REPORT

MADE TO THE

BUREAU OF STEAM-ENGINEERING,

NAVY DEPARTMENT,

AUGUST 9, 1882,

BY

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ON THE

VEDETTE BOATS

CONSTRUCTED FOR THE

BRITISH AND FRENCH NAVIES

BY THE

HERRESHOFF MANUFACTURING COMPANY,
AT BRISTOL, R. I.

WASHINGTON:
GOVERNMENT PRINTING OFFICE.
1882.

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NEW YORK, *August* 9, 1882.

SIR: In obedience to an order from the Navy Department dated the 3d of June, 1882, I proceeded to the city of Bristol, R. I., and there witnessed the trials in Narragansett Bay of the duplicate Vedette boats built for the French Government by the "Herreshoff Manufacturing Company" of that city.

In addition to the results of these trials, I obtained the original data, including indicator diagrams, of some of the trials made at Bristol and also in England, of two Vedette boats built in 1880–'81 for the British Admiralty by the Herreshoff Manufacturing Company," which were the exact duplicate in all respects of those above referred to as constructed by that company for the French Government.

Having made from the data and diagrams, kindly furnished by the company, all the calculations that could be based upon them, I herein submit the results, together with the exact dimensions, descriptions, &c., of the hull and machinery, as well as such observations and deductions as the case seemed to warrant, to the bureau for its information.

The boats and their machinery are, in my opinion, so remarkable for their speed, qualities, and constructive details that, in view of their prospective utility to the Navy in time of war, I consider my duty requires me to make as full a presentation of the subject as my limited opportunities permit.

I had no assistants with whose aid I could execute a proper and exhaustive series of experiments, such as should have been made; and the company, though offering every facility, was so overloaded with work to be done in insufficient time that it could afford me but slight help. No indicator diagrams were taken or weighing of coal made except in two trials, and all the determinations of power, &c., are from diagrams taken during experiments with the boats built for the British Admiralty. The data that are furnished, however, may be relied on as correct.

The two Vedette boats, whose trials were witnessed, were constructed under contract with the French Government to make, with stipulated weights on board and at a stipulated draft of water, 42 geographical miles in smooth water during three consecutive hours without stopping or interruption of the machinery. The contract trials were made by a commission of French naval officers from the corvette Chasseur, sent to Bristol for that purpose. This commission also made rough-water trials, and experiments on the stability of the boats, besides closely scrutinizing their material, workmanship, and strength. The result was unqualified approval and acceptance.

These Vedette boats are a novelty in naval warfare, and have been developed from the necessity of providing means to counteract the attempts of torpedo boats upon large vessels or upon a squadron requiring much time and space for maneuvering when in confined positions or anchored at night in an enemy's waters. Under such circumstances large vessels, either singly or in squadron, are peculiarly exposed to torpedo attacks, and the services of extremely fast Vedette boats, forming a cordon around the large vessel or squadron, are indispensable for intercepting torpedo boats or warning of their approach. The bow of the Vedette boat can be strengthened to act as a ram against small torpedo boats; and effective machine guns can be mounted upon the rails abaft the machinery. The services to be rendered by so small and light draft a vessel, with exceptionally high speed and such formidable capabilities, are so numerous and important to a squadron or to a large iron-clad, under the complicated conditions of modern maritime war, that it must hereafter become an element of every naval system, and its management and use a part of naval instruction.

The following are descriptions and dimensions of the hull and machinery, followed by an account of the experiments made. All are given with as much detail and with as many calculations as the data at command could furnish.

HULL.

The hull is of wood, and uncoppered. The frames are of white oak, $1\frac{5}{8}$ inches square, and placed 12 inches apart from center to center. The planking is of pine in two courses, double-battened, one upon the other. The inner course is three-eighths of an inch thick, and is placed at an angle of 45° with the horizon; the outer course is one-half inch thick, and is placed horizontally. The two courses of planking are secured to each other and to the frames by brass screws, and are calked with cotton yarn. The stem is of white oak, molded 9 inches, and sided 4 inches. The stern-post is 8 by 4 inches in cross-section, and the keel is 4 inches wide. The deck-planks are of mahogany, three-quarters of an inch thick. At the stern a wooden skeg shod with a brass casting descends below the bottom of the keel for the protection of the screw; from the bottom of the keel to the bottom of the casting is 11 inches vertically. This casting is extended horizontally beneath the screw, its after extremity supporting the lower end of the rudder-post. The rudder is of wood, is of the ordinary type, and 20 inches wide. The side elevation of the skeg is a right-angled triangle of 11 inches vertical height at the after side of the stern-post, whence it extends forward along the bottom of the keel 4 feet as the base, the bottom of the skeg forming the hypothenuse. The width of the skeg is 4 inches, which is also the width of the rudder-post.

There are five transverse water-tight bulkheads, dividing the vessel into distinct compartments. In addition to these bulkheads there are two fore and aft water-tight bulkheads, one on each side of the machinery in the compartment containing the latter, and forming the two coal-bunkers, each of which contains one ton of coal.

The compartments that are forward of the one containing the machinery are decked over. The one immediately in front of that compartment has a hatch and a conveniently arranged steering gear, the head of the steersman protruding from the hatch, so that the vessel can be steered from forward as well as from aft.

The compartment containing the machinery is amidship, and is decked over by easily removable hatches made in convenient pieces and bolted down, but no particular care is taken to make their joints air-tight further than accurate fitting. The cabin, which is formed by the compartment next abaft the one containing the machinery, is similarly covered over by easily removable hatches; so that when all these hatches are in place, the vessel is decked over from the stem to the after bulkhead of the cabin. The compartment next abaft the cabin is the stern sheets; it is open, and can accommodate a dozen persons. The after compartment is decked over, and contains a well for a steersman, who, sitting in it, steers directly by means of a tiller.

The vessel had neither guns nor torpedoes, but could be easily fitted for either or both.

The following are the principal dimensions and proportions of the vessel:

Extreme length of hull on deck, from the forward edge of the stem to the after edge of the stern	48 feet.
Length on load water-line, from forward edge of stem to after side of stern-post	46 feet.
Extreme breadth of hull	8 feet 10 inches.
Extreme breadth of hull on load water-line	7 feet 5 inches.
Depth of hull amidship from lower edge of rabbet of keel to top of deck beams	5 feet.
Depth of hull at stem from load water-line to lower edge of rabbet of keel	2 feet.
Depth of hull amidship, from load water-line to lower edge of rabbet of keel	1 foot 10 inches.
Depth of hull at stern-post, from load water-line to lower edge of rabbet of keel	1 foot 8 inches.
Depth of keel forward below lower edge of its rabbet	$5\frac{1}{2}$ inches.
Depth of keel amidship below lower edge of its rabbet	8 inches.
Depth of keel aft (exclusive of skeg) below lower edge of its rabbet	$10\frac{1}{2}$ inches.
Siding of keel	4 inches.
Load draught of water forward	2 feet $5\frac{1}{2}$ inches.
Load draught of water aft, exclusive of skeg	2 feet $6\frac{1}{4}$ inches.
Load draught of water aft, inclusive of skeg	3 feet $5\frac{1}{2}$ inches.
Area of the load water section	217.76 square feet.
Area of the greatest immersed transverse section above the lower edge of the rabbet of the keel	8.48 square feet.

Area of the greatest immersed transverse section, including projected areas of keel and skeg	9.08 square feet.
Displacement to load water-line of body, stem and stern-post	247.84 cubic feet.
Displacement to load water-line, including keel, skeg, rudder-post, and rudder	260.00 cubic feet.
Total displacement (34.932 cubic feet per ton)	7.44306 tons.
Area of external wetted or immersed surface of hull, to lower edge of rabbet of keel, including stem and stern-post	266.24 square feet.
Wetted surface of keel	71.00 square feet.
Wetted surface of rudder post and rudder	11.36 square feet.
Wetted surface of the skeg	6.90 square feet.
Aggregate wetted surface of the vessel	355.50 square feet.
Ratio of the length of the hull to its breadth on load water-line	5.20755
Ratio of the load water section to its circumscribing parallelogram	0.63828
Ratio of the greatest immersed transverse section to its circumscribing parallelogram	0.62366
Ratio of the displacement above lower edge of rabbet of keel to its circumscribing parallelopipedon	0.39624
Weight of hull and fittings	6,993 pounds.

The stem and stern-post are chamfered to the vessel's form.

During the trials, to be hereinafter described, the vessel's load for a mean, consisted of nine men weighing 1,450 pounds, and of 1,209½ pounds of coal, with which weights she had the draught of water given above.

The displacement, greatest immersed transverse section, and wetted surface of the hull, as given above, are for the vessel at rest, but they changed to unknown quantities when the vessel was in motion, and the greater her speed the greater was the change. At the speed of fourteen geographical miles per hour, her trim was so excessively altered that the bow was 22 inches higher above the stern than when she was at rest.

ENGINE.

There is one compound condensing engine, with vertical cylinders placed side by side above the crank-shaft and having their axes in the vertical plane passing through its axis. The cylinders are direct-acting. The large cylinder works the air-pump and the feed-pump by means of a lever which makes the stroke of the pistons of these pumps less than that of the steam-pistons. Both pumps are vertical and single-acting. The air-pump has no foot-valve; it has a receiving valve in its piston and a discharging-valve into a closed reservoir. The feed-pump piston is a plunger. The feed-pump takes its water from a closed reservoir and delivers it into the bottom of the outer coil of the boiler. The circulating-pump is a piston-pump, double-acting and horizontal; it is worked from a pin on the forward end of the crank-shaft. From the same pin the bilge-pump is worked; this pump being in all respects an exact duplicate of the feed-pump.

The crank-shaft is of steel and has its two pairs of cranks forged with it. The crank-shaft journals are four in number.

The steam-valve of each cylinder is the ordinary three-ported slide with sufficient steam-lap to cut off the steam at two-thirds from the commencement of the stroke of the piston of each cylinder. There is no separate cut-off valve, and the steam-valve works with the boiler pressure upon its back. The gear for working the valves and for reversing is the usual Stephenson link with its two eccentrics and their rods, the link acting directly upon the end of the valve-stem.

The engine-frames are of wrought iron bolted to a cast-iron bed-plate extending beneath the entire length and breadth of the engine. This bed-plate has a semicircular bottom, and its side flanges are bolted to side keelsons. The crank-shaft pillow-blocks are cast in the bed-plate, the after pillow-block serving also as the thrust pillow-block, the after crank-shaft journal having collars made upon it to transmit the thrust of the screw to the bed-plate and thence to the vessel.

The engine works with surface condensation. The surface-condenser is composed of two bent copper pipes, one of which is placed on the outside of each side of the vessel beneath the water, and just about at the garboard strake. Each pipe commences 8 feet forward of abreast of the large cylinder, extends nearly to the stern-post, and is thence returned to abreast of the large cylinder. The diameter of the pipe continuously decreases from the end at which it receives the exhaust steam from the large cylinder to the end at which it delivers the water of condensation and the uncondensed vapor and air into the air-pump, whence they are thrown into the hot-well, from which the feed-pump forces the water of condensation into the bottom of the outer coil of the boiler. It is essential for satisfactory working that the delivering end of this tube should not exceed one-half the diameter of its receiving end.

The cylinders and their valve-chests, including covers of both, are incased with polished brass, between which and the iron are air spaces. The steam-pipes are coated with mineral wool or steam-blown melted glass, as a non-conductor of heat.

The steam pistons are packed by a cast-iron ring $\frac{3}{8}$ by $\frac{11}{32}$ of an inch in cross-section, set out by a spiral spring. The total depth of the piston at its circumference is $\frac{7}{8}$ inch.

The metal of the cylinders is $\frac{7}{16}$ inch thick; of their ends, $\frac{3}{8}$ inch; and of their pistons, $\frac{5}{16}$ inch.

The following are the principal dimensions of the engine:

Number of cylinders	2
Diameter of the small cylinder	8 inches.
Diameter of the piston-rod of the small cylinder	1¼ inches.
Net area of the piston of the small cylinder	49.65205 square inches.
Stroke of the piston of the small cylinder	9 inches.
Space displacement of the piston of the small cylinder per stroke	0.2586044 cubic foot.
Space in clearance and steam passage at one end of small cylinder	0.0278956 cubic foot.
Fraction which the space in clearance and steam passage at one end of the small cylinder is of the space displacement of the piston of the small cylinder per stroke	0.10787.

Length of steam-port of small cylinder	7 inches.
Breadth of steam-port of small cylinder	$\frac{15}{16}$ inch.
Area of steam-port of small cylinder	6.5625 square inches.
Length of exhaust-port of small cylinder	7 inches.
Breadth of exhaust-port of small cylinder	2 inches.
Area of exhaust-port of small cylinder	14.0000 square inches.
Clearance of piston of small cylinder	$\frac{1}{8}$ inch.
Aggregate area of the inner cylindrical surface of the small cylinder, of its two steam passages, of its two ends, of the two faces of its piston, and of half its piston-rod	666 square inches.
Diameter of the large cylinder	14 inches.
Diameter of the piston-rod of the large cylinder	$1\frac{1}{4}$ inch.
Net area of the piston of the large cylinder	153.32485 square inches.
Stroke of the piston of the large cylinder	9 inches.
Space displacement of the piston of the large cylinder per stroke	0.798567 cubic foot.
Space in clearance and steam passage at one end of large cylinder	0.067533 cubic foot.
Fraction which the space in clearance and steam passage at one end of the large cylinder is of the space displacement of the piston of the large cylinder per stroke	0.084568.
Length of steam-port of large cylinder	12 inches.
Breadth of steam-port of large cylinder	$1\frac{1}{4}$ inches.
Area of steam-port of large cylinder	15 square inches.
Length of exhaust-port of large cylinder	12 inches.
Breadth of exhaust-port of large cylinder	$2\frac{1}{2}$ inches.
Area of exhaust-port of large cylinder	30 square inches.
Clearance of piston of large cylinder	$\frac{1}{8}$ inch.
Aggregate area of the inner cylindrical surface of the large cylinder, of its two steam passages, of its two ends, of the two faces of its piston, and of half its piston-rod	1,240 square inches.
Diameter of the air-pump (single acting)	$4\frac{1}{2}$ inches.
Diameter of the piston-rod of the air-pump	$\frac{5}{8}$ inch.
Stroke of the piston of the air-pump	$3\frac{1}{8}$ inches.
Space displacement of the air-pump piston per stroke	0.028207 cubic foot.
Diameter of the plunger of the feed-pump (single acting)	$1\frac{1}{4}$ inches.
Stroke of the plunger of the feed-pump	$5\frac{1}{4}$ inches.
Space displacement of the plunger of the feed-pump per stroke	0.00372818 cubic foot.
Diameter of the circulating-pump (double-acting)	$1\frac{1}{2}$ inches.
Stroke of the piston of the circulating-pump	$1\frac{1}{2}$ inches.
Space displacement of the piston of the circulating-pump per stroke	0.00148611 cubic foot.
Depth of the packing-ring in both steam-pistons	$\frac{3}{8}$ inch.
Length of each of the condensing-pipes	43 feet.
Inside diameter of condensing-pipes at exhaust-steam end	$3\frac{1}{2}$ inches.
Continuously decreasing to inside diameter at air-pump end of	$1\frac{1}{4}$ inches.
Thickness of the metal (copper) of the condensing-pipes	$\frac{1}{32}$ inch.
Aggregate exterior surface of the condensing-pipes	54.87982 square feet.
Aggregate interior surface of the condensing-pipes	53.47265 square feet.
Length of the connecting-rods between centers	$22\frac{1}{2}$ inches.
Breadth of the necks of the connecting-rods	$1\frac{3}{4}$ inches.
Width of the necks of the connecting-rods	$\frac{3}{4}$ inch.
Diameter of crosshead journals	$1\frac{3}{4}$ inches.

BOILER

Made for

VEDETTE BOATS Nos 85 & 86

Built By The

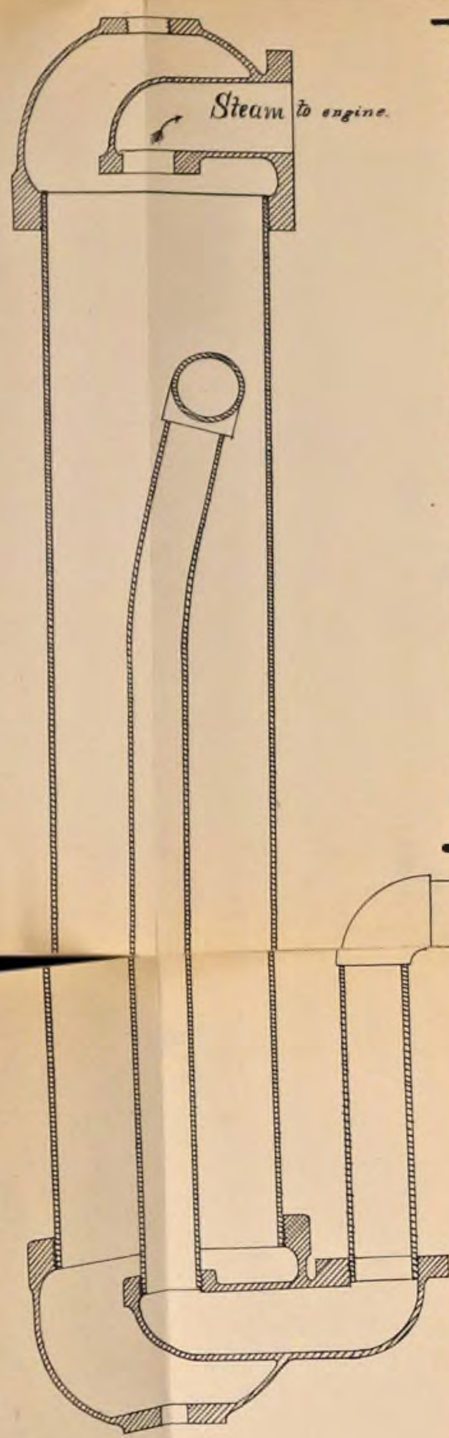
Herreshoff Manufacturing Co.

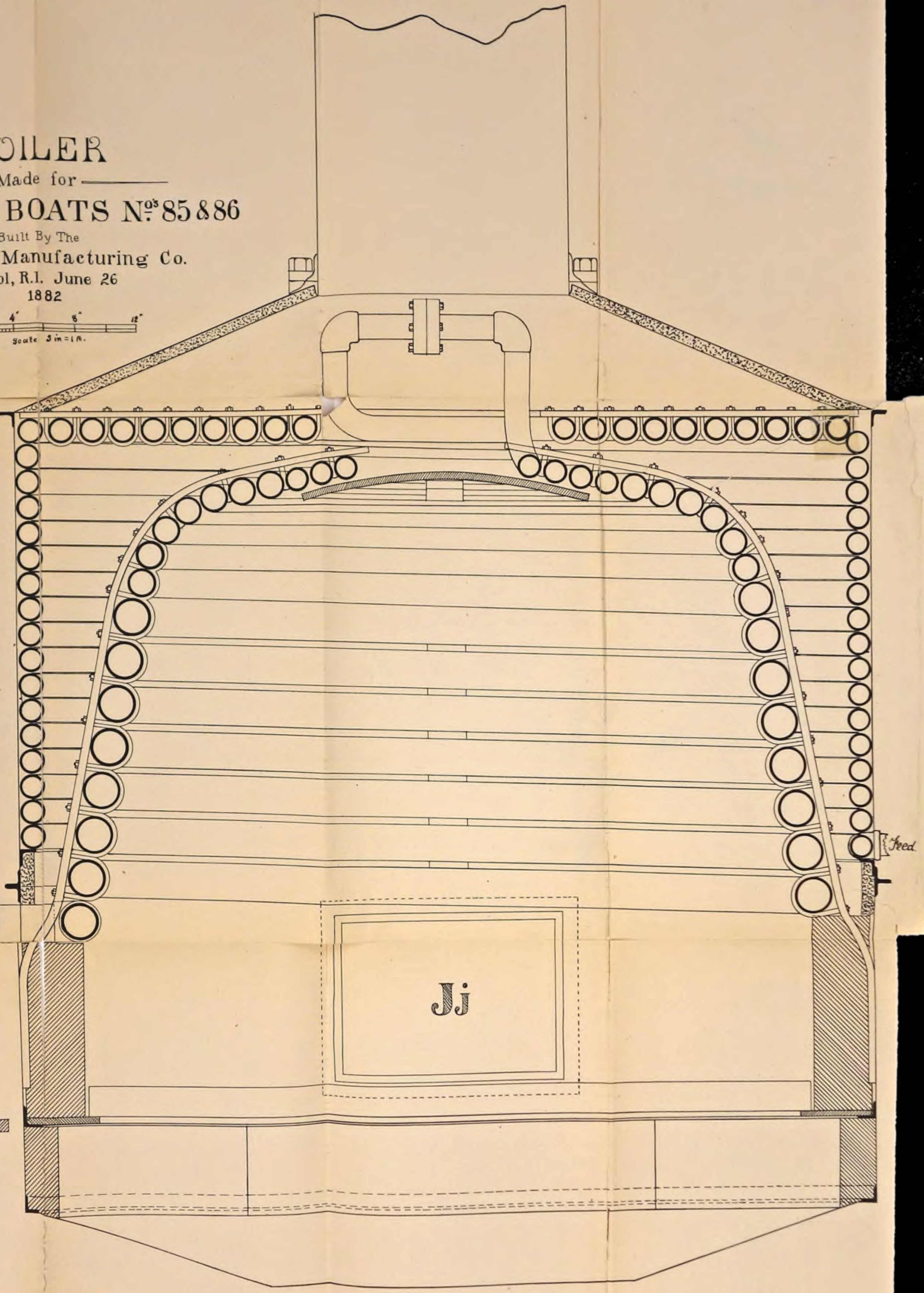
Bristol, R.I. June 26

1882

4" 8" 12"

Scale 3 in = 1 ft.





Length of cross-head journals	3 inches.
Number of crank-shaft journals (steel crank-shaft)	4
Diameter of crank-shaft journals	$2\frac{1}{2}$ inches.
Length of crank-shaft journals	$4\frac{1}{2}$ inches.
Diameter of crank-pin journals	$2\frac{1}{2}$ inches.
Length of crank-pin journals	$3\frac{1}{2}$ inches.
Width of cranks	$1\frac{1}{2}$ inches.
Number of thrust-rings on after journal of crank-shaft	4
Projection of thrust-rings from journal	$\frac{13}{32}$ inch.
Width of thrust-rings	$\frac{3}{8}$ inch.
Diameter of line shaft (steel)	$2\frac{1}{4}$ inches.
Length in the vessel occupied by the engine	4 feet.
Breadth in the vessel occupied by the engine	$2\frac{1}{2}$ feet.
Height of the engine above axis of crank shaft	4 feet.
Ratio of the space displacement of the piston of the large cylinder, per stroke, to that of the small cylinder	3.0879863.
Weight of the engine complete to after end of crank-shaft.	1,600 pounds.
Weight of stern-bearing, line-shafting, and condenser pipes	500 pounds.
Total weight of motive engine and appurtenances	2,100 pounds.

BLOWING ENGINE.

The fuel on the boiler grate is consumed by a forced draught, caused by a blast of air driven mechanically into the closed stoke-hole or fire-room of the vessel; this room being the water-tight compartment containing the engine and boiler. The blowing engine producing the air-blast consists of a separate vertical steam-cylinder, $2\frac{1}{2}$ inches in diameter and 5 inches stroke of piston, which operates directly an ordinary centrifugal fan-blower placed against the after bulkhead of the fire-room or machinery compartment of the vessel, through which bulkhead the fan receives the air from the cabin compartment. The fan-blower, made of thin galvanized sheet-iron, is 42 inches in exterior diameter, and receives the air in one side only through an opening of 18 inches diameter. The width of the blower at the center is 6 inches, tapering to $1\frac{1}{2}$ inches at the periphery. The connecting-rod of the cylinder articulates directly upon a crank secured to the shaft of the blower, so that the blower makes one revolution to each double stroke of the engine-piston. The engine also works an auxiliary plunger feed-pump of 1 inch diameter and 3 inches stroke, the plunger making stroke for stroke with the steam-piston.

The vessel has no independent steam-pump. Aggregate weight of blowing-engine, blower, and auxiliary feed-pump, 145 pounds.

BOILER.

(See drawing.)

There is one Herreshoff coil boiler with a single furnace; both boiler and furnace being circular in horizontal projection. The boiler is formed of a continuous wrought-iron pipe of different diameters, composing the heating surface, and is coiled spirally and symmetrically around and over the furnace.

The continuous pipe is arranged in three different kinds of coil: (1) An outside vertical coil wound around an imaginary right cylinder and forming the outside of the boiler; in this coil the spirals of pipe touch or are in contact, that is, no space is left between them. (2) A horizontal coil forming the top of the above imaginary cylinder; the spirals of this coil are separated by a slight distance, in order to allow the passage of the gases of combustion between them. (3) An inside coil lying within the imaginary cylinder referred to and so inclined over the grate as to form the frusta of two cones, one superimposed upon the other. The spirals of this coil are separated by slight distances to allow the passage of the gases of combustion between them. The upper diameter of the upper frustum, is made as small as the iron pipe can be coiled to, and the opening thus left is completely closed by a plate of boiler iron, so as to prevent the passage of the gases of combustion through the opening.

The pipe composing the coils contains the water to be vaporized, and the hot gases of combustion act on its exterior. The spirals of the first or outside coil have only half their exterior thus acted on, but those of the second and third coils are completely enveloped by the gases.

The fire-grate, circular in plan, is inclosed by a circular wall of brick masonry, on the top of which the coils rest; and the latter are surrounded by a single sheet-iron cylindrical casing in contact with the outside of the first coil. The uptake is a frustum of a right cone and rests upon the casing; it is double and the inclosed space is filled with mineral wool. The chimney rises from the center of the top of the uptake, the vertical axis of the chimney and of the boiler being in the same line. The whole of the gases of combustion passes first between the spirals of the inside coil into the space between that coil, the outside coil, and the top coil, and then between the spirals of the latter into the uptake, whence they are delivered by the chimney.

The feed water enters the lower spiral of the outside or cylindrical coil, passing upwards successively through all the spirals of that coil into the outer or greater spiral of the second or horizontal coil at the top of the boiler, thence successively through all the spirals of that coil, after which it enters the top spiral of the third or inclined inside coil, descending successively through all the spirals of that coil. In its passage through these coils, the feed water is first heated from the temperature of the hot well to that of the boiler, and then vaporized. According to the quantity of heat thrown upon the coils, the feed water may not be entirely vaporized until it arrives in the lower spiral of the third or inside coil, or it may be entirely vaporized at any previous spiral, in which case the remaining spirals will act as steam superheating surface. As it is desirable to avoid excessive superheating, both on account of injury to the metal of the spirals of the inside coil which are

exposed to the intense direct heat of the furnace, and on account of the valve chests and steam cylinders of the engine, it is necessary to have the means of keeping whatever surface is desired, covered with water, and for this purpose recourse is had to a forced circulation of what may be called a superfluous quantity of feed water by a circulating pump, which by continually drawing this superfluous feed water from the delivering or lower end of the third or inside coil and forcing it into the receiving end or upper spiral of the same coil keeps that coil filled to any desired degree with water, and thus controls the degree of superheating, or entirely prevents it, if enough superfluous feed water be employed. The pipe of the inside coil is filled under all conditions with a mixture of steam and water, and the degree of superheating will depend on the proportion of the two. As the circulation of the superfluous feed water is a tax on the power of the engine, that water is entered into the top of the inside coil only, so that the circulating pump may absorb as little power as possible.

The pressure in the boiler decreases gradually from the receiving end of the first or outside coil to the delivering end of the third or inside coil. It is at the maximum where the feed water enters, and at the minimum where the mixed steam and water are delivered. This difference of pressure, caused mainly by the surface resistance of the spirals and their continual deflection of the water from a straight course, reacts against both the feed pump and the circulating pump, causing the feeding of a Herreshoff boiler to be slightly more expensive in power than the feeding of other boilers. Were it not for this difference of pressure the only power expended in working the circulating pump would be that due to overcoming the friction of its piston and the resistance of its water to the inner surfaces and bends of the pipe between the "separator" and the pump, and between the pump and the receiving end of the third or inside coil.

The mixed water and steam are projected from the delivering end of the third or inside coil into the "separator," which is merely a closed cylindrical vessel wherein the water, by its greater gravity, separates from the steam and falls to the bottom, while the steam is carried off from the top by the main steam pipe which conducts it to the valve chest of the small cylinder of the engine. Of course, the superfluous feed water thus collected in the bottom of the "separator" is again pumped by the circulating pump into the upper spiral of the third or inside coil.

The "separator" acts both as the vessel in which the separation of the water and steam takes place, and as a steam drum or reservoir for maintaining an approximately constant pressure in the boiler during the intermittent drafts of steam made from it by the engine. Without a "separator," and a circulating pump or equivalent to perform its functions, a coil boiler could not be used. In this boiler the water and steam occupy exactly opposite positions to what they do in all other boilers, the water being in the top of the boiler and above the steam, instead of, as in other boilers, being at the bottom of the boiler with the steam above. This reversal of the usual relative position of the water and steam in a boiler is rendered possible in a coil boiler by its being composed of a single pipe of excessive length in proportion to inner diameter, coiled with a very slight inclination or pitch, and by the very small quantity of water in it, which flowing slowly along the spirals of the coil has time to become vaporized in the progress.

Upon the "separator" are placed the safety-valve, the steam-pressure gauge, and a glass water-gauge for showing the height of the water in the lower portion of the "separator." This height is the water level to be carried, and its maintenance regulates the quantity of superfluous feed water to be pumped in by the circulating pump. By properly proportioning that pump any quantity of superfluous feed water can be kept in circulation, and the current forced over the heating surfaces in such a torrent as to sweep off the steam bubbles as fast as formed, and to change and mix the water with such rapidity as to obtain the maximum heating efficiency from a given area of those surfaces in a given time. The glass water-gauge on the "separator" answers the same purpose as the gauge cocks on boilers of the usual construction, and requires to be as closely watched, for on the continuous passage through the coil pipe of an excess of feed water over what is vaporized depends the preservation of the metal from burning, and of the steam from too much superheating.

The furnace consists of a circular grate, 4 feet in diameter, surrounded by a circular vertical wall of fire-brick laid in fire-clay. The grate-bars are 2 inches in depth, with their top and bottom parallel. The inside diameter of the wall is 4 feet, and its outside diameter is 4 feet $7\frac{7}{8}$ inches. The thickness of the wall is $3\frac{15}{16}$ inches, and its height above the bottom of the grate-bars is $12\frac{3}{4}$ inches; both it and the grate-bars rest upon a wrought-iron ring, 3 feet 10 inches in inner diameter, 4 feet 7 inches in outer diameter, and three-eighths of an inch in thickness. Below this ring, which forms its cap, is a circular wall of common brick masonry, 4 feet $3\frac{1}{2}$ inches in inner diameter, 4 feet $7\frac{7}{8}$ inches in outer diameter, $2\frac{3}{16}$ inches thick, and $5\frac{1}{2}$ inches high. This wall incloses the ash-pit, the bottom of which is of sheet-iron, formed like an inverted frustum of a cone. The diameters, top and bottom, of the sheet-iron part of the ash-pit are 4 feet $3\frac{1}{2}$ inches and 2 feet; extreme depth below bottom of grate 11 inches. The opening for the ash-pit door is 27 inches wide and 6 inches high. Except this opening the ash-pit is entirely inclosed. The furnace has but one door; it is made in the brick masonry above the grate-bars, and its opening is 14 inches wide by 10 inches high. The outer top and bottom corners of the brick wall below the grate-bars are protected by angle iron bent to the ring form.

Upon the top of the brick wall inclosing the furnace rest the three coils, formed of one continuous wrought-iron pipe. The first or outside coil is composed of seventeen spirals wound around an imaginary right cylinder, 4 feet 4.555 inches in diameter, and 2 feet $8\frac{1}{4}$ inches in height above the top of the brick wall inclosing the furnace; the pipe of these spirals is 1.66 inches in exterior diameter, which makes the outside diameter of the spirals 4 feet $7\frac{7}{8}$ inches. The inside diameter of this pipe is 1.272 inches, and all these spirals touch, so that only half of their surface is heating surface, the remaining half not being subjected to the gases of combustion. The feed-water enters the lowest of these spirals, between the bottom of which and the top of the brick wall inclosing the furnace is a wrought-iron circular box, 4 feet $7\frac{7}{8}$ inches outside diameter, $\frac{7}{8}$ inch thick, and 3.78 inches high, filled with mineral wool. The top of this box is protected by an angle iron, and its center is hooped by a girder iron bent to ring form. The length of the axis of this coil is 242 feet 6 inches; half the exterior surface of this coil is 52.693795 square feet; half its interior surface is 40.377414 square feet, and its content is 2.140003 cubic feet. The feed-water traverses successively each spiral of this coil, ascending from bottom to top.

The second coil, which is a continuation of the first, is horizontal, forms the top of the boiler, and is composed of ten spirals or convolutions. The outside diameter of this coil is $52\frac{1}{2}$ inches, and its inside diameter is $15\frac{1}{2}$ inches. The pipe of which it is made is 1.66 inches outside diameter, and 1.272 inches inside diameter. The length of the axis of this coil is 89.012 feet; the exterior surface of the coil is 38.683547 square feet; its inner surface is 29.641851 square feet, and its content is 0.785509 cubic foot. The space for the passage of the gases of combustion between adjacent spirals is 0.3222+ inch wide, making the calorimeter or area for the passage of these gases through the second coil, 2.4 square feet.

The third or inside coil is made by winding spirally the pipe of which it is composed, around a cast-iron shaper formed of the frusta of two cones, the smaller frustum being superimposed upon the larger one, and their angle of junction rounded. The lower or larger frustum is 46 inches diameter at base, $36\frac{1}{2}$ inches diameter at top, and $26\frac{1}{2}$ inches high. The upper or smaller frustum is $36\frac{1}{2}$ inches diameter at base, 10 inches diameter at top, and 3 inches high. The angle of junction of these two frusta is rounded on a radius of 6 inches.

This third coil, which is a continuation of the second coil, is composed, commencing at the top, of a length of 7 feet of pipe, 1.66 inches outside diameter and 1.272 inches inside diameter; outside area 3.042116 square feet, inside area 2.331067 square feet, content 0.061773 cubic foot. The space for the passage of the gases of combustion between adjacent spirals of this pipe is $\frac{3}{32}$ inch wide, making the calorimeter or area for the passage of these gases 0.018229 square foot. Next, of a length of 54 feet of pipe, 1.900 inches outside diameter and 1.494 inches inside diameter; outside area 26.860680 square feet, inside area 21.120977 square feet, content 0.657390 cubic foot. The space for the passage of the gases of combustion between adjacent spirals of this diameter pipe is $\frac{1}{8}$ inch wide, making the calorimeter or area for the passage of these gases 0.562500 square foot. Last, of a length of 84 feet 6 inches of pipe $2\frac{3}{8}$ inches outside diameter, and 1.933 inches inside diameter; outside area 52.539987 square feet, inside area 42.762019 square feet, content 1.722062 cubic feet. The space for the passage of the gases of combustion between adjacent spirals of this diameter pipe is 0.4644 inch wide, making the calorimeter or area for the passage of these gases 3.270150 square feet.

The length of the axis of the pipe composing the three coils is $242.5+89.012+145.5=477.012$ feet. The exterior heating-surface of this pipe is 173.820125 square feet; its interior heating-surface is 136.233328 square feet, and its content is 5.366737 cubic feet.

On the top of the second or horizontal coil, there are four bars of flat wrought iron laid radially and equi-spaced, to which are secured by nuts the round stirrup irons or staples which keep the spirals of that coil at the desired distance apart.

On the outside of the third or inner coil there are four bars of flat wrought iron laid vertically and equi-spaced circumferentially, to which are secured by nuts the round stirrup irons or staples which keep the spirals of that coil at the desired distance apart.

In order to prevent the third or inner coil from straightening by the pressure within it, it is held together by four iron straps of $\frac{1}{2}$ inch diameter, arranged upon its outside diagonally, passing slantwise from top to bottom and crossing at the center.

The three coils and the brick walls of the furnace and ash-pit, are inclosed by a cylindrical casing of sheet iron $\frac{1}{16}$ inch thick and 56 inches external diameter. The height of the casing is 52 inches, and it is in contact with the brick walls of the furnace and ash-pit and with the first or outside coil, which it hoops and thus keeps in position against the tendency of the inside pressure to straighten it.

The uptake rests symmetrically upon the casing and is composed of two parallel sheet-iron plates $\frac{1}{16}$ inchthick with a $\frac{3}{4}$-inch intervening space filled with mineral wool. The form of the uptake is a frustum of a right cone of 56 inches outside diameter at bottom, $16\frac{1}{2}$ inches outside diameter at top, and $8\frac{1}{4}$ inches height outside.

The chimney rests upon the uptake, is $16\frac{1}{2}$ inches in diameter, and 13 feet high above the top of the grate.

The "separator" is placed by the side of the boiler with a space of 7 inches in the clear between them. It is a hollow cylinder of $\frac{3}{16}$ inch thick boiler plate, $8\frac{5}{8}$ inches in outside diameter, and 39 inches in height. The top and bottom of this cylinder are screwed into hemispherical cast-iron ends. On the top of the upper hemisphere the safety valve is placed, and from the bottom of the lower hemisphere a pipe leads to the

circulating pump for the boiler. On the side of the cylinder near its bottom is placed an ordinary glass water-gauge. Inside the cylinder is a standing pipe of $\frac{3}{16}$-inch thick boiler plate, the upper extremity of which is $5\frac{3}{4}$ inches below the upper end of the cylinder; the bottom of this pipe is screwed into a cast-iron partition forming a cylindrical cavity in the interior of the lower hemisphere, which cavity communicates with the bottom of the third coil and receives from it the water and steam from the boiler. The standing-pipe is $2\frac{1}{4}$ inches in outside diameter. From the upper hemisphere the steam is conducted to the engine. The bottom of the lower hemisphere is fitted with a blow-off pipe and cock for draining the "separator" and blowing out any sediment that may collect in it. The total height of the separator is 52 inches.

Beneath the boil.r, immediately under its ash-pan, is a wrought-iron water-tank, averaging 34 by 84 inches and 7 inches in depth. This tank is filled with fresh water for renewing any losses of water from the boiler due to any cause whatever.

The following are the principal dimensions and proportions of the boiler:

Diameter of the boiler to outside of casing	56 inches.
Height of the boiler from bottom of ash-pit to top of uptake	65 inches.
Diameter of the furnace	48 inches.
Area of the grate surface	12.5664 square feet.
Area of water-heating surface measured on outside of coil pipe.	173.8201 square feet.
Area of water-heating surface measured on inside of coil pipe.	136.2333 square feet.
Aggregate area of the spaces between the spirals of the inside or third coil, for the passage of the gases of combustion	3.8509 square feet.
Aggregate area of the spaces between the spirals of the second horizontal coil, for the passage of the gases of combustion	2.4000 square feet.
Cross-area of the chimney	1.4849 square feet.
Diameter of the chimney	$16\frac{1}{2}$ inches.
Height of the chimney above top of grate	13 feet.
Steam-room in the separator	0.95 cubic foot.
Height of steam-room in the separator	32 inches.
Water-room in the separator	0.3333 cubic foot.
Water-room in the coil pipe, supposing latter to be filled	5.3667 cubic feet.
Square feet of water-heating surface, measured on outside of coil pipe, per square foot of grate surface	13.83213
Square feet of water-heating surface, measured on inside of coil pipe, per square foot of grate surface	10.84108
Square feet of grate surface per square foot of space between the spirals of the inside or third coil, for the passage of the gases of combustion	3.26325
Square feet of grate surface per square foot of space between the spirals of the horizontal or second coil, for the passage of the gases of combustion	5.23600
Square feet of grate surface per square foot of cross-area of chimney for the passage of the gases of combustion	8.46279
Weight of boiler, including separator, brick walls, grate, and chimney	3,562 pounds.

WEIGHTS.

The following are the weights which make up the displacement of the Vedette boat at the draft of water hereinbefore given in the description of the hull, and at which the experiments were made:

	Pounds.
Weight of hull and fittings	6,993
Weight of engine complete to end of crank-shaft	1,600
Weight of condenser-pipes, line shafting, and stern bearing	500
Weight of screw	85
Weight of feed-pump attached to blowing-engine	15
Weight of blower	50
Weight of blowing-engine	80
Weight of boiler complete, with separator, grate, chimney, brick walls, &c., but exclusive of water	3,562
Weight of water-tank and miscellaneous	78
Weight of water in boiler and separator	330
Weight of water in tank beneath boiler	720
Total weight of hull, machinery, water, and appurtenances	14,013
Mean weight of coal in bunkers during trials	1,209½
Mean weight of persons on board during trials	1,450
Total displacement	16,672½

SCREWS.

Three different screws, all of brass, have been employed at different times in the propulsion of the Vedette boats. These screws had all the same diameter, namely, 35½ inches; the same diameter of hub, namely, 4 inches; the same number of blades, namely, four; and the same fraction of pitch, namely, 0.4; but with different lengths and different pitches; and, consequently, with different areas of helicoidal and projected surfaces. In all three screws the blades were at right angles to the axis, and had their forward and after edges parallel when viewed in projection on a plane parallel with the axis. The pitches were uniform for the entire blade, and were obtained by measurement of each blade in each case after the screws were cast. The weight of each screw was about 85 pounds.

The three screws will be designated as screw A, screw B, and screw C. The following are the principal dimensions and proportions of each:

SCREW A.

Diameter of screw	35½ inches.
Diameter of hub	4 inches.
Length of the screw in the direction of its axis (uniform)	4.2 inches.
Number of blades	4
Pitch (uniform) of the screw	3½ feet.
Fraction employed of the pitch	0.4
Helicoidal area of the blades	3.2060 square feet.
Projected area of the blades on a plane at right angles to the axis of the screw	2.7145 square feet.

SCREW B.

Diameter of the screw	35½ inches.
Diameter of the hub	4 inches.
Length of the screw in the direction of its axis (uniform)	4.9 inches.
Number of blades	4
Pitch (uniform) of the screw	4.0833+ feet.
Fraction employed of the pitch	0.4
Helicoidal area of the blades	3.4298 square feet.
Projected area of the blades on a plane at right angles to the axis of the screw	2.7145 square feet.

SCREW C.

Diameter of the screw	35½ inches.
Diameter of the hub	4 inches.
Length of the screw in the direction of its axis (uniform)	5.3 inches.
Number of blades	4
Pitch (uniform) of the screw	4.42 feet.
Fraction employed of the pitch	0.4
Helicoidal area of the blades	3.4977 square feet.
Projected area of the blades on a plane at right angles to the axis of the screw	2.7145 square feet.

PERFORMANCE WITH SCREW A.

FIRST TRIAL.—The Vedette boat, when drawing 2 feet 2 inches forward, and 3 feet nine inches aft, making her mean draft of water 2 feet 11½ inches, gave, when propelled by screw A, in Narragansett Bay, on the 23d of June, 1881, in smooth water and calm air, the following mean performance during two consecutive runs over a base of 3 geographical miles:

ENGINE.

Steam pressure in boiler in pounds per square inch above the atmosphere	95
Position of the throttle-valve	Wide open.
Vacuum in the condenser in inches of mercury	24
Back pressure in the condenser in pounds per square inch above zero	2.91
Fraction completed of the stroke of the pistons when the steam was cut off; both cylinders	0.6666+
Fraction completed of the return stroke of the pistons when the back pressure was cushioned; both cylinders	0.85
Number of times the steam was expended	4.32444
Number of double strokes made per minute by the pistons of the engine	415.

SPEED.

Speed of the vessel per hour in statute miles of 5,280 feet	13.5347
Speed of the vessel per hour in geographical miles of 6,086 feet	11.7422
Slip of the screw (3.5 feet pitch) in per centum of its speed	18.0000

STEAM PRESSURES IN SMALL CYLINDER PER INDICATOR.

Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero	105.0

Pressure on piston of small cylinder at point of cutting off the steam, in pounds per square inch above zero	90.5
Pressure on piston of small cylinder at the end of its stroke, in pounds per square inch above zero	61.5
Mean back pressure against piston of small cylinder during its stroke, in pounds per square inch above zero	28.0
Back pressure against the piston of the small cylinder at the point where the cushioning began, in pounds per square inch above zero	25.0
Mean back pressure against the piston of the small cylinder before the cushioning began, in pounds per square inch above zero	26.0
Indicated pressure on the piston of the small cylinder, in pounds per square inch	60.0
Net pressure on the piston of the small cylinder, in pounds per square inch	57.0
Total pressure on the piston of the small cylinder, in pounds per square inch	82.1

STEAM PRESSURES IN LARGE CYLINDER PER INDICATOR.

Pressure on piston of large cylinder at commencement of its stroke, in pounds per square inch above zero	23.0
Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero	18.0
Pressure on piston of large cylinder at the end of its stroke, in pounds per square inch above zero	14.0
Mean back pressure against piston of large cylinder during its stroke, in pounds per square inch above zero	6.0
Back pressure against the piston of the large cylinder at the point where the cushioning began, in pounds per square inch above zero	4.0
Mean back pressure against the piston of the large cylinder before the cushioning began, in pounds per square inch above zero	4.4
Indicated pressure on the piston of the large cylinder, in pounds per square inch	13.5
Net pressure on the piston of the large cylinder, in pounds per square inch	10.5
Total pressure on the annular space of the piston of the large cylinder remaining after subtracting from that piston the piston of the small cylinder, in pounds per square inch above zero	17.24

HORSES-POWER.

Indicated horses-power developed in the small cylinder	56.19697
Indicated horses-power developed in the large cylinder	39.04557
Aggregate indicated horses-power developed by the engine	95.24254
Net horses-power developed in the small cylinder	53.38712
Net horses-power developed in the large cylinder	30.36877
Aggregate net horses-power developed by the engine	83.75589
Total horses-power developed in the small cylinder	76.89618
Total horses-power developed in the large cylinder	33.71534
Aggregate total horses-power developed by the engine	110.61152

CYLINDER PRESSURES REDUCED TO LARGE CYLINDER ALONE.

Indicated pressure in pounds per square inch that would be on the piston of the large cylinder, were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder and the quotient added to the experimental indicated pressure on the piston of the large cylinder	32.93

Net pressure in pounds per square inch that would be on the piston of the large cylinder, similarly calculated to be immediately above	28.96
Total pressure in pounds per square inch on the piston of the large cylinder, corresponding to the aggregate total horses-power developed by the engine	38.24
Equivalent back pressure against the piston of the large cylinder, being the difference between the above total and indicated pressures, in pounds per square inch	5.31

RATIOS.

Per centum which the aggregate indicated horses-power developed by the engine are of the aggregate total horses-power	86.12
Per centum which the aggregate net horses-power developed by the engine are of the aggregate total horses-power	75.73

DISTRIBUTION OF THE POWER DURING THE ABOVE PERFORMANCE.

The distribution of the indicated horses-power developed by the engine during the above performance of the vessel with screw A is calculated according to the following assumptions:

Of the indicated horses-power, a portion is expended in working the engine *per se*, or the engine alone independently of its load; and this portion which consists of the friction of the packings and moving parts of the engine including the screw shafting must be first deducted because no power can be applied to the screw or developed externally of the engine until the friction of the engine itself is counterbalanced by an equivalent pressure on the piston; in fact, until this pressure is exceeded, the engine cannot move. In the case of the engine of the Vedette boat, the pressure required to work it *per se* is taken at 3 pounds per square inch of each steam piston.

After the deduction of the horses-power required to work the engine *per se*, from the indicated horses-power developed by the engine, the remainder, called the net horses-power, is applied to the crank-pin, and does work external to the engine. A friction attends the development of the net horses-power, additional to that of the unloaded engine and proportional to the net power, let the latter be what it may. This friction is that which is produced by the articulations of the engine moving under the net pressure, and is quite independent of the friction due to the mere weight of the moving parts and to the tightness of the packings. It is assumed to be 7.5 per centum of the net pressure or net power.

Then, there are the horses-power expended in overcoming the resistance of the water to the surface of the screw-blades; that is to say, in overcoming the skin resistance experienced by the screw-blades during their helical passage through the water. This resistance is taken to be 0.45 pound avoirdupois per square foot of helicoidal surface moving in its helical path with a velocity of 10 feet per second, and for other velocities this 0.45 pound is modified in the ratio of their squares to the square of 10. When the dimensions of the screw are known, and the number of revolutions it makes per minute, the horses-power expended in overcoming the surface resistance of its blades can be calculated by means of the above data.

There still remain to be determined the portions of the net horses-power expended in the slip of the screw and in the propulsion of the vessel. These are ascertained as follows: The sum of the horses-power expended in overcoming the friction of the load and in overcoming the resistance of the water to the surface of the screw-blades, being deducted from the net horses-power, the remainder is divided between the horses-power expended in the slip of the screw and in the propulsion of the vessel in the ratio of the speeds of the two, the pressure exercised by the screw forward in propelling the vessel, and backward upon the receeding mass of water constituting the slip of the screw being the same. Hence, if the aforesaid remainder of power be multiplied by the speed of the slip expressed in fractions of the axial speed of the screw, the product will be the horses-power expended in the slip, which, being subtracted from the above remainder, leaves the residue as the horses-power expended in the propulsion of the vessel.

The necessary calculations having been made in accordance with the foregoing assumptions give the following for the distribution of the power during the experiment with screw A on the 23d of June, 1881:

	Horses-power.	Percentum.
Indicated horses-power developed by the engine	95.24254	
Horses-power expended in working the engine *per se*	11.44665	
Net horses-power applied to the crank-pins	83.75589	100.00
Horses-power absorbed by the friction of the load	6.28169	7.50
Horses-power expended in overcoming the resistance of the water to the surface of the screw-blades	7.52091	8.98
Horses-power expended in the slip of the screw	12.59159	15.03
Horses-power expended in the propulsion of the vessel	57.36170	68.49
Totals	83.75589	100.00

THRUST OF THE SCREW.

The thrust of the screw, as it would have been measured by a dynamometer directly applied to the shaft during the above performance, calculated from the data therein given and in the distribution of the power, is as follows:

The horses-power expended in the propulsion of the vessel according to the distribution of the power, being 57.36170, is equal to $(57.36170 \times 33000 =)$ 1892936.1 foot-pounds of work per minute; and the speed of the vessel being 11.7422 geographical miles per hour, is equal to $\left(\frac{11.7422 \times 6086}{60} = \right)$ 1191.050487 feet per minute; hence the resistance of the vessel at that speed, or its equivalent the thrust of the screw, is $\left(\frac{1892936.1}{1191.050487} = \right)$ 1589.3 pounds.

DETERMINATION OF THE POWER EXPENDED IN OVERCOMING THE RESISTANCE OF THE WATER TO THE IMMERSED EXTERNAL OR WETTED SURFACE OF THE HULL.

Taking the resistance of the water to one square foot of rolled copper surface, moving in it with the velocity of 10 feet per second, to be 0.45 pound; and at other velocities to be this quantity modified in the ratio of their squares to the square of 10; and deducing from the speed of the vessel the mean speed of its immersed surface due to the inclination of its horizontal water-lines to its longitudinal central plane, there results for that speed 19.635 feet per second; and, consequently, a surface resistance of $(10^2 : 0.45 :: 19.635^2 :)$ 1.7349 pounds per square foot moving with that velocity.

As the immersed external or wetted surface of the vessel during the above performance was 355.5 square feet, the power expanded in overcoming its resistance was $\left(\frac{355.5 \times 1.7349 \times 19.635 \times 60}{33000} = \right)$ 22.01822 horses-power; consequently, of the 57.36170 horses-power required to propel the hull alone at the experimental speed, $\left(\frac{22.01822 \times 100}{57.36170} = \right)$ 38.385 per centum, were expended in overcoming the resistance of the water to the wetted surface, and the remaining 61.615 per centum were expended in the displacement of water by the immersed solid of the hull, irrespective of the resistance of its immersed surface.

SECOND TRIAL WITH SCREW A.—The Vedette boat, when drawing 2 feet 2 inches forward and 3 feet 9 inches aft, making her mean draught of water 2 feet $11\frac{1}{2}$ inches, gave on her second trial with screw A, in Narragansett Bay, on the 24th of June, 1881, the following mean performance during two consecutive runs over a base of one statute mile in smooth water and calm air.

ENGINE.

Steam pressure in boiler, in pounds per square inch above the atmosphere	38.
Position of the throt le-valve	Wide open.
Vacuum in the condenser, in inches of mercury	23.5
Back pressure in the condenser, in pounds per square inch above zero	3.15
Fraction completed of the stroke of the pistons when the steam was cut off; both cylinders	0.66666+
Fraction completed of the return stroke of the pistons when the back pressure was cush'oned; both cylinders	0.85
Number of times the steam was expanded	4.32444
Number of double strokes made per minute by the pistons of the engine	306.723

SPEED.

Speed of the vessel per hour in statute miles of 5,280 feet	10.55232
Speed of the vessel per hour in geographical miles of 6,086 feet	9.15482
Slip of the screw (3.5 feet pitch) in per centum of its speed	13.5

STEAM PRESSURES IN SMALL CYLINDER PER INDICATOR.

Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero	47.50

Pressure on piston of small cylinder at point of cutting off the steam, in pounds per square inch above zero 38.00
Pressure on piston of small cylinder at the end of its stroke, in pounds per square inch above zero 16.75
Mean back pressure against piston of small cylinder during its stroke, in pounds per square inch above zero 16.25
Back pressure against the piston of the small cylinder at the point where the cushioning began, in pounds per square inch above zero 14.80
Mean back pressure against the piston of the small cylinder before the cushioning began, in pounds per square inch above zero 15.00
Indicated pressure on the piston of the small cylinder, in pounds per square inch 23.75
Net pressure on the piston of the small cylinder, in pounds per square inch 20.75
Total pressure on the piston of the small cylinder, in pounds per square inch 36.50

STEAM PRESSURES IN LARGE CYLINDER PER INDICATOR.

Pressure on piston of large cylinder at commencement of its stroke, in pounds per square inch above zero 14.00
Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero 9.25
Pressure on piston of large cylinder at the end of its stroke, in pounds per square inch above zero 7.10
Mean back pressure against piston of large cylinder during its stroke, in pounds per square inch above zero 5.50
Back pressure against the piston of the large cylinder at the point where the cushioning began, in pounds per square inch above zero 4.25
Mean back pressure against the piston of the large cylinder before the cushioning began, in pounds per square inch above zero 4.60
Indicated pressure on the piston of the large cylinder, in pounds per square inch 5.75
Net pressure on the piston of the large cylinder, in pounds per square inch 2.75
Total pressure on the annular space of the piston of the large cylinder remaining after subtracting from that piston the piston of the small cylinder, in pounds per square inch above zero 9.66

HORSES-POWER.

Indicated horses-power developed in the small cylinder 16.44086
Indicated horses-power developed in the large cylinder 12.29148
Aggregate indicated horses-power developed by the engine 28.73234
Net horses-power developed in the small cylinder 14.36417
Net horses-power developed in the large cylinder 5.87853
Aggregate net horses-power developed by the engine 20.24270
Total horses-power developed in the small cylinder 25.26700
Total horses-power developed in the large cylinder 13.05349
Aggregate total horses-power developed by the engine 38.32049

CYLINDER PRESSURES REDUCED TO LARGE CYLINDER ALONE.

Indicated pressure in pounds per square inch that would be on the piston of the large cylinder, were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder and the quotient added to the experimental indicated pressure on the piston of the large cylinder 13.44

Net pressure in pounds per square inch that would be on the piston of the large cylinder, similarly calculated to the immediately above 9.47
Total pressure in pounds per square inch on the piston of the large cylinder, corresponding to the aggregate total horses-power developed by the engine 17.9265
Equivalent back pressure against the piston of the large cylinder, being the difference between the above total and indicated pressures, in pounds per square inch 4.4865

RATIOS.

Per centum which the aggregate indicated horses-power developed by the engine are of the aggregate total horses-power 74.979
Per centum which the aggregate net horses-power developed by the engine are of the aggregate total horses-power 52.825

DISTRIBUTION OF THE POWER DURING THE ABOVE PERFORMANCE.

The distribution of the indicated horses-power during the above second trial of the vessel on the 24th of June, 1881, with screw A, calculated in the manner previously described, is as follows:

	Horses-power.	Per centum of the net horses-power.
Indicated horses-power developed by the engine	28.73234	
Horses-power expended in working the engine *per se*	8.48964	
Net horses-power applied to the crank-pins	20.24270	100.00
Horses-power absorbed by the friction of the load	1.51820	7.50
Horses-power expended in overcoming the resistance of the water to the surface of the screw-blades	3.03654	15.00
Horses-power expended in the slip of the screw	2.11787	10.46
Horses-power expended in the propulsion of the vessel	13.57069	67.04
Totals	20.24270	100.00

THRUST OF THE SCREW.

The thrust of the screw as it would have been measured by a dynamometer directly applied to the shaft during the above performance, calculated in the manner previously described, is as follows: $13.57009 \times 33000 = 447812.97$ foot-pounds of work per minute. Speed of vessel per minute $= \frac{9.15482 \times 6086}{60} = 928.6039$ feet. And thrust of screw $= \frac{447812.97}{928.6039} =$ 482.24 pounds.

DETERMINATION OF THE POWER EXPENDED IN OVERCOMING THE RESISTANCE OF THE WATER TO THE IMMERSED EXTERNAL OR WETTED SURFACE OF THE HULL.

The power expended in overcoming the resistance of the water to the wetted surface of the hull during the second trial of the vessel on the 24th of June, 1881, with screw A, calculated in the manner previously described, is as follows:

Mean speed of the watery molecules over the immersed surface of the hull = 928.602 feet per minute or 15.4767 feet per second, at which velocity the resistance of the square foot of wetted surface would be $(10^2 : 0.45 :: 15.4767^2) : 1.077876$ pounds. And the horses-power required to overcome this resistance for 355.5 square feet of immersed surface would be $\left(\frac{1.077876 \times 355.5 \times 928.602}{33000} = \right)$ 10.78261; consequently, of the 13.57009 horses-power required to propel the hull alone at the experimental speed, $\left(\frac{10.78261 \times 100}{13.57009} = \right)$ 79.459 per centum, were expended in overcoming the resistance of the water to the wetted surface, and the remaining 20.541 per centum were expended in the displacement of water by the immersed solid of the hull irrespective of the resistance of its immersed surface.

COMPARISON OF THE RESULTS OF THE FIRST AND SECOND TRIALS WITH SCREW A

In the first trial with screw A, the speed of the vessel was 11.74220 geographical miles per hour, and the slip of the screw was 18 per centum of its axial speed or 2.57756 geographical miles per hour; the screw's thrust being 1,589.30 pounds.

In the second trial with screw A, the speed of the vessel was 9.15482 geographical miles per hour, and the slip of the screw was 13.5 per centum of its axial speed or 1.42879 geographical miles per hour; the screw's thrust being 482.24 pounds.

The resistances of the vessel in the two trials should compare as the squares of the slips of the screw in geographical miles per hour, or as $\left(\frac{2.57756^2}{1.42879^2} = \right)$ 3.25440 to 1.0000: they actually did compare as $\left(\frac{1589.30}{482.24} = \right)$ 3.29566 to 1.00000.

Again, the speeds of the vessel in the two trials compare as $\left(\frac{11.74220}{9.15482} = \right)$ 1.00000 and 1.28262, the squares of which are 1.00000 and 1.64511; consequently, had the resistance of the vessel in the two trials been as the squares of the speeds, the immediately above numbers should have been in the ratio of the respective thrusts of the screw. These thrusts were 482.24 and 1,589.30 pounds, or as 1.00000 and 3.29566, which is nearly double what should have been the case had the resistances of the vessel at the two experimental speeds been in the ratio of the squares of those speeds.

In accordance with this enormous increase of the resistance of the vessel at its higher speed above the law of the squares of the speeds, the slip of the screw instead of being constant at both speeds, as it would have been had the vessel's resistances been as the squares of those speeds, is found to increase from 13.5 per centum of its axial speed to 18 per centum, the recession of the watery fulcrum of the screw in the latter case being 2.57756 geographical miles per hour. Had the slip of the screw at the vessel's higher speed of 11.74220 geographical miles per hour been 13.5 per centum, the same as at its lower speed of

9.15482 geographical miles per hour, the recession of the watery fulcrum would have been only 1.83260 geographical miles per hour instead of the 2.57756 which it actually had with the experimental slip of 18 per centum. Now, the resistances of the vessel in the two cases are measured by the squares of the recessions of the watery fulcrum in geographical miles per hour; consequently, these resistances should compare as 1.83260^2 and 2.57756^2, or as 1.0000 and 1.9784, or nearly double.

The vessel's resistance at the speed of 9.15482 geographical miles per hour being 482.24 pounds, it should have been, according to the law of the squares of the speeds, $(482.24 \times 1.64511 =)$ 793.34 pounds at the speed of 11.74220 geographical miles per hour, but at that speed it actually was 1,589.30 pounds, or nearly double what it should have been according to the law, thus showing that the experimental slips of the screw in the two cases are exact. Also, that the method of calculation and the constants employed are correct.

THIRD TRIAL WITH SCREW A.—The Vedette boat when drawing 2 feet 2 inches forward and 3 feet 9 inches aft, making her mean draught of water 2 feet 11½ inches, gave on her third trial with screw A in Narragansett Bay on the 24th of June, 1881, the following mean performance during two consecutive runs over a base of one statute mile in smooth water and calm air.

ENGINE.

Steam pressure in boiler in pounds per square inch above the atmosphere..	19.5
Position of the throttle-valve	Wide open.
Vacuum in the condenser in inches of mercury	23.75
Back pressure in the condenser in pounds per square inch above zero....	3.03
Fraction completed of the stroke of the pistons when the steam was cut off; both cylinders........	0.66666+
Fraction completed of the return stroke of the pistons when the back pressure was cushioned; both cylinders	0.85
Number of times the steam was expanded	4.32444
Number of double strokes made per minute by the pistons of the engine..	220.

SPEED.

Speed of the vessel per hour in statute miles of 5,280 feet........	7.7000
Speed of the vessel per hour in geographical miles of 6,086 feet........	6.6803
Slip of the screw (3.5 feet pitch) in per centum of its speed........	12.

STEAM PRESSURES IN SMALL CYLINDER PER INDICATOR.

Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero	30.60
Pressure on piston of small cylinder at point of cutting off the steam, in pounds per square inch above zero	24.20
Pressure on piston of small cylinder at the end of its stroke, in pounds per square inch above zero	17.42
Mean back pressure against piston of small cylinder during its stroke, in pounds per square inch above zero........	12.42
Back pressure against the piston of the small cylinder at the point where the cushioning began, in pounds per square inch above zero	10.85

Mean back pressure against the piston of the small cylinder before the cushioning began, in pounds per square inch above zero	11.60
Indicated pressure on the piston of the small cylinder, in pounds per square inch	14.00
Net pressure on the piston of the small cylinder, in pounds per square inch	11.00
Total pressure on the piston of the small cylinder, in pounds per square inch	23.86

STEAM PRESSURES IN LARGE CYLINDER PER INDICATOR.

Pressure on piston of large cylinder at commencement of stroke, in pounds per square inch above zero	9.75
Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero	7.50
Pressure on piston of large cylinder at the end of its stroke, in pounds per square inch above zero	5.70
Mean back pressure against piston of large cylinder during its stroke, in pounds per square inch above zero	4.85
Back pressure against the piston of the large cylinder at the point where the cushioning began, in pounds per square inch above zero	4.45
Mean back pressure against the piston of the large cylinder before the cushioning began, in pounds per square inch above zero	4.60
Indicated pressure on the piston of the large cylinder, in pounds per square inch	3.50
Net pressure on the piston of the large cylinder, in pounds per square inch	0.50
Total pressure on the annular space of the piston of the large cylinder remaining after subtracting from that piston the piston of the small cylinder, in pounds per square inch above zero	7.41

HORSES-POWER.

Indicated horses-power developed in the small cylinder	6.95129
Indicated horses-power developed in the large cylinder	5.36637
Aggregate indicated horses-power developed by the engine	12.31766
Net horses-power developed in the small cylinder	5.46173
Net horses-power developed in the large cylinder	0.76662
Aggregate net horses-power developed by the engine	6.22835
Total horses-power developed in the small cylinder	11.84698
Total horses-power developed in the large cylinder	7.68217
Aggregate total horses-power developed by the engine	19.52915

CYLINDER PRESSURES REDUCED TO LARGE CYLINDER ALONE.

Indicated pressure in pounds per square inch that would be on the piston of the large cylinder, were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder, and the quotient added to the experimental indicated pressure on the piston of the large cylinder	8.034
Net pressure in pounds per square inch that would be on the piston of the large cylinder, similarly calculated to the immediately above	4.062
Total pressure in pounds per square inch on the piston of the large cylinder, corresponding to the aggregate total horses-powðr developed by the engine	12.738
Equivalent back pressure against the piston of the large cylinder, being the difference between *the above total and indicated pressures*, in pounds per square inch	4.704

RATIOS.

Per centum which the aggregate indicated horses-power developed by the engine are of the aggregate total horses-power	63.07
Per centum which the aggregate net horses-power developed by the engine are of the aggregate total horses-power	31.89

DISTRIBUTION OF THE POWER DURING THE ABOVE PERFORMANCE.

The distribution of the indicated horses-power during the above third trial of the vessel on the 24th of June, 1881, with screw A, calculated in the manner previously described, is as follows:

	Horses-power.	Per centum of the net horses-power.
Indicated horses-power developed by the engine	12.31766	
Horses-power expended in working the engine *per se*	6.08931	
Net horses-power applied to the crank-pins	6.22835	100.00
Horses-power absorbed by the friction of the load	0.46713	7.50
Horses-power expended in overcoming the resistance of the water to the surface of the screw-blades	1.12033	17.99
Horses-power expended in the slip of the screw	0.55691	8.94
Horses-power expended in the propulsion of the vessel	4.08398	65.57
Totals	6.22835	100.00

THRUST OF THE SCREW.

The thrust of the screw as it would have been measured by a dynamometer directly applied to the shaft during the above performance, calculated in the manner previously described, is as follows: $4.08398 \times 33000 = 134771.34$ foot-pounds of work per minute. Speed of vessel per minute $= \frac{6.6803 \times 6086}{60} = 677.6051$ feet. And thrust of screw $= \frac{134771.34}{677.6051} = 198.89$ pounds.

DETERMINATION OF THE POWER EXPENDED IN OVERCOMING THE RESISTANCE OF THE WATER TO THE IMMERSED EXTERNAL OR WETTED SURFACE OF THE HULL.

The power expended in overcoming the resistance of the water to the wetted surface of the hull during the third trial of the vessel on the 24th of June, 1881, with screw A, calculated in the manner previously described, is as follows:

Mean speed of the watery molecules over the immersed surface of the hull = 669 feet per minute or 11.15 feet per second, at which velocity the resistance of the square foot of wetted surface would be ($10^2 : 0.45 :: 11.15^2 :$) 0.55945125 pound. And the horse-power required to overcome this resistance for 355.5 square feet of immersed surface would be $\left(\frac{0.55945125 \times 355.5 \times 669}{33000} = \right)$ 4.03194. Now, from the foregoing distribution of the power, it appears that the horses-power required to propel the vessel, pure and simple, was 4.08398; hence, the resistance of the hull of the Vedette boat at the speed of 6.6803 geographical miles per hour, with her experimental draft of water, was just what was due to her immersed external or wetted surface; the power expended in the displacement of the water by her fore body being just recovered by the propelling action of the ascending column of water against her after body.

COMPARISON OF THE RESULTS OF THE SECOND AND THIRD TRIALS WITH SCREW A.

In the second trial with screw A, the speed of the vessel was 9.15482 geographical miles per hour, and the slip of the screw was 13.5 per centum of its axial speed or 1.42879 geographical miles per hour; the screw's thrust being 482.24 pounds.

In the third trial with screw A, the speed of the vessel was 6.68030 geographical miles per hour, and the slip of the screw was 12 per centum of its axial speed or 0.91093 geographical miles per hour; the screw's thrust being 198.89 pounds.

The resistances of the vessel in the two trials should compare as the squares of the slips of the screw in geographical miles per hour, or as $\left(\frac{1.42879^2}{0.91093^2}=\right)$ 2.46019 to 1.00000; they actually did compare as $\left(\frac{482.24}{198.89}=\right)$ 2.42466 to 1.00000.

Again, the speeds of the vessel in the two trials compare as $\left(\frac{9.15482}{6.68030}=\right)$ 1.00000 and 1.37042, the squares of which are 1.00000 and 1.87805; consequently, had the resistance of the vessel in the two trials been as the squares of the speeds, the immediately above numbers should have been in the ratio of the respective thrusts of the screw. These thrusts were 198.89 and 482.24 pounds, or as 1.00000 and 2.42466, which latter is about $\left(\frac{2.42466 - 1.87805 \times 100}{1.87805}=\right)$ 29.15 per centum greater than what it should have been had the resistances of the vessel at the two experimental speeds been in the ratio of the squares of those speeds.

In accordance with this considerable increase of the resistance of the vessel at its higher speed above the law of the squares of the speeds, the slip of the screw instead of being constant at both speeds, as it would have been had the vessel's resistances been as the square of those speeds, is found to increase from 12 per centum of its axial speed to 13.5 per centum, the recession of the watery fulcrum of the screw in the latter case being 1.42879 geographical miles per hour. Had the slip of the screw at the vessel's higher speed of 9.15482 geographical miles per hour been 12 per centum, the same as at its lower speed of 6.68030 geographical miles per hour, the recession of the watery fulcrum would have been only 1.24839 geographical miles per hour instead of the 1.42879 which it actually had with the experimental slip of 13.5 per centum. Now, the resistances of the vessel in the two cases are measured by the squares of the recessions of the watery fulcrum in geographical miles per hour; consequently, these resistances should compare as 1.24839^2 and 1.42879^2, or as 1.00000 to 1.30988. By the preceding method they compared as 1.0000 to 1.2915 or very nearly the same.

The vessel's resistance at the speed of 6.68030 geographical miles being 198.89 pounds, it should have been, according to the law of the squares of the speeds, $(198.89 \times 1.87805) = 373.53$ pounds at the speed of 9.15482 geographical miles per hour, but at that speed it actually was 482.24 pounds, or 29.01 per centum more than what it should have been according to the law, which is a very close agreement with the preceding results.

OTHER TRIALS WITH SCREW A TO ASCERTAIN ITS SLIP AT HIGHER SPEEDS.—After the completion of the previously described trials with screw A, which were made at speeds sufficiently low to enable indicator diagrams to be taken, two other trials were made at higher speeds on the 24th of June, 1881, over a base of 3 geographical miles in Narragansett Bay, the water being smooth, the tide slack, the air calm, and the vessel at her draught of water of 2 feet 2 inches forward, and 3 feet 9 inches aft, mean 2 feet $11\frac{1}{2}$ inches. Each trial consisted of two consecutive runs over the base, with the conditions maintained uniform throughout each pair of runs, and was made to ascertain the slip of the screw at higher rates of speed. The fuel consumed during these experiments, as well as during the previous ones, was ordinary anthracite. The results were as follows:

MEANS OF THE FIRST TWO CONSECUTIVE RUNS.

Steam pressure in boiler in pounds per square inch above atmosphere	120.
Position of throttle-valve	Wide open.
Fraction of stroke of piston of both cylinders completed when the steam was cut off	0.66666+
Fraction of return stroke of piston of both cylinders completed when the back pressure was cushioned	0.85
Vacuum in the condenser in inches of mercury	22.
Number of double strokes made per minute by the pistons	485.5949
Speed of the vessel per hour in statute miles of 5,280 feet	15.79279
Speed of the vessel per hour in geographical miles of 6,086 feet	13.70127
Slip of the screw in per centum of its speed	18.2290
Air pressure in fire room above that of the atmosphere, measured by the height in inches of a water column at the temperature of 75 degrees Fahrenheit	2.875
Air pressure in fire-room above that of the atmosphere, in fraction of a pound per square inch	0.10354
Revolutions of the blower per minute	850.

MEANS OF THE SECOND TWO CONSECUTIVE RUNS.

Steam pressure in boiler in pounds per square inch above the atmosphere	140.
Position of the throttle-valve	Wide open.
Fraction of stroke of piston of both cylinders completed when the steam was cut off	0.66666+

Fraction of return stroke of piston of both cylinders completed when the back pressure was cushioned	0.85
Vacuum in the condenser in inches of mercury	21.
Number of double strokes made per minute by the pistons	530.16087
Speed of the vessel per hour in statute miles of 5,280 feet	17.28977
Speed of the vessel per hour in geographical miles of 6,086 feet	15.00000
Slip of the screw in per centum of its speed	18.0000

Performance with screw B.

No indicator diagrams were taken during the trials of the vessel with screw B; consequently the only results with it were its slip and the speed of the vessel.

These trials were made in June, 1882, with the vessel at the following draught of water, namely: Forward, 2 feet 5½ inches; aft, 3 feet 5½ inches; mean, 2 feet 11½ inches. This draught of water differing from that of the previous trials with screw A, in being 3½ inches more forward and 3½ inches less aft, no comparison can be made between the slips of the two screws.

At all speeds below 7 geographical miles per hour the slip of screw B was constant at 13¼ per centum of its axial speed. At 10 geographical miles per hour the slip had gradually increased to 22 per centum; and at 12 geographical miles per hour it had reached its maximum of 24 per centum, after which it remained constant up to the maximum experimental speed of 14 geographical miles per hour.

Three principal trials were made with screw B over a base of 12 statute miles in Narragansett Bay, the water being smooth and the breeze very light. Each trial consisted of four runs made consecutively over the base with the conditions maintained as uniformly as possible. In one of these trials the fuel was anthracite; in the other two it was briquettes d'Anzin, obtained from the French corvette Chasseur. These briquettes are made of an excellent semi-bituminous coal found in the northeast corner of France. This coal, being pulverized, washed to separate its earthy matter, and mixed with a very slight proportion of coal tar to give it cohesion, is formed into large rectangular masses or bricks by a powerful press. These briquettes contain about 5 per centum of light ash; they are very free-burning, form no clinker, make scarcely any smoke, and as a steam-generating fuel are but little if any inferior to the best Welsh coal.

The anthracite was slow-burning, and contained 17 per centum of earthy matter, of which one-half formed a strongly adhesive slag or clinker upon the fire-grate. The comparison furnished by the results of the trials with anthracite and with the briquettes d'Anzin places the relative merits of these fuels in a very clear light as regards their value for vessels in the generation of steam, and shows most clearly why, other things being equal, the steamships of the United States Navy burning anthracite require more boiler for equal speed than the steam-

ships of the British and French navies, which burn the best Welsh coal or its equivalent, the briquettes d'Anzin.

Trial of screw B with anthracite.—The total time occupied by this experiment, including the turnings at the ends of the base, was 3 hours 49 minutes consecutively. The results will be given for each run over the base separately, the speed being calculated in each case from the slip of the screw as previously determined. Date of trial June 13, 1882.

First run.

Steam pressure in boiler in pounds per square inch above atmosphere	115.
Number of double strokes of pistons made per minute	438.33333
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	1.29
Revolutions of the blower per minute	541.
Speed of the vessel in statute miles per hour	15.45790
Speed of the vessel in geographical miles per hour	13.41072

Second run.

Steam pressure in boiler in pounds per square inch above atmosphere	110.
Number of double strokes of pistons made per minute	420.55555
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	2.58
Revolutions of the blower per minute	757.
Speed of the vessel in statute miles per hour	14.83096
Speed of the vessel in geographical miles per hour	12.86682

Third run.

Steam pressure in boiler in pounds per square inch above atmosphere	71.5
Number of double strokes of pistons made per minute	363.20000
Slip of the screw in per centum of its speed	23.
Air pressure in fire-room above atmosphere in inches of water	2.48
Revolutions of the blower per minute	742.
Speed of the vessel in statute miles per hour	12.97683
Speed of the vessel in geographical miles per hour	11.25824

Fourth run.

Steam pressure in boiler in pounds per square inch above atmosphere	52.
Number of double strokes of pistons made per minute	299.00000
Slip of the screw in per centum of its speed	18.
Air pressure in the fire-room above atmosphere in inches of water	1.25
Revolutions of the blower per minute	526.
Speed of the vessel in statute miles per hour	11.37672
Speed of the vessel in geographical miles per hour	9.87005

Means of the above four runs.

Steam pressure in boiler in pounds per square inch above atmosphere	87.125
Number of double strokes of pistons made per minute	380.27222
Slip of the screw in per centum of its speed	22.5818
Air pressure in fire-room above atmosphere in inches of water	1.90
Revolutions of the blower per minute	649.77
Speed of the vessel in statute miles per hour	13.66060
Speed of the vessel in geographical miles per hour	11.85146

TRIAL OF SCREW B WITH BRIQUETTES D'ANZIN.—The total time occupied by this experiment, including the turnings at the ends of the base, was exactly three consecutive hours. The results will be given for each run over the base separately, the speed being calculated in each case from the slip of the screw as previously determined. The date of the trial was June 21, 1882.

FIRST RUN.

Steam pressure in boiler in pounds per square inch above atmosphere	132.
Number of double strokes of pistons made per minute	478.21101
Slip of the screw in per centum of its speed	24.
Air pressure in the fire-room above atmosphere in inches of water	2.
Revolutions of the blower per minute	667.
Speed of the vessel in statute miles per hour	16.86419
Speed of the vessel in geographical miles per hour	14.63077

SECOND RUN.

Steam pressure in boiler in pounds per square inch above atmosphere	130.
Number of double strokes of pistons made per minute	471.42857
Slip of the screw in per centum of its speed	24.
Air pressure in the fire-room above atmosphere in inches of water	2.
Revolutions of the blower per minute	667.
Speed of the vessel in statute miles per hour	16.62500
Speed of the vessel in geographical miles per hour	14.42326

THIRD RUN.

Steam pressure in boiler in pounds per square inch above atmosphere	130.
Number of double strokes of pistons made per minute	454.75914
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	2.8
Revolutions of the blower per minute	788.
Speed of the vessel in statute miles per hour	16.03715
Speed of the vessel in geographical miles per hour	13.91327

FOURTH RUN.

Steam pressure in boiler in pounds per square inch above atmosphere	126.
Number of double strokes of pistons made per minute	460.40724
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	2.6
Revolutions of the blower per minute	760.
Speed of the vessel in statute miles per hour	16.23633
Speed of the vessel in geographical miles per hour	14.08607

MEANS OF THE ABOVE FOUR RUNS.

Steam pressure in boiler in pounds per square inch above atmosphere	129.5
Number of double strokes of pistons made per minute	466.20149
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	2.35
Revolutions of the blower per minute	722.54
Speed of the vessel in statute miles per hour	16.44067
Speed of the vessel in geographical miles per hour	14.26334

ANOTHER TRIAL OF SCREW B WITH BRIQUETTES D'ANZIN.—Another trial was made with screw B on a duplicate vessel at the same draft of water, the fuel being the briquettes d'Anzin. This trial lasted 3 hours 4 minutes and 11 seconds consecutively, including the turnings at the end of the base. The results will be given for each run over the base separately, the speed being calculated in each case from the slip of the screw as previously determined. The date of the trial was June 28, 1882.

FIRST RUN.

Steam pressure in boiler in pounds per square inch above atmosphere	126.
Number of double strokes of pistons made per minute	462.12222
Slip of the screw in per centum of its speed	24.
Air pressure in the fire-room above atmosphere in inches of water	1.7
Revolutions of the blower per minute	615.
Speed of the vessel in statute miles per hour	16.29681
Speed of the vessel in geographical miles per hour	14.13854

SECOND RUN.

Steam pressure in boiler in pounds per square inch above atmosphere	129.
Number of double strokes of pistons made per minute	453.35265
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	2.3
Revolutions of the blower per minute	715.
Speed of the vessel in statute miles per hour	15.98755
Speed of the vessel in geographical miles per hour	13.87024

THIRD RUN.

Steam pressure in boiler in pounds per square inch above atmosphere	118.
Number of double strokes of pistons made per minute	455.06356
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	2.4
Revolutions of the blower per minute	730.
Speed of the vessel in statute miles per hour	16.06683
Speed of the vessel in geographical miles per hour	13.93901

FOURTH RUN.

Steam pressure in boiler in pounds per square inch above atmosphere	140.
Number of double strokes of pistons made per minute	470.00000
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	4.
Revolutions of the blower per minute	943.
Speed of the vessel in statute miles per hour	16.57462
Speed of the vessel in geographical miles per hour	14.37956

MEANS OF THE ABOVE FOUR RUNS.

Steam pressure in boiler in pounds per square inch above atmosphere	128.25
Number of double strokes of pistons made per minute	460.13461
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	2.6
Revolutions of the blower per minute	760.11
Speed of the vessel in statute miles per hour	16.23145
Speed of the vessel in geographical miles per hour	14.08184

MEAN RESULTS OF THE TWO TRIALS BURNING BRIQUETTES D'ANZIN.—The following are the mean results of the two trials, each of three consecutive hours, made with the duplicate Vedette boats, the fuel being the briquettes d'Anzin. The power developed by the engine during these trials could not be ascertained by the indicator on account of the great number of double strokes made per minute by the pistons, but it can be approximated on the supposition that the resistance of the hull at speeds above about 12 geographical miles was in the ratio of the squares of the speeds—a supposition undoubtedly very near the truth, as evidenced by the constancy of the slip of the screw at all speeds above 12 geographical miles per hour.

Now, the power required to propel the hull alone at the speed of 11.7422 geographical miles per hour is known from the previously detailed experiment with screw A, made at that speed, to be 95.24254 horses; and as the slip of screw B, at all speeds above about 12 geographical miles per hour, was constantly 24 per centum, the indicated horses-power developed by the engine for any higher speed can be calculated from the data according to the method already herein described. This calculation has been made for the mean speed of the two trials burning the briquettes d'Anzin, and the result is given below. The indicated horses-power so obtained agrees substantially with what is furnished by an ideal indicator diagram constructed from the boiler pressure, condenser vacuum, and point of cutting off known to exist in the case.

During all the trials with screw B the throttle valve was carried wide open, the vacuum in the condenser averaged 20 inches of mercury, and the temperature of the air in the fire-room averaged about 10° Fahr. above that of the external atmosphere.

Steam pressure in boilers in pounds per square inch above atmosphere	128.875
Number of double strokes of pistons made per minute	463.16805
Slip of the screw in per centum of its speed	24.
Air pressure in fire-room above atmosphere in inches of water	2.475
Revolutions of the blower per minute	742.
Speed of the vessel in statute miles per hour	16.33606
Speed of the vessel in geographical miles per hour	14.17259
Indicated horses power developed by the engine	169.4733
Net horses-power applied to the crank-pins	156.6534
Pounds of briquettes d'Anzin consumed per hour	700.
Pounds of briquettes d'Anzin consumed per hour per indicated horse-power	4.1304
Pounds of briquettes d'Anzin consumed per hour per net horse-power	4.4684
Indicated pressure in pounds per square inch that would be on the piston of the large cylinder, were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder and the quotient added to the experimental indicated pressure on the piston of the large cylinder	52.5017
Net pressure in pounds per square inch that would be on the piston of the large cylinder, similarly calculated to the immediately above	48.5302
Pounds of briquettes d'Anzin consumed per hour per square foot of grate surface	55.704
Number of indicated horses-power developed per square foot of grate	13.4862

DISTRIBUTION OF THE POWER DURING THE MEAN PERFORMANCE OF THE ABOVE TWO TRIALS, BURNING BRIQUETTES D'ANZIN.

	Horses-power.	Per centum of the net horses-power.
Indicated horses-power developed by the engine	169.4733	
Horses-power expended in working the engine *per se*	12.8199	
Net horses-power applied to the crank-pins	156.6534	100.00
Horses-power absorbed by the friction of the load	11.7490	7.50
Horses-power expended in overcoming the resistance of the water to the surface of the screw-blades	12.1863	7.78
Horses-power expended in the slip of the screw	31.8523	20.33
Horses-power expended in the propulsion of the vessel	100.8658	64.39
Totals	156.6534	100.00

THRUST OF THE SCREW.

The thrust of the screw during the above development of power, calculated in the manner previously described, was 2,315.37 pounds.

PERFORMANCE WITH SCREW C.

FIRST TRIAL.—With screw C two consecutive runs were made by a duplicate vedette boat on the 2d of August, 1881, in smooth water and calm air, over the base of one geographical mile at Stokes Bay, England, the vessel drawing 2 feet 2 inches of water forward, and 3 feet 9 inches aft; mean, 2 feet 11½ inches. No indicator diagrams were taken. The best Welsh coal was used. The following were the results:

Steam pressure in boiler in pounds per square inch above atmosphere	145.
Position of the throttle-valve	Wide open.
Vacuum in the condenser in inches of mercury	20.29
Back pressure in the condenser in pounds per square inch above zero	4.726
Fraction completed of the stroke of the pistons when the steam was cut off: both cylinders	0.6666+
Fraction completed of the return stroke of the pistons when the back pressure was cushioned: both cylinders	0.85
Number of times the steam was expanded	4.32444
Number of double strokes made per minute by the pistons of the engine	453.07
Speed of the vessel per hour in statute miles of 5,280 feet	17.4327
Speed of the vessel per hour in geographical miles of 6,086 feet	15.1240
Slip of the screw in per centum of its speed	23.395

SECOND TRIAL WITH SCREW C.—On the 4th of August, 1881, the vedette boat at the Isle of Wight, England, gave the following moderate speed performance in smooth water and calm air, the draught of water being 2 feet 4 inches forward and 3 feet 9½ inches aft; mean draught, 3 feet ¾ inch. The best Welsh coal was used, and indicator diagrams were taken.

ENGINE.

Steam pressure in boiler in pounds per square inch above atmosphere	93.2
Position of the throttle-valve	Wide open.
Vacuum in the condenser in inches of mercury	24.

Back pressure in the condenser in pounds per square inch above zero	2.906
Fraction completed of the stroke of the pistons when the steam was cut off: both cylinders	0.66666+
Fraction completed of the return stroke of the pistons when the back pressure was cushioned: both cylinders	0.85
Number of times the steam was expanded	4.32444
Number of double strokes made per minute by the pistons of the engines.	333.

SPEED.

Speed of the vessel per hour in statute miles of 5,280 feet	12.6444
Speed of the vessel per hour in geographical miles of 6,086 feet	10.9700
Slip of the screw in per centum of its speed	24.40

STEAM PRESSURES IN SMALL CYLINDER, PER INDICATOR.

Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero	101.30
Pressure on piston of small cylinder at point of cutting off the steam, in pounds per square inch above zero	88.00
Pressure on piston of small cylinder at the end of its stroke, in pounds per square inch above zero	58.60
Mean back pressure against piston of small cylinder during its stroke, in pounds per square inch above zero	27.40
Back pressure against the piston of the small cylinder at the point where the cushioning began, in pounds per square inch above zero	26.00
Mean back pressure against the piston of the small cylinder before the cushioning began, in pounds per square inch above zero	26.26
Indicated pressure on the piston of the small cylinder, in pounds per square inch	59.25
Net pressure on the piston of the small cylinder, in pounds per square inch	56.25
Total pressure on the small piston of the cylinder, in pounds per square inch	81.571

STEAM PRESSURES IN LARGE CYLINDER, PER INDICATOR.

Pressure on piston of large cylinder at commencement of its stroke, in pounds per square inch above zero	21.77
Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero	18.11
Pressure on piston of large cylinder at the end of its stroke, in pounds per square inch above zero	11.84
Mean back pressure against piston of large cylinder during its stroke, in pounds per square inch above zero	4.78
Back pressure against the piston of the large cylinder at the point where the cushioning began, in pounds per square inch above zero	3.52
Mean back pressure against the piston of the large cylinder before the cushioning began, in pounds per square inch above zero	3.78
Indicated pressure on the piston of the large cylinder, in pounds per square inch	13.83
Net pressure on the piston of the large cylinder, in pounds per square inch	10.83
Total pressure on the annular space of the piston of the large cylinder remaining after subtracting from that piston the piston of the small cylinder, in pounds per square inch above zero	17.043

HORSES-POWER.

Indicated horses-power developed in the small cylinder	44.5294
Indicated horses-power developed in the large cylinder	32.0964
Aggregate indicated horses-power developed by the engine	76.6258
Net horses-power developed in the small cylinder	42.2748
Net horses-power developed in the large cylinder	25.1340
Aggregate net horses-power developed by the engine	67.4088
Total horses-power developed in the small cylinder	61.3048
Total horsce-power developed in the large cylinder	26.7444
Aggregate total horses-power developed by the engine	88.0492

CYLINDER PRESSURES REDUCED TO LARGE CYLINDER ALONE.

Indicated pressure in pounds per square inch that would be on the piston of the large cylinder, were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder and the quotient added to the experimental indicated pressure on the piston of the large cylinder	33.017
Net pressure in pounds per square inch that would be on the piston of the large cylinder, similarly calculated to the immediately above	29.046
Total pressure in pounds per square inch on the piston of the large cylinder, corresponding to the aggregate total horses-power developed by the engine	56.110
Equivalent back pressure against the piston of the large cylinder, being the difference between the above total and indicated pressures, in pounds per square inch	23.093

RATIOS.

Percentum which the aggregate indicated horses-power developed by the engine are of the aggregate total horses-power	87.0261
Percentum which the aggregate net horses-power developed by the engine are of the aggregate total horses-power	76.5581

DISTRIBUTION OF THE POWER DURING THE ABOVE PERFORMANCE.

The distribution of the indicated horses-power during the above second trial of the vessel, on the 4th of August, 1881, with screw C, calculated in the manner previously described, is as follows:

	Horses-power.	Per centum of the net horses-power.
Indicated horses-power developed by the engine	76.6258	
Horses-power expended in working the engine *per se*	9.2170	
Net horses-power applied to the crank-pins	67.4088	100.00
Horses-power absorbed by the friction of the load	5.0557	7.50
Horses-power expended in overcoming the resistance of the water to the surface of the screw-blades	4.9698	7.37
Horses-power expended in the slip of the screw	14.0015	20.77
Horses-power expended in the propulsion of the vessel	43.3818	64.36
Totals	67.4088	100.00

THRUST OF THE SCREW.

The thrust of the screw as it would have been measured by a dynamometer directly applied to the shaft during the above performance, calculated in the manner previously described, is as follows: $43.3818 \times 33000 = 1431599.4$ foot-pounds of work per minute. Speed of vessel per minute $= \frac{10.97 \times 6086}{60} = 1112.72366+$ feet. And thrust of screw $= \frac{1431599.4}{1112.72366+} = 1286.57$ pounds.

DETERMINATION OF THE POWER EXPENDED IN OVERCOMING THE RESISTANCE OF THE WATER TO THE IMMERSED EXTERNAL OR WETTED SURFACE OF THE HULL.

The power expended in overcoming the resistance of the water to the wetted surface of the hull, during the second trial of the vessel, on the 4th of August, 1881, with screw C, calculated in the manner previously described, is as follows:

Mean speed of the watery molecules over the immersed surface of the hull = 1,260 feet per minute, or 21 feet per second, at which velocity the resistance of the square foot of wetted surface would be ($10^2 : 0.45 :: 21^2 :$) 1.9845 pounds. And the horses-power required to overcome this resistance for 364.5 square feet of immersed surface would be $\left(\frac{1.9845 \times 364.5 \times 1260}{33000} = \right)$ 27.6188; consequently, of the 43.3818 horses-power required to propel the hull alone at the experimental speed, $\left(\frac{27.6188 \times 100}{43.3818} = \right)$ 63.664 per centum were expended in overcoming the resistance of the water to the wetted surface, and the remaining 36.336 per centum were expended in the displacement of water by the immersed solid of the hull irrespective of the resistance of its immersed surface.

ROUGH-WATER TRIAL WITH SCREW C.—A trial was made on the 3d of August, 1881, of the duplicate Vedette boat with screw C, consisting of two consecutive runs over the base of one geographical mile at Stokes Bay, England, in rough water. The wind was a very strong breeze alternately on the bow and quarter. The vessel was tried at low speed, and indicator diagrams were taken. The draft of water was 2 feet 2 inches forward, and 3 feet 9 inches aft; mean, 2 feet $11\frac{1}{2}$ inches.

The steam was cut off at two-thirds of the stroke of the pistons of both cylinders, and in both cylinders the back pressure was cushioned at 0.85 of the return stroke. The throttle-valve was wide open. The following are the mean results:

Steam pressure in boiler in pounds per square inch above atmosphere	52.35
Vacuum in the condenser in inches of mercury	22.50
Back pressure in the condenser in pounds per square inch above zero	3.642
Number of double strokes made per minute by the pistons	273.2
Speed of the vessel per hour in statute miles of 5,280 feet	9.9534
Speed of the vessel per hour in geographical miles of 6,086 feet	8.6350
Slip of the screw in per centum of its speed	27.467
Indicated pressure on the piston of the small cylinder in pounds per square inch	34.22
Net pressure on the piston of the small cylinder in pounds per square inch	31.22
Indicated pressure on the piston of the large cylinder in pounds per square inch	8.02
Net pressure on the piston of the large cylinder in pounds per square inch	5.02
Indicated horses-power developed in the small cylinder	21.0996
Indicated horses-power developed in the large cylinder	15.2701
Aggregate indicated horses-power developed by the engine	36.3697
Net horses-power developed in the small cylinder	19.2499
Net horses-power developed in the large cylinder	9.5581
Aggregate net horses-power developed by the engine	28.8080
Indicated pressure in pounds per square inch that would be on the piston of the large cylinder, were the indicated pressure on the piston of the small cylinder divided by the ratio of the small to that of the large cylinder and the quotient added to the experimental indicated pressure on the piston of large cylinder	19.102
Net pressure in pounds per square inch that would be on the piston of the large cylinder, similarly calculated to the immediately above	15.130

DISTRIBUTION OF THE POWER DURING THE TRIAL IN ROUGH WATER WITH SCREW C.

The distribution of the indicated horses-power during the above trial with screw C in rough water, on the 3d of August, 1881, the wind being a very strong breeze on the bow and quarter alternately, was as follows, the calculations being made as previously described:

	Horses-power.	Percentum of the net horses-power.
Indicated horses-power developed by the engine	36.3697	
Horses-power expended in working the engine *per se*	7.5617	
Net horses-power applied to the crank-pins	28.8080	100.00
Horses-power absorbed by the friction of the load	2.1606	7.50
Horses-power expended in overcoming the resistance of the water to the surface of the screw-blades	2.7444	9.53
Horses-power expended in the slip of the screw	6.5654	22.79
Horses-power expended in the propulsion of the vessel	17.3376	60.18
Totals	28.8080	100.00

THRUST OF THE SCREW.

The thrust of the screw as it would have been measured by a dynamometer directly applied to the shaft during the above performance, calculated as previously described, is as folows: $17.3376 \times 33000 = 572140.8$ foot-pounds of work per minute. Speed of vessel per minute $= \frac{8.635 \times 6086}{60} = 875.876833+$ feet. And thrust of screw $= \frac{572140.8}{875.876833+} = 653.22$ pounds.

DETERMINATION OF THE POWER EXPENDED IN OVERCOMING THE RESISTANCE OF THE WATER TO THE IMMERSED EXTERNAL OR WETTED SURFACE OF THE HULL.

The power expended in overcoming the resistance of the water to the wetted surface of the hull during the above trial with screw C in rough water on the 3d of August, 1881, calculated in the manner previously described, is as follows:

Mean speed of the watery molecules over the immersed surface of the hull=861 feet per minute or 14.35 feet per second, at which velocity the resistance of the square foot of wetted surface would be (10^2: 0.45:: 14.35^2): 0.92665 pound. And the horses-power required to overcome this resistance for 355.5 square feet of immersed surface would be $\left(\frac{0.92665 \times 355.5 \times 861}{33000}=\right)$ 8.5950; consequently, of the 17.3376 horses-power required to propel the hull alone at the experimental speed $\left(\frac{8.5950 \times 100}{17.3376}=\right)$ 49.575 per centum, were expended in overcoming the resistance of the water to the wetted surface, and the remaining 50.425 per centum were expended in the displacement of water by the immersed solid of the hull, irrespective of the resistance of its immersed surface.

A considerable portion of the slip of screw C in this experiment was doubtless due to the violent pitching of the vessel, which at intervals threw a more or less considerable portion of the screw out of water, and this lessening of the screw's propelling effect during the sending was by no means compensated by its deeper immersion during the time the bow was lifting. The result was that the slip was much greater than was due to the increased resistance of the vessel arising from the roughness of the water.

REMARKS.

In order to obtain the greatest speed possible from a boat of given lineal dimensions, it must be constructed of as little weight as possible, with the view of lessening to the utmost its displacement of water and the area of its external immersed or wetted surface. But the lessening of the weight, which is only possible by reducing the scantling of the material, must not be carried so far as to jeopardize the strength of the vessel necessary for its intended use. The exact point at which the minimum of material can be combined with adequate strength is determinable only by practice; and the extensive experience of the "Herreshoff Manufacturing Company" in the construction of vessels of the Vedette boat type has enabled that firm to mark this point with precision. To place it as near lightness as possible, only the best and most carefully selected materials are used, while no pains or expense are spared to secure strength, durability, and excellence in the fastenings. Only the most skillful workmen are employed, selected from mechanics who make the building of such hulls a trade by itself.

The form of the vessel and the distribution of its weights, on which so much of its satisfactory performance depends, has been determined for given dimensions and speed by accurate trial and observation, every boat built being subjected to careful study and well-executed experiments. In this manner such great excellence has been attained as is only possible when a particular kind of manufacture is made a specialty, and success depends on producing the most superior article, irrespective of cost. The hulls of these boats, in whatever light considered, are beautiful specimens of scientific design and practical execution. They cannot be produced by persons accustomed to designing and building only vessels of much greater dimensions, to be employed for different purposes and in different places, however skillful they may be in such constructions.

In this connection there may be stated that as regards strength to resist abnormal shocks, concussions, &c., resulting from collision and rough handling, the light vessel, by reason of its less weight, is less liable to injury than the heavier vessel, its momentum, which under these circumstances is the cause of injury, being at equal speed proportional to the weight alone.

The boats are constructed of wood by preference, having been found to be much stiffer with equal weight than if built of iron or steel.

The resistance of the hulls of the Vedette boats was experimentally ascertained from the indicated horses-power developed by the engine at four different speeds, and the following table shows the power of the speed in which the resistance increased from the one speed to the other:

Speed of the vessel in geographical miles per hour.	Thrust of the screw in pounds.	Power of the speed in which the resistance increased.
6.68030	198.89	
9.15482	482.24	2.8107
10.97000	12·6.57	5.4250
11.74220	1589.30	3.1064

With a vessel of the small dimensions, light weight, and relatively high speed of these Vedette boats, the power of the speed in which the resistance of their hulls varies at different speeds changes abruptly and greatly, owing to the change of the form of the immersed solid of the vessel relatively to the water level, due to the excessive changes of trim undergone.

With the short after body, relatively to draft of water and speed, the ascending column of water which fills the space left by the advance of the vessel does not reach the surfaces of the after body in time to support it, and. as a consequence, this body sinks until it does meet the ascending column, thereby greatly varying the trim and increasing the resistance above what is due to the law of the square of the speed. The high wave observed abaft the stern is the capital of the ascending column of water, which has not only reached the water level, but is projected above it in virtue of its momentum.

The water refilling the space voided by the vessel's advance is mainly and often wholly, supplied vertically from the bottom line of the greatest immersed transverse section, as that is the line of greatest pressure. If this vertically ascending column of water strikes the bottom of the after body while still endowed with momentum, a certain portion of the latter, or the whole, will be utilized in the propulsion of the vessel. In this manner, when the length and depth of the after body, the form of its immersed surface, and the speed of the vessel are properly proportioned each to the other, the power expended by the fore body in displacing the water may be entirely regained by the work done propulsively upon the after body by the ascending column, so that the resistance of the vessel may be that alone which is due to its immersed external or wetted surface, irrespective of displacement.

The power expended by the fore body in displacing the water is what is due to lifting the water displaced from the center of gravity of the greatest immersed transverse section to a certain height above the water level. The water so lifted is then diffused in a wave of translation in all open directions over the general water level.

When the bow of a moving vessel is flaring or V-shaped beneath the water, the reaction of the latter tends to lift it with a force or pressure, other things equal, proportional to the square of the speed of the vessel. To counteract this lifting tendency there is only the constant force of the weight of the bow, which is unaffected by the speed. Now, it is evident that, let this weight be as great as it may, as it does not increase with the speed, while the lifting force increases as the square of the speed, there will be a speed when the bow will be lifted, and the less the weight of the bow the less speed will be required to lift it. When that speed is reached and the bow lifts, its *acting* weight in consequence of that lifting will be increased, less and less of it being supported by the water as it becomes more and more emersed, so that the lifting will be less than what is due to the upward reaction of the water with constancy of acting weight of bow. The effect of the lifting of the bow is to lessen the wetted surface, and the displacement of the vessel, and therefore to lessen proportionally its resistance.

Again, as the velocity of the ascending column of water, refilling the void left by the advance of the vessel, is due to gravity, and therefore constant for given conditions at all speeds of vessel, the greater that speed the farther abaft the stern will the head of the ascending column appear, and the greater will be the dropping or depression of the stern, the result of which dropping is that the bottom of the vessel presents an inclined surface to the water, whose reaction tends to lift it with a force in the ratio of the square of the speed of the vessel. To counteract this force there is only the constant weight of the vessel, so that when the speed becomes sufficiently great the vessel will be bodily lifted. In this manner the vertical reaction of the water against the bow and the inclined surface of the bottom of the vessel lessens the wetted surface and the displacement, and proportionally reduces the resistance of the vessel.

In the case of the Vedette boats, the alteration of trim commenced as soon as the very moderate speed of 7 geographical miles per hour was attained, and continued increasing with increased speeds, the resistance increasing in higher and higher ratio than the square of the speeds as the speed became greater and greater. When the speed of 12 geographical miles per hour was attained the vessel began to lift notably, and the ratio of its resistance in function of speed remained about constant up to the speed of 14 geographical miles per hour, when the vessel began to lift in a very marked manner, so that at the speed of 15 geographical miles per hour its resistance in function of speed had greatly lessened. The slip of the screw at this speed was about the same as at the speed of 10 geographical miles per hour. It is entirely owing to the lessened resistance of the vessel, due to her lifting, that the high speeds could be attained with the power developed by the engine. Had the resistance of the vessel in function of speed continued to be the same at high speeds that it was at medium speeds, the former would have been unattainable by the power actually applied. The exact height which the vessel lifted at the high speeds could not be measured, but it was a considerable fraction of her draught of water when at rest. The alteration of trim may be appreciated from the fact that at the speed of 14 geographical miles per hour the bow was 22 inches higher above the stern than when the vessel was at rest. Now, the draft of water forward from the lower edge of the rabbet of the keel was only 24 inches when the vessel was at rest.

The sea-going qualities of the Vedette boats were excellent, and their performance in rough water satisfactory. They were dry, buoyant, and quite stiff enough. The commission of French officers from the Chasseur, who experimented with the vessel to ascertain her stability, pronounced it sufficient. The remark may here be made that no heavy vessel of the dimensions of the Vedette boats, having so little free board, could be serviceable in a rough sea; their sea-going qualities were largely due to their lightness, their inertia being so small that they rose with the wave as it struck them, instead of being submerged by it.

At the speed of 14 geographical miles per hour, and with the helm hard over, the Vedette boats turned in a circle of about 300 feet diameter. The careening or heeling due to this turning, though marked, was not sufficient to be inconvenient.

The vessel, when at maximum speed, could be stopped in a few feet by backing the engine. The reversal of the latter was practically instantaneous, being done by a single pull of the hand on a lever. In a few seconds sternway was acquired. This rapidity of maneuvering is also due to the lightness of the hull relatively to the thrust of the screw.

The vibration of the vessel's stern, due to the pulsations of the screw, was hardly appreciable.

The vessel can make 14 geographical miles per hour for 6.43 hours with her bunkers filled with the best steam coal. This is equivalent to a distance of 90 geographical miles. Of course, by carrying more coal in bags, this distance can be proportionally increased. As the engine uses steam with condensation, there is no noise other than the hum of the blower to give notice of approach; and, as the combustion is maintained by a mechanically produced draft, the loss of the chimney, if shot away, would not affect the speed.

The machinery is very simple, light, and economical, the boiler especially being admirably adapted for vessels of this description, and superior for the purpose to any of which the writer has knowledge. It gives an excellent economic vaporization, inferior to none; is very durable, and will bear the roughest handling and most ignorant usage without suffering damage. It is lighter than any other boiler of equal grate surface, and immensely safer, an explosion being scarcely possible under any conceivable circumstances. Should one happen, the injurious effects would be reduced to a minimum, because the boiler contains the minimum weight of water. This is a very important fact for a boat of this description, as its boiler is liable to be pierced by a shot at any moment when it is on war service.

The Herreshoff coil boiler of the Vedette boats cannot be made to foam or prime under any conditions or with any rate of combustion, and works as well with greasy as with clean water, no care being necessary in that respect, so that the lubrication of the steam valves and pistons may be as copious as desirable. The only boiler in competition with the Herreshoff coil boiler for Vedette, torpedo, and similar boats is that of the locomotive type; but great difficulty is experienced with the latter when the combustion is highly forced. The priming or foaming is then excessive, and it is almost impossible to keep the tubes from leaking, whereas the Herreshoff coil boiler is reliable at all times and under all circumstances. To palliate the foaming of the locomotive type of boiler perfectly clean water, free of grease, has to be used in it, which prevents the proper lubrication of steam valves and pistons, and is a very troublesome and inconvenient condition, not easily fulfilled under the conditions of actual service. The radiation of heat from the Herreshoff boiler, owing to a peculiar arrangement of its coils recently introduced, is less, without external non-conducting clothing, than from any other type of boiler, even when clad; and this is an important feature when the confined space is considered in which the boiler has to be fired and the engine supervised.

At the close of the three consecutive hours' trials at the speed of over 14 geographical miles per hour the machinery required no adjustment, and appeared to be able to steam on indefinitely at the same rate. The want of endurance is in the men, not in the machinery, and the three hours' trial is more of a test of the former than of the latter. Of course, when the fire has to be thoroughly cleaned, the boiler having but a single furnace, the speed falls until the new fire comes into steady action.

With the briquettes d'Anzin, however, this cleaning was not required in the three hours, although in that time 168 pounds of briquettes had been burned on each square foot of grate surface. This is equivalent to steaming ten consecutive hours without cleaning with the ordinary combustion of nearly 17 pounds of coal per hour per square foot of grate surface.

The engine is excellently designed and built, with numerous peculiar and admirably arranged details. It has been carefully studied, with a view to lightening it to the utmost by the omission of every ounce of weight not indispensably necessary. The surface condenser consisting of only immersed outboard pipes, there are saved the weight and bulk of the shell and appurtenances of an inboard surface condenser, and also the weight and bulk of a pump for circulating the condenser water. The condensing surface itself, by reason of its more advantageous position, requires to be only about one-fifth what it would have to be if placed within the shell of an inboard condenser. The engine can be safely worked under a pressure of 150 pounds per square inch of boiler pressure.

With some modifications to the engine and screw, a considerable increase of speed can be obtained for the boats with the same boiler. The cylinders should be increased in diameter, and an independent adjustable cut-off valve placed on each, by means of which an important addition of power could be had from the same consumption of fuel per hour, not so much as a consequence of using the steam more expansively as of the ability to make a more economical distribution of the pressure in the two cylinders by lessening the loss of pressure due to the transfer of the steam from the one to the other. For the most economical production of power by the compound engine the pressure at the end of the stroke of the piston of the small cylinder must not exceed, except by an insignificant amount, the back pressure against it. This requires the steam to be considerably expanded in the small cylinder, if anything like equality of power is to be maintained between it and the large cylinder. To regulate the pressure in the two cylinders, so as to produce the desired balance and proportion, an adjustable cut-off valve on both is requisite.

The diameter and pitch of the screw should be increased, whereby its losses of useful effect at high speeds could be sensibly lessened.

All the pumps should be worked by a separate steam cylinder, whereby the proper reciprocating speed for their efficient action could be always commanded.

The blowing engine should be able to maintain an air pressure in the fire-room above the atmosphere, with the boiler in use, of not less than the equivalent of a column of water 5 inches high.

The modifications would very slightly increase the weight of the machinery, but only to an insignificant degree in proportion to the benefit obtained. Even this increased weight would probably be more than offsetted by the decreased weight of fuel required to be carried for the production of a given power during a given time.

The air joints of the hatches could be made tighter by means of India-rubber seats, whereby, as less air would be leaked from the fire-room, the same blowing engine could supply the furnace with more air and give a correspondingly increased combustion of fuel per hour. Provision should be made to consume about 100 pounds of best Welsh coal per hour per square foot of grate.

With these changes there would be no difficulty in permanently maintaining a speed of at least 15 geographical miles per hour, with a much higher speed, say 16 geographical miles, for short intervals of time, say from half an hour to an hour, the best Welsh coal being supposed in use.

Also, the vessel if smoothly sheathed with very thin copper, would have a less resistance and correspondingly greater speed, although its displacement and cost would be more by the weight of the copper; the resistance of the surface unit of smooth copper to the water being less than the similar resistance of wood, however smooth and well painted, especially after the vessel has been some time in water.

When the value of the vessel depends almost wholly on its exceptionally high speed, and when that speed must be so great that it cannot be exceeded by any other vessel of the size, no mere money consideration can be permitted to enter into production, particularly as the cost of a Vedette boat at the highest is but an insignificant item in the expenditures of a great naval establishment.

As Vedette boats, torpedo boats, and other small vessels of the kind, with the highest attainable speed, will be really important and indeed indispensable adjuncts to large ironclads and squadrons in future naval warfare, answering to light cavalry, scouts, &c., for an army, bringing intelligence of and intercepting similar vessels of the enemy when approaching under cover of darkness, or in shoal water, or from favorable position to attack a more formidable but unwieldy antagonist, a sufficient number for experiment and instruction should be supplied to the Navy, even in peace, for it is too late to begin learning a trade when the necessity of practicing it is at hand.

The total weight of the machinery and all its appurtenances, including water in boiler and tank, is 7,020 pounds, and admitting that the engine developed 169.4733 horses-power during the three consecutive hours' performances with briquettes d'Anzin, the vessel's speed being 14.17259 geographical miles per hour, there results that one horse-power was obtained from every 41.42 pounds of weight.

The experiments of three consecutive hours made with screw B, burning first anthracite and then briquettes d'Anzin, show strongly the *potential* superiority of the latter for generating steam in a given boiler in a given time.

There is but little difference between the *economic* value of the pound

of combustible matter of each fuel; that is, of the pound of what remains after deducting its earthy matter. The pound of combustible of the one fuel will vaporize about the same weight of water as the pound of combustible of the other fuel, other things being equal; but the very much more free-burning quality of the briquettes, that is, the greater affinity for oxygen of their constituents, than in the case of the anthracite, which is comparatively a very slow-burning fuel, enabled a very much greater weight of them to be consumed on a square foot of grate surface per hour, and consequently they generated in a given boiler a very much greater weight of steam per hour.

The earthy matter in the two fuels also exercised a great influence on the combustion. In the case of the briquettes it was only about 5 per centum of light ash, with scarcely any clinker, while in the case of the anthracite it was about 17 per centum, of which one-half formed a strongly adherent slag or clinker upon the grate, requiring considerable time for removal, and during its continuance preventing the passage of air through the grate into the fuel.

When anthracite was consumed the mean speed of the vessel during the three consecutive hours' trial was 11.85146 geographical miles per hour. When the briquettes were consumed the mean speed of the vessel was 14.17259 geographical miles per hour. In the case of the anthracite there was a steady falling off in the supply of steam from the commencement, which increased with accelerated rapidity as the trial proceeded, and had it lasted much longer the engine would have stopped for want of steam. The labor of the fireman was also most ar[illegible]ous. In the case of the briquettes, as much steam could be commanded at the end as at the beginning of the trial, and the labor of the fireman was moderate.

The difference of speed obtained from the two fuels, as a mean, during three hours was 2.32113 geographical miles per hour. That is to say, 19.585 per centum more speed was given by the briquettes than by the anthracite, an enormous difference, which would require the power obtained from the anthracite to be doubled to be made good. In other words, another boiler would have been required to obtain from anthracite the same quantity of steam per hour that was obtained from the briquettes.

The pound of briquettes, however, it must be remembered, did not produce twice the weight of steam produced by the pound of anthracite. The difference in the efficiency of the two fuels was not economic, but potential, a great many more pounds of briquettes being consumed per hour on each square foot of grate than of anthracite.

The air pressure produced in the fire-room by the blower above that of the external atmosphere was in the ratio of the square of the number of revolutions made by the blower in equal time. The maximum pressure that could be maintained was about equal to the pressure of a column of water at the temperature of 70° Fahrenheit $4\frac{1}{2}$ inches high, which is equivalent to a pressure of 0.16216 pound per square inch. To produce this pressure the blower had to make exactly 1,000 revolutions per minute. Conversely, the number of revolutions made by the blower per minute was in the ratio of the square root of the air pressure in the fire-room above the atmospheric pressure.

Respectfully submitted by, sir, your obedient servant,

B. F. ISHERWOOD,
Chief Engineer, U. S. Navy.

Engineer-in-chief WM. H. SHOCK, U. S. Navy,
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