Compound Corliss Engines.

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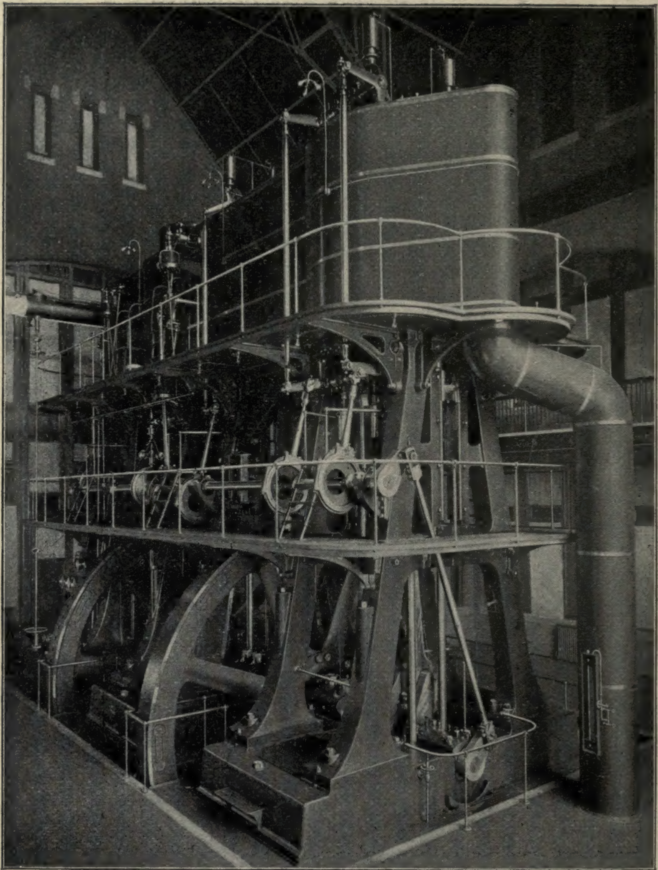

WORLD'S RECORD

30 Million Gallon Pumping Engine, Chestnut Hill, Boston, Mass.

DUTY PER 1,000 LBS. DRY STEAM, 178,497,000 FOOT POUNDS.

THERMAL EFFICIENCY, 21.63%.

COMPOUND CORLISS ENGINES

—BY—

JAMES TRIBE

MEM A. S. M. E.

AUTHOR OF

"COMPOUND ENGINES"

ILLUSTRATED BY DIAGRAMS AND CUTS

FIRST EDITION
1903

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PREFACE

THIS volume was intended as the second edition of my former work "Compound Engines," but so much new material has been added and so thorough has been the revision of the old, that it has virtually become a new book. A new name also seems to be appropriate, for since the nearest approach to perfect thermal efficiency is realized in the Corliss type of Engines, and the variable cut-off features embodied in the same are assumed in all the examples, it has become desirable to adopt the name "Compound Corliss Engines."

In its present form it is an elementary Text-Book on the generation and utilization of heat and the transformation of heat energy into mechanical energy by means of the multi-cylinder Corliss Steam Engine.

Although the work is theoretical in treatment it nevertheless aims to be thoroughly practical in its purpose, all of the examples being based upon actual up-to-date practice, and the data from engines which are actually built and running.

It is written expressly to meet the needs of the steam engine designer whose knowledge of higher mathematics may be limited, and who finds himself handicapped by the complex formulæ usually found in works on this subject. Preference is therefore given to plainly written rules, rather than algebraic formula ; and all figuring is kept well within the range of the ordinary rules of arithmetic, hence it may not be beyond criticism of the technical graduate.

Most of the subject matter and all of the examples are original in treatment, yet great assistance has been had by consulting other authors, among whom are C. H. Haswell, T. M. Goodeve, R. H. Thurston, D. K. Clark, G. C. V. Holmes, William Kent, C. H. Peabody and The Transactions of the Am. Soc. Mech. Engineers, for which due acknowledgment is here made.

JAMES TRIBE.

1903.

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ILLUSTRATIONS

INTRODUCTION.

Every steam engine designer should have before him an ideal purpose at which to aim, and toward which he should make every effort to approach. The ideal of the Steam Engineer is the perfect utilization of every heat unit contained in the fuel burned. And that of the engine Designer to build a machine capable of converting into useful work, all the available heat-energy contained in the steam furnished him. Of course such an ideal is beyond even the most successful results that can ever be attained, and indeed the very anticipation of ever reaching such perfection would in itself be absurd, for the reason that in the steam engine losses are positively inevitable. Nevertheless, the more earnestly we try to approach the ideal, the more likely we are to reap the rewards of our efforts.

It is not intended to slight the importance and value of higher mathematics, yet, fearing that many good rules and formulae lose their usefulness by their complex setting, the author has avoided all unnecessary algebraic calculations, and either omitted or approximated many minor points, and used only such as are strictly necessary for ordinary and practical purposes. Nevertheless, rules and simple formulæ are of paramount importance, and in fact almost indispensible, especially when made and confirmed by actual experience and practice; yet unless a judicious application of them be made, the lack being through, perhaps, want of a knowledge of the circumstances under which the engine is to operate, or inability on the part of the designer, from lack of experience, serious error and possible disaster may be the result. Common sense, therefore, and general knowledge of the natural laws that relate to steam, heat, condensation, expansion, evaporation, etc., together with such rules and tables as have been thoroughly tested, is of the greatest importance to the steam engine designer.

The fundamental principles dealt with in this treatise are, of course, applicable to the study of any type of Compound engine, such as the Compound Locomotive, the Marine engine, or the High-Speed slide valve engine; but all the examples, tables, and general reasoning refer more strictly to modern stationary engines, having a first-class cut-off mechanism of the Corliss principle, by which there can be secured a good, sharp, clean cut-off, automatically controlled to take place at any point of the stroke as the varying load may require.

The student who desires to get the greatest benefit from this treatise should take pencil and paper, and follow all the examples step by step himself, and not be content to simply read them over as given. In this way the method of figuring will soon become comparatively easy and the general principles more thoroughly comprehended. He should also furnish himself with the necessary drafting instruments, board and T square, and lay out for himself the theoretical diagrams as he finds them, beginning with the simple ones and advancing step by step as fast as he understands them. Again, he should not only read over the articles on steam, heat, condensation, etc., but he should study them until he becomes thoroughly familiar with the principles implied. The time thus spent will prove to be of great value when he faces new problems in his professional work. Try it.

The author having had twenty years experience in the design of Corliss engines, and learned the importance of thoroughness in the study of the subject treated, and at the same time the uselessness of overdone technicalities, has endeavored to compile the following pages in such a way as to be of the greatest practical benefit to the student, whoever he may be, if he is only willing to apply himself to the task.

CHAPTER No. 1.

PRELIMINARY INVESTIGATION.

The most skilled and experienced designers have found that a thorough preliminary investigation leading to a full knowledge of all the conditions and peculiarities under which his engine is to operate is indispensable to a wise and proper conclusion, both as to type and also design of the machine.

Every steam plant has a set of conditions peculiar to itself. They may be favorable to best economy or otherwise, but to obtain the most economical results that the peculiar conditions will allow, the design and proportion, the speeds, and ratios of cylinder areas, must all be in keeping with the particular conditions as the designer finds them.

The questions which naturally suggest themselves might be put in the following order, and a few words of explanation under each head may prove helpful:

1. Class of work.
2. Engine-room space.
3. Horse-power required.
4. Number of revolutions per minute.
5. Pressure of steam in engine-room.
6. Condensing or non-condensing.
7. Number of expansion stages preferred.
8. Fluctuation of load.
9. Financial limitations.

§1. **Class of Work.** The many different classes of work to which the steam engine is put, demands many different types and designs. Hoisting machinery with its reversing features, its short and rapid successive periods of go and stop, make the compound engine impracticable; while for pumping machinery of large units, the opposite is true, and especially where uniform load can be maintained for long periods, and the necessary high steam pressure can be provided. Here the multi-cylinder engine finds its greatest usefulness.

Rolling mill engines, with their alternate excessive overload and underload, have much the same objectionable features as hoisting machinery and the advisability of more than single stage expansion is questioned.

The direct connected generator class, which has almost entirely displaced the belt connected, and justly so, is that class of work which stands second only to the pumping engine, for profitable employment of the compound engine. And as the units increase in size, and the sudden load changes decrease in number, the advisability of multiplying the number of expansion stages is also increased.

Each class of work therefore should be carefully considered, and allowed to have proper influence in the choice of the type of engine to be designed.

§ 2. **Engine-Room Space.** Where floor space is limited and height of room unlimited, the vertical engine has the preference, and fortunately so, since the vertical has less friction than the horizontal.

Vertical tandem engines are often installed because of limited floor space, but are not desirable on account of inaccessibility of the low pressure cylinder.

With moderate floor space and moderate height, the horizontal vertical type (high pressure horizontal and low pressure vertical) has advantage over every other style. See § 104.

§ 3. **Horse Power Required.** Careful investigation should be made to ascertain the exact horse-power required, both maximum and minimum. This is especially important in Compound engine design, as the range of load variation should be less in a two stage expansion engine than in a single, and less again in a triple than in a two stage; and still less in a quadruple. Then ascertain the horse-power required when the load is most nearly constant and what proportion of the time it remains so. Figure the minimum so that the expansion line never falls below the back pressure, and avoid forming a loop which is objectionable. § 80.

Figure the indicated horse-power to include that required for friction of engine also. The percentage of friction varies with the style of engine. A good horizontal requires from 8 to 12 per cent. of total. A good vertical, from 6 to 10 per cent. The Glasgow St. R. R. Allis Engine has 94 per cent. mechanical efficiency.

§ 4. **Number of Revolutions Per Minute.** In large units with long stroke, say 6 feet, and Corliss valve gear, the piston speed may be considered safe up to 800 feet or 66 revolutions per minute, but the class of work must be also considered, for instance:

Pumping Engines	17 to 20 R. P. M.
Blowing Engines	40 to 60 R. P. M.
Cotton and Woolen Mills	70 to 80 R. P. M.
Electric Generators	85 to 95 R. P. M.
Smaller Units up to 1,000 K. W. with Corliss valve gear	120 R. P. M.

Milwaukee Street Railway has a 22″ and 46″ × 42″ Corliss belt connected running 120 R. P. M.

§ 5. **Pressure of Steam in Engine-Room.** After ascertaining the boiler pressure near boilers, then measure the length of steam pipe necessary to connect with engine. Assuming the pipe will be properly covered, and of proper size, allow a drop of 1 pound for every 25 feet in length. Excessive length of pipe needs a steam drum, or reservoir, near the engine, with volume equal to that of the high pressure cylinder, to avoid wire drawing in the pipe; but never provide unnecessary area of pipe. The best authorities recommend that the area of the steam pipe should be such as to make the mean velocity of steam not to exceed 6,000 feet per minute. This is done, 1st to prevent loss of pressure due to friction, 2nd to prevent loss of heat by radiation due to excessive surface.

§ 6. **Condensing or Non-Condensing.** There are two conditions which determine whether an engine should be condensing or non-condensing; first the available supply of water for condenser injection, and second the use of the exhaust steam for other purposes. *Water supply.* About 20 feet lift and 500 feet of pipe should be considered as the greatest safe working distance through which the injection water can be drawn by vacuum. If the distance exceeds the above, but the water level is not more than 15 feet below the condenser inlet, then the water can be made to flow by gravitation through pipes, not necessarily strictly air tight, into a dry well made for that purpose near to the condenser. An injection pipe with a foot-valve on the lower end can then be suspended in the center of this well with the valve covered by 3 or 4 feet of water and about 12 inches from the bottom. This reduces the liability of losing the vacuum by leaky joints. The objection to this arrangement is the expense of the necessary excavation, but when once done it costs nothing to maintain. But if the distance to and especially if the lift exceeds the above limit, then condensing should not be attempted. Any idea of pumping the water into the vicinity of the condenser is positively unprofitable, on account of the quantity.

Exhaust Steam. In many cases the exhaust steam may be considered as more profitably used for other purposes outside of the engine, such as heating buildings, cooking, etc., and may not pay to install a condensing apparatus, then the non-condensing type is the only alternative. But it need not be confined to a single cylinder engine, for, providing the necessary boiler pressure (say 150 pounds) can be secured, a compound non-condensing engine of good proportions, and especially where the load is nearly uniform, would be a wise choice.

§ 7. **Number of Expansion Stages.** There are two particular elements which enter into the consideration of, and to some extent determine, the number of cylinders, or stages of expansion. First, the purposes for which the power is to be used and the consequent degree of load fluctuation. § 8. Second, and more important, the limit of steam pressure. § 20. If the desired pressure can be secured and best economy is sought, then use as many stages of expansion as mechanical limitations and variation of load will permit. § 81.

§ 8. **Fluctuation of Load.** Load variation is an enemy to steam economy; but absolute uniformity of load is a practical impossibility. All classes of work must of necessity be subject to more or less fluctuations, the degree of the same differing according to the nature of the work performed; three of the most unfavorable being that of the Rolling Mill, Hoisting Machinery, and Electric Street Railway. In each of these the load changes are very sudden and severe, changing from a great degree of overload to all off almost instantly. Such conditions produce excessive variation of the cylinder temperature and consequently loss by cylinder condensation. On the other hand the most favorable class of work is that of the Pumping Station, especially where the head of water is kept practically uniform, as in reservoir work. This accounts to some extent for the fact that the best records of steam economy are made by Pumping Engines. Between these two extremes lie numerous classes of work having many degrees of load variation. Too much care and thoughtful consideration cannot be given to these conditions. In case of extreme variation, the Quadruple expansion engine, or even the Triple, would be a very unprofitable investment, indeed even the two cylinder compound might be questionable under extreme circumstances. But in the case of pumping machinery, where sufficiently high steam pressure can be had and other conditions are favorable, nothing short of the Triple expansion should be installed.

Mr. Edwin Reynolds says: "For multiple-expansion engines to do their best work requires that the conditions under which the engine is to work must remain constant for long periods, and that these conditions must be definitely known when designing the engine. The marine engine on long voyages illustrates these conditions, and a pumping engine on reservoir work has as nearly perfect conditions as any machine, while the arrangements for obtaining high economy can be more readily supplied than at sea."

§ 9. **Financial Limitations.** This is the problem which often troubles the best of designers. The most economical plant must of necessity have the greatest first cost. First cost limitation means limitation of economy.

Hints to the Purchaser.

Many a steam plant has been permanently crippled by cutting out some of the essential features designed to improve economy. Better sacrifice some of the high polish, than to take chances on leaky valves and pistons. Never install a non-condensing engine to save the expense of a condensing apparatus, if that is the only reason. Put in the most economical engine that circumstances will allow, to do otherwise is false economy.

CHAPTER No. 2.

STEAM.

Steam is said to be "The vapor of water." The quality of steam may be expressed by one of the three following expressions:

Wet Steam.

Dry Saturated Steam.

Superheated Steam.

A few words of explanation under each head:

§ 10. **Wet Steam.** This expression indicates the condition of steam when it is overcharged with moisture, such for instance as when a given volume while doing external work has expanded into a larger volume without additional heat being supplied; or when water is injected into steam, causing partial condensation. Receiver steam is usually wet unless the receiver is fitted with a reheater to re-evaporate the moisture. § 123.

§ 11. **Dry Saturated Steam.** "Steam which is not surcharged with heat after leaving the water from which it was generated," "Dry," because not overcharged with moisture. The word "saturated" implies that it is neither "Wet" nor "Superheated," therefore "Dry Saturated" means dry steam which is not superheated, and is often referred to as "Commercially dry."

All steam tables in general use show temperatures, total heat, latent heat, etc,, of steam which is supposed to be "dry saturated," and if additional heat is introduced, the properties are thereby changed. If increase of temperature without increase of pressure, the steam is then said to be superheated. See § 12.

Definition, Saturated Steam—"The normal condition of steam, generated in free contact with water."

§ 12. **Superheated Steam.** When saturated steam is surcharged with additional heat, and the temperature thereby increased without any increase in pressure, it is then said to be superheated. Def. *"Steam, the temperature of which is higher than the water from which it was generated."*

The rate of cylinder condensation decreases in proportion as the amount of moisture decreases, therefore steam used in compound engines should be superheated if possible.

By superheating, the energy of steam is increased without increasing the pressure beyond its safe limit.

Gaseous Steam. If a volume of steam is isolated from the boiler and raised in temperature by additional heat without even increasing its pressure it behaves more like a perfect gas, and is known sometimes as gaseous steam, and the higher the temperature the more gaseous it becomes. Hine found that by isolation and an increase of temperature of 16 degrees, that it expands almost uniformly as a permanent gas, and approaches more nearly to the adiabatic conditions.

Steam should be superheated for

Single cylinders	50 to 60 degrees.
Double expansion	60 to 80 degrees.
Triple expansion	80 to 100 degrees.

When superheated steam can be furnished, the steam jacket is of less importance, as a greater number of expansions are possible without increasing the cylinder condensation.

By superheating, the friction on the walls of the pipes and passages is reduced, and the areas figured down to increase the velocity, and that in proportion to the degree of superheat. For instance, saturated steam can be figured as 100 feet per second, but with 100 degrees superheat the velocity may be increased to 110 feet per second.

Where to Superheat. Steam cannot be successfully superheated while it remains in contact with the water from which it was generated, but in some types of vertical boilers, in which the upper ends of the tubes project 3 or 4 feet above water level, the steam on passing out to the engine, collects some heat which it retains because it is immediately removed from the water, but at best it cannot be more than a few degrees. The place to superheat should be as near the engine as possible.

Pointers—With highly superheated steam always use
Metallic piston rod packing.
Poppet valves in high press. cylinder.
Mineral oils of high fire test.

§ 13. **Steam Separator.** If superheating cannot be done, the next best thing is to install a good Steam Separator (Fig. 1), to take care of the condensation water which may have accumulated while on the way from the boiler. Let it be placed as near the engine as possible, and with proper drain pipe and steam trap connections. As the steam leaves the separator, having deposited its water, it may be considered "Commercially Dry" and the density, etc., for any given pressure will agree very closely to Table No. 1.

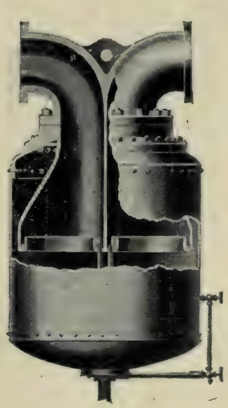


Fig. 1.

TABLE No. 1.

PRESSURES AND WEIGHTS OF STEAM.

PRES. ABSO.	WT. OF ONE C. FT.	PRES. ABSO.	WT. OF ONE C. FT.	PRES. ABSO.	WT. OF ONE C. FT.	PRES. ABSO.	WT. OF ONE C. FT.	PRES. ABSO.	WT. OF ONE C. FT.
1	.003	41	.0996	82	.1913	123	.2801	164	.3674
2	.0058	42	.102	83	.1935	124	.2822	165	.3695
3	.0085	43	.1042	84	.1957	125	.2845	166	.3715
4	.0112	44	.1065	85	.198	126	.2867	167	.3736
5	.0138	45	.1089	86	.2002	127	.2889	168	.3756
6	.0163	46	.1111	87	.2024	128	.2911	169	.3777
7	.0189	47	.1133	88	.2044	129	.2933	170	.3798
8	.0214	48	.1156	89	.2067	130	.2955	171	.3818
9	.0239	49	.1179	90	.2089	131	.2977	172	.3838
10	.0264	50	.1202	91	.2111	132	.2999	173	.3859
11	.0289	51	.1224	92	.2133	133	.302	174	.3879
12	.0314	52	.1246	93	.2155	134	.304	175	.3899
13	.0338	53	.1269	94	.2176	135	.306	176	.3921
14	.0362	54	.1291	95	.2198	136	.308	177	.3943
14.7	.0380	55	.1314	96	.2219	137	.3101	178	.3965
15	.0387	56	.1336	97	.2241	138	.3121	179	.3987
16	.0411	57	.1364	98	.2263	139	.3142	180	.4009
17	.0435	58	.138	99	.2285	140	.3162	181	.4031
18	.0459	59	.1403	100	.2307	141	.3184	182	.4053
19	.0483	60	.1425	101	.2329	142	.3206	183	.4074
20	.0507	61	.1447	102	.2351	143	.3228	184	.4096
21	.0531	62	.1469	103	.2373	144	.325	185	.4117
22	.0555	63	.1493	104	.2393	145	.3273	186	.4138
23	.0580	64	.1516	105	.2414	146	.3294	187	.4159
24	.0601	65	.1538	106	.2435	147	.3315	188	.4180
25	.0625	66	.156	107	.2456	148	.3336	189	.4201
26	.065	67	.1583	108	.2477	149	.3357	190	.4222
27	.0673	68	.1605	109	.2499	150	.3377	191	.4243
28	.0696	69	.1627	110	.2521	151	.3395	192	.4264
29	.0719	70	.1648	111	.2543	152	.3417	193	.4285
30	.0743	71	.167	112	.2564	153	.344	194	.4306
31	.0766	72	.1692	113	.2586	154	.3462	195	.4327
32	.0789	73	.1714	114	.2607	155	.3484	196	.4348
33	.0812	74	.1736	115	.2628	156	.3506	197	.4369
34	.0835	75	.1759	116	.2649	157	.3528	198	.4390
35	.0858	76	.1782	117	.2652	158	.3549	199	.4411
36	.0881	77	.1804	118	.2674	159	.357	200	.4431
37	.0905	78	.1826	119	.2696	160	.359	225	.4947
38	.0929	79	.1848	120	.2738	161	.3611	250	.5464
39	.0952	80	.1869	121	.2759	162	.3632	300	.6486
40	.0974	81	.1891	122	.278	163	.3653	350	.7498

For close figuring, consult Steam Tables by C. H. Peabody.

TABLE No. 2.

LATENT HEAT OF STEAM.

ABSO. PRESS. LBS.	LATENT HEAT. UNITS.	ABSO. PRESS. LBS.	LATENT HEAT. UNITS.	ABSO. PRESS. LBS.	LATENT HEAT. UNITS.	ABSO. PRESS. LBS.	LATENT HEAT. UNITS.
0	1082	41	925.4	83	892.5	130	868.7
1	1043	42	924.4	84	891.9	135	866.6
2	1026	43	923.3	85	891.3	140	864.6
3	1015.3	44	922.3	86	890.7	145	862.6
4	1007.2	45	921.3	87	890.1	150	860.6
5	1000.7	46	920.4	88	889.5	155	858.7
6	995.2	47	919.4	89	888.9	160	856.9
7	990.5	48	918.5	90	888.4	165	855.1
8	986.2	49	917.5	91	887.8	170	853.3
9	982.4	50	916.6	92	887.2	175	851.6
10	979.0	51	915.7	93	886.7	180	849.9
11	975.8	52	914.9	94	886.1	185	848.2
12	972.8	53	914.0	95	885.6	190	846.6
13	970.0	54	913.1	96	885.0	195	845.0
14	967.4	55	912.3	97	884.5	200	843.4
14.7	965.7	56	911.5	98	884.0	210	840.4
15	965.0	57	910.6	99	883.4	220	838.6
16	962.7	58	909.8	100	882.9	230	835.8
17	960.5	59	909.0	101	882.4	240	833.1
18	958.3	60	908.2	102	881.9	250	830 5
19	956.3	61	907.5	103	881.4	260	827.9
20	954.4	62	906.7	104	880.8	270	825.4
21	952.6	63	905.9	105	880.3	280	823.0
22	950.8	64	905.2	106	879.8	290	820.6
23	949.1	65	904.5	107	879.3	300	818.3
24	947.4	66	903.7	108	878.8	325	812.8
25	945.8	67	903.0	109	878 3	350	807.5
26	944.3	68	902.3	110	877.9	375	801.7
27	942.2	69	901.6	111	877.4	400	797.9
28	941.3	70	900.9	112	876.9	425	793.5
29	939.9	71	900.2	113	876.4	450	789.1
30	938.9	72	899.5	114	875 9	500	781.0
31	937.2	73	898.9	115	875.5	550	773.5
32	935.9	74	898.2	116	875.0	600	766.3
33	934.6	75	897.5	117	874 5	650	759.6
34	933.4	76	896.9	118	874.1	700	753.3
35	932.2	77	896.2	119	873.6	750	747.2
36	931.0	78	895.6	120	873.2	800	741.4
37	929.8	79	895.0	122	872.3	850	735.8
38	928.7	80	894.3	124	871.4	900	730.6
39	927.6	81	893.7	126	870.5	950	725.4
40	926.5	82	893.1	128	869.6	1000	720.3

§ 14. **Latent Heat of Steam.** *Def.—"Latent heat of steam, is that heat which is communicated to water while being evaporated into steam without changing its temperature'," or "Heat which is not sensible to the thermometer."* For instance, one pound weight of steam at 212° Fah. and atmospheric pressure contains 965.7 units of latent heat. Of this quantity, 894 units are required for the formation of steam at atmospheric pressure, and 71.7 units for expansion against the atmospheric pressure. Again, for higher pressure, say 200 pounds abso., one pound weight 381.6° Fah. contains 843.4 units of latent heat. Of this quantity 759.4 units are required for the formation of steam at 200 pounds pressure, and 84 units for expansion against the surrounding steam, thus:

200 pounds per sq. in=28,800 pounds per sq. ft.$\times$ 2.27 cubic feet of volume = 65,376 foot pounds, which when divided by 778, the mechanical equivalent of 1. unit, = 84 units and 843.4—84 = 759.4 units for the formation of steam against the 200 pounds pressure.

To demonstrate that one pound weight of steam at 212° Fah. contains 965.7 units of latent heat, take 1 pound of water at 212°, and evaporate it into steam, at 14.7 pounds pressure, then take 9 pounds of water at 105° Fah., and inject it into the steam, thus condensing it, and the result will be 10 pounds of water at 212° F., each pound of the 9 pounds having absorbed 107.3 units or a total of $107.3 \times 9 = 965.7$ units, which was latent.

The quantity of latent heat per pound of steam decreases as the total heat increases; that is to say the higher the temperature and pressure of the steam, the less will be the latent heat. This decrease of latent heat in saturated steam is at the rate of .721 for each 1 degree Fah. increase of temperature. For example, test two different pressure, first atmos. 14.7 at 212° Fah., latent heat being 965.7, and compare with 200 pounds abso. at 381.6 F. and latent heat 843.4, we see that

$$381.6 - 212 = 169.6$$

$$\text{and } 169.6 \times .721 = 122.28$$

The difference in latent heat (Table No. 2) is

$$965.7 - 843.4 = 122.3.$$

Try another example by comparing atmospheric pressure with 400 abso. pressure, (Table 2).

$$444.9 - 212 = 232.9$$

$$\text{and } 232.9 \times .721 = 167.9$$

Difference in latent heat is

$$965.7 - 797.9 = 167.8$$

The steam engine is incapable of transforming Latent heat into mechanical energy. When steam has done its work, and has expanded down to the lowest pressure practicable, there still remains in it a quantity of heat (Latent) which is exhausted either into the atmosphere or into the condenser as the case may be. For instance, if expanded down to 5 pounds abso. there is still 1,000 heat units in each pound weight to be absorbed by the injection water, and which was consequently unavailable for performing work. Part of this heat, however, may be utilized by providing a hot well to retain part of the overflow water from the condenser, and pumping it back to the boiler as feed water; but since the overflow water is about 17 to 20 times more than the boiler requires, only a comparatively small quantity can be thus saved. If this low pressure steam (exhaust) could be picked up before entering the condenser, and returned to the boiler, the latent heat could be saved, but how to do it is the question.

§15. **Specific Heat of Steam.** The expression "Specific Heat" means the capacity for heat. The specific heat of saturated steam is .305, and expresses the ratio of the heat capacity of steam, to that of water which is 1.

The specific heat of steam which is generated in contact with water (not superheated) is constant for all pressures; and may be verified by following rule and examples:

Rule.—"Total heat, minus latent heat at zero pressure, viz., 1082, divided by temperature corresponding to total heat."

Example.—Find specific heat of steam at one pound absolute pressure. Total heat = 1113.1 units. Temperature=102.1. See table No. 4.

$$\therefore \quad \frac{1113.1 - 1082}{102.1} = .305$$

Again for atmospheric press., viz., 14.7 in which total heat = 1146.6 and Tem. = 212

$$\therefore \quad \frac{1146.6 - 1082}{212} = .305$$

and again, for 1,000 pounds abso. in which total heat = 1248.7 and Tem. Fah. = 546.8

$$\therefore \quad \frac{1248.7 - 1082}{546.8} = .305$$

The above examples therefore show that while water requires 1 unit of heat to raise its temperature 1 degree Fah., Saturated Steam requires only .305 of a unit of additional heat to raise its temperature 1 degree. Not that .305 of a unit is equal to 1 degree, but that by virtue of adding the .305 of a unit there has been a giving up from the steam .695 of a unit of latent heat, hence

$$.305 + .695 = 1 \text{ unit.}$$

The above is true only with saturated steam. Regnalt found the specific heat of superheated steam to be .475 under constant pressure.

In round figures, therefore, for every one unit of heat imparted to saturated steam, the temperature is raised 3 degrees, and for superheated steam 2 degrees.

§ 16. **Temperature of Steam.** The word temperature does not express the total quantity of heat in steam, but only that part of it which is sensible to, and can be indicated by, the thermometer, the other part being latent.

When by reason of heat, water is evaporated into steam at atmospheric pressure, there is no change of temperature during the process.

All temperatures referred to in this book are by the Farenheit Thermometer, it being the one most generally adopted in English speaking countries.

The temperature of water at freezing point is taken at 32° Fah. and from this temperature, by common consent, the total heat in steam is reckoned, and not from zero. Absolute zero has been fixed at 461.2 degrees below the zero of the Fahrenheit scale.

The temperature of boiling water at sea level is 212° Fah., and 1° Fah. less for every 550 feet of elevation. For instance, water will boil on Pike's Peak, Col., an elevation of 14,216 feet above sea level, at a temperature of 186.2° Fah.

Temperature, Def.—"The temperature of a substance is a quantity of sensible heat possessed by it, and capable of being communicated to other bodies."

CAUTION. Avoid confusion between the words "Degrees of heat" and "Units of heat." Thermometer readings are always "Degrees," not "Units."

§ 17. **Available Heat in Steam.** The available heat in steam is that portion of its heat which is free and capable of passing from the steam into other substances of lower temperatures, without changing its state from steam to water. We therefore understand that the term "available heat" refers to heat in steam which is available as energy for exerting force upon a moving piston. Available heat is sensible to the thermometer, latent heat is not. All heat in steam is not available for energy.

The total heat in steam, atmospheric pressure, is 1146.6 units, but the temperature is only 212° Fah., and if reckoned from 32° F., common practice, then $212°—32°=180°$ F. available. But in the steam engine even though it is expanded down below 6 pounds abso., or

TABLE No. 3.

AVAILABLE HEAT IN STEAM.

Heat Convertible into Mechanical Energy by the Steam Engine.

ABSO. PRESS.	Convertible Units with Back Press. of		ABSO. PRESS.	Convertible Units with Back Press. of		ABSO. PRESS.	Convertible Units with Back Press. of	
	5 LBS. ABSO.	6 LBS. ABSO.		5 LBS. ABSO.	6 LBS. ABSO.		5 LBS. ABSO.	6 LBS. ABSO.
14.7	49.7	41.9	60	130.2	122.4	110	172.2	164.4
15	50.7	42.9	61	131.3	123.5	115	175.5	167.7
16	54.0	46.2	62	132.4	124.6	120	178.7	170.9
17	57.1	49.3	63	133.4	125.6	125	181.8	174.0
18	60.1	52.3	64	134.5	126.7	130	184.8	177.0
19	62.9	55.1	65	135.5	127.7	135	187.7	179.9
20	65.6	57.8	66	136.5	128.7	140	190.5	182.7
21	68.2	60.4	67	137.5	129.7	145	193.2	185.4
22	70.7	62.9	68	138.5	130.7	150	195.9	188.1
23	73.1	65 3	69	139.5	131.7	155	198.4	190.6
24	75.5	67.7	70	140.4	132.6	160	201.0	193.2
25	78.7	69.9	71	141.4	133.6	165	203.2	195.4
26	79.9	72.1	72	142.3	134.5	170	205.7	197.9
27	82.0	74.2	73	143.3	135.5	175	208.0	200.2
28	84.0	76.2	74	144.2	136.4	180	210.3	202.5
29	86.0	78.2	75	145.1	137.3	185	212.8	205.0
30	88.9	80.1	76	146.0	138.2	190	215.0	207.2
31	89.8	82.0	77	146.9	139.1	195	217.2	209.4
32	91.7	83.9	78	147.8	140,0	200	219.3	211.5
33	93.4	85.6	79	148.6	140.8	205	221.4	213.6
34	95.2	87.4	80	149.5	141.7	210	223.4	215.6
35	96.9	89.1	81	150.4	142.6	215	225.4	217.6
36	98.5	90.7	82	151.2	143.4	220	227.4	219.6
37	100.2	92.4	83	152.1	144.3	230	231.3	223.5
38	101.7	93.9	84	152.9	145.1	240	235.0	227.2
39	103.3	95.5	85	153.7	145.9	250	238.6	230.8
40	104.8	97.0	86	154.5	146.7	260	242.1	234.3
41	106.3	98.5	87	155.4	147.6	270	245.5	237.7
42	107.8	100.0	88	156.2	148.4	280	248.7	240.9
43	109.2	101.4	89	157.0	149.2	290	251.9	244.1
44	110.6	102.8	90	157.7	149.9	300	255.1	247.3
45	112.0	104.2	91	158.5	150.7	350	269.7	261.9
46	113.4	105.6	92	159.3	151.5	400	272.6	274.8
47	114.7	106.9	93	160.1	152.3	450	284.3	286.5
48	116.0	108.2	94	160.8	153.0	500	305.1	297.3
49	117.3	109.5	95	161.6	153.8	550	315.2	307.4
50	118.6	110.8	96	162.3	154.5	600	324.6	316.8
51	119.8	112.0	97	163.1	155.3	650	333.4	325.6
52	121.0	113.2	98	163.8	156.0	700	341.8	334.0
53	122.2	114.4	99	164.5	156.7	750	349.8	342.0
54	123.4	115.6	100	165.3	157.5	800	356.3	349.5
55	124.6	116.8	101	166.1	158.3	850	363.5	356.7
56	125.8	118.0	102	166.8	159.0	900	371.4	363.6
57	126.9	119.1	103	167.5	159.7	950	377.0	370.2
58	128.0	120.2	104	168.2	160.4	1000	384.5	376.7
59	129.1	121.3	105	168.8	161.0	......	......	......

170° Fah., before being thrown out into the condenser, in which case, out of the 1146.6 units in steam at atmospheric pressure, there is only $212 - 170 = 42$ heat units available for energy. Again, taking one pound of steam at 180.3 pounds gauge or 195 abso., the total heat is 1197.7 units, the temperature is 379.5° Fah. Reckoning from 32° the available heat would be 379.5 less $32 = 347.5$ units. All that can be transformed into mechanical energy, allowing the steam to expand to 6 pounds abso., as may be the case in a Compound condensing engine, would be as per Table No. 3, page 28,

$$347.5 - 138.1 = 209.4 \text{ units.}$$

Herein lies the extreme limit of economy of an engine having absolutely no waste and capable of converting the whole of the available heat into mechanical energy. In such an engine the value of one pound of steam of 195 pounds abso., with back pressure down to 6 pounds abso., as above, and if expressed in foot pounds, and on the basis of 778 foot-pounds per heat unit (§ 39) would be

$$209.4 \times 778 = 162{,}913 \text{ foot-pounds.}$$

Sometimes the back pressure is as low as 4½ pounds abso., but the same reasoning holds good.

§ 18 **British Thermal Unit.** The British Thermal Unit is a thermal unit in which the unit of weight is taken as 1 pound, and the unit of heat as 1 degree Fah.

Definition. "The quantity of heat required for raising the temperature of one pound of water, at or near the temperature of greatest density, viz., 39.1 Fah: through one degree Fah." This unit is usually written as B. T. U.

The efficiency and economy of the compound engine should be measured and expressed by the Ratio between the B. T. Us. consumed, and the horse power developed.

The British Thermal Units in Steam as reckoned

TABLE No. 4.

HEAT IN STEAM.								
ABSO. PRESS.	TEM. FAHR.	TOTAL HEAT ABOVE 2°	ABSO. PRESS.	TEM. FAHR.	TOTAL HEAT ABOVE 32°	ABSO. PRESS.	TEM. FAHR.	TOTAL HEAT ABOVE 32°
0	32.	1091.7	43	271.5	1164.7	88	318.5	1179.1
1	102.1	1113.1	44	272.9	1165.2	90	320.0	1179.6
2	126.3	1120.5	45	274.3	1165.6	92	321.6	1180.0
3	141.6	1125.1	46	275.7	1166.0	94	323.1	1180.5
4	153.1	1128.6	47	277.0	1166.4	96	324.6	1181.0
5	162.3	1131.4	48	278.3	1166.8	98	326.1	1181.4
6	170.1	1133.8	49	279.6	1167.2	100	327.6	1181.8
7	176.9	1135.9	50	280.9	1167.6	102	329.0	1182.3
8	182.9	1137.7	51	282.1	1168.0	104	330.4	1182 7
9	188.3	1139.4	52	283.3	1168.4	106	331.8	1183.1
10	193.2	1140.9	53	284.5	1168.7	108	333.2	1183.6
11	197.8	1142.3	54	285.7	1169.1	110	334.5	1184.0
12	202.0	1143.5	55	286.9	1169.4	115	337.8	1185.0
13	205.9	1144.7	56	288.1	1169.8	120	341.0	1185.9
14	209.6	1145.9	57	289.1	1170.1	125	344.1	1186.9
14.7	212.	1146.6	58	290.3	1170.5	130	347.1	1187.8
15	213.0	1146.9	59	291.4	1170.8	135	350.0	1188.7
16	216.3	1147.9	60	292.5	1171.2	140	352.8	1189.5
17	219.4	1148.9	61	293.6	1171.5	145	355.5	1190.4
18	222.4	1149.8	62	294.7	1171.8	150	358.2	1191.2
19	225.2	1150.6	63	295.7	1172.1	155	360.7	1192.0
20	227.9	1151.5	64	296.8	1172.4	160	363.3	1192.7
21	230.5	1152.2	65	297.8	1172.8	165	365.7	1193.5
22	233.0	1153.0	66	298.8	1173.1	170	368.2	1194.2
23	235.4	1153.7	67	299.8	1173.4	175	370.5	1194.9
24	237.8	1154.5	68	300.8	1173.7	180	372.8	1195.7
25	240.0	1155.1	69	301.8	1174.0	185	375.1	1196.3
26	242.2	1155.8	70	302.7	1174.3	190	377.3	1197.0
27	244.3	1156.4	71	303.7	1174.6	195	379 5	1197.7
28	246.3	1157.1	72	304.6	1174.8	200	381.6	1198.3
29	248.3	1157.7	73	305.6	1175.1	205	383.7	1199.0
30	250.2	1158.3	74	306.5	1175.4	210	385.7	1199.6
31	252.1	1158.8	75	307.4	1175.7	215	387.7	1200.2
32	254.0	1159.4	76	308.3	1176.0	220	389.7	1200.8
33	255.7	1159.9	77	309.2	1176.2	230	393.6	1202.0
34	257.5	1160.5	78	310.1	1176.5	240	397.3	1203.1
35	259.2	1161.0	79	310.9	1176.8	250	400.9	1204.2
36	260.8	1161.5	80	311.8	1177.0	260	404.4	1205.3
37	262.5	1162.0	81	312.7	1177.3	270	407.8	1206.3
38	264.0	1162.5	82	313.5	1177.6	280	411.0	1207.3
39	265.6	1162.9	83	314.4	1177.8	290	414.2	1208.3
40	267.1	1163.4	84	315.2	1178.1	300	417.4	1209.2
41	268.6	1163.9	85	316.0	1178.3	325	424.7	1211.4
42	270.1	1164.3	86	316.8	1178.6	350	432.0	1213.7

from water at 32° F., is the sum of sensible heat and latent heat. For instance,

One pound of water heated from 32° to 212° requires	180 units
Specific heat of water at 212° above that at 32° is	.9 "
One pound of water at 212° requires 965.7 units of latent heat for its evaporation into steam at same temperature.	965.7
Total B. T. Us. . .	1,146.6

§ 19. **Economy of High Pressure Steam.** The economy of using high pressure has become more evident every year and especially since the introduction of the modern compound engine.

History. In the year 1800 boiler pressure seldom exceeded atmospheric pressure. In 1814 one of the finest ships, "Arrogant," was designed for 6 pounds above atmosphere, and soon after Woolf built a two cylinder engine for 40 pounds gauge. But it was not until 1850 that it exceeded 40 or 45 pounds. By the year 1870 increase was made to 80 pounds gauge, which was seldom exceeded until the more general introduction of the multicylinder engine with the consequent higher rate of expansion.

Therefore, with the desire for increase in the rate of expansion, together with the modern improvements in boiler design and construction, the demand for steam pressure as high as 300 and 350 pounds is justifiable.

Comparative Duties and Pressures. A comparison of duties in relation to steam pressures can be seen by a brief historical review.

In Pumping Engines, based on million foot pounds per 100 pounds of Coal or 1,000 pounds of water or dry steam:

1806	Newcomb Engine	15 lbs. press.....	10.5 million
1876	Leavitt, Lynn	75 " "	32 "
1878	Corliss, Pawtucket	120 " "	123 "
	Reynolds, Detroit	125 " "	143 "
1898	Snow, Indianapolis	155 " "	167 "
1898	Allis, Boston	185 " "	178 "

Comparative Economy of Ocean Steamers. Based on weight of coal per H.-P. per hour:

1840	Cunard, Britanica,	12 lbs. press. =	5 lbs. coal
1871	White Star, Oceanic,	65 lbs. press. =	2 lbs. coal
1889	White Star, Teutonic,	180 lbs. press. =	1.6 lbs. coal

United States Navy.

1861	S. S. Warrior,	22 lbs. press.	= 5 lbs.
1872	" Devastation,	30 " "	= 4.45 "
1878	" Inflexible,	61 " "	= 2.74 "
1888	" San Francisco,	135 " "	= 2.40 "
1892	" Oregon,	156 " "	= 2.14 "
1897	" Quail,	220 " "	= 1.64 "
1898	" Diadem,	291 " "	= 1.59 "

Notice above that where pressure is increased the economy is improved.

Gain in Mean Eff. Pressure. The economy of high pressure is also seen by comparing the value of any two pressures. For instance, the economy of using 90 pounds press. in preference to 80 pounds, to perform the same work in each case, but cutting off earlier in proportion, will be seen by following example, showing gain in M. E. P. by the same weight of steam:

Assume same terminal press. in each case, viz., 14.9
Steam at 104 lbs. abso. cutting off $\frac{1}{5.5}$ = 36 lbs. M. E. P.
Steam at 94. lbs. abso. cutting off $\frac{1}{5}$ = 34 lbs. M. E. P.

Refer to Table No. 1.

Now, since one c. ft. at 94 lbs. abso. = .2176 lbs. weight

$$\therefore \quad \tfrac{1}{5} \text{ of c. ft.} = \frac{.2176}{5} = .0435 \text{ lbs.}$$

and since one c. ft. at 104 lbs. abso. = .2393 lbs. weight

$\therefore \quad \frac{1}{5.5}$ of a c. ft. $= \frac{.2393}{5.5} = .0435$ lbs. also.

Showing that a given weight of 90 pounds steam, compared with the same weight of 80 pounds steam, a gain in work, measured by M. E. P. $= 36 - 34 = 2$ pounds, nearly 5.6 per cent. theoretically, but practically say about 5 per cent. gain.

It may be interesting and instructive to also make a comparison between the work done by another weight of steam, viz., .0171 pounds, under two different pressures and volumes, 1st, with 50 pounds abso., 2nd, with 250 pounds abso. pressure, by the following example:

Ex. 1st. Steam at 50 pounds abso. cutting off at $\frac{1}{7}$ stroke and exhausting against back pressure of 4 lbs., equals 17 pounds M. Eff. P., and

2nd. Steam at 250 pounds abso. cutting off at $\frac{1}{32}$ stroke and exhausting against back pressure of 4 lbs., equals 30.8 pounds M. Eff. P.

Again, 1st. Since one c. ft. of steam at 50 pounds $= .1202$ pounds. Table No. 1.

then $\frac{1}{7}$ of a c. ft. $= \frac{.1202}{7} = .0171$,

and 2nd. Since one c. ft. at 250 pounds $= .5464$,

then $\frac{1}{32}$ of a c. ft. $= \frac{.5464}{32} = .0171$ also,

showing that with the same weight of steam, viz., .0171 pounds, under two different conditions, the work performed, theoretically, is as 17 is to 30.8, or a gain of 13.8 M. E. P., which is equal to 81 per cent. in favor of the 250 pounds pressure.

Now, of course, the above number of expansions, viz., 32, would not be practicable in one cylinder, but by the introduction of more cylinders, that number may easily be made, and the loss by condensation considerably reduced, and also the work more evenly distributed, hence the importance of compounding.

§ 20. **Desirable Steam Pressures.** Often the engine designer has no choice as to the pressure of the steam to be used, but is exoected to take it as given; but in so far as he has a choice the following hints may be of value: *Minimum Pressures.* It would be of but little value commercially, and would not pay to compound, unless the steam pressure in the engine room is at the very least.

MINIMUM.

For double expansion .... 75 lbs. gauge.
For triple expansion ....135 lbs. gauge.
For quadruple expansion ......185 lbs gauge.

For instance, in a two-cylinder compound condensing engine, with steam at 75 pounds, the greatest number of expansions commercially profitable would not be more than twelve, while a single cylinder condensing, other things being equal, may be as much as eight, and the little difference gained would not warrant the extra first cost of construction. The same is true in the case of a triple-expansion engine; steam at 135 pounds gauge would at best be expanded twenty-four times, while as much as 22 to 24 can reasonably be expected from a double-expansion condensing engine. And, again, in a quadruple-expansion, with steam as low as 185 pounds gauge, about 32 expansions would be the minimum limit, and quite that number can be made in a triple with equal commercial economy. *Best Pressures.* For Compound engines, the most desirable steam pressures, other things being equal, would be about as shown by Table No. 5.

TABLE No. 5.

DESIRABLE STEAM PRESSURES FOR COMPOUND ENGINES.		
TYPE OF ENGINE.	CONDENSING COMPOUND.	NON-COND. COMPOUND.
Double Expansion........................	130 lbs. G.	170 lbs. G.
Triple Expansion........................	185 "	250 "
Quadruple Expansion........................	350 "	

§ 21. Reason Why High Pressure Steam is More Economical Than Low. The reason why high pressure is more economical than low, is due to the fact that under higher pressure a greater part of the latent heat becomes convertible into energy, that is to say a certain portion of the heat becomes available under high pressure which under low pressure would remain latent. By reference to Table 4 it may be seen that as the pressure increases, the total heat increases; but by Table 2 that while the total heat increases, the latent heat decreases, and that a larger percentage of the total heat becomes available and capable of being transformed into useful work. For instance, the total heat of steam at 14.7 pounds abso. is 1146 units, and at 200 pounds abso. it is 1198, an increase of 52 units. But the latent heat of steam at the same pressures is 966 and 843 units, showing a decrease of 123 units; by which we see that for the additional 52 units in total heat, 123 units of latent heat has become available.

At first thought the above may be objected to on the ground that when steam is expanded down to a lower pressure, and then discharged into a condenser, there is more latent heat in the steam thus discharged after expansion, than there was before expansion. For instance, Table No. 2 shows that at 5 pounds abso. the latent heat is 1,000 units, while that at 200 pounds abso. is only 843.4 units. The objection is true when considering the latent heat per pound weight of steam, but it must be remembered that all the steam which has been actually condensed during expansion has given up all of its latent heat, and is no longer steam but water, consequently there is less weight of steam proper, after expansion, than there was before.

§ 22. Value of One Heat Unit Under Different Pressures. A single heat unit added to high pressure steam, will increase its pressure much more than if added to low pressure steam, as may be seen from the following two examples, adding 13° in each case:

1st example. Add 13 degrees F.,

to steam of 14.7 pounds press., which is 212°,
thus $212° + 13° = 225°$,
and steam at $225° = 19$ pounds press.,
increase $19 - 14.7 = 4.3$ pounds.

2nd example. Add 13 degrees F.,

to steam of 300 pounds press., which is 417° F.,
thus $417° + 13° = 430°$,
and steam at $430° = 350$ pounds,
increase $350 - 300 = 50$ pounds,

showing a difference of

$50 - 4.3 = 45.7$ pounds,

in favor of the higher steam pressure.

§ 23. **Velocity of Steam.** In providing proper areas of pipes, and ports, note the following:

Steam flows into vacuum at a velocity of 1,550 feet per second. Into atmosphere at 650 feet per second.

Live steam through piping should not exceed, for standard, 100 feet per sec., and for highly superheated 110 feet per sec.

Live steam through inlet ports to a cylinder should not exceed 130 feet per sec., and exhaust 90 feet per sec.

These areas are figured on the basis of the area and speed of piston.

WATER FOR STEAM.

"In the study of steam, water plays an important part."

1 Cubic inch of water $= \begin{cases} .0361 \text{ pounds.} \\ .0043 \text{ gallons.} \end{cases}$

1 Cubic foot $= \begin{cases} 62.42 \text{ pounds.} \\ 7.48 \text{ gallons.} \end{cases}$

1 U. S. gallon = 8.34 pounds.

1 pound per sq. inch = 2.31 feet head.

1 Foot head = .433 lbs. per sq. inch.

1 pound weight = 27.7 cubic inches.

All water used in the boiler should be as pure as possible.

§ 24. **Impure Water.** All waters are more or less impure. River water contains some organic matter, such as clay, sand, etc., which should be filtered out before feeding to the boiler. Lime contained in feed water, deposits a scale on the boiler plates, and being a bad conductor of heat reduces the efficiency in proportion to its thickness. Lime scale $\frac{1}{16}$ of an inch thick robs the heating surface of 10 per cent. of its conductivity, and if deposited on that part of the shells nearest to the furnace it is liable to cause blisters and prove serious. One cubic foot of pure water weighs 62.5 pounds.

§ 25. **Sea Water.** Sea water contains 3 per cent. of its own weight of salts, principally common salt. Deposits from sea water occur upon evaporation, being precipitated at a temperature of about 230 degrees. A cubic foot of sea water weighs 64 pounds.

§ 26. **Expansion of Water.** "Increase in volume of a given weight of water, as heat is absorbed by it." For instance, between 39.1° and 212°, the volume increases from 1 to 1.04332.

§ 27. **Specific Heat of Water.** "The additional heat (more than the sensible) which is absorbed by water while being raised in temperature from 32° to 212° Fahr." For instance, the specific heat of one pound of water is $\frac{9}{10}$ of one unit greater at a temperature of 212° than at 32°, hence the heat required to raise one pound of water from 32° to 212° is

$$.9 + 212 - 32 = 180.9 \text{ units. } \S\ 18.$$

CHAPTER No. 3.

HEAT—ENERGY.

The study of the Compound Engine introduces and compels a brief study of some of the fundamental laws relating to Heat and Heat-Energy. The science of heat, when considered as a form of energy, can only be properly and fully treated under the recognized methods discussed in standard works on Thermodynamics; but such discussion involves the higher branches of mathematics and consequently would be beyond the purpose of this treatise; nevertheless our subject under discussion must include at least a brief reference to some of the simple features of the science, but only so far as are easily within the comprehension of the ordinary designer. This section therefore is intended to set forth in a simple and elementary manner (not exhaustive) and in plain language, some of the laws above referred to, so far as they relate to the transformation of heat energy into mechanical energy by means of the steam engine, and will be introduced under three subheads, viz.: 1st, "Thermal"; 2nd, "Dynamics"; 3rd, "Heat Energy Value."

THERMAL.

Definition of the word "Thermal": *"Relating to Heat."*

§ 28. **Heat.** The nature and substance of heat is unknown, but its presence and power can be unmistakably realized, and when Volume, Pressure and Temperature are known, the value of its energy can be accurately computed. *Source of Heat.* So far as the steam engine is concerned, heat has its source in the combustion of fuel, Coal, Coke, Wood and Oil; the effect of such combustion being the evaporation of water and the generation of steam under pressure. The quantity of heat obtained from a given quantity of fuel, depends upon two things: 1st, the quality of the fuel; 2nd, the perfectness of combustion.

§ **29. Quality of Fuel.** The quality of fuel is determined by the percentage of carbon it contains, as carbon is the principal element of combustion; the other elements of combustion are hydrogen and sulphur.

Bituminous coal of first-class quality such as that mined in Pittsburg and Virginia has an average composition of about

Carbon.	Hydrogen.	Sulphur.	Oxygen.	Nitrogen.	Ash.
80	5	1.25	8	1.25	4.5

Poor fuel is the most expensive in the end. That fuel having a high percentage of carbon and low percentage of ash is the most profitable commercially.

Standard Coal. The term "standard coal" is adopted by the Am. Soc. Mech. Engineers, and refers to a coal which imparts to steam 10,000 B. T. Us. for each pound of dry coal consumed. It is any coal having a calorific value of 12,500 B. T. Us. when used under a boiler which has an efficiency of 80 per cent. This quality agrees very closely to the average coals of the United States.

§ **30. Calorific Value of Fuel.** The term "Calorific Value" has reference to heat value as determined by the coal calorimeter, how determined is explained below. A definition of the term "Heat Value of Fuel" would be "The heat caused by the combustion of a given quantity of fuel and expressed in British Thermal Units.

1 pound of carbon contains about......14,500 B. T. U.
1 pound of hydrogen contains about....62,000 B. T. U.
1 pound good anthracite coal, about....13,500 B. T. U.
1 pound wood, about.................. 5,800 B. T. U.

Viz.: One pound wood will evaporate about 6 pounds water from and at 212°, or contains an equivalent to $6 \times 965.7 = 5{,}794$ B. T. U.

How Determined. Calculation for determining the

Calorific (Heat) value of coal, should be made on the basis of "Dry Coal." That is to say, the sample to be tested is powdered to a fineness that it will pass through a sieve of about 60 meshes to the inch, and then dried for one hour at 230° Fahr. The heat value is determined by a Coal Calorimeter. If the Mahler Bomb Calorimeter is used, which is a very reliable instrument, one grain of powdered dry coal is weighed into a platinum pan and placed in the steel bomb of the calorimeter, the bomb is then immersed in 2,000 grains of water. Oxygen is admitted into the bomb until the pressure reaches 367.5 lbs. absolute, and by means of an electric current the charge is ignited. The increase in temperature of the surrounding water determines the heat produced by the combustion.

§ 31. **Unit of Evaporation.** The recognized unit of evaporation is "one pound of water at 212 degrees Fahr. evaporated into dry steam at the same temperature, and is equivalent to 965.7 B. T. Us." Therefore, since one boiler horse-power is the evaporation of 34½ pounds of water per hour; then One Boiler Horse-Power expressed in B. T. Us. is equivalent to

$$965.7 \times 34.5 = 33{,}317 \text{ B. T. Us. per hour.}$$

The above was recommended by the A. S. M. Engrs. in the year 1899. The old Centennial equivalent was 33,305 B. T. Us.

§ 32. **Boiler Efficiency.** Definition, "*Heat absorbed per one pound of combustible, divided by calorific value of one pound of combustible.*" The heat absorbed being the evaporation of water from and at 212° per pound of combustible multiplied by 965.7. The calorific value is explained in § 30. This is used for determining the efficiency of the boiler proper.

§ 33. **Thermal Efficiency Ratio.** The thermal efficiency ratio is the proportion which the "heat equivalent of power developed" bears to the "total heat actually consumed." As it is preferable to figure the heat units consumed "per hour" rather than "per minute," and since the mechanical equivalent of heat is 778 (§ 39), then, the heat equivalent of power developed per horse-power per hour is

$$\frac{33{,}000 \times 60}{778} = 2{,}545 \text{ B. T. Us.}$$

The total heat actually consumed per hour is determined by actual test, and the thermal efficiency ratio is expressed thus

$$\frac{2{,}545}{\text{B. T. Us. per H. P. per hour}} = \text{Thermal Eff.}$$

§ 34. **Combustion.** To generate steam economically, the conditions necessary for the most perfect combustion of the fuel are of first importance. The principal element of combustion is carbon, § 29, and when exposed to the heat in a furnace it becomes volatilized and passes off. If oxygen is present the carbon and oxygen become chemically united causing combustion. The first requirement, therefore, for producing perfect combustion of fuel is a proper supply of air.

§ 35. **Air Supply.** For producing perfect combustion, provision must be made for furnishing the necessary supply of air, neither too much nor too little, and always warm air, if possible. The next condition is that the air should be mixed thoroughly; and further, that the combustible gases shall maintain a high temperature. One pound of bituminous coal needs about 140 cubic feet of air. If the supply is too great, loss results from gases passing off unused to the chimney. If, on the other hand, it is not sufficient, a greater loss is the result, due to incomplete combustion, hence the importance of a proper and uniform supply.

One pound of Carbon with 11.6 pounds of air is claimed by experts to be the exact quantity for perfect combustion, and where obtained will develop 14,500 British Thermal Units, and produce a temperature of 4939 degrees Fahr., the product being carbonic acid gas, but an insufficient quantity, say one-half, would develop only 4500 B. T. U., and the product would be carbonic oxide. Generally speaking, not more than about 8½ pounds of air per pound of coal can be caused to unite, and this amount would develop 10,625 B. T. U., a loss of about 27 per cent. of perfect combustion, although the same weight of coal is consumed.

In general practice, loss through incomplete combustion, radiation and absorptions, amounts to about 33 per cent. That is to say, coal with evaporative power of 15 pounds of water will, under good conditions, actually evaporate about

10 pounds of water per 1 pound of coal.

With a view of producing perfect combustion by the bringing together of the above two elements, viz., Carbon and Air, much skill and experience is needed in the design and construction of the steam boiler which can most effectively accomplish this end. The water-tube boiler is the only type in which all these essential features are embodied, and the same may be found in "The Sterling" as described on the last page of this book, in which type of boiler the conditions for thorough combustion are well provided for in the large combustion space over the furnace arch.

DYNAMICS.

§ 36. **Dynamics of the Heat Engine.** The word dynamics is defined as "The science of moving power, or matter in motion, or of the motion of bodies that mutually act upon one another." Therefore, so far as it relates to the heat engine, we combine the two words, "Thermal," relating to heat, and "Dynamics," relating to moving powers, and we get the word "Thermodynamics," the subject of which treats of the action of heat-energy upon the piston and other moving parts of an engine.

§ 37. **First Law of Thermodynamics.** This law is defined thus: "Heat and mechanical energy are mutually convertible." That is to say, whatever work is performed by the agency of heat, an amount of heat disappears which is equivalent to the work done, on the basis of 778 foot pounds being equivalent to one B. T. U. This 1st law sets no limit, and if considered alone and unqualified might lead some into the mistaken belief that it is possible to convert the whole of the heat taken in by the engine, into mechanical energy. This, however, is an impossibility in a real engine, as part of the heat chargeable to the engine, is absorbed by other bodies of lower temperatures with which the heat of the steam comes in contact, viz., the walls of the cylinder, piston and surrounding air. The heat thus diffused, though not destroyed, could not be accounted for by this first law, hence the need of a knowledge of the second law. § 38. But, if it were possible for an engine to fill the requirements of the first law, then such an engine would be known as a *Reversible Engine;* that is to say, an engine that can run either in the usual way transforming heat into work, or the reverse making the same cycle in the opposite direction transforming work into heat. Such an engine would be in accordance with the principle stated by Carnot, thus "all engines working between the same source of heat and the same refrigerator, give the same efficiency."

The Reversible Cycle is the cycle of a reversible engine, and a cycle is defined as "A series of successive states of the volume and pressure of a working gas, which may be represented by a continuous line returning into itself. Or the boundary line of a definite area enclosed, representing either the external work done during a series of transformations by the enclosed gas, or, in a reverse order, representing the work done upon the gas while passing through these transformations."

§ 38. **Second Law of Thermodynamics.** Definition: *"It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature."* The first law, § 37, sets no limit to the convertibility of heat into work, and requires the full heat value of the whole cycle of operation; but the second law does set a limit, and makes it impossible to convert into work, more than a fraction of the heat supplied. All steam engines are similar in the respect that they receive heat from some external source, transform part of it into work, lose part of it by radiation and leaks, and discharge the remainder into the atmosphere or condenser, wherefore the *Efficiency Ratio* of an engine is the ratio of the heat converted into work, to the total heat supplied, and is expressed thus

$$\frac{\text{Heat taken in by the engine}}{\text{Heat converted into work}} = \frac{}{\text{Fraction}}\text{Ratio}$$

And when figured on the basis of heat units, according to the law of efficiency of thermodynamic engines, and from absolute zero, would be expressed

$$\frac{T_1 - T^2}{T_1} = \text{Percentage of Efficiency}$$

in which $T1$ = temperature of heat received,

and $T2$ = temperature of heat rejected.

For instance, in a perfect engine, that is to say one without loss by radiation, leakage, etc., having an initial pressure of 185 pounds gauge, 200 pounds abso., and back pressure in low pressure cylinder of 3.5 abso., we have

200 pounds abso. = 381 Degrees Fahr.

3.5 " " = 149 " "

Absolute zero = —461.2 " "

The following equation gives theoretical Efficiency

$$\frac{(381 + 461.2) - (149 + 461.2)}{381 + 461.2} = 28 \text{ per cent. nearly.}$$

In the actual engine the above high efficiency of course never can be realized, on account of leakage, radiation, etc., but the method of figuring theoretical efficiency is very important as a means of comparison.

The pumping engine at Chestnut Hill, Boston, affords a good illustration, in which the above steam pressures are true, and by actual test shows an efficiency of 21.63 per cent. as against 28 per cent. for perfect.

§ 39. **Mechanical Equivalent.**

Def.—"The mechanical force necessary for raising the temperature of one pound of water one degree Fahr."

Dr. Joule, in the year 1847, by means of a paddle-wheel working in a vessel filled with liquid, demonstrated that to raise one pound of water at or near freezing, through 1° Fahr., required 772 units of work; therefore, 772 is known as "Joule's equivalent." More recent experiments seem to show that 778 at a temperature of 59° Fahr. is more nearly correct, and is adopted.

HEAT ENERGY VALUE.

§ 40. **Energy of Steam.** All energy in steam, so far as the reciprocating engine is concerned, is due exclusively to the heat it contains, it being the sole cause of all work performed. The water in the steam exerts no force except by the inertia of its mass, which is not herein considered, and apart from which inertia the water simply acts as a convenient medium by which the heat may be collected from the fuel and conveyed to the cylinder of the engine, where its energy, during expansion, is spent upon the moving piston; the heat being considered as the only effective agent. Owing to the Law of Conservation of Energy, no form of energy can be produced except at the expense of some other form, neither can it be destroyed. It can be transformed, or transferred from one body to another, but the sum total must ever remain the same. It can neither be created nor annihilated, but it may be disposed of. In the steam engine it is disposed of in three different ways: 1st, by radiation and conduction, or contact with other cooler bodies; second, by transformation into mechanical energy, useful work; third, by exhaust, either to the atmosphere or the condenser.

§ 41. **Water Accounted for.** After all work is done, the free heat expended, and the latent heat absorbed by the injection water, then were it not for leakage (by use of the surface condenser) precisely the same quantity of water could be collected, as was fed to the boiler before the work began. Showing, as above stated, that all energy imparted to the piston is due to the heat of the steam and not the water.

§ 42. **Dynamical Energy of Steam.** Definition, *"Energy imparted by steam to matter."* The energy conveyed by one pound of water when converted into steam at atmospheric pressure, if measured in foot-pounds of work would be as follows:

One pound of water at 212° occupies .016 cubic ft.; when converted into steam at atmos. pressure, viz., 14.7 pounds per sq. in., it occupies a space of 26.33 c. ft. Since 1 c. ft. of steam at atmos. pressure equals $144 \times 14.7 = 2{,}116.8$ foot-pounds, therefore $2{,}116.8 \times 26.33 = 55{,}735$ foot-pounds. The equivalent of above foot-pounds if expressed in heat units would be, since 1 heat-unit is equivalent to 778 foot-pounds, then $55{,}735 \div 778 = 71.6$ units.

§ 43. **Equivalent British Thermal Units per One Horse Power.** Since 778 foot-pounds of energy is found to be the dynamical equivalent of one B. T. U., then 42.416 B. T. Units represent the thermal equivalent of one horse-power. Thus:

$$\frac{33{,}000}{778} = 42.416 \text{ B. T. Us.}$$

or, if the time unit is one hour, then the B. T. Us. per horse-power per hour would be

$$\frac{33{,}000 \times 60}{778} = 2{,}545 \text{ B. T. Us.}$$

§ 44. **Maximum Theoretical Duty of Steam.** The maximum theoretical duty of the heat contained in one pound of steam, is 778 foot-pounds for every heat unit contained therein. For instance, table No. 4 shows that the total heat in steam at 14.7 pounds abso. is 1,146.6 units, therefore the maximum duty would be

$$778 \times 1{,}146.6 = 892{,}055 \text{ foot-pounds.}$$

or for steam at 200 pounds abso. the total heat is 1198.3 units, and maximum duty

$$778 \times 1198.3 = 932{,}277 \text{ foot-pounds.}$$

CAUTION.—Care should be taken not to confuse maximum duty of "Steam" with maximum duty of "Engine." Study carefully § 44 in connection with § 38, in which it is shown that all that is possible within the range of the engine cycle, even with a perfect actual engine, is about 28 per cent. of maximum theoretical efficiency.

CHAPTER No. 4.

ECONOMY OF USING STEAM EXPANSIVELY, AND THE LAW OF EXPANSION AS APPLIED TO STEAM.

§ 45. **History.** In the earlier history of the steam engine, steam was used without any regard whatever to the economy due to expansion. Such was the Newcomb single-acting beam engine, in which the upper end of the cylinder was open to the atmosphere, and the lower end closed. Steam was admitted under the piston, and allowed to follow it with open valve to the end of the stroke, and upon the inlet valve being closed and the exhaust opened, the piston was caused to descend, partly by its own weight and partly by vacuum caused by injecting a jet of cold water directly into the cylinder. In the year 1782 James Watt demonstrated the fact of economy due to expansion, and made application for letters patent, in which he said:

"My improvement in steam engines consists in admitting steam into the cylinder or steam vessels of the engine only during some part or portion of the descent or ascent of the piston of said cylinder, and using the elastic forces, wherewith the said steam expands itself in proceeding to occupy larger spaces as the acting powers on the piston through the other parts or portions of the length of the stroke of said piston."

That is to say, the inlet valve was to remain wide open, and the steam maintain a uniform pressure up to some definite point reached by the piston, when the valve was to close and remain so for the rest of the stroke, and the enclosed steam permitted to expand as strictly in keeping with the law of expanding gas (§49) as the influence of other conditions would allow. This was Watt's ideal, but owing to lack of efficient means for automatically controlling the cut-off for varying loads, it was not realized until the invention of Geo. H. Corliss (§47).

§ 46. **Fly Ball Governor.** Variation of load demands a variation of effective pressure. James Watt first invented automatic means of controlling the pressure of the initial steam, and this led to his invention of the fly ball governor (regulator) which he attached directly to the throttle in the steam pipe. The objection to this device was made on the claim of excessive loss due to wire-drawing, and expansion of steam between the throttle and the slide valve which, together with the slow cut-off of the same, left considerable room for improvement.

§ 47. **Corliss Cut off.** In the year 1849 Geo. H. Corliss introduced the most valuable improvement of that century, and which completely revolutionized steam engine design in which first-class economy is sought. The object of his invention was, 1st, to maintain the full boiler pressure to any point of the stroke without wire-drawing, and then to cut it off sharp and clean, leaving the steam thus enclosed within the cylinder, to expand independent of that without. 2nd, to vary the point of cut-off, thus admitting a larger or smaller volume of steam as the immediate load required, but at full boiler pressure up to the point of cut-off. For the purpose of accomplishing his object, he employed rotary or rocking valves in preference to the old slide valve, and used two for steam inlet, and two for exhaust. He connected the two steam valves, one for each end of the cylinder, with a releasing mechanism capable of suddenly releasing and cutting off the steam supply at any point required. To this releasing gear was connected the fly ball regulator *direct,* which brought the point of cutting off completely under the control of the regulator, and varied the same upon the slightest change of speed, without the loss due to wire-drawing.

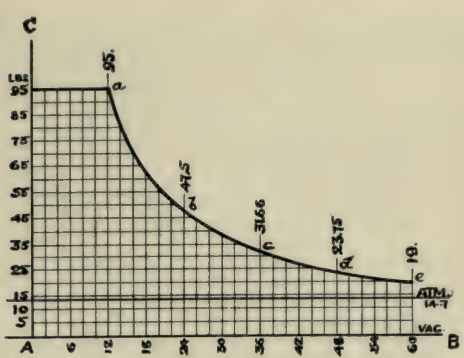
The expansion curve actually produced in steam cylinders fitted with the Corliss valve gear approaches very closely to the Hyperbolic curve, and other things being equally favorable, at least twenty-five per cent. of the fuel can be saved as compared with the slide valve arrangement. Thus the name of Geo. H. Corliss became famous, and the Corliss valve gear almost universally adopted for land service.

§ 48. **Heat Energy Saved by Expansion.** If a given volume of steam of uniform pressure be admitted during, say $\frac{1}{5}$ of the stroke, then suddenly cut off and allowed to exhaust at that point, the mechanical energy thrown away and wasted in the escaping steam, would be equal to what might otherwise have been the total force exerted by the same while expanding into five times its volume, by virtue of the piston being acted upon while moving through the last $\frac{4}{5}$ of its stroke; the mean effective force of which would be, in a non-condensing engine, exhausting against a back pressure of one pound above the atmosphere, equal to about 29 per cent of the effective force in the first $\frac{1}{5}$ of the stroke. In other words, as compared to a non-expansive engine, by the expense of adding to the cylinder four times its length a saving of 29 per cent in power can be gained without any further expenditure of steam. This conclusion may be arrived at by the following example and reference to Diagram Fig, 2.

Example.—Assume steam press. at 80 pounds gauge or 95 pounds abso., and exhausting against a back pressure of 1 pound above atmosphere or 16 pounds abso.

The mean eff. press. during the first $\frac{1}{5}$ of the stroke would equal 95 pounds abso., minus 16 pounds back press. equals 79 pounds.

Then the mean eff. press. during the last $\frac{4}{5}$ of the stroke can be determined by measuring the pressures shown on the 25 vertical ordinates covered by the curve



STROKE IN INCHES.

Fig. 2.

proper, the sum of which is 975; and dividing by the number of ordinates 25 we have $975 \div 25 = 39$ as a mean press., then substracting the back press. of 16 pounds abso., we have $39 - 16 = 23$ pounds M. E. P.

The comparison can now be made between the M. E. Press. of the last $\frac{4}{5}$ of the stroke, viz., 23, with that of the first $\frac{1}{5}$, viz., $95 - 16 = 79$ M. E. P., and found to be about 29 per cent. additional power.

A still greater saving will be found in the case of a Condensing Engine working expansively in comparison with the non-condensing non-expansion engine referred to, shown by the following example, and reference again to Diagram Fig. 2.

Example.—Assume steam press. at 95 pounds abso., as before, but exhausting into a condenser against a back press. of 7 pounds abso., here we have

Mean Press.		Back Press.		Mean Eff. Press.
39	—	7	=	32

If compared with a non-expansive, non-condensing engine, in which the mean effective press. was shown to be 79 pounds, will show a saving of 40 $\frac{5}{10}$ per cent.

Purely from a theoretical point of view, the above reasoning might be continued indefinitely, by raising the steam press. and cutting off earlier in proportion, and with apparent increase in economy, but owing to the variation of temperatures within the cylinder, and the consequent high percentage of cylinder condensation, together with certain mechanical objections, it is generally considered commercially unprofitable to expand more than seven times in one cylinder, or to expand the steam down to less than about 8 or 9 lbs. abso. pressure as a terminal in a single cylinder.

§ **49. Boyle's Law.** The law of expansion of gas had been discovered by Robert Boyle more than a century before Watt's experiments, but Watt was the first to demonstrate the theory so far as it could be applied to steam engines. The law was known as the "First Law of Expanding Gases," and is stated thus:

"The pressure of a portion of gas at a constant temperature varies inversely as the space it occupies."

Expanding steam, however, while performing actual work upon a moving piston within an enclosed cylinder, does not act in strict acordance with this law, for the reason that it is influenced by the changing temperature, consequently interfering with one of the fundamental properties on which the law was based, as well as being influenced by condensation, evaporation, clearance volume and compression, all of which are experienced in actual practice. Nevertheless, the law may be used, and such provisions and allowances made as will approximately compensate for the effects of these foreign influences.

One of the best provisions made to this end is in the use of the Steam Jacket, §89, but even independent of this improvement the theoretical diagram formed after the law and aimed at by every designer, if compared to the actual curve found on the indicator card, will approximate very closely indeed, by simply neglecting, in our calculations, the work done by the steam occupying the clearance space, the heat of which will help to compensate for the loss. § 61 and § 87.

§ **50. Pressure Volume Constant.** On the basis of the first law of expanding gas, §49, we have a pressure volume constant, usually referred to as the P. V. CONSTANT, that is to say the pressure multiplied by the volume equals a constant, which, for instance, if the volume is 12 and the pressure 95 we have

Press.		Vol.		Constant
95	$\times$	12	$=$	1140

If then the volume increases by expansion, the pressure decreases. Assuming then that the constant is 1140, as above, and the pressure has fallen to 47.5, then the volume is:

$$\frac{1140}{47.5}=24$$

§ **51. Pressure at any Point of Stroke.** Referring again to Fig. 2, §48, and on the basis of the P. V. Constant explained § 50, the pressure at any point of the stroke may be found thus: Multiply the initial press., in this case 95 lbs. abso., by distance in inches from start to point of cut-off, and divide by distance from start to point where pressure is required. If, for instance, cut-off is 12″ from start, and point of required press. is 24″ then we have

$$\frac{95 \times 12}{24}=47.5 \text{ abso.}$$

If required at 36″ from start, same cut-off,

$$\frac{95 \times 12}{36}= 31.66 \text{ abso.}$$

If at 60″, which is end of stroke, same cut-off, viz. 1-5,

$95 \div 5 = 19$, terminal press.

The above is true if without back pressure, which is an impossible condition in a steam engine; the effective pressure therefore, is the above, minus the back pressure.

§ 52. **Graphic P. V. Constant.** In Fig. 3 let A C represent the pressure line, and A B the volume, then by constructing a series of triangles A a a^1a^2, A b b^1b^2, etc., each one of exactly the same area, then the length of pressure side multiplied by volume side is constant for all.

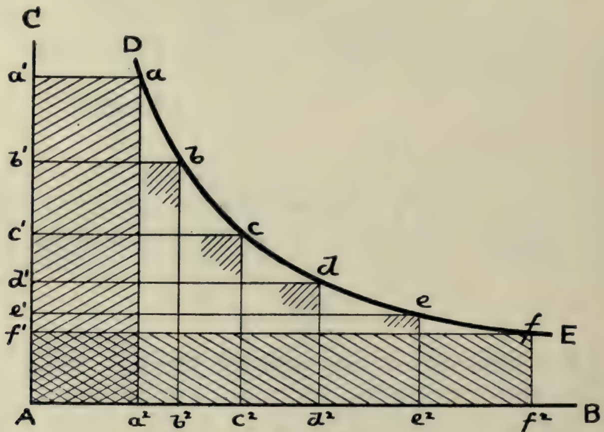


Fig. 3.

§ 53. **Expansion Line.** Referring again to Fig. 3 the curved line D E passing through points a b c d e f, is a rectangular hyperbola, and represents, theoretically, the expansion line of a gas expanding according to Boyle's law, § 49. This curve is a close approximation to the actual expansion line of indicator diagrams taken from first class Corliss engines, having steam jacketed cylinders, and superheated steam.

§ 54. **Mean Effective Pressure.** Referring again to Fig. 2, § 48. If a number of points in a curve, such as a b c d e, be measured on their vertical ordinates from the base line A B, and the sum of their readings in pounds be divided by the number of ordinates, the quotient will be the average pressure during the period of expansion. Or if ordinates are set up covering the whole diagram, including the admission period also, and divided by the number of ordinates as before, the quotient will be the average forward pressure during the whole stroke, then by deducting the mean back pressure the answer will be the mean effective pressure, or the M. E. P.

The diagram shows spacing of 2 inches, but the more ordinates there are the better. The above operation is comparatively slow, and while it is of value in training the beginner, the engineer prefers to use a measuring instrument called the planimeter, which facilitates very much, especially when the diagram is irregular. But if the expansion curve is a rectangular hyperbola, and the back pressure line uniform, then the M. E. P. can be more quickly found by the use of hyperbolic logarithms, as explained in §55.

§ **55. Use of Hyperbolic Logarithms.** To the beginner this name "Hyperbolic Log." is sometimes an unfortunate one, as it may be suggestive that the subject is beyond his depth. But with a little patience and by carefully following step by step through the following examples, the process will soon become clear and easy. Hyperbolic logarithms are common logarithms multiplied by 2.3, and by their use, the average pressure exerted upon a moving piston during the admission and expansion of a given volume of steam at a known initial pressure, can easily be figured by ordinary rules of arithmetic and the tables Nos. 6 and 7, as per following explanation:

Knowing the required number of expansions, say, for instance, 6, look down the column of numbers, and opposite will be found the Hyp. Log. 1.7918; to this number add 1, representing the initial value, making it

$$1.7918 + 1 = 2.7918.$$

Multiply this 2.7918 by the steam pressure in pounds abso., thus for instance, 80 gauge + 15 = 95 abso.

$$2.7918 \times 95 = 265.22,$$

then divide this by the number of expansions, which in this case is 6, and we have

$$265.22 \div 6 = 44.2 \text{ pounds press. abso.,}$$

from which deduct the back pressure of, say, 1 pound above atmosphere or 16 pounds abso., thus

$$44.2 - 16 = 28.2 \text{ M. E. Press.}$$

As an equation it would be

$$\frac{95 \times 2.7918}{6} - 16 = 28.2 \text{ M. E. P.}$$

Try four examples for finding the mean eff. pressures, cutting off at four different points, using the same initial pressure of 90 lbs. abso., and exhausting against the same back pressure of one pound above atmosphere, viz., 16 lbs. abso.

Use the following rule in each case:

Rule.—"Initial press. abso. multiplied by Hyp. Log. of number of expansions + 1, divided by number of expansions, and from quotient subtract back pressure."

First example, cutting off at ⅕ stroke, Fig. 4.

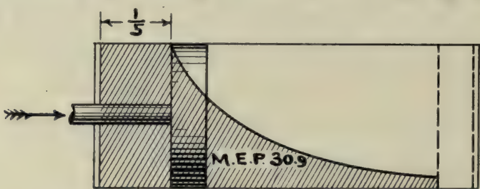


Fig. 4.

Hyp. log. of 5 = 1.6094. Table No. 7.

Hyp. log. + 1 = 2.6094.

$$\frac{90 \times 2.6094}{5} - 16 = 30.9 \text{ M. E. P.}$$

Second, cutting off at ⅓ stroke, Fig. 5.

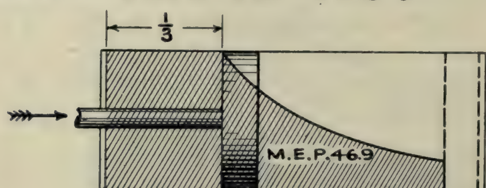


Fig. 5.

$$\text{Hyp. log. of } 3 = 1.0986.$$
$$\text{Hyp. log.} + 1 = 2.0986.$$
$$\frac{90 \times 2.0986}{3} - 16 = 46.9 \text{ M. E. P.}$$

Third, cutting off at ½ stroke, Fig. 6.

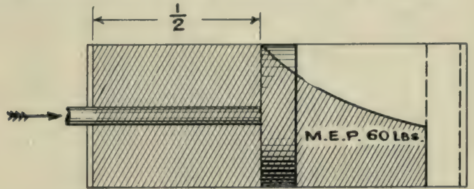


Fig. 6.

$$\text{Hyp. log. of } 2 = .6931.$$
$$\text{Hyp. log.} + 1 = 1.6931.$$
$$\frac{90 \times 1.6931}{2} - 16 = 60 \text{ M. E. P.}$$

Fourth, cutting off at ¾ stroke, Fig. 7.

NOTE! As in the last three examples, so in this, find the number of expansions by dividing the denominator of the fraction by the numerator, for instance, at ½ stroke, $2 \div 1 = 2$ expansions.

For ¾ stroke we have $4 \div 3 = 1.333$ expansions.

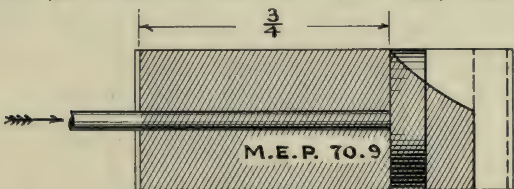


Fig. 7.

$$\text{Hyp. log. of } 1.333 = .2852. \quad \text{Table No. 6.}$$
$$\text{Hyp. log.} + 1 = 1.2852.$$
$$\frac{90 \times 1.2852}{1.333} - 16 = 70.9 \text{ M. E. P.}$$

The same reasoning holds good for any other point of cut-off, and any other back pressure, either single or compound.

TABLE No. 6.

SHORT TABLE.

HYPERBOLIC LOGARITHMS OF NUMBERS.

NO.	HYP. LOG.	NO.	HYP. LOG.	NO.	HYP. LOG.
1.	.0000	1.35	.3001	1.7	.5306
1.05	.0488	1.4	.3365	1.75	.5596
1.1	.0953	1.45	.3716	1.8	.5878
1.15	.1398	1 5	.4055	1.85	.6152
1.2	.1823	1.55	.4383	1.9	.6419
1.25	.2231	1.6	.47	1.95	.6678
1.3	,2624	1.65	.5008	2.	.6931
1.33	.2852	1.66	.5068		

Continued in Table 7.

§ **56. Isothermal Line.** The isothermal line is an expansion line of constant temperatures. That is to say, from the time of cut-off, the volume of steam enclosed in a cylinder, expands to the end of the stroke, and ordinarily the terminal temperature of the steam is considerably lower than was the initial temperature; but if it were possible by steam jacket or any other means, to supply sufficient additional heat to the steam as would maintain the same temperature throughout the whole expansion period, then the expansion curve of the indicator card would be an isothermal line; hence the definition: *"A curved line representing the varying pressures of an expanding gas, having a constant temperature, but partially under the influence of moisture."*

This condition of course is not possible in practice, as in such a case the heat energy in the steam at the end of the stroke would be the same as at the beginning, and the work done on the piston would be due to the additional heat supplied.

§ **57. Adiabatic Line.** The adiabatic line is the expansion curve (not of constant temperature as the isothermal, but the opposite) of a volume of steam enclosed in a cylinder, and which during expansion is

TABLE No. 7.

HYPERBOLIC LOGARITHMS OF NUMBERS.

NO.	HYP. LOG.	NO.	HYP. LOG.	NO.	HYP. LOG.	NO.	HYP. LOG.
2.	.6931	4.	1.3863	6.	1.7918	8.	2.0794
2.05	.7178	4.05	1.3987	6.05	1.8001	8.05	2.0857
2.1	.7419	4.1	1.4110	6.1	1.8083	8.1	2.0919
2.15	.7655	4.15	1.4231	6.15	1 8165	8.15	2.0980
2.2	.7885	4.2	1.4351	6.2	1.8245	8.2	2.1041
2.25	.8109	4.25	1.4469	6.25	1.8326	8.25	2.1102
2.3	.8329	4.3	1.4586	6.3	1.8405	8.3	2.1163
2.35	.8544	4.35	1.4702	6.35	1.8485	8 35	2.1223
2.4	.8755	4.4	1.4816	6.4	1 8563	8.4	2.1282
2.45	.8961	4.45	1.4929	6.45	1.8641	8.45	2.1342
2.5	.9163	4.5	1.5041	6.5	1.8718	8.5	2.1401
2.55	.9361	4.55	1.5151	6.55	1.8795	8.55	2.1459
2.6	.9555	4.6	1.5261	6.6	1.8871	8.6	2.1518
2.65	.9746	4.65	1.5369	6.65	1.8946	8.65	2.1576
2.7	.9933	4.7	1.5476	6.7	1.9021	8.7	2.1633
2.75	1.0116	4.75	1.5581	6.75	1.9095	8.75	2.1692
2.8	1.0296	4.8	1.5686	6.8	1.9169	8.8	2.1748
2.85	1.0473	4.85	1.5790	6.85	1.9242	8.85	2.1804
2.9	1.0647	4.9	1.5892	6.9	1.9315	8.9	2.1861
2.95	1.0818	4.95	1.5994	6.95	1.9387	8.95	2.1917
3.	1.0986	5.	1.6094	7.	1.9459	9.	2.1972
3.05	1.1151	5.05	1.6194	7.05	1.9530	9.05	2.2028
3.1	1.1314	5.1	1.6292	7.1	1.9601	9.1	2.2083
3.15	1.1474	5.15	1.6390	7.15	1.9671	9.15	2.2138
3.2	1.1632	5.2	1.6487	7.2	1.9741	9.2	2.2192
3.25	1.1787	5.25	1.6582	7.25	1.9810	9.25	2.2246
3.3	1.1939	5.3	1.6677	7.3	1.9879	9.3	2.2300
3.35	1.2090	5.35	1.6771	7.35	1.9947	9.35	2.2354
3.4	1.2238	5.4	1.6834	7.4	2.0015	9.4	2.2407
3.45	1.2384	5.45	1.6956	7.45	2.0082	9.45	2.2460
3.5	1.2528	5.5	1.7047	7 5	2.0149	9.5	2.2513
3.55	1.2669	5.55	1.7138	7.55	2.0215	9.55	2.2565
3.6	1.2809	5.6	1.7228	7.6	2.0281	9.6	2.2618
3.65	1.2947	5.65	1.7317	7.65	2.0347	9.65	2.2670
3.7	1.3083	5.7	1.7405	7.7	2.0412	9.7	2.2721
3.75	1.3218	5.75	1.7492	7.75	2.0477	9.75	2.2773
3.8	1.3350	5.8	1.7579	7.8	2.0541	9.8	2.2824
3.85	1.3481	5 85	1.7664	7.85	2.0605	9.85	2.2875
3.9	1.3610	5.9	1.7750	7.9	2.0669	9.9	2.2925
3.95	1.3737	5.95	1.7834	7.95	2.0732	9.95	2.2976

For Hyp. Log. of numbers less than 2, see Table No. 6.

doing external work by overcoming the resistance of the piston, and during which time no heat is allowed to pass neither in nor out through the walls of the cylinder; thus assuming that the cylinder, piston, etc., were made of such non-conducting material as would neither absorb nor radiate any heat whatever; then the expansion curve of the indicator card would be an adiabatic line, hence the definition: *"A curved line representing the expansion of a gas, confined in a cylinder which possesses the imaginary property of not suffering any heat to pass through its walls, while the gas is doing external work."*

Professor Thurston wisely said, "Maximum expansion, as nearly adiabatic as practicable, is the secret of maximum efficiency."

§ 58. **Comparison of Isothermal and Adiabatic Curves.** Fig. 8 shows a comparison between the isothermal and adiabatic curves. Line A B is the line of perfect vacuum. A C represents the line of pressures. Both curves commencing at D, supposedly the point of cut-off in the cylinder. Then D F represents the isothermal, and D E the adiabatic. G H being the initial pressure line.

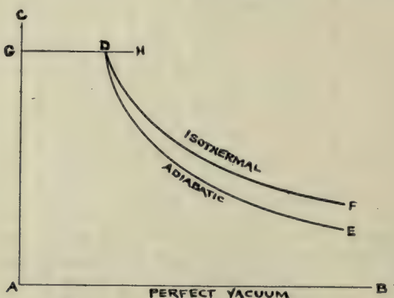


Fig. 8.

Observe that the isothermal is higher than the adiabatic, representing higher pressures. This is due to the additional heat supplied to the steam during expansion, and is always true relatively, when expanding; but when gas is being compressed, as in the case of cushioning, or in an air compresser, then the adiabatic curve is the higher, as shown in Fig. 9, both compression curves commencing at M, and the space between them varying as the conditions influencing their formation changes.

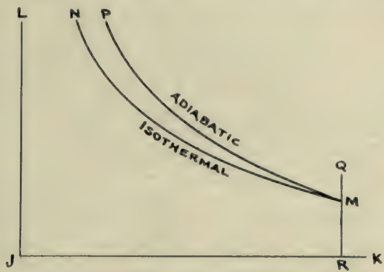


Fig 9.

CHAPTER No. 5.

THEORETICAL DIAGRAMS.

The theoretical diagram is an ideal indicator card, representing the work performed in a cylinder under ideal conditions. The combined diagram is a combination of two or more theoretical diagrams representing the work done in a compound engine, but as though it was all done in one cylinder. It is also a graphic means of predetermining such ratios of cylinder areas and positions of cut-off in each cylinder as will ensure a continuous hyperbolic expansion curve, commencing at the point of cut-off in the high pressure cylinder, and continuing down to the terminal point in the low.

These predetermined diagrams are of course theoretical, as above stated, but in this treatise in which is assumed first-class conditions, such as steam jackets, superheated steam, and reheaters in the receivers, they represent very nearly what can be reasonably expected in actual practice.

§ 59. **Indicator Diagram.** The diagram, Fig. 10, represents a theoretical indicator card, and the several points and lines are named as follows:

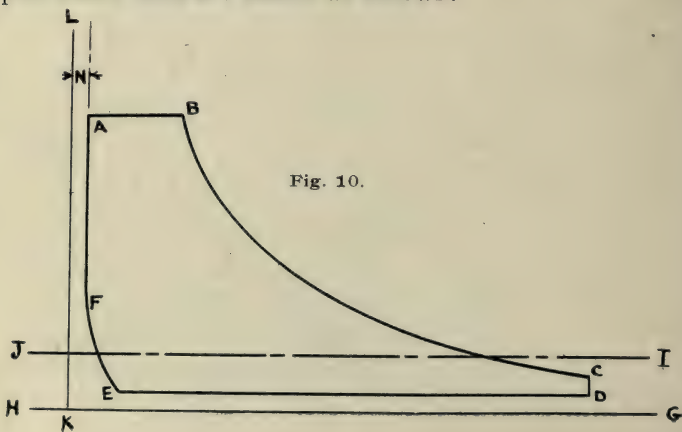


Fig. 10.

A B—Admission period.
B—Point of cut-off.
B C—Expansion line (Rectangular hyperbola).
C—Point of release (Exhaust opens).
C D—Terminal drop.
D E—Back pressure line.
E F—Compression of exhaust.
F—Opening of steam valve.
F A—Steam line.

The above is the diagram proper, the other lines are:
H G—Perfect vacuum.
K L—Clearance line.
N—Clearance volume.
J I—Atmospheric line.

The initial pressure abso. is the height of A B above perfect vacuum line, as measured by a scale corresponding to the indicator spring. For instance, if taken with a 60 spring, one inch in height is 60 pounds abso.

§ 60. **Hyperbolic Curve.** The hyperbolic curve or rectangular hyperbola is regarded in this treatise as the Standard Expansion Line. Referring to Fig. 3, § 52, curve D E is a rectangular hyperbola, and is constructed by means of rectangles as explained. There are also

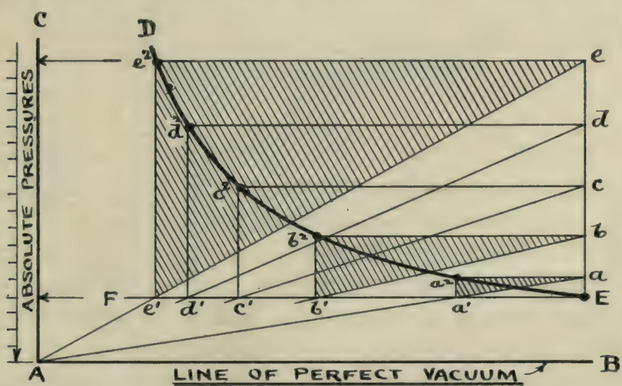


Fig. 11.

two other methods of construction, the first from the terminal point, and the second from the point of cut-off.

First Method, Fig. 11. Let A B and A C represent the vacuum and pressure lines. As the curve is to be constructed from the terminal point E, draw vertical line E e, and horizontal E F. In line E e, prick off several points, any number, such as a b c d e, and to each draw as many lines from point A, cutting F E at a^1 b^1 c^1 d^1 e^1, from which draw horizontals and verticals, meeting in points a^2 b^2 c^2 d^2 e^2 the curve D E passing through these points is a rectangular hyperbola.

Second Method, Fig. 12. From point D, which represents the position of cut-off, draw vertical line D F

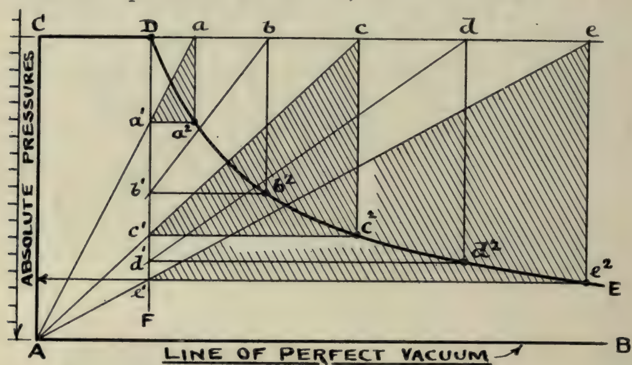


Fig. 12.

and produce C D to e. Then prick off several points a b c d e and complete the diagram as before, finding curve D E which is also a rectangular hyperbola.

§ **61. Clearance Neglected.** Fig. 13 represents the work done in the high and low press. cylinders of a compound engine, cutting off at $\frac{1}{4.8}$ and $\frac{1}{4.4}$ respectively, showing for comparison, first the expansion curves, dotted lines A E and B F, which includes clearance, and second, full line curves A C, B D without clearance.

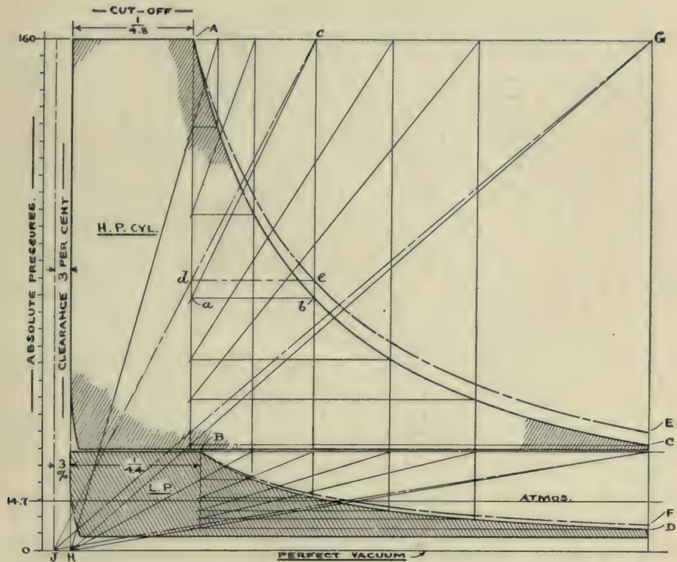


Fig. 13.

When the effect of clearance is to be included, point J is established at such a distance from H as will represent relatively the percentage of the clearance volume to that of the piston displacement, and from which the radial construction lines are drawn. When the effect of clearance is omitted, the radial construction lines are drawn from point H. The curves are plotted by means of a number of triangles such as c a b and c d e in same manner as described in Fig. 12, § 60.

The technically correct method is to allow for clearance; but in practice this higher curve is never fully realized, for the reason that every pound of water drained from a cylinder represents a certain volume of steam which has given up its heat, consequently the total volume has been reduced accordingly and that in proportion to the amount of condensation. The extent of this loss cannot be predetermined, but assuming that the volume of the same is exactly equal to the clearance volume, then the pressure curve would be the same as though there was no clearance and no condensation. Therefore, so far as the expansion curves are concerned it is quite safe to neglect the clearance, and by doing so it simplifies the reasoning, and also enables us to figure the mean effective pressures by hyperbolic logarithms with less complication. But in determining the correct point of cut-off in the low pressure cylinder, the clearance should be included in the layout. It therefore follows that the diagram which most nearly represents the actual work done, should include both features, and for this reason diagram Fig. 20, § 68, is submitted. But to lead the student up step by step, he is asked to study the next six diagrams first, to prepare him to better understand what follows.

COMBINED DIAGRAMS.

§ **62. Method of Combining Indicator Cards.** The method of combining actual indicator cards, is to reproduce them by bringing both to the same pressure scale. That is to say, the low pressure card may have been taken with a 12 or 20 spring, and the high pressure with a 60 or 80 spring. By reproducing them they will have the appearance of having been taken with the same spring. The length of the card to remain the same. Next, arrange them with the high pressure card above the low, and at such heights above the perfect vacuum line as that their respective pressures shall agree with the same pressure scale. Then contract the high presure diagram horizontally at various levels, and in proportion to the ratio of cylinder areas, for instance, as line A B, Fig. 15, § 64. The foregoing refers particularly to actual cards taken from actual engines, but the layout of a theoretical combined diagram requires a little more preliminary work, as the cards must be first constructed for each of the cylinders, representing the work to be done in each. The Initial pressure, the Terminal pressure in the low pressure cylinder, and the Ratio of Areas, must be first decided upon, and the lines laid down accordingly. See § 63.

§ 63. **Ratio of Cyl. Areas, Relative to Initial Pressure.** In constructing a combined diagram, the first thing required is the initial steam pressure, and second the ratio of cyl. areas in keeping with the first. The diagram, Fig. 14, illustrates the method, showing for comparison three different initial pressures, viz., 100, 150 and 200 lbs. abso. respectively, with one common terminal pressure of 6½ lbs. and assuming no clearance.

CONSTRUCTION. Draw a rectangle, say 10 inches long, representing length of diagram, their heights scaling to the initial pressure proposed. Next draw

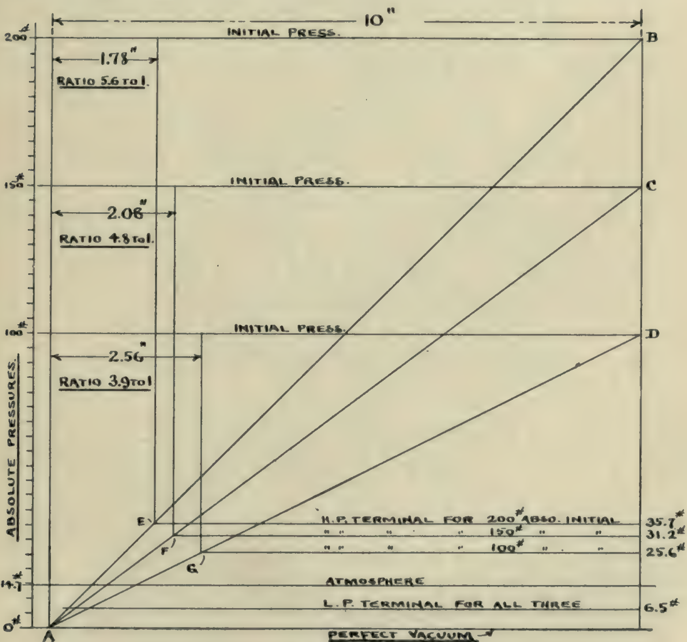


Fig. 14.

diagonals A B, A C, A D. Now it is necessary that the high pressure terminal point for each of the three initial pressures shall be in its corresponding diagonal line, for instance, the 200 lbs. in E, 150 lbs. in F, and 100 lbs. in G. To locate these points figure first for the ratio of areas, and from them the high pressure terminals, the two must agree.

Rule.—"Square root of initial press., divided by L. P. terminal," thus

$$\sqrt{\frac{200}{6.5}} = 5.6 \text{ Ratio.}$$

Rule—"Initial press., divided by ratio of areas," thus

$$\frac{200}{5.6} = 35.7 \text{ H. P. terminal.}$$

To prove the above graphically, draw vertical lines from E F G and measure horizontally in inches and divide length of diagram by it, for instance, for 200 lbs. it measures 1.78 inches, thus

$$\frac{10}{1.78} = 5.6.$$

Try it also at other pressures as shown.

§ 64. **Compound Condensing Diagram.** To construct a combined diagram for double expansion condensing engines, with following data:

Initial steam press., 150 pounds abso.

Terminal press. in L. P. cylinder 6.5 abso.

Total expansions $\frac{150}{6.5} = 23.1.$

Ratio of cylinder areas $\sqrt{23.1} = 4.8$ to 1.

Expansions in H. P. cylinder $\sqrt{23.1} = 4.8.$

Terminal press. in H. P. Cylinder $\frac{150}{4.8} = 31.2$ abso.

Terminal drop in in H. P. cylinder 2.5 pounds.

Back press. in H. P. cylinder 28.7 pounds.

Expansions in L. P. cyl. $\frac{28.7}{6.5} = 4.4.$

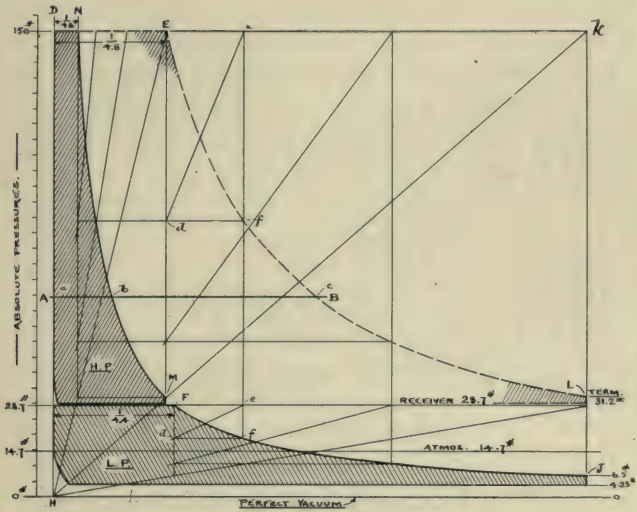


Fig. 15.

With the above data construct diagram, Fig. 15.
1st, draw horiz. line of perfect vacuum.
2nd, draw vert. line of H. D. from point H.
3rd, draw vert. line J K from point J.
4th, draw horiz. line to K from 150 lbs. press.
5th, draw diagonal line H K from point H.
6th, draw horiz. line for receiver press. 28.7 lbs. abso.
7th, lay off L. P. terminal on J K 6.5 abso.
8th, lay off H. P. terminal on J K 31.2 abso.
9th, lay off points of cut-off, H. P. $\frac{1}{4.8}$ and L. P. $\frac{1}{4.4}$
10th, draw hyperbolic curves E L and F J.

11th, contract H. P. diagram, horizontally, on a series of lines such as A B, in proportion to the ratio of cylinder areas, making in this diagram, a b equal to $\frac{a\ c}{4.8}$

12th, verify all three curves, prove them to be rectangular hyperbolas, by examining the right angle triangles, such as d e f, to see that curves pass through points f in every case.

§ 65 **Compound Non-Condensing Diagram.** Construct a compound diagram for double expansion non-condensing engine, with following data:

Initial steam pressure 154.7 pounds abso.

Terminal pressure in L. P. cylinder 17 abso.

Total expansion $\frac{154.7}{17} = 9.2$

Expansion in H. P. cylinder $\sqrt{9.2} = 3.05$

Ratio of cylinder areas $\sqrt{9.2} = 3.05$

Terminal press. in H. P. Cyl. $\frac{154.7}{3.05} = 50.4$

Terminal drop in H. P. Cyl. 2.2 pounds.

Back pressure in H. P. Cyl. 48.2.

Receiver pressure 48.2 pounds abso.

Expansion in L. P. Cyl. $\frac{48.2}{17} = 2.82$

With above data construct diagram, Fig. 16.

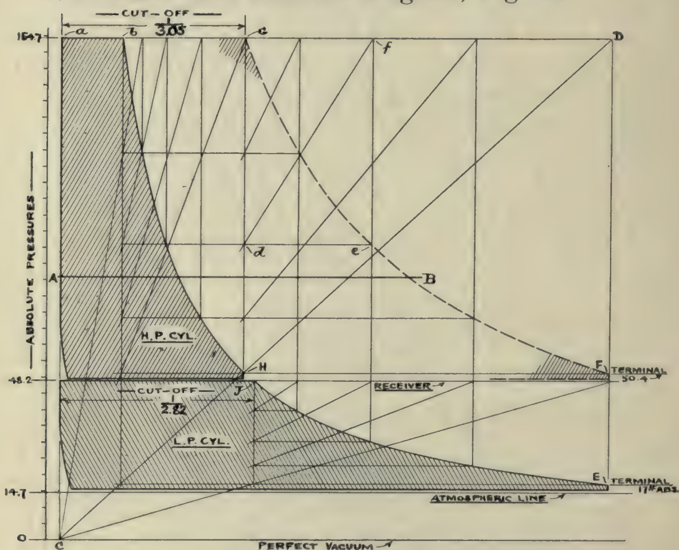


Fig. 16.

1st, draw line of perfect vacuum.
2d, draw vertical line from point c.
3d, draw vertical line E D.
4th, draw horz. line to D. from 154.7 lbs. abso.
5th, draw diagonal line C D.
6th, draw horz. line for receiver press. 48.2 abso.
7th, lay off L. P. terminal E equal to 17 abso.
8th, lay off H. P. terminal F equal to 50.4
9th, draw off points of cut off H. P. $\frac{1}{3.05}$ L. P. $\frac{1}{2.82}$
10th, draw hyperbolic curves F G and E J.

11th, Contract H. P. diagram horizontally at various levels, such as at A B, in proportion to the ratio of cylinder areas, viz.: in this case making

$$a\ G \div 3.05 = a\ b.$$

12th, verify hyperbolic curves, by examining right angle triangles, such as d f e, to see that curves pass through points e in every case.

§ 66. **Triple Expansion Diagram.** Construct a combined diagram for Triple Expansion Condensing, following data:

Initial steam pressure 200 pounds abso.
Terminal pressure in L. P. cyl. 6 pounds abso.

Total Expansions $\frac{200}{6} = 33.3$.

Expansions in H. P. cyl. $\sqrt[3]{33.3} = 3.2$

Ratio of areas, H. P. to I. P. areas $\sqrt[3]{33.3} = 3.2$

Terminal press. in H. P. cyl. $= \frac{200}{3.2} = 62$ pounds

Terminal Drop in H. P. Cyl. = 2½ pounds
Initial in Intermediate Cyl. $= 62 - 2.5 = 59.5$

Expansions in Intermediate $= \sqrt{\frac{59.5}{6}} = 3.1$

Ratio of Intermediate to Low $= \sqrt{\frac{59.5}{6}} = 3.1$

Terminal in I. P. Cyl. $= \frac{59.5}{3.1} = 19.2$ abso.

Terminal Drop in I. P. Cyl. 2.1 assumed.
Initial in L. P. Cyl. $19.2 - 2.1 = 17.1$ abso.

Expansions in L. P. Cyl. $\frac{17.1}{6} = 2.85$.

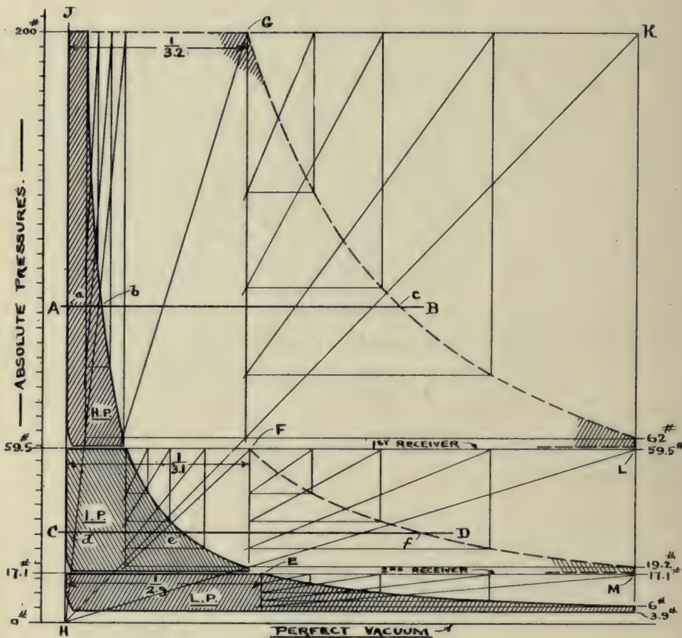


Fig. 17.

Fig. 17. Construct curves from points G, F, E as heretofore, and contract H. P. Diagram as at A, B to the ratio of High to Low, viz.: $3.2 \times 3.1 = 9.92$.

Also contract I. P. Diagram as at C D to the ratio of 3.1 to 1.

The rest of the diagram is similar to § 64 and § 65.

§ 67. **Combined Diagrams with Clearance.** Heretofore clearance volume was omitted for reasons given in § 61, but this article is to explain the ordinary method of laying out a combined diagram with clearance considered. Two diagrams are submitted, one for double expansion and another for triple.

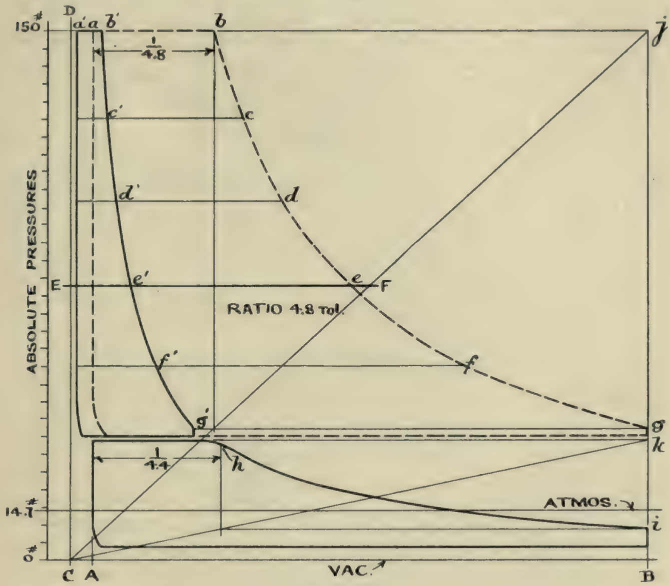


Fig. 18.

Double Expansion.

Steam pressure 135 lbs. gauge = 150 abso.

Clearance 4 per cent. Ratio of cyl. areas 4.8 to 1. On line of perfect vacuum lay off A B equal to length of proposed diagram. Draw vertical line C D at a distance from A which has the same ratio to the length of diagram A B as the clearance volume has to the piston displacement, in this diagram 4 per cent. Next construct two single diagrams by means of diagonal lines such as C j and C k, and further described in § 64, Fig. 15, except that they must radiate from C and not from A, as heretofore. Then contract the high pressure diagram horizontally toward vertical line C D in proportion to the ratio of cylinder areas, viz.: 4.8 to 1, by drawing any number of lines parallel to base, such E F and from points of intersection such as b c d e f g find new points b′ c′ d′ e′ f′ g′, also set back steam line a toward clearance line C D in the same proportion.

Triple Expansion. On the same principle, construct a triple expansion combined diagram, referring to Fig. 19. Initial press. 190 lbs. abso. Clearance of each cyl. 6 per cent. Ratio of areas high to low 1 to 9. Intermediate to low 1 to 3. Cut-off ⅓ in each cylinder.

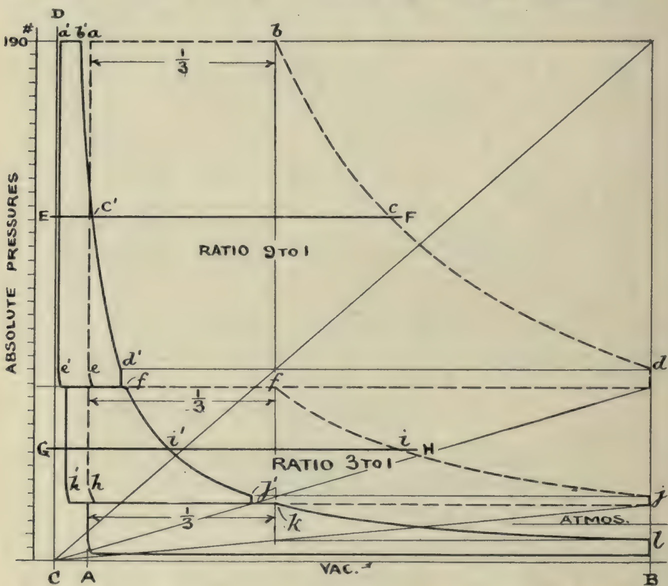


Fig. 19.

This diagram, Fig. 19, is simply to illustrate the method of combining triple exp. diagrams. The steam pressure and ratio assumed is simply to explain the point under consideration. This ratio with 190 lbs. steam pressure is not recommended, but the principle holds good.

§ 68. **Improved Method of Combining.** To construct a theoretical diagram which shall most nearly represent the actual work done in an actual engine the following method is recommended. Referring to Fig. 20.

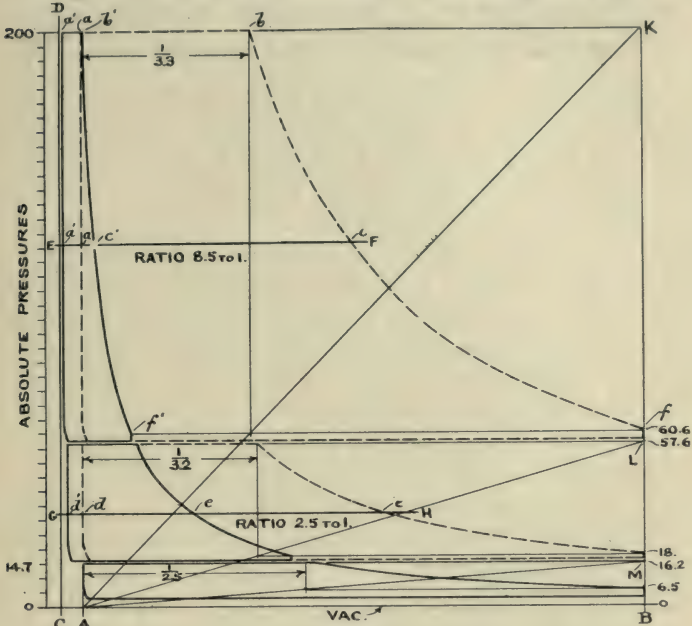


Fig. 20.

The expansion curves in dotted lines are laid out from point A as though there was no clearance, but the contraction of the diagrams is toward line C D, which includes the clearance, each point being contracted in proportion to the ratio of areas, as shown on lines E F and G H.

If a continuous expansion line was drawn from point b^1 down to the low press. terminal, as though all the work was done in one cylinder, it would be found that the three expansion curves shown in Fig. 20 would not exactly coincide, but that is of second importance to that of having a more exact representation of the work actually done.

CHAPTER No. 6.

HORSE-POWER AS A STANDARD MEASUREMENT OF WORK PERFORMED IN A STEAM ENGINE.

§ 69. **Work.** No attempt will be made to fully discuss the subject of work, but simply to treat it sufficiently to show how it is produced, and how it can be measured by what is universally termed in steam engine practice as Horse-Power. Work is the overcoming of resistance by force. Force is energy exerting itself, but energy is not power until its force as an element is combined with two other elements, viz.: Space and Time. We will briefly consider the subject of work in the next five successive articles, viz.:

Energy.
Force.
Space and Time.
Power.
Horse-Power.

§ 70. **Energy.** Energy is defined as "The ability of an agent to perform work and exists in two forms, viz.: Potential and Kinetic." The first of these two forms is inactive and of no value until transformed into the latter. Potential Energy, so far as the steam engine is concerned, exists in the fuel to be burned, for the purpose of making steam. The carbon contained in the fuel has a powerful affinity for oxygen, but so long as it remains unsatisfied it cannot perform any service whatever, and is powerless to produce the slightest physical effect, as it must remain inert until permitted to assume conditions in which the carbon can be supplied with the necessary oxygen for combustion, and the stored-up energy released. The chemical action of combustion transforms the potential-energy into Kinetic-Energy, which energy becoming actual by the burning of the coal and is rendered available by the transfer of its resultant heat to the water enclosed within the boiler; thus generating steam under pressure in which the heat-energy is stored ready to be transformed into mechanical-energy through the medium of the engine.

§ 71. **Force.** *Definition, "That which produces, or tends to produce motion or change of motion in bodies, and is measured dynamically by the velocity which it produces, acting on a unit of mass in a unit of time."*

Newton's first law of motion reads: "Every body continues in its state of rest or uniform motion in a straight line, except in so far as it may be compelled by impressed forces to change that state."

§ 72. **Resistance.** Force and Resistance, when equal to each other, and acting upon opposite sides of a body, will not move it from its state of rest; but whatever force may be exerted upon that body to move it through space at any velocity must be opposed by an opposite resistance, or sum of resistances, which shall exactly equal the force applied in imparting movement to it, before it can be brought to rest again. This resistance may be entirely due to the friction of a machine, or in part due to friction and part to the useful product of that machine. Such is true of the steam engine, wherein the effective force exerted upon the piston by the pressure of the steam sufficient to impart to it certain momentum, must sooner or later find an equal and opposite resistance before it can be brought to rest again; but so long as the effective force is maintained, just so long will the machine be capable of producing useful results.

In steam engine practice, force is expressed in pounds exerted per square inch, and is expressed as pressure per square inch, hence a unit of force is a pressure of one pound.

§ 73. **Space and Time.** Force is not power until combined with the two other elements, space and time. Space or distance through which a piston moves is measured in feet. The standard foot, which is one-third of the Imperial standard yard, is kept in the Exchequer of Westminster, England. Time is expressed as per second, per minute or per hour as the case may be; for instance, a falling body feet per second. Piston travel feet per minute. Heat units consumed per I. H. P. per hour, etc.

§ 74. **Power.** *Definition, "The rate of doing work, or the number of foot-pounds exerted in a unit of time."* If, then, a unit of work is one-pound operating through one-foot of space, one foot-pound, and if power is the rate at which work is done, and the unit of time is taken as one minute, then we have the three necessary elements for measurement of power, viz.: Pound, Foot, Minute. So, also, the power of a steam engine can be calculated by obtaining the three necessary requirements, viz.:

1. Mean Eff. Pressure in Pounds on Piston.
2. Effective Area of Piston in Square inches.
3. Piston Speed in feet per minute.

A few words are necessary here upon what is understood by "Mean Eff. Pressure" (M. E. P.). The steam engine assumes four kinds of pressure exerted upon a piston, viz.:

(1) Forward Pressure, (2) Back Pressure, (3) Effective Pressure, (4) Mean Eff. Pressure.

Forward pressure is that force which is exerted upon one side of a piston, tending to move it in the direction for performing useful work. Back pressure is a force acting upon the opposite side of the same piston in resistance to the forward pressure, and Effective pressure is the difference between the forward and backward pressures, or the measure of force by which the forward pressure predominates over the backward. For illustration, steam at the atmospheric pressure when admitted to one side of a piston can perform no work or exert no useful force, except in so far as the atmosphere is removed from the opposite side. Likewise in a compound engine, where two or more cylinders are employed, the first receiving steam direct from the boilers, and exhausting it into a receiver ready to supply the next cylinder. No work whatever can be done in the first cylinder unless the forward pressure exceeds the receiver pressure, even though the steam is not cut off but remains uniform to the end of the stroke, for the reason that if the initial pressure is equal to the terminal pressure in the first cylinder and the receiver pressure is also equal to the terminal as under some conditions it might be, the one would counteract the other; consequently the first cylinder would be of no value whatever except to conduct the steam to the next; but where the steam is made to expand from a higher initial pressure to a lower terminal pressure, the mean or average force or pressure produced thereby is opposed only by a resistance equal to the terminal or receiver pressure, and that being less than the average forward pressure, leaves an effective force capable of performing useful work upon the first piston. Mean Eff. Pressure is really the effective pressure, but the word "mean" takes into account the varying pressure of steam due to its expansion, after the supply of steam is cut off, and considers the average forward effect.

§ **75. Horse-Power.** It is explained above that one pound pressure moving a body through a distance of one foot is regarded as one foot-pound. Now the work done upon the piston of a steam engine is the average pressure of steam exerted upon it, and causing the same to move through a certain distance; but unless the time occupied in passing through that space is taken into account no definite idea can be expressed to indicate the capacity of the agent employed.

Watt therefore established a standard measurement by which the work performed might be expressed, and as the result of some experiments made with powerful horses, he concluded that one horse was capable of performing 33,000 foot-pounds of work in one minute, which he called "one horse-power," and which has since that time been almost universally accepted as such. Therefore "a quantity of work equivalent to the raising of 33,000 pounds, through one foot of space, in one minute, is established as one horse-power," or, a piston having an effective area equal to 330 square inches, and moving at a velocity of 100 feet per minute, is equal to one horse-power for each pound effective pressure.

Definition of horse-power: "A unit of Mechanical power."

Brake horse-power (B. H. P.) is a measurement of power determined by a brake applied to the rim of a wheel upon which the brake shoes maintain a constant resistance. It therefore takes into account the delivered horse-power only, and does not include the friction of engine.

Indicated horse-power (I. H. P.) refers to the actual horse-power developed in the cylinders of an engine including friction of engine also, and is determined by the use of the steam engine Indicator. All examples in this treatise when horse-power is given is assumed as indicated horse-power.

Approximately One Brake H. P. is equal to .88 of one Indicated H. P.; for instance, the B. H. P. equivalent for 100 I. H. P. is $100 \times .88 = 88$.

§ 76. **Horse-Power Formula.** The work done in foot-pounds on a piston may be expressed thus,

Work = P L A N, in which

P = Pressure in pounds per square inch.

L = Length in feet of piston travel per revolution.

A = Area of piston in square inches.

N = Number of revolutions per minute.

Since one horse-power unit is 33,000 foot-pounds, the work done in horse-power, on a piston, will be

$$\frac{\text{PLAN}}{33{,}000} = \text{H. P.}$$

§ 77. **Boiler Horse-Power.** Error has often been made by supposing that the word "Horse-Power" expresses a measurement of capacity which is common to both Boiler and Engine. This is not so; the expression is an unfortunate one when applied to boiler capacity. The term "Engine Horse-Power" correctly expresses a measurement of engine capacity for performing work. And for the want of a better term, "Boiler Horse-Power" is used to express a measurement of boiler capacity for evaporating water into steam, but the two have no analogy whatever.

The horse-power of a Boiler should always be figured upon an independent basis from the horse-power of an Engine, for it does not follow that the steam necessary for a 100 H. P. engine will need a 100 H. P. boiler for its generation, for the capacity of the accompanying boiler depends relatively upon the Efficiency as well as the capacity of the engine, for which it is to furnish steam. For instance, a single cylinder engine of the non-condensing type, may need 30 pounds of steam per I. H. P. per hour; but the modern triple expansion engine will probably need no more than 10½ pounds per I. H. P. per hour, which shows that the capacity of the accompanying boiler would be very different in the two cases.

The old method of testing the efficiency and capacity of an engine, by the amount of fuel consumed under the boiler, has proved to be a mistake, and should never be resorted to; but the capacity and efficiency of the boiler should be tested independently of the engine, and upon no other basis than the quantity of water evaporated into steam, per hour, per pound of fuel.

From the year 1876 until 1899 the unit of Boiler Horse-Power, known as the Centennial Standard, was:

"One Boiler Horse-Power is equivalent to evaporating 30 pounds of water per hour, from 100 degrees F., into steam at 70 pounds pressure."

But in 1899 the American Society of Mechanical Engineers adopted a new and more desirable unit, which is:

"One Boiler Horse-Power is equivalent to evaporating 34½ pounds of water from and at 212 degrees F., into steam of the same temperature, or 14.7 pounds abso. or atmospheric pressure."

For a first-class boiler for moderate pressures see description of the "Freeman" on last page but one in this book.

CHAPTER NO. 7.

MULTIPLE EXPANSION.

§ 78. **History.** The principle of the multi-cylinder engine is not by any means a new discovery. The practice was anticipated as early as the year 1781 and is set forth in the specifications of the Hornblower patent of that date, in which he says: "First, I use two vessels in which the steam is to act, and which are generally called cylinders. Second, I employ the steam after it has acted in the first vessel, to operate in a second, by permitting it to expand itself, which I do by connecting the vessels together, and forming proper channels and apparatus whereby the steam shall occasionally go in and out of said vessels."

But, even though the principle was anticipated, it remained undeveloped for nearly a century, during which time every effort made was attended with numerous discouragements and consequently doubt. To-day, however, there remains no question of doubt as to its utility, and confidence in the principle has encouraged perseverance until that which was at one time only a theory, has now become a fact, confirmed by both science and experience.

Prof. Carpenter, in one of his papers, says: "The Compound condensing engine of the best type and with improved valve gear, although costing somewhat more, is so much more economical to operate, that taking all things into consideration, it gives much better financial returns than an engine of any other class."

§ 79. **Secret of Success.** The secret of success in multi-expansion engines lies in a three-fold requirement: 1st, in using steam of the highest pressures obtainable; 2nd, in expanding the steam down to the lowest terminal pressure commercially profitable; 3rd, in reducing to a minimum the loss caused by cylinder condensation. Each of these features is discussed in § 21, § 80 and § 89.

In a general way the conditions favorable to best economy are:

High Pressure Steam, Superheated.
Proper Ratio of Cylinder Areas.
High Piston Speed.
Low Terminal Pressure.
Good Vacuum, 27 or 28 inches.
Uniform Load.
Long Stroke.
Steam Jackets.
Uniform Boiler Pressure.

§ 80. **Terminal Pressures and Loop.** Since the total rate of expansion depends upon the terminal pressure in the low press. cyl., it is important to ascertain the conditions which will effect it. For instance, if it is a condensing engine, and the condenser is to be furnished by another builder, the first question to ask is, how many inches of vacuum are guaranteed? The "Inches of Vacuum" being ascertained, refer to table No. 15, § 126, and find the absolute press. in pounds which correspond to the number of inches named; then to this number of pounds add

5.5 to 6 lbs. abso. for Single engines.
5 to 5.5 lbs. abso. for Double expansion.
4.5 to 5 lbs. abso. for Triple expansion.
4 to 4.5 lbs. abso. for Quadruple expansion.

For example, if the number of inches of vacuum guaranteed is 27, this corresponds to 1.47 lbs. abso., then the terminal pressures would be as shown in table No. 8, which are very good where first-class economy is expected.

Loop. In figuring the terminal pressure in either a high press. cyl. of a compound, or a low press. non-condensing, care should be taken to allow for some terminal drop when running with normal load; so that when the load is reduced to minimum, the expansion curve is not carried down below back press. line. When such occurs there is formed in the indicator diagram what is known as a "negative loop." See Fig. 21, with following explanation:

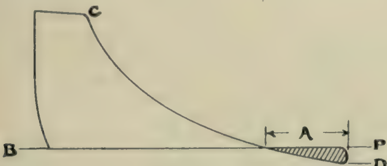


Fig. 21.

Upon opening the exhaust valve as at D, the line rises to P, and returns above terminal pressure, cutting the expansion line at a distance A from commencement of return stroke, thus forming the objectionable loop. This usually indicates two things, 1st that the capacity of the cylinder exceeds that necessary for the work performed, 2nd that too large a percentage of the work performed is done simply in overcoming the friction of the engine. There is also a serious mechanical objection to the loop, which is that of "slamming" of the exhaust valves as they are lifted from their seats and fall again every time the loop is formed.

TABLE No. 8.

TERMINAL PRESSURES IN LOW PRESSURE CYLINDERS.

Type of Engine, Condensing.	Absolute Pressure.
Single Cylinder	7. to 8. lbs.
Double Expansion	6.5 to 7. lbs.
Triple Expansion	6. to 6.5 lbs.
Quadruple Expansion	5.5 to 6. lbs.

§ **81. Rate of Expansion.** High pressure steam would be of but little value commercially without a corresponding increase in the total expansions. The total number of expansions is determined by the ratio between the initial abso. pressure in the high press. cyl., and the terminal pressure in the low, irrespective of the number of cylinders or stages employed.

Having the initial pressure given, then to determine the *Total Number of Expansions* first decide upon the best terminal pressure (§ 80), after which use the following rule:

Rule.—"Divide the absolute Initial press. by the absolute Terminal pressure."

Try the four following examples:

NOTE.—The number found will be the "total" expansions, including the expansion in the receivers as well as in the cylinders.

First Ex. Initial press. 45 lbs. gauge, 60 abso.
Terminal press 7 lbs. abso.

$$60 \div 7 = 8.5 \text{ expansions}$$

Second Ex. Initial press. 130 lbs. gauge, 145 abso.
Terminal press. 6.5 lbs. abso.

$$145 \div 6.5 = 22.3 \text{ expansions}$$

Third Ex. Initial press. 170 lbs. gauge, 185 abso.
Terminal press. 6 lbs. abso.

$$185 \div 6 = 30.8 \text{ expansions}$$

Fourth Ex. Initial press. 230 lbs. gauge, 245 lbs. abso.
Terminal press. 5.5 lbs. abso.

$$245 \div 5.5 = 44.5 \text{ expansions}$$

§ **82. Expansions in Each Cylinder.** The question of "terminal drop" decides whether or not the expansions in each cylinder should be equal. When there is no terminal drop, except in the low press. cylinder, then there is no good reason why the number of expansions cannot be the same in each cylinder; but when drop is considered necessary (§ 84) then the cut-off should be later in the low pressure.

Without Terminal Drop. The approximate number of expansions in each cylinder, without drop, except in

TABLE No. 9.

NUMBERS, THEIR SQ. ROOTS AND CUBE ROOTS.

SQUARE ROOT.	NO.	CUBE ROOT.	SQUARE ROOT.	NO.	CUBE ROOT.	SQUARE ROOT.	NO.	CUBE ROOT.	SQUARE ROOT.	NO.	CUBE ROOT.
3.16	10.	2.15	4.24	18.	2.62	5.10	26.	2.96	5.83	34.	3.24
3.19	10.2	2.16	4.26	18.2	2.63	5.12	26.2	2.96	5.84	34.2	3.24
3.22	10.4	2.18	4.28	18.4	2.64	5.14	26.4	2.97	5.86	34.4	3.25
3.25	10.6	2 19	4.30	18.6	2.64	5.16	26.6	2.98	5.87	34.6	3.26
3.28	10.8	2.20	4.33	18.8	2.65	5.18	26.8	2.99	5.89	34.8	3.26
3.31	11.	2.22	4.35	19.	2.66	5.19	27.	3.00	5.91	35.	3.27
2.34	11.2	2.24	4.38	19.2	2.67	5.21	27.2	3.01	5.92	35.2	3.27
2.37	11.4	2.25	4.40	19.4	2.68	5.23	27.4	3.01	5.94	35.4	3.28
3.40	11.6	2.27	4.43	19.6	2.69	5.25	27.6	3.02	5.96	35.6	3.28
3.43	11.8	2.28	4.45	19.8	2.70	5.27	27.8	3.03	5.98	35.8	3.29
3.46	12.	2.29	4.47	20.	2.71	5.29	28.	3.03	6.00	36.	3.30
3.49	12.2	2.30	4.50	20.2	2.72	5.30	28.2	3.04	6.01	36.2	3.30
3.52	12.4	2.32	4.52	20.4	2.72	5.32	28.4	3.04	6.03	36.4	3.31
3.55	12.6	2.33	4.54	20.6	2.73	5.34	28.6	3.05	6.04	26.6	3.32
3.58	12.8	2.34	4.56	20.8	2.74	5.36	28.8	3.06	6.06	36.8	3.32
3.60	13.	2.35	4.58	21.	2.75	5.38	29.	3.07	6.08	37.	3.33
3.63	13.2	2.37	4.60	21.2	2.76	5.39	29.2	3.07	6.09	37.2	3.33
3.66	13.4	2.38	4.63	21.4	2.77	5.41	29.4	3.08	6.11	37.4	3.34
3.69	13.6	2.39	4.65	21.6	2.78	5.43	29.6	3.08	6.12	37.6	3.34
3.71	13.8	2.40	4.67	21.8	2.79	5.45	29.8	3.09	6.14	37.8	3.35
3.74	14.	2.41	4.69	22.	2.80	5.47	30.	3.10	6.16	38.	3.36
3.76	14.2	2.42	4.71	22 2	2.80	5.49	30.2	3.10	6.17	38.2	3.37
3.79	14.4	2.43	4.73	22.4	2.81	5.50	30.4	3.11	6.19	38.4	3.37
3.82	14.6	2.44	4.75	22.6	2.82	5.52	30.6	3.12	6.20	38.6	3.38
3.85	14.8	2.45	4.77	22.8	2.83	5.54	30.8	3.13	6.22	38.8	3.38
3.87	15.	2.46	4.79	23.	2.84	5.56	31.	3.14	6.24	39.	3.39
3.90	15.2	2.48	4.81	23.2	2.84	5.58	31.2	3.14	6.25	39.2	3.39
3.92	15.4	2.49	4,83	23.4	2.85	5.60	31.4	3.15	6.27	29.4	3.40
3.95	15.6	2.50	4.85	23.6	2.86	5.61	31.6	3.16	6.28	39.6	3.41
3.98	15.8	2.51	4.87	23.8	2.87	5.63	31.8	3.17	6.30	39.8	3.41
4.00	16.	2.52	4.89	24.	2.88	5.65	32.	3.17	6.32	40.	3.42
4.03	16.2	2.53	4.9	24.2	2.88	5.67	32.2	3.18	6,33	40.2	3.42
4.05	16.4	2.54	4.92	24.4	2.89	5.68	32.4	3.18	6.35	49.4	3.43
4.08	16.6	2.55	4.95	24.6	2.90	5.70	32.6	3.19	6.36	40.6	3.43
4.10	16.8	2.56	4.97	24.8	2.91	5.72	32.8	3.19	6.38	40.8	3.44
4.12	17.	2.57	5.00	25.	2.92	5.74	33.	3.20	6.40	41.	3.45
4.14	17.2	2.58	5.02	25.2	2.92	5.76	33.2	3.20	6.41	41.2	3.45
4.17	17.4	2.59	5.04	25.4	2.93	5.77	33.4	3.21	6.43	41.4	3.46
4.19	17.6	2.60	5.06	25.6	2.94	5.79	33.6	3.22	6.45	41.6	3.46
4.22	17.8	2.61	5.08	25.8	2.95	5.81	33.8	3.23	6.46	41.8	3.47

the low, that is to say "without receiver drop," would be as follows:

For Double Expansion Engines. Rule.—"Extract the square root of total expansions." $\sqrt{\text{Total}}$.

For Triple Expansion. Rule.—"Extract the cube root of total expansions." $\sqrt[3]{\text{Total}}$.

For Quadruple Expansion. Rule.—"Extract the fourth root of total expansions." $\sqrt[4]{\text{Total}}$.

For instance, for Double exps., with 145 lbs. abso. initial, and 6½ abso. terminal; and since the total is equal to initial divided by terminal (§ 81), we have

$$\sqrt{\frac{145}{6.5}} = 4.75 \text{ exps. in each cylinder.}$$

For Triple, with 185 abso. initial, 6 abso. terminal, we have $\sqrt[3]{\frac{185}{6}} = 3.15$ exps. in each.

For Quadruple, with 245 abso. initial, and 5½ terminal, we have $\sqrt[4]{\frac{245}{5.5}} = 2.55$ exps. in each.

With Terminal Drop. The process is a little more complicated when terminal drop is considered, as each cylinder will have a different number of expansions, each must be figured independently, beginning with the high pressure in each case.

Double Expansion. For expansion in high press. *Rule.—"Extract square root of total expansions."* For expansions in low press. *Rule.—"Divide receiver pressure by low press. terminal."* For instance, with 145 abso. initial, and 6½ abso. terminal, we have $\sqrt{\frac{145}{6.5}} =$ 4.75 Expansions in H. P. Cyl. Receiver pressure equals initial press. in high press. cyl. divided by expansions in same, minus the terminal drop, thus, allowing say, 2½ lbs. drop

$$\frac{145}{4.75} - 2.5 = 28 \text{ pounds abso. Receiver}$$

thus L. P. terminal being 6.5 abso. we have

$$28 \div 6.5 = 4.3 \text{ Exps. in L. P. Cyl.}$$

Triple Expansion. For expansions in H. P. Cyl. *Rule.—"Extract cube root of total expansions."* For expansions in Intermediate Cyl. *Rule.—"Divide first receiver pressure by L. P. terminal and extract square root of quotient."*

For expansions in L. P. Cylinder:

Rule.—"Divide second receiver pressure by L. P. terminal."

For example, with 185 abso. initial, and 6 abso. terminal, we have $\sqrt[3]{\frac{185}{6}} = 3.1$ Exps. in H. P. Cyl.

First receiver press. equal terminal in high, minus terminal drop of say 2½ pounds, thus

$$\frac{185}{3.1} - 2.5 = 57.5 \text{ abso. 1st Receiver}$$

then for Intermediate expansions we have

$$\sqrt{\frac{57.5}{6}} = 3.05 \text{ Exps. in I. P. Cyl.}$$

and for L. P. expansions.

Second receiver press. equals terminal in I. P. Cyl. minus terminal drop of say 2 pounds.

$$\frac{57.5}{3.05} - 2 = 16.8 \text{ abso. 2nd Receiver}$$

$$\frac{16.8}{6.} = 2.8 \text{ Exps. in L. P. Cyl.}$$

Quadruple Expansion. For expansions in H. P. Cly.:

Rule.—"Extract fourth-root of total expansions."

For expansions in 1st Intermediate Cylinder:

Rule.--"Divide 1st receiver press. by low press. terminal and extract cube-root of quotient."

For expansions in 2nd Intermediate:

Rule.—"Divide 2nd receiver press. by low press. terminal, and extract square-root of quotient."

For expansions in L. P. Cylinder:

Rule.—"Divide 3rd receiver pressure by low press. terminal."

In each case the receiver pressure would be equal to the preceding terminal, minus drop.

§ **83. Temperature Ranges.** By this expression "temperature ranges" we mean that inasmuch as that the temperature of the steam falls as expansion progresses, it passes through a total change from that due to the initial pressure in the H. P. cyl., down to that due to the back pressure in the L. P. cyl. For instance, if the initial steam is 200 lbs. abso., and the back press. in the low is 3.5 abso., then by Table No. 4 we see that there is a total range or fall in temperature from 381° to 146°, viz., 235° Fahr. reduction. Now if all this expansion was done in one clyinder, and the walls of the same became alternately heated and cooled to nearly this extent, there would be great loss of condensation. Herein lies one of the chief values of the multiple expansion principle, which provides for dividing the total expansion range into two or more stages by the employment of two or more cylinders, thus reducing the temperature range in any one cylinder to the minimum, and so far as other conditions will allow, to the same range in each.

But usually where the temperature ranges are exactly divided, the work done in the low press. is greater than that of the high as shown by diagram Fig. 22, in which with 150 lbs. abso., cutting off at ¼ in the high, and 3 lbs. receiver drop, the temperature ranges from 358° to 258° in high, and from 258° to 158° in low, making

100° fall in each cylinder. This condition is less objectionable in tandem engines than in cross-compound where the division of work is more important.

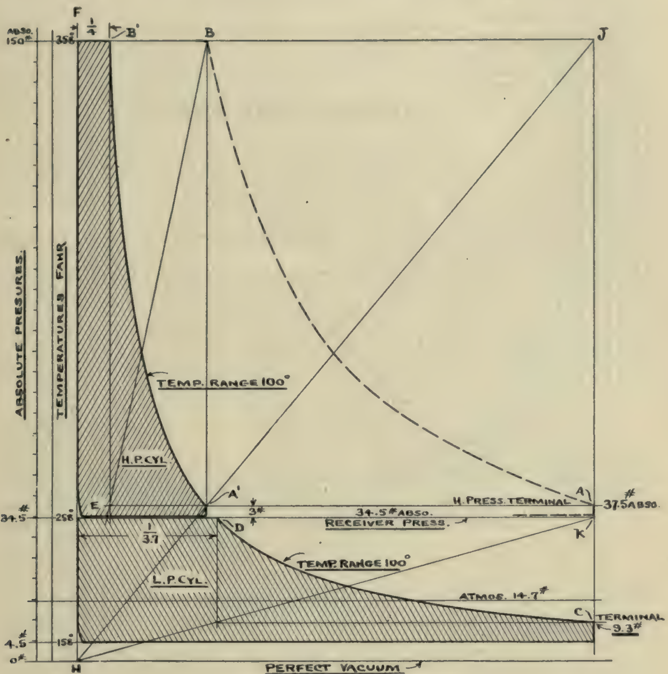


Fig. 22.

But if we regard the equal division of work as of more importance than the equal division of temperatures, and considering that the high press. cylinder has less wall surface for condensation than the low, then we can afford to make the temperature range a little the greater in the high. In view of this refer to Fig. 23 which gives a good illustration of good practice with a small amount of receiver drop.

§ 84. **Terminal Drop.** By the expression "terminal drop" or "receiver drop" is meant the drop in pres-

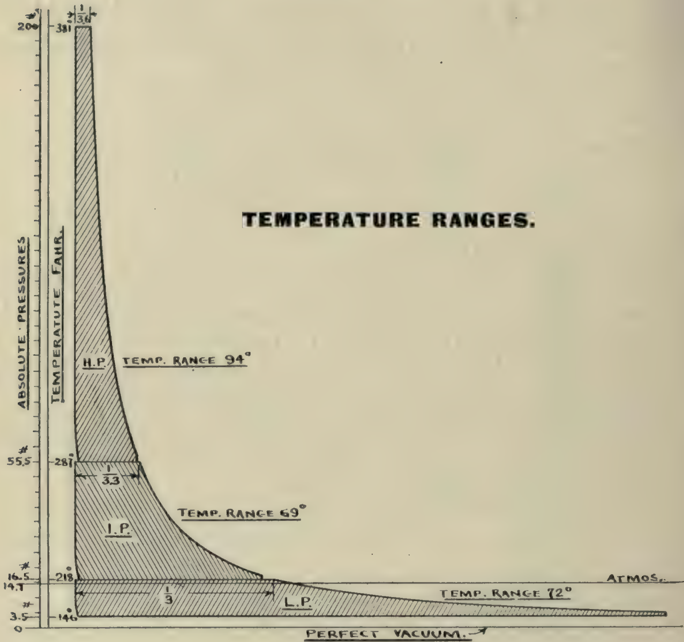


Fig. 23.

sure at the end of the stroke from terminal press. to back press.

There should always be some terminal drop when the engine is running with normal load, for two reasons, Fig. 24. First: If the expansion line is carried down

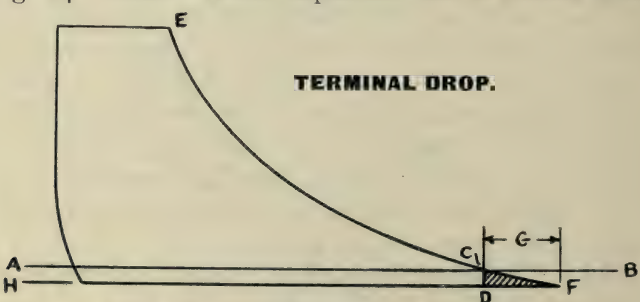


Fig. 24.

to a sharp point F when load is normal, then on the slightest decrease of load a loop is formed which is objectionable. § 80. Second: If the work done in the last few inches of the stroke represented by the shaded portion F C D is only enough to overcome the friction of the engine, then there is no useful work performed during that part of the stroke G, and the stroke might as well be shortened up by that amount, and the terminal drop become C′ D. If, for instance, the mean eff. press. is 33 pounds and the friction 6 per cent. then the average pressure of the shaded portion F C D is 2 lbs., and the terminal drop should be at least that much.

Later cut-off. By reference to Fig. 25 it will be seen that where there is terminal drop, the cut-off in the low press. cylinder should be later than in the high.

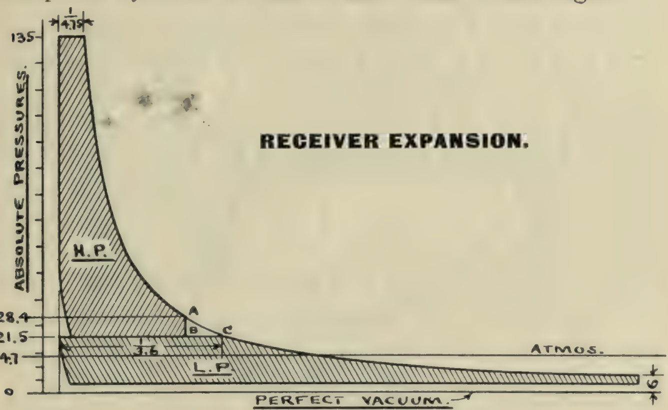


Fig. 25.

It is important for best economy, to make a continuous expansion curve in the combined diagram, as though all expansion was done in one cylinder. Therefore if there is terminal drop, there must be some expansion in the receiver by taking out more cubic feet than is put in, stroke for stroke, and this is effected by a later cut-off in the low, thus removing from the receiver enough volume to cause the exhaust from the high to expand to a lower pressure than its terminal.

This receiver expansion performs no work but there is nothing lost, as the heat given out during this time helps to re-evaporate the moisture in the steam. The space A B C is known as free expansion, and the greater the drop A B becomes, the further should point C run down the curve, making a later cut-off in proportion.

Terminal drop increases the percentage of the work done in the high pressure cylinder, This explains the peculiar fact, that the later the low press. cut-off, the less will be the work done in that cylinder relatively to the high.

§ **85. Relative Volumes.** In order that the initial pressure in the low be equal to the terminal in the high, that is to say, no terminal drop, the low press. cut-off must take place at such a point as to take from the receiver an exactly equal volume to that discharged into it from the high in a given time. For if the volume is greater the pressure must be less, if the volume is less the presure must be greater; on the theory that the pressure is inversely as its volume. § 49. If therefore, the low press. cut-off is too early, the back pressure will be higher than the terminal and form the objectionable loop. § 80, Fig. 21. Or if later there must be terminal drop. § 84, Fig. 24. To find the correct low press. cut-off where no terminal drop is required, work following example. Assume a 20 and 40 x 60 compound, clearance in high 2 per cent. and low 3 per cent. of the piston displacement. Fig. 26.

HIGH PRESSURE.

20″ dia. = 314 sq. inches area.
2 per cent. = $60 \times .02 = 1.2$ inches
$60 + 1.2 = 61.2$
$314 \times 61.2 = 19216$ c. in.

LOW PRESSURE.

40″ dia. = 1256 sq. inches area.

Volume discharged into receiver = 19216 c. in., loss by condensation, say 166 c. in.

Vol. to be taken out by low press. cyl., 19050

$\therefore$ 19050 ÷ 1256 = 15.2 lineal inches.

Clearance 3 per cent. = 60 × .03 = 1.8

15.2 — 1.8 = 13.4

60 ÷ 13.4 = 4.48 expansions.

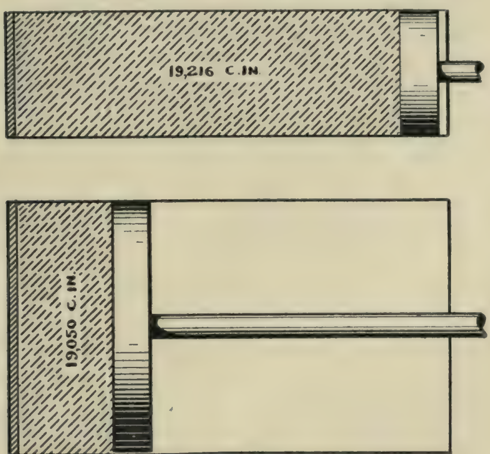


Fig. 26.

CLEARANCE.

Clearance is a necesary evil and positively unavoidable. But to reduce it to the minimum is the object of every good designer. The reducing of clearance is of more importance in the low press. cylinder than in the high, as the latter passes to the receiver for further use, but the former to the condenser and is lost.

§ 86. **Percentage of Clearance.** A good Corliss cylinder with valves in the barrel will average 3½ to 5 per cent. of piston displacement. With valves in the heads 2½ to 4 per cent. Usually the low press. cylinder figures a little higher percentage than the high. Short-stroke engines also figure higher than long-stroke, as the piston may run as near the heads in either case, but the port areas being larger the volume is increased and the percentage of displacement decreased. With pistons and heads faced off, the space between may be reduced to ¼ inch, though ordinarily ½″ is allowed.

§ **87. Effect of Clearance upon M. E. P.** The steam necessary to fill the Clearance Space is not entirely lost as it helps to compensate for the volume lost by condensation (see § 61), and tends to increase the mean eff. pressure, but the same volume of steam if used in a cylinder having no clearance space, would produce more for the heat expended.

TABLE No. 10.

STEAM PRESSURES AND VOLUMES.

Abso. Press. Lbs.	Volume of 1 lb. in cu.ft.	Abso. Press. Lbs.	Volume of 1 lb. in cu.ft.	Abso. Press. Lbs.	Volume of 1 lb. in cu.ft.	Abso. Press. Lbs.	Volume of 1 lb. in cu. ft.
1	330.36	41	10.03	82	5.23	156	2.85
2	172.08	42	9.81	83	5.17	158	2.82
3	117.52	43	9.59	84	5.11	160	2.79
4	89.62	44	9.39	85	5.05	165	2.71
5	72.66	45	9.18	86	5.	170	2.63
6	61.21	46	9.	87	4.94	175	2.56
7	52.94	47	8.82	88	4.89	180	2.49
8	46.69	48	8.65	89.	4.84	185	2.43
9	41.79	49	8.48	90	4.79	190	2.37
10	37.84	50	8.31	92	4.69	195	2.31
11	34.63	51	8.17	94	4.6	200	2.26
12	31.88	52	8.04	96	4.51	205	2.21
13	29.57	53	7.88	98	4.42	210	2.16
14	27.61	54	7.74	100	4.33	215	2.11
14.7	26.36	55	7.61	102	4.25	220	2.06
15	25.85	56	7 48	104	4.18	225	2.02
16	24.32	57	7.36	106	4.11	230	1.98
17	22.96	58	7.24	108	4.04	235	1.94
18	21.78	59	7.12	110	3.97	240	1.9
19	20.70	60	7.01	112	3.9	245	1.87
20	19.72	61	6.9	114	3.83	250	1.83
21	18.84	62	6.81	116	3.77	255	1.8
22	18.03	63	6.7	118	3.71	260	1.76
23	17.26	64	6.6	120	3.65	265	1.73
24	16.64	65	6.49	122	3.59	270	1.7
25	15.99	66	6.41	124	3.54	275	1.67
26	15.38	67	6.32	126	3.49	280	1.64
27	14.86	68	6.23	128	3.44	285	1.61
28	14.37	69	6.15	130	3.38	290	1.59
29	13.90	70	6.07	132	3.33	295	1.57
30	13.46	71	5.99	134	3.29	300	1.54
31	13.05	72	5.91	136	3.25	310	1.5
32	12.67	73	5.83	138	3.2	320	1.46
33	12.31	74	5.76	140	3.16	330	1.42
34	11.97	75	5.68	142	3.12	340	1.38
35	11.65	76	5.61	144	3.08	350	1.33
36	11.34	77	5.54	146	3.04	360	1.29
37	11.04	78	5.48	148	3.	375	1.25
38	10.76	79	5.41	150	2.96	400	1.18
39	10.51	80	5.35	152	2.93	450	1.05
40	10.27	81	5.29	154	2.89	475	1.

For closer figuring, consult Steam Tables by C. H. Peabody.

CHAPTER No. 8.

STEAM JACKETS.

§ 88. **Purpose of Steam Jackets.** The purpose of the steam jacket is to hold the temperature of the cylinder walls up to, or a little higner than, the temperature of the incoming steam, so that during expansion, while the temperature of the steam is falling, the metal in contact may maintain as near as possible a uniform temperature, and be as hot at the end of the stroke as it was at the beginning; which if successfully done will tend, first to reduce the degree of condensation, and second to re-evaporate the moisture of the enclosed steam, by the heat from the surrounding jacket steam, rather than by the heat of the expanding steam. For there is no thermal value in re-evaporation during expansion unless caused by external means.

§ 89. **Cylinder Condensation.** During the expansion of steam from the higher to the lower pressures, and the consequent change of temperature, the cylinder, heads, and piston become alternately heated and cooled to an extent varying according to the number of expansions. Therefore, with excessive expansion comes excessive condensation. § 83. For if the metal could be reduced in temperature as rapidly as the steam, then on the admission of the volume of steam necessary for the next stroke the temperature being higher due to the initial pressure, and coming in contact with the cooler surfaces effects a loss. This is an argument against a high rate of expansion in a single cylinder.

In the year 1769 Watt discovered this loss due to the change of temperature in the cylinder, as caused by what was customary previous to that time, viz., that of injecting the cold water for condensing, directly into the steam cylinder. He said: "That vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in common fire engines, and which I call the steam vessel, must, during the whole time the engine is at work, be kept as hot as the steam which enters it." Again: "In those engines that are to be worked wholly or partially by condensation of steam, the steam is to be condensed in vessels distinct from the cylinders, though occasionally communicating with them. These vessels I call the condensers, and whilst the engines are working they ought to be kept as cool as the air in that neighborhood, by the application of water or other cool bodies." This discovery led, not only to the introduction of a separate condenser, but also to that of the Steam Jacket.

§ 90. **Economy of Steam Jackets.** Judging from a number of expert reports, of tests made under various conditions, the saving effected by the steam jacket seems to vary from 3 per cent. to about 20 per cent. The higher percentage of saving being found in those cases where the rate of expansion in any one cylinder is greater; and the lower percentage where the cut-off is late and the rate of expansion low. No rule can be given, but some guidance may be found by the following approximation:

Cut-off	1/10,	saving	25	per cent.
"	1/8,	"	16	"
"	1/6,	"	12	"
"	1/4,	"	9	"
"	1/3,	"	6	"
"	1/2,	"	3	"

Upon the strength of the above showing, the steam jacket would appear to be of some value even in the quadruple expansion engine where the expansion in each cylinder is of low rate. And while it may appear small in comparison with the higher rate, nevertheless where greatest economy is sought and the first cost not too strictly curtailed, 5 per cent. or even 3 per cent. is good return for the extra money invested.

§ **91. Jacket Steam.** Live steam only should be used in the steam jacket, and never exhaust, for the reason that the exhaust cannot be hotter than the steam within the cylinder at its terminal pressure, and since the terminal temperature is lower than the initial temperature it is evident that the cylinder on the whole must be hotter than the exhaust steam; therefore, if exhaust steam is admitted to the jacket it must have the effect of cooling the cylinder rather than heating it, thus producing the opposite result from that sought for by jacketing.

Steam on its way to the cylinder should never be allowed to pass through the jacket, for the reason that to be of any value the jacket steam must liquify, whereas the steam supplied to the cylinder should be kept very dry, and the less there is of moisture the better will be the result.

Jacket steam, therefore, should be taken direct from the main steam pipe and discharged either into the receiver for the next lower pressure cylinder, or trapped back to the feed water as the case may demand.

CHAPTER No. 9.

SINGLE CYLINDER ENGINES.

Single cylinder engines are divided into two classes, viz., Condensing and Non-Condensing. The more economical being the condensing, and usually shows a saving of about 25 per cent. But when the exhaust steam is needed for purposes outside of the engine, or difficulties about injection water are encountered (§ 6), then the non-condensing engine becomes the choice. The accompanying cut, Fig. 27, shows a very substantial design for a single engine.

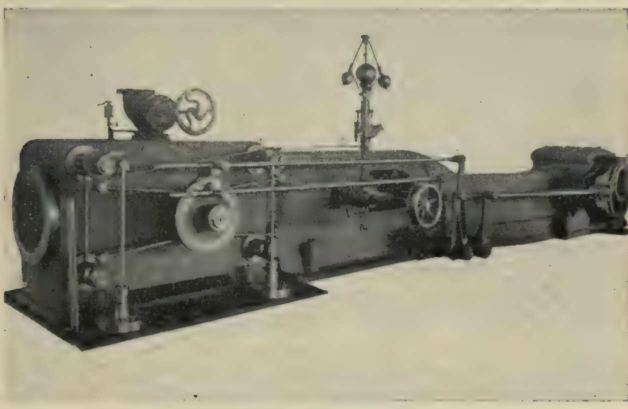


Fig. 27.

As a preliminary study, preceding the more advanced examples in multi-expansion, it will be advisable to work out some examples of finding the mean eff. pressures in single expansion engines, first in non-condensing and next in condensing.

§ 92. **Condensing, Single Cylinder.** Try three examples at different points of cut-off, in each example find the mean eff. press. of a condensing engine, with initial steam 50 pounds gauge and back press. 4 pounds abso.

First Example, cut-off $\frac{1}{7}$ stroke, Fig. 28.

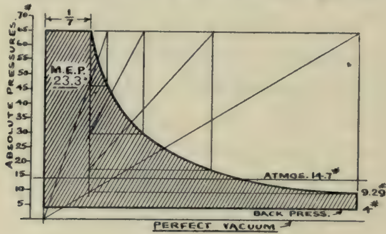


Fig 28.

1st. Find Forward Press. per sq. inch.

Rule.—"Initial press. abso. multiplied by Hyp. Log. of the number of expansions plus 1, and divide by the number of expansions."

thus, Hyp. Log. of 7 is 1.9459. Table No. 7.

Hyp. Log. + 1 = 2.9459.

Boiler press. given at 50 pounds gauge, the abso. press. is

$$50 + 15 = 65 \text{ pounds abso.}$$

$$\therefore \frac{65 \times 2.9459}{7} = 27.3 \text{ forward press.}$$

2nd. Find Mean Eff. Press.

Rule.—"Forward press. minus back press."

Back press. given as 4 pounds abso.

$\therefore$ 27.3 — 4 = 23.3 pounds M. Eff. Press.

Second Example, cut-off $\frac{1}{6}$. Same as first example except point of cut-off.

1st. Find Fd. Press. per sq. inch.

Hyp. Log. of 6 is 1.7918.

Hyp. Log. + 1 = 2.7918.

Boiler press. given as 50 gauge or 65 pounds abso.

$$\therefore \frac{65 \times 2.7918}{6} = 30.2 \text{ forward press.}$$

2nd. Find M. Eff. Press.

Back press. given at 4 pounds abso.

$\therefore$ $30.2 - 4 = 26.2$ M. Eff. Press.

Third Example, cut-off $\frac{1}{5}$. Same as first and second, except point of cut-off.

1st. Find Fd. P. per sq. in.

Hyp. Log. of 5 is 1.6094.

Hyp. Log. + 1 = 2.6094.

Boiler press. given as 50 pounds gauge or 65 pounds abso.

$$\therefore \quad \frac{65 \times 2.6094}{5} = 33.9 \text{ forward press.}$$

2nd. Find M. Eff. Press.

Back press. given as 4 pounds abso.

$\therefore$ $33.9 - 4 = 29.9$ M. Eff. Press.

§ 93. Non-Condensing Engine, Single Cylinder.

Example.—Required the diameter of a single cylinder necessary for a 175 horse-power, single cylinder, non-condensing engine, stroke 42″, with 80 pounds steam pressure gauge, to run at 80 rev. per minute, and exhaust against a back pressure of one pound per sq. in. above the atmosphere.

1st. Find Forward Press. per sq. inch.

Rule.—"Initial press. abso. multiplied by Hyp. Log. of the number of expansions plus 1, and divide by the number of expansions."

thus, Hyp. Log. of 5 is 1.6094.

Hyp. Log. +1 = 2.6094.

Boiler press. given 80 pounds gauge, therefore absolute pressure is

$$80 + 15 = 95 \text{ pounds.}$$

$$\therefore \quad \frac{95 \times 2.6094}{5} = 49.5 \text{ pounds.}$$

2nd. Find Mean Eff. Press.

Rule.—"Forward press. minus back press."

Back press. given as 1 pound above atmos., or 16 pounds abso.

$\therefore\ 49.5 - 16 = 33.5$ M. E. P.

3rd. Find Effective Area of piston.

Rule.—"Total foot-pounds, divided by the product of piston speed multiplied by Mean Eff. Press."

Piston speed given is $80 \times 3.5 \times 2 = 560$ feet,

and, foot-pounds = 175 horse-power × 33,000,

viz., $175 \times 33{,}000 = 577{,}500$ foot-pounds,

$$\therefore\ \frac{577{,}500}{560 \times 33.5} = 307 \text{ sq. inches effective area.}$$

4th. Find Actual Area of cylinder.

Rule.—"Effective area, plus half area of rod.

Assume area of rod as 14 inches.

$$307 + 7 = 314 \text{ sq. inches,}$$

and for diameter see Table No. 11.

314 inches area = 20 inches diameter.

NOTE! It is advisable for the student at this stage of the study to work out for himself five or six examples for various steam pressures and horse-power, all of which he can make for himself, but being guided by the two preceding examples.

CHAPTER No. 10.

DOUBLE EXPANSION NON-CONDENSING ENGINES.

§ 94. **Preliminary Questions.** In a general way there are four questions to consider before figuring the ratio of cylinder areas for non-condensing compound engines. 1st, the Back Pressure; 2nd, Terminal Pressure; 3rd, Total Expansions; 4th, Initial Pressure.

BACK PRESSURE. In a non-condensing engine, the exhaust may either be free to the atmosphere of say 15 pounds abso., or it may be subject to a higher back pressure due to frictional resistance caused by long exhaust pipe. Or in case of using the exhaust steam for heating or other purposes, the back pressure may be several pounds above atmosphere; this is the first thing to ascertain, as the proper terminal pressure should be established in relation to it.

TERMINAL PRESSURE IN L. P. CYLINDER. Having ascertained the back pressure against which the low pressure cylinder is to exhaust; then allow a terminal drop as explained § 84, which added to the back pressure establishes the low pressure terminal. For instance, if back press. is 5 pounds gauge and terminal drop 2 pounds, then the terminal pressure would be $15 + 5 + 2 = 22$ pounds abso.; or again, if the back pressure is ½ pound above atmosphere and the terminal drop 1½ pounds, then the terminal pressure is $15 + .5 + 1.5 = 17$ pounds abso.

TOTAL EXPANSIONS. For profitable investment from a commercial point of view, the non-condensing compound engine should never expand less than 9 times, and preferably 12, as a total; although in a slide valve engine, double expansion, even nine is not possible, as a slide valve cannot cut off earlier than half stroke, but we are assuming Corliss valve gear. Just how many expansions can be had depends upon two things; first, the terminal pressure, and next, the initial pressure. Having decided upon a terminal of say 17 abso. as above, and assuming an initial pressure of 200 abso., then the total number of expansions would be as per rule.

Rule.—"Initial pressure, divided by terminal press."

$$200 \div 17 = 11.7$$

Initial Pressure. But it may be desired to establish the initial pressure by the number of expansions preferred. Say for some reason the preferred total expansions are specified, and the back pressure also, then to figure the necessary initial press. use the following rule:

Rule.—"Terminal pressure, multiplied by number of expansions."

Example. For 9 expansions, and say 15.5 pounds terminal,

$$9 \times 15.5 = 139.5 \text{ pounds abso.,}$$

$$\text{or } 139.5 \text{ less } 15 = 124.5 \text{ gauge,}$$

allowing a little for loss, say 2.5 pounds, then boiler pressure would be

$$124.5 + 2.5 = 127 \text{ pounds gauge.}$$

Another example with say 3 pounds back press., 2 pounds terminal drop, and 12 expansions:

$$(15 + 3 + 2)\ 12 = 240 \text{ abso.,}$$

allow 3 pounds for loss, leaks, etc., then boiler pressure in engine room would be

$$240 + 3 - 15 = 228 \text{ pounds gauge.}$$

Note! The above rule may be used for any number of expansions, and any terminal pressure, but with a slight increase of allowance for loss, as the number of expansions increase.

Caution. The law of expansion is the same in a non-condensing engine as in condensing; therefore, always figure in absolute pounds, and not pounds according to gauge; for the reason that gas expands to absolute vacuum. Serious errors have been made by the oversight of this fact.

§ 95. **Throttling Governor in Non-Condensing Compounds.** To secure the best economy in non-condensing compound engines, the terminal pressure in the low press. clyinder should be kept as near constant as possible. But with a variable cut-off it is impossible to have a constant terminal as well as a constant initial pressure; therefore, one or the other must be variable. Of the two, the variable initial in the high press. cylinder is preferable to a variable terminal in the low, providing such can be done wthout wiredrawing. To accomplish this a throttling governor has, in some instances, been employed, by which the initial steam pressure can be raised or lowered as the load varies; the effect being that of cutting a slice off the top of the high press. diagram, as shown in Fig. 29, line M, or at any other level as the load may require.

With this arrangement instead of the usual method of varying the points of cut-off in both cylinders, a fixed cut-off is made in the high pressure cylinder and a variable cut-off in the low, but instead of cutting off shorter as the load is thrown off, the cut-off is later, which is the reverse of common practice; thus as the load decreases the admission period is lengthened. To follow the effect of a change in load, study the diagram.

Suppose that due to a change of load the governor throttled the steam down from 150 lbs. abso. to 120 lbs.; the cut-off being fixed at A, a slice would be taken off the diagram as at M, and the terminal pressure in the high pressure cylinder reduced, that is to say, the expansion curve would terminate at F instead of D. Then allowing a little drop, the receiver pressure would be at G, and where this line cuts the low press. expansion curve as at J, would be the best point of cut-off for the low, it being lengthened from A′ to B. By this means the combined

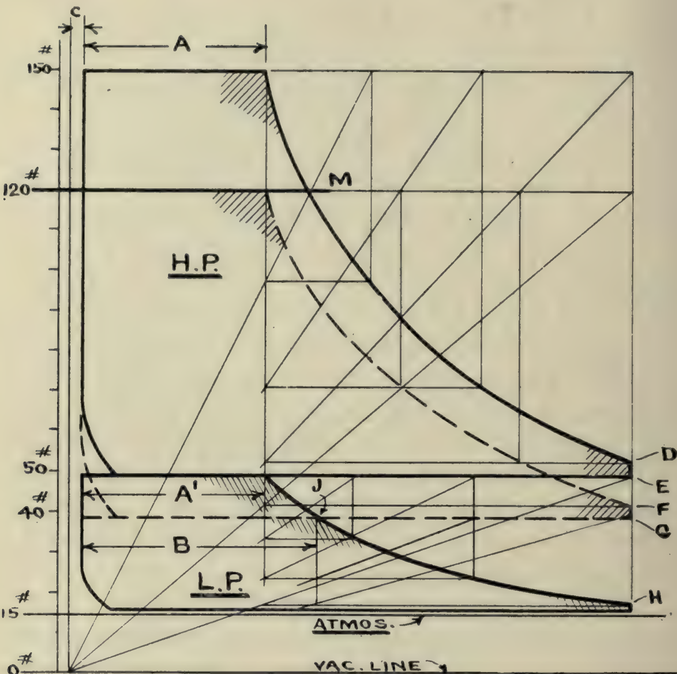


Fig. 29.

diagram would show a continuous expansion curve, and a constant low press. terminal at H.

THREE EXAMPLES WORKED OUT.

In figuring a non-condensing compound, there are two important things to consider, viz., the back pressure in the low pressure cylinder, and the necessity of terminal drop in the high. To make the matter clear three examples are necessary, the first two having no terminal drop but different back pressure, § 96 and § 97, and the third having a terminal drop and a known back pressure. § 98.

§ 96. Example, Without Terminal Drop. In order to have no terminal drop in the high press. cylinder, the initial pressure in the low must be equal to the terminal in the high. Of course this never can be exactly so in actual practice (§ 84) but these examples help to explain and lead up to those which follow.

Required, the diameters of cylinders for a 500 horse-power, double expansion, non-condensing engine, without terminal drop in high press. cylinder.

Assuming Steam press: 175 lbs. gauge, 190 abso.,
Back press. ½ lb. above atmos.,
Piston speed 480 feet per minute.

See diagram Fig. 30.

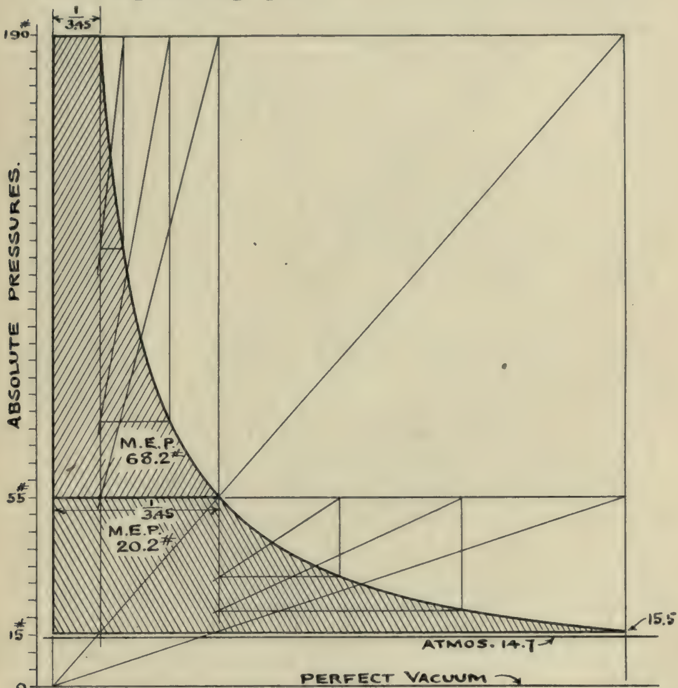


Fig. 30.

Deal with the high pressure first, then with the low, following step by step in the following order:

High Pressure.

1st. Find Total Number of Expansions.

Rule.—"Steam press .abso., divided by terminal press. in low press. cylinder."

$$175 + 15 = 190 \text{ pounds absolute.}$$

$190 \div 15.5 = 12.2$ minus loss $= 11.9$ expansions.

2nd. Find Number of Expansions in Each Cylinder.

Rule.—"Extract the square root of total expansion."

11.9 total expansions.

$$\sqrt{11.9} = 3.45 \text{ expansions in each cylinder.}$$

3rd. Find Forward Pressure per square inch.

Rule.—"Initial press., multiplied by Hyp. Log. of expansions + 1, and divide by number of expansions."

Hyp. Log. $3.45 = 1.238$.

Hyp. Log. $+ 1 = 2.238$.

$$\frac{190 \times 2.238}{3.45} = 123.2.$$

4th. Find Back Pressure per square inch. Back pressure should equal terminal pressure.

Rule.—"Initial press., divided by number of expansions."

$$190 \div 3.45 = 55 \text{ pounds.}$$

5th. Find Mean Effective Pressure per square inch.

Rule.—"Forward press., minus back press."

$$123.2 - 55 = 68.2 \text{ pounds.}$$

6th. Find Effective Area of Piston.

Rule.—"Foot-pounds, divided by the product of piston speed in feet per minute, multiplied by M. E. P."

NOTE.—The work in each cylinder should be equal, viz., 250 horse-power each:

$\therefore\ 250 \times 33{,}000 = 8{,}250{,}000$ foot-pounds.

Piston speed, by mean effective pressure, by effective area.

$$480 \times 68.2 \times \text{effective area.}$$

$$\therefore\ \frac{8{,}250{,}000}{480 \times 68.2} = 252 \text{ square inches.}$$

7th. Find Diameter of Cylinder. Table No. 11, or

Rule.—"Extract square root of effective area of piston, plus half area of rod, divided by .7854."

NOTE.—Assume half area of rod as eight square inches.

$$252 + 8 = 260 \text{ square inches area.}$$

$$\sqrt{\frac{260}{.7854}} = 18.2 \text{ diameter.}$$

Low Pressure.

8th. Find Initial Pressure in pounds absolute.

Rule.—"Terminal press. in High Press. Cylinder."

NOTE.—Assume that receiver is well protected. Initial pressure equals 55 pounds.

9th. Find Mean Forward Pressure.

Rule.—"Initial press., multiplied by Hyp. Log. of expansion plus 1, and divide by number of expansions."

$$\therefore \text{ Hyp. Log. of } 3.45 = 1.238.$$

$$\text{Hyp. Log.} + 1 = 2.238.$$

$$\frac{55 \times 2.238}{3.45} = 35.7 \text{ pounds per sq. inch.}$$

10th. Find Terminal and Back Pressure.

Rule.—"Initial press. divided by number of expansions, minus loss by condensation."

$$55 \div 3.45 = 15.9 \text{ less } .4 = 15.5 \text{ back pressure.}$$

11th. Find Mean Effective Pressure.

Rule.—"Forward press., minus back press."

$$35.7 - 15.5 = 20.2 \text{ pounds.}$$

12th. Find Effective Area of Piston.

Rule.—"Total foot-pounds, divided by the product of piston speed in feet per minute, multiplied by M. E. P."

NOTE! This cylinder also is to develop 250 H. P.; $250 \times 33{,}000 = 8{,}250{,}000$ foot-pounds, = Piston speed, by mean effective pressure by effective area, viz., $480 \times 20.2 \times$ effective area $= 8{,}250{,}000$ foot-pounds.

$$\therefore \frac{8{,}250{,}000}{480 \times 20.2} = 851 \text{ square inches.}$$

TABLE No. 11.

DIAMETERS AND AREAS.

DIA.	AREA.	DIA.	AREA.	DIA.	AREA.	DIA.	AREA.	DIA.	AREA.
1.	.7854	21.	346.36	41.	1320.2	61.	2922.4	81.	5153.
1.5	1.7671	21.5	363.05	41.5	1352.6	61.5	2970.5	81.5	5216.8
2.	3.1416	22.	380.13	42.	1385.4	62.	3019.	82.	5281.
2.5	4.9087	22.5	397.60	42.5	1418.6	62.5	3067.9	82.5	5345.6
3.	7.0686	23.	415.47	43.	1452.2	63.	3117.2	83.	5410.6
3.5	9.621	23.5	433.73	43.5	1486.7	63.5	3166.9	83.5	5476.
4.	12 566	24.	452.39	44.	1520.5	64.	3216.9	84.	5541.7
4.5	15.904	24.5	471 43	44.5	1555.2	64.5	3267.4	84.5	5607.9
5.	19.635	25.	490.87	45.	1590.4	65.	3318.3	85.	5674.5
5.5	23.758	25.5	510.70	45.5	1625 9	65.5	3369.5	85.5	5741.4
6.	28.274	26.	530.93	46.	1661.9	66.	3421.2	86.	5808.8
6.5	33.183	26.5	551.54	46.5	1698.2	66.5	3473.2	86.5	5876.5
7.	38.484	27.	572.55	47.	1734.9	67.	3525.6	87.	5944.6
7.5	44.178	27.5	593.95	47.5	1772.	67.5	3578.4	87.5	6013.2
8.	50.265	28.	615.75	48.	1809.5	68.	3631.6	88.	6082.1
8.5	56.745	28.5	637.94	48.5	1847.4	68.5	3685.2	88.5	6151.4
9.	63.617	29.	660.52	49.	1885.7	69.	3739.2	89.	6221.1
9.5	70.882	29.5	683.49	49.5	1924.4	69.5	3793.6	89.5	6291.2
10.	78.540	30.	706.86	50.	1963.5	70.	3848.4	90.	6361.7
10.5	86.590	30.5	730.61	50.5	2002.9	70.5	3903.6	90.5	6432.6
11.	95.033	31.	754.76	51.	2042.8	71.	3959.2	91.	6503.8
11.5	103.86	31.5	779 31	51.5	2083.0	71.5	4015.1	91.5	6575.5
12.	113.09	32.	804.24	52.	2123.7	72.	4071.5	92.	6647.6
12.5	122.71	32.5	829.57	52.5	2164.7	72 5	4128.2	92.5	6720.
13.	132.73	33.	855.30	53.	2206.1	73.	4185 3	93.	6792.9
13.5	143.13	33.5	881.41	53.5	2248.	73.5	4242.9	93.5	6866.1
14.	153.93	34.	907.92	54.	2290.2	74.	4300.8	94.	6939.7
14.5	165.13	34.5	934.82	54.5	2332.8	74.5	4359.1	94.5	7013.8
15.	176.71	35.	962.11	55.	2375.8	75.	4417.8	95.	7088.2
15.5	188.69	35.5	989.80	55.5	2419.2	75.5	4476.9	95.5	7163.
16.	201.06	36.	1017.8	56.	2463.	76.	4536.4	96.	7238.2
16.5	213.82	36.5	1046.3	56 5	2507.1	76.5	4596.3	96.5	7313.8
17.	226.98	37.	1075.2	57.	2551.7	77.	4656.6	97.	7389.8
17.5	240.52	37.5	1104.4	57.5	2596.7	77.5	4717.3	97.5	7466.2
18.	254.46	38.	1134.1	58.	2642.	78.	4778.3	98.	7542.9
18.5	268.80	38.5	1164.1	58.5	2687.8	78.5	4839.8	98.5	7620.1
19.	283.52	39.	1194.5	59.	2733.9	79.	4901.6	99.	7697.7
19.5	298,64	39.5	1225.4	59.5	2780.5	79.5	4963.9	99.5	7775.6
20.	314.16	40.	1256.6	60.	2827.4	80.	5026.5	100.	7854.
20.5	330.06	40.5	1288.2	60.5	2874.7	80.5	5089.5		

13th. Find Diameter of Cylinder.

Rule.—"Extract square root of effective area of piston plus half area of rod divided by .7854." See Table No. 11.

NOTE! Assume half area of rod as 12 square inches; $851 + 12 = 863$ square inches area.

$$\sqrt{\frac{863}{.7854}} = \sqrt{1{,}099} = 33.15 \text{ diameter.}$$

Collecting the results of the above calculation together, we have:

HIGH PRESSURE.	LOW PRESSURE.
Effective area of piston = 252 square inches.	Effective area of piston = 851 square inches.
Speed of piston = 480 feet per minute.	Speed of piston = 480 feet per minute.
Mean effective pressure per square inch = 68.2 pounds.	Mean effective pressure per square inch = 20.2 pounds.
$\frac{252 \times 480 \times 68.2}{33{,}000} = 250$ H. P.	$\frac{851 \times 480 \times 20.2}{33{,}000} = 250$ H. P.

§ **97. Example, Non-Condensing Compound, Exhausting Against 5 Pounds Back Pressure, Without Terminal Drop.** The preceding example assumes a back pressure of only ½ pound above atmosphere, other conditions may compel a higher B. P., hence the following example with 5 pounds B. P.

Example.—Determine the Diameter of Cylinders for a two-stage non-condensing compound engine, Terminal Press. ½ above Back Press.

Back Press. = 5 pounds above atmos.
Piston Speed = 480 feet per min.
Steam Press. = 175 pounds gauge.
Horse-Power = 500.

High Pressure Cylinder.

1st. Find Total Number of Expansions.

Rule.—"Steam pressure, abso., divided by terminal press. in low press. cylinder."

$175 + 15 = 190$ pounds abso.
$190 \div 20.5 = 9.2$ Expansions.

Allow a little for loss, making total exp. 9.

2nd. Find number of expansions in each cylinder.

Rule.—"Extract the square root of total exps."

$$\sqrt{9} = 3 \text{ Exps. in each cyl.}$$

3rd. Find Forward Pressure per sq. in.

Rule.—"Initial press. multiplied by Hyp. Log. of expansions + 1, divided by number of expansions."

Hyp. Log. of 3 is 1.0986

Hyp. Log. + 1 = 2.0986

$$\frac{190 \times 2.0986}{3} = 132.9$$

4th. Find Back Pressure per sq. inch. Assume back press. as equal to terminal, therefore find terminal.

Rule.—"Initial press., divided by number of exps."

$$190 \div 3 = 63.3 \text{ pounds B. P.}$$

5th. Find Mean Eff. Pressure per square in.

Rule.—"Forward press. minus back press."

$$132.9 - 63.3 = 69.6 \text{ pounds.}$$

6th. Find Eff. Area of H. P. Piston.

Rule.—"Foot-pounds, divided by the product of piston speed in feet per minute, multiplied by M. E. P."

Let the work in each cylinder be equal.

$$\therefore \quad 250 \times 33{,}000 = 8{,}250{,}000 \text{ Foot-pounds}$$

$$\frac{8{,}250{,}000}{480 \times 69.6} = 247 \text{ sq. in. Eff. Area}$$

7th. Find Diameter of H. P. Cylinder.

Rule.—"Extract square root of eff. area of piston plus half area of rod, divided by .7854."

Assume half area of rod as 8 sq. inch.

$$247 + 8 = 255 \text{ sq. inches actual area}$$

$$\sqrt{\frac{255}{.7854}} = 18 \text{ inches Dia.}$$

Low Pressure Cylinder.

8th. Find Initial Pressure in L. P. Cyl.

Rule.—"Terminal press. in H. P. Cyl."

$$= 63.3 \text{ pounds abso.}$$

9th. Find Mean Forward Pressure.

Rule.—"Initial press. multiplied by Hyp. Log. of expansions plus 1, and divide by number of expansions."

Hyp. Log. of 3 is 1.0986

Hyp. Log. + 1 = 2.0986

$$\frac{63.3 \times 2.0986}{3} = 44.28 \text{ Forward press.}$$

10th. Find Back Pressure per sq. inch.

Assume back press., as given, viz., 5 pounds gauge.

$$15 + 5 = 20 \text{ pounds abso.}$$

11th. Find Mean Eff. Pressure.

Rule.—"Forward press. minus back press."

$$44.2 - 20 = 24.2 \text{ pounds.}$$

12th. Find Eff. Area of Piston.

Rule.—"Foot-pounds, divided by the product of piston speed in feet per minute, multiplied by M. E. P."

$$\frac{8{,}250{,}000}{480 \times 24.2} = 710 \text{ sq. inches.}$$

13th. Find Diameter of Cylinder.

710 area + half rod of 12 inches.

710 + 12 = 722 Actual Area.

$$\sqrt{\frac{722}{.7854}} = 30.3'' \text{ Dia.}$$

NOTE! By comparing the last two examples it will be seen that with the same steam pressure, but different back pressures, there is only a small difference in cylinder areas, the principal difference being in the smaller total number of expansions, and consequently a proportional difference in economy. This shows the importance of increasing the boiler pressure in all cases where back pressure is increased. It will be remembered, also, that no provision was made for receiver drop; this was done for simplicity in those two examples, but the next § 98 takes terminal drop into account as is common in practice.

NON-CONDENSING ENGINES, FOR ELECTRIC GENERATORS.

When lack of water supply makes it necessary, the non-condensing compound can be profitably employed, providing sufficiently high pressure steam can be ensured. Boiler pressure of 200 pounds gauge is very desirable, but in case only 140 pounds can be had, it may be considered above the minimum pressure. Assuming then 140 pounds gauge, let us work out an example.

§ 98. Example of a Non-Condensing Compound, for 2000 K. W. Generator.

This example is to be figured on the basis of the following conditions:

Capacity, 2,000 K. W.

Boiler press., 140 pounds gauge.

Rev. per minute, 70.

Stroke, 60 inches.

Back press. in L. P. Cyl., 2 pounds above atms.

Load fairly uniform.

The example is worked out in fifteen successive steps, each numbered and in their proper order, as before.

Horse Power.

1st. Find Horse-Power equivalent to the K. W. specified, thus,

Since 746 Watts = 1 Horse-power (H. P.)
and 1,000 Watts = 1 Kilowatt (K. W.),

$$\text{therefore } 1 \text{ K. W.} = \frac{1{,}000}{746} = 1.34 \text{ H. P.}$$

and 2,000 K. W. = $1.34 \times 2{,}000 = 2{,}680$ H. P.
adding say, 10 per cent. for friction,
$2{,}680 + 268 = 2{,}948$ Horse-power.

Total Expansions.

2nd. Find Total Number of Expansions.

Rule.—"Steam press., abso., divided by terminal press. in L. P. Cyl."

Steam press. = 140 pounds gauge.

$140 + 15 = 155$ abso.

Back pressure specified equals 2 pounds above atms.

$2 + 15 = 17$ pounds abso.

Now allowing that the terminal in the L. P. Cylinder is one pound above back press., then terminal equals 18 pounds abso. and total expansions will be

$$155 \div 18 = 8.6.$$

High Pressure Cylinder.

3rd. Find Expansions in H. P. Cylinder.

Rule.—"Extract square root of total."

$$\sqrt{8.6} = 2.92.$$

NOTE! It will be remembered that where there is no drop in H. P. terminal, the expansions are the same in both high and low pressure cylinders. This is not the case in this example, as we have H. P. terminal drop, and consequently not the same number of expansions in each cylinder.

4th. Find Mean Forward Pressure.

Rule.—"Initial press. multiplied by Hyp. Log. of expansions plus 1, and divided by number of expansions."

Hyp. Log of 2.92 is 1.0716.

Hyp. Log. + 1 = 2.0716.

Initial pressure being 155 pounds. abso., we have

$$\frac{155 \times 2.0716}{2.92} = 110 \text{ pounds Fd. press.}$$

5th. Find Mean Back Pressure.

Assume back pressure as .5 lbs. below terminal, therefore find terminal and minus the drop.

Rule.—"Initial press., divided by number of exps."

$155 \div 2.92 = 53$ terminal.

53, less .5 for drop = 52.5 Back press.

6th. Find Mean Effective Press.

Rule.—"Forward press. minus Back press."

$110 - 52.5 = 57.5$ Mean Eff. Press.

7th. Find Eff. Area of Piston.

First find area, not including piston rod, and then add half area of rod, unless a tandem engine, then the whole area of rod.

Rule.—"Foot-pounds of work, divided by product of piston speed in feet per minute multiplied by M. E. P."

The foot-pounds of work done in the H. P. Cylinder, being half the total, is

$$1474 \times 33{,}000 = 48{,}642{,}000 \text{ Ft.-Pounds.}$$

$$\text{Piston Speed } 5 \times 2 \times 70 = 700 \text{ ft. per min.}$$

$$\text{M. E. P.} = 57.5.$$

$$\therefore \frac{48{,}642{,}000}{700 \times 57.5} = 1208.6 \text{ Eff. Area.}$$

The rod being about $7\frac{1}{4}''$ dia., and for, say, cross compound engines ½, the area is 20 sq. inches.

Therefore, actual area of cylinder equals

$$1208.6 + 20 = 1228.6 \text{ sq. in.}$$

8th. Find Diameter of Cylinder.

$$\sqrt{\frac{1228.6}{.7854}} = 39\tfrac{9}{16} \text{ say } 40'' \text{ Dia.}$$

Low Pressure Cylinder.

9th. Find Initial Pressure in L. P. Cyl.

Rule.—"Receiver Press., minus ½ pound."

The receiver pressure in this case is assumed as 1½ pounds below H. P. terminal.

$$\therefore\ 53 - (1.5 + .5) = 51 \text{ Initial abso.}$$

10th. Find Expansions in L. P. Cyl.

Rule.—"Initial press. divided by terminal in L. P. Cyl."

Allow terminal to be 1 pound above back press. and back pressure is specified as 2 pounds above atmosphere, therefore terminal must be

$$15 + 2 + 1 = 18 \text{ pounds absolute.}$$

Then initial being as above 51, we have

$$51 \div 18 = 2.75 \text{ Expansions.}$$

11th. Find Mean Forward Pressure.

Rule.—" Initial press. multiplied by Hyp. Log. of expansions plus 1, and divided by number of expansions."

Hyp. Log. of 2.75 is 1.0116.

Hyp. Log. + 1 = 2.0116.

$$\therefore \frac{51 \times 2.0116}{2.75} = 37.2$$

Allowing a little loss of volume for condensation in L. P. cylinder, the amount depending upon conditions, but in this case say ¾ of a pound, then 37.2 — .75 = 36.45 Forward press.

12th. Find Mean Effective Pressure.

Rule.—"Forward press. minus Back press."

Back pressure specified as 2 pounds above atmosphere, thus 15 + 2 = 17.

$$\therefore \quad 36.4 - 17 = 19.4 \text{ M. Eff. P.}$$

13th. Find Effective Area of Piston.

Rule.—"Foot-pounds of work done in L. P. Cyl. divided by the product of piston-speed in feet per minute multiplied by M. Eff. P."

Work done in L. P. Cyl.

$$1474 \times 33{,}000 = 48{,}642{,}000 \text{ Foot-Pounds}$$

and piston speed = 700 feet per minute.

$$\therefore \frac{48{,}642{,}000}{700 \times 19.4} = 3{,}600 \text{ sq. in. Eff. Area.}$$

Add 31 sq. inches for ½ area of rod,

$$3{,}600 + 31 = 3{,}631 \text{ sq. in. Eff. Area.}$$

14th. Find Diameter of L. P. Cylinder.

$$\sqrt{\frac{3631}{.7854}} = 68'' \text{ Dia.}$$

15th. Find Ratio of Cylinder Areas.

Rule.—"Divide Eff. Area of L. P. Cyl. by Eff. Area of H. P. Cyl."

$$\frac{3600}{1208.6} = \text{Ratio } 2.98 \text{ to } 1.$$

Collecting above results together, we have

High Pressure Cylinder 40″ × 60″

M. E. P. = 57.5. Eff. Area = 1206.6. Piston Speed 700.

$$\frac{57.5 \times 1206.6 \times 700}{33{,}000} = 1474 \text{ Horse Power.}$$

Low Pressure Cylinder 68″ × 60″

M. E. P. = 19.4. Eff. Area = 3600. Piston Speed 700 ft.

$$\frac{19.4 \times 3600 \times 700}{33{,}000} = 1474 \text{ Horse-Power.}$$

An engine agreeing with the above, in both sizes and conditions, is owned by the Cleveland Electric Railway Co. and has a consumption of 23½ pounds of dry steam per indicated horse-power per hour.

USE OF TABLE No. 12.

With the assistance of this table, the cyl. areas of any compound non-condensing engine may be quickly found.

NOTE.—Of course tables should be used only for approximate figuring, and modified for the various conditions. With this in view, the following formulae and examples Nos. 99 and 100 are given.

Formulae for Areas.

$$\text{High Press. Cyl. Area} = \frac{33{,}000 \times \text{Horse-Power}}{\text{M. Eff. Press.} \times \text{Piston Speed.}}$$

Low Press. Cyl. Area =

Area of H. P. Piston × Ratio of Areas.

§ **99. Example.** Find cylinder areas for a 200 H. P. non-condensing compound, having Steam press. 130 pounds gauge, Piston speed 400 ft. and 100 H. P. in each cylinder, by use of the table and formulae as above.

$$\text{High Press.} \begin{cases} \text{M. E. P. under 130 lbs. gauge} = 53.8 \\ \text{then } \dfrac{33{,}000 \times 100}{53.8 \times 400} = 153.3 \text{ sq. in. area.} \end{cases}$$

$$\text{Low Press.} \begin{cases} \text{Ratio of areas under 130 lbs.} = 2.9 \\ \text{then } 153.3 \times 2.9 = 444.6 \text{ sq. in. Area.} \end{cases}$$

Try another example of different pressure, using same table.

§ **100. Example.** Find cylinder areas of a 500 Horse-power non-condensing compound,

Steam press. 195 lbs. gauge. Piston speed 600 ft. Divide the load equally.

H. P. Cyl. $\begin{cases} \text{M. E. P. under 195 lbs. gauge} = 76.8 \\ \therefore \dfrac{33{,}000 \times 250}{76.8 \times 600} = 178 \text{ sq. in. area.} \end{cases}$

L. P. Cyl. $\begin{cases} \text{Ratio of areas, under 195 lbs. gauge} = 3.55 \\ \therefore \quad 178 \times 3.55 = 631 \text{ sq. in. area} \end{cases}$

NOTE! In the above two examples the areas found are effective areas. Piston rod areas must be added for actual areas of cylinders. To be strictly correct the work done in each end of the cylinder should be figured independently, on account of the effective area being different due to the piston rod, but this makes unnecessary work, therefore, if the rod is in one end of the cylinder only, figure as though one-half of the rod area was in each end.

TABLE No. 12.

DOUBLE EXPANSION, NON-CONDENSING.

BACK PRESSURE 15½ POUNDS ABSO.

With Terminal Drop.

Gauge Press		100	105	110	115	120	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200
Initial Press. Abso.		114.7	119.7	124.7	129.7	134.7	139.7	144.7	149.7	154.7	159.7	164.7	169.7	174.7	179.7	184.7	189.7	194.7	199.7	204.7	209.7	214.7
Expansions	Total	6.75	7.05	7.35	7.65	7.95	8 25	8 5	8.8	9.1	9.4	9.7	10.	10.3	10.6	10,85	11.15	11.45	11.75	12.1	12.35	12.7
	H. P. Cyl.	2.6	2.65	2.7	2.75	2.8	2.85	2.9	2.95	3.05	3.08	3.1	3.15	3.2	3.25	3.3	3.35	3.4	3,45	3.5	3.55	3.6
	L. P. "	2.45	2.49	2.54	2.58	2.63	2.67	2.72	2.76	2.82	2.85	2.9	2.94	2.98	3.02	3.07	3.12	3.16	3.20	3.25	3.30	3.34
Terminal Press. Abso.	H. P. Cyl.	44.	44.8	45.6	46.4	47.2	48.	48.8	49.6	50.4	51.2	52.	52 8	53.6	54.3	55.1	55.9	56.6	57.3	58.	58.7	59.4
	L. P. "	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17
Terminal Drop.	H. P. Cyl.	2.	2.	2.	2.1	2.1	2.1	2.2	2.2	2.2	2 3	2.3	2.3	2.3	2.4	2.4	2.4	2.4	2.5	2.5	2.5	2.5
	L. P. "	1.5	1.5	1.5	1.5	1 5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Mean Back Press. Abso.	H. P. Cyl.	42.5	43.3	44.1	44.8	45.6	46.4	47.1	47.9	48.5	49.4	50.2	51.	51.8	52.4	53.4	54.	54.7	55.3	55.8	56.7	57.4
	L. P. "	15.5	15.5	15.5	15.5	15.5	15.5	15 5	15.5	15 5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5
Mean Eff. Press	H. P. Cyl.	43.2	45.0	46.7	48.5	50.3	52.0	53.8	55.6	57.3	59.1	60.9	62.6	64.4	66.2	67.9	69.7	71.4	73.2	75.	76.8	78.6
	L. P. "	16.6	17 0	17.3	17.6	18.0	18 3	18.6	18.9	19.1	19.4	19.6	19.9	20.1	20 3	20.5	20.8	21.1	21.3	21.5	21.7	21.8
Receiver Press. Abso.		42	42.8	43.6	44.3	45.1	45.9	46.6	47.4	48.2	48.9	49.7	50.5	51.3	51.9	52.9	53.5	54.2	54.8	55.5	56.2	56.9
Ratio of Cyl. Areas.		2.6	2.65	2.7	2.75	2.8	2.85	2.9	2.95	3.05	3.08	3.1	3.15	3.2	3.25	3.3	3.35	3.4	3 45	3.5	3.55	3.6

CHAPTER No. 11.

DOUBLE EXPANSION CONDENSING ENGINES.

§ 101. **History.** The two-cylinder compound condensing engine dates back to the Hornblower patent of 1781, in which two cylinders were arranged side by side, the one shorter than the other, and both connected to one end of a walking beam.

This arrangement provided short connecting pipes but no receiver.

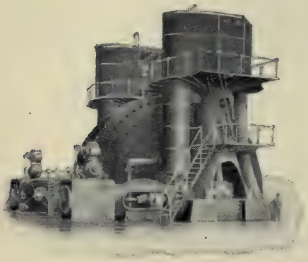
In 1838 E. A. Cowper built a two-cylinder condensing compound with a receiver, which was probably the first receiver engine.

§ 102. **Cross-Compound and Tandem Types.** By the name "Cross-Compound" is meant that there are two engines, one high pressure and the other low; each having its own connecting rod and crank, but the cranks arranged at opposite ends of the same main shaft and at 90 degrees to each other. The two-cylinders are connected by piping and receiver. The exhaust steam from the high pressure cylinder is discharged into the receiver where it is stored, or held in reserve, until the opening of the low pressure inlet valves. See also § 121.

In the "Tandem" the two cylinders are arranged, as the name suggests, one ahead of the other, with one piston rod passing through both cylinders. In this type the exhaust from the high passes directly to the low without waiting.

§ 103. **Two Low Press. Cylinders.** Where large units are required, it frequently happens that the low pressure cylinder figures beyond the capacity of the shop for handling the work. To meet this shop limitation, two low pressure cylinders, each of one-half the total area, may be employed; both cylinders being connected with one receiver, three cranks are employed, one in the center and one on each end of same shaft and keyed at 120 degrees to each other. This is a good arrangement for vertical engines, as the turning effect is thereby improved.

§ 104. **Horizontal Vertical.** Fig. 31 illustrates the horizontal vertical type. It has many advantages, especially in large units. 1st, the low pressure cylinder can be arranged vertically and thereby avoid the excessive friction due to the weight of so large a piston; 2nd, the cylinders being arranged one vertical and the other horizontal and both acting upon the same crank pin, gives four impulses for each revolution of the crank. The cut represents a pair of such engines with two cranks, the cranks being keyed to one shaft and at right angles to each other make eight impulses for each revolution, a still greater improvement in turning effect. This type of engine is particularly valuable in electric generators, with the armature between the engines, and makes a very compact and desirable arrangement.



Manhattan St. Ry., New York City, 10,000 Horse-Power.

Fig. 31.

§ 105. **Good Record.** An example of first-class economy can be found in the Warren Manfg. Co's. engine at Warren, R. I., which is as follows:

32″ and 68″ × 60″.
155 lbs. steam, gauge.
75 revolutions per min.
12.44 lbs. of steam per H. P. per hour.

ELECTRIC GENERATORS.

The double expansion condensing engine is the universal choice for electric generator work. Triple expansion, owing to the excessive load variation, will probably never have the same demand.

The steam pressures best adapted for this class of work may be considered within the range of from 155 pounds to 185 pounds gauge, and the weight of steam per horse-power per hour from 12½ pounds down to 12.

Two examples are sufficient to explain the method of figuring, and other similar cases though somewhat different in detail may be worked out in substantially the same way. The first of the two examples following, being for 155 pounds gauge, § 106, and the other for 185 pounds, § 107.

§ 106. Example, Double Expansion Condensing Engine, for 2,500 K. W. Generator, with Two Low Pressure Cylinders.

Capacity, 2,500 K. W.
Steam at Cyl., 155 pounds gauge.
Rev. per minute, 77.
Stroke of piston, 5 feet.
Vacuum, 27 inches.
Terminal in L. P. Cyl., 6.7 abso.
Terminal Drop in H. P. cyl., 2.7 pounds.

Figure the diameters of cylinders.

HORSE POWER.

As the capacity is specified in Kilowatts, first figure equivalent horse-power thus,

Since 746 Watts = 1 Horse-Power,
and 1,000 Watts = 1 Kilowatt,

therefore $1 \text{ K. W.} = \frac{1000}{746} = 1.34 \text{ H. P.}$

and $2{,}500 \text{ K. W.} = 1.34 \times 2{,}500 = 3{,}350 \text{ H. P.}$

Add 10 per cent. for friction, and

$\text{Total H. P.} = 3{,}350 + 335 = 3{,}685$ Horse-Power.

By developing one-half the work in the H. P. Cyl. and the other half in the two L. P. Cyls. we have

H. P. Cyl................	1,842	H.-Power.
1st L. P. Cyl............	921.5	"
2nd L. P. Cyl...........	921.5	"
Total	3,685	"

Total Expansion.

1st. Find Total Number of Expansions.

Rule.—"Initial press. abso., divided by terminal press. in L. P. Cylinders."

Initial press. $= 155 + 15 = 170$ abso.

Terminal press. specified $= 6.7$ abso.

therefore $170 \div 6.7 = 25.4$ Total Exps.

High Pressure Cylinder.

2nd. Find Number of Expansions in high pressure cylinder.

Rule.—"Extract the Sq.-root of Total Exps."

$\sqrt{25.4} = 5.05$ H. P. Exps.

3rd. Find Mean Forward Pressure.

Rule.—"Initial press., multiplied by Hyp. Log. of expansions plus 1, and divided by expansions in H. P. Cyl."

Hyp. Log. of $5.05 = 1.6194$.

Hyp. Log. $+ 1 = 2.6194$.

$\therefore \dfrac{170 \times 2.6194}{5.05} = 88.1$ Fd. Press.

4th. Find Mean Back Pressure.

Rule.—"Initial press. divided by number of expansions, less terminal drop."

$170 \div 5.05 = 33.6$ Terminal press.

assume H. P. terminal drop as 2.7 pounds,

then $33.6 - 2.7 = 30.9$ Mean Bk. Press.

5th. Find Mean Effective Pressure.

Rule.—"Forward press., minus back press."

$88.1 - 30.9 = 57.2$ M. E. P.

6th. Find Effective Area of Piston.

Rule.—"Foot-pounds, divided by product of piston speed in feet per min., multiplied by M. E. P."

Let the work done in the H. P. Cyl. be one-half of the total, viz., 1,842 H. P.

Expressed in foot-pounds is

$$1{,}842 \times 33{,}000 = 60{,}786{,}000.$$

Piston speed $= 5 \times 2 \times 77 = 770$ ft. per min.

and M. E. P. $= 57.2$ pounds per sq. inch.

$$\therefore \frac{60{,}786{,}000}{770 \times 57.2} = \overset{\text{Eff. Area.}}{1{,}380 \text{ sq. inches.}}$$

Low Pressure Cylinder.

Having two L. P. Cylinders specified, proceed as in other examples except to figure the horse-power for one L. P. Cylinder, and the other is a duplicate.

7th. Find Number of Expansions in L. P. Cyl.

Rule.—"Divide Receiver press. by Terminal press."

Allow that there is a reheater in receiver and that the pressure is equal to the back press. in the H. P. Cylinder. This may not always be the case, and where it is not, proper allowance can be made to suit.

Receiver press. $= 30.9$.

L. P. terminal $= 6.7$.

$$\therefore \quad 30.9 \div 6.7 = 4.6 \text{ Exps.}$$

8th. Find Mean Forward Pressure.

Rule.—"Receiver press., multiplied by Hyp. Log. of expansions + 1, and divided by number of expansions in L. P. Cyl."

Hyp. Log. of 4.6 is 1.5261.

Hyp. Log. $+ 1 = 2.5261$.

$$\therefore \frac{30.9 \times 2.5261}{4.6} = 17 \text{ pounds.}$$

9th. Find Mean Back Pressure.

Rule.—"Receiver press., divided by number of expansions, less terminal drop."

Assume terminal drop as 2.35 or about $\frac{1}{3}$ of terminal press.

$$\frac{30.9}{4.6} - 2.35 = 4.35 \text{ pounds.}$$

10th. Find Mean Effective Pressure.

Rule.—"Forward press., minus Back press."

$$17 - 4.35 = 12.65 \text{ pounds.}$$

But considering that steam is taken from a limited receiver volume, the steam line will fall before cutting off, which together with compression and rounded corners the actual M. E. P. will be about 10 per cent. less than figured, therefore deduct for actual

$$12.65 \text{ less } 1.26 = 11.39 \text{ M. E. P.}$$

11th. Find Effective Area of Pistons.

Rule.—"Foot-pounds of work done in one L. P. Cyl., divided by product of piston speed multiplied by M. E. P."

Foot-pounds $= 921 \times 33{,}000 = 30{,}393{,}000.$

Piston speed $= 5 \times 2 \times 77 = 770$ feet per min.

M. E. Press. $= 11.39$ pounds.

$$\therefore \frac{30{,}393{,}000}{770 \times 11.39} = \overset{\text{Eff. Area.}}{3{,}464 \text{ sq. in.}}$$

Combined area of the two L. P. Cylinders is

$$3{,}464 \times 2 = 6{,}928 \text{ sq. inches.}$$

12th. Find Ratio of Cylinder Areas.

Rule.—"Divide combined area of L. P. Cyls. by area of H. P. Cylinder."

$$\frac{3464 \times 2}{1380} = 5 \text{ to } 1. \text{ Ratio.}$$

§ 107. Example, Pair of Vertical Tandem, Double Expansion Condensing Engines, Electric Generator.

Capacity—1,000 K. W.

Steam pressure—185 pounds gauge.

Rev. per minute—95.

Stroke of Piston—4 feet.

Vacuum—27 inches.

Terminal in L. P. Cyl.—7 pounds abso.

Terminal drop in H. P. Cyl. = 3 pounds.

Figure the areas of cylinders.

NOTE! Being a pair of engines with one common shaft and one generator, each engine should develop equal work, viz., 500 K. W. each.

Horse Power.

As the capacity is specified in Kilowatts, first figure the equivalent horse-power.

Since 746 Watts = 1 horse-power,

and 1,000 Watts = 1 Kilowatt,

therefore, 1 K. W. = $\frac{1000}{746}$ = 1.34 H. P.

and 500 K. W. = $1.34 \times 500 = 670$ H. P.

Being tandem, it is not quite so important to develop the same horse-power in high as in the low press. cylinders, but we will do so in this example, which would be

$$670 \div 2 = 335 \text{ H. Power.}$$

Total Expansions.

1st. Find Total Number of Expansions.

Initial press. = $185 + 15 = 200$ lbs. abso.

Terminal press. specified 7 lbs. abso.

$$200 \div 7 = 28.5$$

High Pressure Cylinder.

2nd. Find Number of Expansions in H. P. Cyl.

$$\sqrt{28.5} = 5.35$$

3rd. Find Mean Forward Pressure.

$$1 + \text{Hyp. Log. of } 5.35 = 2.6771$$

$$\frac{200 \times 2.6771}{5.35} = 100$$

4th. Find Mean Back Pressure.

$$\frac{200}{5.35} - 3 = 34.4$$

5th. Find Mean Effective Pressure.

$$100 - 34.4 = 65.6$$

6th. Find Effective Area of Piston.

Ft.-pounds $= 335 \times 33{,}000 = 11{,}055{,}000$

Piston speed $= 95 \times 4 \times 2 = 760$ ft. per min.

$$\therefore \frac{11{,}055{,}000}{760 \times 65.6} = 222 \text{ sq. inches}$$

Including area of rod $= 17''$ dia.

Low Pressure Cylinder.

7th. Find Number of Expansions in L. P. Cylinder.

Receiver press. equals terminal minus H. P. drop.

$$37.4 - 3 = 34.4$$

Terminal press. equals 7 lbs. abso. specified.

$$\therefore \frac{34.4}{7} = 4.9$$

8th. Find Mean Forward Pressure.

$$1 + \text{Hyp. Log. of } 4.9 = 2.5892$$

$$\therefore \frac{34.4 \times 2.5892}{4.9} = 18$$

9th. Find Mean Back Pressure.

Assume L. P. terminal drop $= 2.4$ lbs.

$$\frac{34.4}{4.9} - 2.4 = 4.6 \text{ lbs. abso.}$$

10th. Find Mean Effective Pressure.

$$18 - 4.6 = 13.4$$

Allow 8 per cent. for loss by condensation.

$$13.4 - 1.15 = 12.25$$

11th. Find Effective Area of L. P. Piston.

Ft.-pounds $= 335 \times 33{,}000 = 11{,}055{,}000$

$$\frac{11{,}055{,}000}{760 \times 12.25} = 1187 \text{ sq. inches}$$

Including area of rod $= 40''$ dia.

12th. Find Ratio of Cylinder Areas.

$$\frac{1187}{222} = 5.35 \text{ to } 1.$$

An engine corresponding with above example is owned by The Calumet & Hecla Mining Co.

DOUBLE EXPANSION PUMPING ENGINES.

§ 108. **Limited Steam Pressure.** As far as it is possible, pumping engines should be of the triple expansion type, but there is one condition which makes such a type commercially unprofitable, and that is too limited a boiler pressure. Triple expansion should never be attempted with less than 135 pounds boiler pressure, and when the maximum is below that, but not less than 90 gauge, double expansion is more profitable. Now let us suppose a case where the pressure is 90 pounds gauge, and a pumping engine of 250 to 300 horse-power is required. There are two ways of figuring such a proposition; 1st, to figure for the mean or average load, viz., 275 H. P., and then let the regulator cut off longer or shorter for load variation. 2nd, to figure for the minimum load of 250 H. P. with equal expansion in each cylinder, but allow no terminal drop in the high press. cylinder; and then for the maximum load increase the length of cut off in each cylinder. The second method is the more simple of the two, but not the more desirable, but for the sake of the practice one example will be worked out by this method (§ 110), and afterward another example by the first method (§ 111).

It is important in § 110 to know positively that the minimum load will not drop below that for which it is figured, for the reason that if no terminal drop is allowed as suggested, and then the load should fall below that specified, the expansion line would fall below the back pressure line and form an objectionable loop. § 80.

§ 109. **Pump Horse-Power Factor.** For convenience in figuring several examples, the following factor is found useful:

One U. S. gallon weighs 8.34 pounds, or 1 million gallons 8,340,000 pounds. Therefore, for 1 million gallons pumped one foot high in 24 hours, the work done is equal to 8,340,000 foot pounds. Now 1 horse power equals 33,000 foot pounds per minute, or in 24 hours $33{,}000 \times 60 \times 24 = 47{,}520{,}000$ ft. lbs. Therefore the horse power required to pump 1 million gallons one foot high in 24 hours is

$$\frac{8{,}340{,}000}{47{,}520{,}000} = .1755$$

§ 110. Example, 250 Horse Power Double Expansion Condensing Engine.

Minimum capacity, 250 Horse Power.
Steam Press. in Engine Room, 90 pounds gauge.
Piston Speed, 240 feet per min.
Vacuum, 27 inches.
Without Terminal drop in H. P. Cylinder.

1st. Find Total Number of Expansions.

Rule.—"Steam press. abso., divided by terminal in L. P. Cylinder."

Initial press. $= 90 + 15 = 105$ abso.
Terminal, assumed $= 6.5$ abso.

$\therefore \quad 105 \div 6.5 = 16$ Expansions.

2nd. Find Number of Expansions in each cylinder.

Rule.—" Extract square root of total expansions."

$$\sqrt{16} = 4 \text{ Exps. in each.}$$

High Pressure Cylinder.

3rd. Find Forward Pressure.

Rule.—"Initial press., multiplied by Hyp. Log. of expansions + 1, and divided by number of expansions."

Hyp. Log. of $4 = 1.3863$
Hyp. Log. $+ 1 = 2.3863$

If steam is not superheated allow for loss by condensation, of say, 1½ pounds in the initial pressure, thus,

$$90 + 15 - 1.5 = 103.5 \text{ abso.}$$

Then for forward pressure we have

$$\frac{103.5 \times 2.3863}{4} = 61.6 \text{ pounds.}$$

4th. Find Back Pressure per sq. inch.

NOTE! Back press. is to equal terminal, therefore find terminal.

Rule.—"Initial press., divided by number of expansions."

$$103.5 \div 4 = 25.8 \text{ pounds.}$$

5th. Find Mean Eff. Pressure per sq. inch.

Rule.—"Forward press., minus Back press."

$$61.6 - 25.8 = 35.8 \text{ Eff. Press.}$$

6th. Find Eff. Area of Piston.

Rule.—"Foot-pounds, divided by the product of piston speed in feet per minute, multiplied by M. E. P."

NOTE! Let the work be equal in each cyl., viz., 125 horse-power in each.

$$\therefore \quad 125 \times 33{,}000 = 4{,}125{,}000 \text{ Foot-pounds}$$

Piston speed multiplied by Mean Eff. Press., multiplied by Eff. Area,

$$\text{viz.,} \qquad 240 \times 35.8 \times \text{Eff. Area};$$

therefore, effective area will be

$$\frac{4{,}125{,}000}{240 \times 35.8} = 480 \text{ sq. inches.}$$

Plus half area of piston rod, say 14 sq. inches,

$$480 + 14 = 494.$$

7th. Find Diameter of Cylinder.

$$\sqrt{\frac{494}{.7854}} = 25.1$$

Low Pressure Cylinder.

Assume initial pressure in L. P. Cyl. as equal to terminal in low, viz., 25.8, but allow that loss by condensation is compensated for by reheater in receiver.

8th. Find Mean Forward Pressure.

Rule.—"Initial press., multiplied by Hyp. Log. of expansion plus 1, and divided by number of expansions."

$$\text{Hyp. Log. of } 4 = 1.3863$$

$$\text{Hyp. Log.} + 1 = 2.3863$$

$$\therefore \frac{25.8 \times 2.3863}{4} = 15.3 \text{ pounds per inch.}$$

9th. Find Terminal and Back Pressure.

Rule.—"Initial press., divided by number of expansions, multiplied by .66."

$$\frac{25.8}{4} \times .66 = 4.26, \text{ say } 4.3 \text{ abso.}$$

10th. Find Mean Eff. Pressure.

Rule.—"Forward press., minus Back press."

$$15.3 - 4.3 = 11 \text{ pounds.}$$

11th. Find Eff. Area of Piston.

Rule.—"Foot-pounds, divided by the products of piston speed in feet per min., multiplied by M. E. P."

$$125 \times 33{,}000 = 4{,}125{,}000 \text{ Ft.-pounds.}$$

Piston speed multiplied by M. E. P., multiplied by Eff. Area should equal foot-pounds, thus:

$$240 \times 11 \times \text{Eff. Area} = 4{,}125{,}000.$$

$$\therefore \frac{4{,}125{,}000}{240 \times 11} = 1562 \text{ sq. inches.}$$

12th. Find Dia. of Cylinder.

Rule.—"Extract sq. root of effective area of piston, plus half area of rod, divided by .7854."

Assume half area of rod as 18 sq. inches.

$$1562 + 18 = 1580 \text{ sq. inches area.}$$

$$\sqrt{\frac{1580}{.7854}} = 44.8'' \text{ diameter.}$$

§ 111. Example, of a Twenty Million Gallon, Double Expansion Pumping Engine.

Find areas of cylinders of a pumping engine of the above capacity and under following conditions:

20 million gallons in 24 hours.
Av. head of water pumped against 100 feet.
Steam pressure at cylinder 135 pounds gauge.
Stroke of piston and plunger 60 inches.
Revolutions per minute 20.

Vacuum 27 inches.

Terminal in L. P. Cylinder 6.5 pounds abso.

Horse Power.

1st. Find Total Horse-Power of Engine.

Rule.—"Number of Million Gallons, multiplied by number of Feet head, multiplied by .1755" § 109.

$$20 \times 100 \times .1755 = 350;$$

add 8 per cent. for friction, viz., 28.

Total, $350 \times 28 = 378$ Horse-power.

Horse-power in each of the two cyls. $\frac{378}{2} = 189$ each.

Total Expansions.

2nd. Find Total Number of Expansions.

Rule.—"Steam press. abso., divided by terminal in L. P. Cyl."

Initial press. $= 135 + 15 = 150$ pounds abso.

Terminal in L. P. Cyl. $= 6.5$ pounds abso.

$\therefore\ 150 \div 6.5 = 23.1$, total expansions.

NOTE! The method of figuring this example where terminal drop in the high pressure cylinder is allowed, requires that the number of expansions should be less in L. P. Cyl. than in the H. P. Cyl., which makes it necessary that the expansions of each cylinder should be figured independently of each other. In this respect this method differs from that used in § 110.

High Pressure Cylinder.

3rd. Find Number of Expansions in H. P. Cyl.

Rule.—"Extract the sq. root of Total Exps."

$$\sqrt{23.1} = 4.8$$

4th. Find Mean Forward Pressure.

Rule.—"Initial press., multiplied by Hyp. Log. of expansions + 1, and divided by number of expansions in H. P. Cyl."

Hyp. Log. of $4.8 = 1.5686$

Hyp. Log. $+ 1 = 2.5686$

$$\frac{150 \times 2.5686}{4.8} = 80.2 \text{ lbs. per sq. in.}$$

5th. Find Back Pressure.

NOTE! Back pressure is assumed as 2½ pounds below terminal, therefore

Rule.—"Initial press., divided by number of expansions, less terminal drop."

$$\frac{150}{4.8} - 2.5 = 28.7 \text{ Back press.}$$

6th. Find Mean Effective Pressure.

Rule.—"Forward press., minus Back press."

$$80.2 - 28.7 = 51.5 \text{ M. E. P.}$$

7th. Find Effective Area of Piston.

Rule.—"Foot-pounds, divided by the product of piston speed in feet per min., multiplied by M. E. Press."

First find foot-pounds of work to be done in H. P. Cyl. which is to develop 189 horse-power, thus,

$$189 \times 33{,}000 = 6{,}237{,}000 \text{ Ft.-pounds.}$$

Next look back for piston speed and M. E. P., and then form equation.

$$\frac{6{,}237{,}000}{200 \times 51.5} = \text{Eff. Area} = 605.5 \text{ sq. inches.}$$

For actual area of cylinder add for piston rod.

Low Pressure Cylinder.

8th. Find Number of Expansions in L. P. Cyl.

Rule.—"Divide receiver press. by terminal press."

NOTE! If there is no reheater in the receiver then allowance should be made for loss; in this example assume that receiver steam is reheated, and that its pressure is ½ pound less than back press. in H. P. cylinder, viz., $28.7 - .5 = 28.2$, then $28.2 \div 6.4 = 4.4$ Expansions.

9th. Find Mean Forward Pressure.

Rule.—"Receiver press. multiplied by Hyp. Log. of

Expansions + 1, and divided by number of expansions in L. P. Cyl."

Hyp. Log. of 4.4 is 1.4816.

Hyp. Log. + 1 = 2.4816.

$$\frac{28.2 \times 2.4816}{4.4} = 15.9 \text{ Forward Press.}$$

10th. Find Mean Back Pressure.

NOTE! Back pressure is assumed as being about 2.25 pounds less than terminal; then find terminal and minus terminal drop.

Rule.—"Receiver press., divided by number of expansions, less terminal drop."

$$\frac{28.2}{4.4} - 2.25 = 4.2 \text{ Back press.}$$

11th. Find Mean Effective Pressure.

Rule.—"Forward Press., minus Back Press."

$$15.9 - 4.2 = 11.7$$

But considering that the steam is taken from a limited volume in the receiver, there is a fall in the steam line before cutting off, which together with compression and other rounded corners makes it impossible to get in actual practice as high a M. E. P. as figured and the difference depends on conditions. In this example allow 1 pound less than figures, making the actual M. E. P. equal to

$$11.7 - 1 = 10.7 \text{ M. E. P.}$$

12th. Find Effective Area of Piston.

Rule.—"Foot-pounds, divided by product of piston speed in feet per minute, multiplied by M. E. P."

$$\frac{6{,}237{,}000}{200 \times 10.7} = 2914 \text{ sq. inches.}$$

For actual cyl. area, add area of rod.

13th. Find Ratio of Cylinder Areas.

Rule.—"Divide area of L. P. Cyl. by area of H. P. Cyl."

$$\frac{2914}{605.5} = 4.8 \text{ to 1. Ratio.}$$

NOTE! For a rough and ready rule, the Ratio of cylinder areas may be found by extracting the square root of total expansion; for instance, in last example,

$$\sqrt{23.1} = 4.8$$

§ **112. Use of Table No. 13.** Page 139.

By use of Table No. 13 the areas of cylinders for any given horse-power can readily be obtained. *Example.*—Find area of each cylinder for 500 H. P. Engine. Piston speed 500 feet per minute, steam 170 pounds abso.

Having the steam pressure, the piston speed, and the horse-power for each cylinder, use the following formula:

$$\text{High Press.} \left\{ \text{Area} = \frac{33{,}000 \times \text{Horse-power}}{\text{M. E. P.} \times \text{Piston Speed.}} \right.$$

$$\text{Low Press.} \left\{ \text{Area} = \text{H. P. Area} \times \text{Ratio of Areas.} \right.$$

NOTE.—The first thing to do is to find from the table the M. E. P. corresponding with the given initial pressure, and then proceed with the example as follows:

$$\text{High Press.} \left\{ \begin{array}{l} \text{M. E. P. for } 170 = 57.3 \\ \dfrac{33{,}000 \times 250}{57.3 \times 500} = 288 \text{ sq. in.} \end{array} \right.$$

$$\text{Low Press.} \left\{ \begin{array}{l} \text{Ratio of areas for } 170 = 5.05 \\ 288 \times 5.05 = 1454 \text{ sq. in.} \end{array} \right.$$

Try another example for 800 H. P. Engine, viz., 400 H. P. in each cylinder, Piston speed 600 feet per min., Steam 150 lbs. abso.

$$\text{High Press.} \left\{ \begin{array}{l} \text{M. E. P. for } 150 \text{ is } 51.5 \\ \dfrac{33{,}000 \times 400}{51.5 \times 600} = 427 \text{ sq. in.} \end{array} \right.$$

$$\text{Low Press.} \left\{ \begin{array}{l} \text{Ratio of Areas for same is } 4.8 \\ \therefore\ 427 \times 4.8 = 2050 \text{ sq. in.} \end{array} \right.$$

TABLE No. 13.

COMPOUND CONDENSING
Terminal Drop in Both Cylinders.

ABSOLUTE PRESSURES IN POUNDS	RATIO OF CYL. AREAS	EXPANSIONS			TERMINAL PRESS. ABSO.		TERMINAL DROP		MEAN BACK PRESS. ABSO.		MEAN EFFECTIVE PRESS.		RECEIVER PRESS. ABSO.
		Total	H.P.	L. P.	H.P.	L. P.	H.P.	L. P.	H.P.	L. P.	H.P.	L. P.	
100	4.05	16.6	4.05	3.8	24.7	6.0	2.0	2.0	22.7	4.0	36.5	9.0	22.7
105	4.15	17.35	4.15	3.85	25 25	6.05	2.05	2.0	23.2	4.0	38.0	9.15	23.2
110	4.25	18.	4.25	3.9	25.9	6.1	2.1	2.05	23.8	4.05	39.5	9.3	23.8
115	4.35	18.7	4.35	3.95	26.45	6.15	2.15	2.05	24.3	4.05	41.0	9.45	24.3
120	4.4	19.35	4.4	4.0	27.3	6.2	2.2	2.1	25.1	4.1	42.5	9.6	25.1
125	4.45	20.	4.45	4.1	28.05	6.25	2.25	2.1	25.8	4.1	43.8	9.85	25.8
130	4.55	20.65	4.55	4.2	28.6	6.3	2.3	2.15	26.3	4.15	45.8	10.0	26.3
135	4.6	21.25	4.6	4.25	29.35	6.35	2.35	2.15	27.0	4.15	47.2	10.25	27.0
140	4.65	21.85	4.65	4.3	30.0	6.4	2.4	2.2	27.6	4.2	48.2	10.35	27.6
145	4.75	22.5	4.75	4.35	30.55	6.45	2.45	2.2	28.1	4.2	49.8	10.5	28.1
150	4.8	23.1	4.8	4.4	31.2	6.5	2.5	2.25	28.7	4.25	51.5	10.7	28.7
155	4.85	23.65	4.85	4 5	31.95	6.55	2.55	2.25	29.4	4.25	52.8	10.9	29.4
160	4.9	24.25	4.9	4.55	32.7	6.6	2.6	2.3	30.1	4.3	54.4	11.1	30.1
165	4.95	24.8	4.95	4.6	33.25	6.65	2.65	2.3	30.6	4.3	55.8	11.25	30.6
170	5.05	25.4	5.05	4.6	33.6	6.7	2.7	2.35	30.9	4.35	57.3	11.39	30.9
175	5.1	26.	5.1	4.65	34.35	6.75	2.75	2.35	31.6	4.35	58.6	11.5	31.6
180	5.15	26.5	5.15	4.7	35.0	6.8	2.8	2.4	32.2	4.4	60.0	11.65	32.2
185	5.2	27.	5 2	4 75	35.55	6.85	2.85	2.4	32.7	4.4	61.5	11.8	32.7
190	5.25	27.5	5.25	4.8	36.2	6.9	2.9	2.45	33.3	4.45	62.9	11.95	33.3
195	5.3	28.	5.3	4.85	36.75	6.95	2.95	2.45	33.8	4.45	64.3	12.1	32.8
200	5.35	28.5	5.35	4.9	37.4	7.0	3.	2.5	34.4	4.5	65.6	12.25	34.4

For examples of the use of this table see page 138.

CHAPTER No. 12.

TRIPLE EXPANSION ENGINES.

Where local conditions are favorable, such as the necessary high pressure steam, ample water for condenser, and a comparatively uniform load such as found especially in high class pumping machinery, the most economical type of engine commercially, at the present time, is the Vertical Triple Expansion Condensing.

§ 113. **World's Record.** The highest record up to date for triple expansion is found in the 30-million gall. pumping engine of Chestnut Hill, Boston, Mass.

Report of test made in the year 1896 shows:

Dia. of cyls..........H. P. = 30″, I. P. 56″, L. P. 87″.
Stroke of engine..............................66″.
Av. steam press. at engine gauge.........185.5 pounds.
Vacuum .... — 13.8 pounds.
Av. head of water pumped against..........140.35 feet.
Av. rev. per minute..............................17.74
Water delivered in 24 hours..........30,313,911 galls.
Indicated horse-power.........................801.57.
Friction ....6.71 per cent.
Dry Steam for I. H. P. per hour.........10.335 pounds.
Duty per 1,000 pounds dry steam.....178,497,000 ft.-lbs.
Duty per million B. T. Us..................163,925,300.
Thermal Efficiency......................21.63 per cent.

§ 114. **Example, Partly Worked Out, of a 20-Million Gallon Pumping Engine, Triple Expansion.**

The figures of this example are based on the actual engine running at the pumping station of the Indianapolis Water Company.

The sizes of cylinders and data of test made October 15, 1898, are as follows:

Dia. of H. P. Cyl. . . . . 29″.
Dia. of Intermediate Cyl . . . . 52″.
Dia. of L. P. Cyl. . . . . 80″.
Stroke of Piston. . . . . 60″.
Rev. per min. . . . . 21.46.
Av. steam press. in pounds gauge. . . . . 155.6.
Total horse-power of engine. . . . . 774.
Dry steam per Ind. H. P. per hour. . . . . 11.26 lbs.
Duty on basis 1,000 lbs. dry steam. . . . . 167,800,000.
Duty on basis one million B. T. U. . . . 150,100,000 ft.-lbs.
Capacity in galls. per 24 hours. . . . . 20,600,000.
Friction of engine. . . . . 4.6 per cent.

Example.—Find cyl. dias. of a 774 H. P. triple expansion pumping engine, with REHEATER in the second receiver only. Piston speed 215 ft., St. Gauge 155 pounds.

$155 + 15 = 170$ lbs. abso., Loss equiv. 5 lbs., Term. in Low 6.1.

$165 \div 6.1 = 27$ Total expansions. $\sqrt[3]{27} = 3$ Ex. in each.

Treat each cylinder in its order.

H. Prs. 258 H.P.

Initial pres. 168 lbs., $\dfrac{168 \times 2.0986}{3} = 117.5$ Forw'd P.

$168 \div 3 = 56$ Back pres., and $117.5 - 56 = 61.5$ M.E.P.

$$\therefore \quad \frac{33{,}000 \times 258}{61.5 \times 215} = 643 \text{ sq. in. area.}$$

Add area of piston rod, and High pres. cyl $= 29''$ dia.

Int. Cyl. 258 H. P.

No REHEATER in first receiver, therefore loss is ab't 3 lbs.

Initial pres. 53 lbs., $\dfrac{53 \times 2.0986}{3} = 37$ Forw'd P.

$53 \div 3 = 17.7$ Back pres., and $37 - 17.7 = 19.3$ M.E.P.

$$\therefore \quad \frac{33{,}000 \times 258}{19.3 \times 215} = 2052 \text{ sq. in. area.}$$

Add area of piston rod, and Intermediate cyl. $= 52''$ dia.

Low Pres. 258 H. P.

REHEATER in second receiver, therefore loss is only ½ lb.

Initial pres. 17.2 lbs., $\frac{17.2 \times 2.0986}{3} = 12$ lbs. Fow'd P.

Assume back pres. 4 lbs., and $12 - 4 = 8$ lbs. M.E.P.

$$\therefore \quad \frac{33{,}000 \times 258}{8 \times 215} = 4950 \text{ sq. in. area.}$$

Add area of piston rod, and Low pres. cyl. = 80″ dia.

NOTE! These figures are not carried out as fine as some other examples; for instance, the expansions are taken as being equal in each cylinder. This is never strictly so, but assumed in this case for simplicity.

§ **115. Example, of a 30-Million Gallon, Triple Expansion, Pumping Engine.**

Compute the cylinder diameters of engine under the following conditions (with Fig. 32):

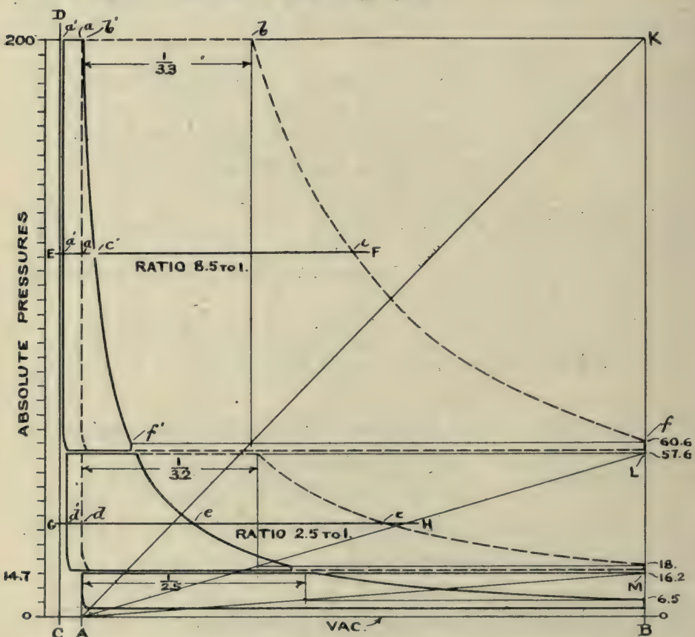


Fig. 32.

Capacity....................30 mill. gall. per 24 hours.
Head of water pumped against....................150 ft.
Steam press. at cylinder................185 lbs. gauge.
Stroke of piston................................5′ 6″.
Rev. per min.....................................17.
Vacuum ....28 inches.

Terminal press. in L. P. Cyl., 5.5 lbs. abso.

Horse Power.

1st. Find Horse-Power of Engine.

Rule.—"Number of million gallons, multiplied by number of feet head, multiplied by .1755." § 109.

$$30 \times 150 \times .1755 = 790$$

add 10 per cent. for friction.

$$790 + 79 = 869 \text{ Total H. P.}$$

Horse-power for each cylinder should be as near equal as possible, as it is assumed that the three water plungers are of equal diameters and strokes, therefore

$$869 \div 3 = 289.7, \text{ say } 290 \text{ H. P. each.}$$

Total Expansions.

2nd. Find Total Expansions.

Rule.—"Initial press. abso., divided by terminal press. in L. P. Cyl."

Initial press. specified is $185 + 15 = 200$ lbs. abso.

Terminal press. in L. P. Cyl., 5.5 abso.

$$\therefore \quad 200 \div 5.5 = 36.3, \text{ Total Exps.}$$

High Pressure Cylinder.

3rd. Find Expansions in H. P. Cylinder.

Rule.—"Extract Cube-root of Total."

$$\sqrt[3]{36.3} = 3.3$$

4th. Find Mean Forward Pressure.

Rule.—"Initial press., multiplied by Hyp. Log. of expansions plus 1, and divided by number of expansions in H. P. Cyl."

$$1 + \text{Hyp. Log. of } 3.3 \text{ is } 2.1939$$

$$\therefore \frac{200 \times 2.1939}{3.3} = 133 \text{ pounds.}$$

5th. Find Mean Back Pressure.

Rule.—"Initial press., divided by number of expansions, minus high press. terminal drop."

$$200 \div 3.3 = 60.6$$

Assume terminal drop of 1.6 lbs.

$$60.6 - 1.6 = 59 \text{ lbs. Back press.}$$

6th. Find Mean Effective Pressure.

Rule.—"Forward press., minus Back press."

$$133 - 59 = 74 \text{ lbs. M. E. P.}$$

7th. Find Effective Area of Piston.

Rule.—"Foot-pounds, divided by product of the piston speed in feet per minute, multiplied by M. E. P."

The work to be done in each cyl. = 290 H. P., which if expressed in foot-pounds is

$$290 \times 33{,}000 = 9{,}570{,}000 \text{ Ft.-pounds,}$$

which must equal

$$\text{Piston speed} \times \text{M. E. P.} \times \text{Eff. Area};$$

therefore, Effective Area must equal

$$\frac{9{,}570{,}000}{187 \times 74} = 691 \text{ sq. inches Eff. Area.}$$

8th. Find Actual Area and Diameter of H. P. Cyl.

Rule.—"Effective area plus half area of Rod."

$$\text{Rod} = 7'' \text{ dia. or } 38'' \text{ area.}$$

For simplicity, figure as though one-half of the rod was on each side of the piston,

$$691 + 19 = 710 \text{ sq. inches.}$$

$$\sqrt{\frac{710}{.7854}} = 30'' \text{ Diameter.}$$

Intermediate Cylinder.

9th. Find Number of Expansions in I. P. Cyl.

Rule.—"Square-root of Initial press., divided by Terminal press. in L. P. Cyl."

Assume initial press. as $1\frac{1}{2}$ lbs. less than receiver press. for the reason that the steam line falls as expansion goes on in receiver.

$\therefore$ Initial press. $= 59 - 1.5 = 57.5$ lbs. abso. and L. P. terminal being 5.5 abso.

$$\sqrt{\frac{57.5}{5.5}} = \sqrt{10.45} = 3.2 \text{ Exps.}$$

10th. Find Mean Forward Pressure.

Rule.—"Initial press., multiplied by Hyp. Log. of expansions plus 1, and divided by number of expansions."

$1 +$ Hyp. Log. of 3.2 is 2.1632

$$\therefore \frac{57.5 \times 2.1632}{3.2} = 38.8 \text{ Fd. press.}$$

11th. Find Mean Back Pressure.

Rule.—"Initial press., divided by number of expansions, then minus terminal drop."

$57.5 \div 3.2 = 18$ lbs.

Assume terminal drop in I. P. Cyl. as $1\frac{1}{2}$ lbs.

$18 - 1\frac{1}{2} = 16.5$ lbs. Back press.

12th. Find Mean Effective Pressure.

Rule.—Forward press., minus Back press.

$38.8 - 16.5 = 22.3$

But in an intermediate cylinder, as in a low press. cylinder, there is always some loss due to round corners, compression, etc., which reduces the M. E. P. below that figured. The exact amount of loss cannot be predetermined, but in this case we will allow 5 per cent.

$22.3 - 1.1 = 21.2$ lbs. M. E. P.

13th. Find Effective Area of Piston.

Rule.—"Foot-pounds, divided by product of piston speed in feet per min., multiplied by M. E. P."

Work done in I. P. Cyl. in foot-pounds is same as in H. P. Cyl., viz., 9,570,000 Ft.-lbs.

$$\therefore \frac{9{,}570{,}000}{187 \times 21.2} = 2414 \text{ Eff. Area.}$$

14th. Find Actual Area and Diameter of Cyl.

Rule.—"Effective Area plus ½ Area of Rod."

$$2414 + 25 = 2439 \text{ sq. inches.}$$

$$\sqrt{\frac{2439}{.7854}} = 55\tfrac{3}{4}''$$

Practically 56″ Diameter.

Low Pressure Cylinder.

15th. Find Number of Expansions in L. P. Cyl.

Rule.—"Initial press., divided by terminal press."

Due to fall in steam line, caused by expansion in receiver, the mean initial pressure may be taken as 15 pounds abso.

$$\therefore \quad 15 \div 6 = 2.5 \text{ Exps.}$$

16th. Find Mean Forward Pressure.

Rule.—"Initial press., multiplied by Hyp. Log. of expansions plus 1, and divided by number of expansions."

$$1 + \text{Hyp. Log. of } 2.5 = 1.9163$$

$$\therefore \frac{15 \times 1.9163}{2.5} = 11.5$$

Allow loss of 5 per cent. for round corners.

$$11.5 - .7 = 10.8 \text{ Fd. Press.}$$

17th. Find Mean Back Pressure.

With 28 inches vacuum and ample air pump the back press. may be assumed as 2 lbs. abso.

18th. Find Mean Eff. Pressure.

Rule.—"Forward press., minus Back press."

$$10.8 - 2 = 8.8 \text{ lbs. M. E. P.}$$

19th. Find Effective Area of Piston.

Rule.—"Foot-pounds, divided by product of piston speed in feet per minute, multiplied by M. E. P."

L. P. Work = 9,570,000 Ft.-pounds.

$$\frac{9{,}570{,}000}{187 \times 8.8} = 5820 \text{ sq. inches Eff. Area.}$$

20th. Find Actual Area and Dia. of Cylinder.

Rule.—"Effective area plus ½ area of Rod."

Rod 9″ dia. equals 64 inches area

$$\therefore \quad 5820 + 32 = 5852 \text{ sq. inches}$$

$$\sqrt{\frac{5852}{.7854}} = 86\tfrac{3}{8}'' \text{ dia.}$$

Practically 87″ Diameter.

NOTE! The above example, § 115, is based on the actual pumping engine at Chestnut Hill, Boston, Mass. The record of which is given § 113 and illustrated in frontispiece of this book.

§ **116. Use of Table No. 14.** Page 148.

This table has been prepared for general reference and comparison, and serves to show approximately the Ratio of Cylinder Areas in triple expansion, which are most suitable for various steam pressures ranging from 115 to 200 lbs. gauge.

It also serves as a rough and ready method of figuring the area of each cylinder for any horse-power, and of any pressure within the range of the table. One example will be sufficient to explain the process.

Example.—Find the cylinder areas for a 1,500 Horse-power triple expansion, condensing,

Steam pressure 175 lbs. gauge.
Piston speed 600 ft. per min.

High Press. 500 H. P.
- In column for 175 lbs.
- M. E. P. = 67.8
- $\therefore \dfrac{33{,}000 \times 500}{67.8 \times 600} = 406$ in. area.

Intermediate 500 H. P.
- In same column
- Ratio of areas = 3.14
- $\therefore \; 406 \times 3.136 = 1273$ in. area.

Low Press 500 H. P.
- Same column
- Ratio of areas = 7.53
- $\therefore \; 406 \times 7.524 = 3055$ in. area.

TABLE No. 14.
TERMINAL DROP IN L. P. CYL. ONLY.

TRIPLE EXPANSION.

	Steam Pressures in Pounds Above Atmosphere.																	
	115	120	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200
Initial Press. Abso.....	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200	205	210	215
Probable Loss Equiv.	2.7	2.8	2.9	3.0	3.1	3.2	3.3	3.4	3.5	3.6	3.7	3.8	3.9	4.0	4.1	4.2	4.3	4.4
Total Expansions......	22.0	22.8	23.6	24.1	24.9	25.7	26.5	27.0	27.7	28.6	29.1	29.7	30.5	31.3	32.1	32.4	33.2	34.0
Expans. in Each Cyl...	2.8	2.84	2.87	2.89	2.92	2.95	2.98	3.00	3.02	3.05	3.08	3.1	3.12	3.15	3.18	3.19	3.21	3.24
Mean Forward Pressure .... High	92.2	95.0	97.9	101.2	104.2	107.2	110.3	113.3	116.4	119.4	121.4	124.4	127.4	130.4	132.6	135.6	138.6	141.5
Mean Forward Pressure .... Int.	32.8	33.5	34.1	35.0	36.2	36.7	37.3	37.8	38.4	38.9	39.5	40.1	40.7	41.2	41.8	42.4	43.0	43.6
Mean Forward Pressure .... Low	11.77	11.86	11.95	12.1	12.2	12.3	12.4	12.5	12.6	12.7	12.8	12.9	13.0	13.1	13.2	13.3	13.4	13.5
Mean Backw'rd Pressure .... High	45.4	46.5	47.7	49.1	50.3	51.4	52.5	53.8	55.2	56.5	57.1	58.3	59.6	60.6	61.4	62.8	64.0	65.1
Mean Backw'rd Pressure .... Int.	16.2	16.4	16.6	17.0	17.2	17.4	17.6	17.9	18.2	18.4	18.6	18.8	19.0	19.2	19.3	19.6	19.9	20.0
Mean Backw'rd Pressure .... Low	3.9	3.9	3.9	3.9	3.9	3.9	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.1	4.1	4.1	4.1	4.2
Mean Effective Pressure .... High	46.8	48.5	50.2	52.1	53.9	55.8	57.5	59.5	61.2	62.9	64.3	66.1	67.8	69.8	71.2	72.8	74.6	76.4
Mean Effective Pressure .... Int.	16.6	17.1	17.6	18.1	19.0	19.3	19.6	19.9	20.2	20.5	20.8	21.1	21.6	21.9	22.4	22.9	23.2	23.6
Mean Effective Pressure .... Low	7.8	7.95	8.2	8.25	8.3	8.35	8.4	8.5	8.6	8.7	8.8	8.9	9.0	9.0	9.05	9.1	9.15	9.2
Terminal Pressure .... High	45.4	46.5	47.7	49.1	50.3	51.4	52.6	53.8	55.1	56.5	57.2	58.4	59.6	60.6	61.4	62.8	64.1	65.0
Terminal Pressure .... Int.	16.2	16.4	16.6	17.0	17.2	17.4	17.6	17.9	18.2	18.4	18.6	18.8	19.1	19.2	19.3	19.6	19.9	20.0
Terminal Pressure .... Low	5.8	5.8	5.8	5.9	5.9	5.9	5.9	5.9	6.0	6.0	6.0	6.1	6.1	6.1	6.1	6.2	6.2	6.2
Receiver Pressure .... First	45.4	46.5	47.7	49.1	50.3	51.4	52.6	53.8	55.1	56.5	57.2	58.4	59.6	60.6	61.6	62.9	64.1	65.0
Receiver Pressure .... Sec.	16.2	16.4	16.6	17.0	17.2	17.4	17.6	17.9	18.2	18.4	18.6	18.8	19.0	19.2	19.3	19.6	19.9	20.0
Ratio of Cyl. Areas.	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
	2.82	2.83	2.84	2.86	2.87	2.89	2.93	3.00	3.03	3.06	3.09	3.12	3.14	3.17	3.19	3.20	3.21	3.23
	6.00	6.10	6.12	6.30	6.49	6.68	6.84	7.00	7.12	7.23	7.30	7.42	7.53	7.75	7.86	8.00	8.15	8.30

CHAPTER No. 13.

QUADRUPLE EXPANSION ENGINES.

§ 117. **Notes.** The highest economy of the quadruple expansion engine cannot be fully realized so long as it remains handicapped by the present limitations in steam pressure. The best records made by quadruple engines up to this time would suggest that they are relatively inferior to the triple. But where economy of fuel is of first consideration and first cost is not restricted too much, then it is only a question of steam pressure. With 350 lbs. pressure of highly superheated steam and a practically uniform load, the quadruple expansion engine will certainly take the lead for superior economy.

To secure the best results the following 10 conditions are desirable:

1. Steam pressure, 350 lbs. gauge.
2. Ratio of cylinder areas in keeping with pressure.
3. Air pumps and condenser capable of 28 in. vac.
4. Superheated steam of 75 to 100 degrees.
5. Reheaters in receivers.
6. Steam jackets, independent steam supply.
7. Long stroke with small clearance.
8. Automatic cut-off mechanism.
9. Uniform load.
10. Vertically arranged cylinders.

Two examples will be worked, first with 250 pounds steam, and then with 350 lbs.

§ 118. **Example, 2,000 K. W. Electric Generator, Quadruple Expansion.**

NOTE! The chief reason why quadruple expansion is not recommended for electric generator work is the liability of load fluctuation, § 8. But assuming a case wherein the load is unusually steady, steam pressure and other conditions being favorable, this example is submitted.

Capacity, 2,000 K. W.
Piston speed, 750 per min.
Boiler pressure, 250 lbs. gauge.
Vacuum, 27 inches.

Find best cylinder areas and ratios.

Horse Power.

The horse-power equivalent to 2,000 K. W. See §106.

$$2000 \times 1.34 = 2680 \text{ H. P.}$$

add 10 per cent. for friction.

$$2680 + 268 = 2948 \text{ H. P. Total.}$$

Load for each cylinder $2948 \div 4 = 737$.

Total Expansions.

Initial press. $250 + 15 = 265$ lbs. abso.
Terminal press. in low press. cyl. 5.5

1st. Find Total Number of Expansions.

Rule.—"Initial press. abso., divided by terminal abso."

$$265 \div 5.5 = 48.2$$

High Pressure Cylinder.

2nd. Find number of expansions in this cylinder.

Rule.—"Fourth root of total expansions."

$$\text{thus } \sqrt[4]{48.2} = 2.6$$

3rd. Find Mean Forward Pressure.

Rule.—"Initial press., multiplied by 1 + Hyp. Log. of expansions, and divide by number of exps."

Thus, $1 + \text{Hyp. Log. of } 2.6 = 1.9555$,

$$\text{and } \frac{265 \times 1.9555}{2.6} = 160 \text{ lbs. M. F. P.}$$

4th. Find Mean Back Press.

Rule.—"Initial press., divided by number of exps., less terminal drop."

$$\frac{265}{2.6} - 2 = 100 \text{ lbs. M. B. P.}$$

5th. Find Mean Eff. Press.

Rule.—"Forward press., minus back press."

$$160 - 100 = 60 \text{ lbs. M. E. P.}$$

6th. Find Eff. Area of Piston.

Rule.—"Foot-pounds, divided by product of the piston speed feet per min., multiplied by M. E. P."

$$\frac{737 \times 33{,}000}{750 \times 100} = 324.7 \text{ Eff. Area.}$$

First Intermediate Cylinder.

7th. Find Number of Expansions.

Rule.—"Cube root of receiver press., divided by low press. terminal."

$$\text{Thus, } \sqrt[3]{\frac{98}{5.5}} = 2.6.$$

8th. Find Mean Forward Press.

Rule.—"Receiver press., multiplied by 1 + Hyp. Log. of expansion, divided by number of exps."

$$98 \, \frac{1 + \text{Hyp. Log. of } 2.6}{2.6} =$$

$$\frac{98 \times 1.9555}{2.6} = 73.7 \text{ lbs. M. F. P.}$$

9th. Find Mean Back Press.

Rule.—"Initial press., divided by number of exps., less terminal drop."

$$\frac{98}{2.6} - 2 = 35.7 \text{ lbs. M. B. P.}$$

10th. Find Mean Eff. Press.

Rule.—"Forward press., minus back press."

$$73.7 - 35.7 = 38 \text{ lbs. M. E. P.}$$

11th. Find Eff. Area of Piston.

Rule.—"Foot-pounds, divided by product of piston speed in feet per min., multiplied by M. E. P."

$$\frac{737 \times 33{,}000}{750 \times 38} = 854.5 \text{ sq. inches Eff. area.}$$

Second Intermediate Cylinder.

12th. Find Number of Expansions.

Rule.—"Square root of receiver press., divided by L. P. terminal."

$$\sqrt{\frac{35}{5.5}} = 2.5$$

13. Find Forward Press.

Rule.—"Receiver press., multiplied by 1 + Hyp. Log. of exps., divided by number of exps."

$$1 + \text{Hyp. Log. of } 2.5 = 1.9163$$

$$\frac{35 \times 1.9163}{2.5} = 26.8 \text{ lbs. M. F. P.}$$

14th. Find Mean Back Press.

Rule.—"Receiver press., divided by number of exps., less terminal drop."

$$\frac{35}{2.5} - 1 = 13 \text{ lbs. M. B. P.}$$

15th. Find Mean Eff. Press.

Rule.—"Forward press., minus back press."

$$26.8 - 13 = 13.8 \text{ lbs. M. E. P.}$$

16th. Find Eff. Area of Piston.

Rule.—"Foot pounds, divided by product of piston speed in feet per min., multiplied by M. E. P."

$$\frac{737 \times 33{,}000}{750 \times 13.8} = 2456 \text{ sq. in. Eff. area.}$$

Low Pressure Cylinder.

17th. Find Number of Expansions.

Rule.—"Receiver press., divided by L. P. terminal."

$$12.7 \div 5.5 = 2.3$$

18th. Find Mean Forward Press.

Rule.—"Receiver press., multiplied by 1 + Hyp. Log. of exps., divided by number of exps."

$$1 + \text{Hyp. Log. of } 2.3 = 1.8329$$

$$\frac{12.7 \times 1.8329}{2.3} = 10.1 \text{ lbs. M. F. P.}$$

19th. Find Mean Back Press.

Rule.—"L. P. Terminal, multiplied by .66."

$$5.5 \times .66 = 3.6 \text{ M. B. P.}$$

20th. Find Mean Eff. Press.

Rule.—"Forward press., minus Back press."

$$10.1 - 3.6 = 6.5 \text{ M. E. P.}$$

21st. Find Eff. Area of Piston.

Rule.—"Foot-pounds, divided by product of piston speed, multiplied by M. E. P."

$$\frac{737 \times 33000}{750 \times 6.5} = 5214 \text{ sq. in.}$$

Ratio of Cylinder Areas.

High to low = 5214 ÷ 324.7 = 16 to 1.
1st Int. to low = 5214 ÷ 854.5 = 6.1 to 1.
2nd Int. to low = 5314 ÷ 2456 = 2.16 to 1.

NOTE! The above example assumes two things; 1st, that the load is unusually steady, and 2nd, that the receivers are equipped with thoroughly efficient reheaters, so as to ensure dry steam and if possible superheated. § 123.

§ 119. Example, 40 Million Gallon Quadruple Expansion Pumping Engine.

Find cylinder areas necessary for a Quadruple expansion pumping engine having:

Capacity of 40 mill. gals. in 24 hours.
Head of water pumped against 150 feet.
Boiler pressure, gauge 350 pounds.
Piston speed per minute, 200 feet.
Vacuum, 27 inches.
Diagram Fig. 33.

Horse Power.

Head of water, multiplied by number of million gals., multiplied by .1755 See § 109.

$$150 \times 40 \times .1755 = 1053 \text{ H. P.}$$

add 8 per cent. for friction,

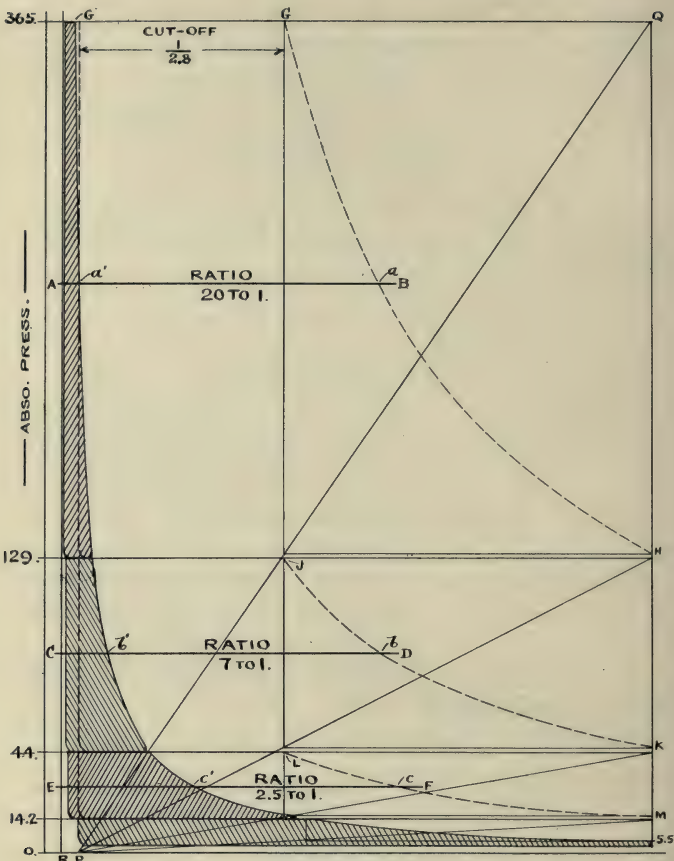


Fig. 33.

$$1053 + 85 = 1138 \text{ Total,}$$
$$1138 \div 4 = 284.5 \text{ in each cyl.}$$

Total Expansions.

Assume L. P. terminal press. 5.5 abso.

$$365 \div 5.5 = 66.36$$

High Pressure Cylinder.

$$\text{Expansions} = \sqrt[4]{66.36} = 2.8$$

Forward press., 1 + Hyp. Log. of 2.8 is 2.0296

$$\frac{365 \times 2.0296}{2.8} = 264.5$$

Back press., assuming terminal drop 1.5 lbs.

$$\frac{365}{2.8} - 1.5 = 128.5 \text{ lbs. abso.}$$

Mean Eff. press.,

$$264.5 - 128.5 = 136 \text{ M. E. P.}$$

Eff. Area of Piston,

$$\frac{284.5 \times 33{,}000}{200 \times 136} = 345.2 \text{ sq. in.}$$

Piston Rod Area.

NOTE! The piston rod being in one end of the cylinder only, add half area of same to eff. area of piston. Then the area of rod for maximum press. and strain on rod of 5,000 lbs. per sq. inch, is:

$$\frac{350 \times 345.2}{5{,}000} = 24 \text{ sq. in.}$$

Actual area of cyl. = 345 + 12 = 357 sq. in.

First Intermediate Cylinder.

Initial press. ½ lb. less than receiver = 128 lbs.

$$\text{Expansions} = \sqrt[3]{\frac{128}{5.5}} = 2.8.$$

Forward press.

$$\frac{128 \times 2.0296}{2.8} = 92.5 \text{ M .F. P.}$$

Back press., assuming terminal drop 1 lb.

$$\frac{128}{2.8} - 1 = 45 \text{ lbs. abso. M. B. P.}$$

Mean Eff. Press.

$$92.5 - 45 = 47.5 \text{ M. E. P.}$$

Eff. Area of Piston.

$$\frac{284.5 \times 33{,}000}{200 \times 47.5} = 986 \text{ sq. inches.}$$

Piston Rod Area.

$$\frac{986 \times 128}{5,000} = 24 \text{ sq. inches.}$$

Actual Area of Cylinder.

$$986 + 12 = 998 \text{ sq. inches.}$$

Second Intermediate Cylinder.

Initial press. 1 lb. less than receiver = 44 lbs.

$$\text{Expansion} = \sqrt{\frac{44}{5.5}} = 2.8.$$

$$\text{Forward Press. } \frac{44 \times 2.0296}{2.8} = 31.8$$

Back Press., assuming terminal drop of 1 lb.

$$\frac{44}{2.8} - 1 = 14.8 \text{ abso.}$$

Mean Eff. Press.

$$31.8 - 14.8 = 17 \text{ M. E. P.}$$

Eff. Area of Piston.

$$\frac{284.5 \times 33,000}{200 \times 17} = 2761 \text{ sq. inches.}$$

Piston Rod Area.

$$\frac{2761 \times 44}{5,000} = 24 \text{ sq. inches.}$$

Actual Area of Cyl. = 2761 + 12 = 2773 sq. inches.

Low Pressure Cylinder.

Initial Press. 14.2 — .7 = 13.5 lbs. abso.

Expansions = 13.5 ÷ 5.5 = 2.5.

$$\text{Forward Press. } \frac{13.5 \times 1.9163}{2.5} = 10.35 \text{ M. F. P.}$$

Back Press for 28″ vac. = 3.55 M. B. P.

Mean Eff. Press. = 10.35 — 3.55 = 6.8 M. E. P.

Eff. Area of Piston

$$\frac{284.5 \times 33,000}{200 \times 6.8} = 6903 \text{ sq. inches.}$$

$$\text{Piston Rod Area} = \frac{6903 \times 17}{5{,}000} = 24 \text{ sq. inches.}$$

Actual Area of Cyl., $6903 + 12 = 6915$ sq. inches.

Ratios of Cylinder Areas.

High to Low	$6903 \div 345 =$	20 to 1
1st Int. to Low	$6903 \div 986 =$	7 to 1
2nd Int. to Low	$6903 \div 2761 =$	2.5 to 1

CHAPTER No. 14.

RECEIVERS.

The receiver is virtually a steam reservoir, and is situated somewhere between the high and low pressure cylinders, with suitable piping connections. It is to the low press. cylinder what the boiler is to the high, viz., its source of steam supply.

§ 120. **Purpose.** The purpose of the receiver is two-fold; first, to receive the exhaust from the high press. cylinder and to hold it in reserve ready to be discharged at short intervals into the low press. cylinder.

For each revolution of the engine, two high press. cylinder volumes are discharged into it, and having but little time between the end of one volume and the commencement of the next, the flow into the receiver is almost continuous; but the flow into the low press. cylinder is not so continuous, since it can only take place during the admission periods, before cutting off, which is $\frac{1}{3}$ or $\frac{1}{4}$ of the stroke as the case may be, therefore, the receiver pressure must be variable, further shown § 121; but the object should be to make the pressure as uniform as possible, and for which we depend largely upon the extent of the receiver volume, § 122.

The further purpose of the receiver is to furnish steam to the low press. cylinder, which shall be as dry as possible or even superheated.

The exhaust from the high press. cylinder carries with it its condensation water and the steam is consequently very wet; the value of the receiver is increased in proportion as the drying or superheating of the steam can be accomplished. § 123.

§ 121. **Receiver Pressures.** The pressure in the receiver must necessarily vary to some extent for reason illustrated in Fig. 34.

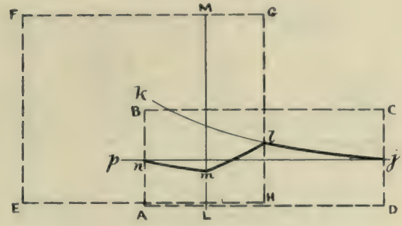


Fig. 34.

Let A B C D represent the high pressure and E F G H the low pressure cylinders of a Cross-Compound. As the cranks are usually set at 90 degrees, the high press. leading, then the cylinders are relatively to each other as shown in the diagram, while the pressure lines j l m n represent the varying pressure in the receiver proper, the pressure scale being exaggerated for the purpose of defining the points.

Let p j represent pressure when high press. piston starts from D C, the exhaust valve being opened at A about the same time; then the pressure in the high press. cylinder falls to the receiver line p j, except that it is increased a little by the additional incoming volume. The high press. piston moving forward, pushes its volume into the receiver, and if none is discharged from the receiver during the same time, the pressure line would rise by compression towards k. But on reaching H, which is about one-half of distance A D, the low press. inlet valve opens, and by the time the low press. piston has reached the point of cut-off, L M, a volume equal to one high press. cyl. volume, has been taken out from the receiver, thus reducing its pressure again from l to m; after which, the inlet to the low press. cylinder being cut off, but the high press. piston completing its stroke, increases the receiver press. again until it returns to n in line p j.

The Tandem-Compound has a somewhat different pressure variation as shown in Fig. 35.

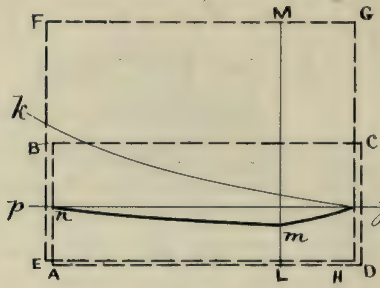


Fig. 35.

Being tandem, the inlet to the low press. cylinder commences at the same moment as the high press. cylinder begins to discharge its volume, and the diagram shows the cylinder so arranged relatively. Let A B C D represent the high pressure, and E F G H the low pressure cylinders, p j receiver pressure line at the beginning of the stroke. As a volume of steam equal to the volume of the high press. cylinder must enter the low press. cylinder by the time the piston reaches L, and since the high pressure cylinder has not by this time discharged as much as in the case of a cross-compound, the receiver pressure drops to m, at which point the low press. inlet being cut off, the receiver pressure again rises as the high press. piston completes its stroke, but m will be lower than in the case of the cross-compound. It is, therefore, a mistake to make the tandem receiver any less than for cross-compound as seen by the study of the two diagrams.

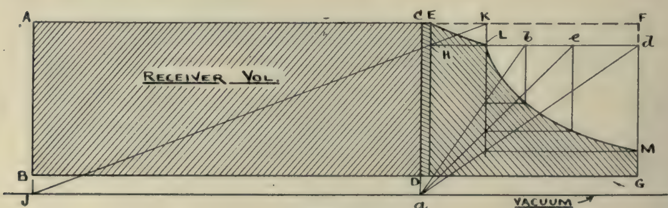


Fig. 36.

§ **122. Receiver Volume.** The greater the volume of the receiver, the less will be the drop in the steam line of the low press cylinder during the admission period,

The best modern practice is based on following rules:

Triple Expansion. *Rule.—"Receiver volume between high press. and intermediate cylinders equals 6 times the high press. cylinder volume, that between the intermediate and low press. cylinders equals 4 times the intermediate."*

For Double Expansion. *Rule.—"Receiver volume between high press. and low press. cylinders equals 5 times the high pressure cylinder."*

Referring to Fig. 36, let A B C D represent a receiver volume, and C D F G, a low pressure cylinder including clearance C E; let K be the point of cut-off. Then to find the drop in pressure during the expansion period, draw a line from K to J, cutting vertical line E at H., from which draw horizontal line to d, and where it cuts vertical line K in L, will represent the pressure at point of cut-off. The expansion curve L. M. may be completed in the usual way (§ 60) by lines radiating from point a.

By carefully studying this diagram it will be observed that as the receiver volume is increased, the less will be the fall in pressure during the expansion period. In other words the larger the receiver volume relatively to the low pressure cylinder, the less will be the fluctuation of the receiver pressure.

§ **123. Reheaters.** The purpose of a reheater is to re-evaporate the moisture in the steam and so far as possible to superheat the same before it enters the low press. cylinder. This of course can only be accomplished at the expenditure of additional heat supplied from some other source. Three methods of supplying heat for above purpose are suggested: First, the common method of steam coil placed within the receiver and through which live steam direct from the boiler can circulate, and so communicate heat to the receiver steam surrounding it. The radiating surface of such reheaters is usually about 2½ square feet per cubic foot of receiver volume.

The second method of reheating is that of utilizing the heat of the partially spent gases from the furnace, by causing them to pass through a coil system within or surrounding the receiver steam. The third suggested method consists of a receiver constructed more on the plan of a steam boiler with combustion chamber in which oil or coal gas may be consumed, thus suppyling the necessary heat for thoroughly drying or even superheating the receiver steam to any desirable degree. The last method is the best.

§ 124. **Drainage.** Even when the reheater can be used, it is of the greatest importance to thoroughly drain the water from the lowest point of the receiver, as it is false economy to attempt to re-evaporate the condensation water while within the receiver, but it should be pumped back to the boiler through properly covered piping and re-evaporated there.

In draining a receiver it is of the greatest importance to install a thoroughly efficient steam trap, great loss may occur unnoticed, by poor steam traps.

CHAPTER No. 15.

CONDENSING APPARATUS.

Calculation for efficient condensing apparatus to suit conditions, and capacity for producing and maintaining a good vacuum is of no small importance in the study of the compound condensing engine.

The following brief articles will be found useful and each one should have due consideration.

§ **125. Good Vacuum.** Exactly what is understood by the words "Good Vacuum" is somewhat uncertain and indefinite. In a general way most engineers look for 26 or 27 inches, but strictly speaking, this is possible only for a limited altitude as shown in § 126, but for ordinary altitude, 27 inches of vacuum is considered good. To exceed this, the quantity of injection water necessary, the extra work caused thereby on the air pump and the increase of power required to operate the same, together with the consequent low temperature of the overflow water which is specially objectionable when part of it is to be used as boiler feed, all tend to offset the apparent advantage of a greater vacuum.

§ **126. Vacuum and Pressures, Below Atmosphere.** To find pressure in pounds abso., corresponding to vacuum in inches, for any given altitude there are three successive steps to take.

First, find barometer reading for any given altitude. *Rule.—"From 30, subtract altitude in feet divided by 1,000."*

Second, find atmospheric pressure for given barometer reading. *Rule.—"Barometer reading in inches, multiplied by .49"*

Third, find absolute pressure corresponding to vacuum in inches at given altitude. *Rule.—"From atmospheric* *pressure at given altitude, subtract vacuum in inches, multiplied by .49.*"

Try two examples of different altitudes.

First Example. Find pressure in pounds abso., corresponding to 26 inches of vacuum at sea level.

1st, Barometer reading at sea level $= 30 - \frac{0}{1000} = 30$ in.

2nd, Atm. press. at 30 in. barometer $= 30 \times .49 = 14.7$ pounds abso.

3rd, Press. abso. corresponding to 26 inches vac. $=$
$14.7 - (26 \times .49) = 1.96$ pounds abso.

Second Example. Find pressure in pounds abso., corresponding to 24 inches of vacuum at 3,000 ft. altitude.

TABLE No. 15.

VACUUM AND ABSOLUTE PRESSURES AT SEA LEVEL.

VACUUM IN INCHES.	ABSO. PRESS. IN POUNDS.	VACUUM IN INCHES.	ABSO. PRESS. IN POUNDS.	VACUUM IN INCHES.	ABSO. PRESS. IN POUNDS.
0.	14.7	11.	9.31	22.	3.92
0.5	14.14	11.5	9.26	22.5	3.67
1.	14.21	12.	8.82	23.	3.43
1.5	13.96	12.5	8.57	23.5	3.19
2.	13.72	13.	8.33	24.	2.94
2.5	13.47	13.5	8.08	24.5	2.7
3.	13.23	14.	7.84	25.	2.45
3.5	12.98	14.5	7.58	25 5	2.2
4.	12.74	15.	7.35	26.	1.96
4.5	12.5	15.5	7.10	26.5	1.71
5.	12.25	16.	6.86	27.	1.47
5.5	12.01	16.5	6.61	27.5	1.22
6.	11.76	17.	6.37	28.	.98
6.5	11.5	17.5	6.12	28.5	.73
7.	11.67	18.	5.88	29.	.49
7.5	11.03	18.5	5.63	29.5	.25
8.	10.78	19.	5.39	30.	. 0
8.5	10.54	19.5	5.14	......	......
9.	10.29	20.	4.9	......	......
9.5	10.05	20.5	4.65	......	......
10.	9.8	21.	4.41	......	......
10.5	9.56	21.5	4.16	......	......

1st, Barometer reading at 3,000 alt. $= 30 - \frac{3000}{1000} = 30 - 3 = 27$ in.

2nd, Atm. press. at 27 in. barometer $= 27 \times .49 = 13.23$ pounds.

3rd, Press. abso., corresponding to 24 inches vac. $=$ $13.23 - (24 \times .49) = 13.23 - 11.76 = 1.47$ lbs. abso.

§ 127. **Back Pressure in L. P. Cylinder.** Never overlook the fact that it is impossible to get the full effect of the vacuum as shown by the gauge readings taken at the condenser, as there is always less vacuum in the cylinder due first to the vapor caused be moisture and also due to friction in the exhaust passages. Then the variation in the vacuum line shown on indicator cards, with rise and fall and rounded corners caused by compression, etc., also tend to make the mean back pressure line several inches above the gauge readings. Now while it is impossible to predetermine the exact back pressure average, yet based on actual observation, an approximate rule can be made to serve the purpose in general figuring, but of course on the assumption of moderate altitude and ample condensing apparatus. The rule suggested is to allow that the average back pressure in the low pressure cylinder shall be considered as two-thirds of the terminal pressure. Upon this rule Table No. 16 is submitted, showing the approximate terminal pressures and back pressures for various rates of expansion.

TABLE No. 16.

APPROXIMATE MEAN BACK PRESSURE IN LOW-PRESSURE CYLINDER.

ABSOLUTE TERMINAL PRESSURE.	MEAN BACK PRESSURE.
7	4.6
6.5	4.3
6	3.9
5.5	3.6

Example.—Find the approximate back pressure upon the low pressure piston of a two cylinder compound condensing engine.

Table No. 8 gives the terminal pressure as equal to 6.5 pounds absolute; then for mean back pressure, which is allowed as two-thirds of that, we have:

$$6.5 \times .66 = 4.3.$$

If the load is allowed to exceed that for which it was designed, and thus the piston is permitted to travel farther before cutting off, then of course the terminal pressure is likewise increased, and the condensing apparatus is taxed beyond its rated capacity, consequently the vacuum is diminished accordingly. Therefore always provide rather more than enough; it pays to be liberal in this particular respect. Table No. 16 is figured with this in view.

TYPES OF CONDENSING APPARATUS.

There are many different styles of condensing apparatus—that is to say, as far as mechanical combinations are concerned—but there are only three distinct types, viz., the Bulkley type, the Jet condenser type, and the Surface condenser type. The type to use to give the best results depends upon the conditions under which it is to be placed, and here, like as in other things herein discussed, the good judgment of the engineer must be exercised.

§ **128. Bulkley Condenser.** This type of condensing apparatus requires no air pump, consequently it is lower in first cost than others. When there is sufficient cold water, 24 inches of vacuum can easily be maintained. Fig. 37 shows the condenser head for regulating the amount of water necessary for condensing the steam. Fig. 38 shows the ordinary setting, the condenser head being placed at a height of 34 feet above the hot well. The water is supplied by an ordinary force pump, but the condenser itself will syphon nearly 17 feet, the remaining 17

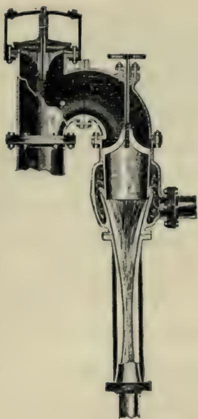


Fig. 37.

Fig. 38.

feet being the head against which the force pump has to work for ordinary temperatures of, say, 45 to 50 degrees. The quantity of water per minute per horse-power may be roughly estimated as 1.4 gallons for single exps., 1.2 gallons for double exps., 1.0 gallon for triple exps. The water is fed in the side of the condensing chamber, and passes down through the contracted throat with such velocity as to carry with it all the air passing over with the steam; thus the condenser clears itself of its discharge water and air by gravity alone, without any air pump.

SURFACE CONDENSING APPARATUS.

§ 129. **Surface Condenser.** The principle and construction of the surface condenser is such that the exhaust steam and the cooling or circulating water are kept separate. This is important when salt sea water or other waters containing impurities which would prevent their use as feed to the boilers. By preventing the mixture, the condensation water can be thus made use of, while the circulating water only passes to waste.

The construction provides a cooling surface on the outside of a number of tubes with which the exhaust steam comes in contact on its way to the vacuum pump, the cooler surface being maintained by the cold water circulating within the tubes. The efficiency depends upon the more or less perfect manner of construction and the extent of cooling surface.

§ 130. **Quantity of Water for Surface Condensers.** The surface condenser requires a greater quantity of water per pound of steam, by about 15 per cent., than does the jet condenser, for the reason that not being mixed with the steam it cannot absorb the heat as rapidly. In figuring the quantity of water the same reasoning may be applied as under the head of jet-condensing, § 137, and then add the 15 per cent.

§ 131. **Pumps, Vacuum and Circulating.** In connection with the surface condenser type of condensing apparatus, two pumps are required, viz.: 1st, the vacuum-pump which is connected on the steam side to remove the condensation water and air; and 2nd, the circulating-pump which simply pumps the cooling water through the tubes, the capacity of the same being figured as an ordinary water pump when the quantity of water, the lift and the head are known.

§ 132. **Jet Condensing Apparatus, Construction.** This type of condensing apparatus is no doubt the most common one for land service, where fresh water can be had. It consists generally—though not always—of a vertical single-acting air-pump, with a bucket containing the suction valves, and moving below a fixed delivery plate and valves, at a velocity of about one-sixth of the engine piston speed. The condenser consists of a vertical hollow vessel, the upper end communicating with the cylinders by the exhaust pipe, and the lower end with the air-pump by the channel way, and the whole system made perfectly air-tight. Into the condenser and near the exhaust steam opening (above, if possible), is introduced the cold water injection sprinkler or spray pipe, causing a shower of cold water to enter and mingle with the exhaust steam immediately upon its admission, thus producing a mixture of injection water and condensation water which is conducted through the channel way to the air-pump, and by it pumped out as overflow water—so called —to the hot well. Of course, there often is occasion to use impure water and such as cannot be used for boiler feed, but it must be at the sacrifice of the heat, which might otherwise be saved, and which is of considerable value from point of economy, when we remember that it usually reaches about 110 degrees, which means possibly a saving of 50 or 60 degrees.

Condenser Volume. To provide a condenser of such a volume as shall be in proprtion to the volume of the air-pump, figure roughly, and not necessarily with the same exactness as for the air pump; but it should nevertheless be governed by some simple rule proportionately with it. The following rule approaches very closely to the correct proportion:

Rule.—"Volume of air-pump multiplied by 4, equals the volume of jet condenser."

Example.—Find the diameter of a jet condenser 6 feet long, connected with a 31″x12″ air-pump.

Here we have air-pump whose volume is 9,057 c. in., and as the condenser is said to be 72″ long, the diameter will be

$$\frac{9{,}057 \times 4}{72} = 503 \text{ sq. in.} = 25\tfrac{3}{8}'' \text{ dia. nearly.}$$

INJECTION WATER.

The quantity of injection water required for a given volume of steam depends upon four conditions, each of which have their importance.

1st. Temperature of injection water.

2nd. Temperature of overflow water.

3rd. Volume and density of exhaust steam.

4th. Total heat units to be absorbed.

Treat each in their order as follows:

§ 133. **Temperature of Injection.** The temperature of injection water before entering the condenser is, in the majority of cases, subject to change at the different seasons of the year, depending, of course, upon ungovernable local conditions. Assume, therefore, that for example we have a case where in winter the temperature is 35 degrees and in summer 80 degrees, and assume that the desired temperature of overflow is 110 degrees. Then one pound of it will absorb in winter, say, $110° - 35° = 75$ degrees or units, from every one pound of steam, and in summer $110° - 80° = 30$ units only. See §137.

§ 134. **Temperature of Overflow.** There is no arbitrary rule that fixes an exact temperature for overflow, but every engineer has decided in his own mind what he believes to be the best in his particular engine. It may be desirable in some instances, where water is of greater value than in others, to use as small a quantity as possible, even though it be at a sacrifice of a little better vacuum, the result of which will be to raise the temperature of the overflow, whereas to maintain a better vacuum, more water must be injected, and a lower temperature of the overflow would be the result, as well as the increased work upon the air-pump. It is an open question whether the small additional vacuum gained compensates for other losses, but the best experience places the temperature of overflow at about 110° Fahr.

§ 135. **Volume and Density of Exhaust Steam.** The weight of a given volume of steam, as compared with the same volume of water, is understood as the density of steam, and since the density and temperature vary in proportion to the pressure, if the pressure is known, the density or weight can be found. For instance, by referring to Table No. 1 we find the weight of one cubic foot of steam at a pressure of 6 pounds abso., is .0163 pounds. Therefore, the weight of a given volume of steam at a given pressure is the cubic feet multiplied by the density, that is to say, if we have 100 cubic feet at 6 pounds abso.,

$$\text{Weight} = 100 \times .0163 = 1.63 \text{ lbs.}$$

§ **136. Absorption of Heat Units.** In addition to the heat indicated by the thermometer, the steam contains a large quantity of latent heat to which the thermometer is not sensible. The total heat, therefore, is the sensible and latent together, all of which has to be absorbed by the injection water, § 14. Referring to Table No. 4, one pound weight at one pound abso. pressure has a total heat of 1,112 units, and that as the pressure increases the total heat increases. If the terminal pressure of the steam after expansion is 7 pounds abso., the total heat units is seen to be 1,135.3. Now, as all of this heat must be absorbed by the injection water, each pound of water so used must carry off a certain quantity of heat in addition to its own, and the number of pounds of such water necessary to absorb sufficient heat units to condense the steam and produce a mixture (overflow) of any desired temperature can easily be figured as follows:

§ **137. Weight of Injection Water.** On the ground of what has been said above, if the injection water is 50° and the overflow required is 110° each pound of such water can absorb 110 — 50 = 60 units, and as one pound weight of steam at, say, 7 lbs. abso., contains 1,135.3 units, to reduce it to 110° it needs as many pounds of water to reduce the whole as there are sixties in 1,135.3 less 110.

$$\text{viz.,} \quad \frac{1{,}135.3 - 110}{60} = 17 \text{ pounds,}$$

expressed in algebraic terms would be

X = pounds of steam, weight.
H = total heat units in the steam.
T = temperature of mixture.
Y = pounds of injection water.

$$X\,(H - T) = Y\,(110 - 50).$$
$$1\,(1{,}135.3 - 110) = Y\,60.$$
$$1{,}025.3 = Y\,60.$$
$$Y = \frac{1{,}025.3}{60} = 17 \text{ pounds.}$$

The weight of overflow water equals the weight of injection water plus weight of steam, which in the above case is 17 + .0189.

For rough figuring, the approximate quantity of injection water is from 18 to 20 times the quantity of feed water used by the boiler for engine only.

TABLE No. 17.

QUANTITY OF INJECTION WATER FOR JET CONDENSERS.

Injection Tem. 50°. Overflow Tem. 110°.

LOW-PRESSURE CYLINDER.	WATER PER REV.					
	SINGLE EXP. ENGINES.		DOUBLE EXP. ENGINES.		TRIPLE EXP. ENGINES.	
	Lbs.	Galls.	Lbs.	Galls.	Lbs.	Galls.
20″ x 36″	4.2	.5	3.9	.47	3.6	.43
22″ x 36″	5.1	.61	4.8	.57	4.4	.53
24″ x 42″	7.	.84	6.6	.79	6.	.72
26″ x 42″	8.3	1.	7.8	.93	7.2	.87
28″ x 48″	11.	1.45	10.4	1.24	9.5	1.14
30″ x 48″	12.6	1.52	11.7	1.41	10.8	1.3
32″ x 54″	16.2	1.95	15.	1.81	13.9	1.68
34″ x 54″	18 3	2.2	17.0	2.05	15.8	1.9
36″ x 60″	22.8	2.75	21.2	2.55	19.6	2.36
38″ x 60″	25.5	3.07	23·7	2.85	21.9	2.64
40″ x 66″	31.	3.73	28.8	3.45	26.7	3.2
44″ x 66″	37.5	4.51	34.8	4.2	32.2	3.8
48″ x 72″	48.5	5.84	45.	5.42	41.7	5.
52″ x 72″	57.	6.89	53.1	6.4	49.2	5.9
56″ x 72″	66.	7.9	61.5	7.41	57.	6.8
60″ x 72″	75.6	9.	70.5	8.5	65.3	7.8
64″ x 72″	85.	10.	80.	9.6	74.	8.9

§ 138. **Velocity of Injection Water.** The theoretical velocity of water, through a pipe having no friction, is based on the formula

$$\sqrt{2\,g\,h} = \text{vel. in feet per second.}$$

in which $g = 32.2$ and $h =$ head in feet, or pressure per sq. in. multiplied by 2.3.

FORCE INJECTION.

For force injection with a pressure in the pipe together with the vacuum in the condenser, we should have the formula

$$\sqrt{2\,g\,(h + h^1)} = \text{vel. feet per second}$$

in which $h =$ head equivalent to vacuum and $h^1 =$ head due to pressure in the pipe.

For instance, with a vacuum of 26.5 inches we should have a pressure of 55 pounds per inch, and

$$h = 26.5 \times .49 \times 2.3 = 29 \text{ feet}$$
$$h^1 = 55 \times 2.3 = 126 \text{ feet}$$
$$h + h^1 = 29 + 126 = 155 \text{ feet head}$$

$$\therefore \sqrt{2\,g\,(h + h^1)} = \sqrt{64.4 \times 155} = 100 \text{ ft. per sec.}$$

But in practice never figure for force injection more than ½ the theoretical velocity, viz.,

$100 \div 2 = 50$ ft. per sec. or 3,000 feet per min.

SUCTION INJECTION.

Suction Injection, with, say 26.5 inches of vacuum, 19 ft. lift from water level to center of inlet, and a friction head of 4, we have

$$\sqrt{2\,g\,(h - h^2)} = \text{vel. in feet per sec.}$$

in which $g = 32.2$, $h =$ head equivalent to vacuum, viz., 26.5 inches $= 13$ pounds, and $13 \times 2.3 = 29$ and $h^2 =$ lift from water level, viz., 19 ft.

Velocity $= \sqrt{64.4\,(29 - 19)} = \sqrt{64.4 \times 10} = 25$ ft. per sec.

But in practice, due to leaks, friction, and other liabilities, never figure more than ⅕ of the theoretical, viz., 5 ft. per second, or 300 feet per minute.

§ 139. **Diameter of Injection Pipes.** The diameter of an injection pipe, for a given quantity of water in galls. per minute, depends upon the velocity at which it actually flows from the pipe into the condenser, but assuming the velocity to be 3,000 feet per min. for Force injection, and 300 feet per min. for Suction injection, then the following rule may be used for quickly finding the diameter of an injection pipe for any required number of gallons per minute.

"Diameter squared, multiplied by Factor, equals number of gallons required per minute."

This rule expressed in the form of an equation would be

$$d^2 \text{ Factor} = \text{galls. per min.}$$

The factor for 3,000 feet per min., is 192 and for 300 feet per min. 12.

The following examples may be helpful:

FORCE INJECTION.

Then on the basis of water flowing into the condenser at a velocity of 3,000 feet per min., which is true for 55 pounds pressure and 26.5 vacuum, § 138, the diameter of a force injection pipe should be, with a factor of 192

$$d^2\ 192 = \text{galls. per min.}$$

For instance to inject 775 galls. per min.,

$$d^2\ 192 = 775$$

$$\text{or } d^2 = 775 \div 192$$

$$\text{and } d = \sqrt{\frac{775}{192}} = 2'' \text{ dia.}$$

or, to inject say 1,200 galls. per min.

$$d = \sqrt{\frac{1200}{192}} = 2\tfrac{1}{2}'' \text{ dia.}$$

NOTE.—It is advisable for the student at this point to go back to §138 and for practice figure the velocity under different pressures and establish for himself other factors in accordance with the same, all of which will prove useful.

SUCTION INJECTION.

On the basis of a velocity of 300 feet per min. the diameter of a suction injection pipe should be, with factor of 12

$$d^2\ 12 = \text{galls. per min.}$$

For instance, to inject 775 galls. per min.,

$$d^2\ 12 = 775$$

$$d^2 = 775 \div 12$$

$$d = \sqrt{\frac{775}{12}} = 8'' \text{ dia.}$$

or, to inject 1,200 galls. per min.

$$d = \sqrt{\frac{1200}{12}} = 10'' \text{ dia.}$$

VACUUM PUMPS.

§ 140. **Volume of Pump.** To insure a good proportion and yet not to enter too minutely into the less important features, we can figure the necessary volume of a vacuum-pump by considering only two of the more important conditions, viz.: First, the total volume of the low pressure cylinder; second, the density of the exhaust steam. The volume of a single-acting pump can be determined by the following rule:

Rule.—"Volume of low pressure cylinder discharge in cubic feet per rev. of engine, multiplied by 3.5, and divided by the number of cubic feet contained in one pound of exhaust steam," thus:

$$\frac{\text{Vol. of Cyl. in C. ft.} \times 3.5}{\text{C. ft. in one lb. of ex. steam}} = \text{Vol. of air-pump in C. ft.}$$

Example.—Find the volume of a single acting air pump for a triple expansion engine, of which the low pressure cylinder is 3055 square inches in area by 60 inches stroke, and terminal steam pressure 6 pounds abso.

Following above rule we should have,

$3055 \times 60 \times 2 = 366{,}600$ c. in. $= 212$ c. ft. of steam, then the number of cubic feet in one pound of steam at 6 pounds abso. pressure is 61.21. See Table No. 10.

$$\therefore \frac{212 \times 3.5}{61.21} = 12.1 \text{ cubic ft. of air pump.}$$

To find the diameter of above pump, with 15-inch stroke, we should have,

$$12.1 \text{ c. ft.} = 20{,}908 \text{ c. inches,}$$

$$\text{and,} \quad \frac{20{,}908}{15} = 1393 \text{ sq. in.} \qquad 42.1 \text{ in. dia.}$$

Therefore a $42'' \times 15$ Single Acting Air Pump is required, at a speed of one stroke per revolution of engine.

For the sake of experience try another example as follows:

Example.—Determine the diameter of a single-acting air-pump for a triple expansion engine with a low pressure cylinder $52''$ diameter $\times$ $60''$ stroke, assuming the stroke of pump to be $15''$.

NOTE! The air-pump being single-acting, there will be for each stroke a volume of exhaust steam equal to twice the volume of a low-pressure cylinder.

$52''$ diameter $= 2{,}123$ sq. inches area $= 14.74$ sq. feet.

Then the exhaust steam volume for each stroke of the air-pump equals $14.74 \times 5 \times 2 = 147.4$ cubic feet.

Table 8 shows the terminal pressure in low-pressure cylinder of triple expansion as 6 pounds.

Now, by reference to Table No. 10, Saturated Steam, we find that in one pound of steam at 6 pounds absolute pressure there are 61.21 cubic feet; therefore we have for the volume of air-pump,

$$\frac{14.74 \times 10 \times 3.5}{61} = 8.45 \text{ cubic feet.}$$

The stroke of pump being assumed as $15''$ or 1.25 cubic feet, we get

$$\frac{8.45}{1.25} = 6.76 \text{ sq. feet}$$

or 973 sq. inches $= 35\frac{1}{4}$ diameter.

For a double-acting air-pump the same rule will apply, but the volume of steam for each stroke of the pump will be but one-half.

Should the pump be driven independently of the engine, then the relative speeds must be carefully considered. Table No. 18 is based on the above rule.

TABLE No. 18.

SIZE OF AIR-PUMPS.
Single Acting.
One Stroke of Pump per Rev. of Engine.

LOW PRESS. CYLINDER.	SIZE OF PUMP.		
	SINGLE EXP. ENGINES.	DOUBLE EXP. ENGINES.	TRIPLE EXP. ENGINES.
DIA. STROKE.	DIA. STROKE.	DIA. STROKE.	DIA. STROKE.
20″ x 36″	15½″ x 8″	15 ″ x 8″	14½″ x 8″
22″ x 36″	15¼″ x 10″	16½″ x 8″	16¼″ x 8″
24″ x 42″	17¾″ x 10″	17¼″ x 10″	16½″ x 10″
26″ x 42″	19½″ x 10″	18¾″ x 10″	18 ″ x 10″
28″ x 48″	22½″ x 10″	21½″ x 10″	20¾″ x 10″
30″ x 48″	21¾″ x 12″	22¾″ x 10″	22 ″ x 10″
32″ x 54″	24¾″ x 12″	24 ″ x 12″	23 ″ x 12″
34″ x 54″	26¼″ x 12″	25½″ x 12″	24½″ x 12″
36″ x 60″	29¼″ x 12″	28¼″ x 12″	27¼″ x 12″
38″ x 60″	31 ″ x 12″	30 ″ x 12″	28¾″ x 12″
40″ x 66″	34¾″ x 12″	33 ″ x 12″	31¾″ x 12″
44″ x 66″	33½″ x 15″	32½″ x 15″	34¾″ x 12″
48″ x 72″	38 ″ x 15″	37 ″ x 15″	35½″ x 15″
52″ x 72″	41½″ x 15″	40 ″ x 15″	38½″ x 15″
56″ x 72″	44½″ x 15″	43 ″ x 15″	42 ″ x 15″
60″ x 72″	47¾″ x 15″	46¼″ x 15″	44½″ x 15″
64″ x 72″	51 ″ x 15″	49½″ x 15″	47 ″ x 15″

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