Chapter XXX.—LINE SHAFTING.

LINE SHAFTING.—A line of shafting is one continuous run or length composed of lengths joined together by couplings. The main line of shafting is that which receives the power from the engine or other motor, and distributes it to other lines of shafting, or to the various machines to be driven. In some practice each line of shafting is driven by a separate engine or motor, so that it may be stopped without stopping the others. This same object may be obtained by providing a clutch for each line. It is obvious that in each line of shafting the length nearest to the driving motor transmits the whole of the power transmitted by the line, and that the diameter of the shafting may, therefore, be reduced as it proceeds from the engine in a proportion depending upon the degree to which the power it is required to transmit is reduced. It is desirable, therefore, so far as the shafting is concerned, to place the machines requiring the most power to drive as near as possible to that end of the shafting that receives power from the motor. Line shafting is supported in bearings provided in what are termed hangers, which are brackets to be bolted to either suitable framing, to walls, posts, or to the ceiling or floor of the building. The short lengths of shafting that are provided to effect changes of speed, and to enable the machine to be stopped or started at pleasure, are termed countershafts. When there is interposed a countershaft between the motor and the main line of shafting, it is sometimes termed a jack shaft.

Shafting is usually made cylindrically true either by special rolling processes as in what is known as "cold-rolled," or "hot-rolled" shafting, or else it is turned up in the lathe. In either case it is termed bright shafting. What is known as black shafting is simply bars of iron rolled by the ordinary process and made cylindrically true only where it receives its couplings, and for its journal bearings, &c. The diameter of black shafting varies by a quarter of an inch, and is usually above its designated diameter by about $\frac{1}{32}$ inch.

The main body of the shafting not being turned cylindrically true and parallel, the positions of the pulleys cannot be altered upon the shafts, nor can pulleys be added to the shaft as occasion may require without the sections being taken down and seatings turned for the required pulleys to be added. Furthermore black shafting does not run true, and is in this respect also objectionable. Nevertheless, black shafting is used for some special cases where extra pulleys are not likely to be required and the shafting is exposed to the weather, as in the case of yards for the manufacture of building bricks.

The diameters of bright or turned shafting (which is the ordinary form in which shafting is made, unless otherwise specified) vary by $\frac{1}{4}$ inch up to about $3\frac{1}{2}$ inches in diameter; but the actual diameter is $\frac{1}{16}$ inch less than the denominated commercial diameter, which is designated from the diameter of the round bar iron from which the shafting is turned; thus a length of what is known as 2-inch shafting will have an actual diameter of $1\frac{15}{16}$ inches, being parallel, or as nearly parallel as it is practicable to turn it in the ordinary lathe.

Cold-rolled shafting has its actual diameter agreeing with its designated or commercial diameter, and is parallel throughout its length.

In England the diameters of shafting vary by eighths of inches for diameters of an inch and less, and by quarters of an inch for diameters above an inch, the commercial and the actual diameters being alike.

The strains to which a line of shafting is subject are as follows: The torsional strain due to rotating the line of shafting, independent of the power transmitted; the torsional strain due to the amount of the power transmitted; and the transverse strain due to the unequal belt pressures and distances from the bearings of the driving or transmitting pulleys. The first and the last are, however, so intimately connected in practice that they may be considered as one: hence we have, 1st, the torsional strain due to driving the whole load, and, 2nd, the transverse strain due to the belt pressures being exerted more on one side than on another of the shaft, and to the belt pulleys being at unequal distances from the hanger bearings.

The first may be reduced to a minimum by so proportioning the strength of the line of shafting that it shall be capable of transmitting the required amount of power at the various sections of its length without suffering distortion of straightness beyond certain limits, and shall be at the same time as light as is consistent with this duty and a certain factor of safety.

Referring for a moment to the above limitation, the weight of the shaft itself will cause it to deflect between the hanger bearings, and the amount of this deflection will depend upon the distance apart of the points of support, or, in other words, of the distance apart of the hanger bearings.

The second may be reduced to a minimum by so regulating the distance apart of the hanger bearings that the deflection of the shaft from the belt pressures shall not be sufficient to produce sensible irregularities in the axis of rotation of the shaft; by so connecting the bearings to the hangers that they shall be rigidly held, and yet capable as far as possible of automatically adjusting their bores to be true with the shaft axis, notwithstanding its deflection from any cause; by placing the pulleys transmitting the most power as near to the hanger bearings as practicable; by so disposing the driving belts as to deliver the power as near as possible equally on all sides of the shaft; and by having the shafting and the pulleys balanced so as to run true, so that the strains on the pulleys shall be equal at each point in the shaft rotation. From this it appears that the distance apart of the shafting hangers may vary according to the amount of power transmitted by a shaft of a given diameter. The following table (given by Francis) gives the greatest admissible distances between the bearings of continuous shafts subject to no transverse strain except from their own weight, as would be the case were the power given off from the shaft equally on all sides, and at an equal distance from the hanger bearing.

Diameter of shaft in inches.	Distance between bearings, in feet.	
	Wrought-iron shafts.	Steel shafts.
2	15·46	15·89
3	17·70	18·19
4	19·48	20·02
5	20·99	21·57
6	22·30	22·92
7	23·48	24·13
8	24·55	25·23
9	25·53	26.24

These conditions, however, do not usually obtain in the transmission of power by belts and pulleys, and the varying circumstances of each case render it impracticable to give any rule which would be of value for universal application.

For example, the theoretical requirements would demand that the bearings be nearer together on those sections of shafting where most power is delivered from the shaft, while considerations as to the location and desired contiguity of the driven machines may render it impracticable to separate the driving pulleys by the intervention of a hanger at the theoretically required location. The nearer together the bearings the less the deflection either from the shaft's weight or from the belt stress, and since the friction of the shaft in its bearings is theoretically independent of the journal-bearing area, the closer the bearings the more perfect the theoretical conditions; but since it is impracticable to maintain the true alignment of the shaft, and as the friction due to an error in alignment would increase with the nearer proximity of the bearings, they are usually placed from about 7 to 12 feet apart, according to the facilities afforded in the location in which they are to be erected.

It is to be observed, however, that the nearer together the bearings are the less the diameter, and, therefore, the lighter the shafting may be to transmit a given amount of power, and hence the less the amount of power consumed in rotating the shafting in its bearings.

COLD-ROLLED SHAFTING.—This is shafting made cylindrically round and parallel by means of cold rolling, which leaves a smooth and bright surface. The effects of cold rolling upon the metal have been determined by Major Wm. Wade, U.S.A., Sir William Fairbairn, C.E., and Professor Thurston, of the ~~Stevens~~ Cornell Institute, as follows :—

The experiments were made upon samples of cold-rolled shafting submitted by Messrs. Jones and Laughlins, of Pittsburgh, Pennsylvania.

"It is made exactly to gauge diameter, and for many purposes requires no further preparation.

"The cold-rolled metal resists stresses much more uniformly than does the untreated metal. Irregularities of resistance exhibited by the latter do not appear in the former; this is more particularly true for transverse stress.

"This treatment of iron produces a very important improvement in uniformity of structure, the cold-rolled iron excelling common iron in density from surface to centre, as well as in its uniformity of strength from outside to the middle of the bar.

"This great increase of strength, stiffness, elasticity, and resilience is obtained at the expense of some ductility, which diminishes as the tenacity increases. The modulus of ultimate resilience of the cold-rolled iron is, however, above 50 per cent. of that of the untreated iron.

"Cold-rolled iron thus greatly excels common iron in all cases where the metal is to sustain maximum loads without permanent set or distortion."

SUMMARY OF THE RESULTS OBTAINED BY MAJOR WADE FROM NUMEROUS EXPERIMENTS WITH ORDINARY HOT-ROLLED BAR IRON, COMPARED WITH THE RESULTS OBTAINED FROM THE SAME KINDS OF IRON ROLLED AND POLISHED WHILE COLD BY LAUTH'S PATENT PROCESS.

		Iron rolled while		Ratio of increase by cold rolling.	Average rate per cent. of increase.
		Hot.	Cold.		
TRANSVERSE.—Bars supported at both ends; load applied in the middle; distance between the supports, 30 inches. Weight which gives a permanent set of one-tenth of an inch, viz.	1½ inch square bars	3,100	10,700	3·451	162½
	Round bars, 2 inch diameter	5,200	11,100	2·134	
	Round bars, 2¼ ,, ,,	6,800	15,600	2·294	
TORSION.—Weight which gives a permanent set of one degree, applied at 25 inches from centre of bars. Round bars, 1¾ inch diameter, and 9 inches between the clamps		750	1,725	2·300	130
COMPRESSION.—Weight which gives a depression, and a permanent set of one-hundredth of an inch to columns 1½ inches long and ⅝ inch diameter		13,000	34,000	2·615	161½
Weight which bends and gives a permanent set to columns 8 inches long and ¾ inch diameter, viz.	Puddled iron	21,000	31,000	1·476	64
	Charcoal bloom iron	20,500	37,000	1·804	
TENSION.—Weight per square inch, which caused rods ¾ inch diameter to stretch and take a permanent set, viz.	Puddled iron	37,250	68,427	1·837	95
	Charcoal bloom iron	42,439	87,396	2·059	
Weight per square inch, at which the same rods broke, viz.	Puddled iron	55,760	83,156	1·491	72
	Charcoal bloom iron	50,927	99,293	1·950	
HARDNESS.—Weight required to produce equal indentations		5,000	7,500	1·500	50

NOTE.—Indentations made by equal weights, in the centre, and near the edges of the fresh cut ends of the bars, were equal; showing that the iron was as hard in the centre of the bars as elsewhere.

GENERAL SUMMARY OF THE RESULTS OBTAINED BY SIR WILLIAM FAIRBAIRN'S EXPERIMENTS.

	Condition of bar.	Breaking weight of bar in lbs.	Breaking weight per square inch.		Strength, the untouched bar being unity.
			In lbs.	In tons.	
1	Untouched (black)	50,346	58·628	26·173	1·000
3	Rolled cold	69,295	88·230	39·388	1·505
4	Turned	47,710	60·746	27·119	1·036

NOTE.—In the above summary it will be observed that the effect of consolidation by the process of cold rolling is to increase the tensile powers of resistance from 26·17 tons per square inch, to 39·38 tons, being in the ratio of 1 : 1·5, one-half increase of strength gained by the new process of cold rolling.

Extract from the general conclusions arrived at by Professor R. Thurston from experiments.

"The process of cold rolling produces a very marked change in the physical properties of the iron thus treated.

"It increases the tenacity from 25 to 40 per cent., and the resistance to transverse stress from 50 to 80 per cent.

"It elevates the elastic limit under torsional as well as tensile and transverse stresses, from 80 to 125 per cent. . . .

"It gives the iron a smooth bright surface, absolutely free from the scale of black oxide unavoidably left when hot rolled.

From this it appears that cold-rolled iron is peculiarly adapted for line shafting. Suppose, for example, a given quantity of power to transmit, and that a length of cold-rolled and a length of hot-rolled iron be connected together to form the line. Then the diameters of the two being such as to have equal torsional strength, we have—

1st. That the weight of the cold rolled will be the least, and it will, therefore, produce less friction in the hanger bearings.

2nd. That the cold rolled will be harder, and will therefore suffer less from abrasion of the journals.

3rd. That being of smaller diameter the journals are more easily and perfectly lubricated.

The resistance to transverse stress (say) 50 per cent.; but the elastic limit under transverse stress is increased from 80 to 125 per cent., accepting the lesser amount we have in the case of the two shafts.

4th. That the resistance to permanent set or bend will be 30 per cent. more in the cold rolled.

5th. The accuracy to gauge diameter enables the employment of a coupling having a continuous sleeve, and gives an equal bearing along the entire coupling bore.

6th. The reduction of shaft diameter enables the employment of a smaller and lighter coupling; and

7th. The hubs of the pulleys may be made smaller and lighter, are easier to bore, and may be bored to gauge diameter with the assurance that they will fit the shaft.

The friction between the journals of a line shafting and its bearings depends so intimately upon the distance apart of the bearings, upon the alignment of the same, upon the accurate bedding of the shaft journals to the bearings, and upon the amount of transverse strain; and this latter is so influenced by the amount of power that may be delivered from one side of the shaft more than from another, that the application of rules for determining the said friction under conditions of perfect alignment rigidity would be practically useless. The conditions found in actual practice are so widely divergent and so rarely alike, or even nearly alike, that the consideration of this part of the subject would, in the opinion of the author, be of no practical value. The reader, however, is referred to the remarks made with reference to the friction of journals.

To prevent end motion to a line of shafting it is necessary that there be fixed at some part of the line two shoulders, or collars, on relatively different sides of a bearing, or of the bearings, these collars meeting the side faces of the bearing. If shoulders are produced by reducing the diameter of the journal bearing of the shaft, the strength of the shafting is reduced to that at the reduced bearing, because the strength of the whole can be no greater than its strength at the weakest part. If collars are placed one on each end of the line of shafting, the difficulty is met that the collars will permit end motion to the shaft whenever the temperature of the shaft is greater than that which obtained at the time at which the collars were adjusted, which occurs on account of the increased expansion of the shaft. On the other hand the collars will bind against the side faces of the bearing boxes whenever the shaft is at a lower temperature than it was at the time of setting the shaft, because of the contraction of the shaft's length, and this would cause undue friction, abrasion, and wear.

It is preferable, therefore, to place such collars one on each side of one bearing, so that the difference in contraction and expansion from varying temperatures shall be confined to the difference in expansion between the metal of which the bearing and shaft respectively are composed in the length of the bearing only, instead of being extended to the difference in expansion between the shaft throughout its whole length and that of the framework to which the hangers, or bearings, are bolted.

The collars may be shrunk on to the shaft so as to avoid the necessity of set-screws, or if set-screws are used they should be as short as is practicable so as to avoid the liability to catch against the lacings, &c., of belts, which, on slipping off the pulley may come into contact with the set-screw head. The Lane and Bodley Co., of Cincinnati, employ a collar (for loose pulleys, &c.) in which the radius of the collar for a width equal to the diameter of the set-screw head, is equal to that of the set-screw head thus projecting from

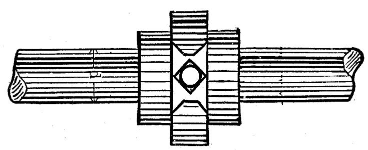

Fig. 2593.

the centre of the collar circumference, a slot in the ring affording access to the set-screw head, as shown in Fig. 2593. By this means the head of the set-screw is protected from contact with a belt, in case the latter should be off the pulley and resting upon the shaft.

As a rule it is preferable that the collars, to prevent end motion to the shaft, be placed at the bearing nearest to the engine or motor; and this is especially desirable where bevel-wheels are employed to drive the shaft, because in that case the pitch lines of the wheels are kept to coincide as nearly as practicable, and the teeth are prevented from getting too far into or out of gear.

Diameters of Line Shafting.—The necessary diameters of the various length of the shafts composing a line of shafting, should be proportioned to the quantity of power delivered by each respective length, and in this connection the position of the various pulleys upon the length and the amount of power given off by the pulley is an important consideration. Suppose, for example, that a piece of shafting delivers a certain amount of power, then it is obvious that the shaft will deflect or bend less if the pulley transmitting that power be placed close to a hanger or bearing than if it be placed midway between the two hangers or bearings.

The strength of a shaft to resist torsion is the cube of its diameter in inches, multiplied by the strength of the material of which the shaft is composed, per square inch of cross-sectional area, giving the strength in statical foot-pounds. The application of this rule is to find the necessary strength of the shaft to convey power irrespective of the distance from its centre at which it delivers such power.

But since the point at which the power to produce torsion is applied is at the rim of the pulley, the amount of torsion produced upon a shaft by a given stress must be obtained by multiplying the given amount of stress by the radius of the pulley in inches and parts of an inch. Example: the static stress upon a pulley, 24 inches diameter, is 100 lbs., what static torsion does it exert upon the shaft?

Here, stress 100×12 (radius of the pulley) $= 1200 =$ static torsional stress.

In the following rules for finding the necessary diameters and strengths of shafts, the margin of extra diameter for strength necessary for safety is included, so that the given sizes are working diameters.

To find the necessary diameter of shaft from a given torsional stress.—Rule, divide the torsional stress expressed in statical foot lbs., by 57·2 for steel, by 27·7 for wrought iron, or by 18·5 for cast iron, and the cube root of the quotient is the required working diameter of shaft expressed in inches.

To find the maximum amount of horse-power capable, within good working limits, of being transmitted by a *shaft* of a given diameter.—Rule, multiply the cube of the diameter of the shaft, in inches, by its revolutions per minute and divide by 92 for steel, by 190 for wrought-iron, or by 285 for cast-iron shafts, and the quotient is the amount of horse-power.

Since, in this rule, the horse-power is a given quantity, the diameter of the pulley is of no consequence, since with a given stress it must have been taken into account in obtaining the horse-power.

To find the revolutions per minute a shaft will require to make to transmit a given amount of horse-power.—Rule, multiply the given amount of horse-power by 92 for steel, by 190 for wrought-iron, or by 285 for cast-iron shafts, and divide the product by the cube of the diameter of the shaft expressed in inches, and the quotient is the required revolutions per minute for the shaft.

The rule adopted by William Sellers and Co. to determine the size of shafts to transmit a given horse-power is:—Rule, divide the cube root of the horse-power by the revolutions per minute and multiply the quotient by 125, the product is the diameter of shaft required.

This gives a shaft strong enough to resist flexure, if the bearings are not too far apart. The distance apart that the bearings should be placed is an important consideration. Modern millwrights differ slightly in opinion in this respect: some construct their mills with beams 9 feet 6 inches apart, and put one hanger under each of the beams; others say 8 feet apart gives a better result. We are clearly of opinion that with 8 feet distance, and shafting lighter in proportion, the best result is obtained.

The following table (from "Machine Tools," by Wm. Sellers and Co.) gives the strength of round wrought iron as given by Clark:—

TABLE SHOWING STRENGTH OF ROUND WROUGHT-IRON SHAFTING.

Diameter of shaft.	Torsional Action.					Transverse Action.		
						Under the gross distributed weight.		Under the net weight of shaft.
	Ultimate resistance.	Working stress.	Work for one turn per minute.	Horse Power at the rate of one turn per minute.	Speed in turns per minute for one-horse power.	Distance of bearings for the limiting deflection.	Gross weight for the span.	Distance of bearings for the limiting deflection.
1	2	3	4	5	6	7	8	9
Inches.	Stat'l ft. tons.	Stat'l ft. lbs.	Ft. lbs.	H. P.	Turns.	Feet.	Lbs.	Feet.
1	·42	27·7	174	·00526	190	6·6	30	7·9
1¼	·82	54·1	340	·01028	97·3	7·7	55	9 2
1½	1·42	93·5	587	·01779	56·2	8·6	89	10·3
1⅝	1·80	118·9	746	·02259	44·3	9·2	112	11·0
1¾	2·25	148·4	932	·02820	35·4	9·6	134	11·5
1⅞	2·77	182·6	1,147	·03469	28·8	10·1	163	12·1
2	3·36	221·6	1,391	·04211	23·7	10·5	193	12·7
2⅛	4·00	265·8	1,669	·05062	19·8	11·0	227	13·2
2¼	4·80	315·5	1,981	·05995	16·7	11·4	264	13·7
2⅜	5·62	371·1	2,330	·07051	14·2	11·8	305	14·2
2½	6·56	432·8	2,718	·08224	12·2	12·5	359	15·0
2¾	8·73	576·1	3,618	·1094	9·14	13·0	450	15·6
3	11·3	747·9	4,697	·1421	7·04	13·7	566	16·5
3¼	14·4	951·0	5,972	·1807	5·54	14·5	701	17·4
3½	18·0	1,188	7,458	·2257	4·43	15·2	854	18·3
3¾	22·1	1,461	9,173	·2775	3·60	16·0	1,029	19·2
4	26·9	1,773	11,136	·3368	2·97	16·7	1,225	20·1
4¼	32·2	2,127	13,345	·4040	2·48	17·4	1,439	20·9
4½	38·2	2,524	15,851	·4796	2·09	18·1	1,679	21·7
4¾	45·0	2,969	18,635	·5642	1·77	18·8	1,943	22·6
5	52·5	3,463	21,750	·6579	1·52	19·4	2,220	23·3
5¼	60·7	4,008	25,177	·7616	1·31	20·0	2,525	24·0
5½	69·8	4,609	28,936	·8758	1·14	20·6	2,854	24·7
5¾	79·8	5,266	33,077	1·000	1·00	21·2	3,210	25·4
6	90·6	5,983	37,584	1·137	·880	21·6	3,600	26·2
6½	117	7,606	47,780	1·445	·692	22·9	4,421	27·5
7	144	9,501	59,682	1·805	·554	24·2	5,426	29·0
7½	177	11,680	73,254	2·220	·450	25 3	6,518	30·4
8	215	14,180	89,088	2·694	·371	26·5	7,774	31·8
8½	258	17,010	106,836	3·232	·309	27·6	9,133	33·1
9	306	20,190	126,846	3·837	·261	28·7	10,650	34·4
9½	360	23,750	149,118	4·512	·222	29·8	12,320	35·7
10	420	27,700	174,000	5·260	·190	30·8	14,100	36·9
11	559	36,870	231,594	7·005	·143	32·8	18,180	39·4
12	725	47,860	300,672	9·095	·110	34·7	22,880	41·7
13	922	60,860	382,278	11·83	·0865	36·6	28,330	44·0
14	1,152	76,010	477,456	14·44	·0693	38·5	34,560	46·2
15	1,417	93,490	587,250	17·76	·0563	40·3	41,530	48·4
16	1,720	113,500	712,704	21·56	·0464	42·1	49,330	50·5
17	2,062	136,100	854,862	25·85	·0387	43·3	57,970	52 6
18	2,447	161,500	1,014,768	30·69	·0326	45·5	67,490	54·6
19	2,880	190,000	1,193,466	36·10	·0277	47·2	78,040	56·6
20	3,360	221,600	1,392,000	42·11	·0237	48·8	80,660	58·5

NOTE.—To find the corresponding values for shafts of cast iron or steel, multiply the tabular values by the following multipliers:

Cast iron .	$\frac{2}{5}$	$\frac{2}{3}$	$\frac{2}{3}$	$\frac{2}{3}$	1·5	·86	·81	·86
Steel .	1 2	2·06	2·06	2·06	·48	1·05	1·07	1·05

"It is advantageous that the diameter of line shaft be kept as small as is possible with due regard to the duty, so as to avoid extra weight in the shafting hangers, pulley hubs and couplings, whose weights necessarily increase with the diameter of the shafting.

"SPEEDS FOR SHAFTING.—The speed at which shafting should run is determined within certain limits by the kind of machinery it is employed to drive. Shafting to drive wood-working machines may, for example, be made to rotate much faster than that employed to run metal-cutting machines, because the motions in the wood-working machines themselves are faster than those in metal-cutting machines. In a general sense, the rotation of shafting is greater in proportion as the movements of the machines driven require to run faster.

"This occurs because in proportion as the driving pulleys of the machines require to rotate faster than the line shaft, the diameters of the pulleys on the line shaft must be larger than the diameters of those on the machines; hence a great variation in speed would demand a corresponding increase of diameter of pulley on the line shaft, and the extra weight of this pulley would be so much added to the weight causing friction, as well as so much added to the cost. If small pulleys were used and countershafts employed to multiply the speed the cost would be increased, extra room would be taken up; indeed, this is so obvious as to require no discussion, further than to remark that the faster the shafting rotates the smaller may be its diameter to transmit a given horse-power. From deflection and weakness to resist transverse strains and other obvious causes it is not found in practice desirable to employ line shafts of less than about 1¼ inches in diameter, and the diameters of shafting employed are usually arrived at from a calculated speed of about 120 revolutions per minute for metal-cutting machines such as used in machine shops, 250 revolutions per minute for wood-working machines, and from 300 to 400 revolutions per minute for cotton and woollen mills, and the countershafts for the machines usually have pulleys of the requisite diameters to convert this speed of rotation into that required to run each respective machine. Tubular or hollow shafting has been made to run at 600 revolutions per minute, but this kind of shafting has been of very limited application because of its expensiveness.

"It is obvious that since the speed of a line shaft is used as a multiplier in the calculation of the horse-powers of shafts, a given diameter of shaft will transmit more power in proportion as its speed is increased. Thus a shaft capable of transmitting 20 horse-power when making 120 revolutions per minute will transmit 40 horse-power if making 240 revolutions per minute.

"There are now running in some factories lines of shafting 1,000 feet long each. The power is generally applied to the shaft in the centre of the mill and the line extended each way from this. The head shaft being, say, 5 inches in diameter, the shafts extending each way are made smaller in proportion to the rate of distribution, so that from 5 inches they often taper down to $1\frac{3}{4}$.

"When very long lines of shafting are constructed of small or comparatively small diameter, such lines are liable to some irregularities in speed, owing to the torsion or twisting of the shaft as power is taken from it in more or less irregular manner. Shafts driving looms may at one time be under the strain of driving all the looms belted from them, but as some looms are stopped the strain on the shaft becomes relaxed, and the torsional strain drives some part of the line ahead, and again retards it when the looms are started up. This irregularity is in some cases a matter of serious consideration, as in the instance of driving weaving machinery. The looms are

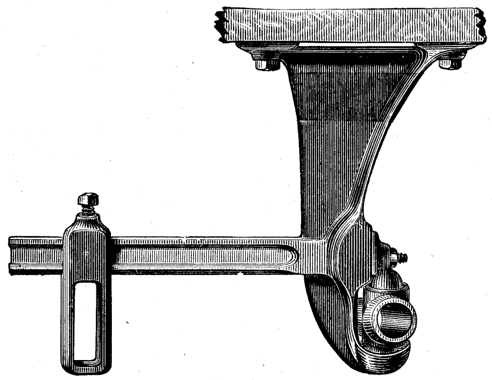

Fig. 2594.

provided with delicate stop motion, whereby the breaking of a thread knocks off the belt shifter and stops the loom. An irregular driving motion is apt to cause the looms to knock off, as it is called, and hence the stopping of one or more may cause others near to them to stop also. This may in a measure be arrested by providing fly-wheels at intervals on the line shaft, so heavy in their rim as to act as a constant retardant and storer of power, which power is given back upon any reaction on the shaft, and thus the strain is equalized. We mention this, as at the present time it is occupying the thoughts of prominent millwrights, and the relative advantage and disadvantage of light and heavy fly-wheels are being discussed, and is influencing the proportions of shafting in mill construction.*"

Countershafts are separate sections of shafting (usually a short

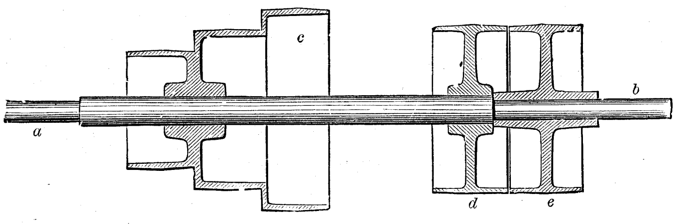

Fig. 2595.

section) employed to increase or diminish belt speed, to alter the direction of belt motion, to carry a loose as well as a fast pulley (so that by moving the belt on to the loose pulley it may cease to communicate motion to the machine driven), and for all these purposes combined.

An excellent form of countershaft hanger is shown in Fig. 2594, the guide for the slide being adjustable along the arm, and fixed in its adjusted position by means of the set-screws. The bearing is self-adjusting horizontally for alignment. The countershaft is shown in Fig. 2595, *a b* being the bearings, *c* the cone pulley, *d* the fast and *e* the loose pulley, which is placed next to the bearing, so that it may be oiled without having to reach past the belt and fast pulley. By reducing the journal for the loose pulley no collar is needed, the shaft shoulder and the face of the bearing serving instead.

* From "Machine Tools," by William Sellers and Co.

When the direction of rotation of the cone pulley on the countershaft requires to be occasionally reversed, there are two belts, an open one and a crossed one, from the line shaft to the countershaft, and there are three pulleys on the countershaft, their arrangement being as shown in Fig. 2596. L L′ are two loose pulleys, one receiving the open and the other the crossed belt, both these pulleys being a little more than twice the width of the belt; F is a fast pulley. By operating the belt skipper or shifter in the requisite direction either the open or the crossed belt is brought upon the fast pulley, the other belt merely moving across the width of its loose pulley, which must be twice that of the fast one. In the position of the belt shifter shown in the cut, both belts would be upon the loose

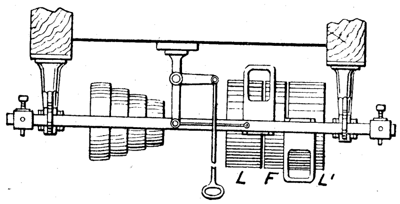

Fig. 2596.

pulleys L L′, hence the countershaft would remain at rest. If the direction of rotation of one pulley is required to be quicker than the other, two fast pulleys, each slightly more than twice the width of the belt, may be placed upon the line shaft, one of them being of enlarged diameter, to give the requisite increased velocity.

In Fig. 2597 Pratt's patent friction clutch is shown applied to a countershaft requiring to rotate in both directions, but quicker in one direction than in the other; hence, one of the pulleys is of smaller diameter than the other. The pulleys are free to rotate upon the countershaft unless engaged by the clutch, which is constructed as follows:—

The inside surface of the pulley rim is bored and the end surface of the shoes is turned to correspond. The shoes are in the form of a bell crank, upon the exposed end of which is provided a small lug, clearly shown in the cut. To prevent end motion of the pulley a collar is placed on one side of it and secured to the countershaft, while, on the other, the sleeve to which the shoes are pivoted is also secured to the countershaft; upon the shaft between the two pulleys there is a sleeve, having at each end a conical hub. When this sleeve is moved to the right, its right-hand coned hub passes between the lugs on the exposed ends of the shoes, forcing these lugs apart and causing the shoes to grip the bore of the large pulley, which thereupon rotates the shaft through the medium of the sleeve upon which the shoes are pivoted. Similarly, if the engaging (and disengaging) sleeve be moved to the left it will pass between the

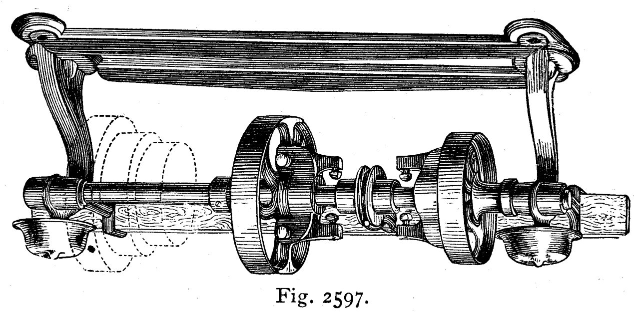

Fig. 2597.

lugs of the shoes on the left-hand pulley, which will, therefore, be caused to drive the shaft. In the position shown in the cut the engaging sleeve is clear of the ends of all the shoes, hence the pulleys would be caused to rotate (by their belts), but the shafts, &c., would remain stationary.

In yet another form the inner face of the pulley rim is coned, and in place of shoes a disk, whose circumference is coned to fit the pulley rim, is fast upon the shaft. The shaft is provided with a fixed collar, and from this collar, as a fulcrum, the pulley and disk are (by means of short levers attached to a sleeve upon the countershaft) brought into contact, the thrust on the other side of the pulley being sustained by a conical surface on the sleeve, fitting to a similar cone on the hub of the pulley. Thus the pulley is gripped between two coned surfaces, one on each side, and is released by moving the sleeve laterally so as to relieve the grip, which it does noiselessly.

By this means motion to the shaft is communicated from the pulley without the sudden shock incidental to the impact of two fixed pieces, because the grip of the cones is gradual, and a certain amount of slip may occur until such time as the grip of the surfaces is sufficient to drive by friction.

Fig. 2598* represents a cone friction clutch pulley. The outer half is a working fit upon the shaft, but is secured against end motion by the collar D. The sliding half is coned and covered with leather as shown at C C, the outer half being coned to correspond. The sliding half is driven by a feather fast in its bore, and sliding in a feather-way or spline in the shaft.

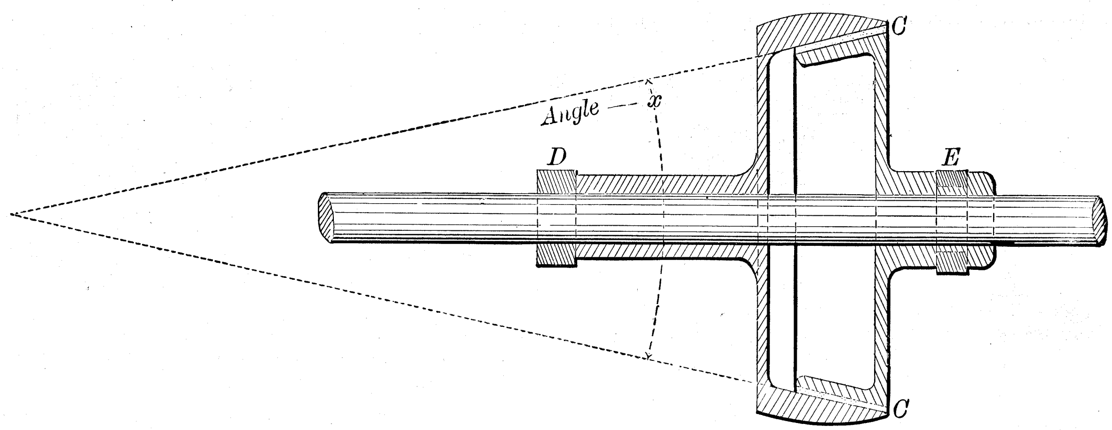

Fig. 2598.

The driving power of the device is obtained by means of the friction of the coned surfaces. The less the angle x of the cones the more power transmitted with a given pressure of the internal to the external cone.

On the other hand, however, this angle may be so little that the external cone will not release the internal one when the end pressure on the latter is removed.

The object is, therefore, to so proportion the angle x of the cones that their friction will be a maximum, while the internal cone may be moved endwise and unlocked from the external without undue effort or strain at the moving clutch bar E. If the angle be 30 degrees, the clutch will release itself when the lateral pressure is removed. If the angle be 25 degrees the internal cone will require a slight lateral pressure to release it. If the angle be 20 degrees, the internal cone cannot be released by end pressure applied by hand.

The transmitting capacity of the clutch depends upon the pressure applied to maintain the cones in contact, and therefore upon the leverage of the clutch bar, whose fork end is shown in section at E.

It is desirable that the end pressure be as small as possible, because of the friction between E and the hub of the sliding half of the pulley.

The hangers which carry the bearing boxes supporting shafting may be divided into four principal classes :—Those in which the bearing boxes are permitted to swivel, and to a certain extent to adjust themselves, to the axial line of the shafting, and having means to adjust the vertical height of the bolts.

Those in which the bearings are incapable of such adjustments.

Those in which the bearing boxes are supported on each side; and those in which the bearing is supported on one side only, so that the shafting may be taken down without disturbing the couplings.

The first named are desirable in that they eliminate to a certain extent the strains due to the extra journal bearing friction which occurs when the shafting is sprung out of its true alignment, and obviate to a great extent the labor involved in fitting the bore of the bearing boxes to the journals of the shafting, so as to hold the same with its axis in a straight line, while they permit of vertical movement to attain vertical alignment.

Fig. 2599 represents Wm. Sellers & Co.'s ball-and-socket hanger which has come into extensive use throughout the United States: *a* represents the frame of the hanger threaded to receive the cylindrical threaded plungers *d e*, which therefore by rotation advance or recede respectively from the centre of the bearing boxes *b c*.

The ends of these plungers are concave, and the top and bottom halves of the bearing boxes are provided with a spheroidal section fitting into the concaves of the plungers, so that when the plungers are adjusted to fit (a working fit) against the boxes, the latter are held in a ball-and-socket or universal joint, which permits motion in any direction, the centre of such motion being central to the spherical concaves on the ends of plungers *e d*.

To adjust the vertical height of the bearings or boxes, it is simply necessary to rotate the plungers *d e*, in the threaded holes in the frame. F is simply a dish to catch the lubricating oil after it has passed through the bearing.

It is obvious that if a shaft be aligned axially true, and held in a box of this design, the centre of a length of shaft on either side

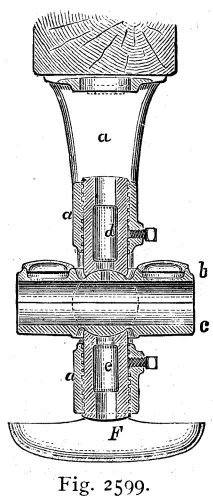

Fig. 2599.

of the box may be sprung or deflected out of alignment, and that the box will adjust itself so that its bore will be parallel with the axis of the shaft thus deflected, hence the friction between the shaft journal and the bearing box will be at all times a minimum.

This feature of self-adjustment permits of the employment of longer bearings, which reduces the wear, as well as the friction, and by providing sufficient bearing and wearing area, enables the bearings to be composed of cast iron, which is the cheapest as well as the very best material of which a bearing can be made, provided that its area of bore is sufficiently large in proportion to

* From *The American Machinist.*

the duty, or load, to have a pressure of not more than about 60 lbs. per square inch of area.

Again, if the alignment of the shaft should require readjustment from the warping or sinking of beams, as is a very common occurrence where hangers are fixed to the joists of ceilings, the adjustment is readily and easily effected by means of the plungers, nor need the boxes be fitted to the shaft more than to see that when

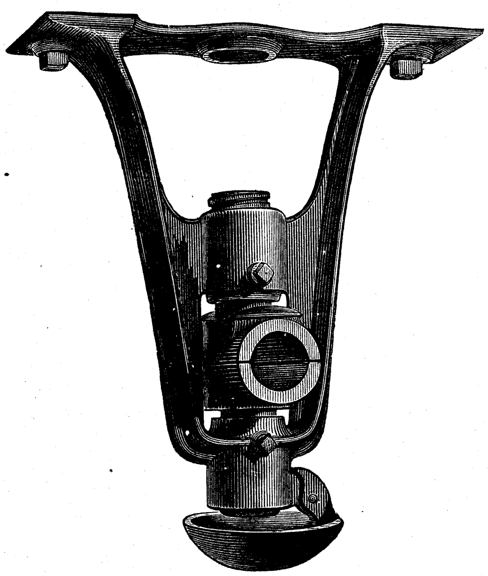

Fig. 2600.

free from the hangers they bed firmly down until the crowns of their bore have contact with the shaft. The hangers themselves require no refinement of alignment, because that may be secured by means of the plungers, and the boxes require no fitting to the shafts after the hangers are erected.

In hangers in which the self-adjusting ball-and-socket feature

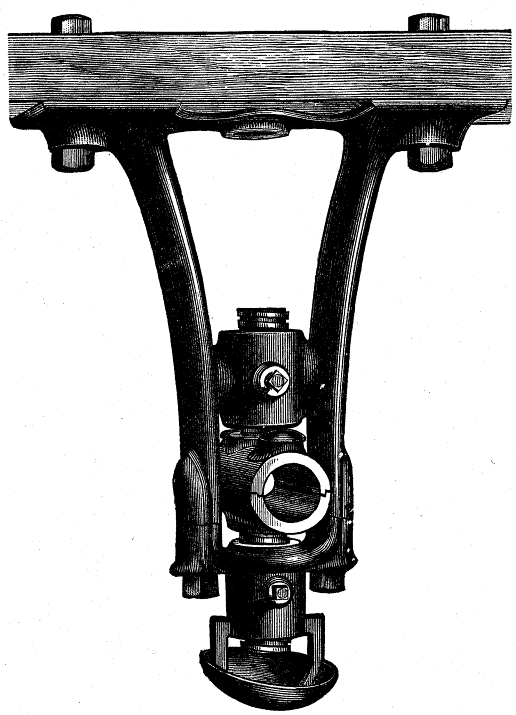

Fig. 2601.

is omitted, the bottom hangers must not only be accurately aligned, but the boxes must, to avoid friction and undue wear, all be fitted to the shaft, and the latter must, during such fitting, be tried in the boxes; the operation, if properly performed, costing far more in labor than is equivalent to the difference in the first cost of the ball-and-socket adjustable hangers and those solid or not self-adjustable, especially if the boxes be long ones, as about, or not less than, three times the diameter of the shaft, as they should be.

An external side elevation of this hanger is shown in Fig. 2600,

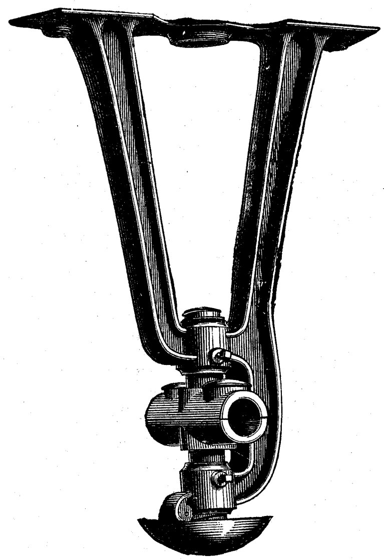

Fig. 2602.

it being obvious that the hanger is designed for bolting to timbers, or framing overhead.

Fig. 2601 represents a hanger of this class. In this the lower part carrying the bottom bearing is held to the upper by two bolts, as shown, the object being to enable the same to be placed in position on a line of shafting without disturbing the pulleys or the couplings. The lower section with the bottom bearing is removed and again put on after the hanger is set over the shaft.

Fig. 2602 represents an open-sided ball-and-socket hanger in which the plungers can be retired, the bearings removed, and the hanger erected on an existing line of shafting without

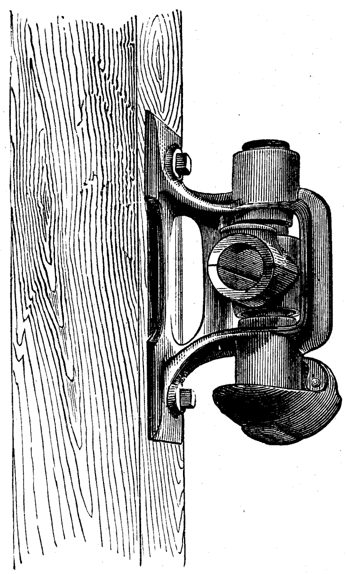

Fig. 2603.

removing the pulleys or couplings, or disturbing the line of shafting.

When the face of the framing to which the hangers are to be bolted stands vertical, the hangers are formed as in Fig. 2603, in which the ball-and-socket or swivelling feature is maintained as before.

Fig. 2604 represents a wall hanger, which is open in front similar to the hanger shown in Fig. 2602, and for the same purpose.

The section of shafting receiving power from the engine or prime mover is usually supported in bearings or pillow blocks.

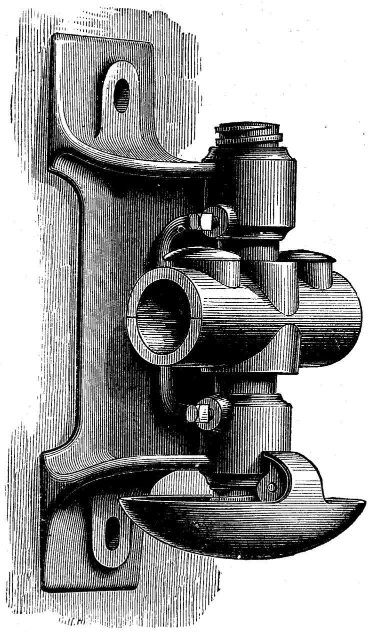

Fig. 2604.

Pillow blocks are also used for vertical shafts, and in cases where the foundation or framing is not liable to lose correct horizontal adjustment.

Fig. 2605 represents a pillow block, in which the ball-and-socket principle shown in Fig. 2602 is embodied. The bearings

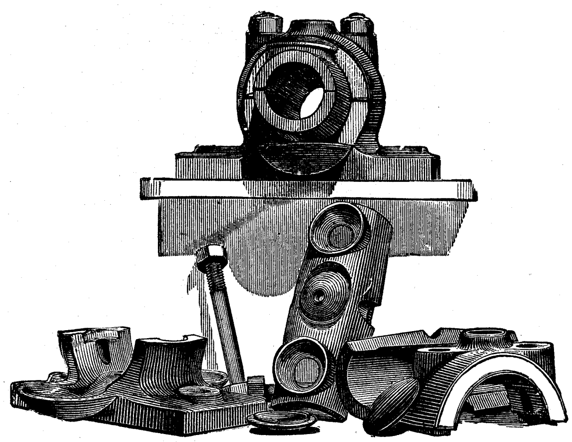

Fig. 2605.

have each a ball section fitting into spherical recesses or cups provided in the body of the block, and in the cap, so that the bearings are capable of swivelling as already described with reference to the hanger Fig. 2599.

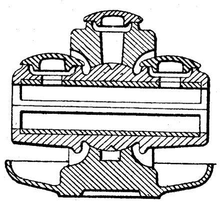

Fig. 2606.

A sectional view of a pillow block having this adjustable feature is shown in Fig. 2606. To provide increased seating bearing, and also means of side adjustment to pillow blocks, they are sometimes bolted to base plates as in Fig. 2607, room being left in the bolt holes to permit of their being moved and adjusted upon the plate. The adjustment may be made by means of wedges, as at A, B in Fig. 2607. These base plates are usually employed when the pillow block is to be held against a wall.

An inverted pillow block of similar construction, but designed for the head line (as the length receiving power from the engine

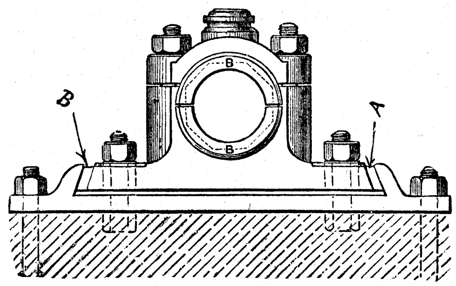

Fig. 2607.

or motor is termed) of the shafting, is shown in Fig. 2608, but an improved form of the same has plungers so as to effect a vertical adjustment of the bearings.

When a pillow block requires to be enveloped by a wall it is provided with a wall box as shown in Fig. 2609, and within this box is set the pillow block as shown, space being sometimes left

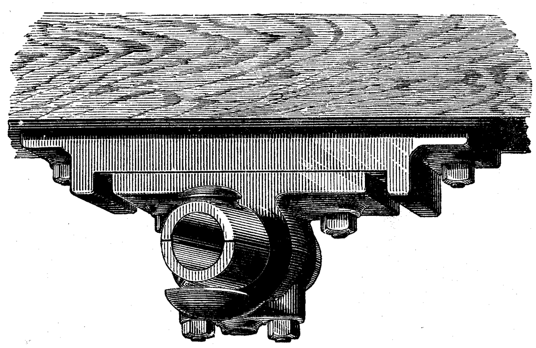

Fig. 2608.

to adjust the pillow block laterally within the box by means of a wedge as shown.

In cases where the shafting requires to stand off from a wall to allow room for the pulleys, brackets or knees, such as shown in Fig. 2610, are employed.

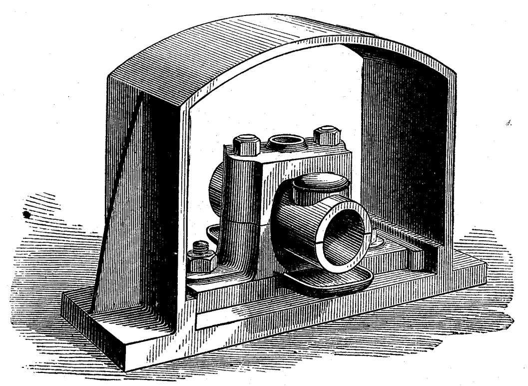

Fig. 2609.

COUPLINGS FOR LINE OR DRIVING SHAFTS.—The couplings for connecting the ends of line shafts should accomplish the following objects :—

1. To hold the two shaft ends axially true one with the other.

2. To have an equal grip along the entire length of shaft enveloped by the coupling.

3. To have a fastening or locking device of such a nature that

it will not be liable to work loose from the torsional strains due to the flexure of the shaft, which is caused by the belts springing or straining the axial line of shafting out of the straight line.

4. To be capable of easy application and removal, so as to permit the erection or disconnection of the lengths of shafting

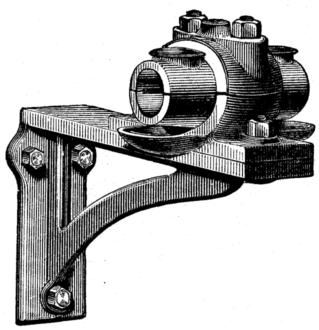

Fig. 2610.

with as little disarrangement of the hangers and bearings as possible, and to be light, run true, and be balanced.

To these requirements, however, may be added that, since it is well-nigh impracticable to obtain lengths of lathe-turned shafting of exactly equal diameter, couplings for such shafting require to fill the following further requirements :

5. The piece or pieces gripping the shaft ends must be capable of concentric and parallel closure along the entire area, enveloping the end of each shaft, and must do this at each end indepen-

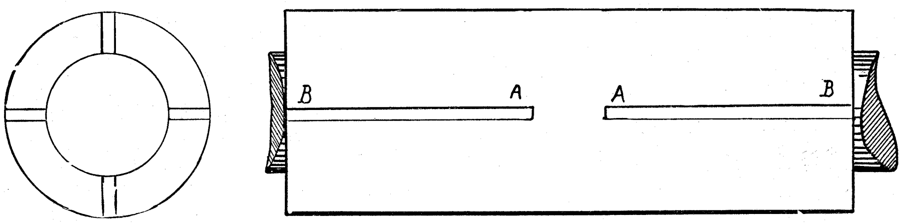

Fig. 2611.

dently of the other, and the piece or pieces exerting the closing or compressing pressure must grip the closing piece or pieces, enveloping the shafting over the entire area.

If, for example, a sleeve be split at four equidistant parts of its circumference, and from each end nearly to the middle of its length, as in Fig. 2611, any pressure that may be applied to its circumference to cause it to grip the shaft it envelops will cause it to grip the shaft with greater force at one part than at

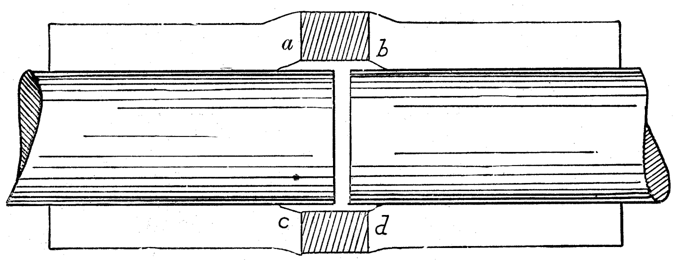

Fig. 2612.

another, according to the diameter of the shaft and the location of the external pressure. Thus, if the pressure be applied equally along the length A B, the weaker end B will close most readily, while at A the support afforded by the unsplit section offers a resistance to closure at the ends A of the split, hence the shaft, even though a working fit to the sleeve bore, will be gripped with least force at the end A. If the shaft were simply a close fit, as, say, just movable by hand on the sleeve bore, the form of the coupling bore would, when compressed upon the shaft, be as shown in Fig. 2612, the bend on the necks *a*,*b*,*c*,*d*, being magnified for clearness of illustration. If the compressing piece covered the compressed sleeve for a lesser distance, the grip would be more uniform, because there would be a greater length of the sleeve to afford the curves *a*,*b*,*c*,*d*, as shown in Fig. 2613. The grip may be more equalised by boring the sleeve of slightly smaller diameter than the shaft.

Fig. 2614 represents a sleeve carrying out this principle. It is composed of two halves, as shown, bored slightly smaller than the shaft diameter, and is to be compressed on the shaft, which, acting as a wedge, would spring open the sides of the bore until the crown of the bore bedded against the shaft. This, in the case of parallel shaft ends of equal diameter, would hold them with great

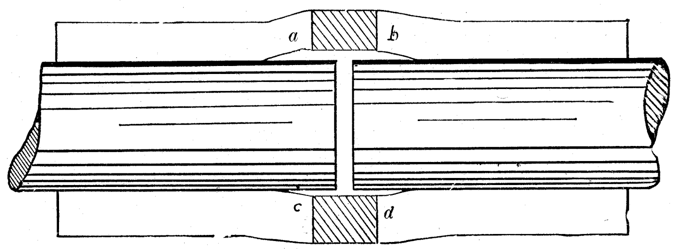

Fig. 2613.

force axially true, and with equal force and bearing, thus meeting all the requirements. If, however, the end of one shaft were of larger diameter than the end of the other (as has hitherto been supposed to be the case), the end accomplished by boring the sleeve of smaller diameter than the shaft is, that the end of the sleeve is afforded the extra elasticity due to the transverse spring of the sleeve, which permits the edges of each half of the sleeve to bear along a greater length of the shaft end than would otherwise be the case ; but the bearing is in this case mainly at and near the edges of the split.

It will be perceived, then, that under this principle of construction, when applied to shaft ends of varying diameters, the metal is left to spring and conform itself to the shape of the parts to be connected, and that there is nothing outside of the condition of relative diameter of shaft to sleeve bore to determine what the direction of the spring or closure of sleeve shall be ; but, on the other hand, the principle possesses excellence in that the sleeve being cylindrical and its closure taking place equally at similar points of contact the shafts will be held axially true, one with the other ; or in other words, the movements of the metal while sleeve

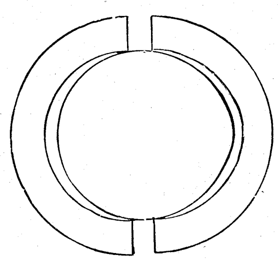

Fig. 2614.

closure is progressing are equally radial to the axis of the sleeve, and there is no element tending to throw the shaft axis out of line one with the other.

If a sleeve have a single split, the manner in which it will grip a shaft smaller than the sleeve's bore depends upon the manner in which the compression is effected.

In Fig. 2615, for example, is a ring supposed to be compressed by a pressure applied at A and at B, causing the ring to assume the form shown by the dotted lines. The centre of the ring bore would therefore be moved from C to D. Now, suppose that the end of one section of shafting were to fit the sleeve bore, then compressing the sleeve upon it would not practically move the centre of the bore; but if the shaft at the other end of the sleeve were smaller than the sleeve bore, the compression of the sleeve to grip the shaft would move the centre of the bore, and, therefore, of the shaft towards D, hence the axial lines of the shafts would

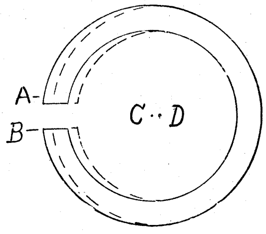

Fig. 2615.

not be held true one with the other. To accomplish this latter object, the compression must be equal all round the sleeve, or it may be applied at the points E and F, Fig. 2616, although it is better to have the compression area embrace all the circumferential area possible of the sleeve, and to have the movement that effects the compression simultaneous and equal at all points on the circumference of the ring or sleeve, because if these movements are independent, more movement or compression may be given at one point than at another, and this alters the centre of the bore; thus, if more pressure were exerted at E than at F, in figure, the centre of the bore would be thrown towards F, or *vice versâ*. If the

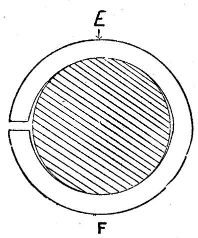

Fig. 2616.

pressure be concentric, the single split ring or sleeve grips the shaft all round its circumference; hence it is only necessary in this case to maintain the circumference of the sleeve in line to insure that the shaft ends be held axially true one with the other; and if the pressure on the ring be applied equally from end to end its closure will also be parallel and equal, and the shaft will be held with equal force along that part of its length enveloped by the coupling. It is obvious, however, that the piece or sleeve gripping the end of one shaft must be independent of that gripping

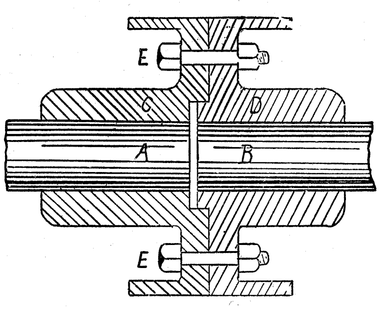

Fig. 2617.

the other, so as to avoid the evils shown in Fig. 2612, while at the same time the casing or guide enveloping the two independent rings or sleeves must guide and hold them axially true, one with the other.

In Fig. 2617 is shown an excellent form of *plate coupling*, in which most of the requirements are obtained. A and B are the ends of the two lengths of shafting to be connected, C and D are the two halves of the coupling driven or forced on the ends of the shafting, and further secured by keys. The end of one half fits into a recess provided on the other half, so as to act as a guide to keep the shafts axially true one with the other, and also to keep the two halves true one with the other, while drilling the holes to receive the bolts E which bolt the coupling together. The objections to this form are, that it is costly to make, inasmuch as truth cannot be assured unless each half coupling is fitted and keyed to the shaft, and turned on the radial or joint faces afterwards. Furthermore, if the coupling were taken off in order to get a solid pulley on the shaft, the coupling is apt to be out of true when put together again, and, therefore, to spring the shaft out of true. Also, that the bearing, support, or hanger must be open-sided to

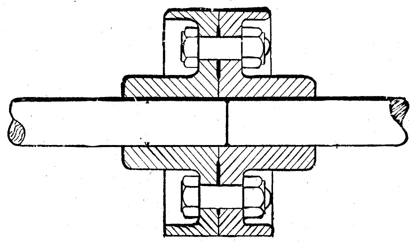

Fig. 2618.

admit the shaft, and that each coupling, being fitted and turned to its place, would be apt to run out of true if removed and applied to another shaft, whether the same be of equal diameter or not; but if each half coupling be provided with a feather instead of the usual key, the coupling may be readily removed and will remain true when put on again.

Fig. 2618 represents a plate coupling, in which one end of the shaft passes into the bore of the half coupling on the other length of shaft, which serves to keep the shafts in line one with the other.

Fig. 2619 represents a single cone coupling composed of an external sleeve having a conical bore and a split internal sleeve

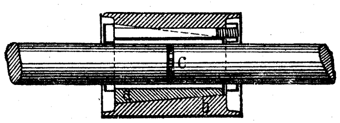

Fig. 2619.

bored to receive the shaft, and turned on its outer diameter to the same cone as the bore of the outer or encasing sleeve. The bolts pass through the inner sleeve, the bolt head meeting the radial face of the inner sleeve while the nut meets the radial face of the outer sleeve, so that screwing up the nut forces the inner sleeve into the outer and closes the bore of the former upon the shaft. This coupling is open to the objection that it cannot grip the ends of the shafts equally unless both shafts be of exactly equal diameter, and the bearing on the smaller shaft will be mainly at the outer end only, as explained in Fig. 2611. As a result, the

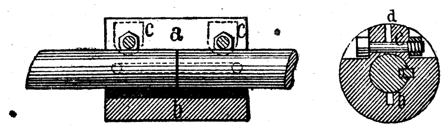

Fig. 2620.

transverse strains on the shaft will cause the couplings to come loose in time.

Fig. 2620 represents a coupling composed of a cylindrical sleeve split longitudinally on one side, as at *d*; the bolts *c* pass through the split. Diametrally opposite is another split passing partly through as at *b*. A key is employed at right angles to the two splits as shown. Here, again, the pressure on a shaft that is smaller than the other, of the two shafts coupled, will be mainly at one end, but separation of the shaft ends is provided against by means of two cylindrical pins on the ends of the key fitting into corresponding holes drilled in the shaft, as shown in the side elevation in the figure.

In Fig. 2621 is shown a coupling whose parts are shown in Fig. 2622. It consists of a cylindrical ring turned true on the outside and bored conical from each end to the middle of its length, as shown. The split cones are bored to receive the shaft and contain a keyway to receive a spline provided in the shaft ends, and are turned on the external diameter to fit the conical borings in the sleeve. Three square bolts pass through the split cones, which, being square, are prevented from rotating while their nuts are being screwed up.

To put the coupling together one split cone is passed over the end of one shaft and the other over that of the other. The sleeve is then put between the ends of the shaft, the position of the shaft adjusted for length and the split cones pushed up into the sleeve; the bolts are then passed through and screwed up. The forcing of the split cones into the conical borings of the sleeve causes the former (from being split) to close upon the shaft ends and grip them equally tight, notwithstanding any slight difference in the diameters of the shaft, there being left between the ends of the split cones sufficient space to allow them to pass through the

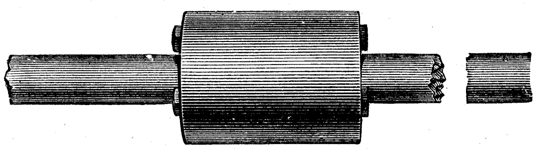

Fig. 2621.

conical borings sufficiently to close upon the respective ends of the shafts; the pressure being parallel and equal on each shaft end, because when the cone has gripped the largest shaft the whole movement due to screwing up the nuts is transferred to the cone enveloping the smaller shaft, and by reason of the cones fitting, the closure of the holes in the cones is parallel, giving an even grip along the shaft end and an equal amount of grip to each shaft end.

To remove the coupling the bolts are removed, and the sleeve being moved endways the cones open from their spring and relieve the grip upon the shaft.

It is evident that in their passage through the sleeve casing the cones will move with their axial lines true with the axial line of the casing; and it is equally evident that the taper on the cone accurately fitting the taper in the sleeve bore, the closure of the cone bores must be equal; while at the same time the pressure on the two cones upon the respective shaft ends must be equal, because it is the friction of the cone bores upon the shaft ends

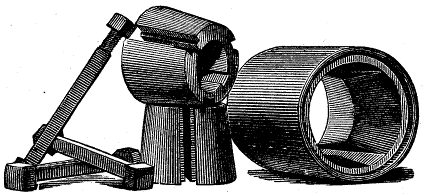

Fig. 2622.

which equally resists the motion of both, while the pressure applied to the respective cones is derived from the same bolts, and hence is equal and simultaneous in its action.

To loosen this coupling for removal the bolts must be stacked back and a few blows on the exterior of the outer shell with a billet of wood may loosen the coupling; but if not, a wedge or a cold chisel may be driven in the splits of the cones to loosen them, but such wedge or chisel should not have contact with the sides of the split, either near the bore or near the perimeter, for fear of raising a burr.

In Fig. 2623 is shown a patent internal clamp coupling. It is formed of a cylindrical piece containing a pair of separate clamps, and between these clamps and the outer casing are four screws, two to each clamp; these screws are tapered so as to close the clamp when screwed up and release it when screwed outwards. The holes to receive the shaft ends are bored somewhat smaller than the shafts they are to fit, and the clamps opened to permit the easy insertion of the shaft ends by means of wedges A driven in the split B of each clamp, as shown in Fig. 2624.

The lower edge of the wedges should be slightly above the bore of the clamp to prevent the formation of a burr or projection of metal when the wedge is driven in. When placed upon the shaft ends and in proper position the wedges are removed and the clamp bore will have contact at and near the edges of the longitudinal split and on the opposite sides of the bore where the keyway is shown, but the pressure of the tape screws will spring the clamps on the side of the longitudinal splits, and increase the bearing area at those points.

The main features of this device are that though the bore be made a driving fit to the shaft, it can, by the employment of the

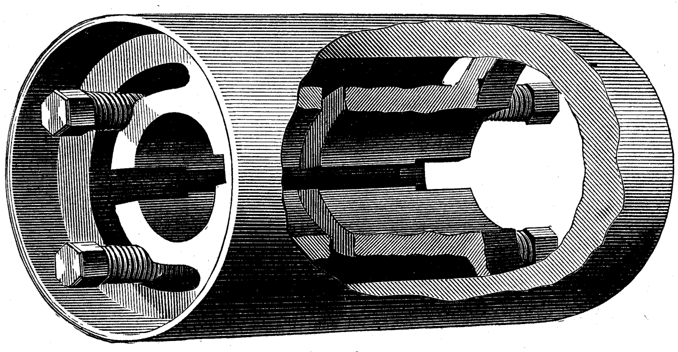

Fig. 2623.

wedges, be put on the shaft with the same ease as if it were an easy fit, while the clamps being separated by a transverse groove may open and close upon the shaft independently of each other, so as to conform separately to any variation in the diameters of the two shaft ends it couples. But it may be noted that since the circumference of each shaft end has a bearing along the line of the coupling bore diametrally opposite to the longitudinal splits, the shafts will not be held quite axially true one with the other unless there be as much difference in the diameters of the separate clamp bores as there is in the diameters of the shaft ends; because to hold two shafts of different diameters axially true one with the other the longitudinal planes of the two circumferences must not at any part of the circumferences form a straight line, as would be the case at that part of the coupling bore at and near the keyway.

It is to be noted, however, that this coupling is formed of one solid piece, and that the strain on the tightening bolts or screws is one of compression only, which tends to hold them firmly and prevent their coming loose.

If the workmanship of a plate coupling, such as in Fig. 2617, be

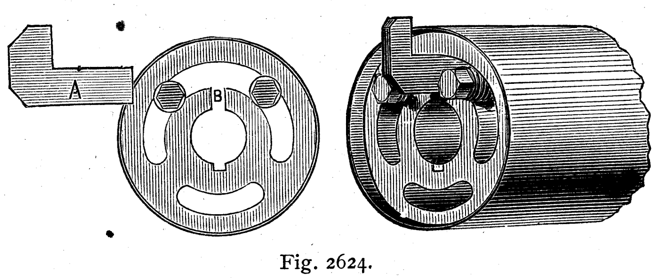

Fig. 2624.

accurately and well done, and the proportions of the same are of correct design, so that the strain placed on the same in keying and coupling it up does not distort it, the coupling and the shaft will run true, because the strain due to the key pressure will not be (if properly driven) sufficient to throw the coupling out of true. But the degree of accuracy in workmanship necessary to attain this end is greater than can be given to the work and compete in the market with work less accurately made, because the difference in the quality of the workmanship will not be discernible save to the most expert and experienced mechanic, and not to him even unless the pieces be taken apart for examination. If the bore of the coupling be true and smooth and of proper fit to the shaft the key pressure, if the key fits on its top and bottom, will not, as stated, be sufficient to throw the coupling out of true. It is true, however, that such pressure is exerted on one half the bore *of the coupling* only, being the half bore opposite to the key. On the other diametral side of the coupling the strain due to the key is exerted on the top face of the key.

If, therefore, the key seats in the shaft and in the couplings are in line or parallel, and both therefore in the same plane, the strain due to the key may throw the coupling out of true to the amount that the key pressure may relieve the bore of the coupling (on the half circumference of the shaft of which the key is the

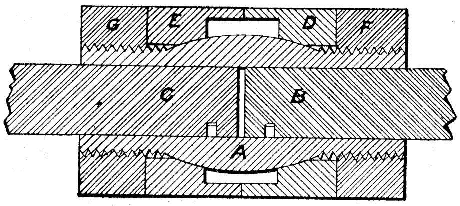

Fig. 2625.

centre) from contact or pressure with the shaft. As a result, the coupling may run to that extent out of true, but the shaft would run true nevertheless so long as the nature of the surfaces on the shaft and on the coupling bore was such that the key pressure caused no more compression or closer contact in the case of one half coupling than in the case of the other.

It is obvious that a plate coupling will require at least as much force to remove it from the shaft as it took to put it on, and sometimes, from rusting of the keys, &c., it requires more. If it be removed by blows it becomes damaged, and damage is likely to be also caused to the shaft, while the surfaces having to slide in

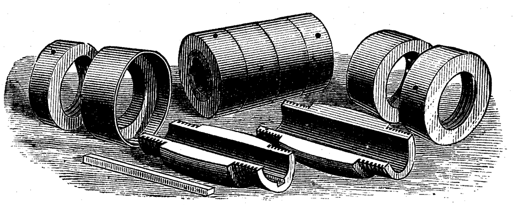

Fig. 2626.

contact under the pressure of the fit the surfaces abrade and compress, and the fit becomes impaired. But in couplings such as shown in Fig. 2621, the gripping pieces are relieved of pressure on the shaft by the removal of the bolts, and the removal of the coupling becomes comparatively easy.

The interchangeability of plate couplings is further destroyed by the fact already stated, that turned shafting is not, as a rule, of accurate gauge diameter, and the least variation in the pressure or fit of the coupling to its shaft is apt to cause a want of truth when the key bears on its top and bottom. The fit of the coupling to its shaft may be, it is true, relied on to do the main part of the driving duty, and the key fitting on the sides only may be a

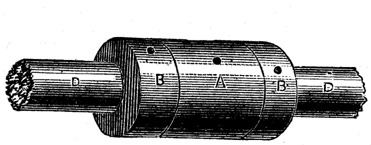

Fig. 2627.

secondary consideration, but in proportion as the fit is relied on to drive, that fit must be tighter, and the difficulty of application and removal is increased.

Another and important disadvantage of the plate coupling in any form is that it necessitates the use of hangers open on one side to admit the shaft, because the couplings must be fitted upon the shaft before the same is erected and should not be removed after being fitted, as would be necessary to slide the end of the shaft through the bearing.

When plate couplings are constructed as in Fig. 2617, the removal of a section involves either the driving back of one-half of the coupling so that the other half will clear it, or else the moving endwise of the whole line to effect the same object.

With a plate coupling the half coupling on one end of the shaft must be removed when it is required to put an additional pulley on the shaft, unless, indeed, a split pulley be used, whereas with a clamp coupling, such as shown in Fig. 2621, the half coupling at each end may be slacked and moved back, one end of the shaft released, a solid pulley placed on the shaft and the coupling replaced, when it will run as true as before, and the pulley may be adjusted to its required position on the length of shafting.

It is to be remarked, however, that a well-made plate coupling, such as in Fig. 2618, makes a good and reliable permanent job that will not come loose under any ordinary or proper conditions.

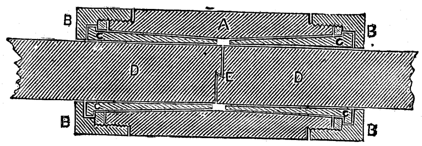

Fig. 2628.

In Fig. 2625 is shown a patent self-adjusting compression clamp, which is peculiarly adapted to connect shafting that is of proper gauge diameter. It consists of a sleeve A made in two halves, each embracing nearly one-half of the shaft circumference and being bored parallel and slightly smaller than the diameter of the shaft ends. Over this sleeve passes at each end a ring D E, bored conical and fitting a similar cone on the external diameter of the sleeve. On each end of the sleeve is the nut F G, which by forcing the cone ring up the taper of the sleeve causes the two halves of the latter to close upon and grip the shaft. For shafts

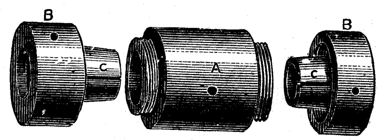

Fig. 2629.

less than two inches in diameter there are provided in the sleeve two pins to enter holes in the shaft ends in place of keys, but for sizes above that keys are employed. All parts of this coupling being cylindrical it is balanced. The separate parts of this coupling are shown in Fig. 2626.

In Figs. 2627 to 2630 are shown a side elevation and sectional view of another form of shaft coupling. A is the sleeve, B B nuts on the ends of the sleeve, and C C cones fitting taper holes in the sleeve. These cones are split, as shown in Fig. 2629, to permit them to close upon the shaft ends. The shaft ends themselves are

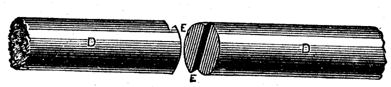

Fig. 2630.

matched with a half dovetail, as in Fig. 2630, which dispenses with the employment of a key.

In coupling shafts of different diameters it is usual to reduce the diameter of the end of the larger to that of the smaller shaft, and to employ a size of coupling suitable for the smaller shaft; but in this case it is necessary that the coupling be placed on the same side of the hanger or bearing as the smaller shaft, otherwise it is obvious that the strength of the larger would, between its bearings, be reduced to that of the smaller shaft.

The couplings for line shafting are usually placed as near to the bearings or hangers as will leave room for the removal of the couplings by sliding them along the shaft.

The couplings on the length of shaft receiving power from the motor are placed outside the bearings, hence on the succeeding lengths there will be one coupling between each pair of bearings, the couplings being in each case as close to each bearing as will allow the coupling to be moved towards the bearing sufficiently to permit the length to be removed without disconnecting the adjacent length from its bearings.

Fig. 2631 represents a very superior form of coupling for line shafts. The ends of the line shaft are reduced to half diameters

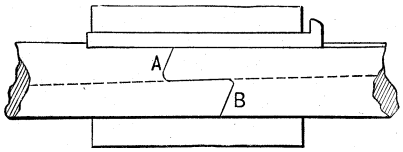

Fig. 2631.

as shown, and lapped with a horizontal joint at an angle to the axis of the shaft as denoted by the dotted line, which prevents end motion ; the ends of the shaft and their abutting surfaces are dovetailed, as shown A and B, and, therefore, perform driving duty. A sleeve envelops the whole joint and is secured by a key. This coupling accomplishes all that can be desired, but requires very accurate workmanship, and on this account is expensive to make.

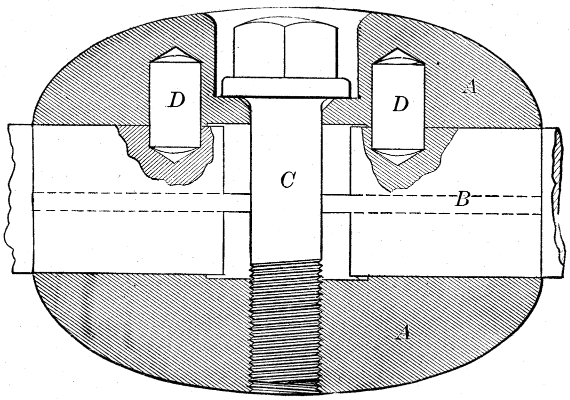

Fig. 2632.

Fig. 2632 represents a form of coupling suitable for light shafting. It consists of two halves A A, of cast iron, which are drawn together by the bolt C ; the centre of the coupling is recessed to enable the coupling to take a better hold on the shaft, which is prevented turning by the pins D D. This coupling has no projections to catch clothes or belts, and is quickly applied or removed.

Fig. 2633* represents a form of coupling for heavy duty, the transmitting capacity only being limited by the strength of the projections A. If the shafts are not axially in line, this form of coupling accommodates the error, since the projections A may slide in their recesses, while if the axial lines of the shafts should vary from flexure of the bearings or foundations, as in steamships, clearance between the ends of A and the bottom of the recesses may be allowed, as shown at B.

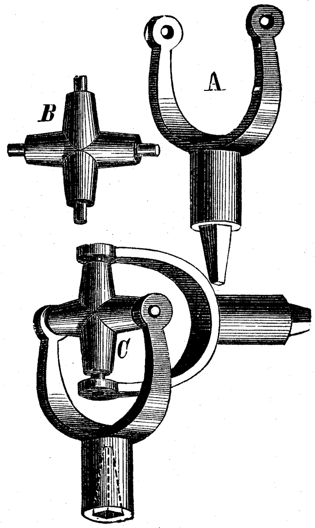

Fig. 2634.

In Fig. 2634 is shown a coupling (commonly known as the universal joint coupling) which will transmit motion either in a straight line, or at any angle up to 45°.

It is formed of two double eyes, such as A, connected to a yoke or crosspiece B as shown at C. It is mainly used for connecting shafts or arms carrying tools of some kind, such as rubbers for polishing stone, tools for boring, or other similar purposes in which the tool requires to be rotated at varying angles with the driving shaft.

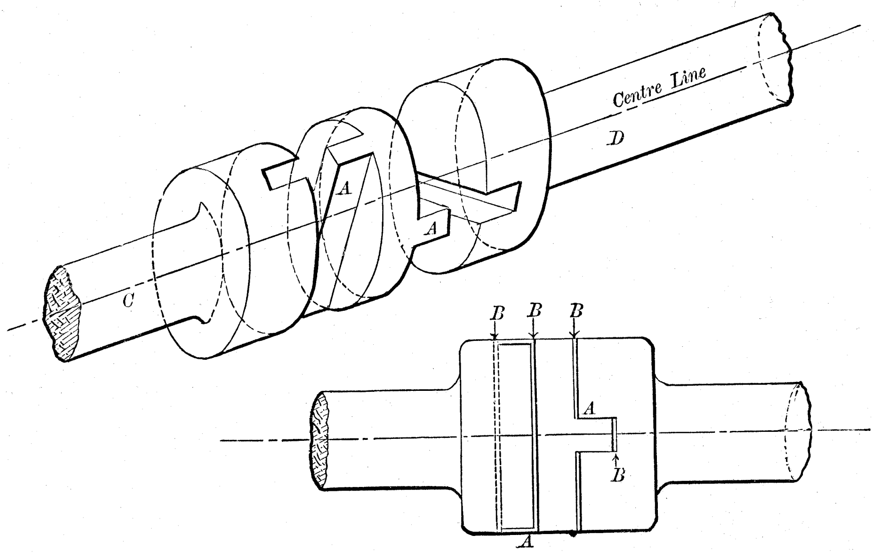

Fig. 2633.

* From Rankine's "Machinery and Millwork."