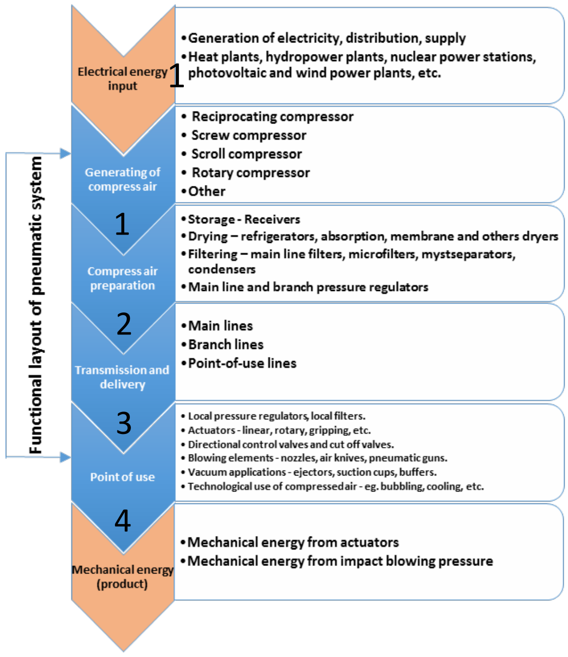


Figure 1. Energy flows in the compressed air system

2. The findings and potential of saving

2.1. Stage of generating (compressor room)

In general, compressor controls seek to maintain the compressed air discharge pressure within a specified range. There are at least six common types of compressor control modes for reciprocating and rotary air compressors: start/stop, load/unload, inlet modulation, auto-dual, variable displacement and variable speed control [3].

For small-scale and middle-scale air compressor applications in Bulgarian industry, the share of rotary screw compressor with fixed speed is about 80%. For new compressed air systems, and big-scale compressor application, a rotary screw compressor with fixed and variable speed is used.

Load/unload is the most common control method - the compressor runs fully loaded, producing compressed air at maximum efficiency until the discharge pressure reaches the upper activation pressure setting which causes the compressor to unload. When unloaded, the compressor no longer adds compressed air to the system, but the motor continues to run. If a compressor unloaded time is much of the time, the specific power consumption will go up, resulting in wasted energy.

Variable-speed compressors vary the speed of the compressor motor to meet the compressed air demand. In general, for variable torque loads such as air compression, the load on the motor varies with the cube (cubic dependence) of the motor speed.

Thus, decreasing the speed of the motor during periods of low compressed air demand significantly decreases the load on the motor, and variable-speed compressors operate very efficiently at low loads.

During this study, we have specified two major problems with big potential of improvement in the stage of generating compressed air. They refer mainly to the majority of installed rotary screw compressors with fixed speed with load/unload control.

2.1.1. Over-sized compressor

Typical relationships between fraction input power to the compressor (*FrP*) and fraction compressed air output (*FrC*) for load/unload control type is:

$$FrP = FrP_0 + (1 - FP_0)FrC \; .$$

where: FrP_0 is representing the percentage of peak power which compressor draws in unload mode. It depends of type and model of compressor and it varies between 20% and 60% [2].

At full output capacity (*FrC*=100%), compressors draw full power (*FrP*=100%). The power draw at less than full output capacity is a function of the type of part-load control.

It is clear that there is direct relation between load/unload ratio of the peak power of the compressor and the level of operating efficiency. The operating efficiency can be shown as quantity of compressed air (Nm^3) per kWh or vice-versa – how many *kWh* is needed to be produced 1Nm^3 compressed air in desired pressure.

The level of operating efficiency can be found by direct measurement of input power on the compressor (or compressors in the compressor room) and the flow on the entrance of usage part of the pneumatic system for certain period of time. However, while in most enterprises, electrical energy is measured accurately and in details, the installation of flow meters into compressed air even on main line or in the entrance is uncommon. In this case, the operation efficiency of compressors can be calculated on the base only of electrical measurements and consideration of technical features and parameters of the compressors, given in the producer documentation. Many compressors producers provide free calculators or services to figure out the operating efficiency of installed compressors or during the design of compressor system stage. In any cases, the results confirm that load/unload ratio of the peak power have a significant impact of efficiency and production price of the compressed air. For example, a compressor with load/unload ratio of the peak power 90% can produce more than 8Nm^3 compressed air with 1*kWh* electrical energy. The same compressor with load/unload ratio of the peak power of 37% produce only 4,3Nm^3 with 1*kWh*. Variable speed compressors have better part-load performance in smaller duty cycles [3].

The conclusion is that over-sized compressors are always inefficient and such cases should be avoided.

The overall potential of savings by improving compressor management can't be shown generally. It should be calculated case by case, but in all cases, it is significant.

One of the approaches can be: limit the number of compressors to the minimum according to the air consumption, and then operate all except one, at 100% load, and control them so that just one is not operated (unload). This results in diagrams of the type shown in Fig.2.

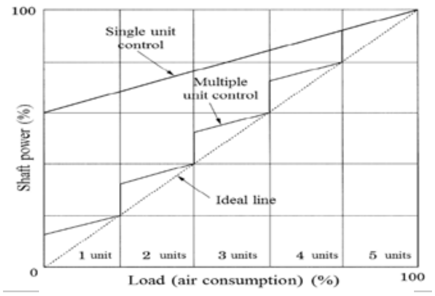


Figure 2. Load characteristics

2.1.2. Overpressure compressed air production

The pressure at which operates the system had a significant effect on the energy efficiency of the compressors.

During our research, we have found that in most of the cases, the average pressure at the exit of the compressor room exceeds significantly the demanded pressure at end user side (Table 2.).

Table 2. Real average data by industry

Mid-High producer by industry type	Average pressure (bar)	Required pressure (bar)	Potential % of saving
Electronic industry	7,2	6	9,78%
Plastic molding	7,2	6	9,78%
Tobacco producer	7	6	8,36%
Machinery	8,5	6,5	13,77%
Beverage industry	6,2	5,7	4,72%
Bottling industry	6	5,7	2,92%
Beverage industry	6,5	5,7	7,25%
Tailoring industry	6,9	6	7,62%
Wood working industry	8	6,5	10,89%
Tailoring industry	6	5,5	4,91%
Plastic molding	7	6	8,36%
Machinery	7	6	8,36%
Automotive industry	6,7	5,5	10,70%
Automotive industry	7	6	8,36%
Tailoring industry	6,5	6	4,46%
Electronic industry	6,7	5,5	10,70%
Sports equipment producer	7,5	6,3	9,30%

In some cases the reasons of that was big pressure drop in some parts of the system on user side, caused by inadequate main and branch line layout (see next chapter), which put the facility managers or operators to boost the discharge pressures in the compress room and generate extra flow (artificial demand). In some cases, they keep pressure high "just in case" justifying that they avoid the artificial demand on the usage site, keeping local regulators on low pressure.

It is true that the benefits of lowering system pressure can be attributed to two separate actions: reduced pressure at the compressor and a reduction in pressure delivered to production equipment which directly lowers the demands of flow.

The benefits of lowering the pressure at the point of use can be calculated or measured easily and it is a good practice. However, let us see what the benefits to lower pressure at the compressors are:

With centrifugal blower, turbo compressor and screw compressor, internal air compression takes place. The power demand of which is given by [6]:

$$Wp = \frac{\rho_a * G_s * RT_a}{K * e_0} * \left(\left(\frac{P_a + \Delta p}{P_a}\right)^k - 1\right)$$

where:

k = (K-1)/K=0,2857; K=1,4 (adiabatic exponent); ρ_a – air density (1,293 *kg/m*^{3}); T_a – inlet gas temperature (*K*); P_a – inlet pressure (Absolute, *bar*); R – gas constant (286,88 *J/kg.K*); e_0 – efficiency coefficient.

Let's find the fractional differences with two different pressure P_H and P_L, where $P_H > P_L$.

If $\frac{\rho_a * G_s * RT_a}{K * e_0} = C$ and P_a – atmospheric pressure = 1; $P_a + \Delta p_L = P_L$; $P_a + \Delta p_H = P_H$,
then: $Wp_H = C * (P_H)^{0,2857}$ and $Wp_L = C * (P_L)^{0,2857}$

The fractional difference is as follow:

$$\frac{(Wp_H - Wp_L)}{Wp_H} = \frac{{P_H}^{0,2857} - {P_L}^{0,2857}}{{P_H}^{0,2857} - 1}$$

(P_H = P_H gauge +1, P_L = P_L gauge +1).

For example – if the discharge pressure of the compressor is set to 7,0 *bars* and we achieve to lower it to 6,0 *bars*, then expected savings in power consumption in percentage will be:

$$100*(8{,}0^{0,2857} - 7{,}0^{0,2857})/ 8{,}5^{0,2857} = \mathbf{8{,}36\%} .$$

In case that this expectation can be applied only during the load mode of the compressor.

2.1.3. Other steps in energy efficiency in compressor side

a) Reducing temperature of intake air – Up to 6% of a compressor's power can be saved by using cool inlet air that requires less energy to compress. If inlet air is currently taken from a hot compressor house, consider ducting cool air from a shady outside area. Situate compressors in well-ventilated area with hot compressor air ducted away from the inlet feed. For every 3°*C* reduction in inlet temperature there is a one per cent reduction in energy usage [9].

b) Drain - Removing condensed water from the receivers is an important part of the work of the pneumatic system. At the same time drainage leads to great losses. So, automatic drainage with optimal duty cycle should be applied. It is recommendable to use „zero-loss“ condensate drain traps.

2.2. Compressed air preparation

Whether designing new systems, or reviewing the existing ones, the important step is to define exactly what are the compressed air purity requirements because, inadequate or unnecessary levels of treatment can significantly increase the associated energy costs in use. There is a very wide range of requirements for air quality, all of which can be met with the right equipment.

ISO8573 is the group of international standards relating to the quality (or purity) of compressed air. Considering compressed air preparation and purification for automated system, ISO8573-1:2010 is applied which specifies the amount of contamination - as Solid Particulate, Water and Oil allowable in compressed air. The purification class is given as a Class: X.Y.Z, where X refers to the particulate count in one cubic meter and their size, Y refers to the moisture of compressed air, Z refers to oil contamination. The purity levels for each contaminant are shown separately in tabular.

In the process of our study we have defined eight types of applications with corresponding air purification requirements and corresponding percentage of compressed air usage (Table 3.).

2.2.1. Dryers

The energy saving potential in compressed air drying process is commonly underestimated and neglected. But in our research we have defined considerable potential of savings during air preparation process.

As it is shown by the data in the table, more than 95% of total applications required pressure dew point in a range of $+3^0C$ to $+13^0C$ and more, which means that refrigerated dryers are applicable to majority of the systems.

In the rest of the systems, refrigerated dryers are usually used in a compressed room preparation system, and desiccant, membrane or other type is used in a point-of-use in order to reach lower pressure dew point. We have found only one company from pharmaceutical industry, where all the used compressed air in a factory was prepared in a compress room with dew point of -24^0C.

The refrigerated air dryers lower dew point by cooling the air in the heat exchanger by meaning of compressor with refrigerant agent usage.

Usually the sizing of refrigerated dryers is done regarding the maximum flow of compressed air that can be produced by compressor/s, which will guarantee that independents of the demands, the air will reach corresponding dew point.

Investigating the producer documentation and recommendation, we have found that recommended power of refrigerated dryer is in-between 3% and 5% of the nominal power of the compressor/s. Here is not taken into account the internal and filter pressure drops that can occur.

The most used type (95% of the cases in our study) is a refrigerated dryer with hot air bypass, in which regardless of the demand, the compressor motor is running on full capacity. These types have their own advantage as stable dew point irrespective of varying load demand.

But let’s take a look of the flow demand profile in a highly automated production factory which we measure (Fig.3.). In a typical pneumatic system, the flow is not constant and depends on the load of the production. The green area represented the pure possibilities of energy saving potential in dryers.

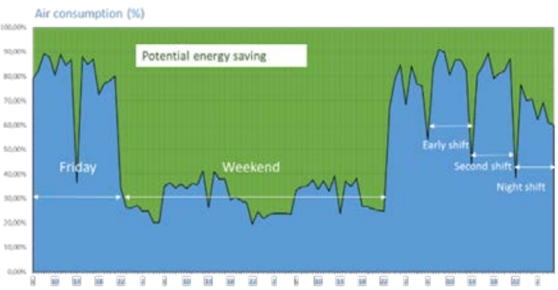


Figure 3. Demand profile in typical pneumatic system

In Fig. 4. are presented different energy consumption profiles in different types of refrigerated dryers (the data is taken from the producers documentation).

The best case is the refrigerators with profile close to the ideal line to be used. However, the differences in price of investment can be a factor in decision taken process.

In the observed situation we can assume that up to 30% of the energy related to refrigerated dryers in Bulgaria can be saved with proper management of the system and using dryers with thermal mass or VSD control. This represents more than 1% of the total electrical energy used in a compressed air production and preparation.

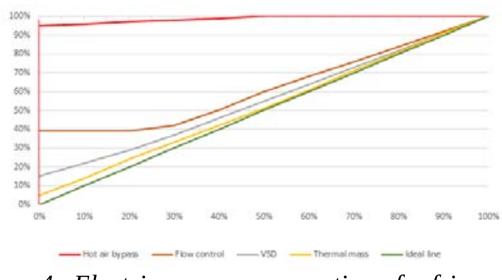


Figure 4. Electric power consumption of refrigerated dryer

Table 3. Air purification requirements

TYPE	Application example	% of total consumption	Air quality class (ISO8573-1:2010)	Impurity of compressed air			
				Nominal filtration (95% filtered practical size)	Moisture		Oil mist concentration
					Dew point	Water contents	
I	**Water removed air** • Air blowing (Simple removal of particles) • General pneumatic tool	2%	3,-,-	3 μm	AP dew point: 6°C 0.7 MPa dew point: 40°C	7 g/m3 (ANR) (At 0.7 MPa 25°C)	---
II	**Dry Air** • Used for the same as "I" and when there is a large temperature drop in the middle of a pipe.	6%	3,4,- 3,5- 3,6,-		Atmospheric pressure dew point: –15 to –23°C · 0.7 MPa pressure dew point: 13 to 3°C	1.7 g/m3 (ANR) to 0.8 g/m3 (ANR)	
III	**Dry air** • General pneumatic equipment • General painting	54%	2,4,3 2,5,3 2,6,3	0,3 μm			Max. 1 mg/m3 (ANR) 0.8 ppm
IV	**Dry and Clean Air** • High grade painting • Sequence control • Measurement device • Instrumentation • Drying and cleaning (Precision parts) • Machine tools (Pneumatic bearings)	9%	1,4,2 1,5,2 1,6,2	0,1 μm			Max. 0.1 mg/m3 (ANR) 0.08 ppm
V	**Dry and Clean Air** • When a refrigerated air dryer is provided in sub line. • Integrated into the equipment (Machine tools, 3D measurement device, etc.)	10%	1,4,1 1,5,1 1,6,1				Max. 0.01 mg/m3 (ANR) 0.008 ppm
VI	**Deodorised air** • Stirring, transporting, drying and packaging • Food industry (Except for direct blowing to foods)	16%					Max. 0.004 mg/m3 (ANR) 0.0032 ppm
VII	**Low dew point clear air** • Drying electric and electronic parts • Drying a filling tank • Transporting powders • Ozone generator • Activation device in a low temperature room	<3%	1,1,1 1,2,1 1,3,1		Atmospheric pressure dew point: –40 to –60°C · 0.7 MPa pressure dew point: –18 to –42°C	0.5 g/m3 (ANR) to 0.02 g/m3 (ANR)	Max. 0.01 mg/m3 (ANR) 0.008 ppm
VIII	**Low dew point clear air (for clean room)** • Blowing semiconductor parts in a clean room	<1%					Max. 0.004 mg/m3 (ANR) 0.0032 ppm

2.2.2. Filtration

Usually filtration is considered in various scientific studies and articles [3], in terms of insufficient filtering and reflection on different elements of the pneumatic system. As this is an important topic, the opposite situation – over filtration is often observed and by energy point of view is more important as every filter in the system brings corresponding pressure drop. When choosing elements for filtration system, the number and thickness of the filters should be considered among with the peak design flow and pressure differential given by the producer. Fig. 5. is an example of information given by the filter producer concerning flow characteristics and pressure drops. As can be found on the graph, pressure drop nearly 0,8*bars* can occur in improper sizing.

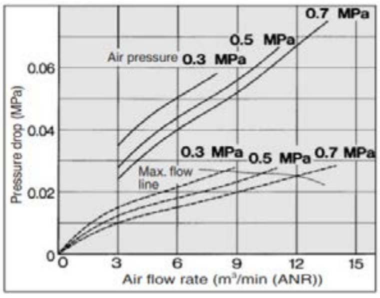


Figure 5. Flow characteristics and pressure drops

In some cases, because of the contamination classes required by the equipment, filtration sequence can reach five or more different stages (pre-filter, filter, micro filter, micro filter with mist separator, odor filter etc.). The designer should work with maximum potential flows. The pipe work in mounting of filters also should be considered to avoid unwanted pressure drops. The screwed and flanged connection to the filter housing is often with much smaller diameter than the up and downstream pipes.

Maintenance of the filter is also very important. It is a good practice for every filter housing to be equipped with pressure differential sensor or indicator which will alert the users of the need of element replacement. The differential pressure of the main line filters should never reach 1bar.

2.3. Transmission and deliver

A pneumatic network is providing compressed air supply to point of use. It is essential in the project phase avoiding pressure for the drops to be considered.

The network consists of main lines, branch lines and supply lines. Usually, factory piping for pneumatic systems is steel or aluminum piping, and within a system, flexible plastic tubing is used. Main lines are usually the bigger pipes, forming the shape of the system. The supply lines connecting main lines or branch lines with end user - point of use (Fig. 6.).

The pressure drops or conductance of the network depends mainly of three factors: diameters and length of tubes in main line and branch lines (in minor consideration for a purpose – material of the tubes), layout and shape of the main line system and connecting elements and type and size of the supply lines. A ring main setup is considered best practice, however, a single main setup may also be useful in many applications. For example, when there is one considerable bigger consumer connected directly to the compressor room with big enough tube and all other consumers require much smaller flow to work. Beside the exceptional proper use of single main line, during the sturdy we have find out that in more than a half of the observed pneumatic systems a significant pressure drops occur due to inadequate shape of the network system.

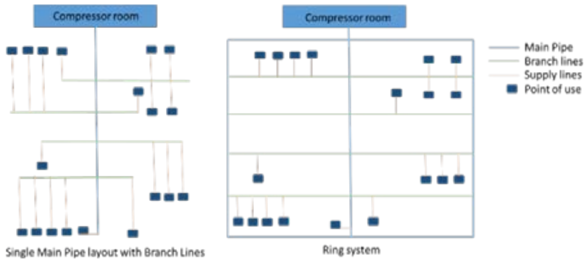


Figure 6. Points of use

It is important to evaluate the pressure losses along the line that can be provoked by the length of the tubes or by the shape of the system layout. The evaluation of the flow-rate parameters is possible by an experimental investigation, using an instrumented test rig, as well as by means of theoretical formulations.

The pressure losses of the pipes can be calculated using the following empirical expression [8]:

$$\Delta P = 1{,}6 * 10^3 * Q^{1{,}85} * \frac{L}{D^5 * P_1}$$

where: Q - flow rate of compressed air [m^3/min (ANR)]; P_1 - upstream pressure of pipe [MPa]; P_2 - downstream pressure of pipe [MPa]; $\Delta P = P_1 - P_2$ pressure drop [MPa]; D - diameter of pipe [mm]; L - length of pipe [m].

This expression can be used to calculate pressure drop at the end of main line (in case of Single Main Line system) or at the end of Branch and Supply lines, applying the pressure at the beginning as an upstream pressure. Such calculation does not take in account the dynamics caused by the actual consumption of the users in the system. Fig. 7. graphically presents the results of the pressure and consumption measurement of a production line, made by us during the study. The blue line is a pressure. It varies between 0,7 and 0,6*MPa*, presenting a pressure drop by 1bar. It is clear that this pressure variation does not correspond with variation of flow (red line) even the flow is significant (between 11 and 15m^3/min). This is a typical case when the pressure drop is caused by complex factors and mainly by consumption of other machines nearby. The problem occurs due to the fact that system was of the type – single main line with branches and the same branch line supply other consumer with significant consumption rate.

In many cases even the system network is with proper shape and size of the tubes, the ignorance of importance of supply lines is observed.

In order to use the simulator, we need to know maximal consumption of compressed air for every consumer and the corresponding place in the production facility. Different type of pipes with different diameters can be simulated as a Main, Branch or Supply lines. The result of the simulation is the value of pressure drop in every point of use. This simulator makes a calculation with assuming that all consumers draw flow at maximum capacity simultaneously, which in real life is not the case. That means that in sizing of the pipes, significant reserve will be shown.

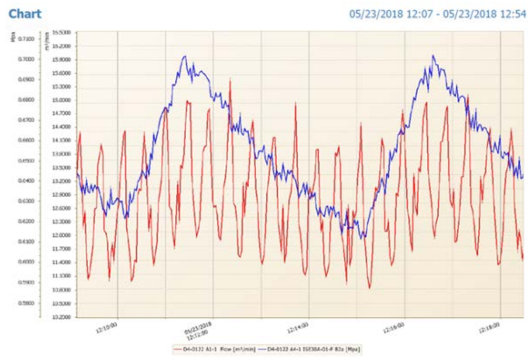


Figure 7. Results of the pressure and consumption measurement of a production line

Fig. 8. presents such a case. The pressure (blue line) vary between 7,3 and 5,5*bars* presenting 1,8*bars* pressure drop corresponding directly to the flow variation (which in this case is relatively small – between 380 up to 650*l/min*). The machine was fed directly from the Main line ring. The problem here is that the supply line (plastic house) was unnecessarily long with small inner diameter.

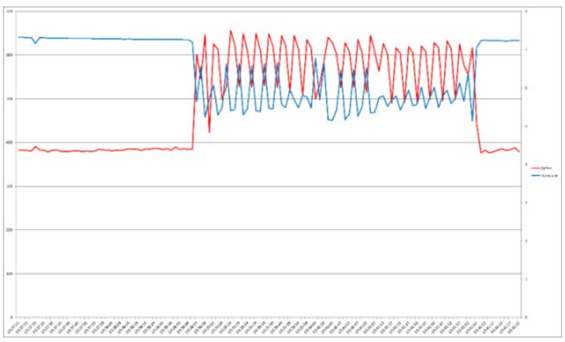


Figure 8. Pressure and consumption

While in this case the result could be predicted and calculated theoretically, in the above case it would be very difficult to predict all the influencing factors.

In case of uncertainty during the design phase, we recommend usage of special software and simulators.

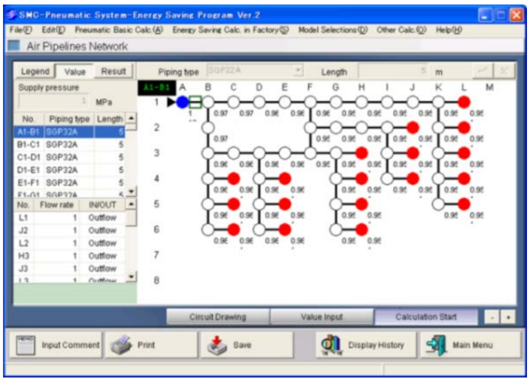


Figure 9. Pressure drop simulation

In Fig. 9. is an example of such simulating software part of Energy saving packet provided by SMC.

2.4. Consumption part

For the purpose of the study it is essential to make a classification of compressed air according to its use. We have defined following roles of usage of compressed air:

A. *Supply air (or automation air)* - used to drive tools and pneumatic machinery. The energy from the compressed air is transferred to mechanical energy. Pilot air (e.g. controlling valves) is also attributed to Automation air.
B. *Blowing air* – The air that is use to blow, clean, drying etc. by means of open tubes, nozzle or air guns. It could be a part of Automation air, as the energy is also transformed in mechanical energy, but with regards to some major differences in use and potential of saving we have put it in separate class.
C. *Vacuum air* – is the portion of compressed air which is used for creating underpressure for some purpose.
D. *Transportation air* - is used to transport substances, media etc. It can be further broken down into conveyor air and vacuum conveyor air which actually transports materials, and active air in a broader sense, when the air is used to blow substances out of a tool or a machine (e.g. for surface treatment). In many cases this type of air is generating separately of the system compressed air by means of low pressure blowers.
E. *Process air* – the air that is used in integration in the processes (e.g. stirring, bubbling, blowing bottles etc.).
F. *Test air* – special air used for testing purpose, to keep clean rooms or cabinets over pressured etc.

In our study we have focused in usage of the first three types of compressed air, as for transportation and process air the savings are matter of change in technology and the test air doesn't generate significant portion of usage in regular production facility.

2.4.1. Air leakages

All compressed air systems have leaks. Leaks are the most visible and most significant contributors to compressed air losses. During our research we made an evaluation of 21 full factory compressed air systems and found an approximately percentage of leakage losses (Table 4.). The amount of leakages presenting was found significant – between 20 and 50% of total air consumption in the system.

Two types of leakages were defined:

Dynamic leaks: These are located at the actuator level (the simplified system is: valve-fittings-tubing-actuator) and do not make an impact when the actuator is on standby. It will only consume during the machine activity periods and only in the part of the cycle in which the actuator abandons its starting position.

Static leaks: These are the leaks which consume most since they are either in the main lines or at the machine level on standby. In the first case they always consume and, in the second, during periods of inactivity and when the actuator is in the starting position.

Table 4. Approximately percentage of leakage losses

Mid-High producer by Industry type	Production/y m^3	Losses in leakages/y m^3	%losses of leakages
Wood working industry	1 596 691	756 422	47,37%
Tailoring industry	430 946	184 608	42,84%
Automotive industry	16 609 398	6 866 325	41,34%
Sports equipment producer	18 363 840	7 415 197	40,38%
Tobacco producer	12 874 680	5 111 974	39,71%
Automotive industry	3 424 320	1 311 319	38,29%
Machinery	51 768 000	18 355 599	35,46%
Tailoring industry	1 873 238	662 468	35,36%
Plastic molding	5 720 585	1 830 587	32,00%
Machinery	2 174 400	668 335	30,74%
Plastic molding	4 375 901	1 283 889	29,34%
Beverage industry	7 125 606	1 976 530	27,74%
Wood working industry	32 956 470	8 655 066	26,26%
Electronic industry	14 857 920	3 833 820	25,80%
Automotive industry	5 228 739	1 319 197	25,23%
Electronic industry	7 362 000	1 853 816	25,18%
Beverage industry	11 169 042	2 727 250	24,42%
Food industry	3 421 440	763 488	22,31%
Bottling industry	5 470 800	1 068 664	19,53%
Bottling industry	6 197 855	1 174 281	18,95%
Tailoring industry	5 332 944	958 330	17,97%

Leaks measurement

a) Direct flow measurement

This method is focused on evaluation that the total amount of air goes through Static type of leakages.

Measurement by a flow meter is necessary in order to quantitatively grasp air leakage. In the first scenario, the flow meter is fitted at the factory air supply, it is possible to grasp the total air leakage by operating an air compressor at nighttime or on holiday when the operation of equipment is stopped and by reading a flow rate at that time.

The second possibility is to measure the flow on every production line or machine, or on a sector or branches (depend of the application and system layout). It is possible to install permanent digital flow meter on every machine supply line and to collect data continuously. This scenario becomes more and more popular as the benefits of it are easy to find and approve, more over the information of compressed air consumption on the machine can be easily redirected to energy cost center and included in calculation of energy cost per product. In addition, creating an ideal model of airflow rate in the machine duty cycle and comparing it with the moment one, giving possibilities to detect presents of dynamic leakages as well.

In Fig. 10. the flow measurement on the supply line of the machine is presented. It clearly shows the value of static leakages – the flat line of end of the graphic when the machine was halted of the operation but still fed with compressed air. The average flow rate during the production cycle is about 450*l/min* of which 260*l/min* are presented by static leakages. Which means that more than half of the consumption of this machine goes to feed leakages.

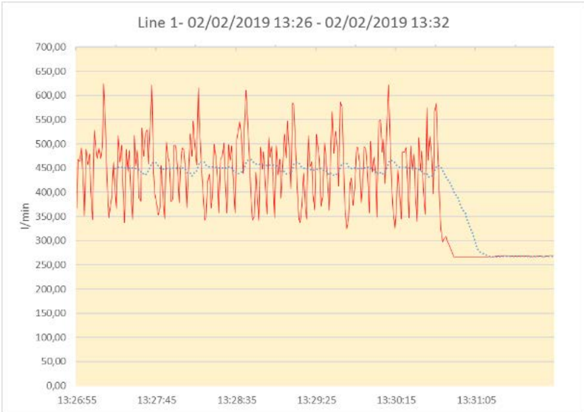


Figure 10. Flow measurement of the machine

Using permanent flow meters on all parts of the system will give exact amount of losses caused by static leakages as a sum of founded and measured static leakages of every flow meter installed into the system.

Although the benefits of such an approach are visible, it is still necessary to make additional investments in measuring instruments, data collection and processing, and so on.

A compromise approach, to maintenance needs, would be to use a mobile flow meter with data collector and at convenient times to measure the flow rate at the input of the machines periodically.

Another approach for flow measuring concerning static leakages is shown in Fig. 11. One flow meter exclusively for leakage tests can be fitted in parallel to a production lines or machines. It makes it possible to measure the air leakage of each line accurately in a short time by switching valves, thus it will facilitate regular maintenance.

The advantages of the latter two approaches to the one with permanent flowmeters are that only one flowmeter is used so, the investment is not so big, and the data can be processed relatively easily.

The disadvantages are that in both cases only data related to leakages can be collected and the measurement can be done with considerable amount of manual work and relatively long time the observed machine should be off duty.

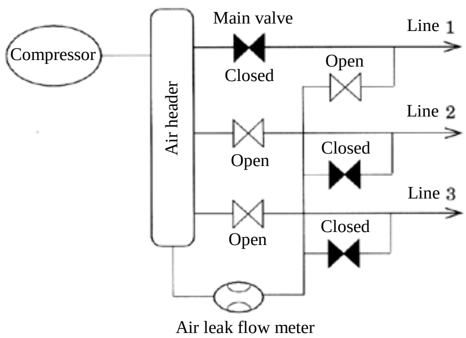


Figure 11. Measuring with one flow meter

b) Ultrasonic detection method

Since direct flow measurement methods give very good results in evaluating the volume or percentage of compressed air flow through the leakages, it cannot give the allocation of leakage on the distribution net or in the machine.

Ultrasound testing belongs to the group of nondestructive methods. For the air leakage, discovering and qualification, a passive ultrasonic method is used. Ultrasounds are generated naturally by fluid turbulence phenomena caused by pneumatic or hydraulic problems (leaks) or by friction phenomena caused by mechanical problems. Electrical problems, such as arcs, corona effects, etc. also generate ultrasounds. In the event of a leak from a compressed air system, the air friction that escapes generates ultrasounds on the sides of the perforation. And it does this whatever the size of the leak, its flow rate and however small the dimension of the hole is [10].

Using ultrasonic detector equipped with appropriate microphones (for short, mid and long distance) makes it possible to find every single leakage wherever it appears. Also with manual manipulating of cylinder systems, it is possible to find dynamic leakages, not only the static ones.

The dependence of sound level, pressure and flow generated by the leakage is given by detector vendor and vary by model to model. However, the evaluation and estimated flow that the leakage is generating is a matter of additional computing and calculation.

It is noticed that this dependence is very strong for small leak diameters, especially in area of up to 1.0*mm* leak diameters. When leak diameters are from 1.0 to 1.3*mm*, correlation is weakening and with diameters of 1.5*mm* and above, the evaluation is hardly done with sufficient precision [12].

In our research we have used the combined method – direct flow measurement for evaluation of losses generated by leakages in machines, production lines, branches or with measurement in the compressor rooms together with ultrasonic method for finding and evaluating of every single leakage, which gives us an exact amount of losses and good estimation where, how often and in which size the leakages appears. The results of such full factory audit are shown in Table 5.

Table 5. Audit results

Summary Plant 1	**Number of leakages**	**Leakage *l/min***	**%Total losses**	**Average single leakage *l/min***
Fittings	76	1681,30	40,81%	22,12
Air preparation	22	557,14	13,52%	25,32
Valve Island	19	378,53	9,19%	19,92
Quick connector	16	391,41	9,50%	24,46
Blow gun	13	225,88	5,48%	17,38
End cap	4	121,16	2,94%	30,29
Speed regulator	5	96,59	2,34%	19,32
Valve	5	93,07	2,26%	18,61
Bord	8	85,29	2,07%	10,66
Cylinders	4	82,77	2,01%	20,69
OTHER	16	406,45	9,87%	25,40
Summary	**188,00**	**4119,59**	**100,00%**	

2.4.2. Air blowing applications

'Air blow' is one of the forms in which compressed air is used. Compressed air is released into the atmosphere either through a nozzle installed in the machine, or by a hand-operated air gun, etc., and the air jet is blown directly onto work-pieces to do work such as draining, removal of machining chips, cooling, transfer and so on. This is one form of actuator, but it is not always recognized as a machine, and its air consumption has tended to be underestimated.

The potential of saving in this application vary within the different applications and it is difficult to be evaluated as a percentage of savings in a factory level. We recommend application to application approach with goal reducing the air consumption of air blow applications to its optimum level.

We can define several issues with using of blowing air we have found during our research:

a) Continues blowing where intermittent usage is possible

In majority of cases where continuous blowing is used, the need is not as it is done. For example – removing water drops from bottle surface just before industrial printing to be applied. It is obvious that this process can be converted to intermittent blowing with small technical improvement – applying proximity sensor-valve system, which will demand blowing only when the bottle appears in to the spot. In other cases where intermittent cycle does not depend of product presents (for example: cleaning of plywood from wooden practices during conveyor movement) the pulse blowing pneumatic valve can be used with adjustable duty cycle and compressed air savings can be realize without using additional electricity and external control (Fig. 12.).

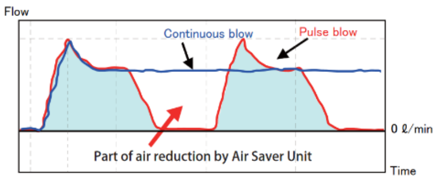


Figure 12. Continuous and pulse blowing

b) Nozzle type and shape

Different types of nozzles were found to be in use during our investigation. In blowing, commonly is used the open tube with inner diameter 2,5 to 4*mm*. Tapered nozzles with variations of diameters and shapes are also widely used, particularly with manual air guns.

The stepped, converging and converging-diverging nozzles (Fig. 13.) use around 5% less compressed air for the same impact force compared to a straight pipe with corresponding diameter.

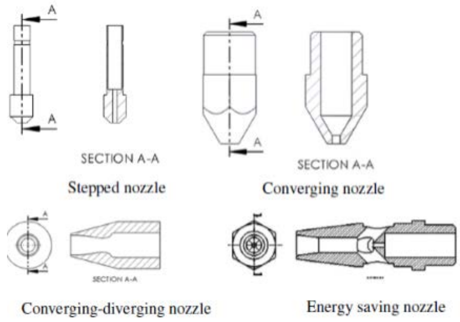


Figure 13. Types of nozzles

A specially designed energy efficient nozzle uses even 10% less air. This nozzle utilizes entrained air form the surrounding to decrease the mass flow used [13].

c) Some additional recommendation which were given during the investigations

Recommended system. A recommended air blow system is shown in Fig.14. A 2-port valve is installed after a check valve, a filter and a regulator at a branch section of the factory line and connected to a blow nozzle by piping. The filter is for removing solid foreign substances, oil and moisture in compressed air to prevent clogging of the blow nozzle and for maintaining the work piece quality. It is necessary to set air quality appropriate for the blow purpose. The regulator is essential to set pressure just before the nozzle and obtain an appropriate air blow effect and airflow rate. The 2-port valve plays a pivotal role for blowing air only when necessary thus preventing wasted air consumption.

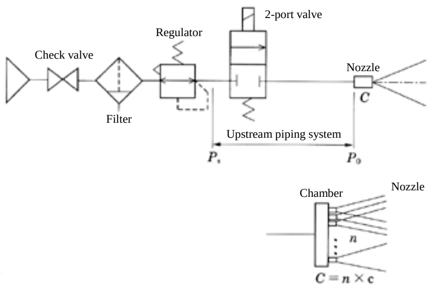


Figure 14. A recommended air blow system

Nozzle spacing. When blowing is done against long and thin work-pieces using multiple tapered nozzles set in line as shown in Fig. 15., it is important to space the nozzles appropriately.

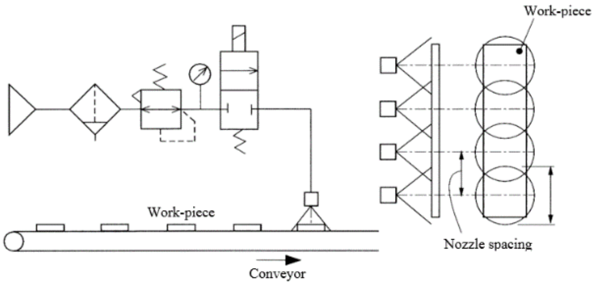


Figure 15. Nozzle spacing

In addition, considerable amount of flow can be saved using proper sizing and manage the pressure at front of the nozzles and distance between nozzle and the impact point.

2.4.3. Vacuum systems

There are systems, which use the power from the pressure difference (vacuum to atmosphere) to pick up and convey work pieces. The vacuum source can be an electrical vacuum pump, or a vacuum ejector using compressed air. A vacuum pump is set up in the factory vacuum line, whereas a vacuum ejector is part of the pneumatic equipment. In our study we have defined how energy can be saved in a system with the vacuum ejectors.

Typical construction of vacuum ejector is shown in Fig. 16.

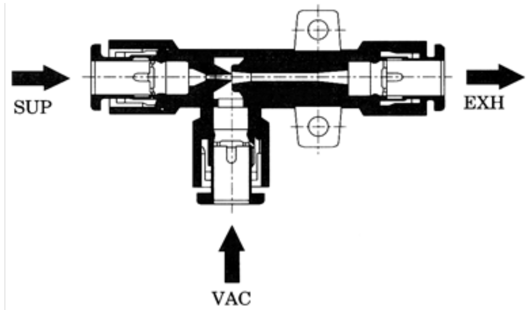


Figure 16. Typical construction of a vacuum ejector

Compressed air is supplied, high-speed flow is generated from the nozzle towards the diffuser, and surrounding air is sucked in and then exhausted to the atmosphere from the diffuser. Two characteristics are important - vacuum pressure and suction rate. The vacuum pressure is related to the capability of suction of the object and it depends mostly of the ejector type and construction and the supply pressure. The suction rate is related to the time that is needed for the vacuum pressure to reach desired level and it depends of the supply pressure and the size of the ejector nozzle.

When we are investigating the possibilities of compressed air savings in ejectors vacuum system, the task can be defined as how to keep vacuum pressure and suction on a best working level with minimum compressed air usage.

In Fig. 17. is presented the suction rate and the exhaust characteristics of vacuum ejector with 1mm nozzle.

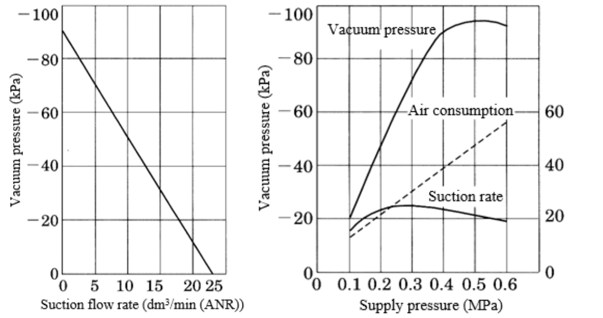


Figure 17. The suction rate and exhaust characteristics of vacuum ejector

It can be seen that the air consumption increases proportionally to the supply pressure, until the vacuum pressure and suction rate reach its peak at about 0,4 *MPa* and then start to decrease [5].

a) Usage of multistage ejectors

One of the possibilities to increase suction rate and keep or decrease compressed air consumption of the ejector system is to use multistage ejector (Fig. 18.).

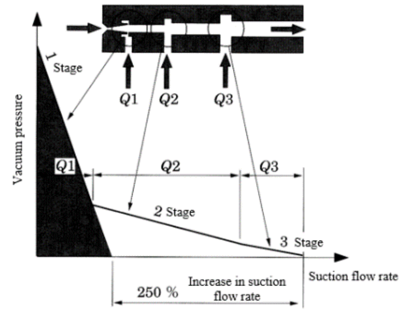


Figure 18. Multistage ejector

Thus the adsorption response time can be speeded up, so compared to the usual single nozzle type, a smaller nozzle diameter, meaning smaller consumption rate, can be applied.

b) Over pressured supply to the vacuum ejector

This is the most common misunderstanding that we have found during our study, that when the suction rate of the vacuum ejector is not enough to increase the supply pressure doing nothing good, but increasing compressed air consumption. If it is the case, the exchange of the ejector with one with bigger suction rate is the solution.

The relationship between the consumption rate and the maximum suction flow rate at the rated point for each size of ejector is proportional which means – the bigger the nozzle is, the higher compressed air consumption will be.

c) Decreasing the suction rate demand

As we have mentioned, the suction rate is related to the absorption respond time. In Fig. 19. is given typical standard circuit of adsorption conveyor system. When the supply valve is in operation, the suction process begins. The time to reach the desired vacuum pressure on the pad side depends of suction rate of the ejector and the volume of air that should be sucked, which on other hand depends of the volume of air in the tubes between ejector and the pad and by the leakages that can be generated at pad side during the absorption.

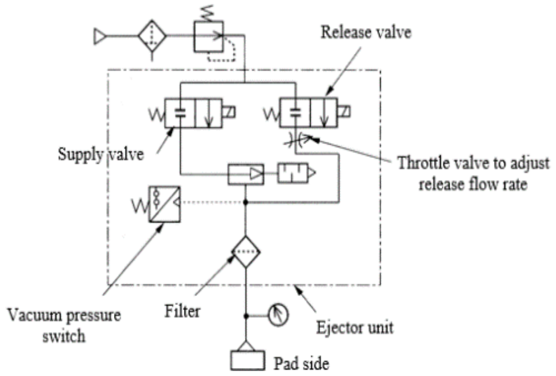


Figure 19. Standard circuit of adsorption

So, we can define two factors which directly reflect to the suction rate demand– the length and diameters of the tube between ejector and vacuum pad and appropriate choice of type and diameter of the pad itself. The smaller volume of air in the tube results in faster emptying and faster vacuum creation, which means the smaller suction rate demand.

d) Ejector with check valve and pressure switch combination (Fig. 20.)

When the object is absorbed and the pad holds it, there is relatively long time to the end of the "pick and place" cycle, in which the ejector continues to consume compressed air. If we fit a check valve to the suction port of the ejector, and a vacuum pressure switch detects work adsorption and closes the supply valve, cutting off the supply of compressed air, the consumption can be decreased dramatically bringing the possible up to 90% savings (depends of the duty cycle). Work pieces are held with vacuum maintained between check valve and work pieces, and if the degree of vacuum falls, the supply valve opens again. It can be modified to the optimum efficiency so that only the required amount of compressed air is consumed.

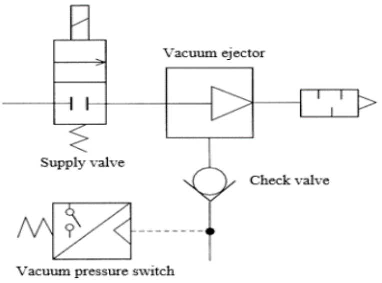


Figure 20. Ejector with check valve and pressure switch

In Fig. 21. is shown the supply valve signal and vacuum pressure dynamic in case of regular ejector system and energy saving ejector system.

The shown system can be equipped with multistage ejector for more savings to be realized. If we consider only the compressed air consumption aspect there is no doubt that the energy saving system should be applied. However, during the design or application optimization stage, the initial price, the maintenance cost and the complexity of both systems should be taken into consideration.

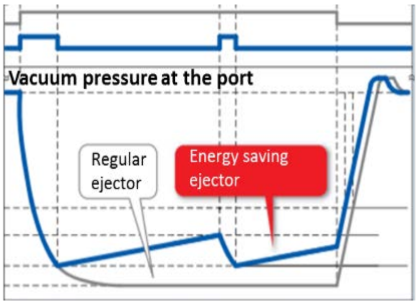


Figure 21. Supply valve signal/compress air consumption

2.4.4. Actuator driving systems

Pneumatic systems are mainly used for converting compressed air energy into mechanical energy such as force, displacement or speed. Actuators used mostly are pneumatic cylinders making linear reciprocating motion. Several hundred to several thousands of cylinders can be used in one business establishment. Therefore, it goes without saying that this driving system forms the significant potential of energy saving for pneumatic systems. The essential point of energy saving is how to construct a pneumatic cylinder driving system that satisfies the required output and cycle time with the most compact products and least air consumption. This chapter describes some subjects of the actuator driving systems in terms of lowering air consumption.

a) Oversized actuators diameter and tube lengths

Simplified driving circuits of typical double acting cylinders are shown in Fig. 22. There are two types of circuits: a meter-out circuit (a) where speed controllers restrict the exhaust flow and a meter-in circuit (b) where they restrict the supply flow. The meter out system is more often used in an application as it has the advantage in smoothness of speed regulation.

For the purpose of this study, we will not take into account the differences in energy usage in both systems, as the air consumption in both cases is similar [11].

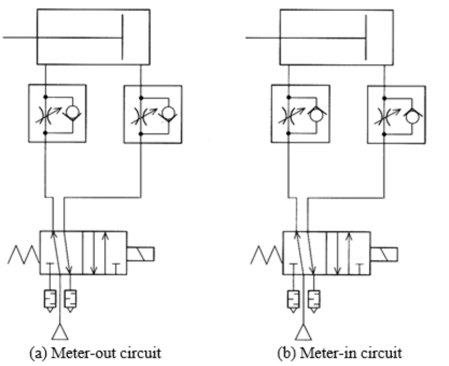


Figure 22. Simplified driving circuits of typical double acting cylinders

For simplicity we can assume that the quantity of compressed air (m^3, ANR) used in both cycles of cylinder work is:

$$q = (V_1 + V_2) * \frac{P_S + 0{,}1}{0{,}1},$$

where: q-quantity of air used in full cycle (m^3,ANR);

$V_1 = V_{1c} + V_{1t}$ – sum of the volumes of the first chamber of the cylinder and the volume of tube, connecting this chamber and the valve (m^3);

$V_2 = V_{2c} + V_{2t}$ – sum of the volumes of the second chamber of the cylinder and the volume of tube, connecting this chamber and the valve (m^3);

P_s – supply pressure (Mpa);

On the other hand $V_{1c} = \frac{D^2}{4} * \pi * l$ and $V_{2c} = \frac{D^2 - d^2}{4} * \pi * l$, where: D-diameter of the cylinder tube; d - diameter of the cylinder piston rod; l - the stroke of the cylinder.

In order to lower the consumption ratio in a cycle of the cylinder we should try to lower (assuming the stroke l of the cylinder is a constant and it is defined in order to fulfil the duty):

- The supply pressure;
- The diameter of the cylinder;
- The volume of air in tubes;

The force in retracted direction (push) is given by

$$F = \frac{P_S * \pi \frac{D^2}{4}}{10} \text{ (N)}, D(mm), P_s(bar).$$

The lowering the diameter of the cylinder will low considerably the volume of air used in a cycle, but at the same time will reduce the force. The same rule is adequate for lowering supply pressure. Precise calculation should be done before optimization (or sizing on design phase) to be started. In fact, in majority of the cases, the designer puts the unnecessary reserve in sizing of cylinders and tubes diameter in order to avoid future claims by the user. As well as the tube length is not every time considered or judgment is done with favor to the fashion outlook and order. For example – six meters long assembly machine in automotive industry on which all valve islands (valve manifolds) are placed in fashion cabinet on one side. Cylinders on the far end of the machine consume remarkable amount of compressed air, emptying 7,5m long tubes on every single movement.

In Table 6. is shown one case that we manage to optimize. In initial situation, more compressed air was used inside the tubes than for operating the cylinder. We have decreased the inner diameter of the tube, without any consequences for the cylinder speed and managed to move the valve closer to the cylinder, reducing tube length and inner diameter. For this cylinder, about 42% saving was realized and on yearly base more than 1500m^3 were saved. Although in the absolute digits, the reduction seems to be relatively small if we assume that there is tenths or in some case hundreds of them, the saving will become significant.

Such problems should be discussed and solved during the machine design phase, as when the machine is already in operation in many cases the optimization is impossible or time and work consuming.

Table 6. Savings for a specific example of a cylinder

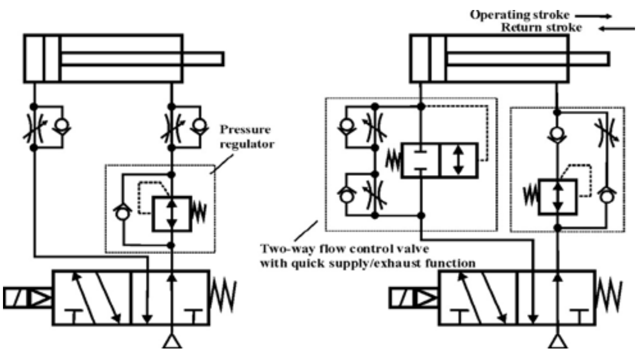
System 1	**value**	**Air volume (I, ANR) full cycle**		
Cylinder bore (mm)	20	Push	Pull	All
Stoke (mm)	100	0,19	0,16	0,35
Rod d (mm)	8	Tube		
Tube inner d (mm)	**4**	0,23	0,23	0,45
Tube length d (mm)	**3000**	Total for cycle		
Pressure (bar)	6	0,80		
Cycles/min	15	Consumption/y (m^3)		
Wh/y	5000	3594,67		

System 2				
Cylinder bore (mm)	20	Push	Pull	All
Stoke (mm)	100	0,19	0,16	0,35
Rod d (mm)	8	Tube		
Tube inner d (mm)	**2.5**	0,06	0,06	0,12
Tube length d (mm)	**2000**	Total for cycle		
Pressure (bar)	6	0,46		
Cycles/min	15	Consumption/y (m^3)		
Wh/y	5000	2089,83		

b) Reducing pressure in "pull" movement of the cylinder

In normal cylinder driving, the cylinder driving speed is adjusted by fitting a speed controller consisting of a throttle valve and check valve between the solenoid valve and cylinder. Both cylinder chambers are filled with the same supply pressure, which is then released. However, in common cylinder applications such as clamping, crimping, pushing and transferring heavy loads in one direction, the specified output is in most cases required only for one side. The supply of lower pressure compressed air is sufficient for the return stroke.

The conventional dual pressure circuit is shown in Fig. 23a. A pressure regulator is mounted on the speed regulator on return port of the cylinder. The pressure level of the return stroke is simply adjusted by means of this regulator to the lower possible operating level. This circuit allows the savings up to 35% of total air consumption of full cycle. However, there are some disadvantages. „Stick-slip" effect occurs in a back movement as the maximum friction force appears immediately after the beginning of cylinder operation. Wakasawa in [16] states that operation delay in a return stroke is significant (see Fig. 25.)



a) b)

Figure 23. a) The dual pressure circuit; b) Two-way flow control valve with quick supply/exhaust function

In order to deal with this problems, we can fit a two-way flow control valve with quick supply/exhaust function on the operation side that requires high output (Fig. 23b.). The supply air from the solenoid valve flows into the cylinder through the meter-in speed control valve.

When the cylinder reaches the stroke end, the relief valve will be opened fully to complete filling at once. When the solenoid valve switches to release, the cylinder side compressed air will start to be released quickly with the relief valve fully open and when it is reduced to the pressure balancing with the spring force by the setscrew, the relief valve will be closed. Afterwards the cylinder will complete the stroke at the speed set by the meter-out speed control valve. The respond time comparison is shown in Fig. 25. [14].

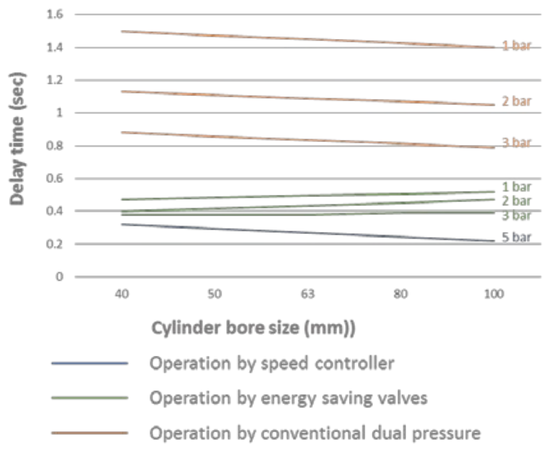


Figure 25. The respond time comparison

c) Additional possibilities of saving compress air in actuator driving systems

There are many actuator circuits developed with regards to additional air usage save, which can be applied to specific application and cases as: *Exhaust air recovery system*, *Accumulator operating system*, *Supply stop driving system* etc., with the energy reducing by 10% up to 40% [15] provided both by pneumatic equipment manufacturers and by researchers in the field.

3. Conclusion

Compressed air energy is one of the main energy in industrial production. The paper attempts to summarize and analyze the main factors - from production to consumption, leading to a loss of compressed. As can be seen, their number determines and reveals the broad potential of the opportunities to limit them. The analysis and evaluation were performed on the basis of quantitative indicators from leading industrial companies in Bulgaria. Audit results for compressed air losses usually show a tendency for energy-efficient units to be problematic. In most cases, successful loss reduction activities require better synchronization and interaction between design engineers, operators, systems maintenance professionals and knowledge of modern tools to create energy-efficient solutions and monitor production and use compressed air.

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