by the "mechanical efficiency" of a compressor? There is rather a tendency to confuse the losses due to (a) the production, and (b) the use of compressed air. These losses should be kept entirely distinct, in the manner in which the inefficiencies of boiler plant, engines, and the plant driven by the engines are separated.

The losses due to the system of compressing air may be divided into several sections, which are covered by the general expression "loss of volumetric and mechanical efficiency." In its accepted sense, as regards compressors, "mechanical efficiency" simply means the proportion that the horse-power shown by the air cards bears to that shown by the steam cards. Thus the percentage of loss of efficiency simply covers the internal friction of the compressor and omits altogether those other mechanical losses which we know must occur. To find the "volumetric efficiency" a measurement is made of the length (on the atmospheric line of a low-pressure card), between the expansion and compression curves, and the percentage that this length bears to the total length of the card shows the "volumetric efficiency;" the effects of clearance and low initial pressure are in a sense allowed for, but the losses due to semi-adiabatic compression, high temperature, and leakage are ignored. It will be readily seen that the least efficient system of compression may easily appear to have the greatest efficiency if such a basis of comparison is accepted. And yet on such absurd premises as these, a comparison is frequently attempted of the efficiencies of plants. Of course, many engineers compare their cards with cards set up on a basis of (a) isothermal compression; (b) the absence of clearance; and (c) the absence of losses between stages, etc. I hope to prove that even this system it not satisfactory. Before proceeding I wish to allude to the great

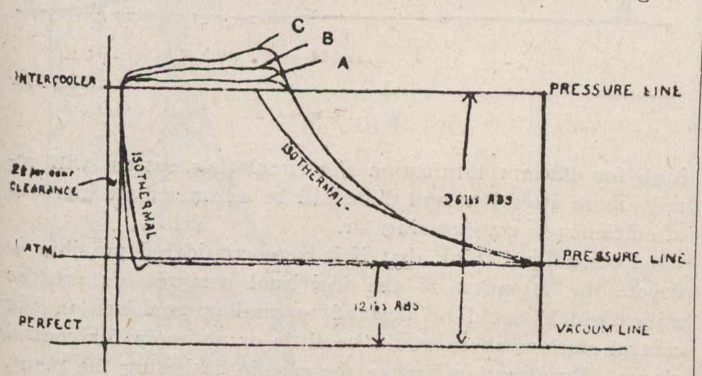

FIG. 1.

difficulty that there is in obtaining full and reliable information on which to base conclusions. Invariably when engines are indicated, there is not sufficient data providel from which to deduce the results. Anyone can obtain air cards by the gross, but it is the exception to find one on which any useful information is noted. In the ordinary course of taking cards it is not necessary that any particular care or trouble should be taken; the engineer only requires the cards so that he may see that valves are in order and that details of that kind are normal. Therefore, for his purpose, it is hardly necessary that even the "scale' 'to which cards are drawn should be given; but, if anything in the nature of efficiency is to be determined, the card might just as well be omitted as the following details:—1. Barometric pressure. 2. (a) Temperature of atmosphere; (b) temperature of low-pressure air inlet; (c) temperature of low-pressure air outlet; (d) temperature of high-pressure inlet; (e) temperature of high-pressure air outlet; (f) temperature of circulating water inlet; (g) temperature of circulating water outlet. [Note.—The latter (f and g) should be taken at the low-pressure cylinder, high-pressure cylinder, and intercooler.] (h) Quantity of circulation water. 3. Pressure at intercooler. 4. Pressure at receiver.

The necessity of obtaining some of these details will be realized on referring to Figs. 1 and 2, which give typical cards for the low-pressure and high-pressure cylinders respectively of a two-stage compressor, three delivery lines being shown on each card for different kinds of outlet valves. Unless the intercooler anl receiver pressures are shown on the card the losses due to high outlet pressure cannot be ascertained. It will be seen that the low-pressure cylinder delivery pressure is considerably above the intercooler pressure, whilst the high-pressure inlet is slightly below intercooler pressure, the respective amounts above and below the intercooler line showing the losses due to the friction of air in pipes and the raising of valves from the seats at outlet and inlet respectively.

In Fig. 3 a combined card is given showing the following:—

	Pressure.	Temperature.
Atmospheric	12 lbs. abs.	80° F.
Intercooler inlet	mean 31 lbs. abs.	240° F.
Intercooler outlet	mean 31 lbs. abs.	110° F.
Receiver inlet	92 lbs abs.	230° F.

An isothermal curve is given for purposes of comparison. The area A M L D and M B C L are measures of

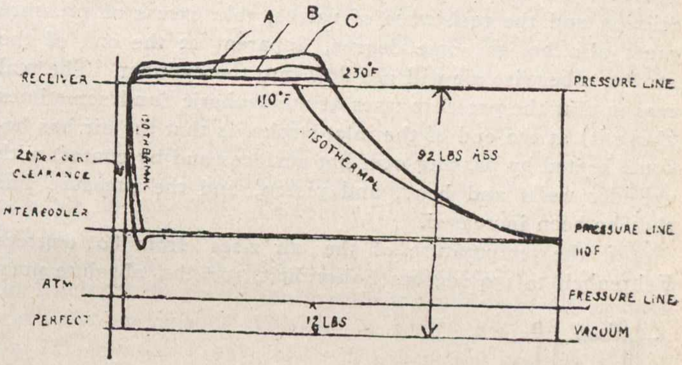

FIG. 2.

the theoretical work of compression; the areas enclosed by A E L F and G H I L show the work actually performed in the two cylinders. Clearances are again taken as being 2½ per cent.. In the theoretical card there is no loss shown between (or in) cylinders. The amount of overlap of the actual cards shows clearly the great loss due to (a) pipe friction, and (b) the excess pressure required to overcome spring loaded valve resistance.

In Fig. 3 the theoretical card is shown based on the assumption that there is no cylinder clearance, thus favoring the compressor. Fig. 4 shows the correct manner of sketching the theoretical cards, and in order that the differences between it and Fig. 3 may be made more apparent, the latter is repeated in the diagram in Fig. 4, and it will be seen that by omitting clearance the theoretical work of one stroke is

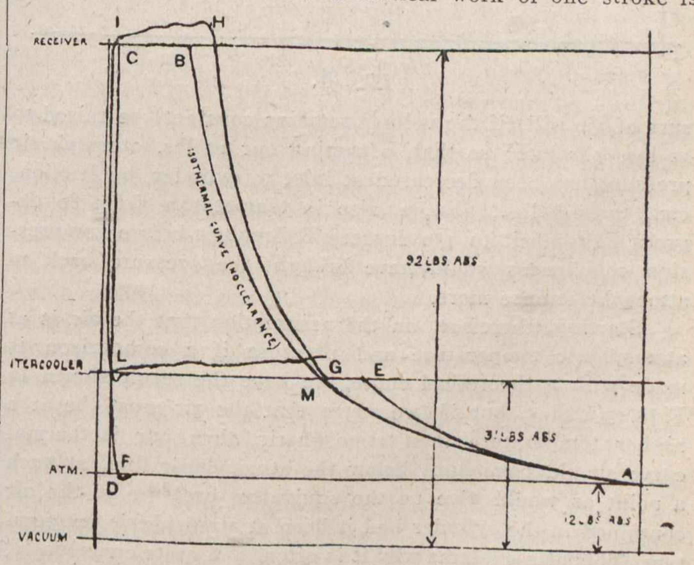

FIG. 3.

made to appear appreciably greater than it really is.

As will be seen from Fig. 5, which combines the work cards of Fig. 3, and the theoretical cards of Fig. 4, the actual losses in comparison are very considerable, the losses being "hatched in." It is my object to show that even the very considerable losses shown in Figs. 3 and 5 are below, rather than above, the actual losses.