MODIFICATIONS AND IMPROVEMENTS OF TWO SEED COTTON UNLOADING SYSTEMS

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Abstract

Pneumatic conveying systems are commonly used in the cotton ginning industry. Inefficiencies in those systems can reduce productivity, create choke ups and result in high operating costs. The fact that inefficient pneumatic conveying systems are costly is nothing new to the ginning industry. But how can system losses be determined and how accurate are the equations recommended for calculating those losses? The purpose of this paper is two-fold: 1) document modifications that were made to the seed cotton unloading system at the USDA-ARS Lubbock, TX ginning laboratory and compare those changes to values that were obtained from using standard friction loss calculations found in the literature, and 2) report the effect that reducing or eliminating the inefficiencies had on power consumption. The results showed that the equations ranged from less than 1% to 19% of the actual measured system losses depending upon the calculation method used. Modifications resulted in a 37% reduction of power usage in the unloading system.

Introduction

A critical aspect of any cotton gin is the seed cotton unloading system. It is the unloading system that is responsible for transferring seed cotton from the trailer or module into the gin for processing. An essential requirement of the unloading system is to convey the material into the gin at as constant and uniform of a rate as possible (Laird et al., 1994). General components of a seed cotton unloading system, depending on the type of cotton being handled, are: 1) a means of introducing seed cotton into a suction conveying pipe, 2) vertical and horizontal conveying pipes, 3) a seed cotton separator, 4) a green-boll separator (optional), 5) an airline cleaner (optional), and a 6) centrifugal suction fan or fans (Baker and Griffin, 1984). Currently, there are two types of seed cotton unloading systems predominately in use; 1) pneumatic suction through a pipe, and 2) module feeding systems. The first system is one of the earliest mechanical means of bringing seed cotton into the gin (Bennett, 1962). Over the years, automation has improved the early pneumatic unloading systems into the more common swinging telescope suction system still in use today. The second type of unloading system is the Cotton Module Feeder. This system came about as a result of storing seed cotton in modules. With the advent of cotton modules, a means of bringing the seed cotton into the gin that addressed problems specific to module storage and handling was needed. Regardless of the seed cotton unloading system used, pneumatic conveying of seed cotton is utilized in the transfer of seed cotton into the gin for processing.

Materials handling in a cotton gin is primarily performed by either centrifugal and/or axial type fans. Both types of fans represent one of the largest power consuming elements of a cotton gin, with pneumatic systems consuming 40 to 60 percent of the total power required to operate a cotton gin (Mangialardi, 1977). The seed cotton handling system is one of the main energy consumers in a cotton gin, of any of the materials handling systems (Watson et al., 1964, Anthony, 1989). Energy consumed from the handling of seed cotton will vary dramatically depending on factors ranging from size and design of the gin, moisture and foreign matter content of the seed cotton, to operating and management procedures (Baker and McCaskill, 1979). One of the factors affecting energy consumption of any pneumatic conveying system is leaks. Some of the equipment utilized in cotton gins today have a certain amount of inherent "leak" associated with their operation. For example, the green-boll traps, vacuum droppers, air line cleaners, and suction separators are all devices which will experience some air leakage. Typical leakage rates have been established for most seed cotton processing equipment. These standard leakage rates are based on manufactures' data or on practical field experience. However, excessive leaks over and above the standard rates can create problems with proper conveying and handling of seed cotton, resulting in less efficient energy use, and pose potential maintenance and operational problems resulting in increased downtime and reduced capacity.

A leak does not have to be very large to have a significant impact on proper operation of an unloading system. For example, a general rule-of thumb states that 15 to 20 cubic feet of air is needed to convey one pound of seed cotton. Likewise, the design air velocity should be 5,500 to 6,000 fpm in telescope pipes and 3500 to 5000 fpm in horizontal and vertical conveying pipes (Baker et al., 1994). For a 20 bale/hour gin processing 2200 lbs of stripper harvested cotton per bale of lint and using 15 cubic feet of air per pound of seed cotton, the volumetric flow rate needed for conveying would be 11,000 cfm. To convey the seed cotton at a velocity of 5,500 fpm, the diameter of the telescope pipe should be 19 inches. Assume the gin has a 60" air line cleaner with four lids. If each lid had a 1/16" crack along one edge, the total gap area would be 15 square inches. If the volume of air leaking into the air line cleaner was only a modest 1000 cfm,

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the resulting upstream volumetric flow rate would be 10,000 cfm. This would result in the velocity in the 19-in. dia. (19-inch diameter) suction line being reduced from the desired 5500 fpm to 5079 fpm. Thus, even a small often overlooked leak can begin to chip away at the design velocities needed for optimum performance of the conveying system.

In addition to air leaks, another factor affecting efficiency of an unloading system is pressure drop due to friction and dynamic losses in the ducts. These losses are important since the fan has to produce sufficient static pressure to overcome the friction in the various system components. The total friction losses through any round duct are directly proportional to pipe length (the longer the pipe the greater the loss), inversely related to the diameter of the duct (the larger the pipe the smaller the loss), and proportional to the square of the velocity of air moving through the duct (higher velocity results in double the loss). For dynamic losses, pressure drops are dependent upon the number and types of elbows as well as the frequency with which the velocity of air changes as it flows through the piping system (ACGIH, 1998 , Murdock, 1996) . Air flow friction losses in galvanized pipe are commonly determined by use of Air Friction Charts, Figure 1, which were constructed using the basic flow equation for pressure and loss in circular ducts. Values in the chart are based on standard air at a density of 0.075 lbs/ft^3 flowing through clean round galvanized ducts having 40 slip joints per 100 foot. Using the chart requires knowledge of the duct diameter, and either the volumetric flow rate or velocity of air in the duct. If the chart is unavailable, the following *Darcy-Weisbach Friction Coefficient* equation could be used to compute the pressure losses (Bleier, 1998):

(1) $\Delta P = K * (L/D) * Vp$

where: ΔP = Pressure loss (in H_2O)
K = 0.0195 (friction factor)
L = Length of pipe (ft)
D = Diameter of pipe (ft)
Vp = Velocity pressure (in H_2O).

The friction factor value of 0.0195 is for Reynolds numbers and Roughness values ranging from 80,000 to 2 million and 0.00075 to 0.00013, respectively. The range for the Roughness was determined by using an Absolute Roughness value (ϵ) for galvanized iron or steel air ducts of 0.0005 ft (Piggott, 1950, ACGIH, 1998) and dividing it by the range of duct sizes that are more commonly encountered in the cotton ginning industry (8 to 48 inches).

In the case of Dynamic losses, two methods can be employed: 1) the equivalent length of duct method or 2) the loss coefficient method. For the equivalent length method, the dynamic loss for the fitting (elbow, transition, branch, etc.) is replaced with a length of duct that has an equivalent loss. A friction chart (Figure 1) is then used to determine friction loss. Table 1 (Baker, 1994) shows an example of tables that can be used in obtaining equivalent length values.

The loss coefficient method uses a dynamic loss coefficient, C, that relates the pressure loss in the fitting to the velocity pressure at a given cross-sectional area of the fitting (AMCA, 1995). Equation 2 can be used to determine pressure losses in a fittings with known dynamic loss coefficients (Severns and Fellows, 1958, Bleier 1998, Fan Engineering, 1983):

(2) $\Delta P = C * Vp$

where: ΔP = Dynamic pressure loss (in H_2O)
C = Dynamic loss constant (dimensionless)
Vp = Velocity pressure (in H_2O)

The Dynamic loss constant is empirically determined for components such as elbows, expansions, and transitions encountered in air duct systems. Figure 2 (ACGIH, 1998) shows an example of dynamic loss constants for elbows of various ratios of centerline radius and diameter.

An example of the effects of friction loss can be seen by further utilizing data from the previous example. For instance, if the cotton gin mentioned above had 20 ft of 16-in. dia. pipe for the suction telescope instead of 19-in. dia. pipe and they maintained a flow rate 11,000 cfm, the velocity in the telescope would increase from 5,500 fpm to approximately 7,800 fpm. This would result in an increased pressure loss, due to friction, in the telescope. From Figure 1, 16-in. dia. pipe at 7,800 fpm yields a friction loss of 4 in. H_2O per 100 ft of pipe. Since we have only 20 ft of pipe instead of 100 ft, the friction loss is a fifth of the value read from the chart (0.8 in. H_2O). The friction loss for the 19-in. dia. pipe at 5,500 fpm would be 1.8 in. H2O per 100 ft of pipe times 20 ft of pipe or 0.36 in. H_2O, a 55% decrease from the 16-in. dia. pipe.

This report describes the process and results of modifications made to improve the capacity of the unloading system at the USDA-ARS cotton ginning laboratory in Lubbock, TX. The modifications were made in an attempt to increase the velocity in the suction line while reducing the energy consumption. Modifications included replacing pipes with larger, more correct sizes, installing smooth transition sections, replacing series fans with a single high efficiency design fan, replacing an undersized Y-valve, and sealing leaks. The results obtained are compared with theoretical friction loss calculations commonly found in the literature and used in the industry.

Objectives

To date, there have been few documented cases showing the results of practical applications involving air line modifications in cotton gins, although proper sizing of pipe to handle increased air volumes is widely recommended. Even though air line modifications occur throughout the industry on a yearly basis, documentation has been limited on the application of theoretical calculations to "real world" results. Therefore, the primary objective of this project was to improve the efficiency of the telescope suction system at the USDA-ARS cotton gin lab in Lubbock, TX while documenting effects the changes had on pressure and energy consumption and compare the findings to commonly used friction loss calculations.

Equipment and Procedures

The equipment used to collect the data consisted of four pressure gages, a standard pitot tube, tachometer, power factor meter, and a volt/amp meter. The four pressure gages used had upper limits of 5, 15, 25 and 50 inches of water. An 18 inch long standard pitot tube was used to measure the velocity pressure along with a commercial tachometer for fan rpm. Power factor, volts, and amps were measured using commercial power factor, volt, and amp meters.

Procedures involved drilling holes into the duct work at selected locations and taking measurements while all equipment normally used in unloading of seed cotton was in operation. Ambient temperatures, relative humidities, and barometric pressures were obtained from the local National Weather Service (Lubbock Airport, two miles south of the gin lab) for each day and time measurements were taken. Velocity pressures were taken using either an 8 point or 10 point traverse across the cross sectional area of the duct depending on the length of straight pipe located before and after the measurement location and the diameter of pipe. Static pressures were also measured at several additional locations by means of small holes drilled in the sides of the pipe. After recording the static and velocity pressures, the average air velocity was calculated using the following equation:

$$V = Kp * Cp * (\Delta P * (Ts/(Ps * Ms))^{1/2} \quad (3)$$

where: V = Velocity (fps)

Kp = Pitot tube constant (ft/sec[(lb/lbmole)(in Hg)/(°R)(in H_2O)]$^{1/2}$)

Cp = Pitot tube coefficient (dimensionless)

ΔP = Average velocity pressure of air stream (in H_2O)

Ts = Absolute duct temperature (°R)

Ps = Absolute duct pressure of air stream (in Hg)

Ms = Molecular weight of air, wet basis (lb/lbmole).

Calculations were corrected for pressure and temperature using a pitot tube coefficient of 0.99 and a pitot tube constant of 85.49. In conjunction with pressure readings, measurements of rpm, power factor, volts, and amps were performed on select days.

When calculating the dynamic losses for the expansions, a conservative approach was used for each expansion-type transition. These transitions were evaluated as abrupt enlargements rather than gradual expansions. This approach was used in an effort to compensate for those suspected, but not quantified, air leaks in the system that are not accounted for in the calculations. Dynamic loss constants for contractions and expansions were obtained from figures similar to figure 2 found in the literature previously referenced.

Pressure losses associated with the various equipment used in the unloading system (i.e. rock and green-boll trap (R&GBT), air line cleaner, separator, etc.) were obtained from actual measurements since losses in these devices can vary dramatically depending upon circumstances ranging from their operational settings, physical condition, etc.

Unloading System

The USDA-ARS South Plains Cotton Gin Lab has two unloading routes for seed cotton (Figure 3). Figure 3 illustrates the two unloading system routes before any changes occurred. The routes are referred to as the 1) normal suction leg and 2) belt dryer leg. Both routes start with the suction telescope unloading either a trailer or module and bringing seed cotton through a free-air valve and R&GBT. From the R&GBT, seed cotton is conveyed in the normal suction leg through three Y-valves, V1, V20 and V3, to an airline cleaner and then to the suction separator over the automatic feed control. The belt dryer leg is identical to the normal suction leg with the following exceptions: 1) from the R&GBT the seed cotton travels through Y- valves V1 and V2 to the separator over the belt dryer, and 2) from the belt dryer the seed cotton drops into the feed control thus bypassing the airline cleaner. The unloading system air from either separator is conveyed through Y-valve 27 (V27), the unloading fan, and to the Unloading System Cyclone. Table 2 shows the Y-valves in the unloading system and their respective functions. The primary valves critical to the unloading system regardless of the unloading leg taken are V1, V3, and V27. The other valves (V2 and V20) are used for research purposes and are not needed in the normal operation of the unloading system, but are included in the discussion due to leaks that can occur as a result of valves being in the system.

There have been numerous changes to ARS Lubbock gin lab since it was first constructed in 1968. Some of the changes have involved the unloading system. For the work detailed in this paper, the initial system refers to the system as it was in July of 1995.

Initial System Layout

The initial system layout was comprised of 13, 14, 16, and 20-inch diameter galvanized steel pipe. The length of duct used for each diameter pipe varied according to the unloading leg. The 13 and 20 inch diameter pipe had 20 and 40.5 feet of straight pipe regardless of the unloading leg. For the normal suction leg, the length of 14 and 16 in. dia. straight pipe used throughout the system was 10.6 and 85 feet, respectively. Whereas the belt dryer leg consisted of 14.2 and 133.6 feet of the 14 and 16-inch straight pipe, respectively. In addition to the straight pipe, there were several elbows, transitions, and Y-valves. The quantity, size, and degree of the other components in the initial normal unloading system are listed in Table 3. Table 3 only includes those components used in the “every day” operation of the unloading leg (i.e. the airline cleaner bypass is not listed). The components in the belt dryer leg are not shown since they are similar to the normal unloading system with only minor changes in dimensions.

The initial system used two No. 45 fans in series to move the air. The fans were powered by 60 and 75 Hp motors which operated at 1945 and 1975 rpm, drawing 54 and 75 amps, respectively. The initial static pressure readings taken on both unloading legs along with their corresponding velocity and volumetric flow rates are shown in Tables 4 and 5.

Modifications

The changes made to the unloading system occurred over a 13 month time period from July 1998 to August 1999. The changes are referenced in the order they occurred from Change I to Change VI. Locations of the various elements changed can be visualized by referring to figure 3.

The first modification, Change I, involved replacing the 14-inch diameter pipe transition segments between the free-air valve, the R&GBT, and V1, with section of rectangular duct. Other changes included replacing the 14-in. dia. pipe to V20 with 16-in. dia. pipe and replacing the 14-inch pipe 90° elbow after V1, in the belt dryer leg, with a 13-inch flat back elbow.

Change II involved changes to the normal suction leg only. Modifications consisted of replacing all 16-inch pipe between V27 and the fans with 20-inch pipe as well as replacing V27 with a larger Y-valve. The new Y- valve had 16 inch square openings for the incoming lines from the two separators and an 18-square-inch opening exiting into the 20-inch line. This resulted in enlarging the area in and out of V27 by 34% and 48%, respectively. The 16-inch pipe between the fans was not replaced in this modification. These changes resulted in elimination of eight 16-inch elbows and increased the inlet and outlet area in V27 from 169 square inches to 256 and 324 square inches, respectively.

The third modification involved replacing the dual No. 45 fans with one No. 60 fan. To power the new fan we used the 75 Hp motor that had previously been used to drive the second dual fan. Since the new fan was larger, we changed the belt drive ratio in order to operate this fan at the desired speed (1565 rpm). Since we were changing drives, we also decided to replace the existing V-belt drives with a cog-belt drive to eliminate slip. As a result of changing the fans, all pipe from V27 to the cyclones became 20 inches in diameter. A total of 4 elbows and 71 feet of straight pipe between V27 and the fan completed the new installation. The fourth elbow was installed at the fan entrance. Even though common practice states that either a straight run of pipe or a banjo should be used at a fan inlet, we installed a 60° elbow at the inlet to determine its overall effect on the system. In addition to replacing the fans and piping, the R&GBT was set to operate with a 1" pressure drop. This allowed proper operation while reducing excess air leakage.

After changing the fan(s), the next two modifications involved slight adjustments to the pipe entering the No. 60 fan. Change IV consisted of installing 6 feet of straight pipe between the 60°elbow and fan entrance. The next change, V, involved inserting straight pipe within half an inch of the fan wheel inside the fan inlet so as to force the air into the blast wheel.

The last change, Change VI, entailed removing the straight pipe insert in the fan inlet and sealing up all leaks in the system. Leaks were sealed in both unloading legs. The primary leaks sealed were the flanges on the rectangular ducts to and from the R&GBT, lids on the airline cleaner and suction separator, and V1 and V20 flange connections.

Results

For all intents and purposes, the belt dryer leg is the same as the normal unloading leg with respect to conveying the seed cotton to the respective separators. Other than a few differences in transitions, the primary difference between the two legs was in the use of the airline cleaner. In the belt dryer leg the seed cotton does not pass through the airline cleaner whereas in the normal suction leg it does. Therefore, the primary discussions of results will focus on the normal unloading system with similar measurements having been obtained for the belt dryer leg (Table 5).

Table 4 shows the static pressure readings, along with the calculated air velocity and volumetric flow rate values, taken after each modification to the system. Using the initial system data in Table 4, the friction losses, according to Figure 1, associated with the straight runs of 13, 14, 16, and 20-inch diameter pipe would be 0.48, 0.51, 3.06, and 0.57 , respectively. Friction loss for the same pipe as calculated by Equation 1 would be 0.48, 0.50, 3.26, and 0.63, respectively. As would be expected, the results using the two methods are almost identical. Differences between the values are more likely a result of error in reading Figure 1 rather than an error associated with the friction factor constant “K”.

To determine the dynamic losses associated with the system, the equivalent length and loss coefficient methods were used to calculate the friction losses. The first approach used the velocities at the various locations and Table 1 values in conjunction with Figure 1. For consistency, and due to the fact that velocity pressure readings were not taken at all possible locations along the piping, the velocity or velocity pressures (Vp) needed to calculate the dynamic losses were assumed to be constant for a given pipe diameter based on the first reading taken for that size pipe. For example, the velocity used in calculating the dynamic loss for all the 16-in. dia. pipe in the initial normal suction system was calculated using the Table 4 value of 7130 fpm. The second approach used Equation 2 along with the pressure loss coefficients, C, in Figure 2.

The measured pressure drop across the system, from the fan inlet to the suction telescope, before any changes were made was 25 in. H2O. Using equations 1 and 2, the calculated losses in in. H2O, using the conservative approach for the transitions, were 3.19 (for fittings and transitions), 7.23 (elbows), 4.87 (pipe), and 9.65 for the equipment. Where the equipment consisted of the pivot for the suction telescope, free air valve, R&GBT, airline cleaner, and separator. The total loss calculated was 24.94 in. H2O. Thus yielding a calculated estimate that was within 99.7% of the losses actually measured. When the losses were calculated using figure 1 in conjunction with table 1, a calculated loss of 22.59 in. H2O was obtained. This value was 9.6% less than the actual loss measured. The primary difference between the calculated values was due to the sum of the elbow losses. The sum of Table 1 losses was calculated at 4.87 in. H2O compared to 7.23 obtained from equation 2, a difference of 32%.

Pressure loss across the system after all modifications and changes had occurred was 13.75 in. H2O. Using the conservative estimate for transitions, equation 2 resulted in calculated losses in in. H2O of 2.49 (fittings and transitions), 3.52 (elbows), 3.24 (pipe), and 4.52 (equipment) for a total calculated loss of 13.77. Figure 1 and table 1 calculations resulted in a total loss of 13.2 in. H2O. After modifications, the conservative equation 2 calculation results were less than 1% higher than the actual losses. Whereas the figure 1/table 1 results yielded values that were 4% under actual losses.

Using the literature recommendations for gradual expansions, after modifications, produced a calculated pressure loss of 12.9 in. H2O which is within 93.5% of the actual loss which was opposite of what was expected. The initial assumption was that the variation between the conservative approach and the literature recommendation for gradual expansions, before and after modifications, would be attributed to the leaks in the system. The thought was, before any changes were made, system leaks such as pipe not being joined properly, worn out gores in the elbows, improperly operating valves, etc., would result in leaks not accounted for in the calculations. Thus, the conservative approach would produce a closer estimation of pressure loss. Conversely, it was believed that after modifications, the leaks would be repaired or at least minimized to the point that the “recommended” equations would yield the closest estimation to the measured system pressure loss. However, the results indicated otherwise. The difference is a combination of the non-quantified leaks and the assumptions used in the equations.

The equations and the coefficients used are based on straight runs smooth runs before the elbows or transitions. Deviations from the conditions for which the coefficients were established will change the factors thus changing the pressure loss. For example, using the equivalent length and loss coefficient calculation methods, the initial system losses for the belt dryer leg were 15.4% and 3.9 % below actual measurements, respectively. Whereas the final system loss calculations, without using the conservative approach for the transitions, indicated an overestimation of only 1.5% by the equivalent length method and 13.5 % by the friction loss coefficient equations. It should be noted that fewer measurements were taken in the belt drier leg than in the normal suction leg. This only compounded the error resulting from the calculations since one measurement was used for long runs of pipes and elbows in the belt drier leg while two to three measurements were used for the normal leg.

On an individual component basis, when comparing the actual pressure drop across the 16 inch 90° elbows (3 in. H2O) with those calculated, equation 2 yielded the closest results (83 % of actual) versus the method used in the Ginners’ Handbook which was only 58 % of actual. However, when comparing the loss for the three 20-in. dia. elbows with the calculated values, the methodology outlined in the Ginners’ Handbook resulted in a 26% overage whereas Equation 2 yielded a 61% overage. These results indicate that any specific method will be limited by assumption used in the calculations, measurement locations, and the presence of flow disturbances before or after the elbows. The variations illustrated for the elbows can be attributed to all the factors listed. For example, the dynamic loss coefficients are measured with straight runs of pipe before the elbows. However, in our system there were at least three instances where an elbow was attached to another elbow which would have resulted in a change not accounted for in figure 2.

Energy Savings

Table 6 shows the power usage and fan operating data for the unloading system before modifications, after Change II, and after the final modifications were made to the system. The data emphases the benefit of eliminating restrictions, excess elbows, and leaks. Initially the system used 88.8 kW per hour to operate the two No. 45 fans in series. The mechanical efficiencies for the fans were below 50%. After Change II, the power requirements increased to 97.3 kW per hour due to less restrictions in the line resulting in moving more air. The mechanical efficiencies improved slightly as a result of the modification. After the final modification, the reduction in system static pressure allowed the suction system to be operated by one 75 Hp motor instead of the two motors that were used initially (60 Hp and 75 Hp). The power required was decreased to 55.8 kW per hour without compromising the volume of air necessary to move the seed cotton. The mechanical efficiency was improved approximately 5 % percentage points from the initial efficiency.

Throughout the process of improving the unloading system, the items that had the largest effect on pressure drop were: 1) V27 not properly closing thus resulting in a 3 to 10 inch pressure drop across the valve depending upon the suction leg being used, 2) the original suction line was plumbed in such a way that it "dodged" the other pipes in the basement causing an additional eight elbows to be used in the line between V27 and the fan, 3) many of the elbows in the suction line were worn to point of leaking at the gores, and 4) the R&GBT and the lids on the airline cleaner were not set or sealed as they needed to be to optimize their use.

As a result of eliminating restrictions, excess elbows, and leaks we were able to increase the velocity in the telescope from 4950 fpm initially to a final of 6166 fpm using 33 less kW per hour. Assuming a cost of seven cents per kilowatt, this improvement in system performance would result in an energy savings of $2.31 per hour while improving the efficiency of the motor.

Summary

Restrictions, leaks, and excess elbows or fittings can greatly effect the performance of any pneumatic conveying system. Even though this is commonly understood and acknowledged throughout the cotton ginning industry, there is limited amount of documentation illustrating the effect that leaks, restrictions, and excess or improperly sized fittings can have on system performance compared to values obtained using standard calculations. In this paper, we illustrated how standard pressure loss calculations compared with actual pressure losses encountered while making modifications to the unloading system at the USDA-ARS Lubbock, TX ginning laboratory. Since the unloading system in the gin lab can either route the seed cotton through the "normal" suction leg or the belt dryer leg, both of these legs were measured during the modifications. The overall measured static pressure drop from the suction to the fan for both suction legs before and after changes in inches of H2O were 25.0 (normal suction leg), 25.3 (belt dryer leg), 13.8 (normal suction leg), and 12.5 (belt dryer leg), respectively. This represents a system pressure reduction of 11.3 in H2O for the normal suction leg and a 12.8 in H2O for the belt dryer leg. Using standard equations to calculated the pressure loss, the equations generated pressure losses that were within 0.4 to 19 % of actual losses. Overall, the equations produced values that would allow someone evaluating a system to estimate losses very accurately. It should be noted that variations between the calculated values and those actually measured can be attributed to a number of factors including: 1) accuracy of the instrumentation used to obtain the measurements, 2) variation in the layout of the duct work from that assumed in the literature (i.e. elbows connected to elbows, etc.), 3) seed cotton cleaning machinery not properly operating, 4) leaks in elbows, worn pipe, and valves, and 5) fan performance and wear. All factors considered, pressure losses in a system can be calculated reliably. However, when optimizing a pneumatic conveying system in a cotton gin, experience and knowledge of the seed cotton cleaning equipment are vital in obtaining the most effective system performance.

The effects of leaks and system inefficiencies can be very costly. A simple routine evaluation of any pneumatic system in a gin would be prudent. Potential problem areas include: 1) undersized fittings and pipe (most commonly brought about by increasing the fan speed and air flow without being aware of how it affects the rest of the system), 2) worn and malfunctioning valves, 3) plumbing (have the runs as straight as possible), and 3) leaks in elbows, worn pipe, and seed cotton processing equipment. Optimizing any materials conveying system can only result in a more cost effective operation. Conventional engineering techniques commonly used to calculate static pressure losses, along with experience and knowledge of the system, are effective means of accurately estimating system losses and performance.

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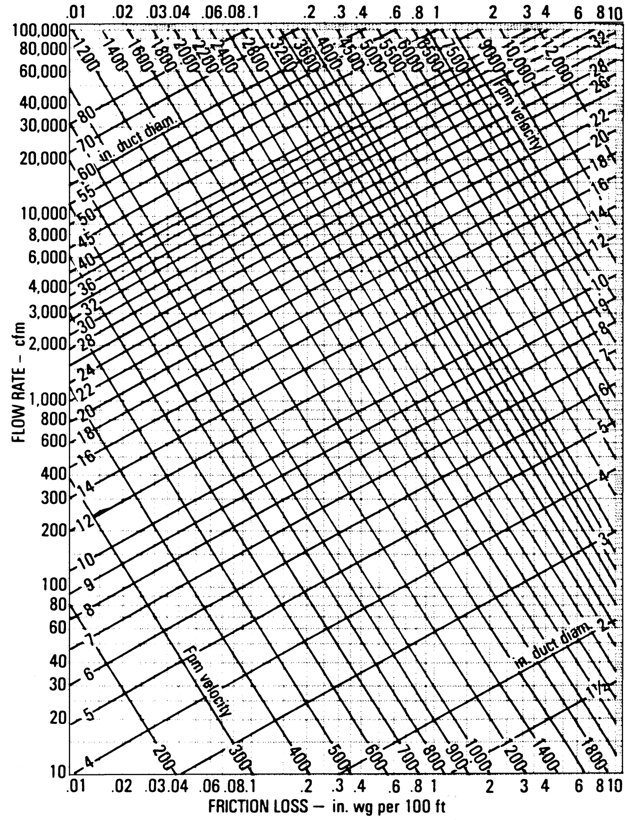
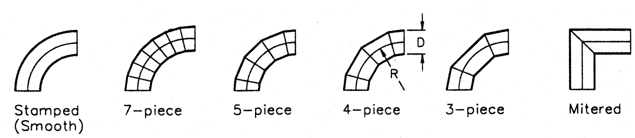


Figure 1. Duct friction chart for round pipe in inches per 100 feet.



	R/D					
	0.5	0.75	1.00	1.50	2.00	2.50
Stamped	0.71	0.33	0.22	0.15	0.13	0.12
7–piece	–	–	–	0.23	0.17	0.15
5–piece	–	0.46	0.33	0.24	0.19	0.17*
4–piece	–	0.50	0.37	0.27	0.24	0.23*
3–piece	0.90	0.54	0.42	0.34	0.33	0.33*

* extrapolated from published data

OTHER ELBOW LOSS COEFFICIENTS

Mitered, no vanes	1.2
Mitered, turning vanes	0.6
Flatback (R/D = 2.5)	0.05

Figure 2. Elbow loss coefficients for 90 degree round pipe.

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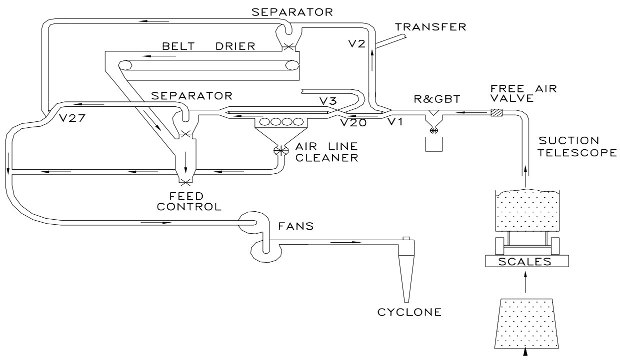

Figure 3. Schematic of the two routes of seed cotton unloading system for the USDA-ARS Lubbock, TX ginning laboratory.

Table 1. Straight pipe equivalent for 90° elbows of various throat radiuses.*

Throat radius of elbow in pipe diameters.	1/2	3/4	1	2	3
Length of straight pipe offering equivalent resistance in diameters.	17	14	12	8.5	6.5

* For elbows less than 90°, the equivalent straight pipe resistance is proportional to the bend (i.e. 30° elbow will be one-third the value shown).

Table 2. Unloading system Y-valves and their function in the USDA Lubbock, TX gin lab.

Y- valves	Unloading Leg	Function
V1	Both	Allows selection of either the normal suction or belt dryer leg.
V2	Belt dryer	Selects whether seed cotton goes to belt dryer separator or into the transfer system for moving seed cotton from one trailer to another.
V3	Normal	Allows the airline cleaner to be bypassed.
V20	Normal	This valve allows small lots of seed cotton in the gin to be picked up and introduced into the ginning system without going out under the suction shed or through the R&GBT.
V27	Both	Works in conjunction with V1. Allows the pull of the suction fan to be routed to the separator of either unloading leg.

Table 3. Unloading system components quantity, size, and degree.

Normal Suction Leg				
Component	**Quantity**	**Size**	**Degree**	**Throat Radius in Diameters**
Free-air valve	1	16" x 14"	--	--
R&GBT	1	24" x 10"	--	--
Y-valves (V1, V3, V20 and V27)	4	13" x 13" openings	--	--
Transition (into ALC)	1	13" x 13" to 10" x 22"	--	--
Transition (from ALC)	1	10" x 50" to 16" pipe	--	--
Transition (into Separator)	1	16" pipe to 6" x 71"	--	--
Reducer from Separator blow box	1	20" round to 16" pipe	--	--
Transitions (to and from Y-valves)	3	13" x 13" to 14" pipe	--	--
Transitions (to and from Y-valves)	2	13" x 13" to 16" pipe	--	--
Increasors (to first fan)	1	16" pipe to 19" round	--	--
Reducer (from fan 1 to fan 2)	1	20" pipe to 19" round	--	--
Transitions (from fans)	2	15" x 18" to 20" pipe	--	--
Transitions (to cyclones)	1	20" pipe to 45" x 11.25"	--	--
Elbows	2	14"	90	3/4
Elbows	4	16"	90	1
Elbows	1	16"	45	1
Elbows	1	16"	60	1
Elbows	4	16"	30	1
Elbows	3	16"	15	1
Elbows	3	20"	90	1

Table 4. Pressure drops, pipe velocities and flow rates for the normal suction leg before and after changes.

Location	Pipe Diameter (in.)	Sp (in. H_2O)	Velocity (fpm)	Flow rate (CFM)	Air Leakage (% of final flow)
Initial System Measurements and Calculations					
4' after open end of suction	13	-3.5	4950	4562	57.3
1' before valve 20	14	-13.25	7318	7845	26.6
1' before air line cleaner	Transition	-13.75			
1' after air line cleaner	16	-15.5			
Before transition into #2 separator	16	-16.5			
1' after #2 separator	16	-20.25			
1' above valve 27	16	-21.5	7130	9989	6.5
At fan inlet transition	NA	-28.5			
8' after 2nd fan	20	5.75	4897	10683	--
After Change I					
1' above valve 27	16	-21.75	7325	10228	6.2
1' below valve 27	16	-26			
At fan inlet transition	Transition	-27.5			
8' after 2nd fan	20	5.5	4999	10906	--
After Change II					
4' after open end of suction	13	-4	6685	6224	48.1
1' before valve 20	16	-12.5	6279	8814	26.5
1' above valve 27	16	-23.5	7612	10629	11.4
At fan inlet transition	Transition	-25.5			
8' after 2nd fan	20	7.5	5501	12000	--
After Change III					
4' after open end of suction	13	-3	5025	4632	56.8
1' before valve 20	16	-8.5	5286	7380	31.2
1' above valve 27	16	-16.5	6473	9038	15.7
At fan inlet transition	Transition	-18			
8' after fan	20	6.5	4915	10722	--
After Change IV					
4' after open end of suction	13	-3.5	5754	5304	50.4
1' before valve 20	16	-7.5	5685	7937	25.8
1' above valve 27	16	-16.5	6691	9342	12.6
At fan inlet	20	-17.5			
8' after fan	20	5.8	4901	10691	--
After Change V					
4' after open end of suction	13	-3.5	5671	5227	50.1
1' before valve 20	16	-8	5480	7651	26.9
1' above valve 27	16	-17	6589	9201	12.2
At fan inlet	20	-17.25			
8' after fan	20	5.5	4803	10478	--
After Change VI					
4' after open end of suction	13	-3.5	6166	5683	45.9
1' before valve 20	16	-8.5	5507	7690	26.8
1' before air line cleaner	Transition	-11			
1' after air line cleaner	16	-12			
Before transition into #2 seperator	16	-12.5			
1' after #2 seperator	16	-15			
1' above valve 27	16	-16	6594	9207	12.4
At fan inlet	20	-17.25			
8' after fan	20	5.5	4817	10510	--

Table 5. Pressure drops, pipe velocities and flow rates for the belt dryer leg before and after changes.

Location	Pipe Diameter (in.)	Sp (in. H20)	Velocity (fpm)	Flow rate (CFM)	Air Leakage (% of final flow)
Initial Measurements and Calculations					
4' after open end of suction	13	-3.2			
3' before belt dryer separator	14	-9.5	6083	6502	42.5
3' after belt dryer separator	16	-11.5	6523	9109	19.5
1' above valve 27 (belt dryer leg)	16	-15.5	6114	8537	24.5
At fan inlet transition	Transition	-28.5			
8' after 2nd fan	20	6	5185	11312	--
Final Measurements and Calculations					
4' after open end of suction	13	-3	5422	4998	55.3
3' after belt dryer separator	14	-12.5	6812	9511	14.9
1' above valve 27 (belt dryer leg)	16	-14.5	6961	9720	13.1
At fan inlet transition	Transition	-15.75			
8' after fan	20	7	5125	11182	--

Table 6. Power and fan data before and after modifications to the unloading system.

	60 HP motor	75 HP motor
Before Any Changes - Initial		
Volt Readings	464	464
Amp Readings	54.00	74.67
Power Factor Measurements	0.94	0.80
Fan RPM	1945	1975
Fan Total Pressure	14.50	17.50
Air Horsepower	24.27	29.30
Kilowatts	40.79	48.01
Brake Horsepower	54.66	64.33
Mechanical Efficiency	44.41	45.54
After Change II		
Volt Readings	464	464
Amp Readings	60.50	80.23
Power Factor Measurements	0.94	0.80
Fan RPM	1945	1975
Fan Total Pressure	14.50	17.50
Air Horsepower	27.37	33.03
Kilowatts	45.70	51.58
Brake Horsepower	61.24	69.12
Mechanical Efficiency	44.69	47.79
After all Changes - Final		
Volt Readings	—	474
Amp Readings	—	82.00
Power Factor Measurements	—	0.83
Fan RPM	—	1565
Fan Total Pressure	—	22.75
Air Horsepower	—	37.61
Kilowatts	—	55.88
Brake Horsepower	—	74.87
Mechanical Efficiency	—	50.23