spindle at once becomes apparent. The factors affecting the deflection of a turbine-pump spindle are:—

Static

(a)—The weight of spindle and distribution of diameter change;

(b)—The weight and distribution of impellers, balancer, and parts;

(c)—The number and span of supporting bearings; and

Dynamic

(d)—In the dynamic condition, other incalculable forces entering into the account, such as, centrifugal forces due to the out-of-balance masses, and finally certain hydraulic disturbances.

(a)—Should be as light as possible consistent with the necessary stiffness; in common practice spindles are practically parallel the whole length.

(b)—Impellers and balancer should be grouped together as closely as possible, only allowing sufficient space for the water passages between the stages; also, the entire weight should be brought as close to the supporting bearings as possible.

(c)—The span of supporting bearings plays such an extremely important part in the durability of the pump; in the possibility and preservation of fine clearances; in the whirling of the shaft, and in the determination of the most economical size of spindle, that special notice will be taken of it.

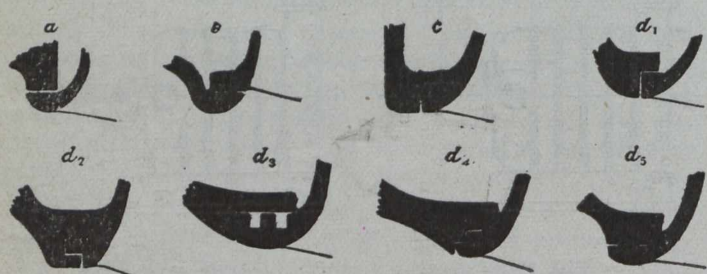

Fig. 31.—Types of Impellers

(d)—A loaded shaft supported horizontally between two bearings will "sag," and when rotated must suffer bending at every revolution. Also, there are bound to be certain out-of-balance masses in the rotor due to keys, heterogeneous composition of material, and the unavoidable variation in thickness of castings, etc. In addition, a shaft has vibrational periods due to its length and diameter, the whole question of vibration being intensely complicated by the loading and supports.

In addition to the disturbing factors mentioned above, vibrations are set up from the reaction of the impeller vane passing the guide vane. It is usual to reduce the intensity of this disturbance by arranging an odd number of vanes in the impeller relative to the guide vane, and by setting successive impellers on the shaft in such a way that the point of passing-vanes is progressive throughout the series, i.e., a "lead" is given. The resultant of these arrangements is to produce a vibrating influence, which may be torsional or combined torsional and transverse, of very high frequency and enfeebled intensity and which, when properly carried out, is free from practical disadvantage. The damping effect of the water film in neck bushes is appreciable in overcoming slight vibrations, as is conclusively shown by the different behavior of a spindle when running a pump dry and when running it filled with water. In order for a shaft to oscillate, it must force liquid out of one side of a neck-ring, and the retardation offered by this action has a noticeable effect. For this reason, quite apart from the lubricating effect gained, it is always a wise precaution to fill a pump with water before running it, otherwise if the shaft is not a stiff one seizing-up will probably occur. The remedy for spindle vibration due to any cause whatever is always a stiff shaft.

The whole matter of whirling shafts has been examined by several investigators, prominent among whom is Professor Dunkerley, but it is not proposed to discuss the matter further than to refer to the general proportions affecting the critical speeding of whirling. We find that the length (1) of span of bearings, and the diameter (d) of the spindle have an effect varying as:—

$$\frac{d^4}{l^3}$$

Turbine-pumps do not usually run at speeds approaching the critical speed, but the maximum safe speed for any rotor will bear a direct relation to the critical speed, in the sense that it will be an equal factor for equal safety. The above relation shows then, that for a fixed critical speed a reduction in the span of the bearings results in a material reduction in the necessary shaft diameter. The effect of axial thrust—always present in

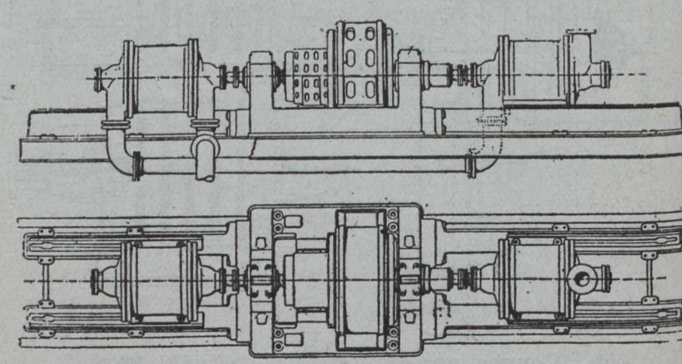

Fig. 32.—Bed for "Cylindrical" Type of Casing

a turbine-pump—in lowering the critical speed of a spindle should always be borne in mind. With a view to considering the effect of the spindle diameter and span of bearings on the degree of fineness of the running clearances, we will examine the extent of the deflections. The following arrangements of bearings are possible and are illustrated in Fig. 30:—

1.—Two outside bearings.

2.—One outside and one inside bearing.

3.—One outside and two inside bearings.

A typical example was taken, and it was found, after drawing out the different arrangements, that the ratios of the spans of the bearings supporting the loaded shaft were, for the three cases, 1.86, 1.44, and 1 respectively, calling the shortest span unity.

As the deflection of a circular spindle is proportional to $\frac{l_3}{d^4}$, the deflections themselves for the above spans are proportional to 6.5, 3, and 1 respectively, or, the necessary sizes of spindles to maintain the same deflections are proportional to 1.6, 1.3, and 1. These results are conveniently tabulated:—