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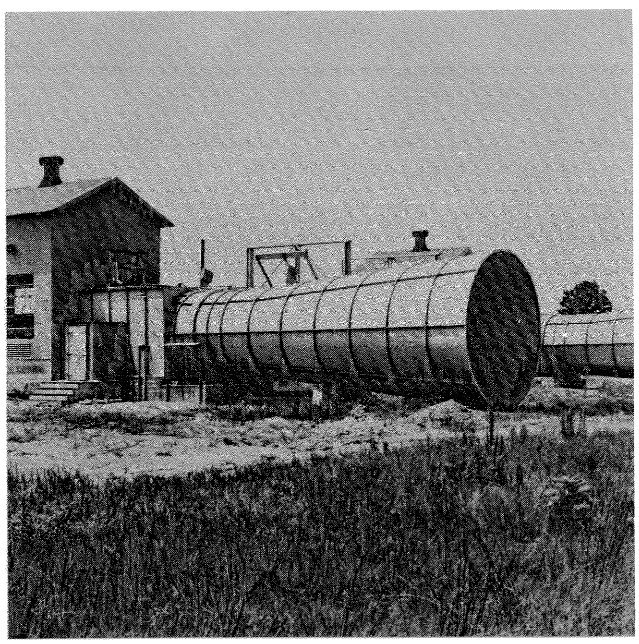
Aspects of Coal Mine Ventilation

By Raymond Mancha



Foreword

Mr. Raymond Mancha, formerly Vice-President of Ventilation with Joy Manufacturing Company, was respected as a leading authority on mine ventilation. Over the years, he published numerous articles on ventilation which were collected and published under one cover by Joy Manufacturing Company.

Authorization for reprinting of certain articles was given by Mechanization, Inc., and the American Institute of Mining and Metallurgical Engineers. These include:

Section VII of the 1945 issue of Mechannual

Transactions, Coal Division, A.I.M.E.
- Volume 149 (1942), Pages 178 and 183
- Volume 157 (1944), Page 108
- Volume 168 (1946), Page 106
- Volume 169 (1946), Page 119

This brochure proved to be a valuable tool to those interested in mine ventilation. Even though some of the cost evaluations used to illustrate points are out of date, the validity of the concepts remains. We continued to receive many requests for these articles which prompted this reprinting exactly as Ray Mancha wrote them. We hope others will find them of considerable value in ventilation planning.

Sincerely,

William D. Meakin
Senior Product Manager
Underground Fans

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Measurements of Air Volume. The time is propitious for exercise of greater care in measurement of underground air volumes at various stations along the air circuit. These readings are customarily obtained by means of an uncalibrated rotating-vane anemometer held in the hand of the observer for one minute at the center of the air course. Sometimes in an attempt at improved precision the instrument operator will try to traverse the entire section during the minute interval by moving the anemometer back and forth and up and down over the plane of the measuring station but some observers will actually seek out the point at which the anemometer rotates fastest and will use that point as a measure of the average velocity passing the station.

The likelihood of obtaining air volume measurements which are even approximately accurate by these methods is remote. It would appear desirable to adopt a standard method of air measurement which would insure a high degree of accuracy and which would require no more time, provided that such a method is available. The following is a discussion of a simple procedure which can be easily followed by any mining company employee who is at all familiar with air volume measurements.

Having selected the station for air volume measurement and having accurately established the true cross-sectional area at the station, the instrument operator should stand in the center of the air course at a distance slightly less than arms' length down stream from the measuring plane. A calibrated anemometer should be attached to the end of a stick or rod of sufficient length to permit the observer to reach both ribs while standing in one place. This stick or rod can be made in short sections which can be easily carried underground. During the traverse the instrument should be held in each of twelve positions for a period of five seconds per position. These twelve instrument positions should correspond to the centers of twelve rectangles which would be obtained by stretching two strings horizontally across the air course spaced between top and bottom at the one quarter and the three quarter points, and three strings stretched vertically from top to bottom spaced at the one quarter, one half, and three quarter points between ribs. Actually, it is not necessary to install the strings, since with a little practice one can judge the approximate location for each anemometer position with sufficient accuracy by eye.

At the start of the traverse the operator should stand in a position from which he can reach the anemometer with his hand and thereby start the registration of the instrument. Immediately following the start of registration the observer should take his position in the center of the air course down stream from the measuring plane and permit the anemometer to run in the center of each rectangle for five seconds as timed by a stop watch held in the hand. The instrument should be permitted to register continuously throughout the traverse without interruption. When the time of operation in the last rectangle approaches five seconds the operator should move sufficiently close to the instrument that he can reach over and turn it off at the prescribed time. In this manner the influence of the observer's body upon the air flow past the measuring station remains unchanged through practically the entire time of measurement so that a fairly accurate average velocity is obtained. It is important that the instrument correction be applied as indicated by the calibration chart.

By following this recommended procedure the customary inaccuracies are either eliminated or reduced to a minimum. Such inaccuracies include errors resulting from:

1. A 1-position reading;
2. Improperly timed traverse rate;
3. Holding the instrument close to the body throughout the traverse in higher than average air velocities;
4. Changing the flow pattern past the measuring plane during the time of traverse by changing the location of the observer's body;
5. Neglecting the anemometer calibration.

When measuring air-flow at a section that is obstructed by one or more roof props located away from the rib, the question is often raised as to the advisability of deducting for the projected area of the obstructions. A simple rule to follow is that no deduction should be made unless the width of the obstruction is equal to, or greater than the horizonal distance between anemometer positions.

Air Pressure Surveys; Purpose. The purpose of an accurate underground survey of air pressure at a mine is to obtain a pressure gradient along the circuit being investigated which will show the pressure drop between various observation points along the route. This makes it possible to locate regions of excessive resistance and to determine the economic feasibility of correcting such conditions by cleaning existing airways or driving additional passages.

A pressure gradient alone is useful for predicting the effect upon the remainder of the circuit of increasing or decreasing air regulation on any split or of applying a booster fan on a high resistance split. When contemplating sinking a field airshaft or driving a drift to the outcrop, a pressure gradient is required to predict results. Often a single-compartment field airshaft gives disappointing results in reducing water gage because it relieves a portion of the circuit that is a minor contributor to the overall mine pressure. It is therefore of value to know how the circuit pressure drop is distributed, which is best accomplished with a pressure gradient.

The underground pressure survey can be used to determine friction coefficient for various types of airway. This requires high precision work to establish useful data for prediction of mine pressure-volume characteristics for use in projecting new mine developments.

Air Pressure Surveys; Methods. Regardless of the type of instrument used to conduct a circuit pressure survey, the work should preferably be performed at a time when the mine is idle to avoid error introduced by moving trips, cages and opening and closing doors, etc. It is also preferable to select a period when the barometer is comparatively steady to restrict variation in the specific weight of the air at each observation station during the time of the circuit traverse.

The two most commonly used instruments for underground pressure surveys are the altimeter and the inclined manometer with hose extensions. The altimeter is sufficiently accurate for establishment of an underground pressure gradient and it is preferable to the manometer for such work because it has greater flexibility and a shorter time required for a traverse. With the altimeter it is possible to select more widely separated observation stations than is practical with the manometer due to restricted practical lengths of hose extensions. Also, use of the altimeter makes it possible to travel along the haulage road between airway observation stations which is easier than traveling a badly obstructed airway.

On the other hand, the inclined manometer is preferable to the altimeter for establishing friction coefficients for mine airways. The manometer readings are unaffected by differences in elevations between stations which introduce a correction that must be applied to altimeter data. Also, the manometer reading needs no correction for temperatures which may or may not be the case with the altimeter, depending upon the make of the instrument used. Elimination of correction factors combined with instantaneous, simultaneous and direct reading of the air-flow pressure drop between two successive stations gives a higher degree of accuracy with the inclined manometer than is possible with the altimeter. This is highly desirable for friction coefficient determination.

Altimeters are at best, imperfect instruments whose accuracy depends on mechanical fidelity or continued adherence to the calibration scale. The manometer, on the other hand, indicates pressure directly in terms of the height water column that the pressure will support. Altimeters are subject to some lag or creep during a traverse. This means that an instrument will not return exactly to the original reading when returned to the starting station even if the pressure at the starting station remains constant throughout the traverse. The degree of this discrepancy depends upon the quality of the instrument used but it can be kept within acceptable working tolerances by careful selection. Altimeters are available that are supposedly compensated for effects of temperature on the accuracy of the instrument. However, an instrument furnished with a temperature correction chart is likely to give more accurate results than a supposedly correct instrument without a temperature calibration chart.

Appendix A presents a detailed explanation of the proper use of altimeters and the inclined manometer for making underground pressure surveys.

Air Velocity Surveys. At any particular mine there is an approximate value for the maximum allowable underground air velocities. This depends upon various local conditions such as power consumption, roof conditions and air volume requirements. Air courses can be easily policed with a deflecting vane anemometer known as the velometer, an instrument that indicates air velocity instantaneously. By its use, regions of unusually high air velocity can be quickly determined and the seriousness of excessively high velocities can be evaluated by using an inclined manometer equipped with rubber tubing, before correcting the condition by cleaning the air course. An underground velocity survey with the velometer is a quick and fairly reliable means of ferreting out trouble, which should be generally used.

Apportioning Airways. When projecting new development at an existing mine, or development of a proposed mine, the question arises as to the most economical number of intakes and returns which should be driven. Obviously, an insufficient number of air courses is costly because of excessive power requirements for ventilation. On the other hand, an excessive number of air courses becomes costly due to the difference in profit derived from development coal on the one hand and room coal on the other. A formula for the most economical number of air courses is

$$N = 9.25\sqrt[3]{\frac{KQ^3OC}{EA^4wta}}$$

In which K is the friction coefficient suitable to the local conditions (such as $150x10^{-10}$); Q is the volume of air handled by all entries, in cubic feet per minute; O is the perimeter of each air course, in feet; C is the cost of electrical energy, in cents per kilowatt hour; E is overall unit efficiency of ventilating equipment from electricity to air, in percent; A is cross-sectional area per entry, in square feet; w is the specific weight of coal, in pounds per cubic foot; t is the difference in profit per ton of entry coal and room coal, in dollars; and a is the amortization rate including depreciation and interest on investment, in dollars annually per dollar invested.

This formula is obtained by setting up an expression for the cost per foot of entry system in terms of the cost of power annually per foot of entry system, plus the annual cost of carrying charges per foot of entry system. These expressions involve the terms just defined. The first derivative of this cost taken with respect to the number of entries and equated to zero, permits solution for the optimum number of entries as set forth.

The formula can be a useful guide as illustrated by the following example:

Suppose it is desired to drive entries 12 feet wide in coal four feet thick to handle 100,000 cubic feet of air per minute. The cost of power is one cent per kwh and the overall unit efficiency of the ventilating equipment is 60 percent. Assume further that the sum of interest on investment plus depreciation is 12 cents annually per dollar invested, and that room coal nets a profit of \$1.00 more per ton than does development coal. The nature of the coal and roof condition is such that an average rubbing coefficient of $70x10^{-10}$ is applicable. The formula gives 3.95 air courses as the most economical number under the prevailing conditions. Therefore, four air courses would be driven. This example serves to illustrate the usefulness of the formula in projecting mine development.

Effect of Leaky Stoppings. One effect of leaky stoppings is to increase the pressure and power requirements for ventilating a particular area with a specified amount of air. It can be shown theoretically and proved experimentally that the pressure increase that results from presence of leaky stoppings is approximately directly proportional to the total leakage air thru the stoppings expressed as a percent of the air volume delivered to the area to be ventilated.

For example, assume a condition in which it is necessary to deliver 100,000 cubic feet of air to the inby end of an entry along which the stopping leakage amounts to 25,000 cfm. A volume of 125,000 cfm must enter the entry. From the above stated relationship, the pressure across the entry will be 25 percent higher than it would be when delivering 100,000 cfm at the far end of the entry without stopping leakage. Since the pressure across the entry and the volume delivered to the entry are increased 25 percent, it follows that the power for ventilating the section in question will be 56 percent ($1.25x1.25 = 1.5625$) higher than without leakage. This illustrates the desirability of maintaining stopping leakage at as low a level as is economically justifiable.

The theoretical analysis of the above mentioned relationship is discussed in Appendix B.

Splitting the Air. The most effective and economical splitting of underground air circuits is accomplished by splitting the intake as close as possible to the point at which the air enters the mine and by reuniting the return air currents as close as possible to the point at which the air leaves the mine. This type of splitting permits air control on an individual split, within the limits of its regulator or booster, without affecting the remaining splits beyond the degree to which the pressure created by the fan is reduced or increased by the change in total fan volume. Consequently, the long splits permit greater power savings with installation of boosters and elimination of regulators than are possible when the splits start and terminate far within the mine.

In the case of short splits which start and end well inside the mine, the control of flow thru an individual split will influence the ventilation of the remaining splits due to the increase or decrease in pressure losses on the intakes and returns outby all the splits, which is occasioned by changes in total mine volume. A system of short splitting is therefore more sensitive to adjustment of flow in individual split than is a system of long splits. In addition the short split system offers less possibility of power savings in replacing regulators with boosters.

Location of Split Regulators and Booster Fans. The purpose of regulation of a split is two-fold. In the name of economy it is desirable to introduce artifical resistance on a split in the form of a regulator to prevent passage of an excessive air current on the split when it is subjected to the pressure differential required to ventilate the free split. In the name of safety, regulation is desirable on a split so that the current can be varied within limits to control the contamination resulting from varying rates of gas emission. Location of the regulators on the split should be such as to maintain the lowest possible pressure differentials across the stoppings along the split. Consequently, the regulators should be completely outby on the split, either on the intake or the return, depending upon convenience and system of mining. In case of an exhaust ventilation system in which the haulage road is on intake air it is usually desirable to locate the split regulators completely outby on the return of the split, whereas, in the case of a pressure system the reverse is true. Occasionally location of the regulator will be influenced by the fact that the split parallels the main intakes or returns being separated from one or the other by doors or stoppings. In such a case the split regulator should be located to maintain the minimum pressure difference between the inby portion and the main air course.

It is sometimes desirable to locate a booster fan on one or more splits. For example, instead of placing regulators upon the splits easy to ventilate, booster fans driven by variable speed motors can be installed on splits that are hard to ventilate. Flexibility of this system with respect to control of gas contamination equals that with use of regulators. On the other hand it is sometimes desirable to place a booster upon the free split only, especially when its resistance is such that the pressure required for its ventilation greatly exceeds the pressure required to ventilate the next easier split. Benefits of split boosters are manifold. Among them are reduced power requirements resulting from imposing the free split pressure upon the free split volume only, instead of upon the total mine air. Also, the booster fan lowers the pressure differential across stoppings outby the booster, thereby reducing the stopping leakage and the total volume of air which must be handled by the primary mine fan.

When installing a booster fan on a secondary split there are two precautions which must be rigidly observed. First the booster fan must be operated so that the pressure immediately outby the booster is greater on the intake air course than on the adjacent return air course to eliminate possibility of recirculation. Secondly, in operating a booster fan if the speed of the surface fan is to be lowered, precaution should be taken against robbing splits outby the booster. This can be done by properly adjusting the speed of the surface fan, or by opening regulators on outby splits.

In the simple case of two inside splits ventilated with a regulator on the low resistance split, the proper location of the booster fan on the free split is at a point at which the pressure between the intake and return on the free split is equal to the pressure drop across the regulator in the restricted split. The surface fan should then be operated at a speed which will pass the original volume at a pressure lower than the original pressure by an amount equal to the pressure across the regulator, which will be removed. When thus operating the actual volume will be slightly greater than the former volume on both splits due to benefits derived from reduced stopping leakages outby the booster.

Effectiveness of a Field Airshaft. Location of a field airshaft is usually based on location of an already developed mine section which is hard to ventilate, or by a proposed development area too remotely situated for ventilation with the existing system. Frequently, the field airshaft is sunk with the idea of permitting it to serve as an additional intake or exhaust opening, depending upon whether the mine is on an exhaust or pressure system, respectively. When used in this manner the field airshaft relieves the intake arteries in the case of an exhaust system, or the return arteries in the case of a pressure system. Since haulage roads must be kept open and cleaned, it follows that the haulage road is usually equivalent to at least two air courses of doubtful maintenance. As a result it is customary to find that at least 75 percent of the total mine pressure is attributable to the air courses. A field airshaft applied in the manner previously described fails to effect the relief anticipated or desired, since it relieves a portion of the circuit which is responsible for but a small part of the resistance.

A single-compartment field airshaft can be most effectively used when it is equipped with a fan. When so used the field airshaft gives relief where most needed and it will usually effect a reduction in total mine pressure of approximately 50 percent while maintaining equivalent mine ventilation.

The most effective type of field airshaft as far as reduction of mine pressure is concerned, is a 2-compartment shaft used for both intake and return. A shaft of this kind usually relieves intakes and returns and it may reduce the total mine pressure as much as 75 or 80 percent and yet maintain equivalent mine ventilation. Great care, however, should be exercised in selection and construction of the curtain wall, otherwise it will serve as a constant source of leakage and waste. Two separate airshafts are preferable to a large single shaft with a curtain wall because of lower cost, freedom from air recirculation and improbability of complete air stoppage from cave in. This conclusion is in accord with recommendations of state and federal mining departments.

Size of Airshafts. The size of an airshaft of any desired shape for a particular air volume should be selected so that the sum of annual power cost and amortization charges is minimal. The optimum size is primarily influenced by presence or absence of shaft lining and secondarily by the cross-sectional shape of the shaft. It is risky to use arbitrary limits on the air speed as a yard stick for determining airshaft size. Regardless of shape, unlined shafts should be larger than lined or partly lined airshafts, but they are cheaper in spite of the larger size. Regardless of the amount of lining, circular air shafts are cheaper than elliptical shafts which in turn are cheaper than rectangular shafts. For a detailed analysis of the problem of optimum size of airshafts, reference is made to Appendix C.

Flow on Fan Stoppage. When the fan is stopped at a mine equipped with a single fan the direction of flow of air thru the fan reverses and the flow soon ceases. The reversal is more pronounced at mines ventilated at high pressure and at mines from which a large tonnage has been extracted and in which old workings have been left standing open.

The air reverses motion thru the fan when it is stopped because the pressure differential between the mine and the atmosphere sets up a flow of air into the mine thru an exhausting fan or out of the mine thru a blowing fan which continues until the mine air reaches atmospheric pressure. A quantitative analysis of a closely related phenomenon can be made by assuming that the mine intake is completely sealed in case of an exhausting fan or that the mine outlet is sealed in case of a blowing fan and that the fan is placed in operation at a speed which will maintain a given depression or pressure when the air has stopped flowing from or into the mine. The volume of air which the fan must handle to create the depression or pressure throughout the mine is proportional to the product of the stipulated difference between atmospheric and mine pressures and the volume of the mine opening, the latter being closly related to the tonnage of coal that has been removed from the mine. The relation may be formulated:

$$\mathrm{V} = \frac{\mathrm{kit}}{\mathrm{a}}$$

where V is the volume of air, in cubic feet, to be passed by the fan to build up the specified pressure differential i, in inches of water, below or above atmospheric; a is atmospheric pressure in inches of water, t is the total tonnage which has been removed from the mine and k is the cubic feet per ton of coal on the solid. For further simplification, assume a as 400 inches water gauge and k as 25 cubic feet per ton. The equation then becomes

$$\mathrm{V} = \frac{25\,\mathrm{it}}{400} = 0.06125\,\mathrm{it}$$

which may be rounded to 0.06 *it*. Thus the volume to be moved by the fan under the postulated circumstances is six percent of the product of the water gage and the tonnage. For a 5-in water gage this means that the air volume to be moved is nearly a third of the tonnage removed so that a fan at a mine from which three million tons of coal had been extracted would have to pass nearly a million cubic feet of air to build up a 5-in water gage throughout the mine with the other end of the mine sealed.

Air Conditioning. The term "air conditioning" as applied to the ventilation of coal mines usually suggests the practice of precooling intake air in hot weather to reduce roof and rib deterioration during the early stages of air passage thru the mine. In some extreme conditions it is desired to increase the moisture content of air entering the mine during the winter to reduce creation of dust hazards by absorption of moisture within the mine. Also it is occasionally desired to preheat the air in cold weather so that freezing in the intake air shaft will be reduced or eliminated.

Returning to the major objective of coal-mine air conditioning, that of roof control, it may be explained that roof difficulties in some localities are aggravated during the summer when warm humid air comes in contact with cooler coal and rock surfaces below ground. The effect upon the roof and ribs is two-fold (1) deposition of moisture whenever the dew point temperature of the air entering the mine is higher than the underground rock temperature and (2) expansion of strata caused by warm air coming in contact with cold underground surfaces. Much argument has ensued as to which condition is responsible for scaling that is produced in many localities. Some attribute the scaling to moisture deposition, whereas others attribute it to temperature difference. It is probable that sometimes one factor is responsible, at other times the other factor and usually a combination of both factors. The remedy is the same in any case, that is to precool the air entering the mine until its dew point is equal to or lower than the temperature of the roof and ribs.

There are three methods of accomplishing the desired cooling:

(A) use of spring water when it is available at sufficiently low temperature and in sufficient quantity.

(B) cooling the air with artificial refrigeration.

(C) utilizing the return mine air as a source of refrigeration.

As an example, the problem will be considered of conditioning 100,000 cubic feet per minute of intake air which has a dry bulb temperature of 90 F and 50 percent relative humidity. It is desired to cool this air to a dry bulb temperature of 60 F and 100 percent relative humidity, which corresponds approximately to the condition of return mine air normally encountered. The problem becomes one of extracting 108,000 Btu per minute from the intake air.

(A) Cooling with Spring Water. Let it be assumed that there is adequate spring water available at a temperature of 55°F. By using a dehumidifier consisting of water sprays, pumps, etc., it is theoretically possible to obtain the desired cooling by circulating 528 gallons of cooling water per minute, using a 2-stage counterflow humidifier requiring two 600-gpm pumps to work at a pressure of approximately 30 pounds per square inch. It is obvious that the difficulty with this system is availability of 528 gpm of cooling water at 55° F. While such a supply of water is available in some localities, this is not usually the case.

(B) Cooling with Artificial Refrigeration. Where a natural supply of cooling water is not available, it is possible to use an artificial refrigerating unit or ice machine in addition to the equipment previously listed. The desired cooling will theoretically require a refrigeration machine capable of producing the equivalent of 539 tons of melting ice per 24 hours. In practice, a 600-ton unit would be selected, which in itself might cost in the neighborhood of a half million dollars, and its operation would approximate a 1000 horsepower of connected load in addition to the power required for operating the two 600-gpm pumps of the dehumidifier.

(C) Cooling by Return Mine Air. Where natural spring water is unavailable but where a 2-compartment air shaft is available in which all of the intake air for conditioning enters the mine, it is possible to effect the desired cooling by using a humidifier in conjunction with the dehumidifier described under (A) above and thereby use the cooling effect of the return mine air as a source of refrigeration.

In principle, the humidifier is a replica of the dehumidifier, consisting of cooling sprays and pumps. The function of the humidifier is to cool water discharged from the dehumidifier by spraying it into the return mine air and recirculating it to the dehumidifier. The humidifier thus provides the cooling water required by the dehumidifier at 60° F.

The desired cooling is theoretically available by pumping 667 gpm of 60° water to the dehumidifier sprays with two 700- gpm pumps. Two pumps are used in each case, assuming use of 2-stage units or what might be described as 2-cell dehumidifiers and 2-cell humidifiers. Theoretically, it is necessary to supply 7 gpm of 60° makeup water of which approximately 3 gpm are drained away from the high temperature accumulator to effect the proper heat balance. A flow of approximately 4 gpm supplies the excess water which the return air picks up in the humidifier over and above the water that is deposited by the intake air in the dehumidifier.

It will be seen that the system (C) requires approximately twice the equipment of system (A). However, the pumps must be of 700-instead of 600-gpm capacity as a result of the difference between the temperature of the 60° return air on the one hand and the 55° temperature of the spring water on the other.

A unique way of refrigerating intake air was devised by A. S. Richardson, ventilation engineer, Anaconda Copper Mining Company at Butte, Montana. Mr. Richardson was faced with the problem of having to cool air entering the lower levels of Anaconda mines. These mines are in an ore body which is still in the process of cooling and rock temperatures reach approximately 130°F. Fortunately the summer psychometric conditions at Butte are such that the outside air maintains a comparatively low dew point temperature. This made it possible to apply the cooling tower principle and to obtain a large supply of refrigerant in the form of brine at a temperature approximately half way between the outdoor dew point and the outdoor wet bulb temperatures. Unfortunately this solution would not be applicable in the eastern part of the United States due to the prevailing high dew point temperatures of outside air.

Auxiliary Ventilation with Tubing and Blowers. The use of blower fans and tubing for face ventilation is a controversial subject. Most authorities agree upon the advantage of this system over the maintenance of line brattice for dead end ventilation but they also recognize the system's disadvantages and the fact that there is considerable disagreement as to the proper relation of advantages to disadvantages. The main justification for the use of tubing and blowers for ventilating dead ends is that the installation and maintenance of line brattice is rendered impractical because of long distances desired between crosscuts or because operation of mining equipment limits the space available. It is generally agreed that when blowers and tubing are used the blowers should be powered with permissible motors and that proper ground connection should be maintained. Furthermore, the blowers should be operated continuously to avoid possibility of high gas concentration during shut down intervals. It is also important that the blowers be placed on fresh air in such a manner as to exclude possibility of recirculation of contaminated air.

For a detailed discussion of this subject reference is made to Appendix D.

Exhaust vs Pressure Systems. This subject also is controversial, each system having advantages and disadvantages. On the basis of explosion hazard it can be argued that the exhaust system is preferable because mine gas liberated at the working faces are returned along air courses which are free of open electrical circuits and sparking electric motors. Of course any properly ventilated coal mine would be supplied with sufficient air to dilute the gas below the limits of an explosive mixture. However, this precaution is not always observed and possible existence of local gas pockets argues for the exhaust system.

In case of fan stoppage, the absolute pressure throughout the mine increases slightly with an exhaust system and lowers slightly with a pressure system. A case can thus be made in favor of the exhaust system on the grounds that an increase in absolute mine pressure tends to hold the gas back in the worked-out places during fan stoppage, whereas, the reverse condition prevails for a pressure system. This conclusion is qualitatively correct but it might not hold when analyzed quantitatively. In any event it ceases to be a point at mines where old workings are properly sealed.

Mines fires are most likely to occur on haulage roads or at working places, due to exposed trolley wires being wrecked by roof falls, or coming in contact with wooden doors, etc., and due to locomotive and machine cables being cut. In event of such a fire, the exhaust system takes the smoke and fumes into the mine and out the air courses thru which men must travel for escape so, it may become necessary to reverse the action of the mine fan before the men leave the mine, but this may cause confusion and loss of life. The pressure system, on the other hand takes the smoke out along the haulage road and leaves the air course on fresh air, permitting safe travel by the escaping miners. After all the men are out of the mine, the fan can safely be reversed to permit men and fire fighting equipment to reach the fire.

In cold weather the exhaust system presents the disadvantages of freezing water in the hoisting shaft and causing discomfort for the cage men and other bottom workers. The pressure system conversely, will form ice in the air shafts but will leave the hoisting shaft free of ice and maintain comfortable working temperature at the mine bottom. At some mines, breaks to the surface near a burning refuse dump eliminate consideration of the exhaust system due to the fact that smoke-contaminated air would be drawn into the mine at some point on the intake circuit. In such cases a pressure system with the fan properly located eliminates possibility of contamination.

When it is desirable to condition the intake air for summer roof-control purposes, or winter heating, the pressure system is preferable to the exhaust system because the air conditioning unit can be installed adjacent to the mine fan, thereby passing all of the intake air thru the conditioning unit. This is practically impossible when all of the intake air is drawn thru a hoisting shaft or slope under the exhaust system.

It is possible with a pressure system to place the haulage road on fresh air with the aid of automatic haulage-road doors or a skip hoist, thereby combining some of the desirable features of each system but it is usually felt that the advantages gained do not justify the disadvantages of the automatic doors on the main haulage artery.

Selection of Mine Fan. The problem of choosing the primary fan for a coal mine is one of selecting a fan of the proper size and operating characteristics to perform efficiently the entire range of ventilation duties that will be required during life of the fan. Sometimes it is possible to select one fan to suffice for the life of the mine, especially now that the vaneaxial type of mine fan equipped with adjustable blades is available.

This type of fan has rendered the older centrifugal type of fan practically obsolete for coal mine ventilation.

Figure I shows comparative efficiency ranges of the vaneaxial and centrifugal fans operating over a range of mine resistance. The comparison is between a standard commercial vaneaxial mine fan with adjustable blades and fixed straightener vanes, and the most optimistically rated centrifugal fan with backward curved blades that is available.

It is obvious that the average operating efficiency of the vaneaxial fan far exceeds the average efficiency of the centrifugal fan. As the average fan efficiency during the operating life of the fan determines the power cost of mine ventilation, it is apparent why the vaneaxial fan has superseded the centrifugal fan for coal mine ventilation.

Having selected the primary mine fan for either exhausting or blowing duty, the fan should be installed on the surface with a distance of not less than 25 feet between the fan rotor and the near side of the air shaft, slope or drift. The fan can be direct-connected or driven by belt or gear connection to the prime mover such as electric motor, steam or internal com-

COMPARATIVE EFFICIENCIES OF TYPICAL AXIAL FLOW AND CENTRIFUGAL FANS

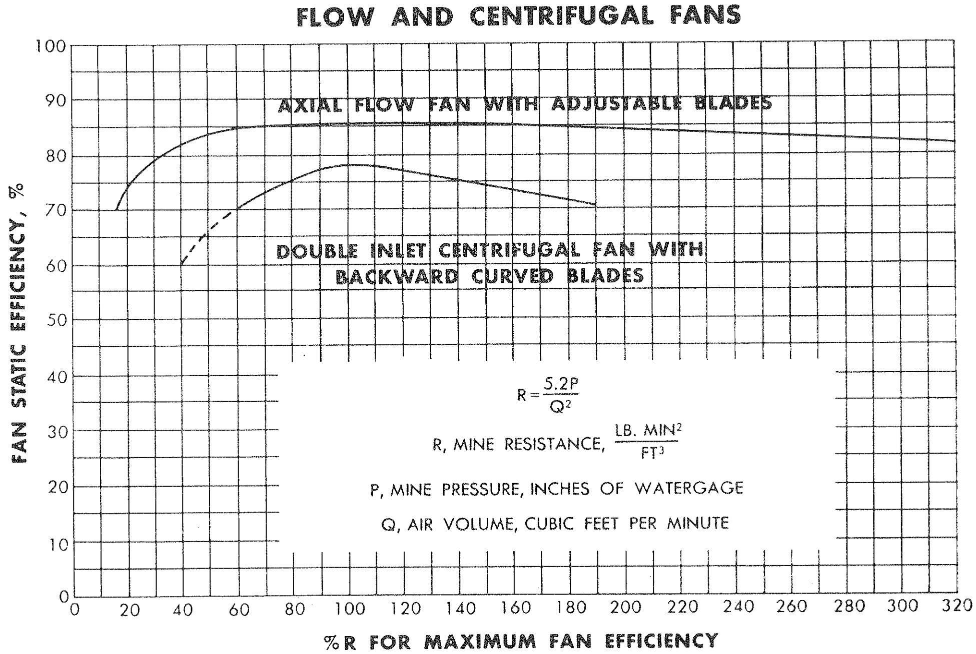


Fig. 1

bustion engine. It is always a good precaution to provide a standby drive unit to insure continued fan operation.

Fan bearings should be equipped with thermal relays designed to stop the fan in event of excessive bearing temperature. A visual or audible signal to indicate continued fan operation is an additional safety feature favored by some state mining departments.

Underground Location of Replacement Fans. Advent of the compact high-pressure efficient vaneaxial mine fan further emphasizes the desirability of locating replacement fans at or near the bottom of the air shaft or slope or just inside the drift mouth at mines already equipped with an outside fan which is maintained in operating condition for emergency use.

Figure II shows satisfactory methods of underground fan installations that insure ventilation of the fan drive-motor by fresh intake air. There are several such installations in this country, altho some state mining laws arbitrarily prohibit the practice of putting a main fan underground. England and Continental Europe have granted the practice statutory approval, recognizing the following safety and economical advantages of such installations:

1) two independent means of mine ventilation instead of one;
2) possibility of rapid scavenging following an inside explosion by simultaneous operation of surface and inside fans;
3) increased protection against damage by lightning and outside forces by virtue of sheltered location of inside fan;
4) instant availability of surface fan virtually assured following the worst possible mine explosion;
5) power reduction possibilities at mines

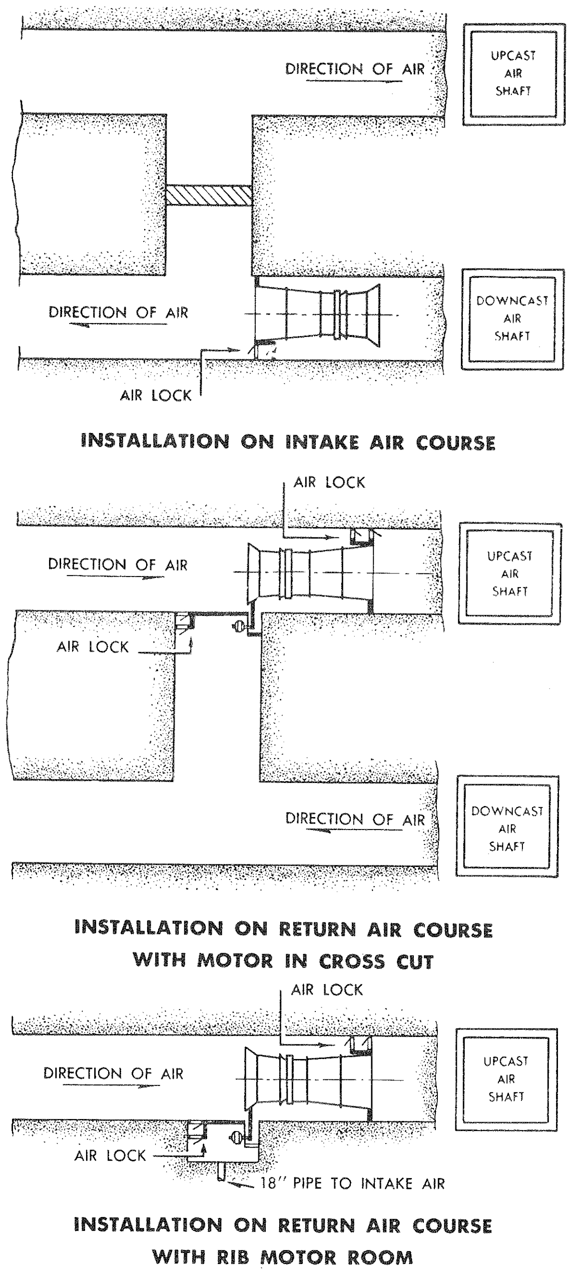


Fig. 2

equipped with a multicompartment air shaft with leaking curtain wall ;

6) power reduction possibilities at mines employing major split regulators;
7) elimination of objectionable fan noise;
8) reduced installation costs;
9) uninterrupted mine operation during fan installation;
10) operating and installation economies encourage improved ventilation at mines difficult to ventilate.

Objection to inside fans based upon possible airway blockage by the wrecked fan following an explosion can be eliminated by installing the underground fan in a run-around parallel to the air-way, with a weak stopping erected in the main air way. Objection on the grounds of air recirculation thru leaking curtain walls disappears when the problem is subjected to quantitative analysis. The principle of inside location for replacement fans is covered by a paper presented before the National Safety Council at the October 1938 meeting held in Chicago, Ill.

Testing Mine Fans. After installation, the primary mine ventilation fan should be carefully tested in place as a check against the accuracy of the fan manufacturer's claims.

When testing a mine fan, accurate measurements of air volume handled by the fan and pressure differential across the fan and power input to the fan shaft are required.

In order to obtain accurate measurements of air volume handled by the fan, it is necessary to conduct such measurements at a point where the air flow is uniform and comparatively free from turbulence. In the case of centrifugal fans, this requirement is satisfied at locations where air velocities are of the order of magnitude of from 1,500 ft. to 3,000 ft. per minute, which is the velocity range best served by an anemometer traverse. In the case of the vaneaxial type mine-fan, however, uniform and turbulent-free flow occurs as the air approaches the fan rotor where air velocities of the order of magnitude of from 4,000 ft. to 10,000 ft. per minute are encountered, which is ideal for a Pitot-tube traverse.

Pressure differentials across the fan should rightfully be taken between the fan inlet and the fan outlet, which requires agreement on definition of these two positions.

The power input to the fan requires knowledge of the motor power output and in the case of a belt driven fan an estimate of the efficiency of the intervening drive.

Appendix E describes in detail the proper method of Pitot-tube field-testing of vaneaxial mine fans.

Appendix A

Surveys of Underground Mine Pressure

The purpose of an accurate underground pressure survey is to obtain a pressure gradient along the circuit or circuits under investigation. The pressure gradient shows the rate of pressure drop between various observation points along a circuit. This determines excessive resistance regions and the economic feasibility of correcting such conditions by cleaning airways, driving additional airways, or modifying existing airways.

A pressure gradient is useful for predicting the effect on other parts of the circuit when adjusting the air regulation on any one split or when planning a field airshaft or slope or a new drift to the outcrop. A single-compartment airshaft often produces disappointing results because it relieves only a part of the circuit instead of the over-all mine pressure. It is valuable to know how the circuit pressure drop is distributed.

The underground pressure survey is used also to determine the friction coefficient for various types of airway. High precision work is required to establish these vital data for predicting the characteristics of mine pressure volume when projecting new mine developments.

Methods

Regardless of the type of instrument used to conduct a circuit pressure survey, the work should preferably be performed at a time when the mine is idle, in order to avoid air-flow disturbances from moving trips and cages or from the opening or closing of trap doors.

Preferably the survey should be made when the barometer is comparatively steady, to restrict variation in the air specific weight at observation stations while the circuit is traversed. The two most commonly used instruments for underground pressure surveys are: (1) the altimeter; (2) the inclined manometer with hose extensions.

The altimeter is sufficiently accurate for determining the underground pressure gradient and is preferable to the manometer for such work because of greater flexibility and shorter time required for the traverse. With the altimeter it is possible to select more widely separated observation stations than is practical with the manometer, with its restricted hose extension lengths. The altimeter also permits easier travel along haulage roads between airway observation stations, avoiding badly obstructed airways.

The inclined manometer is more reliable than the altimeter for establishing friction coefficients in mine airways. Manometer readings are unaffected by differences in elevations between stations, which require corrections to altimeter data. The manometer provides direct reading without temperature correction, which may or may not be true with the altimeter, depending upon the make of instrument used. The elimination of correction factors combined with the instantaneous, simultaneous and direct reading of the air-flow pressure drop between two successive stations permits a higher degree of accuracy with the inclined manometer than is possible with the altimeter. This is highly desirable for determination of friction coefficients.

Altimeters at best are imperfect instruments; their accuracy depends upon mechanical fidelity or persistent adherence to the calibration scale. The manometer indicates pressure directly, in terms of the height of water column that the pressure will support.

Altimeters are subject to some lag or creep during a traverse. With either lag or creep, an altimeter will not return exactly to the original reading when returned to the starting station, even if the pressure at the starting station remains constant throughout the traverse. The degree of this discrepancy depends upon the quality of the altimeter; it can be kept within acceptable working tolerances by the careful selection of instruments.

Altimeters are available that are supposedly compensated for temperature effects upon instrument accuracy. However, an instrument with a temperature correction chart gives more accurate results than one used without benefit of intermediate temperature calibration.

Traversing with the Altimeter

The altimeter is a highly sensitive aneroid type of pressure instrument, calibrated so as to indicate the elevation above sea level of a theoretical 50°F. column of dry air, when the sea-level barometer is 29.90 in. Hg. The pressure-elevation relationship is given in Smithsonian Meteorological Table No. 51 according to the formula:

$$H = 60{,}368\ (1 + 0.0010195 \times 36) \log \frac{29.90}{B}$$

wherein B is the barometric pressure (in-Hg) at any elevation H (ft.) of the theoretical air column. Curve a in Fig. 1, plots B as ordinates against H as abscissas—a useful reference when using an altimeter as a barometer. First, the instrument is adjusted, alongside a mercury barometer, to record the elevation H read from curve a, corresponding to the reading on the mercury barometer. Then the altimeter can be used to measure pressure by referring to curve a to obtain the pressure B for any recorded value of H.

Used to determine a mine-pressure gradient, the altimeter records the instantaneous static pressure at each observation station in terms of the height of the theoretical air column, at which the column pressure is the same as at the observation station. The static pressure difference between two stations is the pressure difference that would exist between the two elevations of a theoretical air column indicated by

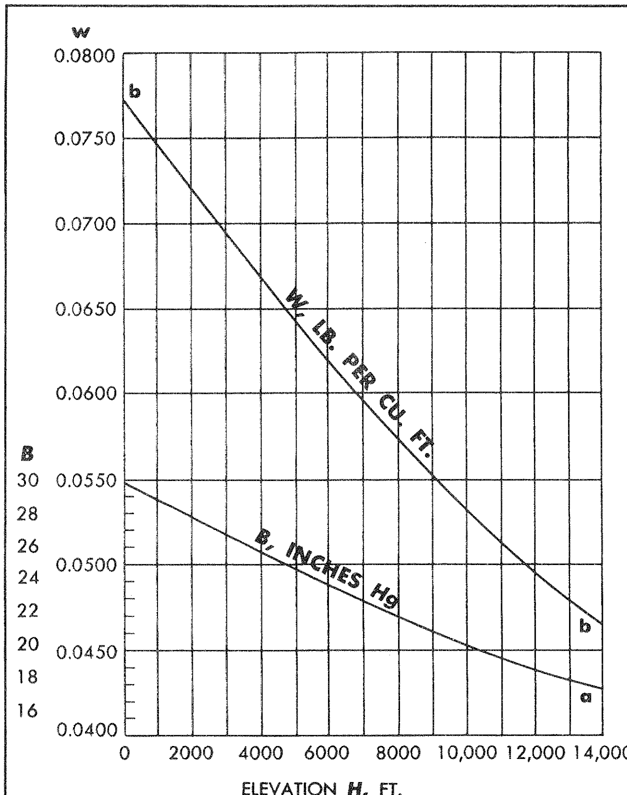


Fig. 1

the altimeter, which in turn corresponds to the weight of that part of the theoretical column.

Usually the pressure difference between two observation stations is too small to read directly from curve a of Fig. 1. To calculate this pressure difference, multiply the height of the portion of the theoretical air column, as indicated from the difference between the two altimeter readings, by the average specific weight of the corresponding portion of the theoretical air column. This pressure difference is expressed in pounds per square foot.

By calculus it can be shown that, when considering a column of dry air at constant temperature throughout, the average specific weight of any portion of the column can be taken as the average of the two specific weights at the top and bottom of the column portion under consideration, with an error of less than 1 per cent for column heights under 9000 ft. Fig. 2 illustrates this point for dry air at 50°F., which serves as the calibration basis for the commercial altimeter.

Curve *b*, Fig. 1, shows specific weight *W* as ordinates with elevation *H* as abscissas for dry air at 50°F., and the specific weight of the theoretical air column for any altimeter reading *H*, when calculating the average specific weight of the theoretical column between the two elevations on the altimeter.

With the static pressure difference between two observation stations determined as described, it is necessary to calculate the amount of differential that is due to difference in station elevation or gravitational effect. Gravitational pressure difference results from the weight of the actual air column, of height equal to the difference in station elevations and average specific weight, approximately equal to the average of the two actual air specific weights

Table 1.

Dry Bulb Temperature Degrees Fahr.	Pounds Per Cubic Foot															
	Barometric Pressure—Inches															Approx. Averg Increase in Weight Per Degree Wet Bulb Depression
	26			27			28			29			30			
	Wt. per Cu. Ft. Saturated Air	Decr's Wt. per Deg. Inc. Dry Bulb	Incr's Wt. per 0.1" Rise in Bar.	Wt. per Cu. Ft. Saturated Air	Decr's Wt. per Deg. Inc. Dry Bulb	Incr's Wt. per 0.1" Rise in Bar.	Wt. per Cu. Ft. Saturated Air	Decr's Wt. per Deg. Inc. Dry Bulb	Incr's Wt. per 0.1" Rise in Bar.	Wt. per Cu. Ft. Saturated Air	Decr's Wt. per Deg. Inc. Dry Bulb	Incr's Wt. per 0.1" Rise in Bar.	Wt. per Cu. Ft. Saturated Air	Decr's Wt. per Deg. Inc. Dry Bulb	Incr's Wt. per 0.1" Rise in Bar.	
0	.07500	.00016	.00029	.07788	.00016	.00029	.08077	.00017	.00029	.08365	.00018	.00029	.08654	.00019	.00029	.000015
10	.07338	.00016	.00028	.07620	.00016	.00028	.07903	.00017	.00028	.08185	.00018	.00028	.08468	.00018	.00028	.000016
20	.07180	.00016	.00028	.07456	.00016	.00028	.07733	.00017	.00028	.08009	.00018	.00028	.08286	.00018	.00028	.000017
30	.07027	.00015	.00027	.07297	.00016	.00027	.07569	.00016	.00027	.07839	.00017	.00027	.08110	.00017	.00027	.000019
40	.06879	.00015	.00026	.07143	.00015	.00027	.07409	.00016	.00027	.07675	.00016	.00027	.07942	.00017	.00027	.000021
50	.06732	.00015	.00026	.06992	.00015	.00026	.07252	.00016	.00026	.07512	.00016	.00026	.07773	.00016	.00026	.000023
60	.06588	.00015	.00026	.06843	.00015	.00026	.07098	.00015	.00026	.07353	.00016	.00026	.07609	.00016	.00026	.000026
70	.06442	.00015	.00025	.06692	.00015	.00025	.06943	.00015	.00025	.07193	.00016	.00025	.07440	.00016	.00025	.000029
80	.06297	.00015	.00025	.06542	.00015	.00025	.06789	.00015	.00025	.07034	.00015	.00025	.07280	.00016	.00025	.000034
90	.06146	.00015	.00024	.06388	.00016	.00024	.06629	.00016	.00024	.06870	.00016	.00024	.07112	.00017	.00024	.000039
100	.05991	.00016	.00024	.06228	.00016	.00024	.06465	.00016	.00024	.06703	.00017	.00024	.06939	.00018	.00024	.000044
110	.05828	.00016	.00023	.06060	.00017	.00023	.06293	.00017	.00023	.06526	.00018	.00023	.06759	.00019	.00023	.000051
120	.05653	.00018	.00023	.05882	.00018	.00023	.06111	.00018	.00023	.06339	.00019	.00023	.06569	.00020	.00023	.000059
130	.05467	.00019	.00023	.05692	.00019	.00023	.05917	.00019	.00023	.06142	.00020	.00023	.06367	.00022	.00023	.000068
140	.05262	.00021	.00022	.05483	.00021	.00022	.05704	.00021	.00022	.05925	.00022	.00022	.06147	.00024	.00022	.000078
150	.05036	.00023	.00022	.05253	.00023	.00022	.05471	.00023	.00022	.05689	.00024	.00022	.05906	.00026	.00022	.000090
160	.04788	.00025	.00022	.05001	.00025	.00022	.05216	.00026	.00021	.05430	.00026	.00021	.05644	.00029	.00021	.000103
170	.04509	.00028	.00021	.04720	.00028	.00021	.04931	.00029	.00021	.05141	.00031	.00021	.05352	.00033	.00021	.000118
180	.04197	.00031	.00021	.04404	.00031	.00021	.04611	.00032	.00021	.04818	.00034	.00021	.05026	.00036	.00021	.000134
190	.03845	.00035	.00021	.04049	.00036	.00021	.04253	.00036	.00021	.04457	.00037	.00021	.04662	.00038	.00021	.000153
200	.03449	.00040	.00020	.03650	.00040	.00020	.03851	.00040	.00020	.04052	.00041	.00020	.04254	.00041	.00020	.000173

Weights of Saturated and Partly Saturated Air for Various Barometric and Hygrometric Conditions

at the two respective stations. The actual specific weight of air at each station can be determined from the dry-bulb and wet-bulb temperatures and from the station barometer by means of Table 1, or it can be separately calculated. The station barometer of course, is read from curve *a*, Fig. 1, using the altimeter reading of *H*. The gravitational pressure difference is the product of the difference in station elevation as determined by the mine surveyor, and the average of the actual air specific weights existing at the two respective stations.

Having calculated the gravitational pressure difference as described, the pressure difference between the two stations, resulting from air flow only, is obtained by adding or subtracting the gravitational pressure correction to or from the actual pressure difference between the two stations, depending upon decrease or increase of actual station elevation in direction of the air flow.

Provided observation stations have been selected, with air speed about the same at each station, the pressure difference between successive stations due to air flow is also the pressure difference due to friction and shock. It is not usually practical to select stations with the same average air speed, and correction for velocity head becomes necessary in order to compute actual pressure drop due to friction and shock alone. The velocity head *VP* (lb. per sq. ft.) existing at each station with average air speed *V* (ft. per min.) and air specific weight *W* (lb. per cu. ft.) can be computed by the formula

$$VP = \frac{WV^2}{231{,}000}$$

The difference in the velocity heads at two successive stations should be added or subtracted, depending upon whether the air speed at the upstream station is higher or lower, respectively, than at the downstream station.

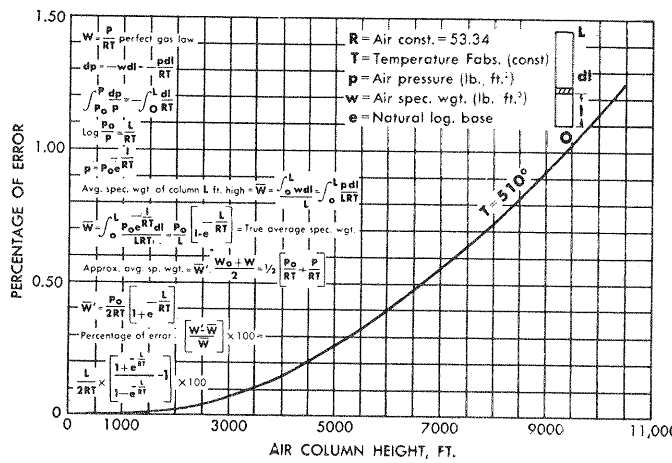


Fig. 2

Two conventional methods of altimeter survey are most commonly used; both requiring the use of two altimeters. The first method involves adjustment of both instruments to the same reading at each station, then moving altimeter *B* to the downstream station leaving altimeter *A* at the upstream station, or vice versa. The two instruments are then read simultaneously by two observers, with the aid of synchronized watches.

The second method leaves altimeter *A* at the first station throughout the entire traverse of the other stations with altimeter *B*. Altimeter *A* is read at regular time intervals of, say, 10 min. or less, thereby providing a benchmark relative to which pressures at the other stations are compared when computing pressure differences between successive stations. Altimeter *B* therefore is a roving instrument, which is read at random times at which both the time and the altimeter reading are recorded for comparison with the stationary or reference altimeter.

Fig. 3 shows a typical data table and computation table required by the first method. Fig. 4 shows the two data tables and the computation

DATA																				
Stations	Elevation (Feet)		Time		Altimeter Read'g. Feet		Barometer Ins. Hg.		Wet Bulb (° F)		Dry Bulb (° F)		Specific Wgt. Actual Air (Lbs/Ft.^{3})		Specific Wgt. Theoretical Air (Lbs/Ft.^{3})		Station Area (Ft.^{2})		Average Air Speed (Ft/Min.)	
	A	B	AM	PM	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
o-n																				
0-1																				
1-2																				
2-3																				
n-o																				

COMPUTATIONS										
Stations	Altimeter Difference (Feet)	Average Specific Wgt. Theoretical Air (Lbs/Ft.^{3})	Pressure Difference (Lbs/Ft.^{2})	Elevation Difference (Feet)	Average Specific Wgt. Actual Air (Lbs/Ft.^{3})	Pressure Diff. Due to Elevation (Lbs/Ft.^{2})	Velocity Head Correction (Lbs/Ft.^{2})	Pressure Diff. Due to Air Flow		Average Air Volume C.F.M.
								(Lbs/Ft.^{2})	(Ins. Wg.)	
o-n										
0-1										
1-2										
2-3										
n-o										

Fig. 3

table required by the second method. The first method requires but one data table because the two altimeter operators meet at each station to synchronize instruments. On the other hand, two data tables are required by the second method because the two altimeter operators do not necessarily meet except at the beginning and completion of the entire circuit traverse.

Each of the two survey methods has its adherents and there is little choice between them as to relative merits. It may be argued that the first method requires more care and is slower because of the necessity of having to synchronize the two instruments at each station and to adhere to a time schedule. However, the first method is less susceptible to error caused by local disturbance on the air circuit in the vicinity of the stations undergoing traverse, because the disturbance can be detected by the upstream instrument. It thus combines an advantage of the inclined manometer with the additional advantages of being able to select more widely separated observation stations and the ability to travel the haulage road between stations.

On the other hand, partisans of the second method argue that the greater speed of the second method offsets the advantages claimed for the first method.

DATA
Stationary Altimeter "A" Elevation (Ft.)

Time 10 Min. Interval* 1	Altimeter Ft. Air 2	Barometer Ins. Hg. 3	Wet Bulb (° F.) 4	Dry Bulb (° F.) 5	Air Specific Wgt. (Lbs./Ft.³) 6

*Note: Shorter Time Intervals when Barometer is Unsteady

DATA
Portable Altimeter "B"

Stations	Elevation (Feet)	Time A.M.	Time P.M.	Altimeter (Ft. Air)	Barometer Ins. Hg.	Wet Bulb (° F.)	Dry Bulb (° F.)	Specific Wgt. Actual Air (Lbs./Ft.³)	Specific Wgt. Theoretical Air (Lbs./Ft.³)	Station Area (Ft.²)	Average Air Speed (Ft./Min.)
0											
1											
2											
3											
n											

COMPUTATIONS

Stations	Time A.M.	Time P.M.	Altimeter Diff. Feet	Average Sp. Wgt. Theoretical Air Column (Lbs./Ft.³)	Pressure Difference (Lbs./Ft.²)	Elevation Difference (Feet)	Average Specific Wgt. Actual Air (Lbs./Ft.³)	Pressure Diff. Due to Elevation (Lbs./Ft.²)	Velocity Head Correction (Lbs./Ft.²)	Pressure Diff. Due to Air Flow (Lbs./Ft.²)	Pressure Diff. Due to Air Flow (Ins. Wg.)	Average Air Volume C.F.M.
0-1												
0-2												
0-3												
0-n												

Fig. 4

Examples of the Usefulness of a Pressure Gradient

Fig. 5 illustrates two examples of the usefulness of a circuit pressure gradient when contemplating the ventilation increase on an inby mine split without altering the ventilation on the remaining splits. The first sketch of Fig. 5 illustrates the initial air-volume distribution, regulation and pressure-drop distribution on a circuit consisting of four principal air splits. Air leakage through stoppings is disregarded. It is assumed that pressure-drop distribution

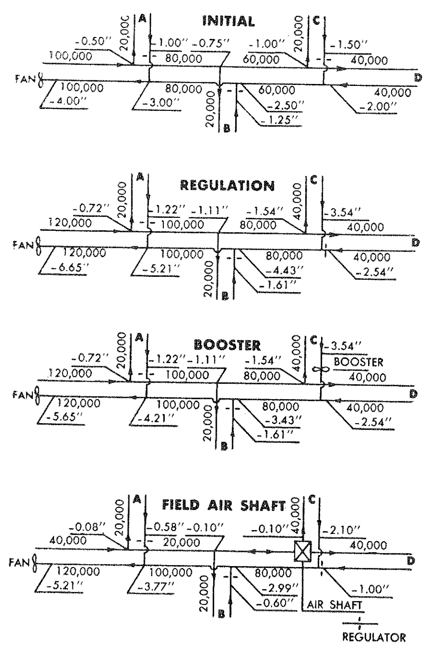


Fig. 5

has been accurately calculated from a careful altimeter survey (Fig. 6) to analyze the problem of doubling the air volume on split *C* without changing the air quantities on splits *A*, *B* and *D*.

Initially the primary exhaust fan handles 100,000 cu. ft. per min. at a 4-in. water-gauge depression and performs useful work upon the air at the rate of 63 air-hp. The split regulation is 2.00 in., 1.25 in. and 0.50 in. water-gauge pressure on splits *A*, *B* and *C*, respectively.

Sketch 2 of Fig. 5 shows the air-volume and pressure-drop distribution and regulation required when doubling the *C*-split ventilation by placing 1.00 in. water-gauge regulation upon the *D* split and by increasing the regulation upon splits *A* and *B* to 3.99 in. and 2.82 in. water gauge, respectively. The fan must now handle 120,000 cu. ft. per min. at 6.65 in. water-gauge depression, which is 126 air-hp. useful work done upon the air, or exactly twice the fan-power output that was required initially.

Sketch 3 of Fig. 5 illustrates the volume and pressure-drop distribution resulting from the use of a booster fan upon the *C* split on which the ventilation is to be doubled. The booster fan must pass 40,000 cu. ft. per min. at 1.00 in. water-gauge pressure across the booster fan and no regulation is required upon the *D* split. But the regulation on splits *A* and *B* becomes 2.99 and 1.82 in. water gauge, respectively; and the primary mine fan must pass 120,000 cu. ft. per min. at 5.65 in. water gauge corresponding to 107 air-hp. The useful work on the air by the booster fan is 6.3 air-hp., resulting in a total of 113.3 air-hp. by the two fans as compared with 126 air-hp. without the booster.

Fig. 7 is a self-explanatory graphical presentation of the air-power distribution with and

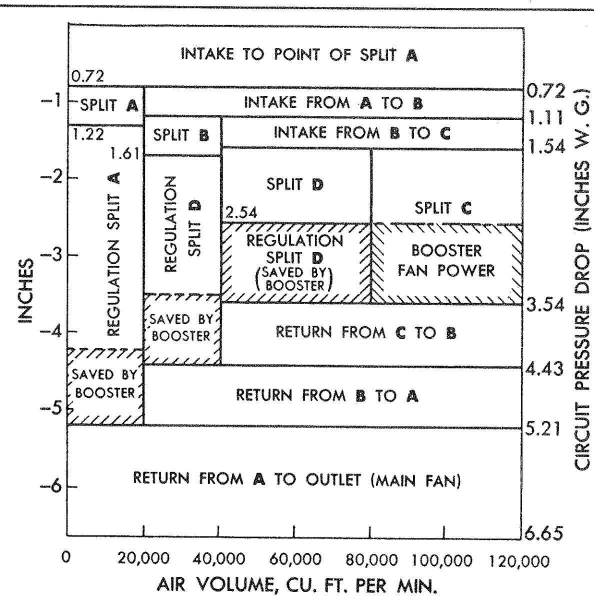


Fig. 7

DATA

Stations	Elevation (Feet)		Time	Altimeter Read'g Feet		Barometer Ins. Hg.		Wet Bulb (° F.)		Dry Bulb (° F.)		Specific Wgt. Actual Air (Lbs./Ft.³)		Specific Wgt. Theoretical Air (Lbs./Ft.³)		Station Area (Ft.²)		Average Air Speed (Ft./Min.)	
	A	B	A.M. P.M.	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
0-7	750	760	- 6:00	800	1073	29.0	28.7	40	60	45	60	.07606	.07280	.07540	.07465	100.0	120.0	1000	833
0-1	750	745	- 6:30	804	834	29.0	29.0	40	45	45	50	.07606	.07523	.07540	.07540	100.0	95.3	1000	1050
1-2	745	748	- 7:00	838	857	29.0	29.0	45	52	50	55	.07523	.07439	.07520	.07520	95.3	94.0	1050	850
2-3	748	750	- 7:30	860	878	28.9	28.9	52	60	55	60	.07413	.07332	.07520	.07520	94.0	95.0	850	632
3-4	750	752	- 8:00	878	948	28.9	28.8	60	60	60	60	.07332	.07306	.07520	.07520	95.0	80.0	632	500
4-5	752	750	- 8:30	945	979	28.8	28.8	60	60	60	60	.07306	.07306'	.07500	.07480	80.0	85.0	500	706
5-6	750	755	- 9:00	975	1016	28.8	28.8	60	60	60	60	.07306	.07306	.07480	.07480	85.0	87.0	706	920
6-7	755	760	- 9:30	1015	1090	28.8	28.7	60	60	60	60	.07306	.07280	.07480	.07470	87.0	95.0	920	1055
7-0	760	750	- 10:00	1090	803	28.7	29.0	60	38	60	40	.07280	.07679	.07470	.07540	95.0	100.0	1055	1000

COMPUTATIONS

Stations	Altimeter Difference (Feet)	Average Specific Wgt. Theoretical Air (Lbs./Ft.³)	Pressure Difference (Lbs./Ft.²)	Elevation Difference (Feet)	Average Specific Wgt. Actual Air (Lbs./Ft.³)	Pressure Diff Due to Elevation (Lbs./Ft.²)	Velocity Head Correction (Lbs./Ft.²)	Pressure Diff. Due to Air Flow (Lbs./Ft.²)	Pressure Diff. Due to Air Flow (Ins. Wg.)	Average Air Volume C.F.M.
0-7	+273	.07502	—20.43	+10	.07443	—.74	+.11	—20.80	—4.00	—
0-1	+ 30	.07540	— 2.25	— 5	.07564	+.38	—.03	—2.60	—0.50	100,000
1-2	+ 19	.07520	— 1.39	+ 3	.07481	—.22	+.13	—1.30	—0.25	80,000
2-3	+ 18	.07520	— 1.34	+ 2	.07372	—.15	+.11	—1.30	—0.25	60,000
3-4	+ 70	.07520	— 5.30	+ 2	.07319	—.15	+.05	—5.20	—1.00	40,000
4-5	+ 34	.07490	— 2.53	— 2	.07306	+.15	—.08	—2.60	—0.50	60,000
5-6	+ 41	.07480	— 3.08	+ 5	.07306	—.37	—.11	—2.60	—0.50	80,000
6-7	+ 75	.07475	— 5.64	+ 5	.07293	—.36	—.08	—5.20	—1.00	100,000
7-0	—287	.07505	+21.57	—10	.07479	+.75	+.02	+20.80	+4.00	—

Fig. 6

without the booster fan on split *C*. The various areas represent the air power absorbed by different portions of the circuit.

The bottom sketch of Fig. 5 shows the results of a single-compartment intake airshaft sunk at the junction of the *C* and *D* splits to ventilate only these two splits; the ventilation for splits *A* and *B* being furnished from the original intake airshaft. The volume and pressure-drop distribution and split regulation is shown in Fig. 5. There is a neutral zone on the intake airways between splits *B* and *C*.

The resultant regulation is 3.19 and 2.39 and 1.10 in. of water-gauge pressure drop on splits *A*, *B* and *D*, respectively. Split *C* becomes a portion of the free split again, as in the case of sketch 2. The primary mine fan functions without the aid of a booster fan and must pass 120,000 cu. ft. per min. at 5.21 in. water-gauge depression, performing approximately 99 air-hp. useful work upon the air. It is obvious that a single-compartment intake airshaft located as suggested would be a poor investment from the standpoint of reduction in power for mine ventilation alone, because of the small pressure reduction that results.

It is apparent that to attempt a solution of the problem just analyzed, without the aid of a predetermined circuit pressure gradient, would be merely a matter of cut and try, which would be both tedious and expensive.

In the first place, the matter of the duty change for the primary mine fan with any one of the three possible methods discussed would involve the question of the adequacy of the existing motor and drive. This is regardless of whether the new duty could be met without: (1) speed change by changing blade adjustment, if practical, with an adjustable bladed vaneaxial type of fan; or (2) if a speed increase would be required. Error in estimating motor or drive capacity could be costly.

It is important for both cost and safety that the booster fan, if adopted, be selected of the proper size and driven at the proper speed to avoid recirculation, from return airways to intake airways, through leaky stoppings outby the booster location.

When contemplating the sinking of a field airshaft for any particular purpose, it is important that the benefits be carefully analyzed before proceding with an expenditure of such major proportions. In the case just cited, the proposed single-compartment airshaft would have been an unwise investment, considering power cost alone. Whereas the comparatively low resistance of the intake portion of the circuit, which includes the haulage road, is responsible for the poor showing made for the proposed airshaft, a similar study might justify a two-compartment airshaft equipped with a mine fan, or two airshafts close together, one of which would be equipped with a fan.

A survey determining the circuit pressure gradient is the only basis for exactly analyzing the type problem just considered.

Traversing with the Inclined Manometer

When determining airway friction coefficients with the inclined manometer, the equipment is comparatively simple. It consists of two legible inclined manometers accurate to .01 in. water column, each equipped with a leveling bubble. Each manometer should be mounted on a tripod, approximately midway between the two observation stations; this will produce a negligible effect upon the airflow pattern, because of the presence of the instrument and the observer, at either station. A sling psychometer and barometer should be used to determine wet-bulb and dry-bulb temperatures with a barometer reading at each station to determine in turn the specific weight of the air at each station. One Pitot tube is required to traverse each station, together with the necessary hose.

An essential precaution is guarding against the formation of condensate in the hose connections when moving from warm, humid air to cold air at a temperature below the dew point of the warm air. This might occur on a hot, humid summer day when moving from the intake airway to the return near the mine entrance. A similar reversed condition prevails on a cold winter day when changing from the return airway to the intake airway near the mine entrance. This condition can best be avoided by selecting a time for traverse when the intake mine air and the return air are at approximately equal temperatures. Limiting the traverse area to mine regions of approximately equal intake and return air temperatures on extremely hot or cold days will also reduce condensation.

It is unnecessary to correct manometer readings for differences in station elevation; the air column in the hose connections exactly balances the gravity head.

When attempting to determine the average friction coefficient c for a length of uniform airway L (ft.) of area A (sq. ft.) and perimeter O (ft.), it is necessary to measure the air volume Q (cu. ft. per min.) passing between the two stations, as well as to determine accurately the total pressure difference P (inches water column) resulting from air-flow friction and shock losses.

The average air specific weight W (lb. per cu. ft.) must also be determined by measuring the wet-bulb and dry-bulb temperatures, and the barometer, at each of the two stations. Air weight must be calculated at each station with the aid of Table 1, averaging the two weights. The relationship existing between the aforementioned properties is given by the formula:

$$c = \frac{5.2 \times A^3 \times P}{W \times L \times O \times Q^2}$$

The factor c forms a useful index to the degree of air-course cleanliness by comparison with published values by competent investigators or, preferably, by comparison with values selected as practical criteria for use at the mine in question.

The friction factor c is actually influenced by the size and shape of the airway, by the air speed, specific weight and viscosity; all of these combine in a relationship known as Reynolds number. The larger the Reynolds number, the smaller the value of c and vice versa. Considering the magnitude of Reynolds number variation in mine ventilation practice, together with the surface irregularities, variations of c with Reynolds number would fall outside the limits of working accuracy.

For determining friction factor c with the inclined manometer, rectangular frames should be erected at each of the two observation stations and each frame should be subdivided into 36 equal-area squares, by means of wires stretched between the sides, and from top to bottom, of each frame. Each frame should be traversed with a Pitot tube held at the center of each square subdivision and the velocity pressure and impact pressure should be recorded by means of the two inclined manometers.

One observer should be stationed at each of the two manometers to record the readings; and an operator is required to locate the Pitot tube at the center of each square subdivision directed into the air flow.

If practicable, it is desirable to use a thin stand or holder for the Pitot tube, so that the operator can move a sufficient distance downstream from the frame to leave the air-flow pattern undisturbed.

If the Pitot stand is not available, the operator should have a Pitot tube of sufficient length to enable him to reach the center of each square subdivision without changing his own location in the airstream. This will tend to preserve the air-flow pattern throughout the traverse, which otherwise might be constantly altered by different positions of the operator.

The manometer, which indicates velocity pressure, should have the suction leg connected to the static leg of the Pitot tube, with the pressure leg connected to the impact leg of the tube. Velocity pressures thus obtained serve a twofold purpose. First, the air speed at the center of each square subdivision can be calculated by the formula:

$$v = 1096.5 \sqrt{\frac{VP}{W}},$$

wherein v is the air speed (ft. per min.), VP is velocity pressure (inch-water column) and W is air specific weight (lb. per cu. ft.), as determined at each station.

The average air speed V (ft. per min.) over the entire frame area is the sum of the individual air speeds divided by the number of square subdivisions of the frame. The air volume Q (cu. ft. per min.) passing the station is the product of the frame area A (sq. ft.) and the average air speed V (ft. per min.).

The manometer indicating the impact pressure at each station has the pressure leg connected to the impact leg of the Pitot tube when traversing the upstream station and the suction leg connected to the impact leg of the Pitot tube when traversing the downstream station. In this manner the impact pressure h (inches w.g.) in each square subdivision of each frame is recorded relative to the static pressure at the manometer.

To compute the average impact pressure H (in. w.g.) at each station relative to the static pressure at the manometer the sum of the products of air speed and impact pressure at each square subdivision is divided by the sum of the individual air speeds in each square, thus:

$$H = \frac{v_1 h_1 + v_2 h_2 \cdot \cdot \cdot v_n h_n}{v_1 + v_2 \cdot \cdot \cdot v_n}$$

The difference in total pressure at the two observation stations P (in. w.g.) is therefore the sum of the two average impact pressures H at the two stations relative to the manometer at the two stations.

$$P = H_1 + H_2$$

Therefore the velocity-pressure readings serve to determine the air volume Q (cu. ft. per min.) at the station and the weighted average total pressure difference P (in. w.g.) between stations.

The inclined manometer is an accurate means of performing a delicate job, when used by competent observers.

A more detailed description of correct procedure when determining mine-airway pressure coefficients, as well as a list of suggested coefficient values for different types of mine airways, is given in United States Bureau of Mines Bulletin No. 285 by H. P. Greenwald and G. E. McElroy and in University of Illinois Engineering Experiment Station, Bull. No. 158.

These references describe the use of the Wahlen gauge, which is a more sensitive type of manometer than the inclined manometer; it permits higher precision work when measuring small pressure differentials. The Wahlen gauge can be accurately read to 0.001 in. and is easily estimated to the nearest 0.0001 in. of gauge liquid.

The wide range of airway-pressure coefficients listed by different investigators is probably due to differences in existing physical conditions rather than experimental accuracy.

The subject of determination of pressure coefficient for mine airways is fully covered in the references given.

Appendix B

Effects of underground stopping leakage upon mine fan performance

When calculating the pressure-volume characteristics of projected mine-ventilating circuits by orthodox methods, certain basic assumptions are required in order to employ the various available empirical data. It is assumed, for example, that the mine air is an incompressible fluid subjected to isothermal flow, an assumption sufficiently accurate for practical purposes since pressure and temperature differentials are small throughout the average circuit. A more erroneous assumption, however, is that all air is accounted for as it travels throughout the mine.

No attempt is usually made to evaluate stopping leakage as it occurs; instead, the air volume required at the last crosscuts is assumed to enter the mine, travel the various intake air courses intact, sweep the workings and travel out by the return air courses to the point or points of exit from the mine. Actually, there is a leakage of air from intake to return at every stopping, the quantity of this leakage depending upon the tightness of the stopping and the pressure difference across the stopping.

Unfortunately, a lack of empirical data and knowledge of the condition of individual stoppings makes an exact analysis of underground stopping leakage impossible. Generally, leakage is most severe through the old stoppings outby the circuit. These are also subjected to higher pressure differences than the newer inby stoppings. Therefore the circuit air volume diminishes at a decreasing rate progressing from outby to inby the circuit.

Consider a simple mine circuit consisting of one intake and one return air course, each of equal and uniform section area and inter-connected at regular intervals with crosscuts equipped with stoppings. At any point X inby the circuit the air volume q on the intake air course equals that on the return air course and decreases with diminishing rate from outby to inby the circuit between the limits of Q_1 completely outby and Q completely inby. Such a volume change is represented by curve *a*, Fig. 1, which is computed with the aid of the equation $q = Q_1 - (Q_1 - Q)\ (X \div L)^N$. This equation results from the general algebraic equation of form $q = k_1 - kX^N$ after solving for k_1 and k with the knowledge that completely outby $X=o$ and $q = Q_1$, also completely inby $X = L$ and $q = Q$. Curves *a*, *b* and *c*, Fig. 1, show values of q as ordinates versus values of X as abscissas, for values of N equal to 0.50, 1.00 and 2.00, respectively.

Distribution of undergound stopping leakage directly affects the air-volume distribution along the circuit, which in turn determines the circuit pressure gradient. To deliver any required air volume inby a circuit or portion thereof requires a higher circuit-pressure differential if stoppings leak than is necessary with tight stoppings. Every mining engineer engaged in the analysis of projected mine-ventilation circuits is therefore confronted with the problem of making proper allowance for the effects of leaking stoppings.

The purpose of this paper is to present evidence in support of the author's suggestion that circuit pressure increase (per cent) due to underground stopping leakage can be considered numerically equal to the leakage air volume expressed as percentage of air volume delivered inby the circuit. Stated somewhat differently, pressure ratio P_1/P is equal to volume ratio Q_1/Q, wherein P_1 and P represent circuit pressure differential with and without leaking stoppings, respectively, and Q_1 and Q represent the air volume outby and inby the circuit, respectively, when stopping leakage is present.

An exact example will best serve to illustrate the use of the principle of pressure correction according to volume ratio as applied to a portion of a typical ventilating circuit. The accuracy of this principle depends upon the assumption of a constant coefficient of unit resistance as well as the assumption that the circuit volume if plotted will on the average lie within the confines of curves *a* and *b*, Fig. 1. If the first assumption is observed when applying the principle, the second assumption will be automatically satisfied in practically every case. Therefore, most accurate results can be expected when applying the principle to individual portions of the ventilating circuit, each consisting of a constant number of uniform intake or return airways separated from the adjacent return or intake airways by equally spaced crosscuts equipped with stoppings of the same general type of construction. The following example is an illustration.

Consider four parallel intake airways, each 4000 ft. long, 5 ft. high and 10 ft. wide. These intake airways form a portion of the intake circuit and lie parallel to a portion of the return circuit with interconnecting crosscuts uniformly spaced equipped with stoppings of similar but questionable construction. It is desired to deliver 100,000 cu. ft. of air per minute at the inby end of that section of intake circuit in question. To do so, however, requires a volume input of 175,000 cu. ft. per min. at the outby end with the difference in the two volumes leaking from intake to return along the circuit. Disregarding leakage, the problem resolves itself into one of passing 100,000 cu. ft. per min. a distance of 4000 ft. by means of four parallel intake air courses each of 50 sq. ft. section area (A) and 30 ft. perimeter (O). The resulting air velocity (V) in each entry is therefore 500 ft.

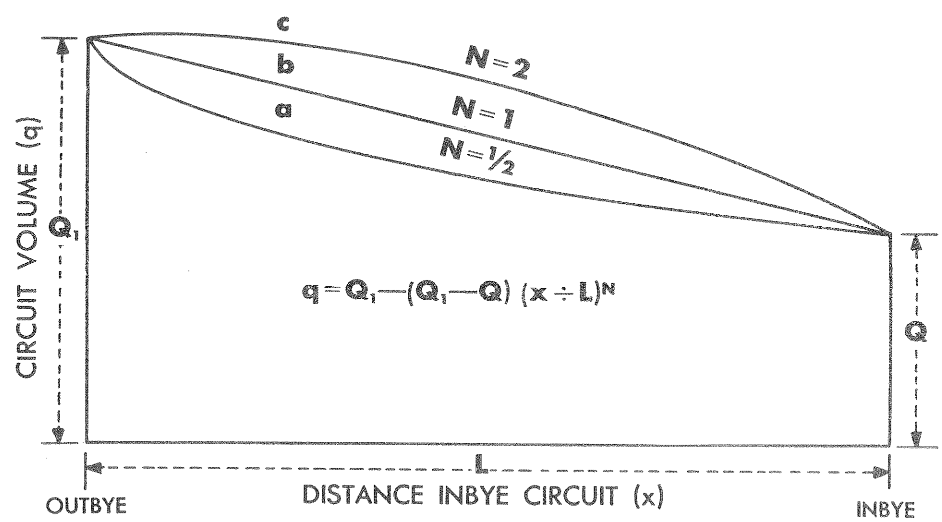


Fig. 1—One Intake and One Return Air Course of Equal and Uniform Section Area and Interconnected at Regular Intervals with Crosscuts Equipped With Stoppings.

Fig. 1

per min. Assuming the well-worn friction coefficient k of numerical value 0.0000000217, the corresponding pressure drop is 2.50 in. water gauge when calculated by the accepted pressure formula $P = \frac{kLOV^2}{5.2A}$. Since normally leakage is disregarded, a pressure drop of 2.50 in. water gauge would be accepted as correct by most investigators.

When considering the effect upon the pressure resulting from the anticipated underground stopping leakage some observers have suggested that the leakage through all of the stoppings be considered equal, resulting in a circuit volume decreasing at a uniform rate along the circuit between the limits of 175,000 and 100,000 cu. ft. per min. This condition is suggested by this author as a limiting case only and is represented by curve *b*, Fig. 1. For a volume ratio of 175,000 ÷ 100,000, or 1.75, it will be seen from curve *b* of Fig. 2 that a pressure ratio of 1.94 results from this assumption. The predicted pressure drop based upon the assumption of uniform stopping leakage becomes $1.94 \times 2.50 = 4.85$ in. water gauge. In the writer's opinion, this pressure is too high, because the assumption of uniform stopping leakage is unjustified by both theoretical and practical consideration. With stoppings of equal area and equal porosity uniform stopping leakage presupposes the same pressure differential across each stopping, which obviously is absurd. Furthermore, careful air-volume measurements taken at regular intervals along any underground circuit with leaky stoppings will always show a circuit volume decreasing at diminishing rate. When plotted along the circuit the circuit volume curves will usually be found to lie between curves *a* and *b*, Fig. 1.

It is therefore the writer's opinion that the pressure ratio can be considered equal to the volume ratio with a high degree of accuracy when correcting pressure for the effect of underground stopping leakage. In the example at hand, which is a volume ratio of 1.75, the true pressure ratio should lie between the limits of 1.59 and 1.94, as shown by curves *a* and *b*, respectively, of Fig. 2. Assuming the pressure ratio equal to the volume ratio results in the selection of a pressure-ratio value of 1.75, which predicts a pressure of $1.75 \times 2.50 = 4.38$ in. water gauge across that portion of the circuit considered in this example. This value should be closer to the truth than that arrived at by any other method so far proposed.

The validity of the relationship $P_1/P = Q_1/Q$, finds mathematical as well as experimental support. For instance, again referring to the simple circuit described in paragraph 4, it is first assumed that all stoppings are completely tight.

The circuit pressure P therefore is equal to the sum of the incremental changes dp, in pressure difference from outby to inby the circuit or

$$P = \int_p^0 dp = -2kQ^2 \int_0^L dx,$$

wherein L is the circuit length from outby to inby. Integrating and substituting limits gives the expression of circuit pressure, $P = 2kQ^2L$.

Again consider the same simple mine circuit previously described, with the exception that the stoppings will now be considered to leak air as occurs in actual practice. Traveling from outby to inby the circuit, at any point X, the pressure difference between intake and return air courses will decrease at the rate $dp/dx = -2kq^2$, wherein k is the circuit coefficient of unit resistance as before but q is the value of the circuit volume at X, varying between the

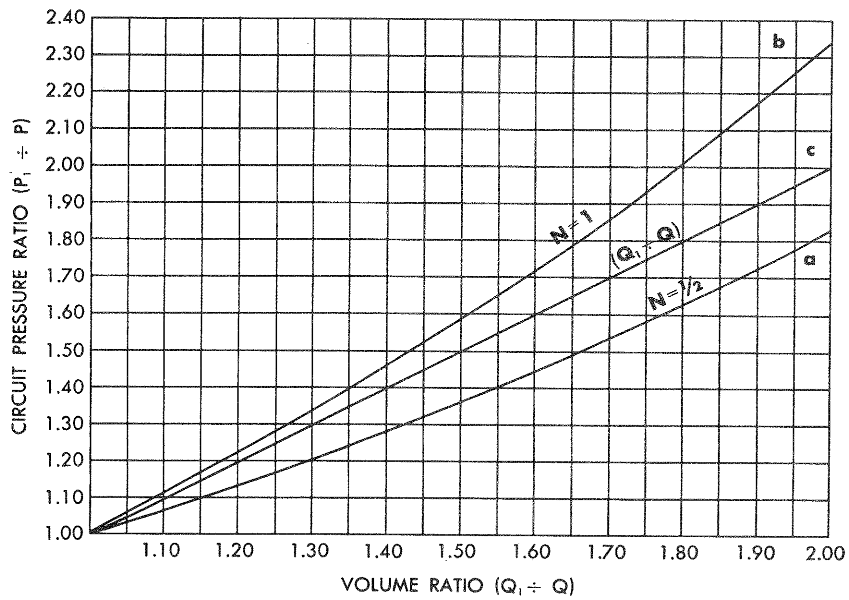


Fig. 2—
Circuit-Pressure Curves

limits of Q_1 completely outby and Q completely inby the circuits.

Again, the circuit pressure P_1 is equal to the sum of the incremental changes dp in pressure difference from outby to inby the circut, or

$$p = \int_{p_1}^{0} dp = -2k \int_{0}^{L} q^2 dx.$$

After substituting for q its value in terms of X, integrating and inserting limits, the following formula for circuit pressure results:

$$P_1 = 2kL[Q_1^2 - 2Q_1(Q_1 - Q) \div (N+1) + (Q_1 - Q)^2 \div (2N+1)]$$

The same formula can be more conveniently expressed by making use of the volume ratio $Q_1 \div Q = R$ and substituting RQ for Q_1, which results in the formula for circuit pressure:

$$P_1 = 2kLQ^2[R^2 - 2R(R-1) \div (N+1) + (R-1)^2 \div (2N+1)]$$

Circuit-pressure ratio, with and without stopping leakage, is shown by the formula:

$$P_1 \div P = R^2 - 2R(R-1) \div (N+1) + (R-1)^2 \div (2N+1)$$

Curves *a* and *b*, Fig. 2, plot values of circuit-pressure ratio, $P_1 \div P$, as ordinates versus volume ratio, $R = Q_1 \div Q$, as abscissas for values of $N = 0.50$ and $N = 1.00$, respectively.

The assumption of values for N between the limits of 0.50 and 1.00 is in close agreement with practical observation. Furthermore, the curves in Fig. 1 show that *n* must have some value less than unity if the circuit volume decreases with diminishing rate from outby to inby.

Therefore, since it is impossible to evaluate N more accurately than to ascribe limits thereto of 0.50 and unity, the relation of the *c* curve to the *a* and *b* curves of Fig. 2 suggests that the use of this *c* curve is an accurate and useful means of correcting circuit pressure for the effects of underground stopping leakage.

Appendix C

Determination of most economical airshaft size

To determine the optimum inside dimension of an airshaft, it is necessary to strike the proper balance between the cost of power for air friction and turbulence losses within the airshaft, on the one hand, and the cost of sinking the airshaft, on the other.

An airshaft larger than necessary is extravagant because the increased amortization rate exceeds the reduction in power cost. Conversely, an airshaft smaller than is needed wastes more power than can be justified by the corresponding reduction in amortization rate.

The airshaft power is a function of the air volume Q cu. ft. per min. and the air-pressure drop p in. water gauge across the airshaft. The airshaft pressure drop is calculated by the formula:

$$p = \frac{R \times l \times O \times Q^2}{5.2 \times A^3}$$

wherein R is the air-friction coefficient for the portion of the airshaft l ft. deep, O ft. in perimeter and A sq. ft. in area, when passing an air volume of Q cu. ft. per minute.

The airpower w kw. dissipated within the airshaft is computed by the formula:

$$w = \frac{0.746Q \times p}{6345}$$

With a combined over-all unit efficiency of the fan and drive of E per cent, the airshaft power requirements become

$$W = \frac{100w}{E}$$

There are 8760 hours per year, so with energy at 1 cent per kilowatt-hour the annual cost per kilowatt for continuous operation is \$87.60; therefore, with energy at C¢ per kw-hr. the annual airshaft power cost M_1 dollars per year becomes:

$$M_1 = 87.60W \times C$$

which, expressed by the component parts, becomes:

$$M_1 = \left(\frac{K \times l \times O \times Q^2}{5.2 \times A^3}\right) \left(\frac{Q \times 0.746 \times 100 \times 87.60 \times C}{6345 \times E}\right)$$

The amortization rate a dollars per year per dollar invested includes both depreciation and interest on the investment. These in turn involve the expected useful life of the airshaft and available monetary interest rate.

For the purpose of this discussion the cost of airshaft sinking is assumed to comprise a fixed move-in charge Y dollars, which covers the cost of transporting, erecting and removing the shaft-sinking equipment and is independent of the airshaft size. Also, it is assumed that the cost of shaft lining, if any, is fixed at \$$X$ per cu. yd. of lining material and that excavation costs \$$c$ per cu. yd. of excavated material. The thickness of the shaft lining is assumed to be one tenth the inside shaft optimum dimension, D ft. as calculated.

Having determined the amortization rate \$$a$ per yr. per dollar invested, the yearly cost of amortization M^2 becomes the sum of lining, excavation and move-in costs.

$$M_2 = \left(\frac{(A_1 - A)\, l_1 \times X \times a}{27}\right) + \left(\frac{(A_1 \times l_1 + A \times l)c \times a}{27}\right) + Y \times a$$

wherein the prime subscript refers to the lined portion of the airshaft.

The total annual cost \$$M$ per yr. is, therefore, the sum of airshaft power and amortization costs:

$$M = M_1 + M_2$$

Optimum Size for Given Shape

To determine the optimum size of airshaft of a given shape, the annual cost M is expressed in terms of an inside principle dimension D, which is a variable. The first derivative of M with respect to D when equated to zero permits solution for the optimum value of D for which M is a minimum.

The Appendixes (P. 33, 34) give the derivation of the formulas for optimum inside airshaft dimension D for circular airshafts, also elliptical, and rectangular airshafts with fixed ratio of sides k, respectively.

It is common practice to select the size of a proposed airshaft by arbitrarily assuming a permissible air speed in the shaft. The following examples will serve to demonstrate the fallacy of such practice and the usefulness of the formulas shown in the Appendixes when deciding upon the proper size of a proposed shaft.

For example, an airshaft is desired for an air volume Q of 200,000 cu. ft. per min. The total depth of the shaft is to be 600 ft. The amortization rate a, based upon 20 yr. expected life and 3 per cent average interest rate, becomes \$0.08 per dollar invested per year. Electric energy for fan operation C is available at 0.8c per kw-hr. and the average over-all efficiency of the ventilating equipment E is assumed to be 60 per cent. Excavation cost c is \$15 per cu. yd. and the cost of concrete lining X is \$30 per cu. yd. of lining material.

The problem will be analyzed on the basis of completely lined, half lined and completely unlined airshafts of circular, elliptical and rectangular design.

The elliptical and rectangular shafts will be selected with a side ratio k of 0.50, which means that the short dimension will be one half the long dimension. Moreover, air-friction coefficients R_1 and R of 0.0000000050 and 0.0000000100 are assumed for the lined and unlined shaft portions, respectively.

Application of the formulas gives the results listed in Table 1.

The wide spread in optimum air speeds should be carefully noted. The principal factor influencing area is type and amount of lining. Shape also influences optimum shaft area, but much less than does lining. This example demonstrates the fallacy of determining the area of a proposed airshaft by blindly selecting what appears to be a reasonable air speed without considering type and amount of lining or shape of the airshaft.

Another point of interest as demonstrated by Table 1 is that whereas for unlined airshafts the circular shape can have the least area, the added cost for lining might mean less optimum area for the elliptical or rectangular airshafts, because the greater ratio of shaft periphery to shaft area—and consequently the greater lining cost per unit area for elliptical and rectangular shafts—out-weighs the more favorable hydraulic radius ratio of the circular shaft.

The annual costs of the airshafts mentioned including power and amortization exclusive of the move-in charge, which would be approximately the same regardless of the shape of the shaft, are listed in Table 1.

Regardless of shape, unlined airshafts are cheaper than lined airshafts, in spite of the greater area required to compensate for the rougher surface.

For a specific depth of lining, a circular airshaft is most economical, elliptical next, and rectangular least.

Rectangular Shaft

Other factors than economy of ventilation sometimes determine the shape of a proposed airshaft. When the shaft is to carry an emergency hoist, or when the air compartment is one compartment of a multicompartment shaft, it is usually desirable to adopt an approximately elliptical or a rectangular shaft instead of circular. The dimensions of the compartment or compartments other than the one solely for air are usually determined by space limitation, and are thereby fixed. The problem of determining the optimum size of air compartment then becomes one of determining the optimum value for the variable side of a rectangular airshaft with the second dimension fixed. Because of mathematical complications, this problem is most easily solved by calculating the optimum sizes of rectangular shafts of different side ratios k by the use of the formula of Appendix 3 until one size is found to include one side dimension that sufficiently approximates the required fixed side.

For example, in the case of the problem considered in computing the data of Table 1, let it be assumed that the upcast air compartment, to handle 200,000 cu. ft. per min., should be one compartment of a two-compartment rectangular shaft, divided by an airtight masonry curtainwall, of which the second compartment must be 9 by 9 ft. in order to accommodate an emergency hoist. Such a shaft should be completely lined from top to bottom and equipped with a strong, tight curtainwall separating the two compartments.

Table 1 discloses that for a side ratio k of 0.50, the dimension of the optimum rectangular airshaft should be 13.75 by 6.87 ft. However, the problem now is to select the optimum value for the second side of a rectangular airshaft with one side fixed at 9 ft. It is obvious that side ratios k greater than 0.50 must be explored.

By the use of the formula of Appendix 3, inserting values of k of 0.70, 0.80 and 0.90 gives optimum airshaft sizes of 11.85 by 8.30 ft., 11.05 by 8.85 ft. and 10.55 by 9.50 ft., respectively. Obviously the optimum size of air compartment would be closely approximated with dimensions of 11 by 9 ft. with a lining and curtainwall thickness of approximately 13.2 in., which is 10 per cent of the 11-ft. side.

When both an intake and return airshaft are required to serve a particular part of a mine, the choice is between sinking two separate airshafts and sinking a single shaft equipped with a strong, airtight curtainwall, thereby providing both an intake and return air compartment.

The separate shaft plan is favored by both State and Federal mining departments on the grounds of freedom from air recirculation and increased insurance against losing both shafts in event of cave-in.

Table 1 — Dimensions and Annual Costs of Shafts

Dimensions and Costs	Circular			Elliptical			Rectangular		
	Lining, (Per Cent Depth)			Lining, (Per Cent Depth)			Lining, (Per Cent Depth)		
	100	50	0	100	50	0	100	50	0
Dimensions:									
Finished, ft.	11.15	12.50	13.85	15.25 × 7.62	17.15 × 8.57	19.75 × 9.87	13.75 × 6.87	15.45 × 7.72	17.80 × 8.90
Lining thickness, in.	13.3	15.0		18.3	20.6		16.5	18.5	
Area, sq. ft.	97.5	124.0	151.0	91.5	115.5	153.5	94.5	119.5	158.5
Air speed, ft. per min.	2050	1625	1325	2190	1695	1310	2110	1675	1260
Costs:									
Excavating	$47,000	$50,000	$50,200	$51,250	$51,500	$51,250	$53,000	$53,500	$53,000
Lining	28,800	18,100		41,500	26,300		42,850	27,100	
Excavating and lining	75,800	68,100	50,200	92,750	77,800	51,250	95,850	80,600	53,000
Excavating and lining per yr.	6,050	5,445	4,020	7,420	6,230	4,100	7,670	6,450	4,230
Shaft power per yr.	2,330	1,950	1,600	2,970	2,480	1,630	3,100	2,580	1,695
Total per yr.[a]	8,380	7,395	5,620	10,390	8,710	5,730	10,770	9,030	5,925

[a] Exclusive of move-in cost.

Cost of Two-compartment Shaft

It is commonly believed, however, that two separate airshafts are more expensive than one large two-compartment shaft with a curtain-wall. This belief is usually erroneous.

When selecting the optimum size of two-compartment air shaft, it is sufficiently accurate to treat the problem as involving two separate rectangular airshafts, lined from top to bottom. This procedure might be criticized on the ground that it pre-supposes four lined walls per shaft or a total of eight lined walls, whereas actually there are only seven lined walls, since the curtainwall is common to both compartments. To offset this apparent defect, however, is the fact that curtainwall construction costs more per cubic yard of material than sidewall lining; therefore the superfluous material, assumed, offsets the extra construction cost of the curtainwall.

For example, consider the problem of providing either two separate airshafts or one two-compartment shaft, 600 ft. deep to handle 200,000 cu. ft. per min. of both intake and return air with excavation and lining costs, amortization rate, power cost and equipment efficiency, the same as were assumed when preparing Table 1.

If the strata permit, two unlined circular shafts, each 13.85 ft. in diameter, could be sunk, which would bear an annual cost (excluding move-in cost) of $5620 per shaft (Table 1) or a total annual cost of $11,240. If lining is required, two circular shafts, each 11.15 ft. in diameter, could be provided for an annual cost of $8380 per shaft, or $16,760 for the two shafts.

On the other hand, a two-compartment rectangular shaft would require lining from top to bottom, regardless of the strata. Therefore the cost of such a shaft, having each compartment 13.75 by 6.87 ft., would be approximately the same as the cost of two individually lined, rectangular shafts of the same dimensions, or $10,770 per year per shaft, which is $21,540 per year for the complete shaft.

Therefore, contrary to popular belief, two separate airshafts are in almost every case cheaper than one two-compartment shaft, and should be given preference.

Summary

Considering the cost of an airshaft to be the annual sum of power cost and amortization rate, there is an optimum size of airshaft of any particular shape that provides the most economical possible shaft for a specific duty.

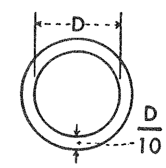
Arbitrary air-speed limitation is a risky yardstick for selecting the proper size of a proposed airshaft. Regardless of shape, unlined shafts should be larger than lined or partly lined airshafts. However, unlined shafts are more economical than lined shafts, in spite of larger size.

Whether lined or unlined, circular airshafts are the cheapest, elliptical shafts more costly, and rectangular shafts the most expensive.

When providing both intake and return air openings for a remote mine section, it is usually cheaper to sink two separate field airshafts than one two-compartment shaft with a curtainwall separating the two compartments. This conclusion is compatible with recommendations of both State and Federal mining departments.

Appendix 1.—Derivation of Formula for Circular Shaft

$$M = \overbrace{\left[\frac{(R_1 l_1 + R \cdot l)\pi \cdot D \cdot Q^2}{5.2\left(\frac{\pi D^2}{4}\right)^3}\right] \cdot \left[\frac{Q \cdot 0.746 \cdot 100 \cdot 87.60 \cdot C}{6345 \cdot E}\right]}^{\text{Power cost 1 yr.}}$$



$$+ \overbrace{\left[\frac{\pi}{4}\left[(1.2D)^2 - D^2\right]\frac{l_1 \cdot X \cdot a}{27}\right]}^{\text{Lining cost 1 yr.}} + \overbrace{\left[\frac{\pi}{4}\left[(1.2D)^2 \cdot l_1 + D^2 \cdot l\right]\frac{c \cdot a}{27}\right]}^{\text{Excavation cost 1 yr.}} + Y \times a \text{ (move-in cost per yr.)}$$

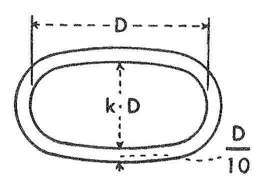
$$\frac{dM}{dD} = O = -\frac{2.04(R_1 \cdot l_1 + R \cdot l)\pi \cdot Q^3 \cdot C}{D^6 \cdot E} + \frac{44 \cdot \pi \cdot D \cdot l_1 \cdot X \cdot a}{54} + \frac{\pi \cdot D(1.44 l_1 + l)c \cdot a}{54} + O$$

$$D = 1.960\sqrt[7]{\frac{Q^3 \cdot C(R_1 \cdot l_1 + R \cdot l)}{E \cdot a(0.44X \cdot l_1 + 1.44c \cdot l_1 + c \cdot l)}}$$

Appendix 2.—Derivation of Formula for Elliptical Shaft

$$K = (1 + k)\left[1 + \frac{1}{4}\left(\frac{1-k}{1+k}\right)^2 + \frac{1}{64}\left(\frac{1-k}{1+k}\right)^4 + \frac{1}{256}\left(\frac{1-k}{1+k}\right)^6 \cdots\right]$$

$$M = \overbrace{\left[\frac{(R_1 \cdot l_1 + R \cdot l)\pi \cdot D \cdot K \cdot Q^2}{2 \cdot 5.2\left(\frac{\pi k D^2}{4}\right)^3}\right] \left[\frac{Q \cdot 0.746 \cdot 100 \cdot 87.60 \cdot C}{6345 \cdot E}\right]}^{\text{Power cost 1 yr.}}$$



$$+ \overbrace{\left[\frac{\pi \cdot D^2(0.2k + 0.24)l_1 \cdot X \cdot a}{108}\right]}^{\text{Lining cost 1 yr.}} + \overbrace{\left[\frac{\pi \cdot D^2 \cdot c \cdot a[(1.2k + .24)l_1 + k \cdot l]}{108}\right]}^{\text{Excavation cost 1 yr.}}$$

$$+ Y \times a \text{ (move-in cost per yr.)}$$

$$\frac{dM}{dD} = O = -\frac{1.023\pi \cdot Q^3 \cdot C(R_1 \cdot l_1 + R \cdot l)K}{D^6 \cdot E \cdot k^3} + \frac{\pi \cdot D \cdot (.2k + 0.24)l_1 \cdot X \cdot a}{54}$$

$$+ \frac{\pi \cdot D[(1.2k + 0.24)l_1 + k \cdot l]c \cdot a}{54} + O$$

$$D = 1.775\sqrt[7]{\frac{Q^3 \cdot C(R_1 \cdot l_1 + R \cdot l)K}{E \cdot a[0.2k + 0.24)X \cdot l_1 + (1.2k + 0.24)c \cdot l_1 + k \cdot c \cdot l]k^3}}$$

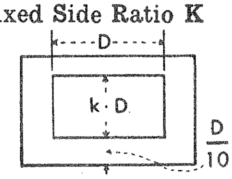
Substituting $(1 + k)$ for K gives the following close approximation:

$$D = 1.775\sqrt[7]{\frac{Q^3 \cdot C(R_1 l_1 + R \cdot l)(1 + k)}{E \cdot a[0.2k + 0.24)X \cdot l_1 + (1.2k + 0.24)c \cdot l_1 + k \cdot c \cdot l]k^3}}$$

Appendix 3.—Derivation of Formula for Rectangular Shaft, Fixed Side Ratio K

$$\overbrace{\left[\frac{(R_1 \cdot l_1 + R \cdot l) \cdot 2 \cdot D(1 + k)Q^2}{5.2(k \cdot D^2)^3}\right] \cdot \left[\frac{Q \cdot 0.746 \cdot 100 \cdot 87.60 \cdot C}{6345 \cdot E}\right]}^{\text{Power cost 1 yr.}}$$

$$+ \overbrace{\left[\frac{D^2(2k + 0.24)l_1 \cdot X \cdot a}{27}\right]}^{\text{Lining cost 1 yr.}} + \overbrace{\left[\frac{D^2 \cdot c \cdot a[(1.2k + 0.24)l_1 + k \cdot l]}{27}\right]}^{\text{Excavation cost 1 yr.}} + Y \times a \text{ (move-in cost per yr.)}$$



$$\frac{dM}{dD} = O = -\frac{1.975(R_1 \cdot l_1 + R \cdot l)Q^3 \cdot C(1 + k)}{D^6 \cdot E \cdot k^3}$$

$$+ \frac{(0.2k + 0.24)D \cdot l_1 \cdot X \cdot a}{13.5} + \frac{[(1.2k + 0.24)l_1 + k \cdot l]D \cdot c \cdot a}{13.5} + O$$

$$D = 1.600\sqrt[7]{\frac{Q^3 \cdot C(R_1 \cdot l_1 + R \cdot l)\,(1 + k)}{E \cdot a[(0.2k + 0.24)X \cdot l_1 + (1.2k + 0.24)c \cdot l_1 + kc \cdot l)k^3}}$$

Nomenclature

A, finished insider airshaft area, sq. ft.
a, amortization rate, dollars per year per dollar cost of airshaft.
C, cost of electric energy, cents per kw-hr.
c, excavation cost, dollars per cu. yd. excavated.
D, inside diameter of circular airshaft or inside dimension of long side of either elliptical or rectangular airshaft, ft.
E, over-all efficiency of ventilating equipment from electricity to air, per cent.
k, ratio of short to long inside dimensions of either elliptical or rectangular airshaft.
l, depth of unlined portion of airshaft, ft.
l_1, depth of lined portion of airshaft, ft.
M_1, cost of electric energy, dollars per year.
M_2, total annual amortization, dollars per year.
M, total annual airshaft cost including both electric energy and amortization, dollars per year.
O, inside perimeter of airshaft, ft.
p, air-pressure drop across airshaft, in. of water.
Q, airshaft design air volume, cu. ft. per min.
R, air-friction coefficient for unlined airshaft.
R_1, air-friction coefficient for lined airshaft.
W, electric power, kw.
w, air power dissipated within airshaft, kw.
X, lining cost, dollars per cu. yd. lining material.
Y, move-in cost, dollars.

Appendix D

Use of tubing and blowers for auxiliary face ventilation

The use of blower fans and tubing for auxiliary face ventilation is a somewhat controversial subject. Most authorities agree upon the advantages of this system over the maintenance of line brattice for dead-end ventilation. They also recognize the system's disadvantages. However, there is considerable disagreement as to the proper relation of the advantages to the disadvantages.

It is the purpose of this report to present the opinions of four authorities of diversified occupation and permit the reader to form his own opinions on the subject.

Observers

Carel Robinson is a mining engineer consultant of wide repute, who has a background of operating experience. (1944)

Thomas G. Fear is a mine operator whose present position is that of chief engineer for the Hanna Coal Co. at St. Clairsville, Ohio. (1944)

E. H. Denny is chief of the Coal Mine Inspection of the U. S. Bureau of Mines, and is a very able engineer whose past experience well qualifies him to appraise this problem. (1944)

Cloyd M. Smith, editor of the periodical **Mechanization**, is a ventilation consultant. For a number of years he was a member of the Mining Department of the University of Illinois and has studied mine ventilation for many years. (1944)

Opinions

It is Mr. Robinson's opinion that closely spaced breakthroughs, although conducive to low gas accumulation past the last open crosscut during the driving of entries, are difficult to maintain and are constant sources of fugitive air. Jute or canvas brattice is suitable for entry driving with closely spaced breakthroughs, but blowers and tubing are required with breakthroughs on long centers.

Mr. Robinson points out that dust hazard is aggravated by mechanized mining, but usually can be controlled in adequately ventilated gassy mines with line brattice. In nongassy mines, however, blowers and tubing prove most effective for rapid removal of dust and smoke.

Mr. Robinson maintains that prejudice against blowers and tubing caused by improper use should not justify blanket condemnation of those properly installed and maintained.

On the other hand, Mr. Fear contends that blowers and tubing should be outlawed because of recirculation, interrupted operation, a feeling of false security and their past record of operation.

Mr. Denny refers to 900 Federal Inspection reports and 366 Bureau of Mines disaster reports, citing the low percentage of blowers with permissible electrical equipment, also the prevalence of intermittent blower operation, and the high percentage of installations that constitute definite fire hazard (Table I).

Five of the 1184 auxiliary fans in use were operated by compressed air and the remainder by electricity.

Only 40 of the 1184 auxiliary fans had a type of motor and control rated by the Bureau of Mines as permissible. This permissible rating means that such motor and control if properly maintained would not ignite gas if inadvertently operated in an explosive mixture of methane and air. Over 95 per cent of all the auxiliary fans reported in use had nonpermissible motors and controls, and constitute potentially dangerous sources of ignition if operated in explosive mixtures.

Of the 1184 fans, 232 were reported as operated continuously, and 936 intermittently. The facts as to continuous or intermittent operation could not be determined for the remaining 16 fans.

"Intermittent operation" usually means

Table I—Figures from Federal Reports

Federal Inspection District	State	Number of Inspection Reports Studied	Mines Using Auxiliary Fans		Number of Auxiliary Fans Used
			Number	Per Cent	
A	West. Pennsylvania, Ohio, Northern W. Virginia, Maryland	381	47	12.3	252
B	East. Pennsylvania, Anthracite	27	18	66.7	258
C	Southern W. Virginia, Virginia, Eastern Kentucky	197	35	17.8	156
D	Tennessee, No. Carolina, Alabama	47	14	29.8	59
E	Indiana, Western Kentucky, Illinois	178	30	16.9	230
F	Michigan, North Dakota	9	3	33.3	7
G	Arkansas, Oklahoma	11	4	36.4	18
H	Colorado, Utah, Washington, Wyoming	50	26	52.0	204
Total		900	177	19.7	1,184

operation only during the time the regular shift is working and cessation of fan operation during the off shift. The term also includes operation only when work is going on at the particular face ventilated by means of the fan and tubing.

Also, all auxiliary fans, whether listed as being operated continuously or intermittently, are subject to stoppage because of local or general mine power interruptions. In some cases the normal ventilation of the working faces by crosscuts and brattices is considered adequate by the mine managements and the fans and tubing are used intermittently for the more rapid dispersal of the smoke and gases developed by blasting.

Mr. Denny emphasizes the number of cases of air recirculation caused by improper location of

Table 2 — Gas Ignitions Causing Fatal or Serious Injuries during the Fiscal Year 1943

State	Number of Explosions	Mines Using Auxiliary Fans	Men Killed	Men Injured
Alabama	3	0	12	8
Arkansas	4	2	5	9
Colorado	1	1	3	0
Illinois	2	0	1	7
Kentucky	6	2	10	15
Montana	1*	0	74	0
Ohio	3	1	7	1
Oklahoma	2	2	7	10
Pennsylvania	3	1	1	15
Tennessee	1	1	10	2
Virginia	1	0	0	4
West Virginia	3	0	21	2
Total	30	10	151	73

* Booster fans used.

blower. He protests against improper tubing installation, permitting constrictions and leakage, as well as instances of blower use without tubing or with tubing terminating too far from the face. Such factors account for the amount of poor face ventilation referred to in the inspection reports.

Mr. Denny states that whereas proper company rules regulating the installation and maintenance of blowers and tubing could prevent recirculation, evidence indicates that such regulation is lacking or is unobserved.

Mr. Denny points out that in 52 instances of gas ignition and mine explosion resulting in injury or loss of life, misuse of blowers and tubing contributed to gas accumulation and ignition, and the use of the equipment was accompanied by inadequate face ventilation, bad air courses, inadequate air at the main fan, or other ventilation short cuts to speed production or cut costs. Mr. Denny cites ten examples of disaster in which blowers and tubing are claimed to have played a contributing part (Table 2).

Since the percentage of inspected mines using blowers (19.7) is less than the percentage of blown-up mines using blowers (33⅓) Mr. Denny concludes that the use of blowers and tubing may present an increased hazard.

Mr. Denny next quotes the Bureau's 1927 recommendations regarding the use of blowers, which relegate their use to emergency work such as driving a cross heading in rock to make connection with distant workings and where distances are excessive for line brattice, and he proceeds to stipulate proper methods of installation and operation. Mr. Denny also refers to the paper by Greenwald and Howarth,* which outlines proper installation precautions for blowers and tubing.

Mr. Denny concludes that the fire hazards due principally to the manner of installation or use of blowers and tubing, plus the examples of explosions and ignition cited where use of these fans is said to have played a contributing part, justifies disapproval by the Bureau of the use of such equipment.

Mr. Smith first differentiates between primary and auxiliary types of ventilation, explaining the greater dependence upon auxiliary ventilation in metal mines than in coal mines, which he explains to be the result of better ventilation in the latter than in the former.

He recognizes the necessity of both safety and effectiveness in the use of auxiliary blowers and tubing in coal mines and states that these objectives are compatible. He emphasizes the necessity of blower location so as to avoid air recirculation and the handling of dust or gas-contaminated air. Also, he insists upon the use of approved and permissible electrical equipment and continuous blower operation and emphasizes the importance of proper precautions, such as suspension of tubing, freedom from kinks, bends, etc.

Mr. Smith explains the advantages of blowers and tubing for driving entries and maintaining crosscuts on long centers, also for reducing the number of crosscuts between conveyor-mined rooms. He again cautions against placing room blowers so as to permit recirculation or dust contamination.

Observations

Whereas these four reports express differences of opinion as to the relative advantages and disadvantages accompanying the use of tubing and blowers for auxiliary face ventilation, they are in complete agreement as to certain basic precautions that should be employed when such equipment is used. For

* H. P. Greenwald and H. C. Howarth: Recirculation of Air and Mine Gas Caused by Auxiliary Fans as Used in Coal Mines *Trans.* A.I.M.E. (1928) 76, 164.

example, all observers agree that such blowers should be powered with permissible Government-approved motors and that proper ground connections should be employed. Furthermore, the blowers should be kept in continuous operation, to avoid high concentration of gas deposits during shutdown periods. It is important also that the blower be placed on fresh air in such a manner as to exclude all possibility of recirculation of air already handled by the blower.

The main justification for the use of tubing and blowers for ventilating dead ends is where the installation and maintenance of line brattice is rendered impractical, either because of long distance desired between crosscuts or the operation of mechanical equipment with space limitations.

By presenting the diversified opinions of four competent observers it is felt that the material contained in this report will prove of most value to the mine operator who is interested in the use of this particular class of equipment.

Appendix E

Pitot-tube field tests of vaneaxial mine fans

A test of any fan requires the determination of such data as fan pressure, air volume handled by the fan, and power input to the fan shaft.

When testing operating mine fans of the centrifugal type, test methods have been employed, for the sake of convenience, that permit great latitude in detail of procedure resulting in wide variance in results by different investigators.

Some controversial aspects of field fan testing are the result of past failure to define clearly the various forms of fan pressure and to a lack of understanding of the accurate procedure for the measurement thereof. Also, the anemometer for air-volume measurement, although convenient, if carelessly used and maintained is subject to a wide error range with variation in method of application and in instrument accuracy.

The appearance of the vaneaxial type of mine fan offers the possibility of refinements in field testing that were impractical with the centrifugal fan. First of all, the fan inlet and outlet are more sharply defined, giving closer agreement as to just where the fan begins and ends; thus there is less disagreement as to the proper location for measurements of fan pressure.

The area of passage into the inlet of a vaneaxial mine fan usually constricts, causing high air velocities with accompanying steadiness of flow. Thus the fan inlet offers an ideal location for accurate air-volume measurements with the Pitot tube, the proper use of which is well established by code. Thereby the controversial aspects of such measurements are eliminated. Use of the Pitot tube also facilitates a more accurate measure of fan pressure, since the individual pressures measured can be properly weighted, as will be explained later.

Determination of power input to the vaneaxial mine fan operating in the field is subject to the same inaccuracies as with the centrifugal fan if the vaneaxial fan is also belt driven. However, because of the higher rates of rotation usual with vaneaxial fans, direct-driven units are more numerous and it is unnecessary to assume a drive efficiency, thus eliminating one possible source of error.

It is the purpose of this paper to define and explain the various types of pressure measurements involved in a field test of a vaneaxial mine fan, and to further explain the proper use of the Pitot tube when measuring pressures and volumes.

Definition of Fan Pressures

The two fan pressures most commonly discussed are fan total and fan static pressure, each of which has the same meaning regardless of whether the fan is blowing, exhausting or blowing and exhausting, as in the case of a fan with ducts connected to both outlet and inlet.

Fan total pressure is the amount by which the total pressure of the air delivered by the fan exceeds the total pressure of the air received by the fan. This pressure is a measure of the maximum amount of the work done on the air by the fan that is available for external use.

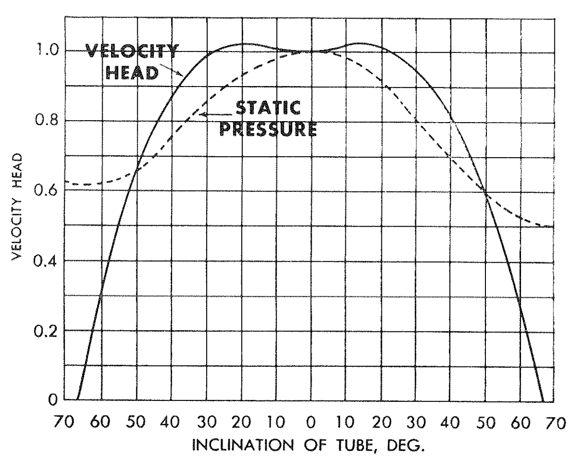


Fig. 1—Pitot Tube With Static Holes At Sides, Top and Bottom.

For a blowing fan, fan total pressure is the total gauge pressure obtained by traversing the fan outlet with a facing Pitot tube connected by a hose to one leg of a manometer of which the other leg is open to the atmosphere.

For an exhaust fan, fan total pressure is the algebraic difference between the total gauge pressures obtained by traversing both the fan outlet and inlet with a facing Pitot tube connected by hose to one leg of a manometer of which the other leg is open to the atmosphere.

With a fan both blowing and exhausting (such as a fan underground), fan total pressure is the algebraic difference between the total pressures obtained by traversing both the fan outlet and inlet with a facing Pitot tube connected by hose to one leg of a manometer of which the other leg is exposed to the same pressure during each traverse.

Fan static pressure is the amount by which the static pressure of the air delivered by the fan exceeds the total pressure of the air received by the fan. This pressure is a measure of the minimum amount of the work done on the air by the fan that is available for external use.

With a blowing fan, fan static pressure is the static gauge pressure obtained by traversing the fan outlet with a static Pitot tube connected by a hose to one leg of a manometer of which the other leg is open to the atmosphere.

For an exhaust fan, fan static pressure is the algebraic difference between the static gauge pressure at the fan outlet and the total gauge pressure at the fan inlet as obtained by traversing the fan outlet and inlet with a static and facing Pitot tube, respectively, connected by a hose to one leg of a manometer of which the other leg is left open to the atmosphere.

With the fan installed below ground, the fan static pressure is the algebraic difference between the static pressure at the fan outlet and the total pressure at the fan inlet as obtained by traversing the fan outlet and inlet with a static and facing Pitot tube, respectively, connected by a hose to one leg of a manometer of which the other leg is exposed to the same pressure during each traverse.

Mine ventilating pressure is the irreducible minimum pressure across a mine required to maintain the desired ventilation. Regardless of the fan location, mine ventilating pressure accounts for all underground pressure losses as well as the loss in the air at the mine exit. With a blowing or underground fan, the exit loss generally consists of velocity pressure only. With an exhaust fan installation, however, the exit loss, if any, is limited to the type of loss associated with changing the direction of air flow at a bend.

For a mine ventilated by a blowing fan, mine ventilating pressure is the total gauge pressure measured just inside the mine entrance, as at the airshaft collar or drift mouth. With the fan exhausting, mine ventilating pressure is the total gauge pressure measured just outside the mine exit in front of the fan. With the fan below

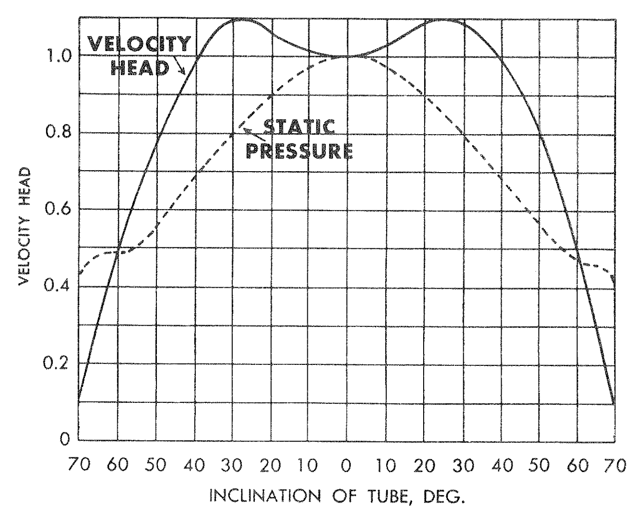


Fig. 2—Brabbee Tube With Static Orifices at the Extremities of Two Diagonals.

ground, mine ventilating pressure is the total pressure difference in the air just ahead of and behind the fan.

Usually, the mine entrance, exit or airway, exceeds in section area the inlet or outlet of a propeller mine fan. Therefore, regardless of the fan location, the air approaching the fan can be expected to undergo accelerated flow, whereas the air leaving the fan will be subjected to retarded flow. The efficiency of the former process justifies neglect of the accompanying pressure losses; however, the latter process—namely, retarded flow—involves pressure losses that must be accounted for.

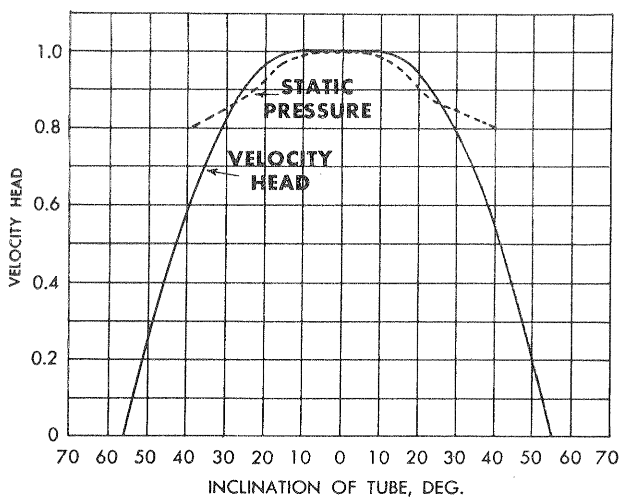


Fig. 3—Prandtl Tube

The fan pressure most closely corresponding to the mine ventilating pressure is obviously the logical reference pressure whenever the ventilating efficiency of a mine fan is considered. Fan total pressure is an unsuitable reference because algebraically it exceeds the mine ventilating pressure by an amount equal to the retarded flow-pressure losses following the fan outlet and, if the fan is blowing, by the bend loss in the airshaft hood as well. The possible magnitude of this discrepancy is best illustrated by considering the effect of eliminating the fan diffuser or stack. On the other hand, fan static pressure can be safely equated to the mine ventilating pressure because in this way the fan outlet velocity pressure is available to compensate for the other losses mentioned.

For field-test purposes, therefore, credit a blowing or underground mine fan with the true fan static pressure measured as already described. The exhaust fan when field-tested should be credited with the total gauge pressure measured by a facing Pitot tube traverse of the fan inlet. This pressure usually closely approximates fan static pressure and equals it exactly for the special case of air flow past the fan outlet entirely free from nonaxial velocity and nonaxial acceleration.

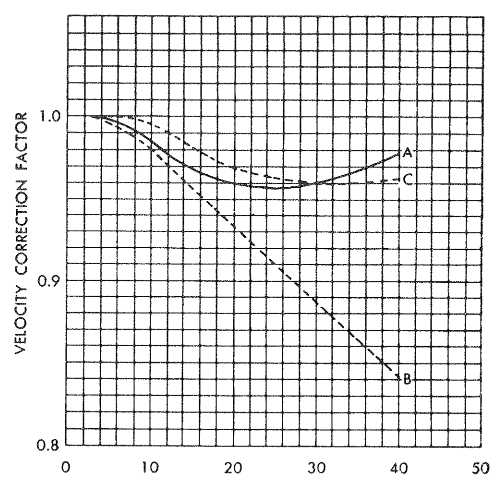


Fig. 4—Correction Factors.

A. Total correction factor for Prandtl tube.
B. Total correction factor for Pitot tube with static orifices at sides, top and bottom.
C. Total correction factor for A.S.H. and V.E. tube.

Pressure Measurements with Pitot Tubes

Air-pressure measurements are accurately and easily performed with the Pitot tube and manometer.

When true total pressure is measured at any point in an air stream, the Pitot tube should be directed so that the total pressure opening coincides with the point in question, and so that the tube points directly into the resultant air velocity at the point of measurement. Any angularity between the tube and the resultant air velocity is known as an angle of yaw, and has

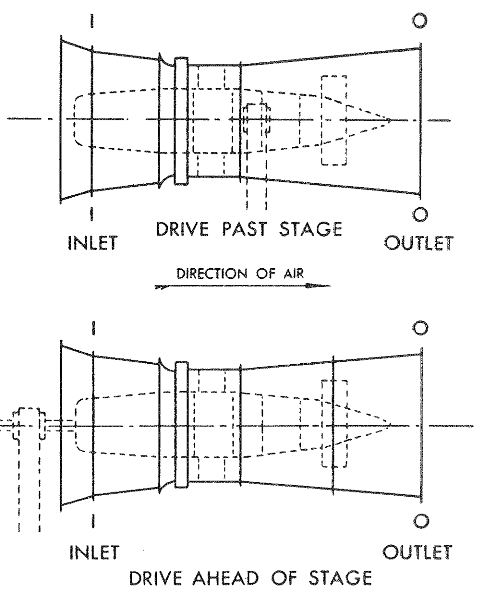


Fig. 5—Proper Location for Pitot Traverse Planes.

the effect of making the total pressure gauge read low. Figs. 1, 2 and 3 show the effects of yaw upon three different kinds of tubes when static or velocity pressure is being measured.

It is common practice to consider total pressure as the algebraic sum of the measured static pressure and the computed velocity pressure referred to the mean axial velocity normal to the plane in which the static pressure is measured. The accuracy of this assumption depends upon the uniformity of the axial velocities over the plane, as well as freedom from nonaxial velocities.

Nonuniform axial flow introduces error because the mean of the squared axial velocities exceeds the mean axial velocity squared. Nonaxial flow produces error because the nonaxial velocity pressure goes unaccounted for. In either case the error leads to an apparent total pressure that is too high or too low, depending upon whether the static pressure is negative or positive.

When determining the total pressure in the air immediately following a bend, or at the outlet of a fan, or at similar places, it is especially important that a complete total pressure traverse of the section be made with a facing Pitot tube. To ensure accuracy, the traverse is always essential unless it has been previously ascertained

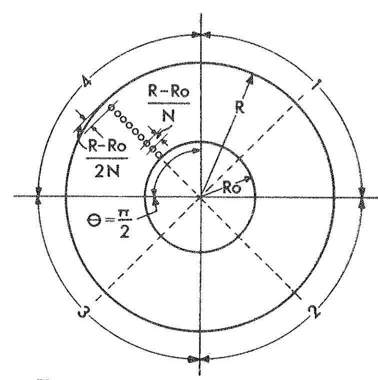


Fig. 6—Traverse Plane.

that uniform pure axial flow exists only across the plane of total pressure measurement.

When measuring static pressure at any desired point in an air stream, the Pitot tube should be placed with the static pressure holes in a plane that includes the point of measurement and is normal to the resultant air velocity at that point. The effect of yaw upon static-pressure measurement is shown by Figs. 1, 2 and 3.

Uniform static pressure over a plane of measurement is possible only when the air flow past the plane is purely axial and of either uniform or nonuniform velocity distribution. The static-

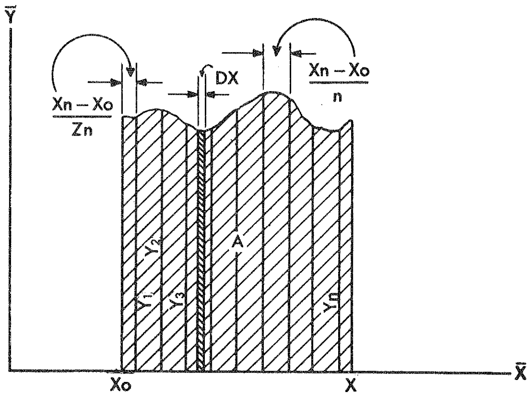


Fig. 7—Calculations for Volume and Pressure.

pressure distribution varies in value whenever nonaxial accelerations or velocities exist in the plane of static pressure measurement. Therefore, to obtain accurate measurements of static pressure at the outlet of any fan it is usually necessary to traverse the outlet with a static Pitot tube.

Total or static-pressure measurements at the inlet or outlet of a high-pressure propeller fan usually can be obtained accurately with a Pitot tube held parallel to the longitudinal fan axis, thereby neglecting the effect of yaw. This is so because the inlet plane of pressure measurement is bounded by almost parallel sides and is either in or immediately following a zone of axially accelerated air flow. The outlet plane of pressure measurement, although following a zone of gradually retarded axial air flow, is fairly free from nonaxial air velocities because of the action of properly designed guide or straightener vanes.

Air-volume Measurements with Pitot Tubes

Measurements of velocity pressure serve as a basis for computation of air volume and therefore are just as essential as measurements of fan pressure. The air velocity v at any point of measurement is expressed as $v = 1097.5 \sqrt{\frac{VP}{d}}$ in which VP represents the measured velocity pressure (inches water) and d is the specific weight of the air (lb. per cu. ft.). The velocity v therefore is expressed as feet per minute.

When measuring velocity pressure, the Pitot tube should be placed with the static-pressure holes in a plane that includes the point of desired velocity measurement and that is normal to the resultant air velocity at that point.

Actually, the Pitot tube measures the difference between the total pressure at the front of the tube and the static pressure at the static holes. The total pressure difference between the two points is negligible, as is likewise the change in direction of air flow. Therefore, the velocity pressure as measured with the Pitot tube is the resultant velocity pressure that exists in the plane of the static-pressure holes.

Similarly, the section area used to compute the air volume should be equal to the section area in the plane of the static holes, normal to the fan axis. The significant air velocity accordingly is the axial component of the resultant velocity measured as described above.

Fig. 4 shows the amount of error with each of three types of Pitot tubes arising from holding the tube parallel to the fan axis with the static holes in the plane of desired velocity measurement, and considering the velocity thus measured as the significant velocity. Such slight errors, when using either the Prandtl, or A.S.H. & V.E. tubes, justify this procedure during field tests.

Selection of Traverse Plane

At a location for pressure or volume measurements, the following conditions are desirable: (1) pure axial air flow, (2) uniform velocities at equal radii, (3) freedom from pulsations, (4) traverse on radial lines located at least three diameters distance from any obstruction in the plane of pressure measurement 10 diameters distance from any obstruction directly ahead, and three diameters distance from any obstruction immediately behind the radial traverse line. Seldom is it possible to strictly observe all of these desirable features, but the location that most nearly satisfies the desired conditions should be selected.

Air volume is measured preferably immediately ahead of the fan rotor, where the air is in accelerated flow and free from turbulence. When testing a pressure-type (blowing) mine fan, however, a period should be selected when there is little or no outside wind. Otherwise the inlet velocities will vary with time.

Fig. 5 shows general arrangement of fan with drive located past and ahead of the fan stage (rotor and vanes). The planes I-I are ideally suited for measurements of either volume or pressure at the fan inlet. The planes O-O correspond to the fan outlet at which it may be desired to measure total or static pressure.

System of Traverse

A suitable plane having been selected for pressure or volume measurement the plane should be divided into at least four equal-angle sectors and the bisector of each sector should be traversed for pressure, velocity, or both, as is desired. Fig. 6 shows such a traverse plane divided into four sectors and with N equally spaced stations selected along each traverse line. As a rule, it is good practice to select at least eight stations on each traverse line between the outer radius R and the inner radius R_0. The spacing of these stations should be determined as shown by Fig. 6.

Obviously, the pressure or velocity reading thus obtained at a particular radius on a particular traverse line represents the average value at that radius over the entire angle θ included within the sector.

Calculations for Volume and Pressure

The area under any curve such as A in Fig. 7 is equal to $\int_{X_0}^{X} y\,dx$, which in turn very nearly equals $\left(\frac{y_1 + y_2 \cdot \cdot \cdot y_n}{n}\right)(X - x_0)$ if n is sufficiently large.

The air volume q passing any sector of angle θ radians between radii R and R_0 is expressed by the equation $q = \int_{R_0}^{R} vr\theta dr$ in which v is the air velocity at radius r. Substituting q for A, R for X and $vr\theta$ for y, the following expression results:

$$q = \theta\left(\frac{v_1R_1 + v_2R_2 \cdot \cdot \cdot v_nR_n}{n}\right)(R - R_0)$$

By averaging the velocities in the different sectors at the same radius and expressing this average velocity as V, the final expression for the air volume passing the entire plane of traverse becomes:

$$Q = 2\pi\left(\frac{V_1R_1 + V_2R_2 \cdot \cdot \cdot V_nR_n}{n}\right)(R - R_0)$$

The rate of energy transfer w by the air passing a traverse section of angle θ radians between radii R and R_0 is expressed by the equation

$$w = \int_{R_0}^{R} p\,dq = \int_{R_0}^{R} pvr\theta dr,$$

in which p and v represent air pressure and velocity at radius r. Substituting w for A, R for X and $pvr\theta$ for y, we have the expression

$$w = \theta\left(\frac{p_1v_1r_1 + p_2v_2r_2 \cdot \cdot \cdot p_nv_nr_n}{n}\right)(R - R_0)$$

By again substituting the average values, in the different sectors, of pressure P and velocity V at each radius traversed R, the final expression results for the rate of energy transfer W by the total air volume Q passing the entire plane of section traverse:

$$W = 2\pi \left(\frac{P_1V_1R_1 + P_2V_2R_2 \cdot \cdot \cdot P_nV_nR_n}{n}\right)(R - R_0)$$

From the relation $W = PQ$ follows the equation

$$P = \frac{W}{Q} = \frac{2\pi}{Q}\left(\frac{P_1V_1R_1 + P_2V_2R_2 \cdot \cdot \cdot P_nV_nR_n}{n}\right)(R - R_0)$$

which is the general expression for the average pressure P at the plane of traverse.

Therefore, when measuring average pressure at any traverse plane it is necessary, for the purpose of proper weighting, that velocities be measured in addition to and simultaneously with the individual pressures.

Power Determination

The mechanical power delivered to the fan shaft is less than the electric power input to the motor terminals by an amount equal to the motor losses and drive losses.

The power input to the motor terminals should be measured directly with an indicating wattmeter or with a recording watt or watthour meter. It is bad practice to measure current and voltage and then attempt a "guess" at the power factor of an alternating-current motor.

The motor losses can be most closely approximated by the motor manufacturer upon receipt of such information as the power input to the motor, the voltage between the different phases and the current flow per phase.

If the fan is driven directly connected to the motor, the motor output can be considered equal to the fan input, thereby neglecting any possible loss in the coupling. If, however, the fan is driven through one or more drives it is necessary to deduct an estimated drive loss.

Flat-belt or V-belt drives with short centers are assumed to be 95 per cent efficient. The true efficiency varies with such factors as belt tension, alignment, load. However, practice has justified the efficiency assumption of 95 per cent, or a power loss of 5 per cent for each pair of pulleys required to drive the fan.

Conclusions

The careful application of the various procedures suggested herein will produce field-test results that are in close agreement with factory fan tests of laboratory precision. This is particularly true of the pressure-volume measurements and applies to a lesser degree to the power measurements, which involve efficiency estimates for motor and drive. However, most of the serious sources of error to which field fan tests have been subjected in the past are hereby eliminated or greatly reduced in magnitude.