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higher education & training

Department:
Higher Education and Training
REPUBLIC OF SOUTH AFRICA

T1460**(E)**(N21)T
NOVEMBER EXAMINATION

NATIONAL CERTIFICATE

POWER MACHINES N6

(8190046)

21 November 2013 (X-Paper)
09:00–12:00

REQUIREMENTS: Steam tables

Calculators and drawing instruments may be used.

This question paper consists of 7 pages and 1 formula sheet.

DEPARTMENT OF HIGHER EDUCATION AND TRAINING
REPUBLIC OF SOUTH AFRICA

NATIONAL CERTIFICATE
POWER MACHINES N6
TIME: 3 HOURS
MARKS: 100

NOTE: If you answer more than the required number of questions, only the required number of questions will be marked. All work you do not want to be marked, must be clearly crossed out.

INSTRUCTIONS AND INFORMATION

1. Answer any FIVE questions.
2. Read AL the questions carefully.
3. Questions may be answered in any order, but subsections of questions must be kept together.
4. Number the answers according to the numbering system used in this question paper.
5. ALL formulae used, must be written down.
6. Show ALL the intermediate steps.
7. Write neatly and legibly.

QUESTION 1

The following heat balance was drawn up after a 60 minute test on a single cylinder, 4-stroke oil engine:

Q in fuel (kJ/min)	Q absorbed (*kJ/min*)		%
4 800	Brake power	= 2 680	35
	Friction power	= 420	8,75
	Cooling water	= 1 176	24,5
	Exhaust gases	= 1 320	27,5
	Unaccounted	= 204	4,25
4 800		4 800	100,00

The following information was also noted during the test:

Swept volume of engine	= 0,02 $m^3 / cycle$
Total number of revolutions	= 21 000
Length of indicator diagram	= 76 mm
Spring number	= 80 kPa/mm
Cooling water used	= 350 kg
Calorific value of oil used	= 45 MJ/kg
Temperature rise of exhaust gases	= 382 °C
Length of pistol stroke	= 1,21 x diameter
Specific heat capacity of cooling water	= 4,2 kJ/kg.K
Specific heat capacity of exhaust gases	= 1,045 KJ/kg.K

Calculate:

1.1 The brake power in kW, the indicated power, the mechanical efficiency, the indicated mean effective pressure in kPa and the average area of the indicator diagram in mm^2 (10)

1.2 The temperature rise of the cooling water in °C, the mass of oil used in kg/h and the air-fuel ration (7)

1.3 The diameter of the piston and the length of the stroke in mm. (3)
[20]

QUESTION 2

A velocity compounded, 2-stage, impulse turbine is designed to discharge the gas axially.
The inlet and outlet angles for the fixed blades are 36° and 17° respectively.
The relatively velocity at exit from the second row of moving blades is 1,125 times the blade velocity.
The velocity of flow at the inlet to the first stage is 1,45 times the blade velocity.
The gas leaves the first stage with a relative velocity of 360 m/s.
The velocity coefficient for all the blades is 0,94.
The gas flows at a rate of 40 k/s.

2.1 Construct velocity diagrams for the turbine in the ANSWER BOOK by using a length of 40 mm for the blade velocity.
Indicate the lengths of ALL the lines as well as the magnitude of the angles on the diagrams. Calculate the scale. (10)

2.2 Determine from the velocity diagrams:

2.2.1 The nozzle angle

2.2.2 The inlet angle of the first row of moving blades

2.2.3 The exit angle of the first row of moving blades

2.2.4 The inlet angle of the second row of moving blades

2.2.5 The exit angle of the second row of moving blades

2.2.6 The blade velocity in m/s

2.2.7 The nozzle velocity in m/s

2.2.8 The velocity of the gas at inlet to the second stage in m/s

2.2.9 The velocity of the gas at exit from the turbine in m/s

2.2.10 The tangential force developed in the turbine in kN

2.2.11 The power developed by the turbine in kW

(10 x 2) (10)
[20]

QUESTION 3

A gas enters a convergent-divergent nozzle at a pressure of 2 030 kPa, a temperature of 627 °C and a velocity of zero and expands to a pressure of 598,9 kPa, at the exit.
The temperature loss in the convergent part is 150 °C.
The area at the throat is 1553,434 mm^2.
The Mach number for the nozzle is 1.3.
Take Cp for the gas as 1,005 kJ/kg.k.

Calculate:

3.1 The velocity in m/s and the absolute temperature at the throat of the nozzle, the value of gamma, the pressure at the throat in kPa, the value of the characteristic gas constant, the specific volume at the throat and the rate at which the gas flows through the nozzle in kg/s. (13)

3.2 The velocity in m/s, the actual absolute temperature, the adiabatic absolute temperature at the exit and the efficiency in the divergent part of the nozzle. (7)
[20]

QUESTION 4

The following readings were taken during a test on a constant pressure, open-cycle gas turbine plant:

Air temperature at compressor inlet	=	29 °C
Ratio of pressures for the turbine	=	6:1
Air temperature at combustion chamber outlet	=	677 °C
Air temperature at plant outlet	=	302,6 °C
Mass of air circulating	=	20 kg/s
Adiabatic efficiency of the compressor	=	84,5%
Adiabatic efficiency of the turbine	=	82,6%
Efficiency of the heat exchanger	=	86%
Specific heat capacity of air	=	1,005 $kJ / kg. K$
Gamma	=	1,4

Calculate:

4.1 The adiabatic and actual absolute temperatures at the compressor outlet and the work done on the compressor in kW. (7)

4.2 The adiabatic and actual absolute temperatures at the turbine outlet and work done by the turbine in kW. (6)

4.3 The absolute temperature of the air at the combustion chamber inlet (3)

4.4 The heat energy supplied in the combustion chamber in kJ/s. (2)

4.5 The thermal efficiency of the plant. (2)
[20]

QUESTION 5

A single acting, 3-stage, reciprocating compressor delivers 15 kg of air per minute to an after cooler at a pressure of 6 720 kPa.
Intercooling is perfect and the stages form a geometric progression.
The compressor sucks air into the low-pressure cylinder at 105 kPa at a rate of 12,6 m^3 / min.
Take R for air as 0.288 kJ kg. K and Cp as 1,005 kJ kg. K.
The index for compression and expansion is 1,31.

Calculate:

5.1 The power to drive the compressor in kW. (3)

5.2 The ratio of pressure for the stage, the absolute temperature at inlet to the low pressure cylinder, the absolute delivery temperature, the heat extracted per intercooler in kW and the heat extracted by the water jackets in each cylinder in kW. (11)

5.3 The effective swept volume of the low-pressure cylinder, the effective swept volume of the intermediate cylinder and the effective swept volume of the high pressure cylinder in m^3 if the rotational frequency of the compressor is 240 r/min. (6)

[20]

QUESTION 6

In order to improve the efficiency of a boiler plant, an air-preheater was fitted.

During tests on the plant the following was noted:

The enthalpy of the generated steam increased from 2 705 kJ/kg to 2 730 kJ due to a slight increase in temperature.
The air supplied per kg of fuel decreased by 0,5 kg.
The feed water temperature remained unchanged at 78.7 °C
The rate of evaporation increased from 39 312 kg/h.
The mass of the fuel burnt per hour remained unchanged at 42 000 kg.
The calorific value of the fuel burnt remained at 28.5 MJ/kg.
The temperature at the chimney base was reduced from 250 °C to 160 °C.
The specific heat capacity for combustion moisture at 160 °C is 2,06 kJ/kg.K.
The heat lost to the moisture remained unchanged.
The boiler room temperature remained unchanged at 25 °C.
The pressure of the gas at the chimney base remained unchanged at 150 kPa.
The mass of the combustion moisture remained unchanged.
The heat lost to the moisture in the flues and the dry flues before fitting the air preheater was 1 577,4 kJ fuel and 4 702,5 kJ/kg fuel respectively.
The specific heat capacity of the dry flues remained at 1,045 kJ/kg.K.
The specific heat capacity of water remained at 4,2 kJ/kg.K.
There was a 2% increase in efficiency after the preheater was fitted.

Calculate by using steam tables only:

6.1 The mass of the air moisture in the flues per kg of fuel and the heat lost to the moisture in the flues per kg of fuel after fitting the air-preheater. (6)

6.2 The mass of the air in the flues per kg of fuel before fitting the air-preheater and the heat lost. (7)

6.3 The thermal efficiency of the plant before fitting the air-preheater and the steam generated in kg/h after the air-preheater was fitted. (5)

6.4 The equivalent evaporation from and at 100 °C of the improved plant. (2)
[20]

QUESTION 7

Carbon dioxide is used as a refrigerant in a vapour compression refrigeration plant.
The plant operates between limits of –12 °C and 24 °C.
Before compression the refrigerant is a wet vapour and at the end of compression it is a dry saturated vapour.
The saturated liquid refrigerant leaves the condenser at a temperature of 20 °C.
The entropy of the refrigerant after throttling is 0,538 kJ/kg.K.
The cooling water temperature rise is 16,6 °C.
The specific heat capacity of the water is 4,2 kJ/kg.K.

The following are extracts from the carbon dioxide table:

SATURATION TEMPERATURE	**SPECIFIC ENTHALPY**		**SPECIFIC ENTROPY**	
(°C)	Liquid *(hf)*	Vapour *(hg)*	Liquid *(sf)*	Vapour *(sg)*
–12	56,62	322,66	0,222	1,241
24	155,58	282.58	0,560	0,986

Calculate:

7.1 The dryness factor of the refrigerant at the compressor inlet, the specific enthalpy at the inlet and the work done by the compressor in kJ/kg. (7)

7.2 The dryness factor of the refrigerant after throttling, the specific enthalpy after throttling and the specific heat capacity of the saturated liquid refrigerant. (7)

7.3 The refrigeration effect in kJ/kg and the actual coefficient of performance. (3)

7.4 The mass of cooling water requires by the condenser in kg per refrigerant. (3)
[20]

TOTAL: 100

FORMULA SHEET

$P_aV_a = mRT_a$

$R = Cp - Cv$

$\gamma = \dfrac{Cp}{Cv}$

$PV = c \quad PV = k$

$PV^n = c \quad PV^n = k$

$PV^\gamma = c \quad PV^\gamma = k$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$\Delta U = m \,.\, Cv \,.\, \Delta T$

$Q = \Delta U + Wd \quad Q = \Delta U + Av$

$$\Delta s = m\left(Cv \,.\, \ln\frac{P_2}{P_1} + Cp \,.\, \ln\frac{V_2}{V_1}\right)$$

$$\Delta s = m \,.\, Cv \,.\, \ln\frac{P_2}{P_1}$$

$$\Delta s = m \,.\, Cp \,.\, \ln\frac{V_2}{V_1}$$

$$\Delta s = m \,.\, R \,.\, \ln\frac{P_1}{P_2}$$

$Q = m \,.\, Cp \,.\, \Delta T$

$Q = m \,.\, Cv \,.\, \Delta T$

$$S_{su} = S_g + Cp \,.\, \ln\frac{T_{su}}{T_s}$$

$S_{fg} = S_g - S_f$

$S = S_f + xS_{fg}$

$h_{su} = h_g + Cp\,(t_{su} - t_s)$

$$h_{ws} = h_f + xh_{fg} \qquad V_{su} = \frac{\frac{n-1}{n}(h_{su} - 1941)}{P_{su}} \qquad h_{ns} = h_f + xh_{fg}$$

$$V_{ws} = xV_g \qquad r = \frac{V_s + V_c}{V_c} \qquad V_{ns} = xV_g$$

$$V_s = \frac{\pi}{4} d^2 \times L$$

$$P_2 = \sqrt{P_1 \times P_3}$$

$$r_{ps} = \sqrt[x]{\frac{P_{x+1}}{P_1}}$$