NATIONAL EXAMINATIONS

May 2013

07-MEC-A6-1 FLUID MACHINERY

Three hours duration

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 2 is on <u>two</u> pages and also that Question 7 and Question 8 are on the same page.

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 10 to 14. **All pages on which answers have been written are to be returned with the answer booklet. Candidates names must be on these sheets.**

11. **Reference formulae and constants** are given on pages 15 to 19.

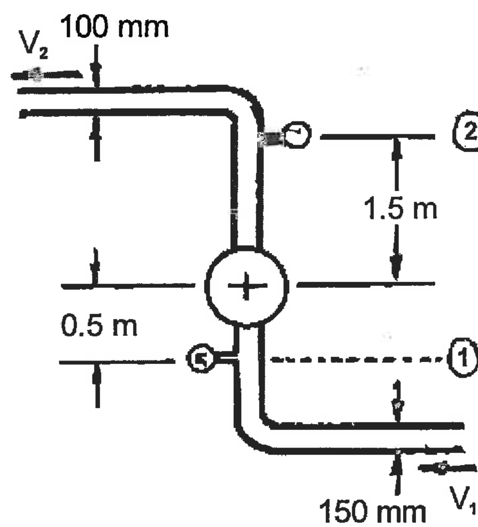
12. **Drawing Instruments** (scale ruler, protractor and sharp pencil) are required for vector diagrams.

SECTION A CALCULATIVE QUESTIONS

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 HYDRO TURBINES

PART I PELTON WHEEL

Refer to the Examination Paper Attachments Page 10 **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

PART II TURBINE SETTING

Refer to the Examination Paper Attachments Page 11 **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)

(5 marks)

[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).

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

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

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 GAS TURBINE BLADES

Refer to the Examination Paper Attachments Page 13 **Acacia and Port Rex Power Stations** and Page 14 **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 at the inlet:

Inlet blade tip diameter	1500 mm
Inlet blade root (hub) diameter	1050 mm

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

Assume that the power turbine has the following efficiency:

$\eta_{turbine}$ = 85%.

Based on these dimensions and conditions at the inlet and exhaust of the power turbine determine the following:

(a) Pressure at power turbine inlet assuming the exhaust is to atmosphere.

(b) Axial velocity of exhaust gas at turbine inlet.

(c) Power turbine output based on actual temperature change.

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

(e) Required whirl velocity (total change in tangential gas velocities) to achieve the power output calculated in (c).

(f) The velocity diagram for the first stage of the turbine, assuming 50% reaction (equal enthalpy drop in fixed and moving blades) and equal enthalpy drop in each stage, drawn to scale as recommended on the next page.

(g) The inlet and outlet angles of both the fixed and moving blade as defined in the diagram on Page 14.

This question is continued on the next page

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 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:

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

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 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 FLOW CHARACTERISTICS

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. Clarify with reasons which parts of pumps and turbines could be damaged due to cavitation.

(5 marks)

PART II NUMBER OF VANES

Centrifugal pump impellers are usually designed for an optimum number of vanes. Explain the effect on performance of a pump having both too many vanes or too few vanes. Clarify how the number of vanes influences the flow through the pump. Explain the reasons for these effects.

(5 marks)

[10 marks]

QUESTION 7 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 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 8 FAN AND COMPRESSOR FLOW CHARACTERISTICS

PART I FAN CONTROL

Describe <u>two</u> of the three methods commonly used for flow control in large centrifugal fans (as for example those installed in large boiler plants). Show in head versus flow diagrams (graphs) for fan and system how the operating point (flow) can be changed in each case and explain why the change has occurred.

(5 marks)

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)

[10 marks]

QUESTION 2 PART I BRIDGE RIVER PLANT

484 *15 Impulse Turbines*

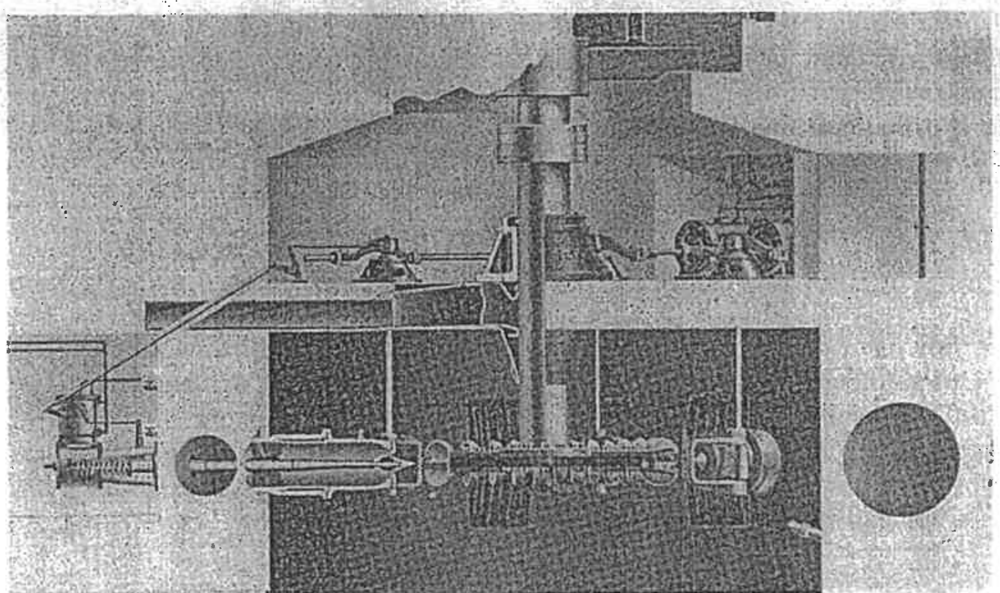
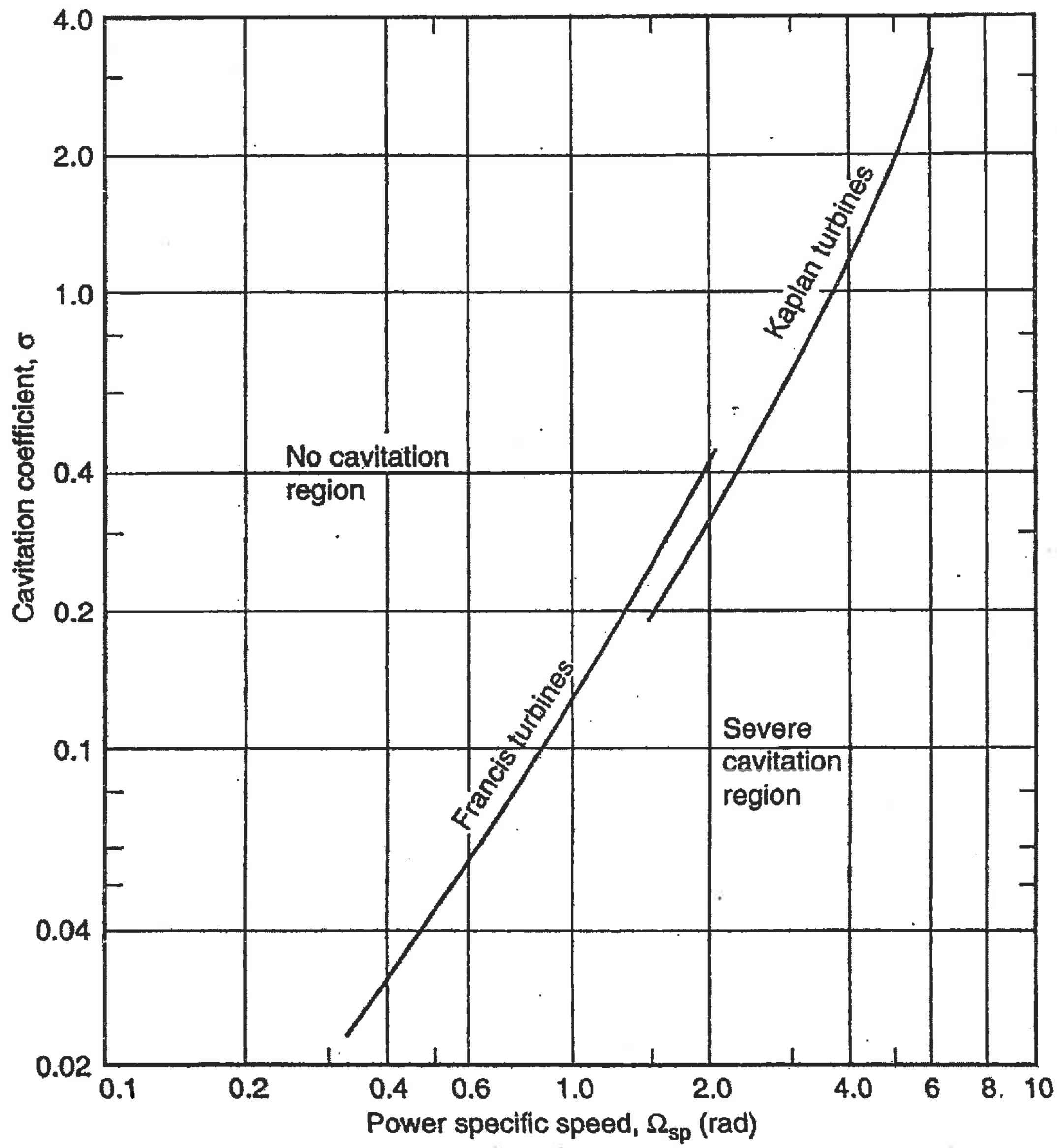


Figure 15.2 Vertical-shaft impulse turbine with six nozzles at Bridge River plant in British Columbia. Gross head = 1,226 ft, net head = 1,118 ft, 62,000 hp, n = 300 rpm, pitch diameter = 95 in.

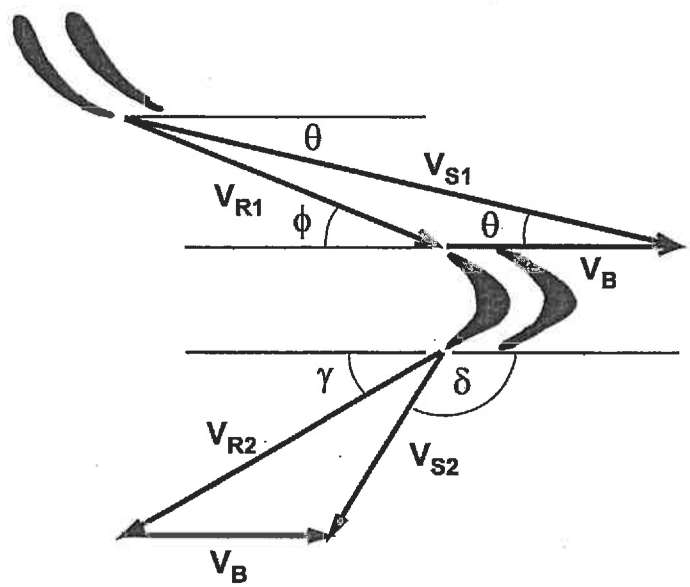
QUESTION 2 CRITICAL CAVITATION PARAMETER



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

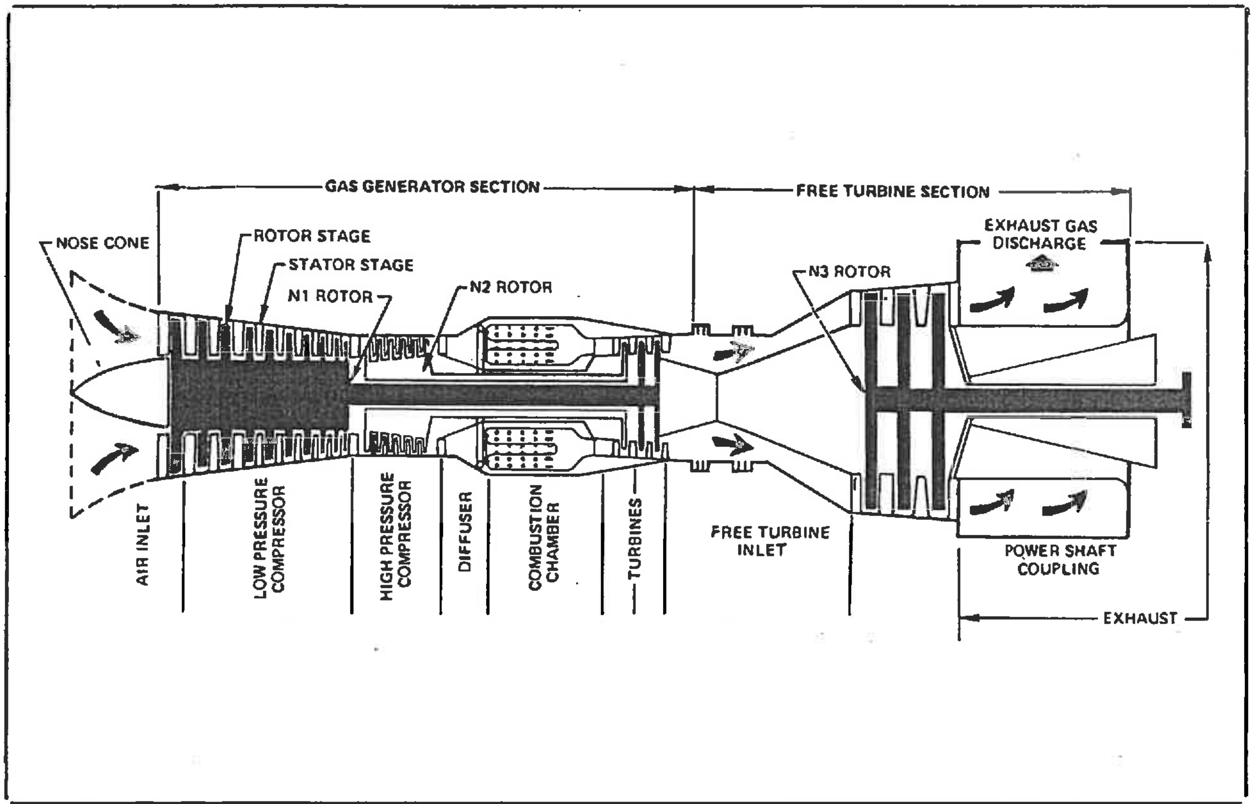
QUESTION 3 STEAM TURBINE VELOCITY DIAGRAM

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

QUESTION 4 ACACIA AND PORT REX POWER STATIONS

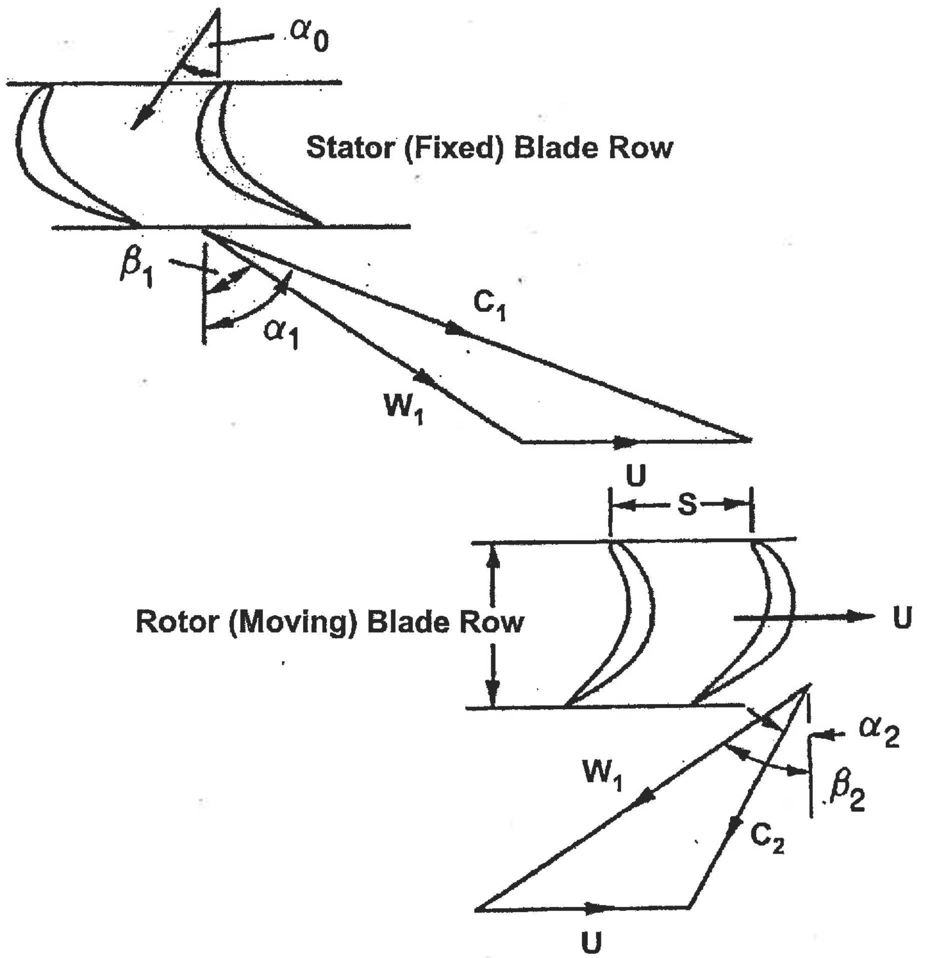


Technical Specifications		Peak Load	Base Load
Net Output	(kW)	60 860	57 100
Heat Rate	(kJ/kWh)	11 791	11 887
Speed - N1 Rotor	(rev/min)	6 805	6 640
Speed - N2 Rotor	(rev/min)	8 395	8 320
Speed - Power Turbine	(rev/min)	3 000	3 000
Temperature - Gas Generator Turbine Inlet	(°C)	1 077	1 043
Temperature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

QUESTION 4 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

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
v	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:	$F = p_1 A_1 - p_2 A_2 - \rho VA(V_2 - V_1)$	(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	$h_1 + \frac{1}{2}W_1^2 = h_2 + \frac{1}{2}W_2^2$
Stator Enthalpy Change	$h_2 + \frac{1}{2}C_2^2 = h_3 + \frac{1}{2}C_3^2$
Isentropic Equation:	$(T_3/T_1) = (p_3/p_1)^{(k-1)/k}$

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