NATIONAL EXAMINATIONS

December 2011

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, each has three (3) questions. Section A is Calculative and Section B is Descriptive.

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 two pages. Note 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. 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. Candidates may use one of the approved Casio or Sharp calculators.

8. Reference data for particular questions are given in the Attachments on pages 10 to 13.

9. Reference formulae and constants are given on pages 14 to 17.

10. Drawing Instruments (scale ruler, protractor and sharp pencil) are required for vector diagrams.
SECTION A  CALCULATIVE QUESTIONS

QUESTION 1  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 = 1226 ft
Net head = 1118 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  KAPLAN TURBINE

Refer to the Examination Paper Attachments Page 10 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:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine-generator speed</td>
<td>112.5 rev/min</td>
</tr>
<tr>
<td>Generator electrical output</td>
<td>110 MW</td>
</tr>
<tr>
<td>Water flow rate</td>
<td>354 m³/s</td>
</tr>
<tr>
<td>Inlet pipe diameter (not in picture)</td>
<td>6.4 m</td>
</tr>
<tr>
<td>Outlet pipe diameter (bottom of picture)</td>
<td>7.0 m</td>
</tr>
<tr>
<td>Inlet water pressure</td>
<td>225 kPa gauge</td>
</tr>
<tr>
<td>Outlet water pressure</td>
<td>-4.5 m H₂O</td>
</tr>
</tbody>
</table>

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

Refer to the Examination Paper Attachments Page 11 Hydro Turbine Characteristics.

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 m³/s
- Maximum water flow: 213 m³/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 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.
QUESTION 3  STEAM TURBINE BLADES

Refer to the Examination Paper Attachments Page 12 Turbine Velocity Diagram

The attached diagram clarifies the nomenclature used in the question below. Use this same nomenclature in your answer.

One stage of a steam turbine operating on the impulse principle has the following blade characteristics:

- Moving blade velocity $V_B = 100$ m/s
- Inlet steam velocity $V_{S1} = 300$ m/s
- Nozzle exit angle $\theta = 25^\circ$
- Steam mass flow rate $M = 24$ kg/s

The moving blades are assumed to be symmetrical and frictionless, that is:

- Blade outlet angle $\gamma =$ Blade inlet angle $\phi$
- Relative velocity $V_{R2} =$ Relative velocity $V_{R1}$

Draw a velocity (vector) diagram to a scale of 1 cm = 20 m/s to show the absolute and relative velocities within the turbine blades. By measuring from this diagram determine the following:

(a) Absolute exhaust steam velocity
(b) Impulse force on the moving blades.
(c) Energy transferred to the moving blades in kJ/kg
(d) Inlet and exhaust kinetic energies in kJ/kg
(e) Blade efficiency
(f) Power developed by the turbine stage

[ 10 marks ]
QUESTION 4 PUMP DESIGN

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

A preliminary analysis for a particular centrifugal pump to deliver water under ambient conditions has yielded the following parameters:

- Impeller inlet radius: 100 mm
- Impeller outlet radius: 180 mm
- Impeller inlet width: 50 mm (in axial direction)
- Impeller outlet width: 30 mm (in axial direction)
- Rotational speed: 1720 rev/min
- Volumetric flow rate: 0.25 m³/s
- Delivery head: 40 m

As a next step in the design of the pump, the impeller blade angles need to be determined.

(a) Calculate the power required by the pump assuming ideal conditions (no friction).

(b) Calculate the mechanical torque required on the shaft.

(c) Calculate the blade inlet and outlet tangential velocities.

(d) Calculate the water inlet and outlet radial velocities.

(e) Calculate the tangential water velocity at the outlet assuming pure radial flow at the inlet.

(f) Draw to scale the inlet and outlet velocity diagrams and determine the inlet and outlet blade angles.

Note that the velocity vector diagrams must be drawn sufficiently large to obtain a suitably accurate answer (1 cm = 5 m/s minimum).

[ 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:

\[
\begin{align*}
K_1 &= 4.5 \times 10^{-6} N^2 \\
K_2 &= 0.0 \\
K_3 &= 16.0 \times 10^{-6} \\
K_4 &= 5.5 \times 10^{-6} \\
N &= 1155 \text{ rev/min}
\end{align*}
\]

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 together to give 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 CAVITATION

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  NET POSITIVE SUCTION HEAD

Explain what is meant by Net Positive Suction Head (NPSH) and how this affects the setting (elevation) of a pump. In particular explain what effect the following parameters have on the NPSH.

- elevation above inlet reservoir
- head loss in inlet pipe
- temperature of liquid
- pressure over inlet reservoir
- design of pump (specific speed)

(5 marks)

[10 marks]
QUESTION 7  BLADE DESIGN

PART I  VANE CHARACTERISTICS

Describe with the aid of sketches the difference between forward curved and backward curved vanes in a centrifugal pump or fan. Explain how these configurations affect the velocity of the fluid leaving the impeller and hence the pressure rise in the diffuser. Sketch a typical head versus flow diagram (graph) for forward and backward curved vanes in a pump or fan.

(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. Explain the reasons for these effects.

(5 marks)

(10 marks)

QUESTION 8  FLOW CONTROL

PART I  FANS

Describe two 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  HYDRO TURBINES

Explain how water flow and hence power output is changed in a typical hydro turbine of the Francis type. Show in a sketch the typical arrangement of components to effect this control. Describe any hydraulic concerns in the penstock or draught tube regarding rapid load changes or low load (low flow) operation of a fixed speed (power generating) machine.

(5 marks)

(10 marks)
QUESTION 1 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.

QUESTION 1 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 2  HYDRO TURBINE CHARACTERISTICS

![Diagram showing efficiency vs. specific speed for impulse and Kaplan turbines.](image-url)
QUESTION 3  TURBINE VELOCITY DIAGRAM

Nomenclature for velocity vectors and angles

\[ V_{S1} \quad \text{Absolute steam velocity entering moving blades} \]
\[ V_{R1} \quad \text{Relative steam velocity entering moving blades} \]
\[ V_B \quad \text{Moving blade velocity} \]
\[ V_{R2} \quad \text{Relative steam velocity leaving moving blades} \]
\[ V_{S2} \quad \text{Absolute steam velocity leaving moving blades} \]
QUESTION 4  PUMP VELOCITY DIAGRAM

Nomenclature for velocity vectors and angles

\[ V_1 \quad \text{Absolute water velocity at inlet} \]
\[ V_{B1} \quad \text{Blade velocity at inlet} \]
\[ V_{1R} \quad \text{Radial water velocity at inlet} \]
\[ V_{1T} \quad \text{Tangential water velocity at inlet} \]
\[ V_2 \quad \text{Absolute water velocity at outlet} \]
\[ V_{B1} \quad \text{Blade velocity at outlet} \]
\[ V_{2R} \quad \text{Radial water velocity at outlet} \]
\[ V_{2T} \quad \text{Tangential water velocity at outlet} \]
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Flow area, Surface area</td>
<td>m²</td>
</tr>
<tr>
<td>c_p</td>
<td>Specific heat at constant pressure</td>
<td>J/kg°C</td>
</tr>
<tr>
<td>c_v</td>
<td>Specific heat at constant volume</td>
<td>J/kg°C</td>
</tr>
<tr>
<td>b</td>
<td>Width</td>
<td>m</td>
</tr>
<tr>
<td>D</td>
<td>Diameter</td>
<td>m</td>
</tr>
<tr>
<td>E</td>
<td>Energy</td>
<td>J</td>
</tr>
<tr>
<td>F</td>
<td>Force</td>
<td>N</td>
</tr>
<tr>
<td>g</td>
<td>Gravitational acceleration</td>
<td>m/s²</td>
</tr>
<tr>
<td>h</td>
<td>Specific enthalpy</td>
<td>J/kg</td>
</tr>
<tr>
<td>h</td>
<td>System head</td>
<td>m</td>
</tr>
<tr>
<td>h_L</td>
<td>Head loss</td>
<td>m</td>
</tr>
<tr>
<td>H</td>
<td>Pump or turbine head</td>
<td>m</td>
</tr>
<tr>
<td>k</td>
<td>Ratio of specific heats</td>
<td>m</td>
</tr>
<tr>
<td>L</td>
<td>Length</td>
<td>m</td>
</tr>
<tr>
<td>m</td>
<td>Mass</td>
<td>kg</td>
</tr>
<tr>
<td>M</td>
<td>Mass flow rate</td>
<td>kg/s</td>
</tr>
<tr>
<td>N</td>
<td>Rotational speed</td>
<td>rev/s</td>
</tr>
<tr>
<td>N_s</td>
<td>Specific Speed</td>
<td></td>
</tr>
<tr>
<td>p</td>
<td>Pressure</td>
<td>Pa (N/m²)</td>
</tr>
<tr>
<td>P</td>
<td>Power</td>
<td>W (J/s)</td>
</tr>
<tr>
<td>q</td>
<td>Heat transferred</td>
<td>J/kg</td>
</tr>
<tr>
<td>Q</td>
<td>Heat</td>
<td>J</td>
</tr>
<tr>
<td>Q</td>
<td>Flow rate</td>
<td>m³/s</td>
</tr>
<tr>
<td>r</td>
<td>Radius</td>
<td>m</td>
</tr>
<tr>
<td>R</td>
<td>Specific gas constant</td>
<td>J/kg K</td>
</tr>
<tr>
<td>s</td>
<td>Entropy</td>
<td>J/kg K</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
<td>K</td>
</tr>
<tr>
<td>u</td>
<td>Specific internal energy</td>
<td>J/kg</td>
</tr>
<tr>
<td>v</td>
<td>Specific volume</td>
<td>m³/kg</td>
</tr>
<tr>
<td>V</td>
<td>Velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>w</td>
<td>Specific work</td>
<td>J/kg</td>
</tr>
<tr>
<td>W</td>
<td>Work</td>
<td>J</td>
</tr>
<tr>
<td>x</td>
<td>Length</td>
<td>m</td>
</tr>
<tr>
<td>z</td>
<td>Elevation</td>
<td>m</td>
</tr>
<tr>
<td>α</td>
<td>Pump blade angle</td>
<td></td>
</tr>
<tr>
<td>β</td>
<td>Pump blade angle</td>
<td></td>
</tr>
<tr>
<td>γ</td>
<td>Turbine blade angle</td>
<td></td>
</tr>
<tr>
<td>φ</td>
<td>Turbine blade angle</td>
<td></td>
</tr>
<tr>
<td>δ</td>
<td>Turbine blade angle</td>
<td></td>
</tr>
</tbody>
</table>
\[ \eta \quad \text{Efficiency} \]
\[ \theta \quad \text{Nozzle angle} \]
\[ \mu \quad \text{Dynamic viscosity} \quad \text{Ns/m}^2 \]
\[ \nu \quad \text{Kinematic viscosity} \quad \text{m}^2/\text{s} \]
\[ \rho \quad \text{Density} \quad \text{kg/m}^3 \]
\[ \sigma_c \quad \text{Critical cavitation parameter} \]
\[ \tau \quad \text{Thrust} \quad \text{N} \]
\[ \tau \quad \text{Torque} \quad \text{Nm} \]
\[ \varphi \quad \text{Peripheral velocity factor} \]
\[ \omega \quad \text{Rotational speed} \quad \text{rad/s} \]
\[ \Omega \quad \text{Heat transfer rate} \quad \text{J/s} \]

**GENERAL CONSTANTS**

*Use unless otherwise specified*

Acceleration due to gravity: \[ g = 9.81 \text{ m/s}^2 \]
Atmospheric pressure: \[ P_{\text{atm}} = 100 \text{ kPa} \]
Density of water: \[ \rho_{\text{water}} = 1000 \text{ kg/m}^3 \]
Specific heat of air: \[ c_p = 1.005 \text{ kJ/kg\degree C} \]
Specific heat of air: \[ c_v = 0.718 \text{ kJ/kg\degree C} \]
Specific heat of water: \[ c_p = 4.19 \text{ kJ/kg\degree C} \]

**GENERAL REFERENCE EQUATIONS**

**Basic Thermodynamics**

- First Law:
- Enthalpy:
- Continuity:
- Potential Energy:
- Kinetic Energy:
- Internal Energy:
- Flow Work:
- Energy Equation:

\[ dE = \delta Q - \delta W \]
\[ h = u + pv \]
\[ pVA = \text{constant} \]
\[ E_{\text{PE}} = mgz \]
\[ E_{\text{KE}} = \frac{V^2}{2} \]
\[ E_{\text{IN}} = U \]
\[ w = \Delta(pv) \]
\[ zg + \frac{V^2}{2} + u + pv + \Delta w + \Delta q = \text{constant} \]
Ideal Gas Relationships

Gas Law: \( pV = RT \)
Specific Heat at Constant Pressure: \( c_p = \Delta h / \Delta T \)
Specific Heat at Constant Volume: \( c_v = \Delta u / \Delta T \)
Isentropic Relations: \( \frac{p_1}{p_2} = \left(\frac{V_2}{V_1}\right)^k = \left(\frac{T_1}{T_2}\right)^{k/(k-1)} \)

FLUID MACHINERY REFERENCE EQUATIONS

Fluid Mechanics

Pressure: \( p = \rho gh \)
Continuity Equation: \( p_1V_1A_1 = p_2V_2A_2 = M \)
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: \( h_1 - h_2 = \frac{(V_2^2 - V_1^2)}{2} \)
Work: \( w = \left[ (V_1^2_{\text{absolute}} - V_2^2_{\text{absolute}}) + (V_2^2_{\text{relative}} - V_1^2_{\text{relative}}) \right] / 2 \)

Gas Turbines

State Equation: \( pv = RT \)
Isentropic Equation: \( \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(k-1)/k} \)
Enthalpy Change: \( h_1 - h_2 = c_p(T_1 - T_2) \) (ideal gas)
Nozzle Equation: \( h_1 - h_2 = \frac{(V_2^2 - V_1^2)}{2} \)

Jet Propulsion

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

Wind Turbine

Maximum Ideal Power: \( P_{\text{max}} = 8 \rho AV_1^3 / 27 \)
Energy Equation

Pump and Turbine
\[ p_1/p_g + z_1 + V_1^2/2g + w_{IN}/g = p_2/p_g + z_2 + V_2^2/2g + w_{OUT}/g \]

With Friction:
\[ p_1/p_g + z_1 + V_1^2/2g = p_2/p_g + z_2 + V_2^2/2g + h_L \]

Hydraulic Machines

Similarity Equations:
\[ Q_M/Q_P = (\omega_M/\omega_P) (DM/DP)^{3/2} \]
\[ H_M/H_P = (\omega_M/\omega_P)^2 (DM/DP)^{3/2} \]
\[ P_M/P_P = (\rho_M/\rho_P) (\omega_M/\omega_P)^3 (DM/DP)^{5/2} \]

Pump Specific Speed:
\[ N_S = \omega Q^{1/2} / (gH)^{3/4} \]

Turbine Specific Speed:
\[ N_S = \omega P^{1/2} / \rho^{1/2} (gH)^{5/4} \]

Moody Efficiency Relationship:
\[ \eta_P = 1 - (1 - \eta_M) (DM/DP)^{1/4} (HM/HP)^{1/10} \]

Power:
\[ P = \rho g Q H \]

Pumps

Hydraulic Torque:
\[ \tau = \rho Q (r_2 V_2 \tau - r_1 V_1 \tau) \]

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:
\[ NPSH = \left[ (\rho_{ATMOSPHERE} - \rho_{VAPOUR}) / \rho g \right] - \Delta z - h_L \]

Peripheral Velocity Factor:
\[ \phi = V_{B2} / (2gh)^{1/2} \]

Critical Cavitation Parameter:
\[ \sigma_C = NPSH / H \]

Steam Turbine

Force on Blades:
\[ F = M (V_{S1} \cos \delta - V_{S2} \cos \delta) \]

Power to Blades:
\[ P = M (V_{S1} \cos \delta - V_{S2} \cos \delta) V_B \]

Power to Blades:
\[ P = M \left[ (V_{S1}^2 - V_{S2}^2) + (V_{R2}^2 - V_{R1}^2) \right] / 2 \]