REPORT

MADE TO THE

BUREAU OF STEAM-ENGINEERING,

NAVY DEPARTMENT,

MARCH 3, 1883,

BY

B. F. ISHERWOOD,
CHIEF ENGINEER IN THE UNITED STATES NAVY,

ON THE

HULL, ENGINE, AND BOILER

OF THE STEAM-YACHT

"SIESTA,"

CONSTRUCTED BY THE

HERRESHOFF MANUFACTURING COMPANY,
AT BRISTOL, R. I.

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ON

THE STEAM-YACHT SIESTA.

NEW YORK CITY, *March* 3, 1883.

SIR: The Herreshoff Manufacturing Company, of Bristol, R. I., having kindly placed at my disposal, for the purpose of making a limited number of experiments, the steam-yacht Siesta, just built by them for Mr. Hulbert H. Warner, of Rochester, N. Y., I accepted the offer and was able to satisfactorily complete the trials that form the subject of this report, which is respectfully submitted for the information of the Bureau.

I regret I had not the assistance of a board of naval engineers, and that the trials could not be longer in their duration and more varied in their conditions. The limit of time was fixed by the fact that the vessel had to be delivered to her proprietor as soon as completed; and the want of the assistance of a board of naval engineers was supplied as far as possible by the Herreshoff Manufacturing Company assigning me a sufficient number of the best qualified persons in their employ for the purpose. During the experiments the carpenters were on board finishing the joiner work, but in all other respects the hull and machinery were in proper condition for trial.

The duration of each experiment was confined to the shortest time in which reliable results could be secured, with the view of making as many experiments under different conditions as possible, and it is believed no better distribution of the limited time could be effected.

Of course the entire, and by no means inconsiderable, expense was borne by the Messrs. Herreshoff, who showed in this their well-known liberality and their desire for the improvement of steam machinery. To them, therefore, the reader is indebted for the very valuable and interesting facts which these trials have established. Among them the entirely novel one of the great influence exercised on the economy of the compound engine by the more or less short cutting off of the steam in the large cylinder. Certainly, the variation of the cut-off of the large cylinder, that of the small cylinder remaining constant, does not theoretically affect the measure of expansion with which the steam is used, but it so affects the distribution of the steam in the two cylinders as to cause a very important difference in the economic result. This fact was first pointed out by me in a report on the Herreshoff system of motive machinery as applied to the steam-yacht Leila, made to the Bureau of Steam-Engineering on June 3, 1881, and it receives in these experiments with the motive machinery of the Siesta the strongest confirmation.

In the following pages will be found the detailed dimensions and descriptions of the hull and machinery of the Siesta; 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 a drawing of the Siesta's boiler, which is an entirely new modification of the well-known coil boiler, patented and heretofore constructed by the Herreshoff Manufacturing Company.

HULL.

The construction of the hull is composite, the frame being of angle iron planked with southern pine. The hull is not coppered, the intended service of the vessel being in the fresh water of the Saint Lawrence River and the great northern lakes. The stem and stern-post are of wood. A house erected upon the deck forms the upper portion of the cabin and of the space occupied by the engine and boiler, forward of which is a large dining-room entirely above deck, and the pilot-house, still more elevated. A narrow passage extends between these constructions and the light bulwark rising about 2 feet above the vessel's side. There are two masts fitted with a light schooner rig. The deck beams are of wood. There is no skeg at the stern descending below the bottom of the keel; the keel itself extends beneath the screw and terminates at a wooden rudder-post to which a metallic rudder is pintaled in the usual manner. The frames are of $\frac{5}{16}$ inch thick angle-iron, molded 3 inches and sided 2 inches, and are placed 18 inches apart from center to center. The bottom planks are 2½ inches thick, and the side planks are 2 inches thick. The deck planks are 1¾ inches thick. The keel and rudder-post are 8 inches broad.

The fore body of the hull has very sharp water lines, but the after body preserves on deck quite to the stern nearly the breadth at the greatest immersed transverse section, which allows an unusually spacious cabin, though causing the water lines aft to appear more than ordinarily full. The two bodies of the hull are made to lap over each other as much as practicable so as to obtain as great a length as possible for each. No speed is sacrificed by this principle, nor are the vessel's qualities in smooth water any way injured.

The following are the principal dimensions and proportions of the hull 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 16 persons embarked and two days' supply of coal.

Extreme length on top of deck	98 feet.
Length on water line from forward edge of stem to after edge of stern-post	90 feet 4 inches.
Extreme breadth on top of deck	17 feet.
Extreme breadth on water line	15 feet 2 inches.
Depth of hull amidship from lower edge of rabbet of keel to top of sheer plank	8 feet 4 inches.
Depth at stem from water line to lower edge of rabbet of keel.	4 feet 9½ inches.
Depth amidship from water line to lower edge of rabbet of keel	4 feet 9½ inches.
Depth at stern from water line to lower edge of rabbet of keel.	4 feet 2½ inches.
Depth of keel, uniform below lower edge of its rabbet	10½ inches.
Siding of keel	8 inches.
Draught of water forward from bottom of keel	5 feet 8 inches.
Draught of water amidship from bottom of keel	5 feet 8 inches.
Draught of water aft from bottom of keel	5 feet 1 inch.
Area of the water section	878.72 square feet.
Area of the greatest immersed transverse section, above the lower edge of the rabbet of the keel	42.48 square feet.
Area of the greatest immersed transverse section, including projected area of the keel	43.0633 square feet.
Displacement above lower edge of rabbet of keel	2178.56 cubic feet.

Displacement, including keel and rudder-post	2234.00 cubic feet.
Displacement, including keel and rudder-post	63.83 tons.
Area of external wetted or immersed surface of hull, exclusive of keel, rudder-post, and rudder	1203.20 square feet.
Area of surfaces of keel	214.80 square feet.
Area of surfaces of rudder	20.00 square feet.
Aggregate area of wetted surfaces of hull, keel, and rudder	1438.00 square feet.
Ratio of the length to the breadth on the water section	5.95604
Ratio of the water section to its circumscribing parallelogram	0.64138
Ratio of the greatest immersed transverse section to its circumscribing parallelogram	0.58453
Ratio of the displacement above the lower edge of rabbet of keel to its circumscribing parallelopipedon	0.33185
Ratio of the displacement above the lower edge of rabbet of keel to a solid having the greatest immersed transverse section for base, and the length on the load-water line from forward edge of stem to after edge of stern-post for height	0.56772

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

The entire bottom of the hull was thickly and smoothly painted, previous to which the planking was carefully sand-papered.

ENGINE.

The Siesta is fitted with a vertical, direct-action, condensing, compound engine, whose two cylinders are placed side by side above the crank-shaft, with their axes in the central vertical, longitudinal plane of the vessel. Each cylinder is supported on a cast-iron frame of two legs, the inner sides of which are bored to segments of a circle and act as guides for the main cross-head. The cast-iron frames are secured upon a cast-iron bed plate which is bolted to the keelsons and supports the crank-shaft pillow-block, pumps, and reservoirs.

There is one piston-rod to each cylinder, extending through only the lower head of the cylinder. The cross-sections of the connecting rods are rectangular, and the forward and after rods have cross-sections of different dimensions.

The valve chests are placed between the cylinders, each chest being distinct and having its own separate cover. Each chest contains the steam valve and the cut-off valve of its respective cylinder; both valves are slides working under the full steam pressure, and the cut-off valve moves on the back of the steam valve. The steam valve is operated by the usual Stephenson link and two eccentrics, the link being attached directly to the valve-stem, while the eccentrics are immediately beneath upon the crank-shaft. The cut-off valve is adjustable by means of a sliding bar connected with the valve-stem, and is operated by an independent eccentric placed directly beneath it on the crank-shaft. The steam valve is a three-ported slide. The cut-off valve working on the back of the steam valve is a plate with two steam passages through it. The throttle valve is a screw disk placed in the steam pipe, just in advance of the valve-chest of the small cylinder. The cylinders are not steam-jacketed, and the steam is not superheated by any special contrivance for that purpose.

The crank-shaft, with its two after cranks and after crank-pin, is a a single forging of steel, the forward crank being single and forged separately. The forward crank-pin is also a separate forging and overhangs the forward crank. Each crank carries its own counterbalance. The crank-shaft is supported by three pillow-blocks. Its after journal has the thrust collars made upon it, and its after pillow-block receives the thrust of the screw.

There is one air-pump; it is vertical, lifting, and single acting; it has

Area of the steam-port of the large cylinder	21 square inches.
Length of the exhaust-port of the large cylinder	14 inches.
Breadth of the exhaust-port of the large cylinder	$2\frac{5}{8}$ inches.
Area of the exhaust-port of the large cylinder	36.75 square inches.
Length of the steam-port in the cut-off valve of the large cylinder	14 inches.
Breadth of the steam-port in the cut-off valve of the large cylinder	$1\frac{3}{16}$ inches.
Area of the steam-port in the cut-off valve of the large cylinder	16.625 square inches.
Length of the passage in the cut-off valve of the large cylinder	$2\frac{1}{2}$ inches.
Aggregate interior surface of the large cylinder and one steam-passage, including half the surface of the piston-rod and the surface of one side of the piston	2,338 square inches.
Ratio of the space displacement of the piston of the large cylinder, per stroke, to that of the small cylinder	2.962275.
Depth of the piston of the large cylinder at the circumference	$2\frac{1}{2}$ inches.
Thickness of the metal of both cylinders	$\frac{11}{16}$ inch.
Depth of packing ring in both cylinders	$\frac{7}{8}$ inch.
Diameter of the air pump (single acting)	6 inches.
Diameter of the air pump piston-rod	$1\frac{1}{8}$ inches.
Stroke of the air-pump piston	$6\frac{1}{4}$ inches.
Space displacement of the air pump piston, per stroke	0.098669 cubic foot.
Diameter of the feed pump and of the circulating pump (single acting)	$1\frac{1}{8}$ inches.
Stroke of the plungers of the feed pump and circulating pump	18 inches.
Space displacement of the plungers of the feed pump and circulating pump, per stroke	0.0103544 cubic foot.
Diameter of the shell of the condenser	20 inches.
Length of the shell of the condenser	$29\frac{3}{4}$ inches.
Number of tubes (brass) in the condenser	731
Outside diameter tubes	$\frac{1}{2}$ inch.
Length of tubes between plates	$29\frac{1}{2}$ inches.
Condensing surface in tubes, measured on outside circumference	235.233 square feet.
Number of crank-shaft journals	3
Diameter of crank-shaft journals	$3\frac{7}{8}$.
Length of crank-shaft journals	8 inches.
Diameter of forward crank-pin journal	$2\frac{1}{4}$ inches.
Length of forward crank-pin journal	$4\frac{1}{2}$ inches.
Diameter of after crank-pin journal	$3\frac{7}{8}$ inches.
Length of after crank-pin journals	$3\frac{1}{2}$ inches.
Diameter of cross-head journal	$2\frac{1}{4}$ inches.
Length of cross-head journal	$3\frac{1}{4}$ inches.
Diameter of line shaft (iron)	$3\frac{3}{4}$ inches.
Length of both connecting rods between centers	$49\frac{1}{2}$ inches.
Breadth of forward connecting rod at cross-head end	$1\frac{3}{4}$ inches.
Depth of forward connecting rod at cross-head end	1 inch.
Breadth of forward connecting rod at crank-pin end	$2\frac{1}{4}$ inches.
Depth of forward connecting rod at crank-pin end	1 inch.
Breadth of after connecting rod at crank-pin end	$2\frac{3}{8}$ inches.
Depth of after connecting rod at cross-head end and at crank-pin end	1 inch.
Width of eccentric straps for the Stephenson links	$2\frac{1}{8}$ inches.
Width of eccentric straps for the cut-off valves	$1\frac{1}{4}$ inches.
Number of thrust collars on crank-shaft	5.
Inside diameter of thrust collars	$3\frac{7}{8}$ inches.
Outside diameter of thrust collars	5 inches.
Breadth, fore and aft the shaft, of the thrust collars	$\frac{1}{2}$ inch.
Width of the forward crank (forced on crank-shaft by hydraulic pressure)	$3\frac{1}{8}$ inches.
Width of the after cranks (forged on crank-shaft)	$2\frac{3}{4}$ inches.
Length occupied in the vessel by the engine	66 inches.
Breadth occupied in the vessel by the engine, exclusive of condenser	36 inches.
Extreme height of engine above the axis of its shaft	96 inches.
Extreme height of engine above the bottom of its bed plate	111 inches.

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 vitae bearing.

The following are the dimensions of the screw:

Diameter of the screw	55 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.

(See drawing.)

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

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

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

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

The feed-water enters the lowest spiral of the first or outside or cylindrical coil, passing upwards successively through all the spirals of that coil into the outer or greatest spiral of the upper of the three horizontal convolutions of pipe forming the second coil at the top of the boiler, thence successively through all the spirals of that convolution, after which it enters the inner or smallest diameter spiral of the central of the above three horizontal convolutions, passing successively through the spirals of that convolution, and then in like manner entering the outer or greatest diameter spiral of the lowest of the above three horizontal convolutions, passing successively through the spirals of that convolution, and finally entering the third or inside coil at the top in the inner or smallest diameter spiral, descending successively through all the spirals of that coil. The feed-water, in its passage through these coils, is first heated from the temperature of the hot well to that of the boiler, and then vaporized. According to the quantity of heat thrown upon the coils, and to the quantity of feed-water pumped into them in a given time, the feed-water may not be entirely vaporized until it arrives in the lower spiral of the third or inside coil, or it may be entirely vaporized at any previous spiral of any of the coils, in which case, except for a device about to be described, the remaining spirals would act as steam superheating surface, and possibly obtain too high a temperature, which latter condition is objectionable, both on account of injury to the unprotected metal of the overheated spirals, especially those of the third or inside coil, which are exposed to the intense direct radiation from the incandescent fuel of the furnace and on account of the valve-chests and steam cylinders of the engine, which would be injured by the steam thus superheated. It is necessary, therefore, to have the means of keeping the whole or whatever part of the heating surface is desired covered with water, and for this purpose recourse is had to a forced circulation of what may be called a superfluous quantity of feed-water by a circulating pump, which, by continually drawing this superfluous feed-water from the "separator" (a vessel to be hereafter described), into which the lowest end of the third or inside coil discharges, and, forcing it into the lower convolution of the second or horizontal coil, keeps the surfaces of all the spirals of all the coils covered with water if enough superfluous feed be employed, thereby entirely preventing superheating, or regulates the degree of superheating if a less quantity of the superfluous feed be pumped in.

The pressure in the coil pipe decreases gradually from the receiving end of the first or outside coil to the delivering end of the third or inside coil. It is at the maximum where the feed-water enters, and at the minimum where the steam and water are delivered. This difference of pressure, caused mainly by the surface resistance of the spirals and their continual deflection of the water from a straight course, reacts against both the feed-pump and the circulating-pump, causing the feeding of a Herreshoff coil boiler to be 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 (which is recovered in most part through the heating of the water by the heat equivalent of this friction), and the resistance of its water to the inner surfaces and bends of the pipe between the "separator" and the pump, and between the pump and the lower convolution of the second coil. This additional power-cost of feeding the Herreshoff boiler, is what is paid for the forced circulation of the water within it.

From the delivering end of the third or inside coil, the mixed water and steam are projected into the "separator," which is merely a closed cylindrical vessel distinct from and situate by the side of the boiler, wherein the water by its greater gravity separates from the steam and falls to the bottom, while the steam is carried off from the top by the main steam-pipe which conducts it to the valve-chest of the small cylinder of the engine. All the feed-water is converted into steam. All the circulating water remains water, collects in the bottom of the "separator," and is again forced by the circulating-pump into the lower horizontal convolution of the second coil, and so on continually.

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 a circulating-pump, or its equivalent, and without a "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 a coil boiler by its being composed of a single pipe of excessive length in proportion to inner diameter, coiled with a very slight inclination or pitch, and by the very small quantity of water in it, which, flowing slowly along the spirals of the coil, has time to become vaporized in the progress. The action of gravity alone upon the water in the coil would produce but a very slow movement, and its circulation is due almost exclusively to the pumps.

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

Owing to the small quantity of water in a coil boiler, and to the small weight of metal in contact with it—being the weight of the coil pipe only—steam can be raised to any working pressure in a few minutes and maintained as soon as raised. The pipe itself is tested by hydrostatic pressure to one thousand pounds per square inch, and its cohesive resistance far exceeds this. The practical limit of pressure is the strength of the engine, the strength of the boiler being so great that it is practically inexplodible.

The furnace of the Siesta's boiler consists of a circular grate 5 feet 9 inches in diameter, surrounded by a circular wall of fire brick laid in fire-clay. The grate bars are of cast iron, 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 $7\frac{13}{16}$ inches. The thickness of the wall is $5\frac{13}{32}$ inches, and its mean height above the bottom of the grate-bars is $14\frac{1}{2}$ inches; both it and the grate-bars rest upon a wrought-iron ring 5 feet 6 inches in inner diameter, 6 feet 7 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 9 inches in inside diameter, 6 feet $7\frac{13}{16}$ inches in outside diameter, $5\frac{1}{2}$ inches in thickness, and $9\frac{1}{2}$ inches in height on the outside. This wall incloses the ash-pit, the bottom of which is sheet-iron lined with common brick masonry $2\frac{1}{4}$ inches thick, and shaped like an inverted frustum of a cone. The inner diameters—top and bottom—of this frustum are 5 feet 9 inches and 2 feet 9 inches, respectively; height of frustum, 5 inches. The opening for the ash-pit door is rectangular, 39 inches wide and 8 inches high. Except this opening the ash-pit is entirely inclosed. The furnace has but one door; it is rectangular, placed in the fire-brick wall above the grate-bars, and has an opening 18 inches wide and 12 inches high. The outer bottom corners of the furnace brick wall and of the ash-pit brick wall are protected by a ring or hoop of angle iron sided $1\frac{1}{2}$ inches, and $\frac{3}{8}$ of an inch thick. The wrought-iron ring supporting the grate-bars rests upon the upper angle-iron, and the sheet-iron of the bottom of the ash-pit is secured to the lower angle-iron.

Upon the top of the brick wall inclosing the furnace rest the three coils, formed of one continuous pipe of different diameters. The first or outside coil is vertical and is composed of twenty-three spirals of wrought iron lap-welded pipe wound around an imaginary right cylinder of 76.0125 inches diameter and 45.6 inches height, the spirals touching, that is, having no openings between them. Between the bottom of this coil and the top of the brick wall inclosing the furnace is an annular sheet-iron box, rectangular in cross-section and filled with mineral wool. This box is $1\frac{1}{2}$ inches wide and 4.4 inches high; its top and part of its outer side are formed of a ring of angle-iron, and the remainder of its outer side is formed of two rings of angle-iron back to back. All these angle-irons are sided $1\frac{1}{2}$ inches and are $\frac{3}{8}$ of an inch in thickness. The pipe of this coil is 1.9 inches in outside diameter, 1.494 inches in inside diameter, with the metal 0.203 inch in thickness. The length of the axis of this coil is 469.3 feet; half the exterior surface of this coil is 116.719603 square feet; half its interior surface is 91.778467 square feet, and its content is 5.7132113 cubic feet. The feed-water from the reservoir enters this coil at the beginning of its lowest spiral, and leaves it at the end of its highest spiral to immediately enter the beginning of the outer spiral of the upper convolution of the second or horizontal coil.

The second coil, which is a direct continuation of the first, is composed of three horizontal convolutions, each containing thirteen spirals. The convolutions are precisely alike. The corresponding spirals of the upper and lower convolutions are placed vertically above each other; those of the intermediate convolution are placed opposite the intervals between the spirals of the other two convolutions. The least vertical distance in the clear between the convolutions is one inch. The outside diameter of each convolution is 78 inches and the inside diameter 17 inches. The wrought-iron lap-welded pipe from which it is made is 1.9 inches in outside diameter and 1.494 inches in inside diameter, with the metal 0.203 inch thick. The length of the axis of the pipe of each convolution is 156 feet, which, together with the two interior connections, makes a total length of pipe, measured on its axis, for the three convolutions of the second or horizontal coil, of 473.5 feet, the exterior surface corresponding to which is 235.528370 square feet, the interior surface 185.199676 square feet, and the content 5.76434165 cubic feet. The calorimeter, or area for the passage of the gases of combustion through the second coil, is 4.000000 square feet.

On the top of each of the three convolutions of the second or horizontal coil are four bars of flat iron, 1¾ inches wide by ⅜ of an inch thick, laid radially and equispaced, to which are secured by nuts the 0.483333 inch diameter round stirrup irons or staples which keep the spirals of each of the three convolutions that distance asunder.

The third or inside coil is made by winding spirally the wrought-iron lap-welded pipe of which it is composed, around a cast-iron shaper formed of the frusta of two cones, the smaller frustum being superimposed upon the larger one, and their angle of junction rounded. The lower or larger frustum is 67 inches diameter at base, 51 inches diameter at top, and 43 inches high. The upper or smaller frustum is 51 inches diameter at base, 12 inches diameter at top, and 7 inches high. The angle of junction of these two frusta is rounded on a radius of 6 inches.

This third coil, which is a direct continuation of the second coil, is composed, commencing at the top, of a length of 3 feet of pipe, 1.9 inches outside diameter, 1.494 inches inside diameter, and 0.203-inch thickness of metal; outside area 1.492260 square feet, inside area 1.173388 square feet, content 0.0365217 cubic foot. The space for the passage of the gases of combustion between this length of pipe and its adjacent spiral is $\frac{1}{16}$ inch wide, making the calorimeter or area for the passage of these gases 0.015624375 square foot.

Next, of a length of 49 feet of pipe of 2.375 inches outside diameter, 1.933 inches inside diameter, and 0.221 inch thickness of metal; outside area 30.466975 square feet, inside area 24.796911 square feet, content 0.9985923 cubic foot. The space for the passage of the gases of combustion between adjacent spirals of this diameter pipe is $\frac{1}{16}$ inch wide, making the calorimeter or area for the passage of these gases 0.765594375 square foot.

Next, of a length of 65 feet of pipe of 2.875 inches outside diameter, 2.315 inches inside diameter, and 0.280 inch thickness of metal; outside area 48.923875 square feet, inside area 39.394355 square feet, content 1.899956916 cubic feet. The space for the passage of the gases of combustion between adjacent spirals of this diameter pipe is ⅜ inch wide, making the calorimeter or area for the passage of these gases 2.03125 square feet.

Last, of a length of 136 feet of pipe 3.500 inches outside diameter, 2.892 inches inside diameter, and 0.304 inch thickness of metal; outside area 124.61680 square feet, inside area 102.969082 square feet, content 6.203877166 cubic feet. The space for the passage of the gases of combustion between adjacent spirals of this diameter pipe is ⅝ inch wide, making the calorimeter or area for the passage of these gases 7.083333 square feet.

The length of the axis of the wrought-iron lap-welded pipe composing the third or inside coil is 253 feet, corresponding to which is an outside area of 205.499910 square feet, an inside area of 168.333736 square feet, and a content of 9.138948 cubic feet. The space for the passage of the gases of combustion through this coil is 9.89580208 square feet.

On the outside of the third or inner coil there are four bars of flat wrought iron, 3 inches wide and ¾ inch thick, laid vertically and equispaced circumferentially, to which are secured by nuts the round stirrup irons or staples which keep the spirals of that coil at the desired distance apart.

In order to prevent the pipe of the third or inside coil from straightening from the pressure within it, the coil is held together by four wrought-iron straps of $\frac{3}{4}$ inch diameter, arranged upon its outside diagonally, passing slantwise from top to bottom and crossing at the center.

The length of the axis of the pipe of different diameters but uninterrupted continuity, composing the three coils of the boiler, is 1195.8 feet; the outside heating surface of this pipe is 557.74788 square feet; its inside heating surface is 445.31188 square feet; and its content is 20.61650 cubic feet.

The three coils and the brick walls of the furnace and ash-pit are inclosed by a cylindrical casing of sheet iron $\frac{3}{32}$ of an inch thick and 80 inches in external diameter. The height of the casing is 7 feet 4 inches, and it is in contact with the brick walls of the furnace and ash-pit and with the exterior of the first or outside coil, which it hoops, and thus keeps in position against the tendency of the pressure within the pipe to straighen it. The height proper of the boiler is from the sheet-iron bottom of the ash-pit at its lowest point to the top of the casing, namely, 7 feet 10 inches.

The uptake rests symmetrically upon the top of the casing, and is composed of two parallel sheet-iron plates $\frac{3}{32}$ of an inch thick with a $\frac{29}{32}$ inch wide intervening space filled with mineral wool, the top of the casing being stiffened with an angle iron hoop sided $1\frac{1}{2}$ inches, with $\frac{3}{8}$ inch thickness of metal, to support the uptake. In form the uptake is a frustum of a right cone of 80 inches outside diameter at bottom, $23\frac{1}{2}$ inches outside diameter at top and 12 inches outside height.

The chimney rests upon the uptake, is $23\frac{1}{2}$ inches in diameter and 25 feet in height above the top of the grate. It is secured to the uptake by an angle iron hoop.

The "separator" is an entirely distinct vessel placed by the side of the boiler, with a space of 10 inches in the clear between them. It is a hollow cylinder of $\frac{1}{4}$-inch thick boiler plate, 15 inches in outside diameter, and $72\frac{1}{2}$ inches in height. The top and bottom of this cylinder are screwed into hemispherical cast-iron ends. The total height of the "separator," including these ends, is 7 feet 4 inches. On the upper hemisphere the safety-valve is placed, and it discharges into the condenser instead of into the air. From the bottom of the lower hemisphere a pipe leads to the circulating pump for the boiler. On the side of the cylinder, near its bottom, is placed an ordinary glass water-gauge. Inside the cylinder is a standing pipe of $\frac{1}{4}$ inch thick boiler plate, $3\frac{7}{8}$ inches in outside diameter, the upper extremity of which is 3 feet below the upper end of the cylinder; the bottom of this pipe is screwed into a cast-iron partition forming a cylindrical cavity in the interior of the lower hemisphere, which cavity communicates with the lower end of the third or inside coil, and receives from it the water and steam from the boiler. From the top of the upper hemisphere the steam is conducted to the valve chest of the small cylinder of the engine. The bottom of the lower hemisphere is fitted with a blow-off pipe and cock for draining the "separator" and blowing out any sediment that may collect in it.

Beneath the boiler, immediately under its ash-pan, is a wrought-iron water tank, $8\frac{1}{2}$ feet long, $3\frac{1}{2}$ feet wide, and 13 inches deep. This tank is filled with fresh water for renewing any losses of water from the boiler due to any cause whatever.

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

Diameter of the boiler to outside of casing	80 inches.
Height of the boiler from bottom of ash-pit to top of uptake	106 inches.
Diameter of the furnace	69 inches.
Area of the grate surface	25.9672 square feet.
Area of water-heating surface measured on the outside of the coil pipe	557.74788 square feet.
Area of water-heating surface measured on the inside of the coil-pipe	445.31188 square feet.
Aggregate area of the spaces between the spirals of the third or inside coil for the passage of the gases of combustion	9.89580 square feet.
Aggregate area of the spaces between the spirals of the second or horizontal coil, for the passage of the gases of combustion	4.00000 square feet.
Cross area of the chimney	3.01202 square feet.
Diameter of the chimney	23.5 inches.
Height of the chimney above top of grate bars	25 feet.
Steam room in the separator	5.67130 cubic feet.
Height of steam room in the separator	5 feet.
Water room in the separator	2.34600 cubic feet.
Water room in the coil pipe, supposing the latter to be entirely filled with water	20.61650 cubic feet.
Square feet of water-heating surface, measured on outside of coil pipe, per square foot of grate surface	21.47894
Square feet of water-heating surface, measured on inside of coil pipe, per square foot of grate surface	17.14901
Square feet of grate surface per square foot of space between the spirals of the inside or third coil for the passage of the gases of combustion	2.63739
Square feet of grate surface per square foot of space between the spirals of the horizontal or second coil for the passage of the gases of combustion	6.49180
Square feet of grate surface per square foot of cross area of chimney for the passage of the gases of combustion	8.62119

THE EXPERIMENTS.

With the vessel and machinery as described, there were made the experiments whose data and results are given in the following tables.

Both cylinders of the compound engine being fitted with cut-off valves variable for certain points of cutting off the steam, the purpose of the experiments, among others, 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. These variations enabled the measure of expansion with which the steam was used to be varied largely, other things remaining the same; and they enabled, by changing the cut-off point in the large cylinder, the back pressure against the piston of the small cylinder to be varied, other things remaining the same. The power developed by the engine, the speed of its pistons, and the initial pressure on the piston of the small cylinder, were also considerably varied with the view of determining their effect upon the economy of the power. As there was no intentional superheating of the steam in the boiler, and no cushioning of the back pressures in the cylinders, there were no variations in the economic results due to these causes to be taken into the account. The experiments, at the same time, determined the economic and potential vaporization of water in the boiler by anthracite consumed with natural draught; the speeds of the vessel with the powers exerted; the slips of the screw at the different speeds, and the corresponding resistances of the hull. The cylinder condensations, both absolutely and relatively to the feed-water, were likewise ascertained at the point of cutting off and at the end of the stroke of the piston of the small cylinder, and at the end of the stroke of the piston of the large cylinder.

Most of the data and results will be found in the following two tables, namely, Table No. 1, and Table No. 1 continued, which two constitute in fact only one table, the division having been made merely for facility of manipulation. The consumption of anthracite is not given in these tables for each of the experiments, because their duration was not long enough to enable it to be determined 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 Tables No. 1 and No. 1 continued.

The speed of the vessel, slip of the screw, resistance of the hull, &c., were obtained for only five of the experiments out of the total twelve, the vessel during only these five having been run over a measured base; the remaining seven experiments, however, were made with the vessel in free route in smooth water, and, with the exception of Experiment G, which was made against a moderate head wind, in nearly calm air; but these five included the greatest variations of the vessel's speed.

To fully and unquestionably determine the various problems in relation to the economic performance of the compound engine, touched by these experiments, the latter should have been much more numerous and of longer duration. The indispensable condition for exactness of deduction, namely, that everything except the point at issue should be precisely the same, could not be commanded with so few experiments; and the mean quantities per hour from experiments so short comparably with the powers developed, do not give as sure a guarantee as could be desired for exactness. In short experiments special causes may be in operation which would be neutralized in long experiments. The results are therefore offered, not as determining the cases absolutely, but as throwing light upon new features of much practical importance in the theory of the compound engine.

MANNER OF MAKING THE EXPERIMENTS.

The experiments were made on the 14th, 15th, 16th, and 22d of June, 1882, but only those of the 22d June were made with the vessel run over a measured base; therefore, in the case of only the latter was the vessel's speed determined in addition to the determinations of the other experiments.

The experiments of June 22 were all made in Narragansett Bay, and each consisted of two consecutive runs between the Bristol Ferry lighthouse, situated on a small rock in the bay and marking the upper end of the base, and the lower end of the South Dumpling Rock lying in the bay near its mouth and marking the lower end of the base. This base was nearly a straight line of exactly twelve statute miles length, or 63,360 feet. The vessel during each run was made to pass within a few yards of each terminus of the base, so there could be no doubt of the accuracy of the distance.

For each run the time of passing each end of the base was taken by two observers with a seconds watch; and, at the same instant, two observers took the number on the engine-room counter, the difference between which gave the number of double strokes made by the steam pistons or of revolutions made by the screw in passing over the base.

Simultaneously with these observations, the number of tanks of feed-water on the tally board was also noted, which gave the weight of feed-water pumped into the boiler during the run.

The weight of feed-water pumped into the boiler was ascertained with absolute exactness by measurement in two duplicate tanks, connected, and used alternately. Each tank contained 37.4 pounds of water at the temperature of 64 degrees Fahrenheit, and was made of tin. The water of condensation was discharged by the air-pump from the condenser through a hose into each tank alternately, and from the tanks this water ran by gravity into the feed-water reservoir of the engine, from which it was forced into the boiler by the feed-pump. Every time a tank was emptied a mark was made on a tally-board kept by a person especially detailed for that purpose.

The two tanks were cylindrical vessels placed side by side, nearly touching, and each had a narrow cylindrical neck surmounted by a funnel-shaped mouth at top. These necks were connected at bottom by a small horizontal cylindrical pipe The center of the bottom of each tank had a small pipe fitted with a stop-cock, which pipe delivered the tank water into the feed-water reservoir; the reservoir end of the pipes being in open sight. As soon as a tank was filled—the cock in its bottom pipe being closed—the discharging hose from the air-pump chamber was turned by an attendant into the other tank, and when the water in the filled tank had fallen to the level of the lower side of the pipe connecting the tanks, the cock was opened in the pipe leading from the filled tank to the feed-water reservoir and that tank emptied, which required but a few seconds, and then the cock was closed. When the other tank became filled the same process was repeated, and so on alternately. The temperature of the water in the tanks was noted every ten minutes from a thermometer kept permanently immersed, and the weight of water was corrected for this temperature.

The weight of anthracite consumed was ascertained for only June 22, and the weight of water vaporized by it was also ascertained. In commencing the experiments of that day, the vessel being at anchor off the city of Bristol, Rhode Island, the boiler was filled with water, and a fire made with pine wood, whose weight was not noted, but which was allowed to burn completely 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. No steam was blown into the atmosphere, nor was the engine stopped until after the vessel was anchored at night. The whole of the water delivered by the air-pump from the first stroke of its piston to the last was measured in the tanks, and this weight was the quantity vaporized by the anthracite and last wood-fire during the day. All the anthracite consumed was accurately weighed on a delicate steelyard, and all the refuse from it in ash, clinker, etc., 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 kept in operation until it stopped for want of steam, when the contents of the furnace were drawn and the unconsumed coal picked out, weighed, and deducted from the quantity of anthracite expended.

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 condenser, the height of the barometer, the temperatures of the air on deck, of the air in the engine and fire room, of the injection or bay water, and of the feed-water in the tanks. 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 pipe of large bore. These instruments gave satisfactory diagrams, the means of all which are recorded in Table No. 1.

The experiments made on the 14th, 15th, and 16th of June were conducted in precisely the same manner as those above described, except that instead of double runs being made consecutively over a base, the vessel was steamed directly onward for a given time; the number on the counter and on the feed-water tally-board being simultaneously noted at the beginning and end of the time. These experiments were made partly in Narragansett Bay and partly in Long Island Sound.

The water was smooth, and the wind never exceeded a gentle breeze except in Experiment G, light airs to light breezes being the average; but, as the effects of breeze and tide, for and against, were not neutralized, the law of the proportionality of the net pressure on the pistons to the square root of the number of double strokes made by the pistons per minute cannot be confidently applied.

EXPLANATION OF TABLE NO. 1.

In Table No. 1 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.

Line 1 contains the dates of the experiments; these latter do not succeed each other in the order of time, but are arranged according to the measure of expansion with which the steam was intended to be used in them combined with the point of cutting off in the large cylinder.

TOTALS.—Line 2 contains the duration of each experiment in hours and decimals of an hour.

Line 3 contains the number of doubles trokes made by the pistons of the engine during the time on line 2.

Line 4 contains the number of pounds of feed-water pumped into the boiler during the time on line 2. This is the weight given by the tank measurement corrected for differences of temperature.

ENGINE.—Line 5 contains the mean steam pressure in the "separator" of the boiler, in pounds per square inch above the atmosphere, during the time on line 2, deduced from observations at ten minutes intervals of a spring gauge.

Line 6 contains the mean steam pressure in the "receiver" between the small and large cylinders, in pounds per square inch above the atmosphere, during the time on line 2, deduced from observations at ten minutes intervals of a spring gauge.

Line 7 contains the mean fraction completed of the stroke of the piston of the small cylinder when the steam was cut off. And line 8 contains similarly the mean fraction completed of the stroke of the piston of the large cylinder when its cut-off valve closed. These quantities are the means of all the indicator diagrams taken, and were obtained by measurement on each diagram, the point at which the curve of throttling reverses into the curve of expansion being taken for the point of cutting off.

Line 9 contains the number of times the steam was expanded, calculated as follows:

Let a = the number of cubic feet from the face of the *cut-off* valve to the point of cutting off in the small cylinder.

Let b = the number of cubic feet from the face of the *steam* valve to the end of the stroke of the piston in the small cylinder.

Let c = the number of cubic feet from the face of the *steam* valve to the end of the stroke of the piston in the large cylinder.

And, let E = the measure of expansion with which the steam is used, or the number of times that the steam is expanded. Then

$$E = \frac{b}{a} \times \frac{c}{b}.$$

This equation would give accurately the number of times the steam was expanded, provided that no part of it were condensed in the cylinders and that its temperature remained constant. The equation really gives less than the measure of expansion for only such portion of the steam as remains in that state from its entrance into the small cylinder until its discharge from the large cylinder, and more than the measure of expansion for such portion of the steam condensed in the cylinders as may be re-evaporated there before the end of the stroke of the piston of the large cylinder. A comparison of the total pressures above zero at the point of cutting off the steam in the small cylinder and at the end of the stroke of the piston of the large cylinder will not give the true measure of expansion with which the steam is used. By the number of times the steam is expanded is meant the number of cubic feet which are finally occupied by a cubic foot of steam after its expansion, its temperature remaining constant throughout the expansion. The comparison is between bulks. Now, as the pressure of a given quantity of steam is not inversely as its bulk, the comparison by pressure cannot be *directly* substituted for the comparison by bulk. By correcting the pressures so as to make them represent bulks at constant temperature, they might be employed to calculate the measure of expansion, provided there was no condensation in the cylinders and partial re-evaporation there of that condensation. It is thus seen that when there is cylinder condensation the measure of expansion with which the steam was used can be given only approximately; the less the condensation the closer the approximation. Only in the case where the steam has been sufficiently superheated to prevent condensation in the cylinder can its true measure of expansion be given.

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 condenser in inches of mercury, obtained from observations made every ten minutes on a spring gauge. 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 condenser (line 12).

Line 12 shows the back pressure in the condenser in pounds per square inch above zero. It is the difference between the quantities on lines 10 and 11, expressed in pounds per square inch, and is a gaseous mixture of steam and air in unknown proportions.

Line 13 gives the number of double strokes made by the pistons of the engine per minute, or of revolutions made by the screw per minute. These quantities are the numbers on line 3 divided by the number of minutes in the time on line 2.

Line 14 contains the number of pounds of feed-water pumped into the boiler per hour. These quantities are the quotients of those on line 4 divided by the numbers on line 2.

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 at the temperature on line 19 and in a pound weight of steam of the boiler pressure on line 5.

TEMPERATURES.—All the temperatures are given in Fahrenheit degrees, and were taken by ordinary mercurial thermometers permanently suspended, the observations being noted every ten minutes. On line 16 is the temperature of the external air in the shade. On line 17 is the temperature of the air in the room containing the engine and boiler. On line 18 is the temperature of the sea water used for the injection or refrigerating water in the condenser. And on line 19 is the temperature of the feed water in the tanks.

SPEED.—The vessel's speed was ascertained during only experiments A, D, I, J, and L; it is expressed on line 20 in statute miles of 5,280 feet per hour, and on line 21 in geographical miles or knots of 6,086 feet per hour. The former were the observed quantities, and the latter were deduced from them. The first measure is employed by landsmen, yachtmen, and for steamers on inland waters; the last is employed by seafaring persons. The speed in all cases is the mean from two consecutive runs over the base of 12 statute miles.

Line 22 contains the slip of the screw in per centum of its axial speed the latter being computed from the product of the pitch of the screw into the number of its revolutions 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 22.

STEAM PRESSURES IN THE SMALL CYLINDER PER INDICATOR.—The quantities on lines 23 to 30, both inclusive, are the means from all the indicator diagrams taken from the small cylinder. These diagrams were taken every ten minutes. Lines 23, 24, and 25 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 26 contains the mean back pressure against the piston of the small cylinder during its stroke (there was no cushioning), and line 27 contains the back pressure against the piston of the small cylinder at the beginning of its stroke; both quantities being expressed in pounds per square inch above zero. The mean back pressure is employed to obtain the total pressure on the piston of the small cylinder (line 30) by adding it to the indicated pressure (line 28). The back pressure at the commencement of the stroke of the piston is required for calculating the weight of steam remaining in the clearance and steam passage at one end of the small cylinder when the exhaust closes, which weight has not to be drawn from the boiler for the succeeding stroke of the piston. Line 28 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 29 contains the net pressure on the piston of the small cylinder in pounds per square inch of its area, and is what remains of the quantities on line 28 after the subtraction of 2 pounds per square inch, as the pressure required to work the small cylinder and its concomitants, *per se*, or unloaded. Line 30 contains the total pressure on the piston of the small cylinder in pounds per square inch of its area above zero; these quantities are the sum of those on lines 26 and 28.

All the pressures in the small cylinder being above the atmospheric pressure, may be taken as composed of nearly pure steam, the air admixture being therefore very small proportionally, none leaking in.

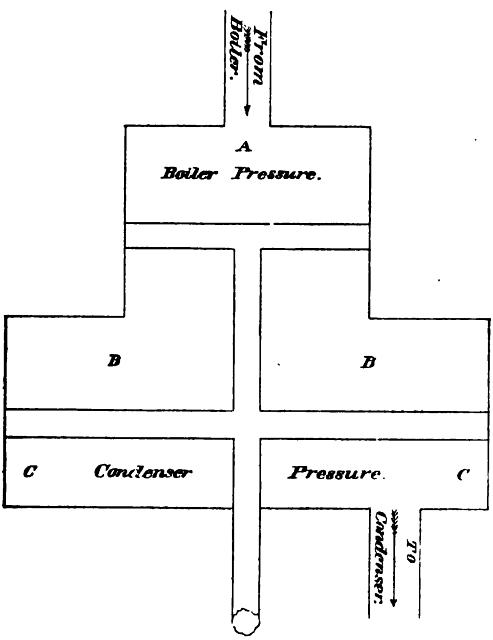
STEAM PRESSURES IN THE LARGE CYLINDER PER INDICATOR.—The quantities on lines 31 to 38, both inclusive, are the means from all the indicator diagrams taken from the large cylinder. These diagrams were taken every ten minutes. Lines 31, 32, and 33, contain, respectively, the pressure on the piston of the 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 of its area above the zero of pressure. Line 34 contains the mean back pressure against the large piston during its stroke, and line 35 contains the back pressure against the large piston at the beginning of the stroke, both quantities being in pounds per square inch of the piston's area above zero. The mean back pressure is employed to obtain the total pressure (line 38) on the piston of the large cylinder by adding it to the indicated pressure (line 36). The back pressure at the commencement of the stroke of the piston of the large cylinder is required to calculate the weight of steam remaining in the clearance and steam passage at one end of that cylinder when the exhaust closes. Line 36 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 37 contains the net pressure on the piston of the large cylinder in pounds per square inch of its area, and is what remains of the quantities on line 36 after the subtraction of 2 pounds per square inch as the pressure required to work the large cylinder, *per se*, or unloaded. Line 38 contains the pressure in pounds per square inch above zero of the annular space or ring remaining of the piston of the large cylinder after the piston of the small cylinder has been subtracted from it; these quantities are the sum of those on lines 34 and 36. Of course, the pressure on line 38 is upon every square inch of the piston of the large cylinder, but for the purpose of calculating the total horses-power developed in that cylinder (line 46), only the annular space just described is employed, as will be hereinafter described.

All the pressures in the large cylinder which are below the atmospheric pressure, must be more or less composed of air mixed with the steam, the air entering by piston-rod stuffing-box and defective joints in unknown quantities.

HORSES-POWER.—Lines 39 to 50, both inclusive, contain the various kinds of horses-power developed by the engine, that are necessary to be known in order to understand the manner in which the total power of the engine is distributed between its two cylinders and for what purposes.

Lines 39 and 40 contain, respectively, the indicated horses-power developed in the small and in the large cylinders. These powers are calculated from the areas of the pistons of the small and large cylinders, from the speed of these pistons in feet per minute, and from the pressures on lines 28 and 36. On line 41 are the indicated horses-power developed by the engine, being the sum of the quantities on lines 39 and 40.

In calculating the indicated horses-power for a compound engine, the common practice—followed above—is to take for factors the area of each piston and the indicated pressure upon it per square inch as given by the indicator diagrams from the respective cylinders. Thus, the indicated horses-power is obtained for each cylinder and their sum is the indicated horses-power developed by the engine. In fact, however, the indicated horses-power developed by the engine are not thus distributed although the aggregate is correct. This method assumes that each cylinder of the compound engine acts independently of the other, as is the case with the simple engine, which is not true. The compound engine is necessarily composed of two cylinders of unequal capacity, while the simple engine is composed of one cylinder only, and if two be used they constitute two independant engines, although coupled to the same shaft and having boiler and condenser in common. The combined cylinders of the compound engine are, in effect, one cylinder only, and their pistons are, in effect, one piston only, nor can their action be separated in reality, however it may be in appearance, by the arrangement of mechanical details adopted. The interiors of both cylinders of the compound engine are always in common, that is, the spaces in the two cylinders between the inner surfaces of their pistons are in common for one stroke, and those between the outer surfaces of their pistons are in common for the return stroke. The accompanying very simple diagram, which exhibits the compound engine in its most elementary conception, will illustrate these ideas. The diagram shows the two cylinders of the compound engine arranged tandemwise with their interiors between the inner surfaces of their pistons in common, and with their pistons rigidly connected to the same piston-rod. With this arrangement the pistons could make only one stroke downward, which is all that is necessary for explanation. To make the return upward stroke, the two cylinders would have to be separated by a partition and the interior of the small cylinder above its piston would have to be made in common with the interior of the large cylinder below its piston by a communicating pipe. In a working engine the valves controlling the alternate flowings of the steam from the bottom of the small cylinder to the top of the large one and from the top of the small cylinder to the bottom of the large one are the movable partitions supposed in the diagram. The space A is supposed to be filled with steam of the boiler pressure; the space BB is supposed to be filled with steam of the boiler pressure expanded with pressures decreasing according to



approximately a hyperbolic curve, from the beginning to the end of the stroke of the pistons, the terminal pressure being as much less than the initial pressure as the capacity of the large cylinder is more than that of the small one; and the space C C is supposed to be filled with steam of the condenser pressure.

Now, evidently, the pressure beneath the piston of the small cylinder and above the piston of the large cylinder being the same, and the two pistons being rigidly connected, this pressure is entirely neutralized against the piston of the small cylinder, and upon so much of the piston of the large cylinder as is immediately opposite and equal to the area of the piston of the small cylinder; hence, the only portion of the piston of the large cylinder that can produce power is the ring remaining after the subtraction of the piston of the small cylinder. As a consequence, the indicated power exerted by the expanding steam in the large cylinder is represented by the indicated pressure there multiplied by the area of the ring remaining after subtracting the piston of the small cylinder from the piston of the large one. This is the true indicated power developed by the large cylinder, and is as much less than the indicated power as habitually calculated for that cylinder, as the area of the ring referred to is less than the area of the piston of the large cylinder. Of course, the true net power and the true total power developed in the large cylinder are represented by the product of the ring area referred to into, respectively, the net pressure and the total pressure, the net pressure being what remains of the indicated pressure after subtraction of the pressure required to work the large cylinder and its concomitant parts, *per se*, or unloaded, and the total pressure being the sum of the indicated pressure and the back pressure against the piston of the large cylinder.

The two pistons of the compound engine being rigidly connected, and the steam pressure between them being the same per square inch of piston, the true back pressure acting against the piston of the small cylinder is evidently the back pressure acting against the piston of the large cylinder; that is to say, the condenser pressure in the space CC, and not the pressure of the expanding steam in the space BB between the pistons; hence, to obtain the true indicator diagram from the small cylinder, the indicator must be placed in connection with the space A or steam side of the piston of the small cylinder and with the space CC or exhaust side of the piston of the large cylinder. The indicated pressure for the small cylinder thus obtained would be as much greater than the indicated pressure habitually calculated for that cylinder, as is due to the mean pressure of the expanding steam in the space BB between the pistons of the two cylinders. Now, this pressure of the expanding steam being what is habitually known as the indicated pressure for the large cylinder, the true method of calculation herein described makes the indicated power of the small cylinder exactly as much greater than what is obtained by the habitual method of calculation as the indicated power of the large cylinder is made smaller, the aggregate indicated power of both cylinders being the same by both methods. The true net pressure for the small cylinder is what remains of the true indicated pressure after subtraction of the pressure required to work the small cylinder and concomitant parts, *per se*, or unloaded, and the true total pressure for the small cylinder is the sum of the true indicated pressure and of the back pressure in the space CC against the large piston.

The indicator diagrams habitually taken from the small cylinder of the compound engine do not correctly represent the effective pressures upon its piston, nor does the habitual co-relation of the indicator diagrams habitually taken from the two cylinders give a correct idea of the true distribution of the pressures. The true co-relation of the diagrams is as follows:

Suppose a straight line taken as base and divided into two parts which have the same relation to each other that the volume of the small cylinder has to the volume of the large cylinder less the volume of the small cylinder. Now, having taken a true indicator diagram from the small cylinder, that is, a diagram made by putting the indicator alternately in connection with the space A or steam side of the piston of the small cylinder, and with the space CC or exhaust side of the piston of the large cylinder, lay off this diagram on the first part of the base representing the volume of the small cylinder. Next, having taken an indicator diagram from the large cylinder by putting the indicator alternately in connection with the space BB or steam side of the piston of the large cylinder, and with the space CC or exhaust side of it, lay off this diagram on the last part of the base representing the volume of the large cylinder less that of the small one; then the two diagrams will form one showing the true action of the steam and the true distribution of its pressure during a stroke of the *combined* pistons, the total length of the base representing the length of the stroke of a piston. Such a diagram, and only such an one, is comparable with the diagrams taken from the cylinder of the simple engine. All other arrangements or combinations of diagrams from the cylinders of compound engines are factitious and misleading.

In the diagram, which has the partition between the two cylinders omitted, the pressure in the space BB must be uniform throughout at any instant, making the back pressure per square inch against the piston of the small cylinder exactly equal to the pressure on the piston of the large cylinder per square inch; but, in a working engine, where the large movable partition between the cylinders is replaced by comparatively a very small valve with contracted openings, this equality of pressure upon the two pistons no longer exists, and the back pressure against the piston of the small cylinder is always greater than the pressure upon the piston of the large cylinder. This difference of pressure in practice varies from 1½ to 3 pounds per square inch, according to the size of the opening in the valve relatively to the capacity of the small cylinder, to the length and tortuousness of the passages connecting the two cylinders, to the speed of the pistons, &c. Now, this difference of pressure, whatever it may be, and it is always something of practical importance, is additional back pressure against the piston of the small cylinder, and must be added to the back pressure in the space CC to obtain the true back pressure against which that piston works. Thus, the piston of the small cylinder of the compound engine must always work practically though not theoretically against a greater back pressure than the surface of the ring works which remains of the piston of the large cylinder after subtraction of the piston of the small cylinder, by the difference between the pressure per square inch on the opposite sides of the movable partition separating the two cylinders. This additional back pressure is a loss of useful effect peculiar to the compound engine and has no analogue in the simple engine.

The foregoing principles governing the compound engine suffer no modification by any variation of the relative position of the cylinders or addition of parts. The interposition of a receiver between the cylinders, whereby their pistons need not come simultaneously to the ends of their strokes, does not in the slightest degree affect the mode of action described; neither does cutting off the steam in the small cylinder, whereby part of the expansion is caused to take place in that cylinder.

All the steam used in the large cylinder is always expanded steam; nor is the measure of expansion affected by placing a cut-off valve on that cylinder closing at any point of the stroke of its piston. The object of such a valve is to increase the pressure between the piston of the small cylinder and the movable partition separating the cylinders, that is to say, between that piston and the cylinder end towards which it is moving. Such an increase of pressure would be nearly useless, theoretically, in a tandem arrangement of cylinders for the compound engine, which arrangement supposes no space between the cylinders to be filled with steam in its transfer from the small to the large cylinder, except that in the connecting pipes; but just in proportion as space exists between the cylinders, as in the case of a receiver arrangement of them, is the necessity for a cut-off valve on the large cylinder. In some tandem arrangement of cylinders the steam from the small one is exhausted into the jacket of the large one, passing thence into the large cylinder, the jacket in which case forms a receiver; and a cut-off valve, therefore, becomes neeessary on the large cylinder. Such a use of the jacket, however, is very injudicious and should never be resorted to. The large cylinder should receive its steam directly from the small one, and the jackets of both should be kept filled with steam of the boiler pressure. With a skilful tandem arrangement of cylinders, whereby the space between them is reduced to the minimum, but very little is to be gained by the application of a cut-off to the large cylinder; such a valve could not be closed judiciously until the stroke of the piston was at least two-thirds completed. In a theoretical tandem arrangement of the cylinders, that is, with no space between them, and without a cut-off valve on the large cylinder, the initial pressure in the large cylinder would be the same as the final pressure in the small cylinder, which is the effect sought to be produced by the application of a cut-off valve to the large cylinder when a receiver intervenes between it and the small cylinder, in which case the cut-off valve on the large cylinder must close earlier and earlier as the cut-off valve on the small cylinder closes later and later. The receiver arrangement of cylinders, therefore, compels, for maximum economic effect as regards pressure alone, the steam to be cut off very early in the small cylinder, so that a large portion of its expansion may be done there, the remainder being done in the large cylinder whose cut-off valve should be so adjusted as to make the initial pressure in that cylinder the same as the final pressure in the small cylinder. The object of the short cutting off in the small cylinder is to enable it to develop, under the preceding condition, a proper power, which it could not do with a late cut-off and the cut-off valve on the large cylinder closing at a point that would make equality of pressure at the end of the stroke of the piston of the small cylinder and at the beginning of the stroke of the piston of the large cylinder. Consequently, in receiver arrangements of cylinders—the steam being used with a given measure of expansion—the ratio of the volumes of the small and large cylinders is much less than *need be* in tandem arrangements. Of course, the steam can be cut off properly as short in the small cylinder ot the tandem arrangement as in the case of the receiver arrangement, but it need not be for maximum economic effect as regards pressure alone; in fact it can be worked without expansion in the small cylinder and the entire expansion done in the large one without sacrifice of economic effect due to pressure distribution, provided there be no space between the cylinders, in which case neither of them would require a cut-off valve, and the mechanism could be much simplified.

In the receiver arrangement of the two cylinders, however, a cut-off upon both is indispensable for maximum economic effect as regards pressure alone. Neither advantage nor necessity requires the receiver to be of greater capacity than the small cylinder, and the pressure in the receiver must be kept the same as the final pressure in the small cylinder, which can only be done by the use of an adjustable cut-off valve on the large cylinder closing at the proper point to produce this effect. Further, this condition prevents the steam from being used without expansion in the small cylinder, as in that case the small cylinder could not develop any power. In order, therefore, that the small cylinder in the receiver arrangement should develop a proper power with the condition of equality between its final pressure and the pressure in the receiver, the steam must be cut off in it considerably before the end of the stroke of its piston. Thus, with the receiver arrangement of the two cylinders, cut-off valves on both are indispensable for maximum economic effect as regards distribution of pressure, and the steam must be used with considerable expansion in the small cylinder to obtain the proper division of power between the cylinders. The manner in which these conditions affect the proportion of the two cylinders is very obvious. If the steam is to be used with a given measure of expansion, say six times, then in a tandem arrangement of the cylinders without cut-off valve on either, the volumes of the two cylinders would be as 1 and 6; but in the case of the receiver arrangement of the two cylinders, cutting off the steam in the small cylinder when one-third of the stroke of its piston was completed, the volumes of the cylinders would compare as 1 and 2, and if the large cylinder were divided into two so as to make a three-cylinder compound engine, all the cylinders could have the same dimensions, a very great mechanical convenience when the engine is a large one.

When the steam is worked expansively in the small cylinder of the compound engine, the application of a cut-off valve to the large cylinder is a source of much economy independently of the proper distribution of the pressure. In all cases of using saturated steam, considerable condensation is found in the cylinders, due to the alternate heating and cooling of the metal of which they are made, owing to the considerable difference of temperature to which the metal is exposed during a double stroke of the piston. This liquefaction of the steam is a minimum when the steam is worked without expansion, other things equal, and increases *pari passu* as it is worked more and more expansively. Now, whatever portion of the steam entering the small cylinder remains liquefied at the end of the steam stroke of its piston, is boiled off or revaporized during the return exhaust stroke under the lessened pressure of the exhaust by the contained heat in the water and in the metal of the cylinder, and this revaporized portion passes to the large cylinder and is used upon its piston without expansion if that cylinder have no cut-off valve; but if it have, then this revaporized portion is worked expansively, and is thus made to give a higher duty. Of course, the less the cylinder liquefaction of steam, the less will be the economic gain due to this cause, an *vice versa*.

The foregoing views of the true theory of the compound engine and of its mode of action, of the advantages of cut-off valves on both the small and the large cylinder, and of the arrangement in proper juxtaposition of the indicator diagrams taken from the two cylinders, are original with the writer and are believed to be novel. They are given with as much brevity as possible, the present purpose being to offer only so much explanation as will make clear the methods of calculation employed in this report; but many additional issues of interest and advantage follow as obvious corollaries from the main propositions herein demonstrated.

Returning to Table No. 1, lines 42 and 43 contain the net horses-power developed respectively in the small and large cylinders of the "Siesta's" engine. These powers are calculated from the speed of the piston, from the entire areas of the cylinders, and from the pressures on lines 29 and 37. On line 44 is the net horses-power developed by the engine, being the sum of the quantities on lines 42 and 43.

Line 45 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 30. This is the power overcoming all resistances to the piston of the small cylinder down to the zero of pressure, and includes internal and external work of all kinds.

Line 46 contains the total horses-power developed in the large cylinder, calculated from the speed of piston, the area of the annular superficies remaining after the subtraction of the area of the piston of the small cylinder from the area of the piston of the large cylinder, and the pressure on line 38. In further explanation of what has already been discussed, there may be remarked that in the compound engine the back pressure (line 26) 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 (line 38) equal to the sum of the indicated and 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.

Line 47 contains the total horses-power developed by the engine. These quantities are the sum of those on lines 45 and 46, and represent the entire dynamic effect of the steam, both useful and prejudicial.

Line 48 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 comprised between the expansion curve and the straight line of no pressure or zero. The power on line 48 is necessary to be known in order to calculate the weight of steam (line 59) 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 piston of the small cylinder (line 58) there may be obtained the weight of steam accounted for at that point by the indicator (line 60), which weight being deducted from the weight of steam evaporated in the boiler (line 14) gives a difference (line 66) 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 23 to those due to the pressure on line 27.

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 49 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 46, because all the steam used in the large cylinder is expanded steam. The thermal equivalent of the powers on line 49, in pounds of steam condensed per hour to furnish the necessary heat, added to the quantities on line 59, forms the quantities on line 62, which added to those on line 61 give the weight of steam accounted for by the indicator at the end of the stroke of the piston of the large cylinder (line 63). This latter weight being deducted from the weight of steam evaporated in the boiler (line 14) gives a difference (line 68) 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 temperature.

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 foot-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 $789\frac{1}{4}$ foot-pounds, and the time T, then, $\dfrac{P \times \frac{33000}{789\frac{1}{4}} \times T}{H}$ = the number of pounds of steam condensed in the cylinder to furnish the heat transmuted into the power P exerted during the time T.

That the latent heat in the above equation should be what is due to the *mean* pressure of the expanding steam is thus derived: At each instant of time the expanding steam develops a power at the expense of the heat in it, and this heat cannot be taken out of the steam below its temperature because its temperature is the boiling-point of water under the insistent pressure; in other words, only the latent heat can be taken out of the steam from instant to instant successively, so that the latent heat to be used in the above equation is the mean of all the latent heats due to all the pressures of the expanding steam. The effect, therefore, of the transmutation of part of the heat of the expanding steam into the power developed by the engine during that expansion, is to produce a weight of water equal to the weight of steam condensed and having a temperature normal to the pressure of the steam from which it was condensed. Now, as the pressures of the expanding steam continue to fall with the expansion, the temperature of the successive waters of condensation continues to fall with them, *pari passu*, and these waters become partly revaporized by the heat in them between their original temperature of condensation and their final temperature, which latter is what is normal to the pressure of the steam at the end of its expansion, that is, at the end of the stroke of the piston. The portion of the waters of condensation thus revaporized will appear as steam at the end of the stroke of the piston and will there be directly measured by the indicator. Thus the expansion curve receives continuously increase of pressure from its commencement to its end, due to the continuous revaporization of part of the waters of condensation caused by the transmutation of part of the heat of the expanding steam into the work done by its expansion. If the latent heat of the steam due to its pressure at the end of the stroke of the piston were taken for H in the foregoing equation, then because the latent heat increases as the pressure decreases, the weight of steam condensed according to the equation would be too small by the difference due to the larger value of H as a divisor when taken normal to the pressure at the end of the stroke of the piston, than when taken normal to the mean pressure of the steam during its expansion. And, as the weight of steam calculated from the pressure at the end of the stroke of the piston would be the same in both cases, the weight of steam accounted for by the indicator at the end of the stroke of the piston would be too small.

The mechanical equivalent of one Fahrenheit unit of heat is taken in the above equation to be 789¼ foot-pounds. It is habitually but erroneously taken at 772 foot-pounds, as given by Joule; for the later experimental researches of H. Tresca and C. Laboulaye, to determine the true mechanical equivalent of one calorie, indorsed by Regnault, Piobert, Combes, Bertrand, and Morin, a commission appointed by the French Academy to investigate and report upon the subject, give 433 kilogrammetres as the mechanical equivalent of one calorie, which is 789.2385 foot-pounds for one Fahrenheit unit of heat. ("Comptes Rendus" of February 13, 1865, page 326.) In reaching this final determination by experiments on air, the experimenters took into consideration every possible correction and used the best methods which the combined science of the time could devise; the apparatus, furnished by the French Government, was on a large scale and as perfect in its design and adjustment as mechanical skill could achieve.

Line 50 contains the total horses-power developed in the cylinders by the expanded steam alone. These quantities are the sum of those on lines 48 and 49. By comparing them with the quantities on line 47, the portions of the total power developed by the engine before and after the closing of the cut-off valve on the small cylinder can be obtained.

ECONOMIC RESULTS.—Lines 51, 52, and 53 give respectively the cost of the indicated, net, and total horse-power developed by the engine, in pounds of feed-water vaporized per hour in the boiler. The quantities on these lines would correctly represent the cost of the powers, provided the boiler pressure and the temperature of the feed-water remained always the same; but as these conditions varied, and as the total heat of vaporization varies with the boiler pressure and with the temperature of the feed-water, the quantities on lines 54, 55, and 56 are added, which give respectively the exact experimental cost in Fahrenheit units of heat per hour of the indicated, net, and total horse-power developed by the engine.

The division of the quantities on line 14 by those respectively on lines 41, 44, and 47, gives the quantities on lines 51, 52, and 53; and the division of the quantities on line 15 by those respectively on lines 41, 44, and 47, gives the quantities on lines 54, 55, and 56.

If the cost of the horse-power in pounds of ordinary anthracite consumed per hour be required, it can be obtained by dividing the cost of the power in pounds of feed-water consumed per hour, by the pounds of the latter vaporized per pound of anthracite. The economic vaporization of the pound of anthracite consumed may be taken under the experimental conditions to average 8.33 pounds of water; hence, the division of the quantities on lines 51, 52, and 53 by 8.33 will give respectively the cost of the indicated, net, and total horse-power in pounds of anthracite consumed per hour. For example, the cost of the indicated horse-power in Experiment D being 16.74107 pounds of feed-water vaporized per hour, its cost in anthracite consumed per hour was $\left(\frac{16.74107}{8.33}=\right)$ 2.00973 pounds. In the cases of the experiments, the cost of the horse-power in pounds of anthracite consumed per hour by direct weighing of the latter could not be given because they were of too short duration to allow the quantity burned to be accurately ascertained, several experiments being made on one day without interval between them. For exact comparisons, under any circumstances, 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 57 to 63, 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 or liquefied in the cylinders 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 cylinders after the closing of the cut-off valve, can be calculated when the power developed by the expanding steam is known, and consequently the foot-pounds of work done by the expanding steam are known, the horse-power being 33,000 foot-pounds of work conceived to be done in one minute, 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 that steam, which abstraction is attended by the liquefaction of such a 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 the 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 57 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 space 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 24; 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 27.

Line 58 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 25, subtracting from the last product the 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 27.

Line 59 shows the weight of steam in pounds condensed or liquefied 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 (line 48). By this 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. The calculation of the quantities on line 59 is made as follows: The mechanical equivalent of one Fahrenheit unit of heat being taken at 789¼ foot-pounds, then $\frac{33000}{789\frac{1}{4}} = 41.811847 =$ 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 developed by the expanding steam after the closing of the cut-off valve; $T =$ the time in minutes during which the power P operated—this time was 60 minutes; $H =$ the latent heat of steam of the mean pressure above zero of the expanding steam after the closing of the cut-off valve; and $\frac{41.811847 \times P \times T \text{ or } 60}{H} =$ the number of pounds of steam condensed in the cylinder to furnish the heat transmuted into the total horses-power developed by the expanding steam.

Line 60 contains the sum of the quantities on lines 58 and 59. 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 61 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 33, 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 35.

Line 62 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 of its own.

The number of pounds of steam condensed in the large cylinder to furnish the heat transmuted into the total horses-power on line 49, 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 49 is calculated is that on line 38, and it acts during the entire stroke of the piston. The quantities thus obtained having been added to those on line 59, the sums are the quantities on line 62.

Line 63 contains the sum of the quantities on lines 61 and 62. 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 IN THE CYLINDER BY THE INDICATOR.—The quantities on lines 64 to 69, both inclusive, show for each experiment the difference between the weight of water vaporized in the boiler and the weight of steam accounted for in the cylinder by the indicator. Were there no cylinder condensation due to other causes than the production of the power, these differences would not exist, supposing the absence of priming or foaming in the boiler; 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 heat from the steam and gives out heat to the resulting water of condensation, the former quantity of heat appearing to be greater than the latter as shown by the indicator, but equal in fact when the heat is included which revaporizes that portion of the water of condensation which is present in the cylinder at the end of the stroke of its piston when the exhaust valve opens, the resulting steam of which passes to the condenser during the exhaust stroke of the piston, and thus escapes detection by the indicator. This revaporization is due to the contained heat in the water of condensation and to the heat of the metal of the cylinder over which this water is spread, and to the less pressure in the condenser than in the cylinder, the interiors of these two vessels being in common as long as the exhaust port 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 (and likewise in the receiver of a compound engine whose large cylinder is fitted with a cut-off valve) when the inrushing steam comes to a 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 (or receiver). But, as regards the cylinder, such heat is lost, and escapes detection by the indicator.

Furthermore, 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 the 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 measurable by the indicator.

The water of condensation due to the transmutation of heat into work when expanding steam is employed as the heat carrier, be that work of what nature it may, internal or external, is not deposited upon the surface of the metal of the cylinder, but remains suspended in the steam and perfectly diffused thoughout the mass in the form of infinitesmal particles which give a cloudy or fog-like appearance to the expanded steam, whereas non-expanded steam, when employed as the heat carrier in doing work, remains transparent, no such water of condensation being present in it. The time of making a stroke of the piston is so short, and the watery particles of condensation are so excessively small and so intimately mixed with the very much greater mass of steam, that both are swept into the condenser together before any separation by gravity can take place.

The water of condensation due to the varying temperature of the metal of the cylinder during a double stroke of its piston, and composing by far the greater portion of the cylinder condensation, is mainly deposited previous to the closing of the cut-off valve, and is spread uniformly in the form of dew over the inner surfaces of the cylinder and its steam passages, whence it is revaporized partly during the expansion portion of the stroke of the piston and the remainder during the exhaust stroke, so that when the cylinder again takes steam these surfaces are dry and comparatively cool. Thus the cylinder acts alternately as a condenser and a boiler, but the reevaporated steam so obtained in the cylinder produces but little dynamic effect in comparison to its heat cost. It is used upon the piston almost without expansion, and its entire effect from the point of the stroke of the piston at which the reevaporation takes place back to the valve face is lost. Of all the components of loss in a steam engine, this cylinder condensation is the greatest; and the economic advantanges of steam jacketing and steam superheating are derivable directly from its prevention.

As regards the enormous influence on the economic production of the power exercised by the metal of which the cylinder is made, an answer may here be properly given to the questions: How, during the exceedingly short time required for a double stroke of the piston, even in the case of the slowest-working engines, can the transfer of so much heat be effected, first from the steam to the metal of the cylinder, whereby a considerable portion of the former is condensed, and then from the metal of the cylinder to the water of condensation, whereby the whole of the latter is revaporized, as is experimentally shown to be the fact? Indeed, does not the exceeding brevity of the time in which these effects are supposed to be produced furnish a sufficient negative to the possibility of such energetic action by the metal of the cylinder?

The answer is, that the weight of metal composing the cylinder is so much greater than the weight of steam or of water within it per single stroke of piston, and the extent of interior cylinder surface exposed in connection therewith is so large comparatively, that no difficulty need be experienced in accepting the fact of the rapidity with which the heat is transferred, notwithstanding that the specific heat of iron at the ordinary temperatures of working cylinders is about 0.115, while that of water at the same temperatures is about 1.015. The heat conductibility of iron is about forty times greater than that of water, and its density about 7.2 times greater.

Taking for illustration the case of the small cylinder in Experiment L, in which the maximum weight of feed-water per hour was vaporized in the boiler, the weight of steam entering the cylinder per single stroke of piston was only 0.12825 pound. The interior surface of the cylinder was 1,200 square inches, and the weight of cast iron in direct connection therewith was about 220 pounds; consequently, not only did the weight of iron exceed the weight of steam 1,716 times, but the latter if reduced to water and spread evenly over the former would form a coating only

0.0031 inch thick. So far from there being any difficulty in conceiving that a considerable percentage of this weight of steam could be condensed to water by the metal of the cylinder having a disposable cooling power represented by its weight, its specific heat, and the difference of temperature in the cylinder during a double stroke of its piston acting through the comparatively large extent of surface within the cylinder, that the real difficulty would be in conceiving the reverse. Besides which, experiments have shown that pure saturated steam in contact with a cooler metallic body condenses almost instantaneously until heat enough is transferred to equalize the temperature of the two bodies. The same considerations show that the water of condensation thus deposited upon the inner surfaces of the cylinder will be revaporized therefrom with exceeding rapidity when the pressure under which it was deposited is reduced, by the disposable heat of the metal of the cylinder and by the contained heat of the water of condensation, the latter being measured by the difference between the total heats of vaporization due to the difference of pressure. The cylinder acts as a condenser during the period of steam admission, as a condenser and boiler during the period of expansion, and as a boiler during the period of the exhaust.

Even in the case of air engines, where the heat carrier is a nonliquefiable gas, instead of, as in the steam engine, an extremely liquefiable vapor, the influence of the metal of the cylinder on the economic performance is so strongly marked that to this cause is to be largely attributed the small percentage practically obtained of the magnificent results theoretically predicable.

The foregoing calculations regarding the horses-power developed by the cylinders, and the weight of steam accounted for by the indicator, cannot be strictly correct, though they are an exceedingly close approximation to the truth, for want of correction for the temperature of the metal of the cylinders in the first case, and for that temperature and the bulk of water of condensation deposited upon the interior surfaces of the cylinders in the last case.

The dimensions of the cylinders, that is, of their diameters and stroke of piston, are from measurement with the metal at ordinary atmospheric temperatures, but, when in use, the temperature is much greater, depending upon the initial steam pressure, point of cutting off, back pressure, cooling by revaporization of the cylinder condensation, &c., and, as it is impossible in any case to know what the increase of the temperature is, it is impossible to apply the proper correction for it.

The diameter of the cylinder is certainly greater when in use than when at the ordinary atmospheric temperatures, so that if account of only its increase of cross area due to increase of temperature be taken, the horses-power should be proportionally larger. Further, not only is the temperature of the cylinder greater when in use than when at ordinary atmospheric temperatures, with corresponding increase of cross area, but the stroke of the piston is also lengthened, due to the fact that the cranks are at a higher temperature when in use than the ordinary atmospheric temperatures. This increase of temperatures in the cranks is caused by their being heated by the adjacent crank-pin and crank-shaft journals, which were first heated by the friction due to the stress upon them; additionally, too, both cylinders and cranks are in air when in use of considerably higher temperature than ordinary atmospheric temperatures. Thus, a further addition should be made to the power for increased length of stroke to the piston. On the other hand, however, the capacity of the cylinders has been lessened by the bulk of the water of condensation deposited upon their inner surfaces, and this lessening tends to compensate their increase of capacity due to increase of temperature, and may wholly compensate it. The two corrections being in exactly opposite directions, must partially and may wholly neutralize each other; any difference must be very small and without practical value.

The same considerations apply to the corrections of the weight of steam accounted for by the indicator. One factor of that weight is the capacity of the cylinder which enlarges with increase of temperature, but which is taken in the calculations at the dimensions under ordinary atmospheric temperatures. Hence the weight of steam, calculated from the cold capacity of the cylinder as present in it, would be too small by the difference between that capacity cold and hot; but then the hot capacity is reduced by the bulk of the water of condensation deposited upon the inner surfaces of the cylinder when in use, and by the water of condensation due to the transmutation of heat into work and suspended in the steam. These two corrections are in exactly opposite directions and may entirely neutralize each other. They certainly do so to such an extent that any remaining difference, if there be any, must be too insignificant to be included in an experimental investigation. For illustration, suppose one-third of the weight of steam entering the cylinder to be condensed there, and suppose the bulk of the water of this condensation to be $\frac{1}{600}$ of that of the steam condensed to form it, then the bulk of the water of condensation in the cylinder will be ($\frac{1}{3} \times \frac{1}{600} =$) $\frac{1}{1800}$ of the capacity of the cylinder. A deduction of $\frac{1}{1800}$ would, in the supposed case, be the correction to be made were there no compensation by the increase in the cylinder's dimensions due to increase in the temperature. It is evident that when this small fraction is greatly reduced by the correction for increased capacity of cylinder owing to increased temperature, the remainder, at the largest possible, must be too small for practical consideration.

Returning to the table. Line 64 contains the difference between the quantities on lines 14 and 57. This difference shows the condensation in the small cylinder at the point of cutting off the steam in pounds of steam condensed therein per hour. Line 65 shows this difference in per centum of the quantity on line 14.

Line 66 contains the difference between the quantities on lines 14 and 60. This difference shows the condensation due to all other causes than the production of the power in the small cylinder at the end of the stroke of its piston, in pounds of steam condensed therein per hour. Line 67 shows this difference in per centum of the quantity on line 14.

Line 68 contains the difference between the quantities on lines 14 and 63. This difference shows the condensation due to all other causes than the production of the power, in the large cylinder and receiver, at the end of the stroke of the piston of the large cylinder, in pounds of steam condensed therein per hour. Line 69 shows this difference in per centum of the quantity on line 14, which is the weight of feed-water pumped into the boiler per hour.

Ratio of the Powers.—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 the 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 70 are the per centum which the indicated horses-power developed by the engine, line 41, are of the total horses-power, line 47.

The quantities on line 71 are the per centum which the net horses-power developed by the engine, line 44, are of the total horses-power, line 47.

CYLINDER PRESSURES REDUCED TO LARGE CYLINDER ALONE.—Lines 72 to 75, 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 72 to 75, both inclusive.

The quantities on line 72 are the quotients of those on line 28, divided by 2.962275 (the ratio of the areas of the pistons of the two cylinders), added to the quantities on line 36.

The quantities on line 73 are the quotients of those on line 29, divided by 2.962275, added to the quantities on line 37.

The quantities on line 74 are the quotients of those on line 30, divided by 2.962275, added to the quantities on line 38 reduced in the ratio of 1.962275. The last reduction is required because the total pressures on the piston of the small cylinder, line 30, 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 38 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 75 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 74 after deduction of those on line 72. 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 was of so great an area and so short a length that no increase of 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 any reasonable approximation 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, with equal vacuum in the condenser, equal terminal pressure in the cylinder, and equal speed of piston. 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 their sinuosities, of the exhaust openings, passages, pipes, &c., the back pressure against the piston exceeding more and more the pressure in the condenser, as these areas are smaller, these lengths greater, and these sinuosities more marked and frequent.

Supposing all other things to be the same, in a comparison of relative capacity of cylinders for the compound and for the simple engine, that is to say, the boiler pressure being the same and used with the same measure of expansion, the speed of piston being equal, the vacuum in the condenser being the same, and no pressure being lost in the transfer of the steam from the small cylinder to the large one of the compound engine, then, if the cylinder of the simple engine is a duplicate of the large cylinder of the compound engine, it will develop precisely the same power as the latter when working against the same resistance or load. The small cylinder of the compound engine is, under all circumstances, whether the steam be used expansively in it or without expansion, or with any measure of expansion, an addition to the cylinder capacity, other things equal, required with the simple engine; because the piston of the small cylinder of the compound engine, making stroke for stroke with the piston of the large cylinder, forces the steam contents of the small cylinder per stroke into the large cylinder, the latter receiving the former and expanding it into a space equal to the difference between the capacities of the two cylinders. Hence the *effective* cylinder capacity of the compound engine is the capacity of the small cylinder added to what remains of the capacity of the large cylinder after deducting from it the capacity of the small cylinder, the sum being equal to the capacity of the large cylinder. But, if there be a loss of pressure between the two cylinders, due to the transfer of the steam from the one to the other, this loss will enable the equivalent cylinder of the simple engine to be correspondingly reduced in capacity and still give equal development of power with the compound engine.

For commercial purposes the reduced capacity, as above obtained, for the cylinder of the simple engine can be still further slightly reduced, owing to the fact that the pressure required with the compound engine to work it *per se*, or unloaded, is greater than with the simple engine, which increased friction pressure is due to the fact that the small cylinder of the compound engine, and consequently the pressure required to overcome the friction of its packings and the weight of its moving parts, is wholly an addition to the friction pressure of the simple engine *per se*, or unloaded. The indicated pressure in all engines includes the pressure required to work the unloaded engine, but the net pressure excludes it, and as only the net pressure is applied to the crank-pins and does external or commercial work, the relative cylinder capacity of the two kinds of engine should be proportioned for the development of the same net power, other things equal, in which case, knowing the capacities of the cylinders of a compound engine and the net pressures upon their pistons, the equivalent cylinder capacity of a simple engine will be obtained as follows: Reduce the net pressure upon the piston of the small cylinder of the compound engine to its equivalent pressure upon the piston of the large cylinder by dividing it by the ratio of the capacities of the two cylinders, and add this equivalent to the net pressure upon the piston of the large cylinder, then multiply the resulting sum by the capacity of the large cylinder. The cylinder capacity for the simple engine will now be given by dividing the product just obtained by the net pressure upon the piston of the cylinder of the simple engine. The pressures referred to are always pressures per square inch. Practically, therefore, other things equal, the cylinder capacity of a simple engine will be less, and always in a marked degree, than the capacity of the large cylinder of the compound engine. And, when the steam is used very expansively, so that the mean total pressure per square inch is small, the back pressure and the friction pressure per square inch remaining the same for all total pressures, the difference in the cylinder capacity for the two kinds of engine developing the same power may become quite large.

The weight of the small cylinder of the compound engine, including that of its supports, its piston, piston-rod, cross-head, connecting-rod, valves, and all appurtenances, is additional to the weight required for the equivalent simple engine; but, the initial stress being less upon the crank-pins (the average stress remaining the same in both cases), the weight of the shaft of the compound engine can be proportionally reduced, as can also the weight of the piston-rod, cross-head, and connecting-rod of the large cylinder, and these reductions neutralize, as far as they go, the weight of its small cylinder. In general, however, the compound engine will be somewhat heavier as well as somewhat bulkier than the equivalent simple engine, and also somewhat more costly. The less initial stress upon the crank-pins of the compound engine, and consequently upon its crank-shaft journals, is a valuable mechanical advantage, as regards practical convenience, in its favor over the simple engine. Its economic advantage, whatever that may prove to be, when the steam pressure, saturated or superheated, is used with the same measure of expansion in equivalent cylinders of the two kinds of engine for the same power developed, the vacuum in the condenser being the same as well as the number of double strokes made by the pistons per minute, the cylinders of both engines being steam-jacketed or not jacketed, must be found entirely in the less cylinder condensation of the compound engine, the extremes of temperature in its cylinders being much less than in the cylinder of the simple engine. To the extent, therefore, that the gain by the lessened cylinder condensation exceeds the loss by the increased back pressure and friction pressure, should there be such excess, will the compound engine exceed the equivalent simple engine in economy of fuel. Under proper and equal conditions, it would seem that the difference could not be much.

The smaller the cylinder and the slower the speed of its piston, other things equal, the greater is the cylinder condensation proportionally to the weight of steam entering the cylinder; therefore, in a comparison of the compound and simple engines, the economic gain by the former should be at a maximum with small cylinders and slow speed of pistons, decreasing with increasing size of cylinders and increasing speed of pistons; also, with steam-jacketing, steam-superheating, and, generally, with all the cases which decrease cylinder condensation; for the loss, in the case of the compound engine, by increased back pressure and friction pressure, is not affected by the causes which lessen cylinder condensation.

Like all physical questions, the comparative economy of fuel with the compound and the simple engine can only be answered by experiments made with all the conditions proper and the same for each. The results will necessarily be as various as the limiting conditions, and will show that no general answer can be given independently of them, each set of which will form a separate problem and require a distinct solution.

The difference between the quantities on lines 72 and 73 shows the pressure per square inch on the piston of the large cylinder alone, 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.

RELATIVE ECONOMY OF THE POWERS.—Lines 76, 77, and 78 contain respectively the quantities representing the relative economy of heat with which the indicated net and total horse-powers were developed by the engine under the different conditions of the experiments. They are the quantities on lines 54, 55, and 56 expressed proportionally, those for Experiment D being taken as unity. They show that very different relations exist between the economical results of the indicated net and total horse-powers, comparing the same experiments; and that while in any two experiments the total horse-power might be developed more economically in the one than in the other, the very reverse might be the fact for the net horse-power. The total work done by the steam and the useful work obtained from it have no fixed relation.

As the cylinder condensation in steam engines exerts so great an influence on their economic efficiency, and as the economic superiority of the compound over the simple engine, other things equal, arises entirely from the less condensation of the steam in its cylinders, the distribution of that steam between its two cylinders should, for maximum economic effect, be made in a manner that will produce the least cylinder condensation.

Now, evidently, if the steam be so distributed that practically speaking all the power of the engine is developed in the small cylinder of the compound engine, then the large one becomes merely an exhaust pipe for the small one. And, similarly, if all the power of the engine is developed in the large cylinder, the small one becomes merely a steam pipe for the large one, so that in either of the supposed cases the compound engine would be converted into a simple engine, and manifestly lose with its character its superior economic efficiency, its cylinder condensation becoming just the same as that of the equivalent simple engine. There must be, consequently, some point between the two extremes, some proportionment of the power between the two cylinders, that would produce the minimum cylinder condensation and give as a result the maximum economic effect. In proportioning the power, therefore, between the two cylinders, care must be taken that not only the steam pressure is properly distributed for producing the maximum economic effect by reducing the loss of pressure to the minimum in the transfer of the steam from the small cylinder to the large one, but that the cylinder condensation shall be at the same time reduced to the minimum, for on the more or less completeness with which these two conditions are combined will be the economic efficiency of the compound engine.

As regards the production of the minimum cylinder condensation, the probability seems that the power should be so divided between the cylinders that the product of the internal surface of one cylinder, including surface of steam passage and piston, and half the surface of the piston-rod, into the difference between the temperature of the initial steam in that cylinder and the temperature of the back pressure at the end of the exhaust stroke of the piston, should equal the similar product in the case of the other cylinder, the weight of steam used in both cylinders being about the same. Of course this problem, like all others in physics, is to be solved by experiment alone, to which end trials cannot be too numerous of the same compound engine with every possible variation of valve-gear, giving corresponding variations of pressure distribution in the two cylinders, and of cylinder condensation. From the discussion of a sufficiently large number of such trials, a general law may be experimentally evolved.

As the difference of economic effect in a compound engine is very great between such a distribution of the pressure in its two cylinders as will give the least loss by producing the minimum loss of pressure in the steam transfer from one cylinder to the other in combination with the minimum steam condensation in them, and such a distribution of the pressure in its two cylinders as will give the greatest loss of pressure in the steam transfer in combination with the greatest cylinder condensation, every compound engine should have determined for it, by actual trial at the speed of piston with which it is to be used, the distribution of pressure that will produce the minimum economic loss. This distribution will probably vary with each engine, and possibly largely, but there is no other way in which it can be ascertained. The proper distribution of the pressure for maximum economy is a far more complex and difficult matter in a compound engine than in a simple engine, because the conditions affecting it are more numerous and more occult.

In ordinary practice the pressure in compound engines is so distributed as to make the indicated horses-power developed by each cylinder the same, the purpose being to equalize the strains upon the mechanism of each cylinder, a purely mechanical effect and a useful one, but not necessarily producing the maximum economic effect. The delicate point is to combine the proper mechanical effects with the maximum economic effects in such a manner as, with a given boiler and condenser pressure, to produce the required power from cylinders of the least dimensions. To accomplish this, the results of many experiments from many compound engines must be carefully and intelligently studied, and the labor will be profitably bestowed, for the difference in the money expenditure involved for the production of a given power from a compound engine skillfully or unskillfully designed is very great.

DISCUSSION OF THE RESULTS.

In comparing the results for different experiments, the comparisons should be confined to those in which the speed of the pistons did not materially vary, as that factor considerably influences the results in the case of engines having a marked cylinder condensation. Also, the initial pressure at the commencement of the stroke of the piston of the small cylinder should not materially differ, nor the back pressure against the piston of the large cylinder, though these factors seem to influence the economical development of the power in a much less degree than the speed of the piston.

The comparisons to be made of the results of these experiments, allow, within their limits and for their conditions, other things being supposed equal, the following determinations, namely:

1st. The economical effect produced in the development of power, by closing the cut off valve of the large cylinder at different points of the stroke of its piston.

, to ascertain their
maining the same.

I	J
June 22.	June 2
2. 21528	0.
23480	6305
4551. 1240	1537.
79. 3	60.
1. 4	32.
0. 4483	0.
0. 8743	0.
5. 84326	3.
30. 15	30.
24. 25	23.
2. 896	3.
176. 65354	193.
2054. 42382	2825.
2326648. 74079	3152834.
76	72
91	94
66	65
80. 0	92.
11. 49365	12.
9. 97147	11.
28. 4304	27.

2d. The economical effect produced in the development of power by using the steam with different measures of expansion.

3d. The economical effect produced in the development of power by using the steam with different initial pressure on the piston of the small cylinder.

4th. The economical effect produced in the development of power by simultaneously increasing the speed of the pistons and the pressures upon them.

Discussing these causes affecting the thermic economy with which the power was developed by the compound engine of the "Siesta," and within the limits and under the conditions of the experiments, the following results will be found:

OF THE INFLUENCE EXERCISED UPON THE THERMIC ECONOMY WITH WHICH THE POWER IS DEVELOPED IN A COMPOUND ENGINE, BY CLOSING THE CUT-OFF VALVE OF THE LARGE CYLINDER AT DIFFERENT POINTS OF THE STROKE OF ITS PISTON.—To determine the magnitude of this influence, the results of Experiments B and C will first be compared. In these two experiments, the piston speed differed but slightly, the number of double strokes made per minute being 169.44444 in Experiment B, and 170.33869 in Experiment C. The initial pressure in the small cylinder in pounds per square inch above zero, was 102.93 in Experiment B, and 105.30 in Experiment C. The pressure at the point of cutting off in the small cylinder in pounds per square inch above zero, was 90.71 in Experiment B, and 90.13 in Experiment C. The final pressure in the large cylinder during Experiment B was 7.64 pounds per square inch above zero, and during Experiment C 7.15. The back pressure against the piston of the large cylinder was nearly the same in both experiments. In Experiment B the steam was expanded 9.91229 times, and in Experiment C 9.61731 times. The mean total pressure in pounds per square inch above zero, reduced for the piston of the large cylinder alone, was 29.90347 for Experiment B, and 28.69970 for Experiment C. The total horses-power developed by the engine, were 116.73975 in Experiment B, and 112.63125 in Experiment C. All the conditions are, therefore, sufficiently near equality except the one at issue, namely, that in Experiment B the cut-off valve of the large cylinder was closed when 0 2511 of the stroke of its piston was completed, and in Experiment C when 0.4648 was completed. In Experiment B, the total horsepower was obtained for 15740.62131 Fahrenheit units of heat consumed per hour, and in Experiment C for 16046.09624 units; the conditions in Experiment C being economically 1.9407 per centum inferior to those in Experiment B.

Again, the results of Experiments E and F may be compared, to determine the influence on the economic development of the power due to closing the cut-off valve of the large cylinder when 0.2374 of the stroke of its piston was completed as in Experiment E, and when 0.4522 of the stroke of its piston was completed as in Experiment F. In these two experiments, the piston speed differed but slightly, the number of double strokes made per minute being 177.94444 in Experiment E, and 178.51667 in Experiment F. The initial pressure in the small cylinder in pounds, per square inch above zero was 90.45 in Experiment E, and 82.18 in Experiment F. The pressure at the point of cutting off in the small cylinder in pounds per square inch above zero, was 75.55 in Experiment E, and 74.02 in Experiment F. The final pressure in the large cylinder during Experiment E, was 9.00 pounds per square inch above zero, and during Experiment F, 9.53. The back pressure against the piston of the large cylinder was nearly the same in both experiments.

In Experiment E the steam was expanded 6.20137 times, and in Experiment F 6.40193 times. The mean total pressure in pounds per square inch above zero, reduced for the piston of the large cylinder alone, was 34.73920 for Experiment E, and 32.96556 for Experiment F. The total horses-power developed by the engine, were 142.42101 in Experiment E, and 135.58420 in Experiment F. All the conditions are, therefore, sufficiently near equality, except the one at issue, namely, that in Experiment E the cut-off valve of the large cylinder was closed when 0.2374 of the stroke of its piston was completed, and in Experiment F when 0.4170 was completed. In Experiment E the total horse-power was obtained for 14922.31862 Fahrenheit units of heat consumed per hour, and in Experiment F for 15933.99601 units; the conditions in Experiment F being economically 6.7796 per centum inferior to those in Experiment E.

The mean of the two preceding determinations shows that the economic development of the total power, when the cut off valve of the large cylinder was closed at $\left(\frac{0.4648+0.4170}{2}=\right)$0.4409 of the stroke of its piston, was $\left(\frac{1.9407+6.7796}{2}=\right)$ 4.3601 per centum inferior to what it was when the cut-off valve was closed at $\left(\frac{0.2511+0.2374}{2}=\right)$ 0.2442 of the stroke.

Extending the comparison to the cases of Experiments F, G, and H, in the latter two of which the cut-off valve of the large cylinder was closed still later, and comparing the data and results of Experiment F with the mean of the data and results of Experiments G and H—Experiment H being nearly a repetition of Experiment G, the principal variation being in the less piston speed of Experiment G, due to a stronger head-wind, the following are obtained:

The number of double strokes made per minute by the pistons in Experiment F was 178.51667, and in Experiments G and H

$$\left(\frac{173.35000+177.12179}{2}=\right)175.235895.$$

The initial pressure in the small cylinder in pounds per square inch above zero, was 82.18 in Experiment F, and $\left(\frac{80.40+78.34}{2}=\right)$79.37 in Experiments G and H. The pressure at the point of cutting off in the small cylinder in pounds per square inch above zero, was 74.02 in Experiment F, and $\left(\frac{74.90+72.34}{2}=\right)$73.62 in Experiments G and H. The final pressure in the large cylinder during Experiment F was 9.53 pounds per square inch above zero, and during Experiments G and H

$$\left(\frac{9.54+9.25}{2}=\right)9.395.$$

The back pressure against the piston of the large cylinder was 4.360 pounds per square inch above zero during Experiment F, and

$$\left(\frac{4.072+3.784}{2}=\right)3.928$$

pounds during Experiments G and H. In Experiment F the steam was expanded 6.40193 times, and in Experiments G and H

$$\left(\frac{6.28256+5.97668}{2}=\right)6.12962$$

times. The mean total pressure in pounds per square inch above zero, reduced for the piston of the large cylinder above, was 32.96556 for Experiment F, and $\left(\frac{29.07594+29.09728}{2}=\right)$ 29.08661 for Experiments G and H. The total horses-power developed by the engine, were 135.58420 in Experiment F, and $\left(\frac{116.12546+118.73924}{2}=\right)$ 117.43235 in Experiments G and H. The conditions throughout are to a small extent more favorable in Experiment F than in Experiments G and H; nevertheless, the difference is not sufficient to prevent comparison between the remaining condition which is the one at issue, namely, that in Experiment F the cut-off valve of the large cylinder was closed when 0.4522 of the stroke of its piston was completed, and in Experiments G and H when $\left(\frac{0.8460+0.8573}{2}=\right)$ 0.85165 was completed. In experiment F, the total horse-power was obtained for 15933.99601 Fahrenheit units of heat consumed per hour, and in experiments G and H for

$$\left(\frac{18799.25957+19064.23068}{2}=\right)18931.74512$$

units; the conditions in Experiments G and H being economically 18.8135 per centum inferior to those in Experiment F.

Again, the results of Experiments E and I may be compared to determine the influence on the economic development of the power due to closing the cut-off valve of the large cylinder when 0.2374 of the stroke of its piston was completed as in Experiment E, and when 0.8743 of the stroke of its piston was completed as in Experiment I. In these two experiments the piston speed differed but slightly, the number of double strokes made per minute being 177.94444 in experiment E, and 176.65354 in Experiment I. The initial pressure in the small cylinder in pounds per square inch above zero, was 90.45 in Experiment E, and 91.30 in Experiment I. The pressure at the point of cutting off in the small cylinder in pounds per square inch above zero, was 75.55 in Experiment E, and 76.30 in Experiment I. The final pressure in the large cylinder during Experiment E was 9.00 pounds per square inch above zero, and during Experiment I 10.82. The back pressure against the piston of the large cylinder in pounds per square inch above zero was 4.328 during Experiment E, and 5.090 during Experiment I. In Experiment E the steam was expanded 6.20137 times, and in Experiment I 5.84326 times. The mean total pressure in pounds per square inch above zero, reduced for the piston of the large cylinder alone, was 34.73920 for Experiment E, and 31.33134 for Experiment I. The total horses-power developed by the engine were 142.42101 in Experiment E, and 127.51792 in Experiment I. The conditions, through a little the most favorable for Experiment E, are sufficiently near equality, except the one at issue, namely, that in Experiment E the cut-off valve of the large cylinder was closed when 0.2374 of the stroke of its piston was completed, and in Experiment I when 0.8743 was completed. In Experiment E the total horse-power was obtained for 14922.31862 Fahrenheit units of heat consumed per hour, and in Experiment I for 18245.66101 units, the conditions in Experiment I being economically 22.2710 per centum inferior to those in Experiment E.

No comparison is proper between the econom c results of Experiment D and those of other experiments in which the steam was expanded about the same number of times, because of the much higher piston speed (190.19443 double strokes of piston per minute), and much greater initial pressure (105.23 pounds per square inch above zero) in the small cylinder during Experiment D.

From the foregoing there appears that each lengthening of the cut-off in the large cylinder, that is to say, each closing of the cut-off valve of that cylinder when a greater and greater fraction of its stroke was completed, from 0.2374 to 0.8743, was attended by a marked decrease in the economic development of the total horse-power; and, consequently, if all the other conditions were exactly the same, there would be a corresponding decrease in the economy of the indicated horse-power and of the net horse-power.

A cut-off valve on the large cylinder of a receiver compound engine is, according to the evidence of these experiments, which is in harmony with theoretical indications, requisite for obtaining the power with the least expenditure of fuel, as well as for convenience in the distribution of the pressures and for the equalization of the powers developed in the two cylinders. Also in the event of accident disabling the small cylinder, and the consequent necessity of using the large cylinder alone, the cut-off valve on the large cylinder is indispensable to economy of fuel, and this latter consideration applies with equal force to both arrangements of the cylinders of the compound engine, namely, the tandem and the receiver.

OF THE INFLUENCE EXERCISED UPON THE THERMIC ECONOMY WITH WHICH THE POWER IS DEVELOPED IN A COMPOUND ENGINE BY THE DIFFERENT MEASURES OF EXPANSION WITH WHICH THE STEAM IS USED.—The measure of expansion with which the steam is used in a compound engine depends, theoretically, entirely on the point at which the cut-off valve closes in the stroke of the piston of the small cylinder. It is not in the least affected, theoretically, by the presence or absence of a cut-off valve on the large cylinder, nor by the point in the stroke of the piston of the large cylinder at which that valve may close. Practically, however, the cut-off valve on the large cylinder enables the steam re-evaporated from the water of condensation in the small cylinder to be used expansively in the large cylinder, instead of being used there without expansion. And it also enables the steam pressure to be so distributed in the two cylinders as to obtain for the greater measures of expansion a higher economy relatively to the smaller ones than can be obtained in the simple engine; because, in the compound engine, when the steam is used with a considerable measure of expansion in the small cylinder, its pressure at the end of the stroke of the piston can be made the same as that of the back pressure and thereby secure the most economical distribution of the pressures without sacrificing the equality of the indicated power developed in each of the two cylinders which cannot be done if the steam be used with an inconsiderable measure of expansion in the small cylinder. In other words, when the steam is used with much expansion in the small cylinder, the expansion curve can be made to touch the line of the back pressure at the end of the stroke of the piston, which enables the maximum economic distribution of the steam pressures to be made and yet allows the small cylinder to develop an equal indicated power with the large one; but when the steam is used with inconsiderable measures of expansion in the small cylinder, then, if the above-described distribution of the pressures be made so that the expansion curve meets the line of the back pressure at the end of the stroke of the piston, the small cylinder would develop only a very small indicated power.

Now, in this case, if the small cylinder must develop an equal indicated power with the large one, its terminal pressure must be kept sufficiently high above the line of the back pressure; and higher and higher as the measure of expansion with which the steam is used is less and less; consequently, there is a greater loss of pressure in the steam transfer from one cylinder to another when the steam is used with a lesser measure of expansion in the small cylinder than when it is used with a greater measure, the same power being developed in the small and in the large cylinder. When, therefore, under this unfavorable condition of a greater loss of pressure in the steam transfer from the small to the large cylinder, that attends the development of equal indicated powers in the two cylinders with less measures of expansion in the small cylinder than with greater ones, which unfavorable condition becomes more and more exaggerated as the measure of expansion becomes less and less, it is evident that the greater economy of using the steam with greater measures of expansion in the same compound engine is not only what may be due to the greater measures of expansion, *per se*, but additionally, to what is due to the less loss of pressure in the transfer of the steam from the one to the other cylinder.

These considerations do not apply to the use of steam with the greater measures of expansion in the simple engine, in which case whatever economy may be found for the greater measures is just what is due to them, *per se*. Hence, in the same compound engine developing about the same indicated power in both cylinders, the greater measures of expansion should produce a greater economy relatively to the smaller measures than in the case of the simple engine; so that under the above condition in the former engine, other things equal, expansion can be beneficially carried further than in the latter one.

In examining the economic results of experiments with steam used with different measures of expansion in the same compound engine, therefore, they will be found to be much governed by the point in the stroke of the piston of the large cylinder at which the cut-off valve of that cylinder closes, as well as by the point in the stroke of the piston of the small cylinder at which its cut-off valve closes.

It is extremely difficult to obtain proper conditions for such comparisons. Obviously, the piston speed, the initial pressure in the small cylinder, the back pressure in the large cylinder, and the effective power developed by the engine should be the same, the latter condition requiring cylinders of greater and greater area of piston—the stroke remaining constant—as the measure of expansion employed was greater and greater. The point at which the cut-off valve of the large cylinder closes in the stroke of its piston should be that which produces the greatest economy in each case. These conditions involve the expenditure of so much money and time, and the trials would have to be so numerous in view of all the changes of valve-gear which could be effected, that important to the industrial arts as the results would be, there is but little hope of such a system of experiments being undertaken.

The experimental results obtained from the compound engine of the "Siesta", though exact for the experimental conditions, do not allow a completely satisfactory comparison to be made of the influence that would be exerted on the economic development of the power by the measure of expansion with which the steam was used if those conditions were what they should be for that purpose. As near as these experiments admit, however, they may be discussed as giving a reasonable indication of the truth. Of course they show the economic effect, in the case of a given compound engine, of using the steam with the different experimental measures of expansions under the experimental conditions.

A comparison of the results of Experiments B and E, whose data furnish the nearest approach to the proper comparable conditions, allows the determination to be made of the relative economy of using steam expanded in the same compound engine, 9.91229 times as in Experiment B, and 6.20137 times as in Experiment E. In the former experiment the cut-off valve of the large cylinder closed when 0.2511 of the stroke of its piston was completed, and in the latter experiment when 0.2374 of that stroke was completed. In Experiment B the pistons of the engine made 169.44444 double strokes per minute, and in Experiment E, 177.94444; the lesser measure of expansion having in this respect the most advantageous condition. In Experiment B the initial steam pressure in the small cylinder was 102.93 pounds per square inch above zero, and in Experiment E, 90.45, the lesser measure of expansion having in this respect the least advantageous condition. The greater piston speed with the lesser measure of expansion may be considered as neutralized in large part, if not wholly, by the smaller initial steam pressure, as regards economic effect. The back pressure against the piston of the large cylinder in Experiment B was 3.765 pounds per square inch above zero, and in Experiment E, 4.328. When the steam was expanded 9.91229 times (Experiment B), the total horse-power cost 15740.62131 Fahrenheit units of heat per hour; and when expanded 6.20137 times (Experiment E), 14922.31862 units; consequently, the greater measure of expansion was economically 5.4837 per centum inferior to the lesser measure. Now, if the same initial steam pressure had been used with both measures of expansion, then the mean total pressure would have been much greater with the lesser measure of expansion than with the greater; and, as the back pressure and the pressure required to work the engine, *per se*, are constant for all measures of expansion, the indicated horse-power would have been obtained by the lesser measure of expansion with considerably greater economy than the above 5.4837 per centum, and the net horse-power with still more economy. It is quite evident from these experiments that a loss instead of a gain in the thermic cost of the power, follows any increase in the measure of expansion beyond six times. And this result is for a piston speed and initial steam pressure considerably above the average in practice, but with saturated steam used in unjacketed cylinders.

Again, making comparison of the results of Experiments C and F, whose data furnish the nearest approach to comparable conditions, the cut-off valve of the large cylinder closing in Experiment C when 0.4648 of the stroke of its piston was completed, and in Experiment F when 0.4522 of that stroke was completed, allows the determination to be made of the relative economy of using steam expanded in the same compound engine 9.61731 times as in Experiment C, and 6.40193 times as in Experiment F. In Experiment C the pistons of the engine made 170.33809 double strokes per minute, and in Experiment F 178.51667; the lesser measure of expansion having in this respect the most advantageous condition. In Experiment C the initial steam pressure in the small cylinder was 105.30 pounds per square inch above zero, and in Experiment F 82.18; the lesser measure of expansion having in this respect the least advantageous condition. The greater piston speed with the lesser measure of expansion may be considered as neutralized in large part, if not wholly, by the smaller initial steam pressure, as regards economic effect. The back pressure against the piston of the large cylinder in Experiment C was 3.340 pounds per square inch above zero, and in Experiment F 4.360. When the steam was expanded 9.61731 times (Experiment C), the total horse-power cost 16046.09624

Fahrenheit units of heat per hour; and when expanded 6.40193 times (Experiment F), 15933.99601 units; consequently the greater measure of expansion was 0.7035 per centum economically inferior to the lesser measure. Had the proper condition of equal initial pressure obtained in both cases, then the back pressure and the pressure required to work the engine, *per se*, being constant for all cases, the indicated pressure and still more the net pressure would have been greater proportions of the mean total pressure with the lesser measure of expansion than with the greater, and the economy with which the indicated horse-power, and still more the net horse-power was developed with the lesser measure of expansion would have been proportionally greater than the above 0.7035 per centum over the economy with the greater measure of expansion. Thus, again, it is experimentally shown that under equal and proper conditions, expanding the steam in a compound engine beyond about six times, involves a loss instead of a gain in fuel.

Experiments A and E would be comparable for determining the influence upon the economic development of the power of extending the expansion of steam in a compound engine from 6.20137 to 12.92622 times, were the difference between the piston speeds in the two cases not so great. In Experiment A, during which the steam had the greater measure of expansion, the pistons made 146.54563 double strokes per minute; and in Experiment E, during which the steam had the lesser measure of expansion, the pistons made 177.94444 double strokes per minute. In the important respect of piston speed, therefore, the greater measure of expansion had the least advantageous condition. But as the initial-steam pressure in the small cylinder during Experiment E, in which the lesser measure of expansion was used, was 90.45 pounds per square inch above zero while in Experiment A, in which the greater measure of expansion was used, it was 110.03 pounds, so that in the important respect of initial pressure the greater measure of expansion had the most advantageous condition the two variations from equality being in opposite directions, may be held to neutralize each other, if not wholly, at least in great part. The back pressure against the piston of the large cylinder was 3.797 pounds per square inch above zero in Experiment A, and 4.328 pounds in Experiment E. When the steam was expanded 12.92622 times (Experiment A), the total horse-power cost 15431.43327 Fahrenheit units of heat consumed per hour; and when expanded 6.20137 times (Experiment E), 14922.31862 units; consequently, the greater measure of expansion was 3.4117 per centum economically inferior to the lesser. In Experiment A the cut-off valve of the large cylinder closed when 0.2627 of the stroke of its piston was completed; and in Experiment E when 0.2374 of that stroke was completed. Had the initial pressure with the lesser measure of expansion been the same as with the greater measure, the back pressure and the pressure required to work the engine, *per se*, remaining constant, the indicated and net pressures would have been so very much larger a proportion of the mean total pressure in the case of the lesser measure of expansion than in the case of the greater measure, that the economic development of the indicated horse-power, and still more of the net horse power, with the lesser measure of expansion would have largely exceeded the economic development of these horse-powers with the greater measure of expansion. In the case of this comparison, as in those of the preceding ones, the experimental evidence shows that an economic loss attends the extension of the measure of expansion beyond about six times, even if considerable be allowed in favor of the greater measure of expansion on account of the less experimental speed of the piston in Experiment A than in Experiment E.

It is to be regretted that Experiment J was of such short duration—only 0.544 of an hour—which makes its data less certain than that of the other experiments of much longer duration; but the error, or rather uncertainty, due to the fewer observations cannot be great, if any, and is as likely to be against as for the economic results. In that experiment the steam was expanded only 3.24468 times, with the cut-off valve of the large cylinder closed when 0.2976 of the stroke of its piston was completed; and the results can be compared with those of Experiment D in which the steam was expanded 5.91860 times, with the cut off valve of the large cylinder closed when 0.2084 of the stroke of its piston was completed. The number of double strokes made per minute by the pistons did not materially vary in the two experiments, being 193.16838 in Experiment J, and 190.19443 in Experiment D. The back pressure in the two experiments against the piston of the large cylinder did not materially vary, but the initial pressure in the small cylinder differed enormously, being 105.23 per square inch above zero with the greater measure of expansion (Experiment D), and 66.70 pounds with the lesser measure of expansion (Experiment J). Indeed, not only was the advantage in this respect for the greater measure of expansion very large, but the ratio of the two cylinders to each other was very disadvantageous for the lesser measure of expansion, as only one-fifth of the indicated horses-power of the engine was developed in the small cylinder during Experiment J. Had the two cylinders been better proportioned to each other, and had the initial pressure been as high with the lesser measure of expansion as with the greater, much better economic results than the experimental ones in regard to the cost of the total horse-power would have been obtained with the lesser measure. When the steam was expanded 5.91860 times (Experiment D), the total horse-power cost 13754.70533 Fahrenheit units of heat per hour; and when expanded 3.24468 times (Experiment J) 16510.63272 units; consequently the lesser measure of expansion was 20.0363 per centum economically inferior to the greater measure. As the per centum of the mean total pressure realized as indicated pressure and still more as net pressure would have been very much greater with the lesser measure of expansion than with the greater measure, the back pressure against the piston of the large cylinder, and the pressure required to work the engine, *per se*, being constant, the same initial pressure being used in both cases, the above economic inferiority of 20.0363 per centum found with the lesser measure of expansion for the total horse-power, would have been largely reduced for the indicated horse-power and still more for the net horse-power.

Experiment L was a repetition of Experiment K, made on a different day, with the view of ascertaining the degree of confidence that might be placed in the experimental determinations. The conditions of these two experiments were almost identical, and so were the results; the cost of the total horse-power in Experiment K being 21866.14355 Fahrenheit units of heat per hour, and in Experiment L 21563.41095 units, a difference of 1.4039 per centum of the smaller number. In Experiments K and L the cut-off valve of the large cylinder closed when $\left(\frac{0.8706+0.8701}{2}=\right)$ 0.87035 of the stroke of its piston was completed; the pistons made per minute $\left(\frac{185.30556+184.44584}{2}=\right)$ 184.87570 double strokes; the initial pressure in the small cylinder was $\left(\frac{65.60+64.98}{2}=\right)$

65.29 pounds per square inch above zero; the back pressure against the piston of the large cylinder was $\left(\frac{5.640+5.259}{2}=\right)$ 5.4495 pounds; and the measure of expansion with which the steam was used, was $\left(\frac{3.26134+3.26033}{2}=\right)$ 3.260835 times.

The experiment furnishing the nearest comparable conditions, excepting the one at issue, namely, the measure of expansion, is experiment H, in which the cut-off valve of the large cylinder closed when 0.8573 of the stroke of its piston was completed; the pistons made per minute 177.12179 double strokes; the initial pressure in the small cylinder was 78.34 pounds per square inch above zero; the back pressure against the piston of the large cylinder was 3.784 pounds per square inch above zero; and the measure of expansion with which the steam was used was 5.97668 times. In this comparison, as in the preceding ones, the lesser measure of expansion has the most advantageous piston speed, and the most disadvantageous initial pressure; to a certain extent without doubt, and possibly entirely, these opposed conditions, as regards economy of power development, counterbalance each other. Assuming such to be the fact, the following result is obtained: The total horse-power with the greater measure of expansion (Experiment H) cost 19064.23068 Fahrenheit units of heat per hour, and with the lesser measure of expansion (Experiments K and L) it cost

$$\left(\frac{21866.14355+21563.41095}{2}=\right) 21714.77725$$

units; consequently, the lesser measure of expansion was 13.9032 per centum economically inferior to the greater measure. In this comparison, too, if the same initial pressure were used with both measures of expansion, the mean total pressure would be much more with the lesser measure of expansion than with the greater, and as the back pressure against the piston of the large cylinder, and the pressure required to work the engine, *per se*, are constant, there would be realized a higher percentum of the mean total pressure in indicated pressure and in net pressure with the lesser than with the greater measure of expansion, so that the indicated and net horse-powers would be obtained in the cases of the two measures of expansion with far less economic difference than the above 13.9032 per centum—possibly with no difference.

The important deduction from these experiments, as regards the influence on the economic development of power of using steam with different measures of expansion, seems to be: That with equal initial pressure in the small cylinder, equal back pressure against the piston of the large cylinder, equal speed of piston of the same stroke, and equal *net* power developed, it may be doubted whether, with cylinders properly proportioned to give the best results with the measure of expansion employed and having the pressures in them distributed in the most advantageous manner for the measure of expansion employed, by the cut-off on the large cylinder, any economic gain could be obtained for the net horse-power by expanding the steam more than from four to five times, even in a compound engine using saturated steam of a very high pressure in unjacketed cylinders with pistons working at a very high reciprocating speed.

This deduction, should it be confirmed by other experiments, is very important in view of the enormous power required in the near future for ocean steam navigation, because the excessive cost, weight, and bulk of engines when such power is developed by using the steam very expansively, limits it below what can be obtained when the steam is used with less expansion.

OF THE INFLUENCE EXERCISED UPON THE THERMIC ECONOMY WITH WHICH THE POWER IS DEVELOPED IN A COMPOUND ENGINE BY DIFFERENT INITIAL PRESSURES IN THE SMALL CYLINDER.—The economic development of power in a compound engine is affected in a marked degree by marked variations of the initial pressure in the small cylinder, other things equal, according to the only pair of experiments made under proper conditions for a comparison, namely, Experiments H and I.

This difference in the economic effect produced by different initial pressures, other things equal, is due to two causes: One, that a given bulk of steam pressure is obtained with a less proportional expenditure of heat with steam of high pressure than with steam of low pressure; the other, that the cylinder condensation is less, relatively to weight of entering steam, with high initial pressure than with low, because the weight of steam passed through the cylinders in a given time is nearly *pro rata* with the higher initial pressure than with the lower, while the refrigerating influences increase in a very much less proportion. The last cause is considerably the most important of the two.

In the comparable Experiments H and I the pistons made 177.12179 double strokes per minute for the former and 176.65354 for the latter; the cut-off valve of the large cylinder closed when 0.8573 of the stroke of its piston was completed in Experiment H and when 0.8743 of the stroke was completed in Experiment I; the back pressure in the large cylinder in Experiment H was 3.784 pounds per square inch above zero and 5.090 pounds in Experiment I; the steam was expanded 5.97668 times in Experiment H and 5.84326 times in Experiment I; the initial pressure in the small cylinder was 78.34 pounds per square inch above zero in Experiment H and 91.30 pounds in Experiment I; the number of total horses-power developed in Experiment H was 118.73924, each of which required the consumption of 19064.23068 Fahrenheit units of heat per hour; while, in Experiment I 127.51792 total horses-power were developed for a consumption of 18245.66101 Fahrenheit units of heat per hour each; consequently the lower initial pressure (Experiment H) was economically 4.4864 per centum inferior to the higher initial pressure (Experiment I). As the mean total pressure is higher with higher initial pressures, other things equal, and as the back pressure against the piston of the large cylinder, and the pressure required to work the engine, *per se*, are constant, there will be a larger per centum of the mean total pressure utilized as indicated pressure and still larger as net pressure with the higher initial pressures than with the lower ones; consequently the indicated and, still more, the net horse-power will be obtained for the higher initial pressure in Experiment I with greater economy than the above 4.4864 per centum, relatively to the cost of the indicated and net horse-powers with the less initial pressure in Experiment H.

OF THE INFLUENCE EXERCISED UPON THE THERMIC ECONOMY WITH WHICH THE POWER IS DEVELOPED IN A COMPOUND ENGINE BY SIMULTANEOUSLY INCREASING THE SPEED OF THE PISTONS AND THE PRESSURES UPON THEM.—The table furnishes but one pair of experiments that can be used to show the influence exercised upon the economic development of the power by simultaneously increasing the speed of the pistons of the same compound engine and the pressure upon them; namely Experiments D and E, in which about the same measure of expansion is used for the steam, with the cut-off valve of the large cylinder closing at about the same point in the stroke of its piston; but with a marked difference in the speed of the pistons given by a marked difference in the pressure upon them.

In Experiment D the steam is expanded 5.91860 times, with the cut-off valve on the large cylinder closed when 0.2084 of its stroke was completed. In Experiment E the steam is expanded 6.20137 times, with the cut-off valve on the large cylinder closed when 0.2374 of its stroke was completed. The initial pressure in the small cylinder during Experiment D was 105.23 pounds per square inch above zero, and during experiment E 90.45 pounds. The back pressure against the piston of the large cylinder was 5.122 pounds per square inch above zero in Experiment D and 4.328 pounds in Experiment E. The number of double strokes made by the pistons per minute was 190.19443 in Experiment D and 177.94444 in Experiment E. The total horse-power in Experiment D was obtained for the expenditure of 13754.70533 Fahrenheit units of heat per hour, and in Experiment E for 14922.31862 units; consequently, the economic result with the slower speed of piston and the lower pressure (Experiment E) was 8.4888 per centum inferior to that with the faster speed of piston and the higher pressure (Experiment D).

As the mean total pressures reduced for the piston of the large cylinder alone, compare in the two experiments as 41.22100 and 34.73920 pounds per square inch above zero, and as the back pressures against the piston of the large cylinder can be made equal in both cases, the pressure required to work the engine, *per se*, being likewise the same in both cases, then, evidently, the indicated power, and still more the net power, will be obtained for the faster piston-speed and higher mean total pressure with greater economy than the above 8.4888 per centum, relatively to the indicated and net horse-powers with the slower speed and lower mean total pressure.

The net horse-power, it will be remembered, is the only power developed by the engine which is commercially valuable; that is, it is the only portion of the total power developed by the engine which does work external to the cylinder. As regards comparison of cost of power for practical purposes—for the objects of steam users—the cost of the net horse-power is the correct data. But, for properly comparing the entire work done by the steam—the absolute performance—which is what scientific engineering requires, the heat cost of the total horse-power must be employed, and from that the engineer deduces, with intelligence and complete command of the subject, the modifications due to any given set of circumstances.

These experiments show, what indeed all comparative experiments made with a given engine have shown, that the economic development of the power in identically the same steam engine varies enormously according to the conditions of its use. The conditions for maximum economy vary also for different engines, according not only to type of engine, but to dimensions of cylinder, thickness of its metal, back pressure against and initial pressure upon its cylinder, reciprocating speed of piston, measure of expansion with which the steam is used, steam superheating and steam jacketing, water trapping, distribution of the pressures by the valve-gear, type of valve-gear, &c., so that deductions accurate for one set of conditions with a given steam engine may not be applicable for another set of conditions with the same engine, and still less with other engines. Each case is a separate problem and requires a distinct solution. And before a complete theory of the steam engine can be formed, there must be obtained the results from a vast number of experiments made on the same and on different engines with all possible variations of attending circumstances. From such a collection of experimental data, properly reduced and discussed, the broad generalizations may be made which alone can be an infallible guide to what can be economically obtained with steam used under given circumstances in a given engine.

From the above may be inferred of how little value is the mere gabble uttered by many superficial writers pretending to give accurate numerical comparisons of the economy with which steam can be used in different types of engines, of the most economical point of cutting off, &c., and in the most general and absolute terms, not only without specifying the limitations, but apparently ignorant that such limitations exist.

RESISTANCE OF THE HULL.

Five of the experiments, namely, A, D, I, J, and L, Table No. 1, were made to ascertain, among other facts, the powers expended to give the vessel the different speeds in those experiments and to ascertain the slip of the screw during the same. From the data of these experiments the resistance of the hull can be determined. The vessel throughout all the trials had, when at rest, sensibly the same draught of water and trim; and if the resistance of the hull at the different speeds varied in the ratio of the square of the speed, the slip of the screw would remain constant or the same at all speeds, which would show that the hull was properly modeled for the highest experimental speed, namely, a little over 11 geographical miles per hour. By forcing the boiler a higher speed could have been obtained, but the Herreshoff manufacturing company did not wish to exceed the limit for which the vessel had been designed.

In the five experiments above referred to, the speed of the vessel and the slip of the screw were as follows:

Designation of the experiment.	Speed of the vessel in geographical miles per hour.	Slip of the screw in per centum of its speed.
A	8.32175	28.0000
I	9.97147	28.4304
L	10.50290	27.8688
D	10.73552	28.4324
J	11.05914	27.4101

From the above is apparent that the slip of the screw was not affected by the variations in the vessel's speed, the experimental differences being very trifling and due probably entirely to differences in wind and current, which are not always exactly neutralized by running the vessel over a base an equal number of times consecutively in opposite directions. From the equality of the slip of the screw, therefore, in the different experiments, the inference is warranted that the resistance of the hull within the experimental limits was in the ratio of the square of the speed. The mean experimental slip was 28.0283 per centum of the axial speed of the screw, which will be taken as the true slip of the screw with the vessel propelled in smooth water and uninfluenced by wind or current.

Again, if the resistance of the hull be in the ratio of the square of its speed, then the net pressure upon the crank-pins of the engine during the above five experiments, the slip of the screw being a constant per centum in all, will also be in the ratio of the square of the speed. The following table shows to what extent the experimental net pressures (line 73, Table No. 1) correspond to this ratio, assuming the quantities of Experiment I as unity:

Designation of the experiment.	Squares of the vessel's speed, proportionally.	Net pressures on the crank-pins of the engine, proportionally.
A	0. 6965	0. 7246
I	1. 0000	1. 0000
L	1. 1094	1. 0756
D	1. 1591	1. 1564
J	1. 2301	1. 2294

The correspondence between Experiments I, D, and J may be called exact. Experiment A is about as much on one side of exact correspondence as Experiment L is on the other. The discrepancy is, moreover, but slight, and might result from a very small error in the vessel's speed—a factor which enters into the result by its duplicate ratio. The sum of the column of the "squares of the vessel's speed, proportionally," is 5.1951; and of the column of "net pressures on the crank-pins of the engine, proportionally," 5.1860; so the fact that the resistance of the hull of the Siesta varied according to the square of its speed may be considered as established within the experimental limits.

To give the vessel the speed of eleven geographical miles of 6,086 feet per hour, the screw with the slip of 28.0283 per centum must make 193.78565927 revolutions per minute; and to make this number of revolutions, supposing the large cylinder alone of the engine to be used, there must be a net pressure upon its piston of 28.75736 pounds per square inch, deduced from the five experiments above referred to, as follows, namely:

Designation of the experiments.	Experimental number of double strokes made by the engine piston per minute. (Line 13 of Table No. 1.)	Experimental net pressure upon the piston of the large cylinder alone, in pounds per square inch, corresponding to the number of double strokes of piston in the preceding column. (Line 73 of Table No. 1.	Net pressure required for 193.78565927 double strokes of piston per minute, deduced from the preceding column in the ratio of the squares of the experimental number of double strokes to the square of 193.78565927. In pounds per square inch of piston of large cylinder alone.
A	146. 54563	17. 09274	29. 8886
I	176. 65354	23. 58914	28. 3864
L	184. 44584	25. 37293	28. 0076
D	190. 19443	27. 27857	28. 3184
J	193. 16838	28. 99944	29. 1858
Mean, or true net pressure for 193.78565927 double strokes per minute.			28. 75736

The difference between the net pressure on the piston of the large cylinder alone and the indicated pressure (difference between pressures on lines 72 and 73 of Table No. 1) being 2.67516 pounds per square inch, this latter, representing the pressure required to work the engine *per se*, or unloaded, must be added to the above 28.75736 pounds net pressure to give the indicated pressure 31.43252 pounds per square inch on the piston of the large cylinder alone. From these pressures there will be obtained for the 193.78565927 double strokes of piston per minute, calculated for the large cylinder alone, 128.392744 net horses-power, and 140.336508 indicated horses-power, corresponding to the speed of eleven geographical miles per hour of the vessel.

To ascertain the resistance of the hull at the speed of eleven geographical miles per hour, it is necessary to find the portion of the above power that was applied to the propulsion of the vessel, for which purpose the distribution of the indicated horses-power will be calculated according to the following assumptions:

Of the indicated horses-power a portion is expended in working the engine *per se*, or the engine alone, independently of its load; and this portion, which consists of the friction of the packings and moving parts of the engine, including the screw shafting, must be first deducted, because no power can be applied to the screw or developed externally of the engine until the friction of the engine itself is counterbalanced by an equivalent pressure on the piston; in fact, until this pressure is exceeded the engine cannot move. In the case of the engine of the "Siesta" the pressure required to work it *per se* is taken at 2.67516 pounds per square inch of the piston of the large cylinder alone.

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

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

There still remain to be determined the portions of the net horses-power expended in the slip of the screw and in the propulsion of the vessel. These are ascertained as follows: The sum of the horses-power expended in overcoming the friction of the load and in overcoming the resistance of the water to the surface of the screw-blades, being deducted from the net horses-power, the remainder is divided between the horses-power expended in the slip of the screw and in the propulsion of the vessel in the ratio of the speeds of the two, the pressure exercised by the screw forward in propelling the vessel, and backward upon the 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.

The necessary calculations having been made in accordance with the foregoing assumptions, give the following for the distribution of the indicated horses-power developed by the engine when the vessel is at the speed of eleven geographical miles per hour:

	Horses-power.	Per centum of the net horses-power.
Indicated horses-power developed by the engine	140. 336508	
Horses-power expended in working the engine, *per se*	11. 943764	
Net horses-power applied to the crank pins	128. 392744	100. 00
Horses-power absorbed by the friction of the load	9. 629456	7. 50
Horses-power expended in overcoming the resistance of the water to the surface of the screw blades	11. 055100	8. 61
Horses-power expended in the slip of the screw	30. 188774	23. 51
Horses-power expended in the propulsion of the vessel	77. 519414	60. 38
Totals	128. 392744	100. 00

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

The horses-power expended in the propulsion of the vessel according to the distribution of the power, being 77.519414, is equal to $(77.519414 \times 33000 =)$ 2558140.662 foot-pounds of work per minute; and the speed of the vessel being 11 geographical miles per hour, is equal to

$$\left(\frac{11 \times 6086}{60} = \right) 1115.76666+ \text{ feet per minute;}$$

hence the resistance of the vessel at that speed, or its equivalent the thrust of the screw, is $\left(\frac{2558140.662}{1115.76666+} = \right)$ 2292.7201 pounds.

DETERMINATION OF THE POWER EXPENDED IN OVERCOMING THE RESISTANCE OF THE WATER TO THE IMMERSED EXTERNAL OR WETTED SURFACE OF THE HULL.—Taking the resistance of the water to one square foot of the smooth wooden surface of the immersed exterior of the vessel moving in it with the velocity of 10 feet per second, to be 0.45 pound, and at other velocities to be this quantity modified in the ratio of their squares to the square of 10, and deducing from the speed of the vessel the mean speed of its immersed surface due to the inclination of its horizontal water-lines to its longitudinal central plane, there results for that speed 18.29 feet per second, and, consequently, a surface resistance of $(10^2 : 0.45 :: 18.29^2 :)$ 1.505358 pounds per square foot moving with that velocity.

As the immersed external or wetted surface of the vessel during the above performance was 1,438 square feet, the power expended in overcoming its resistance was $\left(\frac{1438 \times 1.505358 \times 18.29 \times 60}{33000} = \right)$ 71.986274 horses-power; consequently, of the 77. 519414 horses-power required to propel the hull alone at the experimental speed, $\left(\frac{71.986274\times100}{77.519414}=\right)$ 92.86225 per centum was expended in overcoming the resistance of the wetted surface to the water, and the remaining 7.13775 per centum was expended in the displacement of water by the immersed solid of the hull, irrespective of the resistance of its immersed surface.

PERFORMANCE OF THE BOILER.

In the following table will be found the data and results of the experiment made to ascertain the economic vaporization of water in the boiler of the Siesta by anthracite consumed with natural draught.

The experiment was made with the vessel under way in Narragansett Bay on the 22d of June, 1882, and comprised all the steaming done during that day for the purpose of determining the economic performance of the engine under different conditions of steam pressure, expansion, piston speed, &c. Each of these trials continued from about two hours to three and a half hours, so that the boiler pressure, piston speed, temperatures, &c., are the means for the entire day, from which the extremes differed considerably. The means are not for the time, but for the weight of water vaporized during each of the short trials. The observed data were noted every ten minutes.

The weight of water vaporized was measured as discharged by the air-pump of the engine. For this purpose two tin tanks, both of exactly the same form and dimensions, were securely placed in the vessel side by side. Each tank had a narrow neck, above which was a funnel-shaped mouth. The two necks were joined by an open cross-pipe. From the bottom of each tank a pipe, with a stop-cock close to the tank, led to the open-topped reservoir drained by the feed-pump of the engine. One end of a flexible hose pipe was fitted to the closed top of the air-pump chamber, and the other or free end discharged alternately first into one tank and then into the other, the change being made instantly by an attendant who closed with his hand the discharging end of the hose while it was being shifted from the funnel mouth of one tank to that of the other, the two mouths nearly touching. By this method not a drop of water was lost during the change; as soon as one tank was filled the discharging end of the hose pipe was shifted to the other, and the water allowed to subside to the level of the bottom of the pipe joining the necks of the two tanks, when the stop-cock was opened in the pipe connecting the bottom of that tank with the feed-water reservoir. As soon as the tank was emptied, which required but a few seconds, this cock was closed. The ends of the pipes discharging the tanks into the reservoir were in open sight in order that the exhaustion of the tanks might be apparent as soon as they were empty. The tanks were placed above the reservoir so that the discharge was by gravity and very rapid. As the time of emptying a tank was considerably less than that of filling it, no inconvenience or uncertainty attended any of the operations. Each tankful of water as soon as discharged was tallied by an attendant who had no other duty to perform than this and the opening and the closing of the cocks. The temperature of the feed water in the tanks was noted from thermometers constantly immersed therein, and the weight discharged corrected for it. In all calculations concerning the vaporization of the feed water, its different specific heat at different temperatures is included. Each tank contained 37.4 pounds of water at the temperature of 64 degrees Fahrenheit, by accurate weighing.

The anthracite consumed was of rather less than fair merchantable quality. Every pound of it, and of its refuse in ash, clinker, and dust, was carefully weighed, the refuse in the dry state, on a steelyard. No particular pains was taken with the firing, which was left to the skill and care of an ordinary fireman; nor had the heating surfaces of the boiler been swept since several days' previous use.

At the commencement of the experiment the boiler was filled with water and a fire was made in the furnace with pine wood whose weight was not noted, and 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. The engine was started when enough anthracite was in combustion to maintain the steam pressure. Previously to the starting no steam was blown off at the safety-valve. During the day's experimenting the engine was not stopped, not even when changing from one rate of speed to another, as when one engine trial for economy was succeeded by another under different conditions, but in such cases the succeeding experiment was made with as little change in the rate of combustion as possible, and so sensitive was the boiler with its small quantity of water that a slight closing of the chimney damper was almost instantly followed by the required decrease in the quantity of steam furnished, and *vice versa;* thus, the entire heat from the wood and anthracite consumed acted upon the weight of water vaporized. At the close of the experiment the anthracite was burned entirely out, the engine being kept in operation as long as it would move. When it stopped for want of steam what remained in the furnace was drawn, the small quantity of unconsumed anthracite picked out, weighed, and deducted from the total quantity expended, and the remaining refuse of ash and clinker was weighed and noted as such.

As the fire in the boiler was lighted with wood, the heat from whose combustion contributed to the vaporization of the water, it was necessary to include the wood's thermal equivalent of anthracite in the quantity of that coal consumed. The pound of dry pine wood burned in the furnace of 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, and dust. Consequently, in the table, 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 method of experimenting above described gives somewhat less than the boiler's true economic vaporization, because at the beginning and end of the experiments much too great a quantity of air in proportion to combustible consumed entered the furnace, the rate of combustion being then very slow, the fire very thin and restricted to portions of the grate only, while the air openings were the same as for the whole grate covered with a bed of incandescent coal. The same cause also makes the rate of combustion and the rate of vaporization somewhat too small.

The steam was saturated, but no priming or foaming was at any time observable.

The rate of combustion per hour per square foot of heating surface in the boiler 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 fuel consumed per hour per square foot of the interior surface is much nearest the truth; and this applies also to the weight of water vaporized per hour per square foot of heating surface, for which calculation the inner surface of the coil pipe should be employed. Although the interior surface is in all cases—whether the hot gases of combustion be inside the tube and the water outside, or the reverse—the least gateway through which the heat has to pass, and may therefore be considered the proper measure, yet the difference of area of the corresponding surfaces between the inside and outside of the coil pipe or tube influences the difference of temperature of these sides, and thus influences the rapidity with which the heat is transmitted from the fire side to the water side. When the hot gases are 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 more efficient than in the case of the hot gases being on the inside and the water on the outside of the tube.

The quantity of water experimentally vaporized has been reduced to what would have been vaporized had the vaporization taken place under the standard atmospheric pressure, and from temperatures of feed water of 100 degrees and 212 degrees Fahrenheit. 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 calculation is made by first ascertaining the number of Fahrenheit units of heat—which call H—required to raise one pound of the feed-water from its experimental temperature to the temperature of the water in the boiler and to vaporize it under the experimental boiler pressure. In other words, H is the total heat in Fahrenheit units in one pound weight of steam of the experimental boiler pressure above the experimental temperature of the feed water. Then, as the number of Fahrenheit units of heat required to raise the temperature of a pound of water at 100 degrees Fahrenheit to the boiler temperature when the pressure is one atmosphere above zero, and to vaporize it under that pressure, is 965.7, the problem is simply to modify the experimental weight in pounds of feed water in the ratio of 965.7 to H. Similarly, for the assumed temperature, 212 degrees Fahrenheit of feed water, and the assumed boiler pressure of one atmosphere above zero, the number of Fahrenheit units of heat required to be imparted to the pound of feed water is 1078.533: the experimental number of pounds of feed water must therefore be modified in the ratio of 1078.533 to H; to obtain the required weight from 212 degrees under the pressure of one atmosphere.

The weight of circulating water, or of superfluous feed water, driven through the boiler, is obtained by calculation. The valve between the boiler and the circulating pump was, during the experiment, kept wide open, so that the boiler water had free access to the pump. The latter is therefore assumed to deliver back into the boiler in bulk at each stroke of the plunger 95 per centum of the space displacement of the plunger per stroke. The pump being single acting, made one delivery per double stroke of the engine pistons, so that the weight of circulating water given in the following table is the product of 95 per centum of the space displacement of the plunger of the circulating pump in cubic feet multiplied by the weight in pounds of the cubic foot of boiler water, and again by the number of double strokes made by the engine pistons.

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.

Date of experiment, June 22, 1882.

TOTAL QUANTITIES.

Duration of the experiment in consecutive hours	10¼
Total number of double strokes made by the steam pistons	107000.
Total number of pounds of anthracite consumed	2528.
Total number of pounds of refuse from the anthracite in ash and dust 334 pounds, in clinker 100 pounds	434.
Total number of pounds of combustible or gasifiable portion of the anthracite consumed	2094.
Per centum of the anthracite in refuse of ash, clinker, and dust	17.1677
Total number of pounds of pine wood consumed	160.
Total number of pounds of feed water pumped into the boiler	21057.
Total number of pounds of water circulating through the boiler additional to the feed water	59300.
Ratio of the circulating water to the feed water as 1.00000	2.81617
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	2605.264
Total number of pounds of combustible consumed, including its equivalent of pine wood, estimating one pound of the latter as equal to 0.4 pounds of anthracite combustible	2158.000

PRESSURES.

Steam pressure in boiler in pounds per square inch above atmosphere	80.0
Steam pressure in boiler in pounds per square inch above zero	94.8
Height of the barometer in inches of mercury	30.16

TEMPERATURES.

Temperature, in degrees Fahrenheit, of the atmosphere	74.
Temperature, in degrees Fahrenheit, of the steam in the boiler	323.79
Temperature, in degrees Fahrenheit, of the feed water	84.
Temperature, in degrees Fahrenheit, of the fire-room	93.

RATE OF COMBUSTION.

Number of pounds of anthracite consumed per hour, including pine wood equivalent	248.12038
Number of pounds of combustible consumed per hour, including pine wood equivalent	205.52381
Number of pounds of anthracite consumed per hour per square foot of grate surface	9.55515
Number of pounds of combustible consumed per hour per square foot of grate surface	7.91475
Number of pounds of anthracite consumed per hour per square foot of heating surface, calculated for the exterior surface of the coil pipe	0.44486
Number of pounds of anthracite consumed per hour per square foot of heating surface, calculated for the interior surface of the coil pipe	0.55719
Number of pounds of combustible consumed per hour per square foot of heating surface, calculated for the exterior surface of the coil pipe	0.36849
Number of pounds of combustible consumed per hour per square foot of heating surface, calculated for the interior surface of the coil pipe	0.46153

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	22025.76319
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	24599.26732
Number of pounds of water vaporized from 100 degrees Fahrenheit by one pound of anthracite	8.45433
Number of pounds of water vaporized from 100 degrees Fahrenheit by one pound of combustible	10.20656
Number of pounds of water vaporized from 212 degrees Fahrenheit by one pound of anthracite	9.44214
Number of pounds of water vaporized from 212 degrees Fahrenheit by one pound of combustible	11.39910

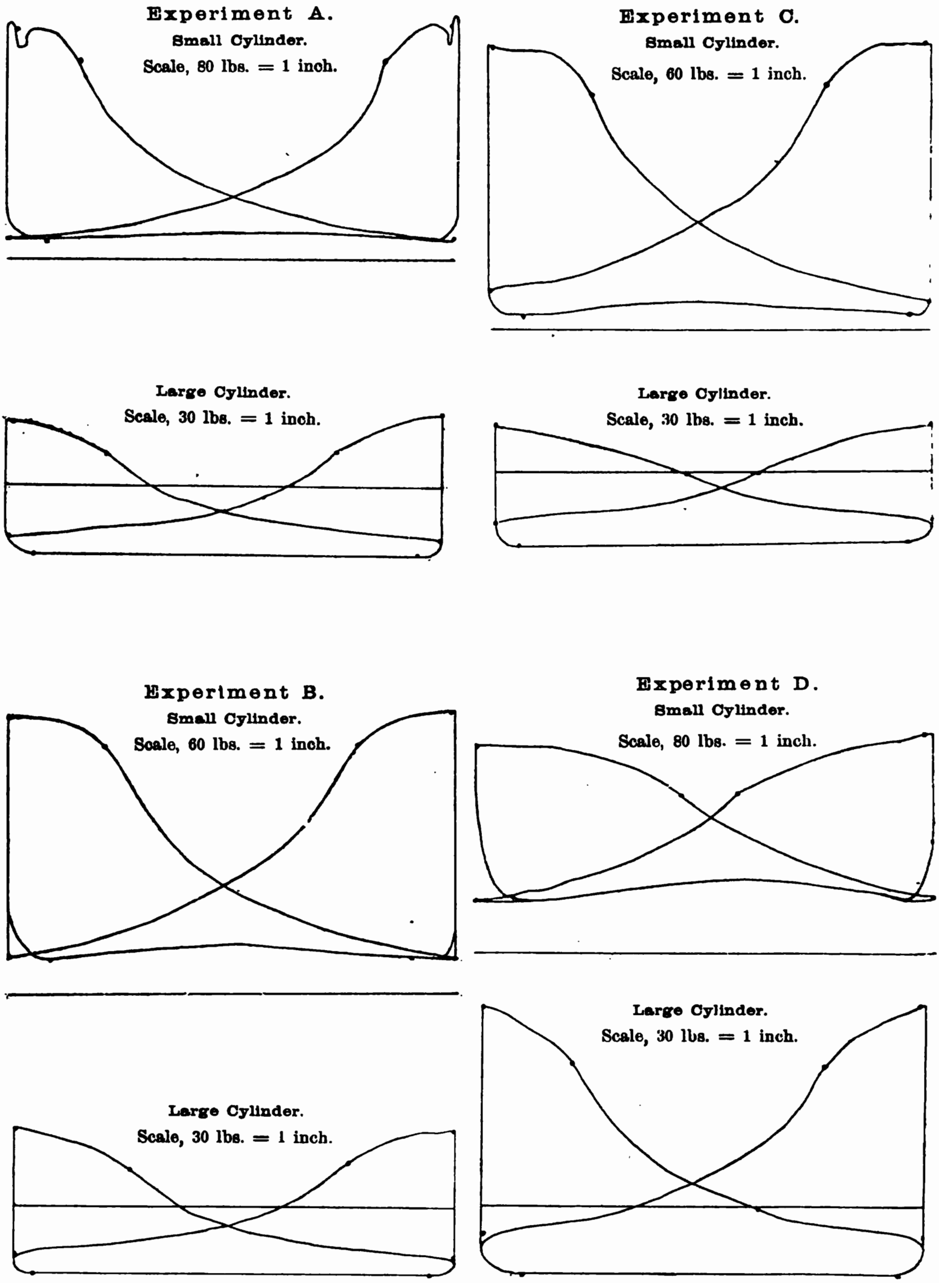
Respectfully submitted.

B. F. ISHERWOOD, *Chief Engineer.*

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

STEAM YACHT SIESTA.

Indicator diagrams, showing the distribution of the pressure in the different experiments.

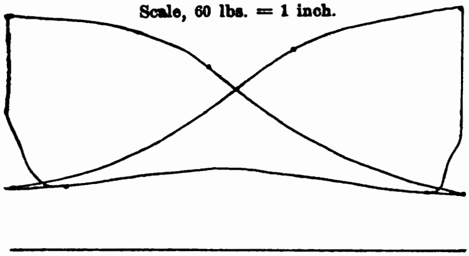


Indicator diagrams, showing the distribution of the pressure, &c.—Continued.

Experiment E.

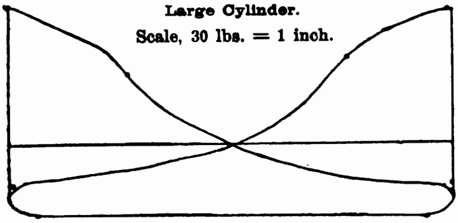
Small Cylinder.

Scale, 60 lbs. = 1 inch.



Large Cylinder.

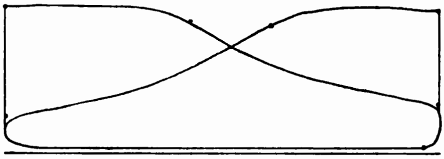
Scale, 30 lbs. = 1 inch.



Experiment G.

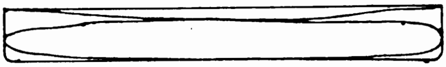
Small Cylinder.

Scale, 80 lbs. = 1 inch.



Large Cylinder.

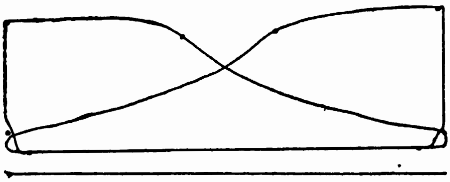
Scale, 40 lbs. = 1 inch.



Experiment F.

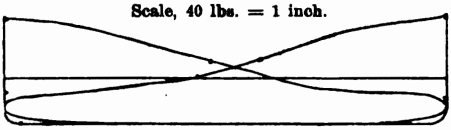
Small Cylinder.

Scale, 80 lbs. = 1 inch.



Large Cylinder.

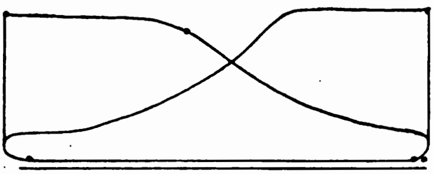
Scale, 40 lbs. = 1 inch.



Experiment H.

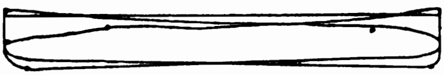
Small Cylinder.

Scale, 40 lbs. = 1 inch.



Large Cylinder.

Scale, 40 lbs. = 1 inch.

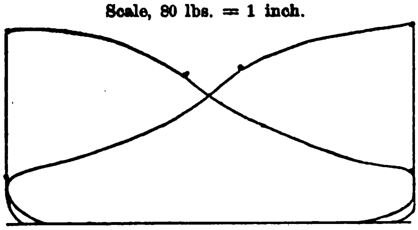


Indicator diagrams, showing the distribution of the pressure, &c.—Continued.

Experiment I.

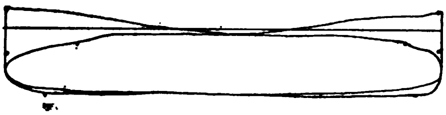
Small Cylinder.

Scale, 80 lbs. = 1 inch.



Large Cylinder.

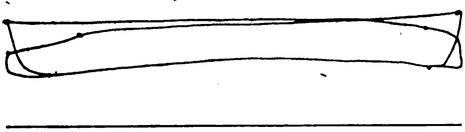
Scale, 30 lbs. = 1 inch.



Experiment J.

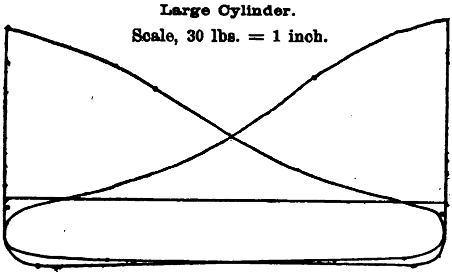
Small Cylinder.

Scale, 80 lbs. = 1 inch.



Large Cylinder.

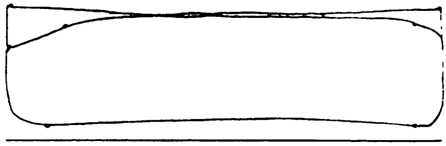
Scale, 30 lbs. = 1 inch.



Experiment K.

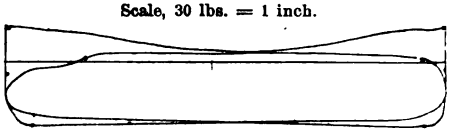
Small Cylinder.

Scale, 60 lbs. = 1 inch.



Large Cylinder.

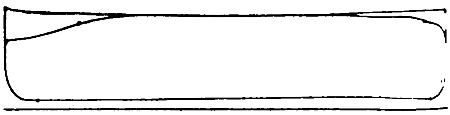
Scale, 30 lbs. = 1 inch.



Experiment L.

Small Cylinder.

Scale, 80 lbs. = 1 inch.



Large Cylinder.

Scale, 30 lbs. = 1 inch.