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UNDERSTANDING AIR SYSTEMS

Processing leaf tobacco involves the use of many air systems**.** There are separation systems, transport systems, cleaning systems and capture systems. Air is a gas and it obeys all the laws of physics applicable to gases. A gas will always fill the vessel containing it and its flow from high to low pressure will always take the path of least resistance. In a moving airstream there are three pressure measurements which define the parameters. Static pressure is the pressure at 90 degrees to the flow and constitutes the difference between vessel pressure and atmospheric pressure. Velocity pressure is the pressure generated by the movement of the air. Total pressure is the combined static and velocity pressure. Velocity pressure cannot be measured directly and is obtained by subtracting the static pressure from the total pressure

In tobacco stemmery operations the most important function of airflow is velocity pressure. Velocity pressure creates the force generated by moving air when it impinges upon another object. It is this force which lifts tobacco particles within a separation chamber and which fluidizes tobacco particles in a transport duct. Velocity pressure is proportional to the density of the air and the square of its velocity according to the equation

$$Pv = 1/2\rho v^2$$

Where Pv = velocity pressure in pascals

ρ = density in kg/m^3

v = velocity in m/sec

From this it can be seen that velocity pressure is affected by density and velocity of the airstream. The density of air is affected by altitude (pressure) and temperature. To correctly calculate required air velocity for a given velocity pressure it is necessary to factor in the altitude and average temperature of the air in order to know the actual density.

Lifting tobacco particles by air requires that the velocity pressure of the upward airstream acting over the surface area of the particle generates a force greater than the weight of the particle. For this reason a piece of free lamina with a high surface area and low weight will rise, and lamina with stem or naked stem will drop at a certain air velocity. The surface area is affected by curling of the lamina and weight is affected by moisture. For most threshed tobacco, airspeeds of 400-600 fpm (2-3 m/s) will give the desired selectivity. Note that a change in airspeed has a squared effect on lift as per the equation above for velocity pressure.

For effective separation the incoming flow of tobacco must be dispersed as well as possible so that each particle can be acted upon independently and airflow in the chamber must be as uniform as possible so that each particle receives the same velocity pressure. Dispersion is accomplished by projecting the particles across a wide chamber by means of a winnower, jet of air or high-speed conveyor. Obtaining uniform airflow is far more difficult. The most effective method is to slow the air down as much as possible in a large plenum to minimize vectored velocity pressure and to create a certain resistance to flow so that the air flows from each aperture equally.

Air systems involve a fan to move the air, a chamber or duct system to channel the air and mechanisms to introduce tobacco into the airstream and then to remove it from the airstream.

Fans

For the pressures and volumes normally used in tobacco plants, the most efficient fan is the centrifugal type. This works on the principle of introducing air at the center and then accelerating it to the periphery by means of a rotating fanwheel with paddles or vanes. As the air is forced out by centrifugal force each revolution imparts a higher velocity as the radius of the path increases.

$$v = 2\,\pi.rN$$

Where v = velocity in m/s

r = radius in m

N = rotations in revolutions/s

The accelerated air is discharged tangentially at a higher pressure than the inlet which causes air to flow in the circuit from higher pressure to lower pressure.

Fan Laws

All fans obey the three fan laws as follows for any given circuit:

1. $V \propto N$
2. $SP \propto N^2$
3. $HP \propto N^3$

Where V = volume in m^3/

SP= static pressure in mm water

HP= power in KW

N = fan speed in rpm

These laws allow accurate calculation of the effects of changes in RPM of a fan.

Fan Wheels

There are many types of fan wheel, each suited t a particular application.

The backward incline fanwheel is one of the most versatile and efficient types but is not suitable for handling airstreams which contain large particles of sand or tobacco. Airfoil blades increase efficiency, but as these are light-gage hollow construction they are only practical in clean airstreams as in downstream of bag-filters. With these wheels the clearance between the inlet cone and fanwheel must be tight but still allow small particles to pass.

Open paddle-wheel for use with large particles in airstream. Rugged but very inefficient

Solid-back paddlewheel Closed backward-inclined

For most separation and transport systems handling relatively clean air the closed backward inclined wheel is the most practical and efficient. For certain special applications the radial-tipped wheel may be the best choice.

Fans are selected for particular applications by means of performance charts supplied by the manufacturer. The chart lists required fan speed and horsepower for a particular airflow and static pressure. Once these have been calculated, the correct fan for the application can be selected. Although fans may look similar, small differences in design can affect performance and so it is important to have the data supplied by manufacturers' actual testing in order to determine performance characteristics.

Radial bladed fan impellers can be used in either rotation by mounting in the correct-handed housing. Backward inclined impellers can only be used in the rotation for which they are built such that the inner edges of the vanes scoop the air. Centrifugal fans will move air in both rotations but when run in reverse the airflow is severely diminished. It is vital that correct rotation is checked after every time the motor or connections are changed.

Measurement of airflow

For transport systems where airspeeds are 20-28 m/s the most common and convenient device to measure air speed is the simple Pitot tube. The Pitot tube has two pressure sensors, one being an orifice at the tip of the tube which is pointed directly into the airstream and the other a series of small orifices around the circumference of the tube at 90 degrees to the airflow. The first sensor measures total pressure and the second static pressure. These two pressures are fed to a manometer or mechanical gage which reads the difference between them. As seen above, total pressure-static pressure = velocity pressure. To convert velocity pressure to velocity it is necessary to apply the formula above and solve for v

$$Pv = \tfrac{1}{2}\rho v^2$$

$$v = \sqrt{(2Pv/\rho)}$$

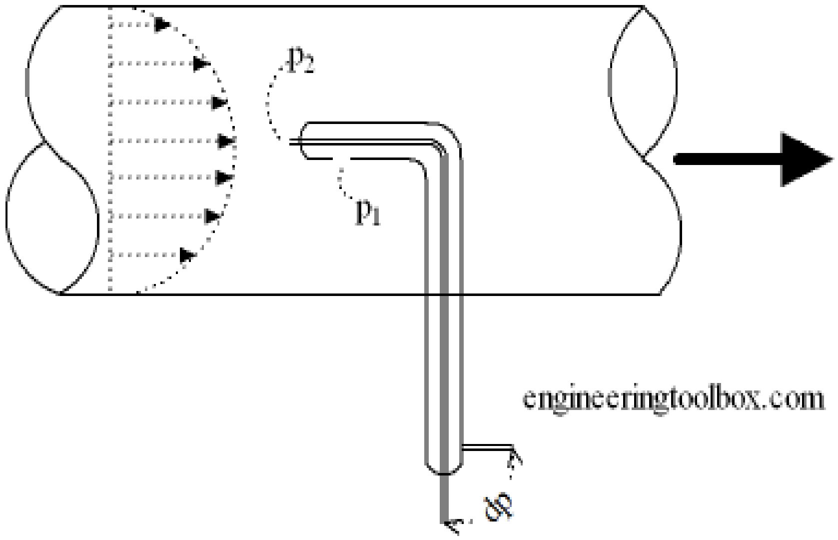
For transport purposes it is not necessary to know the actual velocity, only the velocity pressure so we do not need to apply the formula when simply checking air in ducts. However, to design for correct fanspeed and horsepower we do need the actual velocity because this determines the volume of air the fan must move.

The table blow lists velocity pressures required to effectively transport different products in ducts.

TRANSPORT VELOCITIES FOR TOBACCO

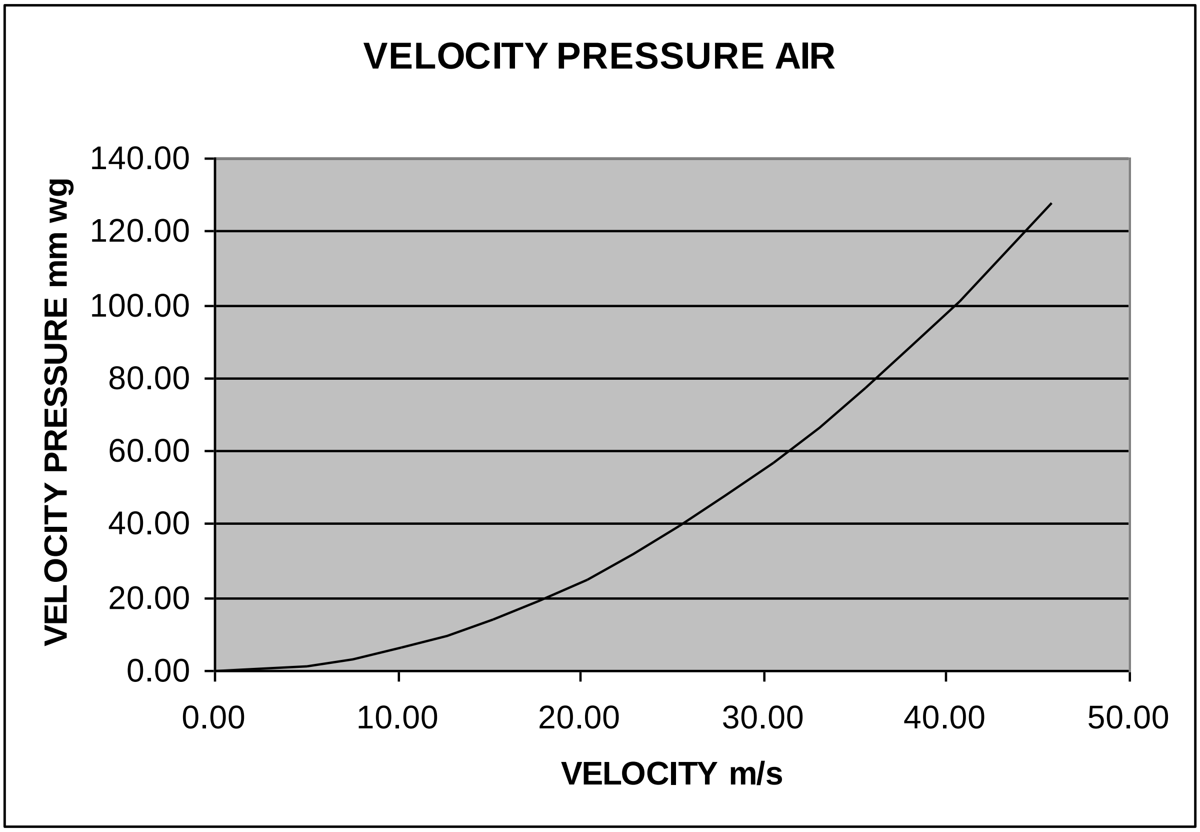
PRODUCT	V fpm	Vp "wg	V m/s	Vp mm wg
Whole leaves < 25% moisture	4500.0	1.3	23.0	31.9
Lamina with some stem	4500.0	1.3	23.0	31.9
Lamina with no stem	4200.0	1.1	21.5	27.8
Scrap	4000.0	1.0	20.4	25.2
Naked stem > 16% moisture	5500.0	1.9	28.1	47.7
Naked stem < 16% moisture	5000.0	1.6	25.6	39.4

When using the Pitot tube it is necessary to take an average of several readings across the width of the duct because the airflow is lower against the walls and highest in the center. The tube must be aligned with the airflow with the tip pointed directly into the flow. Before using each time the orifices must be checked to make sure they are not obstructed in any way. If using a liquid-filled manometer this must be calibrated for zero each time and must be set level. If using a mechanical gage the zero must be set for vertical or horizontal and the gage must be used for the mode selected. When using the Pitot tube to measure velocity pressure it is not necessary to correct for density. For gages which read velocity, this is a direct conversion of pressure using standard air so this reading can be directly applied to the table above.



Pitot tube in duct showing correct alignment. P1 = static pressure, p2= total pressure. dp = p2-p1 = velocity pressure.

For convenient reading in the plant the most common differential pressure gage is the Dwyer Magnehelic. This connects directly to the Pitot tube and is rugged and portable.



For measurement of airflow at velocities < 5 m/s a Pitot tube does not give good accuracy and another type of instrument must be used. The most practical type is the anemometer, which involves a probe with a miniature propeller at the tip. The airflow causes this propeller to spin at a certain speed which is measured electronically and converted to airspeed. The diameter of this probe is generally much larger than the Pitot tube, so larger holes are necessary to insert it into a duct or chamber.

For correct measurement it is important that the probe is clean and the rotor spins freely. Most of these instruments have several functions including an averaging mode. The probe must be placed exactly at 90 degrees to the airflow. For checking air velocity uniformity in separator chambers it is necessary to drill a series of holes so that a grid pattern can be measured and then plotted in three dimensions.

Above is a typical anemometer type instrument by Omega. There are other types of instrument available based on the hot-wire principle or vortex-shedding but these are subject to variations because of air density and therefore less convenient to use.

Ductwork Design

Ductwork for transporting tobacco must be smooth inside and free of any steps, burrs or leaks. The air can only transport a certain weight of tobacco per cubic meter. For dependable results this ration is 3.75 m^3 air/kg tobacco. This means to transport 5000 kg/hr of tobacco, an airflow of 18,750 m^3/hr is required. Once this is known, the table above for transport velocities must be used to calculate duct size.

Example:

What size duct is required to transport 5,000 kg/hr of free lamina?

Airflow required = 5000 x 3.75 = 18,750 m^3/hr = 5.2 m^3/s

Velocity required = 21.5 m/s

Area of duct = 5.2/21.5 = 0.242m^2

Diameter of duct = $2.\sqrt{.242/\pi}$ = 0.55m

If rectangular duct is involved then its area must also be the same $0.242m^2$.

It could therefore be 15 cm x 160 cm.

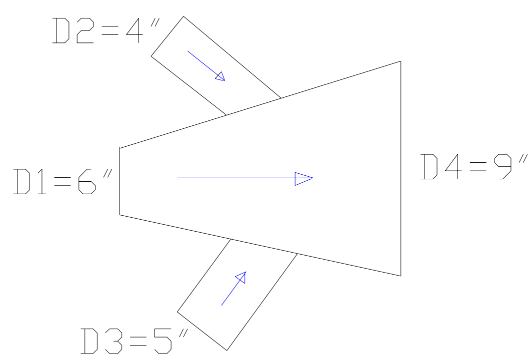
When designing circuits with many branches, the velocity in every branch must be the same. This can be accomplished by simply matching the areas of the ducts.

Example:

A duct of 15cm diameter joins a duct of 25cm diameter to form a common duct. The diameter of the common duct must be:

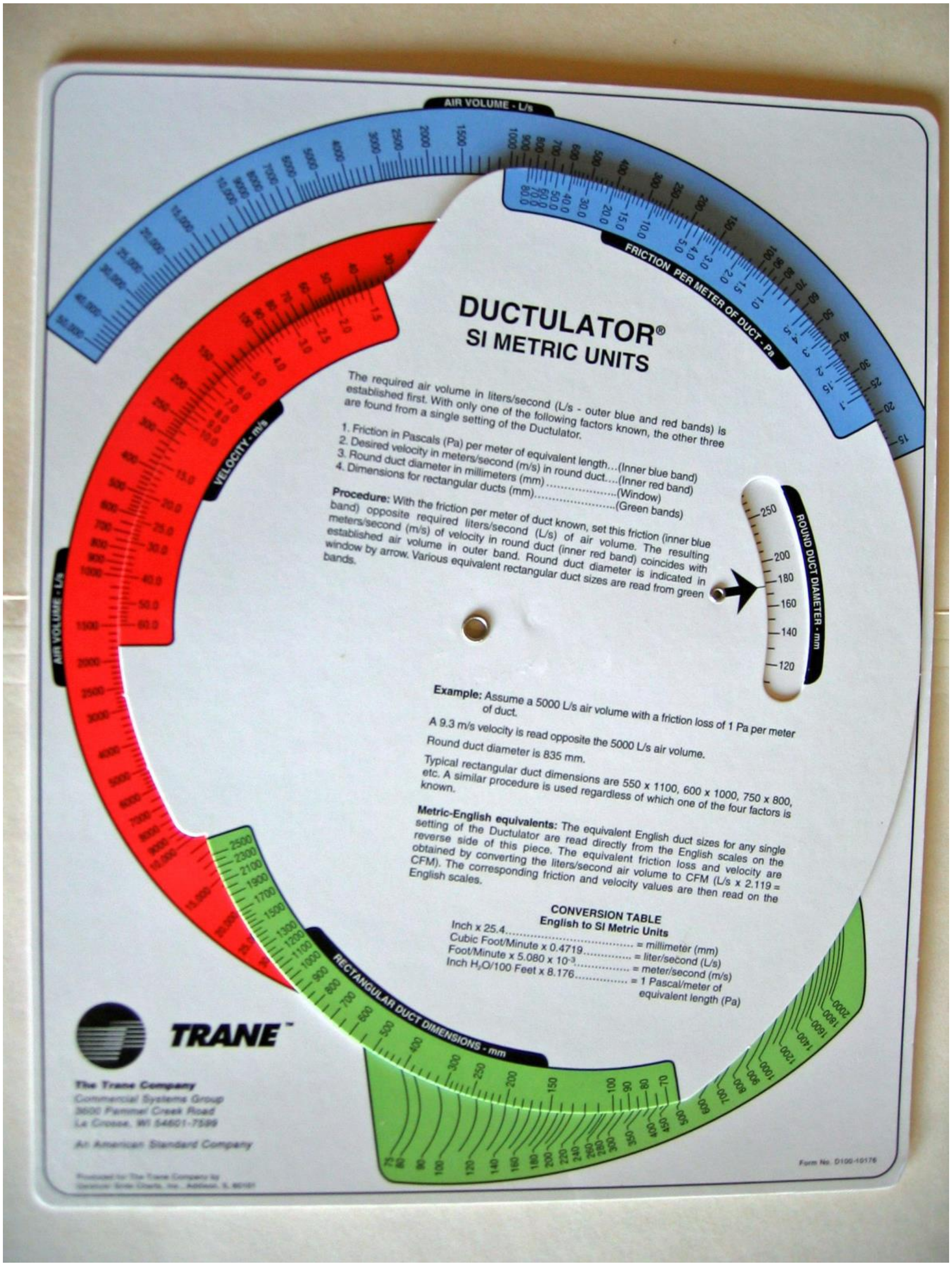
$$\sqrt{(15^2 + 25^2)} = 29.15 \text{ cm.}$$

Combining round ducts is best accomplished by joining the smaller duct to the conical transition at an angle of 45 degrees.



Always A1 + A2 + A3 = A4 so that $D1^2 + D2^2 + D3^2 = D4^2$.

The Trane company provides a convenient tool for ductwork design in the inexpensive and practical Trane Ductulator. This is a cardboard slide-chart in English and metric units which allows the user to easily look up relevant information. If the flow and velocity are known, the Ductulator gives the duct diameter and vice versa. For any given duct size and flow it also provides the friction loss in Pascals per m of duct. It should be noted that the equivalent rectangular duct sizes are for equivalent friction, not area. All the above information can also be arrived at by using charts and tables but the Ductulator is much more convenient. It can be ordered directly from Trane.


AIR VOLUME - L/s
FRICTION PER METER OF DUCT - Pa
DUCTULATOR®
SI METRIC UNITS
The required air volume in liters/second (L/s - outer blue and red bands) is established first. With only one of the following factors known, the other three are found from a single setting of the Ductulator.
1. Friction in Pascals (Pa) per meter of equivalent length....(Inner blue band)
2. Desired velocity in meters/second (m/s) in round duct....(Inner red band)
3. Round duct diameter in millimeters (mm) ....(Window)
4. Dimensions for rectangular ducts (mm)......................(Green bands)
Procedure: With the friction per meter of duct known, set this friction (inner blue band) opposite required liters/second (L/s) of air volume. The resulting meters/second (m/s) of velocity in round duct (inner red band) coincides with established air volume in outer band. Round duct diameter is indicated in window by arrow. Various equivalent rectangular duct sizes are read from green bands.
VELOCITY - m/s
ROUND DUCT DIAMETER - mm
Example: Assume a 5000 L/s air volume with a friction loss of 1 Pa per meter of duct.
A 9.3 m/s velocity is read opposite the 5000 L/s air volume.
Round duct diameter is 835 mm.
Typical rectangular duct dimensions are 550 x 1100, 600 x 1000, 750 x 800, etc. A similar procedure is used regardless of which one of the four factors is known.
Metric-English equivalents: The equivalent English duct sizes for any single setting of the Ductulator are read directly from the English scales on the reverse side of this piece. The equivalent friction loss and velocity are obtained by converting the liters/second air volume to CFM (L/s x 2.119 = CFM). The corresponding friction and velocity values are then read on the English scales.
CONVERSION TABLE
English to SI Metric Units
Inch x 25.4.. = millimeter (mm)
Cubic Foot/Minute x 0.4719.............. = liter/second (L/s)
Foot/Minute x 5.080 x 10-3.............. = meter/second (m/s)
Inch H2O/100 Feet x 8.176.............. = 1 Pascal/meter of equivalent length (Pa)
RECTANGULAR DUCT DIMENSIONS - mm
TRANE
The Trane Company
Commercial Systems Group
3600 Pammel Creek Road
La Crosse, WI 54601-7599
An American Standard Company
Form No. D100-10176

Electronic versions of the Ductulator can also be downloaded. Instructions are supplied with the Ductulator and it is a simple slide-chart to use.

Once the quantity and velocity of air has been calculated as described above, it is necessary to calculate the static pressure loss in the complete system in order to size the fan correctly. Fans are selected from performance tables supplied by the manufacturer. They are based on airflow and static pressure loss which the fan must overcome in order to move the air effectively. There are several components which make up the total static loss in the system

1. Velocity pressure loss. In order to accelerate the air from zero to the required velocity, energy is required. This is measured in mm of water as shown in the velocity pressure table above.

2. Pick-up loss. As air is induced into a duct or hood, turbulence occurs which increases the system losses. Different shapes of entry duct have slightly different characteristics, but in general it is reasonable to use a factor of 0.5 applied to the velocity pressure from (1) above.

3. Ductwork losses. These are the losses caused by friction between the air and the duct walls. They can be read from the Ductulator for each section of duct. Elbows in the duct create additional losses and these are normally translated into equivalent lengths of straight duct. For simplicity it is reasonable to use 3m for 90-degree elbow, 1.5m for 45 degrees etc. The Ductulator gives Pascals loss per m of length so each section must be measured and this number of m applied to the Ductulator reading.

4. Equipment losses. If the air passes though equipment such as separators, screening-separators, venturis, baghouse or cyclones there are losses associated with each which must be taken into account. See later info on typical losses for these items.

5. Fan losses. If the system discharges directly to the atmosphere it entrains ambient air with the high-velocity airstream which creates a back-pressure on the fan. This loss is dependent on the discharge velocity so it is common to fit a section of diverging duct to reduce velocity before the air combines with atmospheric air. This loss does not apply in closed-loop systems.

For strict accuracy the actual velocity of the air must be corrected for conditions where the air is not at standard density because of altitude and temperature differences from standard conditions of 20C and 1 bar. These differences can be looked up in tables to arrive at a density factor which is applied to the velocity pressure equation. As stated earlier, velocity pressure is the important criterion in fluidizing tobacco. We therefore have to correct the velocity by the square-root of the density factor. Fans are constant-volume devices for a given speed and circuit so in order to create the same velocity pressure with lower-density air, the air velocity must be higher and the fan speed greater.

TEMPERATURE C	FACTOR	ALTITUDE m	FACTOR
21	1.00	0	1
27	1.02	150	1.02
32	1.04	300	1.04
38	1.06	460	1.06
43	1.08	610	1.08
49	1.09	760	1.1
54	1.11	914	1.12
60	1.13	1070	1.14
65	1.15	1220	1.16

For example, for a system where the average air temperature is 38C and the altitude is 150m, the combined density factor is 1.06 x 1.02 = 1.08. In order to achieve the same velocity pressure as for standard air, the calculated velocity must be increased by the square root of 1.08 = 1.04. Thus if the calculated velocity is 23 m/s, the actual velocity under the lower-density conditions will be 23 x 1.04 = 23.92.

For most locations this is relatively insignificant, but for plants at high altitudes and high temperatures the density factor should be taken into consideration when calculating fan speeds.

Calculating static pressure in a circuit.

In order to size the fan correctly, total static pressure loss in a circuit must be calculated. In a circuit with many branches it is usually the point most distant from the fan that must be used to determine the maximum pressure loss. If this is not obvious, several different originating points should be examined to arrive at the maximum. As a rule, smaller ducts have far higher friction losses at the same velocity, so long runs of small duct should be avoided.

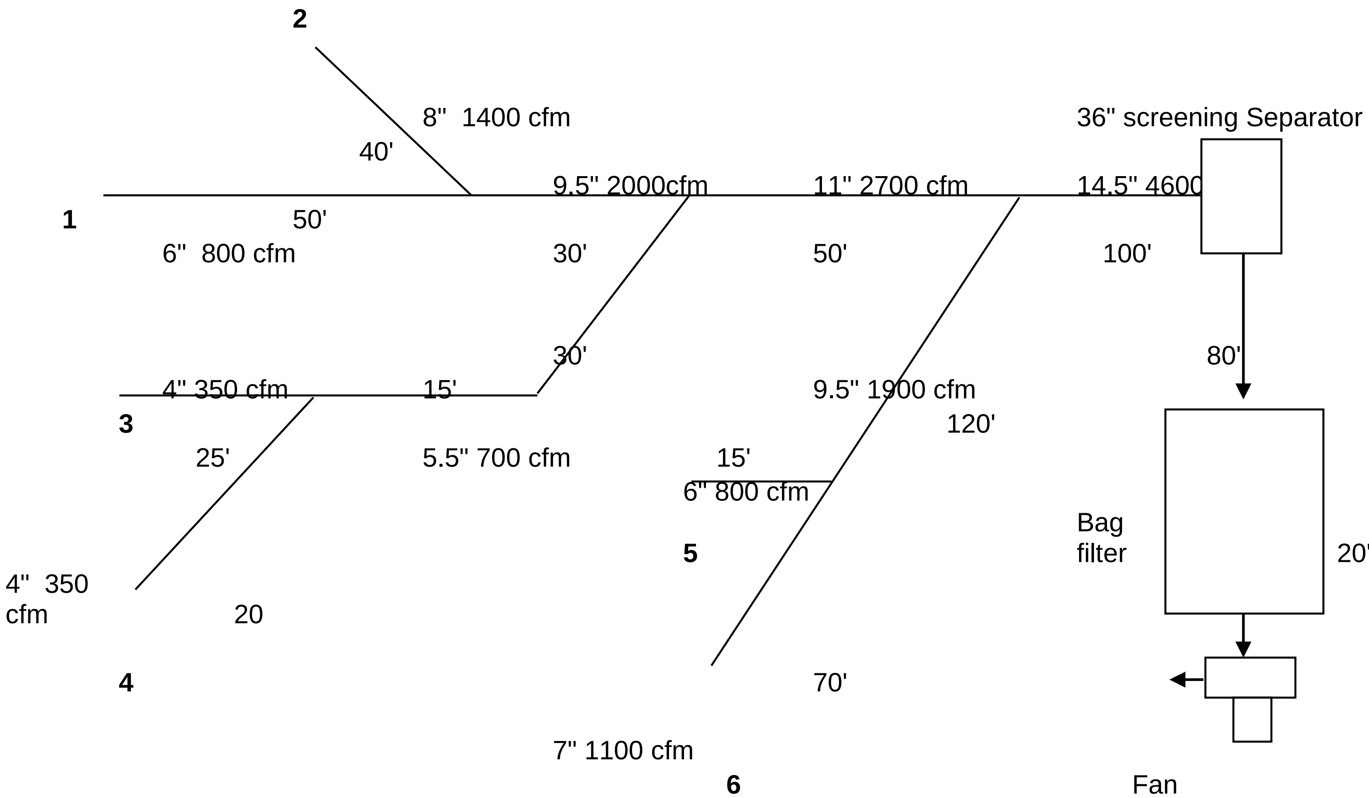
Open circuits

These are circuits that draw air from the atmosphere and discharge it back to the atmosphere via a bag-filter or cyclone. Below is an example of a multi-branch open circuit which would typically be used for dust collection from multiple sources.

OPEN AIR CIRCUIT DESIGN

The system below is used for picking up scrap from multiple shakers and delivering it to one central location.

For scrap we use a transport velocity of 4,000 fpm



First it is necessary to test for the highest duct loss.
This could be circuit 1 or 4 so we test them both to the common junction point

From the Ductulator:

		loss/100'	length	actual loss	
Circuit 1:	800 cfm in a 6" duct 50' long	4.2	50	2.1	inches wg
	2000 cfm in a 9.5" duct 30' long	2.4	30	0.72	inches wg
			Total	**2.82**	inches wg

Circuit 4:	350 cfm in a 4" duct 20' long	6.5	20	1.3	inches wg
	45 elbow in 4" duct	6.5	5	0.325	inches wg
	700 cfm in 5.5" duct 45' long	5.1	45	2.295	inches wg
	45 elbow in 5.5" duct	5.1	5	0.255	inches wg
			Total	**4.175**	inches wg

From this it is clear that circuit 4 will create the highest losses.

Continuing for the remainder of the system:

	2700 cfm in 11" duct 50' long	2.05	50	1.025	inches wg
	4600 cfm in 14.5" duct 100 + 80' long	1.4	180	2.52	inches wg
	2 x 90 14.5" elbows from bag filter to fan	1.4	20	0.28	inches wg
			Total	**8**	inches wg
From table, velocity pressure @ 4000 fpm				1	inches wg
Pick-up loss @ .5 Vp				0.5	inches wg
36" screening separator loss @ 4600 cfm from table				2.7	inches wg
Bag filter loss max				3	inches wg
Fan loss				0.5	inches wg
		Total circuit loss		**15.7**	**inches wg**

Twin City RBA 915 fan, wheel diameter 26.15", 2060 rpm, 17 HP.

Closed Circuits

These are circuits where the air is recirculating in a fixed loop such as through closed-type separators. In fact to reduce dust escaping to the environment, a small percentage of air is continuously removed from the system and this is made up by air entering via any leaks, venture pick-ups or airlocks. This is known as bleedoff air and is normally 5-10% of the amount recirculating. The bleedoff is usually a spade fitted to the periphery of the fanhousing at 45 degrees to the tangent in order to take advantage of the centrifugal effect of the fan which forces dust particles to ride the scroll.

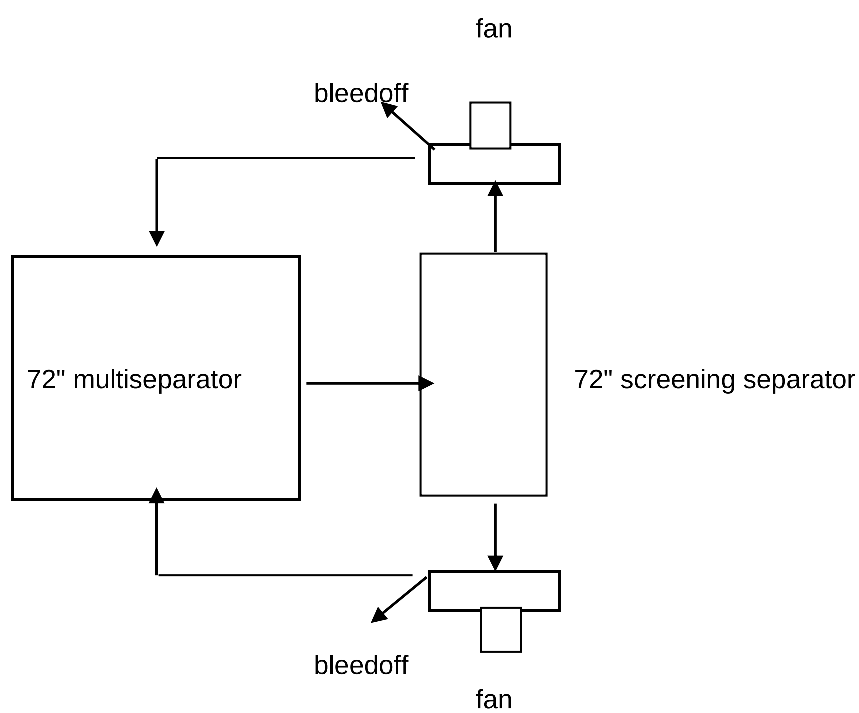
Closed Circuit example:

Consider a separator which requires 600 fpm of uplift velocity in the separation chamber. If the chamber area is 30 ft2 then required CFM is 18,000.

CLOSED-CIRCUIT AIR DESIGN

The circuit below is for a 72" Multiseparator with a lift velocity of 600 fpm

Area of chamber is 6' x 5' = 30 ft2. Total airflow is 30 x 600 = 18,000



In a close-coupled system like the Multiseparator the duct loss is negligible.

As the air is recirculation there is no velocity-pressure loss.

There is no pick-up loss

The multiseparator apron and tobacco particles cause losses	=	1	inch wg
The screening separator causes the largest loss (from table)	=	9.7	inches wg
Fan loss	=	0.5	inches wg
Total loss	**=**	**11.2**	**inches wg**

For symmetry two fans are used, each delivering 9,000 cfm at 11.2" SP

Use Twin City RBA 923 fans, 40" wheel diameter, 1150 rpm, 24 HP

Fan Selection

Unless there are large particles in the airstream passing through the fan, an efficient air-handling fan impeller should be selected. Large means particles > 5mm in any dimension. Large particles can drape over the edges of vanes and build up, causing imbalance and fires. Once the airflow and static pressure requirements for a given circuit have been calculated, the fan can be selected from the manufacturer's tables. Fans of several sizes should be examined in order to arrive at the most efficient.

RBA 923

CFM	OV	1" SP		2" SP		4" SP		6" SP		8" SP		12" SP		16" SP		20" SP		24" SP		28" SP		32" SP	
		RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
3480	1200	349	0.81	473	1.61																		
4640	1600	375	1.17	488	2.14	666	4.32																
5800	2000	406	1.62	511	2.78	678	5.32	816	8.07														
6960	2400	439	2.16	540	3.58	697	6.47	828	9.57	944	12.87												
8120	2800	474	2.82	571	4.50	721	7.79	845	11.21	956	14.85	1151	22.55										
9280	3200	511	3.65	604	5.54	748	9.28	867	13.08	973	17.03	1161	25.46	1326	34.28								
10440	3600	549	4.63	638	6.72	778	10.99	893	15.19	995	19.51	1175	28.52	1335	38.10	1481	48.08						
11600	4000	589	5.82	673	8.10	809	12.85	921	17.52	1019	22.17	1194	31.94	1349	42.19	1490	52.85	1621	63.84				
12760	4400	629	7.18	710	9.73	842	14.91	951	20.09	1046	25.13	1215	35.57	1365	46.43	1502	57.74	1630	69.53	1749	81.44	1862	93.55
13920	4800	671	8.81	747	11.55	875	17.12	982	22.86	1075	28.38	1239	39.53	1385	51.12	1518	63.03	1642	75.40	1758	88.01	1869	101.06
15080	5200	714	10.71	786	13.66	910	19.65	1014	25.83	1106	31.94	1265	43.81	1407	56.09	1537	68.73	1658	81.74	1771	94.99	1879	108.68
16240	5600	758	12.91	826	16.04	945	22.43	1047	29.02	1137	35.67	1293	48.45	1432	61.52	1558	74.76	1675	88.25	1786	102.21	1891	116.45
17400	6000	802	15.39	866	18.68	981	25.51	1081	32.51	1169	39.63	1323	53.52	1458	67.23	1581	81.15	1695	95.28	1804	109.96		
18560	6400	847	18.22	907	21.64	1018	28.94	1115	36.26	1202	43.87	1353	58.77	1485	73.26	1606	87.99	1718	102.89	1823	117.95		
19720	6800	892	21.37	949	24.98	1056	32.71	1150	40.41	1235	48.34	1384	64.35	1514	79.78	1632	95.16	1742	110.81	1845	126.59		
20880	7200	937	24.87	991	28.63	1094	36.77	1186	44.96	1269	53.20	1416	70.24	1544	86.69	1659	102.72	1767	119.09	1869	135.77		

MAXIMUM RPM: CLASS 22 = 1581 CLASS 32 = 1905

Above is an example of A Twin City fan table for 40" RBA airwheel. Assume that the required airflow is 10,000 CFM and the calculated static pressure for the system is 12" wg. Applying these criteria to the chart, it can be seen that this will require a fan speed of about 1175 rpm and a horsepower of 28.52. The underlined numbers indicate conditions of peak efficiency for this fan. To be strictly accurate, the numbers should be interpolated between the actual CFM and the closest match on the table. Thus we have:

9280 CFM 1161 rpm 25.46 HP

10440 CFM 1175 rpm 28.52 HP

Actual rpm @ 10000 CFM = 1161+(1175-1161)*(10000-9280)/(10440-9280)

= 1170

The same interpolation can be applied to the horsepower. However, as can be seen, the differences are normally minor.

If the density of the air is significantly different from standard conditions, the density factor must be applied to the calculation as described above. In the example above, assume the same 10,000 cfm of standard air is required but the altitude is 1200m and the temperature is 40 C. The combined density factor is 1.16 x 1.07 = 1.24. The CFM must be increased by the square root of this in order to maintain the required velocity pressure. Thus the actual CFM will be 10,000 x $\sqrt{1.24}$ = 11135. If this were standard air, the static pressure would increase as well and would need to be calculated based on the higher volume. To find the correct fan speed, these numbers must be applied to the table. However the actual static pressure and horsepower would be those from the fan table reduced by the density factor of 1.24.

It is important to choose the best fan for the calculated duty in terms of energy consumption, noise and wear. Using the fan tables, several different sizes and impellers can be compared. In general, closed impellers of large diameter give the best results as they work at lower rpm, create less noise and have a longer life. Fan design is a specialized science and fans that look alike might perform very differently. Beware of suppliers that simply copy fans designed by others, small differences in shape and clearances can have large differences in performance. In the USA accredited fan suppliers must test their fans under specific conditions and these fans are certified by the AMCA (Air Movement and Control Association)..

Fans can operate in parallel and in series. For large separators it is easier to obtain uniform airflow in the chamber by using two identical fans in parallel, each delivering half the total airflow. For long circuits with high static pressure losses two or more fans can work in series. Each handles the same airflow and the sum of each fan's static pressure load equals the total static pressure of the circuit. The static pressure of each need not be the same, e.g. for a circuit with a total SP of 19", one fan may contribute 12" and the other 7".

For closed-impeller fans it is important that the inner ring on the impeller is narrow and does not have a very tight clearance to the inlet housing otherwise dust can accumulate inside the ring and cause fires which are the transmitted to the bag-filter.

Under certain conditions fans require heavy-duty shafts and bearings. The manufacturer's literature should always be checked for this. Usually these conditions are in bold type or highlighted in the performance tables.

Fan performance is affected by inlet ducting. The best situation is five or more diameters of straight ducting before the inlet. If this is not possible, large-volume plenum inlets slow down the incoming air and allow better uniformity.

Fan drives involve certain belt losses and so where variable fan speed is required an impeller mounted directly to the motor shaft has less maintenance and losses than a belt drive. In these cases the motor rpm must be selected to handle the torque required by the fan at operating speed. Correct fan speed can be accomplished using a variable-frequency drive. This can be set to a higher frequency than the local utility supplies as long as motor amps do not exceed the maximum designated on the motor nameplate. If motor is run at less than the nameplate frequency the horsepower will be reduced by the same percentage. However, by selecting a motor with higher torque and lower nameplate rpm and running it at higher than nameplate frequency full power can be attained. See example below.

Fan requirement is 1350 rpm and 28 HP. Required torque is 109 lb-ft

For a 4-pole motor rated at 1750 rpm, running a 30 HP motor at 1350 rpm (46 Hz) will give only 23.14 HP.

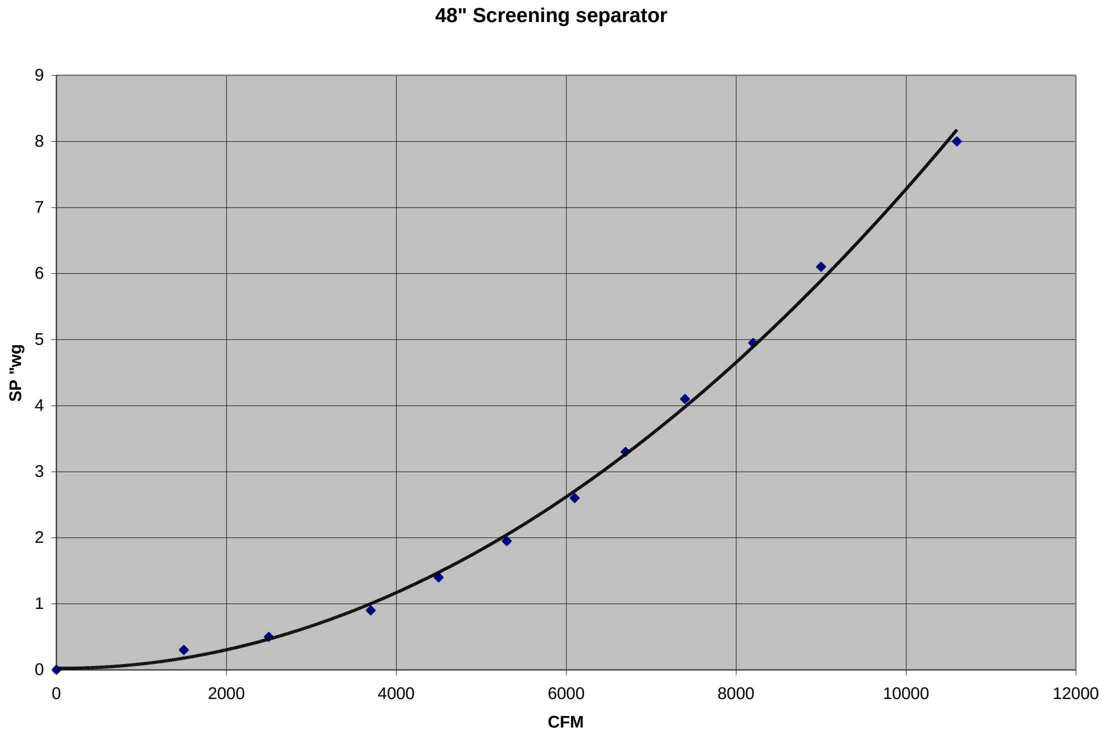
For a 6-pole motor rated at 1150 rpm, a 30 hp motor delivers 137 lb-ft of torque.

Running it at 1350 rpm or 70 Hz and 109 lb-ft torque gives the required 28 HP without overloading the motor.

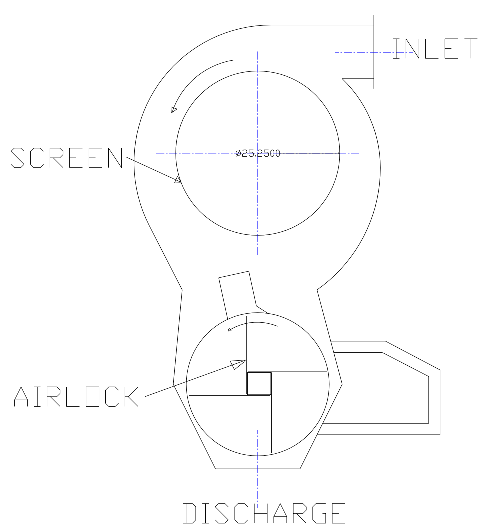
Screening Separators

The most common device for removing tobacco from an airflow is the screening separator. This is essentially a horizontal cyclone with an airlock mounted along the bottom periphery. A rotating screen regulates the particle size that can pass through to the fan and so a lower separation velocity than a cyclone can be used. Cyclones have very high pressure losses which are costly in terms of energy. The rotating screen is continuously cleaned by the action of the incoming tobacco striking the top radius tangentially.

The screening separator size must be selected for a given airflow to give a minimum entry velocity otherwise the centrifugal effect will be insufficient to separate the tobacco from the airstream. When this occurs, the tobacco sticks to the rotating screen and eventually chokes the flow. Experience has shown that the static loss should be above 2” which, in the case of the 48” machine in the table above, requires an airflow of > 5,400 fpm. Screening separators can handle broad ranges of flow but static losses > 10” should be avoided.

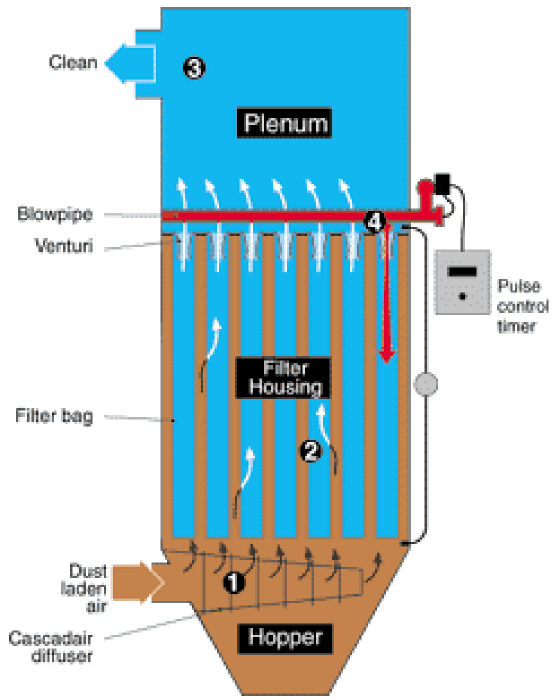


The chart above shows the static-pressure loss for a 48” screening separator.



SCREENING SEPARATOR SIDE SECTION

Bag filters



The most popular means of filtering air before discharging to atmosphere is the pulse-jet bag-filter. This is essentially a large metal plenum containing an array of felted polyester bags which filter out 99.9% of particulate matter in the air. The bags are cleaned periodically by a blast of air from the inside which dislodges particles adhering to the outside. Although simple in principle, bag filter design is a science and certain criteria must be followed if efficient operation is to be accomplished.

The first and most important criterion is average air velocity through the bag material. For typical tobacco applications this should be no more than 10 fpm (3m/m) and ideally about 8 fpm (2.44 m/m). Secondly, the upward velocity of the air as it passes between the bags should be no greater than 400 fpm (2 m/s) because lift velocities above this will hold particles in suspension and prevent the system from cleaning itself.

The lift velocity is the total flow through the system divided by the plenum area minus the combined areas of the blank bag bottoms.

For example a baghouse has a cross-sectional area of 12’ x 12’ and 440 4” diameter bags each 10’ long.

Total filter area is $440 \times 4\pi/12 \times 10 = 4608$ ft2

At a velocity of 8 fpm, the capacity would be 36,864 cfm

The lift area (between bags) would be 12 x 12 –(440 x 4π/144) = 106 ft2

The lift velocity would be 36,864/106 = 348 fpm.

Bags longer than 12' are discouraged because the pulse-jet system cannot clean the furthest points effectively. The pulse-jet requires a properly designed venturi above each bag. A jet-bar above this supplied with compressed air at 90-100 psi injects short blast of air through the venture and entrains surrounding air with it, giving a shock-wave down the bag to clean it.

Bag filters are best installed to run at negative pressure so that the fan handles only clean air. They can run positive when the system design favors this. In both cases, the structure must be designed to comfortably withstand the maximum possible pressure it is likely to see. A typical system runs at 12-20" wg negative which means the structure is subjected to a force of roughly 0.8 psi. This does not sound like much, but when applied over the flat surface of the plenum 10' x 12' in this case, the total force is 10 x 12 x 144 x 0.8 = 13,824 lb. (6,283 kg). The plen faces must therefore be reinforced with ribs to withstand this force without buckling.

With a negative system, any leaks detract from the designed flow at the entry points. Both the duct system and baghouse must be completely free of leaks and airlocks and access doors must be tight. Welded baghouses are best as any bolted panels may leak.

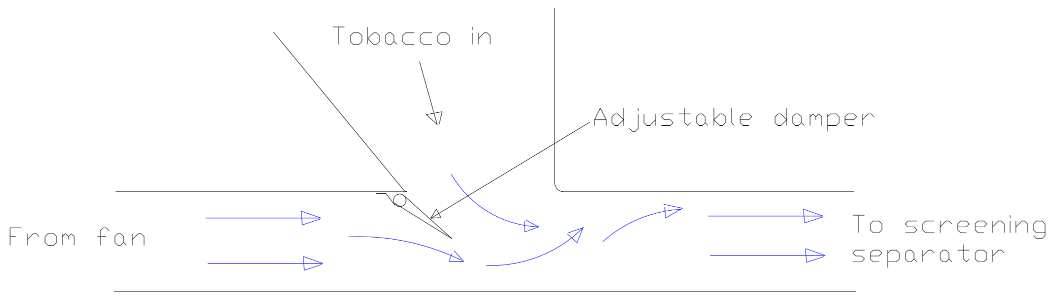
Baghouses in tobacco plants are typically designed for maximum pressure loss across the bags of 3" wg (75mm). It is important that this pressure differential be regularly monitored via Magnelic gage mounted to the baghouse. New clean bags may have a drop as low as 1" wg but as dust particles become impregnated in the fabric, this rises. The pulse system is controlled electronically and can be set for pulse duration and interval between pulses. For most systems a duration of 0.5 seconds is adequate, but interval varies according to the dust loading of the incoming airstream. This can range from 30 seconds to several minutes. By monitoring the differential pressure the interval can be set to maintain this at between 2" & 3" wg. If it drops below 2" the interval time can be increased. If it rises above 3" the interval time must be decreased. Baghouses are major consumers of compressed air and so from the energy standpoint it is important to optimize the pulse program.

In humid climates it is normal for bags to become clogged with dirt after 2 or 3 seasons and regardless of pulse frequency the pressure loss increases. When this occurs the bags must be replaced.

Venturis and pickups

Proper design of venturis and open pick-ups is important for trouble-free operation. Venturis are used in closed circuits to introduce tobacco to the airstream. Open pickups are used to introduce tobacco into an open airstream.

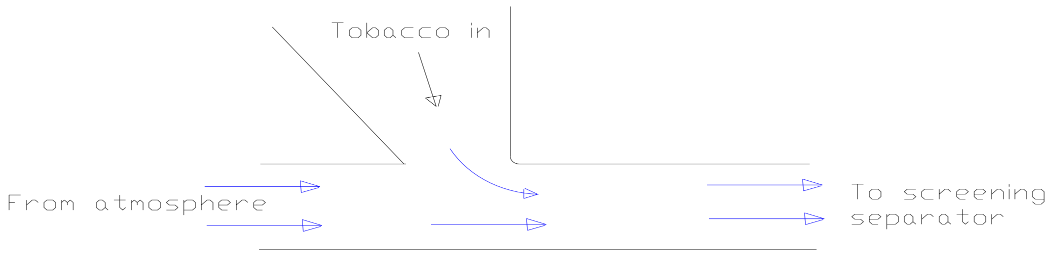
A venture involves a constriction in a duct which increases velocity and drops pressure. If the area above the constriction is open to the atmosphere or chamber, the low pressure in the venture induces air from the atmosphere or chamber into the closed circuit. The amount of air induced depends on the relative pressure difference.



.Above is a typical venture in a rectangular duct where tobacco is introduced into the airstream via the chute above. The damper vane is normally about 60% of the height of the duct and adjusted to an angle of about 45 degrees. Final adjustment is performed by feeling the air in the throat of the venture with the fan running. With correct adjustment a slight negative pressure should be felt. The best way of judging is to hang a piece of light, flexible fabric such as a handkerchief in the throat. It should be puled gently downwards. If it is blown upwards or flutters, close the damper a little. If it is sucked strongly downwards, open the damper a little. When adjusted correctly, tobacco should flow smoothly into the throat.

It is important that the damper is sealed tightly at top and sides. Any leakage will cause turbulence. As air is added to the system at the venture, an equal amount must be removed from the system via a bleedoff. The bleedoff should be set to remove about 10% of the circulating air.

As a general rule-of-thumb the opening in the venture throat should not be wider than the height of the duct. The compressed air expands rapidly and will cause blowback. It also helps to make a radius on the downstream corner of the chute to avoid tobacco particles being trapped on a sharp corner.



Above is a typical open inlet system. Air takes the path of least resistance and therefore prefers a straight path. By extending the duct upstream of the inlet most of the air enters here and a lesser amount through the chute, thus ensuring ample velocity where the tobacco falls. Again, the width of the opening should not be greater that the height of the duct or dead spots will occur and tobacco will accumulate there.

For horizontal pickups such as floor-sweep systems the duct can simply be open at the end provided that the entry velocity meets the criteria listed earlier for specific products.

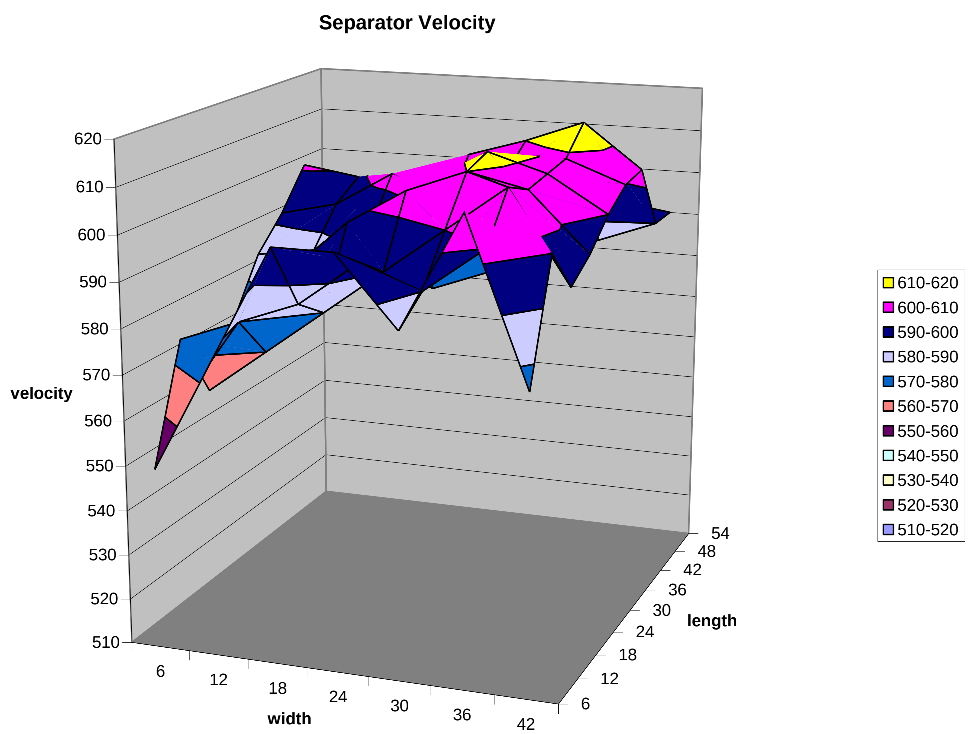
General Considerations for Air Systems

For any fluid, whether gas or liquid, flow can be laminar or turbulent. For transport and separation systems it is important that the flow is laminar because turbulence creates inconsistencies which affect smooth operation. Turbulence can be caused by excessive velocity, intersecting airflows, obstructions in ducts and rotating equipment within the air chamber.

For separators it is vital that upward airflow is uniform in velocity and direction and is laminar. Perforated plate aprons create a small resistance to flow which smoothes out some inconsistencies as long as the plenum below is large enough to reduce velocities to low levels. The ducts feeding air to the plenum must be as large as possible to keep velocities low and avoid vectored air which will create high velocity pressure at some point in the plenum. The perforated plates also vector the air so that direction is constant.

To evaluate the air pattern within a separator chamber, two methods can be used. In the first, thin rods are placed across the chamber at intervals of about 15cm and at the same height the tobacco enters. Pieces of cotton thread about 15cm long are tied to the wires about 15cm apart. Start the fan, apron and winnower and observe the movement of the threads. If they all stream vertically upwards the pattern is good. If some are up, others down or sideways the velocities are inconsistent. If the threads have wild changes in direction there is turbulence.

The other method is to drill holes in the chamber large enough to accept an anemometer probe and measure the chamber area velocity on a grid pattern, also 15 x 15 cm. This can then be plotted on a 3D graph so that inconsistencies can be seen clearly. A good separator should have velocities of about 3 m/s +/- 0.25.



The surface chart above is a typical example of a velocity profile for a 48" separator. As can be seen, there are low points in the corners, as in a rectangular plenum it is difficult to get high flow in the corners. Despite the apparent variation because of scaling, this is not a bad profile. Perfection is almost impossible.

Separators rely on the differences between surface-area-to-weight ratios of particles to lift some and allow others to drop. The velocity pressure of the air acting on the exposed area of the particle creates lift and gravity creates down force (weight). If the lift is greater than the weight the particle rises and is separated.

Good separation depends on good particle dispersion and uniform lift velocity. Dispersion depends on the projection device, the size of the chamber and the load through it.

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