Humidity Control Key to Optimizing TADs for Energy, Production Benefits

Thru-Air Dryer response to humidity is a significant factor in maximizing system performance for increased production and/or reduced utility costs.

— BY JOE PILSBURY

Operating experience in tissue and towel production has shown that the overall thermal efficiency of a Thru-Air Dryer (TAD) air system is proportional to the humidity of the supply air. The majority of Metso TAD air systems have been designed based on this principle, assuming that the fresh air intake damper will be kept closed during steady state operation.

However, depending on the configuration of the TAD machine and the specific utility costs for electricity and natural gas, operating a TAD with a lower supply air may have advantages such as increased production or reduced utility costs.

This article, the first of a two-part series, examines a case study and shows the reaction of system dynamics as a certain production rate of paper is dried by an air stream with varying levels of humidity. It examines drying from a TAD operational standpoint, concentrating on the overall energy consumption of the entire TAD air system rather than delving into the previously documented science of heat and mass transfer (that explains the application of TAD).

Part 2 in September 2003 issue of PaperAge will include further analysis and discussion of the case example established in this article as well as question-and-answer scenarios based on results and conclusions of this study.

Air System Configuration

TAD air systems consist of four key elements:

- Burner that heats the process air
- Fan that moves the process air through the system
- TAD roll
- Stream to exhaust air into the atmosphere.

As illustrated in Figure 1, the fourth element (stream to exhaust air into the atmosphere) distinguishes the two most common configurations being installed today. While Configuration A has a second fan for exhausting air to the atmosphere, commonly called an "exhaust" or "dump" fan, Configuration B uses positive pressure at the main fan discharge to send exhaust air into the atmosphere.

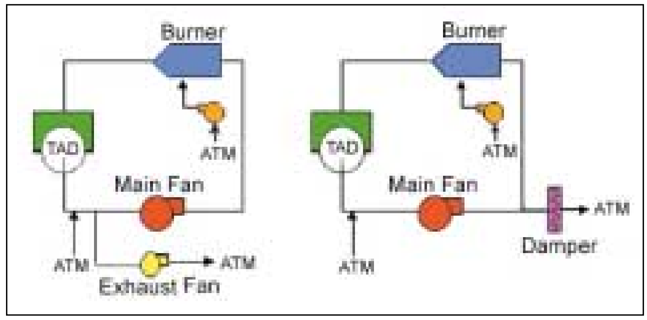

Figure 1. Two most common configurations of air systems.

Calculations of how system dynamics change as air system humidity is varied have been performed for various air system configurations. Cost and energy benefits vary from system to system, but the basic dynamics are consistent. For simplicity, this article only examines the system dynamics of TAD air systems without an exhaust fan (Configuration B).

Case Study

Table 1 defines the product selected for this analysis, with production on a typical 4.8-m-dia. TAD.

Production Rate	186	Metric tpd
TAD Speed	850	M/min
Product Width	4000	Mm
Basis Weight	38	g/sm
Consistency In	25	%
Consistency Out	90	%
Permeability	$V = 0.369\ \Delta P^{0.63}$	m/s, mbar
Supply Temperature	210	°C

Table 1. Product and product assumptions for case study

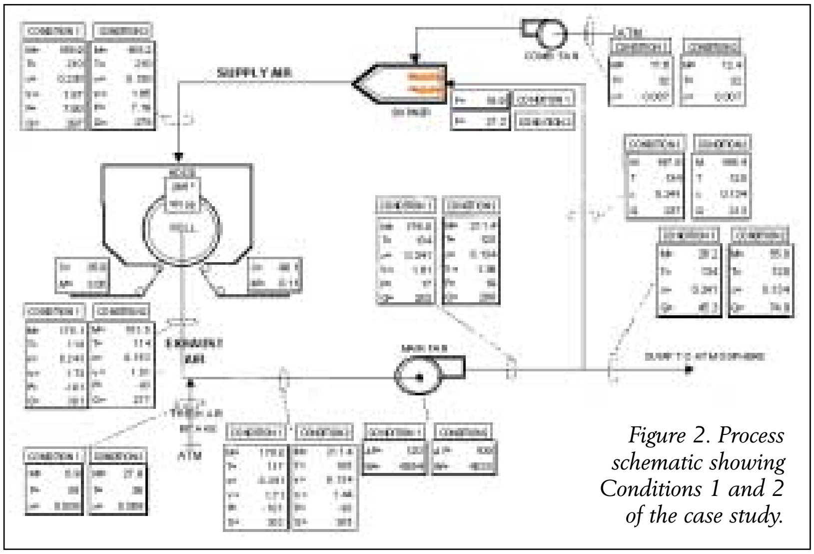

Figure 2. Process schematic showing Conditions 1 and 2 of the case study.

Calculations are based on a supply air stream humidity range from 0.08 to 0.23 kg H_2O/kg dry air. Supply air humidity was varied by introducing different amounts of fresh air into the air system via the fresh air make up duct.

For this analysis, two operating scenarios within the case study are defined. Conditions 1 and 2 have absolute humidity values of 0.23 and 0.13 kg H_2O/kg dry air, respectively. As Figure 2 illustrates, many differences exist between the two scenarios for the fixed production rate.

FIGURE 2 KEY

M = Dry mass air flow, kg dry air/s
T = Temperature, °C
X = Humidity, kg H_2O/kg dry air
P = Static pressure, mbar
Q = Actual volumetric air flow, m^3/s
V = Specific volume, m^3/kg dry air
W = Fan Power, kw
C = Product consistency, %
M = Product moisture, kg H_2O/kg pulp
F = Burner energy, mw

Conditions 1 and 2 in Figure 2 are based on case study machine production data defined in Table 1 above. Table 2 shows the energy input for Figure 2.

Energy	Condition 1	Condition 2
Burner Duty, kw	19,900	21,200
Fan Power, kw	4,354	4,033
Total Thermal Input, kw	24,254	25,233

Table 2. Energy input for Conditions 1 and 2 in case study.

Some key differences to note about Condition 2 relative to Condition 1 are:

- Although the supply temperature is the same, exhaust temperature is lower.
- Higher mass flow in the supply and exhaust streams, but lower volumetric flow.
- Fan differential pressure is 9% less.
- Exhaust to atmosphere mass flow nearly doubles.
- Burner output increases by 6.5%, while fan power decreases by 7.4%.
- Total thermal energy (fan & burner) increases by 4.0%.

Humidity

There are differences in opinion on how a TAD machine should be operated with respect to humidity. The reasons for this are the effects of humidity in an already complex drying process. The following statements illustrate the many effects that humidity has on drying:

- Increased air humidity increases specific volume, which in turn reduces the dry air mass flow rate delivered by the fan., i.e., a fan dictates volumetric flow, and lower humidity allows the fan to move more mass for a fixed volumetric flow.
- Water vapor has nearly twice the specific heat of air.
- Increased air humidity increases the heat transfer coefficient.
- Under steady state conditions, the sheet temperature during evaporation is equal to the adiabatic saturation temperature (AST). The AST is determined by the humidity and temperature of the supply air stream. Increasing the humidity in the air stream increases the AST, effectively lowering the mean temperature difference between the supply temperature and the sheet temperature.

Energy Consumption

The total energy (fans and burners) required to evaporate a given amount of water from the sheet is inversely proportional to system humidity. This can be proved by performing a simple energy balance of the TAD air system.

Figure 3 shows the total energy input required per kg of water evaporated for the production described in Table 1, as a function of humidity. To lower the system humidity, more fresh air must be introduced to the air system, which adds extra duty to the burner. This adjustment also has an impact on the fan power requirements, but the response of the fan is not quite as direct as for the burner.

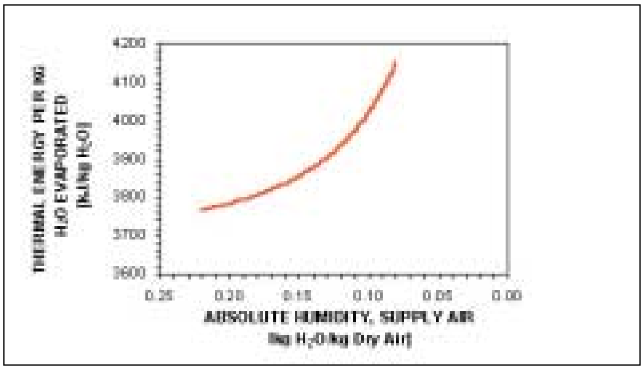

Figure 3. Overall thermal input per kg of water evaporated as a function of humidity in the case study.

Burner

In the case study, reducing the supply air humidity from 0.23 to 0.13 kg H2O/kg dry air caused the burner duty to increase by nearly 7%. This relationship is shown in Figure 4, with Conditions 1 and 2 noted. Regardless of the product being made or the air system configuration, burner input increases as system humidity decreases.

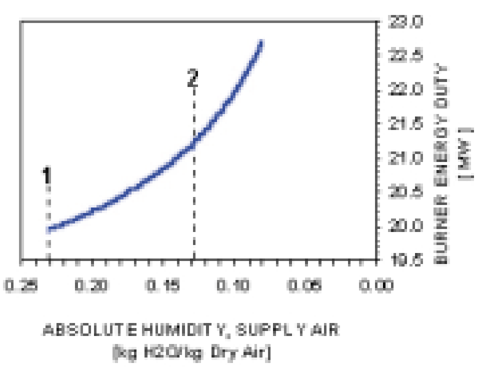

Figure 4. Burner energy duty as a function of supply air humidity in the case study.

Fan Power

It is intuitive to think that adding fresh air to the air stream before the main fan would increase the power input of the fan motor. However, the changes in air stream properties that occur due to the intake of fresh air actually reduce the fan power requirements.

To understand why this happens, notice the effect that the relatively cold and dry fresh air has on the hot and moist exhaust air. Fan power requirements are determined only by volumetric flow and differential pressure across the fan:

$$PWR_{FAN} \approx \frac{Q \times \Delta P}{10 \times \eta}, \text{ where:}$$

Q = Volumetric Flow [m³/s]

ΔP = Fan Differential Pressure [mbar]

η = Fan Efficiency [% as decimal]

PWR_{FAN} = Fan Power [kW]

A significant change occurs in the specific volume of the exhaust air stream after the fresh air is brought in through the fresh air intake duct. As derived from the ideal gas law, the relationship of specific volume to temperature and humidity is:

$$\upsilon = \frac{4.62\,(x + 0.622)(T + 273)}{P_{atm} + P_{stat}}$$

υ = specific volume, m³/kg dry air

x = absolute humidity, kg H_2O/kg dry air

P_{atm} = Atmospheric Pressure, mbar (1013.25 at sea level)

P_{stat} = static pressure, mbar

T = temperature, °C

From the equation above, it is clear that changes in humidity and temperature have a profound impact on specific volume. This point is well illustrated in Figure 2. In Condition 2, the air stream at the fan inlet is 11°C cooler and 0.107 kg H_2O dryer than Condition 1. These seemingly small differences in air stream properties cause the specific volume to decrease by nearly 16%.

The change in specific volume allows the fan to move about 10% more total mass flow (air and water vapor), or 20% more dry air mass flow, while the volumetric flow remains relatively unchanged. By looking at the volumetric flows around the system, it can be seen that the stream entering the fan is the only stream (with exception of the exhaust stream) that the volumetric flow is higher than in the lower humidity condition. Figure 5 shows that the hood supply and roll exhaust volumetric flows increase with system humidity.

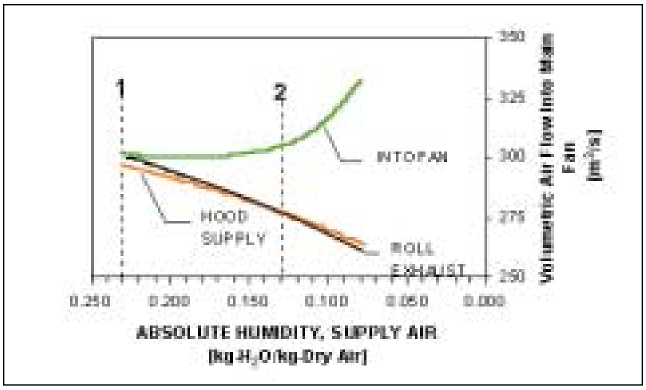

Figure 5. Supply, exhaust, and fan volumetric flow rates as functions of supply air humidity. Points labeled 1 and 2 correspond to Conditions 1 and 2 in the study.

Lower duct velocity and system pressure loss result from reduced volumetric flow rates. The impact of the reduction of system pressure losses is illustrated in the reduction in differential pressure across the main fan from Condition 1 to Condition 2. As a result of reduced fan pressure differential, the fan power decreases by 320 kw (about 7%).

As Figure 6 illustrates, initially a sharp decrease in fan power results when humidity is reduced. If humidity is reduced excessively, however, the fan power actually starts to increase again. This phenomenon is due to the rates of change of the required dry air mass flow and the specific volume entering the fan. As the system humidity decreases, the specific volume decreases and the dry air mass flow rate increases.

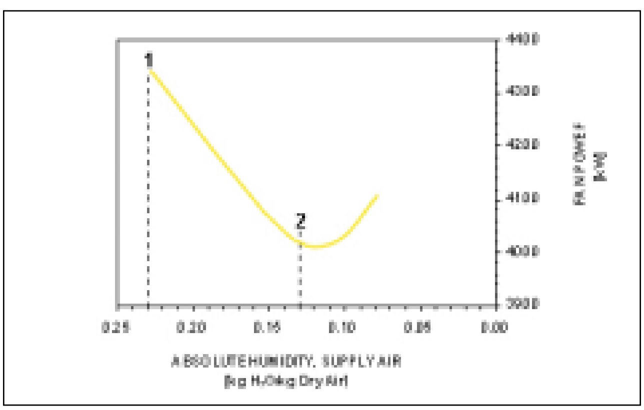

Figure 6. Fan Power versus supply air humidity, representing data from the case study.

At the early stages of this reduction in humidity (from 0.23 to 0.18 kg H_2O/kg dry air), the specific volume decreases at a slightly higher rate than the dry air mass flow rate into the fan increases. Thus, the decreasing rate of specific volume dominates over the increase in dry air mass flow and reduces the fan power by reducing both volumetric flow and the system static pressure losses.

Eventually, as the humidity becomes relatively low (from 0.18 to 0.08 kg H_2O/kg dry air), the rate of increase of the dry air mass flow into the fan becomes much higher and dominates over the rate of decrease of the specific volume. This point is illustrated in Figure 7.

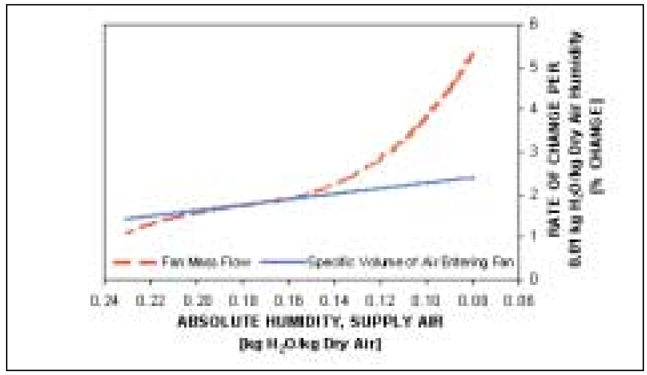

Figure 7. Case study data showing percent change of fan mass flow and specific volume versus decreasing system humidity (absolute value of change). Note: Rate of change is shown as an absolute value (+ only) to illustrate the comparison between the two data sets. Specific volume decreases with a decrease in humidity (would normally show a negative slope).

Once the rates of change start to diverge, the decrease in specific volume as humidity drops is not enough to compensate for the large increases in fan mass flow rate. Consequently, the fan volumetric flow rate begins to climb, thereby increasing the fan power required. Figure 5 shows the volumetric fan flow response.

Supply Air

Another interesting point to study in regard to reducing the system humidity is the dry air energy rate requirements of the supply stream. It is a common opinion that to produce a given amount of paper at a certain production rate with a relatively lower humidity than what is normally used, the total energy in the supply air stream would need to remain constant. Consider the following relationships:

$$C_a = 0.9976 + 85.48 \times 10^{-6} T$$

$$C_v = 1.848 + 36.68 \times 10^{-5} T$$

$$C_p = C_a + x\, C_v$$

$$h = \int_{273}^{T} C_p dT$$

$$H = w \int_{273}^{T} dh$$

Where:

$$T = \text{Absolute Temperature, [K]}$$

$$C_a = \text{Specific Heat of Dry Air,} \left[\frac{kJ}{kg \cdot K}\right]$$

$$C_v = \text{Specific Heat of Water Vapor,} \left[\frac{kJ}{kg \cdot K}\right]$$

$$x = \text{Absolute Humidity of Air Stream,} \left[\frac{kg-H_2O}{kg-DryAir}\right]$$

$$C_p = \text{Combined Specific Heat (air + water vapor),} \left[\frac{kJ}{kg-DryAir \cdot K}\right]$$

$$h = \text{Enthalpy,} \left[\frac{kJ}{kg-DryAir}\right]$$

$$w = \text{Dry Air Mass Flow,} \left[\frac{kg-DryAir}{s}\right]$$

$$H = \text{Dry Air Energy Rate,} \left[\frac{kJ}{s} \text{ or kW}\right]$$

When humidity is reduced, the enthalpy is reduced. This commonly leads to the perception that reducing the system humidity would make it necessary to increase either the supply temperature or the supply dry air mass flow to a point where the dry air energy rate is maintained.

Holding the supply temperature constant, the total dry air energy rate of the supply stream actually decreases with decreasing humidity, even though the supply dry air mass flow increases as well as the moist air flow. Figure 8 illustrates how Hsupply and Hexhaust decrease with decreasing humidity, but that the difference between the two remains relatively constant.

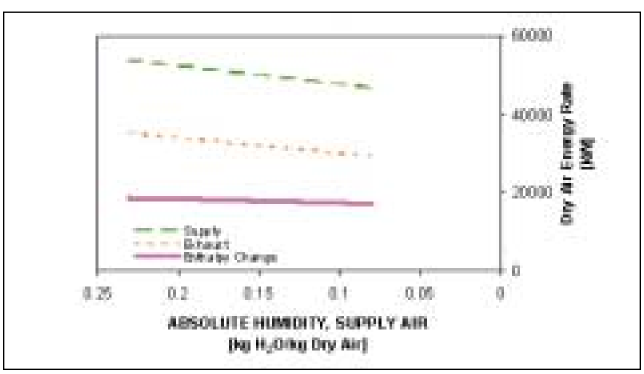

Figure 8. Trends of dry air energy rates in supply, exhaust, and the difference between the two as a function of humidity

Since the dry air energy rate of the supply stream decreases with a decrease in humidity, it can be concluded that the heat transfer rate must be higher under the lower humidity condition. The reason for this is the temperature differential between the supply air and the temperature at which the sheet evaporates—the adiabatic saturation temperature (AST).

Figure 9 illustrates that the AST decreases as humidity decreases, effectively increasing the temperature differential between the supply air and the evaporating sheet under steady state conditions.

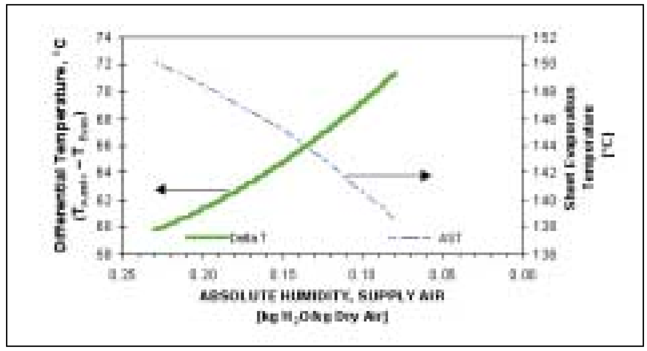

Figure 9. AST and differential temperature versus supply air humidity.

The temperature differential between the supply air and the sheet evaporation temperature has a direct impact on heat transfer:

$$Q = hA\Delta T$$

Where:

$$Q = \text{heat transfer rate [kW]}$$

$$h = \text{heat transfer coefficient} \left[\frac{kW}{m^2K}\right]$$

$$A = \text{surface area, } [m^2]$$

$$\Delta T = \text{Temperature Differential } (T_{supply} - T_{evap}) \text{ [K]}$$

The surface area remains constant and the heat transfer coefficient does not significantly change with changes in system humidity. With the increase in temperature differential comes a substantial increase in the heat transfer rate. Since the heat transfer rate increases so rapidly as humidity is decreased, the total mass flow to the hood (air and water vapor) decreases. This results in a lower exhaust temperature because the system is using less mass flow to transfer an equivalent amount of energy to the sheet. This behavior has been measured and documented on machines in production today.

About the Author:
Joseph Pilsbury is Process Engineer, Metso Paper, Thru-Air Systems, Biddeford, Maine.