the Navy,—has made an experiment to illustrate the above principles. The apparatus used consisted of two rubber tubes of equal lengths, with *circular sections* curving from a common rubber bulb between them. When subjected to the hydrostatic pressure produced by filling the bulb with water, and placing weights upon it, the tubes showed no perceptible tendency to straighten; although, according to my paper, if the cross-section had enlarged by the stretching of the material, *then* a slight straightening of the tubes should have resulted. This crucial test is conclusive of the correctness of the theory I have mathematically demonstrated.

For the Journal of the Franklin Institute.

On the Steam and Exhaust Ports of Steam Cylinders—The practicability of a material reduction in their dimensions suggested. By D. M. GREENE, U. S. Naval Engineers.

The rules employed by designers of steam machinery in determining the areas of steam ports and passages, seem to be generally, if not always, entirely arbitrary; indeed, the “*rule of thumb*” appears to be not only the principal one used, but to find advocates in quarters and under circumstances where they would be least expected. The ever-varying conditions of steam pressure and velocity of piston appear to be ignored, and the unwarrantable assumption made, that the relation between the areas of the ports and passages, and of the piston once determined, for an engine working under a given steam pressure, and with a given velocity of piston, furnishes a reliable precedent in designing all engines whatever may be the conditions under which they are to perform their functions.

In engines of great power, having large cylinders, and using steam of more than ordinary high pressure, great difficulty has been experienced in securing a proper action of the valves. This has been owing to the excessive weight of the valve, together with the enormous steam pressure upon its back, causing a rapid destruction of its face by friction. Much ingenuity has been expended in providing remedies for this evil, and as the results we have, among other contrivances, the “vacuum ring,” the “Wadell plate,” and the steel roller, all of which possess merit; but which, while affording partial relief, are objectionable on account of the difficulty attending their proper fitting up and adjustment, and especially on account of their liability to derangement after having been once adjusted.

It seems strange that, while so much has been done to mitigate the effect of the evil of large and heavy valves, nothing appears to have been done to reduce the *evil itself* to its minimum practicable proportions. Why has not some one suggested that the ports, and consequently the valves, as now proportioned, may be twice or even three times as large as necessity requires?

The writer, in constructing a formula for his own use in determining the proper areas of opening for steam ports, has had his attention forcibly directed to this matter by the result obtained; and, feeling

that the subject is one of profound interest to the profession, will now indicate the course pursued by him in its examination.

Steam passes from the steam chest to the cylinder by virtue of an excess of pressure in the former over that in the latter. What this excess of pressure is, however, in practice, we are without reliable information; no experiments, to our knowledge, having been made with a view to determine it.

Indeed, such information is not essential; as we may assume any admissible excess of pressure, and base our determination upon such assumption. We will, therefore, assume 0·3 of a pound as an excess which is not only entirely admissible, but even considerably less than may be allowed in practice. This assumption being made, the velocity with which the steam will enter the cylinder through the port will be that due to the height of a column of the entering steam, the weight of which is equal to the assumed excess of pressure; in other words, the velocity of the current will be that which a body would acquire, in falling freely through the height of a column of the entering steam, having a base of one square inch, and weighing 0·3 of a pound. The volume of steam admitted through the port will depend upon the area of the port and upon the velocity of the current; while the volume of steam required to be admitted in the same time, will depend upon the area of the piston and its velocity.

In order to express these conditions in the form of an equation, let

D = diameter of cylinder in inches.

A = area of piston in inches.

v = velocity of piston in feet per second.

a = area of steam port in square inches.

$K a$ = effective steam port in square inches.

v = velocity of entering steam in feet per second.

s = stroke of piston in feet.

p = pressure of steam in steam chest above zero.

δ = difference of pressure in steam chest and cylinder.

h = height due to v .

c = specific volume of steam of pressure p .

r = number of revolutions per minute.

Then will

$$v = \sqrt{2 g h},$$

and

$$K a v = K a \sqrt{2 g h};$$

which represents the volume of steam admitted to the cylinder per second. But this volume is also represented by $A v$; hence

$$K a \sqrt{2 g h} = A v;$$

whence

$$a = \frac{A v}{K \sqrt{2 g h}}, \quad . \quad . \quad . \quad (1.)$$

$$\text{Now, } A = .7854 D^2, \text{ and } v = \frac{2rs}{60} = \frac{rs}{30}.$$

Substituting in (1) we get

$$a = \frac{.7854 D^2 rs}{30 \kappa \sqrt{2gh}}, \quad (2.)$$

To determine h , in terms of c and δ , we have but to remember that a column of steam of the pressure p , having a base of one square foot, and a height of c feet, weighs 62.5 pounds; while if the base be but one square inch, the weight of the column will be $\frac{62.5}{144}$ pounds; then to find how much of this column will be required to weigh δ pounds, we have the proportion,

$$\frac{62.5}{144} : \delta :: c : h;$$

whence

$$h = \frac{144 \delta c}{62.5} \\ = 2.304 \delta c.$$

Substituting now this last value of h in (2) we get

$$a = \frac{.7854 D^2 rs}{30 \kappa \sqrt{2g \times 2.304 \delta c}} \\ = \frac{.7854}{30 \sqrt{2} \times 32.16 \times 2.304} \times \frac{D^2 rs}{\kappa \sqrt{\delta c}} \\ = \frac{.7854}{30 \times 12.17} \times \frac{D^2 rs}{\kappa \sqrt{\delta c}} = .00216 \frac{D^2 rs}{\kappa \sqrt{\delta c}}, \quad (3.)$$

If we make $\delta = 0.3$ of a pound, as already indicated, (3) becomes

$$a = \frac{.00216}{\sqrt{0.3}} \times \frac{D^2 rs}{\kappa \sqrt{c}} = .004 \frac{D^2 rs}{\kappa \sqrt{c}}, \quad (4.)$$

For long and narrow rectangular ports, κ may be taken as .8, and (4) will become

$$a = .005 \frac{D^2 rs}{\sqrt{c}}, \quad (5.)$$

For circular ports, as when the puppet valve is employed, κ may be taken as .62, in which case (4) becomes

$$a = .00645 \frac{D^2 rs}{\sqrt{c}}, \quad (6.)$$

The following table furnishes a few values of c . Where the values of p differ from those given, recourse may be had to a more complete

table; or, in its absence, c may be determined with sufficient accuracy by the "Law of Mariotte."

p (In pounds above zero).	c	p (In pounds above zero).	c
15	1669	105	282
30	883	120	251
45	610	135	224
60	470	150	205
75	383	165	187
90	325	180	174

To exemplify the application of our results, let $D = 100''$, $r = 40$, $s = 4'$, and $p = 60$ pounds. Then will (5)

$$a = .005 \times \frac{100^2 \times 40 \times 4}{\sqrt{470}} = 368.75 \text{ square inches.}$$

Thus we see that, the ports being rectangular, the aggregate opening at each end of the cylinder should be 368.75 square inches. If the ports be double, and their lengths be 82 inches, their widths will evidently be $\frac{368.75}{2 \times 82} = 2.25$ inches.

Again, let $D = 60''$, $r = 65$, $s = 2\frac{1}{8}'$, and $p = 45$. Then will (5)

$$a = .005 \times \frac{60^2 \times 65 \times 2\frac{1}{8}}{\sqrt{610}} = 102.6 \text{ square inches.}$$

The data for these examples are taken from the "one hundred inch" and the "sixty inch" engines, now being constructed for the Navy Department; and the results are practically identical with the areas given to the ports of the cylinders of those engines. These areas were determined by the usual method, *i. e.* by making them a certain function of the areas of the pistons; whereas, it is apparent, from the formula, as well as suggested by common sense, that they should also have been made functions of the number of revolutions and the strokes of the pistons.

To prevent any misapprehension in regard to the apparent coincidence of the results given by the two methods, it is proper to remark that, had the number of revolutions proposed in the first example been 50, instead of 40, the resulting area of the port given by the formula would have been 25 per cent. greater; while the result given by the "rule of thumb" would have been the same.

It may be said that a larger cylinder would have been required, but even then the increased speed of piston would have rendered the result unreliable.

Now, to ascertain whether or not it be practicable to reduce the areas of these ports, let us solve (3) with respect to δ . We get

$$\delta = \frac{.00216^2 D^4 r^2 s^2}{a^2 K^2 c};$$

whence it appears that, other things being equal,

$$\delta \propto \frac{1}{a^2}.$$

Let now a_1 and δ_1 represent another area of port, and the corresponding value of δ , respectively; then we have from the relation just noticed,

$$\delta : \delta_1 :: \frac{1}{a^2} : \frac{1}{a_1^2};$$

whence

$$\delta_1 = \delta \cdot \frac{a^2}{a_1^2}.$$

If $a_1 = \frac{1}{2} a$, this becomes

$$\delta_1 = \delta \cdot \frac{a^2}{\frac{1}{4} a^2} = 4 \delta;$$

but we have constructed our formula, and applied it upon the hypothesis that $\delta = 0.3$;

$$\therefore \delta_1 = 4 \times 0.3 = 1.2 \text{ pounds.}$$

Thus it appears that, in order to realize the advantages of a port only one-half as large as that now employed, we have only to increase the boiler pressure 0.9 of a pound. If a_1 had been made $= \frac{1}{3} a$, we should have found

$$\delta_1 = 9 \delta = 9 \times 0.3 = 2.7 \text{ pounds,}$$

which shows that, as the area of the port is made considerably less than one-half that which is at present employed, the excess of pressure in the steam chest over that in the cylinder soon becomes too large to be admissible.

Among the important advantages which would result from the adoption of a port one-half the present size would be: first, a saving of nearly or quite one-half in the weight of the valve, and in the steam pressure in its back; and second, a very large saving in the clearance space in the nozzles.

To best secure these advantages, the present *width* of port should be preserved, while the *length* should be reduced one-half. (5) and (6) for this case would become

$$a = .0025 \cdot \frac{D^2 r s}{\sqrt{c}} \quad . \quad . \quad . \quad (5a,)$$

for rectangular ports, and

$$a = .0032 \cdot \frac{D^2 r s}{\sqrt{c}} \quad . \quad . \quad . \quad (6a,)$$

for circular ports.

In this connexion, it may not be amiss to call attention to the (to

our mind) absurd practice of making the exhaust openings and passages twice as large as the steam passages. It will be remembered that, in the first example given, the velocity with which the steam enters the cylinder is due to an excess of pressure in the steam chest of only 0.3 of a pound; while the absolute pressure, at the end of the stroke, which forces the exhaust into the condenser is—allowing for

the reduced pressure due to expansion—about 51 pounds. Now, $\frac{51}{0.3} =$

170; or the pressure head to which the velocity of discharge is due is about 170 times as great as that to which the velocity of the steam entering the cylinder is due; and since these velocities are proportional to the square roots of their respective pressure heads, they will be to each other as $\sqrt{1} : \sqrt{170}$; or, 1 : 13. Hence, in this particular case, the velocity of egress will be 13 times as great as the velocity of ingress.

With the more usual pressures of 30 to 40 pounds above zero, the velocity of egress will be about 10 times that of ingress. Even with the reduced port, in the case of the “hundred inch” cylinder, the velocities would be $\sqrt{1} : \sqrt{\frac{60}{1.2}}$; or, 1 : 7.

Why, then, make the exhaust opening larger than the steam opening?

The suggestions contained in this paper are offered especially for the benefit of the younger members of the profession; and also in the hope that individuals having facilities may be induced, by the importance of the subject, to institute experiments with the view of ascertaining whether the advantages claimed for the reduced port may or may not be realized.

Washington, D. C., February 15, 1866.

Phillips' Patent Girder and Fire-proof Flooring.

From the London Mechanics' Magazine, December, 1865.

On Tuesday last some important experiments took place at Mr. Kirkaldy's testing works, the Grove, Southwark, with the above inventions, in the presence of a large number of practical engineers and other scientific gentlemen. Mr. Kirkaldy, who has long been known in connexion with experiments on iron and steel, has recently erected an enormous machine, possessing testing powers from ten pounds to one million pounds, which records its own progress and conclusions with the certainty and exactitude of a chronometer. He conducted the experiments in person, assisted by Mr. Homan. The patentees of the girder, Messrs. W. and T. Phillips, of the Coal Exchange, were also present. The peculiarity of the new invention consists in the flanges being rolled solid, instead of being riveted as heretofore, whilst the web is somewhat thicker than usual. The first experiment was with a light girder 8 inches deep, with $2\frac{1}{2}$ inches flanges, and 20 feet between the supports; the weight was 22 pounds per foot run, the iron being equal to ordinary Staffordshire plate. For the purpose of