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Recommended Oils for use with Norgren Lubricators

Oils listed in the following tables may be used in Norgren Lubricators. All oils, however, may not be compatible with the equipment to be lubricated. Contact your lubricant supplier and the builder of the equipment to be lubricated to obtain specific lubricant recommendations.

Light Duty Tool Lubricants (ISO 3448 Viscosity Grades 10 to 22) for Small Air Tools

Manufacturer	Product	ISO 3448 Viscosity Number
BRITISH PETROLEUM	Energal HLP10	10
CASTROL	Hyspin AWS10	10
MOBIL	Velocite Oil No. 6	10
SHELL	Tellus R10	10
TOTAL	Azolla ZS10	10
CENTURY	P198	15
DUCKHAMS	Zeroflo 15	15
ELF	Elfolna 15	15
ESSO	Nuto H15	15
MOBIL	Velocite Oil No. 8	15
ROCOL	MO-4	15
MOBIL	Velocite Oil No. 10	22
SHELL	Tellus 22	22

Heavy Duty Tool Lubricants (ISO 3448 Viscosity Grades 32 to 68) for Standard Air Tools and Cylinders

Manufacturer	Product	ISO 3448 Viscosity Number
BRITISH PETROLEUM	Energal HLP32	32
CASTROL	Hyspin AWS32	32
CENTURY	PWLA	32
ELF	Elfolna 32	32
ESSO	Nuto H32	32
GULF	Harmony 32	32
MOBIL	SHC 524	32
MOBIL	DTE 24	32
MOBIL	DTE Oil Light	32
MOBIL	Gargoyle Arctic Oil Light	32
SHELL	Tellus 32	32
TEXACO	Rondo 32	32
SHELL	Tellus 37	37
GULF	Hydrasil 46	46
BRITISH PETROLEUM	Energal HLP46	46
SHELL	Tellus 46	46
TEXACO	Regal Oil R & 046	46
ROCOL	MO-10	46
BRITISH PETROLEUM	Energal HLP68	68 *

Low Temperature Applications - For use when temperature remains below 32°F (0°C)

Manufacturer	Product	ISO 3448 Viscosity Number
MOBIL	SHC 624	32
DUCKHAMS	Zeroflo 32	32
MOBIL	S4C 524	32

Antifreeze Lubrication

Manufacturer	Product
CASTROL	Antifreeze Mist Lubricant

* For certain applications requiring heavy oil.

Assembly Greases

The following greases may be used in the reassembly of Norgren Airline Equipment.

Industrial Compressed Air Applications

Manufacturer	Product
BRITISH PETROLEUM	Energrease LS2 (non-silicone)
DOW CORNING	Molykote BR-2 Plus (non-silicone)
CARLETON-STUART	Magnalube G (non-silicone)
DOW CORNING	DC44 (silicone)
DOW CORNING	DC55 (silicone)

Low Temperature Industrial Compressed Air Applications below -20°F (-29°C)

Manufacturer	Product
KLUBER	Unisilicon L50/2 (silicone)

Food and Potable Water Applications

Manufacturer	Product
DOW CORNING	Molykote 111 (silicone)

Activated Carbon
A filter material (media) used to remove odors and oil vapors from compressed air.

Aerosol
A suspension of fine liquid particles in gas, i.e., smoke, fog, and mist

After Cooler
A heat exchanger mounted on a compressor outlet to extract the heat of compression.

Air-Piloted Pressure Regulator
See Pilot Operated Regulator

Ambient
The normal condition of air usually defined by temperature or humidity in the vicinity of the equipment being evaluated.

Automatic Drain
A mechanical float used to open and drain excess fluids from a filter bowl (these drains are normally open and use pressure to close)

Average Flow
Typical rate of instantaneous flow over a given timeframe (used to determine compressor capacity requirement). See Instantaneous Flow.

Back Pressure Regulator
See relief valve

Balanced Valve, Regulator
A term used to denote the use of equal pressure on both sides of regulator valve to balance and minimize effect of changing primary pressure.

Bearing Lubricator
Micro-Fog lubricator (See Micro-Fog Lubricator) normally used to lubricate machine bearings.

Beverage Regulator
Regulator normally used for tank systems, often high-pressure, of inert gases (i.e., CO2) and used in conjunction with beverage dispensing equipment.

Check Valve
A device which allow flow in one direction only.

Coalescing:
The action causing small oil aerosols to combines through adsorption to form larger, heavier, droplets for removal form the compressed air stream.

Coalescing Filter
Filter using coalescing element to remove oil aerosol particles.

CFM
Cubic Feet per Minute, a volumetric flow measurement at a specified pressure.

Cv
A dimensionless number expressing the flow capability or conductance of a fixed orifice of a pneumatic device

Constant Bleed
A method used to reduce or eliminate droop in a regulator to provide higher performance, more precise flow and faster regulation characteristics.

Creep
Used to describe increase in secondary pressure (P2) due to a leak from primary pressure (P1).

Critical Back Pressure Ratio (CBPR)
CBPR is equal to 53% with compressed air; CBPR is when downstream pressure is less than 53% of the upstream pressure reducing downstream pressure further will not increase flow.

Dead End Service
Implies a secondary volume is pressurized, however, under normal circumstances the volume does not flow air but acts as a holding vessel.

Deliquescent Dryer
An air dryer utilizing chemicals for purposes of dewpoint suppression by absorbing water from a compressed air stream.

Delta-P Indicator
See Service Life Indicator, Pressure Drop indicator

Desiccant
A material creating adherence, or adsorption, of water to its surface. Often used in regenerative or desiccant dryers to lower the dewpoint for removal of water from compressed air.

Dew Point
The temperature at which moisture begins to condense to liquid at a constant pressure. The temperature at which air is saturated is the dew point and at this point relative humidity is equal to 100%.

Drip Leg Drain
A device placed at the low point of a main air distribution system used to removed condensed liquid water from the compressed air stream where water is likely to accumulate.

Droop
A term used to describe the initial pressure drop seen in a regulator flow curve (caused by seal impingement).

Dump Valves
See Exhaust Valves

Electronic Regulator
Electronically controlled regulator used to reduce inlet pressures to a regulated process range output.

Emulsion
A mixture of oil and water.

Excelon
A Norgren trade name used to describe a modular FRL product using a patented one-piece clamping mechanism called a Quikclamp.

Exhaust Valves
Valve used to evacuate downstream air of the valve itself (warning- air can be trapped by other pneumatic devices downstream of the valve).

Feedback Regulator
A special pilot regulator designed to sense and control pressure more accurately at a remote point downstream of the pilot operated regulator to provide more consistent pressure control.

Free Air
Air at ambient conditions usually measured at a compressor intake (compressors are rated in CFM of Free Air @ Intake).

Flow
The rate of air allowed to pass through a device or orifice at a given pressure, usually in SCFM or dm3/sec. Two expressions are commonly associated with flow: instantaneous flow and average flow.

Flow Curve
A performance chart showing various flow rates for a specific component at specific pressure settings and pressure drop.

General Purpose Filter
Filter used for common liquid extraction and particle removal, normally filtered to 40 or 5 micron particle size.

General Purpose Regulator
A common regulator used to provide normal industrial pressure regulation.

Instantaneous Flow
Flow rate demanded within a single cycle of a compressed air system (used to size pneumatic components of valves, air preparation, fittings, and conductors)

Instrument Regulator
A higher precision regulator used to provide exceptionally close tolerance pressure regulation.

Integral Filter/Regulator
A unit combining the function of a filter and regulator into one space-saving unit.

Integral Pilot Operated Regulator
A unit combing the integral pilot operated regulator with the main control regulator

I/P Pressure Converter
Essentially a precision electronic regulator (see also Electronic Regulator). Also know as E/P, P/I, and P/E converters (I = current, E = voltage, P = pressure).

Machine Lubricator
See Micro-Fog or Bearing Lubricator.

Manual Drain
A valve that requires manually open and closure to drain liquids from a filter bowl (sometimes used with lubricator bowls).

Manifolding Regulator
A regulator with two common P1 ports and two common P2 ports which can be assembled in a single or multiple output fashion for various pressure settings using a solitary inlet air line.

Manifold Block
Block used to connect multiple air line products.

Master Regulator
A term sometimes used to refer to the controlling regulator of a Pilot Operated Regulator.

Membrane Dryer
A variable dewpoint suppression device using hollow fibers which has an affinity for water, whereby moisture-laden air enters into the fibers and the water vapor permeates through the walls to be removed from the air stream.

Micro-Fog Lubricators
A Norgren trade name for lubricators designed to atomize oil to an aerosol within the airflow, usually less than 2 microns in size, to provide longer travel distance at a higher consistency of suspended oil in pipes. Typically used with distances up to 50 feet or when more than one tool requires lubrication from a single lubricator.

Micron
A measurement of size equal to one-millionth of a meter (.000039 inch).

NACE
National Association of Corrosion Engineers specifying standards used to determine corrosion resistance requirements of products.

Non-Return Valve
See Check Valve

Oil-Fog Lubricator
A Norgren trade name for a lubricator designed to heterogeneously disperse oil mist, having particles size usually larger than 2 microns to small droplets. Typically used for short distances or when only one tool requires lubrication.

Odor Removal filter
See Oil Vapor Removal filter.

Oil Vapor Removal Filter
Filter utilizing an activated carbon to remove oil vapor. Also used to remove odors from air stream.

Olympian
A Norgren trade name to describe European manufactured FRL products utilizing a modular yoke design.

P1
See Primary Pressure

P2
See Secondary Pressure.

Pilot Operated or Slave Regulator
A regulator, which has its outlet pressure, controlled by air pilot pressure instead of a spring. Requires another pressure regulator to control the pilot pressure.

Porting Block
Modular device providing a pipe port(s) from a main airflow stream or pneumatic control mechanism.

Precision Pressure Regulator
A regulator used to provide the most precise pressure regulation without electrical feedback (usually a constant bleed regulator).

Pressure Switch
Devices to monitor pressure and/or provide an output signal in a system.

Pounds Per Square Inch Absolute (PISA)
Pressure including atmospheric pressure. Atmospheric pressure is caused by the weight of the atmosphere pushing down on the earth's surface. (Atmospheric pressure is equal to 14.7 at sea level.)

Pounds Per Square Inch Gauge (PISG)
Pressure related and visual on a gauge, not taking into account atmospheric pressure

Pounds per Square Inch Drop (PSID)
The amount of pressure loss, in PSI, incurred by the flow of air through a device (losses usually due to friction and other factors).

Pressure Reducing Device
See Regulator, Pressure

Primary Pressure
Supply or inlet pressure, P1.

Prismatic Lens
A Norgren trade name used to describe a special bowl liquid level indicator, a bowl sight lens.

Quikclamp
Norgren trade name used to describe the patented one-piece clamping mechanism for Norgren EXCELON(r) products

QuikDrain
A Norgren trade name for the patented manual 1/4 turn drain for Norgren Bowls.

Refrigeration Dryer
An air dryer utilizing refrigeration to cool air for purposes of dewpoint suppression in order condense and remove water from a compressed air stream.

Regenerative Dryer
An air dryer utilizing desiccant for purposes of dewpoint suppression by method of adsorption in order to accumulate and remove water from a compressed air stream.

Regulation Curve
A performance chart of a regulator used to determine the expected fluctuation of secondary pressure under constant flow with varying primary pressure.

Regulator, Pressure
A device used to reduce (regulate) air pressure in a pneumatic system to a desired working level.

Relief Valve
A device used to reduce likelihood of over pressurization of a system by relieving air at a given pressure set point.

Relative Humidity
The ratio of actual water present in a volume of air, as measured as a percentage of air completely saturated at the same temperature.

Remote Fill Device
A device used to fill lubricator bowls from a remote location.

Reverse Flow Regulator
A regulator designed to exhaust downstream air with loss of upstream pressure.

SCFM
Standard Cubic Feet per Minute, SCFM converts CFM to standard conditions: 14.7 PSIZ, 68oF ambient temperature, and 35% relative humidity (See also CFM).

Secondary Pressure
Regulated pressure downstream of a specific device, P2.

Semi-automatic Drain
A drain that requires manual operation to drain when pressurized, but opens automatically in the absence of pressure. Sometimes referred to as an overnight drain.

Service Life Indicator
Device used to visually or electronically indicate pressure drop (Delta-P), variance in P1 to P2, across a filter used to warn of filter clogging.

Sight Feed Dome
The control device adjustment on a lubricator used to increase or decrease oil feed rate.

Slave regulator
See Pilot Operated Regulator

Smooth Start Valve
A valve allowing slow pressure build-up of a system to an intermediate pressure before permitting a step-up to full line pressure. Used to prevent sudden movement, damage, and potential failure of equipment during start-up.

Soft Start Valve
See Smooth Start Valve

SSU or Viscosity
Sabolt Universal Seconds, the time in seconds for 60 milliliters of oil to flow through a standard orifice at a given temperature.

Standard Conditions
The basis of comparison of ambient conditions, the specification used to convert CFM as ambient conditions for direct comparison @ SCFM. (For compressed air: temperature of 680- 720F, relative humidity of 35%, and atmospheric pressure of 14.7 psia.)

Terminal Velocity
In compressed air applications, terminal (critical) velocity is reached when the downstream pressure (P2) is less than 53% of the upstream pressure (P1). See also Critical Back Pressure Ratio.

Tool Lubricator
See Oil-Fog Lubricator.

Unbalanced Valve
A term used to denote where primary pressure is allowed to act as a closing force across the valve seat area. Changes in primary pressure (P1) can impact the secondary pressure setting (P2).

Yoke
A term used to describe the connection or mounting method used in the Olympian Series Product.

IMPROVING THE "BOTTOM LINE" WITH PNEUMATICS.
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Abstract:
The cost of compressed air continues to be a very elusive subject in industrial applications. The notion that compressed air is free is a common misconception. Countless attempts at making end users aware of the costs to produce compressed air have largely been ignored. It seems that Operations Managers are concerned with the power consumption of a couple of sixty-watt light bulbs, but couldn't care less about the power required to run a 400 horsepower compressor! Let's put this into perspective, one horsepower equals 745.7 Watts. You can light 4,971 sixty-watt light bulbs with the same power it takes to run a 400 horsepower compressor! Indeed, the single largest electrical appliance in a manufacturing plant may be the compressor motor.

This paper will provide some common sense approaches to determine the cost of compressed air required in a typical pneumatic circuit. It is the discovery of the approach and the requisite considerations that we hope to explore. By clearing up several popular pneumatic myths, we will be able to make better decisions to reduce the cost of compressed air. Compressed air costs are typically hidden in the operating overheads of most companies. **Any overhead cost reduction immediately falls to the bottom line in the form of profit.** The real test of this improved bottom line is determining the additional sales a company must generate to produce a similar profit.

Real Costs

Experts in the compressed air field suggest the cost to produce compressed air varies from $.15 to $.40 per 1000 Standard Cubic Feet (SCF) (1), (2), depending on geographical location. In spite of these estimates, a large sector of the user public fails to complete the simple calculations to determine what a machine will require in terms of Standard Cubic Feet per Minute (SCFM) of compressed air, let alone for an entire year or the life of the equipment. In order to improve the bottom line with pneumatics, we must also expose three popular pneumatic myths!

Myth #1: Compressed Air is Free
Myth #2: Pipe Size = Right Size
Myth #3: If a little bit's good, a whole lot's better

Myth #1: Compressed Air is Free!

In the past most people wouldn't take the time to determine the cost of the compressed air required by an actuator for a year. It is ironic that hydraulic system designers have to do the calculations in order to determine the size of the power unit required to operate hydraulic cylinders. For far too long, little effort has been made to recognize the similarities between hydraulic and pneumatic systems. Typically, fluid power people have defended the differences between hydraulic and pneumatic systems. By recognizing the similarities, we can design and service the two mediums from similar perspectives, including the need for safety, conservation, component sizing, and cost justification.

There is a developing interest in determining the cost of compressed air in a pneumatic system. We are beginning to see users specify 60 PSIG as the maximum pressure range for a pneumatic system. Pneumatic component manufacturers are somewhat paranoid about discussing the cost of compressed air in fear they might encourage customers to apply a substitute for compressed air. Since compressed air is readily available, affordable, clean, and has less force hazards than hydraulic power, it seems reasonable that compressed air will continue to be applied in the industrial sector.

Real Common Sense

To properly apply pneumatic components in a system, the first component to be considered is the actuator. We have seen a large number of cylinders grossly oversized resulting in poor actuator performance, wasted compressed air, and high initial component costs. **Oversizing an actuator by one bore size can result in a fifty percent increase in the cost of compressed air required for the application.** If the cylinder is sized to move more than twice the load at the design pressure, the cylinder speed will be adversely affected, and the cost of compressed air will also increase. Using this simple observation can result in significant savings

If care is taken at this step of the design process, every component upstream of the actuator (valves, conductors, fittings, filters, regulators, and lubricators) will have a better chance of being correctly sized and applied. A good and correct start in the process is essential to having an efficient operating system.

Real Calculations

A Cylinder Flow Calculation is required for a number of reasons. It takes into consideration the force required to move the load at the specified pressure, the extend and retract stroke volumes of the cylinder, the cycles per minute, the operating air pressure, and a conversion to Standard Cubic Feet per Minute (SCFM). SCFM is the value used by most pneumatic fluid power manufacturers to apply the correct components in a system. SCFM is also linked to valve sizing using Coefficient of Flow (Cv)

Once the cylinder flow calculation is completed, the designer can determine the correct tubing, fittings, valves, and the Filter, Regulator, and Lubricator (FRL) for the application. In spite of the obvious benefits this information provides, we find few designers attempting these critical calculations.

In a typical circuit (Figure 1.) comprised of a double-acting pneumatic cylinder, a five-port, four-way valve, and two flow controls, the typical approach to sizing exposes another popular pneumatic myth.

Myth #2: "Pipesize = Right Size".

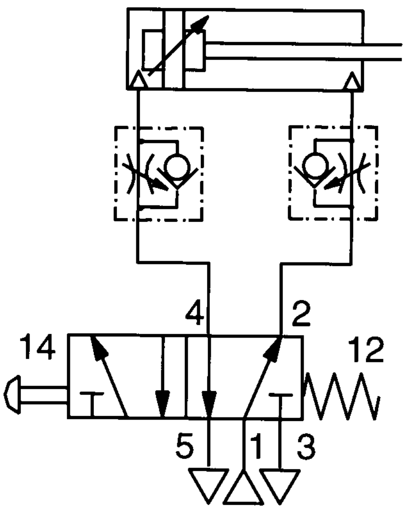


Figure 1. Typical valve/cylinder/flow control circuit

If the cylinder selected had a 1/2" pipe port, most installers would apply 1/2" flow controls, 1/2" pipe and fittings, a 1/2" valve, and quite possibly, a 1/2" FRL! This approach leads to oversized, high priced components and higher long term operating compressed air costs over the life of the equipment. As a result, oversizing components occurs frequently and quickly leads to another popular pneumatic myth.

Myth #3: If a little bit's good, a whole lot's better!
Real Life Example?

Let's consider the following example. We have a cylinder load that requires 500 pounds of force to move in the extend direction only. The retract stroke has no effective load. We want to move the load twelve inches and be able to do this at 30 cycles per minute, eight hours per day, five days per week, 50 weeks per year. The cylinder will be operated on a horizontal plane. Compressed air pressure is 80 PSIG. Of course, we want this cylinder to move as fast as possible (AFAP)!

Specifications:
Load: 500 pounds
Operating Pressure: 80 PSIG
Stroke Length: 12 inches
Cycle Rate: 30 cycles per minute (CPM)
Time to Extend: 1 second
Time to retract: 1 second
Cylinder Velocity: AFAP
Real Solution?

There are five steps required to accurately calculate a cylinder flow rate (in SCFM). The calculations are necessary to achieve accurate results. After the calculations are completed the system designer has the information needed to make sound, cost effective decisions that include downsizing components and conserving compressed air. (Calculations rounded to two decimal places.)

Step #1: Size the Cylinder for Max Performance

This is the stage in sizing a cylinder where we find many designers throwing in a little extra safety factor to cover breakaway forces of a cylinder! (If a little bit's good, a whole lot's better!)

Based on common practice and the orientation of the cylinder, we have found the range of the force multiplier to be between 1.25 to 2.00 times the load (3) being moved, at the specified pressure. This range of values will provide adequate force compensation in the calculations and need not be exceeded.

To size the cylinder for maximum performance (quickest stroke time) we will apply the X2 rule. Multiply the load by two and apply the correct cylinder at the specified pressure. In this case, 500 pounds X 2 = 1,000 pounds. Using the Formula, Force is equal to Pressure times Area (F=PA) we solve for the cross-sectional area we need for the cylinder bore. Written another way to solve for the Area, the formula looks like A=F/P. 1000(80=12.50 in.”. 12.50 in.” is the cross-sectional area of the bore needed to move our 500 pound load as fast as possible! This is very close to a standard NFPA bore of 4"(12.57 in.”). It is IMPORTANT to note here that any larger bore size will move slower at 80 PSIG, and any smaller bore size will also move slower. Use a 4" bore cylinder.

Step #2: Calculate total volume per cycle

Total volume per cycle requires some examination of the cylinder we will be applying. We need to recognize that the extend stroke volume will be more than the retract stroke volume on a typical double acting, single-rod, cylinder, due to the volume displacement of the rod.

Since we've selected an NFPA 4" bore cylinder we will apply the standard 1" rod after we've checked to avoid cylinder rod buckling! **Extend Volume** is equal to the bore cross-sectional area times the stroke length. 12.57 in.” X 12 in. = **150.84 $in.^3$** **Retract Volume** (compensating for the rod) calculation is (12.57 $in.^2$ - .79 $in.^2$) X 12 in. = 141.36 $in.^3$ **Total volume per cycle is:**
150.84 $in.^3$ + 141.36 $in.^3$ = **292.20 $in.^3$**

For a 5" bore the total volume per cycle is:
235.68 in.“ + 226.20 in.“ = 461.88 in.“ or a 58% increase in the volume of the 4" bore cylinder!

Step #3: Calculate total volume per minute

Multiply the total volume per cycle by the number of cycles per minute. 292.20 in.“/cycle X 30 CPM = **8,766 $in.^3$ per minute (CIM)**

Step #4: Convert CIM to Cubic Feet/Minute (CFM). (NOTE: this calculation step is often overlooked) 8,766 $in.^3$ ÷ 1728 $in.^3$ = 5.07 CFM

Step #5: Convert CFM to SCFM

This conversion reduces the cylinder flow calculation to the necessary and required terms.
To make this conversion we must recognize the compression ratio of the compressed air being used in the application. Compression ratio is the working pressure expressed in absolute terms and converts compressed air to standard

conditions. (14.7 PSIA, 36% Relative Humidity, and 68º Fahrenheit temperature). In most industrial applications, the ambient temperature and relative humidity can be ignored since these variables have little impact on the calculations.

Our **Compression Ratio** (CR) calculation:
(80 + 14.7) ÷ 14.7 = 6.44

Multiplying the CFM by the CR = SCFM,
5.07 CFM X 6.44 CR = 32.65 SCFM!

32.65 SCFM for this application seems fairly harmless until you complete the compressed air cost evaluation.

The cylinder flow calculation provides the necessary information (SCFM) to more accurately determine the correct valve (Cv) and the proper FRL for the system. Without the cylinder flow calculation, sizing the rest of the components in the system will be accomplished empirically, (trial and error approach), or by using Myth #2.

With the cylinder flow calculation complete, we can move on to the real cost of compressed air for the application and sizing the rest of the components in the system.

Real Horsepower (HP) Requirements

Most compressor representatives will use a few rules of thumb to determine the compressor capacity required for an application. Depending on the type of compressor used, compressors are typically rated to deliver four to five SCFM per horsepower (rule of thumb). Most compressor representatives will strongly recommend a duty cycle of 50% to 75% (rule of thumb), again depending on the type of compressor. Duty Cycle is the percentage of time the compressor motor is generally running under loaded conditions. **In our application, at 50% duty cycle, and at 4 SCFM/ HP, a 32.65 SCFM application will require an additional compressor capacity of 16.32 HP!** (32.65 SCFM÷4 SCFM/HP)÷50% Duty Cycle = 16.32 HP

Real Compressed Air Costs!

If we carry our SCFM calculations out to the number of SCF per year, per shift, pretty soon we are talking about some serious compressed air usage! Consider how many minutes there are in an eight-hour day, five days a week, fifty weeks in a year? (60 minutes/hour X 8 hours/day X 5 days/week X 50 weeks/year) =
120,000 minutes/year per eight hour shift! (That's 360,000 minutes/year for three shifts!)

32.65 SCFM X 120,000 Minutes = 3,918,000 SCF/Year!
The 5" bore cylinder would require 51.6 SCFM, or 6,196,890 SCF/Year!

If your average cost per 1,000 SCF is only $.25, the cost of compressed air to operate this one 4" bore cylinder for one shift, for the year is an incredible, **$979.50!** [(3,918,000÷ 1,000) X .25]
Obviously, if your cost of compressed air is $.50/1,000 SCF, your annual cost would be **$1,959.00!**

For the 5" bore, air consumption costs jump to $1,549.22 (at $.25/1,000 SCF) and $3,098.45 (at $.50/1,000 SCF). Over a 50% increase for one bore size increase!

Real Concern

The cost of compressed air actually used is a major concern for most manufacturing companies. If we could reduce the compressed air consumption in our system by 30%, most Chief Executive Officers, Chief Financial Officers, and plant engineers would leap at the opportunity! Let's consider another approach to our application.

Since our 500 pound load is only being moved in the extend direction, we could consider lowering the air pressure to return the cylinder. For example, say we were able to lower the return pressure from 80 PSIG to 20 PSIG. How would that impact on the total system air consumption?

If you recall from Step #2 above, the retract stroke volume was **141.36 in.^{3}** per cycle, or about 48.4% of the total cycle volume. Without taking you through the additional math, **the compressed air cost for the extend stroke at $.25/1000SCF is (51.6% of $979.50) $505.42/year.**

By changing the pressure on the return stroke to 20 PSIG the compressed air consumption is reduced from 15.8 SCFM to 5.8 SCFM. The impact on the compressed air bill for the retract stroke is a reduction from $474.08/year to $174.03, **a reduction of $300.00 or 30%.**

Even if the price of a 1/2" regulator was $50 you would be able to expect a payback on the cost of the regulator in about two months. That is, IF we use Myth #2: Pipe Size = Right Size, or Myth #3: If a little bit's good, a whole lot's better!

Please keep in mind this example is for only one 4" cylinder. How many cylinders are you applying? What is your cost of compressed air?

When we examine the application even closer, we are able to save even more in initial costs by properly sizing the valve, the fittings, and tubing by the use of Coefficient of Flow (Cv). It is sufficient to say the use of Cv's to size a system is reasonably accurate and provides an element of cushion in most system calculations.

There are other approaches to conserving compressed air in typical applications. If cylinder speed is not important, using a force multiplier between 1.25 and 2.0 times the load will result in smaller cylinders and less air consumption. Applying single-acting cylinders could significantly reduce the long-term cost of compressed air

Real System Cv?

Using Cv's we can evaluate the typical circuit (Figure 1.) for potential bottlenecks! Each component in the circuit has a Cv. With some effort it is possible to determine the Cv of the cylinder port, Cv of the flow control (in both the free flow direction as well as the WIDE OPEN controlled flow direction), Cv of the piping/tubing and fittings, and the Cv of the directional control valve. **The System Cv is ALWAYS going to be less than the component with the smallest Cv in the system!** A strong recommendation here is to make the most restrictive component ("the weakest link"!) **(4)** in the system, the most expensive component (usually the directional control valve). This will minimize the initial cost of the components in the system. It is fair to say that a 1/2" directional valve will cost more than a 1/4" valve. The difference in pipe or tubing cost is marginally different.

Real Serious Savings?

With dual pressure savings, you can see the fast payback on applying additional regulators. More impressive savings can be realized by finding the leaks in the compressed air system, and eliminating them! We have seen reports suggesting compressed air losses due to system leaks and artificial demand range from 20% to 45%! **(5)** (Perhaps the high loss due to leaks is why compressor representatives suggest the 50-75% duty cycle for compressors?) Compressed air leaks, left unattended, will continue to grow in size and flow due to the abrasive effect of the air-line contamination and particulate matter continuing to attack the leak orifice. (6) The sooner leaks are discovered and repaired, the less waste there is in power required to produce the compressed air. Less wasted air reduces operating costs, and can justify the expense of a maintenance patrol to quickly repair air leaks.

As you can see in Figure 2., even small diameter leaks can be very expensive.

Orifice Size	Leak Rate (SCFM) @100 PSIG	Annual Cost @$ .25/1000 SCF
1/16" (.0625)	5.02	$633
1/8" (.125)	20.1	$2,533
1/4" (.25)	80.4	$10,130
3/8" (.375)	180.8	$22,781
1/2" (.50)	321.4	$40,496

Figure 2. Flow & Cost of Leaks per Orifice Size

The values in Figure 2, are based on twenty-four hours per day, seven days per week, fifty weeks (8400 hours).

Real Maintenance

After all of this discussion to reduce the operating costs associated with wasting compressed air we must mention another, less obvious, source of waste. Users should regularly check for excess pressure drop across air filters. By applying pressure drop indicators, (also called service life indicators, or delta P indicators) and changing filter elements with greater frequency, you will avoid the escalating cost of the pressure drop across the filter element. Electronic and mechanical pressure drop indicators are commercially available to provide reminders to service the filter elements on a regular basis. Using pressure switches to monitor regulated pressure in the system will avoid surpassing the X2 multiplier, ensure efficient use of compressed air, and provide optimum performance of the system.

In hydraulic systems, a pressure drop across a filter has serious consequences effecting the entire hydraulic power unit adversely. Poor maintenance on hydraulic filters results in catastrophic failures. In hydraulic systems, leaks are quickly repaired due to the obvious hazards and cost associated with hydraulic oil.

Rarely does poor maintenance on a pneumatic filter result in catastrophic failure. However, excessive pressure drop across a pneumatic filter is an ongoing operating cost that is hidden from view. In pneumatic systems, leaks are often ignored until they become so annoying (uncomfortably loud), or they have caused such a significant system pressure drop, that they MUST be repaired!

Summary

If you don't care about the cost of compressed air in your plant, don't do the calculations, and you will perpetuate the three pneumatic myths! Consider the similarities between pneumatic and hydraulic systems, rather then the differences.

If increased profits are of interest to you, than you'll find a hidden profit center in the cost-effective use of compressed air. If you reduce your compressed air overhead costs, avoid over - sizing components, and design your systems to operate at an optimum pressure, you can improve system performance, and you will improve the bottom line. The first step to recognizing the potential savings available to you is completing these simple calculations.

With a **little** time and effort you can make a **big** improvement on your bottom line with pneumatics.

References

1) "Compressed Air Handbook ", Electric Power Research Institute, Palo Alto, CA. 1994, pp12.
2) Wagner, H.H., "Your Total Compressor Cost May Be Too High" Plant Engineering, May 1, 1999.
3) Fleischer, H., "Stop Oversizing Pneumatic Components" Machine Design, June 3, 1999, pp 101-106.
4) Fleischer, H., "Manual of Pneumatic Systems Optimization" Chapter 2. Conductance (Cv), 1st ed., McGraw-Hill, New York, 1995, pp 25-66
5) Foss, R.S. "Improving Air System Efficiency" Part 1, Hydraulics & Pneumatics, April 1998, pp 41-68
6) Foss, R.S. "Improving Air System Efficiency" Part 7, Hydraulics & Pneumatics, July 1999, pp 33-79

Acknowledgements

A special thanks to my Norgren colleagues, Jim Dietvorst, Brad Dittmer, Doug Kelly, Jerry Martin, and Joseph Quinn, for their patience, support and technical expertise. Cf.

Determining Proper Air Valve Size

Most manufacturers catalogs give flow rating C_V for the valve, which was established using proposed National Fluid Power Association (NFPA) standard T3.21.3. The following tables and formulas will enable you to quickly size a valve properly. The traditional, often used, approach of using the valve size equivalent to the port in the cylinder can be very costly. Cylinder speed, not port size, should be the determining factor.

The following C_V calculations are based upon simplified formulas which yield results with acceptable accuracy under the following standard condition:

Air at a temperature of 68°F (20°C)

Absolute downstream or secondary pressure must be 53% of absolute inlet or primary pressure or greater. Below 53%, the air velocity may become sonic and the Cv formula does not apply. To calculate air flow to atmosphere, enter outlet pressure p_2 as 53% of absolute p_2. Pressure drop Δ P would be 47% of absolute inlet pressure. These valves have been calculated for a C_V = 1 in Table 3.

Nomenclature

B	Pressure drop factor	
C	Compression factor	
C_V	Flow factor	
D	Cylinder Diameter	(IN)
F	Cylinder Area	(SQ IN)
L	Cylinder Stroke	(IN)
p_1	Inlet or Primary Pressure	(PSIG)
p_2	Outlet or Secondary Pressure	(PSIG)
Δ P	Pressure differential (p_1 - p_2)	(PSID)
q	Air flow at actual condition	(CFM)
Q	Air flow of free air	(SCFM)
t	Time to complete one cylinder stroke	(SEC)
T	Absolute temperature at operating pressure Deg R = Deg F + 460	(R°)

Valve Sizing for Cylinder Actuation – (Method A)

$$C_V = \frac{\text{cylinder area (Sq In) (see Table 1) } \mathbf{F} \times \text{cylinder stroke (IN) } \mathbf{L} \times \text{Compression factor (see Table 2) } \mathbf{C}}{\text{Pressure drop factor (See Table 2) } \mathbf{B} \times \text{time to complete cylinder stroke (SEC) } \mathbf{t} \times \mathbf{29}}$$

Example:
Cylinder size 4" Dia. x 10" stroke. Time to extend: 2 seconds. Inlet pressure 90 psig. Allowable pressure drop 5 psid. Determine C_V.

Solution:	Table 1	F = 12.57 sq in
	Table 2	C = 7.1
		B = 21.6

$$C_V = \frac{12.57 \times 10 \times 7.1}{21.6 \times 2 \times 29} = .7$$

Select a valve that has a C_V factor of .7 or higher. In most cases a 1/4" valve would be sufficient.

It is considered good engineering practice to limit the pressure drop Δ P to approximately 10% of primary pressure p_1. The smaller the allowable pressure drop, the larger the required valve will become.

After the minimum required C_V has been calculated, the proper size valve can be selected from the catalog.

Bore Size D (in)	Push Bore F (sq in)	Bore Size D (in)	Push Bore F (sq in)
3/4"	.44	4"	12.57
1"	.79	4-1/2"	15.90
1-1/8"	.99	5"	19.64
1-1/4"	1.23	6"	28.27
1-1/2"	1.77	7"	38.48
1-3/4"	2.41	8"	50.27
2"	3.14	10"	78.54
2-1/2"	4.91	12"	113.10
3-1/4"	8.30	14"	153.94

Table 1: Cylinder push bore area F for standard size cylinders

Inlet Pressure (psig)	Compression Factor C	Pressure Drop Factor B for various Pressure Drops Δ P 2 psig	5 psid	10 psid	15 psid	20 psid
10	1.7	6.5				
20	2.4	7.8	11.8			
30	3.0	8.9	13.6	18.0		
40	3.7	9.9	15.3	20.5	23.6	
50	4.4	10.8	16.7	22.6	26.4	29.0
60	5.1	11.7	18.1	24.6	29.0	32.0
70	5.8	12.5	19.3	26.5	31.3	34.8
80	6.4	13.2	20.5	28.2	33.5	37.4
90	7.1	13.9	21.6	29.8	35.5	39.9
100	7.8	14.5	22.7	31.3	37.4	42.1
110	8.5	15.2	23.7	32.8	39.3	44.3
120	9.2	15.8	24.7	34.2	41.0	46.4
130	9.8	16.4	25.6	35.5	42.7	48.4
140	10.5	16.9	26.5	36.8	44.3	50.3
150	11.2	17.5	27.4	38.1	45.9	52.1
160	11.9	18.0	28.2	39.3	47.4	53.9
170	12.6	18.5	29.0	40.5	48.9	55.6
180	13.2	19.0	29.8	41.6	50.3	57.2
190	13.9	19.5	30.6	42.7	51.7	58.9
200	14.6	20.0	31.4	43.8	53.0	60.4
210	15.3	20.4	32.1	44.9	54.3	62.0
220	16.0	20.9	32.8	45.9	55.6	63.5
230	16.7	21.3	33.5	46.9	56.8	64.9
240	17.3	21.8	34.2	47.9	58.1	66.3
250	18.0	22.2	34.9	48.9	59.3	67.7

Table 2: Compression Factor C and pressure drop Factor B.

Valve Sizing with C_V = 1 Table (Method B)

(For nomenclature see previous page)

This method can be used if the required air flow is known or has been calculated with the formulas as shown below:

1. $$Q = .0273\ \frac{D^2L}{t} \times \frac{p_2 + 14.7}{14.7} \quad \text{(SCFM)}$$

Conversion of CFM to SCFM

2. $$Q = q \times \frac{p_2 + 14.7}{14.7} \times \frac{528}{T} \quad \text{(SCFM)}$$

Flow factor Cv (standard conditions)

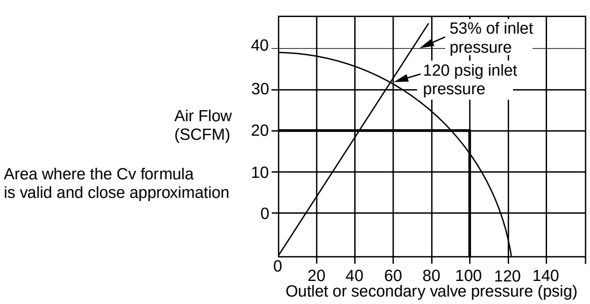
3. $$C_V = \frac{1.024 \times Q}{\sqrt{\Delta P \times (p_2 + 14.7)}}$$ Proposed NFPA Standard T3.21.3

Maximum pressure drop Δp across the valve should be less than 10% of inlet pressure p_1.

Inlet Pressure (psig)	Air Flow Q (scfm) for various Pressure drops Δ P at Cv = 1					Air Flow Q (scfm) to atmosphere
	2 psid	5 psid	10 psid	15 psid	20 psid	
10	6.7					12.0
20	7.9	11.9				16.9
30	9.0	13.8	18.2			21.8
40	9.9	15.4	20.6	23.8		26.6
50	10.8	16.9	22.8	26.7	29.2	31.5
60	11.6	18.2	24.8	29.2	32.3	36.4
70	12.3	19.5	26.7	31.6	35.1	41.2
80	13.0	20.7	28.4	33.8	37.7	46.1
90	13.7	21.8	30.0	35.8	40.2	51.0
100	14.4	22.9	31.6	37.8	42.5	55.9
110	15.0	23.9	33.1	39.6	44.7	60.7
120	15.6	24.9	34.5	41.4	46.8	65.6
130	16.1	25.8	35.8	43.1	48.8	70.5
140	16.7	26.7	37.1	44.7	50.7	75.3
150	17.2	27.6	38.4	46.3	52.5	80.2
160	17.7	28.4	39.6	47.8	54.3	85.1
170	18.2	29.3	40.8	49.3	56.0	90.0
180	18.7	30.1	42.0	50.7	52.7	94.8
190	19.2	30.9	43.1	52.1	59.4	99.7
200	19.6	31.6	44.2	53.4	60.9	104.6
210	20.1	32.4	45.2	54.8	62.5	109.4
220	20.5	33.1	46.3	56.1	64.0	114.3
230	21.0	33.8	47.3	57.3	65.5	119.2
240	21.4	34.5	48.3	58.6	66.9	124.0
250	21.8	35.2	49.3	59.8	68.3	128.9

Table 3: Air Flow Q (scfm) for C_V = 1

Flow Curves: How to read them



Example 1: Find air flow Q (scfm) if C_V is known.
C_V (from valve catalog) = 1.8
Primary pressure p1 = 90 psig
Pressure drop across valve Δ P = 5 psid

Flow through valve from Table 3 for C_V = 1:21.8 scfm

$$Q = C_V \text{ of valve} \times \text{air flow at } C_V = 1 \text{ (scfm)}$$

Q = 1.8 x 21.8 = 39.2 scfm

Example 2: Find Cv if air flow Q (scfm) is given.
Primary pressure p1 = 90 psig
Pressure drop Δ P = 10 psid
Air flow - Q = 60 scfm

Flow through valve from Table 3 for C_V = 1:30 scfm

$$C_V = \frac{\text{Air flow Q (scfm)}}{\text{Air flow at } C_V = 1 \text{ (scfm)}}$$

$$C_V = \frac{60 \text{ scfm}}{30} = 2.0$$

A valve with a C_V of minimum 2 should be selected.

Example 3: Find C_V if air flow Q (scfm) to atmosphere is given (from catalog).
Primary pressure p_1 = 90 psig
Air flow to atmosphere Q = 100 scfm

Flow to atmosphere through valve from Table 3 for C_V = 1:51 scfm

$$C_V = \frac{\text{Air flow to atmosphere Q (scfm)}}{\text{Air flow to atmosphere at Cv = 1 (scfm)}}$$

$$C_V = \frac{100}{51} = 2.0$$

Flow given in catalog is equivalent to a valve with C_V = 2. This conversion is often necessary to size a valve properly since some manufacturers do not show the standard C_V to allow a comparison.

Example 4: Find C_V if cylinder size and stroke speed is known, using the formulas 1 and 3.
Primary pressure = 90 psig
Pressure drop across valve 5 psid
Cylinder size 4" dia. x 10" stroke
Time to complete stroke 2 sec.

$$Q = .0273\ \frac{4^2 \times 10}{2} \times \frac{85 + 14.7}{14.7} = 14.81 \text{ scfm}$$

$$C_V = \frac{1.024 \times 14.81}{\sqrt{5} \times (85 + 14.7)} = .7$$

Conversion Tables

VOLUME

from/to	cm^3	liter	in^3	ft^3	fl oz	pt.	qt.	gal
cm^3	1	.001	0.06102	3.53 x 10-5	.03381	.00211	0.106	2.64 x -10
liter	1000	1	61.02	0.03532	33.81	2.113	10.057	.2642
in^3	16.39	0.01639	1	5379 x 10-4	.5541	.03463	0.01732	.00433
ft^3	2.83 x 104	28.32	1728	1	957.5	59.84	29.92	7.481
fl oz	29.57	0.02957	1.805	0.00104	1	.06250	.03125	.00781
pt	473.2	0.4732	28.88	0.01671	16	1	0.500	0.1250
qt	946.4	0.9463	57.75	0.03342	32	2	1	0.2500
gal (US)	3785	3.785	231	0.1337	128	8	4	1

TEMPERATURE COMPARISON

-100°C to +300°C

°C	°F	°C	°F
-100	-148	29	84.2
-90	-130	30	86.0
-80	-112	31	87.8
-70	-94	32	89.6
-60	-76	33	91.4
-50	-58	34	93.2
-40	-40	35	95.0
-35	-31	36	96.8
-30	-22	37	98.6
-25	-13	38	100.4
-20	-4	39	102.2
-15	5	40	104.0
-10	14	45	113
-5	23	50	122
0	32	55	131
1	33.8	60	140
2	35.6	65	149
3	37.4	70	158
4	39.2	75	167
5	41	80	176
6	42.8	85	185
7	44.6	90	194
8	46.4	95	203
9	48.2	100	212
10	50	110	230
11	51.8	120	248
12	53.6	130	266
13	55.4	140	284
14	57.2	150	302
15	59	160	320
16	60.8	170	338
17	62.6	180	356
18	64.4	190	374
19	66.2	200	392
20	68	210	410
21	69.8	220	428
22	71.6	230	446
23	73.4	240	464
24	75.2	250	482
25	77	260	500
26	78.8	270	518
27	80.6	280	531
28	82.4	290	554
		300	572

PRESSURE

from/to	mm Hg	in Hg	in H_2O	ft H_2O	atm	lb/in^2	kg/cm^2
mm Hg	1	0.03937	0.5353	0.04460	.00132	0.01934	.00136
in Hg	25.40	1	13.60	1.133	.03342	0.4912	0.03453
in H_2O	1.868	0.07355	1	0.08333	0.00246	0.03613	0.00254
ft H_2O	22.42	0.8826	12	1	0.02950	0.4335	0.03048
atm	760	29392	406.8	33.9	1	14.70	1.033
lb/in^2	51.71	2.036	27.67	2.307	0.06805	1	0.07031
kg/cm^2	735.6	28.96	393.7	32.81	0.9678	14.22	1
bar	750.0	29.53	401.32	33.46	0.98592	14.504	1.01978

Length

from/to	cm	m	km	in	ft.	mile
cm	1	0.01	1 x 10-5	0.3937	0.03281	6.21 x 10-6
m	100	1	0.001	39.37	3.281	6.21 x 10-4
km	1 x 105	1000	1	3.94 x 104	3281	0.6214
in	2.540	0.02540	2.54 x 10-5	1	0.08333	1.58 x 10-5
ft	30.48	0.3048	3.05 x 10-4	12	1	1.89 x 10-4
mile	1.61 x 105	1609	1.609	6.34 x 104	5280	1

ENERGY

from/to	BTU	Cal	Joule	Hp. hr.	Kw hr.
BTU	1	252.0	1055	3.93 x 10-4	2.93 x 10-4
Cal	0.397	1	4.186	1.56 x 10-5	1.16 x 10-5
joule	9.48 x 10-4	0.2389	1	3.73 x 10-7	2.78 x 10-7
Hp hr	2545	6.41 x 105	2.68 x 106	1	0.7457
Kw hr	3413	8.60 x 105	3.60 x 106	1.341	1

AREA

from/to	cm^4	m^2	km^2	in^2	ft^2
cm^2	1	0.0001	1 x 10-10	0.1550	0.00108
m^2	1 x 104	1	1 x 10-5	1550	10.76
km^2	1 x 1010	1x 105	1	1.55 x 109	1.08 x 107
in^2	6.452	6.45 x 10-4	6.45 x 10-10	1	0.00694
ft^2	929.00	0.09290	9.29 x 10-8	144	1

TEMPERATURE CONVERSION

°C = 5/9 (°F - 32)
°F = 9/5 (°C + 32)
°K = °C + 273.2
°R = °F + 459.7

WEIGHT

from/to	gm	kg	oz	lb
gm	1	0.001	0.03527	0.00220
kg	1000	1	35.27	2.205
oz	28.35	0.02835	1	0.06250
lb	453.6	0.4536	16	1

M5 A metric machine screw thread (M5 x 0.8) 5 mm in diameter with 0.8 mm between threads. Used on miniature pneumatic products it is nearly identical in size to a 10-32 UNF thread. Port sealing is effected by a thermoplastic washer which creates a static face seal.

ISO G Designed to mate with threads conforming to ISO Standard 228/1, a parallel pipe thread available in both male and female forms. ISO G ports feature a flat surface perpendicular to the axis of the port on which a washer incorporated into the male fitting establishes a static face seal, (reference ISO 1179).

ISO G series threads are usually designated in the following manner, with a 1/4 size pipe thread being indicated in this example: G1/4. This format is always used with female threads, however, in some, but not all instances, male threads may include an “A” or “B” suffix following the port size indicating the thread class, (typically “A”). Example: G 1/4A.

In addition to itself, the ISO G thread form will also mate with British Standard Pipe Parallel (BSPP) and Japanese Industrial Standard Parallel per JIS B 0202. As both the BSPP and JIS B 0202 standards do not define an area for the ISO G static seal to engage, some form of thread sealant must be applied to the thread to prevent spiral leakage.

Note:

Norgren male ISO G threads have always included a thermoplastic washer to create the static face seal. During 1998/9 this design will be changed to provide an o-ring in an under-cut groove. This improved design will still mate with port forms shown opposite but will provide an improved seal.

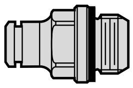
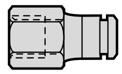
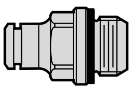
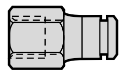
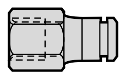
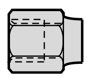
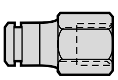
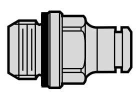
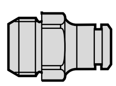
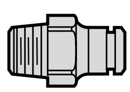
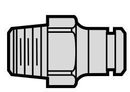
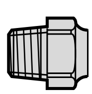
ISO R Designed to mate with threads conforming to ISO Standard 7/1, a tapered pipe thread available in male form. ISO R ports are similar to NPTF threads, they form a pressure tight joint via an interference fit as they are tightened, (thread sealant is recommended). To insure a pressure tight seal, Pneufit ISO R male pipe threads are supplied with Precote 5 thread sealant.

ISO R male series threads are usually designated in the following manner, with a 1/4 size pipe thread being indicated in this example: R 1/4.

ISO R female series threads are usually designated in the same manner, but with a “c” subscript. Example: R_c1/4.

In addition to itself, the ISO R thread form will also mate with British Standard Pipe Tapered (BSPT) and Japanese Industrial Standard Tapered Pipe (PT) per JIS B 0203. Some form of thread sealant must be applied to the thread to prevent spiral leakage.

ISO R male threads will also effect a seal with ISO G, BSPP and JIS B 0202. In all cases, some form of thread sealant must be applied to the thread to prevent spiral leakage.

Thread Form	Mates with Thread Form	Method of Sealing
M5 (male) 	M5 (female) 	Molded thermoplastic sealing washer effects static face seal
ISO G (male) 	ISO G (female) 	Molded thermoplastic sealing washer effects static face seal
	BSPP (female) 	
	JIS B 0202 (female) 	
ISO G (female) 	ISO G (male) 	Molded thermoplastic sealing washer effects static face seal
	BSPP (male) 	Sealing washer should be used to effect static face seal
	JIS B 0202 (male)	
	ISO R (male) 	Thread sealant must be applied to prevent spiral leakage
	BSPT (male) 	
	JIS B 0203 (male) 	

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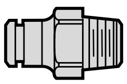
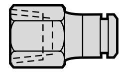
Thread Form	Mates with	Thread Form	Method of Sealing

ISO R (male)

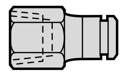


Mates with:

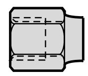
ISO R_C (female)



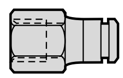
BSPT (female)



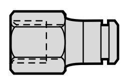
BSPT (female)



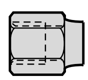
ISO G (female)



BSPP (female)

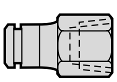


JIS B 0202 (female)



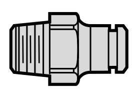
Thread sealant must be applied to prevent spiral leakage

ISO R_C (female)

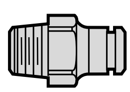


Mates with:

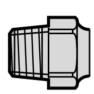
ISO R (male)



BSPT (male)

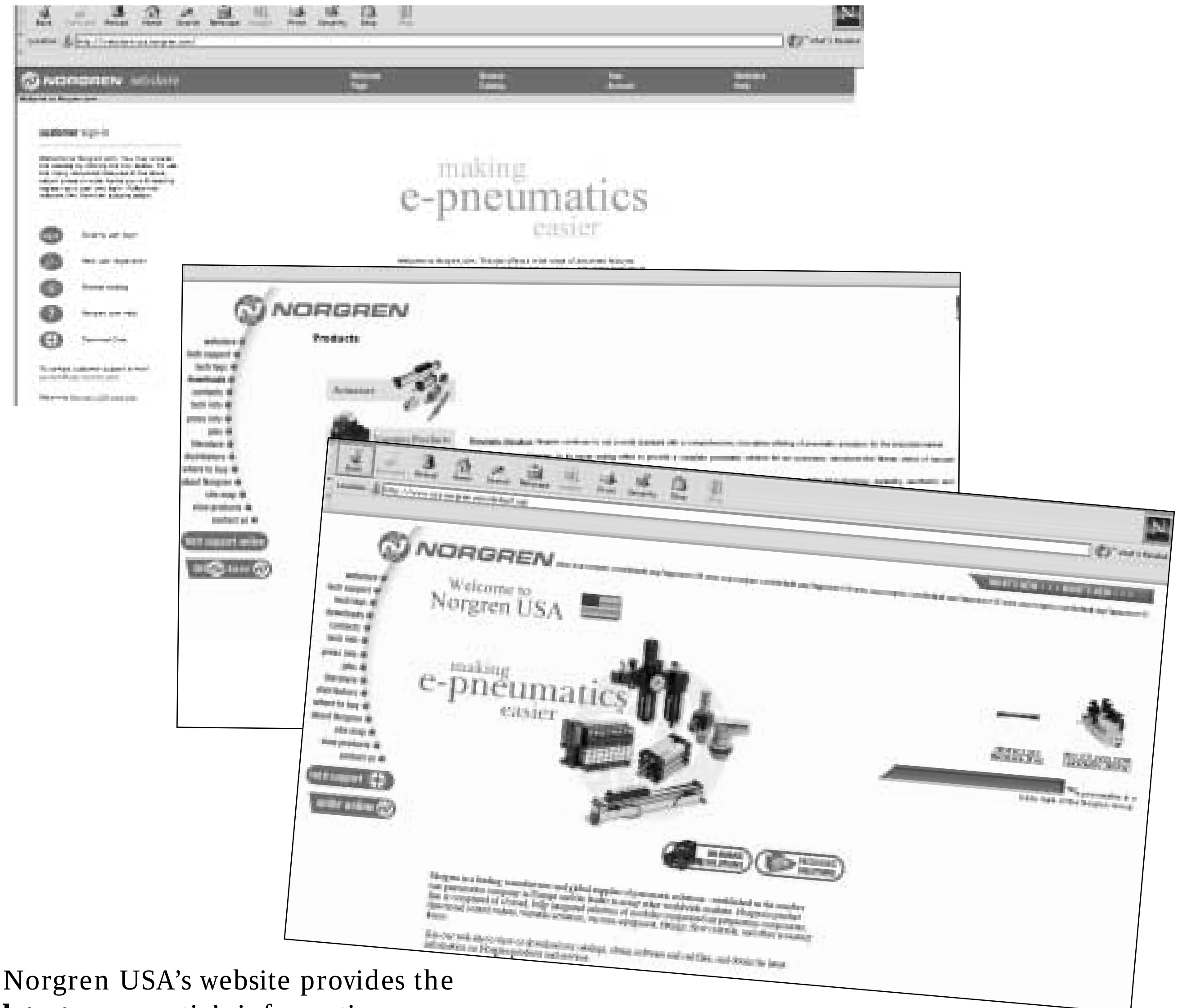


JIS B 0203 (male)



Thread sealant must be applied to prevent spiral leakage

www.norgren.com is your complete source for the latest Norgren product information, technical tips, and other valuable pneumatic information.



Norgren USA's website provides the **latest** pneumatic's information available from Norgren USA. It is **the source** for additional pneumatic product catalogs, CAD files, circuit design software, distributor locations, customer service and application engineering contacts.

To access our website, set your web browser to:
www.norgren.com

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e-mail: inquiry@usa.norgren.com

Warning

These products are intended for use in industrial compressed air systems only. Do not use these products where pressures and temperatures can exceed those listed under **Specifications**.

Before using these products with fluids other than those specified, for nonindustrial applications, life-support systems, or other applications not within published specifications, consult Norgren.

Through misuse, age, or malfunction, components used in fluid power systems can fail in various modes. The system designer is warned to consider the failure modes of all component parts used in fluid power systems and to provide adequate safeguards to prevent personal injury or damage to equipment in the event of such failure modes. **System designers must provide a warning to end users in the system instructional manual if protection against a failure mode cannot be adequately provided.**

System designers and end users are cautioned to review specific warnings found in instruction sheets packed and shipped with these products. System designers should also provide for all OSHA requirements including Title 29 CFR 1910.147 Lockout/Tagout.

It should be recognized that warnings are valid for any product, regardless of manufacturer, and are not restricted to products manufactured by Norgren. Norgren's reputation for product quality and performance is well established. We feel we have the additional obligation to provide information or warnings to customers to assist them in applying our products in a reasonable and safe manner.

Warranty

Items sold by Norgren are warranted to be free from defects in materials and workmanship for a period of two years from the date of manufacture, provided said items are used according to Norgren's recommended usages. Norgren's liability is limited to the repair of, refund of purchase price paid for, or replacement in kind of, at Norgren's sole option, any items proved defective, provided the allegedly defective items are returned to Norgren prepaid. The warranties expressed above are in lieu of and exclusive of all other warranties.

There are no other warranties, expressed or implied, except as stated herein. There are no implied warranties of merchantability or fitness for a particular purpose, which are specifically disclaimed. Norgren's liability for breach of warranty as herein stated is the exclusive remedy, and in no event shall Norgren be liable or responsible for incidental or consequential damages, even if the possibility of such incidental or consequential damages has been made known to Norgren.

Norgren reserves the right to discontinue manufacture of any product or change product materials, design, or specifications.

For complete Valve information order:
APC-102 - Norgren Actuator Products and Accessories

Nugget 30
Port SIze: Port fittings in 5/32", 1/4", 4 mm, or 6 mm
Flow: Cv of .27
Valve Type: 5-port/2-position and 5-port/3-position
Electrical Connections: 25-pin D connector and plug-in wireway, or pinch plugs with leads
Subbase Mounting: lightweight plastic manifolding subbase with plug-on common wireway and 25-pin D connector provides up to 24 stations, or Aluminum fixed length 10-32 UNF ported subbase (1 to 12 stations).
Operator Types: Solenoid pilot
Fieldbus Compatible

Super X Valves
Port SIze: 1/8" and 1/4" NPT
Flow: 1/8" (Cv of .34), 1/4" (Cv of .98)
Valve Type: 3-port/2-position, 5-port/2-position, and 5-port/3-position
Mounting Options: Two through holes for wall or control cabinet mounting. Many operators are available for panel mounting.
Operator Types: Manual or mechanical

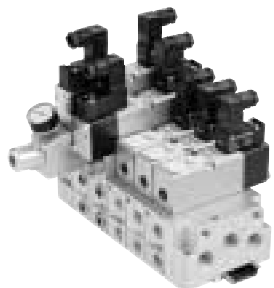
Nugget 40
Port SIze: Inline ports - 10-32 UNF
Subbase ports - 1/8" NPT
Endplate ports - 1/4" NPT (inlet), 1/8" NPT (exhaust)
Flow: Inline - Cv of .26, subbase Cv of .41
Valve Type: 3-port/2-position, 5-port/2-position, and 5-port/3-position
Mounting Options: Inline, inline on a fixed length manifold, inline modular, single subbase valve, and subbase-modular manifolds
Electrical Connections: 25-pin D connector, or 1" conduit electrical connection
Operator Types: Solenoid pilot or air pilot
Fieldbus Compatible

Nugget 70
Port SIze: 1/8" NPT, or ISO G
Flow: 3/2 and 5/2 Cv of 0.76, 5/3 Cv of 0.67
Valve Type: 3-port/2-position, 5-port/2-position, and 5-port/3-position
Mounting Options: Two through holes for wall or control cabinet mounting
Operator Types: Solenoid pilot or air pilot
Electrical Connections: DIN 43650 Table "C"

Nugget 120
Port SIze: Inline ports - 1/4" NPT
Subbase ports - 1/4" NPT
Subbase endplates - 3/8" NPT (inlet), 1/4" (exhaust)
Flow: 3/2 and 5/2 Cv of 1.17, 5/3 Cv of 0.79
Valve Type: 3-port/2-position, 5-port/2-position, and 5-port/3-position
Mounting Options: Inline - through holes in valve bodies
Subbase valve - through holes to subbase and manifold
Manifold mounted - through holes in end plate or to a 35 mm DIN rail
Operator Types: Solenoid pilot or air pilot
Electrical Connections: 25-pin D connector, or 1" conduit electrical connection
Fieldbus Compatible

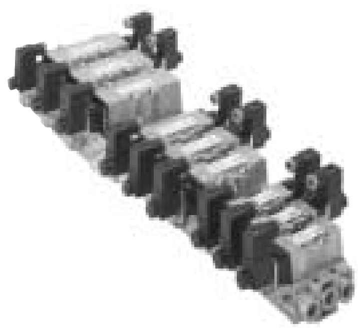
Nugget 200

Port SIze: 1/4", 3/8", or 1/2" NPT or ISO G
Flow: Cv of 1.6
Valve Type: 3-port/2-position*, 5-port/2-position, and 5-port/3-position
Mounting Options: Inline valves, stacking valve assembly, stacking valve assembly with junction box, or inline fixed length manifold.
Operator Types: Solenoid pilot, air pilot, manual, or mechanical
Electrical Connections: DIN 43650 Table "B", 1/2" NPT conduit connector, or 1" conduit (junction box)
*Not available in 1/2" ports



Mini ISO Star Valves

Port Size: ISO 15407-1/VDMA 24 563, 26 mm
Flow: Cv of 0.92, 1.22, 1.17
Valve Type: 5-port/2-position and 5-port/3-position
Mounting Options: Subbase
Operator Types: Solenoid pilot, or air pilot
Electrical Connections: Cable grip standard. Optional: electrical connectors are cable grip with indicator light, a 1/2" NPT conduit connector, 10-foot molded cable, and 10-foot molded cable with surge suppression, reverse polarity protection, and indicator light.



ISO Star Valves

Port Size: 1/4" (size 1), 3/8" (size 2), 1/2" (size 3), NPT or ISO G
Flow: Cv of 1.2, 2.4, 4.0
Valve Type: 5-port/2-position and 5-port/3-position
Mounting Options: single subbase, or manifolding subbase
Operator Types: Solenoid pilot, or air pilot
Electrical Connections: Cable grip standard. Optional: electrical connectors are cable grip with indicator light, a 1/2" NPT conduit connector, 10-foot, three-conductor molded cable, and 10-foot, three-conductor molded cable with surge suppression, reverse polarity protection, and indicator light.

Nugget 500

Port Size: 3/8", or 1/2" NPT or ISO G, and 3/4" ports NPT (inline valves only)
Flow: Cv of 5.0
Valve Type: 5-port/2-position and 5-port/3-position
Mounting Options: Inline have two sets of mounting holes through the body. Fixed length manifolds, single subbase, and subbase valves attached to a manifolding subbase.
Operator Types: Solenoid pilot or air pilot
Electrical Connector Options: cable grip with indicator light, a 1/2" NPT conduit connector and a 5-foot, three-conductor, molded cable connector with or without indicator light and surge suppression.

Poppet Valves

Port Size: 1/4" through 2"
Flow: Cv of 2.1 to 49.5
Valve Type: 2-port/2-position, 3-port/2-position, 4-port/2-position, and 3-port/2-position multi-directional
Mounting Options: Inline or single subbase
Operator Types: Solenoid pilot, air pilot, manual, or mechanical
Electrical Connector Options: cable grip with indicator light, a 1/2" NPT conduit connector and a 5-foot, three-conductor, molded cable connector with or without indicator light and surge suppression.

For complete Actuator information order:
APC-103 - Norgren Actuator Products and Accessories

Section 1 – Series 90000 Compact Actuators

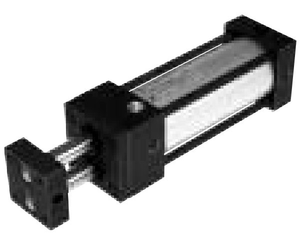
Fluids: Compressed air, filtered, lubricated or non-lubricated
Operating Pressure: 14.5 to 145 PSI (1 to 10 bar)
Operating Temperature: 23°* to 176°F (-5°* to 80 °C)
* Consult Technical Service for use below 35°F (2° C)
Bore Sizes: 0.50"(12mm), 0.625" (16mm), 0.75" (20mm), 1.0" (25mm), 1.25" (32mm), 1.50" (40mm), 2.0" (50mm) 2.50" (63mm), 3.125" (80mm), 4.0" (100mm)

Section 2 – Roundline Stainless Steel Body Actuators

Fluids: Filtered, lubricated or non-lubricated, compressed air
Maximum Operating Pressure: 250 psig (17.2 bar)
Temperature Range:
Standard Nitrile Seals: -20° to 200°F (-29° to 93°C)
Viton Seals: -20° to 400°F (-29° to 205°C)
Bore Sizes: .438" (11mm), .563" (14mm), .75" (19mm), 1.063" (27mm), 1.25" (32mm), 1.50" (40mm), 2.0" (50mm), 2.50" (63)

Section 3 – Series SS Stainless Steel Actuators

Fluids: Filtered compressed air to 250 PSI
Petroleum based hydraulic fluid to 400 PSI
Lubrication: None required
Operating Pressure:
250 psig air(17.2 bar)
400 psig hydraulic (27.6 bar)
Operating Temperatures:
Series SS: -40° to 200°F (-40° to 93°C)
Series D: -20° to 250°F (-29° to 204°C)
with Viton Seals: -20° to 400°F (-29° to 204°C)
Bore Sizes: 1.125" (29mm), 1.50" (38mm), 2.0" (51mm), 2.50" (64mm), 3.25" (83mm), 4.0" (102mm), 5.0" (127mm), 6.0" (152mm), 8.0" (203mm)

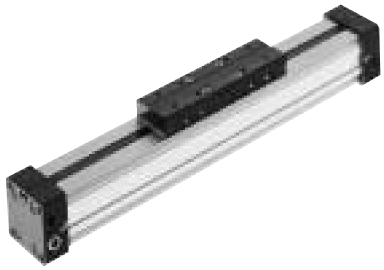


Section 4 – Series N Non-Rotating Actuators

Fluids: Filtered compressed air to 250 psi
(For hydraulic service consult factory.)
Operating Pressure: 250 psig (17.2 bar)
Operating Temperatures: -20° to 200°F (-29° to 93°C)
Bore Sizes: 1.125" (29mm), 1.50" (38mm), 2.0" (51mm), 2.50" (64mm), 3.25" (83mm), 4.0" (102mm)

Section 5 – 8000 & 8000/M ISO VDMA Actuators

Fluids: Compressed air, filtered, lubricated and non-lubricated
Operating Pressure: 1 to 16 bar (14.5 to 232 psig)
Operating Temperature: -20° to 80°C max (-4° to 176°F max)
(Consult Technical Services for use below 2°C (35°F)
Bore Sizes: 32, 40, 50, 63, 80, 100, 125, 160, 200, 250, 320 mm
Note: Corrosion Resistant available.

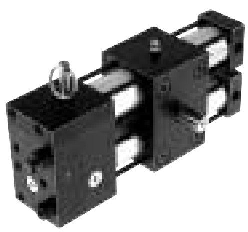


Section 6 – Lintra Series 46000 Rodless Actuators

Fluids: Compressed air filtered to 50µ and lubricated
Operating Pressure
16 mm: 22 to 150 psig (1.5 to 10 bar)
20 mm to 80 mm: 15 to 150 psig (1 to 10 bar)
Operating Temperatures
-22° to 180°F* (-30°C to 80°C)
*With dewpoint of supply air less than ambient air temperature.
Bore Sizes: 0.63" (16mm), 0.79 (20mm), 0.98 (25mm), 1.26" (32mm), 1.57" (40mm), 1.97" (50mm),2.48" (63mm), 3.15" (80mm)

Section 7 – Lintra-Lite Series A440000 Rodless Actuators

Fluids: Compressed air, filtered to 50µ and lubricated
Operating Pressure: 15 to 118 psig (1 to 8 bar)
Operating Temperatures: -22° to 180°F* (-30° to 80°C)
*With dewpoint of supply air less than ambient air temperature at cylinder, consult our Technical Service for use below 36°F (2°C)
Bore Sizes: 1.0" (25mm), 1.25" (32mm), 1.50" (40mm)

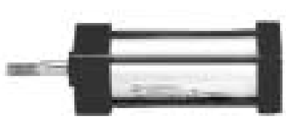


Section 8 – Rack and Pinion Rotary Actuators

Fluids: Filtered compressed air to 150 psi (10 bar)
Operating Pressure: 150 psi (10 bar)
Operating Temperature:
-20° to 200°F (-29° to 93°C) with Standard Nitrile Seals
-20° to 400°F (-29° to 204°C) with Viton® Seals
-20° to 250°F (-29° to 121°C) with Low Friction Seals
Bore Sizes: 0.50" (12mm), 0.75" (20mm), 1.125" (29mm), 1.50" (38mm), 2.00" (50mm), 2.50" (64mm)

Section 9 – Rotary Vane Actuators

Fluids: Lubricated or non-lubricated, filtered, compressed air
Operating Pressure:
M/60280, M/60281: 44 to 102 psig (3 to 7 bar)
M/60282, C/60283: 29 to 102 psig (2 to 7 bar)
C/60284, C/60084/TI: 29 to 145 psig (2 to 10 bar)
Operating Temperature: 40° to 140°F (5° to 60°C)



Section 10 – Tiny Tim Actuators

Fluids: Filtered compressed air to 150 psi (10 bar)
Operating Pressure: 150 psi (10 bar)
Operating Temperature:
-20° to 225°F (-30° to 110°C) with Standard Seals
-40° to 200°F (-40° to 90°C) with Buna N Seals
-40° to 350°F (-40° to 175°C) with Viton Seals
Bore Sizes: 3/4", 1", 1-1/8"