COMPRESSED AIR PRACTICE IN MINING

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BY

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PREFACE TO SECOND EDITION

The First Edition of this book was published in the middle of the Great War and, notwithstanding the absence of Students from the College, it found a ready acceptance, and, with the cessation of the War and return to industrial life, the Edition was soon exhausted. The Author's appointment to a Professorship in India just when the book ran out of print prevented more than slight revision, but the opportunity has been taken to add important Appendices with many new illustrations.

December, 1921

PREFACE

Compressed air has proved itself in very many ways an exceedingly useful agent in coal and metalliferous mining; for transmitting energy into the working-face for drilling and cutting, and for tunnelling operations, haulage, pumping, and other purposes. The introduction of electricity for mining threatened for a time to supersede compressed air, but experience has proved that electricity is, under certain conditions, a source of grave danger, so much so that stringent legislative restrictions have been found necessary. Compressed air, in regard to safety, convenience, and general suitability, has many advantages over electricity and other modes of transmitting power, and should in no wise suffer in comparisons of efficiency if more attention is given to the proper application of the system.

Mining engineers in Britain and her colonies do not appear to have given the attention to this subject which it deserves, and possibly the absence of a textbook dealing with it may partly account for this. An endeavour is here made to supply this need, in the first place for students in mining schools and colleges, but it is also hoped the book may prove serviceable to those in practice. It is based upon lectures given to students, and whilst in the first chapter a knowledge of elementary mathematics is assumed, and in the second the calculus is used as far as thought necessary, the remainder is for the most part descriptive. A thorough understanding of the principles underlying the generation and use of compressed air in mining practice is most important, and to help students to test their knowledge numerous examples of fully worked out calculations are included in the book.

Thanks are tendered to many firms who have willingly given information and blocks descriptive of ingenious devices now in use.

DAVID PENMAN

Cowdenbeath

November 1916

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CHAPTER I

FUNDAMENTAL PRINCIPLES

In order that air may be available as a motive power for operating machinery it requires to be compressed or raised in pressure above the pressure of the atmosphere.

This is done in the air-compressor, which takes in the air from the surrounding atmosphere and compresses it to a predetermined pressure above atmospheric. The air is delivered at this pressure to a receiver, whence it passes through pipes to the air-engine or motor. This latter works in the same way as the ordinary steam-engine, using pressure-air instead of steam.

On the other hand, the air-compressor works in the reverse way to the steam-engine, since in it the working substance is being raised from a lower pressure to a higher pressure. The air-compressor then requires to be driven, since work is being done on the air and it is being put into a condition in which it will be able to do work against an external resistance. The prime mover may be a steam- or gas-engine or an electric motor.

In considering what happens when air is being compressed, it should be borne in mind that during compression heat is produced which if not removed raises the temperature of the air; conversely, when the pressure-air is allowed to expand against an external resistance, heat is necessary to do the work of expansion, and this heat may come from some source external to the air or from the air itself, in which case the air falls in temperature as it expands.

The work done in an air-compressor cylinder therefore results in

(*a*) An increase in the pressure of the air.
(*b*) The production of heat.

The work converted into heat is lost. This would not be the case if the air absorbed and retained all the heat produced during compression. This, in general, is not possible, however, for unless the distance through which the air has to be taken is small, the heat is completely lost by radiation whilst the air is traversing the conducting pipes.

In addition, the resulting temperature in all but comparatively low pressures would be such as to render the compressor unworkable if no provision were made to get rid of at least some of the heat.

Moreover, as will be shown, if it is impossible or impracticable to retain the heat of compression, it is more economical to endeavour to extract as much of the heat as possible *during the compression* than to allow it to dissipate afterwards. Thus it is necessary to employ some cooling agency to keep down the temperature of the air while it is being compressed.

It is impossible, however, to get rid of all the heat produced *before the pressure-air leaves the working cylinder*, which is the ideal aimed at, and all that can be done is to employ a cooling system, which, considering also the other points necessary to a satisfactory and efficient installation, will get rid of as large a proportion of the heat as is thought desirable.

ISOTHERMAL COMPRESSION AND EXPANSION

If we could slowly compress the air in a cylinder of perfectly conducting material, capable of continuously absorbing and carrying off the heat of compression as quickly as it was produced, we might attain a condition of working in which the air would be completely prevented from rising in temperature.

Compression carried out under such a condition is called *isothermal compression*, the temperature of the air throughout the entire process remaining unchanged.

Similarly, if during expansion in the air-engine cylinder—work then being done by the gas instead of on it—we could supply heat to the working stuff at such a rate that the temperature of the air whilst expanding in or escaping from the cylinder was the same as when entering the cylinder, the expansion would be called isothermal.

Obviously under these conditions the air would obey Boyle's Law, which states that at constant temperature the volume of a gas is inversely proportional to the pressure.

Hence the isothermal curve of compression will satisfy the relation

$$p\,v = \mathrm{K}$$

where p = pressure
v = volume
K = a constant

This relation can be written as

$$
\begin{aligned}
p_1 v_1 &= p_2 v_2 \\
\text{where } p_1 &= \text{original pressure} \\
p_2 &= \text{new} \quad\quad ,, \\
v_1 &= \text{original volume} \\
v_2 &= \text{new} \quad\quad ,,
\end{aligned}
$$

The value of the constant K will, however, be different at different temperatures, but if T denotes the absolute temperature we know, from a combination of Boyle's and Charles's Laws, that

$$\frac{p\,v}{T} = R \text{ (foot-pounds)},$$

where R is a constant.

The value of R for air is 53·2 if p is in lbs. per square foot, v in cubic feet, and T in degrees Fahrenheit, 1 lb. of the stuff being considered.

If p is in lbs. per square inch, the other quantities being as before, then R = ·37.

The value of R is different for different substances and varies inversely as the density: thus for oxygen we have

$$R = \frac{53{\cdot}2 \times 14{\cdot}4}{16} = 48$$

The values of R for four different substances are as follows:

TABLE I

Gas.	Relative Density.	R.
Air	14·4	53·2
Oxygen . . .	16·0	48·0
Hydrogen . . .	1·0	766·0
Carbon dioxide . .	22·0	34·9

From the equation $\frac{p\,v}{T} = R$

we have the following important relations:

(1) With T constant, $p \propto^{*} \frac{1}{v}$

(2) With v ,, $p \propto T$

(3) With p ,, $v \propto T$

(4) With all three varying, $p \propto \frac{T}{v}$

* For α, read *is proportional to*, or *varies as*.

EXAMPLE I.—Ten cubic feet of air at atmospheric pressure (15 lbs. per square inch absolute) are compressed into a space of $2\frac{1}{2}$ cubic feet. Find the new pressure assuming no change in the temperature.

$$p_1 v_1 = p_2 v_2$$

$$\therefore p_2 = \frac{p_1 v_1}{v_2} = \frac{15 \times 10}{2\frac{1}{2}} = 60 \text{ lbs. per square inch}$$

EXAMPLE II.—Ten cubic feet of air at 60° F. and at 15 lbs. per square inch are compressed to a pressure of 100 lbs. (absolute) per square inch when the temperature is 425° F. Find the new volume.

$$\frac{p_1 v_1}{T_1} = R.$$

This can be written

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}$$

$$\text{or } p_1 v_1 T_2 = p_2 v_2 T_1$$

$$\therefore v_2 = \frac{p_1 v_1 T_2}{p_2 T_1}$$

$$\therefore v_2 = \frac{15 \times 10}{100} \times \frac{(425 + 460)}{(60 + 460)}$$

$$= 1{\cdot}5 \times \frac{885}{520} = 2{\cdot}553 \text{ cubic feet}$$

ISOTHERMALS.—If a given volume of air at any temperature be taken and compressed or allowed to expand without change of temperature, then the curve showing graphically the relation between the pressure and the volume of the gas throughout the process is called an *isothermal*. (Fig. 1.)

If the initial temperature of the air were different, then the isothermal at that temperature would not coincide with that at the other temperature, although both would satisfy the same relation, viz.:

$$p\,v = K$$

Hence for any given volume of gas we might have an infinite number of isothermal curves, depending upon the initial temperature of the stuff, which would each represent the relation between the pressure and the volume at one definite temperature.

In the case of a gas which does not quite conform to Boyle's Law the isothermal curve is of the same general form as that for a perfect gas, but the curve, as is shown by the dotted line, falls below that for the gas completely obeying the law.

Considering the isothermal, Fig. 1, if we commence compressing the air at the point *A* and cease at the point *B*, then the pressure will be given by the perpendicular distance of *B* from the volume line, *i.e*

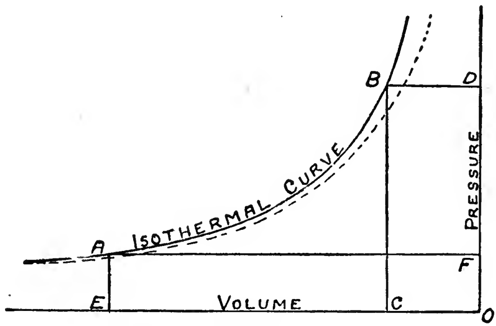


FIG. 1 Isothermal Curve.

B C, while the volume will be given by the perpendicular distance of *B* from the pressure line, *i.e. B D*. Conversely, if we now allow the air to expand isothermally from *B* to *A* we shall have brought the stuff back to its original pressure and volume, viz. *A E* and *A F*.

Since the isothermal curve complies with the law $p\ v = K$, the rectangle *A F O E* is equal to the rectangle *B D O C* in area.

ADIABATIC COMPRESSION AND EXPANSION

If, during the process of compression, we could so surround the working cylinder with some completely non-conducting wrapping that *all* the heat produced would be confined to the interior of the cylinder and thus go to raise the temperature of the air, the compression would be what is called *adiabatic*. Similarly, if, during the expansion period in the air-engine, the air was completely prevented from absorbing heat from any source whatever, the condition for adiabatic expansion would be secured.

It is clear that the relation between pressure and volume under such conditions will be totally different from that holding in isothermal compression and expansion. For since it is impossible to compress a gas without the production of heat or to have a gas expand, doing work, without the dissipation of heat, it is obvious that in either operation the temperature of the working substance cannot remain unchanged, which is the condition necessary to conform to the isothermal law.

The difference between isothermal and adiabatic compression and expansion may be readily understood in a general way by considering the two curves in Fig. 2.

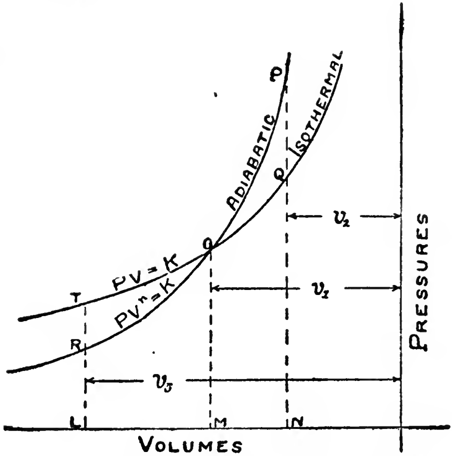


FIG. 2. Isothermal and Adiabatic Curves.

Suppose a volume of air v_1 be compressed isothermally to the volume v_2. Then the curve T O Q represents the relation between pressure and volume, so that the pressure during compression is changed from that represented by M O to that represented by N Q. On the other hand, suppose the air be compressed adiabatically, then under this condition all the heat produced goes to raise the temperature of the stuff, with the result that the pressure increases more rapidly for the same rate of decrease of volume, and if the final volume is v_2 as before, then the pressure is that indicated by N P, which is *greater* than that reached on the isothermal curve.

In the same way, if the volume v_1 at the pressure M O be allowed to *expand* isothermally to v_3 the resultant pressure is L T, while if the air expands adiabatically to v_3 the pressure will be L R, which is less than L T, the reason being that in the adiabatic expansion heat must be abstracted from the air to do the work of expansion, no other source of heat being permissible, and so the temperature falls and for a given volume the pressure is less.

The adiabatic curve is therefore *steeper* than the isothermal curve. The isothermal curve conforms to the law $p\,v =$ constant, while the adiabatic curve will be shown to conform to the relation $p\,v^{\gamma} =$ constant (*see* p. 21). In Fig. 2, n is substituted for γ.

The index γ has the value for air of 1·408, and is equal to the ratio of the specific heat of air at constant pressure to the specific heat at constant volume. The value of γ is also equal to the ratio of the adiabatic elasticity of air to its isothermal elasticity.

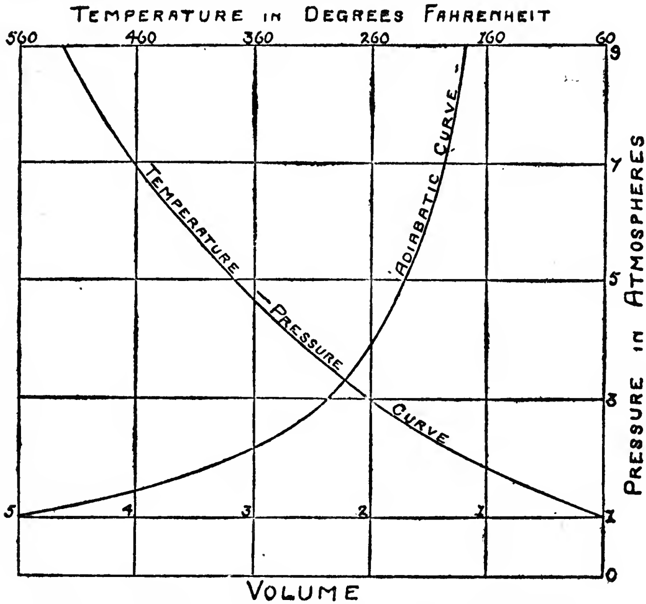


FIG. 3. Adiabatic Curve showing Temperature Rise during Compression.

The modulus of elasticity of any substance is in general the ratio of the stress per unit area to the corresponding strain. In the case of gases the strain is, of course, volume strain.

Now during isothermal compression for a given stress we have a certain diminution in volume, but during adiabatic compression, when all the heat produced goes to raise the temperature of the gas, the consequent tendency to expansion produces a force which opposes the compression; and thus for the same stress the volume strain will be less during an adiabatic change than during an isothermal change. Consequently stress $\div$ strain will be greater for adiabatic than for isothermal compression.

The adiabatic elasticity modulus is equal to 1·408 times the isothermal modulus for air.

ADIABATICS.—These are curves which conform to the law $p\,v^{\gamma} =$ constant, and, as has already been pointed out, represent the relation of pressure and volume when the heat of compression enters completely into the substance. As for isothermals, we can have an infinitude of adiabatics for a given initial pressure and volume depending upon the initial temperature of the air.

The final temperature of the air, moreover, after an adiabatic change will depend upon the initial temperature; the lower the initial temperature the lower will be the final temperature.

It is clear, therefore, that the colder the air taken into the compressor the lower will the delivery temperature be, whether we have no cooling or whether we cool the air during compression by some of the methods to be considered. This is a point of some importance and shows that the air going to the compressor should be drawn from the outer atmosphere and not from the heated air of the engine-room. Further, the compressor should be so designed as to admit the air to the compressor-cylinder with the minimum of heating during its passage through the suction-pipe and the inlet valves. The lower the initial temperature of the air the higher will be the efficiency of compression obtained.

Fig. 3 shows graphically the adiabatic curve for compression from one atmosphere absolute up to 9 atmospheres absolute, with the rise of temperature throughout the operation, the initial temperature of the air being assumed to be 60° F.

COMPRESSION AND EXPANSION UNDER WORKING CONDITIONS

THE COMPRESSION CURVE.—If we suppose, as in Fig. 4, a volume of air equal to $f\ a$ to be compressed adiabatically until the pressure above atmospheric is as indicated by $f\ e$, then the volume will be reduced to $f\ h$.

If we could extract all the heat of compression during the operation the pressure-volume curve would be the isothermal and the volume would be reduced to $f\ l$.

Thus by compressing adiabatically we get an increased volume due to the heating up of the air for the same delivery pressure. But in most cases all this heat is lost before the air reaches the air-engine, and so the volume eventually falls to $f\ l$.

Now if $a\ f$ represents the whole distance travelled by the air-piston in one stroke, then the area of the figure $a\ b\ e\ f$ is a measure of the work done during the adiabatic compression; also the area $a\ d\ e\ f$ measures the work done if the compression were isothermal. Since eventually the pressure and volume would become the same in both cases, it is evident that a saving is effected by compressing isothermally. This saving is represented by the shaded area *a b d*. In actual practice the compression curve lies between the adiabatic and the isothermal.

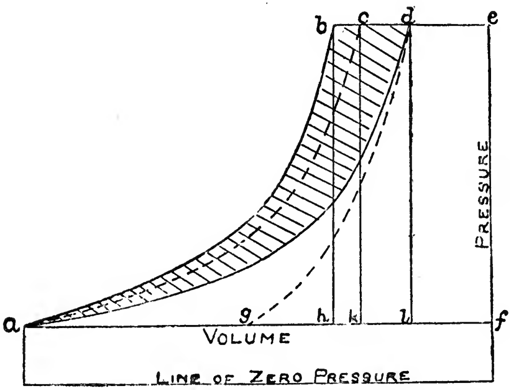


FIG. 4. Graphical Representation of Work Lost during Compression and Expansion.

With water-jacket cooling, which is the most general method, the index n is approximately 1·3 and the curve is *a c*, and the loss is that given by the area *a c d*.

This matter is considered further in Chapter II.

THE EXPANSION CURVE.—Suppose the temperature of the pressure-air at the place where it is being used is the same as that at the beginning of compression, and there is no loss of volume or pressure during transmission; then the pressure in the air-engine cylinder will be that indicated by *f e* and the volume will be that indicated by *f l*.

Now since no attempt is made in an air-engine cylinder to supply heat to the expanding air, and the operation is performed much too quickly to admit of the air extracting any appreciable quantity of heat from the surrounding objects, the curve of expansion will be very nearly a purely adiabatic one. Thus the temperature of the air will fall rapidly and the curve of expansion will be that represented by *d g*. The work done is therefore represented by the area *f e d g*, which is less than the area under the isothermal curve.

If, on the other hand, we re-heat the air, before it is used to do work, up to the temperature that would have been reached in the adiabatic compression, the volume would be increased to *f h*, the pressure remaining constant, and the expansion would take place along the curve $b\ a$.

Thus the work done by the same *mass* of air would be greater. In the re-heating we are, of course, simply supplying *energy* to the air in the form of heat.

In the case of adiabatic compression and then re-heating of the air to the final compression temperature before using expansively, we would get the same work done *by* the air as was done *on* the air. The efficiency would not, however, be 100 per cent., since the heat of compression is lost, and that, of course, means lost work.

If we could cause the air to retain all the heat of compression and so have the initial temperature on the expansion curve equal to the final temperature on the compression curve the efficiency would be the highest possible. Attempts are made to approximate to this in some cases where the air can be compressed close to where it is to be used, but generally this is not at all practicable.

The efficiency, allowing only for losses in compression and expansion,

$$= \frac{\text{area } f\ e\ d\ g}{\text{area } f\ e\ c\ a.}$$

In Fig. 5 it will be noticed that the further the compression of the air is carried the greater is the divergence of the adiabatic and isothermal curves, and thus in *single-stage* compression the loss per cent. represented by the heat of compression becomes greater the higher the final or delivery pressure. It would therefore appear that to obtain a high efficiency it is necessary to use low pressures. Although this is quite correct from the purely thermo-dynamic point of view, when the increased cost of compressors, pipes, and air-motors required if using low-pressure air is considered, it is found more economical to use moderately high pressures—60 to 70 lbs. per square inch above atmospheric—even in single-stage compressors.

CONSTRUCTION OF THE ISOTHERMAL DIAGRAM

For the purpose of comparison with the indicator diagram taken from the cylinder of the compressor or of the air-engine, it is often necessary to construct the isothermal diagram corresponding to the cylinder volume and the inlet and delivery pressures.

The cylinder volume equals the piston displacement plus the clearance volume. The piston displacement is proportional to the length of stroke, and if the clearance volume is known it can also be represented by a line of a given length.

Referring to Fig. 5, let $o\ a$ be the atmospheric line. If the air behind the piston on the suction stroke be at atmospheric pressure, then the suction line will coincide with the atmospheric line. Let $o\ e$ equal clearance to scale, and $e\ a$ equal length of stroke of piston to the same scale.

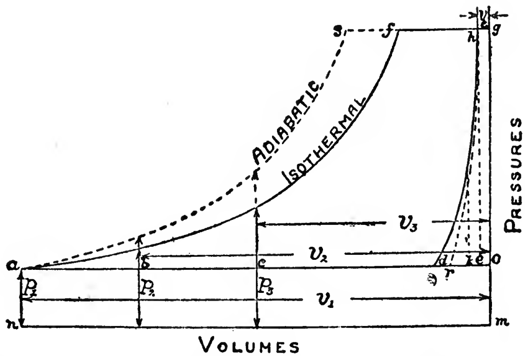


FIG. 5. Construction of Isothermal and Adiabatic Diagrams.

At the end of the suction stroke, therefore, the total volume is $o\ a$ and the pressure is atmospheric. Let this volume be v_1 and the pressure p_1. Then for isothermal compression, and assuming no leakage past valves or piston, if p_2 is the pressure at any point on the compression curve up to the delivery-point f, and v_2 the corresponding volume we have

$$p_1\,v_1 = p_2\,v_2\,;$$

and hence, selecting any point on the line $o\ a$—say b, when the volume is $v_2 = \frac{3}{4}\,v_1$—then

$$p_2 = \frac{p_1\,v_1}{v_2} = \frac{p_1\,v_1}{\frac{3}{4}\,v_1} = \frac{4}{3}\,p_1$$

Similarly, selecting some other point—say c, when the volume is $v_3 = o\ c = \frac{1}{2}\,o\ a = \frac{1}{2}\,v_1$—we have

$$p_3 = \frac{p_1\,v_1}{v_3} = \frac{p_1\,v_1}{\frac{1}{2}\,v_1} = 2\,p_1$$

Calculating other points on the curve in the same way and joining up the points by a smooth curve, we get the part $a\,f$ of the diagram.

Then draw $f\,h$ parallel to $o\ a$ and to meet the perpendicular drawn from e. This is the delivery line. The distance $h\,g = e\,o =$ clearance (v_c).

Assuming again isothermal expansion, the compressed air in the clearance space on the return stroke of the piston expands down the curve *h d*, the points on which can be calculated in the same way as those on the compression curve, starting from p_{f1}, the final or delivery pressure, and v_c, the compression volume.

Thus at the point *k* on *o a*, where *o k* is twice *o e*, and calling p_{f2} the pressure at that point, we have:

$$o\,k = o\,e \times 2 = 2\,v_c$$

$$\therefore\ p_{f1} \times v_c = p_{f2} \times 2\,v_c$$

$$\therefore\ p_{f2} = \frac{p_{f1} \times v_c}{2\,v_c} = \tfrac{1}{2}\,p_{f1}$$

And similarly for other points. Thus we complete the diagram ***a f h d***.

It should be noted that the pressures have to be measured from the line of absolute zero pressure (*m n*). For accuracy, about ten points should be calculated and the best procedure is to divide the volume line *o a* into ten equal parts and calculate the pressure at every point up to the delivery pressure. The various calculations are easily and quickly made, and on squared paper the plotting is done in a few minutes.

THE ADIABATIC DIAGRAM

The points on the compression and expansion parts of the adiabatic diagram can be calculated in a similar way to that shown for the isothermal diagram but not quite so simply.

In Fig 5 (p. 11) let $o\,a =$ the stroke of the piston + clearance, as before. Call this v_1 and the pressure p_1 as before.

Then at the point *b* the volume will be $o\,b = v_2 = \frac{3}{4}\,v_1$, as before. Let the pressure at this point be p_2.

$$\text{Then } p_1\,v_1^{1\cdot41} = p_2\,v_2^{1\cdot41}$$

$$p_1\,v_1^{1\cdot41} = p_2\,\tfrac{3}{4}\,v_1^{1\cdot41}$$

$$\therefore\ p_2 = \frac{p_1\,v_1^{1\cdot41}}{(\frac{3}{4}v_1)^{1\cdot41}} = \frac{p_1\,v_1^{1\cdot41}}{(\frac{3}{4})^{1\cdot41} \times v_1^{1\cdot41}} = \frac{p_1 \times 1^{1\cdot41}}{(\frac{3}{4})^{1\cdot41}} = \frac{p_1 \times 4^{1\cdot41}}{3^{1\cdot41}}$$

$$\therefore\ \frac{p_2}{p_1} = (\tfrac{4}{3})^{1\cdot41}, \text{ and, using logarithms, log. } \left(\frac{p_2}{p_1}\right)$$

$$= 1{\cdot}41\,\{\text{log. } 4 - \text{log. } 3\} = 1{\cdot}41\,\{\cdot6021 - \cdot4771\}$$

$$\therefore\ \frac{p_2}{p_1} = 1{\cdot}501, \text{ whence } p_2 = 1{\cdot}501\,p_1$$

Similarly, when the volume is reduced to $o\ c = v_3 = \frac{1}{2}\ v_1$ we find

$$p_3 = \frac{p_1 \times 2^{1.41}}{1^{1.41}} = p_1 \times 2^{1.41}$$

$$\text{whence } p_3 = 2{\cdot}657\ p_1$$

and similarly for other points. Then we can draw the dotted curve $a\ s$.

Also when the compressed air in the clearance volume expands we get the dotted curve $h\ r$.

It will be sufficient to calculate one point on this curve, say that when the volume $o\ k = 2\ v_c$ as before.

We have:

$$p_{f1} \times v_c^{1.41} = p_{f2} \times 2\ v_c^{1.41}$$

$$\therefore\ p_{f1} = \frac{p_{f1} \times v_c^{1.41}}{2\ v_c^{1.41}}$$

$$p_{f2} = \frac{p_{f1}}{2^{1.41}}$$

$\therefore\ p_{f2} = \dfrac{p_{f1}}{2{\cdot}657}$, which is less than the pressure at the same point on the isothermal curve, so the adiabatic falls below the isothermal in expansion, as has been seen elsewhere.

MULTIPLE STAGE COMPRESSION.—So far we have considered the compression to take place in one stage only—as, for example, in Fig. 4, where the curve $a\ c$, representing compression, under ordinary water-jacketing conditions rises unbroken from the inlet pressure to the discharge pressure.

Single-stage compression is generally considered satisfactory up to about 60 lbs. per square inch gauge pressure, but for higher pressures the compression is now generally performed in two or more operations or stages.

The initial pressure in the first cylinder is atmospheric as in single-stage, but the inlet pressure in the second cylinder is the same as the delivery pressure from the first; or, allowing for a slight loss of pressure between the two cylinders, only a pound or two lower. After the pressure-air leaves the first cylinder, but before it enters the second, it passes through an intercooler (*see* p. 54), where it is made to give up a very large portion of the contained heat, and so the initial temperature of the air at the beginning of the second operation is brought down considerably, nearly to that of the atmosphere.

The result is, as can be readily understood from the curves in Fig. 6, that the curve of compression in the second stage is very much nearer to the isothermal than it would have been throughout the same range of pressure continued as a single-stage compression. This, as will be evident from what has previously been said, represents a considerable saving on the total work expended in compressing a given weight of air to a given pressure.

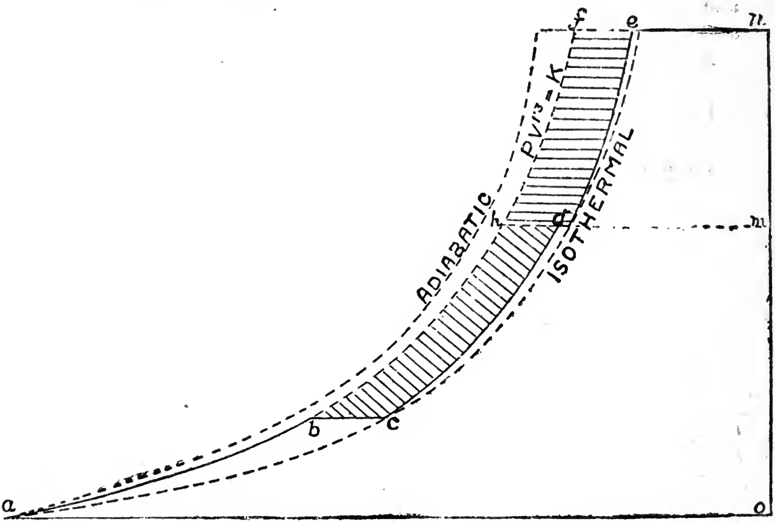


FIG. 6. Two-stage and Three-stage Compression.

Considering Fig. 6, suppose we compress in the low-pressure cylinder along the curve *a b*, assuming water-jacket cooling. At *b* the air is passed through an intercooler which reduces its temperature, say, to what it was at *a*, *i.e.* to the initial temperature. In the second-stage cylinder, compression starts at *c* on the isothermal curve and follows the curve *c d*.

If the compression had been done in one stage along the water-jacketed curve (*i.e.* $P\ V^{1.3} = K$) the work done during the operation would be represented by the area *a h m o*, while in the case of the two-stage compression with intercooling the work done in compressing the same volume or weight of air to the same pressure is that represented by the area *a b c d m o*, which is less than the work done in single-stage compression by an amount represented by the hatched area *b c d h*.

It is therefore evident that a considerable saving may be effected by compressing in two stages instead of in one stage.

The diagram also shows the effect of compressing in three stages, with a further intercooler between the second and third stages. In the third stage the saving as compared with the single-stage compression is represented by the hatched area *h d e f*. Three-stage compression is comparatively rare in mining practice, and is generally only adopted for pressures over 200 lbs. and up to 750 lbs. per square inch

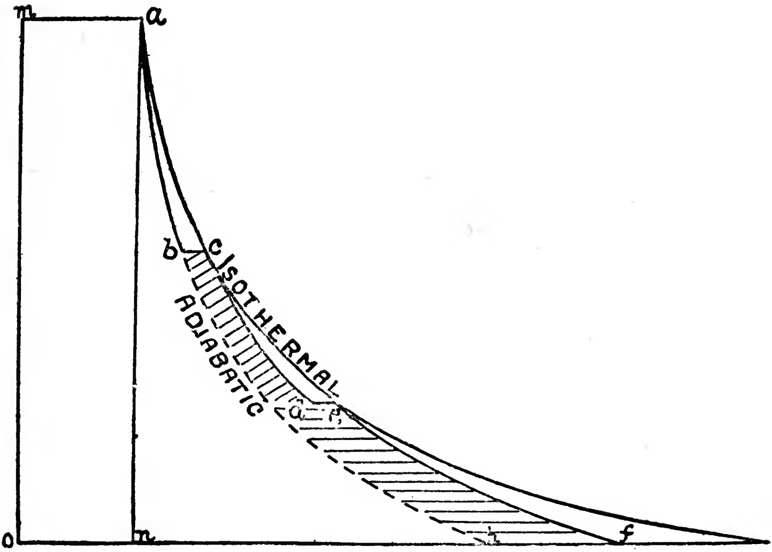


FIG. 7. Two-stage and Three-stage Expansion.

For pressures above 750 lbs. per square inch four and even five stages are employed. So far these high pressures have only been used for locomotive work.

In addition to giving a higher efficiency and keeping down the final temperature of the air, multiple-stage compression with intercooling greatly reduces the stresses on the working parts—since smaller cylinders can be used for the higher pressures—and so makes for lighter machines and greater durability.

MULTIPLE STAGE EXPANSION.—For the most part little or no attempt is made in mining practice to utilize the expansive properties of the pressure-air, the full pressure being kept on throughout the major portion of the stroke.

That considerable advantage as regards efficiency may be gained by using the air expansively where practicable is readily demonstrated. Referring to Fig. 7, let air at a pressure represented by *o m* displace a piston a distance equal to *o n*. The area of the rectangle ***o m a n*** will be a measure of the work done. Now suppose the supply of air to be cut off and the air allowed to expand, forcing the piston forward. As already pointed out in considering Fig. 4, this expansion will be adiabatic, and if we assume the pressure in the cylinder to fall to atmospheric as represented by the line *o h*, the piston will now have been displaced to *h*. Thus the air with expansion has accomplished a total amount of work represented by the area *o m a h*.

Now consider the case where we have, say, expansion in three stages, to take a case which will include two-stage expansion and can be extended if desired. In the first expansion the pressure falls off along the adiabatic curve from *a* to *b* ; then the air, escaping, passes, say, through a re-heater, which brings back the temperature to the initial value, and so the air expands from *b* to *c at constant pressure*. Then in the second expansion, which would take place, of course, in another cylinder, the air expands *adiabatically* from *c* to *d*. It now passes through a second re-heater which brings back the temperature again to the initial temperature, and the volume expands from *d* to *e*. Finally adiabatic expansion, *e* to *f*, takes place in a third cylinder. The gain over single expansion is represented by the shaded area, the second and third stages being separately indicated.

Underground the re-heating is a difficulty, but if there was an appreciable distance between the cylinders representing the different stages it might be practicable to allow the natural heat of the strata—which is considerable in deep mines—to bring back the temperature of the air to the isothermal.

The excessively low temperatures produced, with the consequent tendency to the formation of ice, is the objectionable feature in using air expansively, but a thorough drying of the air between compression stages and before use would remove this objection.

It should be noted too that in two- or three-stage expansion, as described above, the initial pressure would have to be considerable—from about ten atmospheres and upwards—before the method could be adopted with success.

RE-HEATING THE AIR.—We have seen that the highest efficiency obtainable from compressed air can only be secured by using the air expansively.

In order to obtain the highest degree of expansion without increasing the tendency to freezing, the compressed air is sometimes re-heated before it enters the air-engine.

Various forms of re-heaters are in use. Most of them consist in passing the air through a spiral pipe or tube or some other form of chamber, which is heated by the hot gases from a coal or coke fire.

The air takes up the heat and increases in temperature and volume, and the increase is proportional to the increase in the absolute temperature of the air.

For example, let the temperature of the air before entering the heater be T_1 and on leaving T_2,

$$\text{Then} \frac{\text{new volume}}{\text{original volume}} = \frac{T_2}{T_1}$$

Suppose the air enters at 70° F. (530° absolute) and leaves at 00° F (760° absolute), and let the original volume be V_1,

$$\text{Then new volume} = V_1 \times \frac{760}{530}$$
$$= 1{\cdot}43\ V_1$$

Thus the vol me has been increased 4 3 per cent.

Unless the re-heater is very close to the air-motor some of the heat is again lost, and in the most favourable cases in practice the increase in volume does not amount to more than about 30 per cent.

However, this 30 per cent. increase in volume means 30 per cent. increase of possible work got out of the air, since the pressure remains unaffected by the heating.

Further, as has already been mentioned, it may be possible to expand the air in the cylinder right down to atmospheric pressure without lowering the temperature at the exhaust below the freezing-point of water. This, of course, greatly enhances the efficiency.

A point of extreme importance is that the heating results in an increase of energy, which is obtained five or six times as efficiently as that obtained in the compressing cylinder. Therefore, wherever possible, re-heating should be done, but, unfortunately, it is not very practicable to re-heat underground.

In the case of portable machines such as coal-cutters, conveyers, and rock-drills re-heating cannot be profitably carried out. For stationary machines, driving pumps, and haulages, it may be possible occasionally to use re-heaters. The ordinary form of re-heater is, however, generally unsuitable.

An internal electrical re-heater, consisting of a resistance coil or a series of these placed in a small receiver in the piping close to where the air is to be used, has been employed. An electric current passing through insulated wires from a source of supply to the resistance coils heats these, and so the air passing over the coils is increased in temperature and volume.

In metalliferous mines ordinary re-heaters may sometimes be used, but in coal-mines, gassy or dusty, their use is out of the question.

CALCULATIONS ON THE ADIABATIC LAW

We have seen that the equation $p\,v^{\gamma} = K$ gives the relation between the pressure and volume on an adiabatic curve.

If the compression or expansion is not purely adiabatic but only approximates thereto, we can substitute n for γ, and the equation becomes $p\,v^n = K$.

This can be written

$$p_1\,v_1^{\,n} = p_2\,v_2^{\,n}$$

where p_1 and v_1 = pressure and volume at any one point on the curve.
p_2 and v_2 = pressure and volume at some other point on the curve.

EXAMPLE III.—Two cubic feet of air at atmospheric pressure (say 15 lbs. per square inch absolute) are compressed to 90 lbs. gauge. If the compressor is water-jacketed and the compression index is 1·3, find the volume after compression.

$$p_1\,v_1^{\,n} = p_2\,v_2^{\,n}$$

substituting values—

$$15 \times 2^{1\cdot3} = (90 + 15) \times v_2^{\,1\cdot3}$$

$$\text{or } 2^{1\cdot3} = 7\,v_2^{\,1\cdot3}$$

$$\therefore \quad v_2^{\,1\cdot3} = \frac{2^{1\cdot3}}{7}$$

Taking logarithms of both sides—

$$1\cdot3 \log v_2 = 1\cdot3 \log. 2 - \log. 7$$

$$\log. v_2 = \frac{1\cdot3 \log 2 - \log. 7}{1\cdot3}$$

$$= \bar{1}\cdot6510$$

$$\text{whence } v_2 = \cdot4477 \text{ cubic feet.}$$

If the compression were purely adiabatic,

$$v_2 = \cdot5021 \text{ cubic feet;}$$

and if purely isothermal,

$$v_2 = \cdot2857 \text{ cubic feet.}$$

EXAMPLE IV.—Two cubic feet of compressed air at 60 lbs. per square inch gauge pressure expand in a cylinder down to atmospheric pressure (say 15 lbs., as before). Find the volume after expansion if expansion index = 1·4.

$$p_1\,v_1^{\,1\cdot4} = p_2\,v_2^{\,1\cdot4}$$

$$75 \times 2^{1\cdot4} = 15 \times v_2^{\,1\cdot4}$$

Dividing both sides by 15 as before—

$$5 \times 2^{1.4} = v_2^{1.4}$$

$$\therefore \log. 5 + 1{\cdot}4 \log. 2 = 1{\cdot}4 \log. v_2$$

whence in the same way, as before—

$$v_2 = 6{\cdot}314 \text{ cubic feet.}$$

If the expansion had been isothermal, as is nearly the case in a steam-engine, v_2 would have been 10 cubic feet.

Relationship between Temperature, Volume, and Pressure on an Adiabatic Curve.

Let $p_1\ p_2\ v_1\ v_2$ be as before (p. 18) and

let T_1 = initial absolute temperature

and T_2 = final " "

$$\text{Then } p_1 v_1^n = p_2 v_2^n$$

$$\text{or } p_1 v_1 v_1^{n-1} = p_2 v_2 v_2^{n-1}$$

$$\text{Now } p_1 v_1 = R\, T_1$$

$$\text{and } p_2 v_2 = R\, T_2$$

$\therefore$ By substitution we have—

$$R\, T_1 v_1^{n-1} = R\, T_2 v_2^{n-1}$$

$$\text{or } T_1 v_1^{n-1} = T_2 v_2^{n-1}$$

$$\therefore \frac{T_1}{T_2} = \frac{(v_2)^{n-1}}{(v_1)^{n-1}} = \left(\frac{v_2}{v_1}\right)^{n-1} \quad \ldots \quad \text{(A)}$$

$$\text{Also} \quad \frac{v_2}{v_1} = \left(\frac{T_1}{T_2}\right)^{\frac{1}{n-1}}$$

Again, from $p_1 v_1^n = p_2 v_2^n$ we get

$$\frac{p_1}{p_2} = \left(\frac{v_2}{v_1}\right)^n$$

$$\therefore \frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}$$

$$\text{but } \frac{v_2}{v_1} = \left(\frac{T_1}{T_2}\right)^{\frac{1}{n-1}}$$

$$\therefore \left(\frac{T_1}{T_2}\right)^{\frac{1}{n-1}} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}$$

$$\text{or } \frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}} \quad \text{(B)}$$

Equation (B) is usually the more useful

EXAMPLE V.—Ten cubic feet of air at atmospheric pressure (say 15 lbs. per square inch absolute) and 50° F. are compressed in a water-jacketed compressor to 90 lbs. gauge (*i.e.* 105 lbs. absolute). If the index n in the relation $p\ v^n =$ const. is 1·3, find the temperature of the air at the end of compression.

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}$$

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{1\cdot 3-1}{1.3}} = \left(\frac{p_1}{p_2}\right)^{\frac{\cdot 3}{1\cdot 3}}$$

$$= \left(\frac{p_1}{p_2}\right)^{\cdot 23}$$

$$\therefore \frac{T_1}{T_2} = \left(\frac{15}{105}\right)^{\cdot 23}$$

$$= \left(\frac{1}{7}\right)^{23}$$

$$\therefore T_2 = T_1 \times 7^{(\cdot 23)}$$

$$\therefore \log. T_2 = \log. T_1 + \cdot 23 \log. 7$$

$$\text{whence } T_2 = 798° \text{ absolute}$$

$$= 798 - 460 = 338° \text{ F.}$$

If the index of compression had been 1·4 instead of 1·3 the final temperature would have been 429·2° F., showing that water-jacketing, though a very inefficient method of cooling, yet effects a considerable reduction on the final temperature.

EXAMPLE VI.—Compressed air at 45 lbs. per square inch is used in a coal-cutter motor. Find the final temperature of the exhaust air, assuming expansion in the cylinder down to atmospheric pressure. Temperature of pressure-air 70° F.

Here the curve of expansion is very nearly purely adiabatic and the index will be, say, 1·4.

$$\therefore \frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{1\cdot 4-1}{1.4}}$$

$$= \left(\frac{p_1}{p_2}\right)^{.2857}.$$

$$\therefore \log T_2 = \log. T_1 + \cdot 2857 \log. p_2 - \cdot 2857 \log. p_1$$

$$p_1 = 45 + 15 = 60 \qquad p_2 = 15$$

$$T_1 = 460 + 70 = 530°$$

$$\therefore \log. T_2 = \log. 530 + \cdot 2857 \log. 15 - \cdot 2857 \log. 60$$
$$= \log. 530 + \cdot 2857 \log. 1 - \cdot 2857 \log. 4$$
$$\text{whence } T_2 = 356{\cdot}7^\circ \text{ absolute}$$
$$= -103{\cdot}3^\circ \text{ F.}$$

Note.—When the air is exhausted at or near full pressure the temperature at the exhaust ports is not nearly so low as that obtained above. The temperature of the exhaust depends chiefly on the degree of expansion.

To show that γ = the Index of Compression or Expansion on an Adiabatic Curve.

When unit weight of air is heated at constant volume the whole of the heat added goes to increase the internal energy, for as there is no increase in volume there is no external work done.

$\therefore$ The gain in internal energy = heat supplied = $C_v(T_2 - T_1)$ where C_v = specific heat of air at constant volume, and T_1, T_2 are the initial and final absolute temperatures.

If the air, however, be heated from T_1 to T_2 and allowed to expand in order to keep the pressure constant, then

$$\text{Heat supplied} = C_p(T_2 - T_1)$$

where C_p = specific heat at constant pressure.

But there is external work done in this case

$$= p \text{ (change in volume)}$$
$$= p\,(v_2 - v_1)$$

where v_2 and v_1 = final and initial volumes.

$$\text{Also, since } p\,v = R\,T, \text{ and } v \propto T,$$
$$p\,(v_2 - v_1) = R\,(T_2 - T_1)$$

Thus the gain in internal energy

$$= \text{heat supplied} - \text{work done}$$
$$= C_p\,(T_2 - T_1) - R\,(T_2 - T_1)$$
$$= (C_p - R)\,(T_2 - T_1)$$

Since between the same limits of temperature the gain in internal energy is the same in both cases,

$$(C_p - R)\,(T_2 - T_1) = C_v\,(T_2 - T_1)$$
$$\therefore C_p - C_v = R$$
$$\frac{C_p}{C_v} - 1 = \frac{R}{C_v}\,.$$

now the ratio $\frac{C_p}{C_v}\left(=\frac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}}\right)$ is called γ.

$$\therefore\ \gamma - 1 = \frac{R}{C_v}$$

$$\text{or} \qquad C_v = \frac{R}{\gamma - 1}$$

Now it is shown on p. 25 that during an adiabatic compression or expansion the work done on or by the gas is given by

$$W = \frac{p_2 v_2 - p_1 v_1}{n - 1}$$

$$= \frac{R\,T_2 - R\,T_1}{n - 1} = \frac{R\ (T_2 - T_1)}{n - 1}$$

Also during such an operation the work done = gain in internal energy if compressing or loss if expanding

$$= C_v\,(T_2 - T_1)$$

$$= \frac{R\ (T_2 - T_1)}{\gamma - 1} = \frac{R\ (T_2 - T_1)}{n - 1}$$

$\therefore\ n = \gamma$ on an adiabatic curve and $\gamma = \frac{C_p}{C_v}$

Regnault found $C_p = 0{\cdot}238$ B.Th.U. * per lb.
and $C_v = 0{\cdot}169$ B.Th.U. per lb.

or $K_p = 185{\cdot}1$ ft. lb. per lb.
$K_v = 131{\cdot}5$,, ,,

where $K_p = C_p \times 778$ (Joule's Equivalent) and similarly for K_v.

Thus $$\frac{C_p}{C_v} = \frac{\cdot 238}{\cdot 169} = 1{\cdot}408 = \gamma$$

$\therefore$ An adiabatic curve conforms to the law $p\,v^\gamma =$ constant.

γ is not constant for all temperatures, since C_v varies slightly with variation in temperature; but for all working ranges of temperature in compressed air γ can be taken equal to **1·41**.

* One British Thermal Unit is the quantity of heat required to raise 1 lb. of water from a temperature of 39° F. to a temperature of 40° F.

CHAPTER II

EFFICIENCY OF COMPRESSORS AND AIR-MOTORS

In estimating the efficiency of compressors or air motors it is necessary to be able to compute, graphically or mathematically, the work done if the compression had followed the law $p\ v^n =$ constant, where the exponent n may have any value from 1 to 1·41.

When $n = 1$ the compression is isothermal, when $n = 1·41$ it is adiabatic.

Before showing how the efficiency is calculated, therefore, it will be necessary to obtain formulæ that can be used in these calculations.

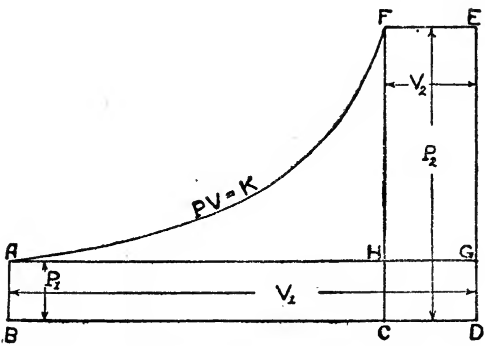


Fig. 8. Work done in Isothermal Compression.

WORK DONE IN COMPRESSING AIR ISOTHERMALLY.—Let the volume of air in the cylinder $= V_1$ and the pressure P_1 (Fig. 8), and let this volume be compressed to volume V_2 and pressure P_2.

Let W = work done on the air while the volume changes from V_1 to V_2 and the pressure from P_1 to P_2.

Then $$W = \int_{V_2}^{V_1} P\,d\,v$$

But $P V = K$ $\qquad \therefore P = \frac{K}{V}$

$$\therefore W = K \int_{V_2}^{V_1} \frac{d v}{V} = K \quad \log._e \frac{V_1}{V_2}$$

$$= P_1 V_1 \log._e \frac{V_1}{V_2} \quad (1)$$

$$= P_1 V_1 \log._e \frac{P_2}{P_1} \quad (2)$$

$$= R T \quad \log._e \ r \quad (3)$$

where $r = \frac{\text{original volume}}{\text{final volume}}$

$= \frac{\text{final pressure}}{\text{original pressure}}$

If pressures are in lbs. per square inch and volumes in cubic feet we have, using (2),

$$W \text{ (foot-lbs.)} = 144 \ P_1 V_1 \log._e \frac{P_2}{P_1}$$

or changing to common logarithms,

$W = 331{\cdot}2 \ P_1 V_1 \log._{10} \frac{P_2}{P_1}$; which is the most convenient **form.**

Now W = area A B C F. The work done during expulsion of air to receiver = $P_2 V_2$ and work done on the other side of the piston by the atmospheric air = $P_1 V_1$.

$$P_2 V_2 = P_1 V_1;$$

$\therefore$ area A B C H = area E F G H, since area C D G H is common to both.

$\therefore$ Area A B C F = area A F E G,

$\therefore$ whole work done during one stroke of the piston for isothermal compression

$$= W = 144 \ P_1 V_1 \log._e \frac{P_2}{P_1}$$

$$= 331{\cdot}2 \ P_1 V_1 \log._{10} \frac{P_2}{P_1}$$

WORK DONE IN COMPRESSING AIR ADIABATICALLY OR ACCORDING TO THE LAW $p \ v^n$ = CONSTANT.

Here as before,

$$W = \int_{V_2}^{V_1} P \, d \, v$$

Now $P_1 V_1^n = P_2 V_2^n = K \qquad \therefore \; P = \frac{K}{V^n}$

$$\therefore \; W = K \int_{V_2}^{V_1} V^{-n} \, dv$$

$$= \frac{K}{1-n} (V_1^{1-n} - V_2^{1-n})$$

$$= \frac{1}{1-n} (P_1 V_1 - P_2 V_2)$$

$$= \frac{P_2 V_2 - P_1 V_1}{n-1} \quad . \quad . \quad . \quad (1)$$

And if P is in lbs. per square inch and V in cubic feet

$$W \text{ (foot-lbs.)} = \frac{P_2 V_2 - P_1 V_1}{n-1} \times 144 \quad . \quad . \quad . \quad (2)$$

This work W = area A B C F as before (Fig. 9), and whole work

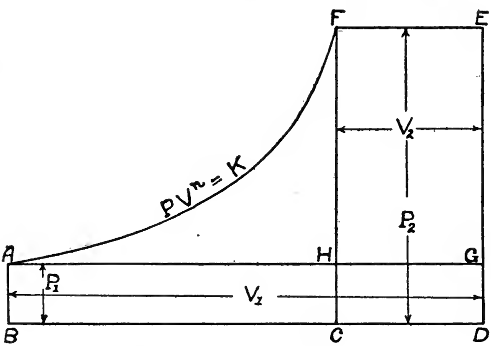


FIG. 9. Work done in Adiabatic Compression.

done by piston during one stroke, *i.e.* moving from A to G = area A F E G

$$= A\,B\,C\,F + E\,F\,C\,D - G\,A\,B\,D$$

$$= \left\{ \frac{P_2 V_2 - P_1 V_1}{n-1} + P_2 V_2 - P_1 V_1 \right\} \times 144$$

$$= \frac{n}{n-1} (P_2 V_2 - P_1 V_1)\, 144 \quad . \quad . \quad . \quad (3)$$

which is the simplest and often the most convenient expression for the whole work done in one stroke of the piston with adiabatic compression.

It can be given in another form as follows:

$$P_2 V_2^n = P_1 V_1^n$$
$$P_2 V_2 V_2^{n-1} = P_1 V_1 V_1^{n-1}$$
$$\text{or } P_2 V_2 = P_1 V_1 \left(\frac{V_1}{V_2}\right)^{n-1}$$
$$= P_1 V_1 \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

and substituting this in (3) we get

$$W \text{ (foot-lbs.)} = \frac{n}{n-1}\left[P_1 V_1 \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - P_1 V_1\right] 144$$
$$= \frac{n}{n-1}\left\{P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]\right\} 144$$

EXAMPLE I.—Find the work done in compressing 2 cubic feet of air from 15 lbs. absolute to 60 lbs. absolute—(*a*) assuming isothermal compression, (*b*) assuming adiabatic compression ($n = 1{\cdot}4$).

$$(a) \quad W = 331{\cdot}2\, p_1 v_1 \log. \frac{P_2}{P_1}$$
$$= 331{\cdot}2 \times 15 \times 2 \times \log. \frac{60}{15}$$
$$= 5982 \text{ foot-lbs.}$$

$$(b) \quad W = \frac{n}{n-1}\left\{p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]\right\} 144$$
$$= \frac{1{\cdot}4}{{\cdot}4}\left\{15 \times 2\left[(4)^{{\cdot}29} - 1\right]\right\} 144$$
$$= \frac{1{\cdot}4 \times 15 \times 2 \times (1{\cdot}495 - 1) \times 144}{{\cdot}4}$$
$$= 7484 \text{ foot-lbs.}$$

Showing a gain of $7484 - 5982 = 1502$ foot-lbs. in compressing isothermally as compared with adiabatic compression.

This loss—for in most cases all the heat of compression is lost—represents a percentage of

$$\frac{1502}{7484} \times 100 \text{ per cent.} = 20 \text{ per cent. approximately.}$$

The comparison between isothermal and adiabatic compression may be viewed in another light as follows:

For isothermal compression.—Two cubic feet compressed to 60 lbs. absolute pressure requires 5982 foot-lbs. of work.

∴ Cubic feet of air compressed to 60 lbs. per sq. in. per horse-power in the compressor cylinder $= \frac{2 \times 33000}{5982} = 11$ cubic feet nearly.

For adiabatic compression.—Two cubic feet of air compressed to 60 lbs. requires 7484 foot-lbs.

∴ Cubic feet of air compressed per horse-power in the compressor cylinder

$$= \frac{2 \times 33000}{7484} = 8{\cdot}8 \text{ cubic feet nearly.}$$

The following table gives the horse-power required, as calculated from the isothermal and adiabatic diagrams, to compress 1000 cubic feet of free air per minute from atmospheric pressure at sea level up to various pressures:

TABLE II.—HORSE-POWER REQUIRED PER 1000 CUBIC FEET OF FREE AIR PER MINUTE COMPRESSED FROM ATMOSPHERIC PRESSURE AT SEA-LEVEL UP TO VARIOUS GAUGE PRESSURES

Final Gauge Pressure lbs. per sq. in.	Horse-Power for Isothermal Compression.	Horse-Power for Adiabatic Compression	Brake Horse-Power for Actual Practice.
40	85	103	135
50	95	120	155
60	105	134	175
70	113	147	190
80	120	159	205

It should be noticed, of course, that the second and third columns give the theoretical horse-power inside the air-compressing cylinders as represented by the isothermal and adiabatic diagrams, whereas the fourth column gives the horse-power overcoming friction and mechanical defects of the compressor in addition to that absorbed in compressing the air.

The figures in the fourth column are based on the performances of different compressors, and may be taken as a roughly approximate estimate for modern machines.

The table refers to single-stage compressors only

CONNEXION BETWEEN HEAT AND WORK.—The first law of thermo-dynamics states that *heat and work are mutually convertible.*

Thus a given amount of work in foot-lbs. can be expressed as so many British Thermal Units.

It has been found that one B.Th.U. is equal to 778 foot-lbs. of work, and this number is called *Joule's Equivalent of Heat.*

Thus the work done in compressing air, calculated above, may be expressed in heat units instead of foot-lbs. In the above numerical example the heat units required for isothermal compression are:

$$\frac{5982}{778} = 7{\cdot}68 \text{ B.Th.U.};$$

and for adiabatic compression,

$$\frac{7484}{778} = 9{\cdot}62 \text{ B.Th.U.}$$

The loss is therefore

$$9{\cdot}62 - 7{\cdot}68 = 1{\cdot}94 \text{ B.Th.U.}$$

Consequently the loss per horse-power by compressing adiabatically to 60 lbs. per sq. in. absolute

$$= 1{\cdot}94 \times \frac{33000}{7484}$$

$$= 8{\cdot}55 \text{ B.Th.U.}$$

COMPOUND COMPRESSION.—In Fig. 6 (p. 14) the saving that may be effected by compressing in two or more stages is shown graphically. It is seen that at the end of each stage the air is cooled to the initial temperature and that the curve of compression in each stage is represented by the law $p\ v^n$ = constant.

Thus the work done in the first stage is given by

$$\frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \times 144 \text{ foot-lbs.} \quad . \quad . \quad . \quad (1)$$

and the work done in the second stage by

$$\frac{n}{n-1} p_3 v_3 \left[\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} - 1\right] \times 144 \text{ ft-lbs.} \quad . \quad . \quad . \quad (2)$$

and similarly for further stages.

Now in the second stage

$$p_3 v_3 = p_1 v_1,$$
$$\text{and } p_3 = p_2,$$

and so, substituting these values for $p_3\ v_3$, and for p_3, in (1) and (2) we have:

Total work done on the air in two-stage compression

$$= \frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_4}{p_2}\right)^{\frac{n-1}{n}} - 2 \right] \times 144 \quad . \; . \; . \; (3)$$

as compared with

$$\frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] \times 144 \quad . \; . \; . \; (4)$$

if the compression up to the value p_4 had been done in one operation.

EXAMPLE II.—Determine the percentage saving in two-stage compression up to 9 atmospheres absolute as compared with single-stage to the same pressure, assuming compression according to the law $p\,v^{1.4} = K$ in each case.

In the two-stage operation we will assume a first stage up to three atmospheres absolute for reasons explained under the next heading.

In equations (3) and (4) we may leave out the factors common to both, since their omission will not affect the ratio in any way.

Thus for two-stage we have the work done:

$$\propto \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_4}{p_2}\right)^{\frac{n-1}{n}} - 2 \right]$$

$$\propto \left[\left(\frac{3}{1}\right)^{\cdot 285} + \left(\frac{9}{3}\right)^{\cdot 285} - 2 \right]$$

$$\propto 0{\cdot}736$$

And for single-stage the work done is:

$$\propto \left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

$$\propto \left[\left(\frac{9}{1}\right)^{\cdot 285} - 1 \right]$$

$$\propto 0{\cdot}87$$

Therefore the percentage saving

$$= \frac{0{\cdot}87 - 0{\cdot}736}{0{\cdot}87} \times 100$$

$$= 16 \text{ per cent.}$$

For $n = 1{\cdot}3$ the percentage saving is 13 per cent.

If the compression were done in three stages a still greater saving would be effected.

CONDITION FOR HIGHEST EFFICIENCY IN TWO-STAGE COMPRESSION.—In equation (3), p. 29, for maximum efficiency we require the expression

$$\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_4}{p_2}\right)^{\frac{n-1}{n}} \text{ to be a minimum}$$

Let $p^{\frac{n-1}{n}} = \text{P}$ and put

$$y = \frac{\text{P}_2}{\text{P}_1} + \frac{\text{P}_4}{\text{P}_2}$$

Then differentiating y with respect to P_2, which is the only variable on the right-hand side, we have

$$\frac{d\,y}{d\,\text{P}_2} = \frac{1}{\text{P}_1} - \frac{\text{P}_4}{(\text{P}_2)^2}$$

and equating to O and transposing, we have

$$(\text{P}_2)^2 = \text{P}_1\,\text{P}_4$$

$$i.e.\ (p_2)^2 = p_1\,p_4$$

Thus in order to get the minimum work done on the air we must have the pressure at the end of the first stage, viz. $p_2 = \sqrt{p_1\,p_4}$.

EXAMPLE III.—A two-stage compressor is to deliver air at a pressure of 10 atmospheres absolute (147 lbs. per square inch). Find the terminal pressure in the first stage for maximum efficiency.

$$\begin{aligned} p_2 &= \sqrt{p_1\,p_4} \\ &= \sqrt{1 \times 10} \\ &= \sqrt{10} \qquad = 3{\cdot}16 \text{ atmospheres absolute.} \\ & \qquad\qquad\;\; = 46 \text{ lbs. per square inch nearly.} \end{aligned}$$

EFFICIENCY IN SINGLE-STAGE COMPRESSION AND EXPANSION.—In this case we will assume compression and expansion according to the law $p\,v^n =$ constant, n being the same for both compression and expansion. We will assume no loss through leakage or friction in the passage of the air from the compressor to the motor, so that the operation will be similar to that which would occur in an air-tight cylinder in which we have first a volume of air $= v_1$ at a pressure $= p_1$ and temperature $= \text{T}_1$, and after compressing to v_2, p_2, and T_2 the air is allowed to cool to v_3 and T_1 at the pressure p_2, and then, lastly, allowed to expand to v_4, p_1, and T_4 (Fig. 10).

Then the work represented by the compression figure *a b c d*

$$= \frac{n}{n-1}\,p_1\,v_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \times 144$$

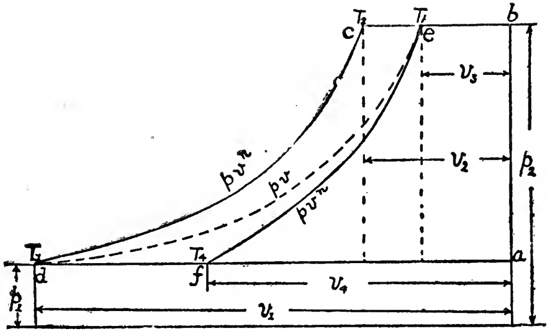


FIG. 10. Efficiency in Single-stage Compression and Expansion.

and the work represented by the expansion figure *a b e f*

$$= \frac{n}{n-1} p_1 v_4 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] \times 144$$

$$\therefore \text{ Efficiency} = \frac{\text{work given out}}{\text{work put in}}$$

$$= \frac{\frac{n}{n-1} p_1 v_4 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] \times 144}{\frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] \times 144}$$

$$= \frac{v_4}{v_1}$$

Now $$p_1 v_1 = p_2 v_3$$

and $$p_2 v_3^n = p_1 v_4^n$$

or $$p_2 v_3 v_3^{n-1} = p_1 v_4 v_4^{n-1}$$

$$\therefore \quad p_2 v_3 = p_1 v_4 \left(\frac{v_4}{v_3}\right)^{n-1}$$

$$\therefore \quad p_1 v_1 = p_1 v_4 \left(\frac{v_4}{v_3}\right)^{n-1}$$

or $$\frac{v_4}{v_1} = \left(\frac{v_3}{v_4}\right)^{n-1} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}} = \textit{Efficiency}$$

EXAMPLE IV.—An air-compressor delivers air to an underground haulage-engine, which uses the air expansively. The delivery pressure is equivalent to five atmospheres absolute (*i.e.* 58·8 lbs. per square inch gauge).

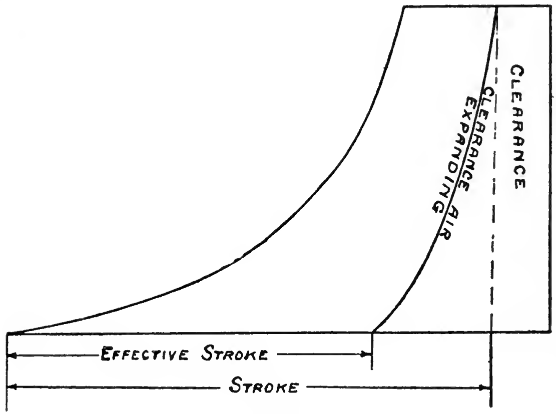


FIG. 11. Effect of Clearance.

If the index of compression and of expansion is 1·35 and the air enters the cylinders underground at the same temperature as it enters the compressor cylinder, determine the maximum theoretical efficiency.

$$\text{Efficiency} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}$$

$$= \left(\frac{1}{5}\right)^{\frac{1\cdot35-1}{1\cdot35}}$$

$$= \left(\frac{1}{5}\right)^{\cdot26}$$

$$= \cdot658 = 65\cdot8 \text{ per cent.}$$

Using the same value for n, the student can find the efficiencies for delivery-pressures of 2, 3, 4, and 6 atmospheres absolute. The results are 83·5, 75·1, 69·3, and 62·7 per cent. respectively.

It is thus seen, as pointed out elsewhere, that the higher the pressure the lower the efficiency. This, of course, only holds for single-stage compression and expansion.

EFFECT OF CLEARANCE.—The effect of clearance is to reduce the length of the *effective* stroke of the piston (Fig. 11). The air in the clearance volume has to expand down to atmospheric pressure before the inlet valves open. Thus the clearance space reduces the volumetric efficiency.

Let C = clearance fraction.
V = volume of piston displacement.
$C\,V$ = clearance volume.
a = volume occupied by pressure air in clearance on expanding.

Then the air in the cylinder at the end of the suction stroke = $V + C\,V = V\,(1 + C)$ and air actually drawn in and discharged, assuming no leakage past valves or piston = $V\,(1 + C) - a$.

Assuming isothermal expansion of the air in the clearance, and calling P_1 the pressure of the air at the commencement of compression, and P_2 the final or delivery pressure,

$$\text{Then } a\,P_1 = C\,V\,P_2$$

$$\therefore \quad a = C\,V\,\frac{P_2}{P_1}$$

Substituting this value of a in the above expression we obtain the volume discharged:

$$= V\,(1 + C) - C\,V\,\frac{P_2}{P_1}$$

Thus the Volumetric Efficiency (*see* p. 34) may be stated as

$$\frac{\text{Actual Volume discharged}}{\text{Volume of Piston displacement}}$$

$$= \left\{\frac{V\,(1 + C) - C\,V\,\dfrac{P_2}{P_1}}{V}\right\} \times 100$$

$$= \left\{(1 + C) - C\,\frac{P_2}{P_1}\right\} \times 100$$

If the expansion of the clearance air is assumed adiabatic, the expression for the efficiency is

$$\left\{(1 + C) - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right\} \times 100$$

It should be pointed out that the volumetric loss due to clearance is *less* in two-stage compression than in single-stage for the same final pressure, as the clearance air in the L. P. cylinder expands from a lower terminal pressure.

EFFICIENCIES OF COMPRESSORS.—The efficiency of a compressor is tested in three ways:

(1) Volumetric Efficiency.
(2) Efficiency of Compression or Thermal Efficiency.
(3) Mechanical or Overall Efficiency.

The thermal efficiency is sometimes called the *air efficiency.*

Volumetric Efficiency.—The ratio of the actual volume of air delivered per minute by the compressor to the theoretical volume corresponding to the displacement of the air-compressing piston in the same time is called the *volumetric efficiency* of the compressor.

The quantity of air delivered by the compressor is frequently measured by noting the number of strokes of the piston required to raise the pressure in the receiver from atmospheric or other pressure to some higher value. The volume so obtained must be reduced to the volume of equal mass at atmospheric pressure and temperature.

Or the quantity delivered can be measured by some form of air-meter, the necessary corrections being again made.

Let A = cross-sectional area of air-cylinder or piston in square inches.
a = cross-sectional area of piston-rod in square inches.
L = length of stroke in feet.
N = number of strokes per minute.
V = actual volume delivered per minute corrected to atmospheric pressure and temperature.

Then, assuming that the compressor is double-acting and that the piston-rod is on one side of the piston only, as is usual, we have:

$$\text{Volumetric Efficiency} = \frac{\text{air delivered}}{\text{piston displacement}}$$

$$= \frac{\text{V (cubic feet)}}{\left(\frac{\text{A} + (\text{A} - a)}{2} \times \frac{1}{144}\right) \times \text{L} \times \text{N}}$$

Efficiency of Compression or Air Efficiency.—We have seen that heat is produced during the process of compression, and that it is necessary to extract as large a proportion of this heat as possible from the working stuff while the air is still in the cylinder.

If all the heat could be removed as rapidly as it was generated, compression would be performed under ideal conditions and the efficiency would be the maximum. This, however, is not obtainable in practice, and the measure by which we fall short of the ideal represents the loss in efficiency as regards the work of compression.

Let E_c = efficiency of compression per cent.
W_i = work done per stroke if the air had been compressed isothermally.
W_a = work per stroke actually done as obtained by scaling the air indicator diagram.

$$\text{Then } E_c = \frac{W_i}{W_a} \times 100 \text{ per cent.}$$

The isothermal diagram can be easily constructed, and the areas of it, and the indicator diagram, obtained from the compressor, can be computed by a planimeter or by some graphical method.

Then we have

$$E_c = \frac{\text{area of isothermal diagram}}{\text{area of air indicator diagram}} \times 100 \text{ per cent.}$$

The value of E_c can also be calculated if we know the value of n, the index of the compression curve, which can be obtained from the actual diagram, several diagrams being taken so as to get an average value.

Then if p_1 = initial pressure in lbs. per square inch,
p_2 = final or delivery pressure,
v_1 = initial volume in cubic feet,
n = compression index,

the actual work done in compression is given by

$$\frac{n}{n-1}\left\{p_1 v_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]\right\} 144$$

while the work done had the air been compressed isothermally is

$$331{\cdot}2\, p_1\, v_1 \log_{\cdot 10} \frac{p_2}{p_1}$$

$$\text{Then } E_c = \frac{331{\cdot}2\, p_1\, v_1 \log_{\cdot 10}\left(\frac{p_2}{p_1}\right)}{\frac{n}{n-1}\left\{p_1 v_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]\right\}144} \times 100 \text{ per cent.}$$

When no cooling is done, as in blowing-engines or in compressors where the final pressure is not more than about four atmospheres absolute, and the air is used warm, the efficiency of compression is referred to the ideal adiabatic curve.

The net theoretical work in foot-lbs. can be calculated from the formula forming the denominator in the above fraction by substituting γ (= 1·408) for n.

$$\text{The ratio } \frac{\text{actual work done on air}}{\text{theoretical work done}} = \text{efficiency.}$$

Further, as practically all the losses reappear in heat which raises the temperature of the discharged air, the efficiency in the cases mentioned can be measured by the thermometer as follows:

If T_1 = temperature of air at inlet,
T_2 = ,, ,, outlet,
T_a = theoretical adiabatic temperature rise,

$$\text{Efficiency} = \frac{T_1 - T_2}{T_a}$$

Two or three per cent. should be deducted from the efficiency as obtained in this way to allow for radiation and bearing friction, depending upon the design.

Mechanical or Total Efficiency of Compressor and Engine.—The total efficiency of a compressor plant is generally measured by the ratio

$$\frac{W}{I}$$

where W = useful work done, as given by the isothermal diagram,
I = indicated horse-power of the engine.

This method, of course, is open to the objection that it saddles the compressor with the inefficiency of the engine, but as the two are inseparable the objection is not a strong one.

Another method is to take the ratio

$$\frac{W}{B} = \text{mechanical efficiency of compressor}$$

where B = the brake-horse-power of the engine.

This, of course, is fairer to the compressor, but the B.H.P. of the engine is not always easily measured.

A third method of measuring the mechanical efficiency of the compressor and engine is to take the ratio $\frac{A}{I}$ where A = the *actual* work done on the air as obtained from the air-indicator diagram.

This, again, is open to the objection that it does not take into consideration the efficiency of compression or air efficiency.

The first method stated is the most general.

The useful work W can be calculated from the equation

$$W = 331{\cdot}2\, p_1\, v_1 \log._{10} \frac{p_2}{p_1} \text{ foot-lbs. per stroke,}$$

or the isothermal diagram can be constructed and the work represented by it computed. Also the mean effective steam pressure can be obtained from the steam-indicator diagram, and the I.H.P. of the engine calculated.

Or, if the drive is electrical the E.H.P. of the motor can be obtained by

$$\frac{\text{watts}}{746}.$$

Then mechanical efficiency of the plant

$$= \frac{331{\cdot}2\ p_1\ v_1 \log\left(\frac{p_2}{p_1}\right) \times \frac{\text{N}}{33000}}{\text{I.H.P. of engine or E.H P. of motor.}}$$

TEST OF A REAVELL COMPRESSOR

The under-noted particulars were obtained on a test of a Reavell quadruplex, single-stage air-compressor

The compressor was of the portable motor-driven type described in Chapter VIII.

Rated size of machine	230 cubic feet of free air per minute
Diameter of cylinders	10 inches
Length of stroke (effective) . . .	$4\frac{3}{4}$ inches
Average revolutions per minute . . .	325
Average delivery pressure	70 lbs. per square inch
Capacity of air receiver	93 cubic feet
Revolutions taken to fill receiver . .	595
Mean temperature of air in receiver . .	100° F.
Mean temperature of air in suction . .	65° F
Mean temperature of air at delivery of compressor	160° F.
Actual mean effective pressure from diagram at 65° F. inlet and 160° F. outlet temperature	30 lbs.
Average volts	460
Average amperes	68
Motor efficiency	·88

CALCULATIONS

(*a*) *Volumetric efficiency.*—Each cylinder is fitted with a pilot piston in front of the main piston, the pilot piston being 3 inches in diameter. There are four cylinders.

Thus the volume swept out by the machine per minute

$$= \frac{100 - 9) \times {\cdot}7854 \times 4{\cdot}75 \times 4 \times 325}{144 \times 12}$$

$$= 255 \text{ cubic feet per minute.}$$

The equivalent volume of free air delivered to the receiver, reduced to the inlet temperature of 65° F., was 226·5 cubic feet per minute.

Thus the volumetric efficiency

$$= \frac{226{\cdot}5 \times 100}{255} = 89 \text{ per cent.}$$

(*b*) *Efficiency of Compression.*—Delivery pressure is 70 lbs. per square inch (= 84·7 lbs. absolute), and if we assume the volume delivered to the receiver to be the actual volume handled by the compressor, then the horse-power for isothermal compression

$$= \frac{331{\cdot}2\, p_1\, v_1 \log._{10}\left(\frac{p_2}{p_1}\right)}{33000}$$

$$= \frac{331{\cdot}2 \times 14{\cdot}7 \times 226{\cdot}5 \times \log._{10}\left(\frac{84{\cdot}7}{14{\cdot}7}\right)}{33000}$$

$$= 25{\cdot}5 \text{ H.P.}$$

Also the H.P. represented by the indicator diagram

$$= \frac{P\,L\,A\,N}{33000} \times (4 \text{ cylinders})$$

$$= \frac{30 \times 255 \times 144}{33000} = 33{\cdot}3 \text{ H.P.}$$

$$= \text{Indicated Horse Power.}$$

Hence the Efficiency of Compression

$$= \frac{25{\cdot}5}{33{\cdot}3} = 72 \text{ per cent.}$$

(*c*) *Overall Efficiency of Plant.*

E.H.P. of motor

$$= \frac{460 \times 68}{746} = 41{\cdot}95 \text{ H.P.}$$

B H.P f motor (with ·88 efficiency)

$$= 41{\cdot}95 \times {\cdot}88$$
$$= 36{\cdot}9 \text{ H.P.}$$

Thus efficiency from the power supplied to the motor to isothermal compression

$$= \frac{25{\cdot}5}{41{\cdot}95} \times 100 = 60{\cdot}8 \text{ per cent.} = \left\{ \begin{array}{c} \text{Overall Efficiency} \\ \text{of Plant} \end{array} \right.$$

and the efficiency from actual power on the motor shaft to isothermal compression

$$= \frac{25 \cdot 5}{36 \cdot 9} \times 100 = 69 \text{ per cent.} = \begin{cases} \text{Overall efficiency} \\ \text{of Compressor.} \end{cases}$$

The separate mechanical efficiency of the compressor is

$$\frac{\text{Air indicated H.P.}}{\text{B.H.P. of motor}}$$

$$= \frac{33 \cdot 3}{36 \cdot 9} = 90 \text{ per cent.}$$

CHAPTER III

INDICATOR DIAGRAMS

INDICATOR diagrams are taken from an air-compressor cylinder in much the same way as they are obtained from a steam-engine.

It is advisable to take indicator diagrams at regular intervals—say every alternate week-end. Very valuable information may be gained thereby as to the conditions inside the compressor, *e.g.* the state of the valves and piston, the efficiency of the cooling arrangements, the extent of wire-drawing or throttling in the suction pipe and inlet valves, etc.

The actual diagram representing the pressure and volume during compression is similar in general outline to either the isothermal or the adiabatic diagram. Fig. 12 shows a typical diagram from a single-

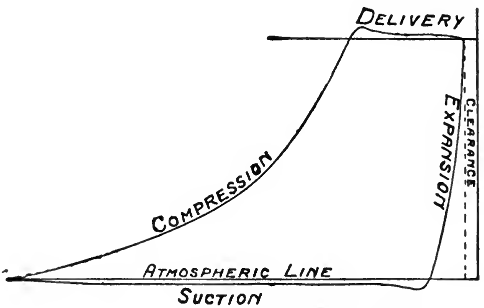


FIG. 12. Typical Indicator Diagram.

stage compressor having automatic valves, in good condition. It will be noticed that on the suction stroke the pressure falls somewhat below the atmospheric line before the inlet valves open, but thereafter if the design is a good one the suction line closely approximates to the atmospheric line and may even coincide with it, and at any rate at the end of the stroke the two will meet.

The compression curve will lie somewhere between the pure isothermal and adiabatic, but nearer the latter than the former; and on the delivery valve opening, a slight hump on the pressure line will be followed by a fall nearly to the straight line of the receiver pressure.

It is questionable if the straight-line delivery is the ideal, for on the delivery valve opening there will be a surge towards the receiver,

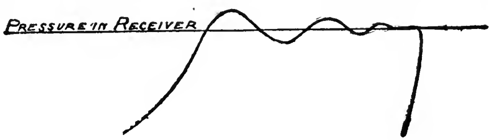


Fig. 13. Ideal Line of Discharge.

and this will be followed by a backward surge, and so instead of a straight delivery we shall have a wavy pressure line during discharge, the amplitude of the wave falling off towards the end of the stroke (Fig. 13).

The next diagram (Fig. 14) exhibits several bad features. The considerable dip on the diagram at *a* indicates that there is too great a resistance to the opening of the inlet valves, *e.g.* springs too tight.

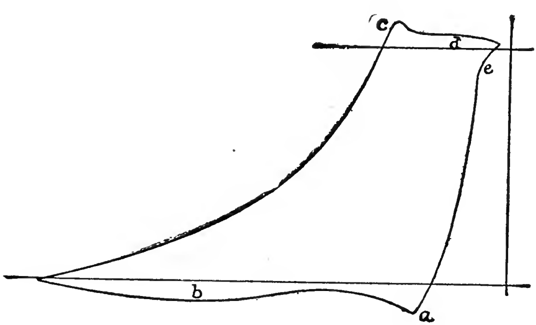


Fig. 14. Diagram showing Defects of Compressors.

The fact that the suction line *b* is well below the atmospheric line throughout nearly the whole stroke, indicates that there is throttling or wire-drawing of the air either through too small opening of inlet valves or too long and narrow a suction pipe. Tight springs are again indicated at *c* at the top of the diagram, and the fact that the pressure line *d* only slowly falls to the receiver pressure, shows too small an area of opening of delivery valves or too small a delivery pipe. Also the abrupt concavity at *e* means that the delivery valves close too sluggishly, thus allowing some of the pressure-air that has been forced through the valves to surge back as the piston starts the next suction stroke. It is not likely that the defects mentioned would occur on one and the same diagram, and the figure is intended to illustrate defects that might occur in different compressors.

In contradistinction to the valve having too tight springs, if the latter are too slack there will be chattering of the valves, and this is an objectionable feature and has to be prevented.

The proper tension of the springs is found by trial.

The next diagram (Fig. 15) illustrates the effect of a leaky piston. The pressure towards the end of the suction stroke has risen to *above* atmospheric at ***a*:** this shows that as the pressure on the other side

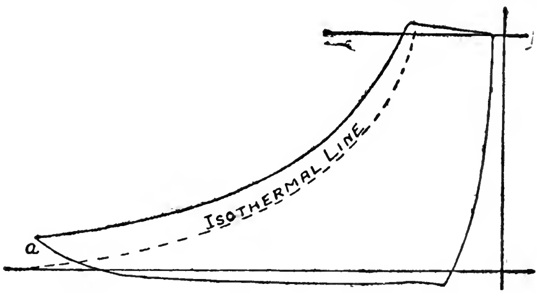


FIG. 15. Diagram indicating Leaky Piston.

of the piston increases, some air is forced past the side of the piston and so the pressure rises on the suction side. This leakage of air from the compressing side of the piston means that the compression curve will approximate more closely to the isothermal, implying a higher efficiency of compression than is actually the case.

This latter defect might, however, exist without the pressure on the suction stroke on the indicator diagram being above the atmospheric line, for in the case of excessive throttling of the inlet air the leakage past the piston might just compensate for the shortage due to throttling. It is therefore advisable to plot the isothermal line on the diagrams taken from the compressor, or, as an alternative, to determine the compression index.

Leaky suction valves will also cause the compression line to approach the isothermal, and the expansion line of the clearance volume will fall too steeply, giving an apparently longer effective suction stroke than is actually the case

If the compression curve or the index approaches abnormally close to the isothermal, the condition of the piston should be examined. On the other hand, if the compression curve is hardly distinguishable from the pure adiabatic either something is wrong with the cooling arrangements or else the discharge valves are leaking badly, and this should be put right at once. By trying the temperature of the delivery air it can be seen if the cooling is defective.

It should be pointed out that the ordinary indicator springs used in obtaining the diagram are much too stiff for giving an accurate record of the suction line, and it is best to take the suction line separately with a weak spring, *e.g.* one of about 5 lbs. to the inch.

TO DETERMINE THE INDEX OF THE COMPRESSION CURVE ON AN INDICATOR DIAGRAM

It is sometimes required to find the compression index on an actual diagram to get an idea of the efficiency of the cooling.

The method is as follows: Let Fig. 16 represent the indicator diagram. To the same scale as that of the diagram itself set down at a distance

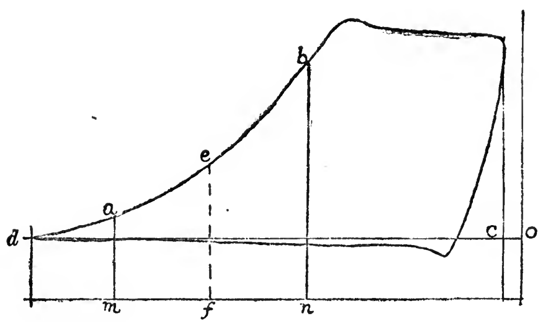


FIG. 16. Method of finding Value "n" from the Indicator Diagram.

representing 14·7 lbs. per square inch, or if at some altitude above sea-level, the pressure of the atmosphere at the place, the absolute zero pressure line as indicated. Choose some smooth portion of the compression curve and, taking two points a and b, drop perpendiculars $a\ m$ and $b\ n$ to the zero pressure line. Knowing the clearance, O C, the volumes when the absolute pressures are $m\ a$ and $n\ b$, will be $o\ m\ (= v_1)$ and $o\ n\ (= v_2)$ respectively.

Then we have, calling $m\ a = p_1$ and $n\ b = p_2$,

$$p_1\, v_1{}^n = p_2\, v_2{}^n$$

Taking logarithms of both sides:

$$\text{Log. } p_1 + n \log. v_1 = \log. p_2 + n \log. v_2\,;$$

$$\therefore\ n (\log. v_1 - \log. v_2) = \log\ p_2 - \log. p_1$$

$$\text{or } n = \frac{\log. p_2 - \log. p_1}{\log. v_1 - \log. v_2}$$

Knowing the value of the piston displacement and the clearance volume, we know that the length of the line *o d* corresponds to this, and hence we can find the actual volume corresponding to v_1 and v_2 We can also measure *m a* and *n b*, and, knowing the scale of the diagram, we can determine the values of p_1 and p_2. The value of n is then easily obtained.

Or we can just take the lengths v_1, v_2, p_1, p_2 as measured from the diagram, and so calculate the value of n.

EXAMPLE I.—Let the measured values be as follows. $v_1 = 1{\cdot}82$, $p_1 = {\cdot}15$; $v_2 = {\cdot}84$, $p_2 = {\cdot}41$.

Multiplying these throughout by 100 for convenience, we have

$$n = \frac{\log. 41 \ - \log. 15}{\log. 182 - \log. 84}$$

$$\text{whence } n = \frac{{\cdot}4367}{{\cdot}3358} = 1{\cdot}3$$

The clearance volume can be obtained from the manufacturers, or it can be determined once for all by finding the *volume of water* that the cylinder holds behind the piston when the latter is at one end of its stroke.

Another way of finding "*n.*"—The value of n can also be obtained as follows:

$$p\,v^n = \text{C}\,;$$

and since volume $\propto$ length of cylinder we can write

$$p\,l^n = \text{C}$$

$$\therefore\ \log. p + n \log. l = \log. \text{C} = \text{constant.}$$

Thus by taking p and l from the indicator card for several points on the smooth part of the compression curve, and taking the logarithms of these points and plotting log. p against log. l we get the value of n from the slope of the line.

To obtain accurate results, from 5 to 8 points should be chosen. To show how the method is carried out, suppose 3 points on the compression curve in Fig. 16 be chosen.

Let $a\,m = p_1$, $e\,f = p_2$, $b\,n = p_3$, and let $o\,m = l_1$, $o\,f = l_2$, $o\,n = l_3$.

Tabulate as follows:

Points.	Pressures.	Lengths.	Log. p.	Log. l.
	lbs. abs.	feet.		
1	26	2·87	1·4150	0·4579
2	34	2·20	1·5315	0·3424
3	57	1·53	1·7559	0·1847

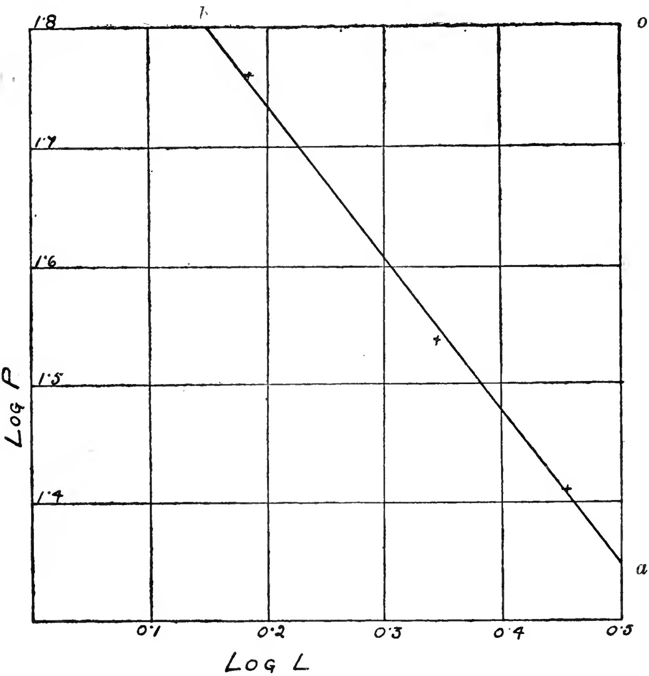


Fig. 17. Graphical Method of finding Value of "n."

Then, plotting log. p against log. l as in Fig. 17, we get

$$\text{“}n\text{”} = \frac{o\,a}{o\,b} = \frac{4\cdot5}{3\cdot5} = 1\cdot285.$$

Or "n" may be found from the graph as follows:

When log. $p = 1{\cdot}8$, log. $l = 0{\cdot}15$,

$\therefore\ 1{\cdot}8 + 0{\cdot}15\,n = \text{constant} \quad . \quad . \quad . \quad (1).$

When log. $p = 1{\cdot}35$, log. $l = 0{\cdot}5$,

$\therefore\ 1{\cdot}35 + 0{\cdot}5\,n = \text{constant} \quad . \quad . \quad . \quad (2).$

Subtract (1) from (2) and we get

$$- {\cdot}45 + {\cdot}35\,n = 0$$

$$\text{or } {\cdot}35\,n = {\cdot}45\,;$$

$$\therefore\ n = \frac{{\cdot}45}{{\cdot}35} = 1{\cdot}285.$$

This is an accurate method if several points are used, and a straight-line graph is drawn evenly amongst them.

TO OBTAIN THE MEAN EFFECTIVE PRESSURE FROM THE INDICATOR DIAGRAM

It is often necessary to determine the mean effective pressures on the steam piston and on the air piston, in order to be able to calculate

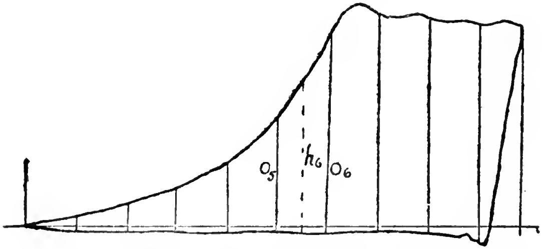


FIG. 18. Method of finding Mean Effective Pressure.

the indicated horse-power of the engine and of the compressor. The method in each case is the same, so it will be sufficient to explain the procedure in connexion with the compressor.

Method (1).—Measure up the area of the indicator diagram by means of a planimeter, or otherwise, and divide this by the length of the diagram corresponding to the stroke of the piston. This will give the *mean* or *average* height of the diagram. The mean effective pressure is then found by multiplying the average height by the number of the indicator spring.

Method (2).—Divide the diagram into ten or more portions by equidistant ordinates, as in Fig. 18.

Then measure the height of each portion *at its mid-point,* as indicated by the dotted line between one pair of ordinates. Add these heights together and divide by their number. Or the height of each strip at the mid-point can be determined by adding together the ordinates bounding it and dividing by 2.

Thus $h_6 = \dfrac{o_5 + o_6}{2}$

Then mean height $= \dfrac{h_1 + h_2 + \cdots\cdots\cdots + h_n}{n}$

Then multiplying the average height so obtained by the number of the spring of the indicator, the mean effective pressure is obtained as before.

If A = area of piston in square inches,
L = length of stroke in feet,
N = number of strokes per minute,
P = mean effective pressure in lbs. per square inch,

Then Air Indicated Horse-Power

$$= \frac{\text{P L A N}}{33{,}000}$$

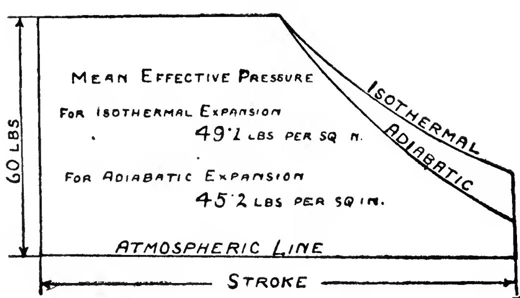


FIG. 19. Mean Effective Pressure in Expansion.

MEAN EFFECTIVE PRESSURE DURING EXPANSION.—It is well known that by taking advantage of the expansive properties of steam more work can be got out of a given weight than would otherwise be the case. The same fact is applicable to compressed air, and hence in the air-engine, wherever practicable, provision should be made for cutting off the supply of air to the cylinder at some more or less early point in the stroke.

There is a difference, however, between the mean effective pressures during expansion in the two cases, because whereas steam expands approximately on the isothermal curve, air expands adiabatically in the cylinder. Thus, as is shown in Fig. 19, for the same initial pressure and the same cut-off point the mean effective pressure on the piston throughout its stroke is *less* in the case of air than in the case of steam, since the terminal pressure is less for air.

EXAMPLE II.—In a haulage air-engine cut-off takes place at half-stroke. If the initial gauge pressure is 50 lbs. per square inch, find the pressure at the end of the stroke, assuming (*a*) isothermal expansion, (*b*) adiabatic expansion.

$$(a)\ p_1 v_1 = p_2 v_2 \qquad \therefore\ p_2 = \tfrac{1}{2} p_1$$

$$= \frac{(50 + 14{\cdot}7)}{2} = 32{\cdot}35 \text{ lbs. absolute}$$

$$= 17{\cdot}65 \text{ lbs. (gauge)}$$

$$(b)\ p_1 v_1^n = p_2 v_2^n$$

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^n$$

$$\therefore\quad p_2 = p_1 \left(\frac{v_1}{v_2}\right)^n = 64{\cdot}7\ (\tfrac{1}{2})^{1{\cdot}4}$$

$$= 24{\cdot}5 \text{ lbs. absolute}$$

$$= 9{\cdot}8 \text{ lbs. (gauge)}$$

Since the final pressure is less, the mean effective pressure throughout the stroke will be less in adiabatic expansion than in isothermal expansion, and so the work done will be less.

Table III gives the approximate actual mean effective pressures at various cut-offs and also approximately the volume of free air required for each indicated horse-power developed by the air-engines.

The figures for the mean effective pressures are obtained from calculated values on the assumption that the actual mean value is approximately 75 per cent. of the theoretical.

The economy of using the air expansively will be evident from the figures.

For example, an engine using air at an initial gauge pressure of 60 lbs. per square inch and cutting off at $\frac{1}{2}$ of the stroke would use 13 cubic feet of free air per I.H.P. per minute, and so for a 30 I.H.P. engine the air-consumption would be $30 \times 13 = 390$ cubic feet.

On the other hand, for the same initial pressure but no cut-off the air consumption would be $30 \times 20 = 600$ cubic feet—a difference of 210 cubic feet.

To obtain the quantity of free air per B.H.P. the I.H.P. values requi to be multiplied by 1·25 to 1·3.

TABLE III.—APPROXIMATE ACTUAL MEAN EFFECTIVE PRESSURES AND FREE AIR * USED PER I.H.P. AT VARIOUS CUT-OFFS

Point of Cut-off.	Initial Pressures in lbs. per sq. in. 60 — Mean Effective Pressure.	60 — Cubic ft. of free air per I.H.P. per minute.	70 — Mean Effective Pressure.	70 — Cubic ft. of free air per I.H.P. per minute.	80 — Mean Effective Pressure.	80 — Cubic ft. of free air per I.H.P. per minute.
1	45	20	53	19·5	60	19
$\frac{3}{4}$	43	16	50	15·5	57	15
$\frac{2}{3}$	40·5	15	47	14·5	54	14
$\frac{1}{2}$	34	13	40	12·5	46	12
$\frac{1}{3}$	24	12	28·5	11·5	34	11
$\frac{1}{4}$	18	13·5	22	12·0	25·5	11

TWO-STAGE INDICATOR DIAGRAMS.—These are two separate diagrams, one taken from the L.P. air cylinder and the other from the H.P. cylinder, and so arranged that the H.P. diagram comes on the top of the L.P. diagram.

Fig. 20 shows diagrams from a two-stage compressor of the "Koster" type, and Fig. 21 shows a similar diagram taken from an Alley and MacLellan two-stage machine.

The "Koster" compressor has mechanically operated valves, while the "Sentinel" machine has automatic disc valves. It should be noted that the horizontal or volume scale of the H.P. diagrams is greater than the volume scale of the L.P. diagrams in the ratio

$$\frac{\text{area of L.P. cylinder}}{\text{area of H.P. cylinder.}}$$

* "Free air" means air at the normal atmospheric pressure and temperature.

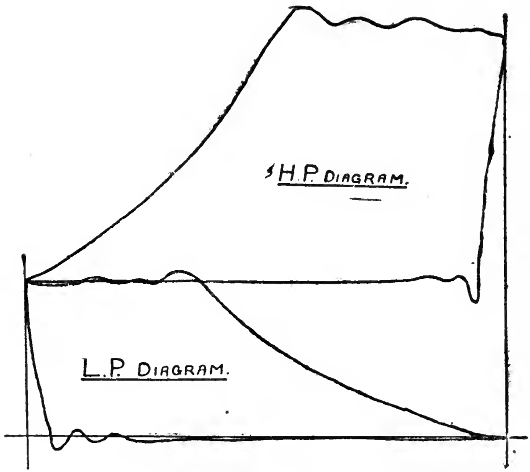


FIG. 20. Diagrams from Koster Two-stage Compressor.

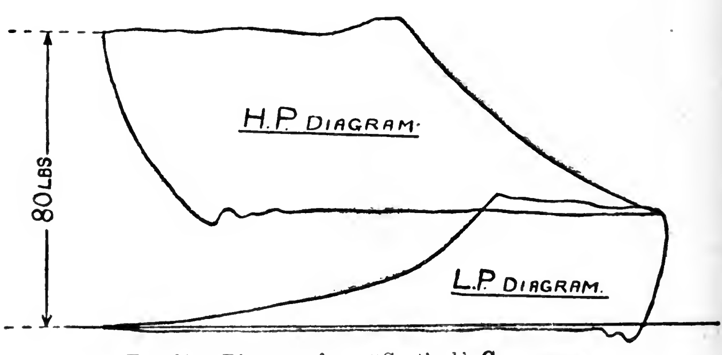


FIG. 21. Diagrams from "Sentinel" Compressor.

CHAPTER IV

RECIPROCATING AIR-COMPRESSORS

THE reciprocating or piston type of compressor, in which the cylinders are placed either vertically or horizontally, is the commonest form at present in use.

The vertical form for a given capacity occupies less floor space and the foundations are less costly. The wear on the piston is less, as it does not rest on, but is only steadied by, the cylinder, and forced lubrication of bearings can be conveniently secured. The speed of revolution is generally higher in the vertical than in the horizontal type.

As regards efficiency there is not much to choose between the two, and although the vertical high-speed engine and compressor has many conspicuous advantages, the modern slow-speed horizontal compressor still has its advocates.

ESSENTIALS OF A GOOD COMPRESSOR.—A good form of compressor should, in a general way, satisfy the following requirements:

(1) Reasonably high mechanical efficiency.

Since this includes the efficiency of the steam-engine driving the compressor it is evident that an inefficient engine will mean an inefficient compressor, so that to obtain a good *overall* efficiency it is essential that the prime mover be satisfactory also.

(2) High volumetric efficiency.

To secure this, the

(*a*) Clearance space should be small.

(*b*) Inlet valves should open readily and have ample area of opening —at least equal to 15 per cent. of cylinder area—so that at the end of the suction stroke the space behind the piston will be completely filled with air at atmospheric pressure.

(*c*) During compression the inlet valves should be air-tight.

(*d*) The delivery valves should open automatically at as little above receiver pressure as possible; they should have ample area of opening, should close smartly, and should be perfectly air-tight during the suction stroke.

(3) High efficiency of compression.

For a single-stage water-jacket compressor the compression index should not be higher than 1·3, showing efficient cooling.

(4) Freedom from valve troubles and break-downs.

(5) The compressor should be so constructed that the inlet air is drawn from as cool a source as possible and is heated to the smallest possible extent in its passage through the suction pipe to the cylinder.

(6) It should not be easy for dust or dirt to get into compressor valves and cylinder. If necessary the air should be strained through canvas or gauze screens to keep out dust.

It is not easy to get a compressor which will completely satisfy all the above desiderata, but a machine which will reach a high standard of overall efficiency and general reliability should and must be the aim of manufacturer and user alike. The reciprocating compressor has been greatly improved in recent years, and, on the whole, modern types, such as those to be described, are very efficient machines.

COMPRESSOR VALVES.—These may be separated broadly into four classes.

(1) Automatic valves, *i.e.* valves which are actuated by difference of air pressure only, the opening and closing being controlled and assisted by springs. Examples of this type are the poppet valve (Ingersoll and Reavell compressors, pp. 64 and 123) and the disc valve (Walker and Sentinel compressors, pp. 57–60).

(2) Mechanically operated valves (Koster compressor, p. 66).

(3) Valves which open automatically by difference of air pressure but are closed mechanically (Riedler compressor, p. 69).

(4) Mechanically operated inlet and automatic delivery valves.

The automatic valve is almost invariably of the multiple type, there being several inlet and delivery valves instead of only one of each kind at either end of the cylinder, as is the case, for example, in the Riedler compressor. By using multiple valves the weight is considerably less for the same area of opening.

One important advantage of the automatic valve is that since it is actuated by difference of air pressure, the inlet valves will not open until the conditions are such that air can only flow *into* the cylinder, not out of it, while the delivery valves will only open when the pressure in the cylinder is greater than that in the receiver. Thus there is no expulsion of pressure-air from the cylinder because of the inlet valves opening too quickly, nor inrush of air from the receiver because of too early opening of the delivery valves, which *might* happen if the valves were opened and closed mechanically at a definite point on the piston stroke. There is, of course, in both cases a loss of energy.

On the other hand, the automatic valve offers some obstruction to the free passage of air, since there must be a definite pressure difference before the valves will open, and this pressure difference may need to be considerable if the valve is heavy or the springs are stiff. Consequently, when the inlet valves close, the air in the cylinder may be at a pressure *less* than atmospheric, meaning a less weight of air drawn in per stroke, and also during delivery the pressure of air in front of the piston will have to be some few pounds greater than the pressure in the receiver.

This is not the case in the mechanically actuated valve, which, opening independently of the air pressure, offers no obstruction to the free passage of the air beyond the frictional resistance common to all air ports, and thus obviates the *under-pressure* during the suction stroke and the *over-pressure* during the delivery stroke referred to above.

Thus obviously there are points for and against both the automatic and the mechanically operated valve.

In the valve which opens automatically but is closed mechanically we have an attempt made to combine the advantages of both types. The Riedler valve is a good example of this form of valve. Descriptions of various kinds of valves are given where the different compressors are described.

COOLING DURING COMPRESSION.—It is necessary, as has already been pointed out elsewhere, to keep down the temperature of the air during compression as much as possible.

This may be done by

(*a*) Having the piston working under water.

(*b*) Spraying water into the cylinder during each compressing stroke.

(*c*) Water-jacketing the cylinder, the piston working dry.

(*d*) Water-jacketing the cylinders and compressing in two or more stages with intercooling between the stages.

Methods (*a*) and (*b*) are now practically obsolete and need not be further considered.

Water-jacketing is the universal method of extracting the heat from the air during compression. The water should be as cold as can be obtained and as free from sediment or impurities as possible.

The last point is important, for if the impurities are such as to deposit a scale on the walls of the water-jacket spaces, a very effective non-conductor of heat is interposed between the water and the metal and the transference of heat to the water is greatly interfered with. Care should therefore be taken to prevent the formation of scale by purifying the water if need be.

The circulation of the water through the jacket should be continuous from a tank overhead or by means of a pump, a float or other means being used to indicate cessation of flow. Every available position around the cylinder should be occupied by water-jackets, and the most effective place is at or near the cylinder ends, as it is there that the temperature of the air during compression reaches its highest point.

INTERCOOLING.—When the working pressure is to be above 60 lbs. per square inch, in any but the smallest sizes of compressors, it is more economical to compress in two or more stages, each being water-jacketed and the air passed through an intercooler between each stage. The gain in efficiency, as is shown in the diagram (Fig. 6), is considerable.

The intercooler consists of a series of tubes through which a current of water is kept flowing (Fig. 24, p. 59). By means of baffle-plates the air discharged from the lower stage is made to pass amongst the tubes, and is thus effectively cooled.

The perfect intercooler would reduce the temperature of the air to the initial temperature, but in practice it is seldom that the cooling is quite so effective as this.

LUBRICATION.—The proper use of the right kind of oil for lubricating the inside of an air-compressing cylinder is a point that requires careful consideration.

In single-stage compression up to, say, 80 to 100 lbs. per square inch, final temperatures in the vicinity of 400° F. may be attained. This temperature is sufficient to cause ignition of the vapours from certain kinds of mineral lubricating oils, some of which have flash-points as low as 300° F. The high compression renders an atmosphere laden with the vapour of such an oil a highly explosive one, and explosions—some attended with loss of life—have occurred in air-compressor cylinders and receivers from this cause, and the matter is consequently one of considerable import.

It should be remembered too that even though an actual ignition may not occur, the high temperature may result in the partial combustion of the carbonaceous elements in the oil, and so result in the production of carbon monoxide—a very poisonous gas.

The remedy is to adopt the precaution of using only high flash-point lubricants (flash-point not less than 550° F.).

The oil should not leave any carbonaceous deposit in the cylinder, and a means of checking this is not to use the oil lavishly but just sufficient to ensure effective lubrication.

METHODS OF DRIVING COMPRESSORS.—The compressor may be direct-driven or through gearing or belting, according as is found convenient. For direct drive the steam-engine is the most suitable for most types of compressors, though some, such as the Reavell (*see* Chapter VIII), can be conveniently driven direct by slow-speed electric motors, and all can be electrically driven with gearing reduction.

The modern compound condensing steam-engine is a very efficient machine, and at collieries where the power-house can be situated close to the boilers the losses in the steam transmission are small. It is, of course, futile to use an inefficient steam-engine to drive a modern air-compressor and still expect a good overall efficiency, and hence in an air-power plant as much consideration should be given to the prime mover as to the air-compressor itself.

Gas-engines are also used to a limited extent for actuating compressors. In many mining districts large quantities of coke-oven or blast-furnace gas are available, and if not, producer plants may be installed. At quite a number of collieries throughout the country gas-engine plants, using coke-oven or producer gas, have been in operation for several years and found very satisfactory as regards economy and reliability. Most of these engines drive electric generators, but there is no reason whatever why similar plants should not be utilized for driving air-compressors wherever supplies of gas are available.

The most suitable type of gas-engine for the purpose is the two-stroke cycle, double-acting engine for single-cylinder engines, but the twin-cylinder, four-stroke cycle type of engine can also be employed.

Automatic control of the speed and output may be secured in a manner similar to that employed with the steam-driven compressors. A heavy fly-wheel is, of course, necessary to secure uniformity of torque, as indeed it is for all methods of driving.

Electric drive through gearing or belt is a favoured method of actuating air-compressors where electric power is available, and for driving compressors placed underground it is practically the only means suitable.

The advantages of the electric drive are many, chief amongst which are:

(1) Compactness.
(2) High efficiency at all loads.
(3) Speed easily varied (D.C. machines).
(4) Starting and stopping simple and instant, and capable of being made automatic.
(5) Uniformity of turning moment

The steam turbine using either live or exhaust steam may also be used for driving high-speed vertical reciprocating compressors through gearing. The turbine is very efficient, and the utilization of exhaust

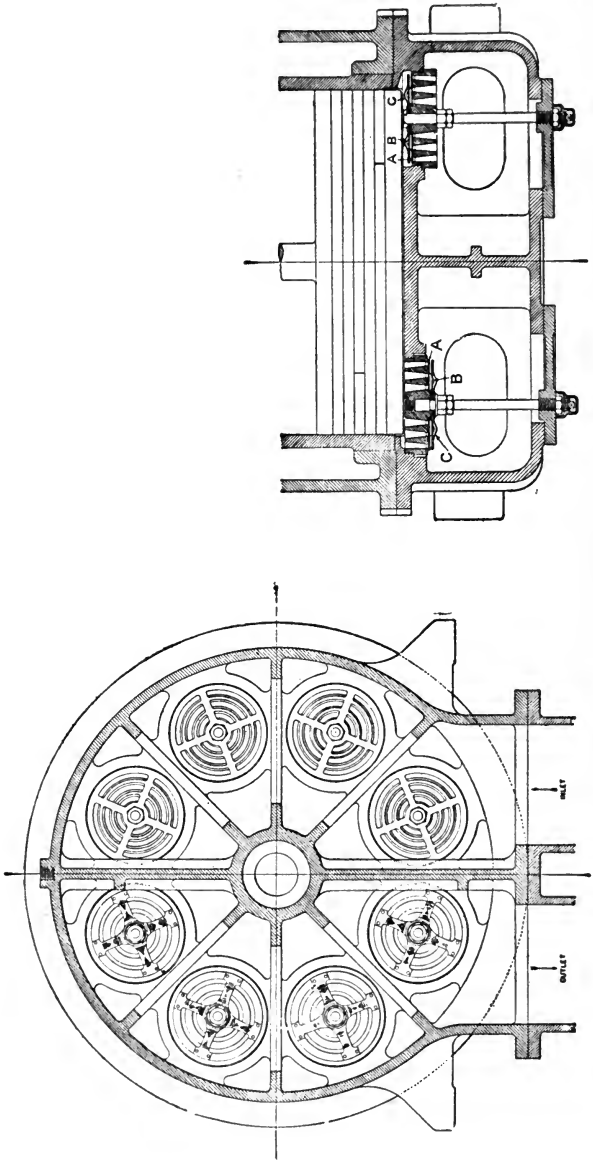


Fig. 22. Disc Valves of Walker Compressor.

steam that would otherwise be wasted constitutes a considerable saving. This method of drive is again considered in connexion with turbo-compressors.

Where water-power is available the compressor may be driven by a Pelton wheel, or water-turbine.

EXAMPLES OF MODERN COMPRESSORS

THE WALKER HORIZONTAL COMPRESSOR

The special feature of this compressor is the patent disc valves employed. These valves, details of which are shown in Fig. 22, consist of speciallv tempered circular plates of steel with concentric openings or ports, and having one side ground to a true face. The seatings are also circular and are provided with openings corresponding with the bars in the valve. A small lift of the valve gives sufficient opening for the free passage of the air.

Referring to Fig. 22, the valve A has four waved or corrugated arms, B, this sinuosity allowing extension or contraction of the arm to take place without distortion of the valve. A steel guard, C, with waved arms similar to the valve gives additional flexibility, and thus resistance to the passage of air is kept very low. The valves work automatically, being actuated by difference of air pressure, and they are placed in the cylinder ends as shown in the figure. The lightness of the valves, the low lift, the freedom from shock, and the ample passage-way combine to give a very satisfactory form of valve.

Fig. 23 shows a Walker horizontal two-stage compressor driven by compound Corliss engines and fitted with the above-described valves.

As will be readily understood from the figures, the piston-rod of the H.P. steam cylinder is continuous with that of the H.P. air cylinder, and the same is the case with the L.P. steam and air cylinders. The steam cylinders are next the fly-wheel, while between the air cylinders is situated the intercooler.

ALLEY AND MACLELLAN'S COMPRESSOR

This compressor is of the vertical, quick-running type.

The action will be evident from the illustration in Fig. 24. In the position shown, the space above the piston forms the low-pressure cylinder, while the annular space below the piston-head is the high-pressure or discharging cylinder.

During the down-stroke of the piston, air from the atmosphere is admitted to the low-pressure cylinder past the multiple-ported valve.

When the motion of the piston is about to be reversed the inlet valve closes the inlet to the low-pressure cylinder, and when further upward motion of the piston has compressed the air the discharge valve opens and permits the discharge of air into the intercooler.

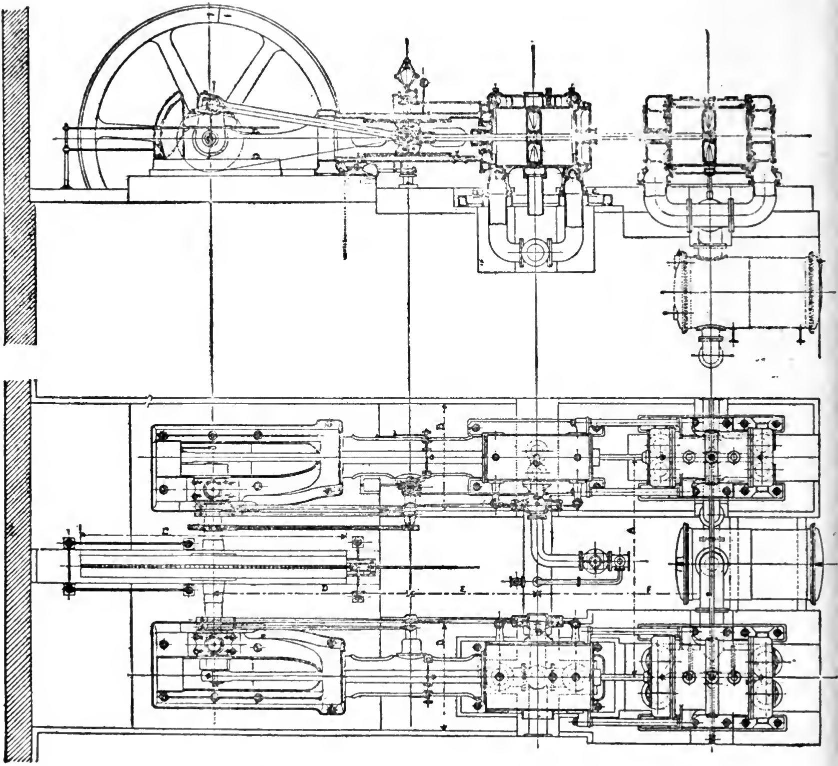


Fig. 23. Plan and Elevation of Two-stage Walker Compressor driven by Compound Corliss Steam-engines.

While the piston has been moving upward, air will have been drawn from the intercooler past the H.P. inlet valve into the annular high-pressure cylinder, and the next down-stroke further compresses the air in that cylinder, until it is discharged at the working pressure past the H.P. discharge valve to the receiver.

Disc valves are used, and their construction will be understood from

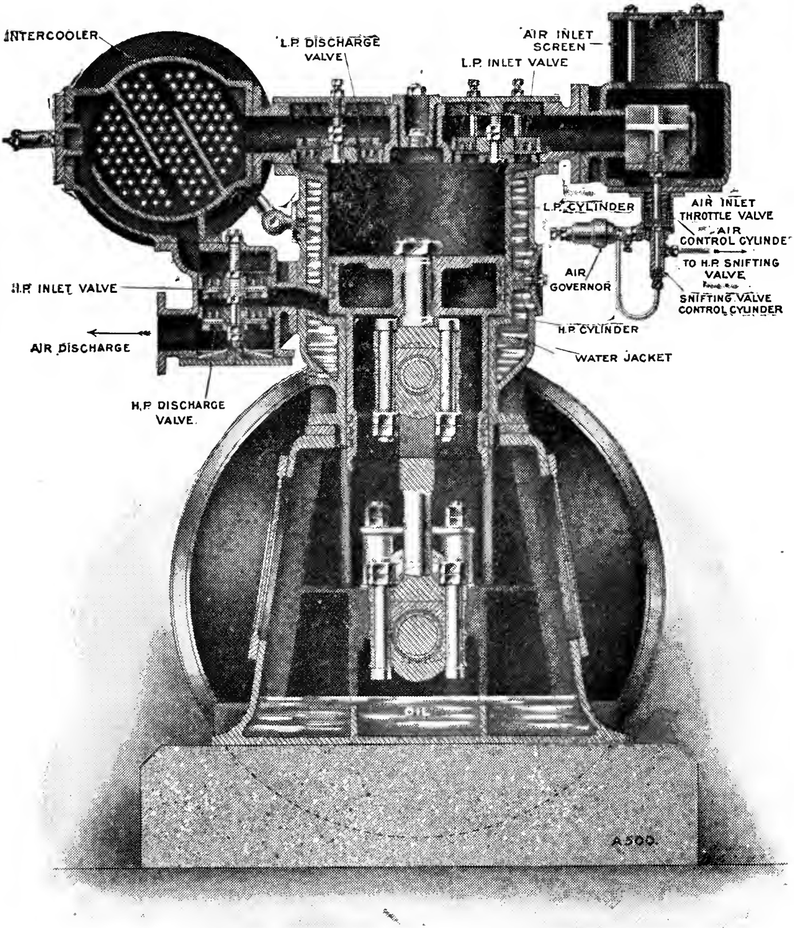


FIG. 24. Sentinel Two-stage Compressor.

Fig. 25. A is the valve, and B its seating; small steel cups, C, working in a guard, D, engage with the valve at E, the extent of opening being controlled by the springs F. The valves are simply circular steel plates with concentric openings which admit air when the plate is raised from its seating. The action is automatic and the valves

are very similar to those in the Walker and Belliss and Morcom Compressors.

By having the valves in the cylinder covers the clearance is kept low, and so a high volumetric efficiency is obtained.

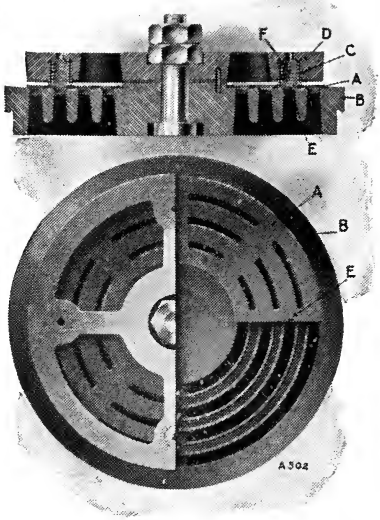


FIG. 25. Section of Valve and Seat (Sentinel Compressor).

BELLISS AND MORCOM COMPRESSOR

A two-stage Belliss and Morcom Compressor is shown in section in Figs. 26 and 27.

Both air cylinders are double-acting, the low-pressure cylinder delivering through an intercooler to the high-pressure cylinder. The arrangement shown is for a belt-driven or motor-driven compressor; in the case of steam-driven compressors the air cylinders are superposed above the steam cylinders and worked by a prolongation of the steam piston-rods. The compressor is very compact and the system of automatic forced lubrication adopted ensures that all bearings are well lubricated, friction and wear and tear thus being reduced to a minimum; a heavy fly-wheel promotes steadiness in running.

The chief feature of the compressor, however, is the use of the Rogler-Hoerbiger valves which are fitted.

Details of the valve are shown in Fig. 28. In this diagram 1 is the valve-seat and 2 the valve-guard, both of cast-iron. The valve-guard

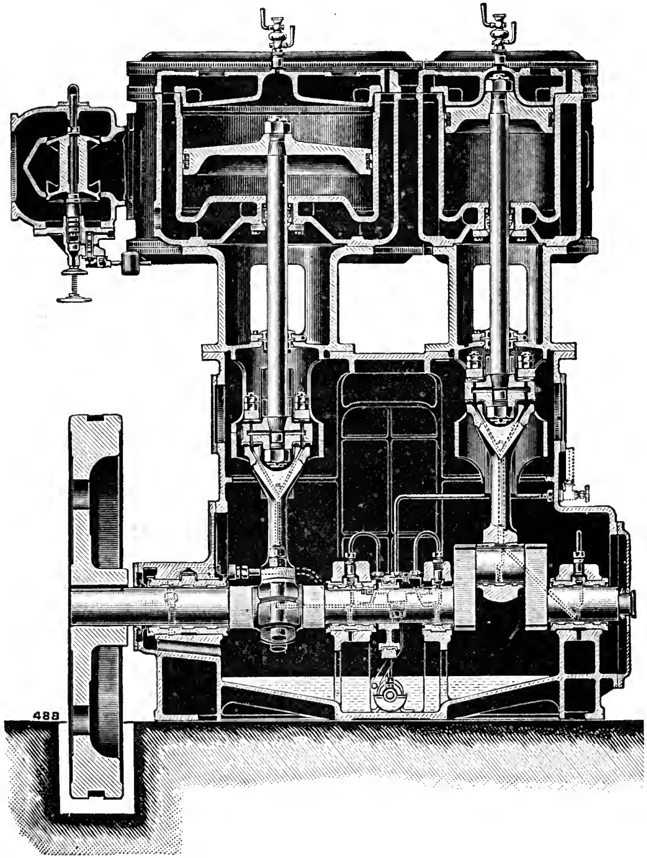


FIG. 26. Belliss and Morcom Compound Compressor.

has recesses for holding the four closing springs, 3, and the valve-plate, 4, is of thin tempered steel perforated to give a multiple opening and rests on the valve-seat. Above the valve-plate comes the cushion-plate, 6, also of thin steel—a lift-washer, 5, being interposed between the cushion-plate and the valve. The function of the cushion-plate is to soften the blow of the valve-plate when the latter lifts.

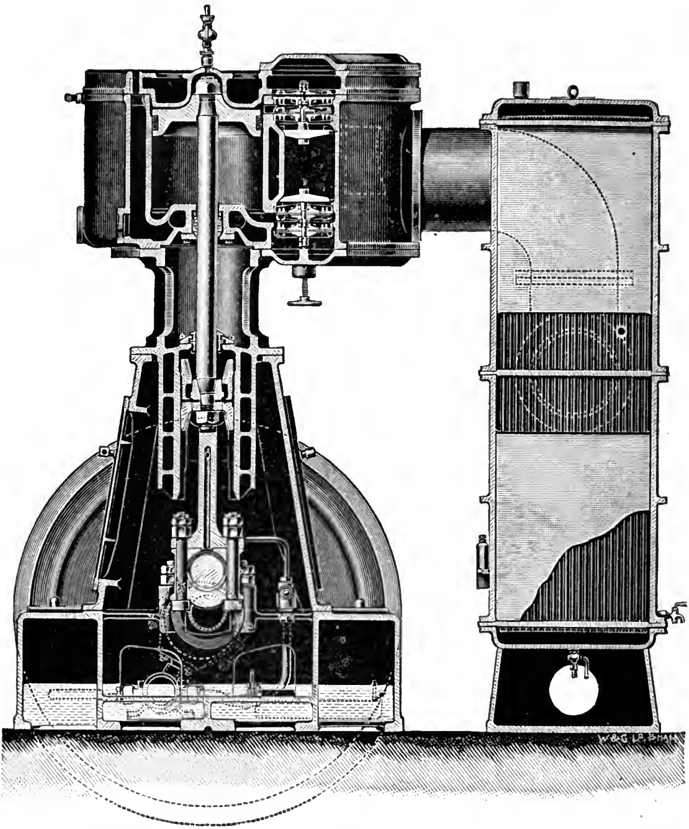


FIG. 27. Belliss and Morcom Compressor.
Section through H.P. cylinder and intercooler.

The complete valve with the various parts in position is shown at 7. The valve-plate, 4, is ground down and specially tempered at 8 to form springs which aid in the movements of the valve.

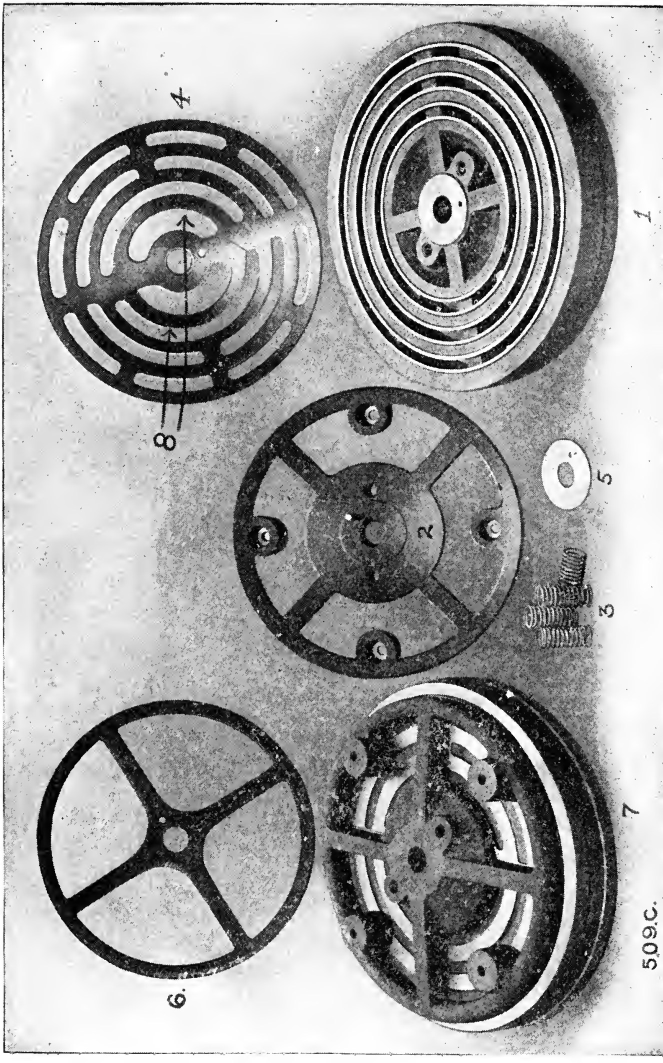


Fig. 28. Details of Valves of Belliss and Morcom's Compressor.

INGERSOLL-RAND COMPRESSORS

The Ingersoll-Rand Company manufacture several forms of compressors; one well-known form is the Imperial Compressor, which is shown in section in Fig. 29.

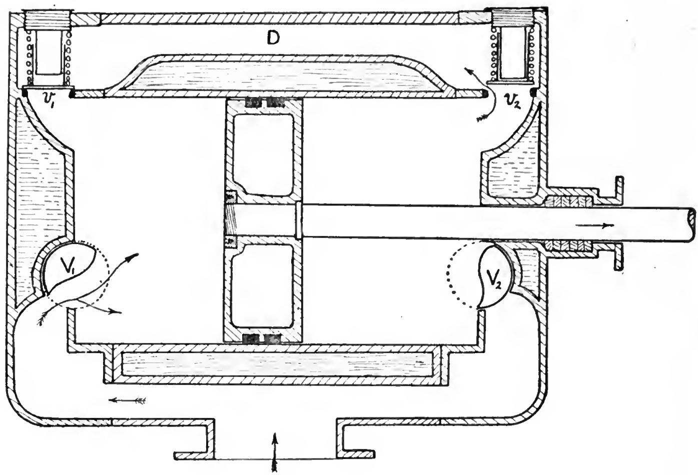


FIG. 29. Imperial Compressor.

The compressor is double-acting, the piston-rod extending on one side only and working through the packing gland as shown.

The inlet valves V_1 and V_2 are mechanically operated from eccentrics on the main shaft, and are semicircular in shape and rotate in a shaped recess.

At the moment the piston is moving in the direction indicated by the arrow marked on the piston-rod, and V_1 is full open, admitting air to the cylinder, while V_2 is closed.

The delivery valves v_1 and v_2—the latter shown open, the former shut—are of the direct-lifting, spring-controlled, poppet type. The air, being discharged through v_2, passes into the space D communicating with the delivery pipe and thence to the intercooler or the receiver, according to whether it has come from the L.P. or the H.P. cylinder, if the compressor is a two-stage one.

With two stages the arrangement is similar to the Walker compressor as regards the steam and air cylinders, *i.e.*, the cylinders are arranged side by side with the fly-wheel between the H.P. and L.P. cylinders. This is indeed the favourite arrangement for compound compressors as giving greater uniformity of turning moment.

HURRICANE INLET COMPRESSOR.—Another type is the so-called Hurricane Inlet Compressor.

In this machine both the inlet and the discharge valves in the high-pressure stage are spring-controlled, direct-lifting.

The delivery valves in the first stage are also of this type. The inlet valves, however, are in the piston itself, and consist of a ring on the front and back of the piston. The piston is provided with a hollow tail-rod, which extends through a stuffing-box in the rear end of the cylinder, and moves inside a larger pipe which leads to the outer atmosphere.

The ring valves in the piston open and close automatically by difference of air pressure, so that on the suction stroke for either side of the piston the valve opens and air is drawn in through the hollow tail-rod and the hollow piston into the cylinder. On the return stroke the pressure of the imprisoned air closes the valve.

The movement of the piston valves is limited by pins which fit into slots cut in the rings, and since the peripheral length of the ring is nearly equal to the circumference of the piston very small movement is sufficient to give ample passage area.

The advantages claimed for the valve are:

(1) Inflow of air not retarded by springs, opening and closing effected by difference of air pressure only.

(2) Since no inlet valves in cylinder covers, more space for water-jacketing.

(3) Cylinder covers simplified.

(4) Air taken in a single stream from outer air.

(5) Valves require little attention.

Obvious objections are:

(1) Piston is hot and air receives more heat on suction stroke than in other types, causing less weight of air drawn in per stroke.

(2) Piston is bulky and hollow tail-rod takes up room. This reduces capacity of cylinder somewhat or means larger cylinder for same volumetric capacity.

(3) An extra stuffing-box required.

Still another type which has recently proved itself a very efficient and reliable machine, made by the same firm, is the Ingersoll-Rogler Compressor.

The valves, which are of the disc type and very similar to those described in connexion with the Belliss and Morcom Compressor, are arranged radially in each end of the cylinder barrel, the inlet valves being at the bottom and the discharge valves at the top. This arrangement allows each cylinder end to be effectively water-jacketed.

BAILEY'S "KÖSTER" AIR-COMPRESSOR

This is a horizontally arranged compressor, and its most noteworthy feature is the method of admitting the air to, and releasing it from, the compressing cylinder.

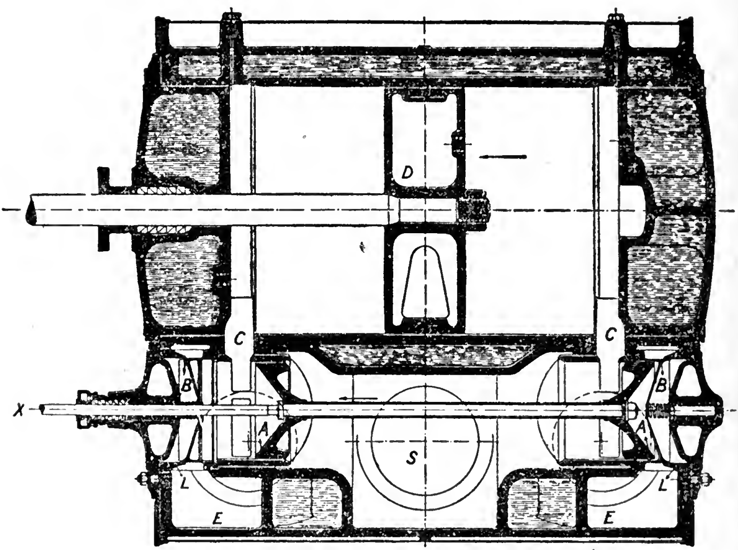


FIG. 30. Köster Single-stage Compressor.

Referring to the diagram in Fig. 30, D is the piston and S the suction pipe through which the air enters to the valve chamber and thence to the cylinder. A A are the two ends of the piston valve, which is moved to and fro by an eccentric rod actuated from an eccentric in the same way as the ordinary slide valve. In the position shown the piston is moving to the left, and the right-hand end of the piston valve has uncovered the inlet port C, so that air enters from S to the right-hand side of the piston.

On the left-hand side the piston valve has also at this stage uncovered the other port, *C*, but the passage to the discharge chamber *E* is still closed by the spring-controlled valve *B*.

When the final pressure is reached, however, due to the further travel of the compressor piston, the valve *B* is forced open against the slight resistance of the spring and the back pressure of the air from the receiver, and so the pressure-air is discharged into the delivery pipe.

The piston valve will now be returning, and when the compressor piston reaches the end of its stroke the piston valve instantaneously closes the left-hand port. At the same time the right-hand end of the piston valve will have moved into the position which closes the port *C* to the suction pipe *S* and opens it to the delivery. Discharge of air to the delivery cannot take place, however, till the air has been compressed sufficiently to force open the spring-controlled valve B, as has been described.

Thus it will be seen that three points on the compression cycle—namely, the opening of the inlet, the closing of the inlet, and the closing of the discharge—are positively and mechanically controlled; while only one point, the opening of the discharge, which is the variable point in the cycle, is controlled by the automatic spring valve. The automatic valve is, of course, relieved from the necessity of quick closing, this being done by the piston valve.

This arrangement is unique and makes for effective and satisfactory working.

In the "Köster" two-stage compressor (Fig. 31) the differential air piston P is actuated by the connecting-rod R, and the piston air valve by means of the eccentric O and rod S. Free air is drawn in through the suction opening A and passes through the port B to the cylinder, the piston valve C being on the right-hand side of the port at the time. When the compressor piston arrives at the end of its stroke, the piston valve C closes the port B, and on the return stroke the air is compressed.

On the proper pressure being reached, the piston valve C having passed to the left-hand side of the port B, the compressed air is discharged through the spring valve D to F. The first compression stage is now completed, and the air passes through the intercooler to H, and thence to the annular space K, to which access is obtained by the piston valve L having unclosed the port I.

Then, on the differential piston moving to the left, the air in the annular space is raised to the final pressure and discharged through the port I to the spring discharge valve and into the delivery pipe at M.

The compression stroke in the one stage is thus the suction stroke in the other.

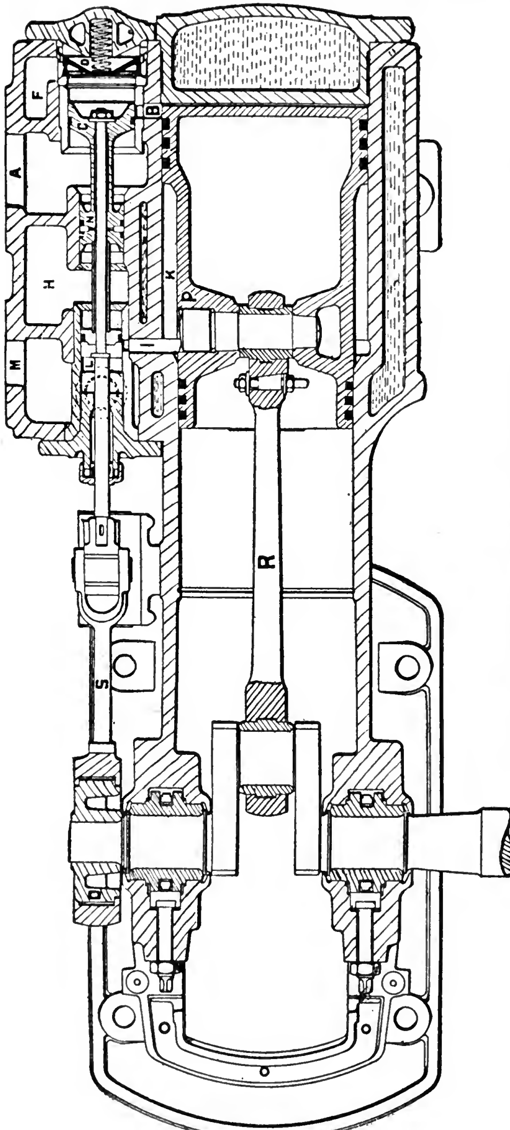


Fig. 31. Köster Two-stage Compressor.

The compressor shown in Fig. 31 is belt-driven from a steam-engine or electric motor. An enlarged sectional view of the valve gear of the low-pressure end is shown in Fig. 32.

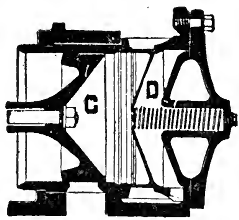


FIG. 32. Enlarged Section of Piston Valve.

THE RIEDLER COMPRESSOR

This compressor is built in both vertical and horizontal forms, and is characterized by the fact that the air valves, both inlet and delivery, are opened automatically and closed mechanically.

There is only one inlet and one delivery valve for each side of a double-acting compressor, the valves being at the ends of the cylinders.

Fig. 33 shows sections through the inlet and outlet valves; their action is as follows:

At the beginning of the suction-stroke of the piston the inlet valve opens automatically by difference of air pressure, a dash-pot arrangement being provided at A to prevent the valve slamming open.

At the end of the stroke the valve is closed mechanically by a cam arrangement on a wrist-plate, causing the valve spindle to act on the valve at the points *a a*.

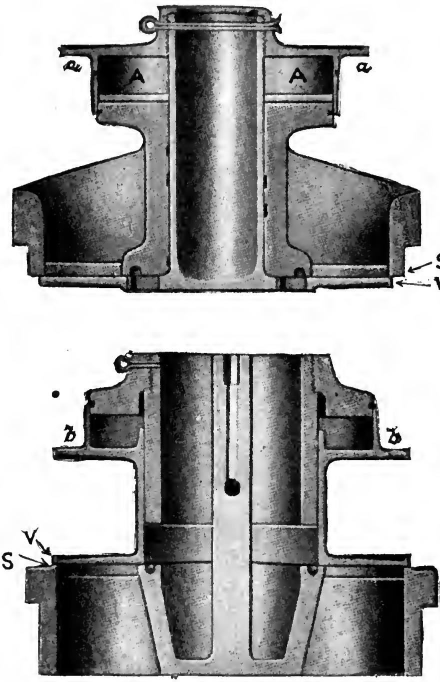
This action closes the valve until only a narrow space is left between the valve V and its seat S, and the valve is closed the remaining distance by the pressure of the air. The valve is shown in the closed position.

The outlet valve, which is also shown closed, opens automatically when the air in the compressor cylinder slightly exceeds the delivery pressure This valve remains full open until the piston has travelled nine-tenths of its stroke, when it begins to close by the spindle acting upon the surface *b b*, and is closed completely at the end of the stroke.

The movement to operate the gear is, in the case of a Corliss engine, taken from a wrist-plate, but may be taken from any convenient part of the motion. All that is required is that the valve spindle shall receive a rocking motion.

This automatic opening and mechanical closing of the valves allows the compressor, whether vertical or horizontal, to be driven at high speeds. There is no tendency for the valve to close until the right moment is reached, springs being totally absent.

Inlet Valve.



Outlet Valve.

Fig. 33. Valves of Reidler Compressor.

THE ROBEY AIR COMPRESSOR

This compressor is made in two forms, horizontal low-speed and vertical quick-revolution compressors.

The distinguishing feature of the Robey Compressor is the disc-plate type of air valve used, which is corrugated in form and is controlled by coiled springs of square section.

It is multiple-ported and thus very small lift is required, and the valves are consequently perfectly silent in action and the wear and tear is low.

THE BROTHERHOOD COMPRESSOR

This compressor is of the vertical type. Forced lubrication is used in the double-acting units, which are used for all but very small outputs. For portable use and outputs up to about one hundred cubic feet per minute single-acting trunk pistons are used.

The valves are of the steel-disc form, held in position by light springs, and situated at the side of the cylinder in the smaller sizes.

In the larger machines the valves are arranged in the cylinder-heads.

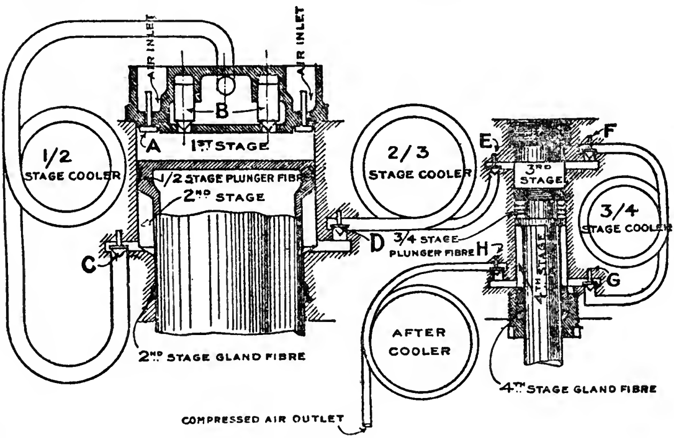


FIG. 34. Four-stage Compressor.

COMPRESSORS FOR VERY HIGH PRESSURES.—Up to 200 lbs. per square inch there is no need to go further than two stages, but for pressures above that three- and four-stage compression has to be employed.

Fig. 34 shows diagrammatically the arrangement in a four-stage Brotherhood Compressor.

Under proper working conditions, and when pumping at full pressure, the gauges should read as follows:

1st stage	50 lbs.
2nd „	200 „
3rd „	800 „
4th „ (receiver) . . .	3000 „

The packing fibres in the plungers and glands are of a very durable character and require to be very accurately fitted.

In Fig. 34 the path of the air can be traced as follows:

From the atmosphere through inlet valves A to first stage, delivery from first stage through valves B, thence through half-stage cooler and valve C to the annular space round the half-stage plunger forming the second stage. Then the air is delivered through D to the two-third-stage cooler and thence through the valve E to the third stage. Its subsequent path is through valve F, three-quarter-stage cooler, valve G, fourth-stage, valve H, and after-cooler.

The coolers and each stage of the compressor are tested by water-pressure to at least double the working pressure before the machine is put into use.

GOVERNING AND UNLOADING DEVICES

It is often required to run air-compressing plant at mines on varying loads, at one time producing perhaps only one-fourth and at another one-half of its full load capacity. It is therefore very frequently necessary to provide the compressor with some means whereby the output may be adjusted automatically from time to time to suit the variation in the demand.

There are several ways in which this can be done, among which are the following:

(1) A relief or blow-off valve on the receiver. This is a very wasteful arrangement, and is only permissible on small plants where the load varies very little.

(2) A combined air and speed governor. There are several efficient devices of this type.

(3) Automatic unloading devices.

(4) Use of variable-speed motors for driving.

(5) Automatic starting and stopping switches on electric drives.

(6) Automatic belt-shifting devices.

COMBINED AIR AND SPEED GOVERNORS.—*The Ingersoll-Rand "Air-Ball" Governor.*—This device combines the functions of an adjustable limit speed governor and an air-pressure regulator.

It is driven by an outside belt from a return crank. The speed-governing element consists of the usual fly-ball arrangement, throttling the steam when the speed exceeds a fixed limit. The pressure-regulating part consists of an air-operated piston, working in connexion with the speed governor, to control the speed of the compressor between maximum and minimum, maintaining air-pressure within a few pounds of normal under all loads within the range of speed.

This device is used on Ingersoll-Rand compressors where the demand for air is continuous and varies between the limits of half-load and full-load output.

Bellis and Morcom Governing Mechanism.—In the case of steam-driven compressors a centrifugal governor directly connected to the crank shaft is used to prevent the speed from exceeding a predetermined maximum, while in conjunction is used a relay air governor which is designed to vary the output of the compressor in accordance with the demand for air.

AUTOMATIC UNLOADING DEVICES.—These are used where the compressor is to be kept running at a practically constant speed—the load on the compressor, however, varying from time to time.

Walker's Unloading Device.—In the Walker Compressor (p. 57) an unloading arrangement is devised by modifying the design of one or more of the inlet valves in each cylinder cover, the ports in both valve and seating being made radial in outline instead of circumferential.

The valve is secured to its spindle so that by a suitable mechanism when there is a reduced demand for air, and consequently increased pressure in the receiver, the valve is rotated by the spindle so that the ports in the valve and seating register with one another and allow air to pass freely to and fro, being merely churned by the compressor piston with practically no load on the engines other than friction of the working parts.

The drawing (Fig. 35) shows the working of this device. The unloading valve A and its guard B are secured in position on the central spindle D, and so arranged that in ordinary working the ports do not register with one another.

The movement of the valve is accomplished by the double-cylinder mechanism G, the primary cylinder of which contains a small plunger loaded by the weight K through the medium of the yoke L. The pipe M places the cylinder in communication with the air receiver, so that when the pressure in the latter rises above the normal, the plunger rises and admits compressed air through an intermediate port to the under-side of the plunger in the secondary cylinder, which, when rising, operates the valve through the lever E. This carries at its outer end an adjustable weight, F, which turns the valve to its ordinary position for working as an inlet valve when pressure falls in the receiver. The pipe N on the side of the cylinder allows compressed air to pass through the secondary cylinder to the other operating mechanism on the second cover of the compressor, it being customary to install only one double-cylinder arrangement for each air cylinder. The valve is shown midway between its extreme positions; the movement generally takes place on the inlet stroke of the piston, and suitable means are

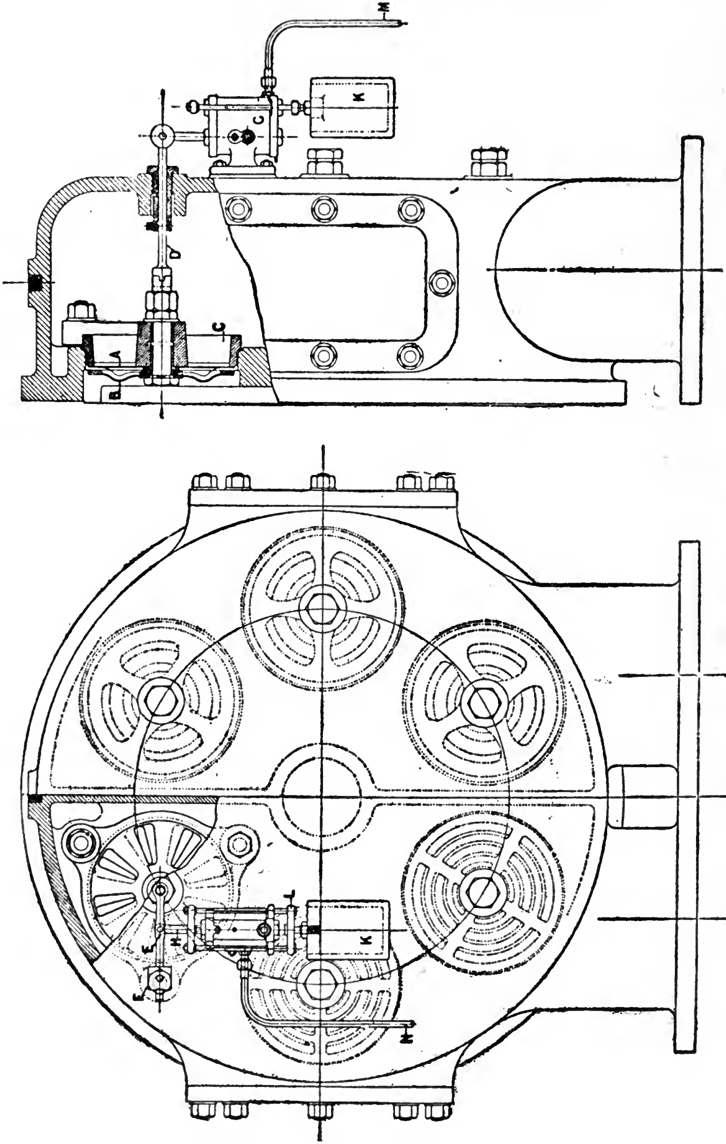


FIG. 35. Walker Unloading Device.

provided for changing the load by degrees to minimize sudden variation and strains on the moving parts, and also for relieving the cylinders of entire compression load when starting up the plant.

Alley and MacLellan's Governor.—In this arrangement, when the air pressure in the reservoir reaches that fixed as the working pressure, an adjustable governor acts on a throttle valve fitted to the air inlet and on a release on the discharge, thus stopping the output of com-

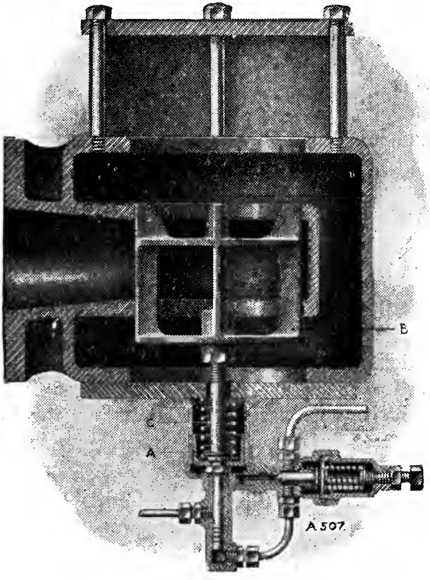


FIG. 36. "Sentinel" Unloader.

pressed air. The machine continues to run thus in a vacuum until the reservoir pressure falls a few pounds below that fixed, when the throttle valve automatically opens and the delivery of air restarts.

The device is shown in Fig. 36. Air from the receiver is admitted below the small piston A. The throttle valve B is loaded by an adjustable spring, C, so that when the pressure overreaches the working value the piston A rises in its cylinder and, slowly lifting the throttle B, stops the entry of air to the compressor. As the throttle valve closes, air is admitted to the small valve which unloads the second cylinder, and so the compressor runs "idle" until a demand is made again for air.

The position of the governor on the compressor will be seen in Fig. 24 (p. 59).

The Ingersoll Unloader.—The Ingersoll combination inlet and discharge unloader consists of a balanced throttle valve on the intake of the compressor, normally held open by a spring. An auxiliary needle valve, normally held closed by a spring, controls admission of air to a piston which operates to close the main valve. The air pressure in the receiver is communicated through a small pipe to the upper side of a diaphragm on the needle-valve stem. When the air pressure rises above normal, the pressure on this diaphragm exceeds the tension of the needle-valve spring, opens the needle valve, and admits air behind the throttle-valve piston. This closes the throttle valve and cuts off the intake to the compressor, thus unloading it.

When the air pressure falls below normal, the needle valve closes, cutting off admission of air to the throttle-valve piston, which allows the throttle valve to be opened by its spring, and so the load is thrown on the compressor again.

This device on the intake is often used alone, but sometimes an unloader is also placed on the discharge. The discharge unloader is operated also by a controlling valve, simultaneously with the intake unloader.

USE OF A VARIABLE-SPEED MOTOR.—The shunt-wound, direct-current motor is very suitable for operating a compressor which has to supply a varying demand for air. Its speed can be easily and efficiently regulated by inserting or cutting-out resistance in the field circuit.

Inserting resistance will increase the speed and cutting-out resistance will make the motor, and consequently the compressor, run slower. A differentially wound compound motor will automatically increase in speed as the load increases, and can be made to work very satisfactorily over a considerable range.

In the case of the shunt motor, the speed-regulation can be made automatically, operated by air pressure.

Three-phase motors cannot be so gradually or so efficiently regulated for speed, but specially designed machines can be obtained for running at two or more different speeds.

AUTOMATIC STOPPING AND STARTING SWITCH.—When the machines to be supplied with compressed air are in nearly constant use, and are either all working or all standing at the same time, it will be found most economical, if the drive is electrical, to use a fixed-speed motor and have an automatic stopping and starting switch.

An arrangement of this sort will stop the machine when the pressure in the mains or receiver reaches a certain maximum owing to a cessation in the demand for air, and will start it when the pressure falls to a predetermined minimum. This means of regulation entails the use of an air reservoir of considerable size.

CALCULATION OF SIZE OF CYLINDERS OF COMPOUND AIR-COMPRESSOR

To illustrate the method of calculating the size of cylinders the following example is taken:

Required the size of L.P. and H.P. air cylinders to deliver 4000 cubic feet of free air per minute at a gauge pressure of 90 lbs. per square inch. Piston speed to be 550 feet per minute and L.P. cylinder to compress the air to 25 lbs. per square-inch gauge.

Answer.—The whole of the air has to be handled by the L.P. cylinder, and therefore the area of cross-section of this cylinder, allowing for a volumetric efficiency of, say, 85 per cent.:

$$= \frac{4000}{550} \times \frac{100}{85} = 8{\cdot}5 \text{ square feet}$$

$$\text{and diam. of L.P. cylinder} = \sqrt{\frac{8{\cdot}5 \times 144}{{\cdot}7854}}$$

$$= 40 \text{ inches diameter.}$$

The volume to be handled by the H.P. cylinder assuming isothermal relations

$$= 4000 \times \left(\frac{14{\cdot}7}{25 + 14{\cdot}7}\right)$$

$$= \frac{4000 \times 14{\cdot}7}{39{\cdot}7} = 1480 \text{ cubic feet per minute;}$$

or, allowing a volumetric efficiency of 85 per cent. as for the L.P cylinder, capacity of cylinder

$$= 1480 \times \frac{100}{85} = 1741 \text{ cubic feet.}$$

$$\text{Then diameter} = \sqrt{\frac{1741 \times 144}{550 \times {\cdot}7854}}$$

$$= 24 \text{ inches.}$$

Also if we take a length of stroke of twice the diameter of the H.P. cylinder, which is common, we obtain a length of stroke of 4 feet.

And thus R.P.M. of engines

$$= \frac{550}{4 \times 2} = 69 \text{ R.P.M.}$$

∴ Approximately, L.P. cylinder	= 40 inches diam.
,, H.P. ,,	= 24 ,, ,,
Length of stroke	= 48 inches.
R.P.M.	= 70
Quantity of free air delivered per minute	= 4000 cubic feet.

Hydraulic Air-Compressing Plant.—In recent years a method of compressing air by the direct action of falling water has been successfully applied in America and on the Continent.

The method consists in sinking a vertical shaft to a depth sufficient to give a head of water equivalent to the pressure to which it is desired to compress the air. In this shaft a vertical pipe of large diameter is placed, terminating in a large circular chamber at the bottom of the shaft.

The top of the pipe enters a somewhat smaller chamber at the surface.

Water is conducted from a large reservoir, and in entering the vertical pipe produces a suction effect which draws in air through an arrangement of small feed pipes.

The air is caught up by the falling water in the form of bubbles, and during the descent is compressed to a pressure corresponding to the head of water.

At the bottom the direction of motion of the water changes first to the horizontal and then upwards in the shaft to the tail-race level, whence it makes its escape.

As the velocity of the water is greatly diminished when it reaches the large chamber at the bottom and while flowing upwards in the shaft, the air bubbles rise and form a layer of compressed air in the upper part of the bottom chamber. By means of a pipe passing down the shaft and communicating with this air space, the compressed air may be drawn off as required. One of the most successful of these hydraulic air-compressing plants is that designed by the Taylor Hydraulic Air-Compressing Co. of Montreal.

The advantages of the system are:

(1) Since the air is thoroughly disseminated through the water during compression, cooling is practically perfect and isothermal compression obtains.

(2) The air is obtained very dry.

(3) The plant, once installed, requires little attention.

Objections are:

(1) An abundant supply of water is necessary.

(2) The cost of sinking the shaft and fitting up the plant is considerable.

CHAPTER V

THE TURBO-COMPRESSOR

THE rotary form of air-compressor has come very rapidly to the front in recent years.

This machine depends upon centrifugal force to compress the air, and to some extent resembles the well-known centrifugal water pump, except that there are certain differences in design necessitated by the fact that air is compressible and has also a much lower specific weight than water.

Because of this lower specific weight considerably higher speeds and a greater number of stages are necessary in the rotary air-compressor than are common in the centrifugal or turbine pump.

PRINCIPLE OF THE TURBO-COMPRESSOR.—The action of the turbo-compressor depends upon the same principles as those involved in the flow of air, or any gas, through a conduit of varying cross-section. In such a pipe the same quantity must flow throughout every part of the pipe, regardless of the size of bore; consequently the velocity will be different at different points, and will be greatest when the air is flowing through the smallest bore.

Now, according to Bernouilli's law the sum of the pressure energy and the kinetic energy will, if loss from friction be neglected, be the same at every point throughout the length of piping considered.

Therefore, when the air under steady pressure flows from the larger to the smaller area of cross-section, the velocity energy is increased at the expense of pressure energy, and conversely, when the flow is from the smaller to the larger, the pressure energy rises and the velocity energy falls.

Now consider a revolving pipe inside a chamber with air entering at its centre of rotation, and moving outwards under the action of centrifugal force. During passage through the pipe the air is accelerated and leaves the pipe at a higher pressure and with considerable velocity. If the air, as it emerges, passes into a chamber with a gradually increasing cross-sectional area, the velocity of the air will fall, and in accordance with the law stated above, a gradual increase of pressure will result.

Thus in this receiving chamber the final pressure will be equal to the pressure on leaving the pipe added to the gain of pressure resulting from the fall in velocity.

This is exactly the action which takes place in a turbo-compressor.

In Fig. 37 a single stage of a turbo-compressor is shown in half-section.

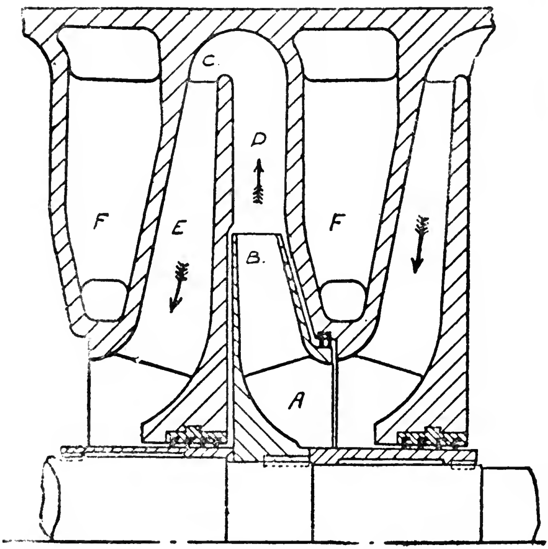


Fig. 37. Element of Turbo-compressor.

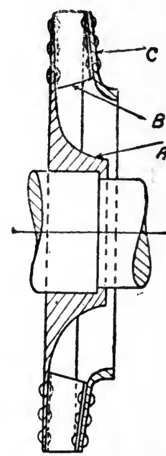


Fig. 38 Impeller of Turbo-compressor.

The portion A B is the revolving part called the *impeller* and takes the form of a wheel constructed of two side discs with partitions or *blades* of steel at intervals, the spaces between the blades forming the air passages. Fig. 38 shows the construction of the impellers in the Westinghouse Turbo-Compressor. The impeller disc A is made of forged steel, as is also the ring C. Between the disc and the ring are riveted the blades B. The blades are turned backwards against the direction of rotation as shown in Fig. 39, this design allowing a lower surging point (*see* p. 89) than straight blades.

Referring now to Fig. 37, the impeller when revolving takes in air at A—from the atmosphere through the inlet pipe, if that is the first stage, or, if not, from the previous stage as indicated by the arrow.

In flowing through the impeller from A to B the air is accelerated and leaves the impeller at B with a higher pressure and with a considerable velocity.

The air now passes into the *diffuser* D, which has a gradually increasing cross-sectional area, and so the velocity falls and the pressure is increased.

At C the air is diverted back through the guide-blades down the passage E and into the next impeller wheel.

In the succeeding stage the pressure is increased still further, and stage follows stage until the air eventually emerges at the delivery

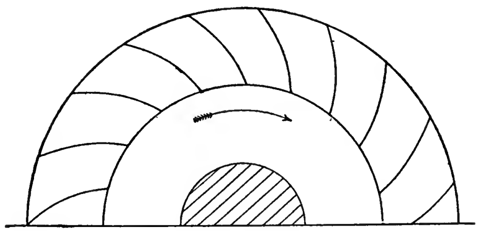


FIG. 39. Shape of Blades in Turbo-compressor.

end of the compressor with the final pressure for which the machine has been designed. The spaces F are for the circulating cooling water.

Advantages of the Turbo-compressor over the Piston Type.

(1) Simpler since there are no valves or valve gear.

(2) More reliable—less chance of breakdown—the valves are often a great source of trouble in reciprocating compressors.

(3) Occupy smaller space, especially for larger units.

(4) Continuous delivery of air.

(5) Automatic lubrication—only the shaft bearings have to be lubricated; this means reduction in attendance.

(6) Less vibration, since parts are revolving continuously in one direction, consequently less massive foundations required.

(7) The turbo-compressor can be used in conjunction with a mixed-pressure steam turbine in which a very large portion of the power is obtained from the exhaust steam from winding engines and other mining steam plant. *This is probably the most important advantage of all.*

(8) The amount of clearance is nil. The turbo-compressor is not well suited for small quantities of air, or for intermittent deliveries, and cannot hope to take the place of the piston type for such work. But for large quantities of air and a steady delivery, the turbo-compressod,

worked by a mixed-pressure turbine using largely exhaust steam, is easily superior to the reciprocating type and bids fair to displace the latter to a considerable extent in the near future.

In justice to the piston compressor, however, it should be pointed out that the efficiency of the turbo-compressor considered as such is considerably below that of the piston type except in the larger sizes; and even in these the turbo-machine still falls somewhat behind its rival as regards efficiency. Moreover the turbo-compressor has a very low efficiency on light load.

On the whole, however, having regard to the advantages enumerated and the lower first cost, the turbo-compressor is an excellent machine where the conditions are suitable.

THE WESTINGHOUSE-RATEAU TURBO-COMPRESSOR

This type of compressor, to which Figs 37, 38, and much of the description already given more particularly refer, is built on the Rateau turbo principle and is becoming largely used where pressure air is required for power and other purposes.

Fig. 40 * shows a typical turbo-compressor installation such as was supplied for Brodsworth Main Colliery by the British Westinghouse Co

This compressor is designed to deliver 7500 cubic feet of free air per minute at 80 lbs. per square inch gauge pressure. There are altogether twenty stages which are arranged in two sections to form a low-pressure and a high-pressure chamber.

Air is drawn in through the inlet pipe A, and passes through a butterfly valve at B_1 to I_1, which is the inlet of the L.P. chamber. After passing through the ten stages of the L.P. chamber, the air now under pressure emerges at D_1, and passes along the connecting pipe through the butterfly valve B_2 to the inlet I_2 of the H.P. chamber. Here it is raised to the final pressure and discharged at D_2.

The path of the compressed air is now through the non-return valve, N.R.V., and sluice valve, S.V., to the distributing mains. The surge valve, S.V., discharges into the silencer S.

The drive is by means of a mixed-pressure steam turbine.

The cooling water for the compressor is circulated through the pipes E to and from the cooling jackets of the two compressing chambers by a centrifugal pump, P, driven by the steam-engines which also work the pumps supplying water to the multi-jet condenser attached to the steam turbine.

* From *Transactions of the South Wales Mining Institute.* Paper by Messrs. Guy and Jones.

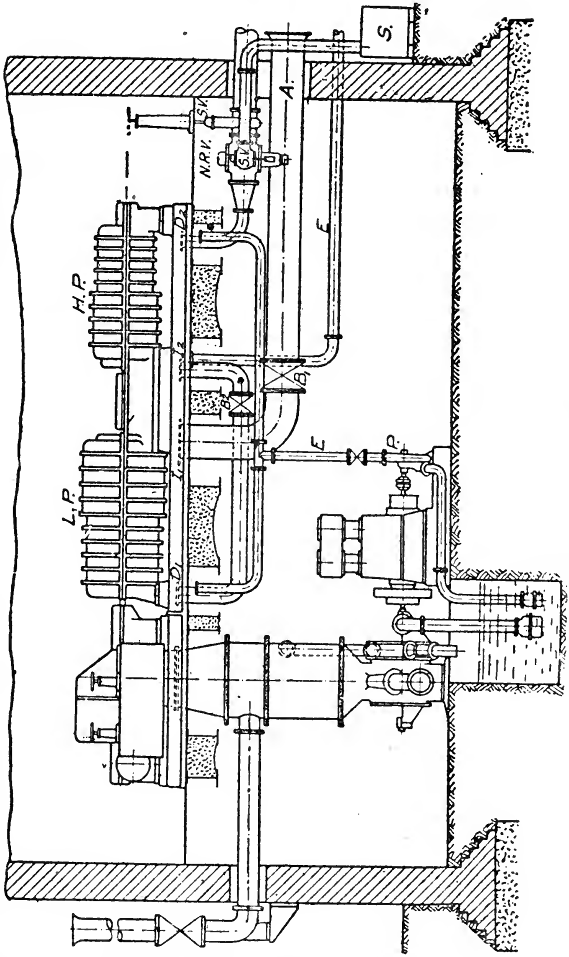


Fig. 40. Typical Turbo-compressor Installation.

The normal driving speed is 4100 R.P.M.

It will be noticed in the figure that the stages are reduced in diameter as the air passes from the inlet side to the delivery side of both the L.P. and H.P. cylinders of the compressor. This is because the air, being compressed, occupies less and less space as it travels through the compressor.

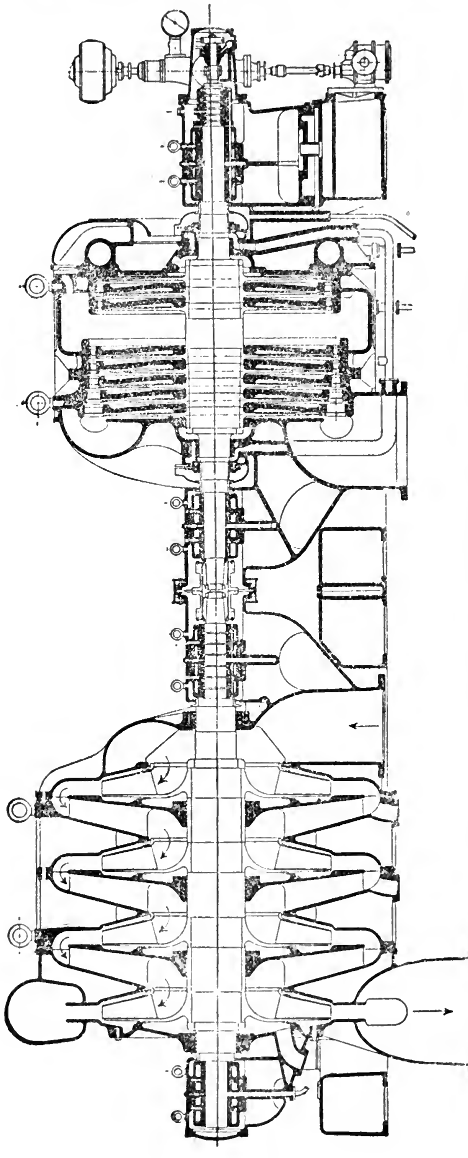


FIG. 41. Turbo-compressor coupled to Impulse Steam Turbine

LOW-PRESSURE TURBO-COMPRESSORS.—These are on the same principle as the turbo-compressor for high pressure; indeed they are identical except that the number of impeller wheels is less.

They are used for two purposes:

(1) To supply pressure air to blast-furnaces, Bessemer converters, and similar plant.

(2) To form the first compression stage where the second and succeeding stages take place in piston compressors.

The pressure may be from 5 lbs. up to 30 or 40 lbs. per square inch, and the compressor may be of the single-flow or double-flow form.

Where the air is to be used as in a blast-furnace the machine passes under the name of a turbo-blower; the design is, however, identical with that of the turbo-compressor, except that the stages are fewer, since the pressure is lower and the volume of air delivered may be greater, and also the turbo-blower is *not water-jacketed,* whereas the compressor is.

A single-flow, low-pressure turbo-compressor made by the firm of Daniel Adamson and Co. is shown in section in Fig. 41. The machine is direct-coupled to a mixed-pressure Rateau steam turbine, the turbine being on the right of the figure and the compressor on the left.

The machine can also be used as a blower; in this case the air-pressure is 8 to 12 lbs. per square inch, and the volume anything up to 40,000 cubic feet of free air per minute; if it is used as a compressor, a single stage can be made to give a pressure of from 5 to 8 lbs. per square inch, so that the four-stage machine illustrated would give a pressure of from 20 to 30 lbs. per square inch, depending on the speed of driving.

If a higher pressure is desired, all that is required is to add additional stages to the machine.

The path of the air into, through, and out of the compressor is shown by the arrows. The principle of this machine is the same as that described in connexion with the Westinghouse-Rateau Compressor and the design is similar.

A machine of the double-flow type is shown in Fig. 42, and is constructed by Messrs. Fraser and Chalmers. The air enters at both ends, and passes through the machine towards the centre, where it passes into the delivery. This type is only used as a blower.

MOTIVE POWER FOR THE TURBO-COMPRESSOR.—The turbo-compressor has to be driven at high speeds and is best driven direct-coupled instead of through gearing.

The steam turbine is especially well suited for high speeds and can be constructed to run at anything from 2000 to 5000 R.P.M.

The steam turbine therefore provides the ideal method of driving the turbo-compressor; especially when it is remembered that it is in conjunction with a mixed-pressure turbine, largely using exhaust steam from engines which would otherwise probably exhaust into the atmosphere, that the turbo-compressor is likely to prove most economical.

Failing the steam turbine, or where the pressure required or the quantity of air is small, high-speed electric motors both of the alternating and direct-current types can be used.

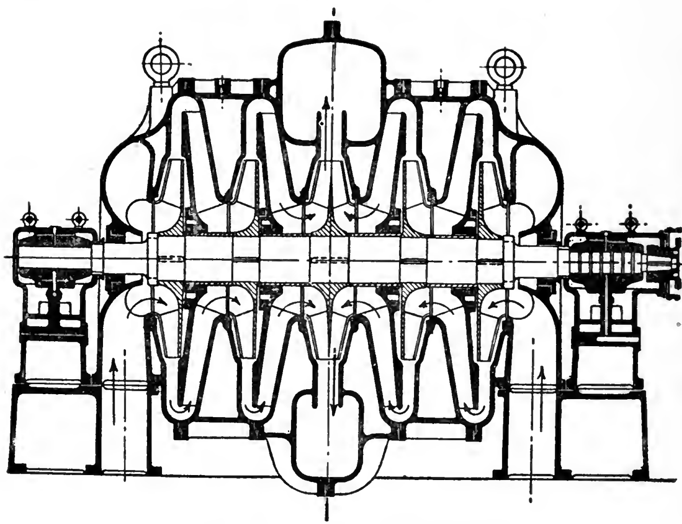


FIG. 42. Double-flow Turbo-blower

If the supply of exhaust steam is considerable, as it often is at collieries where the winding engines, which are generally the largest consumers of steam, are non-condensing, then during the daytime when the engines are fully working the turbo-plant will be run on exhaust steam alone, the live steam part being shut down.

It is necessary, however, that the turbine be designed, and the live steam supply be sufficient, to enable the compressor to cope with the maximum load when the winding or other engines have ceased working, or are standing for any reason.

If a mine or colliery takes power from a central generating station it may be most convenient and economical to use either a synchronous motor or an induction motor. The frequency is generally 50 cycles, and a speed of 3000 R.P.M. can be obtained from a synchronous motor and about 2900 R.P.M. in normal working from the induction motor.

When an induction motor is used, the wound rotor type is preferable as being more suitable for starting under load.

The synchronous motor has two outstanding advantages, namely: (1) the speed is constant from light load to full load, and (2) if the motor is over-excited it will take a leading current from the line and so improve the power factor of the whole distributing system. An objection to the synchronous motor is that it is not self-starting; it also slows down and stops dead if very much overloaded, but this is not likely to be a source of trouble.

Small auxiliary induction motors called "pony motors" are frequently used for starting up, and after synchronous speed has been attained the induction motors are uncoupled and the driving motor continues to run in synchronism with the frequency of the line.

Several sets of turbo-compressors driven by synchronous motors supplied by the Siemens concern were recently installed at the Compressor station of the Robinson Central Deep Mining Company. Each set is capable of dealing with 21,000 cubic feet of free air per minute and the delivery pressure is 130 lbs. per square inch. The compressors are driven by 2150 h.p. synchronous motors started by a special converter, consisting of an induction motor and a three-phase generator.

Direct-current motors can also be made to run at the desired speeds for compressor working, and have the important advantage that they are very easily and economically regulated for speed.

A shunt motor can be varied in speed over a very wide range by a variable resistance in the field circuit, which can be arranged to work automatically. The differentially wound compound motor can also be made to adjust its speed to the load, increase in speed being obtained in this case by an automatic weakening of the field as the load increases.

COOLING.—The method of getting rid of some of the heat in the turbo-compressor is the same as in the reciprocating type, viz. that of water-jacketing. Circulating water is pumped through annular spaces in the casing between each pressure stage, as at F in Fig. 37.

The cooling is more effective in the later stages than in the first stages, owing to the fact that as the density of the air increases it can the more readily give up its heat to the cooling surfaces. Thus the compression changes from nearly adiabatic in the first stage or two to nearly isothermal in the last stages.

It is interesting to note that the index of compression in the first stage in a turbo-compressor may be slightly greater than the purely adiabatic one of 1·408, as the result of the increase of temperature due to the friction of the air at high velocity.

Where the compression takes place in two or more separate casings intercooling may be adopted, by a method similar to that used in connexion with piston types.

GOVERNING.—This is effected in the case of the Westinghouse-Rateau turbo-compressor by means of a small piston connected up

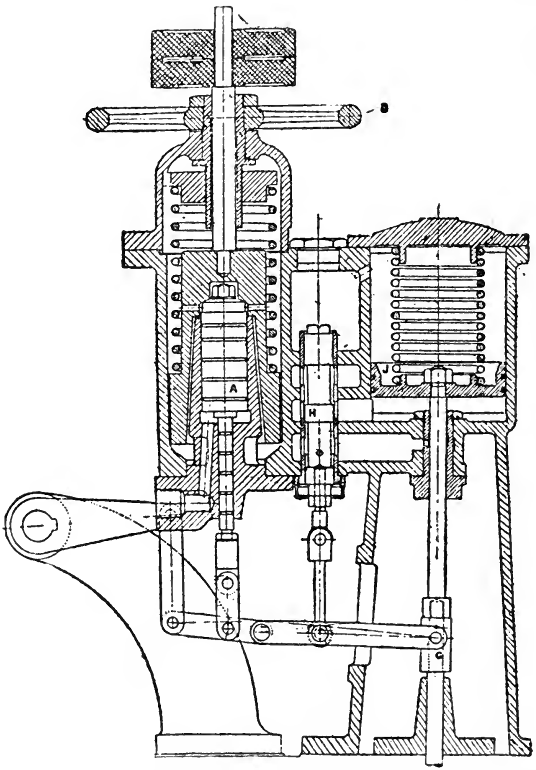


Fig. 43 Governing Arrangement of Turbo-compressor.

on the air-valve and operating on the governor-valve of the steam turbine. In this way it is possible to maintain a *constant pressure* even with a varying quantity of air delivered. Fig. 43 * shows the mechanism of the governor.

* From *Transactions of the South Wales Mining Institute.*

The discharge of the compressor communicates with the lower side of a spring-controlled piston A, and when the pressure of the delivery falls, A drops slightly and by means of the connecting lever pulls the pilot-piston H down from its mid-position. This allows a supply of oil under pressure to get underneath the power piston J, thus raising it and simultaneously raising the turbine governing valve which is attached to the spindle G. This motion returns the pilot valve H to its mid-position again.

In this way the turbine is enabled to increase in speed until the air pressure has increased to the working value.

In order to protect the turbine from overspeeding a centrifugal governor is used which comes into action only when the speed has exceeded a predetermined limit. The amount of variation of air-pressure required to bring the governing gear into action can be varied up to plus or minus 10 per cent. by the hand-wheel D.

If, instead of regulating for a constant pressure at the discharge, *constant-speed* governing is adopted, this is very easily arranged by the use of an ordinary centrifugal governor.

In the case of an electric drive, constant speed can easily be secured by using a synchronous motor or a direct-current shunt motor with a slight differential series winding, just sufficient to compensate for the small drop in speed on load of the ordinary shunt machine.

THE SURGING POINT.—If a turbo-compressor be kept running at constant speed and the volume of air delivered decreased from the normal output, it is found that the pressure rises until a maximum is reached at from 30 to 40 per cent. of the volume which the machine is designed to deliver.

This point is called the "Surging Point." At this point or below it a surging action takes place in the compressor, the air surging to and fro inside it. This action is so serious that the compressor becomes practically unworkable at or below the critical point.

It is therefore necessary to use a surging valve, which will open when the delivered volume reaches the minimum value and so allow the compressor to continue discharging a quantity which is slightly above the critical point. This additional volume permitted by the surging valve is waste, and hence the efficiency of the compressor under these conditions is low.

The use of the butterfly valves on the inlets to the compressor cylinders has the effect of lowering the surging point to about 20 per cent. of the full-load output.

CHAPTER VI

TRANSMISSION OF POWER

THE pressure-air is delivered from the compressor into a receiver or reservoir of a greater or lesser capacity which serves to act as a storage chamber. Before entering the receiver, the air may be made to pass through an after-cooler, or the after-cooler may form an integral part

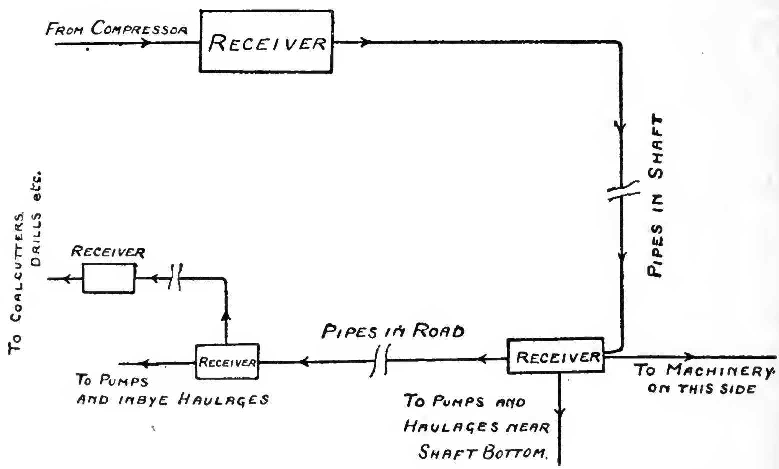


FIG. 44. Diagram of Compressed-air Transmission and Distribution System.

of the receiver itself. After leaving the receiver the air passes through pipes down the shaft and into the workings.

Diagrammatically the transmission and distribution system may be represented as in Fig. 44.

PIPES.—These may be of cast-iron, but wrought-iron or steel pipes are preferable as being lighter and easier to handle.

The proper size of pipe to use is determined by the permissible loss of pressure through friction, and the matter is fully considered under that heading.

The joints require to be very carefully made, rubber or asbestos washers being used to secure air-tightness. Figs. 45 and 46 show two forms of joints.

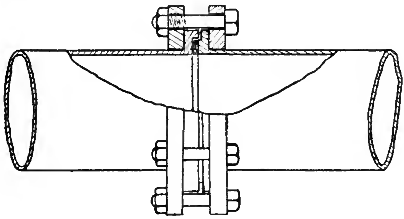


Fig. 45. Jointing of Pipes.

Electrically welded joints are also used, and where the welding can be done conveniently and with safety, this form of joint is the tightest of all. It would not be permissible to do this in many mines,

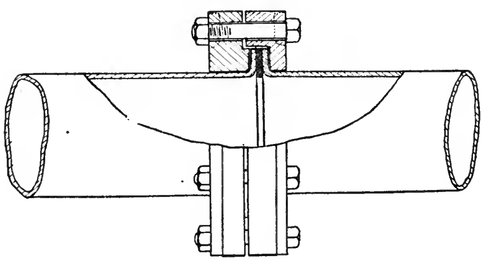


Fig. 46. Jointing of Pipes.

however, although it could, of course, always be done on the surface, and perhaps also in the shaft.

The pipes underground are uncovered, but in climates where the surface temperature in winter falls to the freezing-point of water, surface piping should be protected by some non-conducting covering, unless special precautions are taken to cool the compressed air to atmospheric temperature before it leaves the receivers. This would prevent the collection of water in the surface piping and so reduce the risk of the formation of ice inside the pipes during winter.

PIPE SUPPORTS.—In the shaft it is necessary to support and steady the column of air pipes at regular intervals. One method of doing this consists in using collarings carried on cross-pieces or buntons, as shown in Fig. 47. The cross-pieces *a*, of which there are two, one

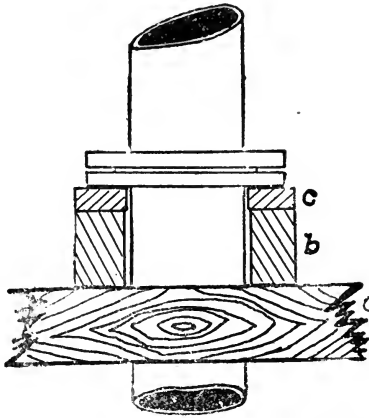


Fig. 47. Supporting Pipes in Shaft.

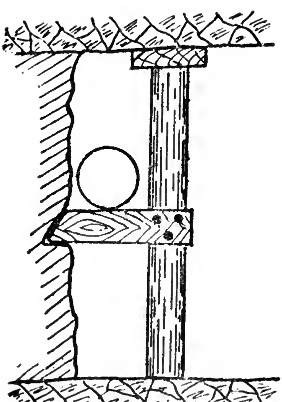


Fig. 48.

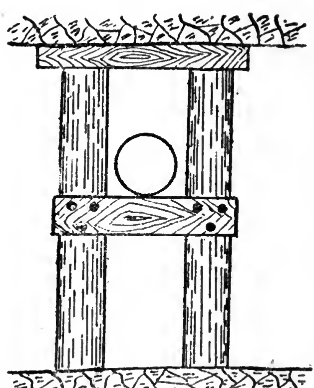


Fig. 49.

Supporting Pipes in Roadways.

on either side of the pipe, are fixed into the timber or brickwork of the shaft side and bolted across, and to these are fixed two other pieces of timber, *b*. Wedging-pieces *c* are driven under the pipe flanges to upport the column firmly.

Another method of support consists in using wrought-iron or steel hoops, or glands, which encircle the pipes and are fixed to the shaft timbering or walling by bolts or wood screws.

In the roads underground, the pipe-line is often laid on the floor at the side of the tram-line. Although this is an easy, convenient, and inexpensive method it is not to be recommended, as the pipes soon become covered with rubbish and it is then very difficult to detect leakages.

A better plan is to carry the pipes on wooden supports after the manner shown in Figs. 48 and 49. The pipes are thus less liable to be affected by movements of the strata and systematic inspection of the condition of the joints can be more easily carried out.

Objections that might be raised against carrying the pipes in this way, such as the space taken up and the necessity for occasionally renewing the timber, are easily outweighed by the easier detection of leakages and consequently the better efficiency obtainable.

RESERVOIRS OR RECEIVERS.—An air reservoir or receiver consists of a steel cylinder or tank, placed either vertically or horizontally as convenience dictates.

It should be placed as near the compressor as possible, and should not be lagged or protected in any way, in fact the more exposed it is the better, since we wish the air to lose as much of its heat as possible before leaving the receiver. Indeed, the perfect receiver would cool the air to atmospheric temperature before letting it go. As the air cools, it gives up the moisture it previously held in suspension; and since it will in any case fall to atmospheric temperature in the pipes, the cooler it becomes before leaving the receiver, the less trouble will there be with water in the pipes.

The functions of the receiver are as follows:

(1) To reduce the pulsating effect of the compressor piston if the receiver is on the surface, or if underground to do the same for the air-engine.

Violent pulsations and fluctuations in the flow of air increase the losses from friction and shock.

(2) To act as a reservoir of power so as to answer sudden demands for air which the compressor would not otherwise be able to supply.

In order to be able to perform this duty, even to a small extent, the receiver should be of a capacity at least equal to three times the output of the compressor for one minute.

(3) To cool the air as far as possible before it is allowed to pass into the pipe-line, and thus enable a large proportion of the moisture in the air to be got rid of before it passes underground.

The receiver should be fitted with the following:

(1) A pressure gauge to record the air pressure in lbs. per square inch.

(2) A safety blow-off valve, generally of the lever type.

(3) Manhole door to enable the inside of the receiver to be inspected and cleaned.

(4) Drain cock and pipe, for running off periodically the collected water.

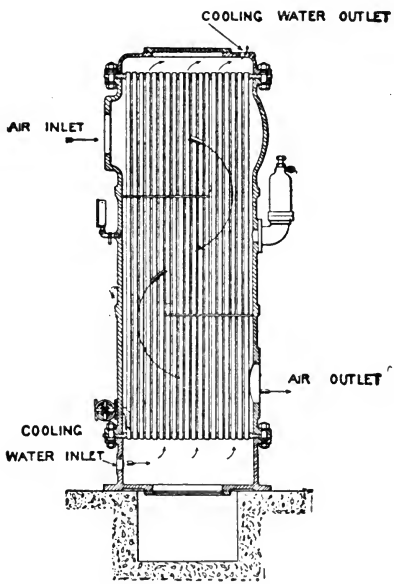


FIG. 50. After-cooler Receiver

A control valve should be fitted to the main pipe leading from the receiver.

An egg-ended boiler, or a Cornish or Lancashire boiler with the flues removed, the ends closed up and the necessary fittings arranged for, may be used as a receiver, or specially constructed reservoirs can be obtained from any of the manufacturers of compressors. The receiver, before being used, should be tested by hydraulic pressure up to twice the working pressure.

The air should be allowed to blow off periodically in order to see that the safety valve is in working order.

AFTER-COOLERS.—Where the capacity of the receiver is insufficient to allow the air to cool down to, say, below 100° F. at least, after-coolers are often placed in the pipe-line somewhere between the receiver and the compressor. These after-coolers are on the same principle as the intercooler between the low- and high-pressure cylinders of the compressor; sometimes the receiver and the after-cooler are combined in one and the same chamber.

The arrangement of the after-cooler used with the Alley and MacLellan Compressors is shown in section in Fig. 50. The air in its passage through the after-cooler is made to impinge again and again on the outer surface of the tubes, and so its temperature is greatly reduced and it is made to give up its contained moisture, which is drained off periodically.

UNDERGROUND RECEIVERS.—These are similar in construction to the surface receiver, though they may be smaller in size, and carry all the fittings except perhaps the safety-valve, which is hardly needed.

If the air has to be taken in more than one direction from the shaft bottom, it is advisable to place a receiver there. If no receiver is used, a drain-cock must be provided on the pipes at the bottom of the shaft, the vertical piping being continued for a few feet below where the horizontal piping branches off; or, if a bend pipe is used, provision should be made so that the water may collect so as not to interfere with the passage of air.

The water may be run off from the collector pipe periodically by hand, or an automatic water-trap may be used.

On the branch pipes receivers are also placed, and in this case they should be as near to the place where the air is being used as is convenient. The inbye receiver is necessarily small, and may often consist simply of a large diameter pipe placed in the pipe-line.

Wherever the gradient changes from downhill to uphill, drain-cocks should be placed on the pipes at the bends.

It should be remembered that although the use of receivers on mains and branches underground is quite good practice and helps materially to smooth down fluctuations of flow in the pipes, as well as providing a suitable means of draining off moisture, no system of receivers, however plentiful, can compensate for too narrow pipes.

The size of pipes required should therefore be determined without any reference as to what extent receivers will be utilized, and thus receivers, wherever introduced, will still further increase the economy of an already efficient system, instead of being a more or less futile attempt to palliate the fundamental error of too small pipes.

AIR-HOSE.—For portable machines flexible rubber air-hose is used to bring the air from the pipe-end to the machine. The hose is generally protected by a spiral of wire round it. The diameter of the hose should be as large as can be conveniently handled and the length should be kept as short as possible. For coal-cutters a usual length is 40 yards, made up of two lengths coupled together as required. A common diameter is 2 to $2\frac{1}{4}$ inches. In general, there is a very considerable fall in pressure due to friction and leakage in the hose-pipe, which can be minimized by keeping the hose even shorter than 40 yards, say 30 yards, and the diameter say $2\frac{1}{2}$ inches. On a longwall face this would necessitate taking piping into three or even four gates on a length of face of 100 to 120 yards instead of only two. But the saving effected would justify the trouble and expense. Careful inspection of the trailing hose during each shift with a report thereon would ensure that leakage from punctures would be detected, and so a check kept on the waste of air. Spare lengths of hose should be kept in readiness so that defective hose can be easily and expeditiously replaced.

In narrow work as short a length of hose as possible should be used and the piping kept close up to the face. The hose-couplings should be of such a design as not to reduce the size of the air-passage at the junction.

AIR LEAKAGE.—Leakage of air during transmission from the surface down the shaft and into the workings constitutes a very considerable source of loss in very many compressed-air installations. The percentage of the volume of free air compressed per minute lost in this way may vary from as low as 5 per cent. to as high as 50 per cent. in some cases.

The chief causes of leakage, apart from defective air-hose, are:

(1) Defective pipe joints.
(2) Defective valves and cocks.
(3) Blow-off cocks and receiver drains left open negligently or on purpose.

The first is undoubtedly the chief cause of air leakage, and although it may be difficult to eliminate this loss altogether, it is certain that if only half as much care were taken to make and maintain effective joints, and so minimise leakage, as is taken in the case of electrical transmission, a very decided gain in efficiency would inevitably result.

The effect of leakage is to bring about a reduction in volume accompanied by a fall in pressure.

The conspicuous leakage, bespeaking gross negligence or incompetence if allowed to continue, since it can readily be found by reason of the noise of the escape, or by the sudden fall of pressure below the normal, is easily dealt with; but the numerous small leakages which are continually occurring, especially on the underground part of the pipe-line, and which, though small in themselves yet amount to a great deal taken in the aggregate, are much more difficult to detect.

In order to facilitate detection of these small leakages the air pipes in the underground roads should be carried on cross-timbers or other supports. This is much better practice than burying the pipes underfoot or laying them on the floor, for not only are leakages more easily discovered but in most cases the pipe column is less affected by movements of the roof or floor, and hence joints are not subjected to the same strain, and are likely to keep longer air-tight.

In the shaft the pipes should be well supported by brackets and collarings, and at some point within 100 yards or so of the surface receiver an expansion joint, preferably of the bend form, should be inserted in the pipe line. The provision for expansion is necessitated by the fact that, unless the receiver is of much larger dimensions than usual, the air is still fairly hot, probably not far short of 200° F., when it leaves the receiver, and so the temperature of the piping, for some distance on, will rise and fall with alternating periods of work and rest. This, of course, causes the pipes to stretch and contract, and the provision of one or more expansion joints will naturally tend to save the pipes from strain and so protect the joints.

The bend form of expansion joint is to be preferred to the straight form with packing, which may be a source of trouble.

Careful and systematic attention should be given to valves and cocks, the gland packings being renewed at regular intervals or whenever signs of leakage occur, and the faces of the valves should be ground to a perfect fit when signs of wear are seen.

The custom of leaving drain-cocks on pipes and receivers slightly open so that the water may be drawn off with a minimum of trouble although a simple and convenient plan is an extremely wasteful one and results in a more or less serious leakage of air.

The extent to which leakage can take place through a puncture or opening may be realized from the curves in Fig. 51, which have been plotted to show graphically the relation between (1) the flow of air through an orifice of varying diameter under a constant pressure of 60 lbs. per square inch (curve A) and (2) the leakage through a hole half an inch in diameter under an increasing pressure (curve B).

In the first case the leakage is approximately proportional to the square of the diameter of the hole, and in the second it increases with the pressure; in both cases the leakage of air is given in cubic feet of free air per minute.

The diagram shows that quite a small opening may permit a considerable escape of air, and although the efflux from one puncture may not amount to much it serves to accentuate the aggregate waste. Every source of leakage, therefore, however small, must be rigorously eliminated.

Where possible, as at main receivers, automatic water-traps should be used; and elsewhere, instead of the drain-cock being left always partly open, it should be kept shut and the collected water run off daily or at other regular intervals as may be found necessary.

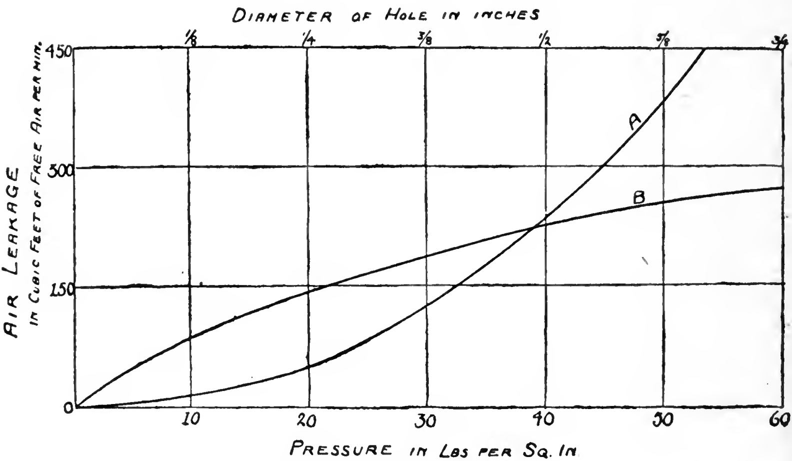


FIG. 51. Curves of Leakage.

TESTING FOR LEAKAGE.—Overall tests for leakage should be carried out periodically, say at every alternate week-end, so as to give a general idea of the waste through leakage between the compressor and the underground receiver.

There are several ways of carrying out such a test, of which the following may be mentioned:

(1) Choose a time when the underground machines are not working and run the compressor until the gauge on the underground receiver registers the normal working pressure. Then stop the compressor and close all stop-valves and -cocks. Note the time taken for the pressure on the underground gauge to fall a definite amount, or, in other words, observe the *rate* of fall of pressure at the underground receiver. This indicates whether the leakage is excessive or not. If the fall of pressure is rapid, the leakage is great and requires immediate attention; if slow, the matter is less urgent. If such a test is

made at regular intervals, comparison can be made between the results obtained at each, and thus data secured by means of which it can be ascertained whether or not improvement is being effected.

(2) A second method is to calculate the capacity of the whole system of pipes and receivers; and then, having increased the pressure up to normal as before and closed all the cocks, to keep a watch on the fall of pressure for a definite interval of time.

Suppose the period to be five minutes, then the leakage in free air per minute is equal to one-fifth of the difference between the volume of free air in the system at the normal pressure, and the volume at the pressure recorded by the gauge at the end of five minutes.

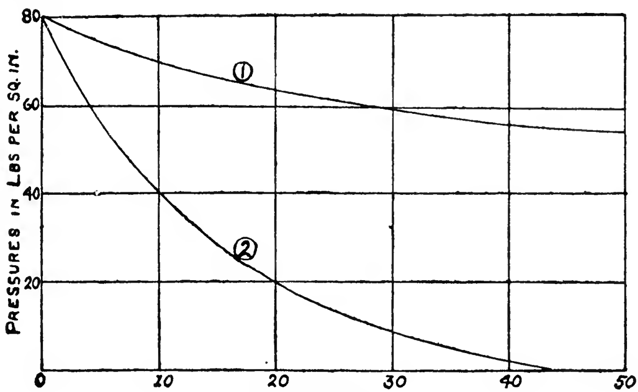


FIG. 52. Curves of Leakage during Testing.

(3) A third method consists in raising the pressure up to the norma in the underground receiver and then running the compressor at such a rate as will just suffice to maintain this pressure on the gauge. The volumetric discharge of the compressor at this speed is a measure of the loss through leakage.

All the stop-valves and -cocks leading to the machines using the air are, of course, closed during the test.

Testing for Leakage at the Pipe-joints.—Comparatively large leaks are easily detected in a systematic search by the hissing noise of the escape.

The smaller leakages can be tested for as follows:

Take a bucket of soap-suds and sprinkle a little over each joint. If there is any leak it will manifest itself by the production of bubbles. The defective joints should be chalked and attended to without delay.

It should be remembered, as is shown in the graph in Fig. 51, that the leakage is greater at high than at low pressures, and consequently the higher the pressure the more imperative it becomes to keep the joints as air-tight as possible.

The curves in Fig. 52 show the fall in pressure due to leakage in an overall test that might be obtained from (1) an efficient and (2) a wasteful system of transmission.

CONTINUOUS CHECKS ON LEAKAGE.—A continuous check on leakage may be kept by the use of rate-of-flow measurement meters.

By having a continuously recording meter at the surface and another at the underground end of the mains, it would be an easy matter to check the leakage between the two points of measurement, since the difference between the quantities passing the two points in a given time will represent the leakage during that interval. The advantages of such a system of checking will be obvious.

If daily records of the readings are taken immediate attention will be drawn to any increase in the leakage loss and the matter put right as quickly as possible.

The effect on the efficiency of the whole system, when methods such as those recommended above are used, as compared with that obtained by the haphazard and unscientific methods of checking the leakage losses used in the majority of mines at the present day, would, to say the least, be very beneficial.

USE OF THE AIR FOR VENTILATION.—A fruitful cause of waste of power is the use of the pressure-air for the purpose of clearing out gases and fumes from working places.

It is a strong temptation to workmen to have means available for quickly clearing out a working place after a round of shots, and the fact of getting back to work a little quicker would seem to justify the waste.

There is no doubt, however, that the abuse of expensive pressure-air or this purpose is a hopelessly extravagant method of ventilating a working place. Indeed it would be no exaggeration to say that the ventilation secured by jets of compressed air is at least ten times more expensive than that obtained by the ordinary method of using a fan.

The extent to which this form of waste may be carried may be judged from the fact that in one case brought to the notice of the author from one-fifth to one-third of the entire volume of air compressed was estimated to be squandered in this way.

The exhaust from the drill or coal-cutter when working helps the ventilation and sweetens the atmosphere, but, in general, this ought not to be relied upon for the sole ventilation of the place. Even in cases where the exhaust air is sufficient for the airing of the place, the necessary waste of air when the machines are standing, and particularly after shot-firing, will in most cases amount in a short time to a greater expenditure than would have provided for the installation of a small fan, giving a much cheaper and more effective ventilation current.

While admitting, therefore, that there may be occasions when the use of the air-jet is convenient and warranted, the practice of allowing workmen to make use of the air for this purpose at their own discretion is strongly reprehensible. It should always be kept in mind that the

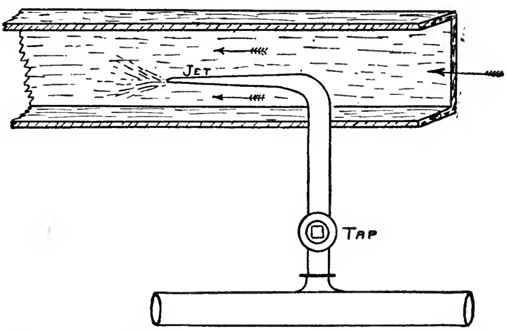


FIG. 53. Ventilation by means of Compressed-air Jets

process of producing compressed air is an expensive one, and that it should be used for power purposes only.

In circumstances where no other method of ventilation is practicable and the use of a compressed jet is required, especially after a round of shots, to clear away the fumes and allow work to proceed, the jet should be made to play into an air tube at the side of the road.

The jet of pressure-air arranged as shown in Fig. 53 causes an appreciable air-current to travel along the tube to the return, the roadway being the intake, or, if convenient, the tube can be made the intake.

FRICTION IN PIPES AND HOSE.—The friction resulting from the flow of air through the pipes and hose produces a diminution of pressure. This fall in pressure may be great or small according to the extent to which regard has been had to the laws relating to the flow of air in pipes.

The frictional loss of pressure is given by D'Arcy's formula as follows:

$$Q = c\sqrt{\frac{d^5\,p}{w\,L}}$$

where Q = quantity in cubic feet per minute discharged under lowest or final pressure—*i.e.* pressure at end of length of piping considered,

c = co-efficient,

d = diameter of pipe in inches,

p = fall in pressure due to friction in lbs. per square inch,

w = weight in lbs. of a cubic foot of air at the initial pressure,

L = length of pipe main in feet (assumed to be straight).

Squaring both sides of this equation and substituting the proportionality for the equality sign, c being dropped, we have

$$Q^2 \propto \frac{d^5 p}{w L}$$

and transposing,

$$p \propto \frac{Q^2 w L}{d^5}$$

from which we obtain the following laws:

(1) Loss of pressure varies directly as the square of the quantity.

(2) Loss of pressure varies directly as the square of the velocity, since $Q^2 = v^2 a^2$ and a is assumed constant.

(3) Loss of pressure varies directly as the density or weight of the air per cubic foot.

(4) Loss of pressure varies directly as the length.

(5) Loss of pressure varies inversely as the *fifth power* of diameter *for a given quantity.*

(6) Loss of pressure varies inversely as the diameter, if the *velocity* remains *unchanged.*

The relations connecting loss of pressure with velocity, quantity, and diameter are the most important.

These show us that if we wish a small loss of pressure in a given length of piping, we must keep down the velocity by using as large diameter pipes as circumstances and other considerations will allow.

The fact that for a given quantity of free air flowing per minute the fall of pressure decreases as the fifth power of the diameter shows that quite a small increase in size will produce a very large diminution in loss due to friction.

It is not quite certain that the frictional loss at all pressures is strictly proportional to the square of the velocity. Recent experiments tend to cast some doubt on this point, but in any case there can be no question but that the loss varies in very close approximation to the velocity squared throughout the range of pressures common in mining.

The laws deduced above may therefore be accepted as correct for practical purposes.

EXAMPLE I.—Compressed air is supplied underground through 10-inch diameter pipes, and the drop in pressure due to friction is found to be 2 lbs. per 1000 yards.

What would it have been if the pipes had only been 8 inches in diameter and the same quantity of air at the same pressure was required?

$$p \propto \frac{1}{d_5} \text{ or } \frac{p_1}{p_2} = \frac{d_2{}^5}{d_1{}^5}$$

Then $p_1 = 2$ lbs. per square inch

$d_1 = 10$ inches

$d_2 = 8$ inches

$$\therefore \frac{2}{p_2} = \frac{8^5}{10^5}$$

or $$p_2 = \frac{2 \times 10^5}{8^5} = 6{\cdot}1 \text{ lbs. per square inch per 1000 yards.}$$

Thus the slight decrease of 2 inches in the diameter of the pipes means over three times the drop in pressure due to friction.

The following losses may be taken as representing approximately what is considered reasonable in modern practice:

Shaft pipes and underground mains .	3 lbs.	drop	per	1000	yards.
Branch pipes	3 ,,	,,	,,	500	,,
Small pipes	3 ,,	,,	,,	100	,,
Hose	3 ,,	,,	,,	50	,,

The value of the co-efficient c varies somewhat with the diameter of the pipe.

Table IV gives the value of c for pipes of from 1 inch up to 12 inches in diameter, and for convenience of calculation the values of d^5 and $c\sqrt{d^5}$ are also given.

In Table V are given the values of w and $\sqrt{w}$ for initial pressures up to 100 lbs. per square inch.*

Effect of Bends.—Bends in the pipes increase the friction considerably and there is also a loss of energy due to shock.

The sharper the bend the greater the loss. Considerable loss is caused in this way by cocks and valves which cause the air abruptly to change its direction of flow; and insufficiency of passage through these introduces an additional loss due to throttling.

The type of valve giving the least drop in pressure due to the passage of air through it is the parallel-flow or sluice type allowing of full-bore passage when open.

* Tables IV and V are taken from Prof. Peele's "Compressed Air Plant for Mines" with the author's permission.

TABLE IV.—GIVING c, ETC., FOR DIFFERENT VALUES OF d

Diam. of Pipe.	c	d^5	$c\sqrt{d^5}$
Inches.			
1 . .	45·3	1	45·3
2 . .	52·6	32	297
3 . .	56·5	243	876
4 . .	58·0	1,024	1,856
5 . .	59·0	3,125	3,298
6 . .	59·8	7,776	5,273
7 . .	60·3	16,807	7,817
8 . .	60·7	32,768	10,988
9 . .	61·0	59,049	14,812
10 . .	61·2	100,000	19,480
11 . .	61·8	161,051	24,800
12 . .	62·0	248,832	30,926

TABLE V.—GIVING w AND $\sqrt{w}$ FOR INITIAL PRESSURES UP TO 100 LBS. PER SQUARE INCH

Gauge Pressure.	w	$\sqrt{w}$	Gauge Pressure.	w	$\sqrt{w}$
lbs. per sq. in.			lbs. per sq. in.		
0	0·0761	0·276	55	0·3607	0·600
5	0·1020	0·319	60	0·3866	0·622
10	0·1278	0·358	65	0·4125	0·642
15	0·1537	0·392	70	0·4383	0·662
20	0·1796	0·424	75	0·4642	0·681
25	0·2055	0·453	80	0·4901	0·700
30	0·2313	0·481	85	0·5160	0·718
35	0·2572	0·507	90	0·5418	0·736
40	0·2831	0·532	95	0·5677	0·753
45	0·3090	0·556	100	0·5936	0·770
50	0·3348	0·578			

In order to give a measure of the friction due to bends, these may be expressed as equivalent lengths of straight piping, offering the same resistance to the flow of air.

Table VI shows the equivalent lengths of piping for various radii of bend, these being expressed in terms of the pipe diameter.

TABLE VI

Radius of Bend. Diameters.	Equivalent length of Straight Pipe. Diameters.	Radius of Bend. Diameters.	Equivalent length of Straight Pipe. Diameters.
5	8	$1\frac{1}{4}$	13
3	$8\frac{1}{2}$	1	$17\frac{1}{2}$
2	9	$\frac{3}{4}$	35
$1\frac{1}{2}$	$10\frac{1}{2}$	$\frac{1}{2}$	121

CALCULATIONS ON LOSS OF PRESSURE DUE TO FRICTION

EXAMPLE II.—Find the loss of pressure in 1000 yards of an 8-inch diameter pipe passing 2000 cubic feet of free air per minute under an initial gauge pressure of 60 lbs. per square inch.

$$Q = c\sqrt{\frac{d^5\,p}{w\,L}}$$

$$Q = \frac{2000 \times 14{\cdot}7}{60 + 14{\cdot}7} = \frac{2000 \times 14{\cdot}7}{74{\cdot}7}$$
$$= {\cdot}394 \text{ cubic feet.}$$

From Tables IV and V:

$$c\sqrt{d^5} = 10988$$
$$\sqrt{w} = {\cdot}622$$

$$\therefore\ 394 = \frac{10988}{{\cdot}622} \times \frac{\sqrt{p}}{\sqrt{L}}$$
$$= \frac{10988}{{\cdot}622} \times \frac{\sqrt{p}}{\sqrt{3000}}$$

and transposing

$$\sqrt{p} = \frac{394 \times {\cdot}622 \times \sqrt{3000}}{10988}$$
$$= 1{\cdot}34 \text{ lbs.}$$
$$\therefore\ p = 2 \text{ lbs. nearly.}$$

Note.—Q has been calculated at the initial pressure instead of the end pressure as required by the formula, but the difference between the two is small.

EXAMPLE III.—Ten 3-inch power drills each taking 130 cubic feet of free air per minute at 75 lbs. pressure are to be fed from a main 1000 yards long. If the pressure drop due to friction is not to be more than 2 lbs., calculate the required diameter of pipe. Assume 10 per cent. leakage.

Quantity of air at atmospheric pressure $= (10 \times 130) + 10$ per cent.
$= 1430$ cubic feet.

$$\text{Quantity at 75 lbs.} = 1430 \times \frac{14{\cdot}7}{(75 + 14{\cdot}7)} = 234{\cdot}3 \text{ cubic feet.}$$

$$\therefore\ 234{\cdot}3 = c\sqrt{\frac{d^5\,p}{w\,\mathrm{L}}}$$

$$\text{or } c\sqrt{d^5} = \frac{234{\cdot}3 \times \sqrt{w} \times \sqrt{3000}}{\sqrt{2}}$$

From Table IV on p. 104: $\sqrt{w}$ for 75 lbs. $= {\cdot}681$
$\sqrt{w}$ for 80 lbs. $= {\cdot}700$
Difference $= {\cdot}019$ for 5 lbs.

$\therefore$ by proportional parts the difference in $\sqrt{w}$ for 2 lbs.
$= {\cdot}019 \times \frac{2}{5} = {\cdot}007$
$\therefore\ \sqrt{w}$ for 77 lbs. $= {\cdot}681 + {\cdot}007$
$= {\cdot}688$

$$\text{Then } c\sqrt{d^5} = \frac{234{\cdot}3 \times {\cdot}688 \times \sqrt{3000}}{1{\cdot}414}$$
$$= 6270$$

From Table IV we see that for this value of $c\sqrt{d^5}$ the size of pipe lies between 6 and 7 inches.

$c\sqrt{d^5}$ for 6-inch pipe $= 5273$
$c\sqrt{d^5}$,, 7-inch ,, $= 7817$

difference $= 2544$

$6270 - 5273 = 997$

$$\frac{997}{2544} = {\cdot}49 = \tfrac{1}{2}\text{-inch}$$

$\therefore$ $6\frac{1}{2}$-inch diameter pipe required.

AIR VELOCITIES.—The velocity of the air in the main should be limited to not more than 25 feet per second, and in some cases 20 feet per second is considered sufficiently high.

In branch pipes velocities of from 30 feet per second up to 50 feet per second are usual, but the latter figure should not be exceeded.

For portable machines such as coal-cutters and rock-drills, where a trailing hose is necessary, velocities very much greater than even 50 feet per second are allowed. In most of these cases the velocity is seldom less than 100 feet per second and may often be as high as 200 feet per second.

Such high velocities, of course, mean very big drops in pressure even in a comparatively short length of hose.

Convenience of handling is, of course, of great importance with these machines, and consequently, in order to obtain a handy size of hose, high velocities are employed. However, if the length of hose is considerable, as is the case with coal-cutters, it is good practice to use as large a hose as is at all convenient.

To show the effect of velocity on loss of pressure, it may be stated that for a 12-inch pipe the percentage pressure loss per 1000 yards for a velocity of 25 feet per second is about 1·5 per cent., for 50 feet per second 6 per cent., and for 100 feet per second 24 per cent.

EXAMPLE IV.—A main air-pipe is to supply four coal-cutters with air at 40 lbs. per square inch, each machine requiring 520 cubic feet of free air per minute. Allowing 10 per cent. extra for leakage, find what size of pipe will be needed if the velocity is not to exceed 25 feet per second.

$$(4 \times 520) + 10 \text{ per cent.} = 2288 \text{ cubic feet per minute.}$$

This quantity, at 40 lbs.,

$$= \frac{2288 \times 14{\cdot}7}{40 + 14{\cdot}7}$$

$$= \frac{2288 \times 14{\cdot}7}{54{\cdot}7} = 615 \text{ cubic feet per minute.}$$

$$= 10{\cdot}25 \text{ cubic feet per second.}$$

$$\text{Then } d^2 \times \frac{11}{14} \times \frac{1}{144} \times 25 = 10{\cdot}25$$

$$\therefore d^2 = \frac{14 \times 144 \times 10{\cdot}25}{11 \times 25}$$

$$= 75{\cdot}1$$

$$\text{whence } d = \sqrt{75{\cdot}1} = 8{\cdot}6 \text{ inches,}$$

say $8\frac{3}{4}$-inch diameter pipe.

A Ready Method of Obtaining the Size of Pipe for a given Quantity of Air.—The curves in Figs. 54 and 55 have been plotted from calculations based on D'Arcy's formula, and give in a convenient way the size of pipe for a given quantity of free air per minute.

In Fig. 54 the curves give the sizes of pipes for quantities up to 9000 cubic feet of free air per minute.

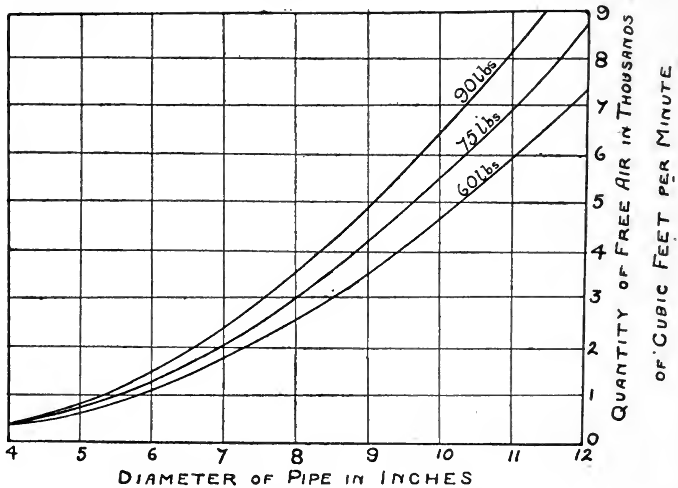


FIG. 54. Curves giving Sizes of Main Pipes.

A fall of pressure of approximately 3 lbs. per 1000 yards is allowed for, and the curves for three different initial gauge pressures are given.

In Fig. 55 the curves A, B, and C have to be referred for size of pipe to the figures at the bottom of the diagram, and for quantity of air to those on the right-hand side.

These curves assume a fall of pressure equal to 3 lbs. per 500 yards, which might be considered reasonable in branch pipes.

The other set of curves are referred to the top and left-hand side of the diagram, and for these a pressure drop of 3 lbs. per 120 yards is allowed for. These are intended for the smaller sizes of pipes.

To show the method of using the curves the following examples are given:

EXAMPLE V.—Quantity of free air per minute, 8000 cubic feet. Pressure at receiver, 75 lbs. per square inch. Find the size of main.

From the curve for, 75 lbs. in Fig. 54 we find that the size of pipe lies between 11 and 12 inches. Say $11\frac{1}{2}$-inch diameter pipe.

EXAMPLE VI.—Find size of branch pipe for 2000 cubic feet of free air per minute at 75 lbs. per square inch, allowing for a pressure drop of 3 lbs. per 500 yards.

From curve B, Fig. 55, we get 6-inch diameter as size of pipe.

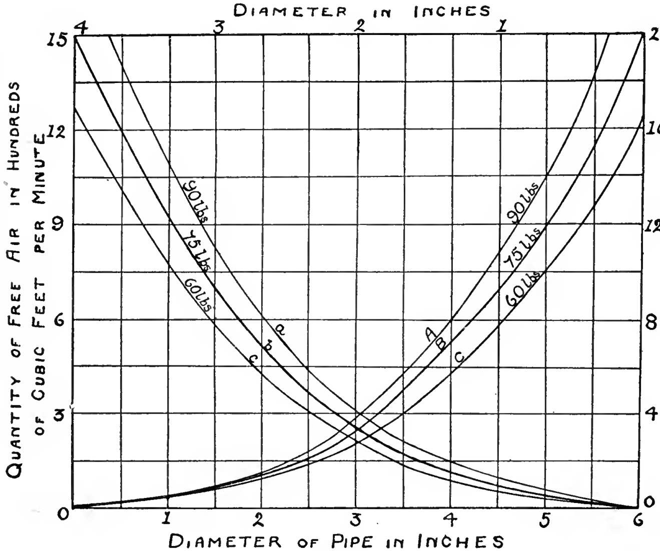


FIG. 55. Curves giving Sizes of Branch Pipes.

EXAMPLE VII.—Give size of pipe at 60 lbs. pressure to suit 800 cubic feet of free air per minute, allowing a fall of pressure due to friction of 3 lbs. per 120 yards.

From curve *c* (left-hand side) we obtain for 900 cubic feet $3\frac{1}{2}$-inch diameter and for 750 cubic feet 3·3-inch diameter. Say, therefore, $3\frac{3}{8}$-inch pipe for 800 cubic feet.

The use of the curves in Figs. 54 and 55 can be further extended to find the quantity of air transmitted for a given size of pipe allowing larger pressure drops than those for which the curves have been constructed.

From D'Arcy's formula we deduce that the quantity a given size of pipe will transmit is proportional to the square root of the permissible pressure drop.

Thus for main pipes, if a fall of pressure due to friction of 6 lbs.

instead of 3 lbs. per 1000 yards be allowed, the same size of pipe will transmit $\sqrt{2}$ (1·414) times as much air at the same gauge pressure.

The following are the multiplying factors for pressure drops up to 12 lbs. per 3000 feet, 1500 feet, and 360 feet respectively:

Permissible pressure drop in lbs. per square inch . . .	1	2	3	4	5	6
Multiplying factor .	0·6	0·8	1	1·15	1·3	1·4
Permissible pressure drop in lbs. per square inch . . .	7	8	9	10	11	12
Multiplying factor .	1·5	1·6	1·7	1·8	1·9	2

To show the use of this multiplying factor, suppose permissible pressure drops are

(*a*) 10 lbs. per 3000 feet (mains)
(*b*) 10 „ „ 1500 „ (sub-mains)
(*c*) 10 „ „ 360 „ (branches)

and suppose initial gauge pressure in each case to be 75 lbs. per square inch.

Let quantity of free air per minute in case (*a*) = 10,000 cubic feet.
case (*b*) = 3,000 „ „
case (*c*) = 1,200 „ „

The multiplying factor for 10 lbs is 1·8, so that the size of pipe got from the curves will be that corresponding to the above quantities *divided by* 1·8.

Case (*a*).—From Fig. 54, size of pipe for 5550 cubic feet is 10-inch diameter. This is the size for 10,000 cubic feet for the pressure drop allowed.

Case (*b*).—Similarly from Fig. 55, curve B, we find size for 1660 cubic feet is $5\frac{3}{4}$-inch diameter pipe. This is the size for 3000 cubic feet with 10 lbs. fall of pressure per 1500 feet.

Case (*c*).—Fig. 55, curve *b*, gives 3-inch diameter pipe for 660 cubic feet. This size will do for 1200 cubic feet for a permissible pressure fall of 10 lbs. per 360 feet of length of pipe.

The proper size of pipe for any other case may be quickly determined in a similar fashion.

It should be noted that the size of pipe as obtained from the curves is independent of the length of piping, although, of course, the total fall of pressure in the piping due to friction will be proportional to the length. This, although not strictly accurate, is sufficiently so for most purposes.

The gauge pressures for which the curves have been drawn are those most common in practice. For other pressures slightly higher or slightly lower than those given, the quantity passed for the same pressure drop can still be read off by assuming a curve diverging from the others to an extent proportional to the difference in pressure.

Thus, for example, from Fig. 54 one can easily see that a 9-inch pipe would pass the following quantities for the pressure drop for which the curves are plotted:

Gauge Pressure. (lbs. per sq. in.)	45	60	75	90	105	120
Quantity of free air per minute (cubic ft.) .	3000	3500	4200	4800	5500	6000

The first and the last two are obtained by estimating whereabouts the curve for the particular pressure would cut the ordinate through 9.*

Factors which Modify the Loss of Pressure in Transmission.—The energy expended in the pipes through friction is converted into heat. Part of this heat enters the pipes and is dissipated and lost, but part of it is taken up by the air and to some slight extent helps to retard the drop in pressure. The extent of this compensation is not easily determined.

Another factor which helps to counteract the loss of pressure in transmission, especially in deep mines, is the aerostatic head due to excess in weight of the vertical column of compressed air in the shaft over the weight of a similar column of atmospheric air of equal height.

The gain from this source in a deep mine is quite appreciable; for

* Professor Richards of the University of Illinois has constructed a more elaborate "Transmission Diagram" giving size of pipes for a wide range of quantities and pressure losses. The diagram is reproduced in the *Transactions of the Institution of Mining Engineers*, vol. li.

example, the increase of pressure due to the aerostatic head for three different working pressures and depths of shafts is as follows:

TABLE VII

Depth of Shaft. Yards.	Working Pressure. lbs. per sq. in.	Aerostatic Head. lbs. per sq. in.
400	60	2·08
600	70	4·50
800	80	6·88

The aerostatic heads have been calculated for a temperature of 65° F. in each case.

AIR MEASUREMENT

One of the outstanding advantages of electricity is the ease and accuracy with which it can be measured, and to this the high efficiency obtainable is very largely due.

On the contrary, little or no attempt has been made, at least in Britain, to make proper use of measuring instruments in connexion with compressed-air installations. This is a matter, however, that will probably be given close consideration in the near future. It is quite certain that continuous measurement of air consumption and of pressure is one of the most effective means of maintaining a check on waste and loss, and an invaluable incentive to efficiency.

One of the most noteworthy instances of the salutary effect of proper measurement on the efficiency is that provided by the Rand Mines Power Supply Company, Ltd.

The company supplies pressure air to the mines of thirteen different companies. The air is metered at the various mines, and as a result of the adoption of the system of measurement the air-consumption has been reduced by about 40 per cent. for the same amount of work done.

Air-measuring instruments are of two kinds, namely: (1) Pressure gauges. (2) Rate-of-flow meters.

PRESSURE GAUGE.—This type of instrument has, of course, been in use for a very long time, if not underground at least on the surface. Its use underground, however, should be much more extensive than it is. There should be one on all receivers, or at least on the more important ones. In the common type the air exerts its pressure in a hollow oval-shaped tube fixed at one end and free to move at the other. The air pressure tends to make the tube circular in shape and thus produces movement of the free end in proportion to the pressure. The free end, through gearing, works a pointer which on a scale of pounds per square inch registers directly the pressure in the receiver or pipe.

In another form of pressure gauge a graphic record of the pressure variation is made on a chart. This form of instrument is extremely useful as giving a continuous record of the pressure.

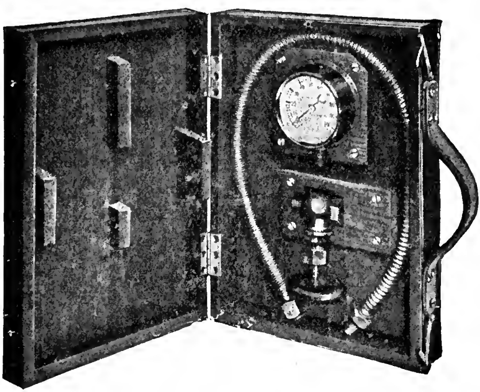


FIG. 56. Portable Pressure Gauge.

Its sphere of usefulness is on the surface and perhaps at the main receiver at the shaft bottom. For other work further inbye the ordinary pressure gauge is preferable.

Portable Pressure Gauge.—The use of a portable gauge is valuable as a means of ascertaining the pressure at the machines and for the localization of fall of air pressure in the pipe-lines.

A convenient form of gauge as recommended by Mavor and Coulson, Ltd., is shown in Fig. 56. The gauge is provided with a short length of metallic flexible tube and a small valve, the whole being in a teak-wood case convenient for carrying.

TESTING FOR PRESSURE DROP DUE TO FRICTION.—The *actual* pressure drop—as distinct from the calculated—should be obtained for the several sizes of pipes used in the distributing system.

The method of testing is to ascertain by means of pressure gauges the fall in pressure over a measured length of the piping considered, while the normal quantity of air is passing through it.

Care must be taken that there is no loss of air due to leakage along the pipe between the points where the pressure is measured, or else the test is valueless as regards frictional loss.

If the pressure fall due to friction is obtained for any length of piping, then for the same rate of flow the total fall for any other length of the same piping is easily calculated, since the pressure drop is proportional to the length.

AIR-METERS.—Meters for measuring the rate of flow of the air are based on the fact that the velocity can be measured or deduced from the velocity-head or pressure producing it.

There are several forms of air-meters, among which the following are the most important:

(1) *The Pitot Tube.*—In this form of meter two tubes project radially into the air-pipe at a point where the velocity head is to be measured. The open end of one of the tubes faces directly against the flow of the air and the other faces across the pipe at right angles to the direction of flow. The two tubes are connected on the outside of the pipe to a manometer. The tube facing the air-flow communicates to the manometer the static pressure plus the pressure due to velocity-head.

The other tube communicates only the static pressure. Thus the difference between the two pressures indicates the pressure producing velocity. The velocity can then be calculated from the formula

$$v^2 = 2\,g\,h$$

$$v = \sqrt{2\,g\,h}$$

where v = velocity in feet per second

g = 32·2

h = height of a column of air corresponding to the measured difference of pressures in inches of water-gauge.

If H = inches of water-gauge

then $v = 66\sqrt{H}$

The diagram (Fig. 57) shows graphically the velocity of flow for water-gauges up to 12 inches.

Then, knowing the area of cross-section of the air-pipe at the point where the velocity is measured, the volume passing per minute can be calculated, or, since the cross-section is constant, the quantity is proportional to the velocity, and so the same graph will also give quantity. The temperature and pressure, however, are not constant in practice and corrections are necessary for this

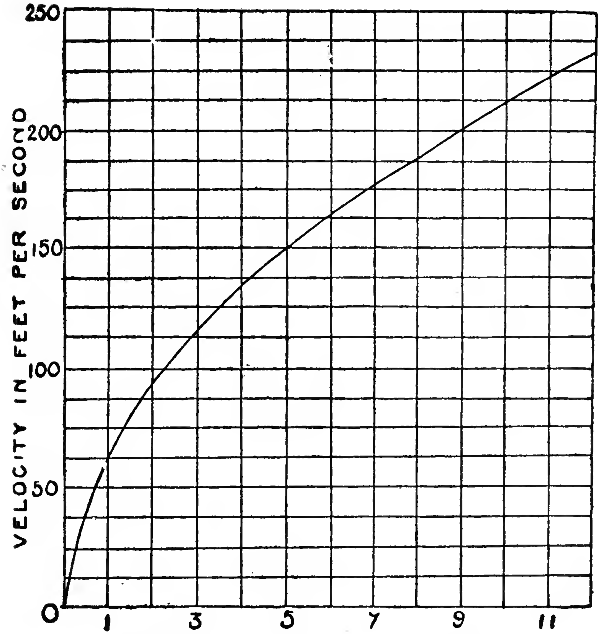


FIG. 57. Curve giving Velocities for Pitot Tube Air-Meter.

(2) *The Venturi Meter.*—This meter is an application of Bernoulli's theorem.

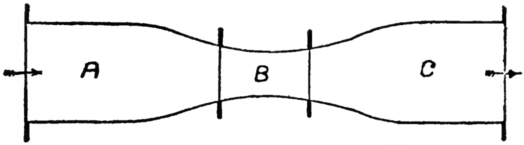


FIG. 58. Principle of Venturi Meter.

In the pipe main two conical pipes, A and C (Fig. 58), are inserted, having a short pipe, B, forming the throat of the meter fixed between them. By Bernoulli's theorem, neglecting friction, the sum of the pressure and velocity heads throughout the pipe remains constant, and since the velocity is greater at the throat and the static pressure therefore less, if we have a manometer or water-gauge connected to the throat B and to the full bore of A, we shall get a difference of static pressure recorded which will be proportional to the square of the velocity.

$$\text{Let } H = \text{inches of water-gauge}$$

$$\text{Then rate of flow} \propto \sqrt{H}$$

In practice the rate of flow can be obtained from a chart supplied by the makers, or the meter can be made to record the rate of flow on a scale.

The rate-of-flow form of meter—either Pitot or Venturi—is most suitable for underground. On the surface a *chart-recording* apparatus can be attached to the meter. A drum is driven by a clock mechanism actuated by a small air-turbine, and upon the chart on the drum a diagram is traced by a pencil actuated by the manometer. The chart gives a continuous record of the rate of flow, and as the abscissæ of the diagram represent time, the area between any two ordinates on the chart gives a measure of the quantity of air that has passed in the given time.

Sometimes a *mechanical integrator* is added, which records upon a series of dials the quantity of air that has passed in a given time.

Other Meters.—Other forms of meters in use are: (*a*) the Standard Orifice meter, which consists of a hole in a diaphragm placed in the pipe, the difference of pressure in the pipe behind and in front of the orifice giving a measure of the rate of flow; (*b*) the Displacement meter; and (*c*) the Swinging-gate meter, consisting of a valve hinged in the air-pipe, whose angular swing under the impact of the flowing air gives a measure of the rate of flow.

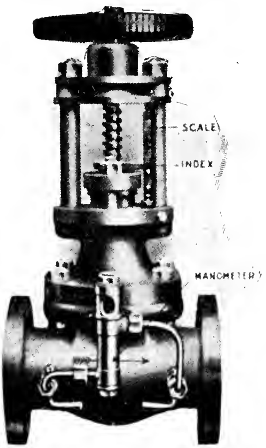
THE "SENTINEL" AIR-METER VALVE.—In this meter, which is made by the manufacturers of the Alley and MacLellan Compressors (*see* Chapter IV), the air-measuring device is fitted to the stop-valve (Fig. 59).

The manometer is shown in section in Fig. 60. The plunger is a free but practically fluid-tight fit in its cylinder.

Pressure air is led to the under side of the plunger from the inlet side of the stop-valve, and the pressure on the outlet side of the valve communicates with the top side of the plunger. The manometer is so constructed that the plunger floats when a certain standard difference of pressure exists on the two sides of the stop-valve, and the movement of the plunger is visible through the glass tube above the plunger.

A scale for indicating the amount of opening of the valve is attached, the opening being indicated by the index pointer.

The area of valve opening and the velocity corresponding to the standard difference of pressure being known, the quantity flowing can be readily ascertained.



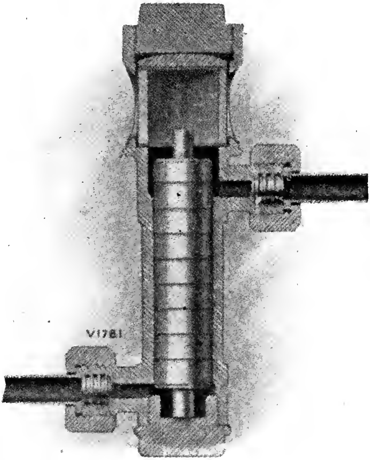


FIG. 59. Meter Valve. FIG. 60. "Sentinel" Manometer for Meter Valve.

OVERALL TESTS OF PLANT

The carrying out of systematic tests on the whole compressed-air system and not merely on isolated parts of it—*e.g.* the compressor—is of the utmost importance. As the outcome of such tests the overall efficiency will be determined, and from the data obtained may be deduced the directions in which improvement should be sought.

Accurate measurement and testing of electrical plant is vital to the system from the point of view of safety as well as efficiency.

While compressed air is free from risk to life and property, the necessity for keeping a check on the efficiency is infinitely more urgent than with electricity.

To secure even a reasonable efficiency constant vigilance is necessary, and neglect soon brings with it an increased loss of power.

The only scientific and practical way of maintaining the highest possible state of efficiency is to institute and practise a rigid system of measurement of quantity and pressure.

By measurements at the surface the performance of the compressing plant will be ascertained.

By measurements at various points on the transmission and distributing pipe-lines the leakage and friction losses will be recorded.

By measurements at the air-engines the consumption of power by these will be obtained.

From the data secured in these tests it will be seen whether the normal efficiency of the various parts and of the system as a whole is being maintained, and if not, attention will be directed to the defective points. Further, the inefficiency of many plants at the present day is notorious, and it is not too much to say that given a little time and patience and a knowledge of the principles of the system, one could, by carrying out a few simple tests and from these ascertaining the weak points, increase the overall efficiency by 50 per cent. or even more in a very short time.

By far the greatest source of loss in compressed-air practice is *leakage.*

The modern compressor is a fairly efficient machine, and the air-motor, though not of the same high-class construction, may still be made to work, if kept in good repair, with a fair measure of efficiency.

The greatest loss occurs in the pipes ; therefore to secure a reasonable efficiency in the use of compressed air one might enunciate the following rule as indicating a sure road to success :

Install sufficiently large piping and hoses to give a low-pressure drop, and thereafter by continuous checks on leakage keep the pipes and hose in such a state of repair as will reduce the loss in transmission to a minimum.

The engineer-in-charge should not be satisfied until the loss between the compressor and the motor is reduced to under 10 per cent.

When it is considered that in many mines the loss in the piping amounts to over 30 per cent. of the total output, it will be readily acknowledged that there is room for improvement.

FREEZING AT EXHAUST PORTS OF AIR-ENGINES

Ordinary atmospheric air always contains a certain amount of moisture, and as long as the air is below saturation-point it will continue to hold the moisture in suspension. This saturation-point is the condition of the air when it carries as large a proportion of water vapour as it is possible for it to hold. A given quantity of air in such a state is unable to take up any more moisture, unless (*a*) its temperature is raised or (*b*) its pressure is reduced. Further, air in a state of saturation will deposit some of its contained moisture if (*a*) its temperature is reduced or (*b*) its pressure is raised. Thus air if compressed isothermally would deposit moisture in the compressor cylinder, but owing to the great increase of temperature that takes place in the actual process, the air is still able to hold the moisture in suspension until cooling takes place in the intercooler, the receiver, and the pipes; there, much of the moisture is trapped and removed.

The compressed air when it arrives at the motor cylinder is still nearly saturated, however, though the weight of moisture per unit mass of air is very much less than when it was received into the compressor cylinder at the pressure of the atmosphere. Now if it were possible to enable the pressure-air to expand isothermally in the engine cylinder or as it was being exhausted, there would be no possibility of deposition of moisture, because, as the air increased in volume, the percentage of water vapour would fall considerably below that producing saturation. But during expansion the temperature falls rapidly, and so in spite of the increase of volume the escaping air is kept up to saturation and a portion of the water vapour is condensed. If this water collects in the exhaust passages of the engine, the low temperature resulting from the expansion of the air causes freezing to take place, occasionally to an extent sufficient to partially or entirely close up the exhaust ports with ice.

With complete expansion the temperatures reached in practice vary from − 100° F. to − 180° F., while, when full pressure throughout the whole stroke is employed, the resulting temperatures, for the same initial temperature and the same pressure, range from − 35° F. to − 70° F. As the force of ejection during exhausting is greater in the latter case than in the former, it is evident, since the temperature is also not so low, that the chances of accumulations of ice in the exhaust ports are less with full pressure throughout than with complete expansion. Thus in coal-cutter and conveyer air-engines and in rock-drills there is generally little trouble from freezing.

Where an attempt is made to use the air expansively, however, the tendency to freezing is more marked.

Re-heating the air immediately before use to such a temperature as will allow expansion in the cylinder without permitting the temperature at the exhaust to fall to the freezing-point of water, *i.e.* 32° F., is the most effective prevention of freezing. Failing this a thorough drying of the air by draining off all collected moisture in receivers and pipes before the air reaches the engines should always be carried out. Where re-heating is impracticable one means of minimizing the risk of freezing is to use relatively high pressures at the compressor, to drain off all deposited moisture as the air falls to the normal temperature of the mine, and then to reduce the pressure by causing the air to pass through a reducing valve. By reason of the resulting increase of volume the air will be relatively much drier when it again attains the mine temperature, and thus when it reaches the air-engine there may be too little moisture in the air to cause freezing, even though low temperatures are produced.*

CUMMINGS' TWO-PIPE SYSTEM OF TRANSMISSION

In this system, instead of allowing the air to pass into the immediate atmosphere on its escaping from the air-engines, as in the ordinary method, it is conducted back again, through a return line of pipes, to the compressor, there to be again raised in pressure and pumped into the out-going pipe-line once more. We have thus a system with a closed circuit, in which, except for leakage, the same air is used over and over again.

The advantages claimed for the system are:

(1) A higher efficiency is obtainable than in the ordinary single line system, because of the fact that compression begins at an initial pressure greater than that of the atmosphere. This advantage, however, only holds if re-heating is impracticable and if the air is not to be used expansively.

(2) Less trouble with freezing at the exhaust ports of the engines as the pressure does not fall to atmospheric.

(3) The efficiency increases with the pressure employed.

Some objections that may be urged against the system are:

(1) The cost of piping is doubled.

(2) The friction losses are doubled and so also are the chances of leakage.

(3) If the air is to be used expansively the pulsations in the return pipe-line become objectionable.

(4) It is not very suitable for portable machines, such as coal-cutters, conveyers, and rock-drills.

The system is best suited for short distances and high pressures, and although not in use in this country, has given satisfaction in several instances in America.

* *See also* under "Haulage," Chapter XI.

CHAPTER VII

INBYE COMPRESSORS

In many mines it is often found advantageous to install compressors not on the surface but in the underground workings. Among circumstances which might make this course advisable are the following:

(*a*) The case of a fiery mine in which it is only possible to install electric motors for power purposes in the main intakes; and where some other motive power must be provided for coal-cutters and other machinery, which have to work in atmospheres which may at any time become explosive.

(*b*) The case of a mine in which electricity is so far the only underground motive power, and in which the need has arisen, temporarily or permanently, for adopting rock-drills in order to expedite the work of cross-measure drives.

(*c*) The case of a deep and extensive metalliferous mine where a great amount of rock-drilling is done. In such a case, if electricity is already installed on the surface, and perhaps also underground, the use of underground compressors at various points in the workings to suit requirements may be made to supplement or eventually supersede the surface compressing plant, which during sinking or in the early stages of development provided all the necessary air-power.

(*d*) In mines in which both compressed air and electricity are being used, instead of adding to the surface air installation when additional demands for power arise, it is often thought better to put in portable inbye compressors to do the work.

The advantages claimed for inbye compression are:

(1) The high transmission efficiency of electricity is taken advantage of to a greater extent than if the air-power were generated on the surface.

(2) Since in such cases the air is used only where electricity would not be permissible or would be unsuitable, we are working under conditions in which the highest efficiency consistent with the other desiderata, viz. safety and convenience, is secured.

(3) As the inbye compressors are almost invariably portable, they are always installed as close to the place where the air is to be used as is possible, and hence the transmission losses are small.

On the other hand, there are several objections to the system, as, for example:

(1) The break in the transmission, and the consequent losses in changing from steam to electricity on the surface, and then from electricity to air underground.

(2) The increased cost of labour. The underground machines require a good deal of attention.

(3) The supplying of clean cooling water is a difficulty. Cooling water is, of course, essential for inbye as for surface compressors, except for pressures under, say, 30 lbs. per square inch, which is somewhat low for general mining requirements. Unless special provision is made for settling and purification the ordinary pit water is too dirty for the purpose, and water must be supplied either from, say, the shaft bottom or from a supply kept in portable tanks and renewed daily.

(4) The difficulty in preventing dust from getting into the cylinder with the air. The underground air, especially in a deep, dry coal-mine, is often extremely dusty, and since dust is destructive to the machine, very efficient filtering must be carried out.

(5) The system is not very suitable for large units, and it is generally considered that if the use of the pressure-air is to be general underground, and hence plant on a large scale becomes necessary, it is better and more efficient to compress the air on the surface than to compress underground even with the aid of electrical transmission.

THE HOT TRANSMISSION SYSTEM.—The hot transmission system has been suggested for inbye working. In this system the air is compressed in unjacketed cylinders and is taken straight to the coal-cutter cylinders or rock drills and used as warm as possible.

The distance the air has to travel must be short, and it is necessary to jacket the compressor and pipes with some heat-insulating material so as to prevent the pressure-air from losing its contained heat.

The advantages of such a system if successfully carried out are:

(1) Larger volume of air obtained for a given working pressure.
(2) The possibility of using the air expansively.
(3) The elimination of freezing at the exhausts.

The disadvantages are:

(1) The impracticability of using high pressures—30 to 40 lbs. per square inch above atmospheric being about the working limit.

(2) The larger cylinders required in coal-cutters and rock-drills consequent on the lower pressure used. Thus the machines would be bulkier and more inconvenient to handle.

Any of the compressors described in Chapters IV and V can be used underground. The motive power is, of course, almost invariably electricity, although in special circumstances it might be possible to use water turbines or wheels or even wire-rope drives.

If the compressor is a fairly large one, it may be installed in a properly constructed fireproof chamber as near as possible to where the air is to be used; but, generally speaking, if at all convenient, it is better that the compressor and motor, and perhaps receiver as well, should be mounted on a bogie, so that it can be moved forward from time to time and so kept as close as possible to the working places.

Although, as has been pointed out, almost any type of compressor can be constructed for inbye work, there are one or two types which, while being also used to a considerable extent on the surface, are specially adapted for underground conditions and these will now be described.

THE REAVELL COMPRESSOR

This compressor is one of the best-known and most efficient machines for underground work.

It possesses the following advantages:

(1) Compactness.

(2) Conveniently arranged to suit direct electric drive.

(3) Delivery of air practically uniform.

(4) Torque on the motor driving shaft very steady, as the result of the radial arrangement of the cylinders.

The general arrangement of the compressor will be understood from Fig. 61.

It consists of four cylinders arranged radially in a circular-shaped casing. Each of the four cylinders is fitted with a trunk piston, and the four connecting-rods are all driven by a common crank pin. The casing contains an annular space through which the cylinders pass, and which is used as a water-jacket.

Each cylinder forms, as it were, a separate single-acting compressor, and as they all deliver into a common delivery passage, a practically continuous stream of air is given out, the compressor being driven at a relatively high speed. This delivery passage is arranged circumferentially, as can be seen from the illustrations.

Very small clearances are used in the cylinders, this being possible by having balance or dummy pistons, D, extending from each main piston, P, and working in a suitable chamber. The air in front of this dummy piston becomes highly compressed near the end of the delivery-stroke of the main piston, and the effect is to hinder the forward thrust on the main piston just at the end of the stroke, and so enable the

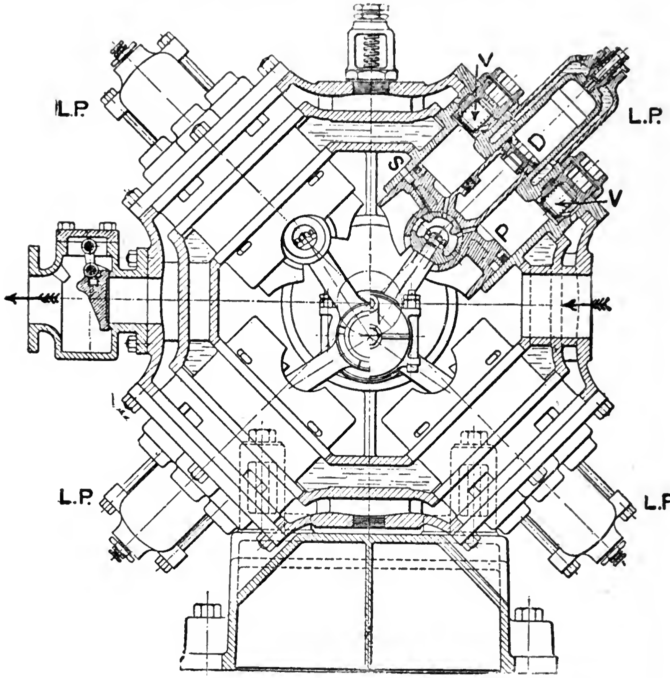


FIG. 61. Reavell Single-stage Quadruplex Compressor.

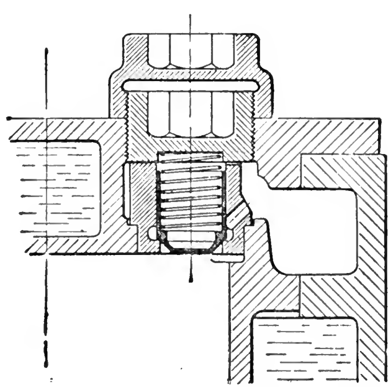


FIG 62. Poppet Valve for Reavell Compressor.

machine to be worked with extremely small clearance. This, of course, increases the efficiency of the machine.

The compressor has no suction valves, air being admitted above each piston by means of a port in the latter, which coincides with a similar

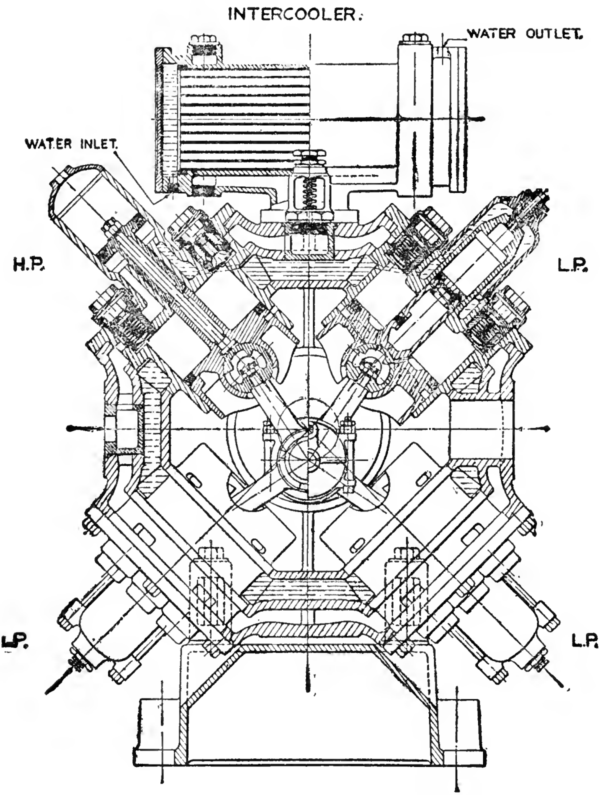


Fig. 63. Reavell Two-stage Compressor.

port in the top of each connecting-rod during the suction stroke; and near the end of the suction stroke the piston overruns the ports (S) cut through the cylinder walls as shown in Fig. 61, thus making direct communication between the cylinder and the inside of the compressor casing, which is arranged to form a suction chamber.

Delivery valves V are fitted at the outer end of each cylinder. These open on the compression stroke on the air slightly exceeding the required delivery pressure, and discharge is made into the circumferential belt or passage which communicates with the delivery. Fig. 62 is an enlarged section of the valve, which is of the poppet type.

Double-ended compressors, consisting of two single-ended compressors with a motor between them, are sometimes used where a single compressor would not meet the requirements. This arrangement

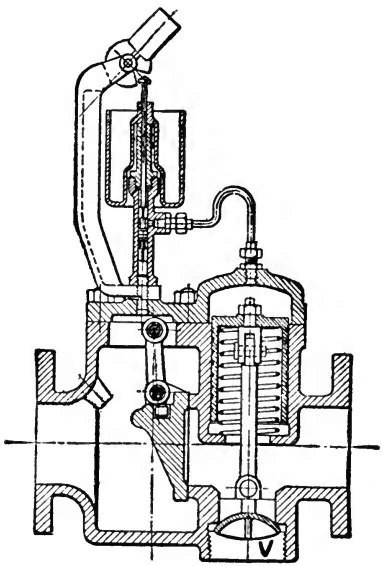


FIG. 64. Reavell Unloading Device.

still further reduces the variation of torque during each revolution of the machine and makes for steady and uniform running.

The Reavell single-stage compressor, as above described, is used for pressures up to 100 lbs. per square inch, but for pressures above that the two-stage type is adopted.

In the two-stage machine one of the cylinders of the single-stage form is replaced by a high-pressure cylinder as shown in Fig. 63. The high-pressure cylinder is fitted with suction valves communicating with the delivery belt from the low-pressure cylinders, and delivers its air through the delivery valves into the delivery outlet indicated by the arrow. An intercooler is fitted on the casing as shown, the air passing from the L.P. cylinders into the intercooler and thence to the H.P. cylinder.

The motor driving the compressor can be fitted with an automatic stopping and starting switch operated by pressure-air from the receiver, and thus the compressor is enabled to readily adjust its output to the varying demand for air. Unloading devices may also be used for a similar purpose. The special unloader used on the Reavell compressor is shown in Fig. 64.

REAVELL UNLOADER.—This device is bolted direct to the outlet and is in the form of a bye-pass valve, making a connexion between the delivery and the atmosphere.

It is automatically controlled by an air relay connected with the receiver, so that when the air pressure reaches the desired limit the compressed air from the receiver, acting on the spring-controlled piston shown in the drawing, forces the by-pass valve V open and the delivery chamber is put in communication with the atmosphere. The load is thus removed. When the pressure has fallen some distance below the normal the valve again closes and the compressor begins to deliver air again.

The Reavell Compressor is, of course, also often installed on the surface, and both above and below ground is an excellent machine because of its suitability for electric drive.

BROOM AND WADE'S COMPRESSOR

In this compressor the inlet valve is opened mechanically against the action of a spring, while the delivery valve is entirely automatic in action.

The piston B (Fig. 65) is of the trunk type, so that the compressor is single-acting. The cylinder C is effectively water-jacketed both at the cylinder-head A, which is wholly available for that purpose, and also for a considerable portion of the cylinder circumference.

The inlet valve I and the delivery valve H are in a pocket on the side of the cylinder. The inlet valve is mechanically operated. At a fixed point in the suction stroke, *i.e.* the down-stroke of the piston, the inlet valve is opened upwards by the action of the cam K acting on the small bell crank, which raises the adjustable tappet J and so opens the valve. The cam is operated by a wiper worked by a link off the connected rod D, as is clearly shown in Fig. 66. A system of forced lubrication is used. A rotary gear oil-pump is mounted direct on the end of the crank shaft F. The pump is positive in its action and has only one pipe connexion to the oil-well G in the base.

The oil is taken through a filter and the pump delivers the oil through its own casing, in a hole drilled through the crank shaft, to each bearing and to the gudgeon-pin bearing at the piston end of the connecting-rod. The oil pump is omitted from Fig. 65 for the sake of clearness,

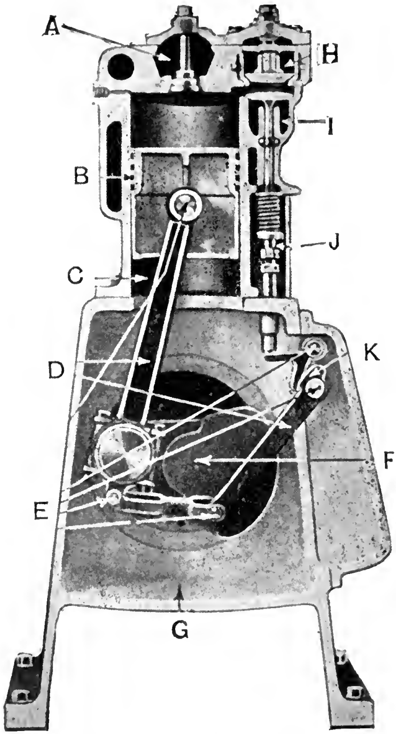


FIG. 65. Broom and Wade's Compressor.

but is shown in outline in Fig. 66. In this figure is also indicated the air-governing device.

ACTION OF AIR-GOVERNOR.—Fig. 67 shows in detail the action of the air-governor. When the desired air pressure is reached in the receiver, the air pressing upon the valve G pushes it from its seat and forces the valve H on to its seat, moving the adjusting screw J and putting a strain upon the spring L through the toggle levers K. The air passing the valve G is admitted to the small cylinder containing

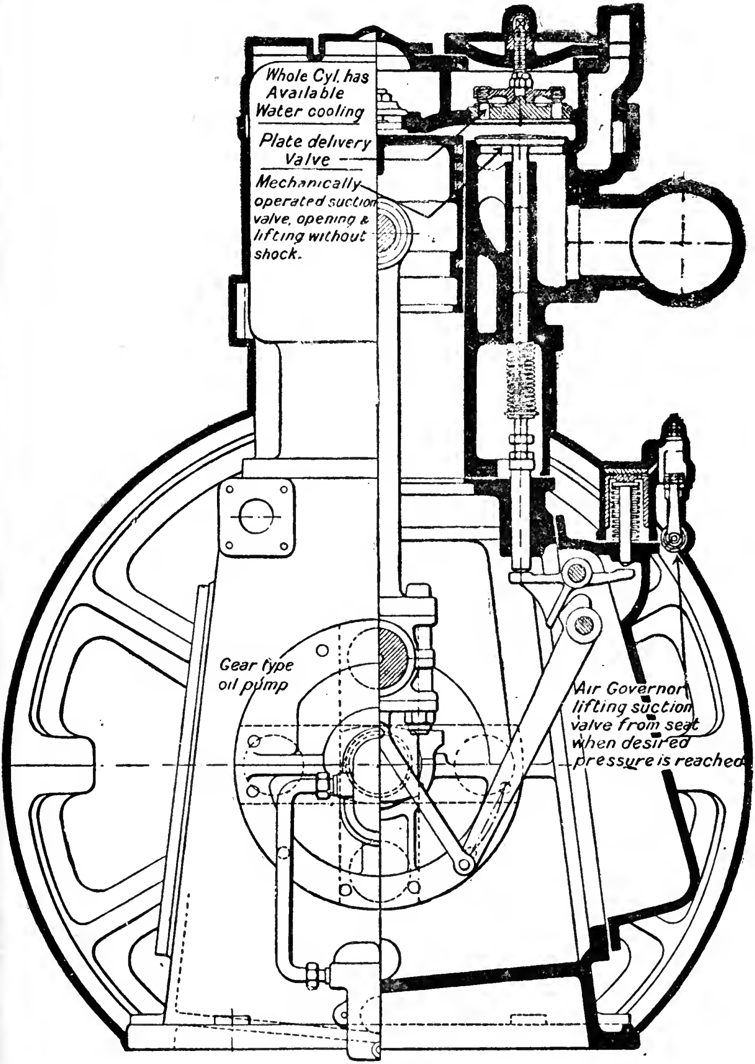


FIG. 66. Broom and Wade's Compressor.

the plunger M, which plunger, acting through a rocking-lever, N, lifts and holds the suction valve of the compressor from its seat until the pressure falls in the receiver. When the pressure drops below the desired point, the valve G returns to its seat, and the air from the plunger cylinder M is exhausted to the atmosphere. It is advisable

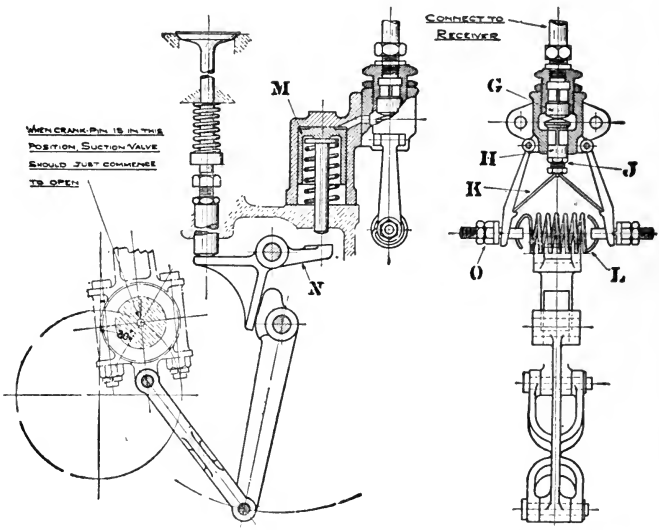


FIG. 67. Action of Automatic Air-governor on Broom and Wade's Compressor.

to take the air for working the automatic air-governor from the top of the receiver. If it is required to adjust the pressure at which the cut-out is to work, the tension upon the spring L is altered by means of the nuts O.

The compressor is made single-stage up to as high as 90 or 100 lbs. per square inch, this being possible owing to the effective water-jacketing. Single-cylinder machines are generally employed for small outputs, but for larger outputs multi-cylinder compressors are used—each, however, being single-stage only.

The compressor is suitable for belt or direct drive, and, being compact, is very suitable for inbye work, being then driven by an electric motor and mounted on a bogie if desired.

INGERSOLL-ROGLER COMPRESSOR

A portable inbye form of this compressor is shown in Fig. 68, from which it can be seen that the arrangement is very compact and extremely portable.

The compressor proper is of enclosed construction and has a hopper type of water-jacket which is open to the atmosphere at the top, and

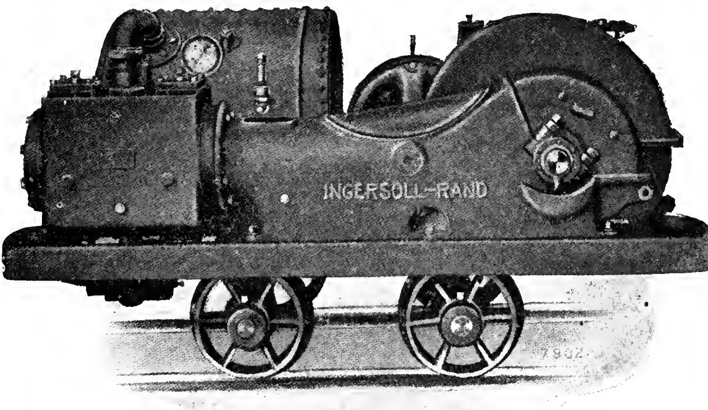


FIG. 68. Portable Compressor on Bogie.

has to be refilled only once a day. The lubrication is automatic. An inlet unloader automatically maintains a constant air-pressure in the receiver, which is mounted on the truck beside the compressor.

The compressor is driven by a motor on the bogie and is suitable for pressures of from 50 to 100 lbs. per square inch.

THE TEMPLE-INGERSOLL "ELECTRIC-AIR" ROCK-DRILL

A very interesting and noteworthy example of a combination of electricity and air as a means of transmitting power is furnished in the Temple-Ingersoll "Electric-air" drill, which is very extensively used for prospecting and development work generally; it is also used in quarrying, and in both metalliferous and coal mining.

In this machine the air is not *used* in the sense that it is taken from the atmosphere at one point, compressed, and then discharged into the atmosphere at some other point, but is only employed, as it were, as an elastic cushion between the motor-driven air-pistons and the operating piston or plunger of the rock-drill.

The general arrangement of the drill and the necessary plant is clearly shown in Fig. 69.

The drill is driven by pulsations of compressed air created by a pulsator actuated by an electric motor. Sections through the pulsator and the drill are shown in the illustration. The air is never exhausted, but is simply used over and over again, playing backward and forward in a closed circuit.

The drill itself is of the simplest possible type—a cylinder containing a moving piston and rotation device, no valves, springs, side-rods, or pawls being used. The cylinder is larger but the piston shorter than that of the corresponding air-drill. The pulsator requires no intake or discharge valves or water-jackets, and two short lengths of hose connect it with the drill, each hose acting alternately as supply and exhaust.

The air is contained in the hose and cylinders between the pistons of the pulsator and the front and back of the piston of the drill itself, The air is slightly compressed, *i.e.* it is at a pressure somewhat greater than atmospheric, so as to give it greater density; and it may be considered as a spring or cushion between the pulsator and the drill, the pressure in the air-supply giving the requisite tension.

There is, of course, some leakage of air during working, and provision is made for this loss by a compensating valve on the pulsator, which is adjusted automatically to maintain the requisite pressure in the circuit. When the pressure, due to leakage, falls below this determined limit, the valve automatically opens and admits a small amount of free air, which is compressed by a differential area on one of the pulsator pistons, until the normal working pressure is restored.

The "electric-air" drill is more economical in power consumption than the ordinary type of air-drill, and its drilling capacity and mudding qualities are very good; it is being largely used where electric power is available.

One specially noteworthy point is that high altitudes, which greatly impair the efficiency of the ordinary air-compressor, make little or no difference to the economical or effective working of this type of air-drill.

A weak point, however, is that it does not strike the same powerful blow as the ordinary rock-drill.

The system described can also be applied to the reciprocating type of coal-cutter, but its scope of usefulness in this connexion would appear to be confined to narrow work and drivages

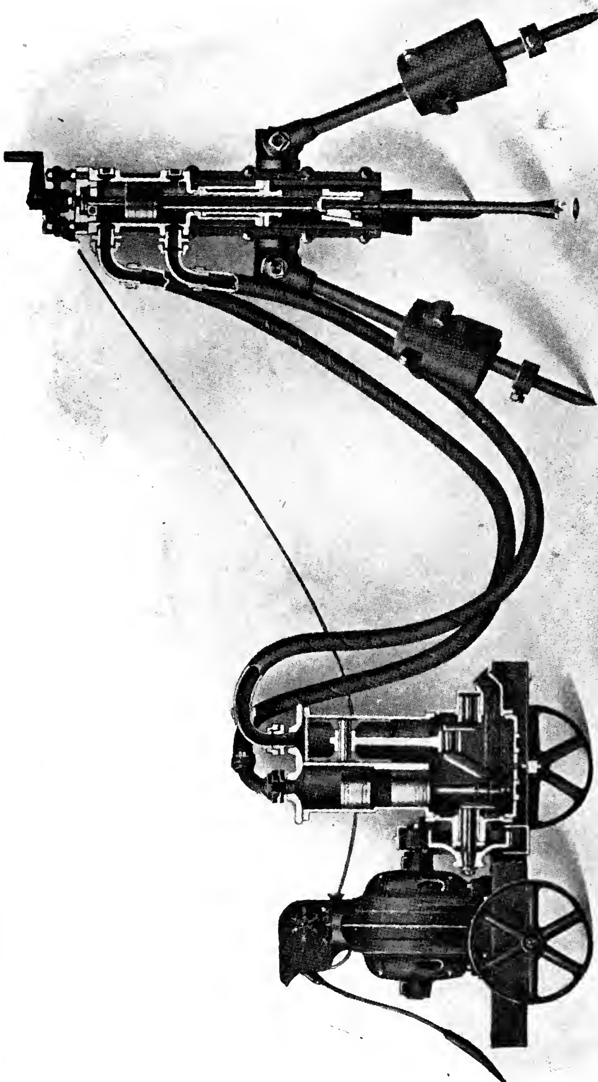


FIG. 69. The "Electric-air" Rock-drill.

CHAPTER VIII

COAL-CUTTING MACHINERY

The laborious process of hand-holing in coal-mining is, every year, to a greater and greater extent being abandoned in favour of under-cutting by means of machines.

The advantages accruing may be summarized as follows:

(1) A larger output per man employed.

(2) A larger tonnage obtained from the same length of face.

(3) More rapid advance of the face and hence better control of the roof is obtained.

(4) Greater profit in working thin seams.

(5) Larger percentage of round coal in soft seams.

(6) Aid in the ventilation due to straighter line of face.

(7) Better division of labour is obtainable—each man does one job only, and this tends to greater efficiency.

(8) Less quantity of explosives needed.

For operating the coal-cutters only two forms of motive-power are used, viz. electricity and compressed air. Compressed air was first in the field as a motive-power in mines, and hence the first coal-cutters were driven by air. When the economical and other advantages of electricity became generally recognized, however, the electrically driven coal-cutter became a serious rival to the compressed-air machine, and, indeed, except for one class of coal-cutter—the percussive type—electricity can now lay claim to be the chief motive-power for coal-cutting machinery. So far, however, as the total number of machines in use is concerned, compressed air is still ahead of its rival—principally, it must be confessed, because of the much greater number of percussive coal-cutters worked by air than by electricity. And when one considers the very large number of rock-drills in use in coal and ironstone mines and in sinking pits—a field in which electricity has not, so far, been very successful—it becomes evident that there is still a considerable scope for compressed air in the various operations of mining.

The curves plotted in Fig. 70 show the mean rate of increase since 1902 in the use of both electrically operated coal-cutters and machines operated by compressed air in British mines.

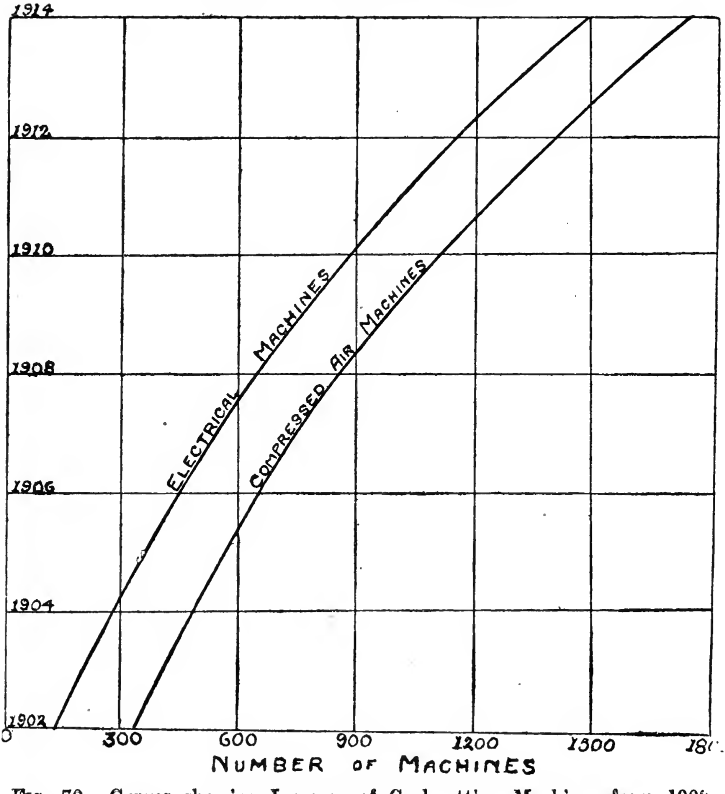


FIG. 70. Curves showing Increase of Coal-cutting Machines from 1902 to 1914.

COAL-CUTTING MACHINES.—Table VIII (p. 136) sets out the number of coal-cutting machines driven by compressed air and electricity, and the quantity of mineral cut by each, in the mines of Great Britain and Ireland.

The particulars are compiled from H.M. Mines Inspectors' Reports for 1914, with the permission of the Controller of His Majesty's Stationery Office.

The bulk of the machine-cut mineral included in the above totals is, of course, coal, but a small percentage comprises ironstone, ganister, etc.

From the table it will be seen that in the longwall types of machines —disc, bar, and chain—electricity is the chief motive-power, while for the narrow work machines—rotary heading and percussive—compressed air is predominant. As regards total amount of mineral cut electricity is an easy first.

TABLE VIII

Description.	Number driven by		Mineral cut by	
	Electricity.	Compressed Air.	Electricity.	Compressed Air.
			Tons.	Tons.
Disc . .	775	487	8,372,450	5,126,690
Bar . .	415	172	3,821,270	1,877,044
Chain . .	219	75	2,000,268	813,715
Percussive .	4	922	11,954	2,203,025
Rotary heading	4	22	2,619	43,488
Totals .	1417	1678	14,208,561	10,063,962

Since the total number of compressed-air machines is greater than the aggregate of the electrical machines, it would appear that as regards quantity cut per machine, compressed air comes out very badly. If we take each type of machine separately, however, as in the accompanying table, we find that the comparison assumes a different aspect.

TABLE IX.—QUANTITY OF MINERAL CUT PER MACHINE

Machine.	Cut by Electricity.	Cut by Compressed Air.
	Tons.	Tons.
Disc	10,800	10,530
Bar	9,200	10,910
Chain . . .	9,130	10,850
Percussive . .	2,990	2,400
Rotary heading . .	655	1,980

The first three machines are primarily longwall machines, and it will be seen that as regards quantity cut per machine compressed air is easily as good as electricity.

The percussive type, which is more particularly suitable for narrow work, has a much lower quantity per unit to its credit than the longwall coal-cutters, this being, however, very largely due to the short width of face worked and the consequent large proportion of idle time during a shift. To the low output of the percussive machine is due the fact of the lower total tonnage produced by compressed air.

The comparison made in the table is, of course, a very general one and is not asserted to be a proper or sufficient criterion. A true comparison could only be obtained by running a test in the same seam under equal conditions over a considerable period, with attention to, amongst others, the following points:

(*a*) Consumption of power per yard or per ton cut.
(*b*) Rate of cutting.
(*c*) Cost for attendance, upkeep, and repairs.
(*d*) Percentage of time lost in stoppages.

With regard to the last three points compressed air would probably keep on at least level terms with electricity, but as regards the first, it is to be feared it would come out rather badly.

The wastefulness of the air machine as compared with the electrical is due chiefly to two causes, viz.:

(1) Coal-cutters work with a very late cut-off and hence make little or no use of the expansive properties of the pressure-air. Thus at the end of every stroke a large proportion of the energy in the air is ejected unused into the atmosphere.

The electric motor, of course, does not waste power in this way.

(2) The air-engine cannot adjust itself automatically, as can an electric motor, to the work it has to do, and thus, *when running unloaded*, it takes a much larger percentage of its full load consumption.

This percentage for the air-engine is often 20 to 50 per cent., while for the electric motor it may be under 10 per cent.

In order to obtain the highest efficiency in power consumption it is therefore much more imperative that the air machine be kept at its maximum cutting speed than is the case with the electrically-driven machine.

It should be remembered too that coal-cutter motors are at work a long distance inbye, and as the percentage efficiency in the ordinary working of compressed-air transmission falls off much more rapidly than in electrical distribution of power, the coal-cutter air-motor is placed at a big disadvantage as compared with the electric motor, since in many cases nearly half of the energy stored in the air may have been lost during the passage from the compressor to the air-motor at the face

By the use of shorter and larger diameter hoses and bigger pipes, together with more attention to keeping down leakage losses, the efficiency of compressed air for coal-cutting could very soon be enormously improved.

That the need for reform is urgent cannot be doubted by any one who has read the recent paper on the subject by Mr. Sam Mavor, in which a deplorable condition of things is brought to light.

TYPES OF COAL-CUTTERS

(1) **THE DISC MACHINE.**—This is the most favoured type of coal-cutter for longwall work in British mines. It consists of a horizontal

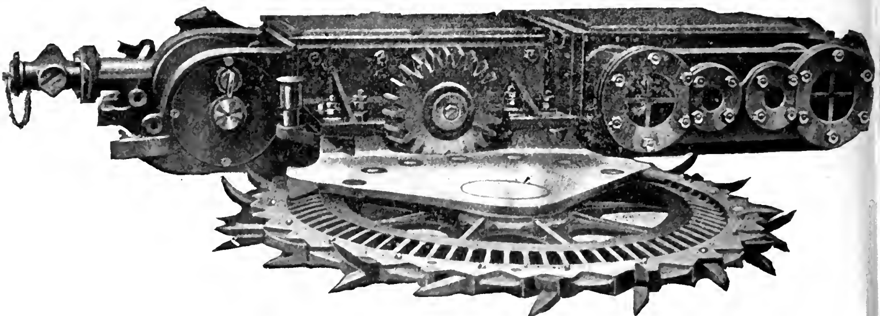


FIG. 71. Anderson-Boyes Disc Machine.

wheel or disc, carrying cutter picks attached at intervals round the periphery of the disc, which is revolved, through gearing, by the air-engines.

The disc itself is toothed and the last wheel of the reduction gearing engages with the teeth on the disc, and so rotation is produced.

In the *Anderson-Boyes* machine (Fig. 71) the cutter picks are single- and double-pronged and are inserted in sockets on the periphery of the disc and held in position by steel cotter pins. In the *Diamond* coal-cutter the picks are single-pronged and are held in cutter boxes which fit over, and are secured to, a lug or projection on the rim of the disc by a feathered steel pin.

Each cutter box holds three single cutters, which are inserted in the box from the inside before the latter is fixed to the disc, and are secured by a small lip forged on the heel. The arrangement of the cutters and the method of attachment will be readily understood from Fig. 72, which shows segments of the disc with and without cutter box pan picks attached. *A* is a part of the disc, *B* is a cutter pick, *C* the cutter box, and *D* and *E* show sections through the disc and cutters.

The disc may be arranged to cut in any position from close to the floor, as in Fig. 73, to eighteen inches or more above floor-level.

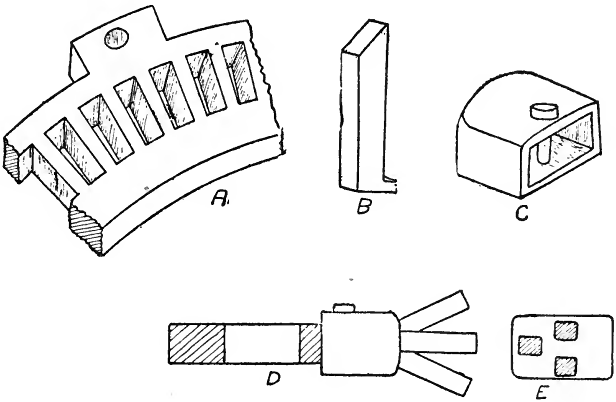


FIG. 72. Details of Cutters for Diamond Machine.

The travel of the machines along the face is secured by the following arrangement:

A haulage drum (shown in front of the machine in Fig. 73) is operated through gearing from the engines. To this drum a flexible steel wire rope is attached. The rope passes along the face, round a small pulley or wheel attached to a firmly set prop, and returns to the machine, to the frame of which it is fixed. As the drum is made to rotate it pulls in the rope, with the result that the coal-cutter, sliding on skids or running on rails, is pulled along the face.

The rate of travel of the machine can be regulated to suit the nature of the material cut by altering the "throw" of the connecting-rod which drives the haulage drum.

The general arrangement of the gearing in the Diamond disc coal-cutter is shown in the diagram in Fig. 74.

Two air-cylinders, C_1 and C_2, are set as shown, and through connecting-rods and cranks actuate the shaft S_1. The pinion P_1, which is keyed to this shaft, drives, through intermediate gearing, the spur wheel P_2, which imparts the motion to the shaft S_2. The bevel wheel B is also

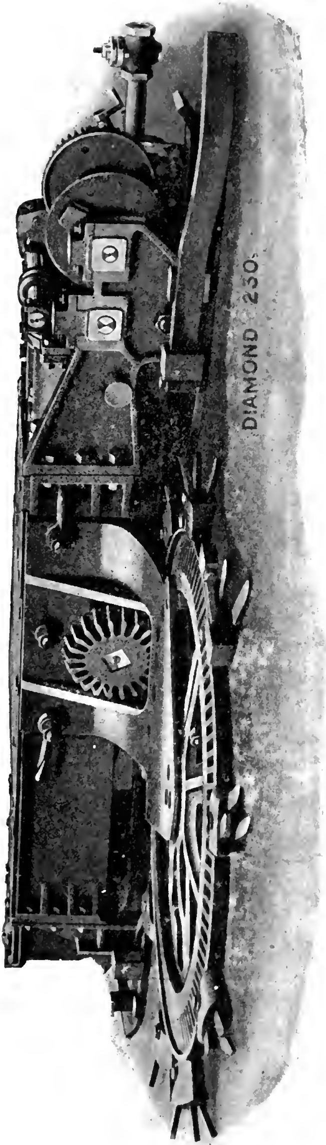


Fig. 73. Diamond Disc Coal-cutter.

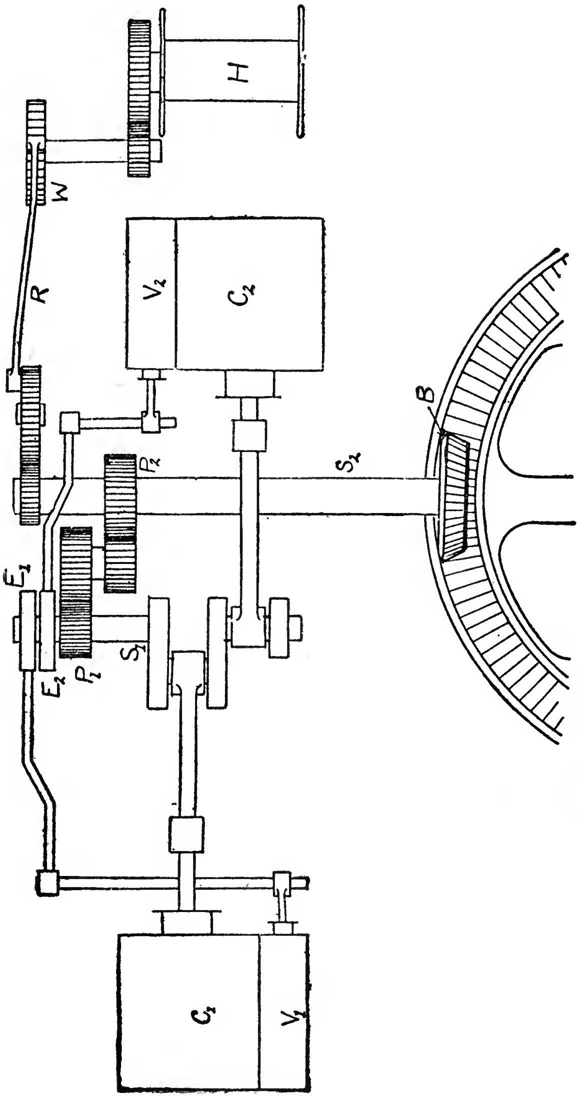


FIG. 74. Arrangement of Gearing in Disc Coal-cutter.

keyed to the shaft S_2, and this wheel gears into the disc, part of which is shown.

The slide valves in V_1 and V_2 are operated by the eccentrics E_1 and E_2 from the first shaft, S_1.

The haulage drum H is driven through gearing from the second shaft, S_2, the connecting-rod R giving a cranked motion to the ratchet-wheel **W**.

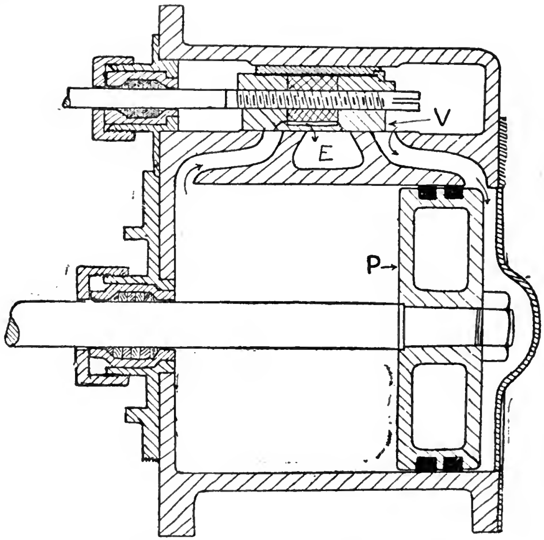


FIG. 75. Slide-valve Air-engine.

The Air-engines.—Air-engines in disc machines, and indeed also in the revolving bar and chain machines, are of two types:

(*a*) Double-acting piston engines with ordinary slide valves.

(*b*) Single-acting trunk piston engines with piston valves.

The first type is shown in Fig. 75 and the second in Fig. 76; in both diagrams P is the piston and V the valve. In Fig. 75 the live air is shown, by the arrows, entering one end of the cylinder, while the air from the other side of the piston is escaping from the cylinder through the exhaust port E.

In the *Diamond* disc machine, which has the first type of engines, the cylinders are 9½ inches in diameter by 9 inches stroke.

The ns *Anderson-Boyes* machine has four cylinders, single-acting trunk pisto , and piston valves.

The Gillott Coal-cutter.—This machine has been in successful use for very many years. It is a light machine and under favourable conditions has done splendid work.

The cutters are fitted to the periphery of the disc in much the same way as in the Anderson-Boyes machine. A pair of slide-valve engines are used, the cylinders being placed side by side at one end of the machine. The cylinders are 9 inches in diameter and about the same length of stroke.

Although a good machine where the conditions are suitable, it is not

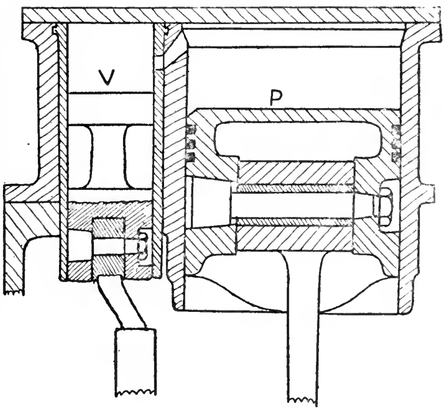


FIG. 76. Piston-valve Air-engine.

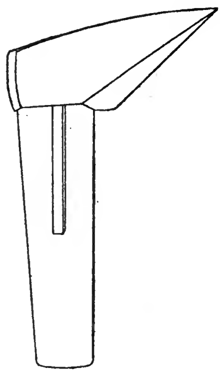


FIG. 77. Cutter Pick for Bar Coal-cutter.

so well suited for heavy work and deep cuts as the Diamond or the Anderson-Boyes coal-cutters

THE BAR COAL-CUTTER.—In this type of machine the cutter picks are attached to a circular steel bar five to six feet long or more, depending upon the depth of undercut.

The portion of the bar carrying the cutters is slightly tapered and is provided with a spiral worm or thread to assist in bringing the cuttings out of the holing. The remainder of the bar enters the gear-head end of the machine and carries a bevel wheel which receives motion through suitable gearing from the air-engines. The length of the portion carrying the cutters determines the depth of the undercut.

The cutter picks (Fig. 77) have tapered shanks and a feather formed at one side, and fit into tapered holes on the cutter bar.

In addition to the rotary motion of the bar when cutting, a slow and short reciprocating motion is given by a worm upon the boss of

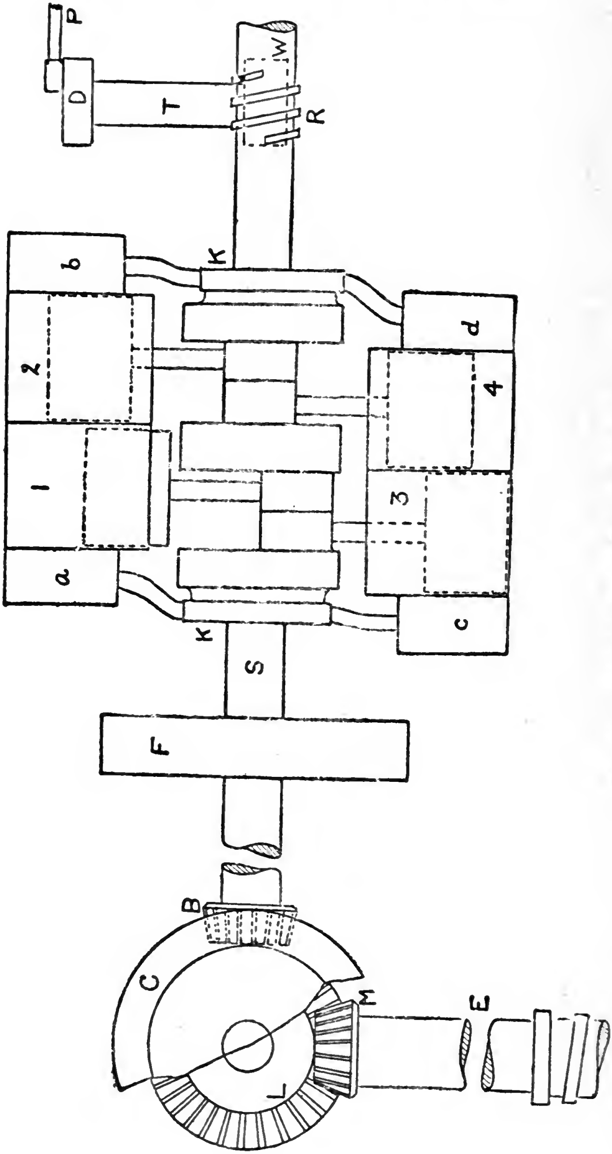


FIG. 78. Arrangement of Bar Coal-cutter

the bar, driving pinion gearing with two small wheels each of which drives a toggle by means of an eccentric pin. The toggles impart a to-and-fro motion to a thrust block on the extreme gear-head end of the cutter bar, and so the latter also moves to and fro during rotation. The reciprocating motion helps to keep the cutter bar from clogging and prevents the cutters from working into grooves.

Fig. 78 shows diagrammatically the general arrangement of the bar machine.

The air-engine consists of four cylinders, 1, 2, 3, and 4, two being placed horizontally on each side of the crank shaft S, which is arranged in the longitudinal axis of the machine.

The pistons are of the single-acting trunk type, being set at different angles on the crank circle so as to secure as near as possible uniformity of turning moment. A small but heavy fly-wheel, F, mounted on the crank shaft helps still further to smooth down the small fluctuations of speed due to the cyclic torque variation.

Admission of air to the cylinders is controlled by piston valves *a*, *b*, *c*, and *d*, worked from eccentrics, K, on the crank shaft. For details of the cylinder and valves *see* Fig. 76.

At the gear-head end of the crank shaft is the bevel pinion B, which gears into the crown bevel wheel C above it. Below C on the same vertical spindle is the bevel wheel L, gearing into M, which is on the end of the cutter-bar E, and so the latter is revolved.

The worm for working the reciprocating-motion gear is on the bar just behind the wheel M, but is omitted in the drawing.

At the other end of the crank shaft is the worm R, gearing into the worm-wheel W, which operates the haulage gear through the shaft T, disc crank D, and connecting-rod P in a similar manner to that shown in Fig. 74 (p. 141) in connexion with the disc machine.

Splash lubrication is used. At each revolution the cranks dip into the lubricant in the crank chamber and splash it on to the pistons, valves, bearings, etc.

The *Pickquick* coal-cutter is the best-known bar machine.

An advantage of the Pickquick bar type of machine is that it can be got to start away a cut without any hand-hewing whatever.

To enable this to be done the cutter-bar can be swung through an angle of more than 90° in a horizontal plane. Fig. 79 shows how the operation of starting a cut is performed. The cutter-bar is swung back into the extreme position, and then as it revolves, cutting in the ordinary way, the bar is slowly swung round until it is in its normal cutting position at right angles to the machine, when the work of cutting the machine " run " proper commences.

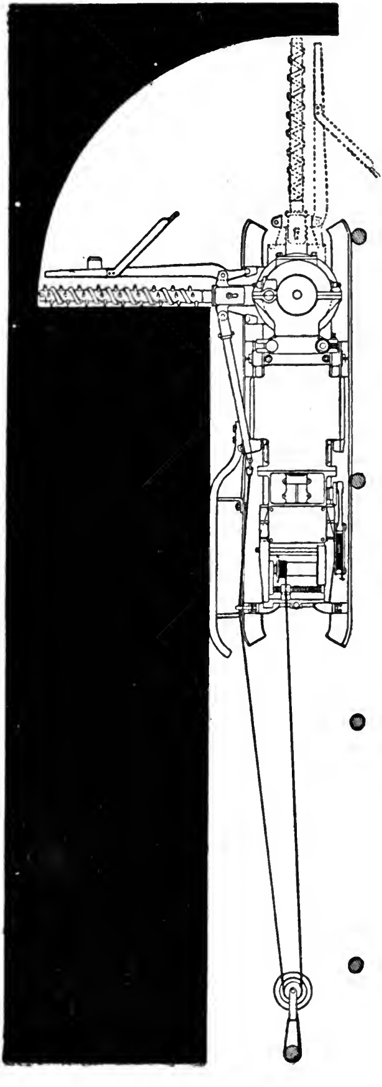


FIG. 79. Bar Machine "Cutting in."

THE CHAIN MACHINE.—In this type of coal-cutter the cutting picks are fixed to the links of an endless steel chain which runs in a groove in a jib or frame. The cutter chain passes round the driving pinion at the rear end of the frame and at the front end passes round two or more small guide pulleys.

The chain machine can be used either for narrow work or for longwall. For the former purpose the frame carrying the cutter chain is movable and can slide inside a stationary frame, at the rear end of which the air-engine or motor is placed. At the commencement of a cut the machine is firmly jacked against the face and roof by means of screw-jacks provided on the stationary frame. The machine is then started and the cutter chain revolves, and at the same time the movable cutter frame is pushed slowly forward by a feed arrangement. The cutters thus cut a rapidly deepening groove under the coal.

When the cutter frame has reached the limit of its travel, the feed is reversed and the sliding frame travels quickly backwards into the position for starting again. The machine is then shifted laterally a distance equal to the width of cut and another similar groove is cut in the coal, and so on until the full width of undercut is obtained; or the cutting may be continuous across the face (*see* Fig. 89).

In the machine as designed to work on a longwall face, the cutter frame is at right angles to the length of the machine when the latter is cutting along the face, although it can be swung out in a straight line with the body of the machine when cutting in at the commencement of a cut or for any special purpose.

The cutter frame does not slide inside another but is in a fixed position, although, as mentioned, it can be swung when sumping under the coal from a position lengthwise of the machine to the right-angle position, being pulled round by a rope wound on a drum worked by the motor.

The feed rope, in the *Jeffrey* and *Sullivan* machines, is guided in such a way that the machine is held up to the face without the use of props or guide rails. The rate of feed is variable, as in the other types of machines.

The Jeffrey Compressed-air Machine.—This is a chain machine and is particularly interesting as regards the method of driving, an air-turbine being used instead of the usual reciprocating engine.

The arrangement of the machine is shown diagrammatically in Fig. 80, the different component parts being indicated.

The turbine actuates the cutting chain through worm gearing, there being also a spur gear between the turbine and the worm gearing. The turbine consists essentially of two rotors and an enclosing case, the

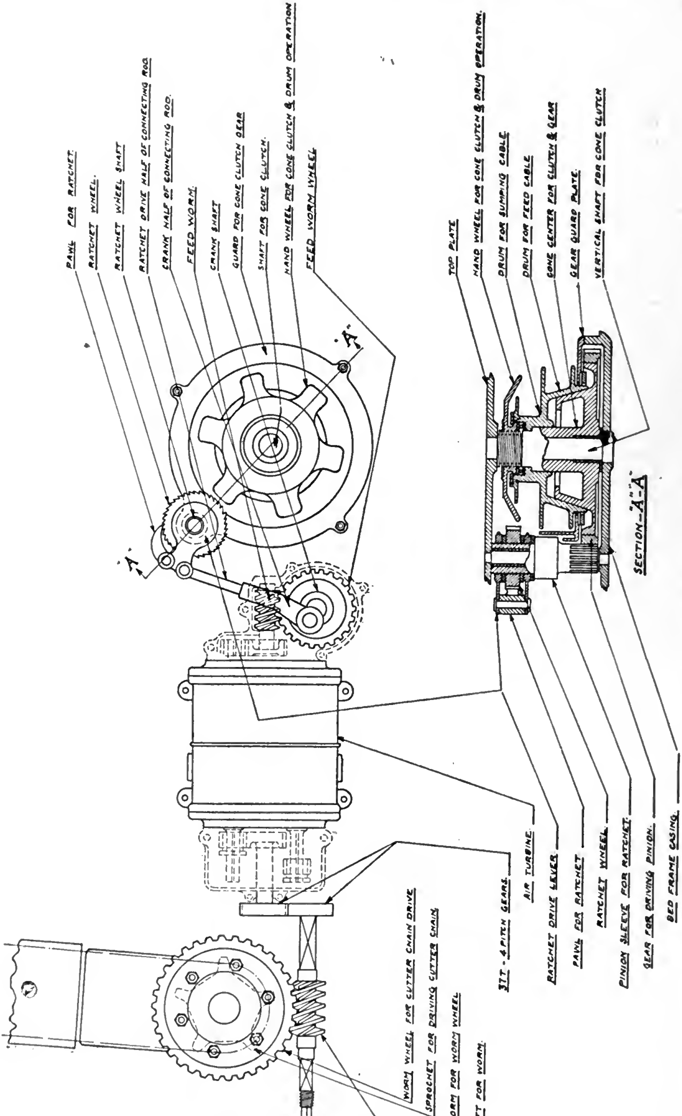


FIG. 80. Arrangement of Parts of Jeffrey Air-turbine Chain Coal-cutter.

rotors being practically helical gears (Fig. 81). This construction is simple and strong, there being no blades or buckets forming a source of weakness.

It is found that after running some time the teeth of the gears have acquired a smooth polished surface of great hardness, which afterwards undergoes no change.

There are no valves, nozzles, connecting-rods, or eccentrics to get out of order and give trouble, and it will thus be seen that very conspicuous advantages attach to this recent development—at least so

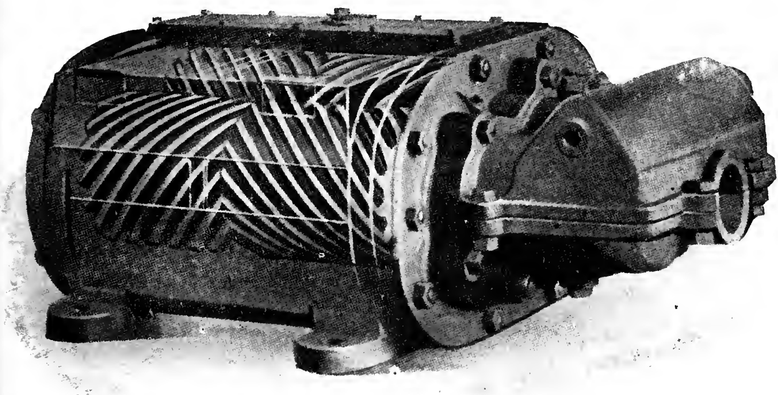


Fig. 81. Air-turbine.

far as British mines are concerned—in the application of compressed air to coal-cutters.

The air gains admission to the interior of the casing under the rotors and in the mid-position. The exhaust is through ports in the sides of the casing and above the rotors.

A relief valve is fitted so as to admit air to assist in balancing the rotors during the period of slowing down when the power has been cut off.

Fig. 82 shows a top view of the machine with the cover removed. It is built in two distinct sizes, one for ordinary working and the other for thin seams, the latter being 15 inches high and weighing 4000 lbs., while the former is 19 inches and weighs 6000 lbs. One of these machines has been doing highly successful work in a South Wales mine, working at 50 lbs. pressure in a 2 feet 10 inches seam on a 150 yards run.

In other forms of the Jeffrey chain machine ordinary piston engines with slide valves are used. The engines, of which there are two, with cranks set 90 degrees apart, are placed above the frame at the rear end of the machine.

The valves are adjusted for three-quarters cut-off, which is found to give the most economical air consumption consistent with satisfactory working.

The Sullivan Chain Machine.—The latest type of this machine is also driven by an air-turbine of a type similar to that used in the Jeffrey machine.

The Sullivan coal-cutter is designed to work at any pressure from 35 to 50 lbs. per square inch. A friction clutch is used between the motor pinion and the driving sprocket of the feed-chain arrangement which propels the machine forward when cutting. The clutch is arranged to slip when the pull reaches 5000 lbs., a convenient margin above the normal, but still within the capacity of the machine. This serves to safeguard the working parts of the machine should a jam occur.

The feed is a distinguishing feature of the machine, being carried out by a continuous chain actuated through a gear train from the turbine; the rate of feed can be varied to suit requirements.

Fig. 83 shows the arrangement of a Sullivan machine on a long-wall face. One end of the feed chain is fixed to an anchor jack or prop, while the other end is attached to an adjustable "take-up rig."

The drive sprocket, at the front of the machine for the direction of travel shown in the figure, grips the chain and propels the machine forward. This arrangement of feed enables the machine to be kept up to the face without the use of props.

PERCUSSIVE MACHINES.—These machines are distinct from those of the other types, in that instead of the undercutting being done by a continuously tearing action, the coal is holed by a rapid series of powerful blows struck by a reciprocating steel bit.

The percussive machine indeed is of a similar design and works in the same way as the well-known reciprocating form of rock-drill, and is often used for the purpose of drilling shot or wedge holes in coal and stone, in addition to the work of coal-cutting.

It is in the percussive type of machine, as is frequently pointed out elsewhere, that compressed air proves its superiority to electricity; it can be used for undercutting, shearing, or for boring holes. It is specially suited for narrow work, and is not very suitable for the ordinary operation of longwall coal-cutting, except on comparatively short lengths of face.

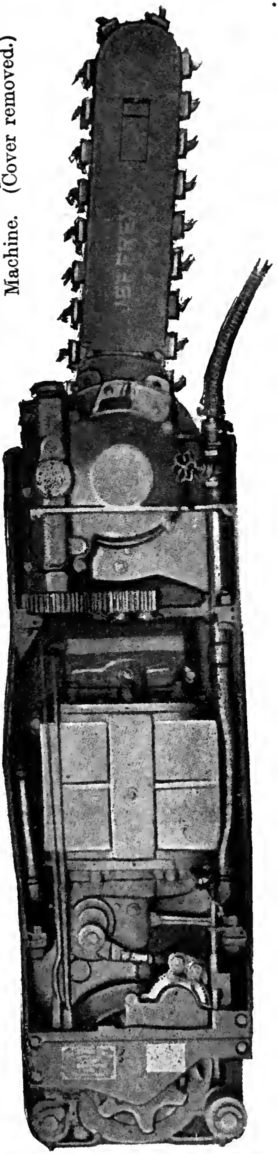


FIG. 82. Top View of Jeffrey Machine. (Cover removed.)

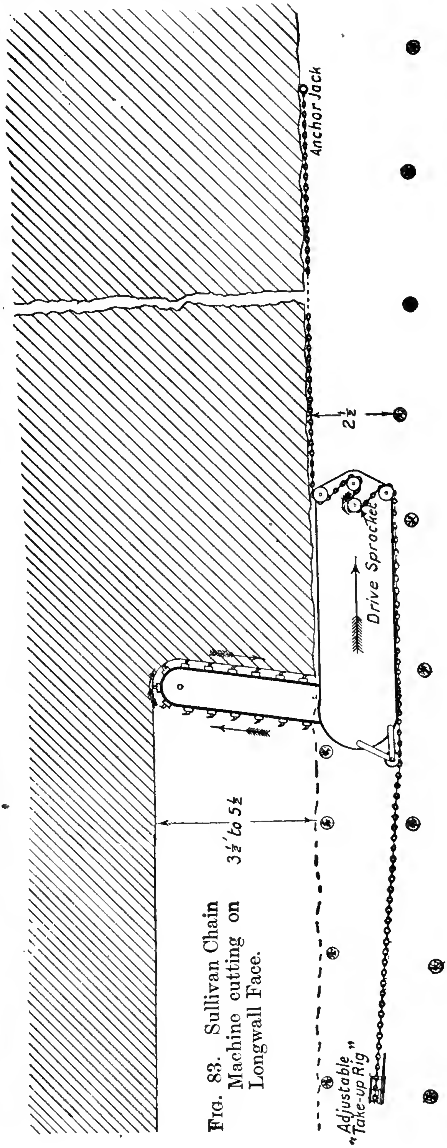


FIG. 83. Sullivan Chain Machine cutting on Longwall Face.

The machines are fixed to a column or standard as shown in Fig. 84, which shows a *Siskol* machine set for holing in mid-seam position.

In shearing the percussive machine is made to cut in the form of an arc of a circle across the whole height of the place, or, in undercutting, across an arc corresponding to an angular swing of anything up to 150° or more. This angular movement is obtained by means of

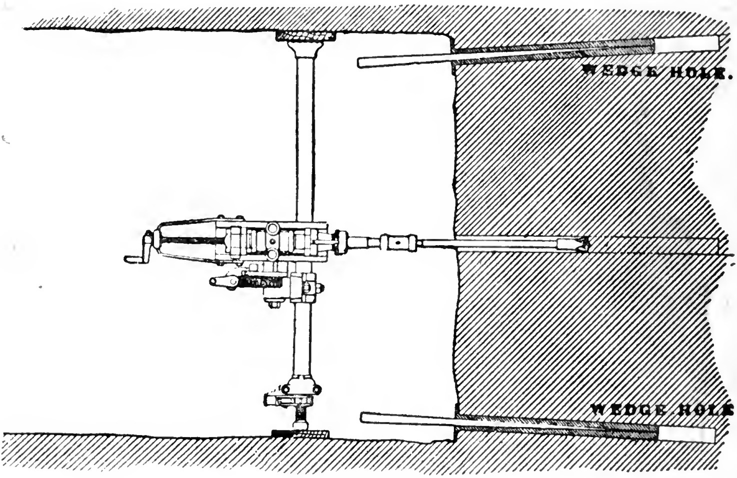


FIG. 84. Percussive Coal-cutter set for Holing.

a worm gearing into a segment of a worm wheel which is bolted rigidly to the column. By turning a handle on the worm the coal-cutter is swung gradually round the column, or in shearing the machine is swung in a vertical plane. A forward movement is secured by a feed screw worked by a handle at the back of the machine. Thus a gradually deepening groove is cut in the coal.

The cutting bit has three, five, or seven cutting prongs, according to the nature of the material in which the machine is working. The depth of cut may be anything from 4 to 7 feet, successive lengths of cutter bar being inserted as required. As much as 18 to 20 feet in width can be cut for one setting of the column. In a wide place or on a longwall face, two or more settings up of the machine may be required.

The coal-cutter proper weighs 240 to 280 lbs., the total weight, including supporting columns, segment, extension rods and cutting bits, being from 500 to 600 lbs.

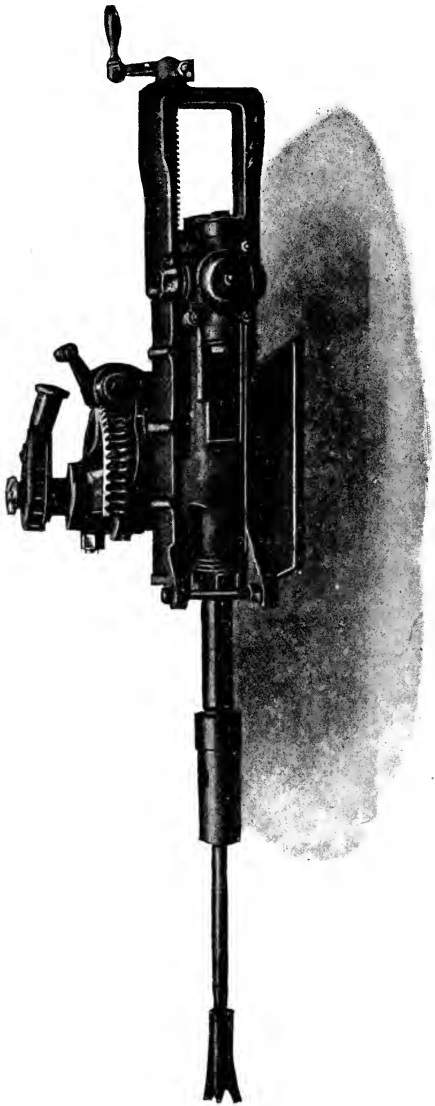


FIG. 85. Longwall Percussive Coal-cutter

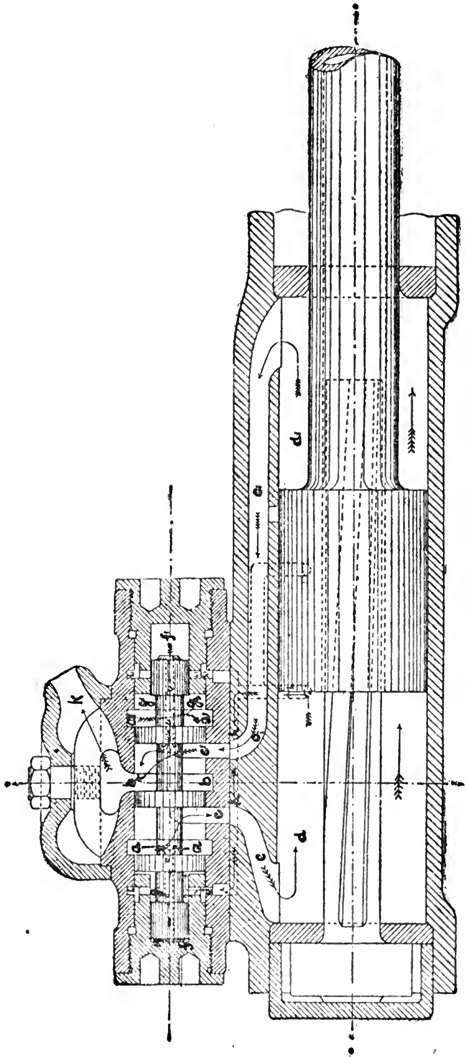


Fig. 86. Siskol Coal-cutter. (First position.)

For very thin seams the design of Siskol coal-cutter used on a longwall face is that shown in Fig. 85.

The "Siskol" Coal-cutter.—This machine is one of the best known British makes of percussive coal-cutters. The piston, cylinder, and valves of the machine are shown in section in Fig. 86.

The action of the valve is as follows:

The piston is on the forward or striking stroke and air is entering the cylinder through the ports a and c. The piston valve is held in the position shown by means of the compressed air which passes through the port a' into the annular chamber g'. At the same time pressure-air through a gets into the passage e (shown dotted) and passes to f at the left-hand end of the piston valve as in the diagram. The surface subjected to the air-pressure at f is, however, smaller than the surface in the annular chamber g', which is subjected to the same air-pressure, so the piston valve still remains stationary.

But when the piston has moved far enough forward to uncover the port h (shown dotted), which communicates with the cylinder d, compressed air rushes from the cylinder through the port h and along the passage shown dotted to the port i and into the annular space g.

In the figure the piston is shown just uncovering the port h.

In this position the air-pressure on the piston-valve face in g' will be balanced by the pressure on the similar face in g, and since the air at the right-hand end of the valve at f_1 is in communication with the exhaust k through e' and b, the force exerted by the pressure-air on the end of the valve at f pushes the piston valve over against the right-hand end of the valve chamber, into the position shown in Fig. 87.

The exhaust of the air in front of the piston on the forward stroke is clearly indicated by the arrows.

On the back stroke the piston is again shown in the position in which reversal of the valve is about to take place, and as the action is the same as that described for the forward stroke, the reader, with the aid of the flow arrows, will be able to see how the "throw-over" of the valve is accomplished.

The usual size of the machine is 3½-inch diameter cylinder, and 9-inch to 12-inch stroke, and works with 50 to 80 lbs. air-pressure. For lower pressures cylinders of 4 and 4½ inches in diameter are used.

The "Hardiax" Coal-cutter.—This is another well-known percussive machine.

The valve action will be understood from the simple diagram in Fig. 88.

In the position shown the valve is hard over against the stop on the left-hand side. This allows the supply air which comes in at L_3

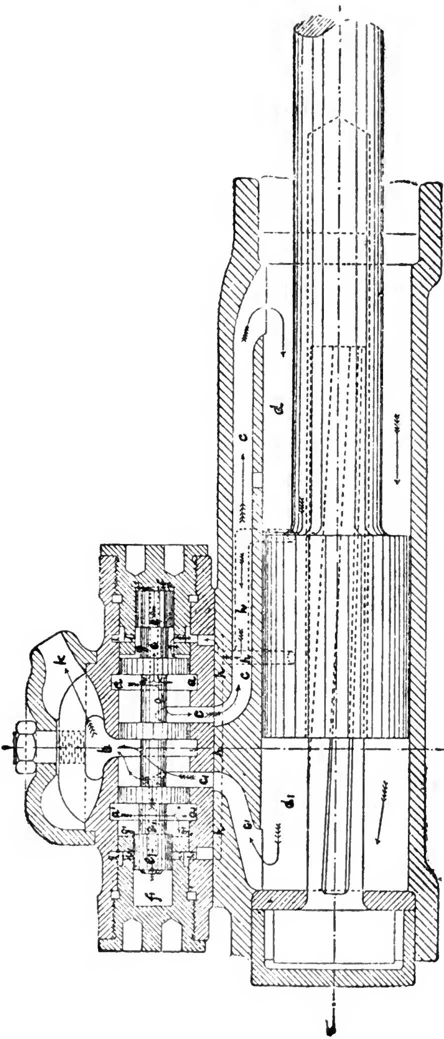


FIG. 87. Siskol Coal-cutter. (Second position.)

to pass down and through the port F_1 into the cylinder and so force the piston P in the direction of F_2. At the same time live air from L_1, which is also in communication with the supply, passes through the port in the valve shown dotted, to the space D, and so holds the valve in the position shown.

When the piston moves forward far enough to uncover the port H, shown dotted, pressure-air passes through H into the valve chamber

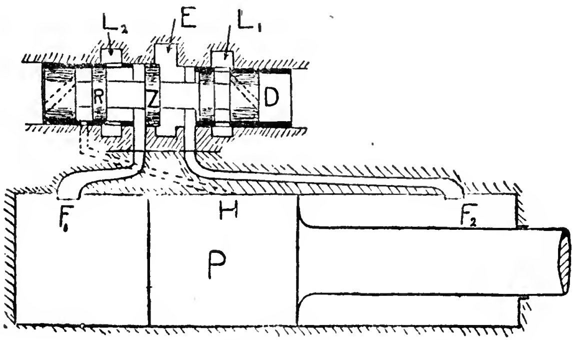


FIG. 88. Action of Hardiax Coal-cutter.

on the left-hand side of R, and thence through the port shown dotted to the left-hand end of the valve. This balances the pressure on the valve ends.

Now the live air at L_2 acts on the faces of the collars R and Z, but the latter has a larger area than the former and so the excess pressure forces the valve over against the right-hand end and puts F_2 in communication with live air through L_1. During the stroke considered the exhaust in front of the piston escapes through F_2 to the exhaust at E. On the return stroke F_1 is placed in communication with E. On this stroke the action of the valve is similar to what has been described. The passage for the other end of the valve corresponding to H is omitted from the drawing.

The Hardiax machine is made in several sizes to suit different conditions, the most popular for undercutting in narrow work being the medium size, weighing about 230 lbs. and suitable for pressures of 45 to 75 lbs. per square inch.

Other types of reciprocating coal-cutters are the *Ingersoll* and the *Meco*. Both of these are used to a considerable extent; their action is similar to that of the two described above.

THE ROTARY HEADING MACHINE.—In this machine the cutters are attached to an arm which revolves in a circle, and so a circular core of coal $4\frac{1}{2}$ to 5 feet in diameter is cut out.

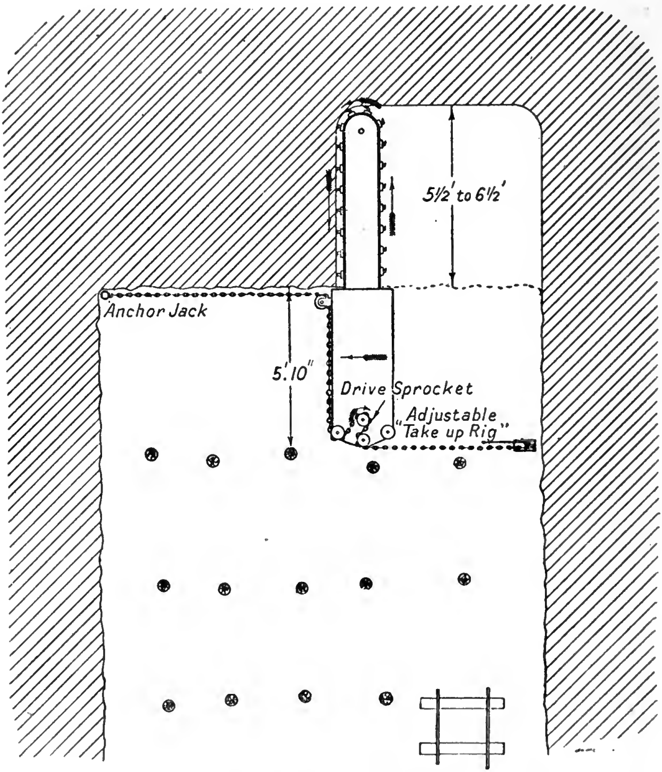


FIG. 89. Chain Machine cutting a Heading.

It is a heavy and cumbersome machine and it has not been used to any great extent in British mines. It may be of service in thick seams where levels and headings in the coal require to be driven at a rapid rate.

The chain machine, however, is very suitable for this kind of work, and its use is increasing for narrow drivages, while the percussive machine is also very extensively used for the same purpose. For this reason there seems little need for a special design of machine for heading work alone.

Fig. 89 shows the method of cutting a narrow heading or stall with a chain machine of the Sullivan type.

AIR-CONSUMPTION OF COAL-CUTTERS.—The pressures used vary considerably, from as low as 30 lbs. up to 70 lbs. per square inch at the machine.

The higher pressures, however, are almost exclusively used for the reciprocating type of machine, and in general the longwall coal-cutter works at pressures somewhere between 35 and 50 lbs. per square inch.

A pressure of 35 lbs. at the machine means, of course, anything from 50 to 70 lbs. at the surface, depending upon the distance the air has to be taken.

As regards air-consumption the following represent average figures:

Reciprocating machines .	120–150 cubic feet of free air per min.
Chain machines . .	450–750 „ „ „
Bar and disc machines .	500–800 „ „ „

The air-consumption will, of course, depend largely on

(1) The condition of the cutter picks.
(2) The state of repair of the machine.
(3) The material cut.
(4) The skill and care of the driver.

TABLE X.—TESTS OF COAL-CUTTERS REFERRED TO IN AIR-BALANCE (TABLE XI). AIR-PRESSURE AT SURFACE, 50 LBS.

(1) Coal-cutter reference number.	(2) Nature of holing material.	(3) Diameter of trailing hose, in inches.	(4) Length of trailing hose, in yards.	(5) Number of observations.	(6) Air temperature, in degrees Fahr.	(7) Pressure at gate-end, in pounds.	(8) Pressure drop in hose, in pounds.	(9) Pressure at coal-cutter, in pounds.	(10) Cutter speed, in inches per minute.	(11) Depth of cut, in inches.	(12) Cubic feet of free air per minute.	(13) Cubic feet of free air per square yard cut.
1	Coal and fireclay	2	26	38	62	36·2	11·5	24·7	12·0	34·0	893	2808
2	Coal and fireclay	2	55	30	61	39·2	19·9	19·3	10·6	33·0	635	2326
3	Coal and fireclay	2	40	18	61	36·0	14·0	22·0	18·5	34·0	475	980
4	Fireclay .	2	55	18	60	38·0	18·0	20·0	20·0	33·0	515	1010
5	do. .	2	39	18	59	30·9	17·4	13·5	26·2	29·0	618	1054
6	do. .	2	27	20	62	35·6	13·0	22·6	19·2	33·4	478	967

The figures given in columns 6 to 13 are averages of the number of observations stated in column 5.

DR. TABLE XI.—AIR-BALANCE CR.

Reference.	Description.	ubic feet of free air compressed in $15\frac{1}{2}$ hours.	Reference.	Description.	Square yards cut.	Cubic feet of free air per sq. yd. cut.	Cubic feet of free air per machine.	Total.
A	Single-stage, single-cylinder, double-acting air-compressor. Cylinder, 30 inches diameter, 48 inches stroke, volumetric efficiency 80 per cent., revolutions of compressor in $15\frac{1}{2}$ hours = 27,900	868,577	1	Coal-cutter. Running light, 12 per cent.	72·55	2808	203,720 24,446	228,166
			2	Coal-cutter. Running light, 12 per cent.	93·00	2326	216,318 25,958	242,276
			3	Coal-cutter. Running light, 12 per cent	50·00	980	49,000 5,880	54,880
			4	Coal-cutter. Running light, 12 per cent.	74·00	1010	74,740 8,968	83,708
B	Single-stage, single-cylinder, double-acting compressor. Cylinder 26 inches diameter, stroke 48 inches, volumetric efficiency, 80 per cent., revolutions of compressor in $15\frac{1}{2}$ hours = 31,620	736,746	5	Coal-cutter. Running light, 12 per cent.	107·00	1054	112,778 13,533	126,311
			6	Coal-cutter. Running light, 12 per cent.	105·00	967	101,535 12,184	113,719
				Loss by leakage at joints, valves, drains, etc., 44·6 per cent. of total air compressed.				716,000
				Unallocated losses, 2·5 per cent. of total air compressed.				40,263
		1,605,323						1,605,323

In a very valuable paper read before the Institution of Mining Engineers,* Mr. Sam Mavor gives particulars of exhaustive tests on compressed-air plants supplying power to coal-cutters. In Table X

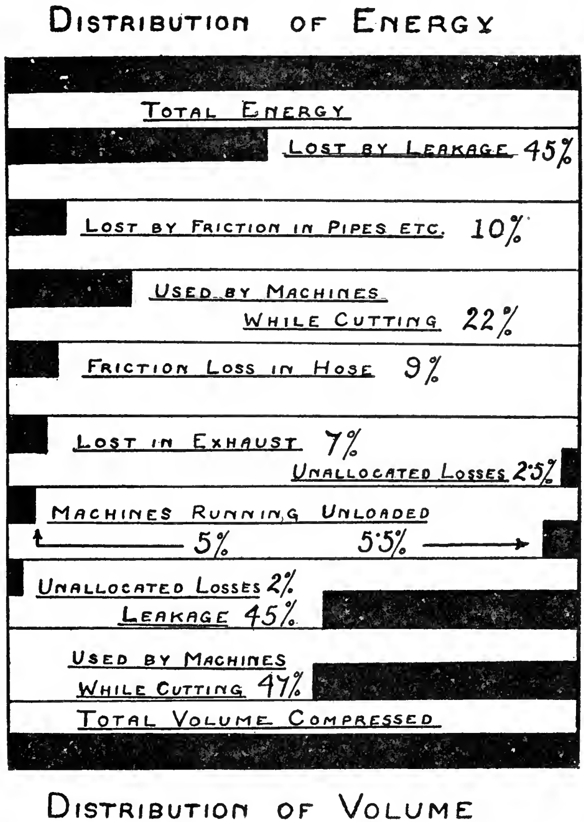


FIG 90. Distribution of Volume and Energy of Air supplied to Coal-cutters

(p. 159) particulars are given of one of the two sets of tests carried out, and in Table XI (p. 160) the " air-balance " for the test is set out.

A graphic representation of the distribution of the volume and energy of the compressed air is given in Fig. 90. The distribution of volume

* *Trans.*, vol. l, p. 625

is shown from the bottom upwards and on the right-hand side, while the distribution of energy is plotted from the top downwards and to the left-hand side.

From the data obtained in this test Mr. Mavor deduces the overall efficiency of the whole compressed-air system as follows:

Total area undercut . . .	502 square yards.
Total volume of air compressed .	1,605,323 cubic feet.
Volume of air compressed per square yard cut . . .	3,200 cubic feet.
Energy required to compress 3200 cubic feet to 50 lbs. per square inch	7·83 brake-horse-power hours.

Assuming that the materials cut require the machine to develop 0·5 brake-horse power-hour per square yard cut, probably a liberal estimate,

$$\text{Efficiency} = \frac{0{\cdot}5 \text{ B.H.P. hour developed by coal-cutter motor}}{7{\cdot}83 \text{ B.H.P. delivered to compressor}}$$
$$= 6{\cdot}38 \text{ per cent.}$$

This value is certainly low enough to call for inquiry, and a contributory cause was found to be the state of the first two machines in Table X; the tests revealing that No. 1 machine was badly in need of overhaul, and that No. 2, although but a short time in use, was in a very defective condition. As a result of the tests both machines were withdrawn.

CHAPTER IX

UNDERGROUND CONVEYORS

THE underground conveyor is an appliance designed to facilitate the loading of the hutches at the coal face.

By its use the coal is conveyed from all parts of the line of face to a common main gate-road, thus dispensing with the need for making and keeping in repair the frequent roadways that would be necessary if no conveyor were used.

The advantages to be gained may be summarized as follows:

(1) Increased output from a given length of face.

(2) Few roads required.

(3) The cost for ripping or brushing is much reduced, as only one main road and perhaps one or two secondary roads have to be kept in repair for a length of face of anything from 60 to 150 yards, whereas in the ordinary method there would be a road every 10 or 12 yards.

(4) Larger tubs or hutches can be used since the main gate-road is the only one to be kept in good repair, and hence it will pay to maintain height and width sufficient to allow of the use of a large size of hutch.

(5) Less cost for "putting" or "tramming," since the hutches all come out one road, and in many cases mechanical or horse haulage can be periodically extended to keep close to the advancing face.

(6) Quicker advance of face.

(7) General economy resulting from the reduction in the cost of producing the coal and in the working and repairing costs.

The employment of conveyors generally calls for a closer supervision of the operations at the face, since a break-down of the roof, a stoppage of the conveyor, or failure to carry out any of the various provisions necessary to enable working to proceed may mean a very serious shortage of output.

TYPES OF AIR-DRIVEN CONVEYORS

There is quite a host of different forms of coal conveyors now in use, amongst which the following are perhaps the most important:

(1) The endless chain conveyor.

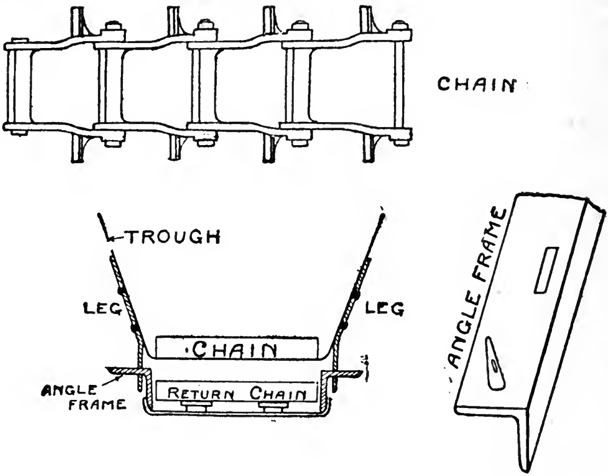
(2) The endless belt conveyor.

(3) The shaker conveyor.

(1) **THE ENDLESS CHAIN CONVEYOR.**—In this type an endless steel chain sliding on the bottom of a continuous line of wrought-iron troughs (Fig. 91), and passing underneath them on the return journey, is used.

The chain is built up of links (Fig. 92) and the motion is obtained from a toothed drum at the delivery end, the teeth of which grip behind the cross-bars of the links and so keep the chain in continuous motion.

The driving drum is operated from an air-motor through suitable gearing.



FIGS. 91, 92, 93. Details of Blackett's Chain Conveyor

At the return end of the conveyor, which is the end farthest from the main gate-road, the chain passes round a similar drum, which revolves in pedestals free to slide for about a foot on the supporting frame; thus by means of a screw and ratchet, tension can be put on the chain by pulling back the drum.

The troughs are in 6-foot lengths and are carried in angle-iron frames (Fig. 93) which rest on the floor or pavement of the seam. On the return journey the chain passes along under the troughs, being carried on cross-pieces in the angle frames.

The coal is thrown into the troughs and the large pieces are carried by the chain, while the small is scraped along on the bottom of the troughs.

The conveyor is anchored at the far end to a stout prop set firmly between roof and floor.

The Blackett conveyor, to which the foregoing more particularly refers, is one of the best known of this type.

The conveyor must, of course, be kept conveniently close up to the face, and this necessitates frequent shifting, which can be done in two ways, (1) shifting bodily by slewing, or (2) breaking up the conveyor into short lengths, carrying forward and rebuilding in the new position.

The first method is the more suitable if the roof is strong.

The alternative method is perhaps better for comparatively weak roofs, where wooden chocks or pillars may frequently have to be put in between the conveyor and the face.

Fig. 94 shows the conveyor in position close to the face, ready for the commencement of the operation of the clearing away of the newly machine-cut coal.

In the figure the coal-cutter is at the top of its run, *i.e.* at the return end of the conveyor. At this end of the machine run, a narrow place is kept constantly a few yards in front of the main line of face so as to enable the coal-cutter to start off with full depth of cut, without the delay that would be necessitated if "cutting in" had to be done by the machine itself.

The road at this end of the run is generally for the purpose of facilitating the bringing in of timber supplies, and also provides an additional means of ingress and egress to the conveyor face.

At the loading end the floor or pavement of the seam is ripped for a sufficient depth to allow the trams to come under the delivery end of the conveyor (Fig. 94). Where the output from the conveyor face requires an uninterrupted supply of hutches, a double line of rails is used with flat sheets or a crossing on the inbye side of the conveyor as shown in the figures.

The road at the discharge end is kept some distance in advance of the conveyor face.

In thick seams it is unnecessary to lift the floor stone, the delivery end of the conveyor being raised to enable the hutch to pass underneath. Conveyors are, however, most common in comparatively thin seams.

(2) **THE ENDLESS BELT CONVEYOR.**—There are several forms of this type of conveyor. In general principle it is similar to the endless chain conveyor, a canvas or wire-cloth belt being substituted for the chain.

In the Sutcliffe conveyor the belt is 20 inches wide and runs between angle-irons bolted to brackets so as to form a trough. The belt runs on rollers carried by the brackets, each of which supports two rollers—one for the loaded side of the belt, the other for the under or return

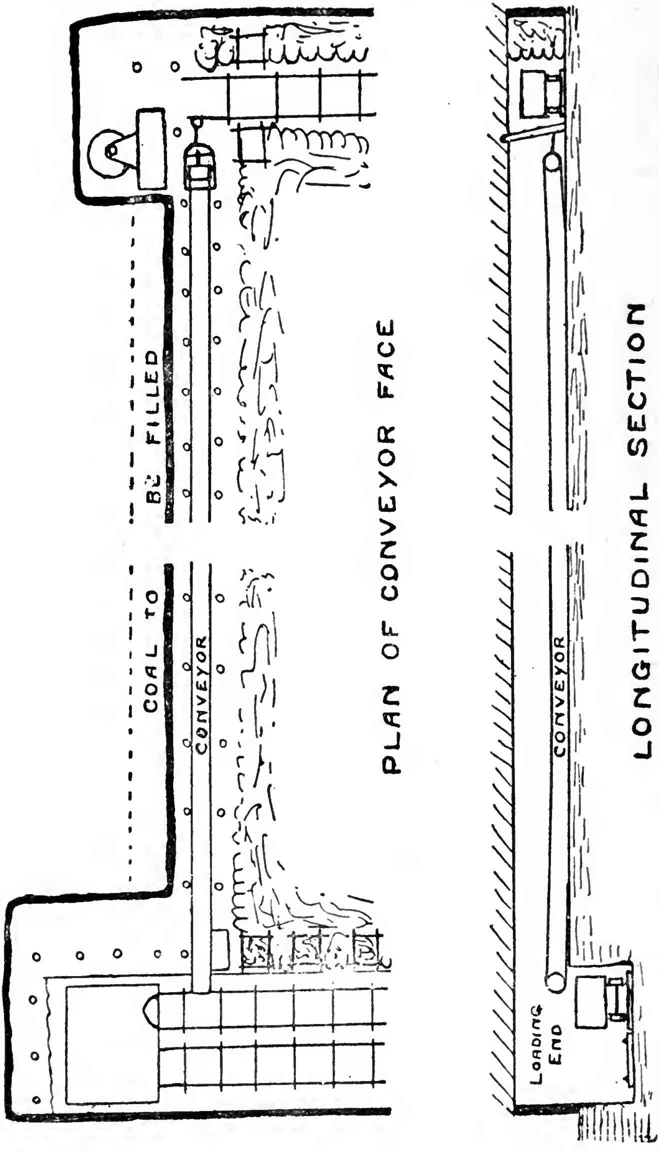


FIG. 94 Plan and Elevation of Conveyor Face.

side. The Spence belt conveyor is similar in design, a canvas belt also being used.

The belt at the driving end, as shown in Fig. 95, passes over and under three equal drums, two of which are driven and the other runs free. The idle drum serves to lift the loaded side of the belt free of

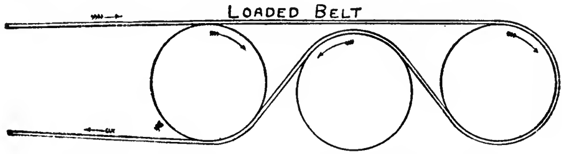


FIG. 95. Driving Arrangement in Belt Conveyor.

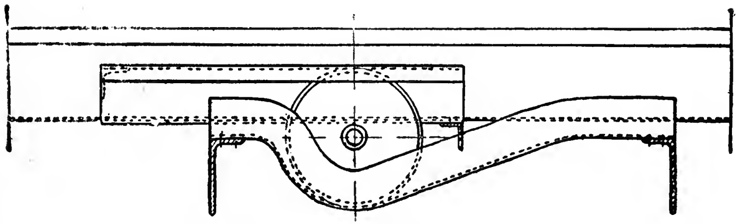


FIG. 96 Side View of Trough and Rollers of Shaker Conveyor

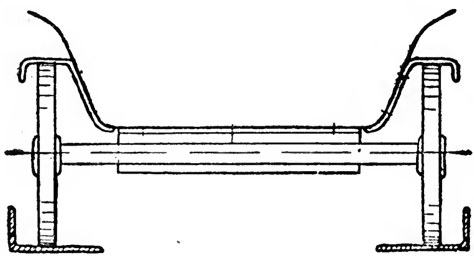


FIG. 97. End View of Trough and Rollers.

the middle drum, which runs in the reverse direction to the leading or endmost drum. The leading drum projects well over the road to facilitate discharge of the coal. For supporting the belt wooden rollers are placed at intervals of about two yards.

The method of driving where compressed air is used is either by means of a reciprocating engine or air-turbine, worm gearing being introduced to give a convenient belt-speed, which is about 100 feet

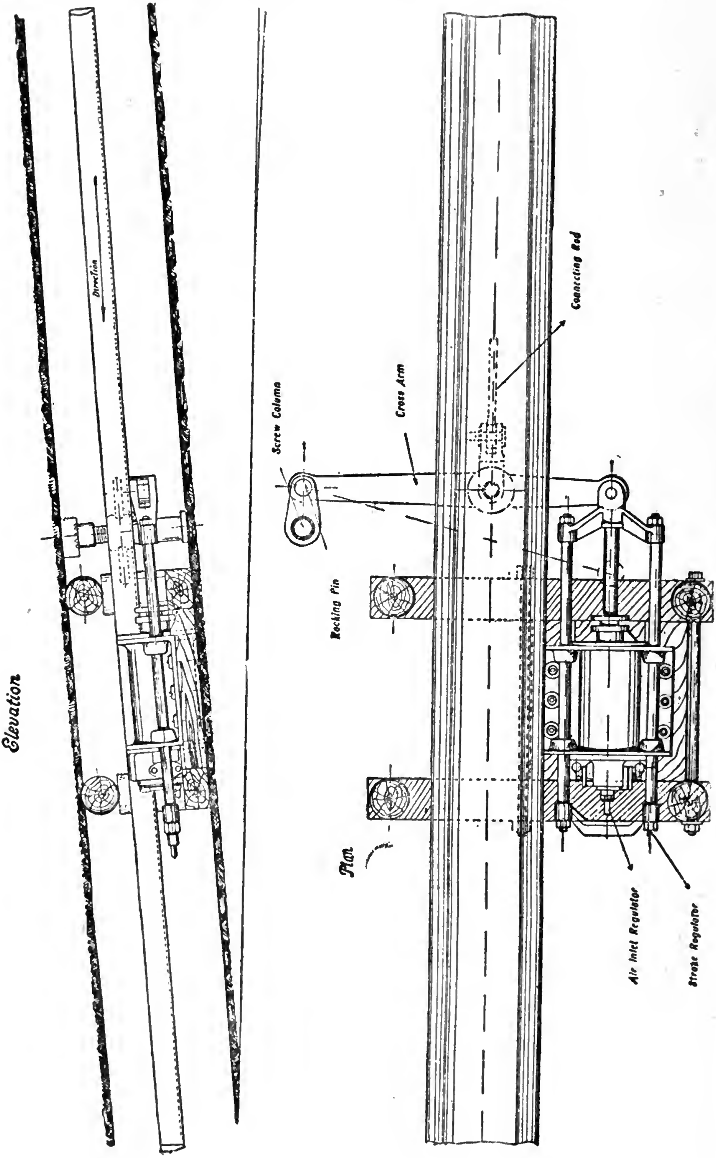


FIG. 98. Driving Arrangements of Shaker Conveyor.

per minute. Tension is maintained at the back end by a drum whose position can be adjusted with screws.

An advantage of the belt conveyor is that there is less noise than with the chain type. The belt is also lighter and easier to handle during shifting.

(3) **THE SHAKER CONVEYOR.**—In this type of conveyor there is neither an endless chain nor belt, but only a long shoot or trough into which the coal is thrown.

In the *Meco* shaker conveyor the trough is built up in 12-feet lengths, carried on rollers running in guides, as shown in Figs. 96 and 97.

By means of a cross-arm (Fig. 98) rocking on a pin which moves on a screw column, motion is transmitted from a reciprocating air-engine to a connecting-rod, which in turn imparts a to-and-fro motion to the trough. In inclined seams a short stroke of 4 to 8 inches is used, but if the coal is wet or the conveyor works in a horizontal position a longer stroke is necessary.

On the forward stroke of the engine the troughs are pushed against the direction of travel of the coal, and so the rollers on the troughs are pushed some distance up the slowly inclined portion of the roller guides. On the return stroke, the recoil of the rollers against the steep front part of the guides, together with the sudden shock of the reversal of the engine, ensures the coal or ore being moved in a forward direction with each stroke.

The air-engine is regulated first of all on the air-inlet, and secondly by adjusting the valve tappets to give the length of stroke required. The cylinder is fitted with bolt-holes to allow of the engine being attached to a wooden frame as shown in Fig. 98.

CHAPTER X

ROCK-DRILLS

One of the most successful applications of compressed air is the operation of drilling shot-holes in rock and stone.

Electrically operated drills have also been tried, but their number is few and their success inconspicuous compared with the number and achievements of the air-drill.

In shaft-sinking, in stoping, in tunnelling, and in stone work generally the use of the compressed-air drill has multiplied to an enormous extent in recent years, and its employment will undoubtedly continue to be extended in the future.

The advantages of power-driven drills over the old method of hand-drilling may be summarized as follows:

(1) The speed of driving in development work is greatly accelerated.

(2) The paying ground per man won from the stopes is increased and so the output from the mine is greater.

(3) The cost per yard of ground won is reduced.

(4) For a given daily tonnage less manual labour is required—an important point where labour is scarce and consequently dear.

(5) A greater proportion of unskilled labour can be made use of, and so the labour charges reduced.

There are few, if any, important objections to the use of power-driven drills, and by skilful attention to improvements in design and in application any disadvantages they may have once possessed have been largely eliminated. In any case, the accruing advantages where they can be conveniently used are so apparent that there is generally little or no question as to the advisability of employing them wherever circumstances permit.

Air-driven rock-drills may be divided into two main classes:

(1) **THE RECIPROCATING DRILL,** in which the drilling steel is fixed in a chuck which forms an extension of a piston working in a cylinder.

The piston and the drilling steel move together, the piston being thrust forward with great force by the pressure-air and causing the drill bit to strike a powerful blow on the rock at the bottom of the drill-hole. On the outstroke the drill steel is pulled backwards a distance equal to the stroke of the piston.

Rotation of the piston, and consequently the steel, is produced automatically, the mechanism for which is described in connexion with the drills included in this chapter.

The feed is by hand. The drill is large and requires some form of support, which renders it unsuitable for some purposes.

It is best adapted for development work, such as sinking shafts, winzes, level drivages, cross-cuts, etc., where tripods or vertical or horizontal standards can be conveniently arranged. It is also probably better than the hammer drill for heavy down-hole drilling, and in large stopes and where deep holes of large diameter are required.

The speed of the piston is generally somewhere in the region of 350 working strokes per minute for the larger sizes of drills.

(2) **THE HAMMER DRILL.**—In this type of drill the drill steel does not reciprocate. On the contrary, it rests loosely in the chuck, and is struck a series of rapid blows by the hammer, which is simply a cylindrical piece of steel forming the piston or hammer of the drill. This hammer is actuated by air-pressure; its stroke is short and speed of oscillation very high, as many as three thousand blows per minute being delivered.

The drill may be made, as in the case of the Leyner type, so as to require mounting on a support, or it may be held simply in the hand of the operator, forward feed being obtained by the maintenance of a steady pressure on the drill.

Another form of the drill is known as the "stoper," and this has an automatic air-feed. It is chiefly used, mounted or unmounted, for up-holes in stopes and rises.

The drill is light and convenient for handling; and because of the relatively negligible inertia of the hammer as compared with the much heavier reciprocating piston and drill, the hammer drill is the more economical as regards consumption of air. The blow struck is, of course, a less powerful one, but the working speed is very much higher, and so the feebler impact is made up for by the greater number of blows per minute.

With the hammer drill, since the drill steel is not in motion in the hole, water or air, or a combination of both, is generally required to keep the bottom of the hole clear of cuttings.

REQUIREMENTS OF A GOOD TYPE OF ROCK-DRILL.—Generally speaking, the successful rock-drill will comply with the following requirements:

(1) Speed of cutting on the average at least one inch per minute under reasonably favourable conditions.

(2) Simple in design and construction.

(3) Durability, *i.e.* ability to stand wear and tear.

(4) Convenient weight for handling and fixing.

(5) Easy to fit up and to repair.

(6) Air consumption reasonably low.

The speed of cutting will depend upon

(1) The nature of the rock.

(2) The effectiveness of "mudding."

(3) The state of the drill as regards repairs.

(4) The skill of the operator.

The skill with which the drill is handled will also affect the extent of the repairs necessary. With regard to air-consumption, this is not so important as the other points, provided the drill is not too wasteful of air. The air is generally used full pressure throughout the stroke, and this, of course, reduces efficiency; but the advantages of powerful stroke, and avoidance of freezing at the exhausts, outweigh the economy that would be effected by using the air expansively.

One or two drills, however, of the valveless type use the air expansively.

METHODS OF MOUNTING THE DRILL.—In the reciprocating drill and also with some hammer drills, the drill has to be mounted on some rigid support.

This can be done in at least three ways:

(1) Mounting on a bar.

The bar is jammed horizontally between the sides or vertically between the roof and floor if underground. The method can therefore be applied in stopes, in drivages, and in sinking shafts. The bar may be sufficiently long, if the place is wide, to accommodate two or more drills at once, and is generally telescopic, so that the length, by screwing out or in, can be adjusted to suit conditions (*see* Fig. 84).

(2) Mounting on a tripod.

This method is suitable where the drill is to be used in places where it might not be convenient to fix a bar, such as quarries, shafts, or wide places. The arrangement works best for downward holes, but the tripod can be arranged so that the drill can be swung from downwards position to a high upward angle.

For downward holes the legs of the tripod have to be heavily weighted to resist the upward thrust on the drill (*see* Fig. 69).

(3) Mounting by having a telescopic adjustment on the drill itself.

This is convenient in stopes and rises or drivages, where by adjusting the telescopic extension the drill can be jammed between the sides or between roof and floor.

AIR-PRESSURES USED.—The pressure at which the air is used varies over a considerable range.

On the Rand a common pressure is 70 to 90 lbs. per square inch, although in one or two cases pressures of 100 to 120 lbs. are employed.

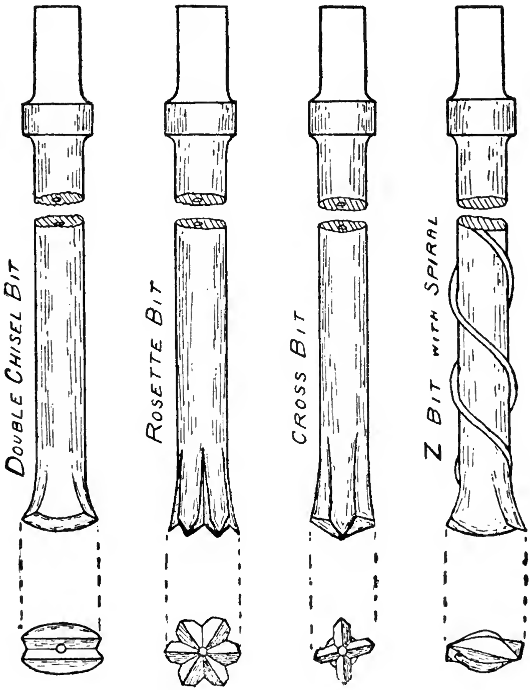


FIG. 99. Forms of Drill Bits.

In this country a somewhat lower pressure—50 to 70 lbs.—is usually employed.

Hammer drills can now be made to work successfully with pressures as low as 40 lbs. per square inch, but the reciprocating type generally requires a somewhat higher pressure.

The advantages of using relatively high pressures are:

(1) A smaller and therefore lighter drill can be employed.

(2) For a convenient size of drill a more powerful stroke can be obtained.

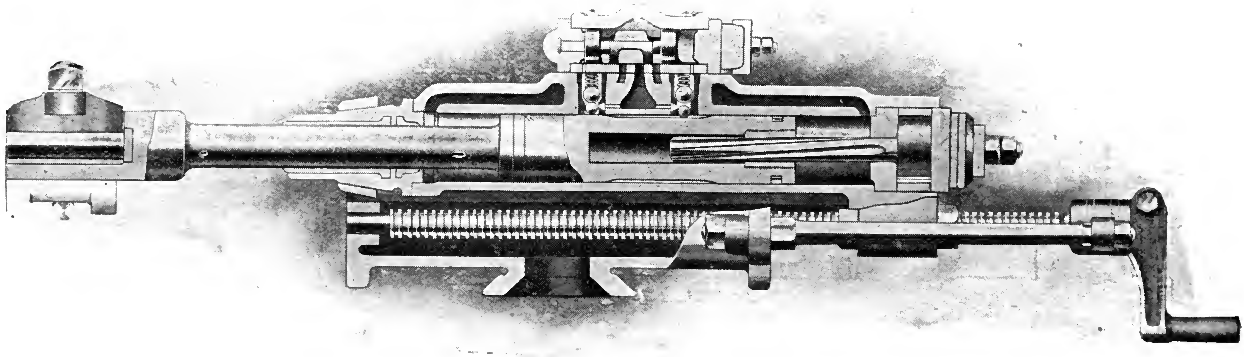


FIG. 100. Holman Reciprocating Drill

(3) More rapid drilling of holes.
(4) Higher efficiency obtained.

The disadvantages are :

(1) The life of the drill is shorter.
(2) Repairs are more frequent.
(3) Tools are blunted sooner and require sharpening oftener.

Too much importance must not be attached to the disadvantages of high pressures, and the general consensus of opinion is that the most economical pressure lies somewhere between 50 and 90 lbs. per square inch.

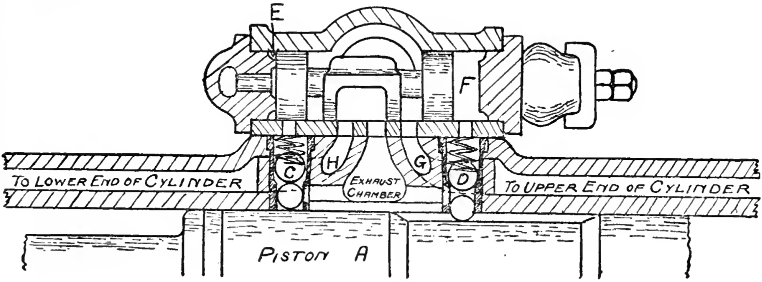


Fig. 101. Valve Action of Holman Drill.

DEPTH OF BORE-HOLES.—A common depth of hole is from 4 to 5 feet.

Occasionally, however, holes outside this range are required, either shallower or deeper as the case may be. The tendency is to increase the depth of the holes, and in some cases, *e.g.* in wide stopes, shaft-sinking and quarrying, holes of 6 to 8 feet are not uncommon.

FORMS OF DRILL BITS.—Success in drilling depends to a very large extent on using the most suitable form of bit, in the best possible condition as regards sharpening and tempering.

Fig. 99 shows various forms of drill steels used in modern power rock-drills.

For hollow steels the double chisel cutter is the best ; for solid steels the cross style of bit or the Z bit is to be recommended. For spiral section steel the Z bit has been found very satisfactory, while for some purposes the rosette form of bit is used.

RECIPROCATING DRILLS

THE HOLMAN DRILL.—This drill is shown in section in Fig. 100. The valve action will be understood from the diagram (Fig. 101).

The piston A is shown on the down stroke. The valve is of the air-thrown type and is shown situated hard over against the left-hand end of its chamber at E. Two auxiliary valves in the form of the steel balls C and D control the movement of the main valve. In the position shown, the ball valve C has been lifted off its seat, against the action of a spring, by the piston lifting the ball underneath it. This has allowed the air in the valve chest at E to exhaust through the port shown between the upper and lower balls. At the same time the ball D is held down on the valve seat by the spring; and the air, entering the end of the valve chest at F *since it cannot get into the exhaust chamber*, forces the valve over against E, thus placing the live air in communication with the upper end of the cylinder through the port G, and the lower end of the cylinder in communication with the exhaust through the port H. At the end of the down stroke the ball D is lifted off its seat, and the ball C is allowed to drop. The valve is therefore forced over to the right-hand side and the reverse action takes place.

The drill steel is held in a *chuck* at the front of the drill. In the Holman drill the chuck consists of a U-bolt with a gripping pad and wedge. Nuts on the two limbs of the U-bolt hold the bushing, wedge, and gripping pad in position. The nuts are chiefly for adjustment and to allow compensation for stretch and wear. The drill steel is inserted and then the wedge pressed home by hand or a slight tap with a hammer. The first few blows of the piston and drill steel against the rock give the wedge the necessary grip and tighten the gripping pad on the drill bit.

The forward *feed* of the drill is accomplished by rotating the handle of the feed screw, which works through a nut in the drill casing under the cylinder.

A spray attachment is used for the allaying of dust when drilling uppers. An open receptacle of water is used from which live air siphons the water, and a coarse spray is produced which is directed towards the mouth of the hole. The spray is found effective, and is one of the best ways of preventing the dread disease of miners' phthisis. Where the attachment is provided the spray comes into operation automatically as soon as air is turned on the drill, and stops when air is turned off.

The rotation device consists of a rifled bar having twisted grooves. Into these grooves fit projections on a bush carried in the hollow part of the piston, as can be seen in Fig. 100. The end of the rifled bar carries a ratchet wheel controlled by pawls, which allow rotation in one direction only. This rotation takes place when the piston is on the forward or striking stroke, the piston shooting straight and the rifled bar and ratchet rotating through a slight angle. On the return stroke the ratchet is prevented from rotating, and so the piston

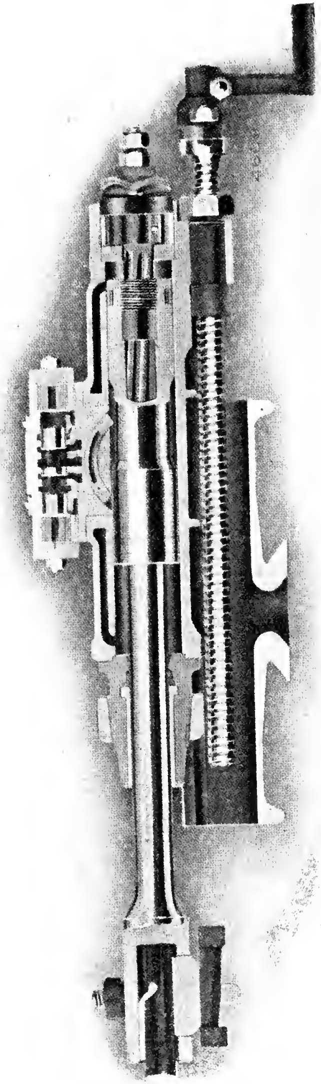


Fig. 102. Ingersoll-Sergeant Drill.

is made to turn in its motion along the rifle bar. Thus at each stroke the piston, and consequently the drill steel, is given a partial rotation.

The Holman drill is made in several sizes, the larger sizes for development work and the smaller for stoping. It has proved itself a reliable and efficient form of drill, and in the South African trials of 1909–10 divided the first prize with another type.

THE INGERSOLL-SERGEANT ROCK-DRILL.—This drill is shown in section in Fig. 102.

The valve action is somewhat similar to that in the Holman drill, except that in this case the movement of the main valve is controlled by a single auxiliary valve.

This valve consists of a light arc-shaped piece of steel working in a groove and having one end or the other projecting slightly into the cylinder bore, as can be seen in the figure.

The raised portion of the piston, going in either direction, pushes against the projecting auxiliary valve and raises it sufficiently to uncover a small port, which releases pressure from one end of the main valve, and allows the live air, which is exerting its pressure on the other end, to throw the valve over, thus opening the main port and admitting full pressure to the piston for the return stroke. The auxiliary valve is thus simply a trigger which releases the main valve, and performs the same function as the steel balls in the Holman drill.

The general make-up of the drill is very similar to that of the Holman, and the description of the rotation device, feed, etc., given with the latter may be taken as applying, with very slight modification, to the Ingersoll drill.

THE CLIMAX DRILL.—This well-known drill is shown in section in Fig. 103.

The tappet valve V is operated by the double-ended piston P P′.

In the position shown the valve admits live air from the supply into the passage F′ and so behind the piston P′. The piston is thus shot forward to deliver the blow. At the same time the valve has put the other passage, F, in communication with the exhaust, and so the piston drives out the air in front of it. When the piston is near the end of its stroke the part P′ strikes the valve, swinging it on its central pin and putting F into communication with live air and F′ with the exhaust. This allows the return stroke, at the end of which P swings the valve back again into the position shown in the figure.

The feed and the rotation device work in a similar way to that described for other drills.

In another form of the Climax drill a spool valve is used instead of the tappet valve. With the Climax drills an atomizer type of dust-layer is provided which takes up water from a bucket or other receptacle at the side of the operator and discharges it in the form of a spray around the orifice of the hole being drilled. The water is atomized, and meeting the dust and cuttings of rock ejected from the hole

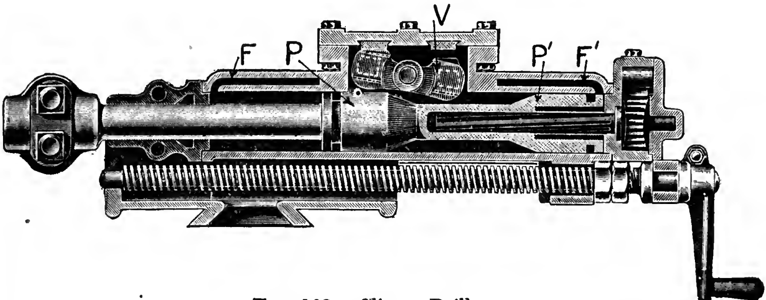


FIG. 103. Climax Drill.

converts them into mud. An air-blast through the hollow drill is used to eject the cuttings.

The *Siskol* reciprocating machine (*see* p. 155) is also largely used as a drill, and its excellence is evidenced by the fact that it shared the first prize in the Transvaal 1909–10 trials.

HAMMER DRILLS

THE FLOTTMANN HAMMER DRILL.—This well-known hammer drill is shown in section in Fig. 104. Its action is as follows:

The piston B moves to and fro inside the cylinder A, striking the drill steel D with its thinner end C at the end of each forward stroke. The number of blows struck is about 2000 per minute.

The thin part C of the piston has a series of straight grooves cut in the front portion of it and twisted grooves in the rear portion. Into these twisted grooves fits the ratchet-wheel E (*see* longitudinal and cross sections), which has female twisted grooves to fit those on the piston. The straight-grooved portion of the piston slides in similar straight grooves in the drill-holder F. The ratchet-wheel can rotate in its case, G, only in one direction, the pawls which are pivoted in the case allowing rotation in one direction but preventing it in the other.

When the piston is on the return stroke, *i.e.* moving towards the handle, the pawls prevent the ratchet-wheel from revolving, and as the twist grooves on the piston are constrained to move in the female twist grooves in the ratchet, the piston is forced to rotate in its chamber. This rotation is transmitted by means of the straight grooves on the front portion of the piston to the drill-holder F, and thence to the drill steel. Thus on every backward stroke a partial turn is given to the drill. On the downward or striking stroke the ratchet-wheel turns freely as the piston shoots forward, and so on the working stroke a straight blow is struck.

The drill is held in the drill-holder by means of the chuck, which in this case consists of the coil spring L, which is screwed on to the front

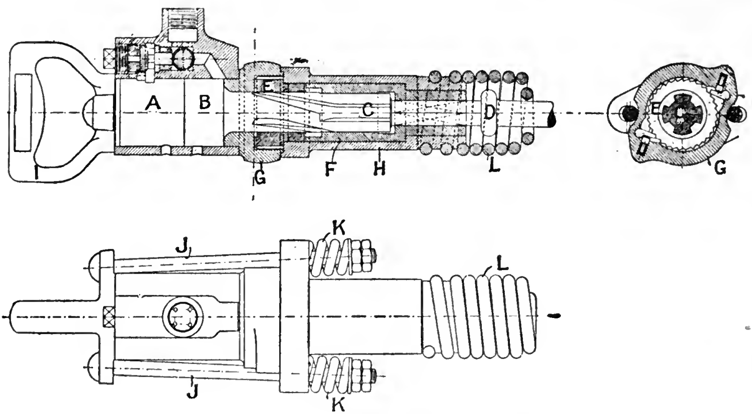


Fig. 104. Flottmann Hammer Drill.

end H. The two tie-rods J keep the parts in position, and the coil springs K prevent breakages if the piston does not strike the drill, as may happen when the drill is withdrawn from the borehole with the machine still running. The valve is very simple and reliable. It consists of a ball of hardened steel, which works in a chamber having communication at both ends through air-passages with the front and rear ends of the portion of the cylinder in which the piston works.

The action of the valve is as follows: Suppose the ball-valve closes the air-passage communicating with the back end of the cylinder, then the pressure-air gets past the valve into the front end, and acting on the fore side of the piston forces it backwards. When the piston has moved a certain distance, it uncovers the lower exhaust ports in the cylinder and the air escapes. This escape of air reduces the pressure on that side of the ball, and so the air-pressure on the other side, due to the compressing action of the piston on the residual air left in front of it, forces the ball over and places the pressure-air supply in communication with the rear end of the cylinder. The full pressure of air from the supply now shoots the piston forward to deliver its blow with great force. As soon, however, as it uncovers the upper exhaust port, the air escapes, while the air in the front end of the cylinder, since the piston will be in a position covering the lower exhaust port, will be compressed; and this pressure forces the valve back over against the passage leading to the rear end of the cylinder once more.

THE MECO HAMMER DRILL.—This drill is shown in section in Fig. 105. There are two valves, V_1 and V_2, at the rear end of the

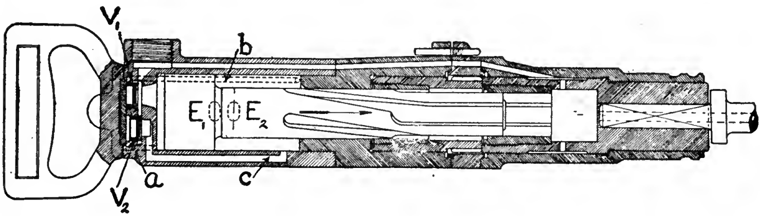


FIG. 105. Meco Hammer Drill.

cylinder. As shown, V_1 is open and the live air entering through it passes into the cylinder and drives the piston forward.

At the same time, the live air enters the valve box of the lower valve by passing from the cylinder through the short air passage (a), (shown dotted on the lower side of the valve), and keeps the valve closed.

Immediately the piston has covered the front exhaust port (E_2), (shown dotted behind the piston), the air in the front end of the cylinder is compressed and forced along the upper and longer air passage (b), (shown dotted), and closes V_1. This takes place before the piston has reached the rear exhaust port.

As the piston travels forward the air behind it expands, since the supply is cut off, and this relieves the pressure from the back of V_2; and as compressed air from the front of the piston is forced along the lower long air passage (c), (shown full), to the front of V_2, this valve is now forced open and admits the live air.

The pressure-air now enters the rear of the cylinder, getting through V_2 and along the lower passage, and drives back the piston. When, on the return stroke, the piston has closed the rear exhaust port E_1, the air is compressed in the rear end of the cylinder, closing the lower valve. As the front exhaust port is uncovered, the air behind V_1 now escapes and the valve itself is forced open by the compression in the rear end of the cylinder. The live air then enters through V_1 and the cycle of operations is recommenced.

The valves are simple and consist of two small pistons of glass-hard steel.

The compression pressure needed to open the valve is small, and so little or no diminution of the force of the blow on the drill steel is experienced.

It is found that by using this form of valve gear, recoil and vibration are considerably reduced, while the economy in air-consumption is very marked.

The rotation of the drill steel is secured in a manner similar to that described for the Flottmann hammer drill.

An air flush through the hollow drill steel is provided in some types of the drill, and can be made continuous or variable as desired. The necessary connexions for the air flush are only partly shown in the illustration.

Another type of the Meco hammer drill which is becoming popular is fitted with a flap valve, which rests on a central pivot and has projections at each side. The valve oscillates on its pivot, first admitting air into one end of the cylinder and then into the other end, the reversals being brought about by the compression of the residual air in the cylinder, in front or behind the piston, towards the end of each stroke. The action is similar to that of the butterfly valve in the Leyner-Ingersoll drill described under the next heading. Otherwise the design of the drill is similar to that of the one shown in the illustration above.

THE LEYNER-INGERSOLL DRILL.—The distinguishing features of this well-known hammer drill are:

(1) The application of water and air under pressure to the bottom of the drill-hole, thus combining the advantages of enabling the steel to strike virgin rock at every blow and of laying the dust.

(2) The automatic rotation.

(3) The butterfly valve.

The drill is shown in section in Fig. 106.

The valve consists of a single piece of steel forming a cylindrical trunnion with two flat wings. The trunnion rests in a bore in the valve chest, and a longitudinal groove in the same accommodates the wings of the valves. The valve is actuated by the mere unbalancing of pressure on each wing alternately. Referring to Fig. 107, the piston or hammer P is on the forward or hitting stroke. Live air at the inlet at the beginning of the stroke has swung the valve over against S_1, closing it and opening the port S_2. The same movement has opened the exhaust port E_1. Thus the piston is thrown forward. When the

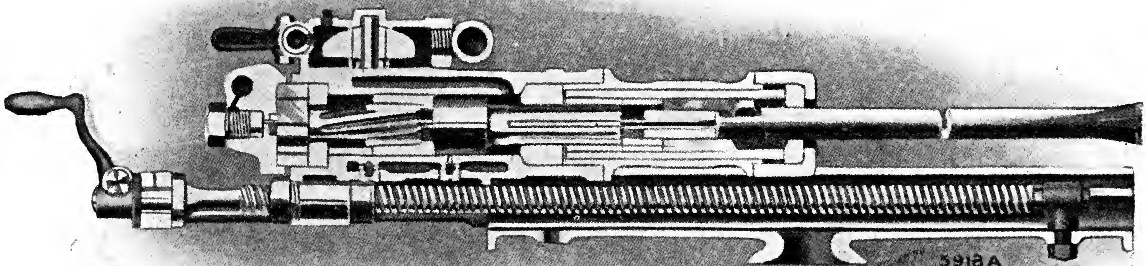


FIG. 106. Leyner-Ingersoll Drill.

port EE_2 is uncovered by the piston, live air passes through it and presses on the lower wing of the valve.

But as air-pressure is still being exerted on the upper wing the valve is not thrown over but merely balanced.

At the same instant, however, that EE_2 is opened EE_1 is closed, and the piston now compresses the air left in the space in front of it. This causes a pressure to be exerted on the upper wing of the valve through S_1, and so the valve is thrown over, uncovering S_1 to live air, and E_2

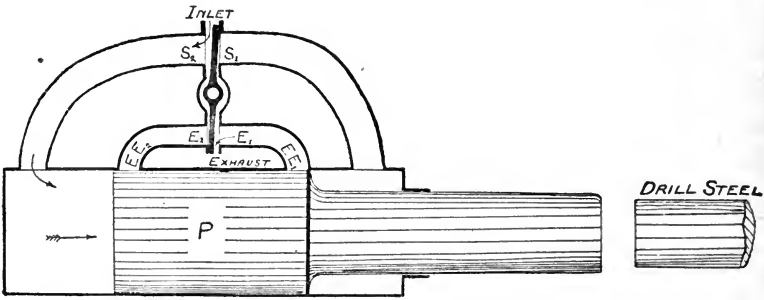


FIG. 107. Valve Action of Leyner Drill.

to the exhaust. The back stroke then commences, the operations being the same as in the forward stroke. At the end of the forward stroke the projecting end of the piston strikes the drill steel a violent blow. The drill steel is shown in position in the figure.

For the cleaning and dust-allaying device water under a pressure of anything from 20 to 150 lbs. is used. The water can be brought to the drill in any convenient way, and if a pressure due to source of supply cannot be obtained it can be kept in a tank for each drill, and a pressure equal to the working pressure of the drill can be got by admitting the compressed air to the tank. The water passes from the tank through a flexible hose to the back-head of the drill, two valves being used, one for regulating the amount of water introduced to the drill and the other for turning on and off the supply. The water passes through the backhead plug and water tube into the shank of the drill steel, where air from the drill mingles with it, and both pass on through the drill steel to the bottom of the drill-hole.

Rotation is transmitted to the drill steel in the following manner:

The chuck, holding the drill steel carries a nut with grooves which fit into the flutings on the front end of the hammer, and thus the latter rotates the chuck and the steel. The hammer, in turn, is rotated by the rifle bar, which fits in a rifle nut in the back end of the hammer. A ratchet and pawl arrangement allows rotation in only one direction.

The drill is a successful one, and the method of allaying the dust has been proved to be very effective.

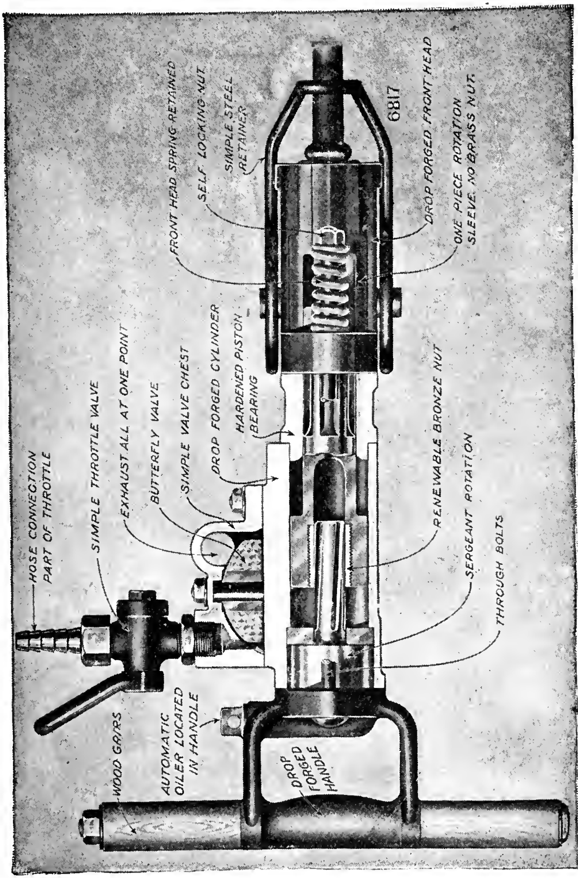


Fig. 108. The "Jackhamer" Drill.

THE "JACKHAMER" DRILL. — The illustration in Fig. 108 shows the arrangement of the different parts of this hammer drill.

The valve is a single piece of steel having two wings and oscillates on a central trunnion by the unbalancing of the pressure on the wings,

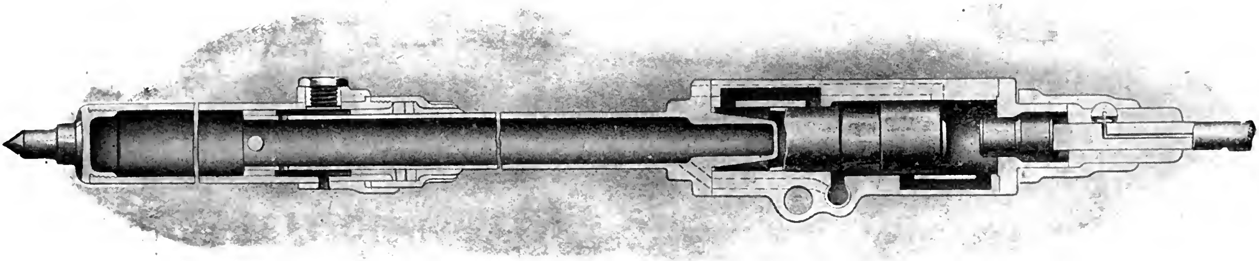
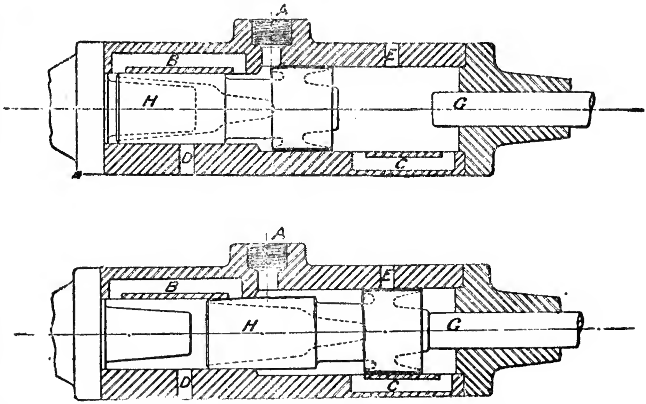


Fig 109. Valveless Stoping Drill

being thus similar to that described in connexion with the Leyner-Ingersoll drill.

Automatic rotation is provided by rifling cut on the piston, running in a rifle nut in the rotation sleeve. The rotation ratchet, which, held by friction, is in the rear of the cylinder, can be held firmly in its place by tightening the through bolts. Rifling on the forward end of the piston fitting into a rifle nut in the chuck rotates the drill steel. If the steel should suddenly stick, however, the whole rotation slips, and the rifle bar is thus prevented from becoming twisted or broken.



FIGS. 110, 111. Action of Valveless Stoper in First and Second Positions.

THE VALVELESS STOPING DRILL.—This type of drill has been recently introduced in metalliferous mines. It is a hammer drill and is provided with a telescopic automatic air-feed. It can be arranged to work with or without a column, but is generally fixed on a bar placed either horizontally or vertically.

The Holman valveless stoper is shown in Figs. 109, 110, and 111.

In Fig. 110 the piston H is shown at the rear end of the cylinder. Air entering at A fills the space between the two half-pistons and also flows through the port B to the back end of the cylinder. As the piston moves forward the air-supply through port B is cut off, and the remainder of the stroke is completed by the expansion of the air in the rear end, assisted by the difference in the two diameters of the piston. When the blow on the drill steel, G, is delivered as indicated in Fig. 111, the air at the rear end of the piston has been exhausted through D, the large end of the piston has uncovered the port C, and so air flows around to the front end of the cylinder, forcing the piston back against the opposing pressure of the larger and smaller areas.

After the piston has travelled a short distance, port C is again closed and the air again works expansively in forcing the piston to the end of its stroke. Towards the end of the backward stroke the piston uncovers the port E and so the air in the front of the cylinder is exhausted.

A section through the complete drill is shown in Fig. 109. At the front the annular space communicating with the small hole running through the drill is the inlet for the water-jet which is used in the drill for cleaning the hole and allaying the dust.

In this type of drill the rotation of the drill and boring steel is not produced automatically as in the other types described, but is done by hand by means of a lever or handle provided on the machine for the purpose. The rotating handle is used in the manner of a ratchet-brace, viz. alternately backwards and forwards during the process of drilling. The forward feeding of the drill steel as the hole deepens is produced automatically by air-pressure in the interior of a hollow cylinder acting on a piston which moves in the cylinder (Fig. 109). Thus a steady pressure is kept on the boring tool.

The advantages of this type of drill are:

(1) Simplicity of design.
(2) Suitability for stoping and drifting.
(3) Efficiency of the water service, thus giving immunity from dust.
(4) Low upkeep charges.

Other makes of hammer drills are the Siskol, the Hardy-Simplex, and the Hydromax drills. The first two are hammer drills of the ordinary type with automatic rotation, while the Hydromax drill is specially designed for use in stopes and is provided with hand rotation, automatic air-feed, and water device similar to the Holman stoping drill described above.

AIR-CONSUMPTION OF DRILLS.—The air-consumption of rock-drills is generally stated in cubic feet of "free air" per minute, *i.e.* air at atmospheric pressure or the pressure at the exhaust.

The quantity used depends upon

(1) The pressure.
(2) The size of drill cylinder.
(3) The character of the rock and the nature of the work.
(4) The condition of the drill
(5) The skill of the operator

The following table gives an average estimate of the quantity of ree air per minute used by various sizes of drills in good condition at various pressures, no allowance being made for loss by leakage or friction in the pipes:

TABLE XII.—AIR-CONSUMPTION AT SEA-LEVEL OF ONE DRILL IN CUBIC FEET OF FREE AIR PER MINUTE.

Air-pressure. lbs. per sq. inch.	Size of Drill Cylinder. Diameter in inches.						
	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$
60 . .	55	60	70	82	92	105	115
70 . .	60	66	80	94	105	120	130
80 . .	68	75	90	106	115	135	145
90 . .	76	84	100	116	130	150	160
100 . .	84	95	110	127	140	162	175

If more than one drill is being worked from a single compressor—as is, of course, almost invariably the case—it does not follow that the total air-consumption will necessarily be equal to the quantity for one drill multiplied by the number of drills.

The greater the number of drills worked by the one plant the more uniform is the load-line, the less is the waste compared with the total power consumed, and the more economical is the working of the whole system.

The economy of the plant will also depend on the percentage of time occupied in drilling by each machine.

With a number of drills worked by one compressor, the air-consumption for all the drills, and consequently the quantity of free air per minute with which the compressor must be capable of dealing, is found by multiplying the air-consumption of one drill by a factor which depends on the number of drills and the height above sea-level.

The table on page 190 gives the multiplying factor for different numbers of drills and various altitudes.

For altitudes and number of drills intermediate between those given, the value of the multiplying factor can be found sufficiently near by interpolation.

TABLE XIII.—MULTIPLYING FACTORS FOR DIFFERENT NUMBERS OF DRILLS AND VARIOUS ALTITUDES.

Number of Drills.	Altitude above Sea-level.					
	0	2000	4000	6000	8000	10,000
1	1	1·05	1·14	1·23	1·30	1·35
2	1·8	1·9	2·02	2·18	2·30	2·40
3	2·7	2·8	3·0	3·20	3·35	3·50
4	3·4	3·6	3·8	4·1	4·3	4·5
6	4·8	5·1	5·4	5·8	6·2	6·5
8	6·2	6·6	6·9	7·2	7·7	8·3
10	7·3	7·7	8·4	9·0	9·5	10·1
15	10·2	10·9	11·7	12·4	13·2	13·8
20	12·5	13·2	13·8	14·9	16·0	16·6
40	22·0	24·0	25·0	26·0	27·0	28·0

CALCULATION OF SIZE OF COMPRESSOR USING THE TABLES

In order to show how the tables may be used for finding the size of compressor for a given number of drills the following example is worked out:

EXAMPLE.—Fifteen $2\frac{1}{4}$-inch stoping drills and ten $3\frac{1}{2}$-inch development drills are to be supplied by a single two-stage compressor. The altitude at the entrance to the mine is 4000 feet above sea-level.

Find the size of the low-pressure cylinder, assuming it to be double-acting and the piston speed 600 feet per minute. Air pressure 70 lbs. per square inch.

From the first table we have that each $2\frac{1}{4}$-inch drill takes 66 cubic feet of free air per minute and each $3\frac{1}{2}$-inch drill 130 cubic feet per minute.

From the second table we have the multiplying factor for fifteen drills at an altitude of 4000 feet is 11·7 and for ten drills 8·4.

$\therefore$ Total air consumption

$$= 66 \times 11{\cdot}7 + 130 \times 8{\cdot}4$$
$$= 772{\cdot}2 + 1092$$
$$= 1864{\cdot}2 \text{ cubic feet per minute}$$

and if we allow 15 per cent. for leakage, and assume a volumetric efficiency of 80 per cent., we find the piston displacement of the low-pressure cylinder per minute to be

$$1864{\cdot}2 \times \frac{100}{85} \times \frac{100}{80} = 2740 \text{ cubic feet}$$

Then area of cross-section of cylinder

$$= \frac{2740 \text{ cubic feet}}{\text{piston speed}}$$

$$= \frac{2740}{600} = 4{\cdot}57 \text{ square feet}$$

$$= 658 \text{ square inches.}$$

$$\therefore \text{ diameter of cylinder } = \sqrt{\frac{658 \times 14}{11}}$$

$$= \sqrt{837} = 28\tfrac{1}{2} \text{ inches}$$

TABLE XIV.—COMPRESSOR EFFICIENCIES AT VARIOUS ALTITUDES. DISCHARGE PRESSURE 80 LBS. PER SQ. INCH GAUGE.

Altitude above Sea-level. Feet.	Barometric Pressure.		Volumetric Efficiency. Per. cent.	Loss of Capacity. Per cent.
	Inches Mercury.	Pounds per sq. inch.		
0	30·00	14·75	100·0	0
1,000	28·88	14·20	96·5	3·5
2,000	27·80	13·67	93·0	7·0
3,000	26·76	13·16	90·0	10·0
4,000	25·76	12·67	86·0	14·0
5,000	24·80	12·20	83·0	17·0
6,000	23·86	11·73	80·0	20·0
7,000	22·98	11·30	78·0	22·0
8,000	22·12	10·87	75·0	25·0
9,000	21·29	10·46	72·5	27·5
10,000	20·49	10·07	70·0	30·0
12,000	18·98	9·34	65·0	35·0
14,000	17·59	8·65	60·0	40·0

Air-compression at Altitudes above Sea-level.—The density of the atmosphere decreases as the height above sea-level increases, and thus at high altitudes the effective capacity of an air-compressor is less than that at sea-level, since at each stroke a less weight of air will be drawn into the cylinder. The matter becomes of importance where mining is carried on in elevated regions as in the Transvaal. Thus in the foregoing numerical example and in the tables on which it is based, it is seen that the altitude above sea-level has an important bearing on the size of compressor required for the given duty.

The preceding additional table gives the volumetric or capacity efficiencies of compressors at various altitudes.

The adoption of stage-compression (*see* Chapters I and II) is particularly advantageous at high altitudes, and tends to raise materially the overall efficiency of the compressor.

Mechanically controlled inlet valves are preferable to automatic valves, since, owing to the low atmospheric pressure, the resistance of the automatic valve to opening becomes relatively of greater and greater consequence as the altitude increases, and so tends still further to reduce the efficiency of the compressor.

CHAPTER XI

OTHER APPLICATIONS OF COMPRESSED AIR

HAULAGE.—The methods of mechanical haulage in use in mines may be divided broadly into two classes, namely:

(1) Those in which the hutches and wagons are attached to a rope either singly or in trains, and are pulled outbye by the rope being wound on a stationary revolving drum, operated by an engine.

(2) Those in which no rope is used, the hutches being coupled up to a locomotive which does the work of hauling.

In both of these systems compressed air is used to a considerable extent as the motive power in the engines or locomotives.

Rope haulage may be further subdivided as follows:

(*a*) *Endless Rope Haulage.*—In this system an endless steel-wire rope is kept in motion and the hutches are attached to the rope, singly or in trains, at intervals, depending upon the output, the attachment being by means of clips.

(*b*) *Direct or Single Rope Haulage.*—Here a single rope coiling on a drum is used. The wagons are attached to the end of the rope and pulled outbye by the engine. On the inbye journey the train of wagons runs downhill under the action of gravity, the speed being controlled by a brake on the drum.

(*c*) *Main and Tail Rope Haulage.*—This method is used on undulating gradients. Two drums are required, on one of which the main rope coils and on the other the tail rope. When the rake or train of loaded hutches is ready to be pulled outbye, the main rope is attached to the front and the tail rope to the rear of it.

The engines are then started with the main-rope drum in gear, the rake being thus pulled outbye—the tail rope, which passes round a wheel at the inbye end of the engine plane, being pulled out with it. On the return journey the tail rope does the work of pulling the empty hutches inbye.

There is little difference between the construction of a compressed-air hauling engine and one using steam, except that perhaps the exhaust ports should be ampler and the exhaust pipe bell-mouthed. A pair of main- and tail-rope compressed-air hauling engines by Messrs. Wood and Sons, Ltd., are shown in Fig. 112.

In mines in which it is not permissible to introduce electricity because of the risk from firedamp or coal-dust, compressed-air or rope transmission is the only suitable means of conveying the power from the surface underground for haulage purposes.

But in many mines it may be possible to use electricity near the bottom of the downcast shaft or in parts of the main intake airways with reasonable security, while for *secondary* haulage nearer the working face it might not be altogether free from risk to employ the electric current.

There is therefore in secondary haulage a field for compressed air, and in such a case it might be more advantageous to install inbye compressors than surface plant.

In some cases portable haulage gears are used for inbye haulages, the air-engines, drums, gearing, etc., being on a bogie carriage.

Freezing at the Exhaust Ports of Air-engines.—Trouble is sometimes experienced with haulage air-engines owing to the fact that the exhaust ports become partially or completely frozen up. This freezing is due to the extremely low temperatures produced by the expanding air as it rushes through the exhaust ports, the particles of water-vapour carried by the air being thus frozen into ice.

The tendency to freeze is much greater when the air is used expansively than when there is no cut-off and the air escapes to the atmosphere at nearly full pressure, the minimum temperature reached in the former case being much lower than that reached when full pressure-air is used throughout the stroke. To reduce this tendency to freezing, the air should be thoroughly drained of moisture at the receivers. This does not remove all the water-vapour, however, and so, if the expansive properties of the air are to be used to any extent, reheating of the air just before it enters the air-engines may be necessary.

To show the extent of this, it may be worth pointing out that for an initial temperature of about 70° F. if the air is used at full pressure at 45 lbs. per square-inch gauge the lowest temperature reached is about − 45° F., while if complete expansion is carried out the minimum temperature reached is − 107° F.

Where reheating is impracticable, as it is in general underground, jets or sprays of water in the exhaust pipes may be found effectual. The water gives up some of its heat to the outcoming air, and so prevents the temperature falling as low as it would otherwise do. Owing to its higher specific heat, the water is able to heat up a weight of air several times greater than its own for an equal change in temperature.

This palliative can only be applied in stationary engines such as

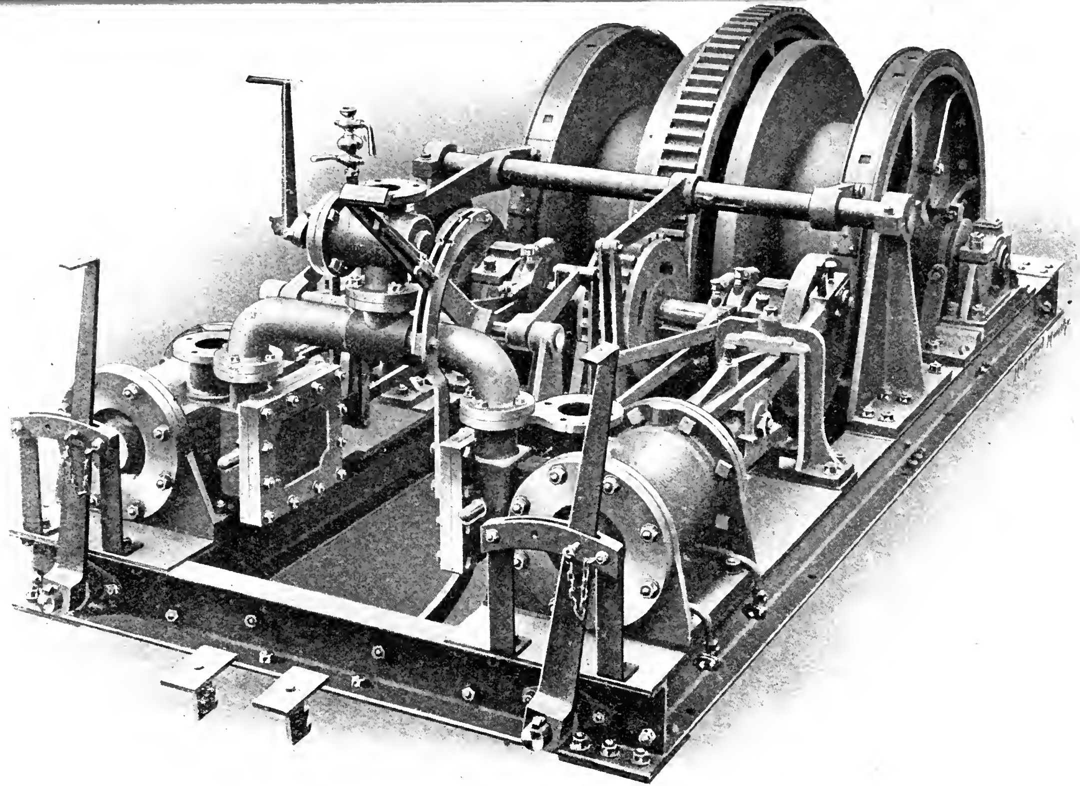


Fig. 112. Main- and Tail-rope Compressed-air Haulage Gear.

hoisting, hauling, or pumping gears. The water underground is warm enough for the purpose, and in addition to giving up heat the scouring action also helps in preventing the formation of ice.

COMPRESSED-AIR LOCOMOTIVES.—These have not been used to any extent, if at all, in British mines up to the present, but in America and on the Continent they are fairly extensively employed.

Very high air-pressures are used for this purpose, 50 to 100 atmospheres being common; while in the more recent installations pressures as high as 200 atmospheres (3000 lbs. per square inch) are employed.

Four- and five-stage compressors are used, and the compressor is kept continuously pumping air into a storage system consisting of the pipe line down the shaft and along the haulage road.

At both ends of the haulage road and at one or more intermediate points depending upon the length of the road, small storage tanks are placed and provision made for coupling up the tank and pipe line to the compressed-air tanks carried by the locomotives.

Where the run is short the pipe line along the road may be dispensed with and a somewhat larger storage tank than usual placed at the outbye end of the road.

The capacity of the storage tanks or pipe line should be at least two and a half to three times the capacity of the locomotive tank, so that the pressure in the pipe line may not be too far reduced when charging is being done. When the locomotive tank requires refilling, it is connected through flexible copper pipes to the compressed-air system. The charging is practically instantaneous.

While the charge of pressure-air is being used by the locomotive, the compressor is refilling the pipes with air up to the working pressure.

From the high-pressure locomotive tank the air passes through a reducing valve which brings down the pressure to anything from 125 to 450 lbs. per square inch, depending upon the type of engines employed. From the reducing valve the air passes into an auxiliary tank, from which the engines draw their supply.

In the older types of engines the air is used at full pressure throughout, the working pressure being usually 120 to 130 lbs. per square inch; but in the most modern types of locomotives double- and even triple-expansion engines are used.

The initial pressure is from 375 to 450 lbs. per square inch, and the air passes through heaters or receivers between the high-, intermediate-, and low-pressure cylinders.

With the modern locomotive a distance of 8000 to 10,000 yards can be run between chargings. The weight is from 5 to 7 tons, the overall length 10 feet 6 inches to 13 feet, the height 55 inches to 65 inches, and the width 36 inches to 40 inches. The track gauge varies from 21 to 36 inches.

The pipe lines are packed at the joints with rolled copper rings, which are found to give air-tight joints even at the high pressures employed.

Locomotive haulage as compared with rope haulage is most suitable in thick and shallow seams, in which it is not difficult to make and keep in repair high and wide roadways. In such mines the hutches or wagons used carry from 3 to 5 tons, and for hauling such the locomotive is admirably suited. The roads have to be high and wide, the railways well and strongly laid, and the gradients low.

HOISTING.—For stationary hoisting engines on the surface compressed air is not generally so suitable or so economical as steam or electricity.

For underground use, however, such as hoisting in blind shafts, and during sinking operations in winzes, or for underhand stoping, the use of compressed-air winches is often found necessary, and for these purposes they are found eminently suitable.

The air winch consists of a pair of coupled engines operating the hoisting drum through spur gearing.

PUMPING.—The pumping plant of a mine is generally of the reciprocating plunger or bucket type or the rotary centrifugal variety.

For the former the compressed-air engine can be used in the same way as the steam-engine or the electric motor, there being no difference in the pump itself. Compound pumping engines with H.P. and L.P. air cylinders are sometimes used, the air being reheated between the cylinders. In the Rix system of reheating, the air, after being exhausted from the H.P. cylinder, is made to pass through a series of corrugated copper tubes placed in the reheating chamber. The water being pumped is caused to pass, in a continuously flowing stream, amongst the copper tubes, and so the air is reheated to the temperature of the water before it enters the L.P. cylinder.

Turbo-pumps can also be driven by compressed air either by an ordinary reciprocating engine working through belting or gearing, or by an air-turbine, which is the more convenient method of driving.

The Pohlé Pump.—In this form of pump compressed air is used to raise water directly without the intervention of an engine or any other actuating mechanism.

It consists simply of a column of pipes, A, open at both ends, one end being immersed in the water to be lifted, and the other forming the delivery (Fig. 113).

The compressed air is introduced into the bottom of the rising main through a small pipe, B, and must be at a pressure slightly greater than that due to the hydrostatic head of water. The compressed air issuing from the small pipe exerts a pressure on the water in the column, and, acting like a piston, forces the water before it. The air divides into bells or bubbles, each displacing a corresponding volume of water.

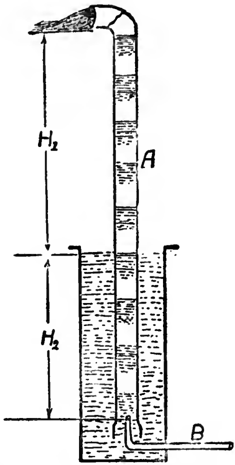


FIG. 113. Pohlé Compressed-air Pump.

The lift H_1 should, if possible, be less than the submergence H_2; in fact, the efficiency is highest when H_2 is one and a half times or even twice as great as H_1. The system will work, however, even when H_1 is greater than H_2, but with reduced efficiency.

The chief merit of the air-lift pump is its simplicity, its application being limited.

In the air-lift pump designed by Mr. George Lansell and installed in a deep mine shaft in Victoria, Australia, the principle of the Pohlé pump is applied to the raising of the water in a series of lifts. In this way the water is raised a total height of 1385 feet.

Another form of pump in which compressed air is used directly for the raising of water is the *displacement pump*. In this pump, the pressure-air is admitted to a chamber containing the water and, depressing the latter, forces it through the discharge valves. The air is now allowed to escape and the chamber again fills with water when the process of displacing it is repeated. The pump chamber must be situated below the level of the water being pumped.

MISCELLANEOUS USES OF COMPRESSED AIR

Compressed air is used to some extent along with water-sprays for laying coal-dust in mines.

At intervals along the haulage roads the air main is tapped and a small branch pipe carried up towards the roof, where it meets with a similar branch pipe communicating with the water main. The two branch pipes run into one which terminates in a nozzle, so constructed as to cause a fine spray to be formed by the issuing air and water. With the sprays sufficiently close a sufficient wetting of the whole of the roof, floor, and sides of the roadway may be obtained.

A non-return valve is placed on both the compressed-air and water branch pipes, so that in the event of any cessation in the supply of air, or water, the rush of air into the water main or vice versa will be prevented.

In another spraying arrangement for laying coal-dust an automatic spraying tub is used.

The tub is filled about two-thirds full of water and compressed air is introduced above it at an initial pressure of 60 to 80 lbs. per square inch, depending upon the pressure available.

When spraying is being done the pressure-air forces water up a pipe extending from near the bottom of the tub to the horizontal spray pipe a few inches above the top cover.

The spray pipe has a series of perforations from which the water under pressure issues in the form of a fine spray, and is so diverted as to strike roof, sides, and floor as the tub is moved along the roadway.

Other uses for compressors include the preparation of liquid air and oxygen for use in one form of rescue apparatus. In other types of breathing apparatus oxygen in steel cylinders is employed at a pressure of 120 atmospheres and special compressing machines are used for the purpose of preparing the supplies of the gas.

A compressor is used in the freezing system of sinking shafts for the purpose of compressing the ammonia gas used in the cooling of the brine solution, which is employed to freeze the watery strata; and in the Caisson method of sinking, compressed air is used to dam back the water.

CHAPTER XII

COMPRESSED AIR COMPARED WITH OTHER MODES OF TRANSMITTING POWER

THE four chief means of transmitting power in mines are, in order of importance:

(1) Electricity.
(2) Compressed air.
(3) Steam.
(4) Wire ropes.

Electricity and compressed air are the only two means which can be used for general power purposes underground, and in consequence attention will be chiefly directed to a comparison of these two.

As regards the initial outlay on plant, steam and rope transmission are the cheapest, but the facts that steam becomes extremely inefficient as the distance it has to be conveyed increases, and that both are unsuitable for many purposes—*e.g.* coal-cutting and drilling—completely outweigh the advantage of low first cost.

Thus, although steam will continue to be largely used on the surface, and ropes may find favour in some instances as a means of transmitting power to underground haulages, yet at all collieries and mines of any size the bulk of the energy to be transmitted underground will of necessity be in the form of electricity or compressed air or both.

The expenditure on plant for electricity or compressed air may be taken as approximately the same, while in general the running costs do not differ appreciably.

It is sometimes put down to the credit of compressed air that much of the repair work can be done by the ordinary semi-skilled mechanic, while electrical plant repairs demand expert technical and mechanical knowledge. This doubtful " good point " should not be allowed to carry much weight. It is notorious that compressed-air machinery in many cases does not receive the skilled attention it should, and so efficiency suffers.

To be able to obtain the maximum efficiency from any form of motive power, it is not only necessary that the equipment should be of first-class design and correctly installed, but that adequate and expert attention be given to the keeping of the machinery in the best possible running condition. By having an engineer with a thorough technical and practical knowledge of compressed-air machinery to take charge of the plant and to superintend repairs, it is reasonable to expect that the all-round efficiency obtained will be appreciably higher than if, as is too often the case, the plant is left almost entirely to take care of itself.

The fallacy that any sort of treatment will do for compressed-air engines so long as they are kept running and can be made to do the work required with a minimum of trouble cannot be too strongly condemned. It is to a large extent responsible for the deplorable inefficiency of compressed air in present-day practice.

In comparing pressure-air with the other methods, attention will be directed chiefly to the following points: (1) cost of production; (2) efficiency; (3) safety; (4) suitability for general mining purposes; (5) convenience.

(1) COST OF PRODUCING COMPRESSED AIR.—In general, with a plant of modern construction and in good condition, the cost of generating compressed air compares very favourably with the cost of generating electrical energy.

Dr. Schultze gives the cost at a modern colliery for compressed air as 0·27*d.* per 35 cubic feet or 0·36*d.* per indicated horse-power hour.*

At the same colliery the cost of generating electricity was 0·448*d.* per kilowatt hour. Thus the ratio

$$\frac{\text{cost of compressed air}}{\text{cost of electricity}}$$
$$= \frac{0{\cdot}36 \times 1000}{0{\cdot}448 \times 746} = 1{\cdot}07$$

The cost of the compressed-air supply at the air-engines, however, was very much higher than that at the compressor. The actual cost at the working point was found to be 0·95*d.* per horse-power hour.

The different factors making up the total cost are approximately as follows:

	Cost per H.P. hour.
Cost of compression	0·36*d.*
Interest and depreciation of pipe lines . . .	0·09*d.*
Extension of pipe lines	0·19*d.*
Materials used in extension	0·07*d.*
Repairs	0·03*d.*
Total	0·74*d.*

* Schultze, *Trans. Inst. Min. Eng.*, vol. xlvii, p. 605.

As about 22 per cent. of the total volume of compressed air was lost in leakage the cost at the point of utilization is brought up to 0·95*d.* per H.P. hour.

The cost for extension is certainly excessive, and is partly explained by the fact that the air was chiefly used for operating power drills and oscillating conveyors, which are both engaged in work requiring frequent extensions to the piping.

The leakage loss is also much larger than it should be.

Taking the working costs for the pipe lines at about half that given, and the leakage loss as not more than 10 per cent., which should not be unattainable, it may be said that the cost of compressed air at the working point should come somewhere between 0·6 and 0·8*d.* per horse-power hour.

Thus, taking the cost for electricity at the point of utilization at about ·75*d.* per B.O.T. unit, the ratio

$$\frac{\text{total cost of compressed air}}{\text{total cost of electricity}}$$

$$= \frac{0{\cdot}6 \text{ to } 0{\cdot}8}{0{\cdot}75} \times \frac{1000}{746}$$

$$= 1{\cdot}08 \text{ to } 1{\cdot}5$$

It must be admitted that the figures are at best rough approximations, but for a properly designed and carefully laid-out plant the cost for compressed air relative to the cost for electricity should, at any rate, come well under the higher figure given above.

(2) **EFFICIENCY.**—The extent of the losses in the generation, transmission, and utilization of the power is a matter of the very first importance and very largely affects the economical working of the system.

Transmission by steam is a very inefficient method unless the distance is small. The losses are due to condensation through loss of heat, and to leakage and friction; and if the distance is at all considerable the loss is great.

It is chiefly for this reason that steam is becoming less and less used underground.

Transmission by ropes is very efficient, and from this point of view will compare favourably even with electricity. The scope of its application is, however, limited to haulage, for which it is eminently suited if the shaft is not too deep. It *is* used for other purposes, such as pumping, but seldom, and even for haulage it is being largely superseded by electricity.

We are left, then, with compressed air and electricity, and since these two are undoubtedly the chief means of conveying power underground it is necessary that we compare them closely with regard to the very important point of efficiency.

The outstanding features of electrical transmission are:

(1) The high value of the efficiency.

(2) The uniformity of the efficiency obtained from different plants.

In general, compressed air cannot be said to have merited anything like the strong position of electricity with regard to efficiency. One might almost say that, so far, experience of compressed air underground would lead one to believe that the keynote of compressed air was inefficiency. The lack of uniformity in the efficiencies obtained at different mines seems to be another characteristic of the system.

At one mine one may find that approximately 30 per cent. of the power put into the engine driving the compressor is got on the motor-shaft underground, while at another mine with apparently similar conditions only 10 per cent., or even less, is obtained. The former value might be called a fair return, and the latter a most unsatisfactory one.

What, may be asked, are the reasons for this striking difference between the two? The probable causes are carelessness, neglect, and a paucity of knowledge of the essential principles underlying the proper application of the system.

One of the inherent disadvantages of compressed air, so far as obtaining a reasonable efficiency is concerned, is the possibility of having a very large proportion of the power wasted through leakage, friction, and other causes, with no discomfort or danger to life or property—a condition of things which in electrical distribution would result in immediate and absolute break-down.

If the waste of the compressed air produced effects that could not be disregarded, there would be greater apparent urgency towards its prevention; and it is quite certain that if only half the care were taken with compressed air to prevent leakage and other causes of loss, that *must* be taken in electrical transmission, the general average of the efficiencies obtained would be very much higher.

Another point which keeps down the efficiency with compressed air is the impracticability, in many cases, of utilizing the expansive properties of the air. One difficulty here is the low temperature resulting from expansion in the cylinder and the consequent trouble with freezing.

In coal-cutters it is difficult to see how expansion can be worked to any great extent, even though, as may be suggested, reheating of the air before it enters the cylinder is adopted. For this work a powerful machine with as small a bulk as possible is required, and, further, it would be a very troublesome matter to adjust a variable cut-off on a coal-cutter engine, since the load on the machine may vary abruptly between wide limits.

For these reasons coal-cutters have to work with a fixed cut-off at a late point in the stroke—$\frac{3}{4}$ or $\frac{7}{8}$ stroke.

With engines driving pumps an earlier cut-off may be adopted, provided freezing at the exhaust ports does not interfere.

In hauling engines, many of which are situated near the pit-bottom, it is more practicable to use a variable cut-off gear, and thus be able to adjust the air-supply to the load.

Reheating of the air would allow the expansive properties of the air to be utilized to a greater extent than otherwise, but the ordinary method of reheating cannot well be adopted underground. Electrical reheating inside the pipes and receivers has been suggested, and it might be possible to utilize some method of steam-jacketing the pipes in the shaft as far as the pit-bottom and close to the main hauling engines.

It is quite certain that if the expansive properties of compressed air were exploited to a larger extent than has been the case in the past and greater attention paid to keeping the compressors, pipes, and engines in good condition, the system would to-day have a better reputation.

In this connexion it might be well to consider, as has been suggested by some engineers, the possibilities of using much higher pressures in mining than we in this country are accustomed to use. If we compressed up to, say, 100 atmospheres, *i.e.* nearly 1500 lbs. per square inch, in a multistage compressor and took the air down the mine at this pressure the resulting advantages would be:

(1) For a given power transmitted very much smaller pipes required.
(2) The possibility of using the expansive properties of the pressure-air to a much greater extent.
(3) A very appreciable increase in efficiency.

There would, of course, be a greater necessity for perfect joints as the tendency to leak would be greater, but by using electrically welded joints and paying proper attention to the air-tightness of stop-valves and cocks, this objection would be easily overcome.

The high-pressure air, after passing through a receiver at the surface, where it would be dried as much as possible, would pass down the shaft pipes to a receiver near the pit bottom, where it would be still further drained of contained moisture. After leaving this receiver it might go straight to the hauling and pumping engines or turbines without reduction in pressure, or if this were not considered practicable, it could be passed through a reducing valve, where the pressure would be reduced to (say) 350 lbs. per square inch.

At this pressure it would be used in the hauling or pumping cylinders, an early cut-off being used in each cylinder and the air being allowed to expand during the remainder of the stroke. Instead of this arrangement compound engines could be used, the high-pressure air entering one cylinder and being cut off at, say, half-stroke; and then, after leaving the high-pressure cylinder, passing to the low-pressure cylinder, there expanding further.

The terminal pressure would not be atmospheric but would be, say, 80 to 100 lbs. per square inch. The L.P. cylinder would exhaust into a receiver at this pressure, from which the air would pass through a pipe leading to other machinery, pumps, coal-cutters, conveyors, or rock-drills further inbye.

The temperature of the air exhausted from the L.P. cylinders would be low, but in the receiver and in the pipes during the passage inbye the air would absorb heat, and in a very short time would have again attained the same temperature as the workings—say anything from 70° to 100° Fahr.

The deeper the mine the greater would be the amount of heat taken up by a given weight of the air for the same fall during expansion.

It will thus be seen that much of the loss due to heating in compression would be largely recovered, without any expenditure whatever, from the natural heat of the workings.

Since the coal-cutters and perhaps inbye pumps might be going when the haulage machinery was standing, it would be necessary to have a pass-bye so as to be able to cut out the haulages, the air being passed through a reducing valve so as to suit the inbye plant.

On the other hand, if the inbye low-pressure machines were idle when the high-pressure plant was working, a relief valve on the receiver would allow the excess air to escape, or it might even be arranged to short-circuit the air back to the compressors at the surface, to be recompressed to the higher value.

With such an arrangement as has been outlined there would be little or no difficulty with freezing of the exhausts.

The gain in economy would be very appreciable, and the engines, using the high-pressure air, would be very much smaller and less cumbersome than engines of the same horse-power using low-pressure air.

By using turbo-compressors for, say, the first two stages, and thereafter compressing in reciprocating machines—efficient intercoolers being interposed between each stage—there would be no difficulty in obtaining efficiently the required pressure.

A matter which has a very important influence on the efficiency is *The Load-factor.*

This is the ratio:

$$\frac{\text{Total load per twenty-four hours}}{\text{Full capacity of plant.}}$$

In a modern electrical colliery equipment this ratio may be anything from 35 to 45 per cent.—it rarely exceeds the latter value.

This means that the generating plant may be intermittently loaded, say, for one period at about three-fourths full load, at another period only one-fourth, the average load throughout a complete day being less than half the continuous full-load capacity.

A low load-factor has an adverse effect upon the efficiency, in that the attendance costs and interest and depreciation are practically the same for a low as for a high load-factor.

In general, therefore, a fairly high load-factor is essential to efficiency.

Compressed-air plant is much more sensitive to wide variations of the load-factor than electricity. The loss from leakage in compressed air does not become less as the load falls off, but remains practically constant at all loads. Thus if the loss from leakage at full load amounted to 20 per cent. of the total volume of air compressed, the leakage loss at half-load would be 40 per cent.; or again, if the leakage loss at three-quarter full load was 20 per cent., then the loss at one-third full load would be 45 per cent.

The efficiency of the compressing plant also decreases as the load-factor decreases, but not by any means to the same extent as the efficiency of the transmission system.

Therefore in a compressed-air system, chiefly on account of this constant leakage loss, which, as has been shown, assumes enormously increased relative importance as the load falls off, it is much more essential than in electrical distribution that the load be uniform and the load-factor have a high value.

It is of more importance that the load be fairly uniform during the period of working than that the load-factor be high.

For example, suppose there are two similar compresssors, A and B, each capable of developing 300 horse-power; A runs at three-fourths of its full capacity for sixteen hours and is completely idle for the remaining eight hours; B runs at full load for eight hours and at one-fourth full load for sixteen hours.

The load-factor for A

$$= \frac{\frac{3}{4} \text{ of } 16}{1 \text{ of } 24} = \frac{12}{24} = 50 \text{ per cent.}$$

The load-factor for B

$$= \frac{1 \text{ of } 8 + \frac{1}{4} \text{ of } 16}{1 \text{ of } 24} = \frac{12}{24} = 50 \text{ per cent.}$$

Consider now the relative efficiency assuming a leakage loss of 20 per cent. of the volume at full load for both plants, and the same mechanical and overall efficiencies for both *when running on full load*

Percentage loss of power through leakage in the first case

$$\begin{aligned} &= 20 \text{ in } 75 \\ &= 26{\cdot}7 \text{ per cent.} \end{aligned}$$

Percentage loss of power through leakage in the second case

$$\begin{aligned} &= 20 \text{ in } 100 \text{ for eight hours} = 20 \text{ per cent.} \\ &+ 20 \text{ in } 25 \text{ for sixteen hours} = 80 \text{ per cent.} \end{aligned}$$

Thus, considering leakage loss only, the efficiency in the first case $= 100 - 26{\cdot}7 = 73{\cdot}3$ per cent., and in the second case

$$= \frac{(100 - 20)\,1 + (100 - 80)\,2}{3} = 40 \text{ per cent.}$$

Thus the relative efficiencies, assuming the percentage of power lost in the other parts of the systems the same in both cases, are as 73·3 to 40, and yet the load-factor was the same in both.

But in addition, the wear and tear, the costs for attendance, the expenditure on lubrication, and other things besides, would be less in the case of the compressor running on three-quarter load for sixteen hours and idle the remainder, than where the compressor was kept running the whole twenty-four hours on a varying load.

Further, the opportunities for repair work would probably only occur in the latter case during week-ends or holidays, and the chances of break-down would consequently be greater.

It is therefore clear that to obtain a reasonable overall efficiency in the use of compressed air it is very important so to arrange the work, wherever possible, that the compressor is kept on a fairly regular duty between the limits of, say, three-quarter and full load.

The conditions which would make for the maximum efficiency are those in which the compressor can be kept on at a high percentage of the full load all the twenty-four hours.

This would perhaps necessitate duplicate plant to be run alternately.

Another point in which compressed air differs disadvantageously from electricity is that in the case of the former, when the plant is standing the pipes are full of pressure-air, and during standage continual leakage is going on. Thus at every shut-down of the plant there is considerable loss of power, and although in most cases this source of loss may not be important when compared with the aggregate loss due to the continuous leakage during working, it still represents an objectionable feature from which electricity is altogether free.

The fact that electricity can be readily and accurately measured, and thus a check kept on the power consumed in each district and by each machine, is a conspicuous advantage. This may not be so easily accomplished with compressed air, but with a proper pressure gauge at every necessary point it should not be impracticable, if systematic observations and records are made, to keep a rigorous check on the leakage and friction losses, as evidenced by the loss of pressure at different points in the transmission line.

Quantity measurements by some form of air-meter (*see* Chapter VI) should also be made at the surface in order to have a constant check on the performance of the compressor, while temperature measurements should be made periodically to see if the cooling arrangements are in proper working condition. A proper system of testing for losses and for efficiency by actual measurement is, in the words of a recent writer on the subject, "the key to efficiency."

Considering the question of efficiency as a whole, it may be said with truth that the low efficiencies obtained in practice with compressed air are not the fault of the system itself, but the direct outcome of incorrect design and installation of plant and carelessness and neglect in working. There is no reason why the overall efficiency should not be within measurable distance of 40 per cent. or even 50 per cent., instead of being under 10 per cent., as is often the case.

The following comparison of losses in compressed air and electricity will give some idea of how the former stands with regard to efficiency:

TABLE XV

	Compressed Air.		Electricity.
	Average of good cases in present practice.	Reasonably possible.	Average of good cases.
	Per cent.	Per cent.	Per cent.
Losses in generation .	25	20	10
Losses in transmission .	15	10	10
Losses in utilization .	28	20	12
Total losses . . .	68	50	32
Overall efficiency . .	32	50	68

(3) **SAFETY.**—It is in regard to safety that compressed air shows itself to best advantage.

In mines to which the Coal Mines Act of 1911 applies, "the use of electricity is prohibited in any part of the mine where on account of the risk of explosion of gas or coal-dust such use would be dangerous to life."

This regulation rules out electricity altogether in fiery or dusty mines, and since for general application neither steam nor rope transmission is practicable, it is in such mines as these—*and the number is likely to increase in the future rather than diminish*—that compressed air is used most largely. Further, "in any part of a mine in which inflammable gas, although not normally present, is likely to occur in quantity sufficient to be indicative of danger," the regulations require that such precautions, in the installation and working of electrical plant, shall be taken as will prevent the possibilities of open sparking.

It is evident, therefore, that even in mines where there is only the occasional possibility of accumulations of explosive gas, *an additional danger* is introduced into the mine by the employment of electricity, which would be prevented if compressed air were adopted as the motive power.

It is for this reason that in many mines compressed air is used for coal-cutters and power-driven conveyors at the face, which, in a seam giving off firedamp, is a part where inflammable gas may occur in quantity sufficient to be indicative of danger; while for haulage, and perhaps pumping, electricity is used, the motor-houses being situated in intake airways, where the possibility of dangerous accumulations of firedamp may be extremely remote.

Many mining engineers are, however, strenuously opposed to the employment of electricity for any purpose whatever in a mine or seam giving off firedamp, arguing that no amount of saving through greater efficiency can compensate for the possibility of a disastrous explosion. In addition to the danger of causing an explosion there is also the danger to life and property from shock and fire resulting from accident or defects in the system.

Fires have occurred in mines which have been directly traceable to electricity, and the accidents—often fatal—that occur from time to time give clear proof, if that were needed, that there *are* dangers associated with electricity, apart altogether from the danger of igniting firedamp. In a modern installation, however, carefully installed and protected, these dangers are reduced to a minimum, and in face of the acknowledged superiority of electricity in regard to efficiency and convenience, the argument against electricity is only a strong one when there is the danger from firedamp or coal-dust

As regards comparison with steam, compressed air is safer, and for long distances more efficient.

(4) **SUITABILITY FOR MINING OPERATIONS.**—Steam is only suitable for winding, haulage, and pumping, and its use underground, as already stated, is rapidly being abandoned. The field of rope transmission is also limited, being confined almost exclusively to haulage. Electricity and compressed air, however, can be used for practically every operation of mining.

Electricity has the additional advantage of providing lighting as well as power, but, on the other hand, compressed air is undoubtedly more suitable for rock-drills and coal-cutters of the reciprocating type. The successful *purely* electric rock-drill has still to be invented.

The electric motor has one considerable advantage over the air-engine, in that it can adjust itself automatically to the load—a desideratum difficult, if not impossible, to secure with an air-engine.

The result is that the electric motor is more efficient on light loads than the air-engine, which may take nearly as much air when running lightly loaded or even unloaded as when fully loaded.

The motor has also a greater overload capacity than the air-engine; indeed the latter, unless working with a variable cut-off, has no overload capacity at all. The electric motor, however, can instantaneously and automatically take on a load very much above its full load and work quite satisfactorily provided the overload lasts for only a short time.

(5) **CONVENIENCE.**—Rope transmission for haulage purposes can be installed very readily, and the space taken up in the shafts need not be much. The number of turns to be negotiated adds to the trouble, since guide wheels must be put in at every bend.

Steam is not convenient underground, for unless a plentiful supply of water is near at hand, there is trouble with the exhaust steam. Also there is the heating of the atmosphere due to the steam, and the trouble from blowing joints, etc.

Electrical transmission is very convenient. The cables can easily be put in and fixed both in the shafts and the roads, and they are also easily extended. The motors take up little room, and in many cases, such as in pumping, the motor is on the same bed-plate as the pump.

Compressed air, however, is probably equally as convenient as electricity, for although the fixing of the pipe lines and the extending of them involve much time and labour, there is the additional necessity in electricity for housing switch-boards and distribution boxes, which require more careful attention than air receivers. Also any leakage of compressed air, although it must be deprecated as a fruitful source of inefficiency, is not fraught with danger, as is the case with a leakage of electricity.

The cooling action of the exhaust of compressed-air machines has also a beneficial effect, especially in hot mines, and although one never relies on the exhaust air for ventilation purposes, there is no doubt that it often increases the comfort of the worker considerably, especially in narrow work.

Professor Galloway * makes a calculation to show that 1000 cubic feet of free air per minute compressed to 7 atmospheres (102·9 lbs. absolute per square inch) will, if allowed to expand to a pressure of 18·1 lbs. per square inch absolute (corresponding to a depth of 6000 feet below sea-level), be capable of cooling a continuous flow of 8000 cubic feet of ventilating air per minute through approximately 20° F.

Thus the exhausts from underground air-motors using a total of 10,000 cubic feet of free air per minute would be capable of cooling a ventilating current of 80,000 cubic feet per minute from 120° F. to 100° F., or from 100° F. to about 80° F. The cooling effect is therefore considerable, and in deep, hot mines forms an additional incentive to the adoption of compressed air as a motive power.

In the case of metalliferous mines, where the employment of compressed air for rock-drills is a virtual necessity, it may be often better to install compressors of sufficient capacity to meet the additional requirements for pumping, hauling, and hoisting, than to provide a separate plant for generating electricity.

CONCLUSION

Summarizing the various points dealt with, one may say that as a motive power suitable for the various operations of coal and metalliferous mining, compressed air, even at the present time, is one of the most important agencies at our disposal as regards suitability, safety, and convenience.

With regard to the stigma of inefficiency, it may be, and probably will be, that in the near future the efforts of engineers to improve the status of compressed air will meet with such success that it will take the position it deserves—if not quite equal to electricity, at least a very good second. In any case, there is a place for both, and they need not necessarily be considered as uncompromising rivals.

* *Lectures on Mining : Ventilation*, p. 13.

APPENDIX A

QUESTIONS SELECTED FROM MINING EXAMINATION PAPERS

B.E. indicates Board of Education Examinations.

C.M. indicates Home Office, Colliery Managers' Examinations; the remaining questions are taken from various College and University Examinations.

1. How is compressed air prepared for use underground? (B.E.)

2. Describe some of the principal methods of transferring power from the surface to machinery underground. (B.E.)

3. A compressor has compounded air cylinders 29 inches and 18 inches bore (inside diameter), by 42-inch stroke, and runs at 110 r.p.m. What volume of air will be delivered to the receiver if the gauge pressure is 80 lbs. per sq. inch, assuming the volume to vary inversely as the pressure? What factor is ignored in this assumption? (Assume the atmospheric pressure to be 15 lbs. per sq. inch.) (C.M., 1st Class.)

4. What are the principal kinds of air-compressors used for mining purposes? (B.E.)

5. Give an account of an installation for producing and distributing compressed air underground. (B.E.)

6. State what you know about driving rock-drills by compressed air, and describe any rock-drill of that class with which you are acquainted. (C.M., 1st Class.)

7. What are the advantages and disadvantages of employing compressed air on a large scale, as a means of obtaining power in mines? (C.M., 1st Class.)

8. What is the approximate difference in the percentage of duty given out at the motor or engine between compressed air and electricity, supposing each to be modern and well-designed plants? Give details of the different items of loss. (C.M., 1st Class).

9. Under what conditions would you use compressed air? What are the advantages and disadvantages of its use? (C.M.)

10. Describe a percussive rock-drill actuated by compressed air. Give details of the following parts: (*a*) The means whereby the com-

pressed air is admitted alternately to the opposite ends of the cylinder; (*b*) the appliance which causes the tool to rotate; (*c*) the mechanism which advances the boring engine and tool as the hole is deepened. (B.E., Honours.)

11. Show in what manner a conveyor system may be employed for filling at a longwall working face. (B.E.)

12. How can compressed air be most economically applied to ventilation? (B.E.)

13. Describe, with sketches, two kinds of mechanical coal-cutting machines. (C.M., 1st Class.)

14. The exhaust ports and passages of plant driven by compressed air sometimes get blocked with ice. How do you explain the presence and formation of this ice, and what steps can be taken to avoid getting the ports and passages choked? (C.M.)

15. Compare inbye compression with compression on the surface, where the compressed air is to be used solely for rock-drills.

16. Compare electricity and compressed air as regards (*a*) efficiency; (*b*) safety; (*c*) suitability for operating coal-cutters and rock-drills.

17. Enumerate the various uses to which compressed air can be put underground.

18. What is meant by (*a*) isothermal and (*b*) adiabatic compression and expansion? What advantages accrue from compressing in more than one stage?

19. Explain the action of turbo-compressors, and compare them with reciprocating or piston compressors for colliery work.

20. What are the laws governing the flow of air in pipes? Choosing what you consider to be a reasonable velocity, find the size of main pipe to convey 3000 cubic feet of free air per minute at a pressure of 75 lbs. per sq. inch gauge.

21. What tests would you apply to a compressed-air system in order to keep a check on the loss from leakage?

22. Describe carefully and give sketches of one type of coal-cutting machine.

23. Show, that if air be compressed in an air-compressor, the relation between the temperatures and pressures is given by the equation

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}}$$

Also find the temperature at the end of compression when air is compressed from 15 lbs. per sq. inch absolute and 70° F. to 105 lbs. per sq. inch absolute. Assume $n = 1{\cdot}35$. (B.Sc., Final, Lond.)

24. Describe fully any means of throttling the air-supply as a means of governing an air-compressor.

25. Discuss the relative advantages and disadvantages of electricity and compressed air for driving coal-cutting machines. (C.M., 1st Class.)

26. In a fiery seam suitable for mechanical coal-cutting, what type of machine and power for driving would you prefer, and why? (Mining Diploma, Heriot-Watt College.)

27. How would you test a compressed-air main for leakage? (Mining Diploma, Heriot-Watt College.)

28. How may compressed air be used for spraying water on dusty roads underground?

29. What advantages are to be obtained by pre-heating compressed air before it is used in the air-engines? Discuss the practicability of pre-heating underground.

30. What methods are used for allaying the dust produced in rock-drilling?

31. Describe the principle and action of one form of rate-of-flow meter for measuring the quantity of air flowing in a compressed-air main.

32. What is meant by the "efficiency of compression" or "air efficiency" of an air-compressor? An air-compressor draws in air from the atmosphere (pressure 15 lbs. per sq. inch absolute), and compresses it to 75 lbs. per sq. inch gauge (90 lbs. per sq. inch absolute). The diameter of cylinder is 24 inches, and the length of stroke 3 feet. If the mean effective pressure as got from the air-indicator card is 39 lbs. per sq. inch, calculate the "efficiency of compression."

33. Four 3-inch power drills, each requiring 150 cubic feet of free air per minute, are to be fed from a main 1000 yards long. The pressure of the air at the outbye end of the main is 82 lbs. per sq. inch. It is desired not to have a loss of pressure of more than 2 lbs. per sq. inch through friction. What diameter must the pipe be? (Mining Diploma, Heriot-Watt College.) Tables IV and V, p. 104, were provided.

34. Describe briefly the construction of a compressor suitable for installing in a mine underground. How would you mount the compressor if it is desirable to shift it further inbye from time to time?

35. Compare compressors placed inbye, *i.e.* underground, with plant situated on the surface as regards (*a*) overall efficiency of transmission; (*b*) convenience; (*c*) safety.

36. How may air be compressed to a pressure suitable for power-purposes by the direct action of falling water? What are the advantages of this system and what are the disadvantages?

37. Obtain an expression for the work expended in compressing 1 cubic foot of air at atmospheric pressure (p_1) to a pressure (p_2) and delivering it at this pressure. The compression curve follows the law pv^n = const. Hence calculate the horse-power of the engine required to drive an air-compressor which compresses 1000 cubic feet of free air per minute from 15 lbs. per sq. inch (absolute) to 75 lbs. per sq. inch (absolute). $n = 1{\cdot}3$ and mechanical efficiency of engine and compressor = 80 per cent. Neglect losses due to clearance, etc.

38. Compare the suitability of reciprocating rock-drills and hammer-drills for use in (*a*) shaft-sinking; (*b*) drives; (*c*) stopes.

39. Describe clearly, with sketches, the action of one type of hammer-drill.

40. Describe the Blackett coal-conveyor and show how it is installed on a longwall face.

41. What are the mechanical and thermal advantages of two-stage and three-stage compression as compared with single-stage compression. (B.Sc., Final, Lond.)

42. In a deep and fiery coal-mine compressed air is to be used on a large scale for operating haulages, pumps, coal-cutters, conveyors, and rock-drills. Describe, with the aid of a diagram, how you would lay out the transmission and distribution system from the surface into the workings.

43. Describe the action of a turbo-compressor and mention any plant of the kind which you know to be in operation.

44. Explain a method of determining the index to the compression curve obtained on an air-indicator card.

45. Describe, with sketches, some form of reciprocating power drill such as the Ingersoll-Sergeant or Holman.

46. A compressing plant is to be installed at a colliery to supply air to coal-cutters and rock-drills. The plant is to be capable of dealing with four disc machines and six pick machines. What air-pressure would you adopt and what size of pipes would be necessary? Also estimate the quantity of air the compressor must be able to deal with per minute.

47. Compare compressed air and electricity as the motive power for (*a*) main haulages; (*b*) secondary or inbye haulages; (*c*) underground locomotives.

48. Sketch an indicator diagram such as might be got from the cylinder of an air-compressor and point out the important points on the cycle. How may the state of the valves and piston be deduced from the air-card?

49. Compressed air at 60 lbs. per sq. inch gauge is used in an air-engine. The air is exhausted at 18 lbs. per sq. inch (absolute). If the temperature of the air on entering the cylinder is 70° F., find the temperature at the exhaust. Take $n = 1{\cdot}35$.

50. Define isothermal and adiabatic compression. Show graphically or otherwise the saving in the work done in the compressor cylinder by cooling during compression.

ANSWERS TO NUMERICAL QUESTIONS

3. Assuming a double-acting compressor, the quantity delivered to the receiver at 80 lbs. per sq. inch is 560 cubic feet per min.

20. Choosing a velocity of 25 feet per second, a pipe $7\frac{3}{4}$ inches diameter will be required.

23. 879° absolute or 419° F.

32. 75 per cent.

33. $4\frac{3}{4}$ inches diameter.

37. 160 horse-power.

49. 366° absolute or -94° F.

APPENDIX B

COMPRESSORS

Additional Notes to Chapters III. and IV.

Tests and Efficiency.—The overall efficiency of a compressing plant depends chiefly on: (1) The mechanical efficiency of the engine or motor, (2) the effectiveness of the cooling, (3) the extent of the friction and leakage losses in the compressor, (4) the variation of loading.

The efficiency of the driver and the effectiveness of the cooling have already been adequately considered. The friction losses may be minimised by adopting a good design and by careful attention to lubrication. A light and well-refined hydrocarbon oil is the best for use in the air cylinders, and the quantity used should not be excessive. The cylinders should be examined periodically to see if the quality and quantity used are right. It is difficult to specify the precise amount of oil to use for any type of compressor, but a rough guide for high-speed machines is an allowance of $\frac{1}{10}$th to $\frac{1}{8}$th of a pint of the best oil for every 1,000 H.P.-hours done by the compressor.

The state of the valves and piston may be ascertained in various ways, chief amongst which are: (1) Periodic inspection, (2) intelligent scrutiny of air indicator cards, (3) by measurement of the quantity of the air delivered by the compressor.

The air card forms a convenient means of detecting defects of the compressor. Sample cards showing the effect of leaky valves are given in Figs. 114 and 115. In both figures the full line diagram is that given with good valves and the dotted line diagram that given with leaky valves. By measurement of the quantity of air delivered by the compressor to the receiver an effective check on the performance is obtained. The use of a reliable form of air-meter, of which there are now several satisfactory forms, is to be preferred to the time-honoured method of noting the number of strokes of the compressor to raise the pressure in the receiver from one value to another. The air-meter has the advantages over the "pumping-up" method of giving continuous readings and of not interfering with the delivery of air from the receiver during the period of testing. Integrating meters may be obtained which automatically integrate the flow values on a chart. The B.T.H. air-flow meter is calibrated in cubic feet of free air per minute, at a temperature of 70° F., for a given condition of pipe diameter, pressure, and temperature. The average pressure and temperature at the point in the air-main where it is proposed to fix the meter should first be ascertained before it is ordered. The air-meter is calibrated to operate under steady flow conditions such as occur in the main delivery pipe of a turbo-compressor.

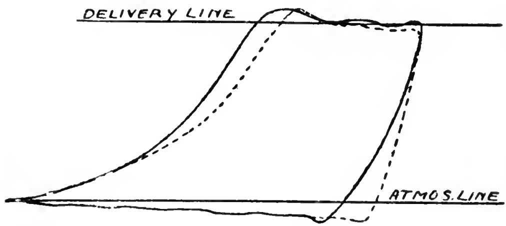


FIG. 114. Diagram showing leaky Inlet Valves.

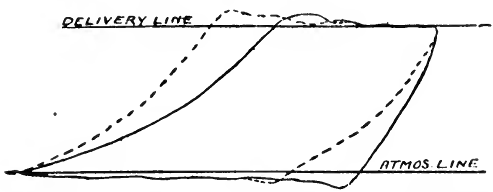


FIG. 115. Diagram showing leaky Delivery Valves.

It will also give accurate results with reciprocating compressors if a reasonably large receiver is used, so that the flow from the receiver is practically unaffected by the intermittency of the working of the compressor. Needless to say, the meter should be installed in the pipe-line on the outflow side of the receiver.

The efficiency of any plant is closely bound up with its load factor. If the demand for air is subject to violent fluctuations —a condition practically always present at mines—the efficiency is sure to be affected thereby. Judged solely from the view-point of efficiency on a greatly varying load, the electric motor takes premier place as a driver of compressing plant. The modern electric motor gives a relatively high efficiency over a considerable range of loading, and is superior to the steam, gas, or oil engine in this respect. Curves are drawn in Fig. 116 showing the relative efficiencies of a steam engine and an electric motor on a varying load. It will be seen that the motor gives much higher efficiencies on low loads. There is, of course, an advantage resting with the steam engines with regard to speed control, referred to elsewhere. The cost of producing compressed air varies greatly with the load-factor. The curve in Fig. 117 shows the relative cost of pro-

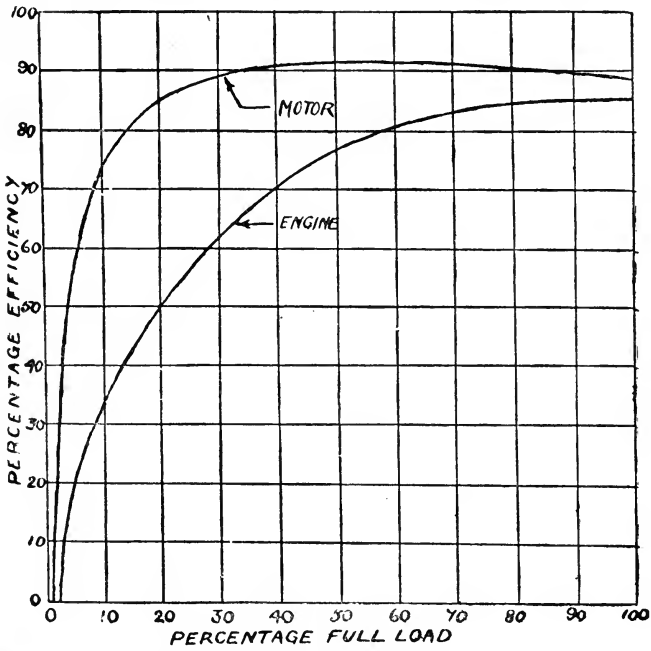


Fig. 116. Relative Efficiencies of Steam Engine and Electric Motor.

duction (which includes fuel, labour, interest, depreciation, etc.) for load-factors from 10 to 50 per cent. If the load-factor is very low the cost becomes excessive. The compressing plant must be taken as a whole, however, and the only satisfactory criterion of its performance is the overall efficiency. This efficiency may be expressed as the ratio quantity of free air in cubic feet per minute compressed to the stated pressure for every indicated horse-power of the steam engine or brake-horse-power of the electric motor.

This efficiency includes all the other partial efficiencies, and the purchaser or user need trouble little about the volumetric, thermal, or mechanical efficiencies as long as the overall efficiency is satisfactory. Of course, in the case of a plant on which a test is being run, it may be necessary to separate the various efficiencies in order to ascertain in what particular the compressor is deficient.

The simplest and most effective method of testing the overall efficiency is to have on the output side of the plant the air-meter

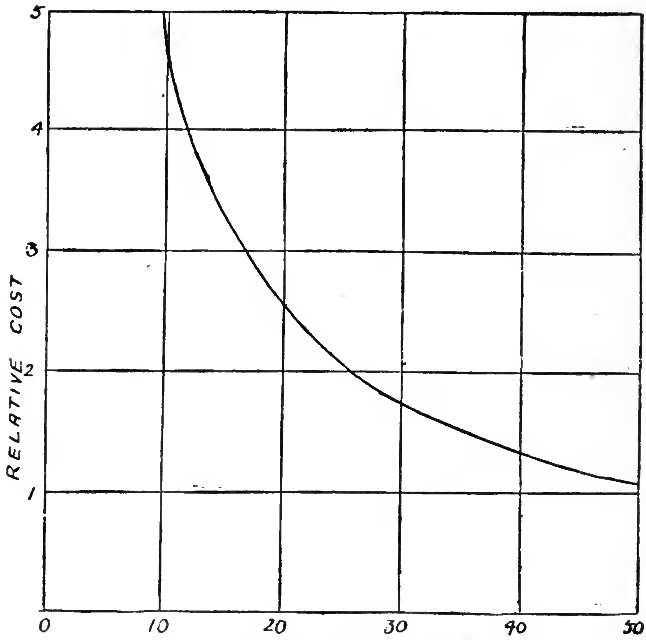


Fig. 117. Effect of Load Factor on Cost of Producing Compressed Air.

and on the input side a steam-flow meter in the case of the steam engine or a wattmeter in the case of the electric motor. In this way a rigid and practically continuous check may be kept on the overall efficiency. An effective check system such as here advocated will always pay in the production of air-power and in conjunction with similar checks on leakages from the pipe-main between the compressor and where the air is used, and a wellorganised and sustained endeavour to reduce those leakages to a minimum would go far to raise the status of compressed air as an efficient means of transmitting power in mines.

A form of compressor recently introduced by the Sullivan Company, and termed by them the "**angle-compound**" **compressor,** is sufficiently novel to be worthy of notice. This compressor is shown in section in Fig. 118. It has the low-pressure

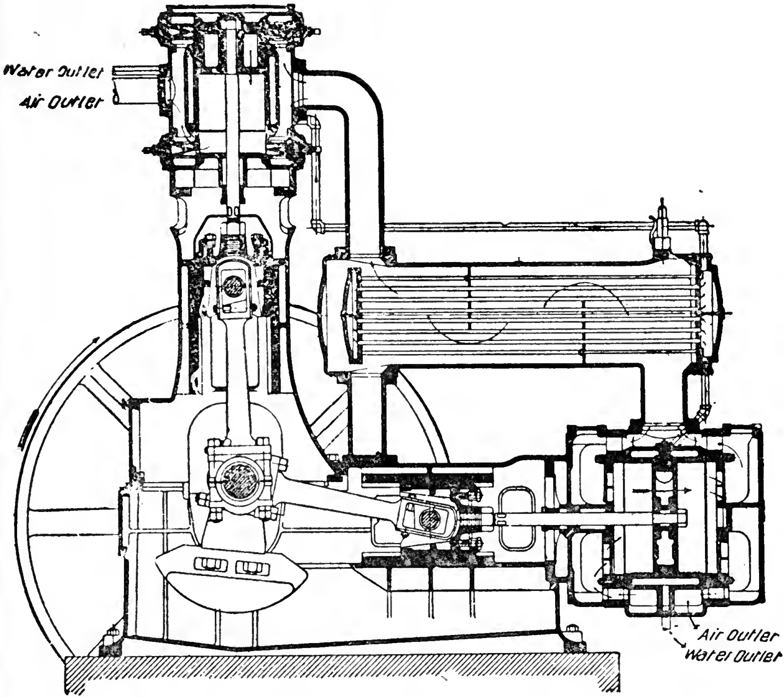


Fig. 118. The Sullivan Angle-Compounded Compressor.

air cylinder placed horizontally and the high-pressure air cylinder vertically. The two connecting-rod ends are placed side by side on the same crank-pin. The pistons as well as the cylinders are thus 90° apart—that is, the high-pressure piston is at one end of its stroke when the low-pressure piston is midway in its cylinder. This design has been adopted to secure accurate balancing of the reciprocating masses. Since the inertia of the moving parts is proportional to the square of the speed of rotation, proper balancing becomes of greater importance in the quick-revolution compressor than in the slow-speed machine. In the angle-compound compressor the disturbing influences of the horizontal and vertical members tend to neutralise each other, the maximum unbalanced effects of the horizontally-moving parts occurring when those of the vertically-moving parts are at their lowest value, and *vice versâ*. Balancing weights attached to the crank still further smooth down the variation of inertia throughout a revolution. The design has proved to be very successful, and has given good results at speeds of from 200 to 250 r.p.m.

Fig. 119. The Sullivan Valves.

The valves on the Sullivan "angle-compound" compressor are of the plate type, but the design is radically different from those described above. It consists of a group of thin, flat fingers, or blades, made from special rolled spring steel (Fig. 119). The valves are rigidly bolted at one end only to a steel guard, and are perfectly free at their other end. The fingers, being very flexible, are bent or lifted by the incoming or outgoing air; and they are bent or rolled over curved stops on the guard plate. Thus, in opening, the fingers exercise a rolling or rocking action against the guard, the lifting commencing first at their outer ends. In closing, the fingers roll back on the grid which forms the seat of the valve. The advantages claimed for this "end rolling" action are that it is free from any bodily lift that would produce a hammering or slapping effect, and that it tends to secure freedom from breaking and leakage. The valves are set in plates or end walls situated between the cylinder barrel and the heads, the inlet valves being in the lower half and the discharge valves in the upper half. In the high-pressure cylinders the valves are placed in cages and arranged radially round the periphery of the cylinder heads.

Output Control.—The regulation of the output of the compressor to the variation in the demand for air is a matter of great importance in mining practice. At one time the number of drills in operation simultaneously may approach the maximum possible, while at another few or none at all may be working. Thus the demand for air may vary from nothing up to the full capacity of the compressing plant. Not only so, but the fluctuations may be very abrupt, so that only a few minutes or even seconds may suffice to produce a change from a small demand to a relatively large one. The pressure in the mains must not be allowed to fall appreciably; indeed, it should be maintained practically constant whatever the quantity of air utilised, so that the compressors must be capable of supplying an increased demand with inappreciable delay. On the other hand, as soon as the demand for air begins to fall off, the output of the compressor must simultaneously decrease, in order to prevent waste of power.

In the case of steam-driven compressors, the method of speed control generally adopted consists of a centrifugal speed-governor actuated from the crank-shaft and operating a throttle valve in the steam inlet. This governor controls the output for all pressures below and up to the full working pressure, the speed of the compressor on that range being a maximum, provided the steam pressure is maintained. If the pressure in the receiver rises above normal, the speed is now controlled by an air governor, which automatically adjusts the speed of the compressor to the demand for air. Such a method of speed control operates between a maximum speed and the permissible minimum speed. For a variation of about 5 per cent. above or below the normal working pressure, the speed may be varied from, say, 250 r.p.m. as a maximum to about 50 r.p.m. as a minimum. The method is extremely economical and can hardly be improved upon.

One disadvantage of electric motors is that in general they cannot be quite so easily or economically regulated for speed as the self-contained steam engine. The direct-current motor may, of course, be automatically varied in speed by having an air relay operated from the receiver, the relay actuating a rheostat which varies the current in the shunt winding of the motor. With the alternating current motor, however, economical variation of speed cannot well be obtained, except in special cases.

In the "Empire" control system, the arrangement of which is shown diagrammatically in Fig. 120, the output is adjusted to the demand in the following manner. The equipment consists of a pressure and a speed regulator and an unloading device. The pressure regulator consists of the Bourdon gauge (A) having

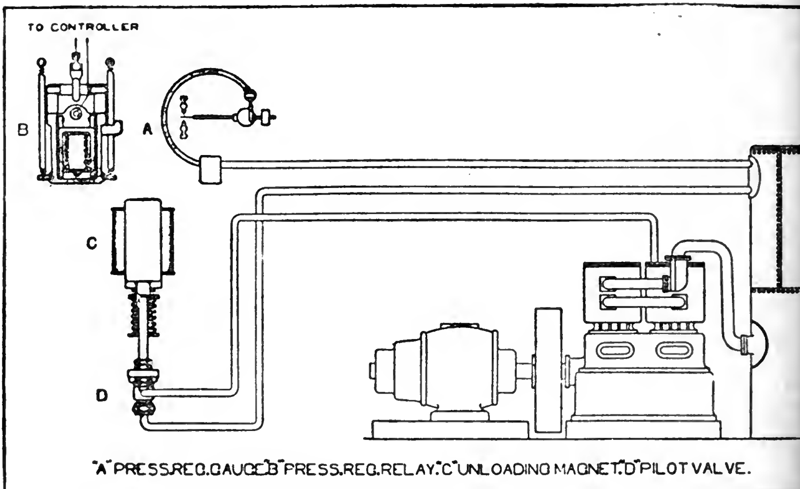


FIG. 120. The "Empire" Control System.

a silver-tipped lever operating between two silver-tipped contacts. The gauge is connected direct to the receiver by piping as shown. When the pressure in the receiver is at the working value, the lever takes up a position between the contacts but touching neither. Suppose the compressor to be standing with the receiver full of air at the working pressure. When the pressure falls a small amount below the working value, the lever swings over to one contact and closes the circuit of the relay (B), which in turn closes the main relay, putting the motor on to the electric supply and so starting the compressor. On the other hand, should the pressure rise above the working value, the lever of the gauge swings over to the other contact, and the relay switch is again energised, but caused to open this time, thus stopping the motor. No electrical circuit is broken at the contacts on the pressure gauge, and thus they cannot be burned or destroyed. The relay is fitted with a device which breaks its own circuit each time it operates, so that the small relay coil is only excited for an instant at a time. The regulator can, of course, be used with either alternating or direct-current motors. As described, the device can only be used on small machines which do not take more than 10 amperes. For larger compressors a controller is required in order to limit the starting current. The starter is operated automatically by having the main contacts of the pressure regulator relay in series with the controller solenoid, so that, when the relay opens, this opens the controller and stops the motor. Similarly, when the relay closes, this operates the controller and starts up the motor. The method of varying the speed of the motor when running consists, in the case of direct-current machines, of automatically varying the shunt resistance of the motor. Various methods, suited to the range of speed regulation required, may be adopted. Speed variations of 200 to 300 per cent. may be obtained in this way. In the case of polyphase induction motors, two economical speeds may be obtained by using a "Cascade" type of motor. The ratio of the two speeds is frequently 2 to 1, the higher speed being used during the periods of the day when the output is high, and the lower speed during less busy intervals. Another system which may be adopted on an alternating current supply is the use of a motor having two sets of pole windings. The speed of an induction motor is inversely proportional to the number of poles, so that with two sets of windings on the stator two distinct speeds may be obtained.

Owing to the extra cost and the greater complexity, however, it is preferable to run electrically-driven compressors at constant speed, and to control the supply of compressed air by means of an automatic output regulator. The device used on "Sentinel" compressors is shown in Fig. 121, a similar arrangement to that used on many other machines. It is fitted in the inlet pipe. Air pressure is communicated from the receiver to the spring-loaded diaphragm (A). When the pressure in the receiver exceeds the predetermined limit, the diaphragm is pressed back, opening the valve (B) and allowing the air pressure to close the throttle valve (C) by pushing the piston upward against the controlling action of the spring which normally keeps the valve open. On the receiver pressure falling below the normal, the pilot valve (B) closes and, as the air pressure is released, the control valve (C) is pulled open. When the throttle valve is shut, the piston simply travels to and fro in the cylinder without drawing in or

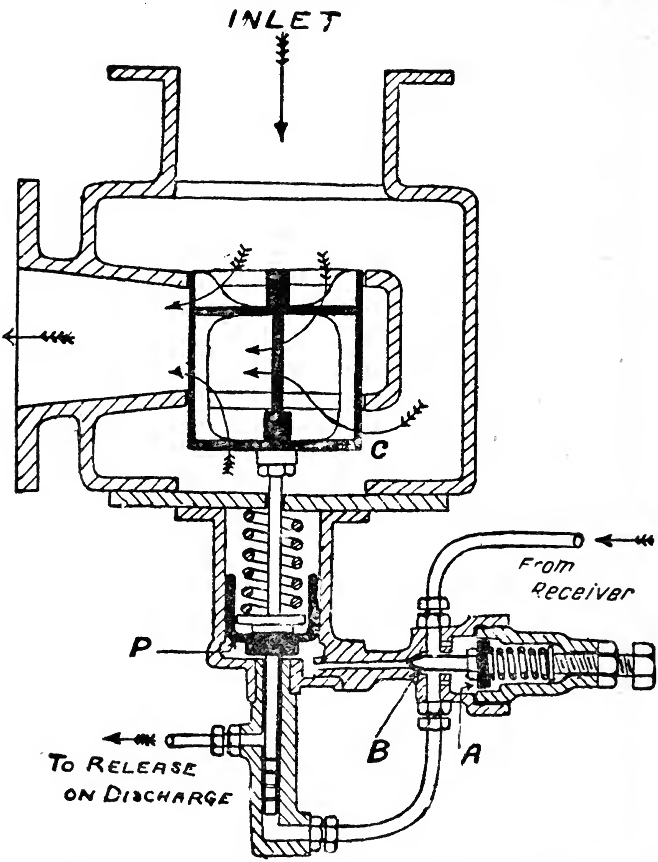


Fig. 121. Throttle-Valve Control for "Sentinel" Compressor.

delivering air. Thus the no-load losses are practically limited to the no-load friction losses. The control valve does not act gradually, but acts so that the inlet passage is either fully open or completely closed. The action is prompt and positive, and this is found to be more economical than a gradually throttling action. The pilot valve may be adjusted so as to operate at any plus or minus pressure from the normal working value.

When the compressor is motor-driven, it is generally found desirable to provide a special unloading device, in order to allow the motor to start easily. Especially is this the case with large motors. The starting current, if the receiver is full, may be anything from one-and-a-half times to twice full-load current. It is important to be able to reduce this consumption of current to well under full-load current. This is best done by using a special unloading valve. In the "Empire" control systems already referred to (Fig. 120), the unloading valve (not shown) is actuated by the pilot valve (D), which is operated by the electro-magnet (C). The pressure regulator (A) closes at the minimum pressure limit, and this energises the pilot-valve magnet, which, operating the unloading valve, releases the pressure on the discharge side of the compressor, and so allows the compressor to start up easily. On reaching full speed the pilot-valve circuit is opened and the load applied.

Air-Filtering.—In the dusty atmosphere which is so frequently met with in mines, it becomes imperative to filter the air effectively before it is drawn into the compressor. Especially is this so if the dust is of a carbonaceous nature as at collieries. Carried into the compressor, the coal-dust becomes permeated with oil and adheres to air-passages, valves, intercooler tubes, etc. Besides obstructing the passage of air and the free action of the valves, it greatly interferes with the efficiency of the intercooler, and of the compressor as a whole. Further, its presence in the cylinder is a source of grave danger, and may be a contributory cause to a disastrous explosion. Dust of any kind interferes with the satisfactory working of a compressor, and to avoid trouble from this source, the air drawn into the compressor should be filtered before passing into the inlet pipe. Probably the most satisfactory arrangement is that employed with such conspicuous success on the Rand and elsewhere. The straining material is composed of cocoa-nut matting of the finest texture obtainable. The matting is held in position by a strong wire gauze of $\frac{3}{8}$-inch or $\frac{1}{2}$-inch mesh, the whole being stretched and supported in a wooden frame. The strainer is built in rectangular box form, generally square, having roof, floor, and sides, the total area of air-passage through the matting being at least 50 times the cross-sectional area of the inlet pipe. The strainer is fitted up outside the engine room so that the air drawn in may be as cool as possible. As is well known, cool inlet air is essential to efficiency, and in those cases, where, to avoid the necessity of filtering, the air is taken from the engine room, efficiency must inevitably suffer. It can be shown that for the same conditions in other respects, a rise of 5° F. in the inlet temperature involves a lowering of the air efficiency by about 1 per cent.

The strainer box is often fixed to the outer wall of the compressor house, being protected from wet weather by a wooden shed. For larger compressors the strainer box may be placed on cross-timbers in a wooden or sheet-iron house on the ground level. A pipe leads from the centre of the filter box through the wall or underneath the floor, and into the inlet of the compressor. This pipe should be ample in size so as to prevent surging on the suction stroke of the compressor. The straining arrangement described can quickly and easily be taken to pieces for cleaning purposes. The several sides are held together by wing nuts, and by undoing these any side may be removed and replaced by a spare one without disturbing the remainder of the box.

Some such arrangement should also be used on compressors at work in the mine itself, where the atmosphere is unavoidably dusty. The use of a wire gauze screen has the objection that if sufficiently small in mesh to be at all effective it soon becomes hopelessly choked up with dirt.

The makers of the "Sirocco" mine fan have designed an air filter working on the wet principle. The supply air is drawn by means of a fan through a washer which cleans and cools the air and it is delivered to the compressor under a small pressure—about 1 inch water-gauge—in order to exclude any unwashed air. The washer consists of a spray chamber where water is discharged from spray nozzles in a very fine atomised condition in the direction of the air current. Much of the solid matter in the air is precipitated down into the tank. The air then passes through a battery of scrubber plates, on which any dust or dirt remaining in the air is deposited. Water sprays playing on the scrubber plates wash the solid matter so deposited to the bottom of the tank. Thereafter the air is dried by being drawn through a triple battery of eliminator plates where buffeting from side to side takes place, causing the air to deposit any free moisture, and enabling it to emerge from the filter in a clean and cool condition with no trace of free moisture. Such an arrangement was recently supplied to the Rothervale Colliery Company, the compressing plant having a total full-load output of 16,000 cubic feet of free air per minute. The makers claim for the wet filter that (1) it cools the air, (2) it does not choke up—a frequent trouble where filter cloths are used—and (3) it is easier to clean and requires less frequent cleaning than the dry form. The fan is not absolutely necessary, but is an advantage, as it maintains a slight positive pressure in the inlet pipe and so ensures the cylinder being filled with air at fully atmospheric pressure. Care has to be taken, however, to design the filter and fan so that they are able to deliver air at a rate equal to the maximum rate of flow of the compressing plant. It is necessary to remember that the rate of flow of air into a compressor cylinder is proportional to the piston speed, and that is at its maximum when the piston is approximately midway in the cylinder.

APPENDIX C

ROCK DRILLS

ADDITIONAL NOTES TO CHAPTER X.

THE Hardy Patent Pick Co., Ltd., has recently introduced a telescopic air-feed hammer-drill specially designed for work in stopes, drifts, and rises. The drill, known as the Water-Jack, is shown in Fig. 123. It possesses several features similar to other drills of the air-feed pattern. The valve is of the air-thrown spool type, and rotation is by hand. An anvil block is interposed between the piston and the drill steel, a feature which is employed in the other air-feed hammer-drills mentioned above. Water is fed under pressure through the anvil and thence through the drill steel to the bottom of the bore-hole. For relatively light work these drills give excellent results.

Rotation of Drill Steel.—In percussive drilling it is imperative that the drill steel be turned through a small angle between successive blows, otherwise the bit will soon begin to stick fast in the hole, and further progress become impossible. In hand drilling the proper rotation of the steel is a simple matter in the hands of a skilled driller, but with the large and clumsy forms of power drills first introduced the turning of the bit was quite a different proposition. Two methods of overcoming the difficulty presented themselves. Either the front head, drill steel included, had to be rotated, or only the piston to which the drill steel was attached was revolved, the rest of the machine remaining in a fixed position. With large reciprocating drills the first method was out of the question, although it has been applied successfully in light modern stoping drills of the hammer type, in which, as will shortly be explained, the drill cylinder and the chisel are rotated by hand. The second method was so obviously the only feasible one that development along the lines of automatic rotation of the piston was rapid. Finally, in 1866, Darlington and Jordan invented the rifle bar and ratchet mechanism, which was destined to prove one of the most noteworthy advances in the whole history of rock drills. The device has since been modified

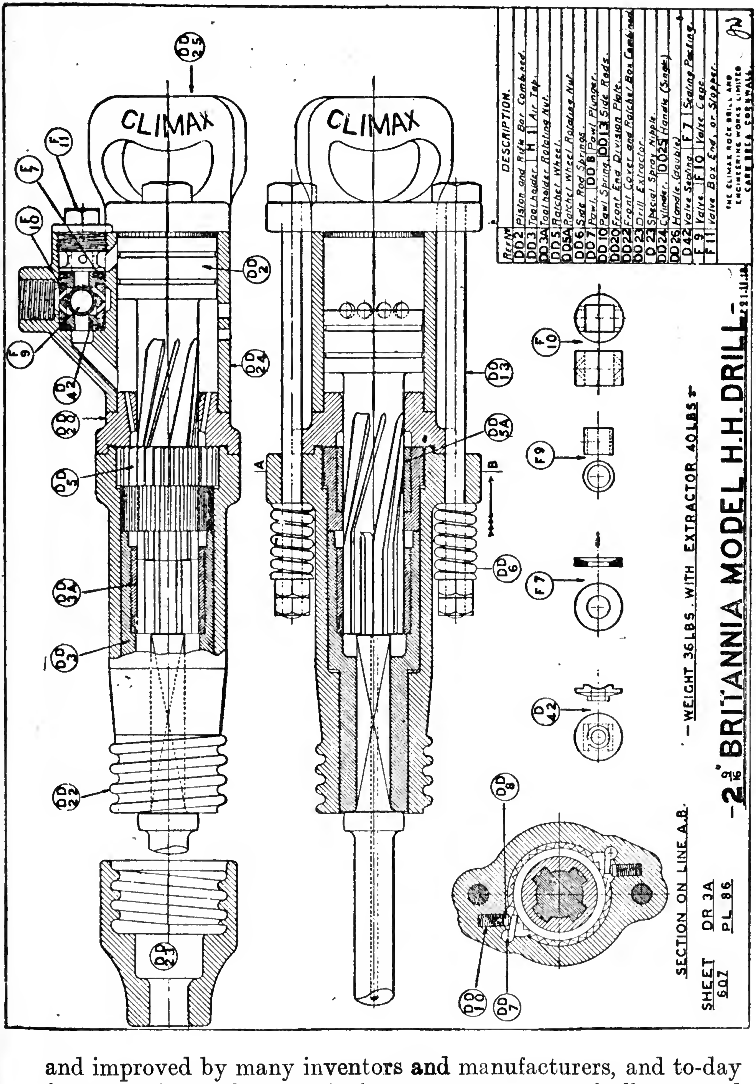


FIG. 122. The Climax-Britannia Hammer Drill.

and improved by many inventors and manufacturers, and to-day forms an integral part of almost every automatically rotated drill on the market.

In machine drills the rifle bar and ratchet has been employed to produce two somewhat different results. In one method there is no possibility of the piston missing rotation when it ought to rotate unless breakage of some of the parts takes place. This is called "non-slip" rotation. In the other system the ratchet wheel is held by friction, and should

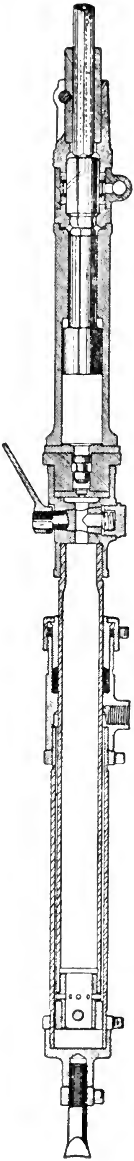


Fig. 123. The Water-Jack Telescopic Air-Feed Hammer-Drill.

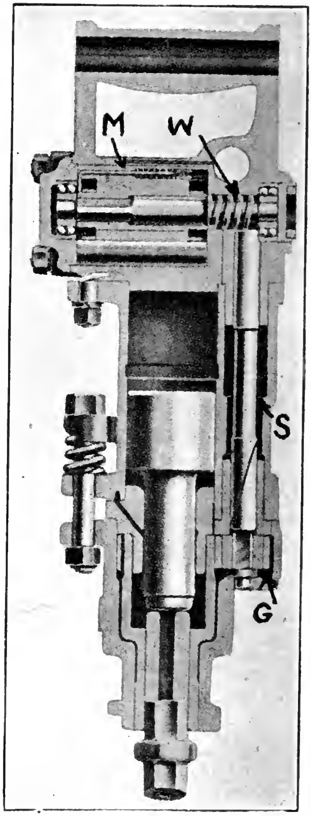


Fig. 124. The Rotation Device of the Hummer Drill.

the drill steel become excessively difficult to turn through sticking or friction in the bore-hole, the rotation system slips and the chisel is not rotated during that stroke. This is termed "slip" rotation. Until a few years ago non-slip rotation was employed extensively, but in modern reciprocating drills, owing to the obvious risk of twisting of the rifle bar and breakage of parts, it has given place to the modification which allows slipping on excessive friction. Practically all modern standard drills employ slip rotation. Rotation of the piston and drill steel always takes place on the back stroke, the forward or hitting stroke being straight. In the Chicago Giant and Slogger drills and others the ratchet teeth are on the inside of the ring surrounding the pawls, while the latter, two or three in number, are placed in the head of the rifle bar. The Sullivan and Siskol drills, however, have the ratchet teeth on the rifle bar head and the pawls in recesses in the slip ring. It is claimed that by having the teeth outside the pawls the teeth are stronger, since their bases fall on a larger circle, and that for a given space they can be more numerous, giving a better rotative effect. A further advantage of the teeth being in the slip ring is that it is the least expensive part to replace.

The commonest arrangement for producing rotation of the drill steel in the hammer type of drill is to use the rifle bar and ratchet as in the piston drill, and in addition to have straight grooves in the front of the piston or hammer which slide in similar straight grooves in the drill-holder. This arrangement is adopted in the Holman cradle hammer drill, the Leyner-Ingersoll, the Jackhamer, Cochise, and Wizard. In hammer drills made by the Climax Company, and in the Flottmann, Waugh, and Sullivan rotator drills, however, the use of a separate rifle bar is dispensed with and the rotating parts are confined to the front end of the tool.

A unique rotation device is employed in the Hummer drill. A section through the drill is shown in Fig. 124. In this machine the ordinary methods of rotation have been superseded by a method which is independent of the movements of the piston. The air is first admitted into a small rotary motor M located at the back head of the drill. The rotation of this motor is transmitted through the worm gearing W, the shaft S and the spur gearing G, to the drill shank. Roller bearings are employed to reduce the friction to a minimum. The advantages claimed for this independent form of rotation are: (1) great smoothness of operation; (2) the free movement of the piston produces greater drilling speed and operation under very low pressure; (3) absence of expensive fluted pistons, rifle bars, and ratchet, with lower cost of repairs. As against these it must be remembered that the necessary shafting and gearing is a complication and a possible source of weakness.

In the hammer drill specially designed for stoping and having automatic air feed for the rotation of the drill steel is accomplished by hand. A lever or handle is provided on the drill for the purpose. The handle is used in the manner of a ratchet brace, being swung alternately backwards and forwards as the work of drilling proceeds.

Dust-Allaying.—Drilling, if carried out in the dry, must of necessity produce clouds of dust more or less fine. The operator of a rock drill would, therefore, if no effective means were employed to allay the dust, inhale great quantities of it into his lungs. It has been found that some dusts, such as those of coal and shale, are not harmful, since after a time the dust begins to be ejected from the lungs. This is not so, however, with quartz and quartzite dust. The finest portions of quartz dust remain in the lungs or only a small proportion of it is expelled. They thus block the minute air cells of the lungs and lacerate the finer tissues, causing silicosis and rendering the operator extremely susceptible to attacks by the tubercle bacillus, so that in bad cases death from phthisis supervenes. Several Royal Commissions have investigated this matter in South Africa and elsewhere in the British Empire, and enactments have been made which render it obligatory to prevent dust clouds.

Numerous devices have been tried to render the dust harmless, but only two may be said to have been completely successful. The first consists in sending a continuous stream of water down the hollow steel of the borer, and the other in spraying the mouth of the shot-hole with water. The first method is exemplified in the Leyner-Ingersoll drill, the Holman water-feed hammer drill, the Hydromax hammer drill of the Climax Company, and the Denver Dreadnaught drill. In these water is forced through a hole in the boring tool right to the face of the bore-hole. The advantages arising from this method in addition to the effective laying of the dust are:—(1) The water effectively clears away the cuttings from the bottom of the hole and allows the bit to strike fresh rock at every blow, and (2) it cools the cutting edge and preserves it. In the Leyner and Holman drills both air and water are used. The water passes from the rear of the drill through a water tube into the hollow steel. Here air from the drill mingles with it, and both pass down the borer to the bottom of the hole. This is a very effective system, and aids the work of the drill. In the Water Jack, Hydromax, Dreadnaught, and Sullivan drills, however, only water is used. In this method, as also with the Leyner drill, the water must be under a pressure of 30 to 50 lbs. per square inch. This pressure may be obtained from a pressure-water pipe or by employing a small closed cistern and using the compressed air which operates the drill to act as the pressure agent on the surface of the water in the cistern. Fig. 125 shows the Sullivan Hyspeed drill with water attachment.

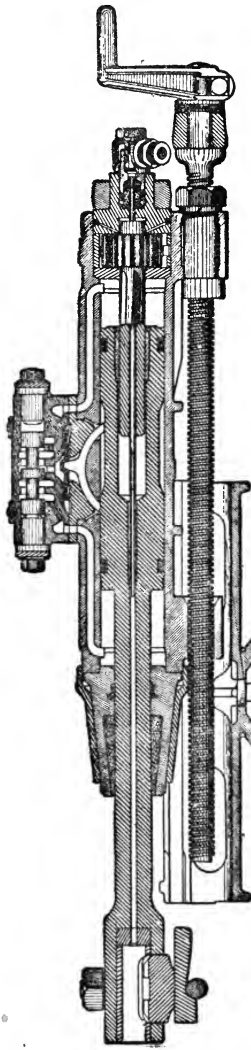


FIG. 125. The Water Attachment of the Sullivan Hyspeed Drill.

In drills which operate dry either with solid borers or with hollow steel and an air flush, the water spray is employed to allay the dust. This is accomplished in a very simple manner. Referring to Fig. 126, the air enters at (*a*) and passes through the nozzle (*n*). Here its velocity increases enormously, and the resulting injector effect sucks water at (*w*) through a flexible hose from a tank or pail. The mingled air and water are ejected at the spray nozzle (*s*) in the form of a coarse spray, which is directed at the mouth of the bore-hole. The spray, meeting the dust-cloud, effectively renders it innocuous. It is necessary that the water be correctly atomised. This is best done, as has been conclusively proved by experiment, by means of compressed air. The pressure air, in issuing from the jet, effectively separates the water into drops. The size of the jet and the proportion of water to air should be proportioned so that the correct degree of atomisation is attained. If the atomising is too fine a dense fog is created, and if it is too coarse large drops of water are formed which readily fall to the ground and do not effectively lay the dust. In drilling a coarser spray is permissible than in sprays which are used for laying dust in the roads, since the distance before deposition is much shorter. Indeed, in the spraying of roads and working faces after blasting, in which operation

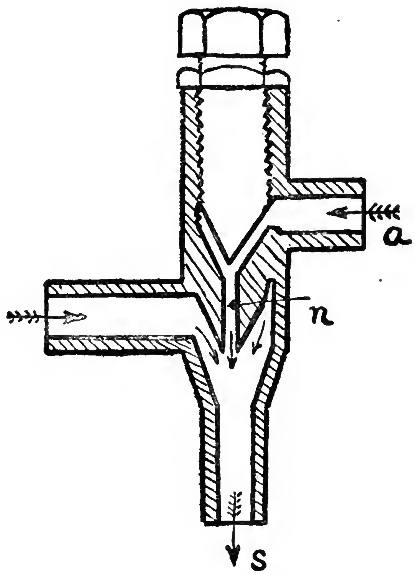


FIG. 126. The Water Spray Producer.

needless to say a vast quantity of dust is produced, it is probable that the best method of laying the suspended dust is a combination of a very fine water spray followed shortly afterwards by a coarser jet.

Drill Bits.—It is not too much to say that the success of a rock drill depends very largely on the proper shape of drill steel and on the skilful sharpening and tempering of the bit. The single chisel bit so much used in hand boring is apt to drill irregular holes when used with machine drills, and the double chisel cutter is generally better, as boring much rounder holes. Moreover, the two-edged bit has the greater advantage of protecting the central hole in the drill steel when air or water flushing is used to clear out the cuttings from the bottom of the hole. Several forms of two-edged bits are used. A common form is that having the two edges parallel; another has the two edges crossing at right angles; and still another like the letter X. The last is a favourite form of bit with many users of drills. The centre of the cross should be raised slightly, a suitable angle of slope being 20° (see Fig. 127). The convex shape tends to keep the drill bit central and to prevent the hole from diverging. A three-edged bit, having the edges shaped like the letter Z, is also much used, while the rosette bit, which has three cutting edges crossing each other on the diameters of a hexagon, is pre-

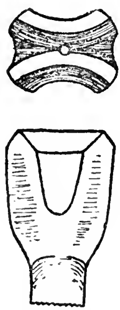


FIG. 127.

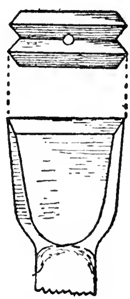


FIG. 128.

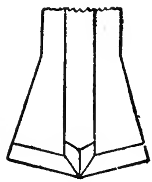


FIG. 129.

ferred for some purposes. The section of the steel is generally cylindrical, either plain or having a spiral, but octagonal and cruciform sections are also used. The function of the spiral is to act as a conveyor which draws the cuttings from the nose of the bit towards the mouth of the hole. It is most suitable for soft rock and for down-holes where there is a tendency for the débris to clog the bit.

Theoretically the best form of cutting edge is that which is so designed as to evenly distribute the work over its whole length. With such a bit in perfectly homogeneous ground the entire cutting edge would become dulled to the same extent. With no bit at present in use is this the case. In all of them the outer fringes of the cutting edge have to do the major portion of the work of drilling. In consequence it is the outer edges which wear most quickly. The contour of a cutting edge which would theoretically ensure equal wear over the whole of the bit would, however, be impracticable. Nevertheless, in designing drill bits this fact should be borne in mind. From this point of view the double-edged chisel bit is better than the cross-bit (see Fig. 128). An even nearer approach to the ideal is the double-arc bit recommended by the Sullivan Company (see Fig. 129).

For soft ground the angle of the cutting edge may be sharper than for drilling in hard ground. For hard rock the angle of cutting edge should not be less than 90°. The shoulders of the bit should be well supported for strength, and properly designed reaming edges are necessary to enable the bit to ream out the hole and maintain the gauge.

The difference in gauge of following drills should not be more than $\frac{1}{8}$ inch, and some writers advocate as little as $\frac{1}{16}$ inch. In hard rock, however, it is probable that at least $\frac{1}{8}$ inch is necessary to ensure the drills following each other easily. It should be remembered that the greater the reduction in gauge in the drills of a set the larger will be the initial diameter of the hole for a given final size. This means additional work the drill has to do for the same effective size of hole.

The steel from which the drills are made should contain from 0·6 to 0·85 per cent. carbon and be free from sulphur and phosphorus. The heating, whether in ordinary blacksmith forges or in oil, gasoline, or electric furnaces, should be properly regulated and the tempering done at the proper temperature. Sharpening is generally done by hand, but machine sharpeners are also largely used. A separate dolly should be used for each size of bit.

Operation of Drills.—The success of a drill depends largely on the care and skill of the operator. The commencement of a hole is the most difficult part, especially if a hand hammer drill is used. With the piston drill securely fixed to its tripod or column the starting of the hole is generally negotiated with comparative ease if care is taken to select a face of rock normal to the line of the drill. But with the hammer drill, even allowing for equal care in the choice of the starting point, there is the necessity for holding the drill up to its work, with the consequent tendency for the bit to spread itself over an area much larger than the proper size of the hole. With screw feed machines the proper rate of feed to suit the particular ground should be the careful study of the drill-man. Too rapid feeding shortens the stroke and reduces the drilling capacity, and under-feeding will result in damage to drill shanks, and produce breakage of chucks, since, if the bit is not up to the face of the rock, the whole force of the blow of the piston expends itself on the steel instead of on the rock. In feeding the bit should always be kept pressed lightly against the bottom of the drill hole. With hammer drills care should be taken throughout the whole period of drilling that the drill steel and the machine are in line, otherwise the piston will not hit the steel fairly but on the edge, and may eventually injure the end of the hammer and chip the shank.

Drilling with blunt bits is bad for any machine, and sufficient sharpened bits should be at hand to replace a damaged or blunt steel when required. The blunter the bit, the greater the shock to the tool and the drill when the blow is struck. Breakages of drill shanks are largely due to drilling with blunt bits. Proper attention should be given to the lubrication of the valve. When inserting a borer which simply fits into the chuck, care should be taken not to force it in, as it may take hours to get it out, and the shank should not be passed by the smith before being gauged to make sure that it will fit into the chuck properly without being too tight. With drills using the U-bolt chuck, the latter must grip the steel securely and in true alignment with the piston extension. The chuck bushing should be renewed when too much worn. It is one of the most important parts of the drill, as if it is much worn it interferes with the correct alignment of the steel. If the shanks of the drills become excessively worn, they should be re-shanked, as ill-fitting shanks are a second source of incorrect alignment. With a new or newly-sharpened bit, the air pressure should be turned on gradually, gently at first, and then afterwards gradually increasing to full pressure. This caution may avoid breaking the corners of the bit. Before coupling up the air hose to the machine it is a good plan to blow air through it for a few seconds, and on disconnecting the hose from the machine the inlet of the latter should be plugged. In this way dust and grit will be prevented from getting into the drill. With drills which have a water feed through the steel to the bottom of the bore-hole, the water should be turned on *after* the air and turned off *before* the air.

When the drill is not in use it should be laid in as clean a place as can be got, and not just laid down anywhere. No drill, however well designed or strongly made, can continue to give satisfaction for an indefinite time unless it is properly looked after and overhauled from time to time. In examining and refitting, particular attention should be given to the valve, the rotation gear, and the piston. Damaged or badly worn parts should be replaced. If the piston should become too slack in its cylinder, the latter should be re-ground and a new piston fitted. In general this should be done when the diameter of the piston is less than the bore of the cylinder to the extent of more than $\frac{1}{64}$ inch. Some makers supply pistons increasing by $\frac{1}{16}$ inch diameter, and when a cylinder has become worn it may be re-bored to fit a piston $\frac{1}{16}$ inch larger than the piston previously used. Some users of drills, however, prefer to purchase new cylinders rather than bore out worn ones. If a cylinder is much worn there is no doubt that the loss through increased air consumption of the drill will in a few weeks equal the cost of a new cylinder. An innovation introduced by the Sullivan Company in 1913 in their Liteweight drill consists in the use of a cylinder fitted with a renewable liner of hardened steel.

Rockers, tappets, and auxiliary valves should be frequently inspected. Excessive or irregular wear has the effect of shortening the valve movement, producing cushioning of the blow of the drill. The feed screw and feed nuts in piston and cradle hammer drills should be kept in order and replaced when worn. Careful attention should also be given to the cradle of the machine. If the cradle guides become greatly worn the machine loses in rigidity, so that the drill bit does not hit true, but strikes a different place in successive blows. This undesirable feature will be most apparent when the machine is run out to the full extent of the feed screw. Whenever instability of the machine is noticeable, inspection of the cradle or of the clamp, arms, or bar to which the machine is fixed should be made and the matter put right without delay. All cradle machines have provision for taking up wear in the cradle guides.

All drills should be brought to the surface for inspection and repair periodically, say every three months. A record should be kept of the condition of each drill at each inspection and the details of the repairs carried out. The importance of maintaining the drill in a high state of efficiency cannot be over-estimated. No type of drill, however good in design and construction, will continue to produce satisfactory results unless it is carefully, skilfully, and systematically overhauled and all the parts maintained in as perfect condition as possible. As one writer on the subject has said, the key to success in rock-drilling may be summed up in the word "maintenance."

Tests and Efficiency.—The success of a rock drill depends upon the following points in order of importance:—(1) Speed of cutting, (2) strength and durability, (3) air consumption, (4) portability, (5) ease of fitting up, (6) simplicity of construction, (7) ease of repair.

From the points of view of portability and ease of fitting up, the hammer drill, of course, has a great advantage over the heavy piston drill, but the two classes have really to be considered separately, as each is to some extent supreme in its own special sphere of work.

A high speed of drilling is a strong point in favour of a rock drill, but that alone is not sufficient to make the machine a success. Nothing could emphasise this more pointedly than the competition arranged in 1907 under the auspices of the *South African Mining Journal* to test the merits of light stoping drills. The machine which outdistanced all others from the point of view of cutting speed was the Gordon, a hammer drill. Yet when put to actual use underground it could not stand the wear and tear of every-day work, and failed to come up to the expectations warranted by its position in the tests. The test clearly proved that high-cutting speed, though greatly to be desired, must also be accompanied by reliability or else the machine is doomed to prove a failure.

It should be remembered, too, that the cutting speed in any given rock material depends very largely on the skill of the operator, the state of the drill as regards repairs, the efficiency of the drill bit, and the effectiveness with which the bottom of the drill hole is kept clear of the cuttings made by the drill. Further, the air pressure used has an important bearing on the speed of drilling. A high air pressure produces a more powerful blow for a given size of drill, but the drill bits are blunted more quickly, the machine itself is subjected to greater stresses, with the inevitable result that breakages occur more frequently, repairs are high, and the life of the drill is shortened. On the other hand, very low pressures are certain to prove uneconomical, especially in hard rock. Air pressures varying from 40 lbs. per square inch up to 120 lbs. per square inch are used. The best practice is to employ pressures of 60 to 80 lbs. per square inch, and to maintain the working pressure as uniform as possible.

The speed of drilling has increased considerably of late years, and it would be safe to say that the average cutting speed has been doubled in the last ten years. But what is of more importance, the reliability and handiness of the machines have been enormously improved. There is no doubt that much of this advancement is due to the enterprise of manufacturers as well as to the insistence of the user on a drill that will stand the rough handling of underground conditions.

Perhaps the most complete series of tests of rock drills carried out in the history of mining were those conducted by the Transvaal Government and the Chamber of Mines in 1909-10. The tests were carried out under ordinary underground conditions, and their exhaustiveness can be realised from the fact that they stipulated for 300 drilling shifts in seven different stopes. The test period was, however, eventually reduced to 215 shifts of eight hours each, owing chiefly to it being found impossible to maintain the requisite air pressure in one of the mines where the drills were to be tested. Twenty-three drills entered for the trials, and nineteen of these started the competition. Only four completed the test, these being two Holman drills, the Siskol, and the Chersen. The prizes were £4,000 for the drill taking first place and £1,000 for the second best. The Holman and the Siskol drills were deemed of equal merit, and the prize money was divided between them. The total cost of the competition was over £17,000, but there can be no doubt about the great value of it, since ordinary working conditions prevailed throughout, and the duration of the test was such as to test the durability and reliability of the machines severely. Elaborate records were kept of footage drilled, costs for labour, drills, sharpening, spares, and stores, air consumption, etc. All the four drills completing the test were reciprocating drills, an altogether different result from the 1907 tests already referred to when a hammer drill was the best. The winning drills were light machines, the weight being limited to 100 lbs., and the competition being intended for stoping drills. Experience in the tests and since has shown, however, that a slightly heavier machine can be conveniently handled in the stopes and a more powerful drill obtained. About 130 lbs. is now considered to be the best weight for a stoping drill of the piston type.

The air consumption of a drill is a matter of great importance. Not that it is quite so important a factor as speed of cutting and durability. Nevertheless the air efficiency of the drill must not be ignored. The compression of air is costly, and no machine operated by air power can be tolerated which does not endeavour to use the air to the best advantage, consistent of course with the other necessary desiderata. During the South African tests the drills were periodically taken to the Johannesburg University Technical College where the air consumption was tested. The air consumption of a drill is usually expressed in cubic feet of free air per minute—that is, air at normal atmospheric pressure. The actual quantity of air measured in terms of free air taken by a drill depends upon (*a*) the size of the drill, (*b*) the air pressure, and (*c*) the condition of the drill. Naturally the larger the drill the greater the quantity of air taken. Also, if the drill is in a state of disrepair it will take a much larger quantity of air to do the same work than if it were in good condition. As regards the air pressure, the quantity of air taken by a given drill is not quite directly proportional to the working pressure. For example, a drill working at 100 lbs. per square inch does not take twice as much air as the same machine working at 50 lbs. per square inch, but only about 80 per cent. more air. The air consumption of piston drills in good condition varies from about 65 cubic feet of free air per minute for a 2-inch drill at 70 lbs. per square inch pressure to about 175 cubic feet per minute for a $3\frac{1}{2}$-inch drill and 100 lbs. per square inch pressure. Hammer drills take anything from 50 to 100 cubic feet per minute.

Most manufacturers and many mine owners test the efficiency of their drills from time to time. It is not a simple matter to measure the efficiency of a rock drill. One can calculate with ease the horse-power represented by the compressed air the drill consumes, but it is a much more difficult problem to estimate the amount of useful work performed in the usual engineering units. As a matter of fact it is hardly possible to estimate the absolute efficiency of a rock drill. All that one can do is to compare one drill with another. In order to carry out such a test fairly, the several drills should be in an equal state of repair, they should be operated by the same skilled drill-man or by men equally skilled in the working of their respective drills, the drill bits should be of the same class of steel, and shaped, sharpened, and tempered with equal care, the drilling should be carried out in the same kind of rock, and the air pressure should be maintained uniform throughout the tests. Such tests, if carried out carefully and with scrupulous fairness, cannot fail to be productive of good results. They will show up the relative merits of the types of drills tried, both as regards cutting speed and air consumption. It should not be forgotten of course that a drill is constantly being tested in the ordinary every-day work of drilling in the mine, and the testimony of the drill-runner or mine foreman is one of the most valuable criteria of the worth of a drill.

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