# Dynamic Air consumption of control valves

**M.T. Coughran**
*Research Specialist, EnTech Control, a division of Emerson Process Management, and formerly with Fisher Controls International PLC*

When comparing new automobiles, would you rely on fuel consumption data measured only while the engines were idling? Probably not, but that is equivalent to the static air consumption data used today by purchasers and vendors of control valve positioners. This paper shows that static energy consumption can be misleading, particularly for pneumatic positioners with low bleed rates, in process loops where the input signal changes frequently.

Energy consumption of the control systems is part of the operating cost of a process plant. However, attempting to minimise energy consumption by restricting energy flow to the final control element may compromise the purpose of the control loop. Degrading the performance of the control loops reduces product quality, reduces plant output, or wastes process fluids far more valuable than the instrument air. The operating cost of the control valve must be weighed against the control valve's effect on the control loop.

For air-operated control valves, the pneumatic power at constant supply pressure is proportional to the mass flow rate of instrument air. On most control valves used in closed loop control, where the input signal is continuously changing, the pneumatic instrument is a positioner. When considering the operating costs of candidate positioners, one is tempted to use the only flow data normally available: static air consumption. S75.13[1] gave procedures for measuring the static consumption, i.e. with the device at rest*. However, S75.13 also gave separate procedures for frequency response (sine wave input) tests, using stem motion to measure performance. Today sensors are available for measuring the dynamic air consumption during a frequency response test. This paper shows that some surprising results can occur with this more-realistic test method.

## Basic Principles

***Zero consumption*** is not a legitimate design goal. The final control element must consume energy if it moves in response to controller output changes, which occur after any disturbance to a closed-loop control system. Furthermore, in many applications the instrument air is bled into actuator enclosures to keep corrosive atmospheres out[2]. The lower limit of this beneficial bleed has not been established by standards.

Control valves that are rarely required to move are an exception to this principle and are not covered by the conclusions of this paper. Even for natural gas pipelines where the control valves are located remotely and powered by the gas, static evaluation of the candidate positioners is reasonable (and will result in lowest emissions) only if the disturbances are small.

***Static consumption*** differences of 0.1-0.2 Nm$^3$/h (4-8 scfh) are not economically significant. *Table 1* compares two digital two-wire positioners intended for use on a single-acting actuator. Positioners such as design B are called "low-bleed" because they use on-off solenoid valves (driven by piezoelectric or electromagnetic actuators) to achieve low static consumption. Pulsing of air to and from the actuator is done by "bang-zero-bang" control using a microprocessor. The third column of the table shows that Positioner B had advertised static air consumption 0.17 normal m$^3$ per hour (Nm$^3$/h) lower than Positioner A.

Unit costs of instrument air vary considerably and are not often revealed by plant operators. One book[3], based on U.S. Department of Energy data, gives a compressor station example with total costs (depreciation plus operation) of $0.66 per 1000 standard cubic feet, or $0.025 per Nm$^3$**. A static consumption difference of 0.17 Nm$^3$/h, then, amounts to only $37/year in operating costs. This difference in consumption is a small fraction of total costs for a control valve. For example, Reference 2 estimates that purchase, installation, operation, repair, etc of a control valve typically totals greater than $700/year in chemical plants.

However, two misconceptions are inherent to this economic comparison. One misconception is the assumption of equal performance. *Table 1* shows poorer performance of the control valve with Positioner B for small amplitudes (dead band), medium amplitudes (frequency response with ±2.5% sine waves), and large amplitudes (stroking time). The second misconception is that static consumption accurately reflects actual consumption in an operating process plant.

***For low-bleed devices,*** static specifications under-predict the actual consumption with a dynamic input signal.

---

**A positioner test standard being developed by IEC65B similarly requires that consumption be measured statically.*

***The standard cubic foot is defined at 60 °F and the normal cubic meter is defined at 0 °C. Therefore the volume conversion is 1 scf = 0.0265 Nm$^3$.*

| Type | Design Basis | Static Air Consumption | | Measured Stem Motion Performance | | | |
|---|---|---|---|---|---|---|---|
| | | Advertised at 1.4 bar supply (Nm3/h) | Measured (present) at 2.4 bar supply (Nm3/h) | Small Amplitude | Medium Amplitude (±2.5%) | Large Amplitude 100% Stroking Time | |
| | | | | Dead Band [1] (%) | Frequency of -3 dB Amplitude (Hz) | Opening (sec) | Closing (sec) |
| A | Nozzle-flapper and proportional relay | <0.26 | 0.25 | < 0.13 | 0.8 | 3.0 | 3.2 |
| B | Solenoid on-off valves with finite dead zone | <0.09 | 0.05 | 0.5 | 0.4 | 11.2 | 4.7 |

*Table 1: Digital positioners installed for the 50-mm valve and 450-cm2 actuator.*

The mean (time-average) air consumption, when operating in closed-loop control, depends on several aspects of the control valve design, the controller tuning, and the frequency and amplitude of loop disturbances. The simplest open-loop signal that can fairly simulate closed-loop control is the sine wave with appropriate frequency. Such tests, shown here, prove there is negligible difference in dynamic consumption of these two positioners in the medium-amplitude range, where Positioner B gives reasonably good performance.

### Test Methods

For a straightforward comparison, positioners A and B were tested sequentially on one valve and actuator. A 50-mm (2-inch) globe valve with PTFE packing was installed for an air-to-open diaphragm actuator with an effective area of 450 $cm^2$ (69 $in^2$) and a supply pressure of 2.4 bar (35 psig). The total sliding friction in the assembly was 130 N (30 lbf), measured from input signal ramps at 2% per second. Each digital positioner was installed according to its manufacturer's instructions. The manufacturer of the air mass flow rate sensor advertised a first-order dynamic response with 3dB of attenuation at 0.8 Hz. The calibrated range of the sensor was 0 to 7.16 $Nm^3/h$ (270 scfh). *Figure 1* shows the three signals recorded. Because the dynamic air flow sensor was the only item not required by S75.13, the test represented minimal additional investment in equipment.

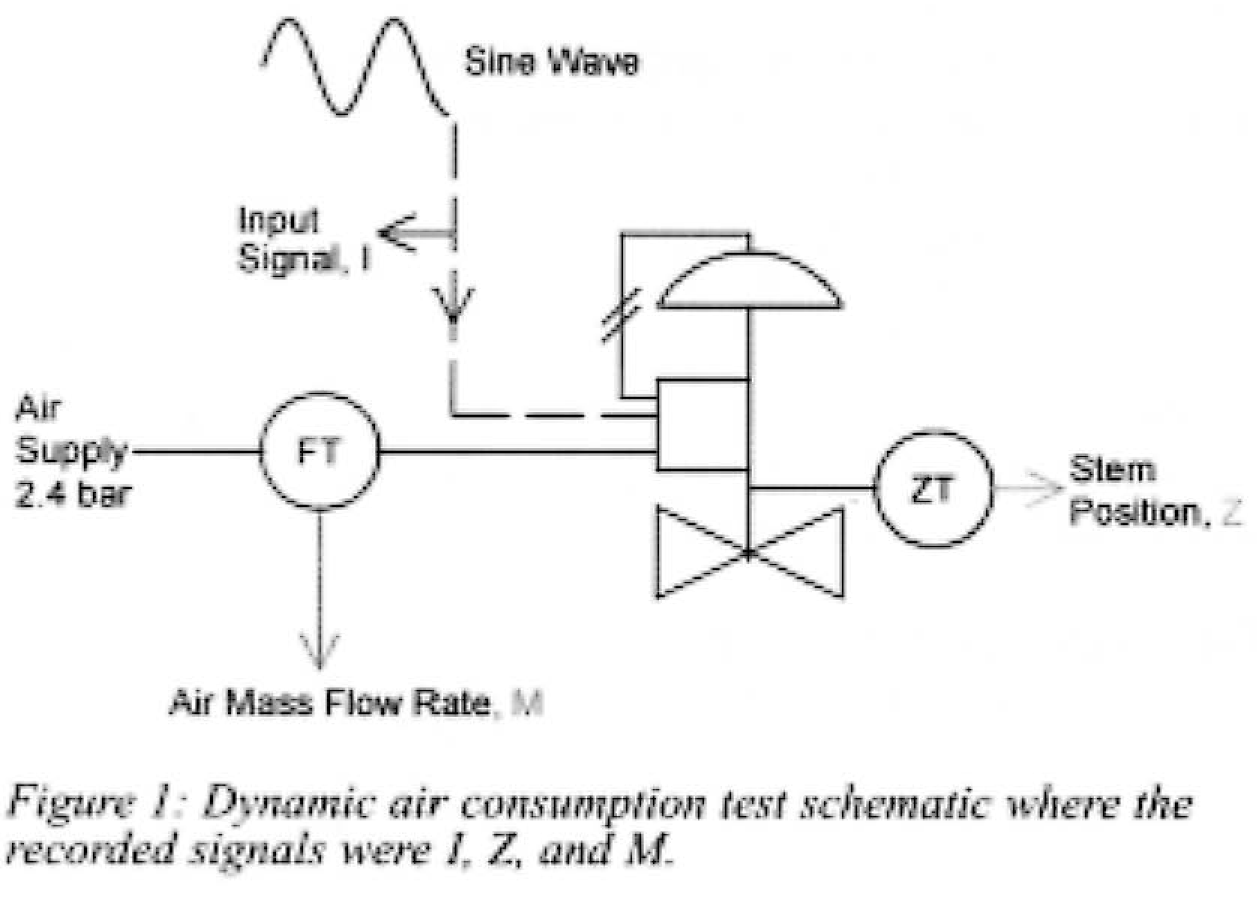

*Figure 1: Dynamic air consumption test schematic where the recorded signals were I, Z, and M.*

Positioner A used a nozzle-flapper followed by a proportional-acting poppet valve, similar to conventional analog designs. Positioner B used on-off valves with variable duty cycle; such designs normally require a programmed dead zone (defined in ref[4]) to prevent chattering near the null state. In this case, the positioner dead zone was 0.5%, which determined the overall dead band of the control valve, listed in *Table 1* (also 0.5%). *Table 1* shows that Positioner A gave better results at all amplitudes: smaller dead band, amplitude response flat to a higher frequency, and faster full-scale stroking.

The input sine wave amplitude was set at ±2.5% to simulate throttling control with a moderate range of controller output. This amplitude also allowed Positioner B to work outside its dead zone, in its optimal amplitude region. Mean consumption was calculated over at least seven cycles of input.

Other signal shapes are available, including steps or square wave, triangle wave, and pulse. However, judging from controller output time series data in actual process plants, sinusoids are more realistic for this purpose. From the principles of Fourier analysis, the controller output can be envisioned as a sum of sinusoids. Of course, it is important to select realistic amplitude and frequency of the signal. The reasons for selecting the "medium amplitude" are given above. Concerning the frequency, this paper shows a range of frequencies reasonable for process plants—less than 1.0 Hz—from which the reader may select the frequency appropriate for a particular control loop. The significant design factors of the valve/actuator were held constant by testing on one common valve/actuator with typical friction, actuator size, and supply pressure.

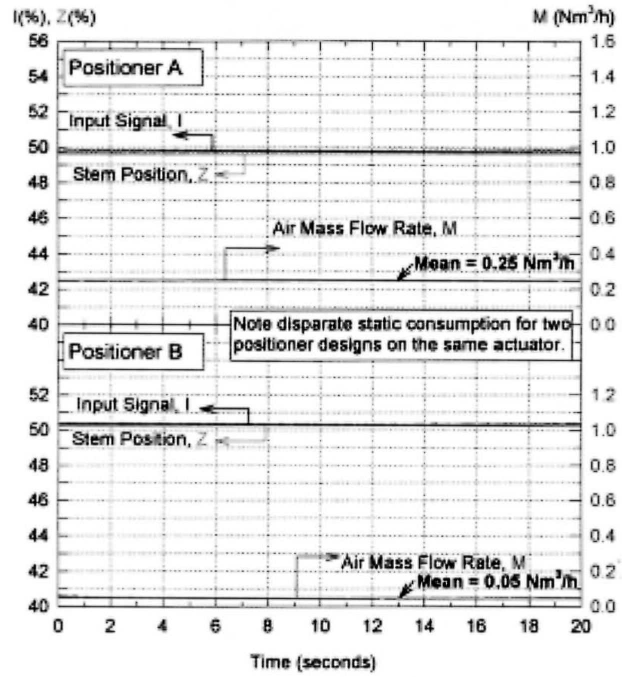

*Figure 2: Static test; constant input signal, per existing standards.*

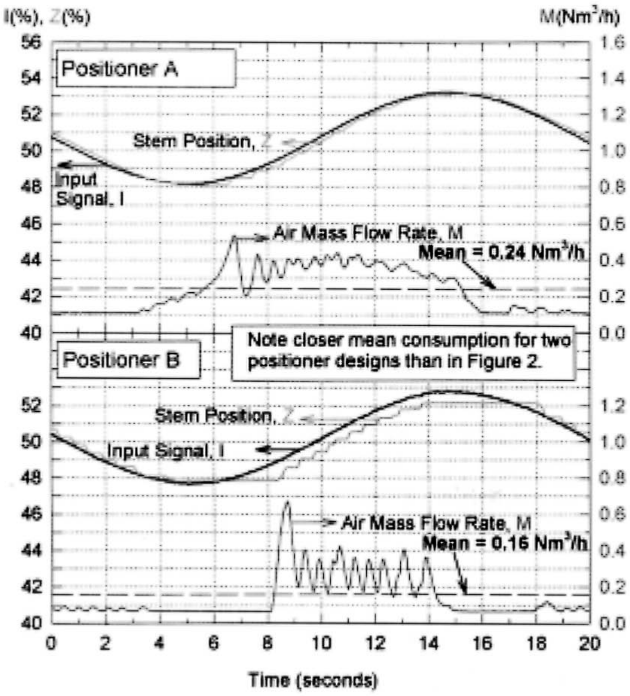

*Figure 3: 0.05 Hz sine wave inputs.*

## Results

*Figure 2* shows results from a static test as prescribed by S75.13. The difference in mean static consumption for the two positioners was 0.20 Nm³/h at 2.4 bar. This is reasonably consistent with the difference in static consumption advertised by the two manufacturers at 1.4 bar (column 3 of *Table 1*).

Dynamic results with a slow (0.05 Hz) input sine wave are shown in *Figure 3*. The lower graph shows the expected result that the instantaneous air consumption for Positioner B had a threshold value similar to the static consumption, with significantly more air consumed when the valve stem was rising (actuator filling).

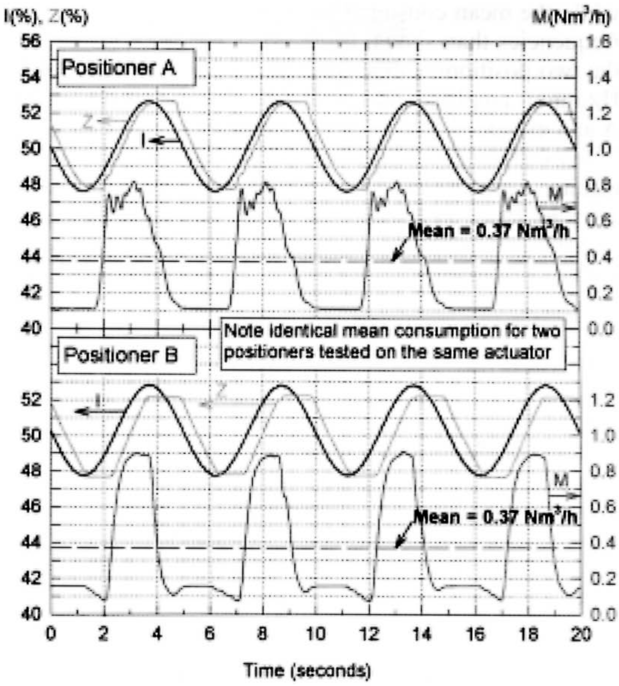

*Figure 4: 0.2 Hz sine wave inputs.*

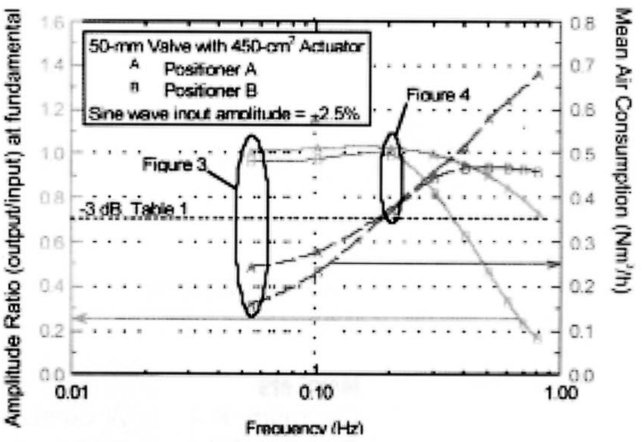

*Figure 5: Power consumption (right) and performance (left) at various frequencies.*

Positioner A, on the other hand, consumed less air than the static value when the valve stem moved downward. The net result was that Positioner A consumed on average (dashed trace) only 0.08 Nm³/h more than Positioner B during 0.05 Hz input. Using the unit cost example from earlier, this is equivalent to paying approximately $18/year for Positioner A's smaller dead band and faster response (*Table 1*). The benefit to the plant of superior performance varies significantly from one control loop to the next, and must be judged by the user; however, in many cases, it is much larger than $18/year.

*Figure 4* shows a higher input frequency, 0.2 Hz. Note the repeatable behaviour from one cycle to the next for each positioner. At this frequency and amplitude, mean consumption of the two positioners was essentially identical at 0.37 Nm³/h.

*Figure 5* condenses the performance and consumption results over a range of frequencies. At each frequency, frequency response analysis was performed and the amplitude ratio at the fundamental frequency is displayed at left,