Civil and Mechanical Engineering.

THE THEORY AND PRACTICE OF THE SLIDE-VALVE.

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THE Slide-Valve, as at present almost universally used on the Steam Engine, was invented by William Murdock of the Soho Works, near Birmingham, about the year 1784. It was first set to work without either lap or lead, but, subsequently, he added a sixteenth of an inch of lap to give lead for the exhaust, and subsequently, again, he added a sixteenth of lead for the admission, and in that state he left it. And although engineers, since that period, have extended the proportions of lap and lead, yet the primary elements, lap, lead and travel, remain as they left the hands of Watt's companion.

Notwithstanding the enormous amount of power required to work this valve and its appendages, engineers prefer it to any other known form, because of the facilities which it affords for a ready admission of steam, any required amount of expansion, and a liberal lead for exhaust, and requiring no other gear to work it than a rod direct from the valve to the eccentric, thus avoiding complication of parts and ensuring simplicity and certainty of action.

The relief of the slide-valve from the load which its mechanical construction necessarily imposes upon it, has engaged, and in some instances absorbed, for a time, the attention of mechanical engineers. The records of the Patent Office of England alone show that upwards of 500 patents have been taken out for the accomplishment of this object, including, amongst them, many names of the highest distinction known in engineering, and this forms but a fractional part of the number who have otherwise tried and failed. Probably on no portion of the Steam Engine has there been spent more time in study, patents and trials, or have more attempts been made which have proved unsuccessful, than upon the slide-valve. In fact, there is no engineer, of any repute, past or present, who has not devoted some of his time in the endeavor to remove the weight

* This paper is now just received from the author, which fact explains the delay in its publication.—Ed.

from the back of the valve; but so fickle and deceitful a piece of mechanism has it proved itself to be, that it has hitherto baffled the united skill of all, until, at length, it was considered impossible to produce a frictionless slide-valve. This is sufficient proof that there must have been some unknown cause, apart from its mere mechanical construction, operating upon the valve to prevent the successful accomplishment of an effect so unanimously desired. That this was so, there is now no longer any reason to doubt. For, upon principles hitherto unknown to engineers, a slide-valve can be constructed which will move as easily with steam of any pressure on as off, be perfectly steam-tight, and bear on the face of it the impress of simplicity.

To the accomplishment of this task the author set himself about nine years ago, and he has labored incessantly in designing and making experiments during the whole period, from that date to the present. For nearly four years of that time the problem completely baffled him, setting all attempts at defiance, and at length compelled him to leave the beaten path of engineers and to seek a new field of discovery in the relation of surface to pressure. For, it has been discovered, that within certain limits of pressure, neither the file, nor the scraper, nor pumice stone, nor the oil stone, will get up two surfaces of metal sufficiently fine to shut out steam. The steam says "I am master and in I will go between your surfaces, and I will not come out until you apply a force greater than the area of the surface in contact multiplied by the pressure." This is the fundamental principle, the discovery of which immediately produced a practically frictionless slide-valve.

All engineers are of one accord in ascribing the loss of a large amount of power in working the slide-valve, but there is a large diversity of opinion as to what that amount of power is, solely because there is no correct data from which they may calculate it as they do other losses of power. A paper on the slide-valve would be incomplete did it not bring within its range, the consideration of two questions:—First,—What is the magnitude of the evil of which it complains, and, secondly, what is the remedy for that evil. And in setting about such a task the first thing to be done is to ascertain the co-efficient of friction under the circumstances attending its practical application, and by that ascertained factor to calculate the amount of work performed by the force expressed by the co-efficient of friction passing through space, and by the time in passing through

that space, and upon the principle of work done find the proportion which that power bears to the mean power of the piston.

With these preliminary remarks we will proceed with the investigation of the subject, and first,—What is the amount of power absorbed in working the slide-valves and appendages of the steam engine?

For that purpose we select (Plate I) where Fig. 1* represents the ordinary valve, as applied by Mr. William Buchanan, master mechanic of the Hudson River Railway, to an Express Engine named W. H. Vanderbilt, which is one of the most improved type of the American locomotive, so far as the distribution of the steam is concerned, as the following dimensions and investigation will show. Diameter of driving wheel $5\frac{1}{2}$ feet, speed of engine 30 miles an hour. Diameter of cylinder 17", stroke 24", pressure 120 pounds. Exhaust port $2\frac{3}{4}$ ", admission port $1\frac{1}{8}$ ", inside bar 1", outside bar $2\frac{1}{4}$ ", length of port 16" bar on end of ports 1", outside lap $1\frac{1}{8}$ ", inside lap $\frac{1}{8}$ ", depth of exhaust cavity of valve $2\frac{1}{4}$ ", travel of valve $3\frac{1}{4}$ " in working position.

Fig. 1 is a section of this valve laid in middle position over the ports. Fig 2 is a plan of the same. The outer and inner edges of the valve, by their crossing the outer and inner edges of the bars, determine the distribution of the steam. If, then, we let fall lines MN, &c., from the outer and inner edges of the valve in this position and on the points EE'E''E''' with half the travel of valve as radius describe semi-circles equal to travel of valve, divide these semi-circles into ten equal divisions, corresponding to the ten equal divisions on the indicator diagram Fig. 3. Let the pressure on the back of the valve be represented as positive pressure, and those on the face as negative pressures, measure the horizontal distances from each division to the edge of port or bar, and find what section of surface of valve face is exposed to negative pressure, that section multiplied by the length of port and by the corresponding pressure on the diagram will give the negative force, which taken for the ten divisions and their sums added and divided by ten to give their mean, and that sum subtracted from the total positive force will leave the force available for friction, which, multiplied by the coefficient of friction and by the space passed through in feet per

* The plates, or drawings, referred to have not been sent to us by the author. We expect them, however, shortly, and hope to have them engraved and published in the next number.—(Ed.)

minute by the valve, and divided by 33,000, will give the horse-power required to work the valve.

Fig. 3 is a diagram representing the power developed by the piston where B A is the scale of pressure, A the point of admission, s the point of suppression, E the point of exhaust, and c the point of compression.

Table I contains negative forces taken off the diagram (*indicator*) corresponding to the ten positions of the valve represented by the ten equal divisions of the semi-circle, each division corresponding also with its relative division on the diagram, from which the pressures are all taken. The first column is the number, the second is the sectional breadth, the third is the length in inches, the fourth is the pressure in pounds per square inch taken from the diagram, except when the valve is over the extreme end bar, and then the boiler pressure of 120 pounds is taken. The fifth column is the 2d, 3d and 4th multiplied into each other, which, being done for the four edges of the valve, their sum is taken, and that divided by ten to give their mean negative force, which being subtracted from the total positive force will leave the force available for friction, thus:—

The last column is the back pressures giving a mean of 7·8 pounds. The second last is the propelling pressures from which subtracting the back pressure, gives the mean propelling force of 72·6 pounds. The third last column is the number of square inches bearing on the cylinder face, giving a mean of 69·5, and the positive force available for friction divided by the number of square inches of bearing surface, gives the pressure per square inch of the valve on the cylinder face, and from which we get the co-efficient of friction, thus:—

$\frac{17695}{69.5} = 255$ pounds per square inch, and this, and not the boiler pressure of 120 pounds, is the pressure from which we get a co-efficient, and with cast-iron valve, on cast iron cylinder, it gives a co-efficient of ·28. Then we have

$\frac{3072 + 2603 + 8937 + 8241}{10} = 2285$ equal mean negative force,
 $18 \times 9.25 \times 120 = 19980 - 2285 = 17695$ equal force available for friction.
 $\frac{17695 \times .28 \times 88}{33000} = \frac{4955 \times 88}{33000} = \frac{436005}{33000} = 13.313$, which multiplied by 2, for double stroke, gives $26.626 = \text{H. P. of valves.}$
 $4956 \times .15 = 743$, then $\frac{743 \times 643}{33000} = 14.47$, which doubled, gives

28.94 = H. P. of both eccentrics. $\frac{227 \times 72 \times 648}{53000} = 322$ which also doubled, gives 644 = H. P. of both cylinders, and $\frac{644 - 55.6 \times 100}{644} = 91$ (nearly) per cent. and $180 - 91 = 9$ full per cent. for the friction of valves and eccentrics.

Space swept through by piston per minute, 648 feet,

“ “ “ valve “ 88 “

“ “ “ periphery of eccentric per minute 643 ft.

The following is the mode of calculating the friction of eccentric.

Let *c* be the centre of rotation, and *A* the centre of formation. *B, D, E* the periphery of the eccentric sheath. *B, E*, is the plane of the periphery, and *B E* multiplied by *b e* is the bearing surface of eccentric sheath and strap, and the position load on the valve available for friction, multiplied by the co-efficient of friction, is equal to the direct pull on the valve spindle, and the pull on the spindle divided by the number of square inches in the eccentrics plane, is equal to the load per square inch on eccentric and from which we get the co-efficient .15. That multiplied by the space passed through by the periphery of the eccentric in feet per minute, and divided by 33,000 will give the horse-power required to work the eccentrics. Thus, let *B E* equal 15 and *b e* equal 2.75, then $15 \times 2.75 = 41.25 =$ square inches in eccentric plane $\frac{4956}{4125} = 120$ pounds per square in.

The combined friction of the valves and eccentrics worked by link motion is practically a constant quantity in all positions of gear, the speed remaining constant, but the power developed by the piston is changed with a change in the position of gear, and thence it follows that the percentage of the power of working the valves and eccentrics varies with the position of gear, being least in full gear and greatest in mid-gear, varying with the engine before us from a minimum in full gear of about $6\frac{1}{2}$ per cent. to a maximum in mid-gear of about 15 per cent.

These percentage bear the same relative value to the marine and stationary engines, whose dimensions are all proportional.

The Remedy.

To remove this load from the back of the valve and so save this per centage to the users of steam power, has been the study and anxious desire of all intelligent engineers, but though many have tried, and tried often, yet only failure to many, and that often,

has been the result, for this one cogent reason: They were not acquainted with the forces involved. One class of engineers say the pressure is equal to the whole area of the valve, another, it is equal to the area of the valve, minus the area of the two admission ports, and another, the most numerous of all, say, with all the confidence of right, that it is equal to the area of the ports. They are all wrong together. For let us investigate the subject, and first of a perfect mathematically balanced slide-valve.

Fig. 7 is a section of a valve laid in middle position over the ports. Fig 8 is a plan of the same which shows it to be a parallelogram frame-work with communication ports A, through it. Let c be the cylinder face and s the steam chest cover, F F F are false ports $\frac{1}{4}$ " deep opposite the true ports. Let the steam be let on to this valve, it will act on the four outer sides pressing them equally. The exhaust pressure acts also equally on the four inner sides pressing them also equally. Let the valve now move from its middle position to the commencement of the stroke, the steam enters the port A and will press the valve from the cylinder face by a force equal to the area of the valve over the port, but it, at the same time, enters the false ports F pressing the valve to the cylinder with an equal and opposite force. Let it now move on to full port gradually decreasing the area of the valve exposed to the pressure in the port, but it as gradually decreases the area of the false port, and now, having got to full port, it returns gradually closing the port and increasing the area exposed to the steam. It has done the same over the false port and now the outer edge of the valve crosses the outer edge of the port, proclaiming the point of suppression and introducing the period of expansion. Commanding the pressure within the port to decrease, according to the law of gaseous expansion; but the communication port A permits of a free intercourse between the pressures in the true and false ports, until the inner edge of the valve crosses the inner edge of the bar, demanding the release of the steam from imprisonment in the cylinder. It rushes out through the port A with a velocity equal to the conditions of its own constitution, and enjoys the freedom of space outside the blast orifice, but it has, at the same instant, let free its companion in bondage in the false port. At the same time the opposite end of the valve closes the port to exhaust, and the steam, which was not released on the previous stroke, cannot now get out, but is compressed before the advancing piston, gradually raising the pressure

in the port B, but the communication port *a*, ever faithful to its designer, again permits the pressures in the true and false ports to hold sweet communion with each other, and thus, it is seen, that the pressure cannot touch this valve on any one point without touching it on an equal and opposite point and practically at the same instant of time, and hence a perfectly mathematically balanced valve.

But, beautiful as the principle of this valve is, it contains the elements of self destruction in a large degree. It is a solid block manipulated by workmanship of the finest kind, and so fitted between two solid surfaces, such that it will move with ease between those surfaces, and at the same time be steam tight. This can be done to begin with conditionally. The author applied this to five locomotives on the London and Brighton and North London Railways, and its faults are: First, it is an inflexible block between two inflexible surfaces, and with the locomotive, whenever steam is shut off, the compressed air in the cylinder acts in front of the piston, and so powerful is it that it throws the driver forward on the fire-box and shakes every passenger in the train, and an engine will not run down a gradient of one in seventy at more than five miles per hour without steam on. Second, steam of high temperature cuts the surfaces rapidly, and soon permits a leak of steam, and thirdly, when it has been fitted and made steam tight the surfaces do touch each other, and no matter with how little force, they will run all right while steam is on, but shut off steam to run down a hill, and at any stroke the little touch generates heat, the heat expands the metal in the valve and does not expand the metal forming the walls of the steam chest. The valve sticks and the spindles are carried away, putting a period to the existence of this beautiful valve.

This is not therefore the valve best suited to practice, but one that will give and take to the inequalities and irregularities of the practice; for which purpose we introduce to your notice the valve of which Fig. 9 is a section and Fig. 10 a plan. Fig. 11 is also a half-section and at right angles to Fig. 9. Where *v* is the valve which is built up cylindrically to receive a ring *D* which displaces the required amount of steam from the back of the valve, so that at the mean position the negative forces shall be equal. *R* is a packing ring to make a steam tight joint between *v* and *D*. *F* is the door or cover of the steam chest, *s* is a three ounce spring to hold *D* against

F, and v against the cylinder face, and H H are two small holes to allow the exhaust pressure of steam acting in the cavity of valve to come through the back of the valve.

Steam being admitted to this valve, it will act under the head of D pressing it hard against F, but it also acts between the head and F with an equal force, no matter how fine the surfaces are scraped, and as the areas above and below are equal, it is held in equilibrium of the forces. The exhaust pressures acting in cavity c pass freely through H H, and acting equally on both sides of D internally, hold it also in balance. In movement also the forces on the valve balance each other.

Within certain limits of the balance of the valve, determined by a series of experiments by the author, extending over a period of four years (and numbering altogether upwards of eleven thousand) it has been determined by the author that, practically, the forces are balanced over the spaces *aa* and *c*, because the spaces are equal and opposite, but the spaces *bb* and *c* are not balanced because they have the full boiler pressure of 120 pounds on them as positive force, while there is but the exhaust pressure of 3 pounds below, it is therefore an unbalanced positive force. Nor are the spaces *b' b'* and *c* balanced either, for while they have the exhaust pressure above, they are exposed to the full initial pressure below. Moreover the spaces are neither equal nor opposite, but our task is to balance them. Fig. 12 is a section of Buchanan's valve, as given to the author by that gentleman, in order that the author's valve may be applied in its stead. It is required to find the diameter of displacement ring such that the forces may balance each other.

Draw the centre lines and let their intersection *c* be the centre of the valve, of which Fig. 13 is a plan. On *c*, as a centre describe any circle *DEF*, draw the line *nn* leaving the spaces *n' n'* equal to the space *m*. We have now got a triangle to calculate which is easy. Bisect the base of the triangle *no* and from it draw a line to the opposite angle *n*, one-third that distance is the centre of action of that space, *N* and the base multiplied by half the perpendicular height gives us the area, and the area multiplied by the distance of its centre of action from the centre of the valve gives the moment of the force.

Similarly for *P* draw the line *ny*, leaving the space inside the line of displacement equal to that outside, draw the diagonals and

the point of intersection in the centre of action of the rectangular space *P* and one side multiplied by the other and multiplied by the distance of its centre of action from the centre of the valve, gives the moment of the force *P*. The moment of the two forces *N* and *P* being equal, the valve will be practically balanced, thus: $3.7 \times 2.85 \times 4 = 42.18 =$ moment of negative force. $2.25 \times 2.85 \times 6.5 = 42.33 =$ moment of positive force, and a valve so proportioned and constructed (Fig. 9) will move as easy with steam of any pressure on, as off.

In proof of the truth of this theory the author purposes to lay before you a few of his experiments bearing directly on the subject, from which you will see the mode of reasoning he adopted in arriving at these conclusions.

Having discovered the law that to hold any two surfaces of metal together, moving in molecular contact against a heated fluid under pressure, it required a force equal to the area of the surface in contact multiplied by the pressure against which the body moves. The following experiments with the displacement ring were to prove that law. Fig 14 is a plain ring *D*, the section of which is three-quarters of an inch thick. Conceive it to be parallel up to the steam chest cover *F*, the bearing surface against *E* is $\frac{3}{4}$ " and a strong spring *S*, of any force, is put under it to hold it tight against *F*. Steam of low pressure is let in on the valve, 3 pounds to the square inch, the valve is in motion and all is tight, the pressure rises to 10, 15, 20, 25, and at 26 pounds a blow takes place; the higher the pressure, the greater the blow. The ring is taken out and a quarter of an inch recessed out of the head, as shown at *E*, leaving only a half-inch bearing against *F*, when steam is again let on and at 26 pounds all is tight, so at 30, 35, and at 39 a blow was the result; the higher the pressure the greater the blow. The ring was again taken out and another quarter of an inch recessed out of its head, as shown at *E'*, leaving only a quarter of an inch now bearing against *F*. When steam was again let on and at 26 and 39 pounds all was tight, and at 78 pounds a blow again took place, the higher the pressure the greater the blow. It was now taken out and the spring weighted to its working deflection, when it showed a difference of 1 pound 14 ounces only per square inch, from the area of the surface in contact multiplied by the pressure at the point of blow.

It was next applied to a Locomotive on the North London Rail-
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way by Mr. William Adams, and at the same time to one on the Great Eastern Railway by Mr. Sinclair, in the form of Fig. 15. The head of the ring bearing against the door is half an inch wide, and the body of the ring five-eighths thick, the head of the ring is recessed a sixteenth of an inch. Two strong springs were put under it to wear the sixteenth away, and so bring the body of the ring up on to the door. It did so, and a blow was the result in both cases, the extra surfaces against the door being $1\frac{1}{2}$ square feet. It was taken out, and as it did not wear equally all around, and a ring with a head, as shown in Fig. 16, put in. In four months we got a blow there also, it was taken out and the top surface measured three thirty-seconds of an inch more than the bottom. It was recessed, as shown in Fig. 17, and will have been four years at work come the first day of August next.

The question will be asked how do we account for this variation in the areas of the top and bottom surfaces of the displacement ring.

Fig. 18 is a plan of the head of the ring, and c the centre of formation. Square the outer and inner radii of the ring, add the sums together, divide by two and extract the square root of the remainder, the quotient will give the radius of the centre of action of the head of ring, and the centre of action of the semi-circle bounded by the outer and inner radii will be $\cdot 707$ of the radius of the centre of action of the area, thus,

Let c be the centre of formation and c' c'' the outer and inner radii, and c''' the centre of action of area, and c'''' the centre of action of the semi-circle.

With motion in the direction of arrow A'' it is evident that the semi-circle A is moving against the full initial pressure of the boiler 120 pounds, while the semi-circle A' is moving against the exhaust pressure of three pounds it is evident that the centre c cannot be the common centre of action of the two semi-circles, but in some point towards the greater of the two pressures.

Let T equal the temperature of 120 pounds, measured from the absolute zero of temperature, equal 794° Fahr. and T' equal the temperature of 3 pounds, also measure from the absolute zero, equal to 713° and let c c'''' be 3.75 inches, then will

$$794 \times 3.75 - 713 \times 3.75 = 204 \text{ of an inch in the direction of the}$$

larger temperature, giving the proportions of 7 to 8 as the propor-

tion of bottom to top surface of head of displacement ring, and this rule will give the correct proportion to all variation of diameters and pressures.

This brings us now to the application of the author's valve to the marine engine, for which purpose we select the Franklin, the Flag ship of the United States Navy, as applied by Chief Engineer W. W. Wood, of the N. Y. Navy Yard.

Fig. 19 is a section which shows it a double ported valve, Fig. 20 is a half plan. Then as before describe any circle B, D, F, set off the spaces N N' N'' N''' as unbalanced negative areas and find their centres of action, then will each area multiplied by the distance of its centre of action from the edge FF of the valve, give the moment of its respective force, and their sums added will give the total negative force.

Similarly for the positive force set off the spaces P, P', P'', P''', and find their centres of action. Their areas multiplied by the distance with which their centres act from FF gives the moment of their forces, their sums added will give the positive force, thus,

$$3.1 \times 1.1 \times .5 = 1.7$$

$$8.7 \times 1.4 \times 4.5 = 52.19$$

$$10. \times 3.2 \times 18.3 = 585.64$$

$$5.7 \times 2.5 \times 22.6 = 287.$$

$$1.7 + 5.2 + 58.6 + 287 = 927 = \text{moment of}$$

negative force.

$$10. \times 2.5 \times 2.25 = 56$$

$$6.3 \times 2.4 \times 6.1 = 92$$

$$8.9 \times 4.7 \times 11.9 = 497$$

$$8.3 \times 1.7 \times 20. = 282$$

$$56 + 92 + 497 + 282 = 927 \text{ equal moment}$$

of positive force. The moments being equal the balance is complete.

It will be seen that this valve is laid at the commencement of the stroke. The effort of the negative forces is to turn the valve off the edge FF as a fulcrum, and they would do it, too, were the moment of their sums greater than the moment of the sum of the positive forces and the valve would leave the cylinder face.

We have next to prove that the pressure on the back of the valve is not equal to the area of the ports. This being the prevailing idea of engineers.

Fig. 21 is Buchanan's valve. The elements of the valve, lap, lead and travel are the same, and the ports are all the same,

length and breadth. The only dimensions that are altered is the width of the inner bar on the cylinder face, in the former case. Fig. 12 the bar is one inch. In the latter, Fig. 21, it is three inches. In Fig. 13, applying the author's rule, the area of displacement is 84.5 square inches. In Fig. 22 it is 154 square inches, which, with 120 pounds of steam, shows a difference of 8.400 pounds on each valve. The same rule being applied in both cases, the prevailing idea is therefore as false as the bottom of a tailor's thimble.

The next rule is that which makes the weight on the back of the valve equal to the whole area of the valve. Such an assertion requires only to be made in order that it may be condemned, for it ignores the existence of any negative force at all.

And, lastly, we have those who say that it is equal to the whole area of the valve, minus the two admission ports.

Fig. 23 shows the valve upon which the author made his experiments, beginning by displacing the whole of the area of valve, minus the area of the two admission ports, represented by the outer diameter of ring, the result was that the valve left the cylinder face, and a blow of steam through the ports was the result, rings of smaller and smaller diameters were then applied, decreasing by $\frac{1}{8}$ or $\frac{1}{16}$ inch at a time. The valve always leaving the cylinder face which showed plainly that the negative force always prevailed over the positive, until, at last, one was fitted and the valve kept the cylinder face. It was taken out and measured, and the circle A, B, C, represents the displacement, from which the rule was made.

Experiments were also made to determine what amount of load was necessary to produce friction, and whether or not an uniform increase of load would be met by an uniform increase in the co-efficient of friction. It was not so, for friction was only produced by a load of seven per cent. with superheated steam 560° temperature, while with saturated steam of 10 pounds pressure it required 34 per cent. to produce friction, and it is not uniform, but sudden.

These experiments leave no doubt whatever of the existence of a film adhering to the surface of the metals, and this film has been the stumbling block and rock of offence which has deceived all engineers; for, give the relation of surface to pressure, and almost any engineer has enough mechanical skill to build the structure. The valve, in moving against it, slides over it, and so long as its temperature is maintained, it possesses all the energy of the surrounding steam. It is an elastic fluid possessed of energy. It

will therefore perform a given amount of work equal to its equivalent of heat, which it does in supporting the load, as the valve moves over it, and when its heat is destroyed friction sets in.

The fulcrum of both forces is continually changing when the valve is in motion. The positive traversing about $1\frac{1}{2}$ inches on either side of the centre of the valve, while the negative traverses the whole length of the valve during one revolution, and thus an incessant war is set up between area on the one hand, and leverage on the other. At one time the greater area of negative force prevails and says to the lesser area of positive force, "I will push the valve from the cylinder face;" when the positive force replies "I will exercise my prerogative of leverage and will not allow you." The area says "I am the biggest and you must obey." The leverage says "I am the more powerful and will be master," and thus by the war of area on the one hand and leverage on the other, the valve is incessantly being thrown from the cylinder face, and as incessantly being thrown to it, and thus it has no peace. The molecules of the valve face can never embrace those of the cylinder face in the bonds of friction, but by it the valve is made to float on the fluid skin of the surfaces, and hence the ease with which it is moved.

London, December 10th, 1869.

Glasgow Harbor Extension.—M. J. F. Bateman, consulting engineer to the Clyde Trust, and Mr. James Deans, the lately appointed resident engineer, have given a large amount of attention to this subject, it being the intention of the Trustees to go to Parliament in the ensuing session for the necessary powers to enable them to proceed with such harbor extensions as the circumstances of the case, present and prospective, demand. Taking in consideration the varying increase of quayage from time to time, the engineers determined to calculate upon 200 yards per annum as the required rate of increase for the next ten years. "At the same progressive rate of increase in the trade" they proceed, "the additional quayage required on the average of the following ten years will be about 800 lineal yards per annum, showing a mean of 250 yards for the next twenty years.—*Engineering*."

Mineralogy of Yosemite Valley.—Garnets, sphene molybdenite, hornblende, and other interesting materials, are reported by Mr. J. H. Casewell to be plentiful in the granite of Yosemite Valley.