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Design and Build of Animal Feed Dryer Machine Heat Pump System with 1 Pk. Power

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ABSTRACT

The need for animal feed in Indonesia is very high considering that there are so many livestock commodities in Indonesia. The number of farms greatly affects the need for feed that will be ready to be eaten by livestock, while the animal feed produced by the industry is still wet or moist. The planned data will be collected and further analyzed in this study include the following:) Animal Feed Mass (M). The mass of the feed was measured when it was wet (Mb) and when it was dry (Mk). (2) Drying time (t). The drying time needed to dry the feed is from wet to dry (wet weight to dry weight). At what time of the fall. (3) Temperature (T). Based on data analysis and discussion, the following conclusions can be drawn: 1. The compressor used in this heat pump system clothes dryer is a Rotary Vane Compressor or sliding blade compressor. With engine specifications: - Condenser Working Pressure (PK) 3.1 Mpa - Evaporator Working Pressure (Pe) 0.62 Mpa -Compressor Power (Wc) 1 Hp 746 Watt - Condenser Temperature (Tk) 72.125 0C - Evaporator Temperature (Te) 7 0C The performance of the refrigeration cycle is 2.90. A high coefficient of performance is desirable because it indicates that a certain amount of refrigeration work requires only a small amount of work and the drying process. From thermodynamic calculations based on engine specifications, the power of the electric motor driving the compressor is 1.03 kW with a compression efficiency of 0.99.

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1. INTRODUCTION

The need for animal feed in Indonesia is very high considering that there are so many livestock commodities in Indonesia (Aedah et al., 2016). The number of farms greatly affects the need for feed that will be ready to be eaten by livestock (Gustiani, 2009), while animal feed produced by industry is still wet or moist (Gustiani, 2009). For this reason, the industry must dry its products using sunlight or a drying machine (Octavia et al., 2019). Animal feed is a substitute for animal feed from nature (Devri et al., 2020). Animal feed is produced from the home industry (home industry) or mass-produced (Hutagalung et al., 2016). In every production, animal feed producers usually dry their products using sunlight (Nabila, 2022). If using sunlight alone, the production results will not meet the demand for animal feed in Indonesia (Ginting et al., 2018). For this reason, the need for a drying machine is needed to support the production of animal feed. Machines that are often encountered in the market use a heater (heater) (Habibi, 2016) and this tool uses a very large electric current. For this reason, the author tries to use a tool that is not commonly used in drying machines, namely AC (Air Conditioner) (Harmawan, 2022). The heat obtained for drying is obtained from the condenser, the air with low water vapor is released by the evavorator. The AC used here is the type commonly found on the market, namely AC Polytron with a power of 1 pk (Wailanduw et al., n.d.).

The objectives of this research are: (1) To produce a portable animal feed drying machine unit which is oriented towards the efforts of electrical energy efficiency that can be applied on a small and large scale (Nadeak, 2014). (2) Knowing the performance coefficient of the heat pump system (Dina et al., 2018). (3) To find out the specifications of the evaporator, condenser, and compressor.

The benefits obtained from the results of this study are: (1) This simple system widely contributes to meeting the drying needs of the livestock, agriculture, and home industry sectors, especially for areas that have high levels of rainfall in Indonesia (Musaad et al., 2020). (2) Utilization of wasted heat energy in the condenser. (3) As a development in the field of renewable energy, especially refrigeration and air conditioning technology (Buang & Pada, n.d.).

2. RESEARCH METHOD

2.1 Research Data

The data planned to be collected and further analyzed in this study include the following: (1) Animal Feed Mass (M). The mass of the feed was measured when it was wet (Mb) and when it was dry (Mk) (Rahman, 2020). (2) Drying time (t). The drying time needed to dry the feed is from wet to dry (wet weight to dry weight). At how many times did you fall? (Saragih, n.d.). (3) Temperature (T). The temperature measured is the air temperature when it enters the evaporator (T1), enters the condenser (T2), exits the condenser (T3) and when it is in the drying channel (T4) (Bernando & Ambarita, 2014). (4) Air humidity (Rh). Air humidity measured at the point when it enters the evaporator (Rh1), enters the condenser (Rh2), exits the condenser (Rh3) and leaves the drying channel (Rh4) (Syalimono et al., 2016). (6) Pressure (P). The refrigerant entering the compressor (P1), leaving the compressor (P2) and entering the evaporator (P3) is measured for pressure (Tyson et al., 2016).

3. RESULTS AND DISCUSSIONS

3.1. Thermodynamic Calculation

From testing the AC engine used, the data obtained are as follows: Initial Planning Data: (1) Condenser Working Pressure (PK) = 3.102 Mpa, (2) Evaporator Working Pressure (Pe) = 0.62 Mpa (3) Compressor Power (Wc) = 1 Hp = 746 Watt, (4) Condenser Temperature (Tk) = 68.275 ^{0}C (5) Evaporator Temperature (Te) = 7 ^{0}C. From the test data, the working conditions of the machine can be analyzed using a Mollier diagram, as shown in the following figure:

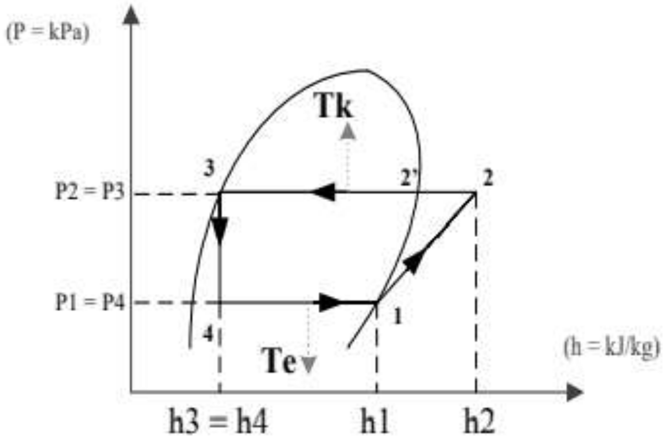

Figure 1. Ph Diagram

The condition of each point on the working condition of the Polytron AC machine, then the data obtained from the table saturation of R22 through the following interpolation process:
Point 1: T1 =7 ^{o}C , P = 0.62 MPa = 90 Psi
h1 = 407.17 kJ/kg
S1 = 1.7401 kj/kg.K
Point 2': T2 = 72.125 ^{0}C , P = 450 Psi = 3.102 MPa
h2 = 415.01 kJ/kg
S1 = S2 = Constant (Isentropy compression process)
Point 2: h2 = 447.09 kj/kg, T2' = T3
S2 = 1.7401 kj/kg.K
Point 3: h3 = 290.18 kJ/kg, P = 3.102 Mpa
T3 = 68.275 ^{o}C
S3 = 1.2863 kj/kg.K
Point 4: T4= T1 and P1= P4
h4= h3.

3.2. Condenser Calculation

Initial planning data: In this design using a condenser with a forced convection system with a fan to take heat. The pipe material in the condenser is made of BWG 16 copper pipe with an outer diameter of 0.5 in and an inner diameter of 0.40 in. Other data used in this design are as follows. (1) Condenser load when superheated = 0.5966 kJ/s, (2) Condenser load when condensing = 2.3218 kJ/s, (3) Total condenser load = 2.9184 kH/s, (4) Refrigerant temperature at when superheated = 72.125oC, (5) refrigerant temperature during condensation = 61.062^{o}C, (6) condenser intake air temperature = 30^{o}C, (7) condenser outlet air temperature = 40^{o}C, (8) refrigerant pressure in condenser = 3.102 Mpa. The state of the condenser can be depicted on the Ph diagram, as shown in the figure below, Preliminary data :
1. Inner diameter of copper pipe = 0.40in = 0.0102 m
2. Outer diameter of copper pipe = 0.5 inc = 0.0127 m
3. Condenser load when superheated;
h2 = 447.09 kJ/kg
h2' = 415.01 kJ/kg check = kg/s
h3 = 290.18 kJ/kg
4. Condenser load when superheated, Qk superheated = (h2-h2') (1)
= kg/s (447.09 kJ/kg - 415.01kJ/kg)
= 0.5966 kJ/s
5. Condenser load at the time of condensation, Qk condensation = (h2' –h3) (2)
= kg/s (415.01kj/kg - 290.18 kj/kg)

= 2.3218 kJ/s
Total condenser load = Qk superheated + Qk condensation (3)
= 0.5966 kJ/s + 2.3218 kJ/s
= 2.9184 kH/s

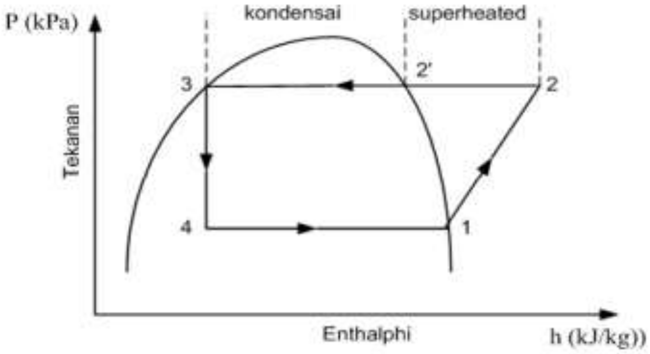

Figure 2. Ph . chart

3.2.1. Condenser Existing Data

Pipe length per pass : 0.68 m
Pipe Outer Diameter: 0.0127 m
Number of pipelines: 20
Fin height : 0.05 m
Fin thickness : 0.0003 m
Number of fins per in : 10/in
Distance between fins: 2 mm
Total length of pipe: 13.73 m
Distance between pipes 1 inch = 0.0254 m
Afr = (sn x Do) (4)
= (0.025 mx 0.0127 x 1)
= 0.0127 m2
Inlet air temperature = 300C
Outside air temperature = 40 ^{0}C.
Pressure R-22 in the condenser = 3.102 Mpa = 3102 Kpa

Design condenser dimensions, based on considerations in terms of construction, a condenser with air cooling media is used. To find the magnitude of the heat transfer effect that occurs, the table 0.03/8T of the compact heat exchanger by Kays and London is used, with the following data:

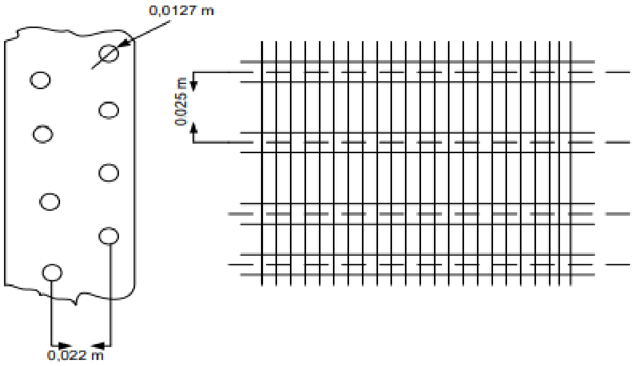

Figure 3. Design Condenser

Outside diameter = 12.7 mm
The hydraulic diameter of the flow passage,

Dh = 3.63 mm
Fin thickness = 0.3 mm
Number of fins/m = 315 per meter
Total heat transfer/total volume = 587 m2/m3
Free-flow area/frontal area,σ= 0.534
Fin area / total area = 0.913
Aluminum material conductivity (k) = 204 W/m2 ^{0}C
Fin height (l) = 0.05 m
Sn = Distance between pipes Afr = (Sn – Do) (5)
= 0.0254 – 0.0127 x 1
= 0.0127 m2

3.3. Evaporator Calculation

In the P–h diagram of a simple vapor compression cycle, the evaporator has the task of realizing lines 4 – 1. After the refrigerant drops from the condenser through the expansion valve it enters the evaporator and is vaporized, and is sent to the compressor. In principle, the evaporator is almost the same as the condenser, which is the same as APK whose function is to change the refrigerant phase. From the results of data collection conditions in the evaporator can be seen in Figure 4.1.
From Figure 4.1 obtained:
Point 4: T4 = 7^{o}C and P = 0.062 Mpa
h4= 290.18 kJkg
Point 1: T1 = 7 ^{0}C
h1= 415.01 kJ/kg

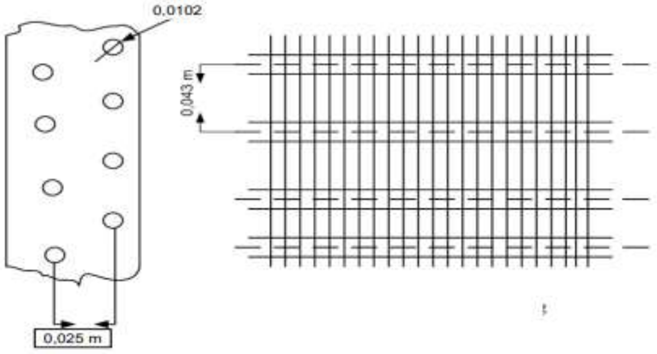

Figure 4. Design Evaporator

From the picture above, the initial data is obtained:
Inner diameter of copper pipe (Di) = 0.0102 m
Outside diameter of copper pipe (Do) = 0.0127 m
Planned n = 16
Hydraulic diameter (Dh) = 0.0127 – 0.0102 = 0.0025 m
Evaporator fin height = 0.2 m
Distance between pipes = 0.025 m
Afr = (Sn – Do) x 1 m (6)
= (0.0127 m – 0.025 m) x 1 m
= 0.0127 m2

3.3.1. For Refrigerant Side R22

The thermophysical properties of R22 at a temperature of 7^{o}C:

$$\rho_l = 1257{,}5 \, {}^{kg}/_{m^3}$$

$$\rho_v = 37{,}91 \, {}^{kg}/_{m^3}$$

$$\mu_l = 0{,}00019475 \, {}^{Ns}/_{m^2}$$

$$K_l = 93{,}2 \times 10^{-3} \, {}^{W}/_{m.K}$$

$$C_p = 1192 \, {}^{kJ}/_{kgK}$$

$$h_4 = xh_g + (1-x)h_l \tag{7}$$

$$290{,}18 = x.415{,}01 + (1-X)208{,}25$$

$$290{,}18 = 415{,}01x + 208{,}25 - 208{,}25x$$

$$198{,}92X = 81{,}93$$

$$x = 0{,}411$$

1. Refrigerant Mass Velocity

$$G = \frac{\dot{m}_{R22}}{\pi \times r_{in}^2} \tag{8}$$

$$G = \frac{0.0186}{3{,}14 \times 0{,}0051^2}$$

$$G = 227{,}941 \, {}^{kg}/_{m^2 s}$$

2. Prandtl Numbers

$$\mathrm{Pr} = \frac{Cp \times \mu_l}{K} \tag{9}$$

$$\mathrm{Pr} = \frac{1192 \times 0{,}0001947}{93{,}2 \times 10^{-3}}$$

$$\mathrm{Pr} = 24{,}941$$

3. Refrigerant Side Heat Transfer

$$\frac{hi \times Di}{Kl} = 0{,}06 \left(\frac{\rho_l}{\rho_v}\right)^{0{,}28} \left(\frac{Di \times G \times x}{\mu_l}\right)^{0{,}87} \times Pr^{0.4} \tag{10}$$

$$\frac{hi \times 0{,}0102}{93{,}2 \times 10^{-3}} = 0{,}06\left(\frac{1257{,}5}{37{,}91}\right)^{0{,}28}\left(\frac{0{,}0102 \times 342{,}837 \times 0{,}296}{0{,}00019475}\right)^{0{,}87} \times 24{,}907^{0{,}4}$$

$$\frac{hi \times 0.0102}{0{,}932} = 0{,}06 \times 2{,}66\ X\ 1625{,}491\ X\ 3{,}61$$

$$\frac{hi \times 0{,}0102}{0{,}932} = 938{,}7417$$

$$hi \times 0{,}0102 = 874{,}9073$$

$$hi = \frac{874{,}9073}{0{,}0102}$$

$$hi = 85775.22\ {}^{W}/_{m^2\,°C}$$

3.3.2 For Air side

Thermophysical properties of air at average temperature:

$$\frac{30 + 16}{2} = 23°C = 296\ K$$

= 1.8264 x 10-5 Ns/m2
K = 0.02592 W/mK
Cp= 1.0056 kJ/kgK
ρ = 1.1962 kg/m3
Pr = 0.70912

1. Air Mass Rate

$$m_{ud} = \frac{Qe}{Cp \times \Delta Tu} \tag{11}$$

$$= \frac{1{,}192}{1{,}0056 \times 14}$$

$$= 0{,}08466$$

2. Velocity of Air Mass per Area

$$G_{ud} = \frac{m_{ud}}{\sigma \times A_{fr}} \tag{12}$$

$$= \frac{0{,}08466}{0{,}534 \times 0{,}0127}$$

$$= 12{,}4847\ {}^{kg}/_{m^2 s}$$

4. Reynolds Number

$$Re = \frac{Dh \times G}{\mu} \tag{13}$$

$$= \frac{0{,}0025 \times 227{,}941}{0{,}00019475}$$

$$= 2926{,}072$$

5. Stanton's Number

For Re = 2926,072 then we get jH = 0.0058

$$jH = st \times Pr^{2/3} \tag{14}$$

$$0{,}0058 = st \times 24{,}941^{2/3}$$

$$st = 0{,}000694$$

6. Heat Transfer For Air Side

$$h_o = st \times G \times Cp \tag{15}$$

$$= 0{,}000694 \times 227{,}941 \times 1{,}192$$

$$= 0{,}1886\ {}^{W}/_{m^2\,°C}$$

3.3.2. Total Heat Transfer Area

The overall heat transfer can be determined by:

$$A = \frac{Q_e}{U \times F \times LMTD} \tag{16}$$

$$= \frac{1{,}192}{985{,}75 \times 1 \times 5{,}73}$$

$$= 0{,}3852\ m^2$$

3.3.3. Pipe Length Per Pass

To find the required pipe length can be determined by:

$$A = \pi \times Di \times L \tag{17}$$

$$L = \frac{A}{\pi \times Di} \tag{18}$$

$$= \frac{0{,}3852}{3{,}14 \times 0{,}0102}$$

$$= 12.02\ m$$

So with this we
can determine the length of the pipe for each path, namely:

$$l = \frac{L}{n} \tag{19}$$

$$= \frac{12{,}02}{16}$$

$$= 0{,}7517\ m$$

4. CONCLUSION

Based on the data analysis and discussion, the following conclusions can be drawn: (1) The compressor used in this heat pump system clothes dryer is a Rotary Vane Compressor or sliding blade compressor. With engine specifications: (1) Condenser Working Pressure (PK) = 3.1 Mpa, (2) Evaporator Working Pressure (Pe) = 0.62 Mpa, ()- Compressor Power (Wc) = 1 Hp = 746 Watt, (3) Condenser Temperature (Tk) = 72.125 0C, (4) Evaporator Temperature (Te) = 7 0C. The performance of the refrigeration cycle is 2.90. A high coefficient of performance is desirable because it indicates that a certain amount of refrigeration work requires only a small amount of work and the drying process. From thermodynamic calculations based on engine specifications, the power of the electric motor driving the compressor is 1, 03 kW with a compression efficiency of 0.99. (2) Based on the thermodynamic calculation, the performance of the refrigeration cycle is 2.90 and the total condenser load is 2.9184 kH/s. The overall heat transfer coefficient when superheated is 38.96 W/m2 0C and at the time of condensation is 71.945 W/m2 0C. The length of the pipe required for the condenser is 10.31 m and the length of the crossing pipe on the condenser is 0.5155 m. (3) From the calculation of the dimensions of the evaporator obtained: For the Refrigerant Side: The length of the pipe required for the condenser is 10.31 m and the length of the crossing pipe on the condenser is 0.5155 m. (3) From the calculation of the dimensions of the evaporator obtained: For the Refrigerant Side: The length of the pipe required for the condenser is 10.31 m and the length of the crossing pipe on the condenser is 0.5155 m. (3) From the calculation of the dimensions of the evaporator obtained: For the Refrigerant Side:

$G = 227{,}941\ {}^{kg}/_{m^2}$

$\mathrm{Pr} = 24{,}941$

$h_i = 85775.22\ {}^{W}/_{m^2\,°C}$

For Air Side:

$m_{ud} = 0{,}08466\ {}^{kg}/_{s}$

$G_{ud} = 12{,}4847\ {}^{kg}/_{m^2 s}$

$\mathrm{Re} = 2926{,}072$
$\mathrm{St} = 0{,}000694$
$h_o = 0{,}1886\ {}^{W}/_{m^2\,°C}$

Effisiensi Sirip :
$\eta_c = 0{,}999$
Koefisien Perpindahan Panas Menyeluruh :
$U = 985{,}75\ {}^{W}/_{m°C}$
Perbedaan Temperatur Rata – Rata (LMTD)
$\Delta Tm = 5{,}73°C$
Luas Perpindahan Kalor Menyeluruh :
$A = 3852 m^2$
Panjang Pipa Per Lintasan :
$l = 0{,}7517\ m$

Based on the research that has been done, the authors suggest the following: (1) It is necessary to redesign this animal feed drying machine by adding a xesos van at the evavorator output and condenser output in the designed machine, to get air pressure that flows faster. (2) The design of the animal feed drying machine that has been made already has an air channel for drying animal feed, presumably in the next research so that the drying channel is isolated to minimize wasted heat.

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