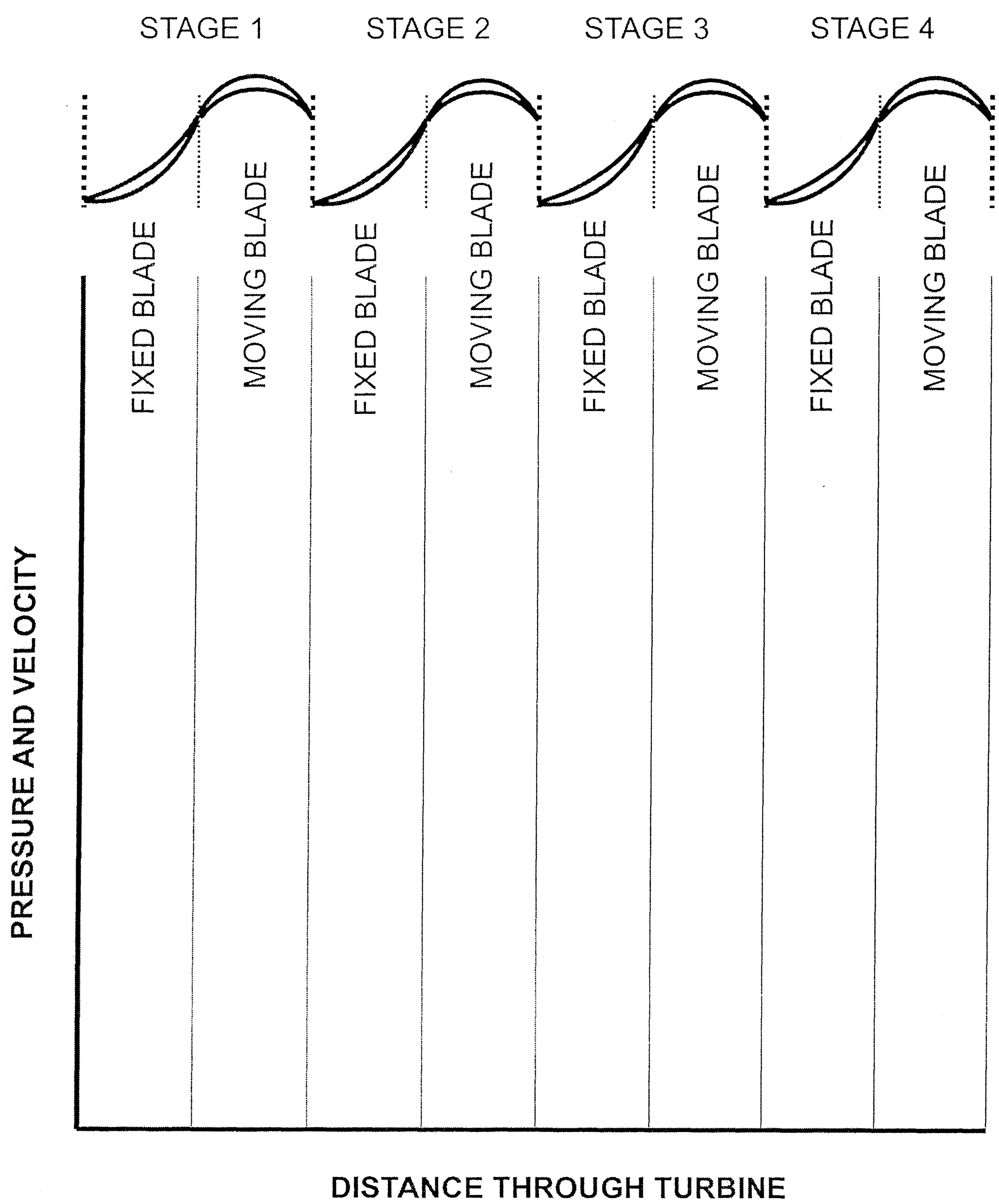
Nomenclature for velocity vectors and angles



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

NAME ....

QUESTION 7 PRESSURE AND VELOCITY VARIATION



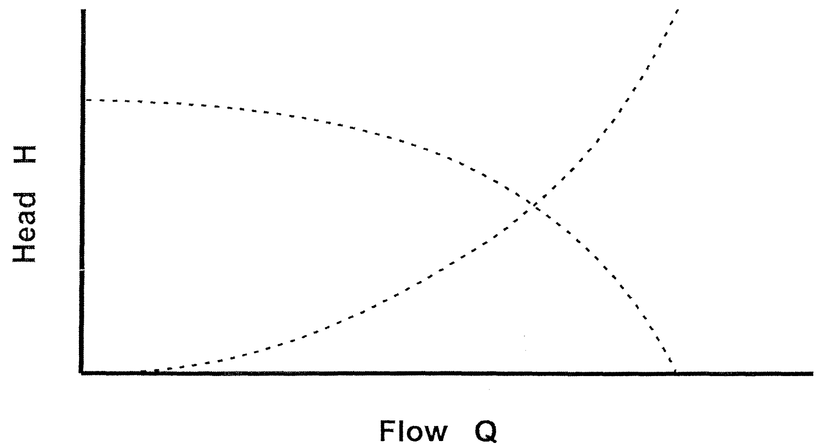
On the above diagram show the pressure variation through the first four stages of a multistage pressure compounded impulse turbine.

Show pressure as a solid line ————————

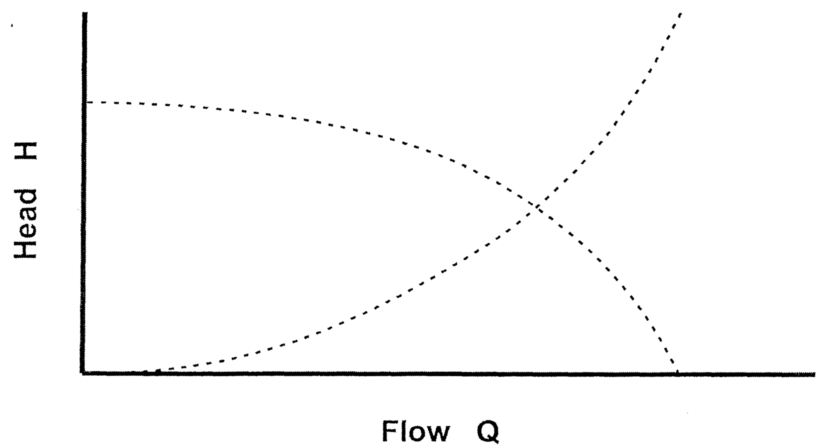
Show velocity as a dashed line ---------------- (absolute velocity relative to turbine casing)

QUESTION 8 FAN CONTROL METHODS NAME ....

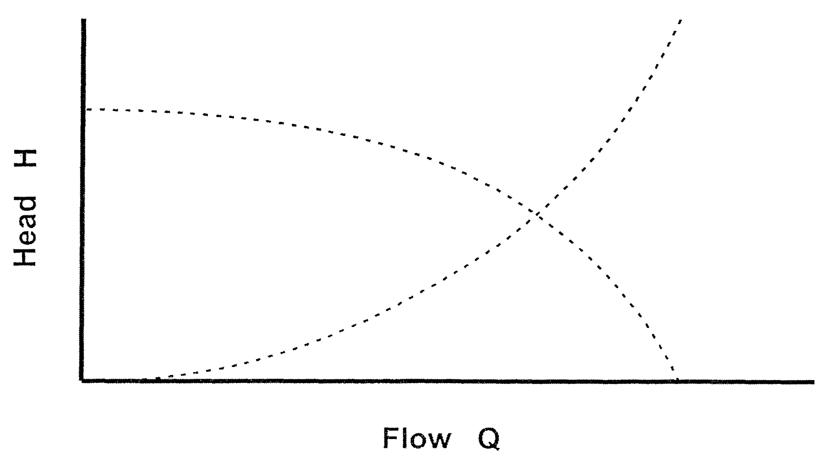
(i) Control by duct dampers



(ii) Control by inlet vanes



(iii) Control by fan speed



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

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:	$p_{vapour} = 2.34$ 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:	$F = p_1A_1 - p_2A_2 - \rho VA(V_2 - V_1)$ (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:	$\tau = \rho Q\,(r_2 V_{2T} - r_1 V_{1T})$
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:	$(1 - \eta_P)/(1 - \eta_M) \approx (D_M/D_P)^{1/5}$

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