NOTE ON STEAM-TURBINES.

By CAPT. H. RIAL SANKEY, R.E. (ret.), M.Inst.C.E.,
M.I.Mech.E.

THE following diagrams illustrate the production of motion energy in steam-turbines of various types, and the conversion of this energy into mechanical work; and it is hoped that they may be of assistance to those who are beginning to study the steam-turbine.

The matter is regarded from the standpoint of a flow of energy into, through, and out of the turbine. A portion of the energy flow is converted into work, another portion leaks out during passage through the turbine in the form of radiation and conduction, but by far the greater amount flows away into the exhaust. Even if the turbine were ideally perfect—that is to say, if there were no radiation or conduction losses, and the conversion into work were the maximum theoretically possible—the portion flowing away in the exhaust would still be very considerable. As is well known, for saturated steam the maximum conversion is that of the Rankine cycle, and is exhibited on the energy chart in Fig. 1 by the shaded area marked R , and the area $a_1 A B b$ represents the total energy flow, or rather, it is the heat energy expended on each pound of steam to raise the pound of water from the exhaust temperature and convert it into steam at the admission temperature. If 100 deg. F. be chosen as the exhaust temperature corresponding to 28 inches of vacuum, or to approximately 1 pound absolute pressure; and if 366 deg. be taken for the admission temperature, corresponding to very nearly 165 pounds absolute pressure, it will be found that $R = 317$ British thermal units, and the heat going away in the exhaust will be represented by the area $a_1 A D b = 804$ British thermal units.

Adapting a diagram originally prepared by the writer for the Introduction to the report of the Thermal Efficiency Committee of the Institution of Civil Engineers to the case in hand, Figs. 2 and 3 are ob-

tained for an actual and an ideally perfect turbine. The admission and exhaust pressures and temperatures are taken at the same values as in Fig. 1, and it will be supposed that 1 pound of steam per second is flowing through the turbine, corresponding to about 200 brake horse-power. This is, of course, a very small turbine; but the reasoning is not affected thereby. The width of the stream of energy is S British thermal units per second when reckoned equal to 1,121 British thermal units in the numerical example under consideration from 100 deg. F. The issuing stream, representing mechanical work, is of width R (317 British thermal units) for the ideal turbine, and of width A (220 British thermal units) for the actual turbine. A/S is the thermal efficiency of the actual turbine, and the ratio A/R is the "efficiency ratio" as defined by the Thermal Efficiency Committee. In the numerical example the thermal efficiency is 19.5 per cent, and the efficiency ratio equals 69 per cent. These figures have been reckoned on what corresponds to the indicated horse-power, so that actually A must be further reduced to the width B to allow for mechanical frictional resistances in the turbine, as shown in Fig. 2; the ratio B/R , equal in this case to 64 per cent, might be called the *brake efficiency ratio*, and in the case of a turbine seems to be the best ratio to take.

Directly the steam enters the turbine it is made to acquire a considerable velocity by being expanded through nozzles or blades of suitable shapes. The motion energy thus created is derived from the molecular energy of the steam, which is reduced by the exact amount thus created. Thus in Fig. 4, if A represents the entry to a nozzle, and R is plotted equal to the heat utilization of the Rankine cycle, at B , the exit from the nozzle, the height BC would represent the motion energy of the issuing jet, and the balance CD would be partly in the form of eddies, the remainder lost as radiation. If the nozzle were perfect, the point C would move to D , and the motion energy imparted to

the steam would be equal to R , and the ratio BC/BD is the efficiency of the jet. A line, AC , would represent the amount of motion energy at every section of the nozzle, and the shape of this line would depend on the shape of the nozzle. If the jet plays on suitably-shaped vanes, the motion energy will be converted into work; and if the steam came to rest on issuing from the mov-

ing vanes, the whole of the motion energy would be converted into work. Actually, however, in the case of a single row of moving blades, this velocity of the steam would be about 1,000 feet per second, corresponding to 20 British thermal units per pound of steam; and this energy is represented in Fig. 5 by the height EF ; then FG is plotted equal to CD , and thus represents the loss due to the inefficiency of the nozzle; and the balance GH represents eddies, radiation, etc. The remainder, HJ , is converted into work, and clearly the ratio HJ/EF is the brake efficiency ratio, equal in the

numerical example under consideration to 64 per cent. CF gives the reduction in motion energy as the steam passes through the openings between the moving blades, and the dotted lines, AD , DE , represent the conditions in the ideally perfect turbine.

In Fig. 6 the idea is extended to a turbine having an additional set of fixed and moving blades, as indicated diagrammatically. The issuing velocity from the first row of moving blades will be considerably greater than in the previous case; consequently, a smaller portion of the motion energy will be converted into work in the first row of moving blades. In passing through the fixed blades no work is done, as shown by the horizontal line, but the motion energy will be somewhat reduced, as shown by the sloping line; the motion energy thus lost will reappear as heat. The second row of moving blades converts a further portion of the motion energy into work; and finally, as before, HJ/EF is the brake efficiency ratio of the turbine.

Fig. 7 represents an element of the Parsons parallel-flow turbine, drawn to a much larger scale than the previous figures, and it will be noticed that in this case motion energy is produced both in the fixed and moving blades; it is, of course, only in the moving blades that mechanical work is done as indicated in Fig. 7. The curves shown, however, are only diagrammatical. The results obtained above have been applied to turbines of various types, following the classification pro-

posed by Mr. Neilson, in a paper he read before the Manchester Association of Engineers. His Classes I. and III. have already been dealt with in Figs. 5 and 6. The Laval and the single-disk Rateau turbine are practical examples of Class I. Class III. is represented by one type of the Riedler-Stumpf turbine. Figs. 8 to 10 deal with the remainder of Neilson's classification, and the names of some of the actual turbines at present being manufactured are noted against each figure.

In Figs. 8 and 9 the loss due to the nozzles is indicated by the vertical distance between the motion-energy line and the lower chain-dotted line, and the loss due to eddies, etc., is measured vertically from

	Neilson's Classification.	Names of Actual Turbines.
Fig. 8.....	Class II.	{ Rateau Multicellular. Zoelly. Hamilton Holzwarth.
Fig. 9.....	Class IV.	{ Curtis. Riedler-Stumpf.
Fig. 10	Class V.	{ Parsons.

the "work" line to the upper chain-dotted line. The distance between the two chain-dotted lines gives that portion of the utilizable energy in the steam (317 British thermal units) still remaining as molecular energy in the steam at the various stages in the turbine. In Fig. 8 the temperature and pressure are also shown, on the supposition that one-third of the total utilizable energy is dealt with by each of the three sections of the turbine. The rapid drop of pressure in the first section is very noticeable.

In the case of the Parsons turbine (Fig. 10) the alternate rows of blades are so numerous (50 to 80) that they cannot be shown to so small a scale; therefore only three sets of fixed and moving blades have been drawn to a larger scale than the rest of the figure. Both the "work" line and the motion-energy line will therefore be in the shape of a saw, following DH and AF respectively. For the sake of comparison the brake-efficiency ratio has been taken as the same as in the case of the other turbines, although for so small a Parsons turbine as 200 brake horse-power this ratio would be less. Similar diagrams can obviously be drawn for any other form of turbine by combining, as may be necessary, the diagrams given in Figs. 5, 6, and 7.—Engineering.

MANUFACTURE OF HYDRAULIC CEMENTS.*

By L. L. STONE.

IN preparing a paper on the subject of cement it is fully recognized that many articles have already been prepared on this subject, and that various descriptions of it have repeatedly been written. Yet, the extensive developments in the cement industry, both from an engineering and financial standpoint, are commanding

such unusual attention that the manufacture of cement is very much discussed. It is then the purpose of this paper to give here as complete a description of the processes of manufacture as possible, free from the cumbersome figures which may be obtained at length from any factory report.

The term hydraulic cement is applied to any cement which possesses the property of hardening under water. Under these cements are included the three chief commercial cements of the present time, viz.: natural, slag and Portland cements. The general process of manufacturing and hardening is similar in the three classes. Each is a compound consisting chiefly of lime, silica, and alumina, calcined at a high temperature and the product reduced to a fine powder. The compounds of lime, silica, and alumina are produced by thoroughly mixing some form of carbonate of lime with the exact proportion of clay.

Natural cement was the first cement manufactured in this country, and was extensively used some years ago when nearly all Portland cement was imported from Europe. It is still used to a small extent but only for work of secondary importance. It is made in several places throughout the country from natural deposits of rock containing nearly the same proportions of lime rock and clay as are used to produce Portland cement. If a uniform deposit of rock containing exactly the correct proportions of rock and shale could be found, a natural cement, equal in grade to Portland cement, could be made of it, by simply burning and grinding; but since a variation of a very few per cent in composition is sufficient to destroy or greatly reduce the value of the resulting cement, there never has been found a natural rock deposit sufficiently uniform to produce a cement of this quality. In many cases it is only by the most careful sorting and mixing of certain portions of the raw material that cement can be made from it at all. The process of manufacture is very simple—the rock is quarried and burned at a comparatively low temperature and ground to a powder. It is slower in setting than Portland cement and does not develop as great strength.

Slag cement is made by mixing blast furnace slag with some form of lime, then burning and grinding.

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