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Department:
Higher Education and Training
REPUBLIC OF SOUTH AFRICA

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NATIONAL CERTIFICATE

POWER MACHINES N6

(8190046)

6 April (X-Paper)
09:00 – 12:00

REQUIREMENTS: Steam tables

Calculators may be used.

This question paper consists of 7 pages and a 6- page 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 FIVE questions, only the first five 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. Questions may be answered in any order, but subsections of questions must be kept together.
3. ALL formulae used, must be written down.
4. Show ALL the intermediate steps.
5. Questions must be answered in BLUE or BLACK ink.
6. ALL the sketches and diagrams must be done in pencil in the ANSWER BOOK.
7. Number the answers correctly according to the numbering system used in this question paper.
8. Write neatly and legibly.

QUESTION 1

Steam expands through a convergent-divergent nozzle at a rate of 300 kg/min to the exit where the isentropic dryness factor is 0,94 and the diameter is 72,2 mm. The specific volume of dry saturated steam at the exit pressure is 0,668 4 m^3/kg. The superheated steam at the inlet has a pressure of 1 500 kPa, a temperature of 250 °C and the velocity is negligible. At the throat the superheated steam has a pressure of 820 kPa, a velocity of 500 m/s and the specific heat capacity is 2,56 kJ/kg.K with an index (n) of 1,31. The isentropic dryness factor is 98,95% of the actual dryness factor.

Calculate the following by using steam tables only:

1.1 The specific enthalpy of the superheated steam and the normal temperature at the throat (5)

1.2 The specific volume, the area in mm^2 and the diameter in mm at the throat (6)

1.3 The actual dryness factor, the specific volume, the area in mm^2, the velocity in m/s and the specific actual enthalpy at the exit (9)

[20]

QUESTION 2

A two-stage, single-acting reciprocating compressor delivers 12 kg of air per minute at 220 r/min. The initial pressure and temperature for the low-pressure cylinder are 96 kPa and 24 °C respectively. The temperature of the air after compression in the low pressure cylinder is 142,633 °C. The temperature of the air leaving the intercooler is 62 °C. The pressure ratio for the low pressure cylinder is 4:1 and for the high pressure cylinder it is 3,5:1. The diameter of the low pressure cylinder is 390 mm and the stroke length is 452 mm. Take R for air as 0,288 kJ/kg.K.

Calculate the following:

2.1 The index (n) for compression (4)

2.2 The power required to drive the compressor in kW (4)

2.3 The effective swept volume in m^3/stroke, the swept volume in m^3/stroke and the volumetric efficiency of the low pressure cylinder (7)

2.4 The effective swept volume and the swept volume of the high pressure cylinder in m^3/stroke if it has the same volumetric efficiency as the low pressure cylinder (5)

[20]

QUESTION 3

An air pre-heater was fitted to a boiler plant to improve its efficiency.
During tests on the plant the following information was noted:

The superheated steam generated, increased from 9 kg/kg to 9,2 kg/kg of fuel used.
The temperature at the chimney base decreased from 200 °C to 150 °C.
The air supplied per kg of fuel decreased by 0,5 kg.
The fuel used is 98% combustible.
The mass of the combustion moisture formed remained unchanged at 0,58 kg/kg of fuel.
The pressure of the gas at the chimney base remained unchanged at 150 kPa.
The feed water temperature remained unchanged at 95,2 °C.
The enthalpy of the steam generated increased by 9 kJ/kg due to a slight increase in temperature.
The calorific value of the fuel remained unchanged at 30,46 MJ/kg.
The boiler room temperature remained unchanged at 20 °C.
The heat carried away by the dry flues before the pre-heater was fitted was 3 762 kJ/kg fuel.
The efficiency of the plant before fitting the pre-heater was 78,004%.
The specific heat capacity of water was 4,2 kJ/kg.K.
The specific heat capacity of the dry flues was 1,045 kJ/kg.K.

Calculate the following by using steam tables only:

3.1 The heat lost to the moisture in the flues per kg of fuel before and after fitting the pre-heater (5)

3.2 The mass of air used per kg of fuel before fitting the pre-heater and the heat carried away by the dry flues after fitting the pre-heater (6)

3.3 The specific enthalpy of the superheated steam before fitting the pre-heater, the efficiency of the plant after fitting the pre-heater and the percentage improvement (5)

3.4 Draw up a heat balance in kJ/kg and as a percentage for the plant after the pre-heater was fitted, to determine the percentage heat unaccounted for (4)

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QUESTION 4

A single cylinder engine operating on the diesel cycle principle has a piston with a diameter of 172,31 mm and the stroke length is 1,12 times the diameter of the piston.
The volumetric compression ratio is 16:1.
The initial pressure is 110 kPa.
The compression and expansion index (n) is 1,32.
The volume after combustion is 14% of the cylinder volume.
The change in temperature during compression is 521,4 °C.
The change in temperature during combustion is 1 150 °C.
The change in temperature during exhaust is 730 °C.

Take R for air as 0,288 kJ/kg.K and Cv as 0,718 kJ/kg.K.

Calculate the following:

4.1 The swept volume, clearance volume, cylinder volume and the volume after combustion in m^3 (7)

4.2 The missing pressures at the principal points in kPa (3)

4.3 The heat received and the heat rejected in kJ/kg gas and the air standard efficiency (7)

4.4 The heat flow through the cylinder wall during compression in kJ/kg gas (3)

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QUESTION 5

The following readings were taken during a test on an open circuit, continuous combustion, constant pressure gas turbine plant:

Air temperature at compressor inlet	=	16 °C
Air temperature at compressor outlet	=	240 °C
Air temperature at combustion chamber inlet	=	400 °C
Temperature of combustion products at turbine inlet	=	840 °C
Temperature of combustion products at turbine outlet	=	483,25 °C
The temperature at exit from the plant	=	233,25 °C
The pressure ratio for both compressor and turbine	=	6:1
The mass flow rate of the air is	=	5 kg/s
The specific heat capacity of air	=	1,008 kJ/kg.K
The value of gamma	=	1,4

Ignore the mass of the fuel and calculate the following:

5.1 The power to compress the air and the power developed by the turbine in kW (4)

5.2 The energy received from the combustion chamber in kW (2)

5.3	The thermal efficiency of the plant	(3)
5.4	The efficiency of the heat exchanger	(3)
5.5	The absolute isentropic temperature and the isentropic efficiency of the compressor	(4)
5.6	The absolute isentropic temperature and the isentropic efficiency of the turbine	(4) **[20]**

QUESTION 6

No axial thrust was developed in the first or second stage of a velocity compounded, two-stage, impulse turbine.

The steam leaves the moving blades of the second stage at an angle of 30° to the rotation of the blades.
The inlet angle of the fixed blades is 35°.
The average blade velocity is 152 m/s.
The velocity of flow at the inlet to the second stage is 90 m/s.
The velocity coefficient for all the blades is 0,96.

6.1 Use scale 1 mm = 4 m/s and construct velocity diagrams for the turbine in the ANSWER BOOK. Indicate the lengths of ALL the lines as well as the magnitude of the angles on the diagrams. (10)

6.2 Determine the following from the velocity diagrams:

6.2.1 The nozzle velocity in m/s
6.2.2 The velocity of the steam entering the fixed blades in m/s
6.2.3 The velocity of the steam leaving the fixed blades in m/s
6.2.4 The inlet angle of the first row of moving blades
6.2.5 The exit angle of the first row of moving blades
6.2.6 The inlet angle of the second row of moving blades
6.2.7 The nozzle angle
6.2.8 The outlet angle of the fixed blades
6.2.9 The diagram efficiency
6.2.10 The power developed in the turbine in kJ/kg steam (10)

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QUESTION 7

A carbon dioxide refrigeration plant operates between temperature limits of -4 °C and 28 °C. The refrigerant enters the compressor as a wet vapour and after compression it has a temperature of 54,9 °C. The condenser undercools the refrigerant to a saturated liquid with a temperature of 23 °C. The specific heat capacity of the superheated refrigerant is 2,5 kJ/kg.K and that of the saturated liquid refrigerant is 5,42 kJ/kg.K.

The following are extracts from carbon dioxide tables.

Saturation temperature(°C)	Specific enthalpy (kJ/kg)		Specific entropy (kJ/kg.K)	
	Liquid (hf)	Vapour (hg)	Liquid (sf)	Vapour (sg)
-4	74,3	320	0,286	1,2
28	178,4	263,8	0,638	0,922

Calculate the following:

7.1 The specific entropy after compression and the dryness factor of the refrigerant after the evaporator (5)

7.2 The specific enthalpy of the refrigerant at inlet to the compressor and the specific enthalpy after compression (6)

7.3 The specific enthalpy of the refrigerant before throttling, the refrigerating effect in kJ/kg, the work done in kJ/kg and the actual coefficient of performance (7)

7.4 The power required to drive the compressor in kW if the mechanical efficiency is 82% and the refrigerant flows at a rate of 18 kg/min (2)

HINT: Entropy after compression = Entropy of dry saturated vapour at 28 °C + Cp. ln (T after compression ÷ T in condenser)

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TOTAL: 100

FORMULA SHEET

Any applicable formula may also be used.

ENGLISH	*GENERAL*	*AFRIKAANS*
	$P_a V_a = mRT_a$	
	$R = Cp - Cv$	
	$\gamma = \dfrac{Cp}{Cv}$	
$PV = c$		$PV = k$
$PV^n = c$		$PV^n = k$
$PV^\gamma = c$		$PV^\gamma = k$
	$\dfrac{T_2}{T_1} = \left(\dfrac{V_1}{V_2}\right)^{n-1} = \left(\dfrac{P_2}{P_1}\right)^{\frac{n-1}{n}}$	
	$\Delta U = m \,.\, Cv \,.\, \Delta T$	
$Q = \Delta U + Wd$		$Q = \Delta U + Av$

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

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

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

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

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

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

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

$$S_{fg} = S_g - S_f$$

$$S = S_f + xS_{fg}$$

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

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ENGLISH	*GENERAL*	*AFRIKAANS*
$h_{ws} = h_f + xh_{fg}$	$V_{su} = \dfrac{\frac{n-1}{n}(h_{su} - 1941)}{P_{su}}$	$h_{ns} = h_f + xh_{fg}$
$V_{ws} = xV_g$	$r = \dfrac{V_s + V_c}{V_c}$	$V_{ns} = xV_g$
	$V_s = \dfrac{\pi}{4} d^2 \times L$	
	$P_2 = \sqrt{P_1 \times P_3}$	
	$r_{ps} = \sqrt[x]{\dfrac{P_{x+1}}{P_1}}$	

Different formulae for work done (Wd)

Verskillende formules vir arbeid verrig (Av)

$$= P \times \Delta V$$

$$= P_1 V_1 \ln \frac{V_2}{V_1}$$

$$= \frac{P_1 V_1 - P_2 V_2}{n - 1}$$

$$= \frac{P_1 V_1 - P_2 V_2}{\gamma - 1}$$

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

$$= \frac{xn}{n-1} P_1 V_e \left[\left(\frac{P_{x+1}}{P_1} \right)^{\frac{n-1}{xn}} - 1 \right]$$

$$= \frac{xn}{n-1} mRT_1 \left[(r_{ps})^{\frac{n-1}{n}} - 1 \right]$$

ENGLISH | *GENERAL* | *AFRIKAANS*

Different formulae for work done (Wd)

= *area of PV-diagram*

= *work done first stage + work done second stage + ...*

Verskillende formules vir arbeid verrig (Av)

= *area van PV-diagram*

= *arbeid verrig eerste stadium + arbeid verrig tweede stadium + ...*

$$Wd_{nett} = Wd_t - Wd_c$$

$$Wd_{nett} = Q_{nett}$$

$$Av_{nett} = Av_t - Av_k$$

$$Av_{nett} = Q_{nett}$$

Different formulae for air standard efficiencies (ASE)

Verskillende formules vir lugstandaardrendemente (LSR)

$$= 1 - \left(\frac{1}{r}\right)^{\gamma - 1}$$

$$= 1 - \frac{r_p r_c^{\gamma - 1}}{r_v^{\gamma - 1}\,[(r_p - 1) + \gamma^{r_p}\,(r_c - 1)]}$$

$$= \frac{\text{heat added} - \text{heat rejected}}{\text{heat added}} = 1 - \frac{\beta^{\gamma} - 1}{r^{\gamma - 1} \times \gamma\,(\beta - 1)} = \frac{\text{warmte toegevoeg} - \text{warmte afgestaan}}{\text{warmte toegevoeg}}$$

Different volumetric efficiencies, θ_{vol}

$$= \frac{\text{Volume of air taken in}}{\text{Swept volume}}$$

$$= \frac{\text{Volume of free air}}{\text{Swept volume}}$$

Verskillende volumetriese rendemente, θ_{vol}

$$= \frac{\text{Volume lug ingeneem}}{\text{Slagvolume}}$$

$$= \frac{\text{Volume vrylug}}{\text{Slagvolume}}$$

$$= 1 - \frac{V_c}{V_s}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]$$

ENGLISH	*GENERAL*	*AFRIKAANS*
Different thermal efficiencies, $0_{therm.} = \dfrac{Wd}{heat\ supplied}$		*Verskillende termiese rendemente,* $0_{term.} = \dfrac{Av}{warmte\ toegevoeg}$
$\eta_{brake\ therm.} = \dfrac{BP}{m_{f/s} \times CV}$		$\eta_{rem\ term.} = \dfrac{RD}{m_{b/s} \times WW}$
$\eta_{ind.\ therm.} = \dfrac{IP}{m_{f/s} \times CV}$		$\eta_{ind.\ term.} = \dfrac{ID}{m_{b/s} \times WW}$
$\eta_{therm.} = \dfrac{m_s\ (hs - hw)}{m_f \times CV}$		$\eta_{term.} = \dfrac{m_s\ (hs - hw)}{m_b \times WW}$
$\eta_c = \dfrac{T_2' - T_1}{T_2 - T_1}$	$\eta_t = \dfrac{T_3 - T_4}{T_3 - T_4'}$	$\eta_k = \dfrac{T_2' - T_1}{T_2 - T_1}$
$\eta_{mech.} = \dfrac{BP}{IP}$		$\eta_{meg.} = \dfrac{RD}{ID}$
Indicated efficiency ratio $= \dfrac{\eta_{ind.\ therm.}}{ASE}$		*Indikateurrendementverhouding* $= \dfrac{\eta_{ind.\ term.}}{LSR}$
Brake efficiency ratio $= \dfrac{\eta_{brake\ therm.}}{ASE}$		*Remrendementverhouding* $= \dfrac{\eta_{rem.\ term.}}{LSR}$
$BP = 2\pi \dfrac{TN}{60}$	$T = F \times r$	$RD = 2\pi \dfrac{TN}{60}$
$BP = P_{brake\ mean}\ LANE$		$RD = P_{rem\ gem.}\ LANE$
$IP = P_{ind.\ mean}\ LANE$		$ID = P_{ind.\ gem.}\ LANE$
$ISFC = \dfrac{m_{f/h}}{IP}$		$ISBV = \dfrac{m_{b/h}}{ID}$
$BSFC = \dfrac{m_{f/h}}{BP}$		$RSBV = \dfrac{m_{b/h}}{RD}$
$COP = \dfrac{T_1}{T_2 - T_1}$		$KVW = \dfrac{T_1}{T_2 - T_1}$
$COP = \dfrac{RE}{Wd}$		$KVW = \dfrac{VE}{Av}$
$P = m\ .\ U\ .\ \Delta Vw$		$D = m\ .\ U\ .\ \Delta Vw$
$F_{ax} = m\ .\ \Delta V_f$		$F_{aks.} = m\ .\ \Delta V_f$

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ENGLISH *GENERAL* *AFRIKAANS*

$$\eta_{dia.} = \frac{2 \,.\, U \,.\, \Delta Vw}{V_1^2}$$

$$P_c = P_1\left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}}$$

$$T_c = T_1\left(\frac{2}{\gamma + 1}\right)$$

$$C_c = \sqrt{2 \times 10^3 \ (h_1 - h_c) + C_1^2}$$

$$C_2 = \sqrt{2 \times 10^3 \ (h_1 - h_2) + C_1^2}$$

$$C_c = \sqrt{2 \times 10^3 \times C_p \ (T_1 - T_c) + C_1^2}$$

$$C_2 = \sqrt{2 \times 10^3 \times C_p \ (T_1 - T_2) + C_1^2}$$

$$A_c = \frac{mV_c}{C_c} \qquad A_2 = \frac{mV_2}{C_2}$$

$$\eta = \frac{h_1 - h_c}{h_1 - h_c^{'}} \qquad \eta = \frac{T_1 - T_c}{T_1 - T_c^{'}}$$

$$\eta = \frac{h_c - h_2}{h_c - h_2^{'}} \qquad \eta = \frac{T_c - T_2}{T_c - T_2^{'}}$$

$$\eta = \frac{h_1 - h_2}{h_1 - h_2^{'}} \qquad \eta = \frac{T_1 - T_2}{T_1 - T_2^{'}}$$

$$EE = \frac{m_s \ (h_s - h_w)}{m_f \times 2\,257} \qquad EV = \frac{m_s \ (h_s - h_w)}{m_b \times 2\,257}$$

$$\eta_{iso.} = \frac{Wd_{iso.}}{Wd_{poly.}} \qquad \eta_{iso.} = \frac{Av_{iso.}}{Av_{poli.}}$$

$$\eta_{rank.} = \frac{Wd}{Q} \qquad \eta_{rank.} = \frac{Av}{Q}$$

$$\eta_{carn.} = 1 - \frac{T_2}{T_1}$$

$$h = u + pV$$

$$gZ_1 + U_1 + P_1V_1 + \frac{C_1^2}{2} + Q =$$

$$gZ_2 + U_2 + P_2V_2 + \frac{C_2^2}{2} + Wd$$

$$gZ_1 + U_1 + P_1V_1 + \frac{C_1^2}{2} + Q =$$

$$gZ_2 + U_2 + P_2V_2 + \frac{C_2^2}{2} + Av$$