REPORT

OF A

BOARD OF UNITED STATES NAVAL ENGINEERS

ON THE

HERRESHOFF SYSTEM

OF

MOTIVE MACHINERY

AS APPLIED TO THE STEAM-YACHT

LEILA,

AND ON THE PERFORMANCE OF THAT VESSEL,

MADE TO THE

BUREAU OF STEAM ENGINEERING,
NAVY DEPARTMENT.

JUNE 3, 1881.

WASHINGTON:
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ON

THE HERRESHOFF SYSTEM OF MOTIVE MACHINERY.

NEW YORK, *June* 3, 1881.

SIR: The undersigned, appointed a board by the department's order of November 5, 1880, to proceed to Bristol, R. I., and there make such trials with the steam-yacht Leila, embodying the latest improvements in the Herreshoff system of motive machinery, as would enable them to give an opinion of its applicability to naval vessels of the Leila's dimensions, and of the advisability of so applying it—reporting the same to the Bureau of Steam Engineering, accompanied by such data and drawings as would enable the department to follow the reasons for the opinions expressed and the recommendations made—have the honor to state that we proceeded as directed, and made a careful study of the system in question, collecting the required data and drawings, and making such experiments as appeared proper to us for a complete elucidation of the qualities of the Herreshoff system of motive machinery.

The entire expense of this investigation was borne by the Herreshoff Manufacturing Company, which furnished the vessel and crew, placing both at our disposal for the purpose of the experiments, and leaving us entirely untrammeled in their conduct, but supplying every facility for making them. The great cost of such trials, which is quite serious for individuals, prevented us from availing ourselves of this liberality any farther than was absolutely necessary for the requisite experimental determinations. We varied the conditions of the machinery as much as possible without resorting to costly alterations or readjustments, and believe our experiments will be found sufficiently exhaustive, not only for the immediate purpose, but also for the solution of some important problems in steam-engineering not hitherto experimentally examined.

The steam-yachts of the Herreshoff Manufacturing Company have long been celebrated for their high qualities. They are remarkable for the lightness of their hulls and machinery, for their economy in fuel, for the excellence of their design, materials, and workmanship, for their speed, for the extreme rapidity with which steam can be raised from cold water, and for their safety and freedom from accidents; all of which qualities have been progressively developed by a very long, costly, and intelligent experience, the practice of this company being to experiment thoroughly and accurately with its work, and to avail itself of the knowledge thus acquired. In this manner its system has been successively perfected, each construction being an improvement on those preceding it, so that the Leila's machinery presents many points of advance on those previously experimented with by boards of naval engineers.

In the following pages will be found the detailed dimensions and descriptions of the hull and machinery of the Leila; an account of the manner in which the experiments were made, and why they were so made; tables containing, *in extenso*, the data and results of the experiments; explanations of the quantities in these tables; and, finally, a discussion of the results, followed by the opinions based upon them. There will also be found drawings of the lines of the Leila's hull, of her engine, and of her boiler.

HULL.

The construction of the hull is composite, the frame being of angle-iron planked with Southern pine and sheathed with copper; the stem and stern-post are of wood. The water-lines of the hull are excessively sharp and the angle of its dead rise proportionally great. A house erected upon the deck forms the upper portion of the cabin and of the space inclosing the machinery, forward of which is the pilot-house. A narrow passage extends between these constructions and the light bulwark rising about 30 inches above the edge of the deck. There are two masts, fitted with a light schooner rig.

The draught of water of the hull proper at the stern was so small that the screw had partly to descend below the bottom of the keel in order to be wholly immersed; this required the addition of a skeg at the stern and below the keel, for the purpose of protecting the screw and of supporting the metallic shoe which extends horizontally beneath it. The aftermost end of this shoe sustains the lower pintle of the rudder; the latter is of copper and counterbalanced with the axis at one-fourth of its breadth from the forward edge.

The skeg is a right-angled triangle of wood, 15½ inches deep below the bottom of the keel, at the after side of the stern-post, and 60 inches long upon the bottom of the keel from the after side of the stern-post; its breadth is 7 inches.

The following are the principal dimensions and proportions of the hull, exclusive of the skeg, and at the experimental draught of water, which was that below given and to which all the dimensions and calculations refer. The vessel carried no load except the *personnel* embarked and one day's supply of coal.

Extreme length on top of deck	100 feet.
Length on water-line from forward edge of stem to after side of stern-post	95 feet 5 inches.
Extreme breadth on top of deck	15 feet 4 inches.
Extreme breadth on water-line	11 feet 9 inches.
Depth of hull amidship from lower edge of rabbet of keel to top of deck-beams	5 feet 10 inches.
Depth at stem from water-line to lower edge of rabbet of keel	2 feet 1 inch.

Depth amidship from water-line to lower edge of rabbet of keel	2 feet 10½ inches.
Depth at stern from water-line to lower edge of rabbet of keel	3 feet 1¼ inches.
Depth of keel forward below lower edge of its rabbet	6¼ inches.
Depth of keel aft, below lower edge of its rabbet	8½ inches.
Siding of keel	7 inches.
Distance from the forward edge of the stem to the greatest immersed transverse section	54 feet 6 inches.
Area of the water-line	699.5 square feet.
Area of the greatest immersed transverse section above the lower edge of the rabbet of the keel	19.0 square feet.
Area of the greatest immersed transverse section, including projected areas of keel and skeg	20.12 square feet.
Displacement above lower edge of rabbet of keel	1199.4 cubic feet.
Displacement (35 cubic feet per ton)	37.27 tons.
Area of external wetted or immersed surface of hull, exclusive of keel and skeg surfaces	901.5 square feet.
Area of surfaces of keel and skeg	34.0 square feet.
Aggregate area of wetted surfaces of hull, keel, and skeg	935.5 square feet.
Angle of dead rise at greatest immersed transverse section	21½ degrees.
Half angle of bow on water-line	8½ degrees.
Half angle of stern on water-line	14 degrees.
Ratio of the length to the breadth on water-line	8.1206.
Ratio of water-line to its circumscribing parallelogram	0.6239.
Ratio of the greatest immersed transverse section to its circumscribing parallelogram	0.5624.
Ratio of the displacement above lower edge of rabbet of keel to its circumscribing parallelopipedon	0.3721.
Distance between centers of angle-iron frames	1 foot 3 inches.
Angle-iron frames, molded	1½ inches.
Angle-iron frames, sided	2¼ inches.
Thickness of iron of angle-iron frames	¼ inch.
Deck-beams, molded	2 inches.
Deck-beams, sided	3½ inches.
Thickness of wooden stem	6 inches.
Thickness of wooden stern-post	7 inches.
Thickness of bottom and side planking	1¼ inches.
Thickness of deck planking	$1\frac{3}{16}$ inches.

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

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 outer end of the piston-rod being secured into a crosshead working between guides in the engine-frame, while the connecting-rod lies in direct extension between the crosshead journal and the crank-pin journal.

The forward or small cylinder operates a lever which works the air-pump, the feed-pump, and the circulating-pump, all of which are vertical, single-acting, and have the same stroke of piston. The axes of these three pumps are in the same vertical plane, which is parallel to the vertical plane passing through the axis of the crank-shaft. The feed-pump and the circulating-pump are plunger-pumps, with brass receiving and delivering valves. The air-pump is a lifting-pump without a foot-valve; its receiving valve is brass, circular, and placed in the piston; its delivering valve is also of brass and discharges into an open-topped hot-well or reservoir placed above the outboard water-line, the top of the air-pump being closed. The air-pump piston is not packed, but ground to a metallic fit in the brass barrel.

The cylinders are separated to allow the valve-chests to be placed between them, with sufficient additional space for the removal of their covers. The valves of each cylinder are a plain three-ported slide with a slide cut-off on its back; these valves are not counterbalanced, but work with the full steam pressure on their backs. The three-ported slides or steam-valves are operated each by a Stephenson link and two eccentrics, which serve as a reversing gear. The cut off valves are each operated by an eccentric. The three eccentrics of each cylinder are immediately beneath its valve-chest. The cut-off valve of the small cylinder is adjustable; that of the large cylinder is fixed to cut off at about one-third of the stroke of the piston from the commencement.

The engine-frames, four in number, are each in a single casting and bolted to a bed-plate, which is also a single casting extending under the entire length and breadth of the engine. The crosshead guides are on these frames, to the top of which the cylinders are bolted. The bed-plate has a semicircular bottom, and its side flanges are bolted to side keelsons. The crank-shaft has three bearings or pillow-blocks cast in the bed-plate, the forward crank being overhung.

The engine works with surface condensation. The surface condenser is composed of a single copper pipe placed on the outside of the vessel, beneath the water, and just about at the garboard strake. This pipe commences on one side of the vessel abreast of the after or large cylinder, extends to and around the stern-post, and thence along the opposite side of the vessel until abreast of the air-pump and forward 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 top of the boiler-coil, where it is revaporized, and the steam, passing first into the small cylinder and thence into the large one, is finally exhausted into the condensing tube. It is essential for satisfactory working that the delivering end of this tube should not exceed one-half the diameter of its receiving end; for if a larger diameter be given to the delivering end, a part of the exhaust steam will pass directly to the air-pump over the water of condensation in the tube. The delivering end of the tube must be small enough to remain completely filled with water for the exclusion of the steam from the pump.

The after main pillow-block serves also as the thrust pillow-block, the after journal of the crank-shaft being made with the necessary thrust rings upon it.

From the "separator," an adjunct of the boiler into which the steam and water from the bottom of the boiler-coil are discharged in order to be separated by gravity, before the steam is carried to the superheater and thence to the cylinders, the circulating-pump continuously draws water and forces it into the top of the boiler-coil, where it enters along with the feed-water from the feed-pump, thus maintaining a forced circulation through the coil of what may be termed "superfluous feed."

The cylinders and their valve-chests, including covers of both, are incased with polished brass, between which and the iron are air spaces.

The following are the principal dimensions of the engine:

Number of cylinders	2.
Diameter of the small cylinder	9 inches.
Diameter of the piston-rod of the small cylinder	$1\frac{5}{8}$ inches.
Net area of the piston of the small cylinder	62.58045 square inches.
Stroke of the piston of the small cylinder	18 inches.
Space displacement of the piston of the small cylinder, per stroke	0.65188 cubic foot.
Space in clearance, and steam passage at one end of small cylinder	0.05752 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	0.08824.
Length of steam port of small cylinder	7.5 inches.
Breadth of steam port of small cylinder	$1\frac{1}{16}$ inches.
Area of steam port of small cylinder	7.97 square inches.
Length of exhaust port of small cylinder	7.5 inches.
Breadth of exhaust port of small cylinder	1.75 inches.
Area of exhaust port of small cylinder	13.125 square inches.
Clearance of piston of small cylinder	$\frac{3}{16}$ 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	690 square inches.
Diameter of the large cylinder	16 inches.
Diameter of the piston-rod of the large cylinder	$1\frac{5}{8}$ inches.
Net area of the piston of the large cylinder	200.02545 square inches.
Stroke of the piston of the large cylinder	18 inches.
Space displacement of the piston of the large cylinder, per stroke	2.08360 cubic feet.
Space in clearance and steam passage at one end of large cylinder	0.14066 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	0.06751.
Length of steam port of large cylinder	13 inches.
Breadth of steam port of large cylinder	$1\frac{7}{16}$ inches.
Area of steam port of large cylinder	18.69 square inches.
Length of exhaust port of large cylinder	13 inches.
Breadth of exhaust port of large cylinder	2.5 inches.
Area of exhaust port of large cylinder	32.5 square inches.
Clearance of piston of large cylinder	$\frac{3}{16}$ 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	1515 square inches.
Diameter of the air-pump	7 inches.

Diameter of the piston-rod of the air-pump	1½ inches.
Stroke of the piston of the air-pump	6 inches.
Space displacement of the air-pump piston per stroke	0.1275 cubic foot.
Diameter of the plunger of the feed-pump	1½ inches.
Stroke of the plunger of the feed-pump	6 inches.
Space displacement of the plunger of the feed-pump, per stroke	0.0061358 cubic foot.
Diameter of the plunger of the circulating-pump	1⅛ inches.
Stroke of the plunger of the circulating-pump	6 inches.
Space displacement of the plunger of the circulating-pump, per stroke	0.0034515 cubic foot.
Width of all eccentric straps	2⅛ inches.
Depth of the packing ring in both steam pistons	⅞ inch.
Length of the condensing pipe	53 feet.
Inside diameter of condensing pipe at exhaust-steam end	5 inches.
Continuously decreasing to inside diameter at air-pump end of	2 inches.
Thickness of the metal of the condensing pipe (copper)	$\frac{1}{16}$ inch.
Exterior surface of the condensing pipe	50.2983 square feet.
Interior surface of the condensing pipe	48.5639 square feet.
Length of the connecting-rods between centers	49½ inches.
Diameter of the necks of the connecting-rods	$1\frac{9}{16}$ inches.
Diameter of crosshead journals	2¼ inches.
Length of crosshead journals	3¼ inches.
Diameter of forward crank-pin journal (overhung)	2¼ inches.
Length of forward crank-pin journal	4½ inches.
Diameter of after crank-pin journal	3⅞ inches.
Length of after crank-pin journal	3½ inches.
Number of crank-shaft journals	3.
Diameter of crank-shaft journals	3⅞ inches.
Length of crank-shaft journals	8 inches.
Diameter of screw-shaft inside of brass casing	3⅞ inches.
Number of thrust rings on crank-shaft	5.
Breadth of each thrust ring	½ inch.
Projection of each thrust ring from shaft	$\frac{9}{16}$ inch.
Length in the vessel occupied by the engine	66 inches.
Breadth in the vessel occupied by the engine	48 inches.
Height of engine above center of crank-shaft	96 inches.
Ratio of the space displacement of the piston of the large cylinder, per stroke, to that of the small cylinder	3.196293.

SCREW.

There is one true screw of brass with uniform pitch and four blades equispaced around the axis. The blades are at right angles to the axis; their forward and after edges when viewed in projection on a plane parallel to the axis are parallel. The outboard end of the screw-shaft is cased with brass, and supported by a lignum-vitæ bearing.

The following are the dimensions of the screw:

Diameter	4 feet 7 inches.
Diameter of the hub	7 inches.
Pitch (uniform)	8 feet.
Number of blades	4.
Length of the screw (uniform from hub to periphery)	9¾ inches.
Fraction of the pitch used	0.40625.
Helicoidal area of the screw-blades	9.4564 square feet.
Projected area of the screw-blades on a plane at right angles to axis	6.5941 square feet.

BOILER.

There is one Herreshoff double-coil boiler, with one furnace, circular in horizontal projection, around and over which the continuous pipe of wrought iron composing the heating surface is coiled spirally and symmetrically in two concentric coils, one coil being immediately on the outside of the other so as to surround it. This pipe contains the water to be vaporized, and the hot gases of combustion act on its exterior, enveloping every part of it from one end to the other. The grate, circular in plan, is inclosed by a circular wall of brick masonry, on the top of which the double coil rests, and the latter is surrounded by two concentric casings of sheet-iron, with an air space between. The uptake and the chimney rest on this casing, all three and the furnace having their axes in the same vertical line. The whole of the gases of combustion pass between the spirals from the inside to the outside of the two coils into the space between the latter and the casing, and from this space these gases ascend the chimney. The feed-water enters the inside coil at the extreme upper end, whence it flows partly by gravity, but mainly by the action of the feed-pump, down to the extreme lower end of this coil; thence into the extreme lower end of the outside coil, up which it ascends to the extreme upper end, being converted in its progress into steam. If the supply of feed-water relatively to the heat of the furnace be such that the former is entirely vaporized by only a portion of the heating surface in the coils, then the remaining portion of that surface will act to superheat the steam. As the latter effect should be avoided on account of the injurious action of the intense direct heat of the furnace on the iron of the pipe when unprotected by water, recourse is had to a forced circulation of a superfluous quantity of feed-water by means of a circulating-pump, which, by continually drawing this superfluous feed-water from the delivering end of the double coil and forcing it into the receiving end, keeps both coils always sufficiently filled with water to prevent steam superheating, let the quantity of water vaporized be what it may. Probably the pipe is filled, under all conditions, with a mixture of steam and water, but if the latter be in a certain proportion there will be no superheating of the former; if this proportion, however, be not attained, superheating will take place. The feed and the superfluous feed both enter simultaneously and at the same point. The mixed water and steam are projected from the delivering end of the coil pipe into the "separator," which is merely a closed cylindrical vessel where 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 a pipe, which, after winding spirally two and five-eighths times around the upper portion of the inside coil, appearing like an extension of the upper portion of the outside coil on which it rests, passes to the engine. All the surface in these two and five-eighths spirals is steam-superheating surface, which, being exposed to the gases of combustion at a high temperature, acts very efficiently for that purpose. The water collected in the "separator" is again pumped by the circulating-pump into the top of the pipe of the 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 draughts of steam made from it by the engine. Without the "separator" 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 the 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 down the spirals of the coil, has time to become vaporized during the progress. The water when acted on by gravity alone requires a long time to traverse the coil from top to bottom. When the feed-pump and the circulating-pump are acting, their pressure is, of course, added to the influence of gravity.

The pressure in the boiler decreases gradually from the receiving end of the coil pipe to the "separator." It is at the maximum where the feed-water enters, and at the minimum where the mixed steam and water are delivered. Under the conditions of ordinary practice the extreme difference of pressure varies from 5 to 10 pounds per square inch, according as more or less feed-water is pumped in during equal times. This excess of pressure at the receiving end of the coil over the pressure in the "separator," reacts against both the feed and circulating pumps, 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 surfaces and bends of the pipe between the separator and the pump and between the pump and the receiving end of the coil pipe.

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 amount 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.

The furnace consists of a circular grate 5 feet and 9 inches 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 5 feet 9 inches, and its outside diameter is 6 feet and 11 inches. The thickness of the wall is 7 inches, and its height above the bottom of the grate-bars is 14 inches; both it and the grate-bars rest upon an iron ring 5 feet 6 inches in inner diameter, 6 feet 10 inches in outer diameter, and half an inch in thickness. Below this ring, which forms its cap, is a circular wall of common brick masonry, 5 feet 6 inches in inner diameter, 7 feet in outer diameter, 9 inches thick, and 7½ inches high. This wall incloses the ash-pit, the bottom of which is of sheet-iron, having a depression at its center and towards its door. The opening for the ash-pit door is 30 inches wide and 11 inches high. Opposite the door is another opening for receiving the blast from a fan-blower. Excepting these two openings 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 18 inches wide by 12 inches high. The outer top and bottom corners of the brick wall below the grate-bars are protected by angle-irons bent to ring form.

Upon the top of the brick wall inclosing the furnace rests the double coil of continuous wrought-iron pipe. The inner coil may be conceived to be wound spirally around the frusta of two right cones, one superimposed upon the other, and having their axes in the same vertical line. These imaginary frusta form the inner dimensions of the inside coil. The lower one is 5 feet 5 inches in diameter at bottom, 4 feet 8 inches in diameter at top, and 4 feet 3 inches in height. The upper one is 4 feet 8 inches in diameter at bottom, 1 foot 4 inches in diameter at top, and 8 inches in height. Above these two frusta the pipe forming the inside coil is extended into a nearly horizontal spiral of 6 feet 10 inches in extreme diameter, formed of eleven and a half circumvolutions, and placed as low as the upper frustum will allow. This horizontal spiral, situated in the uptake and above the coils, is exposed to the hot gases of combustion just before they enter the chimney, and after as much of their heat as possible has been extracted by the coils with which they first come in contact. Consequently, it acts as a heater, the feed-water and circulating water being delivered into it at one extremity, forced round the spirals by the feed and circulating pumps, and emerging from it into the inside coil at the other extremity, with which it is continuous. At the center of the horizontal spiral is an opening 1 foot 6 inches in diameter, immediately over the 1 foot 4 inches diameter opening forming the upper diameter of the upper frustum. These openings are covered by a plate of wrought iron, to prevent the gases of combustion from passing through, and to force them to pass out entirely between the spirals of the double coil.

The pipe composing the horizontal spiral is of wrought iron, lap-welded, $1\frac{7}{8}$ inches outside diameter and $1\frac{1}{2}$ inches inside diameter; thickness of metal $\frac{3}{16}$ inch. The convolutions are separated by spaces $\frac{5}{16}$ inch wide, through which pass the gases of combustion. The form of the horizontal spiral is maintained by four equispaced flat bars of wrought iron, $\frac{5}{8}$ inch thick, laid radially upon its top, to which bars the convolutions of the spiral are held by wrought-iron stirrups of $\frac{5}{16}$ inch diameter. The length of the axis of the horizontal spiral is 150 feet. The exterior surface of the horizontal spiral is 73.68125 square feet, its interior surface is 58.90500 square feet, and its content is 1.84075 cubic feet.

The inside coil, starting from the top, is composed of four turns or helical spirals of wrought-iron lap-welded pipe, $1\frac{7}{8}$ inches outside diameter and $1\frac{1}{2}$ inches inside diameter, thickness of metal, $\frac{3}{16}$ inch; five spirals of $2\frac{3}{8}$ inches outside diameter and 2 inches inside diameter, thickness of metal, $\frac{3}{16}$ inch; eight and a half spirals of $2\frac{7}{8}$ inches outside diameter and $2\frac{1}{2}$ inches inside diameter, thickness of metal $\frac{3}{16}$ inch; and five and a half spirals of $3\frac{1}{2}$ inches outside diameter and 3 inches inside diameter, thickness of metal $\frac{1}{4}$ inch. The four spirals of $1\frac{7}{8}$ inches outside diameter pipe and the five spirals of $2\frac{3}{8}$ inches outside diameter pipe form the top of the furnace and are in close contact, so that none of the gases of combustion can pass between them. The eight and a half spirals of $2\frac{7}{8}$ inches outside diameter pipe and the five and a half spirals of $3\frac{1}{2}$ inches outside diameter pipe form the sides of the furnace, and are separated by spaces through which all the gases of combustion pass. The eight and a half spirals of $2\frac{7}{8}$ inches outside diameter pipe are held $\frac{7}{16}$ inch apart by wrought-iron stirrups of that diameter. The five and a half spirals of $3\frac{1}{2}$ inches outside diameter pipe are held $\frac{1}{2}$ inch apart by wrought-iron stirrups of that diameter. The form of the four spirals of $1\frac{7}{8}$ inches outside diameter pipe and of the five spirals of $2\frac{3}{8}$ inches outside diameter pipe is maintained by eight flat wrought-iron bars, $\frac{5}{8}$ inch thick, equispaced in pairs, four on top and four below the spirals, and arranged radially, the ends of each pair of upper and lower bars being bolted together. The form of the eight and a half spirals of $2\frac{7}{8}$ inches outside diameter of pipe and of the five and a half spirals of $3\frac{1}{2}$ inches outside diameter of pipe is maintained by four flat bars of wrought iron, $\frac{5}{8}$ inch thick, equispaced on the outside of the spirals and arranged vertically, to which the separating stirrups are bolted.

The axis of the four spirals of $1\frac{7}{8}$ inches outside diameter pipe is 25.5 feet long. The exterior surface of these spirals is 12.51731 square feet, the interior surface is 10.01385 square feet, and their content is 0.31292 cubic foot.

The axis of the five spirals of $2\frac{3}{8}$ inches outside diameter pipe is 58 feet long. The exterior surface of these spirals is 36.03295 square feet, the interior surface is 30.36880 square feet, and their content is 1.26537 cubic feet.

The axis of the eight and a half spirals of 2¾ inches outside diameter pipe is 137.5 feet long. The exterior surface of these spirals is 103.49282 square feet, the interior surface is 89.99375 square feet, and their content is 4.68713 cubic feet.

The axis of the five and a half spirals of 3½ inches outside diameter pipe is 95 feet long. The exterior surface of these spirals is 87.04850 square feet, the interior surface is 74.61300 square feet, and their content is 4.66331 cubic feet.

The length of the axis of the inside coil is (25.5+58+137.5+95=) 316 feet. The exterior surface of that coil is 239.12158 square feet, the interior surface is 204.98940 square feet, and its content is 10.92873 cubic feet.

The outside coil, continuous with the inside coil and connecting with it at the bottom, is composed of nine and three-fourths spirals of wrought-iron lap-welded pipe 3½ inches outside diameter and 3 inches inside diameter, thickness of metal ¼ inch. The least space between the outside coil and the inside coil is 1½ inches; that is to say, these coils are separated by an annular space which is 1½ inches wide at its least width. The spirals of the outside coil are held ½ inch apart by wrought-iron stirrups of that diameter. The form of these spirals is maintained by four flat wrought-iron bars ⅜ inch thick, equispaced and placed vertically between the two coils.

The axis of the nine and a half spirals of the outside coil is 188 feet long. The exterior surface of these spirals is 172.26440 square feet, the interior surface is 147.65520 square feet, and their content is 9.22845 cubic feet.

The water-heating surface of the boiler consists of a continuous wrought-iron lap-welded pipe wound in spiral coils around a common vertical axis. The length of this pipe is (150+316+188=) 654 feet. Its outside diameter varies from 1⅞ to 3½ inches, and its inside diameter varies from 1½ to 3 inches; its thickness of metal varies from $\frac{3}{16}$ to ¼ inch. Its exterior surface is 485.01723 square feet, its interior surface is 411.54960 square feet, and its content is 21.99793 cubic feet.

The exterior surface of the water-heating pipe is distributed as follows:

	Per centum.
In the heater in the uptake	15.18
In the inside coil	49.30
In the outside coil	35.52
	100.00

The interior surface of the water-heating pipe is distributed as follows:

	Per centum
In the heater in the uptake	14.31
In the inside coil	49.81
In the outside coil	35.88
	100.00

The superheater for superheating the steam consists of two and five-eighths turns or helical spirals of wrought iron lap-welded pipe $3\frac{1}{2}$ inches in outside diameter and 3 inches in inside diameter, thickness of metal $\frac{1}{4}$ inch. It is placed on the top of the outside coil and is held in form in exactly the same manner as described for that coil, from which, however, it is entirely distinct, receiving the steam from the "separator" and delivering the same to the engine. The length of the axis of the superheater pipe is 48 feet; its exterior surface is 43.9824 square feet, its interior surface is 37.6992 square feet, and its content is 2.3558 cubic feet.

In order to prevent the outside coil and the superheater spirals from straightening by the pressure within the pipe, they are held against that force by four wrought-iron bands of $\frac{5}{8}$-inch diameter metal arranged upon the outside of the coil and superheater, diagonally, passing slantwise from top to bottom and crossing at the center.

The water-heater, the inside and outside coils, and the steam-superheater, as above described, are inclosed by two concentric cylindrical casings of sheet-iron $\frac{1}{16}$ inch thick, separated by a space $\frac{5}{8}$ inch wide, which is filled with mineral wool (steam-blown furnace slag) as a non-conductor of heat. The entire width of the two casings and intervening space is $\frac{3}{4}$ inch. The bottom of the casing is on a level with the bottom of the grate-bars, both resting upon the iron ring previously described, the lower portion of the casings thus embracing the fire-brick wall of the furnace. The outside diameter of the outside casing is 7 feet, and its height is 6 feet 7 inches. All the gases of combustion, after passing between the spirals of the two coils and of the superheater, impinge on the inside casing, which has thus nearly their temperature.

The uptake rests symmetrically upon the casings, and is composed like them of two parallel sheet-iron plates $\frac{1}{16}$-inch thick, with a $\frac{5}{8}$-inch intervening space filled with mineral wool. The form of the uptake is a frustum of a right cone, 7 feet in diameter at bottom, 2 feet in diameter at top, and 7 inches in height.

The chimney rests upon the uptake, and is 2 feet in diameter and 20 feet in height above it.

The "separator" is placed by the side of the boiler, with a space of $7\frac{1}{2}$ inches between them. It is simply a hollow cylinder formed of $\frac{3}{8}$-inch thick boiler-plate, and has both top and bottom closed. It receives at the top the mingled water and steam from the top or delivering end of the outside coil, which is prolonged into the "separator" for about one-third the height of the latter. In the "separator" the water is separated from the steam by gravity and falls to the bottom, while the steam is carried off from the top of the "separator" by the superheating coils which lead it to the engine. The "separator" is thus an independent but essential adjunct of the boiler, intended to act both as a steam-reservoir or steam-drum and as a water-trap. The top is fitted with a safety-valve of the usual construction, and the bottom is fitted with a blow-off pipe and cock for draining the "separator" and blowing out any sediment that may collect in it. On the side of the lower portion of the "separator" is a glass water-gauge of the usual construction, which shows on inspection whether there is an excess of feed-water passing through the coils of the boiler; the proper performance of the boiler requiring always such excess. From near the bottom of the "separator" a pipe proceeds to the circulating-pump, which continually removes this excess of feed-water and forces it back into the boiler by a pipe connecting the delivery of the pump with the receiving end of the heater coil.

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

Diameter of the boiler to outside of casing	84 inches.
Height of boiler from bottom of ash-pit to top of uptake	99 inches.
Height from bottom of ash-pit to top of grate-bars	15 inches.
Height from top of grate-bars to top of brick wall around furnace	12 inches.
Height from top of brick wall to top of the coils	64 inches.
Diameter of the furnace	69 inches.
Area of the grate surface	25.9673 square feet.
Area of water-heating surface measured on outside of coil pipe	485.01723 square feet.
Area of water-heating surface measured on inside of coil pipe	411.54960 square feet.
Area of steam-superheating surface measured on outside of coil pipe	43.9824 square feet.
Area of steam-superheating surface measured on inside of coil pipe	37.6992 square feet.
Aggregate area of the spaces between the spirals of the inside coil for the passage of the gases of combustion	6.5600 square feet.
Aggregate area of the spaces between the spirals of the outside coil and of the superheater coil for the passage of the gases of combustion	11.6600 square feet.
Aggregate area of the spaces between the spirals of the horizontal heater, and between it and the casing, for the passage of the gases of combustion	10.0000 square feet.
Cross area of chimney	3.1416 square feet.
Diameter of the chimney	2 feet.
Height of the chimney above the level of the grate-bars	27 feet.
Interior diameter of the "separator"	14 inches.
Interior height of the "separator"	50 inches.
Height of steam room in the "separator"	41 inches.
Steam room in the "separator"	3.6525 cubic feet.
Steam room in the superheater	2.3558 cubic feet.
Total steam room	6.0083 cubic feet.
Aggregate contents (water and steam) of the heater, and of the inside and outside coils	21.99793 cubic feet.
Square feet of water-heating surface, measured on outside of coil pipe, per square foot of grate surface	18.6780
Square feet of water-heating surface, measured on inside of coil pipe, per square foot of grate surface	15.8488
Square feet of steam-superheating surface, measured on outside of coil pipe, per square foot of grate surface	1.6938
Square feet of steam superheating surface, measured on inside of coil pipe, per square foot of grate surface	1.3112

Square feet of grate surface per square foot of space between the spirals of the inside coil for the passage of the gases of combustion	3.9584
Square feet of grate surface per square foot of space between the spirals of the outside coil and of the superheater coil for the passage of the gases of combustion	2.2270
Square feet of grate surface per square foot of space between the spirals of the horizontal heater and between it and the casing for the passage of the gases of combustion	2.5967
Square feet of grate surface per square foot of cross area of chimney for the passage of the gases of combustion	8.2656

THE EXPERIMENTS.

With the vessel and machinery as described, there were made the experiments whose data and results are given in the following tables, numbered 1, 1 continued, 2, and 3. The purpose of the experiments was to ascertain the manner and extent to which the economic development of the power was affected by such changes of the distribution of the steam in the cylinders as could be made by the possible variations of the valve gear; the effect upon the economy of the power due to different developments of power caused by varying the piston pressures; the manner and extent to which the superheating was affected by the development of these different powers; the comparative economy of producing a given power in the large cylinder alone of the compound engine, and in its two cylinders combined; and the potential and economic vaporization of water in the boiler by anthracite consumed at different rates of combustion.

Tables Nos. 1 and 1 continued constitute in fact one table, the division having been made merely for facility of manipulation. In these two tables will be found all the experimental data and results relating to the performance of the vessel and to the development and economy of the power; they contain, likewise, the necessary computations for showing the behavior of the steam in the cylinders and the degree to which it underwent condensation there. The consumption of anthracite is not given in these two tables, because the duration of the experiments was not long enough to determine it with certainty. The economy of the power is found, however, with certainty from the weight of feed-water pumped into the boiler during each experiment, which weight admitted of being ascertained with absolute precision. With the exception of the weight of anthracite consumed, all the measured quantities for each experiment are given in these tables.

Table No. 2 shows the distribution of the power developed by the engine during the experiments whose data and results are in Tables 1 and 1 continued. By means of this distribution there are known how much of the power was absorbed by the friction of the engine alone, how much by the friction of the load, how much in overcoming the resistance of the water to the surface of the screw-blades, how much was expended in the slip of the screw, and how much was exerted in the propulsion of the vessel, *per se;* the latter power being identical with the power that would have been given by a dynamometer attached to the screw-shaft.

Table No. 3 gives the weight of anthracite consumed and the weight of water vaporized by it during each day of the experiments, whose data and results are in Tables Nos. 1 and 1 continued. It will be observed that these experiments were made on the 11th, 12th, 13th, 15th, 16th, 17th, and 18th of November, but that on each of these days, excepting the 11th and 15th, there was more than one experiment. It was possible to ascertain exactly the total weight of anthracite consumed each day and the total weight of water vaporized thereby, but it was not possible to ascertain the weight of anthracite consumed during any portion of a day, though the weight of water vaporized could be known exactly from minute to minute. In order, therefore, to obtain with exactness the potential and economic vaporization of the boiler with anthracite, the quantities are taken for whole days' experiments, and everything related to these purposes—total and mean quantities—will be found in Table No. 3.

Table No. 4 contains data and results of an experiment, additional to the experiments in Tables Nos. 1 and 1 continued, which was made by a previous board of naval engineers, consisting of Chief Engineers Isherwood, Zeller, and Carpenter, the first two being also members of the board making the experiments whose data and results are given in Tables Nos. 1 and 1 continued. This additional experiment was made on the 3d of August, 1880, and continued 13.5500 consecutive hours, during which all the quantities were ascertained in precisely the same manner as those in Table No. 3, and give, like them, the potential and economic vaporization of the boiler with anthracite under the experimental conditions.

MANNER OF MAKING THE EXPERIMENTS.

In commencing the first experiment for the day—the vessel being at anchor off the city of Bristol, R. I.—the boiler was filled with water, and a fire made with pine wood, whose weight was not noted, but which was allowed to burn entirely out by the time the steam had reached its normal pressure. A new fire was then kindled with a weighed quantity of pine wood, upon which, as soon as it was properly ignited, the anthracite was thrown. When the latter had attained uniform combustion, the vessel was got under way and proceeded to the Bristol Ferry light-house, situated on a small rock in Narragansett Bay, and marking the upper end of the course run. This course was nearly a straight line, extending exactly twelve statute miles of 5,280 feet from the Bristol Ferry light-house to the lower end of the South Dumpling Rock, lying in the bay, near its mouth, and marking the lower end of the course. The vessel during each run was made to pass within a few yards of each terminus of its route, so there could be no doubt of the accuracy of the distance run. This was the base in all the experiments except experiments A and G, Table No. 1. The base for experiment A was a portion of the base above described, and extended 8.6830 statute miles to a point abreast of Gould Island light-house. The base for experiment G was a straight line 3.14394 statute miles long, extending from Bristol Ferry light-house to Sandy Point light-house. These two experiments were made over the shorter bases for want of time to make them over the long base used for the remaining fourteen experiments.

For each experiment the base was run over an equal number of times in each direction so as to equalize the effects of wind and tide, and the mean of the speeds given by the runs was taken as the true speed. In all cases the runs were in pairs. The wind was very variable, but never exceeding a moderate breeze and generally on bow and quarter. Experiments J and K were made in a calm. The condition of the water was also variable, being perfectly smooth in experiments J and K, while in experiments A, B, and O it was rough enough to cause some rolling and a slight pitching in the vessel. As an average there was a gentle breeze and a very light swell.

The vessel's trim and draught of water remained sensibly constant throughout all the experiments. The same weight of coal was on board each morning, and the same persons embarked, no changes being made in the other weights.

For each run the time of passing each end of the course or base was taken by two observers with a seconds watch; and the number of revolutions made by the screw during each run was determined by two observers, also from two counters—one in the engine-room, worked from the engine by a lever, the other, a rotary one, in the cabin, worked from the screw-shaft by a band. These observations always coincided within an insignificant extent.

The weight of feed-water pumped into the boiler was ascertained with absolute exactness by measurement in a standard United States gallon tin measure containing 231 cubic inches, or 8.34007 pounds, of water at the temperature of 39° Fah. under the pressure of 29.92 inches of mercury; the weight of water consumed at the maximum being sufficiently small to allow its accurate measurement by this means. The water of condensation was discharged by the air-pump from the condenser into a large wooden receiving tub conveniently placed, and in which a thermometer was kept immersed. From this tub the water was measured by hand with the gallon measure into a delivering tub, also of wood, from which it was drawn by the feed-pump and forced into the boiler. The two tubs were situated side by side. The water-level in the receiving tub was always kept at the same height, namely, that of the depth of the gallon measure, which as soon as filled was emptied into the delivery tank. Each time the gallon was emptied a mark was made on a tally-board kept by a person especially detailed for that duty only.

The number of gallons of water pumped into the boiler during each run or passage of the vessel over the base was known exactly, the count being marked at the precise moment the vessel passed each terminus of the base. This also gave the number of gallons of water pumped into the boiler during the turning of the vessel at each end of the base. Likewise, the total number of gallons of water pumped into the boiler from the first starting of the engine in the morning to the final stopping of the same in the evening, every gallon being duly tallied between those times. For the purpose of checking any possible error, the number of gallons of water pumped into the boiler was noted every ten minutes in an appropriate column of the log, or tabular record, kept during the experiments.

All the anthracite consumed was accurately weighed on a delicate steelyard, and all the refuse from it in ash, clinker, &c., was similarly weighed in the dry state. After the conclusion of the day's experimenting, the fire, which up to the end of the last run had been kept in uniform condition, was allowed to burn out as the vessel returned to her anchorage, and the engine was operated until it stopped for want of steam, when the contents of the furnaces were drawn and the unconsumed coal picked out, weighed, and deducted from the quantity of anthracite expended. In all the experiments the anthracite was burned with natural draught; and during their continuance from the 11th to the 18th of November the heating surfaces were not swept of soot, nor had they been for several months previously.

While each experiment was being made, there were entered in the appropriate columns of the log, or tabular record, at intervals of ten minutes, the steam pressure in the boiler and in the receiver, the vacuum in the bottom of the air-pump, the height of the barometer, the temperatures of the superheated steam, of the feed-water in the receiving tub, of the air on deck, and of the water in the bay. Simultaneously, at intervals of ten minutes, there was taken from each end of each cylinder an indicator diagram. The means of all these observations are given in Table No. 1.

Four excellent Richardson indicators were used and kept permanently in position, one at each end of each cylinder, with which it was connected by a short straight pipe of large bore. These instruments gave satisfactory diagrams, which were skillfully taken by experts.

The firemen were accustomed to the Herreshoff boiler, and the experienced pilot kept the vessel evenly on her course with but little use of the helm. During all the experiments except those at the maximum, lettered A and B in Table No. 1, the steam was carried with but very slight variation, and, in the case of the two experiments at the maximum, the variation was practically unimportant. In each experiment the throttle-valve was kept wide open. There was no stop-valve on the boiler.

EXPLANATION OF TABLES NOS. 1 AND 1 CONTINUED.

In Tables Nos. 1 and 1 continued, which form, in effect, one table only, the division having been made for convenience of handling, each column contains the data and results of an experiment which, for facility of reference, is designated by a letter at the head of the column. For the same reason, the different quantities are grouped as they stand in natural relation, and the lines containing them are numbered, the numbers extending continuously from Table No. 1 through Table No. 1 continued.

Line 1 contains the dates of the experiments; these latter do not succeed each other in the order of their dates, but in that of the power developed, as a matter of convenience in considering them.

Line 2 gives the number of times the base was passed over, which was not the same for each experiment, varying from twice to ten times. Want of time alone prevented the base from being passed over ten times for each experiment; but so numerous were the observations during each run, so accurate the measurements, and so thorough the organization, that the experiment of but two runs can be relied on as correct. In general, the experiments of the fewest runs were made at the slowest speeds, so that in *time* the experiments varied much less than appears from the number of runs made.

Line 3 gives the length of the base for each experiment. In all but two experiments, this length was twelve statute miles of 5,280 feet, a sufficient distance to reduce to nearly nothing any slight inaccuracies of observation that might creep in at the two ends where the vessel was turned. The shorter bases used in two experiments were in consequence of the want of daylight to run the long base.

Engine.—Line 4 contains the mean pressure in the "separator" of the boiler, in pounds per square inch above the atmosphere, obtained from observations at ten minutes intervals.

Line 5 contains the mean pressure in the receiver between the small and large cylinders, in pounds per square inch above the atmosphere, obtained from observations at ten minutes intervals.

Line 6 gives the position of the throttle-valve, which was wide open throughout each experiment; and, as there was no stop-valve on the boiler, whatever throttling of the steam existed was due to the cross area, length, and sinuosity of the steam-pipe alone.

Line 7 gives the fraction completed of the stroke of the piston of the small cylinders when the steam was cut off, and line 8 gives the same quantity in the case of the large cylinder. These points of cutting off the steam are, in both cases, the means from all the indicator diagrams taken, the measurements being to the point of inflection between the throttle and expansion curve on the diagrams. The point of cutting off varied in the different experiments, even while the valve gear to ordinary observation remained invariable. With everything else the same, the point of cutting off changed slightly with increased speed of piston, *sures, different degrees of superheating,*

The small cylinder disconnected and not in use. The large cylinder used alone with an independent cut-off. Experiments made with decreasing boiler pressure and with lessening degrees of steam superheating.	
O	P
9.44 a. m., Nov. 16.	3.25 p. m., Nov. 16.
6.	2.
12.	12.
44. 93	21. 43
40. 65	19. 43
Wide.	Wide.
....	....
0. 3345	0. 3712
2. 6554	2. 4333
30. 17	30. 19
25. 59	25. 98
2. 248	2. 067
191. 57236	147. 63465
2517. 9443	1354. 3427
2787039. 8214	1520410. 4913
344. 00	304. 00
292. 30	261. 39
51. 70	42. 61
96. 50	71. 00
38. 00	40. 00
47. 00	47. 00
13. 666505	10. 697362
11. 856580	9. 280656
23. 9908	20. 2958
....	....
....	....
....	....
....	....
....	....
....	....
....	....
....	....
54. 53	30. 90
43. 74	24. 90
17. 24	10. 93
6. 060	4. 583
4. 32	4. 07
28. 400	15. 789
26. 400	13. 789
34. 460	20. 372

pressures, different degrees of

he small cylinder disconnected and not in use. The large cylinder used alone with an independent cut-off. Experiments made with decreasing boiler pressure and with lessening degrees of steam superheating.

O	P
....	....
98. 93359	42. 38728
98. 93359	42. 38728
....	....
91. 96643	37. 01806
91. 96643	37. 01806
....	....
120. 04406	54. 69083
120. 04406	54. 69083
....	....
60. 53412	25. 79103
60. 53412	25. 79103
25. 450854	31. 951630
27. 378950	36. 586000
20. 975168	24. 763616
28170. 8146	35704. 3550
30304. 9691	41072. 1278
23216. 8074	27800. 0990
....	....
....	....
....	....
2044. 0928‖	994. 4041 ‖
2228. 5625	1099. 7534
160. 8455	67. 0969
2389. 4080	1166. 8503
....	....
....	....
473. 8515¶	359. 9386¶
18. 8190§	26. 5767§
128. 5363	187. 4924
5. 1048	13. 8438
82. 4144	77. 5034
76. 6106	67. 6860
28. 400	15. 789
26. 400	13. 789
34. 460	20. 372
6. 060	4. 583

ndensed.

with increased boiler pressure which added to both the pressure on the backs of the cylinder-valves and to the difference caused by expansion, and with the more or less keying up of the main journals, &c.

Line 9 shows the number of times the steam was expanded. This was obtained by dividing the sum of the space displacement of the piston of the small cylinder up to the point of cutting off the steam and the space in the clearance and steam passage at one end of that cylinder into the sum of the space displacement of the piston of that cylinder per stroke and the space in the clearance and steam passage at one end of that cylinder, and then dividing this quotient into the sum of the space displacement of the piston of the large cylinder per stroke and the space in the clearance and steam passage at one end of that cylinder.

Line 10 contains the mean height of the barometer in inches of mercury, obtained from observations made every ten minutes on a tested aneroid barometer.

Line 11 contains the mean vacuum in the bottom of the air-pump in inches of mercury, obtained from observations made every ten minutes on a gauge directly applied. The vacuum referred to is the difference between the barometric pressure (line 10) and the pressure of the mingled air and uncondensed steam in the bottom of the air-pump (line 12). Owing to the fact that the surface condenser of the Leila consisted of one continuous pipe through which all the exhaust steam had gradually to pass, the condensation had necessarily to be much more gradual than in the case of the usual surface condenser, placed inboard and composed of many tubes, through which the injection or condensing water passed, while the exhaust steam was exhausted around them and thus instantaneously thrown upon all their surface at once instead of being brought successively along it. From this cause is due, too, the fact that for equal vacuums in the bottom of the air-pump the back pressure against the piston will be greater with condensers like the Leila's than with the ordinary inboard surface condenser. In the continuous pipe surface condenser of the Leila a considerable difference exists between the vacuum at the end of the pipe receiving the exhaust steam from the cylinder and the vacuum at the opposite end delivering the water of condensation, uncondensed steam, and inleaked air to the air-pump, the back pressure gradually decreasing from the receiving to the delivering end of the pipe, so that a gauge would show a different back pressure at every point of the pipe, whereas in the usual surface condenser the back pressure is the same in every portion of it.

Line 12 shows the back pressure in the bottom of the air-pump in pounds per square inch above zero. It is the difference between the quantities on the lines 10 and 11, expressed in pounds per square inch. This pressure is of use when compared with the pressures on lines 36 and 37, to show the considerable difference between the back pressure against the piston of the large cylinder and that in the bottom of the air-pump.

Line 13 gives the number of double strokes made by the pistons of the engine per minute, or of revolutions made per minute by the screw. These quantities are the means of the mean number of double strokes made by the pistons per minute during each run of an experiment obtained from the total number of double strokes given by the counter, divided by the time in minutes of making the run.

Line 14 contains the number of pounds of feed-water pumped into the boiler per hour for each experiment. These quantities are obtained from the number of gallons of water measured out of the receiving tub during the time the vessel for each experiment was running over the base (excluding the turnings) in alternate directions. This number of gallons being corrected for the temperature of the feed-water (line 19) and divided by the duration (exclusive of the turnings) of the experiment in hours, gives the quantities on line 14.

Line 15 contains the number of Fahrenheit units of heat imparted to the feed-water per hour. These quantities are the products of those on line 14 multiplied by the difference between the Fahrenheit units of heat contained in a pound of feed-water of the temperature on line 19 and in a pound weight of steam of the boiler pressure on line 4.

Temperatures.—Line 16 contains the temperature in degrees Fahrenheit of the superheated steam from the boiler, taken by a mercurial thermometer, whose bulb and part of whose stem were immersed in the steam-pipe at a point 18 inches from the valve-chest of the small cylinder. The thermometer, by means of an attached nut, was screwed into the steam-pipe, so that the entire bulb and lower portion of the stem were in direct contact with the steam.

Line 17 contains the temperature which steam of the boiler pressure would have had had it been saturated instead of superheated. This temperature is according to Regnault's experiments.

Line 18 gives the number of degrees Fahrenheit that the steam from the boiler was superheated. It is the remainder of the subtraction of the temperature on line 17 from that on line 16.

Lines 19, 20, and 21 contain, respectively, the temperatures in degrees Fahrenheit, taken by ordinary mercurial thermometers, of the feed-water in the receiving tub, of the air on the vessel's deck, and of the water in the bay. These temperatures, as well as the temperature on line 16, are the means of observations taken every ten minutes.

Speed.—The vessel's speed is expressed on line 22 in statute miles of 5,280 feet per hour; and on line 23 in geographical miles of 6,086 feet per hour. The first is employed by yachtmen, landsmen, and for steamers on inland waters; the last is employed by sea-faring persons.

Line 24 contains the slip of the screw in per centum of its speed; the latter being computed from the product of the pitch of the screw into the number of its revolutions made per hour. The speed of the vessel per hour being deducted from the speed of the screw per hour (both in the same terms), the remainder expressed in per centum of the latter is the quantity on line 24.

Steam pressures in the small cylinder, per indicator.—The quantities on lines 25 to 32, both inclusive, are the means from all the indicator diagrams taken from the small cylinder. These diagrams were taken every ten minutes. Lines 25, 26, and 27 contain, respectively, the pressure on the piston of the small cylinder at the commencement of the stroke, at the point of cutting off the steam, and at the end of the stroke, in pounds per square inch above zero, or the line of no pressure, as given by the barometer (line 10). Line 28 contains the mean back pressure against the small piston during its stroke, and line 29 contains the back pressure against the small piston at the beginning of the stroke, both quantities being in pounds per square inch above zero. The mean back pressure is employed to obtain the total pressure (line 32), by adding it to the indicated pressure (line 30). The back pressure at the commencement of the stroke of the piston is required for calculating the weight of steam present per hour in the small cylinder at the point of cutting off the steam (line 59), and at the end of the stroke of its piston (line 60). Line 30 contains the indicated pressure and represents the mean ordinate of the indicator diagrams in pounds per square inch of the area of the piston of the small cylinder. Line 31 contains the net pressure on the piston of the small cylinder in pounds per square inch, and is what remains of the quantities on line 30 after the subtraction of 2 pounds per square inch, as the pressure required to work the small cylinder, *per se*, or unloaded. Line 32 contains the total pressure on the piston of the small cylinder in pounds per square inch above zero; these quantities are the sum of those on lines 30 and 28.

Steam pressures in the large cylinder, per indicator.—The quantities on lines 33 to 40, both inclusive, are the means from all the indicator diagrams taken from the large cylinder. These diagrams were taken every ten minutes. Lines 33, 34, and 35 contain respectively the pressure on the piston of the large cylinder at the commencement of the stroke, at the point of cutting off the steam, and at the end of the stroke in pounds per square inch above the zero of pressure. Line 36 contains the mean back pressure against the large piston during its stroke, and line 37 contains the back pressure against the large piston at the beginning of the stroke. The mean back pressure is employed to obtain the total pressure (line 40) on the large piston by adding to this back pressure the indicated pressure on the large piston (line 38). The back pressure at the commencement of the stroke of the piston is required in the calculation of the weight of steam present per hour in the large cylinder at the end of the stroke of its piston (line 63). Line 38 contains the indicated pressure and represents the mean ordinate of the indicator diagrams in pounds per square inch of the area of the piston of the large cylinder. Line 39 contains the net pressure on the piston of the large cylinder in pounds per square inch and is what remains of the quantities on line 38 after the subtraction of 2 pounds per square inch as the pressure required to work the large cylinder *per se*, or unloaded. Line 40 contains the total pressure on the piston of the large cylinder in pounds per square inch above zero; these quantities are the sum of those on lines 38 and 36.

Horses-power.—Lines 41 to 52, both inclusive, contain the various kinds of horses-power developed by the engine, which it is necessary to know in order to understand the manner in which the total power of the engine is distributed between its cylinders and for what purposes.

Lines 41 and 42 contain, respectively, the indicated horses-power developed in the small and in the large cylinders. These powers are calculated from the speed of the pistons, from the areas of the pistons, and from the pressures on lines 30 and 38. On line 43 is the indicated horses-power developed by the engine, being the sum of the quantities on lines 41 and 42.

Lines 44 and 45 contain, respectively, the net horses-power developed in the small and in the large cylinders. These powers are calculated from the speed of the pistons, from the areas of the pistons, and from the pressures on lines 31 and 39. On line 46 is the net horses-power developed by the engine, being the sum of the quantities on lines 44 and 45.

Line 47 contains the total horses-power developed in the small cylinder, calculated from the speed of the piston, the area of the piston, and the pressure on line 32. This is the power overcoming all resistances to the small piston down to the zero of pressure, and includes internal and external work of all kinds.

Line 48 contains the total horses-power developed in the large cylinder, calculated from the speed of the piston, the area of the annular superficies remaining after the subtraction of the area of the small piston from that of the large piston, and the pressure on line 40. In explanation of this it is necessary to remark that in the compound engine the back pressure overcome by the piston of the small cylinder is more or less utilized upon or transferred to the piston of the large cylinder, where, for a superficies equal to the area of the piston of the small cylinder, it develops a pressure equal to the sum of the indicated and mean back pressures per square inch upon the piston of the large cylinder. Thus a portion of the indicated horses-power developed *in* the large cylinder is really developed *by* the piston of the small cylinder, this portion being what corresponds to the area of the small piston. The only portion of the indicated horses-power developed *in* the large cylinder that is developed *by* its piston is what corresponds to the annular superficies remaining after deduction of the area of the small piston from that of the large one. The sum of the indicated pressure and of the back pressure upon the piston of the large cylinder (line 40), would exactly equal the back pressure against the piston of the small cylinder (line 28) were the steam-ports, steam-passages, &c., between the two pistons sufficiently large to pass the steam without resistance from the small cylinder to the large one. The want of equality found in actual practice between these pressures for the two cylinders—the difference varying from much to little according to the circumstances mentioned—is due mainly to the smallness of the cross area of the steam-passage of the small cylinder taken in connection with the high pressure of the steam in that cylinder at the end of the stroke of its piston, and the velocity of the piston. Hence the total horses-power developed by the piston of the large cylinder is what is due to the annular superficies remaining after the subtraction of the area of the small piston from that of the large one and to the total pressure upon the superficies composed of the sum of the indicated and back pressure (line 40) upon the piston of the large cylinder per square inch.

Line 49 contains the total horses-power developed by the engine. These quantities are the sum of those on lines 47 and 48, and represent the entire dynamic effect produced by the steam, both useful and prejudicial.

Line 50 contains the total horses-power developed in the small cylinder by the expanding steam alone; that is to say, the total horses-power developed in the small cylinder by the steam after the closing of the cut-off valve. This power corresponds to the portion of the indicator diagram formed by the expansion curve. It is necessary to know the power on line 50, in order to calculate the weight of steam (line 61) condensed in the small cylinder to furnish the heat transmuted into that power, so that by adding it to the weight of steam due to the pressure at the end of the stroke of the small piston (line 60) there may be obtained the weight of steam accounted for at that point by the indicator (line 62), which weight being deducted from the weight of steam evaporated in the boiler (line 14) gives a difference (line 68) that is mainly due to the condensation of steam by the inner surfaces of the cylinder and steam-passages, under the influence of the varying temperatures, from those due to the pressure on line 25 to those due to the pressure on line 29.

For the power developed in the small cylinder previously to the closing of the cut-off valve, there is no condensation of steam in that cylinder similar to what takes place in it after the closing of the cut-off valve, for furnishing the heat transmuted into the power; because such heat has, previously to the closing of the cut-off valve, been furnished to the steam in the boiler by the fuel. A loss of heat equivalent to the power developed by the steam previously to the closing of the cut-off valve has indeed taken place in the boiler, but instead of being at the expense of the steam in the cylinder, which has to be the case after the closing of the cut-off valve, is at the expense of the fuel in the furnace, and is included in the total heat of vaporization.

Line 51 contains the total horses-power developed in the large cylinder by the expanding steam alone. These quantities are, of course, the same as those on line 48, because all the steam used in the large cylinder is expanded steam. The thermal equivalent of these powers, in pounds of steam condensed to furnish the necessary heat, added to the quantities on line 61, form the quantities on line 64, which added to those on line 63, give the weight of steam accounted for by the indicator at the end of the stroke of the piston of the large cylinder (line 65). This latter weight being deducted from the weight of steam evaporated in the boiler (line 14) gives a difference (line 70) which is mainly due to the condensation of steam in the small cylinder, in the large cylinder, and in the receiver between them, by the inner surfaces of those vessels, including the surfaces of the steam-passages of the two cylinders, when affected by the variations of temperature upon them due to the variations of pressure.

By total power developed by the expanded steam alone in a cylinder, is meant the power due to the mean pressure, estimated from the zero line of pressure, of that portion of the indicator diagram which is made by the expansion curve, and for a stroke of piston equal to the length traversed by the piston from the closing of the cut-off valve to the end of the stroke.

To calculate the number of pounds of steam condensed in a cylinder to furnish the heat transmuted into the total power developed by the expanded steam alone, there are required to be known: The total horses-power developed by the expanded steam alone; the latent heat in Fahrenheit units corresponding to the mean pressure above zero in pounds per square inch of the expanded steam; the mechanical equivalent in pounds of one Fahrenheit unit of heat; and the time in minutes during which the power acted. Calling the power P, the latent heat H, the mechanical equivalent 789¼, and the time T, we have: $\frac{P \times \frac{33000}{789\frac{1}{4}} \times T}{H}$ = the number of pounds of steam condensed in the cylinder to furnish the heat transmuted into the power P exerted during the time T.

Line 52 contains the total horses-power developed by the expanded steam alone in the engine. These quantities are the sum of those on lines 50 and 51. By comparing them with the quantities on line 49 the portions of the total power developed by the engine before and after the closing of the cut-off valve can be obtained.

Economic results.—The quantities on line 53 are the quotients of those on line 14 divided by the quantities on line 43. The quantities on line 54 are the quotients of those on line 14 divided by the quantities on line 46; and the quantities on line 55 are the quotients of those on line 14 divided by those on line 49.

The quantities on line 56 are the quotients of those on line 15 divided by the quantities on line 43. The quantities on line 57 are the quotients of those on line 15 divided by the quantities on line 46. And the quantities on line 58 are the quotients of those on line 15 divided by the quantities on line 49.

The quantities on lines 53 to 58, both inclusive, show the cost of the indicated, the net, and the total horses-power developed by the engine, in pounds of feed-water consumed per hour per horse-power, and in Fahrenheit units of heat consumed per hour per horse-power. Of these two measures, the latter only is strictly correct, because it includes the difference due to the different total heats of the steam of different boiler pressures, and to the different temperatures of feed-water, which the former measure does not. If the total heat of steam was the same for all pressures, and if it were allowable to assume the same temperature of feed-water for all the experiments, then measuring the cost of the power by the weight of feed-water consumed per hour would be exact; and it is sufficiently exact for ordinary practical purposes, with the advantages that it can be very easily computed, and that when divided by the economic vaporization of the boiler it gives the cost of the horse-power in pounds of fuel consumed per hour. For example, if the weight of feed-water consumed per hour per horse-power be 18 pounds, and if each pound of coal burned in the furnace vaporized 8 pounds of water, the horse-power would be obtained for ($\frac{18}{8}=$) $2\frac{1}{4}$ pounds of coal. Thus, when the cost of the horse-power in pounds of feed-water consumed per hour is known, its cost with different kinds of fuel and the different kinds of boilers can be ascertained, the economic vaporizations of these fuels and boilers being supposed to be known.

In the cases of the experiments, the cost of the horse power in pounds of coal consumed per hour could not be given, because they were mostly of too short duration to allow the weight of coal burned to be accurately ascertained; several experiments being sometimes made on one day without interval between them; but as the economic vaporization of the coal is given in Table No. 3 for each day of experimenting, all that is necessary for ascertaining the cost of the horse-power in pounds of coal consumed per hour in any case is to divide the weight of feed-water consumed per hour per horse-power by the economic vaporization of the coal when producing the same weight of steam per hour that was consumed during the experiment in question, and the quotient will be the cost of that horse-power in pounds of coal consumed per hour.

For exact comparisons, however, between the cost of power in different cases the number of units of heat consumed in producing it must be taken as the measure.

Weight of steam accounted for by the indicator.—Lines 59 to 65, both inclusive, contain the weight of steam accounted for per hour by the indicator at various points in the process of the engine, which weight, with saturated steam, would necessarily be less than the weight according to the tank, by the weight condensed in the cylinder due to all causes other than the transmutation of a portion of the heat in the expanding steam into the power produced by its expansion. This weight of steam condensed in the cylinder after the closing of the cut-off valve can be calculated when the power developed by the expanding steam is known, as for every 789¼ foot-pounds of work performed by the expanding steam after the closing of the cut-off valve, one Fahrenheit unit of heat is abstracted from the steam, which abstraction is attended by the liquefaction of such weight of steam as is necessary to furnish this unit of heat, the resulting water of condensation having the temperature normal to the pressure of the steam from which it was precipitated. The weight of steam so condensed, added to the weight ascertained from the pressure at the end of the stroke of the pistons of the engine, gives the weight of steam accounted for by the indicator at those points. For the weight of steam accounted for by the indicator at the point of cutting off the steam, no similar addition is necessary, because there has been up to that point no transmutation of the heat in the unexpanded steam into the power developed. Consequently, the weight of steam calculated from the pressure at the point of cutting off is the whole weight accounted for by the indicator at that point. Of course, in these calculations the steam is supposed to be in the saturated state, which is always the case in the cylinder unless it has been very highly superheated before entering.

Line 59 shows the number of pounds of steam accounted for by the indicator per hour at the point of cutting off the steam in the small cylinder. This quantity was calculated for the number of cubic feet displaced per hour by the piston of the small cylinder up to the point of cutting off given on line 7, plus the number of cubic feet in the spaces in the clearance and steam passage at one end of that cylinder multiplied by the number of strokes made by its piston per hour, and by the weight of a cubic foot of steam of the pressure on line 26; deducting from the product the weight of steam per hour already in the clearance and steam passage when steam was admitted from the boiler to the cylinder, this latter weight being calculated from the weight of a cubic foot of steam of the pressure on line 29.

Line 60 shows the number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston, obtained by multiplying the space displacement of that piston per stroke in cubic feet, plus the number of cubic feet in the clearance and steam passage at one end of the cylinder, by the number of strokes made by the piston per hour, and by the weight of a cubic foot of steam of the pressure on line 27, subtracting from the last product the weight of steam per hour already in the clearance and steam passage when steam was admitted from the boiler to the cylinder, this latter weight being calculated from the weight of a cubic foot of steam of the pressure on line 29.

Line 61 shows the number of pounds of steam condensed per hour in the small cylinder to furnish the heat transmuted into the total horses-power developed in that cylinder by the expanding steam after the closing of the cut-off valve. By total horses-power is meant the power due to the mean pressure of the expanding steam above zero, acting through the portion of the stroke of the piston which remains after the closing of the cut-off valve. This is the power on line 50. The calculation of the quantities on line 61 is made as follows; the mechanical equivalent of one Fahrenheit unit of heat being taken at $789\frac{1}{4}$ foot pounds:

$\frac{33000}{789\frac{1}{4}} = 41.811847$ = the number of Fahrenheit units of heat equivalent to one horse-power, or to 33,000 foot-pounds of work per minute.

P = the number of horses-power, computed for the piston-pressure above zero, developed by the expanding steam after the closing of the cut-off valve.

T = the time in minutes during which the power P operated, which was 60 minutes.

L = the latent heat of the mean pressure above zero of the expanding steam after the closing of the cut-off valve.

Then, $\frac{41.811847 \times P \times 60}{L}$ = the pounds of steam condensed in the cylinder to furnish the heat transmuted into the total horses-power developed by the expanding steam.

Line 62 contains the sum of the quantities on lines 60 and 61. These sums are the weight of steam accounted for by the indicator at the end of the stroke of the piston of the small cylinder.

Line 63 shows the number of pounds of steam present per hour in the large cylinder at the end of the stroke of its piston, obtained by multiplying the space displacement per stroke of that piston in cubic feet, plus the number of cubic feet in the clearance and steam passage at one end of the cylinder, by the number of strokes made by the piston per hour and by the weight of a cubic foot of steam of the pressure on line 35, subtracting from the last product the weight of steam per hour already in the clearance and steam passage when steam was admitted from the boiler to the cylinder, this latter weight being calculated from the weight of a cubic foot of steam of the pressure on line 37.

Line 64 shows the number of pounds of steam condensed per hour in the small and in the large cylinder and in the receiver between them to furnish the heat transmuted into the total horses-power developed in the small and large cylinders by the expanding steam after the closing of the cut-off valve of the small cylinder. After the closing of that valve the steam is used expansively during the remainder of the stroke of the piston of the small cylinder and throughout the entire stroke of the piston of the large cylinder, notwithstanding that the latter cylinder is provided with a cut-off valve.

The number of pounds of steam condensed in the large cylinder to furnish the heat transmuted into the total horses-power on line 51, which are the total horses-power developed by the expanding steam in that cylinder, are calculated in precisely the same manner, with substitution of the proper corresponding power and pressure, as in the case already described of the small cylinder. The pressure from which the power on line 51 is calculated is that on line 40, and it acts during the entire stroke of the piston. The quantities thus obtained having been added to those on line 61, the sums are the quantities on line 64.

Line 65 contains the sum of the quantities on lines 63 and 64. These sums are the weight of steam accounted for by the indicator at the end of the stroke of the piston of the large cylinder.

Difference between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator.—The quantities on lines 66 to 71, both inclusive, show the difference for each experiment between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator. Were there no cylinder condensations due to other causes than the production of the power, these differences would not exist; the weight of water and the weight of steam would be equal. Among these other causes is the action of the metal of the cylinder, which alternately takes up and gives out heat, the former quantity being greater than the latter, as shown by the indicator, but equal in fact when the heat is included which vaporizes the water of condensation in the cylinder during the exhaust stroke of the piston. This heat escapes detection by the indicator because it is contained in the steam that is evaporated from the water of condensation present in the cylinder at the end of the stroke of its piston, by its contained heat and the heat in the metal of the cylinder, and passes to the condenser during the exhaust stroke of the piston; the evaporation being made possible during the exhaust stroke of the piston by the less pressure in the condenser than in the cylinder, the interiors of these two vessels being in common as long as the exhaust passage is open.

There is also a condensation of steam in the cylinder due to the heat transmuted into the work of expulsion of the exhaust steam by its own pressure at the end of the stroke of the piston. This heat reappears in the condenser when the inrushing steam comes to the state of rest; the *vis viva* communicated to this steam in the cylinder at the expense of heat, reproducing the same quantity of heat when extinguished in the condenser. But, as regards the cylinder, such heat is lost and escapes detection by the indicator.

Likewise, there is a condensation of steam in the cylinder when the steam is used expansively, owing to the expansion *per se*, and is due wholly to transmutation of heat into interior work on the steam molecules; this transmutation being independent of any mechanical work done on the piston. The heat thus disappearing is not accounted for by the indicator.

Line 66 contains the difference between the quantities on lines 14 and 59; and line 67 shows this difference in per centum of the quantities on line 14.

Line 68 contains the difference between the quantities on lines 14 and 62; and line 69 shows this difference in per centum of the quantities on line 14.

Line 70 contains the difference between the quantities on lines 14 and 65; and line 71 shows this difference in per centum of the quantities on line 14.

Ratios.—As the total horses-power developed by the engine are the entire dynamic effect produced by the steam, of which the indicated and the net horses-power are only fractions, it is frequently of interest to know what these fractions are, for the larger they are, other things equal, the more economically does the engine use its steam. The indicated horses-power relatively to the total horses-power show how properly the organs of the engine function; that is to say, how efficiently the condenser, the air-pump, and the exhaust conduits from the valve-seat of the cylinder to the condenser act. The net horses-power relatively to the total horses-power include, in addition to showing the efficiency of the organs of the engine, the more or less resistance of the moving parts of the engine; that is to say, the more or less friction of the engine *per se*, or unloaded.

The quantities on line 72 are the per centum which the indicated horses-power developed by the engine, line 43, are of the total horses-power, line 49.

The quantities on line 73 are the per centum which the net horses-power developed by the engine, line 46, are of the total horses-power, line 49.

Cylinder pressures reduced to large cylinder alone.—Lines 74 to 77, both inclusive, give single expressions for the pressures on the pistons of the two cylinders, by reducing them to their equivalents on the supposition that the engine consisted of only the large cylinder. As the engine consists of two compounded cylinders of unequal space displacements of piston per stroke, operated by unequal pressures per square inch of pistons, and as the ratios of these pressures are unequal in different experiments, it is necessary to obtain a single expression for the piston pressures, for which purpose the indicated, net, total, and back pressures on the pistons of the two cylinders have been equated to what they would be if applied to the piston of the large cylinder alone. The desired equivalents have been obtained by reducing the experimental indicated net and total pressures per square inch on the piston of the small cylinder, in the ratio of the areas of the pistons of both cylinders, and adding the resulting quantities to the corresponding experimental pressures per square inch on the piston of the large cylinder. The sums of these additions are contained on lines 74 to 77.

The quantities on line 74 are the quotients of those on line 30 divided by 3.196293 (the ratio of the areas of the pistons of the two cylinders) added to the quantities on line 38.

The quantities on line 75 are the quotients of those on line 31 divided by 3.196293 added to the quantities on line 39.

The quantities on line 76 are the quotients of those on line 32 divided by 3.196293 added to the quantities on line 40 reduced in the ratio of

3.196293 to 2.196293. The last reduction is required because the total pressures on the piston of the small cylinder, line 32, are counted from zero, and consequently cover an area of the piston of the large cylinder equal to the area of the piston of the small cylinder, so that the addition must be made to the quantities on line 40 in the proportion of the area of the piston of the large cylinder to the remainder of that area after deducting the area of the piston of the small cylinder.

The quantities on line 77 are the equivalent back pressures against the pistons in pounds per square inch of both cylinders for the large cylinder alone. They are the remainders of the quantities on line 76 after deduction of those on line 74. Abstractly, the back pressure in the compound engine should be only that against the piston of the large cylinder, because the total pressure on that piston per square inch above zero should be equal to the back pressure per square inch above zero against the piston of the small cylinder, and would be equal to it, supposing no cut-off valve for the large cylinder, if the steam-passage connecting the exhaust end of the small cylinder and the steam end of the large cylinder were of so great an area and so short a length that no pressure would be required to force the steam from the one cylinder into the other. But, owing to the fact that the steam-passage in question has not these dimensions nor anything approaching to them, the back pressure in pounds per square inch above zero against the piston of the small cylinder will always be greater than the total pressure per square inch above zero on the piston of the large cylinder. Hence, the back pressure in a compound engine will always be greater than in a simple engine whose cylinder is a duplicate of the large cylinder of the compound engine, and with equal vacuum in the condenser. In the simple engine there is a similar difference between the condenser pressure and the back pressure against the piston, owing to the same cause, namely, the restrictions offered by the too small area and too great length with its sinuosities of the exhaust openings, pipes, &c., the back pressure against the piston more and more exceeding the pressure in the condenser, as these areas are smaller, these lengths greater, and these sinuosities more marked and frequent.

The difference between the quantities on lines 74 and 75 shows the pressure per square inch on the piston of the large cylinder required to work the engine *per se*, or unloaded; that is to say, to overcome the friction of the packings and of the weights of the moving parts attached to each cylinder.

EXPLANATION OF TABLE NO. 2.

Table No. 2 contains the distribution of the pressures on the piston, and of the indicated horses-power developed by the engine during the experiments. The piston pressures here referred to are those on lines 74 and 75 in Table No. 1, continued, being what would have been the indicated and net pressures on the piston of the large cylinder alone had

...ters at the head of the columns in

O	P
... a. m., Nov. 16.	3.25 p. m., Nov. 16.
11. 856580	9. 280656
23. 9908	20. 2958
1551. 4726	814. 0220
191. 57236	147. 63465
28. 400	15. 789
2. 000	2. 000
26. 400	13. 789
1. 980	1. 034
3. 066	1. 821
5. 123	2. 219
16. 231	8. 715
98. 93359	42. 38728
6. 96716	5. 36922
91. 96643	37. 01806
6. 89748	2. 77635
10. 68059	4. 88840
17. 84636	5. 95748
56. 54200	23. 39577
7. 5000	7. 5000
11. 614	13. 206
19. 405	16. 093
61. 481	63. 201
47. 101	42. 778

the experimental indicated and net pressures on the piston of the small cylinder been reduced in the ratio of the areas of the pistons of the two cylinders and added to the experimental indicated and net pressures on the piston of the large cylinder.

For facility of reference the quantities have been grouped and the lines containing them numbered.

Before explaining the manner in which these quantities were obtained, it is proper to state generally that this distribution of the indicated pressure and of the indicated horses-power is necessary in order to understand what portions of each are expended in the different operations connected with the propulsion of a vessel by a screw.

Of the indicated horses-power a portion is expended in working the engine *per se*, or the unloaded engine; and this portion must be deducted first, because until the friction of the packings and of the moving parts of the engine, including the screw-shafting, is overcome, no power can be applied to the screw, or externally of the engine, for it is obvious that until the friction of the engine itself is counterbalanced the piston cannot move.

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 external work. A friction attends the net power, additional to that of the unloaded engine and proportional to the net power, let the latter be what it may; and the power required to overcome this friction is the horses-power absorbed by the friction of the load, the articulations of the engine moving under the net pressure producing greater friction on them than is due to the mere weight of the moving parts and to the pressure of the packings.

Then, there are the horses-power expended in overcoming the surface or skin resistance of the screw-blades experienced from the water in which they move. This power can be calculated independently when the data are known.

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 receding 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.

On these general principles the corresponding quantities in Table No. 2 have been calculated.

Line 1 contains the dates of the experiments.

Line 2 gives the speeds of the vessel per hour in geographical miles of 6,086 feet.

Line 3 gives the slips of the screw in per centum of its axial speed (product of pitch and revolutions).

On line 4 are the thrusts of the screw in pounds during each experiment. These quantities are what would have been given by a dynamometer had one been applied directly to the screw-shaft and fulcrumed on the vessel. They are calculated from the quantities on line 19 by multiplying the latter by 33,000 and dividing the product by the speed of the vessel in feet per minute.

Line 5 contains the number of double strokes made per minute by the pistons of the engine, and of revolutions made per minute by the screw.

The quantities on lines 1, 2, 3, and 5 are taken from Table No. 1 and added for the sake of completeness to Table No. 2.

Distribution of the indicated horses-power.—Lines 13 to 19, both inclusive, show the distribution of the indicated horses-power among the operations contingent on the propulsion of the vessel during the different experiments.

Line 13 contains the indicated horses-power, the quantities being taken from Table No. 1 continued.

Line 15 contains the net horses-power applied to the crank-pin, also taken from Table No. 1 continued.

Line 14 contains the horses-power expended in working the engine *per se*, or unloaded; the quantities on this line being the difference between those on lines 13 and 15; they can, however, be obtained independently by calculation, the data being the speed of the piston per minute in feet obtained from line 5, the area in square inches of the piston of the large cylinder alone, and the pressure on line 7.

Line 16 contains the horses-power absorbed by the friction of the load; these quantities are obtained by multiplying those on line 15 by 0.075, that fraction being the coefficient of the friction of the load.

Line 17 gives the horses-power expended in overcoming the resistance of the water to the surface of the screw-blades. These quantities have been calculated on the assumption that one square foot of the helicoidal surface of the screw, moving in its helical path with a speed of 10 feet per second, has a resistance of 0.45 pound, which resistance increases or decreases in the ratio of the square of the helical speeds. This 0.45 pound also covers the direct resistance of the forward or cutting edges of the blades, which varies as the square of the helical speeds.

Line 18 contains the horses-power expended in the slips of the screw, and the quantities are calculated by multiplying the remainders of the quantities on line 15, after subtraction of the sum of the quantities on lines 16 and 17, by the quantities on line 3 expressed in fractions of the axial speed of the screw considered as unity.

Line 19 gives the horses-power expended in the propulsion of the vessel, which power is all that is utilized of the entire total power developed by the engine. The quantities are the remainders of those on line 15 after subtraction of the sum of the quantities on lines 16, 17, and 18.

Lines 20, 21, 22, and 23 show respectively the per centum which the quantities on lines 16, 17, 18, and 19 are of those on line 15. If the net horses-power applied to the crank-pin be taken at 100, the quantities on lines 20, 21, 22, and 23 show respectively the number of horses-power expended in overcoming the friction of the load, and the resistance of the screw surface to the water, and expended in the slip of the screw and in the propulsion of the vessel.

Line 24 shows the per centum which the quantities on line 19 are of those on line 49 of Table No. 1 continued. Line 49 contains the total horses-power developed by the engine, the mean piston pressure being reckoned from zero. The quantities on line 19 being the horses-power applied to the propulsion of the vessel, a comparison of the two shows how much of the entire dynamic effect of the steam—or of the fuel—has been utilized during each experiment.

Distribution of the indicated pressure on the piston.—Lines 6 to 12, both inclusive, show the distribution of the indicated pressure during the different experiments. A single expression for this pressure being necessary, and the engine having two compounded cylinders—a small and a large one—the desired single expression was obtained by reducing the experimental indicated pressure per square inch on the piston of the small cylinder in the ratio of the areas of the pistons of both cylinders, and adding the quantity thus obtained to the experimental indicated pressure per square inch on the piston of the large cylinder. These are the quantities on line 6, and are taken from line 74 of Table No. 1 continued.

Line 7 contains the pressure per square inch of the large piston alone, required to work the engine *per se.* With both cylinders in use, this pressure was taken at 2 pounds per square inch of the pistons, and the equivalent, if applied to the piston of the large cylinder alone, was obtained by dividing 2 by the ratio of the areas of the two pistons and adding the quotient to 2, which produced the constant 2.626 pounds per square inch for the piston of the large cylinder alone, on line 7.

Line 8 gives the net pressures applied to the crank-pin in pounds per square inch of the piston of the large cylinder alone; these quantities are the remainders of those on line 6 after subtraction of those on line 7.

Line 9 contains the pressures per square inch of the piston of the large cylinder alone, absorbed by the friction of the loads; these quantities are the products of those on line 8 by the fraction 0.075, which is the coefficient for the friction.

Line 10 contains the pressures per square inch of the piston of the large cylinder alone, expended in overcoming the resistance of the water to the surface of the screw-blades. These quantities are the same proportion of those on line 8 that the quantities on line 17 are of those on line 15.

Line 11 contains the pressures per square inch of the piston of the large cylinder alone, expended in the slip of the screw. These quantities are obtained by multiplying the remainders of those on line 8 after subtraction of the sum of the quantities on lines 9 and 10, by the slip of the screw on line 3 expressed in fractions of its axial speed as unity.

Line 12 contains the pressures per square inch of the piston of the large cylinder alone, expended in the propulsion of the vessel. These quantities are the remainders of those on line 8 after subtraction of the sum of the quantities on lines 9, 10, and 11.

Table No. 3, containing the data and results of the experiments made in the month of November, 1880, by a board of United States naval engineers with the coil boiler of the Herreshoff steam-yacht Leila to ascertain its potential and economic vaporization of water by anthracite consumed with natural draught during the experiments made by the same board on the same vessel in Narragansett Bay, whose data and results are contained in Tables No. 1 and No. 1 continued.

Date of the experiments	November 13.	November 15.	November 16.	November 11.	November 17.	November 18.	November 12.
TOTAL QUANTITIES.							
Duration of the experiments, in consecutive hours	10.600	8.150	8.800	10.700	9.050	8.800	12.400
Total number of double strokes made by the steam-pistons	136855.	91886.	90869.	113000.	89411.	66091.	102545.
Total number of pounds of anthracite consumed	3462.5	2414.0	2502.	2223.50	1778.5	1070.0	1476.5
Total number of pounds of refuse from the anthracite in ash, clinker, and dust	522.5	432.5	409.	455.25	370.5	214.5	321.0
Total number of pounds of combustible, or gasifiable portion of the anthracite, consumed	2940.0	1981.5	2093.	1768.25	1408.0	855.5	1155.5
Per centum of the anthracite in refuse of ash, clinker, and dust	15.0903	17.9163	16.3469	20.4745	20.8322	20.0467	21.7406
Total number of pounds of pine wood consumed	128.	175.	168.5	160.	177.5	175.	175.
Total number of pounds of feed-water pumped into the boiler	24723.6125	17489.0592	18566.5013	17805.2519	13396.0018	8175.9461	11340.8879
Total number of pounds of water circulating through the boiler additional to the feed-water	23504.1783	16234.6907	16282.1812	19617.9114	15710.6593	11739.9387	18107.2573
Number of times the weight of circulating water exceeded the weight of feed-water	0.9507	0.9283	0.8770	1.1018	1.1728	1.4359	1.5966
Total number of pounds of anthracite consumed, including its equivalent of pine wood, estimating one pound of the latter as equal to 0.4 pound of anthracite combustible	3522.7993	2499.2788	2582.5709	2304.0000	1868.1829	1157.5512	1565.9461
Total number of pounds of combustible consumed, including its equivalent of pine wood, estimating one pound of the latter as equal to 0.4 pound of anthracite combustible	2991.2000	2051.5000	2160.4000	1832.2500	1479.0000	925.5000	1225.5000
PRESSURES.							
Steam pressure in boiler in pounds per square inch above the atmosphere	124.55	62.91	38.20	98.00	72.24	51.75	62.61
Steam pressure in boiler in pounds per square inch above zero	139.40	77.58	53.02	112.72	87.14	66.48	77.33
Height of the barometer in inches of mercury	30.25	29.886	30.19	30.00	30.36	30.014	30.01
TEMPERATURES.							
Temperature, in degrees Fahrenheit, of the superheated steam in the steam-pipe	414.0	360.00	331.0	372.0	355.0	327.00	338.00
Temperature, in degrees Fahrenheit, of the steam in the steam-pipe, considered as saturated	352.5	308.44	284.6	336.4	317.6	299.33	309.52
Number of degrees Fahrenheit the steam was superheated	61.5	51.56	46.4	35.6	37.4	27.67	28.48
Temperature, in degrees Fahrenheit, of the feed-water	93.0	88.5	89.0	76.0	78.	67.	66.
Temperature, in degrees Fahrenheit, of the atmosphere	48.	38.	38.	55.	43.	50.	50.

Table No. 3, containing the data and results of the experiments made in the month of November, &c.—Continued.

Date of the experiments	November 13.	November 15.	November 16.	November 11.	November 17.	November 18.	November 12.
RATE OF COMBUSTION.							
Number of pounds of anthracite consumed per hour	332.3395	306 6600	293.4740	215.3271	206.4290	131.5399	126.2860
Number of pounds of combustible consumed per hour	282.1887	251.7178	245.5000	171.2383	163.4254	105 1705	98.8307
Number of pounds of anthracite consumed per hour per square foot of grate surface	12.7894	11.8095	11.3017	8.2923	7.9496	5.0656	4.8633
Number of pounds of combustible consumed per hour per square foot of grate surface	10.8671	9.6937	9.4542	6.5944	6.2935	4.0501	3.8060
Number of pounds of anthracite consumed per hour per square foot of heating surface, calculated for the exterior circumference of the coil pipe	0.6852	0.6323	0.6051	0.4440	0.4256	0.2712	0.2604
Number of pounds of anthracite consumed per hour per square foot of heating surface, calculated for the interior circumference of the coil pipe	0.8075	0.7451	0.7131	0.5232	0.5016	0.3192	0.3069
Number of pounds of combustible consumed per hour per square foot of heating surface, calculated for the exterior circumference of the coil pipe	0.5818	0.5190	0.5062	0.3531	0.3369	0.2168	0.2038
Number of pounds of combustible consumed per hour per square foot of heating surface, calculated for the interior circumference of the coil pipe	0.6857	0.6116	0.5965	0.4161	0.3971	0.2555	0.2401
VAPORIZATION.							
Total number of pounds of water that would have been vaporized in the boiler had the feed-water been supplied at the temperature of 100 degrees Fahrenheit and vaporized under the atmospheric pressure of 29.92 inches of mercury	25866.8413	18159.3319	18965.3686	18828.5858	14070.6902	8628.4417	12011.7452
Total number of pounds of water that would have been vaporized in the boiler had the feed-water been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the atmospheric pressure of 29.92 inches of mercury	28889.1421	20281.0816	21181.2962	21028.5317	15714.7200	9636.5952	13415.2064
Number of pounds of water vaporized from 100 degrees Fahrenheit by one pound of anthracite	7.3427	7.2658	7.3436	8.1721	7.5318	7.4540	7.6760
Number of pounds of water vaporized from 100 degrees Fahrenheit by one pound of combustible	8.6477	8.8517	8.7786	10.2762	9.5137	9.3230	9.8015
Number of pounds of water vaporized from 212 degrees Fahrenheit by one pound of anthracite	8.2006	8.1148	8.2016	9.1270	8.4118	8.3250	8.5668
Number of pounds of water vaporized from 212 degrees Fahrenheit by one pound of combustible	9.6580	9.8860	9.8043	11.4769	10.6252	10.4123	10.9467

EXPLANATION OF TABLE NO. 3.

Table No. 3 contains the data and results of the experiments to ascertain the potential and economic vaporizations of water in the boiler of the Leila, by anthracite consumed with natural draught. The experiments continued seven days, and the data and results referred to are given separately for each day from the lighting of the fires to their complete extinction by burning out, during which time there was no stoppage of the machinery.

The rate of combustion varied for each experiment, and for most of the days there was more than one experiment, and sometimes several. The rate given in Table No. 3 is the mean for the whole day, and often varied very largely from the extremes. Frequently, also, in the experiments made with the slow rates of combustion, the furnace-door was carried more or less open, whereby the economic vaporization was decreased by the admission of air in mass, so that the boiler with less than 8 pounds of anthracite consumed per hour per square foot of grate, gave a less economic result than it would have done could the furnace-door have been kept closed.

The quantity of circulating water, or of superfluous feed-water, driven through the boiler, by the circulating-pump is given, as the economic vaporization of the boiler was influenced by it; the greater the quantity of this water, other things equal, the better was the vaporization. In none of the experiments was the quantity of circulating water sufficient to produce the maximum economy, and its deficiency was owing to the too small dimensions of the circulating-pump, the whole capacity of which was used in all cases. In the experiments with the highest rates of combustion the injurious effect of this deficiency of circulating water was the greatest, as in them the deficiency of circulating water relatively to the feed-water was the greatest.

As the fire in the boiler was always lighted with wood, the heat from whose combustion entered into the vaporization of the water, it was necessary to include its thermal equivalent of anthracite in the quantity of that coal consumed. The pound of dry pine wood burned in a boiler is known to give about four-tenths the vaporization of a pound of the combustible portion of anthracite; that is, of a pound of what remains of anthracite after deduction of its refuse in ash, clinker, soot, dust, &c. Consequently, in all cases, four-tenths of the total weight of the wood has been added to the total weight of the combustible or gasifiable portion consumed of the anthracite. The per centum of refuse in the anthracite having been obtained from the anthracite exclusive of the wood, was applied to the combustible inclusive of the wood, and the new weight of anthracite thus obtained is given as the total weight consumed. By this method, the true per centum of refuse is maintained in the new weight of anthracite.

The amount of superheating possessed by the steam during the experiments is given. It varied for each, becoming less and less as the rate of combustion became less and less, but as it was many degrees at the lowest, none of the water could have foamed over or been entrained by the steam, which was delivered to the engine perfectly dry, so that the apparent was also the true vaporization.

The temperature of the feed-water was not the same in all the experiments; it became less and less as the rate of combustion became less and less, and it has an influence on the vaporization beyond what is due to the less number of units of heat required from the fuel to vaporize a given weight of water. The higher the temperature of a given weight of feed-water per hour on entering the boiler, the more efficient becomes the vaporizing surface of that boiler, owing to the less quantity of heat per hour having to pass through it. In other words, the increased temperature of the feed-water allows a slower rate of combustion of the fuel, with its attendant economic advantages.

The weight of water vaporized per hour per square foot of heating surface is given for both the exterior and the interior surface of the coil pipe. Neither of these is the true expression, but of the two the weight of water vaporized per hour per square foot of the interior surface is much the nearest to the truth. Although the interior surface is in all cases the least gateway the heat has to pass through, and may therefore be supposed to be the proper measure, yet the difference of area of the corresponding surfaces between the outside and inside of the coil pipe, or tube, influences the difference of temperature of these sides, and therefore influences the rapidity with which the heat is transmitted from the fire side to the water side. When the fire is on the outside and the water is on the inside of the tube, measuring the efficiency of the heating surface by the interior surface of the tube makes such surface appear less efficient than in the case of the fire being on the inside and the water on the outside of the tube.

The quantity of water experimentally vaporized is, for all the experiments, reduced to equality of feed temperature and of steam pressure for the purposes of accurate comparison; that is to say, from the experimental weight of water vaporized from the experimental feed temperature and under the experimental steam pressure, there is calculated what weight would have been vaporized had the temperature of feed-water been 100 degrees and 212 degrees Fahrenheit, and had the steam pressure been that of one atmosphere above zero. In these calculations allowance has been made for the different total heats of steam of different pressures, and for the different specific heats of water of different temperatures.

The quantities in the table have been grouped for convenience of examination, and they are so fully described on their respective lines that no further explanation is needed.

Table No. 4, containing the data and results of an experiment made on the 3d of August, 1880, by a board af United States naval engineers, with the coil boiler of the Herreshoff steam-yacht Leila, to ascertain its potential and economic vaporization of water by anthracite consumed with natural draught during a trial of the vessel in free route in Narragansett Bay.

Date of the experiment.	August 3, 1880.
TOTAL QUANTITIES.	
Duration of the experiment in consecutive hours	13.5500
Total number of double strokes made by the steam-pistons	142256.
Total number of pounds of anthracite consumed	2760.
Total number of pounds of refuse from the anthracite, in ash, clinker, and dust	503.
Total number of pounds of combustible, or gasifiable portion of the anthracite, consumed	2257.
Per centum of the anthracite in refuse of ash, clinker, and dust	18.2246
Total number of pounds of pine wood consumed	175.
Total number of pounds of feed-water pumped into the boiler	22524.
Total number of pounds of water circulating through the boiler additional to the feed-water	24635.
Number of times the weight of circulating water exceeded the weight of feed-water	1.0938
Total number of pounds of anthracite consumed, including its equivalent of pine wood, estimating one pound of the latter as equal to 0.4 pound of anthracite combustible	2845.6004
Total number of pounds of combustible consumed, including its equivalent of pine wood, estimating one pound of the latter as equal to 0.4 pound of anthracite combustible	2327.0000
PRESSURES.	
Steam pressure in boiler in pounds per square inch above the atmosphere	104.
Steam pressure in boiler in pounds per square inch above zero	118.829
Height of the barometer in inches of mercury	30.21
TEMPERATURES.	
Temperature, in degrees Fahrenheit, of the superheated steam in the steam-pipe	411.
Temperature, in degrees Fahrenheit, of the steam in the steam-pipe considered as saturated	340.32
Number of degrees Fahrenheit the steam was superheated	70.68
Temperature, in degrees Fahrenheit, of the feed-water	101.
Temperature, in degrees Fahrenheit, of the atmosphere	76.
RATE OF COMBUSTION.	
Number of pounds of anthracite consumed per hour	210.0074
Number of pounds of combustible consumed per hour	171.7343
Number of pounds of anthracite consumed per hour per square foot of grate surface	8.0874
Number of pounds of combustible consumed per hour per square foot of grate surface	6.6135
Number of pounds of anthracite consumed per hour per square foot of heating surface, calculated for the exterior circumference of the coil pipe	0.4330
Number of pounds of anthracite consumed per hour per square foot of heating surface calculated for the interior circumference of the coil pipe	0.5103
Number of pounds of combustible consumed per hour per square foot of heating surface calculated for the exterior circumference of the coil pipe	0.3541
Number of pounds of combustible consumed per hour per square foot of heating surface, calculated for the interior circumference of the coil pipe	0.4173
VAPORIZATION.	
Total number of pounds of water that would have been vaporized in the boiler had the feed-water been supplied at the temperature of 100 degrees Fahrenheit and vaporized under the atmospheric pressure of 29.92 inches of mercury	23320.4206
Total number of pounds of water that would have been vaporized in the boiler had the feed-water been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the atmospheric pressure of 29.92 inches of mercury	26045.1958
Number of pounds of water vaporized from 100 degrees Fahrenheit by one pound of anthracite	8.1953
Number of pounds of water vaporized from 100 degrees Fahrenheit by one pound of combustible	10.0217
Number of pounds of water vaporized from 212 degrees Fahrenheit by one pound of anthracite	9.1528
Number of pounds of water vaporized from 212 degrees Fahrenheit by one pound of combustible	11.1926

TABLE NO. 4.

In Table No. 4 will be found the data and results of an experiment made by the senior two members of the present board of chief engineers, to ascertain the potential and economic vaporization of the boiler of the

Leila with anthracite consumed by natural draught, the vessel being in free route in Narragansett Bay over the base employed for the present experiments. This experiment was made a few months previously to those which are the subject of this report, and in precisely the same manner. Although, owing to an accidental leakage of the steam-pistons, the economy of the power could not be determined, yet the data were accurate for ascertaining the potential and economic vaporization of the boiler. Accordingly, the data and results of this experiment, as regards the boiler alone, have been placed in Table No. 4, the same arrangement of the quantities being followed, and the calculations being made in precisely the same manner as in Table No. 3.

DISCUSSION OF THE RESULTS.

An examination of Table No. 1 shows that during Experiments A to H, both inclusive, the steam was cut off by independent cut-off valves in both the small and large cylinders, the extremes of the point of cutting off in the small cylinder not varying much from its average of 0.3613 from the commencement of the stroke of the piston, and in the large cylinder not varying much from the average of 0.3553 of the stroke of the piston from the commencement. In both cylinders the intended point of cutting off was one-third of the stroke of the piston from the commencement. The slight variations in the measure of expansion with which the steam was used during these experiments could produce no effect on the economy of the power, that measure varying from 7.1245 to 8.2126 times.

The throttle-valve in all cases was carried wide open.

The steam pressure in the boiler varied for each experiment, and the number of double strokes made per minute by the pistons of the engine correspondingly varied. The degree of superheating given to the steam varied also in the different experiments, lessening as the boiler pressure lessened; this results from the fact that the lessening boiler pressures were accompanied by a lessening rate of combustion of the coal, whereby less heat in equal times being thrown upon the heating surfaces of the boiler a greater quantity of it was absorbed by them; the rapidity of the absorption being also quickened by the lessening temperature of the water within the boiler, while the temperature of the gases of combustion as they rose from the coal remained constant. It is also probable that more uncombined air, proportionally to coal consumed, passed through the grate during the slower than during the faster rates of combustion; besides which, the inleakage of air remaining nearly constant at all rates was in larger proportion to the gases of combustion at the slower than at the faster rates. The temperature of the gases of combustion when they impinged upon the steam-superheating surfaces was thus much less with the slower than with the faster rates of combustion, lessening as they lessened.

The variations in the three important quantities of boiler pressure, speed of piston, and degree of superheating are shown in the following table, in which the boiler pressure is expressed in pounds per square inch above zero.

Designation of the explriments.	Steam pressure in boiler in pounds per square inch above zero.	Number of double strokes of engine pistons made per minute.	Number of Fahrenheit degrees of superheating added to the steam.
A	144.25	221.50452	61.81
B	142.03	215.89459	62.00
C	119.22	192.18577	37.43
D	106.14	181.09517	34.03
E	89.74	166.53375	30.11
F	69.46	145.41532	17.75
G	47.09	111.50042	1.86
H	36.06	94.73779	0.00

With the boiler pressure, piston speed, and quantity of steam superheating thus widely varying in the above experiments no inference can be made as to the extent to which the economy of the power was affected by each variable. The combined effect of the three, however, was very strongly marked, the higher boiler pressure, faster piston speed, and more degrees of superheating causing the power to be developed with the greater economy of fuel.

The enormous extent to which the economy of the fuel was so influenced appears from the following table, in which the cost of the indicated, net, and total horse-powers is each given in units of heat:

Designation of the experiments.	Cost of the horse-power in Fahrenheit units of heat consumed per hour.		
	Indicated horse-power.	Net horse-power.	Total horse-power.
A	18,437.1605	19,836.2586	13,572.8978
B	18,077.7340	19,459.5666	13,514.9052
C	19,083.9054	21,014.0359	14,478.8082
D	19,958.2801	22,196.7462	14,938.4549
E	21,370.5809	24,423.4212	15,938.3515
F	23,907.5585	28,512.9848	17,281.0583
G	28,408.3902	38,239.7709	18,823.3534
H	36,956.6188	57,245.0744	20,693.5391

In the above table it will be observed that the ratio of the increase of the cost of the horse-power, proceeding from Experiment A to Experiment H, is least for the total horse-power, greater for the indicated horse-power, and greatest for the net horse-power.

This results from the fact that the back pressure against the piston of the large cylinder being nearly constant for all the experiments, and the pressure on the pistons required to work the unloaded engine being absolutely constant, while the remainder of the total pressure after deduction of the first for the indicated pressure, and of the sum of the two for the net pressure, varies as the squares of the piston speed, there follows that the back pressure and the pressure required to work the unloaded engine are greater and greater fractions of the total pressure as the piston speed becomes less and less; whereby the fraction of the total pressure utilized in the indicated and net pressures is less and less as the piston speed is less and less, the total pressure representing the entire dynamic effect of the steam.

A great many comparative experiments with steam machinery show that for the same engine working against a resistance which varies as the square of the piston speed, with steam in the saturated state and used with the same measure of expansion, the cost of the total horse-power is not affected by either the steam pressure or the piston speed. Applying this general result to the experiments in question, in connection with the fact that for Experiment H the steam was used without superheating or in the saturated state, the cost of the total horse-power in that experiment may be taken as what would be its cost in all the other experiments had they been made with saturated instead of superheated steam, the other experimental conditions remaining the same.

Designation of the experiments.	Number of degrees Fahrenheit the steam was superheated in the boiler above the temperature normal to its pressure as saturated steam.	Cost of the total horse-power in Fahrenheit units of heat consumed per hour.	Per centum of the cost of the total horse-power with saturated steam, saved by the use of superheated steam of the experimental degree of superheating.
A	61.81	13,572.8978	34.41
B	62.00	13,514.9052	34.69
C	37.43	14,478.8082	30.03
D	34.03	14,938.8549	27.71
E	30.11	15,938.3515	22.98
F	17.75	17,281.0583	16.49
G	1.86	18,823.3534	9.04
H	0.00	20,693.5391	0.00

In the table immediately preceding the last the cost of the horse-power during the above experiments has been given in Fahrenheit units of heat consumed per hour; but that cost can be expressed in pounds of anthracite consumed per hour by dividing the number of units of heat per hour per horse-power by the number of units of heat which the combustion of each pound of anthracite imparted to the water in the boiler. It will be shown farther on that the heat imparted to the water in the boiler by the combustion of one pound of anthracite having 18.28 per centum of refuse was, as the mean for all the experiments, 8,286.5308 Fahrenheit units. Dividing this number into the number of Fahrenheit units of heat consumed per hour per indicated, net, and total horse-powers, the quotients will be the number of pounds of anthracite consumed per hour per indicated, net, and total horse-power, which quotients will be found in the following table:

Designation of the experiments.	Cost of the horse-power in pounds of anthracite consumed per hour.		
	Indicated horse-power.	Net horse-power.	Total horse-power.
A	2.22495	2.39379	1.63795
B	2.18179	2.34834	1.68095
C	2.30300	2.53593	1.74727
D	2.40851	2.67865	1.80274
E	2.57895	2.94736	1.92340
F	2.88512	3.44076	2.08544
G	3.42826	4.61469	2.27156
H	4.45984	6.90822	2.49725

The results of Experiments I, J, and K compared with the results of Experiments A to H, both inclusive, Tables Nos. 1 and 1 continued, show the effect produced on the economy of the power by the removal of the cut-off valve from the large cylinder alone, all the other conditions remaining the same, substantially. That removal, however, did not affect the measure of expansion with which the steam was used, as in all cases after the communication of the small cylinder with the boiler was intercepted by the closing of the cut-off valve of that cylinder the steam was used expansively, both through the remainder of the stroke of the piston of that cylinder and throughout the entire stroke of the piston of the large cylinder, whether the cut-off valve of the latter was in use or not. Notwithstanding, however, that the cut-off valve on the large cylinder did not affect the measure of expansion with which the steam was used, yet it did affect, owing to other causes, the economy of the power to a very considerable extent.

The strictly comparable experiments for the effect produced on the economy of the power by the use or omission of the cut-off valve on the large cylinder are the mean of C and D with this cut-off in use and I with it omitted; Experiment E with this cut-off in use and J with it omitted; and Experiment F with this cut-off in use and K with it omitted. The experiments cited are comparable, because in them the number of double strokes made per minute by the pistons was nearly the same, and the total horses-power developed were nearly the same; the boiler pressure not varying much. But in Experiments I, J, and K, made with the cut-off valve omitted, the steam was much more superheated than in the comparable Experiments C, D, E, and F, made with this valve in use, thus giving the former set of experiments a considerable advantage over the latter set as regards economy of the power.

The mean of Experiments C and D gave for the piston speed 186.64047 double strokes per minute; in the comparable Experiment I this speed was 188.07645 double strokes. The mean of Experiments C and D gave 123.11637 for the total horses-power developed by the engine; in the comparable Experiment I this power was 116.60943 horses. The mean of Experiments C and D gave for the boiler pressure 112.68 pounds per square inch above zero; in the comparable Experiment I this pressure was 118.56 pounds. The mean of Experiments C and D gave 35.72 Fahrenheit degrees of superheating; in the comparable Experiment I the superheating was 42.85 degrees. The mean of Experiments C and D gave for the cost of the total horse-power 14708.6315 Fahrenheit units of heat consumed per hour; while in the comparable Experiment I the cost of the total horse-power was 16948.8632 Fahrenheit units of heat, showing that the omission of the cut-off valve increased the cost of the total horse-power 15.2307 per centum, notwithstanding the 7.13 degrees Fahrenheit more superheating of the steam in that case.

In Experiment E, with the cut-off valve of the large cylinder in use, the piston speed was 166.53375 double strokes per minute; in the comparable Experiment J, with this valve omitted, the piston speed was 167.31998 double strokes. In Experiment E the total horses-power developed by the engine were 85.29409; and in the comparable Experiment J, 78.84555. In Experiment E, the boiler pressure was 89.74 pounds per square inch above zero; and in the comparable Experiment J, 89.46. In Experiment E the steam was superheated 30.11 degrees Fahrenheit: and in the comparable Experiment J, 36.33 degrees. In Experiment E the cost of the total horse-power was 15938.3515 Fahrenheit units of heat consumed per hour; and in the comparable Experiment J, 17585.7600 units, showing that the omission of the cut-off valve increased the cost of the total horse-power 11.3361 per centum, notwithstanding the 6.22 degrees Fahrenheit more superheating of the steam in that case.

In Experiment F, with the cut-off valve of the large cylinder in use, the piston speed was 145.41532 double strokes per minute; in the comparable Experiment K, with this valve omitted, the piston speed was 145.87832 double strokes. In Experiment F the total horses-power developed by the engine were 59.46890; and in the comparable Experiment K, 54.51471. In Experiment F the boiler pressure was 69.46 pounds per square inch above zero; and in the comparable Experiment K, 70.69 In Experiment F the steam was superheated 17.75 degrees Fahrenheit; and in the comparable Experiment K, 31.58 degrees. In Experiment F the cost of the total horse-power was 17281.0583 Fahrenheit units of heat consumed per hour; and in the comparable Experiment K, 19620.3858 units, showing that the omission of the cut-off valve increased the cost of the total horse-power 13.5370 per centum, notwithstanding the 13.83 degrees Fahrenheit more superheating of the steam in that case. The mean of the above three determinations, under their widely different conditions of data, shows that the omission of the cut-off valve on the large cylinder, cutting off the steam at about 36 per centum of the stroke of its piston increased the cost of the total horse-power $\left(\frac{15.2307 + 11.3361 + 13.5370}{3} =\right)$ 13.3679 per centum of its cost, with the cut-off in use, notwithstanding that the steam was superheated when the cut-off valve was not in use $\left(\frac{7.13 + 6.22 + 13.83}{3} =\right)$ 9.06 degrees Fahrenheit more than it was when the valve was in use, the effect of which additional amount of superheating on the economy of the power being allowed for would make the economic loss due to the omission of the cut-off valve on the large cylinder about 20 per centum of the cost of the power when that cut-off valve is in use.

The enormous importance of the cut-off valve on the large cylinder, in respect to the economy of the power, is thus made apparent by these experiments, and, from considerations farther on, it appears that the shorter the steam is cut off in the large cylinder the greater the economy of the fuel with which the power is produced.

The effect of the greater superheating in Experiments I, J, and K, when the cut-off valve was not in use on the large cylinder, as compared with Experiments C, D, E, and F, when that valve was in use, is shown by the less cylinder condensation in the former set of experiments than in the latter. For example: The mean cylinder condensation in Experiments I, J, and K, in the small cylinder at the point of cutting off the steam, is 19.1111 per centum of the boiler vaporization. The similar mean in Experiments C, D, E, and F is 20.3612 per centum. The mean condensation in Experiments I, J, and K in the small cylinder, at the end of the stroke of its piston, is 2.7052 per centum of the boiler vaporization. The similar mean in Experiments C, D, E, and F is 2.9834 per centum. The mean condensation in Experiments I, J, and K in the large cylinder, at the end of the stroke of its piston, is 8.6511 per centum of the boiler vaporization. The similar mean in Experiments C, D, E, and F is 13.7466 per centum. The above per centums do not express the whole economy produced by superheating. They only show the portion of steam evaporated in the boiler liquefied in the cylinders by other causes than the production of the power due to the expanding steam alone. There may be a considerable shrinkage of the bulk of superheated steam with corresponding loss of mechanical effect, before liquefaction takes place.

Again, the effect of the greater degree of superheating in Experiments I, J, and K than in experiments C, D, E, and F, is shown in the higher terminal pressures in the cylinders relatively to the pressures at the point of cutting off in the former experiments than in the latter.

The mean pressure at the point of cutting off (0.3437 of the stroke of the piston from the commencement) in the small cylinder, for Experiments C, D, E, and F, taking the mean of C and D and averaging it with E and F, as heretofore, is 77.627 pounds per square inch above zero, and the mean terminal pressure in that cylinder is 32.725 pounds per square inch above zero; ratio, $\left(\frac{77.627}{32.725} =\right)$ 2.3721. The mean pressure at the point of cutting off (0.3338 of the stroke of the piston from the commencement) in the small cylinder, for Experiments I, J, and K, is 82.933

pounds per square inch above zero, and the terminal pressure in that cylinder is 34.333 pounds per square inch above zero; ratio, $\left(\frac{82.933}{34.333}=\right)2.4130$, or more than before, notwithstanding the shorter point of cutting off in Experiments I, J, and K, and the effect of the greater re-evaporation in the cylinder when using the less superheated steam of Experiments C, D, E, and F.

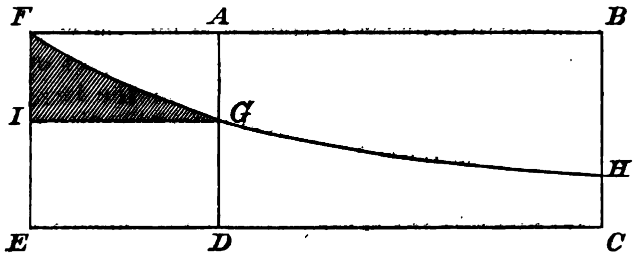
The mean terminal pressure in the large cylinder during Experiments C, D, E, and F was 7.745 pounds per square inch above zero, comparing which with the pressure at the point of cutting off in the small cylinder there is found the ratio $\left(\frac{77.627}{7.745}=\right)$ 10.0229. The mean terminal pressure in the large cylinder during Experiments I, J, and K was 9.180 pounds per square inch above zero, comparing which with the pressure at the point of cutting off in the small cylinder, there is found the ratio $\left(\frac{82.933}{9.180}=\right)$ 9.0341. In Experiments C, D, E, and F the steam was expanded as a mean 8.0561 times; and in Experiments I, J, and K, 8.2506 times; correcting the above 9.0341 in proportion to these numbers there is found 8.8211 for the ratio of the terminal pressure in the large cylinder to the pressure at the point of cutting off in the small cylinder for Experiments I, J, and K, when the steam is used with the same measure of expansion as in Experiments C, D, E, and F. The difference between the ratio 8.8211 and 10.0229 is the result of the greater degree of superheating possessed by the steam in Experiments I, J, and K than in Experiments C, D, E, and F.

There remains to be explained why, with steam of the same boiler pressure used with the same measure of expansion in a compound engine, with the same degree of superheating, and with the same piston speed, vacuum, &c., there should be so great a difference as about 20 per centum in the economic production of the power effected by the simple use or non-use of a cut-off valve on the large cylinder, cutting off at about one-third of the stroke of its piston from the commencement. The presence or absence of this valve does not affect the measure of expansion with which the steam is used, but it does affect the distribution of the power in the two cylinders; the shorter the point of cutting off in the large cylinder the greater is the initial steam pressure in it, and the greater is the back pressure against the piston of the small cylinder; thus, by shortening the cut-off on the large cylinder the power developed in that cylinder is increased and the power developed in the small cylinder decreased simultaneously, by which means the powers developed in the two cylinders can be made equal or with any difference desired. This was the sole end originally sought to be accomplished by the application of a cut-off valve to the large cylinder; and in many compound engines of the most recent date it is omitted as an unnecessary mechanical complication, not worth the cost of its construction. Such a view, however, is entirely erroneous, and the omission of the cut-off valve from the large cylinder, cutting off quite short, is fatal to the production of the power with the maximum economy obtainable in compound engines.

When there is no cut-off valve on the large cylinder, the steam exhausted from the small cylinder expands first into the space between the exhaust port of the small cylinder and the steam port of the large cylinder, continuing to expand as the piston of the large cylinder recedes until it reaches the end of its stroke; the pressures urging that piston being inversely as these increasing spaces. But when there is a cut-off valve on the large cylinder, the steam exhausted from the small cylinder expands first into the space between the exhaust port of the small cylinder and the steam port of the large cylinder, as before, continuing to expand likewise as the piston of the large cylinder recedes, until the latter reaches the point at which the steam is cut off in the large cylinder, after which the steam between the exhaust port of the small and the steam port of the large cylinder undergoes no more expansion. The steam imprisoned in the large cylinder by the closing of the cut-off valve between the steam port and the point of cutting off now completes the stroke of the piston of the large cylinder by its isolated expansion.

The difference in the two cases is, that when the large cylinder has a cut-off valve the steam between the exhaust port of the small cylinder and the steam port of the large cylinder is less expanded than when there is no cut off, and less and less in proportion as the cutting off is shorter and shorter. Now, as the whole of the mechanical effect due to the expansion of the steam in this space between the two cylinders is entirely lost, because not accompanied by a piston moving simultaneously through that space, then obviously the less of expansion the steam undergoes in that space the less loss of mechanical effect will it experience. It is plain that if this space were arranged as an extension of the large cylinder, with the piston of that cylinder capable of an additional movement through it, then there would be added to the power whatever was due to that space, multiplied by the mean pressure above the pressure at the steam port of the large cylinder, developed by the fall of pressure to that port from the pressure at the exhaust port of the small cylinder.

For illustration: in the following diagram let the rectangle A, B, C, D represent the large cylinder, and let the rectangle F, A, D, E represent the space between the small and large cylinders added as an extension of the large cylinder. For simplicity, suppose the two cylinders to be arranged with their axes in the same straight line and their pistons acting on the same crank-pin, so that the exhaust from the small cylinder is simultaneous with the receiving of steam by the large cylinder, and suppose the latter to be without a cut-off valve. Let the straight line E F represent the pressure above zero at the exhaust port of the small cylinder, and the straight line D G the pressure above zero at the steam port of the large cylinder. Let the curved line F G H represent the decreasing pressures due to the continuous expansion of the pressure E F to the pressure C H above zero at the end of the stroke of the piston



of the large cylinder. Let the straight line D C represent the zero of pressure in the large cylinder. As the total dynamic effect of the steam includes the driving out of the back pressure in the cylinder, the total power must be calculated down to D C.

The straight line I G represents the minimum pressure in the space A F D E between the two cylinders at the commencement of the stroke of the piston of the large cylinder. All the pressure between the pressures E F and D G, due to the continuous expansion of the steam from the former to the latter in the space F A D E, has no action whatever on the piston, and produces, consequently, no dynamic effect. The straight line I F represents the useless expansion undergone by the steam in the space (receiver and connections) between the two cylinders. But all the pressure between the pressures D G and C H, due to the continuous expansion of the steam from the former to the latter in the large cylinder, does act on the piston, and consequently produces its corresponding dynamic effect. When the piston reaches B C—the end of its stroke—the entire large cylinder and the space between the two cylinders are filled with steam of the pressure C H. The steam valve of the large cylinder now closes and the exhaust valve opens, thus imprisoning the pressure C H in the space F A D E, while that pressure passes from the large cylinder, or space A B C D, into the condenser.

When the piston of the small cylinder is at the end of its stroke and the exhaust port of that cylinder is opened, there were thrown into communication the interior of the small cylinder and the space between the two cylinders. When the pistons of the two cylinders began to move, as they advanced simultaneously, there were in common the spaces of the interiors of the two cylinders and the space between them; and the aggregate of these spaces augmented as the pistons moved by the difference of their space displacements, not by the space displacement of the piston of the large cylinder alone. All the steam received by the large cylinder is supplied from the small cylinder, *pari passu;* that is to say, as fast as the large cylinder draws a given weight of steam from the space F A D E, just so fast does the small cylinder put the same weight of steam into that space, because its piston advancing simultaneously with the piston of the large cylinder pushes that weight of steam out of it into the space F A D E, by which means there is exactly neutralized the loss of dynamic effect which would otherwise be sustained by the expansion of the steam in the space F A D E below the pressure D G.

The hatched area F I G represents the pressure above I G produced by expanding the pressure E F to the pressure D G; that is, F I G represents the available pressure for work produced by that expansion had there been any piston for it to act upon; but there being none, no dynamic effect is produced.

The power thus lost may be calculated if an indicator diagram be taken from the receiver or space F A D E, and the mean pressure in pounds per square inch above the pressure I G obtained from it. The contents of F A D E in cubic feet being known, and the number of double strokes made per minute by the pistons of the engine, the product of these three quantities, multiplied by 144 and divided by 33,000, will be the indicated horses-power required. There is no analogous loss of power with the simple engine.

When a cut-off valve is applied to the large cylinder, then the steam entering the space F A D E between the two cylinders is less expanded, because the space into which the expansion takes place is less than before by the space in the large cylinder between the point of cutting off and the end of the stroke of the piston. Hence, by the application of a cut-off valve, while the pressure E F remains the same, the pressure D G becomes greater, and the hatched area F I G becomes less. The shorter and shorter the point of cutting off in the large cylinder, the less and less becomes the area F I G, and the less and less of the total pressure due to the steam is lost. Some loss from this cause there always must be, no matter how short the point of cutting off, so that in this respect the arrangement of the small and large cylinders of a compound engine, with their axes in the same straight line and the pistons of both acting upon the same crank-pin—in which case no receiver is required, but only the necessary connecting-pipes conveying the steam from the ends of the small cylinder to the ends of the large one—is more economical than the arrangement of the cylinders side by side with their pistons acting on different crank-pins carried by cranks at right angles to each other, in which case a receiver between them is necessary. But, with a short cut-off on the large cylinder the economic difference in the two cases will not be much, while in the latter case the practical advantage is realized of the two pistons working at right angles to each other upon two crank-pins and four main journals instead of one crank-pin and two main journals, the pressures per square inch of journal being greatly diminished—a most important point in the practical working of an engine. In all cases the space between the two cylinders should be made as small as practicable. When no cut-off is used on the large cylinder in either case, then the arrangement of the cylinders without a receiver between them, and with their axes in the same straight line—their pistons acting on the same crank-pin—will give a marked superiority in economy of fuel over the side-by-side arrangement of the cylinders with a receiver between them. The former arrangement, for equal development of power, requires four cylinders if the cranks are to be at right angles, which would be accompanied by greater friction of the unloaded engines, more cylinder condensation, more weight, more space, and more mechanical complexity than the latter arrangement, which requires but two cylinders, though of double capacity. As long, therefore, as the dimensions of the cylinders with the side-by-side arrangement do not exceed convenience in their practical construction, the balance of advantages will probably be considered to be with the receiver system, in which case the use of a cut-off valve on the large cylinder, cutting off early in the stroke of the piston, is indispensable. For marine engines especially, and when of moderate dimensions, the receiver type will be found more mechanically convenient and better adapted for its uses.

In Experiments I, J, and K, made without the cut-off valve in action on the large cylinder, and with decreasing steam pressures in the boiler, and decreasing speeds of piston, there was a corresponding decrease in the superheating of the steam, accompanied by an increase in the cost of the horse-power in fuel; these results being precisely the same as in Experiments A to H, made under the same variable conditions and with the cut-off valve in use on the large cylinder. The less the consumption of coal per hour, the less was the superheating of the steam, owing to the facts that a less quantity of heat being thrown in a given time upon a given water-heating surface, more of it was absorbed in vaporizing water and less remained for steam superheating; and that with the slower rates of combustion more unconsumed air passed through the furnace and disproportionately cooled the gases of combustion.

The following table will show the extent to which the steam superheating was decreased by decreasing the boiler pressure and the piston speed:

Designation of the experiments.	Steam pressure in boiler in pounds per square inch above zero.	Number of double strokes of engine pistons made per minute.	Number of Fahrenheit degrees of superheating added to the steam.
I	118.56	188.07645	42.85
J	89.46	167.31998	36.33
K	70.69	145.87832	31.58

The combined effect of the widely varying boiler pressures, piston speed, and steam superheating shown in the above table, on the economy of the power, appears in the following table. It is impossible to say to what extent each of these variables entered into the economy as a factor, but it is probable that the degree of superheating was the principal cause influencing the cost in heat of the total horse-power, because many experiments with saturated steam have shown that the cost of the total horse-power is not sensibly affected by reducing the boiler pressure when the piston speed is correspondingly reduced in overcoming resistances, varying as the square roots of the pressures, the reduced cylinder condensation consequent on the less difference of temperature in the cylinder, due to the less difference of pressure there, compensating the increased cylinder condensation following the lessened piston speed.

Designation of the experiments.	Cost of the horse-power in Fahrenheit units of heat consumed per hour.		
	Indicated horse-power.	Net horse-power.	Total horse-power.
I	21,088.3958	23,323.1619	16,948.8632
J	22,989.9592	26,500.7234	17,585.7600
K	27,291.3581	33,186.6325	19,620.3858

In the above table, as in the similar one previously given for Experiments A to H, the ratio of the increase in the cost of the power is least for the total horse-power, more for the indicated horse-power, and most for the net horse-power, which results from the fact that the back pressure against the piston, and the pressure required to work the engine, *per se*, being constant, while the pressure applied to doing external work varies about as the squares of the piston speeds, they form larger fractions of the total pressure for the lower piston speeds than for the higher. The total horse-power includes pressures for all purposes down to zero; the indicated horse-power excludes the back pressure against the piston, but includes the pressure required to work the unloaded engine; and the net horse-power excludes both the back pressure and the pressure required to work the unloaded engine.

If the economy with which the total horse-power is produced in Experiments I, J, and K is affected by only the amount of superheating given to the steam, as before suggested, the following table will exhibit the gain due to the higher degrees of superheating over the lower one:

Designation of the experiments.	Number of degrees Fahrenheit the steam was superheated in the boiler above the temperature normal to its pressure as saturated steam.	Cost of the total horse-power in Fahrenheit units of heat consumed per hour.	Per centum of the cost of the total horse-power with the least superheated steam, saved by the use of the greater experimental degrees of superheating.
I	42.85	16948.8632	13.62
J	36.33	17585.7600	10.37
K	31.58	19620.3858	0.00

The cost of the horse-power during Experiments I, J, and K has been already given in Fahrenheit units of heat consumed per hour; the more familiar measure is pounds of coal consumed per hour; the one is easily changed into the other by dividing the units of heat by 8286.5308, which are the Fahrenheit units of heat imparted to the water in the boiler of the Leila by the combustion of one pound of anthracite containing 18.28 per centum of refuse in ash, clinker, soot, and dust. The quotients of such division will be found in the following table:

Designation of the experiments.	Cost of the horse-power in pounds of anthracite consumed per hour.		
	Indicated horse-power.	Net horse-power.	Total horse-power.
I	2.54487	2.81458	2.04535
J	2.77437	3.19804	2.12219
K	3.29346	4.00489	2.36774

Experiments L and M were made to ascertain the effect on the economy of the power, of using highly superheated steam with a much less measure of expansion than in Experiments A to K, both inclusive. For this purpose the cut-off valve in Experiments L and M was removed from the small cylinder, but kept in use on the large cylinder, so that the results from these two experiments are comparable with such of the experiments from A to H as correspond in speed of piston and extent of steam superheating, in which respects the results of Experiment L may be compared with the mean of those from Experiments B and C, and the results of Experiment M with the mean of those from Experiments F and G.

The mean boiler pressure in Experiments B and C was 130.62 pounds per square inch above zero; it was superheated 49.72 degrees Fahrenheit as a mean, and used with a mean expansion of 7.5271 times, the pistons of the engine making for a mean 204.04018 double strokes per minute. The boiler pressure in Experiment L was 76.32 pounds per square inch above zero; it was superheated 51.56 degrees Fahrenheit, and used with an expansion of 3.1963 times, the pistons of the engine making 197.92534 double strokes per minute. The much greater boiler pressure and the slightly greater piston speed given by the means of Experiments B and C may be considered as balancing the slightly more steam superheating in Experiment L. The mean cost of the total horse-power in Experiments B and C was 13996.8567 Fahrenheit units of heat consumed per hour; and in Experiment L it was 16848.6445 units, showing that, *with equal cylinder condensations obtained by a sufficiently high superheating of the boiler steam for that purpose*, increasing the measure of expansion from 3.1963 times to 7.5271 times, or about $2\frac{1}{3}$ times, decreased the cost of the total horse-power $\left(\frac{16848.6445 - 13996.8567 \times 100}{16848.6445} = \right)$ 16.9259 per centum. In this case the cylinder condensation of steam with the greater measure of expansion was at the end of the stroke of the piston of the large cylinder, 10.5793 per centum of the weight of steam evaporated in the boiler; and with the lesser measure of expansion, this condensation was 7.1600 per centum.

The mean boiler pressure in Experiments F and G was 58.27 pounds per square inch above zero; it was superheated 9.80 degrees Fahrenheit as a mean, and used with a mean expansion of 7.8390 times, the pistons of the engine making for a mean 128.45787 double strokes per minute. The boiler pressure in Experiment M was 35.78 pounds per square inch above zero; it was superheated 43.49 degrees Fahrenheit, and used with an expansion of 3.1963 times, the pistons of the engine making 122.58577 double strokes per minute. The much greater boiler pressure, and the slightly greater piston speed, given by the means of Experiments F and G, go to their extent in offsetting the more steam superheating in Experiment M. The mean cost of the total horse-power in Experiments F and G was 18052.2058 Fahrenheit units of heat consumed per hour; and in Experiment M it was 22383.1019 units, showing that, *with about equal cylinder condensations*, increasing the measure of expansion from 3.1963 times to 7.8390 times, or about $2\frac{1}{2}$ times, decreased the cost of the total horse-power $\left(\frac{22383.1019 - 18052.2058 \times 100}{22383.1019} =\right)$ 19.3490 per centum. In this case the cylinder condensation of steam with the greater measure of expansion was, at the end of the stroke of the piston of the small cylinder, 25.7844 per centum of the weight of steam evaporated in the boiler; and with the lesser measure of expansion this condensation was 17.5861 per centum. Again, the cylinder condensation of steam with the greater measure of expansion was, at the end of the stroke of the piston of the large cylinder, 30.4573 per centum of the weight of steam evaporated in the boiler; and with the lesser measure of expansion this condensation was 30.7887 per centum.

The mean of the two determinations shows that increasing the measure of expansion from 3.1963 times to $\left(\frac{7.5271 + 7.8390}{2} =\right)$ 7.6830 times decreased the cost of the total horse-power $\left(\frac{16.9259 + 19.3490}{2} =\right)$ 18.1374 per centum.

Experiment N was made with the cut-off valves removed from both cylinders, so that the steam was only expanded the number of times due to the dimensions of the small and large cylinders. This experiment is comparable with Experiment I for the purpose of determining the effect upon the economy of the power of using highly superheated steam with a greatly increased measure of expansion, as there was no cut-off valve on the large cylinder during Experiment I.

The boiler pressure in Experiment I was 118.56 pounds per square inch above zero; it was superheated 42.85 degrees Fahrenheit, and used with an expansion of 8.3481 times, the pistons of the engine making 188.07645 double strokes per minute. The boiler pressure in Experiment N was 71.91 pounds per square inch above zero; it was superheated 40.42 degrees Fahrenheit, and used with an expansion of 3.1963 times, the pistons of the engine making 189.52011 double strokes per minute. In this comparison the piston speed and the degree to which the steam was superheated varied but a trifle in the two experiments, but the boiler pressure was necessarily much the highest, with the greatest measure of expansion. The cost of the total horse-power in Experiment I was 16948.8632 Fahrenheit units of heat consumed per hour, and in Experiment N it was 23805.4776 units, showing that *with a sufficiently high degree of superheating to nearly prevent cylinder condensation*, increasing the measure of expansion from 3.1963 times to 8.3481 times, decreased the cost of the total horse-power $\left(\frac{23805.4776 - 16948.8632 \times 100}{23805.4776} = \right)$ 28.8027 per centum.

In this case there was no cylinder condensation at the end of the stroke of the piston of the small cylinder in either experiment. In Experiment I, in which the steam was used with the greater measure of expansion, the cylinder condensating at the end of the stroke of the piston of the large cylinder was 11.0780 of the steam evaporated in the boiler; and in Experiment N, in which the steam was used with the lesser measure of expansion, the corresponding cylinder condensation was 9.7211 per centum.

Experiments O and P were made after the compound engine had been reduced to a simple engine by the omission of the small cylinder, all the valves of which were taken out so as to allow the steam to pass directly through its exhaust passage to the large cylinder. The connecting-rod of the small cylinder was not removed, as the crosshead of that cylinder was required to be in motion in order to work some of the pumps, consequently the piston of the small cylinder continued to move up and down as usual, being worked by the crank-pin of the small cylinder. The steam, of course, was simultaneously upon both sides of the piston of the small cylinder, and the inference is that the pressure on the two sides of that piston was exactly balanced, as the steam passages at the two ends of the small cylinder were exactly alike, so that the admission of the steam was as free at one end as its exit was at the other; nevertheless, when the tortuousness of these passages is considered, and their length and small cross area, in connection with the great number of reciprocations made by the piston per minute, there are just grounds for supposing that the pressure on the side of the piston expelling the steam might have been notably greater than that on the side receiving the steam. In fact, an expansion of the entering steam is possible by the wire drawing due to the rapidity of the piston's reciprocations, and to the cross area, length, and bends of the steam passages; and a compression of the outgoing steam is possible from the same causes. If, however, any such inequality of pressure existed on the two sides of the piston, then the resulting resistance to the expulsion of the steam from the small cylinder was included in the total power developed by the large cylinder, which likewise included the additional frictions of the piston, the crosshead, the connecting-rod, and the main journals belonging to the small cylinder; so that no error can occur by taking the total horses-power developed by the large cylinder as the correct measure of the entire dynamic effect produced by the steam; but there would be an error in the calculation of the power expended in the slip of the screw and in the propulsion of the vessel, did the piston of the small cylinder offer any resistance to motion during Experiments O and P other than its friction. It is possible, therefore, that there may be an error in the distribution of the power exerted during these two experiments, the fractions of the net power expended in the slip of the screw and in the propulsion of the vessel being larger than the truth in the proportion of any resistance which the piston of the small cylinder might have undergone in the expulsion of the steam therefrom.

The economic results of Experiments O and P, made with the large cylinder alone, may be compared respectively with those of Experiments C and F, made with the two cylinders compounded; the piston speeds in the two cases varying but very little. The boiler steam was of much higher pressure in Experiments C and F than in Experiments O and P; but in the latter it was much more superheated, and in the former it was used with an enormously greater measure of expansion. It is therefore impossible, with such widely-varying conditions, to institute a comparison of the economic performances of the engine between the compounding of its cylinders or the use of its large cylinder alone. All that can be done is to compare the cost of the total horse-power in Fahrenheit units of heat consumed per hour in the two cases for the actual experimental conditions.

The boiler pressure in Experiment C was 119.22 pounds per square inch above zero; it was superheated 37.43 degrees Fahrenheit, and used with an expansion of 7.9298 times; the pistons of the engine making 192.18577 double strokes per minute. The boiler pressure in Experiment O was 59.74 pounds per square inch above zero; it was superheated 51.70 degrees Fahrenheit, and used with an expansion of 2.6554 times; the piston of the engine making 191.57236 double strokes per minute. The cost of the total horse-power in Experiment C was 14478.8082 Fahrenheit units of heat consumed per hour; and in Experiment O 23216.8074 units; showing that under the experimental conditions compounding the cylinders decreased the cost of the total horse-power $\left(\frac{23216.8074 - 14478.8082 \times 100}{23216.8074} = \right)$ 37.6365 per centum.

The boiler pressure in Experiment F was 69.46 pounds per square inch above zero; it was superheated 17.75 degrees Fahrenheit, and used with an expansion of 7.8884 times; the pistons of the engine making 145.41532 double strokes per minute. The boiler pressure in Experiment

P was 36.25 pounds per square inch above zero; it was superheated 42.61 degrees Fahrenheit, and used with an expansion of 2.4333 times; the piston of the engine making 147.63465 double strokes per minute. The cost of the total horse-power in Experiment F was 17281.0583 Fahrenheit units of heat consumed per hour; and in Experiment P 27800.0990 units; showing that under the experimental conditions compounding the cylinders decreased the cost of the total horse-power $\left(\frac{27800.0990 - 17281\ 0583 \times 100}{27800.0990} = \right)$37.8381 per centum.

The mean of the two determinations shows that with equal piston speed, and with steam superheated to the degree that it has but a very little cylinder condensation, then using it in compounded cylinders and at a high boiler pressure expanded $\left(\frac{7.9298+7.8884}{2} = \right)$ 7.9091 times, produces the total horse-power for $\left(\frac{37.6365+37.8381}{2} = \right)$37.7373 per centum less fuel than when it is used in a single cylinder with a low boiler pressure expanded $\left(\frac{2.6554+2.4333}{2} = \right)$2.5443 times. The conditions were as favorable as possible for the compounded cylinders, and as unfavorable as possible for the single cylinder. The superheating for Experiments O and P given in the table is that which the steam had in the steam-pipe at a little distance from the boiler, and just before it entered the small cylinder, not just before it entered the large cylinder; consequently, with the compounded cylinders the steam began its work without cooling, while with the single cylinder it was additionally cooled by exposure in the small cylinder, in the valve-chest of that cylinder, and in the pipe connecting the two cylinders, before beginning its work in the large cylinder. As superheated steam loses temperature with great facility, the degree of superheating given in the table for Experiments O and P is too high relatively to the degree of superheating in Tables C and F. Had steam been used in the large cylinder alone, with the same boiler pressure, degree of superheating, and measure of expansion as in Experiments C and F, made with the cylinders compounded, it is obvious from the large economy obtained from the high measures of expansion and high boiler pressures in those experiments that no economic gain would have been shown for the compounded cylinders. The proportions of the details of the Leila's engine would not allow a higher initial pressure to be carried on the piston of the large cylinder than was adopted in Experiment O; nor did the cut-off valve of that cylinder admit of change without much expensive work, which there was no time to execute.

Of the steam condensation in the cylinders.—An examination of the quantities on line 67 of Table No. 1 continued, shows that in all the experiments, let the degree of superheating of the steam be what it might, there was a considerable fraction of the weight of steam evaporated in the boiler, condensed in the small cylinder at the point of the stroke of its piston at which the cut-off valve closed. With the highest boiler pressure, and greatest degree of steam superheating employed (Experiments A and B), this condensation was 9.7614 and 10.5538 per centum of the boiler vaporization for a minimum, increasing, *pari passu*, as the pressures and superheating decreased, until the maximum of 43.5201 per centum (Experiment H) was attained, when there was no superheating. Notwithstanding, however, this condensation at the point of cutting off in the small cylinder, yet the re-evaporation therein, which began as soon as the cut-off valve closed, and continued both during the remainder of the steam stroke of the piston and the whole of the return exhaust stroke, under the resulting continually lessening pressures, and by reason of the contained heat in the water of condensation and in the metallic surfaces on which this water was deposited, was completed in the cases of Experiments A, B, C, and D, and nearly completed in the case of Experiment E, by the time the piston of the small cylinder reached the end of its stroke, as shown by the quantities on line 68. To accomplish this re-evaporation by the time the piston arrived at the end of its stroke, in an unjacketed cylinder having the small dimensions of the small cylinder of the Leila's engine, the minimum superheating required was between 34.03 and 30.11 degrees Fahrenheit, Experiments D and E, say 32 degrees.

In Experiments I, J, L, and N the same fact is found of the sufficiency of the steam superheating to allow whatever steam might be condensed during the early portion of the stroke of the piston of the small cylinder, to be re-evaporated by the time that piston reached the end of its stroke. The minimum superheating in those experiments was 36.33 degrees Fahrenheit.

In all the experiments, let the degree of superheating be what it might, there was a cylinder condensation at the end of the stroke of the piston of the large cylinder, but, as an average, it amounted to only about two-thirds of the similar condensation at the point of cutting off the steam in the small cylinder, nor was it influenced to so great an extent by the superheating in the boiler.

In general, when the steam entered the small cylinder there was a marked condensation up to the point of cutting off, after which came a re-evaporation, more or less complete at the end of the steam stroke, according to the degree of superheating which the steam possessed when it entered the small cylinder, but always complete by the time the return exhaust stroke was made. During the steam stroke of the piston of the large cylinder there was again condensation of the steam, the re-evaporation of which was not completed until some time during the return exhaust stroke of that piston.

Of the action of the single pipe overboard condenser.—The Leila's engine was fitted with a surface condenser consisting of a single copper pipe placed on the outside of the vessel as far below the water-line as practicable. It will be of interest to compare the action of this condenser with that of the ordinary inboard arrangement of tubes in a shell for surface condensation.

In the first place the area of the condensing surface given in the two cases differs enormously. In the Leila it was 0.118 square foot per square foot of water-heating surface in the boiler, while with the ordinary arrangement this proportion varies from 0.5 to 1.0, to 1.0 to 1.0; nevertheless, it was sufficient in the Leila to condense all the steam which could be obtained from the boiler at the maximum rate of combustion with artificial draught produced by the fan-blower, forming as good a vacuum in the condenser as in the case of the ordinary arrangement, with the same rate of combustion in the boiler-furnace. In general, one-fifth of the condensing surface is sufficient with the overboard single pipe condenser that is required with the ordinary inboard arrangement of tubes. And this results from the fact that the entire surface of the single overboard pipe is always simultaneously in contact with the refrigerating water, and consequently has the lowest temperature of that water on every particle of the surface; this water being rapidly renewed, not only by the forward movement of the vessel, but by any rolling and pitching it may have; even the slight movements of the waves, with the vessel steady, contribute to the same result.

But in the case of the ordinary arrangement of many tubes in a shell inboard the vessel, the refrigerating water progresses gradually over the surface, the same water coming in contact, successively, first with the first portion of these tubes, then with the next portion, then with the next succeeding, and so on until the last portion is reached. And during this progress the refrigerating water is continuously receiving accessions of temperature, so that although it may enter upon the tubes, say, at 60 degrees Fahrenheit, it leaves them at, say, 100 degrees, thus causing a unit of their surface to be continuously less and less efficient for steam condensation as it is more and more distant from the first portion of the tubes. For example, suppose the feed-water or water of condensation to be carried at the temperature of 120 degrees Fahrenheit, that of the discharge water, or refrigerating water, when leaving the tubes, being 100 degrees, and when entering them 60 degrees; then the efficiency of unit of surface of the first portion of the tubes may be expressed by ($120° - 60° =$) 60, while that of the unit of surface of the last portion of the tubes will be expressed by ($120° - 100° =$)20, or it will have only one-third of the steam-condensing efficiency of the unit of surface of the first portion.

Besides this cause of inferiority, there is that of the unequal distribution of the refrigerating water among the tubes by the circulating-pump of the inboard surface condenser. The refrigerating water will take the easiest route from the aperture by which it enters the condenser to that by which it leaves the same, passing mainly through or over the tubes in this path, and to a much less extent through or over those lying more remote. Further, there is always more or less air inleakage through the joints on the shell of the inboard surface condenser, but there can be no such inleakage with the single outboard pipe wholly submerged; it has neither shell nor joints nor air surrounding. The outboard single pipe condenser has probably not one-twentieth the weight and cost of an equivalent inboard tubular condenser, composed of shell, tubes, tube plates, injection valve and pipe, outboard delivery valve and pipe, &c. But these great advantages are not obtained for nothing; the outboard single pipe is more liable to accidental damage than the inboard tubular condenser, though it can be so well protected as to make such accidents very rare; when they do happen, however, the vessel must be docked for reparation, which is not needed for repairs to the inboard tubular condenser.

The most serious objection to the outboard single pipe condenser is that it produces a greater back pressure against the steam piston than the inboard tubular condenser produces, the same *minimum* pressure existing in both, thereby making the indicated and net (not the total) horse-powers more costly in fuel. When the steam is exhausted from the cylinder into the shell of the inboard tubular condenser it enters an area vastly greater than the cross-section of the exhaust pipe, and every portion of the interior is filled with sensibly the same pressure, for there can be no greater pressure in one portion of the shell than in another, the exhaust steam expanding almost instantaneously to all parts, so that there is only one pressure in the shell against which the exhaust steam is delivered. But in the outboard single pipe the conditions are very different. This pipe is, in effect, a continuation of the exhaust pipe of the engine, and has about the same cross-section at the point where it receives the exhaust steam, which section continuously decreases until it is about one-half at the opposite end of the pipe attached to the air-pump. This decrease is essential, because at the air-pump end the pipe must be filled completely with the water of condensation, for the uncondensed steam would pass above this water in an incompletely filled pipe and enter the pump, "blowing through," so to speak. Accordingly, the water of condensation has to be pushed along the whole length of the pipe by the pressure of the exhaust steam, and at the intervals of the exhaust, spasmodically, with great velocity and correspondingly great resistance. Besides which the exhaust steam does not come simultaneously on the whole condensing surface as it does in the inboard tubular condenser, but it comes progressively, passing in succession over, first, the nearest portion of the pipe, then over the next, and so on to the end. The condensation, therefore, in the pipe is not practically instantaneous as in the inboard tubular condenser, but comparatively slow, so that there is a great difference of pressure in the two ends of the pipe, that in the end receiving the exhaust steam being much greater than that in the end delivering the water of condensation into the air-pump. The result is, that for the same pressure in the bottom of the air-pump there is a much greater back pressure against the piston of the engine with the outboard single pipe than with the inboard tubular condenser.

The mean of all the experiments with the Leila gives the pressure in the bottom of the air-pump, or at the delivering end of her outboard single pipe condenser, 2.190 pounds per square inch above zero, while the least back pressure against the piston of the large cylinder—that is, against that piston at the commencement of its stroke—was 4.090 pounds per square inch, the difference being (4.090—2.190=) 1.900 pounds per square inch. With the inboard tubular condenser, the difference between the pressure in it and the least back pressure against the piston is about 0.400 pound per square inch, consequently the employment of the outboard single pipe condenser of the Leila increased the back pressure against the piston of the large cylinder (1.900—0.400=) 1.500 pounds per square inch, making it that much more than it would have been with an inboard tubular condenser, and correspondingly lessening the indicated and net pressures utilized.

When the Leila was developing her maximum power (Experiments A and B) the difference between the pressure in the bottom of her air-pump and against the piston of her large cylinder was (6.328—2.110=) 4.218 pounds per square inch, showing a loss of indicated and net pressure on the piston of the large cylinder of (4.218—0.400=) 3.818 pounds per square inch of that piston over what it would have been with an inboard tubular condenser. The economy lost by the outboard single pipe condenser, in function of the greater back pressure it causes against the piston, is about regained by the saving of the power required to work the refrigerating water-pump, and by the saving of air inleakage.

Mean performance of the anthracite.—The total weight of anthracite consumed during all the experiments, as given in Tables 3 and 4, was 18,346 pounds, of which 18.28 per centum was refuse of ash, clinker, soot, and dust; the quality of the coal being in this respect slightly below the fair merchantable average, which is $16\frac{2}{3}$ per centum; consequently of every 100 pounds of anthracite of average quality ($100 - 16\frac{2}{3} =$) $83\frac{1}{3}$ per centum is combustible matter, while of every 100 pounds of the experimental anthracite (100.00—18.28=) 81.72 per centum was combustible; hence the inferiority of the anthracite of the experiments to that of average quality was $\left(\frac{83.33 - 81.72 \times 100}{83.33} = \right)$ 1.932 per centum.

The experimental mean temperature of the feed-water was 82.3 degrees Fahrenheit, but it would have been about 100 degrees had it been pumped directly into the boiler without experiencing the cooling inevitable from its passage through the measuring tanks, consequently the economic vaporization, as computed from the temperature of 100 degrees, will be taken as accurate for the case of regular practice.

The quantity of heat imparted to the water in the boiler was, for a mean, equivalent to the vaporization from the temperature of 100 degrees Fahrenheit, and under the pressure of one atmosphere above zero of 7.68315 pounds of water by one pound of anthracite, and of 9.40176 pounds of water by one pound of the combustible or gasifiable portion of the anthracite, which are fully equal to the economic vaporization in marine boilers by the same coal consumed at the same rate of combustion per unit of water-heating surface. Under these conditions, the pound of crude anthracite imparted to the water 8286.5308 Fahrenheit units of heat, and the pound of combustible or gasifiable portion of the anthracite imparted 10140.1084 units.

The mean rate of combustion corresponding with the above economic vaporization was as follows:

Pounds of anthracite consumed per hour per square foot of grate	8.76985
Pounds of combustible consumed per hour per square foot of grate	7.16672
Pounds of anthracite consumed per hour per square foot of heating surface measured on the outside of coil pipe	0.46960
Pounds of anthracite consumed per hour per square foot of heating surface measured on the inside of coil pipe	0.55336
Pounds of combustible consumed per hour per square foot of heating surface measured on the outside of coil pipe	0.38376
Pounds of combustible consumed per hour per square foot of heating surface measured on the inside of coil pipe	0.45221

The steam was superheated, as a mean, 59.88 degrees Fahrenheit above the temperature normal to its pressure as saturated steam by the hot gases of combustion, which, consequently, thus imparted to it per pound of crude anthracite consumed ($7.68315 \times 59.88 \times 0.4805 =$) 225.8672 Fahrenheit units of heat, and per pound consumed of the combustible or gasifiable portion of the anthracite ($9.40176 \times 59.88 \times 0.4805 =$) 270.5106 Fahrenheit units. The heat employed in superheating the steam was $2\frac{2}{3}$ per centum of that which was employed in evaporating it.

From the above appears that of the heat developed by the combustion of one pound of the crude anthracite, there were utilized in the steam ($8286.5308 + 225.8672 =$) 8512.3980 Fahrenheit units; and from each pound of the combustible or gasifiable portion of the anthracite ($10140.1084 + 270.5106 =$) 10410.6190 units; to which must be added about one per centum for the heat lost by radiation from the casing of the boiler, making the heat accounted for 10515 Fahrenheit units per pound of combustible consumed.

The pound of the combustible or gasifiable portion of anthracite in its commercial condition, including its hygroscopic water, and allowing for its incomplete combustion in ordinary furnaces, generates by its combustion in oxygen, according to the experiments of Scheurer-Kestner, 16470 Fahrenheit units of heat; consequently, there were utilized in these experiments, as a mean, $\left(\frac{10515 \times 100}{16470} = \right)$ 63.84 per centum of the heat generated by the combustion of the coal, leaving 36.16 per centum in the hot gases of combustion escaping from the chimney, this latter percentage of the heat being useful only for the production of the chimney draught.

The mean temperature of the gases of combustion in the chimney, as nearly as it could be obtained from the melting points of metals, was 650 degrees Fahrenheit; consequently, the temperature of these gases when they rose from the incandescent anthracite on the grate was

(36.16 : 650 : : 100.00 :) 1797 degrees Fahrenheit. The temperature of the gases of combustion from anthracite when it is burned in air furnishing exactly the quantity of oxygen required for the complete combustion of the coal, is, according to Bunsen's experiments, 3650 degrees Fahrenheit; hence, the quantity of air which passed through the grate was $\left(\frac{3650}{1797}=\right)$ 2.03 times what was chemically necessary.

It was found impossible to produce the slightest priming or foaming in the Herreshoff coil boiler, even when the anthracite was forced to an excessively high rate of combustion by an artificial draught from a fan-blower delivering air into the closed ash-pit, the blower being driven at its maximum speed by a small independent steam cylinder.

Performance in smooth water uninfluenced by wind or current, deduced from the mean of the experiments.—A careful comparison of the experiments shows that within the experimental speeds the resistance of the vessel was sensibly as the square of its speed; and that if the results of the experiments at low speeds be omitted, because in them the assumed constants enter more largely, so that any error in them affects the final result to a correspondingly greater degree, the thrust of the screw or resistance of the hull at the speed of 10 geographical miles per hour may be taken at 1,175 pounds, the experimental thrusts having value in proportion to the number of runs made in each case over the base. The normal slip of the screw deduced from all the experiments, their slips having value in proportion to the number of runs made in each case over the base, is 22.50 per centum of its axial speed. With this slip, and for a speed of vessel of 10 geographical miles per hour, the number of revolutions made by the screw and of double strokes made by the engine-pistons was 163.602145 per minute. The indicated pressure to produce these revolutions, reduced to the piston of the large cylinder alone, was 21.97257 pounds per square inch; and the pressure, similarly reduced, required to work the unloaded engine was 2.626 pounds per square inch.

The following is the distribution of the power developed by the engine for the vessel's speed of 10 geographical miles per hour under normal conditions and at the experimental draught of water, calculated in the manner hereinbefore described for the quantities in Table No. 2:

	Horses-power.	Per centum of net horses-power.
Indicated power developed by the engine	65. 3676	
Power required to work the engine, *per se*	7. 8123	
Net power developed by the engine	57. 5553	100. 0000
Power absorbed by the friction of the load	4. 3166	7. 5000
Power expended in overcoming the resistance of the water to the surface of the screw-blades	6. 6369	11. 5313
Power expended in the slip of the screw	10. 4854	18. 2179
Power expended in the propulsion of the vessel	36. 1164	62. 7508
Totals	57. 5553	100. 0000

In the above distribution of the power, 36.1164 horses-power were expended in the propulsion of the vessel; that is, in overcoming the resistance of the water to the hull. This resistance is composed of two parts: one, that of the water to the external immersed or wetted surface of the hull; the other, that of the water to the form alone of the hull, abstracted from this surface resistance. The first may be called the resistance to the hull in function of surface; and the last, the resistance to the hull in function of form. Both these resistances follow the same law relatively to the speed of the vessel, increasing or decreasing in the ratio of the square of that speed. The last kind of resistance can only be ascertained experimentally, but the first can be approximated with reasonable accuracy by calculation for any given speed of vessel; and in vessels of exaggerated form in the direction of having the area of the external immersed surface of the hull in enormous proportion to the area of its greatest immersed transverse section, combined with very light draught of water and excessively fine water-lines, as notably in the case of the Leila, the resistance of the wetted surface of the hull forms sensibly the entire resistance of the vessel. As, however, the draught of water deepens, and the breadth becomes in larger proportion to the length with corresponding fullness of the water-lines, the resistance in function of form becomes important and may exceed the resistance in function of surface, as in the case of many armed vessels, particularly those which are also armored.

The speed with which the watery molecules glide over the wetted surface of the hull is less than the speed of the vessel, owing to the inclination of her water-lines to her longitudinal axis. In the case of the Leila, owing to her great length relatively to the mean breadth of her water-lines at her greatest immersed transverse section, and to the large amount of flat surface in the keel, skeg, &c., these two speeds may be taken to compare as 1.000 for the vessel and 0.993 for the watery molecules. The speed of the vessel at 10 geographical miles per hour being equal to $1{,}014\frac{1}{3}$ feet per minute, or to 16.9055 feet per second, that of the watery molecules will be $(16.9055 \times 0.993 =)$ 16.787 feet per second. The resistance of the water to a metallic surface moving in it at the speed of 10 feet per second, is 0.45 pound per square foot of surface, which increased in the ratio of 10^2 to 16.787^2 becomes 1.268 pounds; and as the external immersed surface of the hull, inclusive of keel, skeg, &c., is 935.5 square feet, the horses-power expended in overcoming the resistance of the water to the external immersed surface of the hull at the vessel's speed of 10 geographical miles per hour, is $\left(\frac{935.5 \times 1.268 \times 1014\frac{1}{3}}{33000} = \right)$ 36.4611 horses-power, showing that the resistance of the Leila was sensibly only that of the wetted surface of her hull; from which follows, that the resistance of this hull in function of speed was in the ratio of the square of the speed.

Effect of putting the helm hard over on the speed of the piston.—In making the experiments, the vessel was run at the experimental speed past the termini of the base a sufficient distance to turn and recover this speed on repassing the termini, the helm being put hard over as soon as a sufficient distance was run past the termini to secure this effect. The influence of the helm hard over was very marked on the speed of the engine, and as the pressures were maintained the same during the turnings as during the runs over the base, and as the counter and time were taken each time a terminus was passed, the decrease in the engine speed could easily be ascertained, and it amounted, as a mean for all the experiments, to 9.5 per centum; that is to say, the engine with the same piston pressure made 9½ per centum fewer revolutions per minute during the turnings than during the runs over the base. As during a portion of the time occupied by the turnings, the helm was not hard over, the vessel continuing on her straight course to obtain room to turn and recover her speed, this 9½ per centum is too little for the effect of the helm hard over. The putting over of the helm, however, was commenced at a very short distance past the termini, so that the 9½ per centum decrease in the piston speed may be taken to represent the mean between commencing to put the helm over and its return to the fore-and-aft direction.

THE HERRESHOFF SYSTEM FOR STEAM-YACHTS.

The Herreshoff steam-yachts are modeled and engined for speed alone, the purpose being to obtain the highest possible speed while allowing large cabins for the comfortable accommodation of a considerable number of persons. To this end the hulls are made with as sharp water-lines and as much dead rise as practicable, to secure an immersed solid of the least resistance, which involves a narrow beam proportionally to length; but this proportion, ascertained from a number of accurate experiments made by the Herreshoff Manufacturing Company, is not so great as is found in many transatlantic steamships, and the results of the board's experiments with the Leila show the modeling to have been carried to that degree of perfection which renders the resistance simply what is due to the wetted surface. This perfect form of model has also the important advantage, for vessels of yacht size, that the resistance in function of speed does not increase in a higher ratio than the theoretical one of the square of the speed. Vessels of the Leila's dimensions, when less skillfully modeled and driven above a very moderate speed, have their resistance increased largely above the theoretical law of the square of the speed.

To obtain with a given power the maximum speed for a vessel of given linear dimensions requires not only the modeling of the hull to be the best possible, but that its weight shall be the least, too; and in this latter respect, as well as in the former, the steam-yachts of the Herreshoff Manufacturing Company have no superior, that company having long made the construction of such yachts a specialty and the subject of exhaustive experiments. But the light weight has to be compensated by corresponding excellence of material and workmanship, and it is by a combination of these, with careful and skillful arrangement of the material and its fastenings, that the Herreshoff yachts are produced with as great strength and durability as are commonly found in much heavier hulls.

The Leila is about three years old, and has been in constant service in Chesapeake Bay, Long Island Sound, and Narragansett Bay, without repairs or showing any signs of weakness or change of form, notwithstanding the rough weather encountered and the high speed at which she has always been driven. Her strength appears quite sufficient for the use intended.

Owing to the small draught of these yachts, due to their extreme lightness, the screw has to be lowered a considerable portion of its diameter below the bottom of the keel, and a skeg carried down to protect it.

This skeg is a triangular extension downward of a small portion of the after end of the keel, and has its after side in the same vertical line with the after side of the stern-post; there is no rudder-post. To the bottom of this skeg is secured the brass shoe, extending beneath the screw and supporting the lower pintle of the rudder. The objections to the skeg are, the damage experienced in consequence of it by the vessel when grounding, the direct resistance of its forward edge, the resistance of the water to its surface, the restriction of the vessel to deeper water than it could otherwise navigate, and its increasing the time and space required for turning the vessel. In some cases the skeg has been advantageously replaced by a bent brass arm bolted to the bottom of the keel and thence extending, first, vertically, and afterwards horizontally, beneath the screw, the free extremity of the arm forming the lower pintle of the rudder. The rudder is counterbalanced and of metal, with its axis one-fourth of its breadth from the forward edge. With this proportion the vessel steers easily and promptly, the rudder surface being large.

The screws are of brass, four-bladed, with the diameters, pitches, and fractions of pitch giving the best economic results as determined for the particular cases by trials over a measured base. They are made always of uniform pitch and uniform length, with the edges of the blades at right angles to the axis.

The engine consists of two compounded cylinders, having the same stroke of piston but different diameters, placed vertically, side by side, and connected to cranks on the main shaft at right angles to each other. Each cylinder is fitted with an independent cut-off valve working on the back of the steam-valve, which latter is a three-ported slide, unbalanced, and worked by two eccentrics and a Stephenson link. The cut-off valves are worked by eccentrics. The cylinders are not steam-jacketed, but the steam is used with a considerable degree of superheating.

The engine has a surface condenser formed of a single copper pipe placed on the outside of the hull just above the keel, and extending from abreast of the engine aft, around the stern-post, and forward to the engine again. This pipe is of the greatest diameter at the end which receives the exhaust steam, and thence tapers gradually to about one-half of that diameter at the end which delivers the water of condensation, the uncondensed vapor, and the air, to the air-pump. This is the lightest, least bulky, cheapest, and most efficient surface condenser possible, and is vastly superior to the inboard tubular surface condenser in its freedom from the numerous joints between the tubes and their tube-plates, from the possibility of air-leaks, from the weight and bulk of the shell, from the weight of the tube-plates and excess of tube surface, from the weight and bulk of a pump to supply refrigerating water, and from the weight and bulk of the injection and outboard delivery valves. The outboard condensing pipe has a maximum supply of condensing water of the temperature of the sea water simultaneously on every portion of its surface, thereby requiring much less condensing surface than in the case of the inboard tubular condenser over whose tubes the same refrigerating water passes successively from one end to the other with continuously increasing temperature and consequent decreasing condensing efficiency. Not only is the refrigerating water changed by gravity, being displaced by the colder water as soon as heated, but it is also changed with every movement of the vessel about its center of gravity, as in rolling and pitching, as well as by the direct or backward motion of the vessel. In the latter case so great is the condensing efficiency, owing to the current of water thrown forward by the screw, that, even with the vessel secured to the wharf and the engine backing, as good a vacuum can be maintained with the maximum development of power as when the vessel is going freely ahead.

The disadvantages of the outboard condensing pipe are, that it is more exposed to injury from external accident; that in the case of vessels above a certain size they would require docking for its examination and repair, and that the resistance of the vessel is increased by the projected cross-section of the pipe and by its surface. Further, the amount of refrigerating water thrown upon the condensing surface does not admit of graduation, but remains always the same, let the quantity of steam to be condensed in a given time be what it may; thereby, when small powers are developed, often cooling the feed-water below the degree of maximum economy. And, lastly, there is a greater back pressure against the steam piston with the outboard condensing pipe than with the inboard tubular condenser, owing to the necessity of so reducing the diameter of the pipe at the air-pump end as to keep that end filled with water and prevent the blowing-through of steam. As a general average in practice the loss of fuel with the outboard condensing pipe, due to the less temperature of the feed-water, is about 1½ per centum; and the loss by the increased back pressure 6 per centum, making a total of 7½ per centum, or about one-fourteenth. These figures will of course vary, according to the variable conditions of practice, but the total will remain not far from the truth. When less power is developed by the same engine, there will be less loss from the increased back pressure and more from the decreased temperature of the feed-water. When more power is developed, there will be more loss from the increased back pressure and less from the decreased temperature of the feed-water. To these losses must be opposed the gains in economy of fuel made by the outboard condensing pipe as against the inboard tubular condenser. In the first place, the pipe has less air-leaks, which, as regards back pressure, amounts to about 2 per centum as a practical average; and, in the last place, there is saved the power expended in working the pump supplying the refrigerating water to the inboard tubular condenser, amounting to, say, 5 per centum, making a total of 7 per centum; so that, as regards economy of fuel, the difference can be but insignificant between the two kinds of condensing apparatus, leaving all the practical advantages intact in favor of the outboard pipe, with the *per contra* of its inaccessibility for examination and repair unless the vessel, if of moderate dimensions, be docked. The greater resistance added to the vessel by the projected area and surface of the outboard condensing pipe below water is to a certain extent neutralized by the greater immersion of the hull, due to the greater weight of the inboard tubular condenser and its accessories. It is quite certain, therefore, that for small vessels easily taken out of water anywhere, the outboard condensing pipe is greatly superior to the inboard tubular condenser; nor, in the absence of extended trial, can any limit be properly placed to its use in larger vessels; on board the Leila it was absolutely a success.

The air-pump, feed-pump, and circulating-pump for the boiler are all vertical and single acting, and are worked by a beam articulated to the crosshead of the forward or small cylinder.

In general arrangement and in the details the engine is of the utmost simplicity and accessibility; and every part is made as light as is consistent with strength, and equally strong, all useless weight due to disproportionate strength in particular places having been studiously retrenched. It is secured to a cast-iron bed-plate, which contains the three main pillow-blocks, the after one of which acts also as the thrust bearing by means of collars. The bed-plate is bolted to the keelsons, and the engine is independent of other support.

The principal feature in the Herreshoff system is the boiler, which is entirely different in design, mechanical details, appendages, and method of functioning from any boiler in use. It can have but one furnace, and that must be circular. It has no shell, no braces, no tube-plates, no rivets, no manhole-plates, and no handhole-plates. Its water is on top and its steam is beneath. The water-heating surface is entirely contained in a single wrought-iron pipe coiled in helical spirals around and over the furnace, which thus becomes inclosed, the gases of combustion passing out between the spirals and ascending between them and an outer casing made of two thicknesses of sheet-iron filled in with "mineral wool" or steam-blown furnace slag. All the water contained in the boiler is within this pipe, which it enters at the extreme upper end, and leaves at the opposite end as steam, mingled with whatever of the water may have escaped vaporization, both steam and water being delivered simultaneously into a closed vessel called a "separator," where they are disengaged by the difference of their specific gravity, the steam passing from the top of the "separator" by another pipe, which is carried back to the boiler and wound about the furnace in the uptake in as many spirals as may be necessary to give the proper surface for superheating the steam by the otherwise waste heat of the gases of combustion in the uptake, after which the pipe is carried to the engine. The water falls to the bottom of the "separator," whence it is drawn off by the circulating-pump and forced back into the upper end of the pipe. This circulating-pump is therefore an appendage peculiar to the kind of boiler described. So, also, is the "separator," without which the boiler cannot practically be worked; it is entirely a distinct vessel from the boiler, being merely placed at its side for convenience; its upper portion constitutes the steam-room, and the water-level in its lower portion shows the proper amount of feeding.

The boiler is to be used with a very much larger supply of feed-water than is delivered into it by the feed-pump; all the water pumped by the circulating-pump from the "separator" back into the boiler being what may be termed superfluous feed, not vaporizing, but merely circulating continually through the coil so as to produce a rapid current of water over the heating surfaces, sweeping off the steam bubbles as fast as formed, and keeping the whole surface in contact with water alone, thus rendering each unit of that surface of maximum vaporizing efficiency, and far exceeding in that respect, by this method of forced or mechanical circulation, anything which can be obtained in ordinary boilers where there is only the natural circulation due to the vaporization. The quantity of superfluous feed used depends merely on the capacity of the circulating-pump. The power required to work this pump is more than what is due to overcoming the friction of its piston and the resistance of the water in the pipe connecting pump and boiler, because of the fact that the pressure in the top of the boiler receiving this water is greater than the pressure in the "separator" by several pounds per square inch, being more or less, according to the quantity of water pumped in a given time. The circulating-pump has no analogue in other boiler systems, and the power required to work it is additional to the usual power cost of feeding, but that cost is so small that a considerable addition is insignificant. By blowing the superfluous feed-water from the "separator" directly into the condenser, the use of a circulating-pump can be avoided. In that case, however, there will result a loss of fuel measured by the weight of the superfluous feed-water multiplied by the difference between the temperatures of the boiler and the condenser. The superfluous feed-water having to be pumped out of the condenser by the air-pump and into the boiler by the feed-pump, entails further loss of fuel by the power required for these operations over what is required for working the circulating-pump; and as the air-pump and feed-pump would have to be made correspondingly larger for the additional quantity of water to be passed through them, nothing is gained in weight or bulk by the omission of the circulating-pump. Finally, more surface is required in the outboard condensing pipe by the amount of heat to be extracted from the superfluous feed-water.

If there was no superfluous feed-water pumped into the boiler, and the supply restricted to the quantity pumped in by the feed-pump and entirely vaporized, then, if that vaporization was wholly effected, as it might easily be, in the upper coils of the boiler, the lower coils in near proximity to the fire would be filled with steam only, which would become excessively superheated, and if the steam could not thus carry off the heat the lower coils would be burnt out. The least quantity of superfluous feed which can be used is that which will keep the lowest coil filled with water.

The use of a surface condenser with the Herreshoff boiler is a necessity, in order to obtain distilled water for feeding it, any lime in the water filling the boiler-coil with scale, which cannot be removed, and soon, causing its destruction by the burning out of the scale-coated metal. In all steam machinery employing surface condensers there is a small but inevitable loss of fresh water, partly at the joints of the pipes when they happen to leak—and they are rarely all tight—partly at the air-pump, which at each delivery-stroke pumps out into the atmosphere some uncondensed vapor, and partly, in all systems except the Herreshoff, at the safety-valve whenever steam is blown off there; but in the Herreshoff system the steam from the safety-valve is discharged into the outboard condensing pipe and there condensed, so that this otherwise loss of fresh water is saved and the noise of blowing off steam prevented. The small losses of fresh water with the Herreshoff system are replaced from a tank filled with fresh water, carried for that purpose; the tank being situate beneath the boiler.

The air-pump is not a necessity in the Herreshoff system. The surface condenser can be used without it to merely furnish the boiler with distilled water, in which case the back pressure against the piston will be several pounds per square inch above the atmosphere, and the temperature of the feed-water in the neighborhood of 200 degrees Fahrenheit, or more. This modification involves either a higher boiler pressure or a larger cylinder for the development of a given power.

As the boiler double casing on its inside is in contact with the gases of combustion at a very high temperature, it radiated an excessive amount of heat while its interior contained only air, but the filling of this space with mineral wool—a most excellent non-conductor of heat and unalterable by water or a degree of heat twice that to which it can ever be exposed—has so much impeded the passage of heat that the temperature of the outside of the casing does not exceed the temperature of the felt covering of an ordinary boiler containing the same pressure of steam. A further improvement in this respect has recently been made by the Herreshoff Manufacturing Company, in lining, so to speak, the casing with an extension of the boiler-coil, the spirals of which lining being brought into contact with each other, and with the casing entirely prevent the gases of combustion from reaching the latter. These lining spirals are used as a heater, the feed-water being first pumped through them, so that the temperature within them is greatly less than what is normal to the boiler pressure. With this arrangement and mineral wool in the casing, the heat radiated from the exterior of the Herreshoft boiler is barely perceptible to the hand.

The metal of the coil pipe is much thicker than what is used for ordinary boiler-tubes of the same diameter, in order to secure durability and strength. The coil pipe is first lap-welded in convenient lengths, which are then welded together at the ends into one continuous pipe that is wound spirally around a shaping form as fast as made, passing on the way through a furnace at which it obtains the necessary temperature for bending. When the coil is finished it is tested by hydraulic pressure to 1,000 pounds per square inch. There is no boiler which in cheapness and rapidity of construction approaches this. After the preparation of the lap-welded lengths a boiler can be completed in a few hours.

As the Herreshoff boiler has no shell, but merely a sheet-iron casing to confine and direct the hot gases of combustion, which casing is subjected to only a very trifling external pressure measured by the force of the draught; and as the boiler has no bracing, while its ash-pit and up take are composed of sheet-iron, the entire heating surface being in a continuous pipe of small diameter and thickness of metal, having no joints or connections, manholes or handholes, every portion of which pipe is heating surface, it is evident the arrangement produces for equal grates and heating surfaces a boiler of less weight than any other arrangement can. All the water in the boiler being contained in this pipe, will, with the pipe entirely filled, be an insignificant addition to the weight, the aggregate of which for boiler and contained water about equals half the weight of an ordinary boiler and its contained water. Further, as the steam pressure is wholly contained within the coil pipe, it is evident that, owing to the small diameter and considerable thickness of metal relatively to the diameter, this boiler must have greater strength than any other arrangement can. The "separator," which, in consequence of its diameter being much greater than the largest part of the pipe, is the weakest part of the system, is a plain cylindrical vessel, not exposed to the gases of combustion, and can have its strength brought up without difficulty to that of the pipe by giving adequate thickness of metal or shrinking wrought-iron bands over it.

As there are no joints to be affected by the contraction and expansion of the metal due to changes of temperature, and as the pipe containing the water and steam is a very flexible coil, without rigidity in any portion or direction, being free to come and go without rupture, it is evident this arrangement is one that offers the least possibility of accident from overheating. A low red heat which would destroy the unprotected heating surface of an ordinary boiler by tearing it from its fastenings would produce no apparent injury to a Herreshoff coil. In fact, the Herreshoff boiler can be left with heavy banked fire and open furnace-door without the least probability of accident from injury to the metal or from explosion. It is quite safe in any contingency, the only possibility of rupture being from imperfect welds in the metal, which rupture, however, would not produce an explosion, but only allow an escape of steam for a few moments from the orifice, the coil containing so small a quantity of water. The destroying agent is limited to the minimum in quantity and in surface on which it could act.

The same causes which produce the lightness of the Herreshoff boiler allow steam to be raised in it with wonderful rapidity, from three to five minutes only being required to generate steam enough from cold water to start the engine. There is no large mass of metal and water, as in the ordinary boiler, to be heated to the boiling point before vaporization can commence. The weight of both metal and water, especially the latter, is insignificant, and the time required to heat them is correspondingly brief.

The small weight of metal and water in the Herreshoff boiler makes its use economical for any service in which the time of continuous steaming is short. At each cessation of steaming, the mass of water and iron, which had been heated to the temperature normal to the pressure employed, cools down a certain number of degrees, which have to be again imparted at the expense of the fuel when the steaming recommences. The less the weight of metal and water subjected to this alternation, the less is the waste of fuel.

The coil pipe can be repaired for only ruptures in the line of the welded seam, due to imperfect welding, by putting iron clamps over the aperture. When any considerable surface is at fault, the length of pipe affected can be cut out and a new length inserted, the connections at the two ends being made by a pipe coupling with a right-hand screw at one end on the inside and a left-hand screw at the other end, corresponding threads being cut on the ends of the pipe to be joined.

The interior of the coil pipe is inaccessible for cleaning, scaling, and repair, or even for examination. It can be kept clean, however, when only distilled water from a surface condenser is used, by occasional dosages of a solution of soda and potash, which also neutralizes the corroding effect of the fatty acids from the oils employed to lubricate the cylinders and their valves. This boiler cannot be fed with water containing sulphate or carbonate of lime, because no amount of "blowing off" will prevent those salts from forming scale, which, as the interior of the coil pipe is absolutely inaccessible, cannot be removed by mechanical means. With sea-water the scaling is rapid; with river or well water, much slower, owing to the vastly less percentage of lime salts. When, however, distilled water from a surface condenser is fed, with occasional injections of soda or potash, the interior of the coil pipe after several years' use presents a perfectly clean, hard, smooth, and dark appearance, which is a lining composed of the black magnetic oxide of iron that acts as a protecting coat for the metal beneath. This oxide is formed by the iron at high temperature being in contact with the superheated steam, very frequently produced in the ordinary management of the boiler, or whenever from any cause there is not sufficient water to fill the coils, a condition that always exists when steam is newly raised or when the boiler remains for a short period inoperative, with the fire in action, or when the pumps fail to act for a few minutes. The black oxide does not corrosively affect its base like the ordinary iron-rust, which is the hydrated carbonate of the peroxide of iron, but, on the contrary, is an absolute protection against further corrosion, so that but little or no injury is sustained from this cause by the interior of the coil pipe. The corrosion of this pipe is almost wholly exterior and due to the combined influence of the sea-air, dampness, and the sulphur gases evolved by the combustion of the coal; nevertheless, several years of constant use under the above conditions do not seem to have much injured this boiler, certainly not to any degree requiring its renewal or materially weakening its strength; in some cases the deterioration was quite insignificant.

If, owing to absence of water, superheating occurs in the coil pipe, any grease there present, brought over from the cylinder and valve lubrication, will be partially charred, forming a tarry mass strongly adherent to the surface of the pipe. This substance can be quickly removed with turpentine injected by the feed-pump, only a very small quantity being required; if allowed to remain, it is mainly injurious by clogging up the valves and pistons of the air-pump, circulating-pump, and feed-pump.

The Herreshoff boiler can be used with anthracite for very long periods without requiring the coil to be swept of soot, because no soot remains upon it. Several months' daily use of two Herreshoff boilers by a board of naval engineers, for an average of twelve hours per day, failed to produce soot enough to need sweeping, and in fact that time could be greatly extended without inconvenience. This condition would probably remain the same if bituminous coal were used, because the freedom from soot is due to its removal by burning off every time the coil pipe is exposed without water for a few moments to the fire, which occurs whenever steam is newly raised, or the boiler remains inoperative a few minutes with the fire in action, or when the pumps fail to work for a few minutes. In ordinary boilers freshly formed soot adheres to the water-heating surfaces because it is then a tarry substance, and it retains its adhesiveness for some time because the surface on which it is deposited has not a sufficiently high temperature to quickly drive off the volatile constituents of the tar which causes its viscidity; but even in such cases there is a limit to the thickness of the soot deposit which cannot be exceeded, as after it is reached any further addition, being too far from the water-heating surface to be protected by its low temperature, is burned off by the gases of combustion. Conformably with this description, the soot deposit in ordinary boilers is found more copious and thicker the greater its distance from the furnace. In the Hereshoff boiler, the coil pipe or water-heating surface being frequently overheated—that is, heated considerably above the temperature normal to the steam, due to the absence of water while the fire is in action—the soot deposit has no protection from the low temperature of the water, and scarcely deposits to any sensible thickness before some accidental overheating burns it off, an overheating certain to happen whenever steam is newly raised.

When steam is raised in the Herreshoff boiler a little water is first hand-pumped into the upper spirals of the coil, and a fire of wood kindled, the pumping and the firing being almost simultaneous, so that a sharp heat is then thrown upon the surfaces before they are covered with water, and the soot as a consequence disappears before it, the boiler receiving no injury. In the ordinary boiler such a state of things cannot exist without destroying it. Its heating surfaces must be always kept covered with water, whose low temperature protects the soot, allowing it to increase to a certain thickness and remain.

The Herreshoff boiler, on account of the very small quantity of water it contains, requires more careful management than the ordinary boiler. Both its feed-pump and its circulating-pump must be incessantly watched. A failure of the feed-pump to act for a minute or two would leave the boiler absolutely without water in any part, and the engine would stop for want of steam, while the fire would remain in full action to destroy the unprotected metal. There would, indeed, be no danger of explosion, as there would not be any explosive material in the boiler, nor would there be any danger of explosion from suddenly pumping in feed-water upon the overheated metal, but the metal itself might be fatally damaged by the overheating. A failure of the circulating-pump during a minute or two to act would prevent the boiler from receiving the superfluous feed-water, and the steam would become highly superheated, with the possibility of damaging the coil pipe, cylinders, and valves. Unremitting attention is indispensable, and it is the most difficult quality to obtain in boiler attendants. The management of the Herreshoff boiler is special, and it is more difficult and troublesome than that of the ordinary boiler, which is well known to the entire personnel of steam engineering. Special instructions and experience are required for its use, and a very much greater care than is needed with ordinary boilers. This experience can be quickly acquired by the intelligence of such persons as are usually intrusted with boiler watches; nevertheless, it is something new to be learned; but the unremitting vigilance which must be exercised in the management of this boiler will be more difficult to obtain. There is no more difficulty in maintaining an uniform pressure with the Herreshoff boiler than with any other; all that is required is the proper functioning of the pumps. The experiments with the 'Leila' were all made with but a trifling variation of pressure, even throughout the longest trials, some of which continued during the entire daylight of a single day.

The coil boiler allows a perfectly durable and safe steam superheater to be added, involving no extra trouble in its use. This is effected by simply coiling the steam-pipe in a few spirals over the water-heating coil, where it can be acted on by the heat in the escaping gases of combustion which would otherwise be wasted. A superheating pipe, to be durable, must be arranged in coils with their flexibility and freedom of motion, so that the excessive contraction and expansion of the metal due to the great variations of temperature upon it will not tear it from its fastenings. No superheater made of straight tubes rigidly secured to plates, as in ordinary boilers, will be durable, unless the highest temperature of the gases of combustion impinging upon them is low. With the coil construction, as in the Herreshoff boiler, any temperature can be safely employed below a red heat. Vegetable packings for stuffing-boxes having been entirely superseded, with superheated steam, by mineral packings, asbestos in the form of either sheets or yarns proving perfectly successful, there is no difficulty experienced in using highly superheated steam and obtaining the great economy resulting therefrom.

With the grate 5 feet and 9 inches in diameter, a Herreshoff double-coil boiler, containing 18.6780 square feet of water-heating and 1.6938 square feet of superheating surface to 1 square foot of grate surface, can be constructed in a height of 8 feet. If the diameter be increased, the height must be increased in the simple ratio of the diameter if the above proportions of surface are to be retained, because the area of the grate is as the square of its diameter, while the heating surface for equal heights will be only as the diameter. The height of 8 feet allows this boiler to be placed below the water-line in any class of vessels larger than gunboats.

A most important merit of the Herreshoff boiler is its capability of having its rate of combustion forced to the uttermost by artificial draught from fan-blowers, without priming or foaming. It is impossible to make this boiler prime or lift its water, because the water is already at the top of the boiler.

The Herreshoff system, as embodied in the Leila's machinery, gives the power with an economy of fuel rarely, if ever, equaled; the indicated horse-power being produced for $2\frac{1}{4}$ pounds of common anthracite, having $18\frac{1}{4}$ per centum of refuse, as the cost under the conditions of ordinary practice.

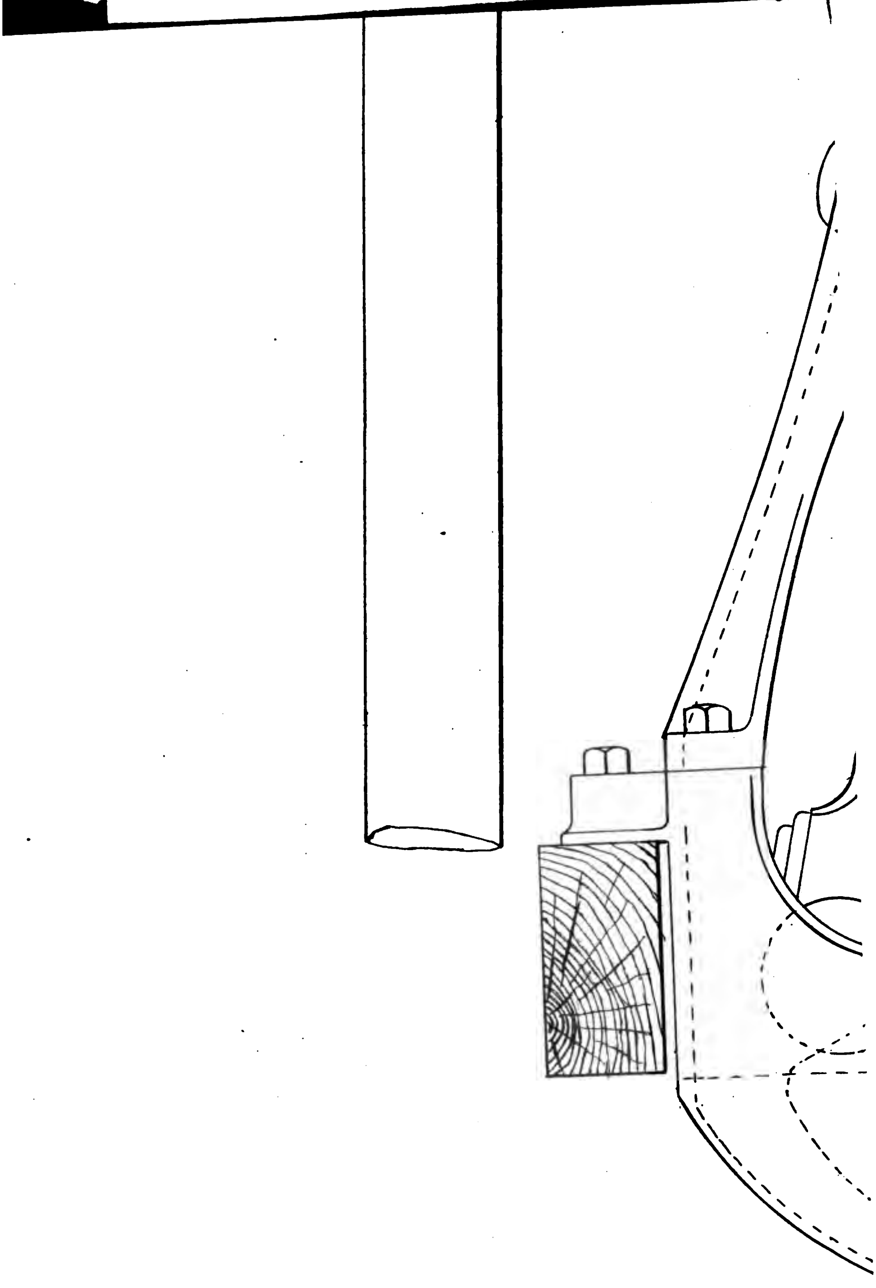
For steam-cutters, launches, gunboats, torpedo-boats, vedette-boats, tug-boats, yachts, dispatch-vessels, small rams for littoral warfare, and small sloops of war, the Herreshoff boiler is very greatly superior to any other known to the undersigned. It is the safest, simplest, lightest, and cheapest, and possesses so great a balance of advantages that, in our opinion, none other can be put in competition with it.

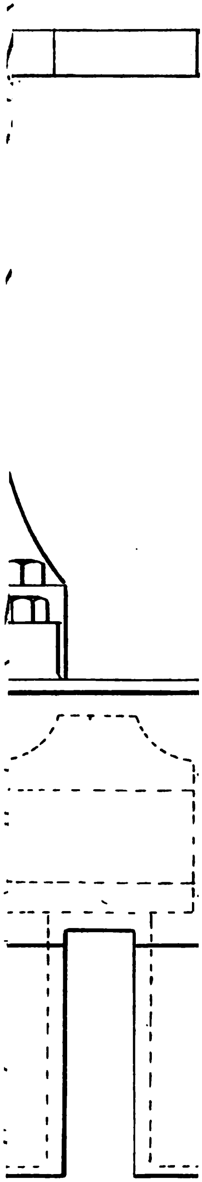
We know of no reason for restricting the use of this boiler to vessels of the above sizes except that it has not been tried on larger ones, but it can doubtless be as advantageously adapted to the largest vessels as to the smallest ones. The saving of one-half the weight, the absolute safety from explosion, the capability of working properly at the highest rate of combustion, with very high pressures superheated as much as the constitution of iron will admit, the rapidity of raising steam, and the simplicity and cheapness of the construction, in all of which it is superior to any boiler we know, while its economic vaporization is as great, are of such importance in steam navigation that its use must extend as its merits become understood.

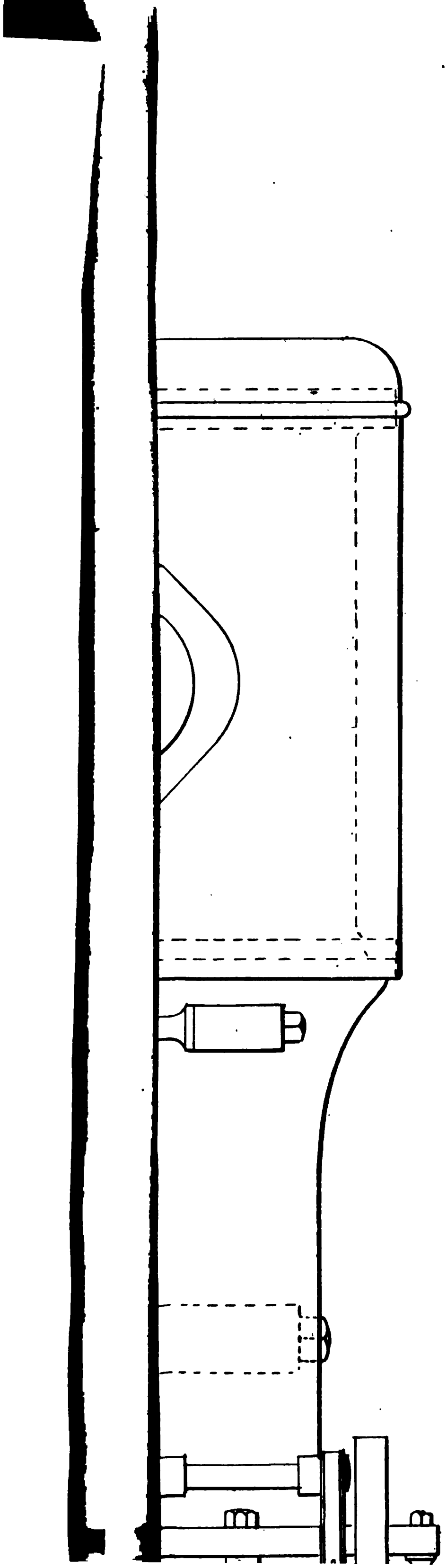
Your obedient servants,

B. F. ISHERWOOD, *Chief Engineer.*
THEO. ZELLER, *Chief Engineer.*
GEO. W. MAGEE, *Chief Engineer.*

Engineer-in-Chief WM. H. SHOCK, U. S. Navy,
Chief of the Bureau of Steam Engineering, Navy Department.







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