

Civil and Mechanical Engineering.

FORMULAS AND TABLES FOR THE SHAFTING OF MILLS AND FACTORIES.

By JAMES B. FRANCIS, Civil Engineer.

THE following investigation was undertaken at the request of General John C. Palfrey, the Agent of the Merrimack Manufacturing Company, of Lowell, Massachusetts, for the purpose of determining the relative fitness of wrought iron and steel for the shafting of a cotton factory now erecting by that company.

Samples of steel, all of American manufacture, were obtained from different makers, and, together with several samples of iron, were subjected to experiment.

The constant expressing the resistance of cylindrical bars to torsion, I deduce from Navier's formula,*

$$T = \frac{16 WR}{\pi d^3}, \dots \dots \dots (1.)$$

in which,

T = a constant for the same material.

W = the weight, in pounds, which, if applied at the distance R , in inches, from the axis, will just fracture the bar.

π = the ratio of the circumference of a circle to its diameter.

d = the diameter, in inches, of the bar at the place of fracture.

The bars subjected to torsion, were finished in the form of the following diagram; the ends being two inches square, and the middle turned down to a diameter of $\frac{3}{4}$ inch, in order to insure the fracture taking place in that part of the bar.



The weight producing the torsion was applied at the end of a lever, of the effective length of 35.975 inches, fitted to the square boss at one end of the bar. The tendency of the bar to revolve

* Résumé des Leçons sur l'application de la mécanique.

under the action of the weight, was controlled by a worm-wheel about fifteen inches in diameter and one hundred and thirty-eight teeth, fitted to the square boss at the other end of the bar. This wheel could be moved through any arc by means of a worm. As the bar became twisted by the torsional strain, the worm-wheel was moved through an arc sufficient to bring the lever to an horizontal position.

A graduated circle on one face of the worm-wheel, furnished the means of measuring the arc of torsion.

The effective weight of the lever and scale at 35.975 inches from the axis, where the scale was hung on a knife edge, was 48.5 pounds, and was the least effective weight which could be applied to produce torsion.

EXPERIMENTS ON TORSION.

DESCRIPTION OF THE BAR.	Mean diameter of the reduced part of the bar, in inches.	Arc of torsion just before fracture.	Weight producing fracture, in pounds.	Mean temperature of the air.	Value of T
English refined wrought iron, from a bar two inches in diameter, marked A, 13.....	0.750	416.8°	113.17	58.8°	49,148
Same, marked 13.....	0.750	596.0°	125.69	66.0°	54,585
Wrought iron, from the Pembroke Iron Works, Maine, marked 14	0.753	641.3°	143.72	62.3°	61,673
Decarbonized steel, from the Farist Steel Company, Windsor Locks, Conn., from a bar two inches square, marked B, 6.....	0.752	390.5°	192.48	70.2°	82,926
Spindle steel, from the same, from a bar two inches square, marked A, 5.....	0.750	284.3°	235.17	68.3°	102,131
Steel, from the Nashua Iron Co., Nashua, N. H., from a bar two inches square, marked 2.....	0.751	611.3°	198.73	65.5°	85,961
Same, marked d, 2.....	0.752	557.0°	203.23	63.7°	87,557
Steel, from same, from 1½ inch octagonal bar, marked 4.....	0.752	475.0°	221.0	67.5°	95,213
Same, marked 3.....	0.751	508.3°	217.25	61.2°	93,972
Steel, from the works of Hussey, Wells & Co., Pittsburgh, from a bar two inches square, marked E, 1.....	0.751	398.0°	202.66	63.6°	87,661
Same, marked 1.....	0.748	297.3°	196.50	68.0°	86,023
Bessemer steel, from the works of Messrs. Winslow & Griswold, Troy, N. Y., from a bar two inches square, marked 16.....	0.748	215.5°	181.97	66.0°	79,662
Same, marked 16 x.....	0.748	268.5°	174.50	67.0°	76,392

The experiments on deflection were made on round bars turned to a diameter of about one inch. The distance between the points of support was forty-eight inches. Observations were made of the deflections produced by a weight of one hundred and fifty pounds suspended at the middle point between the supports. This weight was not sufficient to cause any sensible set in the bar after the weight was removed; and no sensible increase in the deflection was produced by allowing the weight to remain suspended on the bar for several days.

The constant E for deflection, has been computed by Navier's formula,*

$$s = \frac{l^3 w}{6\pi d^4 E}, \quad \dots \dots \dots (2)$$

in which

l = the distance between the points of support, in inches.

w = the weight at the middle point between the supports, in pounds.

π = the ratio of the circumference of a circle to its diameter.

d = the diameter of the bar, in inches.

s = the deflection at the middle point between the supports, in inches.

EXPERIMENTS ON DEFLECTION.

DESCRIPTION OF THE BAR.	Diameter of bar at the middle, in inches.	Deflection, in inches.	Mean tempera- ture of the air.	Value of E
Spindle steel, from the Farist Steel Co., Windsor Locks, Conn., from a bar 1 1-16 inches in diameter, marked A, 7.	0.995	0.2330	48.0°	3,853,590
Same, marked A x 7.	0.977	0.2315	53.8°	3,847,530
Decarbonized steel, extra, from the Farist Steel Co., from a bar 1 1-16 inches in diameter, marked A A x.	0.993	0.2310	53.0°	3,918,360
Same, marked A A 8.	0.995	0.2327	53.7°	3,858,557
Decarbonized steel, from the Farist Steel Co., from a bar 1 1-16 inches in diameter, marked 9 x B.	0.992	0.2330	54.2°	3,900,420
Same, marked 9 B.	0.995	0.2307	53.3°	3,892,008
Steel, from the works of Hussey, Wells & Co., Pittsburgh, from a bar 1 1-16 inches in diameter, marked 15.	0.998	0.2337	52.2°	3,796,060
Same, marked 15 x.	0.996	0.2337	49.8°	3,826,641
Bessemer steel, from the works of Messrs. Winslow & Griswold, Troy, New York, from a bar 1 1-16 inches in diameter, marked 17 x.	1.000	0.2330	49.4°	3,777,095
Same, marked 17.	1.000	0.2315	52.0°	3,801,566

* See the number of this Journal for February, 1862.

Several specimens of the steel have been tested for tensile strength, at the works of the South Boston Iron Company, by Mr. F. Alger, in the apparatus designed by Major W. Wade, for testing metals for cannon, a description of which may be found in *Reports of Experiments on the Strength and other Properties of Metals for Cannon*, published in 1854, by authority of the Secretary of War.

EXPERIMENTS ON TENSILE STRENGTH.

DESCRIPTION OF THE SPECIMEN.	Diameter at the place of fracture, in inches.	Weight producing fracture, in pounds.	Tensile strength per square inch in pounds.	Specific Gravity.
Spindle steel, from the Farist Steel Co., Windsor Locks, Conn., marked A 10 A 1.....	0.597	40,800	145,754	7.8401
Same, marked A 10 A 2.....	0.598	39,500	140,639	7.8287
Decarbonized steel, from the same, marked B 11 B 1.....	0.596	34,500	123,662	7.8583
Same, marked B 11 B 2.....	0.597	35,200	125,750	7.8514
Decarbonized steel, extra, from the same, marked A A x 1.....	0.600	30,500	107,862	7.8417
Same, marked A A x 2.....	0.600	30,900	109,271	7.8579
Same, marked A A x ; ends upset in order to form the specimen.....	0.600	30,890	108,901	7.8484
Same as next preceding specimen, marked A A x 2.....	0.600	29,700	105,053	7.8534
Steel, from the works of Hussey, Wells & Co., Pittsburgh, marked c 12, 1.....	0.594	40,400	145,790	7.8530
Same, marked c 12, 2.....	0.594	40,200	145,070	7.8496

I find on record many experiments on the fracture of iron and steel by torsion, from which I deduce the following values of T; using the above formula for cylindrical bars, and Navier's formula,

$$T = \frac{3\sqrt{2}WR}{b^3}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (3.)$$

for square bars, in which b = the side of the square in inches, and w and R the weight in pounds, producing fracture, and the distance from the axis in inches, at which it is applied.

EXPERIMENTS BY RENNIE, given in the *Philosophical Transactions of the Royal Society*, for 1818.

Bar of English wrought iron, 0.25 inch square,	T = 65,982
Bar of Swedish " 0.25 "	T = 61,909
Bar of shear steel, 0.25 "	T = 111,191
Average of 3 bars of iron cast horizontally, 0.25 in. sq., T =	64,776

EXPERIMENTS given in the fifth edition of *Haswell's Engineers' and Mechanics' Pocket Book*.

Bar of Ulster Iron Co.'s wrought iron, one inch diameter, $T = 87,090$
 Bar of Swedish " " " $T = 93,965$

EXPERIMENTS made at the Royal Gun Factories, Woolwich, England, on many varieties of cast iron. *Parliamentary Document ordered to be printed July 30, 1858.*

Experiments are given on fifty-one varieties of British cast iron, besides several varieties from other countries. I select the experiments on four varieties of British iron, viz: the strongest, two of medium strength, and the weakest; each result being deduced from a mean of several experiments on bars about 1·8 inch in diameter.

From West Hallam Iron Works, Ilkeston..... $T = 38,217$
 " Netherton Iron Works..... $T = 34,490$
 " Butterley " $T = 33,949$
 " Haematite Iron Company..... $T = 22,132$

EXPERIMENTS made at the Fort Pitt Foundry, in 1846, on bars of different forms and dimensions, of common foundry iron, given in *Reports of Experiments, &c.*, above cited.

Bar about one inch square..... $T = 36,846$
 " 1·415 inch square..... $T = 34,443$
 " about 1·749 inch square..... $T = 42,821$
 " 1·135 inch in diameter..... $T = 37,445$
 " 1·595 " " $T = 42,047$
 " 1·955 " " $T = 38,851$

EXPERIMENTS made at the West Point Foundry, in 1851, on Greenwood iron of different grades, mixtures and fusions, given in *Reports of Experiments, &c.*, above cited.

Mean deduced from eighteen experiments on bars about 1·9 inches diameter..... $T = 44,957$

The value of E , for wrought iron, I have previously deduced from English experiments, and tested by a single experiment on a shaft two inches in diameter and about 180 inches between bearings.*
 From these experiments I find..... $E = 3,492,539$

There being such great irregularities in the values of T , it will not be safe, in practice, to take a mean value, but one near the lowest value.

* See the number of this Journal for February, 1862.

The values for wrought iron vary from 49,148 to 93,965. For safety, I take for wrought iron	T = 50,000
The values for steel vary from 76,392 to 111,191. For safety, I take for steel	T = 80,000
The values for cast iron vary from 22,132 to 64,776. For safety, I take for cast iron	T = 30,000
I also take for wrought iron	E = 3,500,000
And for untempered steel	E = 3,800,000

Shafts for transmitting power, are subject to two forces, viz: transverse strain and torsion. In shafts of wrought iron or steel, in which the bearings are not very near to each other, a transverse strain, too small to cause fracture, will produce sensible deflection; if this is too great, it will produce sensible irregularities in the motion, and tend towards the rapid destruction of the shaft and its bearings. This limits the distance between the bearings, as the weight of the shaft itself will produce an inadmissible amount of deflection whenever this distance exceeds a certain amount, which varies with the material and diameter of the shaft.

The deflection of a cylindrical shaft from its own weight, supported at each end, but disconnected from other shafts, is given by the formula (4), which is deduced from Navier's formula for the deflection of a cylindrical bar. See *Journal of the Franklin Institute for February, 1862.*

$$\delta = 0.007318 \frac{l^4}{d^2 E}, \quad . \quad . \quad . \quad . \quad (4.)$$

If the several parts are so connected as to be equivalent to one continuous shaft, it will correspond to the case of a beam fixed at both ends, for which case Barlow* gives δ equal to two-thirds of its value in the case of a beam supported at both ends, given by formula (4). Navier†, taking into account the effect of the deflection in the adjacent divisions, finds δ equal to one-fourth of its value by formula (4). In order to decide which of these eminent authorities to follow, I have appealed to experiment.

Experiment 1. A bar of wrought iron purchased as "English refined," 12 feet $2\frac{3}{4}$ inches long, 0.367 inch deep, 1.535 inch wide, was supported at four equidistant points, four feet apart. When loaded at the middle points of each division with fifty-two pounds, the deflection in the middle division was 0.069 inch, and the

* Report of the third meeting of the British Association for the Advancement of Science.

† Résumé des leçons sur l'application de la mécanique.

mean deflection in the other two divisions was 0.371 inch. The weight on the middle division was then increased until the deflection was alike, viz: 0.281 inch in each division; the weight being 82.84 pounds in the middle division, and 52.00 pounds in each of the other divisions. Four feet was then cut off of each end of the bar, when the deflection, with 82.84 pounds on the middle division, was 1.102 inch.

Experiment 2. A bar of iron of the same quality and length as in experiment 1, 0.551 inch square, was laid on the same supports. When loaded at the middle points of each division with fifty-two pounds, the deflection in the middle division was 0.058 inch, and the mean deflection in the other two divisions was 0.314 inch. The weight on the middle division was then increased until the deflection was 0.241 inch in each division; the weight being 82.84 pounds in the middle division, and 52.00 pounds in each of the other divisions. Four feet was then cut off of each end of the bar, when the deflection, with 82.84 pounds on the middle division, was 0.984 inch.

In the case in which the deflections were alike in the three divisions, the middle division corresponds to the case of a continuous shaft supported by numerous equidistant bearings, and the case where the bar was reduced in length, corresponds to that in formula (4). Comparing the deflections in the two cases in the above experiments, we find by experiment 1, that the ratio of the deflection of the shaft, simply supported at each end, to that of the continuous shaft, is as 1 to 0.255. In experiment 2, the corresponding ratio is as 1 to 0.245; the mean of the two experiments giving a ratio of 1 to 0.25,* which agrees with Navier, and we must adopt for the deflection of a continuous shaft, from its own weight, the formula

$$\delta = \frac{1}{4} \times 0.007318 \frac{l^4}{d^2 E}, \quad . \quad . \quad . \quad (5.)$$

The greatest admissible value of δ in proportion to the length, must be determined by experience. Tredgold assumes that for cast iron, it might be 0.01 inch for each foot in length, or $\frac{1}{1200}$ part of the

* These experiments indicate the effect of connecting the chords of truss bridges over the piers. Assuming that in a bridge of not less than three equal spans, the top and bottom chords have equal resisting powers, and the whole length of the bridge is uniformly loaded, if the chords are continuous throughout the whole length of the bridge, the deflection of any span, except the end spans, will be one quarter of the amount that it would be if the chords were disconnected at the piers.

length, *whatever may be the diameter*; but the transverse strain to produce this deflection, is a greater fraction of the transverse strain that will produce fracture in a large shaft than in a small one. The maximum strains of extension and compression in a shaft, for the same deflection, are in proportion to the diameter, while the deflection itself, from the weight of the shaft, is inversely as the square of the diameter; consequently, the deflection to produce the same maximum strains, must be inversely as the diameter.

Adopting this principle and the assumption that a shaft of wrought iron or untempered steel two inches in diameter, may deflect from its own weight, 0.01 inch per foot in length between the bearings, we may determine the greatest admissible distances between the bearings of shafts of other diameters, as follows:

The greatest admissible deflection for any diameter d , is

$$\delta = \frac{2l}{1200d} = 0.00167 \frac{l}{d} \quad (6.)$$

Substituting this value of δ in (5) and reducing, we have

$$l = \sqrt[3]{0.9128 d^2 E}, \quad (7.)$$

TABLE of the greatest admissible distances between the bearings of continuous shafts, subject to no transverse strain except from their own weights; computed by formula (7).

DIAMETER OF SHAFT, IN INCHES.	Distance between bearings, in feet.	
	If of wrought iron	If of steel
1.....	12.27	12.61
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
10.....	26.44	27.18
11.....	27.30	28.05
12.....	28.10	28.88

In practice, long shafts are scarcely ever entirely free from transverse strains; however, in the parts of long lines which have no pulleys or gears, with the couplings near the bearings, the interval

between the bearings may approach the distances given in the preceding table. Near the extremities of a line, the distances between the bearings should be less than are given in the table. The last space should not exceed sixty per cent. of the distance there given, the deflection in that space being much greater than in other parts of the line. In shafts moving with high velocities, it will usually be necessary to shorten the distances between the bearings, as given in the table, in order to obtain sufficient bearing surface to prevent heating.

In factories and workshops, power is usually taken off from the lines of shafting, at many points, by pulleys and belts, by means of which the machinery is operated. When the machines to be driven are below the shaft, there is a transverse strain on the shaft, due to the weight of the pulley and tension of the belt, which is in addition to the transverse strain due to the weight of the shaft itself. Sometimes the power is taken off horizontally on one side, in which case the tension of the belt produces a horizontal transverse strain; and the weight of the pulley acts with the weight of the shaft, to produce a vertical transverse strain. Frequently the machinery to be driven is placed above the floor to which the shaft is hung in the story below; in this case the transverse strain produced by the tension of the belt is in the opposite direction to that produced by the weight of the pulley and shaft. Sometimes power is taken off in all these directions, from the part of a shaft between two adjacent bearings. To transmit the same power, the necessary tension of a belt diminishes in proportion to its velocity; consequently, with pulleys of the same diameter, the transverse strain will diminish in the same ratio as the velocity of the shaft increases. In cotton and woollen factories with wooden floors, the bearings are usually hung on the beams, which are usually about eight feet apart; and a minimum size of shafting is adopted for the different classes of machinery which has been determined by experience as the least that will withstand the transverse strain. This minimum is adopted independently of the size required to withstand the torsional strain due to the power transmitted; if this requires a larger diameter than the minimum, the larger diameter is, of course, adopted. In some of the large cotton factories in this neighborhood, in which the bearings are about eight feet apart, a minimum diameter of $1\frac{1}{8}$ inch was formerly adopted for the lines of shafting driving looms. In some mills this is still retained, in others $2\frac{1}{8}$ inches and $2\frac{3}{8}$ inches have been substituted.

In the same mills, the minimum size of shafts driving spinning machinery, is from $2\frac{1}{8}$ to $2\frac{1}{16}$ inches. In very long lines of small shafting, fly-wheels are put on at intervals, to diminish the vibratory action due to the irregularities in the torsional strain.

We can deduce from formula (1) the *breaking power*, or, in other words, the power which, being transmitted by a shaft, will produce a torsional strain upon it equal to its total resistance to that force.

Put p = the breaking power, in horse-powers of 33,000 foot-pounds,
 N = the number of revolutions of the shaft per minute.

$$p = \frac{2\pi R N W}{12 \times 33000}$$

from which we deduce,

$$W R = \frac{12 \times 33000 p}{2\pi N}$$

Substituting this value in (1), we find,

$$p = \frac{\pi^2 N d^3 T}{8 \times 33000 \times 12} = 0.000003115 N d^3 T, \quad \dots (8.)$$

Substituting the values of T , adopted above for iron and steel, we have

$$\text{For wrought iron, } p = 0.1558 N d^3, \quad \dots (9.)$$

$$\text{" steel, } p = 0.2492 N d^3, \quad \dots (10.)$$

$$\text{" cast iron, } p = 0.0935 N d^3, \quad \dots (11.)$$

A formula for the wrought iron shafts of prime movers and other other shafts of the same material, subject to the action of gears, which I have adopted in numerous cases in practice during the last twenty years, and found to give an ample margin of strength, is

$$d = \sqrt[3]{\frac{100 P}{N}}, \quad \dots (12.)$$

in which P = the power transmitted, and from which we deduce

$$P = 0.01 N d^3, \quad \dots (13.)$$

For simply transmitting power, the formula I have used is

$$d = \sqrt[3]{\frac{50 P}{N}}, \quad \dots (14.)$$

$$\text{from which we deduce } P = 0.02 N d^3, \quad \dots (15.)$$

Comparing formulas (9) with (12) and (13), and also with (14) and (15), it will be seen that the formulas (12) and (13), used for shafts for prime movers, give a strength 15.58 times the breaking power; and the formulas (14) and (15), for shafts simply transmitting power, give a strength 7.79 times the breaking power.

In applying the rules for the strength of materials to constructions in which there is no movement, it is usual to make the computed strength from three to five times the breaking strain. Bodies in rapid motion, however, usually require a greater margin of strength, in order to provide for the tendency to vibration. In cases where shafting for simply transmitting power, is very accurately finished and firmly supported by bearings at short intervals, an excess of strength two-thirds of that given by formulas (14), (19) and (23) will undoubtedly suffice. In ordinary cases, however, the strength given by these formulas should be adopted.

It must be understood that the shafts to which formulas (12) and (13) are applied, are supported by bearings sufficiently near to each other to guard against the transverse strain caused by the prime mover or gear.

To find formulas for steel shafts of the same strength as those for wrought iron, we have for prime movers $p = 15.58 P$; substituting this value of p in (10), we have

$$P = 0.016 N d^3, \quad . \quad . \quad . \quad . \quad . \quad (16.)$$

from which we deduce

$$d = \sqrt[3]{\frac{62.5 P}{N}}, \quad . \quad . \quad . \quad . \quad . \quad (17.)$$

Similarly, we find for steel shafts for simply transmitting power,

$$P = 0.032 N d^3, \quad . \quad . \quad . \quad . \quad . \quad (18.)$$

and

$$d = \sqrt[3]{\frac{31.25 P}{N}}, \quad . \quad . \quad . \quad . \quad . \quad (19.)$$

Similarly for cast iron, we find for prime movers,

$$P = 0.006 N d^3, \quad . \quad . \quad . \quad . \quad . \quad (20.)$$

$$d = \sqrt[3]{\frac{167 P}{N}}, \quad . \quad . \quad . \quad . \quad . \quad (21.)$$

For simply transmitting power,

$$P = 0.012 N d^3, \quad . \quad . \quad . \quad . \quad . \quad (22.)$$

$$d = \sqrt[3]{\frac{83 P}{N}}, \quad . \quad . \quad . \quad . \quad . \quad (23.)$$

The following table gives the power which can be safely carried by shafts making one hundred revolutions per minute. The power which can be carried by the same shafts at any other velocity, may be found by the following simple rule:

Multiply the power given in the table, by the number of revolutions made by the shaft per minute; divide the product by one hundred; the quotient will be the power which can be safely carried.

DIAMETER IN INCHES.	Horse-power which can be safely carried by shafts for prime movers and gears, well sup- ported by bearings, and making 100 revolutions per minute; if of			Horse-power which can be safely transmitted by shafts making 100 revolutions per minute, in which the transverse strain, if any, need not be considered; if of		
	Wrought iron, computed by formula (13)	Steel, computed by formula (16)	Cast Iron, computed by formula (20)	Wrought Iron, computed by formula (15)	Steel, computed by formula (18)	Cast Iron, computed by formula (22)
1.00	1.00	1.60	0.60	2.00	3.20	1.20
1.25	1.95	3.12	1.17	3.00	6.24	2.34
1.50	3.37	5.39	2.03	6.74	10.78	4.06
1.75	5.36	8.58	3.22	10.72	17.16	6.44
2.00	8.00	12.80	4.80	16.00	25.60	9.60
2.25	11.39	18.22	6.83	22.78	36.44	13.66
2.50	15.62	24.99	9.37	31.24	49.98	18.74
2.75	20.80	33.28	12.48	41.60	66.56	24.96
3.00	27.00	43.20	16.20	54.00	86.40	32.40
3.25	34.33	54.93	20.60	68.66	109.86	41.20
3.50	42.87	68.59	25.72	85.74	137.18	51.44
3.75	52.73	84.37	31.64	105.46	168.74	63.28
4.00	64.00	102.40	38.40	128.00	204.80	76.80
4.25	76.77	122.83	46.06	153.64	245.66	92.12
4.50	91.12	145.79	54.67	182.24	291.58	109.34
4.75	107.17	171.47	64.30	214.34	342.94	128.60
5.00	125.00	200.00	75.00	250.00	400.00	150.00
5.25	144.70	231.52	86.82	289.40	463.04	173.64
5.50	166.37	266.19	99.82	332.74	532.38	199.64
5.75	190.11	304.18	114.06	380.22	608.36	228.12
6.00	216.00	345.60	129.60	432.00	691.20	259.20
6.25	244.14	390.62	146.49	488.28	781.24	292.98
6.50	274.62	439.39	164.78	549.24	878.78	329.56
6.75	307.55	492.08	184.53	615.10	984.16	369.06
7.00	343.00	548.80	205.80	686.00	1097.60	411.60
7.25	381.08	609.73	228.65	762.16	1219.46	457.30
7.50	421.87	674.99	253.13	843.74	1349.98	506.26
7.75	465.48	744.77	279.29	930.96	1489.54	558.58
8.00	512.00	819.20	307.20	1024.00	1638.40	614.40
8.25	561.52	898.43	336.91	1123.04	1796.86	673.82
8.50	614.12	982.59	368.47	1228.24	1965.18	736.94
8.75	669.92	1071.87	401.95	1339.84	2143.74	803.90
9.00	729.00	1166.40	437.40	1458.00	2332.80	874.80
9.25	791.45	1266.32	474.87	1582.90	2532.64	949.74
9.50	857.37	1371.79	514.43	1714.74	2743.58	1028.86
9.75	926.86	1482.98	556.12	1853.72	2965.96	1112.24
10.00	1000.00	1600.00	600.00	2000.00	3200.00	1200.00

Comparing formulas (14) and (19), it will be seen that the diameters of shafts of wrought iron and steel, to transmit the same power, are in the ratio of the cube root of 50 to the cube root of 31.25, or as 1 to 0.855. The weights of the shafts will be as the squares of the diameters, or as 1 to 0.731. The power required to overcome the friction of the shafts in their bearings, assuming that the co-efficient of friction is the same for wrought iron and steel, will be as the products of the weights into the velocities of the rubbing surfaces. The number of revolutions in a given time being the same in both, the velocities of the rubbing surfaces will be as the diameters; and the weights will be as the squares of the diameters; the power required to overcome the friction will therefore be as the cubes of the diameters, or as 1 to 0.625. That is to say, the power which must be expended to overcome the friction of a steel shaft is five-eighths of that required to overcome the friction of a wrought iron shaft of equal strength.

The superiority of steel to resist transversal strain is much less than to resist torsional strain. The relative diameters of wrought iron and steel shafts, to resist equal transverse strains, exclusive of their own weights, are inversely as the fourth roots of the respective values of E, or as $\left(\frac{1}{3500000}\right)^{\frac{1}{4}}$ to $\left(\frac{1}{3800000}\right)^{\frac{1}{4}}$, or as 1 to 0.98. That is to say, steel shafts, to offer the same resistance to external transverse strains, may be two per cent. less in diameter than wrought iron shafts. The weights of such steel shafts will be about four per cent. less than the weights of wrought iron shafts of equal stiffness; and the power required to overcome the friction of the bearings will be about 6 per cent. less.

Lowell Mass., May 4, 1867.

(Continued from page 304.)

THE NEW YORK "CENTRAL PARK."

By WILLIAM H. GRANT, Superintending Engineer.

SHOULD such an amount of rain fall at a time when the grounds were covered with a considerable body of snow, and the grounds not frozen, the accumulation of water might so much surpass the capacity of the gutters and drains, as to cause the gullyng of the roads and grounds. But the simultaneous occurrence of such contingencies,