

**NATIONAL EXAMINATIONS**

**May 2018**

**16-MEC-B6 FLUID MACHINERY**

**Three hours duration**

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**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. Read the entire question before commencing the calculations and take note of hints or recommendations given. Note that Question 5 is on two pages.
7. If doubt exists as to the interpretation of any question, submit 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 16. **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 17 to 21.
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 TURBOJET COMPRESSOR**

Refer to the Examination Paper Attachments Page 11 **Compressor Velocity Diagram**. Refer also to Page 12 **Single Spool Turbojet** for illustrative purposes only.

The top picture on page 12 shows the core engine (compressor, combustion chamber, turbine), while the bottom picture shows the complete engine including afterburner and nozzle.

Specifications for this small turbojet (without afterburner) are as follows:

Number of stages	8
Pressure ratio	6.8
Flow rate	20 kg/s
Rotational speed	16 500 rpm (revolutions per minute)
Static thrust	12.65 kN

Assume the following additional data:

Blade root diameter = one half blade tip diameter for first stage only	
Blade <u>tip</u> velocity for all stages	320 m/s
Air inlet temperature	15°C
Inlet guide (fixed) vane outlet angle $\alpha_1$	10°
First stage moving blade outlet angle $\beta_2$	30°

- (a) Calculate the following:
- First stage blade root and tip diameters.
  - Axial air inlet velocity neglecting blade thickness.
  - Mean blade velocity (velocity at mid-height). (5)
- (b) Draw to scale, the velocity diagrams for the first stage and determine and show all velocities obtained. (3)
- (c) From the velocity diagram determine the power required to drive the first stage. (2)

*Note: The scale drawing should be large enough 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 2 TURBOJET ENGINE**

Refer to the Examination Paper Attachments Page 12 **Single Spool Turbojet** for illustrative purposes only.

The top picture on page 12 shows the core engine (compressor, combustion chamber, turbine), while the bottom picture shows the complete engine including afterburner and nozzle.

Specifications for this small turbojet (without afterburner) are as follows:

Number of stages	8
Pressure ratio	6.8
Flow rate	20 kg/s
Rotational speed	16 500 rpm (revolutions per minute)
Static thrust	12.65 kN (without afterburner)

Assume the following additional information:

Air inlet temperature      15°C  
 Air inlet velocity          200 m/s  
 Equal pressure ratio in all compressor stages.  
 The turbojet engine (or aircraft) is stationary.  
 Ideal thermodynamic and fluid conditions prevail throughout the turbojet.

- (a) From thermodynamic principles, calculate the following:
- Pressure ratio for first stage.
  - Power required to drive the first stage.
  - Power required to drive the whole compressor.      (5)
- (b) From energy and momentum theory, calculate the following:
- Exit jet velocity to give the specified thrust when stationary.
  - Power developed by the turbine of the turbojet.
  - Equivalent net power developed by the gas flow.      (3)
- (c) Sketch a temperature versus entropy diagram of the whole thermodynamic cycle and show where power is produced or consumed. Write the values obtained in (a) and (b) on the diagram in the correct places. (2)

[ 10 marks ]

**QUESTION 3 STEAM TURBINE BLADE EFFICIENCY**

Refer to the Examination Paper Attachments Page 13 **Steam 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:

$$\begin{aligned} \text{Blade outlet angle } \gamma &= \text{Blade inlet angle } \phi \\ \text{Relative velocity } V_{R2} &= \text{Relative velocity } V_{R1} \end{aligned}$$

Draw a velocity (vector) diagram (see note below) to a scale of 1 mm = 2 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.                  | (5) |
| (b) | Impulse force on the moving blades.               | (1) |
| (c) | Energy transferred to the moving blades in kJ/kg. | (1) |
| (d) | Inlet and exhaust kinetic energies in kJ/kg.      | (1) |
| (e) | Blade efficiency.                                 | (1) |
| (f) | Power developed by the turbine stage.             | (1) |

*Note: While calculation of velocities by trigonometric ratios with reference to a sketch is acceptable it is longer and more time consuming.*

[ 10 marks ]

**QUESTION 4 PELTON WHEEL**

An electrical generator is driven by a small single jet Pelton turbine designed to have the following technical parameters:

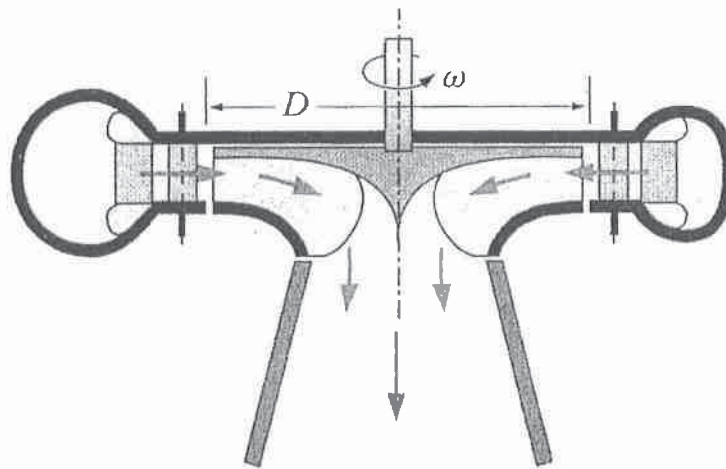
Specific speed	0.20
Effective head at the nozzle inlet	120 m
Nozzle velocity coefficient	0.985
Runner rotational speed	880 rev/min
Blade speed to jet speed ratio	0.47
Overall efficiency of the turbine	0.88 (based on shaft output)

- (a) Calculate the shaft power output of the turbine. (3)
- (b) Calculate the volume flow rate. (2)
- (c) Calculate the jet flow area. (3)
- (d) Calculate the ratio of wheel diameter to jet diameter. (2)

[ 10 marks ]

**QUESTION 5 HYDRO TURBINE**

Refer to the Examination Paper Attachments Page 14 **Hydro Turbine Parameters**. This shows a picture of an actual Francis turbine runner as well as a typical velocity diagram associated with this type of turbine. The diagram below shows how the runner is fitted into the machine.



The specifications, converted to SI units, for the Niagara Falls Francis turbine runners as shown on page 14 are as follows:

Total head	$H$	=	65.227 m
Turbine speed	$N$	=	107 rev/min
Mechanical power	$P$	=	54.063 MW
Turbine efficiency	$\eta$	=	0.938
Runner outer diameter	$D_1$	=	4.470 m (at blade entrance)

Measurements from the picture of the runner indicate approximate parameters as follows:

Runner inner diameter	$D_2$	=	$0.5 D_1$ (at blade exit)
Runner throat diameter	$D_{throat}$	=	$D_1$ (at draft tube entrance)
Runner blade height	$h$	=	$0.2 D_1$ (at blade entrance)
Runner blade thickness	$t_{total}$	=	10% of inlet circumference
Blade inlet angle	$\beta_1$	=	$90^\circ$
Blade outlet angle	$\beta_2$	=	$(180^\circ - 15^\circ) = 165^\circ$

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

**Question 5 Continued**

Using the above (on previous page) information do the following:

- (a) Calculate the water volume flow rate through the turbine. (1)
- (b) Calculate the runner inlet peripheral velocity and radial velocity. (1)
- (c) Draw a velocity diagram at the runner inlet and determine the absolute water velocity  $V_1$  entering the runner and the water inlet angle  $\alpha_1$ . (2)
- (d) Calculate the runner outlet peripheral velocity and radial velocity. Assume that the runner exit area is the same as that of the throat (top of draft tube). (1)
- (e) Calculate the hydraulic power developed by the runner. (1)
- (f) In order to develop the hydraulic power calculated in (e) above determine the angle  $\alpha_2$  at which the water leaves the runner. Assume that the water leaves the runner in a nearly axial direction to minimise loss that is  $V_2 \approx V_{\text{radial}}$ . (2)
- (g) Draw a velocity diagram at the runner outlet and determine the absolute velocity of the water  $V_2$  leaving the runner and the blade exit angle  $\beta_2$ . Compare this value with that given in the data above. (2)

*Note: The velocity diagrams should be drawn to a large enough scale (for example at least 2 mm = 10 m/s at the inlet and 4 mm = 10 m/s at the outlet) and be fully labelled.*

[ 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 COMPRESSOR AND TURBINE BLADE SHAPE

Refer to the Examination Paper Attachments Page 15 **Compressor Inlet Blades**.

Compressor blades at and near the inlet of large compressors have twisted blades as shown in the photograph on Page 15. Similarly turbine blades at or near the exhaust of large turbines also have twisted blades as shown in the adjacent sketch.

- (a) Explain why it is necessary to have twisted blades on large machines. (3)
- (b) For large steam turbines show with the aid of sketches how the velocity diagrams are different at the base and at the tip of twisted blades. (4)
- (c) For large steam turbines explain how the degree of reaction changes from the base to the tip of twisted blades. Clarify what is meant by the degree of reaction. (3)



[ 10 marks ]



## QUESTION 7 COMPRESSOR AND PUMP CHARACTERISTICS

### PART I 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 )

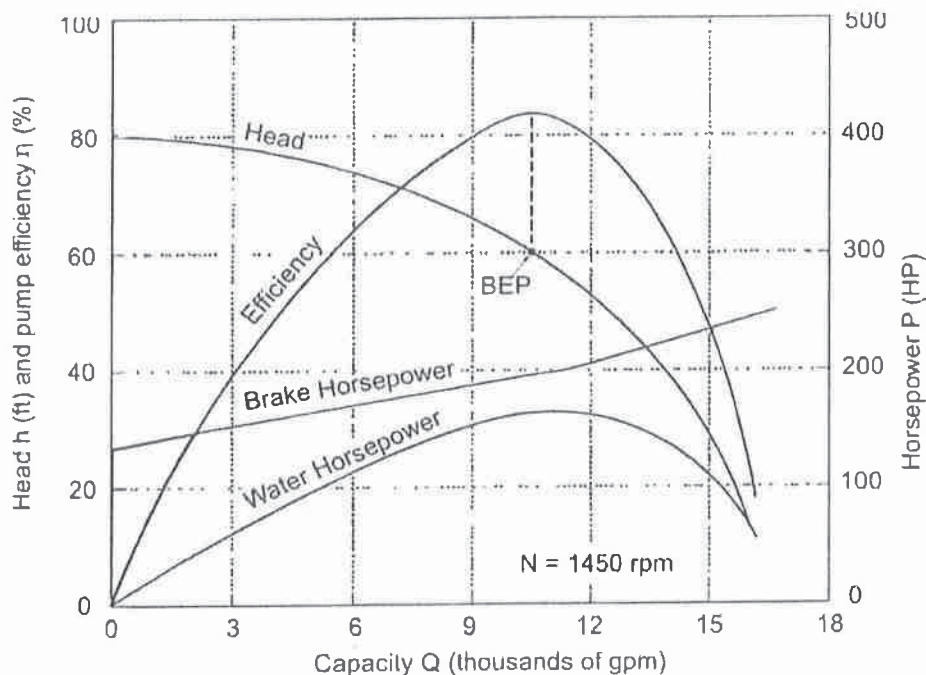
### PART II CENTRIFUGAL PUMPS

With reference to the figure below explain the following:

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

Refer to the Examination Paper Attachments Page 16 **Fan Control Methods**.

The diagrams show (as dotted 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)

Return Page 16 with the examination answer booklet.

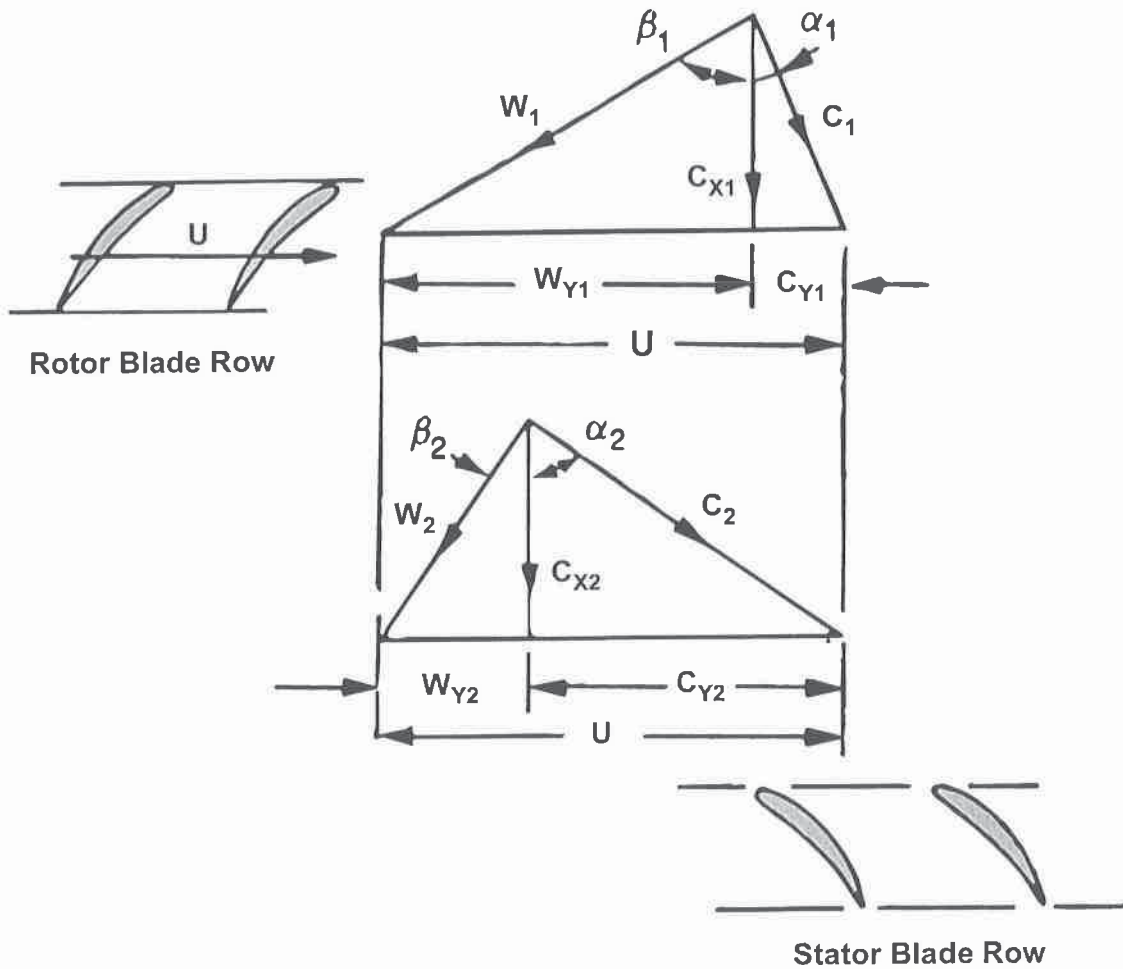
- (b) Explain, with reference to the diagrams, why and how the flow is reduced in each case.
- (i) Effect of duct dampers.
  - (ii) Effect of inlet vanes.
  - (iii) Effect of fan speed. (5)

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

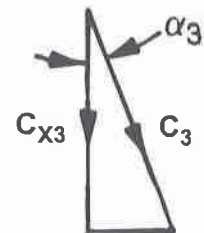
[ 10 marks ]

EXAMINATION PAPER ATTACHMENTS

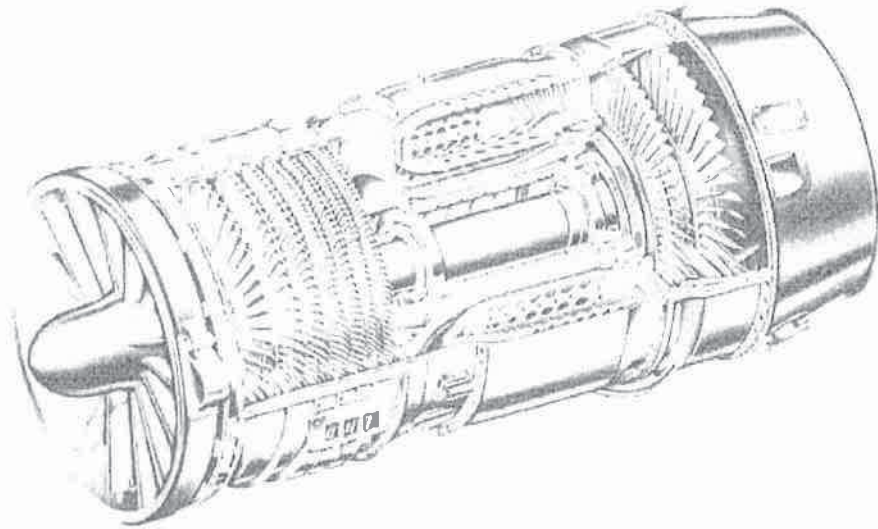
QUESTION 1 COMPRESSOR VELOCITY DIAGRAM



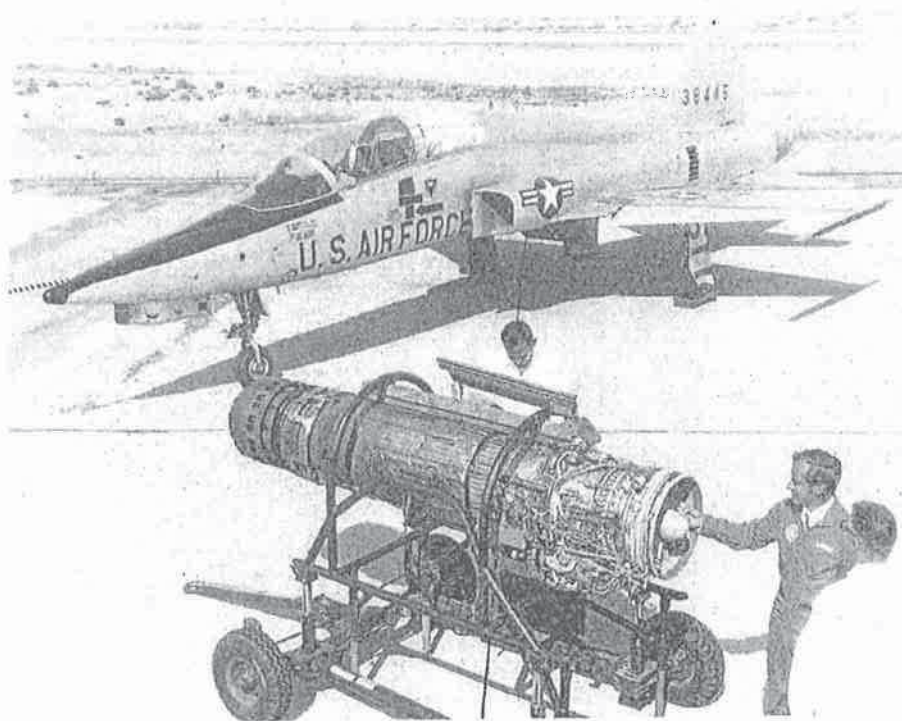
- $U$  Blade velocity
- $C_1$  Rotor blade absolute inlet velocity
- $W_1$  Rotor blade relative inlet velocity
- $C_2$  Rotor blade absolute outlet velocity
- $W_2$  Rotor blade relative outlet velocity
- $C_3$  Stator blade absolute outlet velocity



**QUESTION 1 & QUESTION 2 SINGLE SPOOL TURBOJET**



**Fig 4-21** Single-spool turbojet with low compression ratio (General Electric CJ-610, 8-stage compressor, compressor pressure ratio  $p_{t3}/p_{t2} = 6.8$ , mass flow rate 20 kg/s, rotational speed 16,500 rpm, static thrust 12.65 kN/2800 lb)

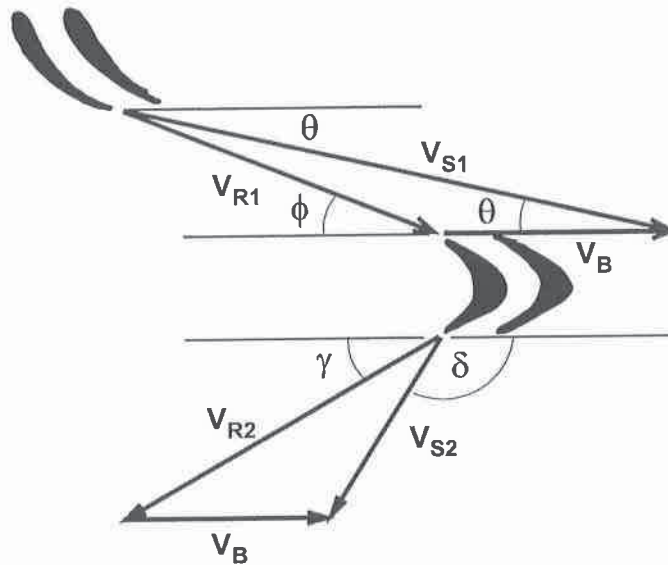


*A J85-21 for the F-5F. Weighing 684 lb, this diminutive afterburning turbojet has an airflow of 53 lb/s and gives 5,000 lb thrust. Installed, it fits downstream of the auxiliary inlet doors seen open near the tail! Note GE overalls.*

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 5 HYDRO TURBINE PARAMETERS

Francis Turbine Runner

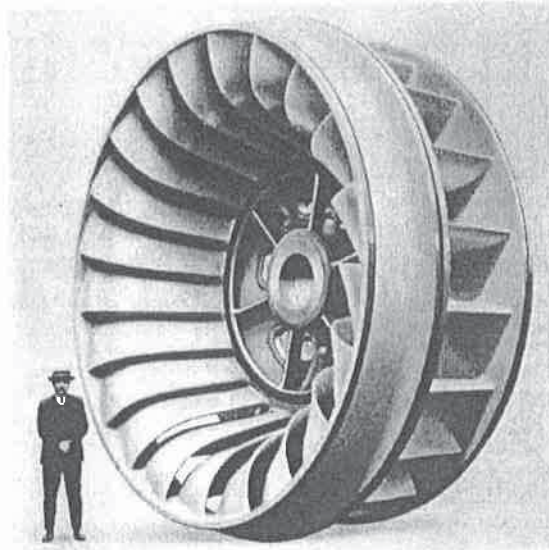
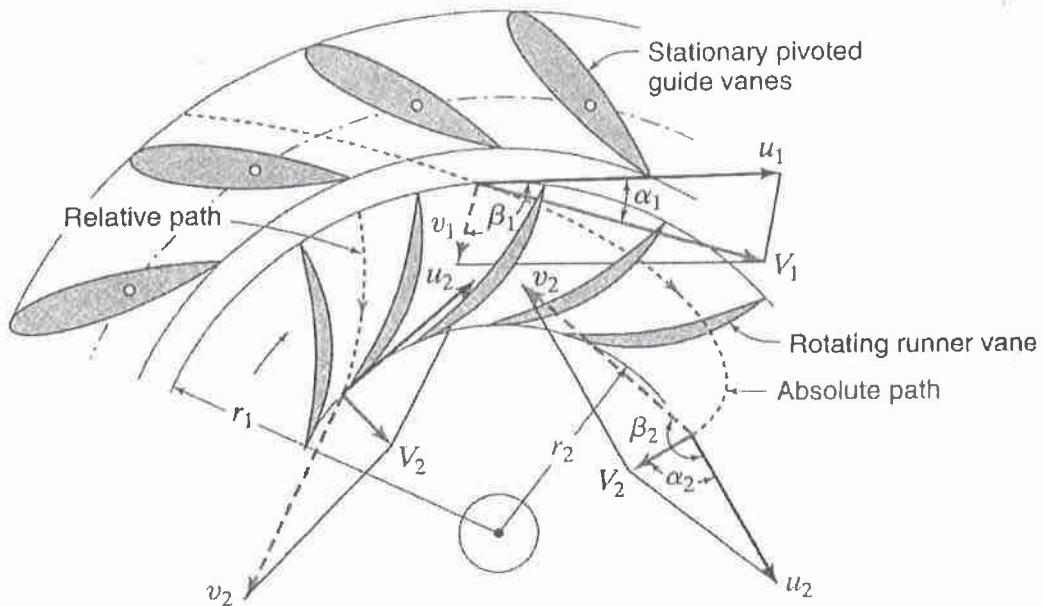


Figure 16.3 Francis runner at Niagara Falls.  $h = 214$  ft,  $n = 107$  rpm,  $\eta = 93.8$  percent at 72,500 hp, diameter = 176 in, overall diameter at band = 183 $\frac{3}{8}$  in. (Courtesy of Allis-Chalmers Mfg. Co.)

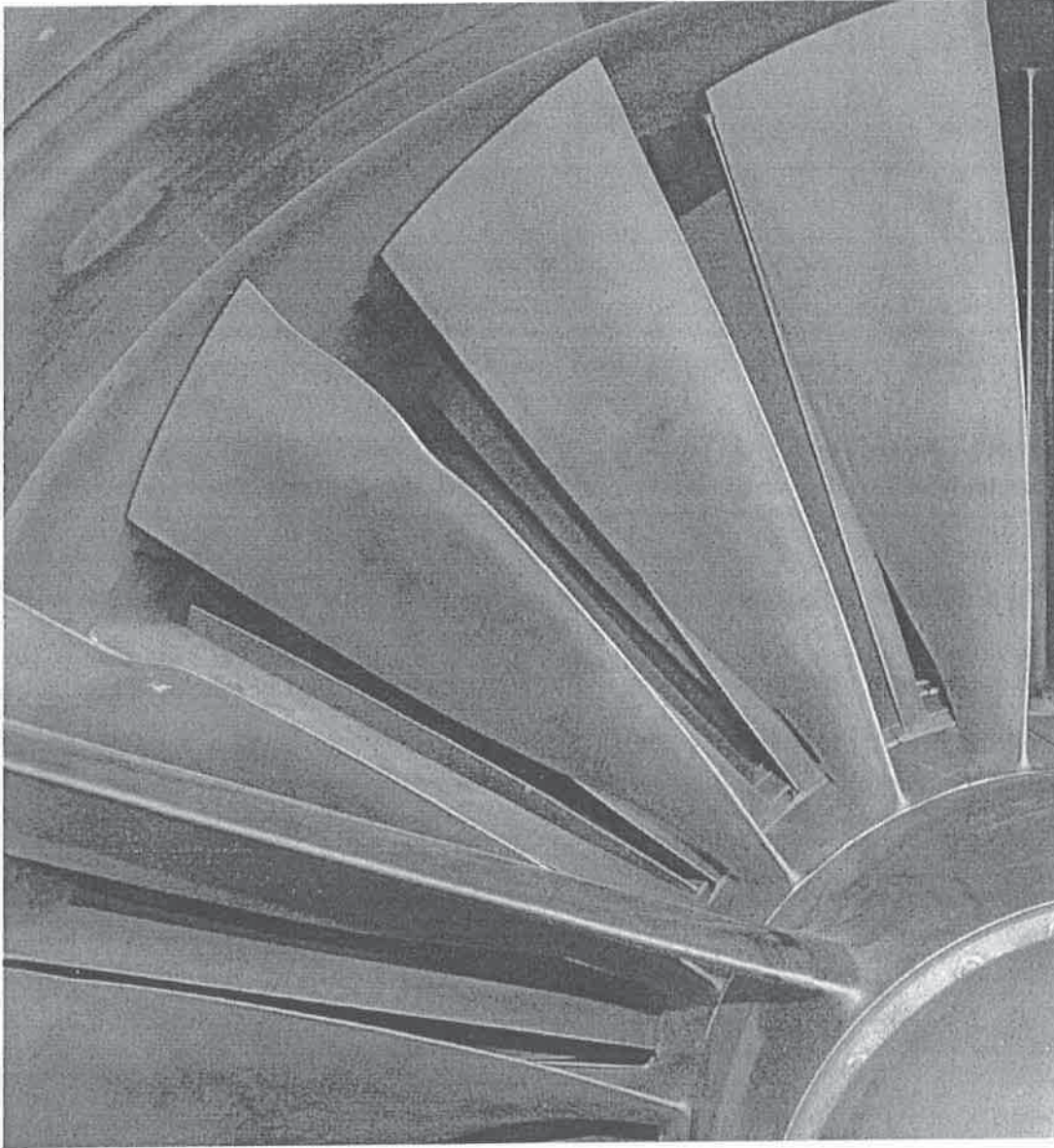
Velocity Diagram



Velocity vectors for radial inward flow hydraulic turbine (Francis type)

**EXAMINATION PAPER ATTACHMENTS**

**QUESTION 6 COMPRESSOR INLET BLADES**

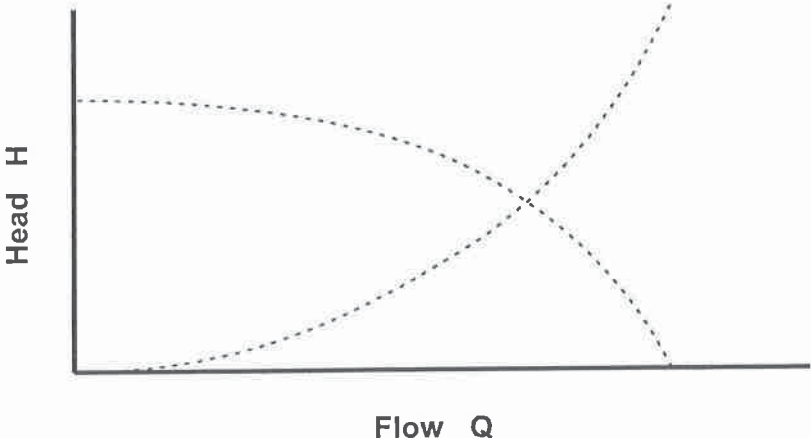


**Rolls-Royce Olympus 593-610 first stage low pressure compressor blades**

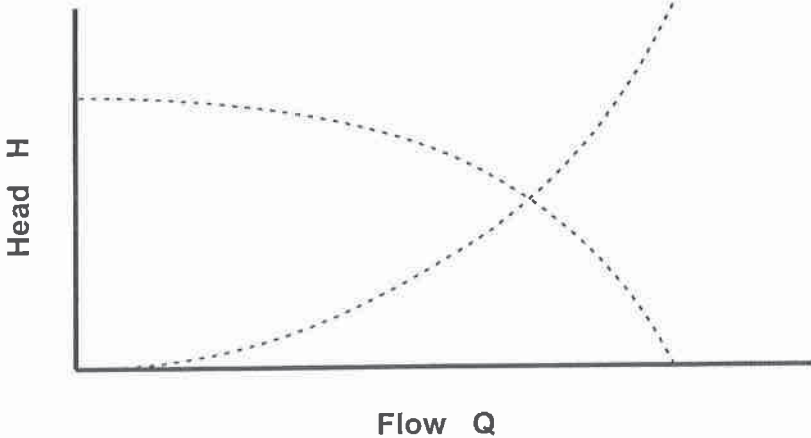
**QUESTION 8 FAN CONTROL METHODS**

**NAME .....**

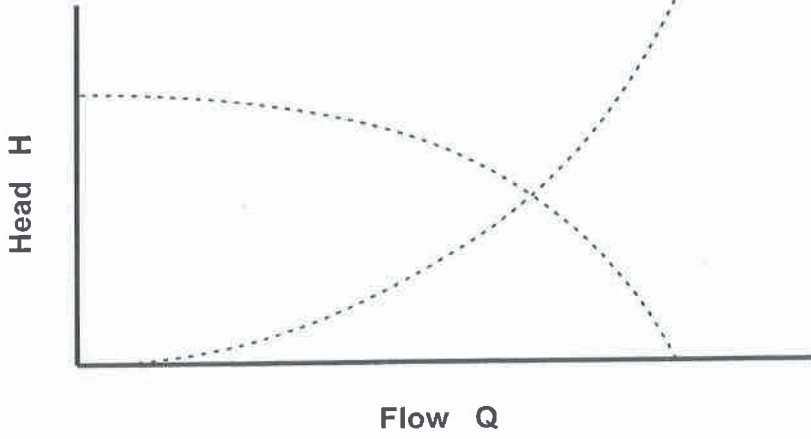
**(i) Control by duct dampers**



**(ii) Control by inlet vanes**



**(iii) Control by fan speed**





**EXAMINATION REFERENCE MATERIAL****NOMENCLATURE FOR REFERENCE EQUATIONS (SI UNITS)**

a	Sonic velocity	m/s
A	Flow area, Surface area	m <sup>2</sup>
c <sub>p</sub>	Specific heat at constant pressure	J/kg°C
c <sub>v</sub>	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 <sup>2</sup>
h	Specific enthalpy	J/kg
h	System head	m
h <sub>L</sub>	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 <sub>A</sub>	Mach number	
N	Rotational speed	rev/min
N <sub>S</sub>	Specific Speed	
p	Pressure	Pa (N/m <sup>2</sup> )
P	Power	W (J/s)
q	Heat transferred	J/kg
Q	Heat	J
Q	Flow rate	m <sup>3</sup> /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 <sup>3</sup> /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	$^{\circ}$
$\alpha$	Compressor blade angle	$^{\circ}$
$\beta$	Pump blade angle	$^{\circ}$
$\beta$	Compressor blade angle	$^{\circ}$
$\gamma$	Turbine blade angle	$^{\circ}$
$\phi$	Turbine blade angle	$^{\circ}$
$\delta$	Turbine blade angle	$^{\circ}$
$\eta$	Efficiency	$^{\circ}$
$\theta$	Nozzle angle	$^{\circ}$
$\mu$	Dynamic viscosity	Ns/m <sup>2</sup>
$\nu$	Kinematic viscosity	m <sup>2</sup> /s
$\rho$	Density	kg/m <sup>3</sup>
$\sigma_c$	Critical cavitation parameter	
$\tau$	Thrust	N
$\tau$	Torque	Nm
$\phi$	Peripheral velocity factor	
$\omega$	Rotational speed	rad/s
$\Omega$	Heat transfer rate	J/s

### GENERAL CONSTANTS

*Use unless otherwise specified*

Acceleration due to gravity:	$g = 9.81 \text{ m/s}^2$
Atmospheric pressure:	$p_{\text{atm}} = 100 \text{ kPa}$
Water vapour pressure:	$p_{\text{vapour}} = 1.71 \text{ kPa}$ (at 15°C)
Water vapour pressure:	$p_{\text{vapour}} = 2.34 \text{ kPa}$ (at 20°C)
Density of water:	$\rho_{\text{water}} = 1000 \text{ kg/m}^3$
Density of air:	$\rho_{\text{air}} = 1.21 \text{ kg/m}^3$ (at 15°C)
Density of air:	$\rho_{\text{air}} = 1.19 \text{ kg/m}^3$ (at 20°C)
Specific heat of air:	$c_p = 1.005 \text{ kJ/kg}^{\circ}\text{C}$
Specific heat of air:	$c_v = 0.718 \text{ kJ/kg}^{\circ}\text{C}$
Specific heat of water:	$c_p = 4.19 \text{ kJ/kg}^{\circ}\text{C}$

**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:	$\rho_1 / \rho_2 = (v_2 / v_1)^k = (T_1 / T_2)^{k/(k-1)}$

**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:	$w = [(V_1^2_{\text{absolute}} - V_2^2_{\text{absolute}}) + (V_2^2_{\text{relative}} - V_1^2_{\text{relative}})] / 2$
Work:	$w = (V_{S1} \cos \theta - V_{S2} \cos \delta) V_{\text{blade}}$
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:	$\sigma = \{[(p_{\text{atmosphere}} - p_{\text{vapour}}) / \rho g] - \Delta z\} / H$
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:	$\phi = V_{B2} / (2gh)^{1/2}$
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	$U_1 = \phi (2gH)^{1/2}$
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:	$T = M (V_{jet} - V_{aircraft})$
Thrust Power:	$T V_{aircraft} = M (V_{jet} - V_{aircraft}) V_{aircraft}$
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:	$P_{max} = 8 \rho A V_1^3 / 27$
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