CHAPTER VI.

PROPORTIONS OF ENGINES.

STEAM PASSAGES.

312. *Q.*—What size of orifice is commonly allowed for the escape of the steam through the safety valve in low pressure engines?

A.—About 0·8 of a circular inch per horse power, or a circular inch per $1\frac{1}{4}$ horse power. The following rule, however, will give the dimensions suitable for all kinds of engines, whether high or low pressure:—multiply the square of the diameter of the cylinder in inches by the speed of the piston in feet per minute, and divide the product by 375 times the pressure on the boiler per square inch; the quotient is the proper area of the safety valve in square inches. This rule of course supposes that the evaporating surface has been properly proportioned to the engine power.

313. *Q.*—Is this rule applicable to locomotives?

A.—It is applicable to high pressure engines of every kind. The dimensions of safety valves, however, in practice are very variable, being in some cases greater, and in some cases less, than what the rule gives, the consideration being apparently as often what proportions will best prevent the valve from sticking in its seat, as what proportions will enable the steam to escape freely. In Bury's locomotives, the safety valve was generally $2\frac{1}{2}$ inches diameter for all sizes of boiler, and the valve was kept down by a lever formed in the proportion of 5 to 1, fitted at one end with a Salter's balance. As the area of the valve was 5 square inches, the number of pounds shown on the spring balance denoted the number of pounds pressure on each square inch of the boiler.

314. *Q.*—Is there only one safety valve in a locomotive boiler?

A.—There are always two.

315. *Q.*—And are they always pressed down by a spring balance, and never by weights?

A.—They are never pressed down by weights; in fact, weights would not answer on a locomotive at all, as they would jump up and down with the jerks or jolts of the train, and cause much of the steam to escape. In land and marine boilers, however, the safety valve is always kept down by weights; but in steam vessels a good deal of steam is lost in stormy weather by the opening of the valve, owing to the inertia of the weights when the ship sinks suddenly in the deep recess between the waves.

316. *Q.*—What other sizes of safety valves are used in locomotives?

A.—Some are as large as 4 inches diameter, giving 12 square inches of area; and others are as small as $1\frac{3}{16}$ inch diameter, giving 1 square inch of area.

317. *Q.*—And are these valves all pressed down by a Salter's spring balance?

A.—In the great majority of cases they are so, and the lever by which they are pressed down is generally graduated in the proportion of the area of the valve to unity; that is, in the case of a valve of 12 inches area, the long end of the lever to which the spring balance is attached is 12 times the length of the short end, so that the weight or pressure on the balance shows the pressure per square inch on the boiler. In some cases, however, a spiral spring, and in other cases a pile of elliptical springs, is placed directly upon the top of the valve, and it appears desirable that one of the valves at least should be loaded in this manner. It is difficult when the lever is divided in such a proportion as 12 to 1, to get sufficient lift of the valve without a large increase of pressure on the spring; and it appears expedient, therefore, to employ a shorter lever, which involves either a reduction in the area of the valve, or an increased strength in the spring.

318. *Q.*—What are the proper dimensions of the steam passages?

A.—In slow working engines the common size of the cylinder passages is one twenty-fifth of the area of the cylinder, or one fifth of the diameter of the cylinder, which is the same thing. This proportion corresponds very nearly with one square inch per horse power when the length of the cylinder is about equal to its diameter; and one square inch of area per horse power for the cylinder ports and eduction passages answers very well in the case of engines working at the ordinary speed of 220 feet per minute. The area of the steam pipe is usually made less than the area of the eduction pipe, especially when the engine is worked expansively, and with a considerable pressure of steam. In the case of ordinary condensing engines, however, working with the usual pressure of from 4 to 8 lbs. above the atmosphere, the area of the steam pipe is not less than a circular inch per horse power. In such engines the diameter of the steam pipe may be found by the following rule: divide the number of nominal horse power by 0·8 and extract the square root of the quotient, which will be the internal diameter of the steam pipe.

319. *Q.*—Will you explain by what process of computation these proportions are arrived at?

A.—The size of the steam pipe is so regulated that there will be no material disparity of pressure between the cylinder and boiler; and in fixing the size of the eduction passage the same object is kept in view. When the diameter of the cylinder and the velocity with which the piston travels are known, it is easy to tell what the velocity of the steam in the steam pipe will be; for if the area of the cylinder be 25 times greater than that of the steam pipe, the steam in the steam pipe must travel 25 times faster than the piston, and the difference of pressure requisite to produce this velocity of the steam can easily be ascertained, by finding what height a column of steam must be to give that velocity, and what the weight or pressure is of such a column. In practice, however, this proportion is always exceeded from the condensation of steam in the pipe.

320. *Q.*—If the relation you have mentioned subsist between the area of the steam passages and the velocity of the piston, then the passages must be larger when the piston travels very rapidly?

A.—And they are so made. The area of the ports of locomotive engines is usually so proportioned as to be from $\frac{1}{10}$th to $\frac{1}{8}$th the area of the cylinder—in some cases even as much as $\frac{1}{6}$th; and in all high speed engines the ports should be very large, and the valve should have a good deal of travel so as to open the port very quickly. The area of port which it appears advisable to give to modern engines of every description, is expressed by the following rule:—multiply the area of the cylinder in square inches by the speed of the piston in feet per minute, and divide the product by 4,000; the quotient is the area of each cylinder port in square inches. This rule gives rather more than a square inch of port per nominal horse power to condensing engines working at the ordinary speed; but the excess is but small, and is upon the right side. For engines travelling very fast it gives a good deal more area than the common proportion, which is too small in nearly every case. In locomotive engines the eduction pipe passes into the chimney and the force of the issuing steam has the effect of maintaining a rapid draught through the furnace as before explained. The orifice of the waste steam pipe, or the blast pipe as it is termed, is much contracted in some engines with the view of producing a fiercer draught, and an area of $\frac{1}{22}$d of the cylinder is a common proportion; but this is as much contraction as should be allowed, and is greater than is advisable.

321. *Q.*—In engines moving at a high rate of speed, you

have stated that it is important to give the valve lead, or in other words to allow the steam to escape before the end of the stroke?

A.—Yes, this is very important, else the piston will have to force out the steam from the cylinder, and will be much resisted. Near the end of the stroke the piston begins to travel slowly, and if the steam be then permitted to escape, very little of the effective stroke is lost, and time is afforded to the steam, before the motion of the piston is again accelerated, to make its escape by the port. In some locomotives, from inattention to this adjustment, and from a contracted area of tube section, which involved a strong blast, about half the power of the engine has been lost; but in more recent engines, by using enlarged ports and by giving sufficient lead, this loss has been greatly diminished.

322. *Q.*—What do you call sufficient lead?

A.—In fast going engines I would call it sufficient lead, when the eduction port was nearly open at the end of the stroke.

323. *Q.*—Can you give any example of the benefit of increasing the lead?

A.—The early locomotives were made with very little lead, and the proportions were in fact very much the same as those previously existing in land engines. About 1832, the benefits of lap upon the valve, which had been employed by Boulton and Watt more than twenty years before, were beginning to be pretty generally apprehended; and, in the following year, this expedient of economy was applied to the steamer Manchester, in the Clyde, and to some other vessels, with very marked success. Shortly after this time, lap began to be applied to the valves of locomotives, and it was found that not only was there a benefit from the operation of expansion, but that there was a still greater benefit from the superior facility of escape given to the steam, inasmuch as the application of lap involved the necessity of turning the eccentric round upon the shaft, which caused the eduction to take place before the end of the stroke. In 1840, one of the engines of the Liverpool and Manchester

Railway was altered so as to have 1 inch lap on the valve, and 1 inch opening on the eduction side at the end of the stroke, the valve having a total travel of $4\frac{1}{4}$ inches. The consumption of fuel per mile fell from 36·3 lbs. to 28·6 lbs, or about 25 per cent., and a softer blast sufficed. By using larger exhaust passages, larger tubes, and closer fire bars, the consumption was subsequently brought down to 15 lbs. per mile.

AIR PUMP, CONDENSER, AND HOT AND COLD WATER PUMPS.

324. *Q.*—Will you state the proper dimensions of the air pump and condenser in land and marine engines?

A.—Mr. Watt made the air pump of his engine half the diameter of the cylinder and half the stroke, or one eighth of the capacity, and the condenser was usually made about the same size as the air pump; but as the pressure of the steam has been increased in all modern engines, it is better to make the air pump a little larger than this proportion. 0·6 of the diameter of the cylinder and half the stroke answers very well, and the condenser may be made as large as it can be got with convenience, though the same size as the air pump will suffice.

325. *Q.*—Are air pumps now sometimes made double acting?

A.—Most of the recent direct acting marine engines for driving the screw are fitted with a double acting air pump, and when the air pump is double acting, it need only be about half the size that is necessary when it is single acting. It is single acting in nearly every case, except the case of direct acting screw engines of recent construction.

326. *Q.*—What is the difference between a single and a double acting air pump?

A.—The single acting air pump expels the air and water from the condenser only in the upward stroke of the pump, whereas a double acting air pump expels the air and water both in the upward and downward stroke. It has, therefore, to be provided with inlet and outlet valves at both ends, whereas the single acting pump has only to be provided with an inlet or foot valve, as it is termed, at the bottom, and with an outlet or delivery valve, as it is termed, at the top. The single acting air pump requires to be provided with a valve or valves in the piston or bucket of the pump, to enable the air and water lying below the bucket when it begins to descend, and which have entered from the condenser during the upward stroke, to pass through the bucket into the space above it during the downward stroke, from whence they are expelled into the atmosphere on the upward stroke succeeding. But in the double acting air pump no valve is required in the piston or bucket of the pump, and all that is necessary is an inlet and outlet valve at each end.

327. *Q.*—What are the dimensions of the foot and discharge valves of the air pump?

A.—The area through the foot and discharge valves is usually made equal to one fourth of the area of the air pump, and the diameter of the waste water pipe is made one fourth of the diameter of the cylinder, which gives an area somewhat less than that of the foot and discharge valve passages. But this proportion only applies in slow engines. In fast engines, with the air pump bucket moving as fast as the piston, the area through the foot and discharge valves should be equal to the area of the pump itself, and the waste water pipe should be of about the same dimensions.

328. *Q.*—You have stated that double acting air pumps need only be of half the size of single acting ones. Does that relation hold at all speeds?

A.—It holds at all speeds if the velocity of the pump buckets are in each case the same; but it does not hold if the engine with the single acting pump works slowly, and the engine with the double acting pump moves rapidly, as in the case of direct acting screw engines. All pumps moving at a high rate of speed lose part of their efficiency, and such pumps should therefore be of extra size.

329. *Q.*—How do you estimate the quantity of water requisite for condensation?

A.—Mr. Watt found that the most beneficial temperature of the hot well of his engines was 100 degrees. If, therefore, the temperature of the steam be 212°, and the latent heat 1,000°, then 1,212° may be taken to represent the heat contained in the steam, or 1,112° if we deduct the temperature of the hot well. If the temperature of the injection water be 50°, then 50 degrees of cold are available for the abstraction of heat; and as the total quantity of heat to be abstracted is that requisite to raise the quantity of water in the steam 1,112 degrees, or 1,112 times that quantity one degree, it would raise one fiftieth of this, or 22·24 times the quantity of water in the steam, 50 degrees. A cubic inch of water therefore raised into steam will require 22·24 cubic inches of water at 50 degrees for its condensation, and will form therewith 23·24 cubic inches of hot water at 100 degrees. Mr. Watt's practice was to allow about a wine pint (28·9 cubic inches) of injection water, for every cubic inch of water evaporated from the boiler.

330. *Q.*—Is not a good vacuum in an engine conducive to increased power?

A.—It is.

331. *Q.*—And is not the vacuum good in the proportion in which the temperature is low, supposing there to be no air leaks?

A.—Yes.

332. *Q.*—Then how could Mr. Watt find a temperature of 100° in the water drawn from the condenser, to be more beneficial than a temperature of 70° or 80°, supposing there to be an abundant supply of cold water?

333. *A.*—Because the superior vacuum due to a temperature of 70° or 80° involves the admission of so much cold water into the condenser, which has afterward to be pumped out in opposition to the pressure of the atmosphere, that the gain in the vacuum does not equal the loss of power occasioned by the additional load upon the pump, and there is therefore a clear loss by the reduction of the temperature below 100°, if such reduction be caused by the admission of an additional quantity of water. If the reduction of temperature, however, be caused by the use of colder water, there is a gain produced by it, though the gain will within certain limits be greater if advan-

tage be taken of the lowness of the temperature to diminish the quantity of injection.

334. *Q.*—How do you determine the proper area of the injection orifice?

A.—The area of the injection orifice proper for any engine can easily be told when the quantity of water requisite to condense the steam is known, and the pressure is specified under which the water enters the condenser. The vacuum in the condenser may be taken at 26 inches of mercury, which is equivalent to a column of water 29·4 ft. high, and the square root of 29·4 multiplied by 8·021 is 43·15, which is the velocity in feet per second that a heavy body would acquire in falling 29·4 ft., or with which the water would enter the condenser. Now, if a cubic foot of water evaporated per hour be equivalent to an actual horse power, and 28·9 cubic inches of water be requisite for the condensation of a cubic inch of water in the form of steam, 28·9 cubic feet of condensing water per horse power per hour, or 13·905 cubic inches per second, will be necessary for the engine, and the size of the injection orifice must be such that this quantity of water flowing with the velocity of 43·15 ft. per second, or 517·8 inches per second, will gain admission to the condenser. Dividing, therefore, 13·905, the number of cubic inches to be injected, by 517·8, the velocity of influx in inches per second, we get 0·02685 for the area of the orifice in square inches; but inasmuch as it has been found by experiment that the actual discharge of water through a hole in a thin plate is only six tenths of the theoretical discharge on account of the contracted vein, the area of the orifice must be increased in the proportion of such diminution of effect, or be made 0·04475, or $\frac{1}{22}$d of a square inch per horse power. This, it will be remarked, is the theoretical area required per actual horse power; but as the friction and contractions in the pipe further reduce the discharge, the area is made $\frac{1}{15}$th of a square inch per actual horse power, or rather per cubic foot of water evaporated from the boiler.

335. *Q.*—Cannot the condensation of the steam be accomplished by any other means than by the admission of cold water into the condenser?

A.—It may be accomplished by the method of external cold, as it is called, which consists in the application of a large number of thin metallic surfaces to the condenser, on the one side of which the steam circulates, while on the other side there is a constant current of cold water, and the steam is condensed by coming into contact with the cold surfaces, without mingling with the water used for the purpose of refrigeration. The first kind of condenser employed by Mr. Watt was constructed after this fashion, but he found it in practice to be inconvenient from its size, and to become furred up or incrusted when the water was bad, whereby the conducting power of the metal was impaired. He therefore reverted to the use of the jet of cold water, as being upon the whole preferable. The jet entered the condenser instead of the cylinder as was the previous practice, and this method is now the one in common use. Some few years ago, a good number of steam vessels were fitted with Hall's condensers, which operated on the principle of external cold, and which consisted of a faggot of small copper tubes surrounded by water; but the use of those condensers has not been persisted in, and most of the vessels fitted with them have returned to the ordinary plan.

336. *Q.*—You stated that the capacity of the feed pump was $\frac{1}{240}$th of the capacity of the cylinder in the case of condensing engines,—the engine being double acting and the pump single acting,—and that in high pressure engines the capacity of the pump should be greater in proportion to the pressure of the steam. Can you give any rule that will express the proper capacity for the feed pump at all pressures?

A.—That will not be difficult. In low pressure engines the pressure in the boiler may be taken at 5 lbs. above the atmospheric pressure, or 20 lbs. altogether; and as high pressure steam is merely low pressure steam compressed into a smaller compass, the size of the feed pump in relation to the size of the cylinder must obviously vary in the direct proportion of the pressure; and if it be $\frac{1}{240}$th of the capacity of the cylinder when the total pressure of the steam is 20 lbs., it must be $\frac{1}{120}$th of the capacity of the cylinder when the pressure is 40 lbs. per square inch, or 25 lbs. per square inch above the atmospheric pressure. This law of variation is expressed by the following rule:—multiply the capacity of the cylinder in cubic inches by the total pressure of the steam in lbs. per square inch, or the pressure per square inch on the safety valve plus 15, and divide the product by 4,800; the quotient is the capacity of the feed pump in cubic inches, when the feed pump is single acting and the engine double acting. If the feed pump be double acting, or the engine single acting, the capacity of the pump must just be one half of what is given by this rule.

337. *Q.*—But should not some addition be made to the size of pump thus obtained if the pump works at a high rate of speed?

A.—No; this rule makes allowance for defective action. All pumps lift much less water than is due to the size of their barrels and the number of their strokes. Moderately good pumps lose 50 per cent. of their theoretical effect, and bad pumps 80 per cent.

338. *Q.*—To what is this loss of effect to be chiefly ascribed?

A.—Mainly to the inertia of the water, which, if the pump piston be drawn up very rapidly, cannot follow it with sufficient rapidity; so that there may be a vacant space between the piston and the water; and at the return stroke the momentum of the water in the pipe expends itself in giving a reverse motion to the column of water approaching the pump. Messrs. Kirchweger and Prusman, of Hanover, have investigated this subject by applying a revolving cock at the end of a pipe leading from an elevated cistern containing water, and the water escaped at every revolution of the cock in the same manner as if a pump were drawing it. With a column of water of 17 feet, they found that at 80 revolutions of the cock per minute, the water delivered per minute by the cock was 9·45 gallons; but with 140 revolutions of the cock per minute, the water delivered per minute by the cock was only 5·42 gallons. They subsequently applied an air vessel to the pipe beside the cock, when the discharge rose to 12·9 gallons per minute with 80 revolutions, and 18·28 gallons with 140 revolutions. Air vessels should therefore be applied to the suction side of fast moving pumps, and this is now done with good results.

339. *Q.*—What are the usual dimensions of the cold water pump of land engines?

A.—If to condense a cubic inch of water raised into steam 28·9 cubic inches of condensing water are required, then the cold water pump ought to be 28·9 times larger than the feed pump, supposing that its losses were equally great. The feed pump, however, is made sufficiently large to compensate for leaks in the boiler and loss of steam through the safety valve, so that it will be sufficient if the cold water pump be 24 times larger than the feed pump. This ratio is preserved by the following rule:—multiply the capacity of the cylinder in cubic inches by the total pressure of the steam per square inch, or the pressure on the safety valve plus 15, and divide the product by 200. The quotient is the proper capacity of the cold water pump in cubic inches when the engine is double acting, and the pump single acting.

FLY WHEEL.

340. *Q.*—By what considerations do you determine the dimensions of the fly wheel of an engine?

A.—By a reference to the power generated, each half stroke of the engine, and the number of half strokes that are necessary to give to the fly wheel its standard velocity, supposing the whole power devoted to that object. In practice the power resident in the fly varies from $2\frac{1}{2}$ to 6 times that generated each half stroke; and if the weight of the wheel be equal to the pressure on the piston, its velocity must be such as it would acquire by falling through a height equal to from $2\frac{1}{2}$ to 6 times the stroke, according to the purpose for which the engine is intended. If a very equable motion is required, a heavier or swifter fly wheel must be employed.

341. *Q.*—What is Boulton and Watt's rule for fly wheels?

A.—Their rule is one which under any given circumstances fixes the sectional area of the fly wheel rim, and it is as follows:—multiply 44,000 times the square of the diameter of the cylinder in inches by the length of the stroke in feet, and divide this product by the product of the square of the number of revolutions of the fly wheel per minute, multiplied by the cube of its diameter in feet. The quotient is the area of section of the fly wheel rim in square inches.

STRENGTHS OF LAND ENGINES.

342. *Q.*—Can you give a rule for telling the proper thickness of the cylinders of steam engines?

A.—In low pressure engines the thickness of metal of the cylinder, in engines of a medium size, should be about $\frac{1}{40}$th of the diameter of the cylinder, which, with a pressure of steam of 20 lbs. above the atmosphere, will occasion a strain of only 400 lbs. per square inch of section of the metal; the thickness of the metal of the trunnion bearings of oscillating engines should be $\frac{1}{32}$d of the diameter of the cylinder, and the breadth of the bearing should be about half its diameter. In high pressure engines the thickness of the cylinder should be about $\frac{1}{16}$th its diameter, which, with a pressure of steam of 80 lbs. upon the square inch, will occasion a strain of 640 lbs. upon the square inch of section of the metal; and the thickness of the metal of the trunnion bearings of high pressure oscillating engines should be $\frac{1}{13}$th of the diameter of the cylinder. The strength, however, is not the sole consideration in proportioning cylinders, for they must be made of a certain thickness, however small the pressure is within them, that they may not be too fragile, and will stand boring. While, also, an engine of 40 inches diameter would be about one inch thick, the thickness would not be quite two inches in an 80 inch cylinder. In fact there will be a small constant added to the thickness for all diameters, which will be relatively larger the smaller the cylinders become. In the cylinders of Penn's 12 horse power engines, the diameter of cylinder being $21\frac{1}{2}$ inches, the thickness of the metal is $\frac{9}{16}$ths: in Penn's 40 inch cylinders, the thickness is 1 inch, and in the engines of the Ripon, Pottinger, and Indus, by Messrs. Miller, Ravenhill and Co., with cylinders 76 inches diameter, the thickness of the metal is $1\frac{1}{16}$. These are all oscillating engines.

343. *Q.*—What is the proportion of the piston rod?

A.—The diameter of the piston rod is usually made $\frac{1}{10}$th of the diameter of the cylinder, or the sectional area of the piston rod is $\frac{1}{100}$th of the area of the cylinder. This proportion, however, is not applicable to locomotive, or even fast moving marine engines. In locomotive engines the piston rod is made $\frac{1}{7}$th of the diameter of the cylinder, and it is obvious that where the pressure on the piston is great, the piston rod must be larger than when the pressure on the piston is small.

344. *Q.*—What are the proper dimensions of the main links of a land beam engine?

A.—The sectional area of the main links in land beam engines is $\frac{1}{113}$th of the area of the cylinder, and the length of the main links is usually half the length of the stroke.

345. *Q.*—What are the dimensions of the connecting rod of a land engine?

A.—In land engines the connecting rod is usually of cast iron with a cruciform section: the breadth across the arms of the cross is about $\frac{1}{20}$th of the length of the rod, the sectional area at the centre $\frac{1}{28}$th of the area of the cylinder, and at the ends $\frac{1}{35}$th of the area of the cylinder: the length of the rod is usually $3\frac{1}{2}$ times the length of the stroke. It is preferable, however, to make the connecting rod of malleable iron, and then the dimensions will be those proper for marine engines.

346. *Q.*—What was Mr. Watt's rule for the connecting rod?

A.—Some of his connecting rods were of iron and some of wood. To determine the thickness when of wood, multiply the square of the diameter of the cylinder in inches by the length of the stroke in feet, and divide the product by 24. Extract the fourth root of the quotient, which is the thickness in inches. For iron the rule is the same, only the divisor was 57·6 instead of 24.

347. *Q.*—What are the dimensions of the end studs of a land engine beam?

A.—In low pressure engines the diameter of the end studs of the engine beam are usually made $\frac{1}{9}$th of the diameter of the cylinder when of cast iron, and $\frac{1}{10}$th when of wrought iron, which gives a load with low steam of about 500 lbs. per circular inch of transverse section; but a larger size is preferable, as with large bearings the brasses do not wear so rapidly and the straps are not so likely to be burst by the bearings becoming oval. These sizes, as also those which immediately follow, suppose the pressure on the piston to be 18 lbs. per circular inch.

348. *Q.*—How is the strength of a cast iron gudgeon computed?

A.—To find the proper size of a cast iron gudgeon adapted to sustain any given weight:—multiply the weight in lbs. by the intended length of bearing expressed in terms of the diameter; divide the product by 500, and extract the square root of the quotient, which is the diameter in inches.

349. *Q.*—What was Mr. Watt's rule for the strength of gudgeons?

A.—Supposing the gudgeon to be square, then, to ascertain the thickness, multiply the weight resting on the gudgeon by the distance between the trunnions, and divide the product by 333. Extract the cube root of the quotient, which is the thickness in inches.

350. *Q.*—How do you find the proper strength for the cast iron beam of a land engine?

A.—If the force acting at the end of an engine beam be taken at 18 lbs. per circular inch of the piston, then the force acting at the middle will be 36 lbs. per circular inch of the piston, and the proper strength of the beam at the centre will be found by the following rule:—divide the weight in lbs. acting at the centre by 250, and multiply the quotient by the distance between the extreme centres. To find the depth, the breadth being given:—divide this product by the breadth in inches, and extract the square root of the quotient, which is the depth. The depth of a land engine beam at the ends is usually made one third of the depth at the centre (the depth at the centre being equal to the diameter of the cylinder in the case of low pressure engines), while the length is made equal to three times the length of the stroke, and the mean thickness $\frac{1}{108}$th of the length—the width of the edge bead being about three times the thickness of the web. In many modern engines the force acting at the end of the beam is more than 18 lbs. per circular inch of the piston, but the above rules are still applicable by taking an imaginary cylinder with an area larger in the proportion of the larger pressure.

351. *Q.*—What was Mr. Watt's rule for the main beams of his engines?

A.—Some of those beams were of wood and some of cast iron. The wood beams were so proportioned that the thickness was $\frac{1}{38}$th of the circumference, and the depth $\frac{1}{375}$. The side of the beam, supposing it square, was found by multiplying the diameter of the cylinder by the length of the stroke, and extracting the cube root of the quotient, which will be the depth or thickness of the beam. This rule allows a beam 16 feet long to bend $\frac{1}{8}$th of an inch, and a beam 32 feet long to bend $\frac{1}{4}$ of an inch. For cast iron beams the square of the diameter of the cylinder, multiplied by the length between the centres, is equal to the square of the depth, multiplied by the thickness.

352. *Q.*—What law does the strength of beams and shafts follow?

A.—In the case of beams subjected to a breaking force, the strength with any given cohesion of the material will be proportional to the breadth, multiplied by the square of the depth; and in the case of revolving shafts exposed to a twisting strain, the strength with any given cohesive power of the material will be as the cube of the diameter.

353. *Q.*—How is the strength of a cast iron shaft to resist torsion determined?

A.—Experiments upon the force requisite to twist off cast iron necks show that if the cube of the diameter of neck in inches be multiplied by 880, the product will be the force of torsion which will twist them off when acting at 6 inches radius; on this fact the following rule is founded: To find the diameter of a cast iron fly wheel shaft:—multiply the square of the diameter of the cylinder in inches, by the length of the crank in inches, and extract the cube root of the product, which multiply by 0·3025, and the result will be the proper diameter of the shaft in inches at the smallest part, when of cast iron.

354. *Q.*—What was Mr. Watt's rule for the necks of his crank shafts?

A.—Taking the pressure on the piston at 12 lbs. pressure on the square inch, and supposing this force to be applied at one foot radius, divide the total pressure of the piston reduced to 1 foot of radius by 31·4, and extract the cube root of the quotient, which is the diameter of the shaft: or extract the cube root of 13·7 times the number of cubic feet of steam required to make one revolution, which is also the diameter of the shaft.

355. *Q.*—Can you give any rule for the strength of the teeth of wheels?

A.—To find the proper dimensions for the teeth of a cast iron wheel:—multiply the diameter of the pitch circle in feet by the number of revolutions to be made per minute, and reserve the product for a divisor; multiply the number of *actual* horses power to be transmitted by 240, and divide the product by the above divisor, which will give the strength. If the pitch be given to find the breadth, divide the above strength by the square of the pitch in inches; or if the breadth be given, then to find the pitch divide the strength by the breadth in inches, and extract the square root of the quotient, which is the proper pitch in inches. The length of the teeth is usually about $\frac{5}{8}$ths of the pitch. Pinions to work satisfactorily should not have less than 30 or 40 teeth, and where the speed exceeds 220 feet in the minute, the teeth of the larger wheel should be of wood, made a little thicker, to keep the strength unimpaired.

356. *Q.*—What was Mr. Watt's rule for the pitch of wheels?

A.—Multiply five times the diameter of the larger wheel by the diameter of the smaller, and extract the fourth root of the product, which is the pitch.

STRENGTH OF MARINE AND LOCOMOTIVE ENGINES.

357. *Q.*—Cannot you give some rules of strength which will be applicable whatever pressure may be employed?

A.—In the rules already given, the effective pressure may be reckoned at from 18 to 20 lbs. upon every square inch of the piston, as is usual in land engines; and if the pressure upon every square inch of the piston be made twice greater, the dimensions must just be those proper for an engine of twice the area of piston. It will not be difficult, however, to introduce the pressure into the rules as an element of the computation, whereby the result will be applicable both to high and low pressure engines.

358. *Q.*—Will you apply this mode of computation to a marine engine, and first find the diameter of the piston rod?

A.—The diameter of the piston rod may be found by multiplying the diameter of the cylinder in inches, by the square root of the pressure on the piston in lbs. per square inch, and dividing by 50, which makes the strain $\frac{1}{7}$th of the elastic force.

359. *Q.*—What will be the rule for the connecting rod, supposing it to be of malleable iron?

A.—The diameter of the connecting rod at the ends, may be found by multiplying 0·019 times the square root of the pressure on the piston in lbs. per square inch by the diameter of the cylinder in inches; and the diameter of the connecting rod in the middle may be found by the following rule:—to 0·0035 times the length of the connecting rod in inches, add 1, and multiply the sum by 0·019 times the square root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches. The strain is equal to $\frac{1}{6}$th of the elastic force.

360. *Q.*—How will you find the diameter of the cylinder side rods of a marine engine?

A.—The diameter of the cylinder side rods at the ends may be found by multiplying 0·0129 times the square root of the pressure on the piston in lbs. per square inch by the diameter of the cylinder; and the diameter of the cylinder side rods at the middle is found by the following rule:—to 0·0035 times the length of the rod in inches, add 1, and multiply the sum by 0·0129 times the square root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches; the product is the diameter of each side rod at the centre in inches. The strain upon the side rods is by these rules equal to $\frac{1}{6}$th of the elastic force.

361. *Q.*—How do you determine the dimensions of the crank?

A.—To find the exterior diameter of the large eye of the crank when of malleable iron:—to 1·561 times the pressure of the steam upon the piston in lbs. per square inch, multiplied by the square of the length of the crank in inches, add 0·00494 times the square of the diameter of the cylinder in inches, multiplied by the square of the number of lbs. pressure per square inch on the piston; extract the square root of this quantity; divide the result by 75·59 times the square root of the length of the crank in inches, and multiply the quotient by the diameter of the cylinder in inches; square the product and extract the cube root of the square, to which add the diameter of the hole for the reception of the shaft, and the result will be the exterior diameter of the large eye of the crank when of malleable iron. The diameter of the small eye of the crank may be found by adding to the diameter of the crank pin 0·02521 times the square root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches.

362. *Q.*—What will be the thickness of the crank web?

A.—The thickness of the web of the crank, supposing it to be continued to the centre of the shaft, would at that point be represented by the following rule:—to 1·561 times the square of the length of the crank in inches, add 0·00494 times the square of the diameter of the cylinder in inches, multiplied by the pressure on the piston in lbs. per square inch; extract the square root of the sum, which multiply by the diameter of the cylinder squared in inches, and by the pressure on the piston in lbs. per square inch; divide the product by 9,000, and extract the cube root of the quotient, which will be the proper thickness of the web of the crank when of malleable iron, supposing the web to be continued to the centre of the shaft. The thickness of the web at the crank pin centre, supposing it to be continued thither, would be 0·022 times the square root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder. The breadth of the web of the crank at the shaft centre should be twice the thickness, and at the pin centre $1\frac{1}{2}$ times the thickness of the web; the length of the large eye of the crank would be equal to the diameter of the shaft, and of the small eye 0·0375 times the square root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder.

363. *Q.*—Will you apply the same method of computation to find the dimensions of a malleable iron paddle shaft?

A.—The method of computation will be as follows:—to find the dimensions of a malleable iron paddle shaft, so that the strain shall not exceed $\frac{5}{6}$ths of the elastic force, or $\frac{5}{6}$ths of the force iron is capable of withstanding without permanent derangement of structure, which in tensile strains is taken at 17,800 lbs. per square inch: multiply the pressure in lbs. per square inch on the piston by the square of the diameter of the cylinder in inches, and the length of the crank in inches, and extract the cube root of the product, which, multiplied by 0·08264, will be the diameter of the paddle shaft journal in inches when of malleable iron, whatever the pressure of the steam may be. The length of the paddle shaft journal should be $1\frac{1}{4}$ times the diameter; and the diameter of the part where the crank is put on is often made equal to the diameter over the collars of the journal or bearing.

364. *Q.*—How do you find the diameter ot the crank pin?

A.—The diameter of the crank pin in inches may be found by multiplying 0·02836 times the square root of the pressure on the piston in lbs. per square inch, by the diameter of the cylinder in inches. The length of the pin is usually about $\frac{9}{8}$th times its diameter, and the strain if all thrown upon the end of the pin will be equal to the elastic force; but in ordinary working, the strain will only be equal to $\frac{1}{3}$d of the elastic force.

365. *Q.*—What are the dimensions of the cross head?

A.—If the length of the cross head be taken at 1·4 times the diameter of the cylinder, the dimensions of the cross head will be as follows:—the exterior diameter of the eye in the cross head for the reception of the piston rod, will be equal to the diameter of the hole, plus 0·02827 times the cube root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches; and the depth of the eye will be 0·0979 times the cube root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches. The diameter of each cross head journal will be 0·01716 times the square root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches—the length of the journal being $\frac{9}{8}$ths its diameter. The thickness of the web at centre will be 0·0245 times the cube root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches; and the depth of web at centre will be 0·09178 times the cube root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches. The thickness of the web at journal will be 0·0122 times the square root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches; and the depth of the web at journal will be 0·0203 times the square root of the pressure upon the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches. In these rules for the cross head, the strain upon the web is $\frac{1}{2 \cdot 225}$ times the elastic force; the strain upon the journal in ordinary working is $\frac{1}{2 \cdot 33}$ times the elastic force; and if the outer ends of the journals are the only bearing points, the strain is $\frac{1}{1 \cdot 105}$ times the elastic force, which is very little in excess of the elastic force.

366. *Q.*—How do you find the diameter of the main centre when proportioned according to this rule?

A.—The diameter of the main centre may be found by multiplying 0·0367 times the square root of the pressure upon the piston in lbs. per square inch, by the diameter of the cylinder in inches, which will give the diameter of the main centre journal in inches when of malleable iron, and the length of the main centre journal should be 1½ times its diameter; the strain upon the main centre journal in ordinary working will be about ⅓ the elastic force.

367. *Q.*—What are the proper dimensions of the gibs and cutters of an engine?

A.—The depth of gibs and cutters for attaching the piston rod to the cross head, is 0·0358 times the cube root of the pressure of the steam on the piston in lbs. per square inch, multiplied by the diameter of the cylinder; and the thickness of the gibs and cutters is 0·007 times the cube root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of its cylinder. The depth of the cutter through the piston is 0·017 times the square root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder in inches; and the thickness of the cutter through the piston is 0·007 times the square root of the pressure on the piston in lbs. per square inch, multiplied by the diameter of the cylinder.

368. *Q.*—Are not some of the parts of an engine constructed according to these rules too weak, when compared with the other parts?

A.—It is obvious, from the varying proportions subsisting in the different parts of the engine between the strain and the elastic force, that in engines proportioned by these rules—which represent nevertheless the average practice of the best constructors—some of the parts must possess a considerable excess of strength over other parts, and it appears expedient that this disparity should be diminished, which may best be done by increasing the strength of the parts which are weakest; inasmuch as the frequent fracture of some of the parts shows that the dimensions at present adopted for those parts are scarcely sufficient, unless the iron of which they are made is of the best quality. At the same time it is quite certain, that engines proportioned by these rules will work satisfactorily where good materials are employed; but it is important to know in what parts good materials and larger dimensions are the most indispensable. In many of the parts, moreover, it is necessary that the dimensions should be proportioned to meet the wear and the tendency to heat, instead of being merely proportioned to obtain the necessary strength; and the crank pin is one of the parts which requires to be large in diameter, and as long as possible in the bearing, so as to distribute the pressure, and prevent the disposition to heat which would otherwise exist. The cross head journals also should be long and large; for as the tops of the side rods have little travel, the oil is less drawn into the bearings than if the travel was greater, and is being constantly pressed out by the punching strain. This strain should therefore be reduced as far as possible by its distribution over a large surface. In the rules which are contained in the answers to the ten preceding questions (358 to 367) the pressure on the piston in lbs. per square inch is taken as the sum of the pressure of steam in the boiler and of the vacuum; the latter being assumed to be 15 lbs. per square inch.