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Design of A Generator Driven Steam Turbine with A Nominal Power of 10 MW For Industrial Electricity Needs

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ABSTRACT

Steam turbine is one type of prime mover that is widely used in industry, including: as a prime mover for electric generators, pumps and compressors, as well as process industries. The things that are considered in the use of the steam turbine are the heat source used to evaporate water can be from gas, liquid, and solid fuels, the efficiency of the steam turbine is greater than the gas turbine. The method used in writing this thesis is as follows: (a)) Field survey, namely in the form of direct observation to the location where the generating unit is located. (b) Literature study, namely in the form of literature study, study of books, and related writings. (c) Discussion, which is in the form of questions and answers with the supervisor, The comparison lecturer who will be appointed by the Department of Cylindrical Mechanical Engineering for small and medium capacity turbines is usually made of gray cast iron JIS G 5501 FC30 with a tensile stress of b 27 kg/mm2 or 2700 kg/cm2 and a safety factor value of k3 (taken) so : b permit 4000/3 1333,33kg/cm2 thus : b permit t , then this construction is safe. From the calculations performed, some conclusions can be drawn in the design of the steam turbine driving the generator: 6.1. Specifications of steam turbine 1. Inlet steam pressure : 42 bar 2. Turbine inlet steam temperature : 480 0 C 3. Turbine outlet steam pressure : 0.1 bar 4. Turbine level : 10 level 5. Extraction amount : 1 level 6. Flow rate steam mass : 12,163 kg/s 7. Turbine power : 12,47 MW 8. Turbine Shaft rotation : 5700 rpm 6.2. Dimensions of the main parts of the turbine: 1.

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1. INTRODUCTION

Steam turbine is one type of prime mover that is widely used in industry, including: (Syarif, 2009)as the prime mover of electric generators, pumps and compressors, as well as the process industry (Manurung, 2009). The things that are considered in the use of the steam turbine are the heat source used to evaporate water can be from gas, liquid, and solid fuels, the efficiency of the steam turbine is greater than that of the gas turbine, and easy maintenance (Dwitama, n.d.). Energy is a very important element in the overall development of a country (Kadir, 2006). Utilization of energy appropriately will be a powerful tool to stimulate the level of the economy (Widiati, 2019).One form of energy that is most needed by humans today is electrical energy, humans need electrical energy for households, industrial sectors, transportation and others. (Putra et al., 2020) Large and continuous electrical energy is not naturally available in nature, therefore a power system is needed to convert energy from one form into electrical energy (Siagian & Fahreza, 2020). The steam turbine is a part of the power system and is also referred to as an energy conversion machine and has a good alternative because it can produce electrical energy with large enough power and high efficiency (Adam et al., 2019).

This plan is intended to plan a power plant with a steam turbine as a driving force for an electric generator with a nominal power (Prayuda, 2014) 10 MW generator at 5700 rpm rotation (turbine rotation), the steam pressure entering the turbine is 42 bar at a temperature of 480^{0}C.

2. RESEARCH METHOD

The method used in writing this thesis is as follows: (a) Field survey, which is in the form of direct observation to the location where the generating unit is located (Silitonga, 2020) (b) Literature studies, namely in the form of literature studies, studies of books, and related writings (Sofiah et al., 2020). (c) Discussion, which is in the form of questions and answers with supervisors, comparison lecturers who will later be appointed by the parties (Sari, n.d.). Department of Mechanical Engineering – FT USU regarding the shortcomings in this undergraduate thesis writing (Akbar & Kurniasari, 2019).

3. RESULTS AND DISCUSSIONS

3.1. Shaft Size Calculation

In this plan, the shaft has a function as a connector that transfers power and turbine rotation as well as a place for mounting discs and blades. In this case the load that will be experienced by this shaft is(Dahlan et al., 2018): (1) Bending load originating from the weight of the blades and discs. (2) The torsional load originating from the disc(Riyadin, 2010). In designing the shaft in terms of mechanical strength, the stresses in the cross-section are taken as the basis for calculations, which include: (1) The cross-section with the greatest bending moment(Rahman et al., 2021). (2) The section with the maximum torsional moment. For the shaft of medium rotation and heavy loads used alloy steel with hardening of the skin. For this purpose, the shaft material is nickel chrome steel JIS 4102 SNC 21 which has a tensile strength of 80 kg/mm2. The allowable shear stress for the shaft material can be calculated based on the equation
a = b / Sf1 x Sf2 where:
Sf1 = factor of safety due to shaft weight (for alloy steel = 6)
Sf2 = factor of safety due to the presence of pegs, terraced shafts, and
stress concentration (1.3 3,0), is taken as 2.7
So the allowable shear stress for the shaft material is:

$$\tau_a = \frac{80kg/mm^2}{6\times 2{,}7} \tag{1}$$

a = 4.94 kg/mm2
The nominal power transmitted in this design is 10,000 kW at 5700 rpm. The magnitude of the torsional moment (T) can be calculated by the equation
T = 9.74 . 105$\frac{N}{n}$

$$T = 9{,}74 \, . \, 10^5 \, \frac{10000kW}{5700rpm}$$

H = 1708771.93 kg.mm
Shaft diameter ds is calculated by the equation:

$$ds = \left[\frac{5{,}1}{\tau_a}\times K_t \times C_b \times M_t\right]^{1/3} \tag{2}$$

where:

Kt = loading factor (1.5 3,0), for shock load and
the large collision is taken 2.6
Cb = flexural loading factor (1,2 2,3) (taken 2,2)

$$ds = \left[\frac{5{,}1}{4{,}94} \times 2{,}6 \times 2{,}2 \times 1708771{,}930\right]^{1/3} = 216 \text{ mm}$$

From the existing standard shaft, the smallest diameter of the shaft used in this plan is 220 mm, while for the terraced shaft, 224 mm is chosen.

3.2. Calculation of the Size of the Nozzle and Blades

The nozzle is a passage that varies in cross section where at the nozzle the potential energy of the steam is converted into kinetic energy in the form of steam emission to the turbine blades. From theoretical and experimental investigations, it turns out that the steam flowing through the nozzle section with a convergent cross section when expanding therein only reaches a certain minimum value called the critical pressure (pkr) which is equal to 0.577 Po for saturated steam and 0.546 Po for superheated steam. . The steam velocity at this pressure is called the critical velocity. If the pressure after the nozzle is greater than the critical pressure P1 > pkr, then the steam expansion occurs only up to the pressure p1 and the steam velocity at the outlet of this pressure is less than the critical velocity, in this case the nozzle is used. convergent,

3.2.1. Nozzle and Blade Height

The state of the steam in the first stage is superheated steam, so the critical pressure is:
pkr = 0.546 x P0
pkr = 0.546 x 39.9 bar = 21.786 bar
Where the pressure after the nozzle P1 = 22 bar, because P1 is greater than pkr, a convergent nozzle is usedNozzle exit side section

$$f_1 = \frac{G_o}{c_1} \upsilon_1 \ (m^2) \tag{3}$$

where:
G0 = mass of steam flow = 12,163 kg/s
1 = specific volume of steam at the outlet side = 0.2030 m3/kg
C1 = actual velocity of steam at the exit cross section = 689.946 m/s
The height of the nozzle, preferably between 10 mm-20 mm, and the degree of partial entry, is not less than 0.2. for large and medium capacity turbines with relatively large blades, the value of the partial intake degree can be up to one. By making the nozzle height ln 16 mm, the degree of partial entry of steam is obtained: The nozzle material is taken from the same steel as the blade material because from the condition the incoming steam is superheated steam, so the nozzle material chosen is stainless nickel chrome steel AISI UNS NO .41400 with total tensile and flexural stress due to centrifugal force which is 2137 kg/cm2n, so the selection of the above material is safe.

3.2.1.1. Blade Width

The blade width ranges from 20 25 mm for medium and large capacity turbines. In this plan, the blade width is set at 25 mm. The arc radius of the first row blade profile can be calculated by the equation:

3.2.1.2. Level 2 Nozzles and Blades

The strength of the turbine blades is calculated on the weakest parts, and if this part is safe, then the other parts will be safer. The magnitude of the tensile stress due to the radial force which has the largest value, namely at the final stage of the blade (level 10), can be calculated by the equation.

$$\sigma = \frac{\pi^2 n^2 \gamma}{900 \times g}\left[t \times r + \frac{F_s}{F_0} \times t_s \times r_s \right] \tag{4}$$

Where: n = turbine wheel rotation = 5700 rpm
= density of blade material = 0.00785 kg/cm3
l" = height of the moving blade level 10 = 41.7 cm
r = average radius of the blade axis = 92.5/2 = 46.25 cm
rs = average radius of blade reinforcement plate
= r + 0.5 x l"+ 0.5 xs ; (s = sheath thickness = 0.2 cm)
= 46.25 + 0.5 x 41.7 + 0.5 x 0.2 = 67.2 cm
ts = length of each sheath blade

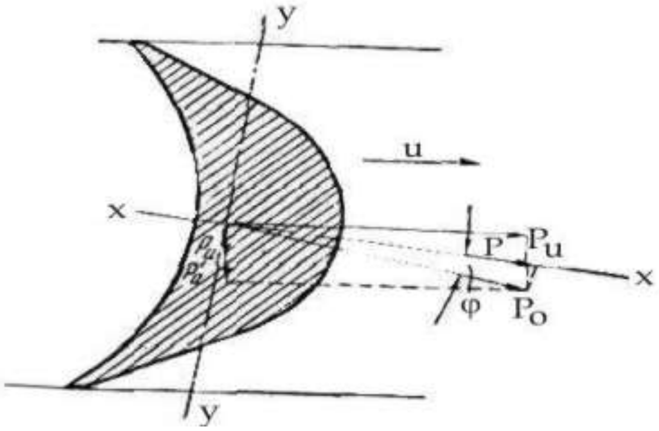

Image 1. Bending forces on the blade[1,292]

The bending stress which has the greatest value occurs along the blade 10, can be calculated by the equation:
b = Mx1/Wy1 (kg/cm2)
where Wy1 = the smallest moment of resistance of the blade relative to yy
= 0.15502 cm3
then : = 21.76/0.15502
= 140.37 kg/cm2
For full intake turbine: b 380 kg/cm2, thus the planned blade construction is safe.

3.3. Blade Check Against Vibration

Vibration that occurs in the turbine is due to the irregularity of the steam flow that comes out of the osel and the guide vane. The dynamic frequency (Fd) of the vibration that occurs can be calculated by the equation:

$$F_d = \sqrt{f_{st}^2 + B.n^2} \qquad \text{(rps)}$$

(5)
with:
Fst = static frequency of blade assembly natural vibration
= 400 rps (for low frequency blades)

B = coefficient that takes into account the influence of rotation

$$\mathrm{B} \quad = 0{,}8\,\frac{Drata - rata}{l''} - 0{,}85 \qquad (6)$$

With: Average = mean diameter of disc = 8029/10= 802.9 mm

$$\text{Maka : B} = 0{,}8\,\frac{802{,}9}{356} - 0{,}85$$

$$\text{B} = 0{,}96$$

$$\text{n} = \text{putaran turbin} = 5700 \text{ rpm} = 95 \text{ rps}$$

$$\text{Maka : Fd} = \sqrt{(400)^2 + 0{,}9526(95)^2}$$

$$= 410{,}6 \text{ rps}$$

The value of Fd has limitations (literature 1, p. 298), namely:

Fd 7n, Fd 7 x 95, Fd 665. Where the value obtained from the calculation is below this limit, the turbine is safe from vibration.

3.4. Discussion on Calculation of Disc Size

3.4.1. Discussion on Calculation of Disc Size

The disc type chosen is the conical disc type because it is suitable for large diameter grades in terms of a more even distribution of stress on the flange. For discs with heavy loads, the radial stress on the radius ro can be taken:

r0 = 100 kg/cm2

The radial stress at the radius r2 due to the centrifugal force of the blades and the rim (rim) is r2 = 1902.96 kg/cm2.

ro = radius in disc= 0.5 dp = 0.5 x 224 = 112 mm

r2 = outer radius of disc= (923/2)-(356/2) = 283.5 mm

r1 = hub radius = r2/2 = 283.5/2 = 141.75 mm

Y1= disc foot thickness = 80 mm (defined)(literature 1 p. 286):

Y = top disc thickness = 20 mm (set)

Y0 = hub thickness = 2.y1= 2 x 80 = 160 mm (defined)

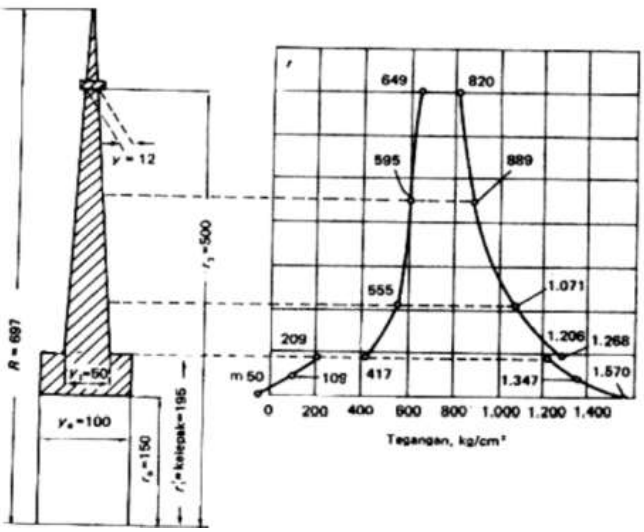

Figure 2. Cross-section of Konis's flap disc[1,311]

The allowable stresses for each case are determined by taking into account the physical properties of the steel and the planned operating temperature of the disc. In general, the allowable stresses are never more than 0.4 times the yield point stress of the material at the design temperature. From the results of the calculation of the stresses on the parts that are important for the planned chakra, the material used is selected from category III where the surrender point is: 75 kg/mm2 (7500 kg/cm2). And the allowable voltage is:
max = t1 0.4 x 7500
t1 = 2284.990 3000 kg/cm2
So that the design of this chakra has fulfilled.

3.5. Bearings and Lubrication

Bearing is the main part of the machine element so that in its selection its role must be considered. The bearings used in this design are sliding bearings, considering that the load experienced is quite large and the rotation is high. The bearings are supplied with lubricating oil which is usually at a pressure of 0.4 to 0.7 atm measurement (gauge). Free space is provided between the shaft and the bearing surface to make room for a layer of lubricating oil. In general, sliding bearings can be described as follows: From the average viscosity of lubricating oil = 0.3 x 10-6 kg.sec/cm2 = 29.418 Mpa (for TZOUT type oil (GOST 32-53) and the temperature T1, T2 can be determined SAE viscosity type

3.6. Turbine House

The turbine stator has such a complex shape, the exact calculation of the cylinder wall will be very difficult. Ignoring the effects of sidewalls, reinforcing ribs, flanges, variations in pressure and temperature according to length, etc., we can assume the cylinder is in the form of a drum. In this case the forces acting on the stator walls can be expressed by the formula:

$$\sigma_t = \frac{D \cdot P}{2\delta} \tag{7}$$

where :
D = cylinder inner diameter = 130 cm
P = Measuring pressure of steam gauge inlet nozzle = 39.9 kg/cm2
= cylindrical wall thickness, set 4.5 cm
so :

$$\sigma_t = \frac{(130) \times (39,9)}{2(4,5)}$$

= 576.33 kg/cm2
Cylinders for small and medium capacity turbines are usually made of gray cast iron JIS G 5501 FC30 with a tensile stress b = 27 kg/mm2 or 2700 kg/cm2 and a safety factor value of k = 3 (taken) so that: b permit = 4000/3 = 1333,33kg/cm2 thus: b permit > t , then this construction is safe.

4. CONCLUSION

From the calculations carried out, several conclusions can be drawn in the design of the steam turbine driving the generator: Steam turbine specifications (1) Steam inlet pressure: 42 bar. (2) Turbine inlet steam temperature : 480 0C (3) Turbine outlet steam pressure : 0.1 bar (4) Turbine level : 10 stages (5) Extraction amount : 1 stage (6) Steam mass flow rate : 12,163 kg/s (7) Turbine power : 12.47 MW (8) Turbine Shaft Rotation : 5700 rpm (6.2.) Dimensions of main parts of turbine: (1)Shaft (1) Diameter : 240 mm (2) Material : JIS 4102 SNC 21.
(2) Nozzles

Tingkat	Jenis	Jumlah	Tinggi (mm)	Lebar(mm)
Tingkat I	Konvergen	44	16	5,48
Tingkat II	Konvergen	87	22	7,66
Tingkat 2	Konvergen	130	22	3,7
Tingkat 3	Konvergen	137	21	3,8
Tingkat 4	Konvergen	139	31	3,8
Tingkat 5	Konvergen	159	46	3,5
Tingkat 6	Konvergen	174	55	3,3
Tingkat 7	Konvergen	196	77	3,3
Tingkat 8	Konvergen	216	104	3,6
Tingkat 9	Konvergen	223	174	3,8
Tingkat 10	Konvergen	216	328	4,1

(3)Movement angle

Tingkat	jumlah	Tinggi sisi masuk(mm)	Tinggi sisi keluar(mm)	Lebar (mm)	bahan
Tingkat I	124	18	20,97	25	AISI UNS S41400
Tingkat II	133	25	34	25	AISI UNS S41400
Tingkat 2	129	16,02	19	25	AISI UNS S41400
Tingkat 3	136	21,31	26	25	AISI UNS S41400
Tingkat 4	139	31,13	37	25	AISI UNS S41400
Tingkat 5	159	46,50	54	25	AISI UNS S41400
Tingkat 6	173	55,3	62	25	AISI UNS S41400
Tingkat 7	196	77	84	25	AISI UNS S41400
Tingkat 8	215	104,58	110	25	AISI UNS S41400
Tingkat 9	223	173,98	181	25	AISI UNS S41400
Tingkat 10	216	327,68	356	25	AISI UNS S41400

(4) Disc

Tingkat	Lebar hub (cm)	Lebar cakram (mm)	Jari-jari hub (cm)	Jari-jari luar cakram (cm)	Bahan
Tingkat I	3	1,5	15,02	30,50	AISI UNS S41400
Tingkat II	3	1,5	15,02	30,50	AISI UNS S41400
Tingkat 2	3	1,5	17,51	35,02	AISI UNS S41400
Tingkat 3	3	1,5	17,66	35,33	AISI UNS S41400
Tingkat 4	3	1,5	17,75	35,51	AISI UNS S41400
Tingkat 5	3	1,5	18,18	36,37	AISI UNS S41400
Tingkat 6	3	1,5	18,25	36,51	AISI UNS S41400
Tingkat 7	3	1,5	19,01	38,02	AISI UNS S41400
Tingkat 8	4	2	19,01	38,02	AISI UNS S41400
Tingkat 9	6	3	19,34	38,69	AISI UNS S41400
Tingkat 10	16	8	19,58	39,17	AISI UNS S41400

(5) Bearing and Lubrication
Type : Gliding Bearing
Inner Diameter: 220 mm
Length : 120mm
Lubricating oil : TZOUT (GOST 32-53)
Viscosity = 0.3 .10-6 kg.s/cm
Lubricant flow capacity : qγ = 3.08 liter/second

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