The increased diameter at the low-pressure end not only allows the steam velocity to be increased but by enlarging the annulus enables a sufficient area of passage to be provided without unduly lengthening the blades. In the very last stages of the expansion, however, the volume becomes so great that it is not practicable to provide sufficient area by lengthening the blades, and the blades there are accordingly shaped so as to face in a more nearly axial direction and are spaced more widely apart.

The area of the steam passage depends on the angle of the blade. If the blades were indefinitely thin it would be equal to the area of the annulus multiplied by the sine of the angle of discharge, and in practice this is subject to a deduction for the thickness of the blade on the discharge side, as well as to a correction for leakage over the tips. Generally the angle of discharge is about 221/2°; and the effective area for the passage of steam is about one-third of the area of the annulus.

Fig. 61 A shows a representative pair of fixed and moving blades of a Parsons turbine, and fig. 61 B the corresponding velocity diagram for the steam, neglecting effects of friction. V1 is the exit velocity from the fixed blades, the delivery edges of which are tangent to the direction of V1. The blade velocity is *u* which is 1/2V1. V2 is consequently the relative velocity with which the steam enters the moving blades. Approxi­mately, the back surface of these blades is parallel to V2, but the blades are so thick near the entrance side that their front faces have a considerably different slope and there is therefore some shock at entrance. Tn passing through the moving blades the relative velocity of the steam over the blades changes from V2 to V3. Allow­ing for the velocity *u* of the blades themselves, this corresponds to an absolute velocity V4, with which the steam enters the next set of fixed blades. In these blades it is again accelerated to V1 and so on.

126. *Calculation of Velocity at each Stage.*—The acceleration of the steam in each row of blades results from a definite heat drop. Or, if we look at the matter from the point of view of the pressure-volume diagram, the acceleration results from the work done on the steam by itself during a drop *δp* in its pressure. The amount of this work per pound is *vδp* where *v* is the actual volume per pound. It is convenient in practice to write this in the form *(pv)δp/p,* for the product *pv* changes only slowly as expansion proceeds. In designing a turbine a table of the values of *pv* throughout the range of pressures from admission to exhaust is prepared, and from these numbers it is easy to calculate the work done at each stage in the expansion, the pressure *p* and drop in pressure *δp* being known. In the ideal case with no losses we should have

(V12-V42)/2*g*=(*pv*)δ*p/p*

or V12=2*g*(*pv*)δ*p/p*+V42 where V4 is the velocity before the acceleration due to the drop *δp* and V1 is the velocity after.

But under actual conditions the gain of velocity is less than this, owing to blade friction, shock and other sources of loss. The actual velocity depends on the efficiency and on the shape and angles of the blades. It appears that under the conditions which hold in practice in Parsons turbines it is very nearly such that

*V12 = 2g(pv)*δ*p/p.*

In this formula, which serves as a means of estimating approxi­mately the velocity for purposes of design, it is to be understood that in. calculating the product *pv* the volume to be taken is that which is actually reached during expansion. The actual volume is affected both by friction and by leakage and is intermediate in value between the volume in adiabatic expansion and the volume corresponding to saturation. In the case of a turbine of 70 % efficiency the actual wetness of the steam is, according to Mr Parsons’s experience, about 55% of that due to adiabatic expansion in the early stages and 60% in the latest stages. In preparing the table of values of *pv* figures are accordingly to be taken intermediate between those for saturated steam and for steam expanded adia­batically, and from these is found as above the velocity for any given drop in pressure, and also the volume per pound, for which at each stage in the expansion provision has to be made in designing the effective areas of passage.

The blade speeds used in Parsons turbines rarely exceed 350 ft. per second and are generally a good deal less. In marine forms, where the number of revolutions per minute is limited by considera­tions of efficiency in the action of the screw propeller, the blade speeds generally range from about 120 to 150 ft. per second, though speeds as low as 80 ft. per second have been used.

127. *Parsons Marine Turbines.—*Marine turbines are divided into distinct high and low pressure parts through which the steam passes in series, each in a separate casing and each driving a separate propeller shaft. The most usual arrangement is to have three propeller shafts; the middle is driven by the high pressure portion of the turbine, and the steam which has done duty in this is then equally divided between two precisely similar low pressure turbines, each on one of two wing shafts. The rotor drum of each turbine has a uniform diameter through­out its length, but the casing is stepped to allow the lengths of the blades to increase as the pressure falls.

The casing which contains each of the two low pressure turbines contains also a turbine for running astern, so that either or both of the two wing shafts may be reversed. Steam is admitted to the reversing turbine direct from the boiler, the centre shaft being then idle. Each astern-driven turbine consists of a comparatively short series of rings of blades, set for running in the reversed direction, developing enough power for this purpose but making no pretensions to high efficiency. The astern turbine, being connected to the condenser, runs *in vacuo* when the ahead turbine is in use and consequently wastes little or no power.

Figs. 62 and 63 are sections of the high pressure and low pressure portions of a typical Parsons marine steam turbine, as designed for the three-shaft arrangement in which the low. pressure portion is duplicated. In each figure A is the fixed casing and B is the revolving drum. Steam enters the high pressure turbine (fig. 62) through J and passes out through H. There are 4 “ expansions ” or steps, with 9 stages or double rows of blades in the first, 9 in the