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Economizer Control Using Mixed Air Enthalpy

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ABSTRACT

Enthalpy economizer can theoretically save more energy than temperature based economizer. However, the requirement of outdoor air humidity measurement in the traditional enthalpy economizer control made it impossible. A novel control sequence using mixed air enthalpy is developed in the paper. Both theoretical and experimental investigation shows that humidity measurement in mixed air duct is very reliable, and the proposed method can achieve true enthalpy economizer saving. A case implementation shows 15.7% more energy saving than temperature based economizer in Omaha, NE.

1. INTRODUCTION

Economizer is recommended by ASHRAE (1) as an energy conservation measure in air conditioning system. Significant saving can be achieved if a system can properly switched over to an economizer cycle (2~4). The control algorithms for switchover are typically classified into dry-bulb temperature based and enthalpy based. (5) Theoretical studies and simulations show that the determination of the switchover set point of dry-bulb economizer is a trade off between energy consumption and indoor comfort, while enthalpy economizer control maximize saving and comfort level by taking into account both sensible and latent heat of outdoor air(6~8). Simulation results by Wacker (1989) show that enthalpy economizer saves 5%~50% more compressor energy compared to dry-bulb economizer with switchover set point of 70°F in different locations.

In spite of the superiority of enthalpy economizer, its application is greatly impeded by the so-far notoriously unreliable humidity measurement of outdoor air, which is required in current enthalpy economizer control strategies (4, 9). Here are some general considerations apply to commercial Humidity sensors.

1. Design to operate at ambient temperature around 68°F or 77°F, and poor accuracy in temperature other than nominal range.
2. Could not handle wet conditions(condensation, rain, fog or spraying) (10)
3. Large drift happens due to wide humidity and temperature cycles. (11)
4. Inadequate sample air flow can 1) allow undisturbed wet gas to remain in sensor for long periods. 2) Accentuate adsorption and desorption effect in the volume of gas passing through sensor (12).
5. Susceptible to contamination.

To overcome the problems stated above, mixed air humidity is measured instead of outside air humidity, and based on this change a new algorithm is developed in the paper.

The principles and features of current enthalpy measurement methods are introduced first, along with a comparison of characteristics between outside air and mixed air in HVAC system. Secondly, the proposed new control algorithm is presented with detailed implementation procedure. The uncertainty analysis of enthalpy measurement is done to study its impact on energy consumption. Finally, a case study of air handler units in a hospital building with mixed air enthalpy-based economizer control demonstrates the operation results.

2. ENTHALPY MEASUREMENT

As a thermodynamic property, enthalpy can not be directly measured but can be expressed from knowledge of two properties as far as air is concerned: dry-bulb temperature and humidity content. Therefore, dew point sensor or relative humidity sensor is usually installed for obtaining humidity content information. This section will investigate the features of some popular commercial hydrometer, and compare air properties in mixed air duct and outdoor air duct in the aspect of enthalpy measurement.

Detailed descriptions on most available hydrometers for HVAC application are given by ASHRAE 2005 and Wiederhold (1997). [13] Researches have been done to test sensor performances, [10~12, 14~17]).). Table 1 summarizes features of some popular relative humidity sensors and dew point sensors. Studies show that no single type of humidity sensor covers the entire humidity span, however, almost any of the standard humidity sensors can be used in the mid-regions of humidity and temperature (Wiederhold 1975). Narrower temperature and humidity range will greatly improve measurement accuracy.

Based on the concept above, the measurement of mixed air humidity is more reliable than outside air humidity measurement. Table 2 lists the typical air conditions in both outside and mixed air sensor locations for typical climates where economizers are used.

From Table 2, we can see that outside air will experience wider temperature and humidity range, while the RH for mixed air is always less than 95%, and goes through smooth change. When the system is on, the air velocity varies in a range of 300~ 2000fpm, that saturation or condensation on the surface of the system shall never occur. For system which is off at night time and weekend, condensation can happen in outside air duct due to possible high humidity content at night time and the low air velocity, while for the sensor in mixed air duct, it is more likely that the mixed air is the same with return air, which is much more favorable to sensor performance.

Table.1 Humidity Sensor Summary

Sensor Family	**Sensor Type**	**Accuracy**	**Features and Considerations**
Relative Humidity (R.H)	resistive	±1 to 2% RH	1.Temperature dependent
			2.Narrower temperature range: 15~180°F
			3. Low accuracy at low humidity
			4. Not least a noticeable hysteresis.
	capacitive	±2 to 3% RH	1.Temperature dependent
			2.Poor in higher humidity levels and elevated temperatures
			3. Not least a noticeable hysteresis.
			4. Fast Response.
Dew Point	chilled mirror	±0.5 to 1°F	1. Most reliable with wide measurement range
			2.Susceptible to Contamination;
			3.Usually used for sensor calibration
	Saturated-salt	±1 to 2°F	1. Not capable to measure low humidity level
			2. Susceptible to Contamination;
			3. Slow response time.

Note: The sensor accuracies are manufacturer data at nominal condition

Table2. Comparison of Air in Outdoor air Duct and Mixed Air Duct

System Status	Parameters	Outdoor air intake duct	Mixed air duct
On	Temperature range	-10~120°F	50~85°F
	Air velocity	0~500 fpm	300~2000fpm
	Humidity range	0~100% (Possible condensation)	20~90%
Off (Night Time/ Weekend)	Temperature range	-10~120°F	40~85°F
	Air velocity	0~20 fpm	0~50 fpm
	Humidity range	0~100% (Possible condensation)	20~90%

3. IMPLEMENTATION

Since the current problem in implementation lies in unreliability of outdoor air humidity measurement, while mixed air property is nature more favorable to humidity measurement, which can greatly increase the practicability of economizer control, the true enthalpy economizer is implemented by measuring mixed air humidity.

Figure 1 presents the schematic diagram of the enthalpy economizer, Figure 2 shows outside air flow ratio against the outside air temperature.
As can be seen in Figure 1, the upgraded system includes both humidity and temperature sensors in the return air duct and the mixed air duct, and one outdoor air dry-bulb temperature sensor.

If the mixed air enthalpy is 5% higher than the return air enthalpy, or the outside air temperature is 5 °F higher than the return air temperature, modulate the outside, return, and exhaust air damper to maintain the minimal outside air intake. If the mixed air enthalpy is lower than the return air enthalpy and the outside air temperature is between supply air temperature set point and the return air temperature plus 5 °F, full fresh air should be used, called Partial Free Cooling Mode. And if the outside air temperature is lower than the supply air temperature set point, modulate the economizer dampers to maintain the supply air temperature at its set point. See Table 3.

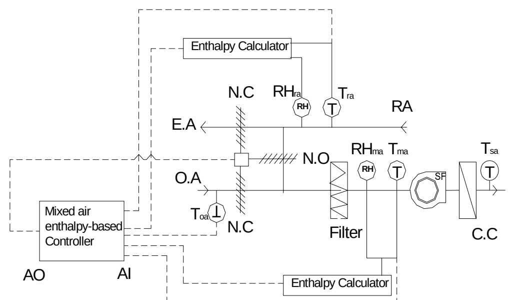
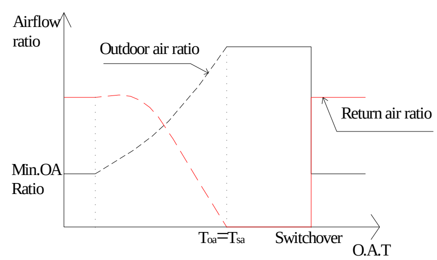


Figure 1 Schematic Diagram of the Enthalpy Economizer



Note: Switchover = [h_{ma}=h_{ra} or T_{oa}>T_{ra}+4}

Figure 2 Air Flow Ratio VS Outdoor Temperature

Table 3 Summary of Enthalpy-Based Economizer Control Using Mixed Air Condition

Condition	Mode of operation
$h_{ma} > (h_r + 5\% \times h_r)$ or $T_{oa} > (T_{ra} + 5)$	Minimum outdoor air intake
$h_{ma} < h_r$ and $T_{oa} < (T_{ra} + 4)$ and $T_{oa} > T_{sa}$	Partial free cooling

$h_{ma} < h_r$ and $T_{oa} < (T_{ra} + 4)$ and $T_{oa} < T_{sa}$	Total free cooling

Since the mixed air enthalpy is used for economizer switchover control, it is ideal to put both sensors in location with better mixing effectiveness, for example, in down stream of mixing chamber or after the air filter. However, there is no perfect mixing effectiveness requirement with this control algorithm, since the economizer will be de-energized whenever the measured enthalpy is higher than the minimum cut-off value, and be energized whenever the measured enthalpy is lower than set point. Therefore, the exact enthalpy value of perfectly mixed air is not a prerequisite.

In the application of the new control algorithm, the switchover happens when measured mixed air enthalpy is 5% higher than return air enthalpy. This slightly higher enthalpy switch algorithm can improve indoor air quality with minimal energy penalty considering the typical sensor's accuracy.

4. IMPACT OF MEASUREMENT UCERTAINTY

The impacts of measurement uncertainty are studied in this section, including the impact of measurement uncertainty on enthalpy calculation, and the impact of this error has had on system energy consumption.

4.1 Instrument Uncertainty Effect on Enthalpy Calculation

Enthalpy is usually obtained from dry-bulb temperature and either relative humidity or dew point temperature in certain pressure level. This section will study the uncertainty sensitivity of enthalpy to relative humidity and dew-point error.

The root sum square method of uncertainty calculation is applied to the individual equations used in calculating enthalpy. It is the preferred method for independent measurement of temperatures and humidity [18, 19]. Rather than sum up the individual contribution of each measurement, the method argues that it is statistically likely that the errors will partially counteract each other most of the time due to their independency such that square root of the sum of the squares of individual uncertainties is a more representative gauge of the overall random uncertainty.

If relative humidity is known, the moist air enthalpy $h_{air}\left(Btu/lb_{da}\right)$ is given by ASHRAE 2005[20]

$$h_{air} = 0.240t + \frac{0.62198 \cdot \phi \cdot p_{ws}}{p - \phi \cdot p_{ws}} \times (1061 + 0.444t) \qquad (1)$$

Where t is the air dry-bulb temperature, °F; ϕ is decimal representation of relative humidity; P_{ws} is the saturation vapor pressure over liquid water for temperature range of 32 to 392°F, psia, given by

$$\ln p_{ws} = C_8 / T + C_9 + C_{10}T + C_{11}T^2 + C_{12}T^3 + C_{13}\ln T$$

(2)

Where, $C_8 \sim C_{13}$ are coefficients can be found in ASHRAE Handbook of Fundamental (2005), T = absolute temperature, °R = °F +459.67

Using the root sum square method, the random uncertainty is expressed in units of Btu/lb_{da} as equation (3):

$$\delta h_{air} = \left[\left(\frac{\partial h}{\partial t}\delta t\right)^2 + \left(\frac{\partial h}{\partial \phi}\delta\phi\right)^2 + \left(\frac{\partial h}{\partial p}\delta p\right)^2\right]^{\frac{1}{2}} \qquad (3)$$

Where δt , $\delta\phi$ and δp are sensor uncertainties in °F, %, and pisa respectively. The partial derivatives represent the sensitivity of enthalpy result to each of the measured parameter and are given by (4), (6) and (7).

$$\frac{\partial h}{\partial t} = 0.24 + 0.444 \times \left(\frac{0.62198 \cdot \phi \cdot p_{ws}}{p - \phi \cdot p_{ws}}\right) + \frac{\partial p_{ws}}{\partial t} \times \left\{(1061 + 0.444t) \times \left[\frac{0.62198 \cdot \phi \cdot p}{(p - \phi \cdot p_{ws})^2}\right]\right\}$$

(4)

Where,

$$\frac{\partial p_{ws}}{\partial t} = p_{ws} \times \left[\frac{-C_8}{T^2} + C_{10} + 2 \cdot C_{11}T + 3 \cdot C_{12}T^2 + \frac{C_{13}}{T}\right]$$

(5)

$$\frac{\partial h}{\partial \phi} = (1061 + 0.444t) \times \left[\frac{0.62198 \cdot p \cdot p_{ws}}{(p - \phi \cdot p_{ws})^2}\right]$$

(6)

$$\frac{\partial h}{\partial p} = -(1061 + 0.444t) \times 0.62198 \cdot \phi \cdot p_{ws} \times \frac{1}{(p - \phi \cdot p_{ws})^2}$$

(7)

If dew-point temperature is known, the moist air enthalpy $h_{air}\left(Btu/lb_{da}\right)$ is given by [20]

$$h_{air} = 0.240t + \frac{0.62198 \cdot p_{ws}(t_d)}{p - p_{ws}(t_d)} \times (1061 + 0.444t)$$

(8)

Using the root sum square method, the random uncertainty is expressed in units of Btu/lb_{da} as:

$$\delta h_{air} = \left[\left(\frac{\partial h}{\partial t}\delta t\right)^2 + \left(\frac{\partial h}{\partial t_d}\delta t_d\right)^2 + \left(\frac{\partial h}{\partial p}\delta p\right)^2\right]^{\frac{1}{2}} \quad (9)$$

The partial derivatives represent the sensitivity of enthalpy result to each of the measured Parameters and are given by (10), (11) and (13).

$$\frac{\partial h}{\partial t} = 0.24 + 0.444 \times \left(\frac{0.62198 \cdot p_{ws}(t_d)}{(p - p_{ws}(t_d))^2}\right)$$

(10)

$$\frac{\partial h}{\partial t_d} = (1061 + 0.444t) \times \frac{0.62198 \cdot p}{(p - p_{ws}(t_d))^2} \times \frac{\partial p_{ws}(t_d)}{\partial t_d}$$

(11)

Where

$$\frac{\partial p_{ws}(t_d)}{\partial t_d} = p_{ws}(t_d) \times \left[\frac{-C_8}{T_d^{\,2}} + C_{10} + 2 \cdot C_{11} T_d + 3 \cdot C_{12} T_d^{\,2} + \frac{C_{13}}{T_d}\right]$$

(12)

$$\frac{\partial h}{\partial p} = -(1061 + 0.444t) \times 0.62198 \cdot p_{ws}(t_d) \times \frac{1}{(p - p_{ws}(t_d))^2}$$

(13)

Figure 3(a) shows the uncertainty in enthalpy vs measured enthalpy at different relative humidity (RH) and temperatures (DB-Temp). It is based on typical instrument uncertainty shown in table 4. In each constant relative humidity line, the temperature change is from 10~95°F, and for each constant Temperature line, the relative humidity is varying from 0.2~1(0.8 for DB=95°F).

Figure 3(b) compares the effect of RH and DP uncertainty on enthalpy uncertainty.

Table 4 Typical uncertainties for temperature, RH and Pressure sensor.

Temperature(d)	Relative humidity(RH)	Pressure
±0.5 °F	±3% rh	±0.019 ps ia

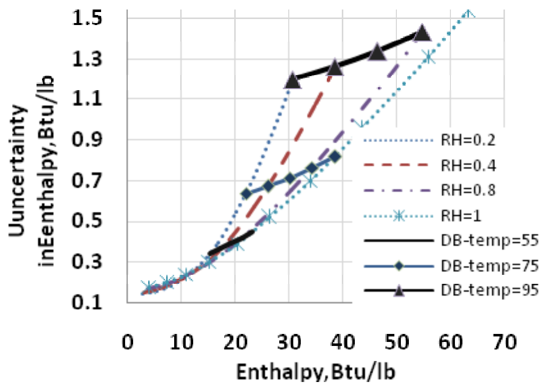


Figure 3 (a) Uncertainty in Enthalpy vs. RH and DB

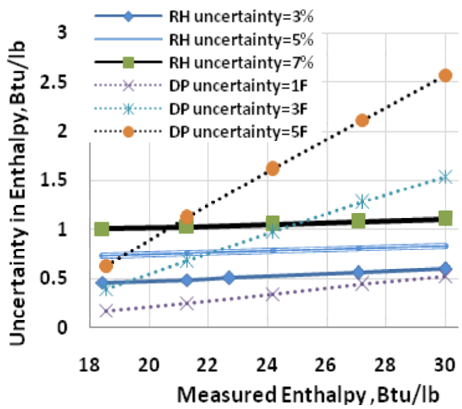


Figure 3 (b) Impact of Sensor Uncertainty on Enthalpy Uncertainty

4.2 Effect of Uncertainty in Enthalpy on System Energy Consumption

The uncertainty in enthalpy will result in discrepancy between the theoretical and practical energy consumption in enthalpy-economizer system. The energy performance is evaluated using energy consumption effect per unit total air flow rate. The energy consumption effect is expressed in ratios of the difference between actual and theoretical consumption over the theoretical value, which is 253507Btu/CFM/Year, based on Omaha, NE weather bin data. The return air condition is assumed to be constant at 75°F and 50%RH. The simulation is based on ideal enthalpy economizer control shown in Table 5. Figure 4 examines the effect of uncertainty in relative humidity measurement on the total annual cooling energy consumption. The supply air temperature is assumed to be 55°F and 95%RH all year round; therefore the measurement uncertainty has no impact on heating energy consumption.

The cooling energy per unit CFM can be calculated as:

$$E_c = \begin{cases} 60 \times \rho \times (h_{ma} - h_{sa}) if h_{ma} > h_{sa} \\ 0 \cdots if h_{ma} \le h_{sa} \end{cases} \quad (14)$$

Table5. Ideal mixed air based enthalpy economizer

Condition	Mode of operation
$h_{ma} > h_r$ or $T_{oa} > (T_{ra}+4)$	Minimum outdoor air =0.2
$h_{ma} \le h_r$ and $T_{oa} < (T_{ra}+4)$ and $T_{oa} > T_{sa}$	Partial free cooling
$h_{ma} \le h_r$ and $T_{oa} < (T_{ra}+4)$ and $T_{oa} < T_{sa}$	Total free cooling

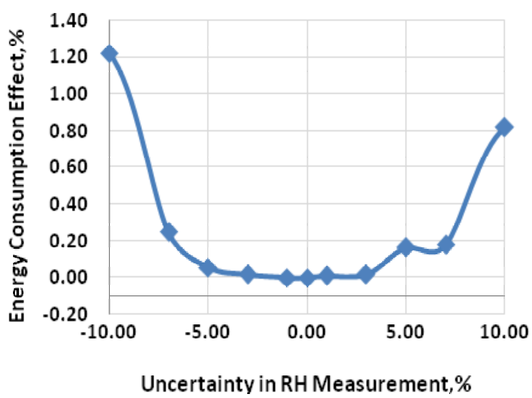


Figure 4 Uncertainties in RH Measurement on Energy Consumption

Fig. 4 shows that when the RH uncertainty range from -10%~+10%, the resulting energy consumption effect ranges from 1.2% ~0.8%. Other parameters' uncertainties are constant based on Table 4. The positive uncertainty in RH results in higher enthalpy value, which equals to the real enthalpy plus enthalpy uncertainty calculated by equation (3). This error cause the potential of earlier switchover to non-economizer mode (MIN.OA mode), and increase mechanical cooling cost by not fully take advantage of free cooling. And the negative uncertainties in enthalpy may cause a late switchover, and results in higher mechanical cooling by using outside air with high enthalpy. As can be seen, uncertainty in RH up to $\pm 10\%$ in mixed air will cause less than 1.2%more energy consumption than ideal enthalpy control. This means even though the possibility of drift after a long term operation, as long as no condensation or malfunction happened on the sensor, the increased energy consumption caused by sensor error is acceptable, considering energy saving of enthalpy economizer over ideal db-temperature based economizer is 17.1% if the switchover is 65°F based on simulation using weather in Omaha, NE.

5. CASE STUDY

The building studied is a hospital located in Tecumseh, NE. This hospital has two major air handler units with similar design capacity and both have economizers, and running 24/7. For comparison, one unit (AHU1) is programmed to use dry-bulb temperature based economizer, and the other one (AHU3) uses mixed-air enthalpy based economizer. For AHU1, the switchover is 64 °F with 2°F control band; and for AHU3, economizer control algorithm based on Table 2 is used. The system is upgraded with required sensors installed. Also, a humidity sensor is installed in outside air intake duct for testing.

The manufacture specifications of the relative humidity sensors installed are listed in Table 6. Both the sensors have been calibrated when installed. Calibrations were done after 4 months and seven months; the drift for both sensors is shown in row 7 of Table 5. The operation condition of the two sensors is summarized in Fig5 (a) and (b). The available data so far is from April 15st, 2007~Aug 22th, 2007.

Table6.Humidity Sensor Operation

Sensing element	Resistance change of bulk polymer		
Accuracy (at 77°F)	$\pm 3\%$ (20~95%)		
Temperature effect	0.06%per°F		
Hysteresis	1%		
Drift	1% per Year		
Measured Drift	In Mixed air Duct	After 4 months	1.8%
		After 7 months	2%
	In Outdoor air Duct	After 4 months	3.5%
		After 7 months	Abnormal

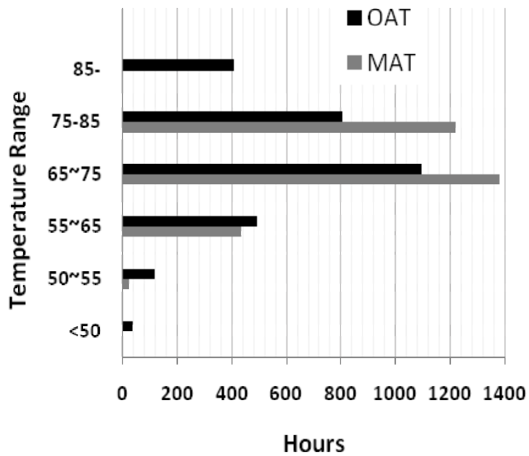


Figure.5 (a) M.A and O.A temperature range

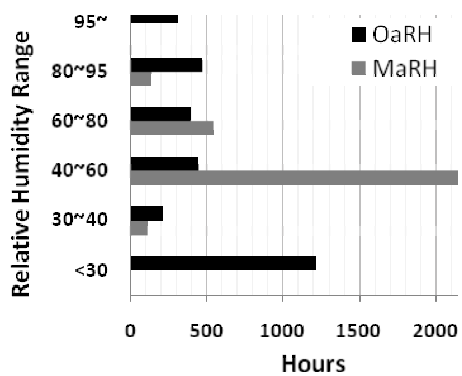


Fig.5 (b) M.A and O.A humidity range

From Table 6, we can see that the measured drifts of sensor in mixed air duct are both in acceptable range, while the sensor in outside air duct gives 1.6% RH compared to measured 61%RH. Trended data shows the sensor has been working abnormal after half year operation, and the malfunction is due to frequent and large humidity and temperature cycle.

In Fig5 (a), it can be found that mixed air temperature range is almost between 55~85°F. More than 80% of the time the temperature band is from 65~85°F, which is close to sensor rating condition, and therefore less temperature effect on humidity measurement and higher accuracy. For outdoor air condition, the temperature range is much wider, and if annual operation data is available, temperature can be from -10~105°F, which means large temperature effect and possible large drift.

In Fig 5(b) we can see that more than 80% of the humidity range in mixed air duct is between 40~60%, which is quite favorable to reliable humidity measurement. Situations that relative humidity is less than 30% seldom happen, and only 5% of the testing period the mixed air humidity are in 80~95% range. For condition of outdoor air sensor, the trended data shows the highest RH is 97.4%, which is out of manufacture rating range. The large amount of hours with outdoor air humidity lower than 30% is due to sensor malfunction.

Table 7 summarizes the performance of the two economizers. The base case in energy saving calculation of enthalpy economizer is db-temperature based economizer.

Table7. Economizer Operation

Testing Period: April.3rd~Aug. 22th ,2007		
	Temperature-based Economizer	Mixed-air enthalpy economizer
Operation hours	888	1251
Energy saving	-	15.7%

6. CONCLUSION

The traditional way to control enthalpy economizer by using outdoor air enthalpy has been largely impeded by poor humidity sensor performance in outdoor air environment. Investigation on principle and testing results of commercial RH sensor shows that air in mixed air duct is more favorable to reliable measurement. This is further supported by a two month field testing, which shows no extreme temperature and relative humidity has ever happened in mixed air duct, while the sensor in outdoor air duct has bear much worse condition. Also, the mixed-air based enthalpy economizer works good with 15.7% mechanical cooling saving.

The study of effect of uncertainty in humidity measurement on enthalpy calculation shows that enthalpy uncertainty is more sensitive to dew point error than error in relative humidity. The resulting uncertainty in enthalpy with 10% uncertainty in relative humidity of mixed air measurement will increase annual energy consumption around 1.2% based on Omaha, weather bin data. This data is quite acceptable if compared with 13% saving using ideal enthalpy economizer instead of temperature based economizer with switchover at 65°F.

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