

## Thermal Anemometer Measurements

The Kurz thermal anemometers use two RTDs, one heated 50 to 100 °C above the ambient, the other monitors the ambient. The current or power required to keep the velocity element heated is the parameter used during calibration.

### Mass Rate

A thermal anemometer, controlled at a fixed temperature above the ambient, responds to convective heat transfer. With forced convective heat transfer, the output is proportional to the sensor's Reynolds number (Re). Looking at the Reynolds number terms we can see how it measures **mass rate** per unit area. It does NOT measure volumetric flow rate but a density weighted version known as **standard flow rate**. The thermal anemometer automatically compensates for density because it responds to the Reynolds number.

Because a thermal anemometer measures the unit-area mass flow, it can be said to measure **mass rate**. In other words, it measures the true velocity, weighted by the actual **density** of the flowing gas. If the mass rate is normalized by a known density, it has velocity units, a term known as **standard velocity**.

### Mass Flow Equations

#### Reynolds Number

Let's look at the Reynolds number since it is proportional to the sensor's power or current when heated X degrees above the ambient:

$$Re = \rho v d / \mu$$

Where

$\rho$  = actual density  
 $v$  = actual velocity  
 $d$  = sensor's diameter  
 $\mu$  = gas viscosity

It is this density and velocity ( **$\rho v$** ) **product** that makes the thermal anemometer a mass flow meter. Density ( $\rho$ ) has units of mass/volume and velocity ( $v$ ) has units of length/time. So the  $\rho v$  product has units of (mass/time)/area or mass rate per unit area.

For example:

$\rho$  is  $\text{kg/m}^3$ ,  $v$  is  $\text{m/s}$

so  $\rho v$  is  $(\text{kg/s})/\text{m}^2$

The sensor is sensitive to the energy that each gas molecules hitting it takes away (convective heat flow) in the form of heat. This energy is proportional to the size and number of molecules that hit the sensor. It does not know about density and velocity, just the number of collision with a cooler gas per unit time all contributing to heat loss. Small light gas molecules like hydrogen ( $\text{H}_2$ ) are more efficient at transferring the vibrational heat energy of the sensor's surface than large heavy molecules like Argon (Ar).

**Standard Velocity** is the  $\rho v$  product normalized to a standard density.

$$\text{Standard Velocity} = \rho v / \rho_s$$

where  $\rho_s$  is the standard gas density. For air this is  $0.07387 \text{ lb/ft}^3$  at  $77 \text{ }^\circ\text{F}$  and  $14.69 \text{ PSIA}$  (or  $1.184 \text{ kg/m}^3$  at  $25 \text{ }^\circ\text{C}$ ,  $101.325 \text{ kPa}$ )

Note: the density units cancel and you are left with velocity ( $\text{m/s}$ ). Typical units are: Standard Feet Per Minute (SFPM) or Standard Meters Per Second (SMPS). If the gas density doubled (you went from  $100 \text{ kPa}$  to  $200 \text{ kPa}$ ) at the same actual velocity, the standard velocity would double. This also means that if the process gas is at the same temperature and pressure as the standard condition the standard velocity and actual velocity are identical.

**Standard Volumetric Flow** is the  $\rho v$  product multiplied by an area (like a pipe cross section), normalized to a standard density

$$\begin{aligned} \text{Standard Volumetric Flow} &= \text{Area} \times (\text{Standard Velocity}) \\ &= A \rho v / \rho_s \end{aligned}$$

where  $A$  is the area:

The units here are volume/time ( $\text{m}^3/\text{s}$ ). Typical Displayed units are:  
 SCFM, Standard Cubic Feet-per-Minute  
 SCMM, Standard Cubic Meters-per-Minute  
 SCFH, Standard Cubic Feet-per-Hour  
 SCMh, Standard Cubic Meters-per-Hour

**Mass Flow** is obtained by simply multiplying the Standard Volumetric Flow by the Standard Density.

$$\text{Mass Flow} = (\text{Standard Volumetric Flow}) \times \rho_s \\ = A\rho v$$

The units here are mass/time (kg/s). Typical units are:  
PPH, Pounds-per-Hour  
KGH, Kilograms-per-Hour

**Standard Density.** Different gases have different standard densities. This is often described using the gas's molecular weight (molar fraction or volumetric % sum of all elements)

$$\rho_s = \rho_{\text{air}}(\text{MW}/\text{MW}_{\text{air}})$$

Then:

$$\text{Mass Flow} = (\text{Standard Volumetric Flow}) \times \rho_{\text{air}}(\text{MW}/\text{MW}_{\text{air}}) \\ = A(v\rho/\rho_s) \rho_{\text{air}}(\text{MW}/\text{MW}_{\text{air}})$$

Air which is about 78% N<sub>2</sub> (MW = 28) and 21% O<sub>2</sub> (MW = 32) plus some trace is

$$0.78 \times 28 = 21.84 \\ 0.21 \times 32 = 6.72 \\ = 28.56$$

when we add the trace elements (CO<sub>2</sub>, Ar, He) we get the established number for Air which is MW<sub>air</sub> = 28.96.

On the other hand a pure gas like methane, CH<sub>4</sub> (C=12 + H=1 x4) MW = 16.04

This is alternately described as a reference density (Air) multiplied by a specific gravity (sg).

$$\rho_s = \rho_{\text{air}}\text{sg}$$

Then:

$$\text{Mass Flow} = (\text{Standard Volumetric Flow}) \times \rho_{\text{air}}\text{sg} \\ = A(v\rho/\rho_s) \rho_{\text{air}}\text{sg}$$

### Conversion between Actual and Standard Flow

Conversion of Standard Velocity or Standard Volumetric Flow to actual requires only scaling the result for the gas density according to the ideal gas law.

$$V_a = V_s (P_s/P_a)(T_a/T_s)$$

or

$$F_a = F_s (P_s/P_a)(T_a/T_s)$$

Where

$V_a$  is actual velocity,  $V_s$  is standard velocity

$F_a$  is actual volumetric flow,  $F_s$  is standard volumetric flow

$P_s$  is the standard pressure in absolute units

$P_a$  is the actual pressure in absolute units

$T_a$  is the actual temperature in absolute units (Kelvin or Rankin)

$T_s$  is the standard temperature in absolute units (Kelvin or Rankin)

Note: °K = °C + 273.16, °R = °F + 459.67

A more accurate conversion also accounts for the compressibility (Z) of the gas (deviation from the ideal gas law). This version looks like this.

$$V_a = V_s (P_s/P_a)(T_a/T_s)(Z_a/Z_s)$$

Where

$Z_a$  is the compressibility of the gas under actual conditions

$Z_s$  is the compressibility of the gas under standard conditions.

All other terms are the same

The compressibility of a gas is a function of temperature and pressure. High pressures and low temperatures cause the largest deviations from the ideal gas law and the value of  $Z= 1.0$ . For small changes in reference conditions, the Z values are very close and can be ignored.

## Known Flow Measurement Error Sources

A short description of known error sources is provided for reference:

- Gas property induced errors
- Flow Profile control
- Acoustic noise sources
- *Gas species decomposition (not verified yet)*

### Gas Property Induced Errors

Here are secondary effects which are known to cause mistracking of the ideal thermal anemometer.

- **Pressure changes** will affect the calibration for some gasses. For example, N<sub>2</sub> has a large 2.5% /100 psi shift in its viscosity which changes its mass flow reading the same amount. By contrast He has nearly no viscosity change with pressure.

The gas compressibility Z can also a factor for some gases. Looking at CH<sub>4</sub> for example:

20 °C, 101.3 kPa, Z= 0.9975

20 °C, 800 kPa or 7.9 bar, Z= 0.9875 or 1% deviation.

- **Temperature changes** will affect the gas thermal conductivity and viscosity so the calibration will drift. This is typically 2.5% /100 °C. The minimum drift occurs near 3000 SFPM where the dynamic temperature compensation is performed. The use of velocity temperature mapping (VTM) or multiple calibration curves for different gas temperatures, largely eliminates this source of error.
- **Temperature profiles** in the pipe will produce flow errors. This is caused by using uninsulated pipe upstream of the sensor where the gas is above or below the ambient temperature.
- **Low flow free convective** heat transfer forces compete with forced convective and conductive heat transfer forces for power. This causes measurable errors (depending on gas type, temperature, pressure, and orientation of sensor to both flow and gravity) starting at about 300 SFPM and becomes significant down at about 100 SFPM.
- **Wet vs. Dry Flow Rate:** The thermal anemometer responds to all gas molecules which hit it. In the case of water vapor(H<sub>2</sub>O) dissolved in Air, it reads what is know as wet standard volumetric flow or WSCFM. For intake combustion processes you want to know the dry standard volumetric flow or DSCFM which is 21% O<sub>2</sub> so your

fuel air ratio can be properly computed. Knowing the specific humidity ratio  $\omega$  you can use the following equation up to 5%  $\omega$  and get results within 1%:

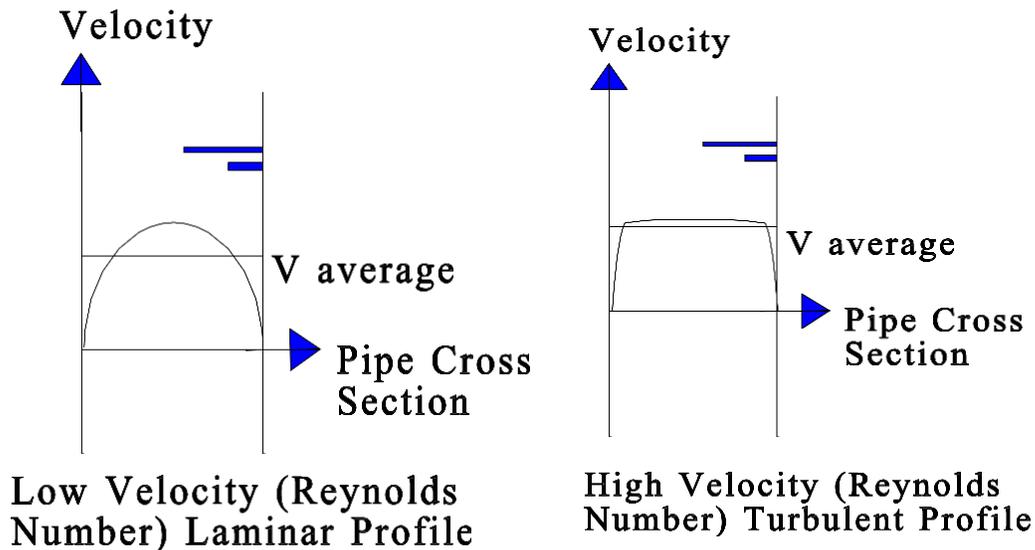
$$\text{DSCFM} = \text{WSCFM} \times 0.622 / (0.622 + \omega)$$

For example. In Monterey California, a typical dew point of 14 °C corresponds to 70 grains/lb of Air based on the Psychrometric chart. Since there are 7000 grains per pound this is 1% specific humidity ( $\omega = 0.01$ ). Therefore  $\text{DSCFM} = \text{WSCFM} \times 0.622 / (0.622 + 0.01) = \text{WSCFM} \times 0.984$ . In other words, 1.6% if the Air by volume is water vapor. For a derivation of the above equation see the technical note [364018](#).

### Flow Profiles And Correction Factors

At low velocity, a laminar velocity profile develops across the pipe cross section as shown in the figure. Note that the peak velocity is about 30% higher than the velocity average (V average).

At higher flow rates, a flatter velocity profile develops where the peak velocity is closer to the average. So depending on



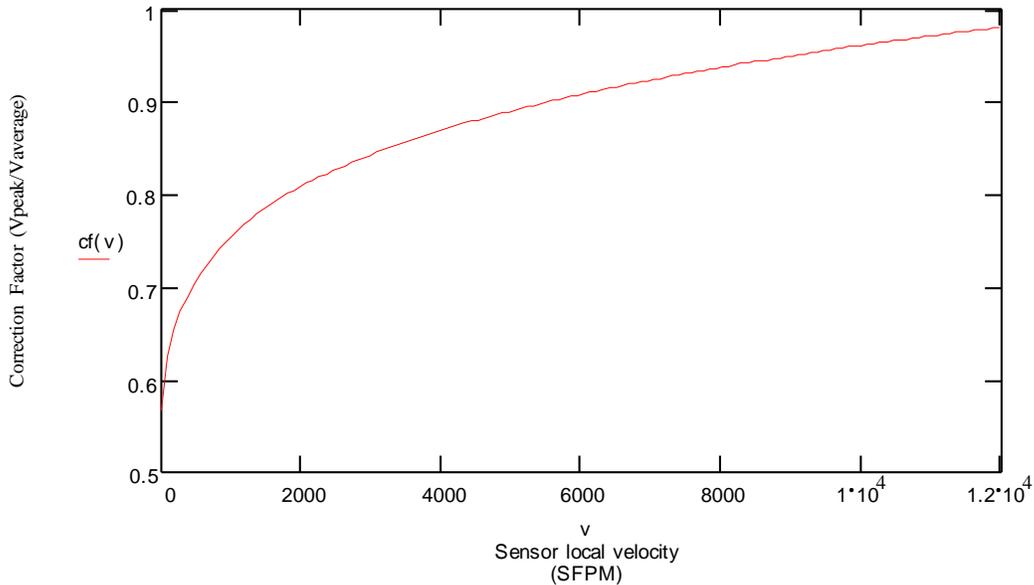
where the sensor is located, it will read a different fraction of the average velocity. It is the average velocity multiplied by the cross sectional area that will obtain the total flow.

### Correction Factors

The use of a velocity dependent correction factor can convert the local velocity measurement to average velocity.

$$\text{Flow} = V_{\text{local}} * \text{Area} * \text{cf}(V_{\text{local}})$$

The above correction factor curve was measured from a 4" ID pipe with a 2" welded support, triple sting CD sensor. For other sized ducts, the data can be scaled by the



Reynolds Number.

### Acoustic, sound or pulsing pressure induced errors.

Pneumatic control valves, reciprocating pumps, sonic nozzles, pressure regulators etc. all tend to induced pressure pulsations in to the piping or ducting. At high enough amplitude, the gas molecules will collide with the sensor more than normal due to the sound waves. This causes a thermal sensor to loose heat to this gas more than normal. So the sensor will read higher flow than is actually flowing. The source of this pressure wave can be upstream or downstream of the sensor.

Valves with a large pressure drop across them (sonic flow in the valve) can generate a lot of wideband (think rocket motor but lower amplitude) acoustic noise downstream of the valve. Placing a control valve with this kind of noise upstream of the thermal sensor will cause it to read high whereas if it is downstream of the sensor the valve noise will have little to no effect on the flow readings.

In some plumbing configurations the pulsations will be sympathetic with the standing wave sound reflections in the pipes which will allow the amplitude to grow very high.

Starting at the fundamental and all the harmonics of the standing wave pattern this phenomena can aggravate a flow measurement with a thermal anemometer.

### **Mitigation of acoustic induced errors**

To prevent a standing wave of sound in the piping, we simply need to introduce enough sound absorption to prevent the buildup of a large sound wave with each reflection from each end of the piping section where the flow meter is in.

Elimination of these acoustic induced errors can be controlled using expansion tanks, mufflers etc. to attenuate the sound waves and return the piping/duct to a smooth one-way flow field so the thermal sensor does not produce false high readings. Inserting a tank between the noise source and sensor is often all that is required to correct this error source. Even a branching tee with a short stub to a tank will attenuate the sound enough to correct most applications which suffer from high acoustic noise. In some applications, a control valve and shut off valve can be swapped so the control valve is not in the same sound-resonator as the sensor.

Valve noise can be attenuated by shear distance between the valve and the flow meter. Rubber hose of several hundred D in length has been shown to be effective at attenuating valve acoustic noise. Often the easiest solution is to place the valve after the flow meter, then is sonic noise is not directed to the thermal sensor. Some systems can have their system pressure adjusted so the control valve upstream of the thermal flow meter is not running sonic. It is also worth mentioning that sonic valves after the flow meter will isolate any acoustic noise downstream of the valve from propagating upstream to the thermal sensor as it cannot travel through the sonic nozzle in the opposite direction.

### ***Gas species decomposition from the velocity sensor heat (Not experimentally verified yet)***

*Some gas species, like ozone ( $O_3$ ) can be induced to decompose from the heated velocity sensor. As it requires heat to split up a molecule, the thermal sensor will read high under this condition. The heat of recombination, O combines with O to make  $O_2$  occurs downstream of the sensor and is not detected. So while the net chemical reaction from  $O_3$  to  $O_2$  is exothermic giving off heat, the first step is endothermic. In this case, the only mitigation is to reduce the application temperature or the sensor overheat to below the activation energy needed to initiate this decomposition. In the case of  $O_3$ , this is 80 to 100 °C.*

*The opposite issue where the sensor surface could act as a catalyst for a recombination and the sensor picks up too much heat, so it reads low, has not been observed but is possible. Higher temperature applications are more likely to observe catalytic heat absorption.*

## Use Of The Flow Equations In The Kurz Mass Flow Computer

**Single Point Insertion Flow Elements** like the 410, 450, 452 and 454FT flow transmitter are calibrated as velocity devices in gas X. You can display standard velocity, or with application specific information you can display standard volumetric flow and mass flow:

- Area,
- Sensor and probe support blockage
- Correction factor (velocity profile)
- Gas specific gravity when reading mass flow

**Multi-point Insertion Flow Elements** (K-BAR) are also calibrated as a velocity devices in gas X. You can display standard velocity or with application specific information you can display standard volumetric flow and mass flow:

- Area
- Sensor and probe support blockage
- Correction factor (velocity profile). This tends to be automatic since the velocity is measured across the duct at equal area locations.
- Gas specific gravity or MW , when reading mass flow.

**In-line Flow Elements** (510, 502, 522UHP, 532 534) and the 504 and 534FT flow transmitter are calibrated as standard volumetric flow devices in gas X. You can display standard volumetric flow or with application specific information it will display standard velocity or mass flow:

- Area,
- Sensor and probe support blockage
- Correction factor (velocity profile)
- Gas specific gravity or MW when reading mass flow

To maintain the Factory calibration on in-line units requires adherence to the recommended L/D upstream and downstream criteria. This ensures the long pipe run velocity profile when used in the field.

Example L/D criteria:

Model 502-16 or 504FT-16

L/D is from the heated sensor to the disturbance	90 ° Elbow at x L/D	Calibration Error
	10	11 %
	20	2.5 %
	30	< 0.5 %

**Built-in Flow Conditioners.**

These lengths are reduced significantly [factors of 16, area-ratio<sup>2</sup>. Area-ratio = Free pipe area/(venturi throat area)] for flow elements and transmitters that have a venturi flow conditioner like the 534FT and 542.

You can close couple elbows and line size changes to the 534FT with less than a 2% change in the reading. Of course, with metering runs before and after the meter you avoid this flow profile induced error source.

**Problems:**

Air flow of 100,000 lb/hr through a 3' x 3' square duct, 90 °F, 20 PSIG

- a. What is mass flow in SCFM \_\_\_\_\_
- b. What is velocity in SFPM \_\_\_\_\_
- c. The actual velocity is \_\_\_\_\_
- d. What range does Kurz calibrate to \_\_\_\_\_

Nitrogen flow of 10 ACFM through a 3" Schedule 40 pipe, 110 °F, 50 PSIG

- e. What is the area \_\_\_\_\_
- f. What is the flow rate in SCFM \_\_\_\_\_
- g. The velocity in SFPM is \_\_\_\_\_
- h. The calibration max. range is \_\_\_\_\_

Answers on the Next Page:

Worked out answers to the previous questions.

Kurz Standard Reference Conditions.

Air density:  $0.0739 \text{ lb/ft}^3$  at  $77 \text{ }^\circ\text{F}$  and  $14.69 \text{ psia}$ ,

a:  $100,000 \text{ lb/hr} \times (1 \text{ hr}/60 \text{ min}) \times 1 \text{ ft}^3/0.0739 \text{ lb} = 22,553 \text{ ft}^3/\text{min}$  which is  $22,553 \text{ SCFM}$  since our density was at standard conditions.

b:  $\text{Velocity} = (\text{Volume rate}) / \text{Area} = 22,553 / (3 \times 3)$   
 $= 2506 \text{ SFPM}$ .

c:  $V_a = V_s (P_s/P_a)(T_a/T_s) = 2506 (14.69/\{20 + 14.69\})(460 + 90)/(460 + 77)$   
 $= 1087 \text{ FPM}$ .

d: Calibrated rates are about 1.5 times the expected maximum.  
 So  $2506 \text{ SFPM} \times 1.5 \sim 4000 \text{ SFPM}$ .

e: ID of schedule 40 3" pipe is  $\sim 3"$   $\text{Area} = 3.14 D^2/4$   
 $= 3.14(3/12)^2 / 4 = 0.0491 \text{ ft}^2$ .

f: Same as C above but the other way around with flow.  
 $F_s = F_a(P_a/P_s)(T_s/T_a) = 10 (50 + 14.69)/14.69 (460 + 77)/(460 + 110)$   
 $= 41.49 \text{ SCFM}$ .

g:  $\text{Volumetric Rate} / \text{Area} = \text{Velocity} = 41.49 \text{ SCFM} / 0.0491 \text{ ft}^2$   
 $= 845 \text{ SFPM}$ .

h: Calibrated rates are about 1.5 times the expected maximum.  
 So  $41.49 \text{ SCFM} \times 1.5 \sim 60 \text{ SCFM}$ .

**Chapter L**