Yankee hoods: development of a theoretical model to optimize thermal and electrical efficiency

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ABSTRACT: For any given production rate, there exists an optimum operating point based on energy costs.

KEYWORDS: Convection heating, costs, development, electrical properties, equations, hoods, models, optimization, theories, thermal efficiency, Yankee machines.

The modern Yankee hood serves an important function. It performs as a convective dryer, driving water out of the sheet by defined heat and mass transfer mechanisms. Also it removes the evaporated water through a dedicated exhaust system. Drying rates on a Yankee dryer hood are varied by two key factors: nozzle air temperature and nozzle air velocity. These are the only variables which control drying rate once a hood is installed. **Figure 1** shows the typical air flows for a Yankee hood system.

The total energy input and operational costs of a Yankee hood are comprised of:

- Electrical/horsepower (fans): related to nozzle velocities
- Thermal/fuel (burners): related to air temperatures.

At what combination of temperature and velocity should a hood operate to be thermally efficient?

In this paper, we will develop a theoretical model which shows that, at any given production rate (drying requirement) and Yankee cylinder diameter, there is a distinct, optimum selection of drying nozzle temperature and velocity which minimizes total energy (operating) cost.

Finally, on a Yankee hood system, it is generally accepted that a proper air balance ensures minimum energy consumption (*1*). This refers to any system operating at a given drying temperature and impingement velocity. This assumption will hold throughout the development of the model.

Convective heat transfer coefficient

Every hood exhibits a particular heat transfer coefficient based on the classical equation for convective drying:

$$Nu = K\,Pr^{b}Re^{n} \tag{1}$$

There are many published articles based on this formula (*2–4*).

Chance (*2*) and Kercher (*3*) confirm that the power (n) of the Reynolds number is dependent on the net (nozzle) open area, nozzle diameter, and spacing. Furthermore, the power of the Prandtl number is relatively constant for convective drying, with a pattern of round holes b = 1/3. Therefore,

$$h = K_1\,Pr^{1/3}\,Re^{n} \tag{2}$$

From such a formula, it is possible to predict the convective heat transfer coefficient for a particular Yankee hood nozzle geometry at any nozzle velocity and temperature. Note that the factor K_1 incorporates the effect of mass transfer on drying. Within the scope of this paper, its formulation has been omitted for simplicity.

Furthermore, a temperature correction model can be set up by a simple ratio:

$$\frac{h_1}{h_2} = \frac{k_1}{k_2}\left(\frac{Re_1}{Re_2}\right)^{n} \tag{3}$$

From such a model, the heat transfer coefficient can be derived at any temperature, based on a given coefficient at a certain (reference) temperature and velocity.

Typical ranges for key variables are:

- Nozzle air temperature: 500–950°F (260–510°C)
- Nozzle air velocity: 10,000–30,000 ft/min (3049–9146 m/min)
- Nozzle open air: 1.5–2.5%
- Nozzle diameter: 0.25 in. (6.4 mm)
- Impingement distance to sheet: 1 in. (25.4 mm).

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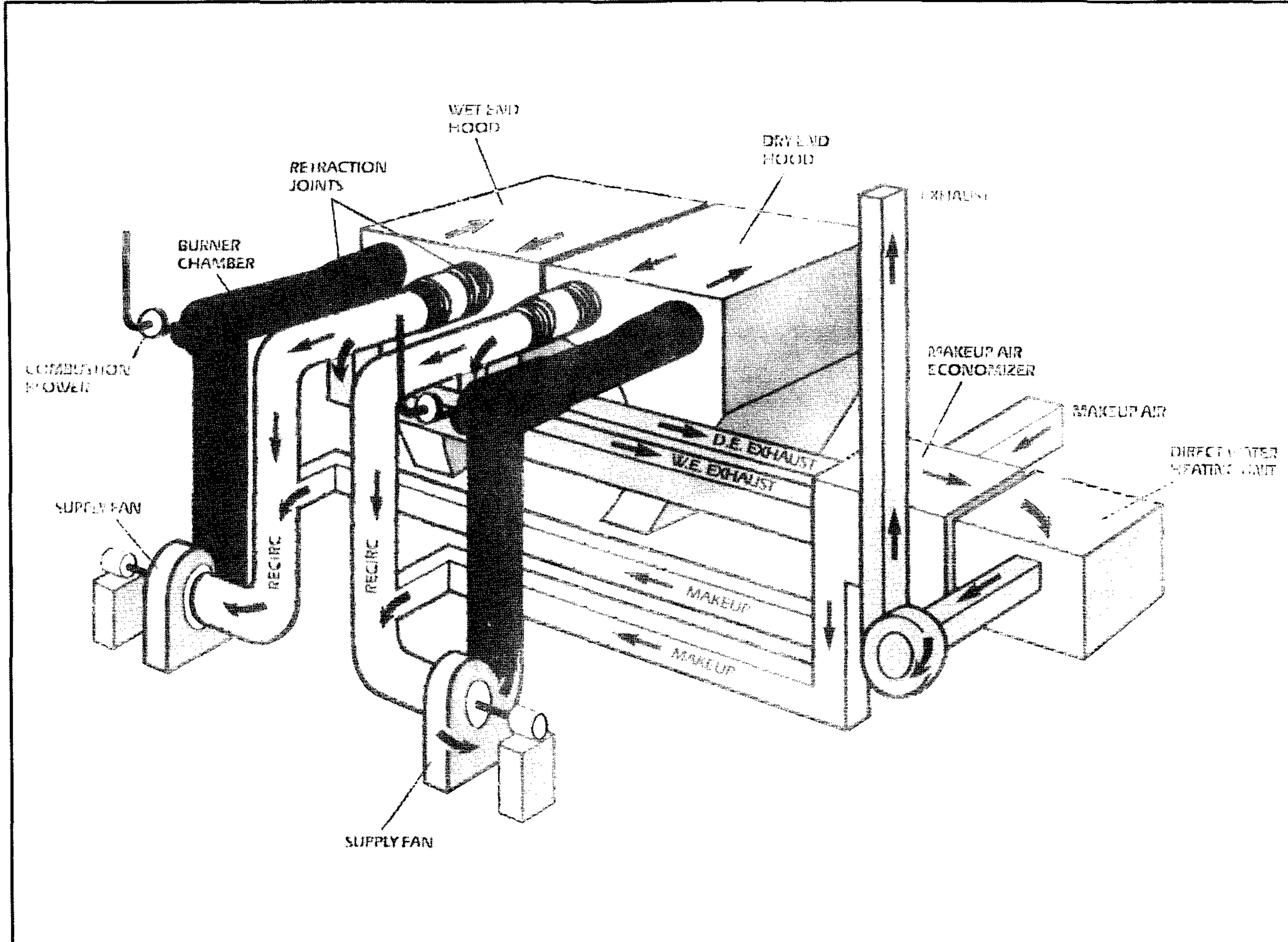
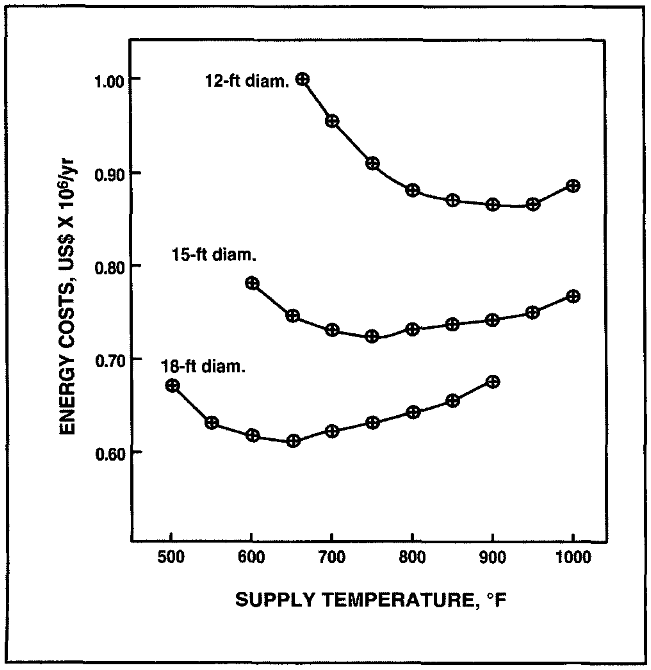
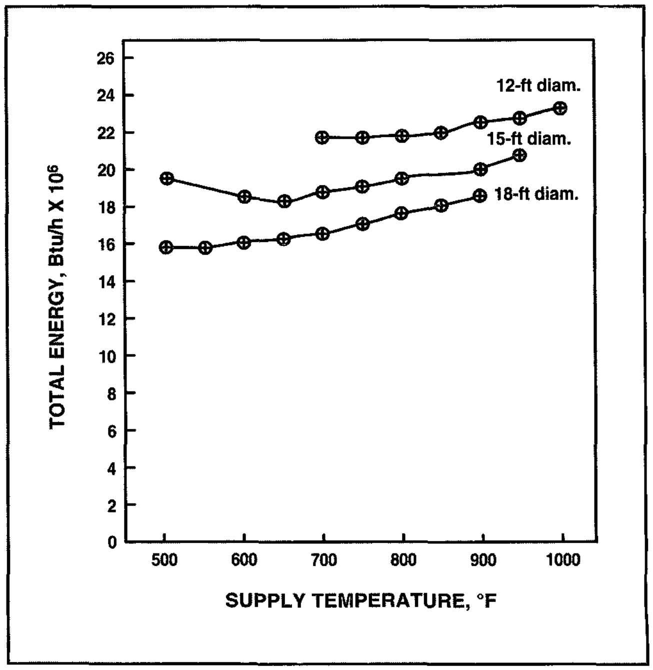
1. Typical air flows for a Yankee hood system



2. Energy costs vs. temperature



3. Energy consumption vs. temperature



Hood-side drying rate

The air-side drying rate can be calculated from the following formula:

$$R_w\,(air) = h \times (T_a - T_s) \times 1/L \qquad (4)$$

Note that the radiation effect is negligible.

Calculation of required air-side drying rate

The following formulas apply:

$$P = B \times S \times W \qquad (5)$$

$$F = \frac{P \times Y}{100} \qquad (6)$$

$$E = F \times [(100 - Z)/Z - (100 - Y)/Y] \qquad (7)$$

$$A = D\pi \times W \qquad (8)$$

$$Rw\,(total) = E/A \qquad (9)$$

$$Rw\,(air) = RW_{(total)} - RW_{(cyl.)} \qquad (10)$$

Based on a required production rate for a given cylinder diameter and cylinder drying rate, one can generate a variety of possible hood nozzle velocities and temperatures to achieve the necessary total water evaporation.

At a given hood temperature, one can use Eq. 4 to calculate the required "h".

Then, using Eq. 1, the required velocity (v) is calculated.

Electrical energy input

There are normally five fans on a Yankee hood system: two supply, two combustion, and one exhaust.

The energy input to each fan follows the general formula:

$$HP = [CFM \times SP \times (0.01573)]/N \qquad (11)$$

$$SP = [\rho(V)^2]/(1096.5)^2 \qquad (12)$$

Thermal energy input

The total burner heat input is equal to the sum of the following loads:

Heat to paper web =
$$Q_{PW} = P \times C_p \times (T_s - T_{ES}) \qquad (13)$$

Heat to evaporate water =
$$Q_{EW} = L \times E \times Rw(air)/Rw\,(total) \qquad (14)$$

Heat to makeup air =
$$Q_{MA} = U \times C_a \times (T_s - T_R) \qquad (15)$$

Heat lost to radiation =
$$Q_R = (0.05)(Q_{PW} + Q_{EW} + Q_{MA}) \qquad (16)$$

Energy optimization model

A computer program was written to incorporate all of the previously mentioned equations. Its purpose was to determine whether, for any given production rate, there exists an optimum (minimum energy) operating point based on energy costs.

Nomenclature

Nu = Nusselt number, hd/k

h = convective heat transfer coefficient, Btu/h/ft²/°F (cal/h/m²/°C]

d = nozzle diameter, ft (m)

k = thermal conductivity (air), Btu/h/ft/°F (cal/h/m/°C)

K = correlation constant which relates to cross flow interference, nozzle open area, nozzle diameter, and impingement distance

Pr = Prandtl number, Cpa/k

m = air viscosity, lb/h/ft (kg/h/m)

Cpa = air heat capacity, Btu/lb/°F (cal/kg/°C)

b = power of the Prandtl number

Re = Reynolds number, ρvd/μ

ρ = air density, lb/ft³ (kg/m³)

v = air velocity, ft/h (m/h)

n = power of Reynolds number

K_1 = Kk/d

Rw (air) = Drying rate of hood, lb/h/ft² (kg/h/m²)

T_a = drying air temperature, °F (°C)

T_s = sheet (average) temperature, °F (°C)

L = latent heat of vaporization, Btu/lb (cal/kg)

P = production (of paper), lb/h (kg/h)

B = paper basis weight, lb/ft² (kg/m²)

S = machine speed, ft/h (m/h)

W = sheet width, ft (m)

F = bone-dry fiber production, lb/h (kg/h)

Y = reel dryness (sheet), %

E = evaporation by cylinder and hood, lb/h (kg/h)

Z = dryness of sheet entering hood, %

A = dryer surface area, ft² (m²)

D = dryer diameter, ft (m)

Rw (total) = drying by both air and cylinder, lb/h/ft² (kg/h/m²)

Rw (cyl.) = drying rate by cylinder, lb/h/ft² (kg/h/m²)

HP = horsepower, hp (kw)

CFM = fan volume, ft³/min (m³/min)

SP = static pressure at fan, in. (mm)

N = fan efficiency, %

C_{pp} = heat capacity of paper, Btu/lb/°F (cal/kg/°C)

T_{ES} = sheet temperature entering hood, °F (°C)

U = makeup air mass flow, lb/h (kg/h)

T_R = makeup air temperature, °F (°C)

The following assumptions were made:

- Paper grades: tissue and towel
- Cost of electrical energy: US$ 300/hp/year (US$ 400/kW/year)
- Cost of thermal energy: US$ 4/million Btu (US$ 16/million kcal).

The following cylinder diameters were modeled: 12 ft (3.66 m), 15 ft (4.57 m), and 18 ft (5.49 m).

Finally, for each diameter, the hood temperature was varied from 500°F to 950°F, and the resulting nozzle air velocity was calculated.

Consequently, the total fan horsepowers and burner loads were determined.

Figures 2 and 3 illustrate the interesting results of this model. Obviously, for a given production rate, there exists a hood temperature at which the system would cost the least to operate. It turns out that the least expensive operating points are associated with the larger hood diameters. Of course, as the equipment gets physically larger, so does the capital cost. It is up to the papermaker to decide on an acceptable return on investment.

No matter which diameter is chosen, it is within the experience of the hood designer to dictate the optimum (energy) operating point.

Furthermore, this model was consistent in its conclusions regardless of the paper grade, tonnage, sheet width, machine speed, or sheet moisture.

Summary

Based on sound theoretical modelling, we have shown that for any given Yankee hood drying application:

- There exists an optimum air temperature and velocity such that the total energy cost is minimized.
- The larger-diameter systems operate at a lower total energy cost, for comparable production rates.
- This model is consistent regardless of paper grade, tonnage, sheet width, machine speed, or sheet moisture. TJ

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