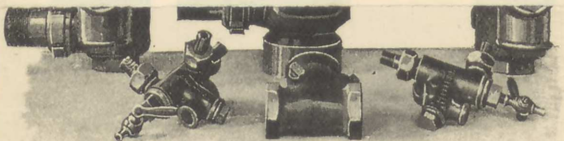
THE STEAM INJECTOR

W. A. B. HUGHES



HOT WATER INJECTORS QUOTED FOR SPECIALLY UPON APPLICATION.

WHITE'S SPECIALITIES.

Shewing Own Standard Designs in Injectors, Elevators, etc., also Interchange Types to Replace Standard Leading Patterns.

EXHAUST INJECTORS

LIVE STEAM INJECTORS

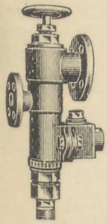


Fig. 1

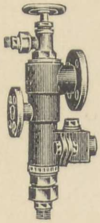


Fig. 2

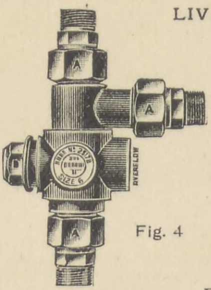


Fig. 4

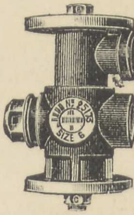


Fig. 5

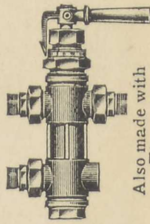

Also made with Flanges

Fig. 3

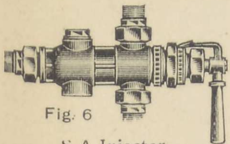


Fig. 6

S A Injector

Live Steam Heater


Fig. 15

Exhaust Heater

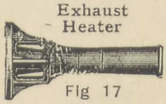


Fig 17

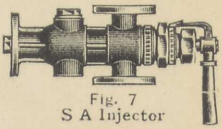


Fig. 7

S A Injector

ELEVATORS

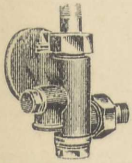

Traction Injector

Fig 8

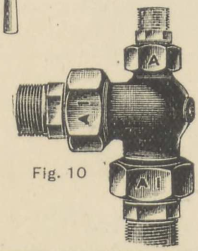


Fig. 10

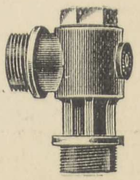


Fig. 10a

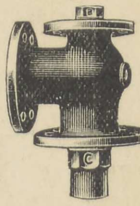


Fig. 11

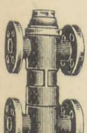

Self-Acting Injector

Fig. 26

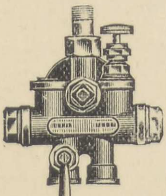

Steam Wagon Injector

Fig. 23a

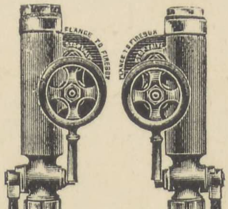

Locomotive Combination Injectors

Fig 22

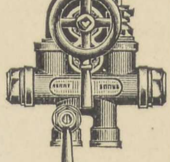

Steam Wagon Injector

Fig. 23c

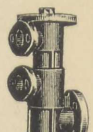

Self-Acting Injector

Fig. 29

Before purchasing any type or kind of Injector, Elevator, or Jet Appliance, be sure to write for Latest General Illustrated Catalogue to—

WHITE'S INJECTOR WORKS, Engineers,

PENDLETON, MANCHESTER, ENGLAND.

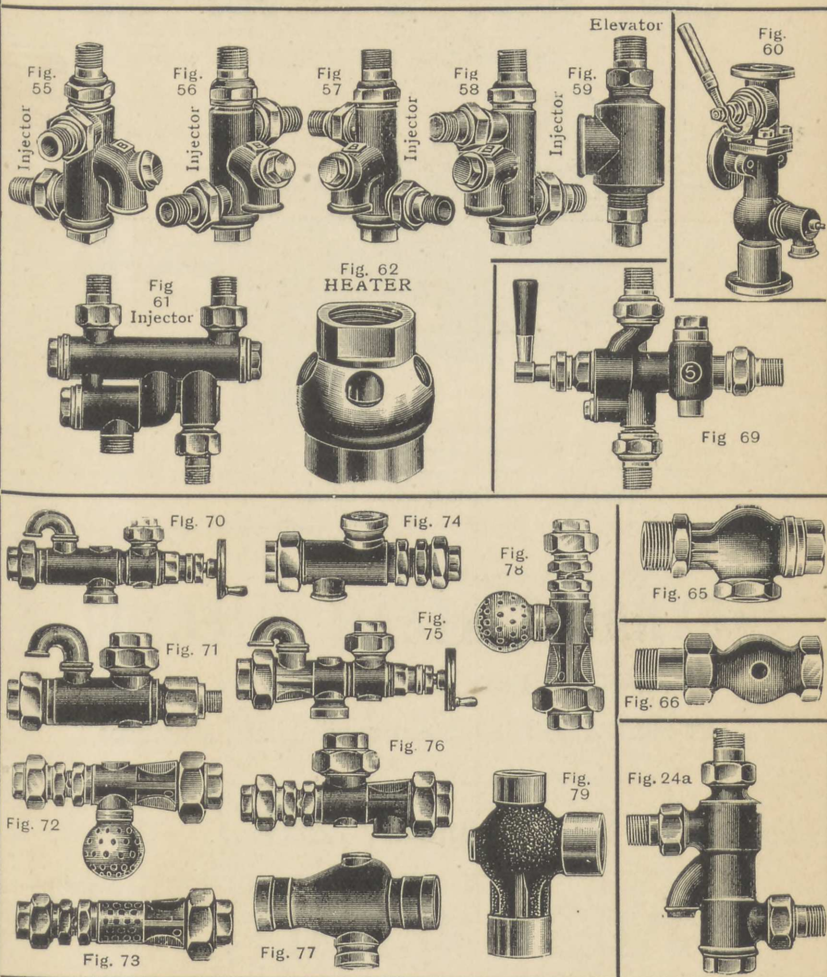
TELEPHONE: No. 58, PENDLETON.

TELEGRAMS: "WHITE'S," PENDLETON, MANCHESTER.

HOT WATER INJECTORS QUOTED FOR SPECIALLY UPON APPLICATION.

WHITE'S SPECIALITIES.

Shewing Interchange Types of Injectors, Elevators, etc., etc., to Exactly Replace Standard Leading Patterns.



Before purchasing any type or kind of Injector, Elevator, or Jet Appliance, be sure to write for Latest General Illustrated Catalogue to—

WHITE'S INJECTOR WORKS, Engineers,

PENDLETON, MANCHESTER, ENGLAND.

TELEPHONE: No. 58, PENDLETON. TELEGRAMS: "WHITE'S," PENDLETON, MANCHESTER.

The Steam Injector:

A THEORETICAL AND PRACTICAL TREATISE
ON THE DESIGN AND OPERATION
OF INJECTORS AND ON THE FLOW OF FLUIDS
THROUGH AND THE DESIGN OF NOZZLES.

BY

V. A. B. HUGHES.

1912.
THE TECHNICAL PUBLISHING COMPANY LIMITED,
55 & 56, Chancery Lane, London, W.C.

PREFACE

The construction of steam injectors has in recent years been brought to a high state of perfection, and the principles underlying the operation of the apparatus under various conditions have been more accurately understood.

The literature devoted to the subject of steam injectors is, with one or two exceptions, of a very perfunctory character; the few works that have been issued do not appear to constitute such a thorough and systematic treatise as will enable the reader to appreciate the reasons for the various details in the design of the apparatus. Thus, the overflow arrangements and also the water control arrangements of injectors have to a large extent been treated as if they played but a very insignificant part in determining the success of the apparatus, whereas the former are of the utmost importance, and are practically the sole features which, in the hands of the inventor and manufacturer, have rendered possible the excellent results now obtainable.

The treatment here adopted is in many respects thought to be original, and to open out the possibilities and probabilities of a difficult subject in a way not previously attempted.

The author has to acknowledge his great indebtedness to Mr. Edward C. R. Marks, A.M.I.C.E., M.I.M.E., for the facilities generously placed at his disposal for the preparation of the numerous drawings here reproduced, to Mr. J. D. Morgan, A.M.I.C.E., A.I.E.E., for kindly reading through the draft of the matter, and also to the numerous makers for the loan of blocks or the supply of particulars relative to their productions.

V. A. B. H.

Manchester, 1912.

CONTENTS.

THE STEAM INJECTOR.

CHAPTER I.

Introduction.

The appliance known to engineers as an injector may be defined as an instrument in which a jet of steam moving at high velocity is caused to carry a jet of water, with which it mingles and by which it is condensed, into a boiler or vessel under pressure.

The general principles of the injector are based upon certain physical laws expressed in the following simple formulæ:—

If H represents the head in feet equivalent of the fluid pressure in a boiler, and g the acceleration due to gravity (which may be taken as 32·2 ft. per second per second), then

$$V = \sqrt{2gH} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(i.)}$$

V being the theoretical velocity in feet per second of a jet of such fluid flowing from the boiler.

The fluid head equivalent of any pressure depends upon the density of the fluid, and may be expressed thus:—

$$\text{Head in feet} = \frac{\text{presssure in pounds per square foot}}{\text{weight per cubic foot}}.$$

It follows, therefore, that the velocities of discharge of different fluids under a given pressure (P) vary inversely as the square root of their densities, the relationship being expressed in the following formula:—

$$V = \sqrt{\frac{2gP}{w}} \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(ii.)*}$$

* It is not strictly accurate to use formula (ii.) for determining the velocity of discharge of an elastic fluid such as steam, as the formula takes no account of the velocity increase due to expansion during discharge. The formula is, however, convenient for determining the approximate relationship between steam and water jets issuing from the same vessel. The case for the steam jet is understated.

w being density in pounds per cubic foot, and P pressure in pounds per square foot.

[*Or if for pressure in pounds per square foot we substitute pressure in pounds per square inch, and take 2·3 ft. as the fluid head equivalent of 1 lb. pressure per square inch, then*

$$V = \sqrt{2 g p \times 2{\cdot}3} \quad . \quad . \quad . \quad \text{(ii. } a.\text{)}]$$

In other words, if steam and water jets issue from a boiler under pressure, then with s denoting the density of the steam and w the density of the water within the boiler,

$$\frac{\text{The velocity of steam jet}}{\text{The velocity of water jet}} = \sqrt{\frac{w}{s}}.$$

If the pipes through which the steam and water jets issue be of the same cross-sectional area, and the discharge in unit time be considered, then

$$\frac{\text{Mass of steam discharged}}{\text{Mass of water discharged}} = \sqrt{\frac{s}{w}};$$

therefore
$$\frac{\text{Momentum of steam jet}}{\text{Momentum of water jet}} = 1,$$

and
$$\frac{\text{Kinetic energy of steam jet}}{\text{Kinetic energy of water jet}} = \sqrt{\frac{w}{s}}.$$

Thus it will be seen that whilst under the conditions aforesaid the momentum of the steam jet is equal to that of the water jet, the velocity and kinetic energy of the former jet is to that of the latter as the square root of the density of the water is to the square root of the density of the steam. The density of steam being very much less than that of water under the same pressure, its velocity will therefore be greater. Even when the steam jet is mixed with and condensed by a water jet, the velocity and kinetic energy of the resultant jet may be very considerably greater than that of a water jet issuing from the same vessel as the steam jet. It will thus be seen that by utilising the kinetic energy of a jet of steam, a very effective force is obtained for carrying or injecting water into a boiler under a pressure equal to or even higher than that of the steam used.

If we consider the relationship between water and steam from a thermal standpoint by comparing the heat energy represented by 1 lb. of steam with that represented by 1 lb. of water at the same temperature and pressure as the steam, we at once appreciate the advantage of the steam over the water as a mechanical force. It is a portion of the superior force or excess of heat energy of the steam, represented by its latent heat, which is utilised in the injector for forcing water into the water space of the boiler in which the steam was generated.

To illustrate the foregoing statements, we will consider a jet of dry saturated steam issuing from a boiler at 10 lbs. gauge pressure, which mingles with, and is ultimately condensed by, a water jet under atmospheric pressure and a head of 2 ft., the ratio by weight of the mixture being 1 lb. of steam to 9 lbs. of water.* At the point where the steam and water unite a vacuum exists (due to the condensation of the steam), and we will assume this to be 20 in. mercury. The theoretical velocity of dry saturated steam at 10 lbs. gauge pressure issuing into a vacuum of 20 in. mercury (5 lbs. pressure per square inch absolute) is about 2,330 ft. per second, and the theoretical velocity of water under atmospheric pressure and a head of 2 ft. into the same vacuum is about 40 ft. per second. The resultant velocity of the combined jet (condensed steam and water) per unit mass (1 lb.) will therefore be 269 ft. per second.† A water jet discharging into a region at an absolute pressure of 5 lbs. per square inch with a velocity of 269 ft. per second would be

* In the rough calculations here given, the effect of the temperature of the water upon the results has been ignored, so as to make the figures as simple as possible. One pound pressure per square inch has been taken as equal to a water head of 2·3 ft. The values have been calculated by formulæ given later.

† For: Momentum of steam jet + momentum of water jet

$$= \text{momentum of combined jet}$$

$$1 \times 2330 + 9 \times 40 = 2690;$$

$$\textit{i.e.}, 10 \times \text{velocity} = 2690;$$

$$\text{therefore velocity} = 269 \text{ ft. per second.}$$

produced by a pressure of 479 lbs. per square inch.* That is to say, if we neglect all consideration of losses, a steam jet issuing from a boiler under a gauge pressure of 10 lbs. per square inch would be able (when mixed with and condensed by water) to carry nine times its own weight of water into a boiler under a gauge pressure of nearly 479 lbs. per square inch. It is hardly necessary to add that it is impossible in practice to obtain anything like this result.

The following particulars of a test upon an injector under conditions approaching those above set out will show something of the possibilities of the appliance:—

TEST OF AN INJECTOR.

Steam pressure. Pounds per square inch absolute.	Head of feed water.	Temperature of		Delivery pressure. Pounds per square inch absolute.
		Feed water.	Delivery water.	
	Feet.	Deg. Fah.	Deg. Fah.	
25	2	60	180	180
1,75 ata		15,5 °C	82 °C	12,6 ata

Briefly, an injector may be said to comprise three parts, performing the following functions:—

(1) A steam nozzle in which the pressure energy of the steam is converted into kinetic energy.

(2) A combining or mixing nozzle in which the steam mixes with and imparts its velocity to the feed water.

(3) A delivery nozzle in which the kinetic energy of the water jet issuing from the combining nozzle is converted into pressure energy.

Fig. 1 is a sectional illustration of an early type Giffard injector. The steam nozzle is shown at a, the water inlet

* For: Pressure above 5 lbs. per square inch absolute

$$= \frac{\text{velocity}^2}{2g \times 2{\cdot}3} - 10 \quad . \quad . \quad . \quad . \quad \text{[See formula (ii. } a\text{).]}$$

$$= \frac{269^2}{148} - 10$$

$$= 479 \text{ lbs. approximate.}$$

at *b*, the combining or mixing nozzle at *c*, and the delivery nozzle at *d*. The slot or gap *e* is to permit an overflow from the end of the nozzle *c*, when the injector is unable

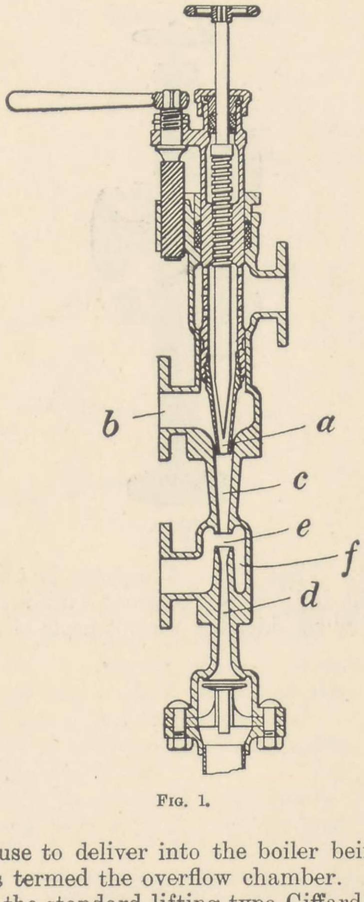


FIG. 1.

from any cause to deliver into the boiler being fed. The chamber *f* is termed the overflow chamber. Fig. 2 is an elevation of the standard lifting type Giffard injector.

The dimensions of an injector are always compared with the diameter of the throat (or section of smallest diameter) of its delivery nozzle. The size of an injector is generally reckoned by the diameter in millimetres of the delivery

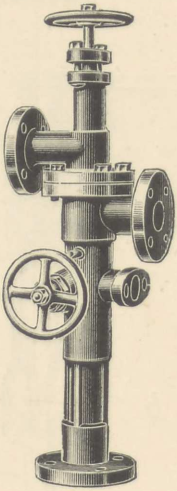


FIG. 2.

nozzle throat. Thus a No. 3 injector is one having the diameter of its delivery nozzle throat 3 millimetres (3 mm.).

We will now consider the various parts of an injector in detail.

CHAPTER II.

The Steam Nozzle.

As the kinetic energy of steam is the force utilised in an injector for propelling the feed water into the boiler or vessel being fed, it is necessary that the injector steam inlet nozzle be so designed as to develop under any given conditions the greatest kinetic energy per pound of steam discharged therefrom.

In the days of the early injectors the properties of steam were not thoroughly understood; engineers treated steam in the same manner as they would treat water or other inelastic fluid. To obtain the maximum velocity of discharge of water from a nozzle, the latter is made of convergent form; so in the early injectors a converging steam

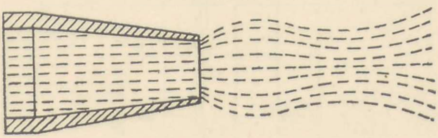


FIG. 3.

inlet nozzle was employed. In the Giffard injector illustrated at fig. 1 there is a converging steam inlet nozzle *a*. With this form of steam nozzle, as with a straight or cylindrical nozzle, the issuing steam is of greater pressure than the medium into which it flows, and expands laterally immediately it leaves the nozzle.

Fig. 3 shows a converging steam nozzle with a live steam jet issuing therefrom.*

Fig. 4 shows in diagram form the fall of pressure in a steam jet issuing from a plain tube, such as indicated at

* This figure must be taken as approximate only, as the exact form of the jet depends on conditions which vary with every nozzle. See Rosenhain's paper referred to on page 9.

the top of the figure.* The initial steam pressure was the same in all the tests, but the terminal pressure was varied in each test. The great pressure drop immediately beyond the tube will be noted; also the pressure oscillations before the exhaust or terminal pressure is reached.

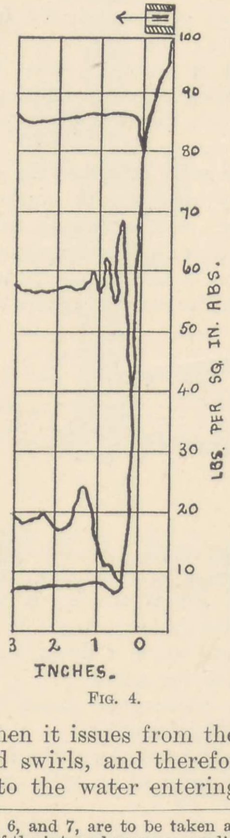


FIG. 4.

If the steam, when it issues from the steam nozzle, expands laterally and swirls, and therefore tends to give a lateral movement to the water entering an injector, it is

* The diagrams, figs. 4, 6, and 7, are to be taken as typical only. For a full treatment of the subject of the internal-pressure conditions of steam jets issuing through and from various forms of nozzle, Dr. Stodola's work on the "Steam Turbine" should be consulted.

obvious that it is not acting in the most efficient manner, for to fulfil the latter condition all the energy imparted to the feed water should be in a direction parallel with the axis of the steam nozzle. It will be understood that for maximum efficiency, eddy motion in the steam jet should be a minimum.

In 1869 it was proposed to make the injector steam inlet nozzle diverge towards its mouth or exit, so that the steam would expand within the nozzle down to the pressure of the medium into which it is flowing. The jet issuing from the nozzle is then of cylindrical form. The expansion of the steam within the limits of the nozzle ensures that the jet shall have maximum velocity as it leaves the same (for the lower the steam pressure at the nozzle mouth with a

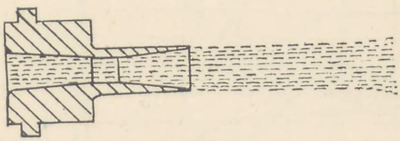


FIG. 5

given initial pressure the greater the velocity of discharge), and also that eddies in the jet shall be reduced to a minimum.

Fig. 5 shows an injector steam inlet nozzle with a diverging mouth piece, and a steam jet issuing therefrom.

In live steam injectors, the angle of divergence of the steam nozzle is generally between 5 deg. and 15 deg. If the angle of divergence be too great, eddies form in the jet and produce loss in velocity, whilst if it be too small the nozzle has to be of excessive length to provide for a proper degree of expansion of the steam issuing therethrough. Eddies commence to form in the steam jet if the angle of divergence exceeds 6 deg. In Rosenhain's experiments* to determine the most efficient form of nozzle for developing the greatest kinetic energy per pound of steam

* Proc. Inst. C.E., vol. cxl., page 199.

discharged, the best all-round results were given by a nozzle diverging at the rate of 1 in 12.

Fig. 6 shows in diagram form the fall of pressure within the diverging nozzle indicated at the top of the figure when the initial pressure is maintained constant, but the terminal pressure is varied. The absence of the violent oscillations observed in connection with fig. 4 will be noted.

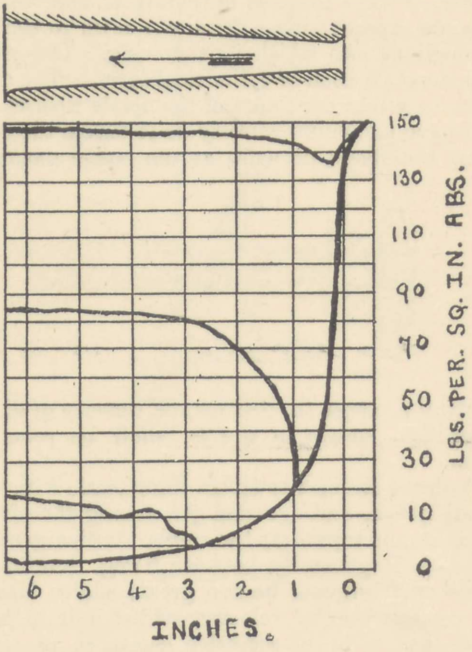


Fig. 6.

The effect of varying the pressure in the exhaust space beyond the diverging nozzle from that of the jet which would issue from the nozzle after full expansion therein, is precisely the same as in the case of the short tube shown at fig. 4. Fig. 7 indicates the pressure oscillations in the exhaust space. The pressure at the nozzle mouth was about atmospheric.

In general it may be said that if the pressure of a steam jet issuing from a nozzle be greater than that in the region into which it is discharging, lateral expansion of the jet takes place immediately beyond the nozzle, and pressure oscillations are set up in said region of lower pressure.

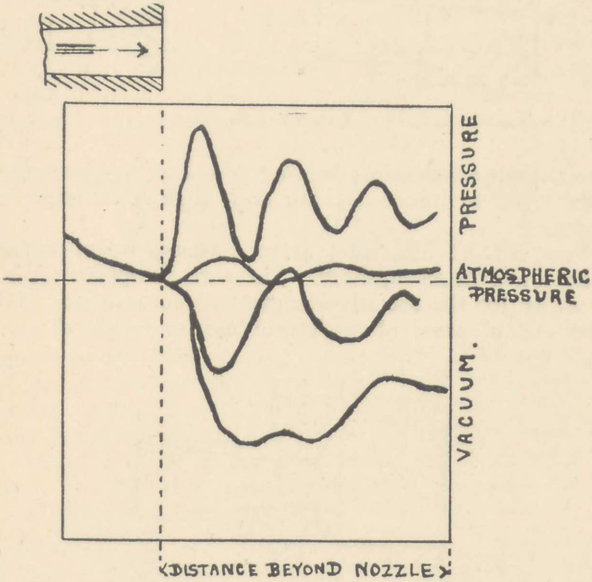
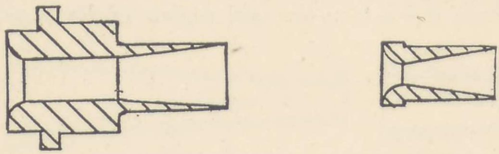


FIG. 7.

Such oscillations become a minimum or vanish if the pressure of the jet leaving the nozzle is the same as the exhaust pressure, or pressure in the exhaust space, whilst if the latter pressure is higher than that which the jet would have after full expansion in the nozzle, the pressure rises in the nozzle at a distance from the mouth depending upon the amount of the exhaust pressure.

To obtain the best results, the inlet edge of the steam nozzle should be slightly rounded, so as to obviate the setting up of eddy motions at the nozzle throat. If the inlet edge is excessively rounded, there is probably a choking action in the nozzle resulting in velocity loss. In



FIGS. 8 AND 9.

Rosenhain's experiments before referred to, the best results were given by nozzles having only slightly rounded inlet edges.

Figs. 8 to 11 illustrate various forms of steam nozzle as used in live steam injectors. There would appear to be no necessity for the excessive length and convergence of the inlet end of some of the steam nozzles employed in in-

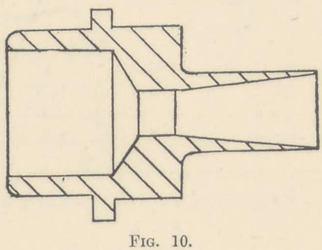


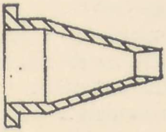
FIG. 10.

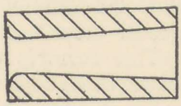
jectors* (except it be that a hexagon is required on the nozzle exterior). Fig. 12 shows the form of nozzle which gave the best all-round results in Rosenhain's experiments.

The velocity of discharge of the steam from the exit

* A nozzle having a well-rounded inlet discharges more steam in a given time than a nozzle having only a slightly-rounded inlet, and likewise a nozzle having the latter form of inlet discharges more steam than one having a sharp inlet.

end of a diverging steam nozzle depends upon the degree of expansion of the steam within the nozzle. If the steam is not expanded within the nozzle down to the pressure of the medium into which it is flowing, then it leaves the nozzle with some of its heat pressure energy unconverted into kinetic energy, and such energy will be absorbed by the feed water, which, in the case of the injector, surrounds the exit end of the steam nozzle. In this way, though the heat energy is not lost, yet it does not assist in increasing the velocity of discharge of the steam from the steam nozzle. On the other hand, if the steam is over-expanded within the steam nozzle, the velocity of the jet will be sacrificed in order to increase the jet's cross sectional area.* It has





FIGS. 11 AND 12.

been found by experiment in connection with the diverging nozzles used in certain types of steam turbine that the velocity losses due to slight over-expansion within the nozzle are greater than those due to slight under-expansion, so that it is better to have the nozzle too short than too long. In fact, it is recommended that the nozzle be designed for slight under-expansion.

The following table, in which A_m and A_t respectively indicate area of nozzle mouth and throat, and p_1 and p_2 boiler and exhaust steam pressure (absolute), gives sufficiently accurate values (for practical purposes) for the ratio of areas of nozzle mouth and throat for different ratios of initial and final steam pressures.

* The value of the ratio of the velocity of the jet at the point where the proper degree of expansion is attained to that of the jet at the mouth of the nozzle may be taken as approximately equal to the ratio of the cross-sectional area of the nozzle at the mouth to that of the nozzle at the said point of correct expansion ; in other words, $\frac{V_x}{V_m} = \frac{A_m}{A_x}$, where V denotes velocity, A area, m nozzle mouth, and x the section of the nozzle where correct expansion is attained.

TABLE I.—TABLE OF RATIOS OF AREAS OF NOZZLE MOUTH AND THROAT FOR DIFFERENT RATIOS OF INITIAL AND FINAL STEAM PRESSURES.

Values of								
$\frac{p_1}{p_2}$	1·732	4	8	10	20	50	70	100
$\frac{A_m}{A_t}$	1	1·35	2·07	2·436	3·966	7·98	11·55	13·8

It is generally held that the best form of the divergent part of a nozzle is slightly concave and not conical, as usually made, but owing to the expense of manufacture, such form is not commercially practicable.

All injector steam nozzles are made of circular cross section. This form is probably the most efficient, as it will not cause any eddy motion in the jet, and it is the most economical to manufacture; but provided a nozzle is well rounded at all parts, the losses due to eddy motion should be negligible.

To provide an injector steam nozzle of theoretically accurate proportions for any service, it would be necessary to have a fixed steam pressure, so as to obtain a uniform density of steam at the nozzle throat and a fixed counter-pressure or pressure at the exit end of the nozzle. Such conditions do not obtain in practice. The nozzle must therefore be designed to give a maximum efficiency at mean steam and counter-pressures. It may be best to have one form of nozzle for high-pressure and another for low-pressure service. The counter-pressure will vary only within very narrow limits.

If a steam nozzle is reasonably correct in design and is working within the limits of pressure for which it was designed, its efficiency as a means of converting pressure energy into kinetic energy may be as high as 95 per cent.

The total steam inlet area of an injector adapted to work with live steam of any usual boiler pressure varies from about 1·8 to 3 times the area of the throat of the delivery nozzle.* In exhaust injectors said ratio of areas is increased to about 16 to 1, on account of the lower density of the exhaust steam. These ratios are experimentally determined and vary according to the mean pressure of the steam dealt with, being greater for low than for high pressures. For example, if an injector be designed to work with a steam pressure of 160 lbs. to 200 lbs. per square inch, the area of the steam inlet nozzle throat may be about 1·96 times the area of the delivery nozzle throat; for 120 lbs. pressure steam the ratio of areas may be 2·56 to 1, and for 60 lbs. pressure steam 3·2 to 1.

The chief factors which have to be considered in deciding 1) upon the size of steam nozzle for any injector are three. In the first place, sufficient steam must be admitted to the injector to give the requisite velocity to the entering feed 2) water. Secondly, if the feed water is cold it will condense more steam per unit weight in a given time than if it is hot, but as condensation must be completed within the limits of the combining nozzle, the ratio of steam to water must be kept as low as possible when the feed water is 3) hot. Thirdly, if the injector has to lift its feed water, the amount of steam admitted must not be sufficient to cause choking within the combining nozzle before the feed water is drawn into the appliance.

As the greatest wear of the steam nozzle takes place at its throat, the latter is frequently made with a straight or cylindrical portion there (see figs. 5 and 10) for a length equal to from about one-third to the diameter of the throat.

It is very usual to divide the total steam inlet area of an injector into two parts, the one part being of annular and the other of circular cross section. Fig. 13 shows one such arrangement. The nozzle *a* is known as the "lifter" or lifting steam nozzle, and the nozzle *b* as the "forcer" or

---

* Formulæ are frequently given for determining the size of steam nozzle for any particular service. These formulæ are interesting, but useless, as they are based upon several assumptions as to ratio of steam to water, velocity of entering steam, velocity of entering feed water, etc, which it is impossible to make with any degree of accuracy. A similar formula has not, therefore, been given here. Later there will be given an approximate method of determining the sizes of the nozzles of an injector from the results of tests on the latter.

forcing steam nozzle. When the injector is lifting its feed water, the steam issuing from the nozzle *a* acts as an ejector to create a vacuum in the feed pipe, so as to lift the feed water and supply it at high velocity to the steam

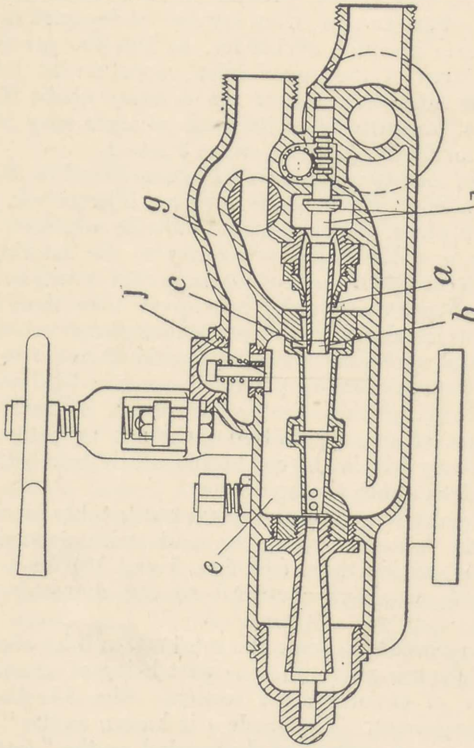


Fig. 13.

jet issuing from *b*. The area of the lifting nozzle is considerably less than that of the forcer. To enable the steam from the lifter to escape freely when the feed water is being lifted, a special overflow aperture as *c* is provided between the lifter and forcer.

A steam control valve such as *d*, fig. 13, is provided for the steam supplies to the lifter and forcer when the injector is to lift its feed water. The first opening movement of said valve allows steam to pass to the annular " lifter " *a*, whilst further movement allows steam also to pass to the " forcer " *b*.

The aforesaid arrangement of double steam nozzles is more powerful for lifting purposes than a single steam nozzle, though the latter is sufficient for most services.

Velocity of Discharge of Steam.

The general formula for obtaining the ideal or theoretical velocity of discharge of steam from an orifice in a vessel is as follows:—

$$V = \sqrt{2\,g\,U} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (\text{iii.})$$

where V denotes velocity in feet per second, g acceleration due to gravity, and U the net amount of work performed by unit weight of steam during admission to the discharging means at constant pressure, expansion to the exhaust pressure, and discharge at that pressure.

If the expansion is adiabatic, then U will be represented by the whole of the available heat energy or heat units between the temperatures of saturation corresponding to the initial and final steam pressures (or the admission and exhaust pressures) multiplied by the dynamical equivalent of a heat unit. That is,

$$U = J\left[\begin{matrix}\text{Heat supplied up to} \\ \text{point of expansion}\end{matrix} - \begin{matrix}\text{Heat rejected} \\ \text{at discharge}\end{matrix}\right] \quad (\text{iv.})$$

$$= J\,\{\,(t_1 + x_1\,L_1) - (t_2 + x_2\,L_2)\,\} \quad . \quad . \quad . \quad . \quad (\text{v.})$$

where J indicates Joule's dynamical equivalent of a heat unit = 778 foot-lbs. per B.T.U.

Equation (iii.) can be expanded to include (v.) when it becomes

$$V = \sqrt{2\,g\,J\,(t_1 + x_1\,L_1 - t_2 - x_2\,L_2)} \quad . \quad . \quad . \quad (\text{vi.})$$

where t_1 and t_2, x_1 and x_2, L_1 and L_2 indicate the thermometric temperatures, dryness fractions, and latent heats of the steam before expansion and at the exhaust pressure respectively.

The value of x_1 in the above formulæ having been obtained by means of a calorimeter, the value of x_2 is given by the following formula :—

$$x_2 = \frac{\frac{x_1 L_1}{T_1} + \phi_1 - \phi_2}{\frac{L_2}{T_2}} \quad . \ . \ . \ . \ . \ . \ . \quad \text{(vii.)}$$

where T_1 and T_2 = initial and final absolute temperatures, deg. Fah.;
L_1 and L_2 = initial and final latent heats of steam;
x_1 = dryness fraction of steam before expansion;
ϕ_1 and ϕ_2 = entropy of water at initial and final temperatures.

Application of Formula (v.).—To find the velocity of discharge of dry saturated steam issuing into the atmosphere from a boiler under an absolute pressure of 200 lbs. per square inch, we first proceed to find the value of x_2 by formula (vii.); x_1 is unity, as the steam is dry.*

$$x_2 = \frac{\frac{844{\cdot}6}{842{\cdot}7} + {\cdot}544 - {\cdot}313}{\frac{966}{673}}$$

$$= {\cdot}86 \text{ approximately.}$$

From formula (vi.) we get

$$V = \sqrt{64{\cdot}4 \times 778\,(381{\cdot}7 + 844{\cdot}6 - 212 - {\cdot}86 \times 966)}$$
$$= 3033 \text{ ft. per second.}$$

If we consider the pressure volume diagram, fig. 14, in which the area $P_1 V_1$ represents the work performed during the admission of unit weight (volume V_1) of steam at pressure P_1, and $P_2 V_2$ the work represented by the steam (at

* The effect of initial wetness of the steam is to reduce the velocity of discharge.

pressure P_2 and of volume V_2) after discharge is completed, and A B is the expansion curve, then the value of U in the equation III. is given by the shaded area of the figure.

The adiabatic expansion curve for steam is a particular case of the general formula $P_1 V_1^n$ = constant (where P denotes pressure in pounds per square foot, V volume in

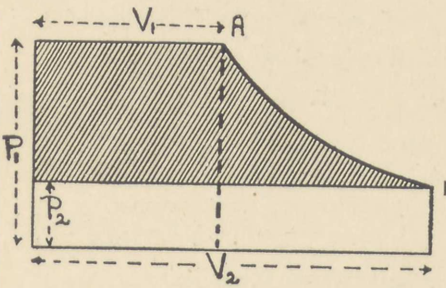


FIG. 14.

pounds per cubic foot of fluid at pressure P_1 and n the coefficient of expansion).* The area bounded by the expansion curve is obtained thus:—

$$\text{area} = \int_{V_1}^{V_2} P\,dV;$$

$$\text{but } PV^n = P_1 V_1^n;$$

$$\therefore\ P = \frac{P_1 V_1^n}{V^n},$$

$$\text{area} = \int_{V_1}^{V_2} \frac{P_1 V_1^n}{V^n}\,dV$$

$$= \frac{P_1 V_1 - P_2 V_2}{n - 1}.$$

$* = \dfrac{\text{Specific heat at constant pressure}}{\text{Specific heat at constant volume}}.$

The value of U is therefore

$$= P_1 V_1 + \frac{P_1 V_1 - P_2 V_2}{n - 1} - P_2 V_2$$

$$= \left\{ \frac{n}{n-1} (P_1 V_1 - P_2 V_2) \right\}$$

$$= \left\{ \frac{n}{n-1} P_1 V_1 - \left(1 - \frac{P_2 V_2}{P_1 V_1}\right) \right\}$$

But since $P_1 V_1^n = P_2 V_2^n$,

$$\therefore \frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$$

$$i.e., \; U = \frac{n}{n-1} \times P_1 V_1 \left\{ 1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right\} \quad . \quad . \quad . \quad \text{(viii.)}$$

$$\therefore \; V = \sqrt{\left(2g \times \frac{n}{n-1} \times P_1 V_1 \left\{ 1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right\}\right)} \quad . \quad \text{(ix.)}$$

Substituting 144 p for P, we get

$$V = \sqrt{2g \times \frac{n}{n-1} \times 144 \times p_1 v_1 \left\{ 1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \right\}} \; . \; \text{(x.)}$$

where V = velocity of discharge in feet per second;
p_1, p_2 = initial and final absolute pressures in pounds per square inch;
v_1 = volume in cubic feet of 1 lb. of fluid at pressure p_1;
n = coefficient of expansion = 1·135 for dry saturated steam, and 1·3 approximately for superheated steam.

This gives for the adiabatic expansion of dry saturated steam

$$V = 279 \sqrt{p_1 v_1 \left\{ 1 - \left(\frac{p_2}{p_1}\right)^{0\cdot12} \right\}} \quad . \quad . \quad . \quad \text{(xi.)}$$

Application of Formula (xi.).—To calculate the velocity of discharge of dry saturated steam issuing into the atmosphere from a boiler under an absolute pressure of 200 lbs. per square inch,

$$V = 279 \left(200 \times 2\cdot26 \left\{ 1 - \left(\frac{15}{200}\right)^{0\cdot12} \right\} \right)$$

$$= 3066 \text{ ft. per second.}$$

The following table gives the approximate theoretical velocity of dry saturated steam into the atmosphere and into a vacuum of 28 in. mercury. It will be noted that the velocity does not increase at anything like the rate of increase of the pressures. When 150 lbs. pressure is reached, the rate of velocity increase is very slow. The great value of the vacuum will be noted, and also the high velocity of even exhaust steam into a vacuum:—

TABLE II.

Pressure of steam. Pounds per square inch absolute.	Velocity in feet per second into atmosphere.	Velocity in feet per second into vacuum of 28 in. mercury.
10	..	2677
15	..	2900
30	1602	3263
50	2108	3510
75	2425	3671
100	2645	3804
125	2777	3905
150	2900	3983
175	3000	4050
200	3075	4100
250	3207	4190

WEIGHT FLOW OF STEAM.

The weight of steam discharged in unit time is proportional to the area of the orifice through which it is discharging and to the velocity of discharge, and inversely proportional to the volume of unit weight of the steam at the orifice. That is,

$$W = \frac{A \times V}{v_2}.*$$

* The dryness fraction of the steam is here neglected. For greater accuracy, for v_2 should be substituted $x_2 v_2 + (1 - x_2)\,\sigma$ where σ = specific volume of 1 lb. of water.

Now
$$p_1 v_1^n = p_2 v_2^n,$$
$$\therefore\ v_2 = v_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}$$

Substituting for V the value given in formula (x.), we get

$$W = \frac{A\,V}{v_2} = A\frac{\sqrt{2\,g \times 144 \times \frac{n}{n-1} \times p_1 v_1 \left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\}}}{v_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}}$$

$$= \frac{A}{v_1}\sqrt{2\,g \times 144 \times \frac{n}{n-1} \times p_1 v_1 \left\{\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_1}{p_2}\right)^{\frac{n+1}{n}}\right\}} \text{ (xii.)}$$

This expression is a maximum for a given initial pressure p_1, when

$$\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}} \text{ is a maximum ;}$$

that is (by differentiating and equating to zero), when

$$\frac{2}{n}\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}-1} - \frac{n+1}{n}\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} = 0,$$

or
$$\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \quad . \quad . \quad . \quad . \quad \text{(xiii.)}$$

For the adiabatic expansion of dry saturated steam, $n = 1{\cdot}135$; W is then a maximum when $p_2 = {\cdot}577\ p_1$.

Other values of the ratio $\frac{p_2}{p_1}$, for different known values of n, are given in the following table :—

TABLE III.

Values of n.	Values of $\frac{p_2}{p_1}$.
1·4	·528
1·3	·546
$\frac{10}{9}$	·582
1	·6

The above results have been verified by experiment. It is found in the case of the flow of dry saturated steam through a nozzle that if the initial pressure (p_1) of the steam be maintained constant whilst the counter-pressure (p_2) is varied, the weight of steam discharged in a given time remains practically constant until the counter-pressure rises to 0·58 of the initial pressure. There is always a certain section of the steam nozzle at or about its throat at which the steam pressure remains at 0·58 of the initial or boiler pressure until the counter-pressure exceeds this amount. If reference be made to figs. 4 and 6, it will be seen how reluctant the pressure at the nozzle throat is to be influenced by varying counter-pressures.

If the values of $\frac{p_2}{p_1}$ given in equation (xiii.) be substituted in equation (xii.), the weight of steam discharged per second, under conditions of maximum weight flow, is given as follows:—

$$W = \frac{A}{v_1}\left(\frac{2}{n+1}\right)^{\frac{1}{n-1}} \sqrt{2g \times \frac{n}{n+1} \times 144\, p_1 v_1}$$

$$= A\left(\frac{2}{n+1}\right)^{\frac{1}{n-1}} \sqrt{2g \times \frac{n}{n+1} \times 144 \times \frac{p_1}{v_1}} \quad . \quad \text{(xiv.)}$$

where W denotes pounds of steam discharged per second, A area of nozzle throat in square feet, n the co-efficient of expansion, p_1 the initial or boiler pressure in pounds per square inch absolute, and v_1 the volume in cubic feet of 1 lb. of steam at pressure p_1.

A simple approximate formula for obtaining the weight of steam discharged per second through an orifice under maximum weight flow conditions is:—

$$W = \frac{p \times A}{70} \text{ (Napier)*} \quad . \quad . \quad . \quad . \quad \text{(xv.)}$$

where W = weight of steam discharged in pounds per second,

p = absolute pressure of boiler steam in pounds per square inch,

and A = area of nozzle throat in square inches.

* A more correct approximation is $W = \cdot01654\, A \times p^{\cdot9696}$.

Application of Formula (xv.).—To calculate the quantity of steam discharged per second into the atmosphere from a boiler under an absolute pressure of 100 lbs. per square inch through an orifice of 2 square inches cross-sectional area. As the exhaust or counter-pressure is less than ·58 of the boiler pressure, there will be a maximum weight flow through the orifice.

$$W = \frac{100 \times 2}{70}$$

$$= 2{\cdot}86 \text{ pounds per second.}$$

By the aid of formulæ (xiv.) and (xv.), the area of an orifice for discharging a known quantity of steam in a given time can be calculated.

If the condition for maximum weight flow does not exist, then to calculate the weight of steam discharged in a given time, the following approximate formula must be used:—

$$W = \frac{A \times V}{v_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(xvi.)}$$

where W denotes weight in pounds of steam discharged per second, A area of orifice or of nozzle throat in square feet, V the velocity of the steam in feet per second as it passes the nozzle throat, and v_2 the volume in cubic feet of 1 lb. of steam at the pressure in the exhaust space or space into which the steam is flowing.* The dryness fraction is neglected.

Application of Formula (xvi.).—To calculate the weight of steam discharged per second through an orifice of 1 square foot cross-sectional area, when the initial pressure is 100 lbs. per square inch absolute, and the final or exhaust pressure is 80 lbs. per square inch absolute. We first calculate the velocity of discharge. Using formula (xi.), we get

$$V = 279 \sqrt{100 \times 4{\cdot}33 \left\{ 1 - \left(\frac{80}{100}\right)^{0{\cdot}12} \right\}}$$

$$= 944 \text{ ft. per second;}$$

* The value v_2 should be multiplied by the dryness fraction x_2 for greater accuracy.

$$v_2 = 5{\cdot}35 \text{ (from "Steam Tables");}$$

$$\therefore\ W = \frac{1 \times 944}{5{\cdot}35}$$

$$= 176{\cdot}4 \text{ lbs. per second.}$$

If the value of $\frac{p_2}{p_1}$, given in equation (xiii.), be substituted in equation (x), the velocity of the steam at the nozzle throat under maximum weight flow conditions is obtained as follows:—

$$V_{\text{throat}} = \sqrt{2g \times \frac{n}{n+1} \times 144 \times p_1 v_1} \quad . \quad \text{(xvii.)}$$

It is perhaps hardly necessary to point out that whilst the velocity of the steam at the nozzle throat does not vary with variations in the pressure at the nozzle exit until the condition for maximum weight flow ceases to exist, the velocity of the steam at the nozzle exit varies with every variation in the pressure at the said exit, as will be readily appreciated from the velocity formulæ already given.

Design of Steam Nozzles.

To determine theoretically the relationship between the areas of nozzle throat and mouth for correct expansion from any one pressure to a lower pressure, the conditions of the steam jet as to velocity, density, and dryness at the said points should be compared.

If A denotes area, V velocity, v volume, the suffixes m and t nozzle mouth and throat respectively, and x the dryness fraction of the steam, then the following will be an approximate formula for calculating the said ratio of areas:—

$$\frac{A_m}{A_t} = \frac{V_t}{V_m} \times \frac{v_m}{v_t} \times \frac{x_m}{x_t} \quad . \quad . \quad . \quad . \quad \text{(xviii.)}$$

Application of Formula (xviii.).—To calculate the ratio of areas of the throat and mouth of a diverging nozzle in order to expand dry saturated steam of 100 lbs. pressure per square inch absolute down to 1 lb. absolute pressure.

The pressure at the nozzle throat will be ·58 of the initial pressure, or 58 lbs. per square inch absolute. The velocity of the jet at the throat and at the nozzle mouth after full expansion can be calculated by formula (xi.) and the dryness fractions by formula (vii.). We then get

$$\frac{A_m}{A_t} = \frac{1553}{3918} \times \frac{330·36}{7·25} \times \frac{·7894}{·9649}$$
$$= 14·7 \text{ approximately.}$$

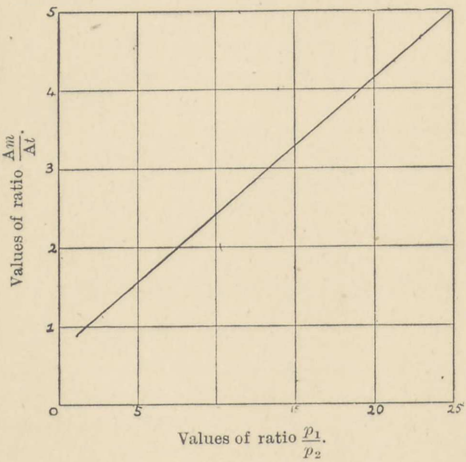


FIG. 15.

The above calculations are somewhat laborious. The following formula, by Zeuner, gives sufficiently correct results for practical purposes :—

$$\frac{A_m}{A_t} = \frac{·155}{\sqrt{\left(\frac{p_2}{p_1}\right)^{1·762} - \left(\frac{p_2}{p_1}\right)^{1·881}}} \quad . \quad . \quad . \quad \text{(xix.)}$$

where A_m = area of nozzle mouth,
A_t = area of nozzle throat,
p_1 = boiler pressure (absolute),
p_2 = exhaust pressure (absolute).

A more simple approximate formula is—

$$\frac{A_m}{A_t} = \cdot 172 \left(\frac{p_1}{p_2}\right) + 0\cdot 7 \quad . \quad . \quad . \quad . \quad . \quad \text{(xx.)}$$

This formula is used for values of $\frac{p_1}{p_2}$, not greater than 25.

For higher values of this ratio the formula is given as

$$\frac{A_m}{A_t} = \cdot 172 \left(\frac{p_1}{p_2}\right)^{\cdot 94} + 0\cdot 7 \quad . \quad . \quad . \quad . \quad \text{(xxi.)}$$

The symbols in the last two formulæ have the same signification as in formula (xix.).

The curve, fig. 15, has been plotted from formula (xx.).

CHAPTER III.

The Combining Nozzle.

The nozzle into which the steam and water issue after they have entered the injector is called the combining or mixing nozzle, for in it the steam and water combine or mix, with the result that the steam is condensed and a water jet alone pases through the nozzle exit. It is impossible to treat of the combining nozzle with anything like mathematical precision, for there are so many different quantities to be considered in relation thereto, and none of them has a fixed value.

In the design of this nozzle, particular regard must be given to the following considerations:—

(*a*) The length of the nozzle must be such as to ensure the complete condensation of the steam jet, so that water alone passes through the nozzle exit.

(*b*) Its walls must converge at a rate corresponding with the rate of increase of density of the jet passing through the nozzle; that is to say, the jet must fill, and only fill, the nozzle at all points in its length.

(*c*) The inlet end of the nozzle must provide a guiding and supporting wall for the water as it enters the injector, and during the impact of the steam thereon.

(The question of the overflow and water control arrangements of the combining nozzle are dealt with in Chapters IV. and V.)

In the consideration of the above points, the following should be borne in mind. In the first place, the volume of 1 lb. of steam varies with the pressure of the steam, as does also the temperature. Secondly, the rate at which steam of a certain temperature can be condensed depends upon the temperature of the water by which condensation is being effected. Thus it would appear that a special length and shape of nozzle is necessary to suit different steam pressures and temperatures, and also different feed-water temperatures. Further, if the proportion of steam to water is increased or diminished, the cross-sectional area of the combined jet will be altered, as may also the rate of condensation of the steam. The velocity of the incoming steam and also of the water may vary either simultaneously or independently. The boiler or counter-pressure against which delivery is being effected may also alter. These points must affect any decision as to the best form of nozzle to be employed.

Thus to set the variable quantities out in full, they are :—

(i.) Density of steam.
(ii.) Temperature of steam.
(iii.) Temperature of feed water.
(iv.) Ratio of steam to water.
(v.) Velocity of incoming steam.
(vi.) Velocity of incoming feed water.
(vii.) Pressure in vessel being fed.

It is obviously impossible to design any one nozzle to meet, with the greatest efficiency, all the conditions which arise even under the most favourable circumstances. The nozzle must therefore be designed to suit certain conditions which are a mean between the extremes likely to occur in service; but this is not easily done. The difficulty in deciding upon the best proportions for the combining nozzle is evidenced by the variety found in the appliances of high-class manufacturers.

Thus, combining nozzles of live steam injectors vary in length from about 9 to 24 times the diameter of the delivery nozzle throat, this measurement being taken between the exit of the steam nozzle or, if two such nozzles are employed (see fig. 1 of the forcing steam nozzle and the throat of the delivery nozzle. The distance between the exits of the lifting and the forcing steam nozzles, which also forms a portion of the combining or mixing area of the injector, varies between four to eight times the diameter of the throat of the delivery nozzle.

For what is known as "hot-water" injectors, that is, injectors designed to deal with high-pressure steam and hot feed water, the length between the forcing steam nozzle exit and the throat of the delivery nozzle is generally from 15 to 24 times the diameter of the throat of the delivery nozzle. It will be understood that the hotter the feed water and the higher the temperature of the steam, the longer will it take to effect complete condensation of the steam.

It would appear to be possible to vary the length of the combining nozzle within fairly wide limits without very materially affecting the efficiency of the injector. Injectors have been tried with the length of the combining nozzle only four times the diameter of the delivery nozzle throat. The jet from the combining nozzle will, however, be given greater stability if passed through a nozzle of

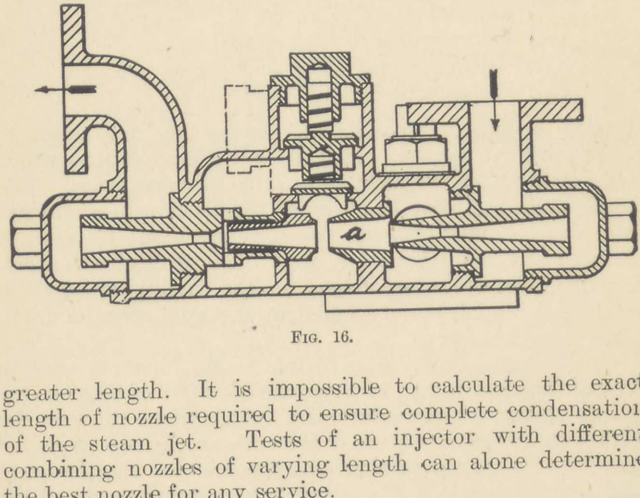


FIG. 16.

greater length. It is impossible to calculate the exact length of nozzle required to ensure complete condensation of the steam jet. Tests of an injector with different combining nozzles of varying length can alone determine the best nozzle for any service.

The rate of decrease in the diameter of the combining nozzle towards its exit end is in most injectors uniform; that is, the difference between the diameters at any two equidistant sections is the same, the nozzle thus being of conical form. In some cases, however, the rate of decrease is more rapid at first, as the condensation of the steam will be more rapid when it first meets the feed water. Thus combining nozzles are made with a taper varying from 15 deg. at the inlet end to 5 deg. at the outlet end.

The combining nozzle is subjected to considerable wear if the feed water contains solid matter in suspension. This is especially marked at the point where the steam first strikes the water and drives the solid matter against the sides of the nozzle. The effect is to groove the nozzle and change its form. The first or inlet portion of the combining nozzle is therefore frequently made separate from the remainder so that it can be renewed when necessary. Such separate inlet portion is termed the "water nozzle," "draft tube," or "lifting cone." This form of combining nozzle is illustrated in fig. 16, where *a* indicates the "draft tube."

The combining nozzle has a long life if the feed water is pure. Hence the advantage of carefully filtering the water passing to the injector.

CHAPTER IV.

The Delivery Nozzle.

The jet of water which leaves the combining or mixing nozzle of an injector is travelling at a high velocity—a velocity higher than that with which a water jet would issue from the boiler being fed—but merely to direct the high velocity jet against the water within the boiler would be a very inefficient method of utilising that jet, for the latter would strike the mass of water in the boiler with great violence, and its energy be to a large extent dissipated in the form of eddies. This effect would be increased in proportion to the increase in the velocities of the water tending to enter and that tending to leave the boiler.

The object which must be had in view in the treatment of the jet leaving the combining nozzle is to reduce its velocity and increase its pressure till the latter exceeds that within the boiler being fed. Such object is attained by causing the jet leaving the converging combining nozzle to pass through a diverging delivery nozzle.

The diverging nozzle has received a degree of attention, apart altogether from its connection with the subject of injectors, on account of the fact that by its aid the quantity of water discharged through an orifice in a given time can be increased, the theoretical rate of increase being in the ratio of the cross-sectional area at the diverging nozzle mouth or outlet to the cross-sectional area at the diverging nozzle throat, if the jet is able to fill the diverging nozzle completely. Many investigations have been made on the best form or taper of nozzle to give a maximum rate of delivery from a tank or vessel being emptied.

In connection with injectors, it will be readily understood that the water passing through the delivery nozzle must completely fill the latter, due to the pressure against which delivery is being effected resisting the free flow through the nozzle, and that if said nozzle be cut through at several points the same weight of water will pass each section in a given time (assuming the water density to remain constant), notwithstanding variations in the areas at the respective sections; that is to say, the velocity of the water jet multiplied by its cross-sectional area is a constant for all sections. It follows, therefore, that velocity and area vary inversely, and since the area changes as the square of the diameters at the various sections, the velocity will change inversely as the square of the diameters.

The velocity at any section of the delivery nozzle is obtained from the equation—

Velocity in feet per second at any section

$$= \frac{\text{volume in cubic feet passing per second}}{\text{area of section in square feet}};$$

or, since $d_1^2 \mathrm{V}_1 = d_2^2 \mathrm{V}_2 = \text{constant}$,

we may write $$\mathrm{V}_2 = \frac{d_1^2 \mathrm{V}_1}{d_2^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(xxii.)}$$

where d_1 = diameter at nozzle inlet,

d_2 = diameter at any section 2 of nozzle,

V_1 = velocity at nozzle inlet,

and V_2 = velocity at any section 2 of nozzle.

By the aid of formula (xxii.) a velocity curve can be plotted for any known form of nozzle if the velocity of the water jet as it enters the nozzle is known.

If the diameter d_1 of, and also the velocity V_1 at, the inlet end of the diverging nozzle are known, then the diameter d_2, which corresponds with the velocity V_2 at any section, is given by the following formula—

$$d_2 = \sqrt{\frac{d_1^2 \mathrm{V}_1}{\mathrm{V}_2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(xxiii.)}$$

This formula enables a nozzle to be designed to give a known velocity curve for the jet passing therethrough.

The pressures at any two sections of the diverging nozzle vary with the squares of the velocities thereat. If the velocity V_1 at the entrance to the nozzle is known, and also the velocity V_2 at any section 2 of the same, then the difference between the squares of these velocities plus the fluid head H_1 at the entrance to the nozzle represents the head at section 2. That is,

Head at section 2

$$= \frac{V_1^2 - V_2^2}{2g} + H_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(xxiv.)}$$

or, as head in feet

$$= \frac{\text{pressure in pounds per square foot}}{\text{weight per cubic foot}},$$

we may write

$$\frac{P_2}{w} = \frac{V_1^2 - V_2^2}{2g} + \frac{P_1}{w}.$$

If we take pressure in pounds per square inch

$$= \frac{\text{head in feet}}{2{\cdot}3},$$

then p_2, or pressure at section 2

$$= \frac{V_1^2 - V_2^2}{2g \times 2{\cdot}3} + \frac{H_1}{2{\cdot}3} \quad . \quad . \quad . \quad . \quad \text{(xxv.)}$$

The pressure of the jet at the entrance to the diverging delivery nozzle varies with the temperature of the jet. As the mixing of the steam and water is assumed to be completed in the combining nozzle, the temperature of the jet entering the delivery nozzle may be taken as the same as that of the water delivered from the injector.

The diagram, fig. 17, shows the diverging portion of an injector delivery nozzle so proportioned as to cause a uniform retardation in the motion of a jet passing therethrough; that is to say, the difference between the velocities of the jet at any two equidistant sections is the same. A velocity curve for an initial velocity of 200 ft. per second, and the corresponding pressure curve (the initial pressure being for convenience considered as zero), are plotted at the upper portion of the figure.

The diameters of the various indicated sections of the nozzle are as follows :—

Section 1	....	Diameter	1 or unity.
,, 2	....	,,	1·069.
,, 3	....	,,	1·154.
,, 4	....	,,	1·265.
,, 5	....	,,	1·415.
,, 6	....	,,	1·632.
,, 7	....	,,	2·000.
,, 8	....	,,	2·828.
,, 9	....	,,	4·472.

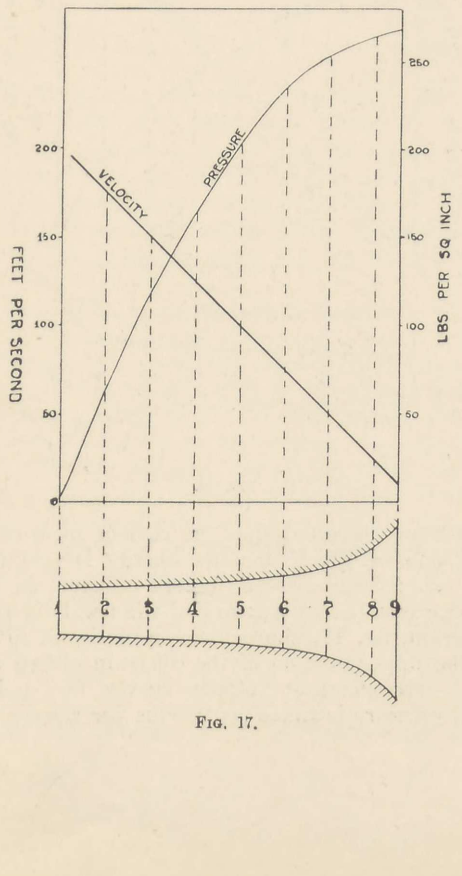


FIG. 17.

It will be noted that the increase in diameter of the nozzle is at a fairly uniform rate up to section 5 and then becomes more rapid, reaching the maximum at the exit of the nozzle. The pressure curve rises very abruptly right from the throat or inlet end of the nozzle.

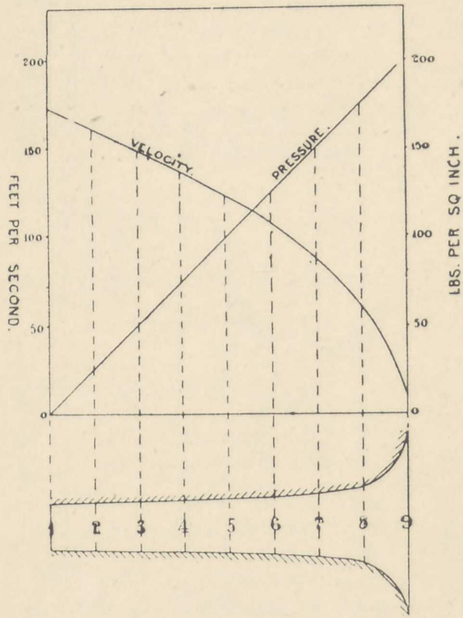


FIG. 18.

It has been suggested that the correct form of delivery nozzle is one in which pressure energy is produced at a uniform rate; that is, the difference between the pressures at any two equidistant sections of the nozzle is the same. The diagram, fig. 18, shows a nozzle designed to give this result, the upper portion of the diagram giving the pressure and corresponding velocity curves for such nozzle. The final pressure is taken as 200 lbs per square inch and the initial pressure as zero. The diameters of the nozzle at the different equidistant sections are as follows:—

Section	1	Diameter	1 or unity.
,,	2	,,	1·034.
,,	3	,,	1·074.
,,	4	,,	1·124.
,,	5	,,	1·188.
,,	6	,,	1·276.
,,	7	,,	1·411.
,,	8	,,	1·673.
,,	9	,,	4·152.

It will be seen that the rate of increase in the diameter of the nozzle is very slow right up to section 8.

It is found in practice that if the jet contain any solid matter, abrasion of the delivery nozzle takes place chiefly around the throat or first portion of the latter. It is usual practice to make the delivery nozzle of cylindrical form at the throat for a length equal to the diameter of the throat, as shown at fig. 19.

If the form of the nozzle be such as to cause a very sudden increase in the pressure of the jet at a point near to the throat of the nozzle where the velocity is high, the jet sets up a strong abrading action on, and tends to groove, the nozzle surface at that point. In this respect the form of nozzle shown at fig. 18 is better than that shown at fig. 17. Towards the delivery end of the nozzle the velocity of the jet is small, and wear is reduced to a minimum.

In most injectors the delivery nozzle has a plain, conical throughway aperture, as this form of nozzle lends itself to economical manufacture. A nozzle of this type is illustrated at fig. 20, which also gives corresponding velocity and pressure curves when the entering velocity is taken as 200 ft. and also as 100 ft. per second. It will be seen that with this type of nozzle there is at first a rapid rise and then a slow rise of pressure which will result in the production of eddying motions in the jet between the sections 1 and 4, causing loss in velocity and probably grooving of the nozzle. The angle of divergence of the walls of conical delivery nozzles is usually about 5 deg. At the mouth or exit the nozzle is rounded, as shown in fig. 20.

Fig. 21 shows a design of delivery nozzle as found in some injectors, the upper portion of the figure giving velocity and pressure curves for the nozzle. It will be seen that the pressure curve is of very even form.

Makers are not in agreement as to the best length and form of nozzle to employ to effect the required conversion of kinetic into pressure energy. The length of the delivery nozzle varies in different injectors from 10 to 16 times the diameter of the nozzle throat.

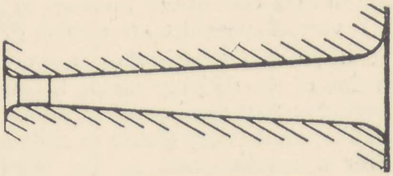


FIG. 19.

The correct proportioning of a diverging nozzle, to reduce the velocity of the jet passing therethrough to a certain amount, depends upon the velocity of the jet as it enters the nozzle. In the case illustrated at fig. 17, where a velocity curve has been plotted for an initial velocity of 100 ft. per second, the final velocity is 5 ft. per second; but when the entering jet travels at 200 ft. per second the final velocity is 10 ft. per second. It would appear, therefore, that different designs of nozzle should be employed for high and low jet velocities in order to obtain the most efficient action; otherwise if the nozzle be designed for a low velocity jet, and a high velocity jet be employed, the latter will leave the delivery nozzle with an unnecessarily large amount of its kinetic energy unconverted into pressure energy. The point is, however, of more importance from a theoretical than from a practical standpoint, as even with such widely different initial velocities as 100 ft. and 200 ft. per second the difference in the final velocities is only 5 ft. per second.

Experiments on injectors with delivery nozzles of various forms clearly demonstrate the importance of having the

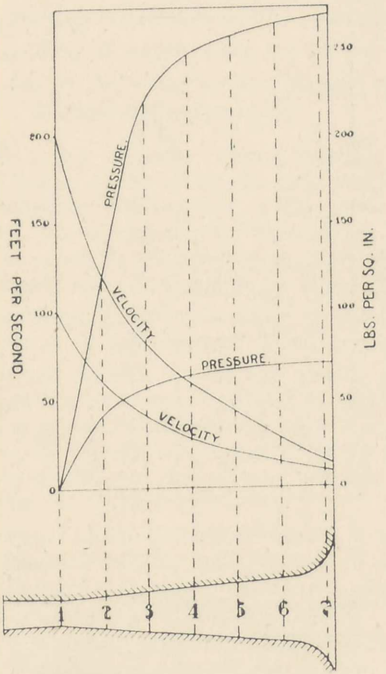


FIG. 20.

angle of divergence or the taper of said nozzles neither too great nor too small. A 5-deg. taper is a very satisfactory one. If no delivery nozzle whatever is employed, the maximum delivery pressure obtainable with the injector is very greatly reduced.

The delivery nozzle of an injector is of the greatest importance, as the diameter of its throat or portion of smallest cross-sectional area is the size by which the capacity of the injector is determined and with which the other dimensions of the injector are compared. The quan-

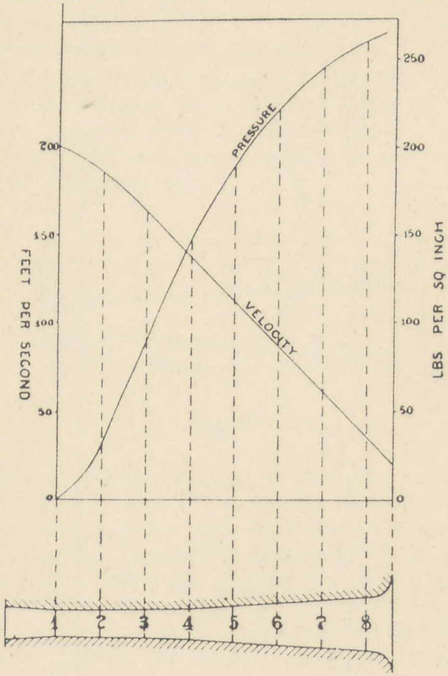


FIG. 21.

tity of water delivered by two injectors under similar conditions varies with the square of the diameter of the throat of the delivery nozzle.

The area of the delivery nozzle throat to pass any particular quantity of water in a given time depends upon the velocity and density of the water as it passes the throat; that is,

$$A = \frac{Q}{V \times D} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(xxvi.)}$$

where A = area in square feet,

Q = quantity of delivery water in pounds per second,

V = velocity of water in feet per second,

and D = density of water in pounds per cubic foot at the delivery temperature.

The actual weight of water passing through the delivery nozzle is, of course, made up by the total weights of steam and water entering the injector. The water velocity at the delivery nozzle throat may for the purpose of the above calculation be taken as 25 per cent greater than that with which a jet would issue from the boiler being fed.

From the foregoing the following general conclusions may be drawn as to the design of a delivery nozzle:—

(1) The nozzle should not be too short, or the quick change of cross-sectional area required to obtain the necessary reduction in the velocity of the jet will set up eddying motions, and so impair the power of the jet. In other words, there should not be a very abrupt rise in the pressure curve.

(2) The form of nozzle should provide for a fairly regular rate of conversion of velocity into pressure so as to produce stability in the jet. In other words, the pressure curve should not have any decided humps upon it.

(3) The nozzle surface should be as smooth as possible, so as not to produce eddying motions in the jet, and to offer a minimum of resistance to its flow.

Table IV. gives the approximate deliveries of different sizes of live steam injector when fitted non-lifting, with the feed water temperature about 60 deg. Fah. An increase in the said temperature, as also the fixing of an injector above the level of the feed water, causes a diminution in the delivery.

TABLE IV.

Size of Injector in Millimetres.	Internal dia. of steam, water, and delivery pipes in ins.	Boiler Pressure in lbs. per Square Inch.																		Size of Injector in Millimetres.
		30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	
		Delivery in Gallons Per Hour at the Above Pressures.																		
2	½	40	50	50	60	60	70	70	80	80	80	90	90	90	100	100	100	110	110	2
3	½	90	110	120	140	150	160	170	180	180	190	200	210	210	220	220	230	240	250	3
4	1	170	200	220	240	260	280	300	310	330	340	360	370	390	400	410	420	430	440	4
5	1	270	310	350	380	410	440	470	490	520	540	560	580	600	620	640	650	660	670	5
6	1¼	390	450	500	550	590	640	670	710	750	780	810	840	870	830	910	930	980	1010	6
7	1¼	530	610	680	750	820	870	920	970	1020	1060	1110	1150	1190	1220	1250	1270	1310	1370	7
8	1½	690	800	890	980	1060	1130	1200	1270	1330	1390	1450	1500	1550	1600	1640	1670	1710	1790	8
9	1½	880	1010	1130	1240	1340	1440	1520	1610	1680	1760	1830	1900	1970	2030	2080	2120	2170	2270	9
10	2	1080	1250	1420	1540	1660	1770	1930	1980	2080	2170	2290	2350	2430	2500	2560	2610	2670	2809	10
11	2	1310	1520	1700	1860	2010	2150	2280	2400	2500	2630	2740	2840	2940	3020	3090	3150	3240	3390	11
12	2¼	1560	1830	2060	2210	2390	2560	2710	2860	3030	3130	3260	3380	3500	3600	3680	3750	3900	4000	12
13	2¼	1830	2100	2340	2600	2800	3000	3180	3350	3500	3650	3850	3950	4040	4220	4320	4410	4620	4740	13
14	2½	2000	2450	2730	3000	3200	3450	3650	3890	4050	4250	4450	4650	4790	4900	5010	5110	5250	5500	14
15	2½	2400	2800	3150	3460	3740	4000	4240	4470	4690	4900	5090	5200	5400	5620	5760	5870	6150	6310	15
16	2¾	2750	3200	3600	3940	4250	4550	4850	5080	5350	5570	5800	6020	6220	6400	6550	6680	6840	7160	16
17	2¾	3120	3620	4050	4450	4800	5130	5450	5740	6020	6290	6540	6790	6840	7220	7390	7540	7750	8070	17
18	2¾	3500	4060	4540	4990	5380	5760	6110	6430	6750	7050	7340	7630	7920	8100	8290	8450	8680	9080	18
19	2¾	3900	4520	5050	5560	6000	6410	6800	7170	7320	7860	8180	8490	8800	9020	9240	9420	9660	10140	19
20	3	4340	5000	5600	6160	6650	7110	7540	7950	8340	8700	9060	9400	9740	10000	10240	10440	10680	11200	20

To find the quantity of water required by a boiler, multiply its nominal horse power by 10. This will give approximately the number of gallons required per hour.

On the subject of the losses in diverging nozzles through which water is flowing, Professor A. H. Gibson, D.Sc., has conducted an extended series of experiments, the results of which are given in an article entitled "The Conversion of Kinetic to Pressure Energy in the Flow of Water through Passages having Divergent Boundaries," which appeared in *Engineering* of February 16th, 1912.

The most important portion of such article, so far as the injector is concerned, is given below, by the kind permission of the proprietors of the above-mentioned journal.

The article also deals with trumpet-shaped pipes or nozzles, and it was found in this case that the most efficient nozzles were those in which the loss of head per unit length of the passage was constant. Compound pipes or passages are also dealt with.

Referring to uniformly tapering pipes, the writer states: "It would naturally be supposed that by enlarging the section gradually, shock and consequent eddy formation and loss of energy would be reduced, and to determine the extent to which this conclusion is justified a series of experiments on uniformly tapering pipes was carried out. Some of these were of circular cross-section, others were square, and others rectangular, with one pair of sides parallel. The ratio of final to initial areas ranged between 2·25 to 1 and 9 to 1, the larger diameters being 3 in., while the larger end of the square and rectangular pipes had the same area as the circular pipes. The mean results of these experiments are shown in figs. 22 and 23, from which the following conclusions are to be drawn:—

(*a*) In a circular pipe, with uniformly diverging boundaries, the loss of head, expressed as a percentage of $(v_1 - v_2)^2 \div 2\,g$, varies somewhat with the mean diameter of the pipe, and with the ratio of final to initial area, as well as with the angle θ, between its opposite faces. For values of θ between 6 deg. and 35 deg. the differences are comparatively small, and the loss of head is given fairly accurately by—

$$\text{Loss} = 0{\cdot}011\ \theta^{1{\cdot}22}\ \frac{(v_1 - v_2)^2}{2\,g}\ \text{feet},$$

where θ is measured in degrees.

The minimum loss of head is attained with a value of θ in the neighbourhood of 6 deg. This is, of course, due to the

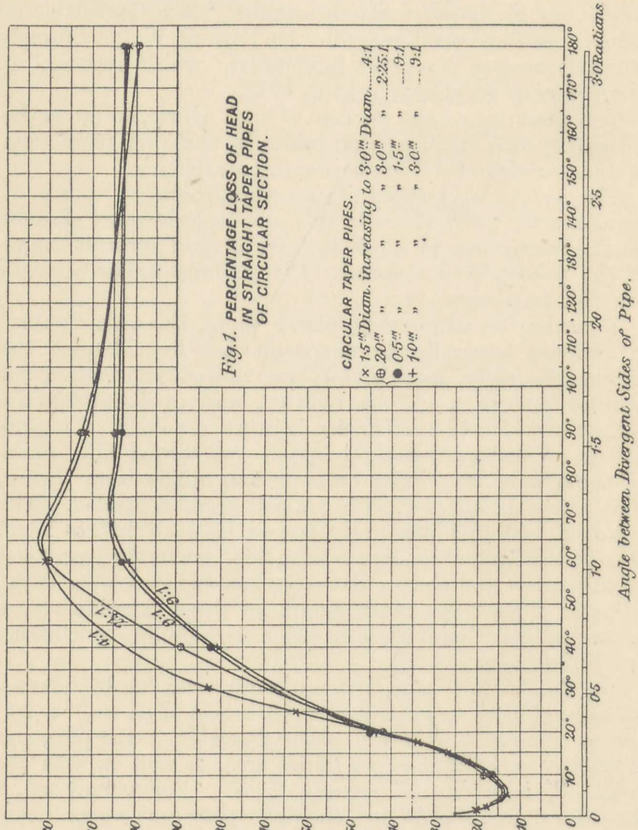


FIG. 22.

fact that the loss is made up of two parts, due respectively to wall friction and to shock following enlargement of section. As θ is reduced, the length of pipe, and therefore the friction loss, is increased; and for values of θ less than 6 deg. the increased friction loss more than counterbalances the reduced shock loss.

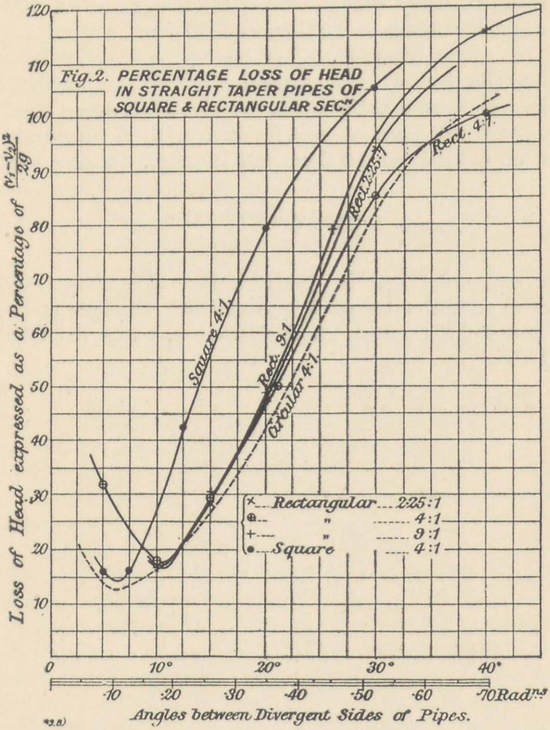


FIG. 23.

As θ is increased, the loss rapidly increases, and attains a maximum, greater than 100 per cent in every case, for a value of θ in the neighbourhood of 65 deg. The value of θ, which makes the loss equal 100 per cent, varies from 40 deg.

to 60 deg., and it is important to note that a sudden enlargement of section is more efficient in the conversion of kinetic into pressure energy than a gradual enlargement in which the angle θ is greater than this critical value.

(*b*) In pipes of square section the loss is a minimum when θ is approximately 6 deg., and attains a value of 100 per cent when θ is between 25 deg. and 30 deg.

(*c*) In rectangular passages having one pair of sides parallel, the loss is a minimum when θ is approximately 11 deg. It varies little with the size of passage and with the ratio of enlargement, and is given with fair accuracy, for values of θ between 10 deg. and 35 deg., by the relationship—

$$\text{Loss} = 0{\cdot}0072\ \theta^{1{\cdot}4}\ \frac{(v_1 - v_2)^2}{2\,g}\ \text{feet.}$$

The maximum loss is obtained when θ is about 70 deg.; while the critical value of θ, above which the loss is greater than at a sudden enlargement of section, varies from 32 deg. to 40 deg."

CHAPTER V.

Overthrow Arrangements.

The object of the overflow arrangements of an injector is to provide for the bye-passing or release of the jet within the combining nozzle, when said jet is unable from any cause to overcome the pressure in the vessel being fed and so travel from the injector through the delivery pipe.

The causes which may operate to prevent delivery against the pressure in the boiler being fed are numerous, but may be briefly summed up by the statement that the jet leaving the combining nozzle has not both the proper density and velocity to effect delivery. Thus the ratio of steam to water passing into the combining nozzle may be too great or too small. In the former case the velocity is high enough, but the density of the jet is too low. In the latter case the density may be right, but the velocity is too low.

The importance of the overflow arrangements is particularly emphasised when the injector is fitted above, and therefore has to lift its feed water. In this case there may be steam only passing into the combining nozzle for a short period.

If no means are provided for the escape of the contents of the combining nozzle when delivery from the injector is not taking place, the entering steam will, in the absence of a non-return valve, force its way down the feed-water pipe and heat the feed water. If such a non-return valve is provided, then the appliance simply remains inoperative.

In the earliest injectors, as illustrated at fig. 1, a gap was left between the combining and delivery nozzles to form the overflow passage. It will be noticed that this gap is placed at the portion of the injector where the nozzles are of smallest diameter. When this type of injector is to lift its feed water, the steam supply has to be reduced for starting purposes (by the spindle valve shown in fig. 1), otherwise the steam would be throttled in the combining nozzle and the production of a vacuum in the feed supply pipe prevented.

If an injector is unable to rapidly clear itself of steam when re-starting, so as to allow of the production of a vacuum in the feed-water pipe without reducing the steam supply, it is a non-automatic appliance.

In 1871 it was proposed to provide an additional overflow outlet at a point in the combining nozzle where the latter is of large diameter. The arrangement is illustrated at fig. 24, *a* indicating the usual overflow aperture between

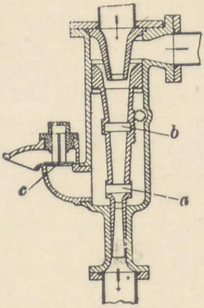


FIG. 24.

the combining and delivery nozzles, and *b* the additional aperture. There is a valve *c* upon the overflow pipe from the injector.

It is of the utmost importance in the consideration of the overflow arrangements of any particular injector to have an accurate knowledge of the pressure conditions at the various points in the combining nozzle where said arrangements are to be provided.

For example, in all injectors a high degree of vacuum exists during the normal working of the appliance at the point where the steam and water first meet (*i.e.*, at the mouth or inlet end of the combining nozzle), due to the rapid condensation of the steam. The intensity of such vacuum gradually diminishes as the jet approaches the delivery nozzle, due to the increase both in the temperature of the water in the jet and in the density of the jet, and to the lessened cross-sectional area of the combining nozzle.

The temperature of the jet at *a* (fig. 24) may, under ordinary working conditions, be equal to, above, or below 212 deg. Fah., and the pressure there, therefore, equal to above or below atmospheric; but at *b* the pressure is always considerably below atmospheric. The injector may therefore create a vacuum in the overflow chamber, and if said chamber communicates freely with the atmosphere draw in air through the overflow pipe. The valve *c* (fig. 24) prevents such inflow. Inflow of air to the

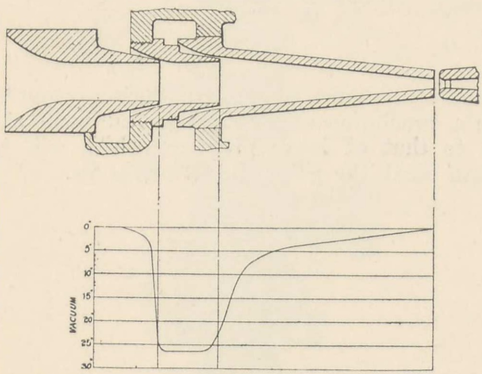


Fig. 25.

combining nozzle, due to leaky overflow valves or other causes, is a very serious hindrance to the proper working of the appliance, and also results in the presence of excessive quantities of air in the exhaust steam of the engine worked from the boiler fed by the injector. Such air impairs the action of the condensing plant, and reduces the vacuum in the exhaust pipe.

Fig. 25 illustrates in diagram form the variations in pressure in the combining nozzle of an exhaust injector.

If the pressure conditions at the various overflow apertures are not considered when deciding upon the nature of the overflow arrangements, there may, for example, be a vacuum of 20 in. mercury (5 lbs. pressure per square inch absolute) at one point (as at *b*, fig. 24), and a pressure above 5 lbs. per square inch absolute at another (as at *a*, fig. 24), and the two points may be freely communicating

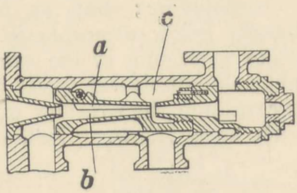


FIG. 26.

with a common overflow chamber. Such a condition must result in a circulation of vapour from the point of higher pressure to that of lower pressure, which will tend to spread and break the jet at the former point.

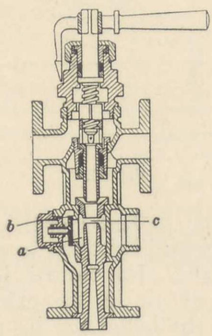


FIG. 27.

When the problem of constructing an injector capable of working with exhaust in place of "live" steam was seriously taken in hand, the importance of the above became evident, for owing to the large volume of steam to be dealt with a very large overflow area was necessary (when the injector was starting) near to the inlet end of the combining nozzle, where a vacuous condition exists during normal working.

The solution of the problem lay in the provision of means in the combining nozzle that would open and provide a large area for overflow purposes but close during normal

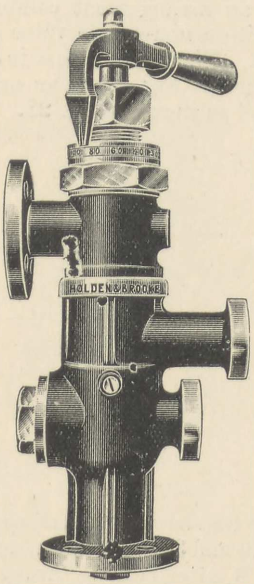


FIG. 28.

working, when the nozzle would present an unbroken passage for the jet flowing therethrough.

One form of the first proposal on the above lines is illustrated at fig. 26. When the injector is starting the hinged flap *a* moves outwardly about its pivot pin, and so provides a large area for the free escape of the contents of the combining nozzle, but when it is working normally the flap is closed, due to the difference of the pressures in the nozzle *b* and in the overflow chamber *c*. There is thus during normal working an unbroken passage for the jet travelling through the combining nozzle.

Many other proposals have been made for automatically opening and closing an overflow aperture in the combining nozzle of an injector. A few of the more important, which have come into extensive use, are given below.

Fig. 27 shows an arrangement comprising a separate compartment *a*, with a non-return valve *b* thereon, around the overflow gap *c*. The valve *b* acts in a similar manner to the flap in fig. 26. Fig. 28 is an external view of an injector constructed as shown in fig. 27.

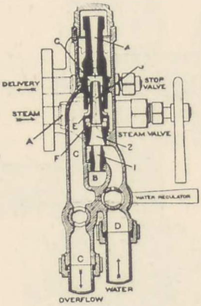


FIG. 29.

One form of the Gresham automatic sliding nozzle injector is shown at figs. 29 and 30. In this arrangement the steam from the passage B, after passing through the steam nozzle 1, enters the lifting cone 2, where it meets the water from the water-inlet pipe D. When the injector is starting, and is unable to deliver into the boiler being fed, the pressure in the space F acts on the sliding combining nozzle 3 to force it towards the delivery nozzle 4, so providing a large overflow gap at E, through which overflow takes place to the chamber C. When, however, the injector commences to work normally, the delivery pressure rises, and, acting in the space G upon the nozzle 3, forces it towards the inlet end of the injector, so closing the overflow gap E, and providing a practically unbroken combining passage. The injector here illustrated is of the combination or self-contained type, comprising steam valve A, delivery stop valve, water regulator, and back-pressure

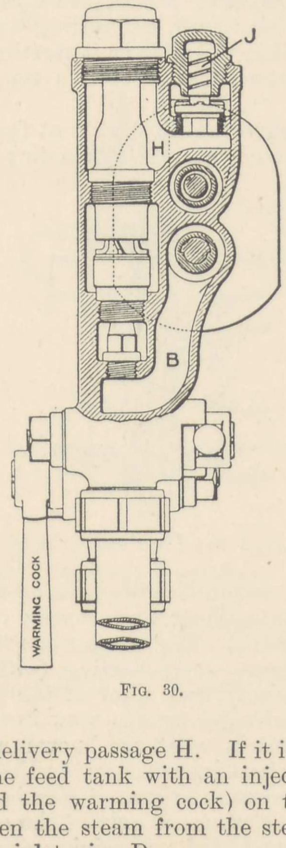


FIG. 30.

valve J on the delivery passage H. If it is desired to warm the water in the feed tank with an injector of this type, the cock (called the warming cock) on the overflow pipe C is closed, when the steam from the steam nozzle passes down the water inlet pipe D.

Another sliding nozzle injector is shown at fig. 31 without the control fittings of the combination type. In this injector *a* is the sliding combining nozzle and *b* the steam-inlet nozzle.

A modification of the sliding-nozzle arrangement is shown at fig. 32. In this a collar or ring *a* slides upon the exterior of the combining nozzle, and opens or closes the outlet from the chamber *b*, containing an overflow gap, according to whether the injector is starting or is working normally. An external view of this type of injector is given in fig. 33.

With the arrangement illustrated at figs. 26 to 33 the action of the injector becomes automatic; that is to say,

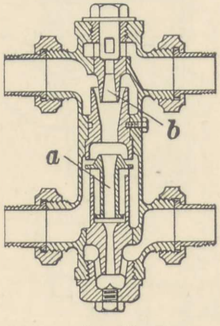


Fig. 31.

the injector can automatically start or re-start after a stoppage, for the opening and closing of the overflow apertures is automatic. The injector can therefore rapidly clear itself of steam when starting (without the steam supply being reduced), and allow of the production of a vacuum in the feed-water pipe to draw in the feed water.

With the advent of what are known as "hot-water" injectors, or injectors intended for dealing with hot feed water and high-pressure steam, it has become necessary to provide for the closing not only of the overflow passage near to the inlet end of the combining nozzle, but also of that near to the outlet end of the latter, for the temperature of the water jet at the latter point is higher than 212 deg. Fah, and its pressure greater than atmospheric.

The following table gives typical examples of the pressure at the overflow gap between the combining and delivery nozzles when the feed water is hot and the steam pressure high. Of course, such overflow pressure should, under perfect conditions, be an amount exactly correspond-

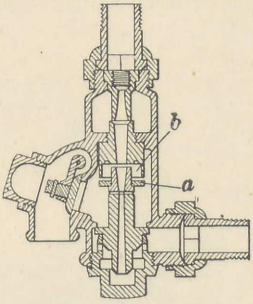


FIG. 32.

ing with the temperature of the jet, but in practice it varies with the pressure against which delivery is being effected.

TABLE V.—SHOWING PRESSURE AT OVERFLOW GAP BETWEEN COMBINING AND DELIVERY NOZZLES OF AN INJECTOR.

Steam pressure. Lbs. per sq. in. absolute.	Feed-water temperature. Deg. Fah.	Pressure at overflow gap between combining and delivery nozzles. Lbs. per sq. in. absolute.
200	126	40
205	126	43
210	126	47
215	126	50
220	126	55

The control of said overflow pressure can be effected by providing a screw-down valve on the overflow chamber of any of the injectors illustrated in figs. 26 to 33, but this will be at the sacrifice of the automatic properties of the appliances, for should the jet passing through the combining nozzle break from any cause, the screw-down overflow valve will prevent its escape; the steam will then force its way down the feed-water pipe.

The object to be had in view with the hot-water injector must be, therefore, to provide automatic means for loading

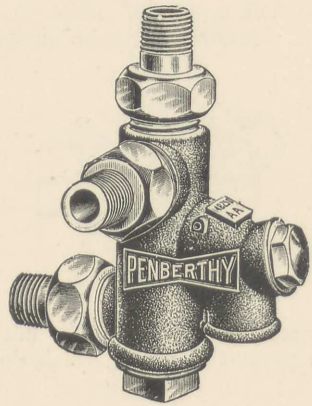


FIG. 33.

or closing (during normal working of the appliance) the overflow aperture or apertures at which the pressure is greater than atmospheric.

The force for such automatic loading or closing of the overflow must be one dependent upon the working of the injector; that is to say, the force must only be operative when the injector is working normally. Such force is, to a certain extent, to be found in the pressure in the delivery chamber of the injector. The limitation in the application of this force is that, even when the injector is not working normally, or no delivery is taking place therefrom but steam and water are flowing thereto, a considerable pressure may exist in the delivery chamber.

Whilst makers generally employ the pressure in the delivery chamber of the injector to load the overflow valve or close the overflow apertures, they adopt various means for neutralising the effect of the pressure existing in that chamber during the starting of the injector.

Automatic loading of the overflow valve by means of the pressure in the delivery chamber is provided for in the injector illustrated at fig. 34, where the valve-like head *b* of the overflow valve *c*, is exposed to the delivery pressure.

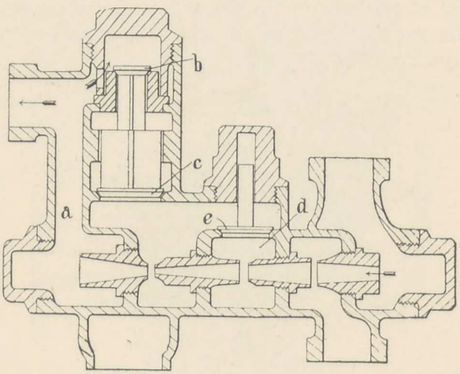


FIG. 34.

No means are provided for neutralising the effect of said pressure upon the valve *c* when the injector is starting, consequently there is a throttling action upon the overflow which impairs the lifting and re-starting powers of the appliance. It will also be seen that all the overflow has to pass the valve *c*. The overflow from the compartment *d*, which has a free valve *e* thereon, is not independent of the overflow from the remainder of the overflow chamber, which it should be for most efficient working.

In the arrangement illustrated at fig. 35 the overflow aperture *a* for the lifting steam nozzle *b* is placed in a separate and entirely independent compartment *c*, having a free non-return valve *d* thereon. A discharge through the valve *d* can take place whenever the pressure at *a* exceeds that of the atmosphere. This will only be when the injector is starting or "knocks off." A second overflow aperture is controlled by a flap *e*, and a third, namely *f*, discharges into the chamber *g*, having a valve *h* thereon loaded through the piston and lever *i j* by the pressure in the delivery chamber *k*. The areas of the piston *i* and valve *h* are such as ensure the opening of the valve when starting. The valve *d* opens independently of the valve *h*. Throttling of the lifting steam jet is thus entirely obviated.

Fig. 36 shows a right-hand vertical combination injector of the type illustrated at fig 35, and fig. 37 a non-lifting injector of this type.

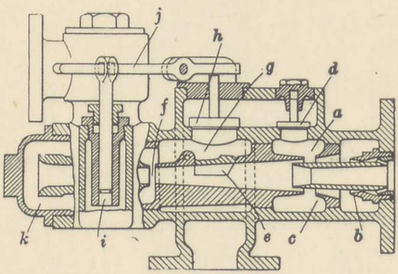


FIG. 35.

In the arrangement shown at fig. 38 the delivery nozzle *a* slides through the wall *b* under the influence of the difference of the pressures at its two ends, that at the delivery end being in part counteracted by the spring *c*. When the injector is starting, the steam from the lifting nozzle *d* escapes freely past the flaps *e*, whilst the nozzle portions *f* and *a* slide towards the delivery end of the appliance, being assisted by the spring *c*. When the injector is working normally the flaps close and the nozzle portions *f a* slide towards the injector inlet end, as shown; all communication between the interior of the combining nozzle and the overflow chamber *e* is then cut off.

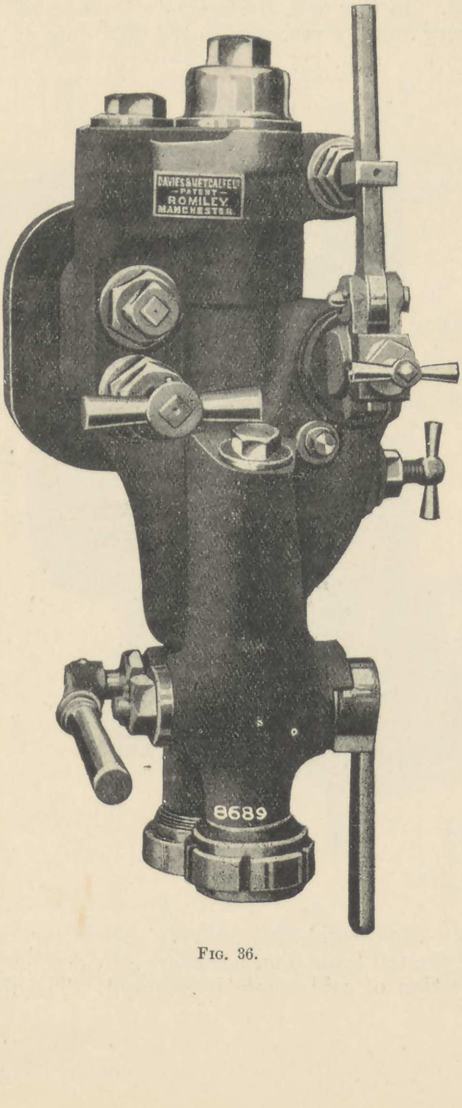


FIG. 36.

Fig. 39 shows a modification of the injector illustrated at fig. 38. In this case the spring *c* is dispensed with, as the delivery nozzle *a* responds to the difference of pressures

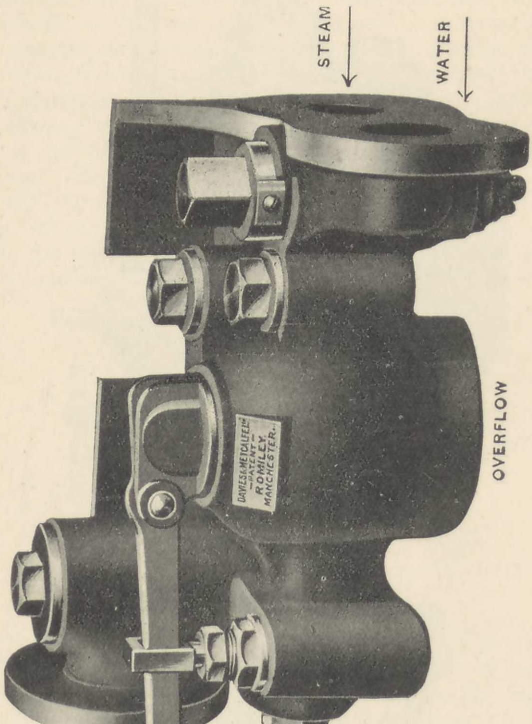


Fig. 37.

at its two ends sufficiently to give satisfactory working. A hand-operated lever *g* is, however, provided for the positive actuation of said nozzle if required. The first overflow apertures are closed by flaps *e* as shown. This injector is of the combination type, adapted for direct connection by the flange *h*, containing steam and delivery passages to the boiler front plate.

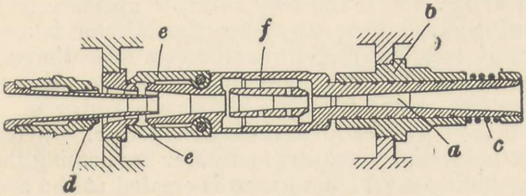


FIG. 38.

Figs. 40 and 41 show a similar type of injector to that illustrated at fig. 39, but of the horizontal pattern for fitting below the water level. Fig. 41 is a plan view of the injector shown in sectional elevation at fig. 40.

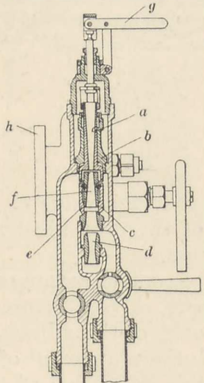


FIG. 39.

Another arrangement is shown at fig. 42. In this the fluid in the delivery chamber *a* cannot obtain free access to the overflow valve stem *b* to load the latter, but is controlled by a spring-loaded valve *c* so as to prevent such access until the pressure in the chamber exceeds a certain amount.

An arrangement in which the delivery chamber forms an overflow chamber when the injector is starting is illustrated at fig. 43. The only overflow aperture from the combining nozzle is at *a*. When the injector is starting, the steam escaping therethrough forces open the valve *b*. The latter then abuts against and opens the valve *c*, permitting free escape from the delivery chamber *d*. The valve *b* is of greater area than the valve *c*. When the correct proportions of steam and water are passing through the combining nozzle, a vacuum is created at the aperture *a*, and the valve *b* is forced on to its seat. The valve *c* is then returned to its seat by the delivery pressure, and

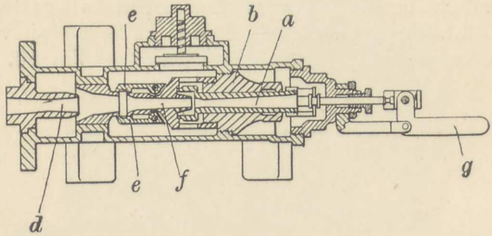


FIG. 40.

delivery takes place past the valve *e*. Fig. 44 is a view showing an injector of the type illustrated at fig. 43, but without the steam valve.

In the overflow valve control arrangements illustrated at figs. 44 to 48, a shuttle valve *s* is provided in a sleeve *h*, its function being to control the admission of steam to the chamber *c*, into which the plunger o^1 of the overflow valve *o* projects. It is by this admission of steam that pressure is put upon the overflow valve and the injector thus transformed into the " hot water " type.

The shuttle valve is arranged to be acted on at one end by the pressure in the delivery chamber through the passages *p*, *p*, and at the other end by steam derived through the passages *v*, *v* from the throat of the steam nozzle, where the pressure is always below that of the boiler. The delivery passage, on the other hand, is always at or above the boiler pressure when the injector is at work. The result of this arrangement is that a large differential pres-

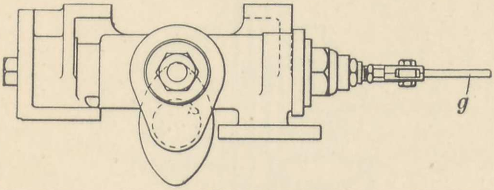


FIG. 41.

sure is ensured under all circumstances for the proper actuation of the shuttle valve.

The action at starting is as follows: When the steam is turned on to the injector, the pressure which is imme-

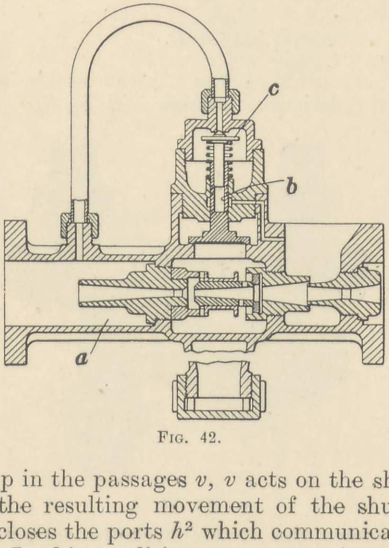


FIG. 42.

diately set up in the passages v, v acts on the shuttle valve at s^2, and the resulting movement of the shuttle in the direction s^3 closes the ports h^2 which communicate with the chamber c. In this condition no pressure is put upon the plunger o^1 and the overflow valve is accordingly free to lift off its seat to allow the usual starting discharge of steam

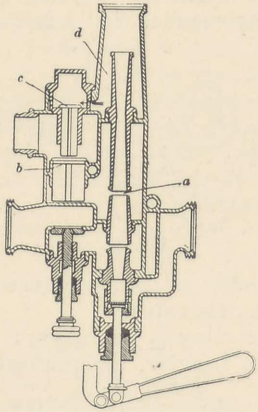


FIG. 43.

and water. As soon as the injector starts to work the pressure in the delivery chamber acts on the shuttle valve

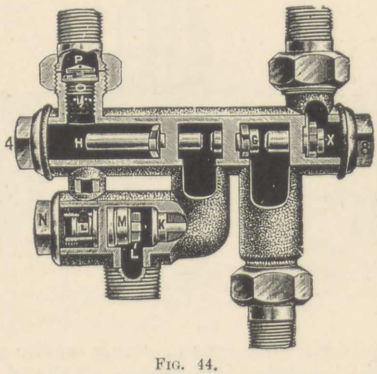


FIG. 44.

at s^3, overcomes the steam pressure acting at s^2, and throws the shuttle over into the position shown. Steam

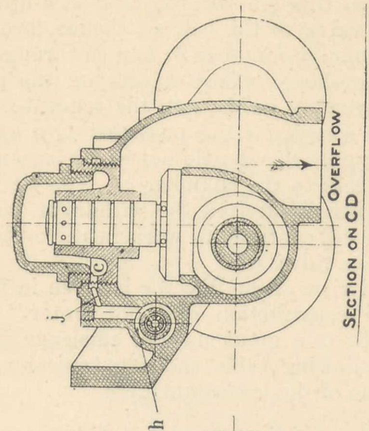


FIG. 46.

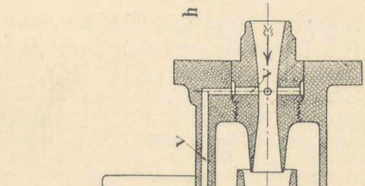


FIG. 45.

can then pass through the passages *v, v* up the centre of the shuttle valve to the belt s^1, thence through the holes h^2, to the annular chamber *h*, and so through the passages *j* to the chamber *c*, where it acts on the plunger *o* and closes the overflow valve with the requisite pressure. By turning the screws *l, l* the passages *p, p* and *v, v* can be closed; this puts the shuttle out of action and cuts off the supply of steam to the chamber *c*. The sleeve and shuttle valve can then, if desired, be withdrawn complete, while the injector will continue to work in the ordinary way with a free overflow valve.

A general view of the injector is given in fig. 48.

That the construction of "hot water" injectors has been brought to a great state of efficiency will be realised from the following table showing typical results guaranteed by some of the leading makers.

TABLE VI.—SHOWING MAXIMUM TEMPERATURE OF FEED WATER FOR INJECTOR SUPPLYING THE BOILER FROM WHICH IT IS RECEIVING STEAM.

Boiler and delivery pressure Lbs. per sq. in. absolute.	Height of lift In feet.	Feed-water temperature. Deg. Fah.
225	6	120
225	2½	125
215	6	125
205	2½	130
195	2½	140
190	6	130
75	2½	140

Under special conditions better results are, of course, obtainable, but the above are typical of ordinary practice. The temperatures are the maximum for automatic restarting.

The total overflow area should in a lifting injector be in excess of the total steam inlet area. If there are a plurality of overflow apertures in the combining nozzle

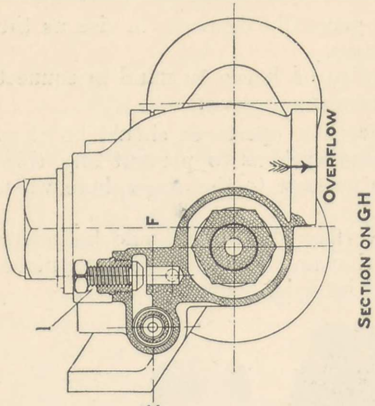

OVERFLOW
F
E
l
SECTION ON GH

FIG. 48.

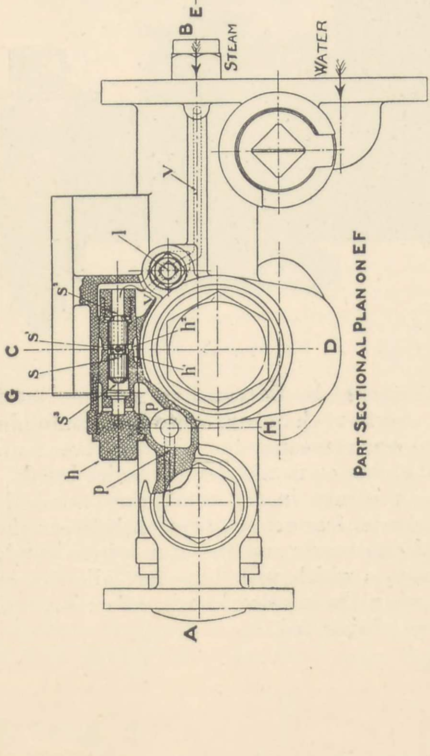

B
STEAM
WATER
V
l
C
G
D
H
h
p
A
PART SECTIONAL PLAN ON EF

FIG. 47.

they should generally decrease in size as the bore of the nozzle decreases.

The points to be borne in mind in connection with the overflow arrangements are:—

(1) The overflow apertures should be of sufficient total area, and also such as to prevent any throttling of the steam in its passage to the atmosphere when the injector is starting.

(2) The overflow apertures should be independent of one another, so that no circulation can take place between one aperture and its neighbours.

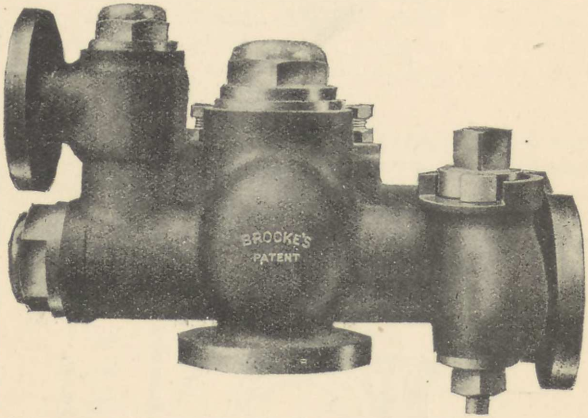


FIG. 49.

(3) Means should be provided to prevent an inflow of air to the overflow chamber or to the combining nozzle interior, but such means should not prevent an outflow from said chamber or nozzle should the jet break.

(4) If the pressure in the combining nozzle is greater than atmospheric, free communication between the interior of the combining nozzle and the atmosphere should be prevented by means which will, however, allow of such communication when the injector knocks off or is not delivering into the vessel being fed.

CHAPTER VI.

Water Control Arrangements.

THE quantity of steam passing through an injector varies with the pressure of the steam supply. It follows that if it is desired to maintain a fairly constant delivery temperature, and to deliver a maximum quantity of water per pound of steam, the quantity of water supplied to the injector must also be varied at the same time as the steam pressure varies. Unless this is done, the temperature and pressure of the jet passing through the combining nozzle is increased upon an increase in the steam pressure. After a certain limit is reached the action of the injector becomes unreliable and then ceases, the water not being able to sufficiently condense the steam within the limits of the combining nozzle. On the other hand, if the steam pressure falls, the water supply may be too great, and the velocity of the jet passing through the combining nozzle be so reduced as to render that jet unable to overcome the pressure in the boiler being fed.

In the earliest Giffard injectors, as illustrated at fig. 1, regulation of the annular water inlet area between the steam nozzle and the combining nozzle was obtained by moving the steam nozzle towards or away from the combining nozzle. This method was found in practice to have many objections, the chief being that the necessary packing around the sliding or adjustable steam nozzle was continually requiring to be renewed.

The use of the adjustable steam nozzle was finally obviated by the introduction in 1864 of a movable combining nozzle. One form of this arrangement is illustrated at fig. 50. The combining nozzle *a* is adjusted by a rack and pinion or other device to increase or diminish the annular water inlet area. Injectors having means for adjusting the latter area are known as "adjusting injectors."

It had also been proposed to control the water supply by a valve or cock placed upon the feed water supply pipe.

This proposal was not received with favour at first, but is now the usual practice.

It will be readily appreciated that from a theoretical standpoint it is decidedly better to vary the annular water inlet area than to have said area fixed and to throttle the water supply in the water pipe, for if said area be too large

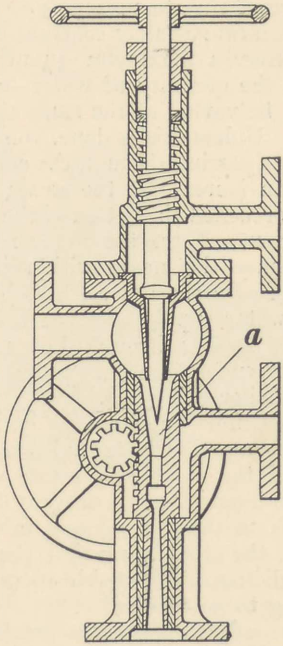


Fig. 50.

and the water does not fill same, the steam jet may not be completely encircled by the water jet, whilst if it be too small a drag is placed upon the water entering the injector.

The form of water regulator shown at fig. 51 is automatic in its action, and is constructed upon the principle that when a correct ratio of water to steam is passing through the injector the pressure at the overflow gap *a* between the combining and delivery nozzles *b, c* is atmospheric, but if said ratio is too great or too small the pressure thereat is respectively above or below that of the atmosphere. If the said pressure is greater than atmospheric, there is an overflow of water into the overflow

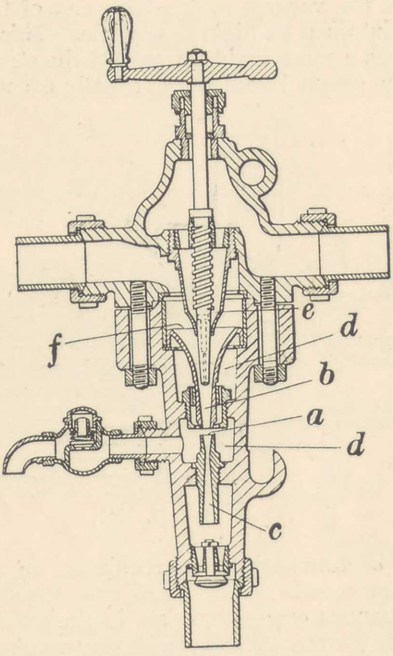


FIG. 51.

chamber *d*; the pressure thereby produced in said chamber forces the combining nozzle *b* towards the steam nozzle *e*, and so restricts the annular water inlet *f*. If, however, the pressure at the gap *a* is less than atmospheric, a vacuous condition will be created in the overflow chamber, and the nozzle *b* will be moved away from the steam nozzle; the water inlet area is then increased.

It may be here remarked that in order to reduce the loss in velocity in an injector due to the impact of the steam upon the water, the water supply should be provided in the form of a thin film moving with a high velocity. The injector water inlet is therefore always made of annular form, and the vacuum at the inlet end of the combining nozzle maintained as high as possible. This arrangement also ensures a more rapid mixture of the steam and water than if the steam issued concentrically around a solid jet of water.

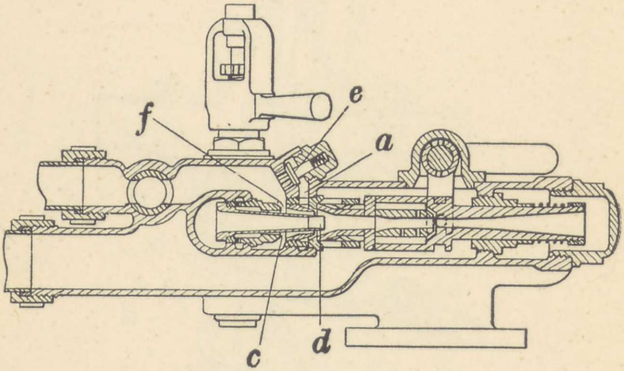


Fig. 52.

A form of automatic water regulator, in which the use of a sliding nozzle is obviated, was shown at fig. 13. When a vacuous condition exists in the overflow chamber *e*, the valve *f* upon the supplementary water inlet passage *g* is opened, and water flows through the overflow chamber into the combining nozzle by way of the slots therein. If, however, the pressure in *e* is equal to or greater than that of the atmosphere, the valve *f* is held on its seat.

A difficulty with the aforesaid type of injector will be at once appreciated if reference be made to the remarks upon overflow arrangements. It was shown that whereas a vacuum always exists during normal working conditions at the overflow slot near to the inlet end of the combining nozzle, a vacuous condition, atmospheric pressure, or a pressure greater than atmospheric, may exist at the gap between the combining and delivery nozzles. Obviously, therefore, one part of the overflow chamber will always be tending to draw in additional water, and the other part may or may not be so doing.

In the arrangement illustrated at fig. 52 the supplementary water inlet passage *a* leads to a point between the lifting and forcing steam nozzles *c* and *d*. If the vacuum

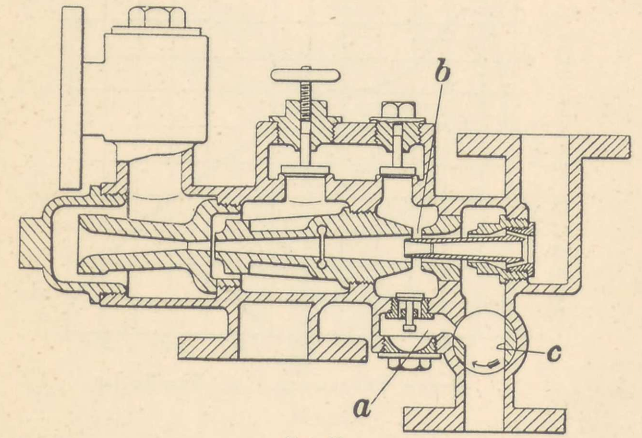


FIG. 53.

at that point is sufficiently great to open the valve *e* a supplementary inflow of water will take place. Otherwise only the water inlet *f* supplies the injector.

Another arrangement of the water inlet connections to overcome the objection aforesaid is shown at fig. 53. In this the supplementary water supply passage *a* communicates with a compartment of the overflow chamber containing the overflow gap *b*, at which during normal working a vacuum always exists. The supplementary water supply can be cut off independently of the main supply by the cock *c*, it being found in practice that a high degree of vacuum always exists at *b*, but that a supplementary water supply is only required when the feed water is so hot that the main supply is unable to properly condense the steam within the limits of the combining nozzle.

The ratio of water to steam passing through an injector decreases with an increase in the temperature of the feed water and in the pressure of the entering steam, but the provision of the supplementary water inlet ensures an

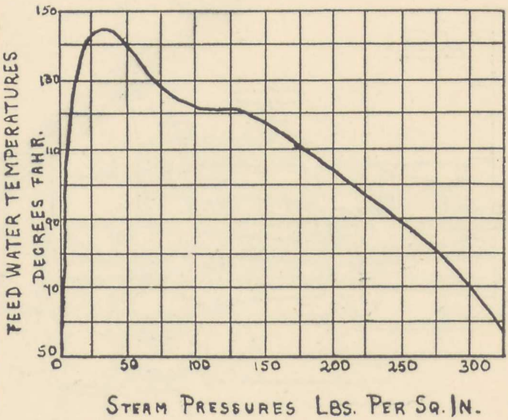


FIG. 54.

increase in the quantity of water passing through the appliance over what would be given by one water inlet only.

With both the forms of injector shown at figs. 52 and 53 the supplementary water supply has a very important effect. The main supply of water is in contact with the steam from the lifting steam nozzle for a considerably longer period than the supplementary supply. The latter therefore exerts a strong cooling action on the steam from the forcing steam nozzle and tends to reduce the temperature within the combining nozzle below what would exist if all the feed water entered at the main water inlet. A reduction of the temperature and pressure at the gap between the combining and delivery nozzles is of the greatest importance with an injector working with high pressure steam and hot feed water, for a very high pressure at that point causes the jet to be unstable and very liable to break.

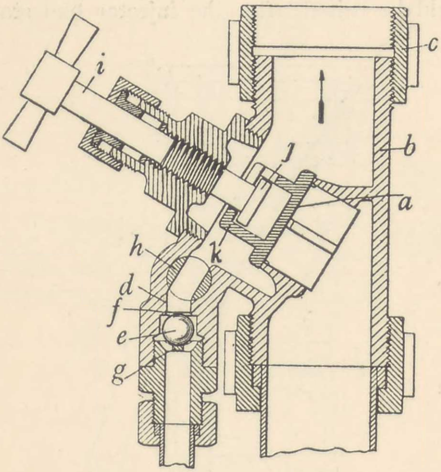


FIG. 55.

The diagram, fig. 54*, shows the maximum allowable temperatures of the feed water with any steam pressure for a typical injector having a supplementary water inlet. The irregular shape of the curve between 75 lbs. to 125 lbs. steam pressure is due to the injector commencing to draw in a supplementary water supply between the said pressures.

It will be understood that when the feed water is hot, and has to be lifted to the injector, a difficulty pre-

* By Kneass, in *Journal of Franklin Institute.*

sents itself in that the water tends, on a reduction of the pressure on its surface, to flash into steam and supply vapour only to the injector. For this reason the maximum temperature for automatic restarting, when the injector is lifting, is considerably below the maximum temperature to which the feed water can be raised after the injector has started to work. Thus, for example, the maximum restarting temperature of the feed water in one test was 104 deg. Fah., whilst the maximum temperature to which the water could be raised after the injector had started was

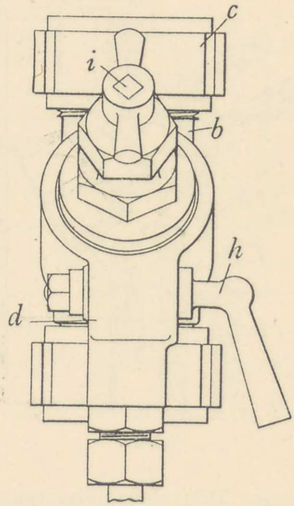


Fig. 56.

125 deg. Fah. The capabilities of any injector as regards the maximum allowable temperature of feed water with any steam pressure cannot be predicted, as they depend very largely upon the design of injector and also upon other indeterminable conditions.

A simple arrangement for dealing with the difficulty before referred to, due to the presence of hot water in the feed pipe at starting, is illustrated at figs. 55 and 56, fig. 55 being a sectional elevation and fig. 56 an end elevation of the device.

A non-return valve *a* is provided in the feed water supply pipe *b*, which opens when the injector is in operation, and tends to draw water from said pipe, but closes immediately the injector ceases working, so as to prevent a return flow down said pipe to the hot well or feed tank. The injector is connected to the fitting by means of the union piece *c*.

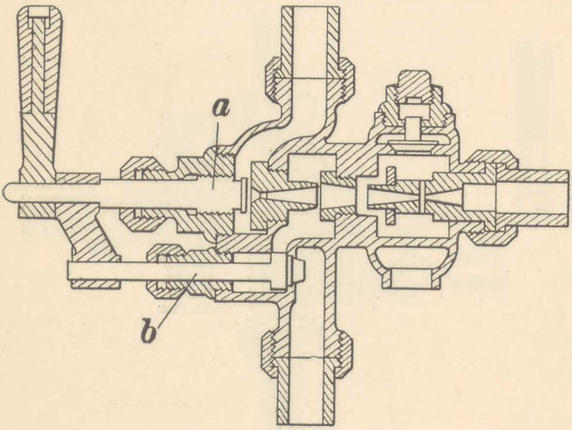


FIG. 57.

Adjacent to and on the injector side of the valve *a* is provided a by-pass or release connection *d* having a non-return valve *e* thereon. The latter closes (upon its seating *f*) when the injector is working normally, due to the difference in the pressures at its two sides, but is opened and held on its support *g* by the pressure of the water in the feed pipe *b* above the valve *a* when the injector ceases working. By the means aforesaid, the escape of the hot water which remains in an injector when the steam supply is cut off is provided for, and the difficulty experienced with lifting injectors when restarting after a short stoppage, due to the presence in the feed pipe of water at practically boiling point, is obviated. The cock *h* is for putting the valve *e* out of action. Means are provided for lifting the main non-return valve *a* in the feed supply pipe *b* from its seat, so that the water in the feed tank or hot well can be warmed with steam from the injector steam nozzle in the usual manner. Such means comprise a screw-threaded valve spindle *i* having a collar *j* upon its operative end which works within a bridge-like exten-

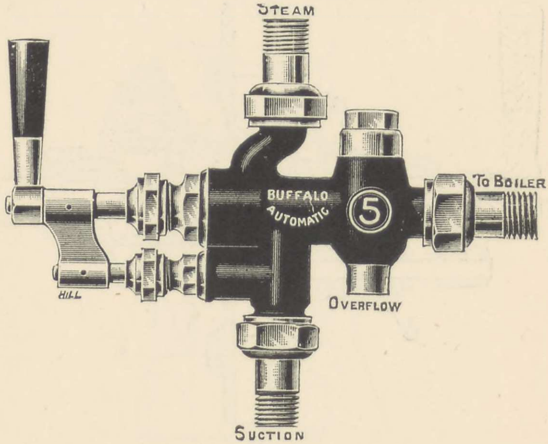


FIG. 58.

sion *k* upon the valve *a*. When the valve spindle is in a mid-way position, as shown, the valve *a* can open and close independently thereof, but if the valve spindle is screwed outwards it lifts the valve from its seat.

To provide for the regulation of the water supply of an injector simultaneously with the steam supply, injectors known as "one movement" injectors are employed. A form of "one movement" injector is illustrated at figs. 57 and 58. The spindle *a* of the steam valve is coupled to the spindle *b* of the water valve, so that when the steam valve is closed the water valve is closed, and when the steam valve is fully open the area of the water inlet passage can be regulated by a further movement of the steam valve. Various forms of "one movement" injectors are in use. Some of these involve the use of a sliding steam nozzle, as shown at fig. 27, but no external water valve, as in fig. 57.

CHAPTER VII.

Principles of the Injector.

In Chapter I. we briefly outlined the general principles of the injector, and in succeeding chapters considered the constructional details of the apparatus. We will now examine the working of an injector more in detail.

Let us consider, as an example, a jet of dry saturated steam from a boiler under a pressure of 75 lbs. per square inch absolute issuing into a region at a pressure of 3 lbs. per square inch absolute. From formulæ (vi. and vii.) we obtain the velocity (V) of discharge of the steam, x_1—the dryness fraction—being unity.

$$x^2 = \frac{\dfrac{x_1 L_1}{T_1} + \phi_1 - \phi_2}{\dfrac{L_2}{T_2}} \qquad \text{(Formula vii.)}$$

$$= \frac{\dfrac{898}{768 \cdot 5} + \cdot 45 - \cdot 2}{\dfrac{1015}{602 \cdot 6}}$$

$$= \cdot 8424$$

$$V = \sqrt{2\, g\, J\ (t_1 + x_1\, L_1 - t_2 - x_2\, L_2} \qquad \text{(Formula vi.)}$$

$$= \sqrt{64 \cdot 4 \times 778\ (307 \cdot 5 + 898 - 141 \cdot 6 - \cdot 84 \times 1015)}$$

$$= 3250 \text{ ft. per second approximately.}$$

Now let us assume that the jet of steam (moving with a velocity of 3,250 ft. per second) enters a nozzle cooled externally, so that the steam is condensed.* The water jet produced will have the same velocity as the steam jet (*i.e.*, 3,250 ft. per second), but its cross-sectional area will be very considerably less than that of the steam jet.

* If a steam jet is condensed by external cooling means, a transverse or lateral contraction of the jet takes place, but the velocity of the jet is not in any way affected.

The velocity of a jet of water issuing into a region at a pressure of 3 lbs. per square inch absolute from the boiler (under a pressure of 75 lbs. per square inch absolute) supplying the steam for the steam jet aforesaid would be an amount due to the fluid head equivalent of the boiler pressure above 3 lbs. per square inch absolute (*i.e.*, of 72 lbs. per square inch). Taking 1 lb. pressure per square inch to be equivalent to a head of 2·3 ft., the total fluid head is 165·6 ft.

Now velocity in feet per second

$$= \sqrt{2g \times \text{head in feet}}$$

$$= \sqrt{64{\cdot}4 \times 165{\cdot}6}$$

$$= 103{\cdot}2.$$

It will thus be seen that the condensed steam jet has a velocity of 3,250 ft. per second, whilst the boiler water jet has a velocity of only 103 ft. per second, the original pressures of the steam and water being the same. Obviously if the two jets were directed against one another in a pipe the condensed steam jet would very easily overcome the boiler water jet and enter the boiler. In this case, however, we are only delivering back into the boiler what has been taken therefrom.

Let us consider the case where, instead of the steam jet being condensed in an externally-cooled nozzle, it is directly mixed with water for the same purpose, and let us assume that the ratio by weight of the mixture is 15 lbs of water to 1 lb. of steam. The water we will assume to be supplied under a head of 2 ft.

Assuming that the pressure in the mixing nozzle is maintained at 3 lbs. per square inch absolute, due to the condensation of the steam by the water, the velocity of the steam jet will be, as before, 3,250 ft. per second.

The velocity of the water entering the mixing nozzle will be an amount due to the head equivalent of the difference between atmospheric pressure plus a head of 2 ft. and the pressure within the nozzle. (If the water had been "lifted" to the nozzle through 2 ft. instead of being supplied under a head of 2 ft., we should have had to "subtract" the 2 ft. instead of to add it, as in this case.) The unbalanced pressure, or the difference between atmospheric pressure and the pressure in the nozzle, is 12 lbs. per square inch approximately. Taking 1 lb. pressure per square inch as the equivalent of a fluid head of 2·3 ft., we obtain the velocity of the incoming feed water as follows:—

Velocity in feet per second

$$= \sqrt{2g \times \text{head in feet}}$$
$$= \sqrt{64 \cdot 4 \times (12 \times 2 \cdot 3 + 2)}$$
$$= 43 \cdot 66.$$

We have therefore the steam entering the nozzle with a velocity of 3,250 ft. per second, and the water entering with a velocity of about 44 ft per second.

The velocity of the water jet resulting from the union of the steam and water is obtained as follows:—

Momentum of steam jet	+	Momentum of water jet	=	Momentum of combined jet
1 × 3250	+	15 × 44	=	3910 units
		i.e., 16 × velocity	=	3910
		∴ velocity	=	244 ft. per second.

The water jet issuing from the nozzle has a velocity, therefore, of 244 ft. per second. But the velocity of a water jet issuing from the boiler supplying steam to the steam jet into a region at a pressure of 3 lbs. per square inch absolute is, as before stated, only 103 ft. per second, so that the combined jet (containing condensed steam and water in the ratio by weight of 1 to 15) will readily be able to overcome a jet tending to issue from the boiler, and to itself again enter said boiler.

From the above the general principles of action of the injector will be readily understood.

In actual practice the jet issuing from the injector is not moving with a high velocity, but is of high pressure. The relation between velocity and pressure is expressed as follows, taking 1 lb. pressure per square inch to be the equivalent of a fluid head of 2·3 ft:—

$$V = \sqrt{2g \times p \times 2\cdot3}$$

$$i.e., \; V^2 = 2g \times p \times 2\cdot3$$

$$p = \frac{V^2}{2g \times 2\cdot3} = \frac{V^2}{148} \quad . \quad . \quad . \quad . \quad \text{(xxvii.)}$$

where p = pressure in pounds per square inch,

g = acceleration of gravity = 32·2 ft. per second^2,

and V = velocity in feet per second.

The following table gives the weight of 1 cubic foot of water at different temperatures, and the fluid head equivalent of 1 lb. pressure per square inch:—

TABLE VII.

Temperature. Deg. F.	Weight, in lbs., of 1 cub. foot.	Head, in feet, equivalent of 1 lb. pressure per sq. inch.	Temperature. Deg. F.	Weight, in lbs., of 1 cub. foot.	Head, in feet, equivalent of 1 lb. pressure per sq. inch.
32	62·418	2·307	125	61·654	2·335
35	62·422	2·307	150	61·201	2·355
40	62·425	2·307	175	60·665	2·374
45	62·422	2·307	200	60·081	2·402
50	62·409	2·307	212	59·64	2·414
60	62·372	2·309	250	58·75	2·451
75	62·313	2·313	300	56·97	2·528
90	62·133	2·318	400	54·25	2·655
100	62·022	2·322	500	51·16	2·81

Let us now examine the action of an injector from a thermal standpoint, which is the most important, as the method previously adopted ignores altogether the question of the heating effect of the steam.

The total heat energy utilised in an injector is that of the entering steam plus that of the supply water. Such heat energy leaves the injector in three forms. In the first place, a portion of same is converted into work (1 B.T.U. being the equivalent of 778 foot-pounds), and provides the propulsive force for imparting velocity to the delivery jet; a second, and the greatest portion, serves to give the delivery jet a high temperature; and a third portion radiates from the injector casing.

We will consider an injector working under the following conditions:—

Steam pressure, lbs. per sq. inch absolute.	Feed-water temperature.	Height of lift.	Pressure in injector, lbs. per sq. inch absolute.	Ratio of water to steam in lbs.	Delivery temperature.	Delivery pressure, lbs. per sq. inch absolute.
	Deg. F.	Ft.			Deg. F.	
95	97	2	5	13 to 1	175	115

We can readily ascertain the quantity of heat energy required to produce a delivery jet of a certain pressure if the weight of the jet and also its temperature are known.

In the present case the weight of unit mass of the jet may be taken as 14 lbs. (1 lb. steam and 13 lbs. water), the delivery pressure is 110 lbs. above the pressure within the injector (which is 5 lbs. per square inch absolute), and the fluid head equivalent of 1 lb. pressure when the fluid temperature is 175 deg. Fah. is 2·375 ft.

Heat energy required to produce the delivery jet

$$= \frac{\text{Mass in lbs.}}{\text{Joule's equivalent}} \times \frac{\text{velocity}^2}{2\,g}$$

$$= \frac{\text{W}}{\text{J}} \times \text{(Head equivalent of pressure of jet, in feet)}$$

$$= \frac{14}{778} \times (110 \times 2{\cdot}375)\ .$$

$$= 4{\cdot}7 \text{ B.T. units approximately.}$$

The total heat in 1 lb. of the steam entering the injector above what is contained in the delivery jet is 1,035 B.T.U. (1,210 – 175). The amount of heat energy actually converted into work in the injector we find to be 4·7 B.T.U., so that there are 1,030 units still to be accounted for. The raising of the temperature of 13 lbs. of water from 97 deg. Fah. to 175 deg. Fah. absorbs 1,014 units, leaving 16 units* as the amount of heat radiated from the injector. We have here neglected the heat energy equivalent of the kinetic energy of the entering feed water, as this is very small, but in the calculations which follow it is considered.

The important point to be noticed is the very small portion of the heat energy supplied to the injector which is effectively utilised as a propulsive force. The reasons for this can be conjectured. In the first place some of the heat energy of the steam is abstracted therefrom during expansion within the steam nozzle by the cold feed water around that nozzle. Secondly, the impact of the steam upon the slowly-moving water will cause violent eddying motions, which will re-convert some of the kinetic energy of the steam jet into heat energy. Thirdly, there is the friction of the jet upon the walls of the nozzles through which it travels, and also the reluctance of the steam and water to quickly unite with one another. A very small portion only of the total heat energy of the steam is actually lost by radiation; the injector may be considered as a perfect appliance from a thermal standpoint, for practically the whole of the heat energy supplied is utilised and returned in the delivery jet.

It now remains for us to show how the above calculations, based upon actual tests, compare with those obtained from a theoretical consideration of the action of the appliance.

The pressure of the steam working the injector is 95 lbs. per square inch absolute, and the pressure within the injector 5 lbs. per square inch absolute. In expanding from 95 lbs. to 5 lbs. pressure per square inch absolute

* It is impossible to give this amount with any great accuracy, as a temperature reading only ½ deg. incorrect makes a difference in this figure of practically 10 degrees.

the steam will give up a certain amount of its heat energy, and a portion of the steam will condense.

Using formula (vii.) to calculate the dryness fraction x_2 when x_1 is taken as unity, we get—

$$x_2 = \frac{\frac{886}{785} + \cdot 47 - \cdot 234}{\frac{1015}{601 \cdot 6}}$$

$$= \cdot 83 \text{ approximately.}$$

That is to say, the steam contains ·17 moisture when its expansion is completed.

The total heat given up by the steam in expanding as aforesaid is therefore as follows, using the symbols of equation (vi.):—

Heat units given up by steam

$$= (t_1 + x_1 \mathrm{L}_1 - t_2 - x_2 \mathrm{L}_2)$$
$$= (324 + 886 - 162 - \cdot 83 \times 1001)$$
$$= 217 \cdot 2.$$

Thus, the steam supplies 217·2 heat units for use as the motive power in propelling the water jet into the boiler.

The water jet provides a very small amount of the propulsive force. The heat energy represented by each 13 lbs. of the water entering the injector where the pressure is 5 lbs. per square inch absolute, after being lifted 2 ft., is obtained as follows, taking 1 lb. pressure as equal to a fluid head of 2·32 ft., and remembering that the unbalanced pressure on the water tending to drive it into the injector is 10 lbs. :—

$$\text{Heat energy} = \frac{\text{Mass in pounds}}{\text{Joule's equivalent}} \times \frac{\text{Velocity}^2}{2g}$$

$$= \frac{\mathrm{W}}{\mathrm{J}} \times (\text{head equivalent in feet of 10 lbs. pressure} - 2)$$

$$= \frac{13}{778} \times (10 \times 2 \cdot 32 - 2)$$

$$= \cdot 355 \text{ B. T. Units.}$$

From our theoretical calculations we therefore ascertain that 217·555 heat units are available for conversion into kinetic energy for carrying the combined condensed steam and water jet into the vessel being fed. From our previous calculations we find that the delivery jet only represented the expenditure of 4·7 heat units, thus showing that 212·8 units must have been utilised otherwise than for the purpose for which they were available. The impact losses, eddying motions, friction, etc., in the injector destroy a large portion of the kinetic energy of the jet and re-convert the same to heat energy, which is absorbed by the feed water. If the 212·8 units are divided between the 13 lbs. of water the temperature of the latter will be raised from 97 deg. Fah. to about 113·3.

The heat units remaining in the steam at 5 lbs. pressure per square inch absolute, after ·17 (dryness fraction = ·83) thereof has been condensed, are as follows:—

$$\text{Latent heat} \times \cdot 83 = 1001 \times \cdot 83 = 831 \text{ units}$$
$$\text{sensible heat} = 162 \text{ units}$$
$$\text{Total} \ldots\ldots\ldots\ldots\ldots \; 993 \text{ heat units.}$$

The excess of heat units in 1 lb. of the steam over what is contained in the delivery jet is therefore:—

$$993 - 175 = 818 \text{ heat units.}$$

If the latter are divided between the 13 lbs. of feed water, the temperature of the latter will be raised from 113·3 deg. Fah to 176·3 deg. Fah; or, since the actual delivery temperature is found to be 175 deg. Fah., we may conclude that the 17 heat units remaining after the water has been raised to a temperature of 175 deg. Fah. represent the loss due to radiation from the injector and its connections.

Calculations in Connection with the Injector.

The most important calculation required to be made in connection with an injector is that for determining the ratio by weight of water to steam passing through the appliance.

Generally speaking, the most efficient injector is one which, when delivering into a boiler under a certain pressure, passes the greatest ratio by weight of water to steam. If, however, we are considering an injector from the point of view of the temperature of the feed water it can deal with when the steam is at a certain pressure, the most efficient injector is one capable of taking the hottest feed water with the steam at that pressure.

If the number of heat units contained in 1 lb. of the steam entering the injector be known, also the temperatures of the feed or supply water and of the delivery jet, then with $t_1 + L_1$ denoting the total heat contained in 1 lb. of the steam, t_2 the temperature of the feed water, t_3 the temperature of the delivery, and W the weight in lbs. of water per lb. of steam,

Gain of heat by feed water $= W\,(t_3 - t_2)$.

Loss of heat of 1 lb. of steam $= t_1 + l_1 - t_3$.

Heat equivalent of the kinetic energy of the jet leaving the injector

$$= \frac{(W + 1) \times V^2}{2\,g \times 778}$$

where V = velocity of delivery jet in feet per second.

If the extremely small amount of heat energy represented by the kinetic energy of the entering feed water be neglected, also the loss of heat due to radiation from the injector, then :—

$$\text{Loss of heat by steam} = \text{Heat equivalent of kinetic energy of delivery jet} + \text{gain by feed}$$

$$t_1 + L_1 - t_3 = \frac{(W \times 1)\,V^2}{2\,g \times 778} + W\,(t_3 - t_2).$$

We have already seen that the heat equivalent of the kinetic energy of the delivery jet is very small, and it can be safely neglected without materially affecting the results. The equation then becomes :—

$$t_1 + L_1 - t_3 = W\,(t_3 - t_2),$$

$$i.e.,\ W = \frac{t_1 + L_1 - t_3}{t_3 - t_2} \quad . \quad . \quad . \quad . \quad \text{(xxviii.)}$$

Application of Formula (xxviii.).—To calculate the ratio by weight of water to steam passing through an injector when the pressure of the steam entering the latter is 15 lbs. per square inch absolute, the feed water temperature is 60 deg. Fah. and the delivery temperature 168 deg. Fah.,

$$W = \frac{213 + 965 - 168}{168 - 60}$$

$$= 9{\cdot}35.$$

That is, 9·35 lbs. of water are mixed with each pound of steam passing through the injector.

If in formula (xxviii.) the numerator and denominator be transposed, the ratio will then be that of the steam to the water. That is:

$$S = \frac{t_3 - t_2}{t_1 + L_1 - t_3} \quad . \quad . \quad . \quad . \quad (\text{xxix.})$$

where S = weight in pounds of steam to 1 lb. of water.

If the quantity of feed water drawn by the injector from the feed tank in, say, an hour is known, then the quantity of steam used by the injector in an hour is given by multiplying the value of S in formula (xxix.) by the weight of feed water in pounds. That is:

$$Q_s = \frac{Q_w (t_3 - t_2)}{t_1 + L_1 - t_3} \quad . \quad . \quad . \quad . \quad (\text{xxx.})$$

where Q_s = quantity of steam in pounds per hour,

and Q_w = quantity of feed water in pounds per hour.

Application of Formula (xxx.).—To calculate the amount of steam used by an exhaust injector per hour, the pressure of the steam entering the injector being 15 lbs. per square inch absolute, the temperature of the feed water 68 deg. Fah., the delivery temperature 190 deg. Fah., and the quantity of feed water used per hour 22,800 lbs. (2,280 gallons):

$$Q_s = \frac{22800 \ (190 - 68)}{213 + 965 - 190}$$

$$= 2815 \text{ lbs. approximately.}$$

In this case it will be seen that the ratio by weight of steam to water is about 1 to 8.

The efficiency of an injector from the point of view of the ratio of water to steam passing therethrough is affected by variations in the steam pressure and in the feed-water temperature.

If the pressure of the steam entering an injector increases whilst the feed-water temperature remains unchanged, or if the pressure of the steam remains constant whilst the feed-water temperature is increased, the temperature of the mixture of steam and water is increased. The vacuum at the inlet end of the injector where the steam and water first meet is therefore impaired, and results in a reduction in the quantity of water flowing into the injector whilst maximum weight flow of the steam is maintained under all normal working conditions.

In the following table the ratio by weight of water to steam is given for a typical example when the feed-water temperature is constant and the steam pressure is varied:

TABLE VIII.

Steam pressure, pounds per square inch absolute	40	65	90	140	215
Pounds of water per pound of steam	28·5	20	17	14	11

The above results would be affected by variations in design of the injector, in the height of lift, or the pressure head of feed water, and must only be considered as typical.

If the pressure of the steam and the temperature of the feed water increase simultaneously, the ratio of water to steam will decrease more rapidly than if the one value only were increasing.

The effect of the height of suction lift or the pressure head of the incoming feed water will be appreciated from the velocity equation for the said water, which is

$$V = 8{\cdot}025 \sqrt{\text{head in feet}},$$

where V denotes velocity in feet per second. If the difference between atmospheric pressure and the pressure at the entrance to the combining nozzle be taken as 10 lbs.

per square inch, and the head equivalent of 1 lb. pressure per square inch as 2·3 ft., then, if the water is on the same level as the injector, the velocity of the water entering the injector is obtained as follows:—

$$V = 8{\cdot}025\sqrt{10 \times 2{\cdot}3}$$
$$= 38{\cdot}5 \text{ ft. per second.}$$

If the water is supplied under a head of 5 ft. its velocity is

$$V = 8{\cdot}025\sqrt{10 \times 2{\cdot}3 + 5}$$
$$= 42{\cdot}5 \text{ ft. per second.}$$

When the water is lifted through a height of 5 ft. the velocity of inflow to the injector is

$$V = 8{\cdot}025\sqrt{10 \times 2{\cdot}3 - 5}$$
$$= 34 \text{ ft. per second.}$$

A reduction in the velocity of the entering feed water naturally results in a reduction in the velocity of the combined steam and water jet, but the effect is practically inappreciable for small variations of lift. The velocity losses due to impact of the rapidly-moving steam upon the water will, however, greatly increase as the water velocity diminishes.

Two effects follow from a variation in the ratio of water to steam passing through an injector, namely, variations in the delivery temperature, and a variation in the over-pressure or excess of pressure of the delivery jet in relation to that of the steam entering the injector. The smaller the ratio of water to steam (within the working limits or range of the injector), the higher the delivery temperature and pressure. Under ordinary conditions the maximum delivery pressure obtainable in a live-steam injector is considerably in excess of the pressure of the steam entering the injector. Thus, with steam at 60 lbs. pressure, the maximum delivery pressure may be about 75 lbs.; at 80 lbs., about 100 lbs.; and 150 lbs., about 175 lbs., and so on.* It is always possible, therefore, when the injector

* These amounts depend entirely upon the feed-water temperature, height of lift, and other conditions.

is feeding a boiler at the same pressure as the steam entering the injector, to increase the ratio of steam to water above that which gives maximum delivery pressure and temperature, whilst obtaining a perfect action of the appliance. The rate of supply of water to the boiler can therefore be varied within fairly wide limits to suit requirements.

We have already seen how very small a portion of the heat energy of the steam entering an injector is actually utilised by conversion into kinetic energy in the propulsion of the water jet through the appliance. It follows, therefore, that a jet apparatus is very inefficient merely for pumping purposes. The actual efficiency of the apparatus is represented by the ratio of the work done by the steam jet to the amount of heat yielded by the steam to perform that work; *i.e.*:

$$E = \frac{U}{t_1 + L_1 - t_3} \quad . \quad . \quad . \quad . \quad \text{(xxxi.)}$$

where E = mechanical efficiency,
U = work done by steam,
t_1 = sensible heat of steam,
L_1 = latent heat of steam,
and t_3 = heat units in 1 lb. of delivery jet.

Now, U is represented by the mechanical work performed by the steam in giving the water jet a certain pressure or in delivering it to a certain height—the equivalent of said pressure—and also in raising the water to the jet pump, if the latter is fitted above the water supply.

That is

$$U = \left(\frac{W \times l + (W + 1)\, h_p}{J}\right) \text{ in heat units,}$$

where W = ratio in pounds of water to steam passing through the apparatus,
l = suction lift in feet,
h_p = head equivalent of delivery pressure,
= delivery pressure in pounds per sq. inch × 2·3.
J = Joule's equivalent = 778 foot-pounds per B.T.U.

The complete formula for jet pump efficiency is therefore—

$$E = \frac{W \times l + (W + 1)\,h_p}{J\,(t_1 + L_1 - t_3)} \quad . \quad . \quad \text{(xxxii.)}$$

If the water is supplied to the pump from a point on a level with the latter, W has zero value, whilst if the water is supplied under a head, W has a negative value, and must be subtracted from the remainder of the numerator.

Application of Formula (xxxii.).—It is desired to calculate the mechanical efficiency of a jet pump; when the steam pressure is 95 lbs. per square inch absolute the incoming water is lifted through a height of 2 ft., the pressure within the pump is 5 lbs. per square inch absolute. the delivery temperature is 175 deg. Fah., the delivery pressure 115 lbs. per square inch absolute, and the ratio in pounds of water to steam passing through the appliance 13 to 1.

Then $$E = \frac{(13 \times 2) + (13 + 1)\,(115 \times 2{\cdot}3)}{778\,(281 + 886 - 175)}$$

$$= {\cdot}0048, \text{ or } {\cdot}48 \text{ per cent.}$$

If the delivery temperature is neglected the efficiency is ·42 per cent.

The advantage possessed by the jet pump over other types of pump is its great simplicity. It is particularly well suited for dealing with chemicals or liquids having corrosive properties, as the nozzles can be suitably protected without affecting the action of the apparatus.

CHAPTER VIII.

Nozzle Dimensions.

The problem of determining the throat areas of the steam and delivery nozzles of an injector from the result of tests is a difficult one, but has been approached in various ways, all of which are subject to error owing to the impossibility of stating with any degree of accuracy the nature and amounts of the various velocity and other losses.

According to one proposed method a result is obtained by comparing the condition of the steam jet as it passes the steam nozzle throat with that of the water jet as it passes the delivery nozzle throat. In this case we require for our calculations the initial steam pressure and dryness fraction, water head at inlet, water temperature and pressure, and the quantity of water received or delivered by the injector in a specified time.

With these particulars we first proceed to determine the dryness fraction at the steam nozzle throat, the pressure there being ·58 of the inlet pressure. This value is always about ·96, but can be calculated by formula (vii.) as follows·

$$x_2 = \frac{\dfrac{x_1 L_1}{T_1} + \phi_1 - \phi_2}{\dfrac{L_2}{T_2}}.$$

We then proceed to determine the velocity of the steam at the nozzle throat and also as it leaves the steam nozzle, for the actual velocity imparted to the incoming feed water is that of the steam leaving the steam nozzle.

These values can be calculated from formula (vi.), namely:

$$V = \sqrt{2 g J \, (t_1 + x_1 L_1 - t_2 - x_2 L_2)}.$$

Proceeding from the test figures given, we obtain the quantity of water delivered per pound of steam, the formula for which is as follows:—

$$W = \frac{t_1 + x_1 L_1 - t_3}{t_3 - t_2} \quad . \quad . \quad . \quad \text{(xxviii.)}$$

The velocity of the incoming feed water is found by taking into account the actual head of water (whether positive, zero, or negative) and the head due to the difference between atmospheric pressure and the pressure within the injector.

With the above results we obtain the theoretical velocity at the delivery nozzle throat.

We then proceed upon the following lines:—

Since from our experimental results we find that 1 lb. of steam delivered B pounds of water to the boiler, then to feed C pounds of water per second will require $\frac{C}{B}$ pounds of steam per second.

The specific volume v of the steam at the steam nozzle throat is—

$$v = x_2 u_2 + (1 - x_2) \sigma \quad . \quad . \quad . \quad \text{(xxxiii.)}$$

$$= x_2 u_2 \text{ approximately} \quad . \quad . \quad . \quad \text{(xxxiv.)}$$

where x_2 = the steam dryness fraction at the nozzle throat, u_2 = the specific volume of 1 lb. of steam at the pressure at the throat, and σ is the specific volume of 1 lb. of water.

From the above the throat area of the steam nozzle is given as follows:—

$$A_s = \frac{C\,(x_2 u_2 + (1 - x_2)\,\sigma)}{B \times V_t} \quad . \quad . \quad \text{(xxxv.)}$$

where V_t = the velocity of the steam at the nozzle throat, which is readily obtainable from the maximum velocity, equation (xvii.) as follows:—

$$V \text{ throat} = \sqrt{2g \frac{n}{n+1} \times 144\, p_1 v_1}.$$

This value is usually about 1,500 ft. per second, and need not be specially calculated for approximate results.

The amount of water delivered by the injector comprises the water taken in at the feed, and also the water resulting from the condensation of the steam, that is

$$\frac{C\,(B + 1)}{B} \text{ pounds per second.}$$

The area of the nozzle (A_d) required to pass this quantity of water, having a density ρ and a velocity V_d feet per second, is given by the following formula :—

$$A_d = \frac{C\,(B + 1)}{B \times \rho \times V_d} \quad . \quad . \quad . \quad \text{(xxxvi.)}$$

Application of Formulæ.—Calculate from theoretical considerations the area of the steam and delivery nozzle throats of an injector taking 4 lbs. of water per second from the hot well, the feed water temperature being 60 deg. Fah., the delivery temperature 140 deg. Fah., the steam or boiler pressure 90 lbs. per square inch absolute, the water head 2 ft., and the vacuum at the steam nozzle mouth or outlet 24 in. mercury.

The pressure at the steam nozzle throat is 90 × ·58 lbs. (52 approximately) per square inch absolute as maximum flow conditions exist.

Taking the initial dryness fraction as unity, the dryness fraction at the nozzle throat is :

$$x_2 = \frac{\dfrac{x_1\, L_1}{T_1} + \phi_1 - \phi_2}{\dfrac{L_2}{T_2}}$$

$$= \frac{\dfrac{889}{781} + \cdot 452 - \cdot 415}{\dfrac{915}{744}}$$

$$= \cdot 956.$$

The steam velocity at the nozzle throat is

$$V = \sqrt{2g\,\frac{n}{n + 1} \times 144\, p_1\, v_1}.$$

Now $n = 1{\cdot}135$,

$$\text{therefore } V = \sqrt{2 \times 32{\cdot}2 \frac{1{\cdot}135}{2{\cdot}135} \times 144 \times 90 \times 4{\cdot}79}$$

$$= 1469 \text{ ft. per second.}$$

The approximate values ·96 and 1500 would have been sufficiently accurate.

The dryness fraction at the steam nozzle mouth, where the pressure is 3 lbs. per square inch absolute, is

$$x_2 = \frac{\dfrac{x_1 L_1}{T_1} + \phi_1 - \phi_2}{\dfrac{L_2}{T_2}}$$

$$= \frac{\dfrac{889}{781} + {\cdot}452 - {\cdot}2}{\dfrac{1015}{603{\cdot}6}}$$

$$= {\cdot}918;$$

therefore the velocity at the steam nozzle exit is

$$V = \sqrt{2\, g\, J\, (t_1 + x_1 L_1 - t_2 - x_2 L_2)}$$

$$= \sqrt{2 \times 32{\cdot}2 \times 778\, (320 + 889 - 141{\cdot}6 - {\cdot}918 \times 1015)}$$

$$= 2606 \text{ ft. per second.}$$

The ratio of water to steam in the injector is:

$$W = \frac{t_1 + x_1 L_1 - t}{t_3 - t_2}$$

$$= \frac{320 + 889 - 140}{140 - 60}$$

$$= 13{\cdot}36.$$

The velocity of the water entering the injector due to a head of 2 ft. and a pressure of 12 lbs. per square inch (the difference between atmospheric pressure and the pressure within the injector) is:

$$\begin{aligned} V &= \sqrt{2\,g\,H} \\ &= \sqrt{2 \times 32{\cdot}2\,(2 + 12 \times 2{\cdot}3)} \\ &= 44 \text{ ft. per second.} \end{aligned}$$

[One pound pressure per square inch is taken as equal to a water head of 2·3 ft.]

The steam nozzle throat area is now obtained thus—

$$\begin{aligned} A_s &= \frac{C\,(x_2\,u_2)}{B \times V} \\ &= \frac{4\,({\cdot}956 \times 8{\cdot}1)}{13{\cdot}3 \times 1469} \\ &= {\cdot}00158 \text{ square foot.} \end{aligned}$$

The velocity imparted to the water is obtained as follows:

$$\begin{matrix} \text{Momentum} \\ \text{of steam} \end{matrix} + \begin{matrix} \text{Momentum} \\ \text{of water} \end{matrix} = \begin{matrix} \text{Momentum of} \\ \text{delivery jet} \end{matrix}$$

$$1 \times 2606 + 13{\cdot}3 \times 44 = 3191\,;$$

therefore velocity of delivery jet = 223 ft. per second. This would give the delivery nozzle throat area as—

$$\begin{aligned} A_d &= \frac{4 \times 14{\cdot}3}{13{\cdot}3 \times 62{\cdot}4 \times 223} \\ &= {\cdot}00031 \text{ square foot.} \end{aligned}$$

We thus find that the ratio of areas of steam nozzle throat to delivery nozzle throat is about 5·1 to 1. In actual practice, for the conditions stated, the said ratio would be about 3 to 1.

We have already shown that the delivery jet of an injector only represents the expenditure of a very small portion of the heat energy of the steam which is available for imparting velocity to said jet. In the example calculated the delivery jet, instead of having a velocity of 223 ft. per second, would probably have a velocity of about 136 ft. per second,—that is, about 25 per cent more than the velocity equivalent of 80 lbs. per square inch above atmospheric pressure—the pressure at the delivery nozzle throat being considered as atmospheric, and in the delivery pipe as 95 lbs. absolute, or 5 lbs. above boiler pressure.

If we consider that the velocity of the incoming water was 25 per cent less than that calculated above, say about 33 ft. per second, then the actual effective velocity required in the steam to give a delivery velocity of 136 ft. per second would be about 1,506 ft. per second. The calculated area of the delivery nozzle throat would be:—

$$A_d = \frac{4 \times 14{\cdot}3}{13{\cdot}3 \times 62{\cdot}4 \times 136}$$

$$= {\cdot}0005067 \text{ square foot,}$$

which gives a ratio of steam nozzle throat area to delivery nozzle throat area of 3·1 to 1, which is near to that adopted in practice

A point about which experimental data is lacking is the density of the jet as it passes the delivery nozzle throat. It is known from experiments that a quantity of air is mixed with the water, but it is not known as to whether the steam is all condensed in the combining nozzle. This knowledge would enable us to give a more accurate figure for density in the formula for area of delivery nozzle throat.

The above method has been set out in full, but it involves several cumbersome calculations.

The determination of the steam nozzle throat area can be effected quite readily by the application of Napier's formula:—

$$W = \frac{p \times A}{70},$$

or

$$A = \frac{W \times 70}{p}.$$

The amount of steam used per second in the injector was $\frac{4}{13{\cdot}3}$ lbs. and the pressure (p) was 90 lbs. per square inch;

therefore

$$A_s = \frac{\frac{4}{13 \cdot 3} \times 70}{90}$$

$$= 0 \cdot 234 \text{ square inch}$$

$$= 0 \cdot 0016 \text{ square foot.}$$

To calculate the delivery nozzle throat area with any degree of ease we must make two assumptions. The first is as to the velocity of the delivery jet at the said throat, and the second is as to the density of the jet there. If we assume that the delivery jet velocity is 25 per cent greater than the velocity equivalent of the difference between the absolute delivery and overflow pressures, which amount errs on the small side, and take the density of the jet as that of hot water at the delivery temperature, which probably errs on the large side, we will obtain a fairly accurate result. This is the method adopted in the earlier calculation, but it is again set out here.

Taking P_d, or the delivery pressure, as 95 lbs. absolute, and P_o, or the overflow pressure, as 15 lbs. absolute, then:

$$V_d = 1 \cdot 25 \sqrt{2\,g \times 2 \cdot 3\ (P_d - P_o)}$$

$$= 1 \cdot 25 \sqrt{64 \cdot 4 \times 2 \cdot 3\ (95 - 15)}$$

$$= 136 \text{ ft. per second.}$$

Now $A_d = \dfrac{W}{\rho \times V}$

where A_d = area in square feet of delivery nozzle throat,
W = weight of water passing in pounds per second,
ρ = density of jet in pounds per cubic foot,
and V = velocity of jet in feet per second;

therefore

$$A_d = \frac{4 \times 14 \cdot 3}{13 \cdot 3 \times 62 \cdot 4 \times 136}$$

$$= 0 \cdot 0005067 \text{ square foot.}$$

A further method of approaching this problem is to consider the momentum equation previously given, and which is as follows:

$$\begin{array}{ccccc} \text{Momentum of} & + & \text{Momentum of} & = & \text{Momentum of} \\ \text{steam jet} & & \text{water jet} & & \text{delivery jet} \\ W_s \times V_s & + & W_w \times V_w & = & W_d \times V_d. \end{array}$$

But W_d = area of delivery nozzle throat (A_d) × density of delivery jet at the throat (ρ_d) × velocity of the jet at the throat (V_d);

therefore $W_s \times V_s + W_w \times V_w = A_d \times \rho_d \times V_d^2$.

Now the velocity of the delivery jet (V_d) at the delivery nozzle throat is, if we neglect all considerations of losses, an amount equivalent to that due to the difference between the pressure in the vessel being fed (P_d) and the pressure at the said throat (P_o) or

$$\text{since} \quad V^2 = 2\,g\,H,$$

$$\text{and} \quad H = \frac{P}{\rho},$$

$$\text{then} \quad V_d^2 = \frac{2\,g\,(P_d - P_o)}{\rho_d}.$$

The above momentum equation now becomes:

$$W_s \times V_s + W_w \times V_w = A_d \times \rho_d \times \frac{2\,g\,(P_d - P_o)}{\rho_d}$$

$$= A_d \times 2\,g\,(P_d - P_o).$$

From formula (xiv.), which is as follows, we know that

$$W = A_s\left(\frac{2}{n+1}\right)^{\frac{1}{n-1}} \sqrt{2\,g \times \frac{n}{n+1} \times \frac{P_s}{v_s}}.$$

Taking $\qquad n = 1{\cdot}135,$

then $\qquad W_s = 3{\cdot}6\ A_s \sqrt{\frac{P_s}{v_s}}.$

If we neglect $W_w \times V_w$, the equation then becomes:

$$3{\cdot}6\ A_s \sqrt{\frac{P_s}{v_s}} \times V_s = A_d \times 2\,g\,(P \quad - P_o);$$

or

$$\frac{A_s}{A_d} = \frac{2\,g\,(p_d - p_o)}{3{\cdot}6\sqrt{\frac{p_s}{v_s}} \times V_s}$$

The losses in the delivery nozzle and connections may be taken as about 25 per cent, so that an approximation is:

$$\frac{A_s}{A_d} = 18 \frac{(1{\cdot}25\, p_d - p_o)}{V_s \sqrt{\frac{p_s}{v_s}}}.$$

The velocity V_s is that of the steam at the steam nozzle throat, as we are virtually comparing the momentum at the said throat with that at the delivery nozzle throat. Now, it is known that the velocity at the throat of a well-designed nozzle under maximum weight flow conditions is equal to:

$$V_s = \sqrt{2\, g \frac{n}{n+1}\, p_s\, v_s}$$

$$= 5{\cdot}85 \sqrt{p_s\, v_s}$$

when $n = 1{\cdot}135$;

then the formula becomes:

$$\frac{A_s}{A_d} = \frac{3\,(1{\cdot}25\, p_d - p_o)}{p_s} \quad . \quad . \quad . \quad . \quad (\text{xxxvii.})$$

Applying this formula to the problem previously worked out, and taking the delivery pressure as 95 lbs. and the overflow pressure as 15 lbs., we get:

$$\frac{A_s}{A_d} = 3 \frac{(1{\cdot}25 \times 95 - 15)}{90}$$

$$= 3{\cdot}125.$$

If, in our previous calculations, we had not discarded the value $W_w \times V_w$, the formula would have been:

$$A_s = \frac{2\,g\; A_d\; (1{\cdot}25\; p_d - p_o) - W_w \times V_w}{20{\cdot}78\, p_s} \quad . \quad (\text{xxxviii.})$$

These notes are more or less schematic, but are useful as giving an insight into the problems surrounding the injector.

CHAPTER IX.

Exhaust Steam Injectors.

Even before the advent of the "Giffard" injector in 1858 engineers had endeavoured to make practical use of the heat energy represented by the exhaust or waste steam from non-condensing steam engines. Such heat energy might be utilised as a heating medium, or, by conversion into work, as a propulsive force, or both as a heating medium and a propulsive force. Exhaust or waste steam also represents a great amount of water which might be recovered.

The chief proposals of the early engineers had, however, for their object not the utilisation of the exhaust steam as a propulsive force for boiler feed purposes, but the employment of such steam as a feed-water heating medium.

The principles upon which the construction of exhaust injectors is based were first enunciated with a degree of accuracy in 1876. The following gives a summary thereof:—

"It is known that when steam is condensed a vacuum is created, the degree of which is dependent upon the temperature of the water of condensation; and when this temperature is low, and the vacuum therefore good, it is found that the velocity of steam of different pressures flowing into it does not differ in anything like the proportion of the pressures.

"The exhaust injector differs from ordinary injectors in having the final cross-sectional area of the steam-inlet passage or nozzle much larger in proportion to that of the water passage and to that of the 'throat' or smallest section of the delivery nozzle than is the practice in ordinary injectors, so as to provide for the larger volume of the exhaust steam in order to pass about the same weight of exhaust as would have been used of 'live' or boiler steam; for it will be understood that, though the velocity of the particles of exhaust steam on leaving the steam nozzle and entering the vacuum formed in the combining cone may be approximately as great as if boiler steam of considerable pressure were used, yet the weight or quantity of steam passed through a given area will be less in proportion to the inferior density of the exhaust steam."

A typical exhaust steam injector is illustrated at fig. 59. The following is a detailed description of the parts of the appliance:—

The Steam Nozzle.—The ratio of cross-sectional areas of steam inlet nozzle *a* and delivery nozzle throat adopted for exhaust injectors is 16 to 1, as compared with from between 2 and 3 to 1 in the case of ordinary live-steam injectors.

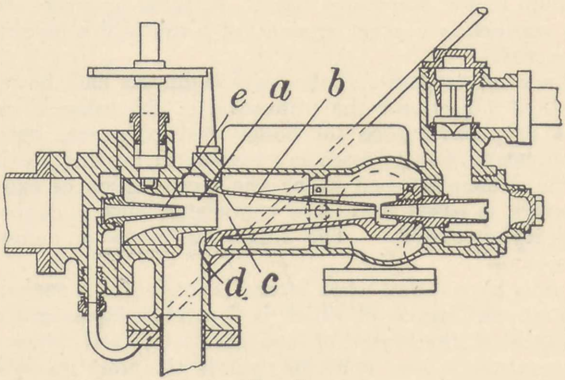


Fig 59.

As the pressure of the entering steam is but small, namely, that of the atmosphere, it is unnecessary to allow for any transverse or lateral expansion in the steam nozzle *a*. The latter is therefore a plain cylindrical tube with a rounded or smooth inlet edge.

The Combining Nozzle.—The length of the combining nozzle is usually not less than 1.5 times the diameter of the delivery nozzle throat, so as to ensure satisfactory condensation of the large volume of steam dealt with.

The Delivery Nozzle of an exhaust injector is similar to that employed in a live-steam injector.

Overflow Arrangements.—The remarks made under this heading in connection with live-steam injectors apply also to exhaust injectors.

The great difficulty presented by exhaust steam is its large volume per unit weight. The overflow area must therefore be very large to permit a free escape of the jet from the combining nozzle when the injector is starting,

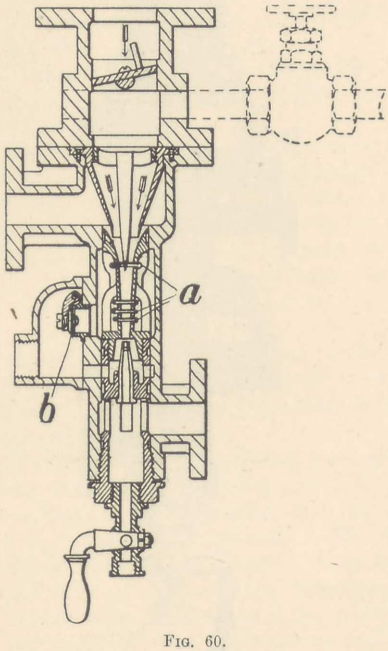


FIG. 60.

but the provision of such a large overflow area must not be at the expense of the continuity of the support provided for the jet by the combining nozzle. It was not until 1880 that the problem of providing satisfactory arrangements was solved, the result being the production of the first automatic injector. The solution then provided lay in dividing the combining nozzle longitudinally into two portions and pivoting one portion, termed the flap, upon the other. A "flap" nozzle is shown in fig. 59. The flap *b* opens when the injector is starting, but closes when the injector is working normally, due to the difference of pressures at its opposite sides.

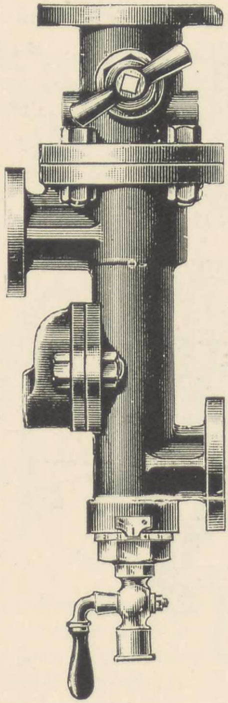


FIG. 61.

Another overflow arrangement as employed upon exhaust injectors is illustrated at fig. 60. The combining nozzle is provided with several slots *a* at right angles to its length, and a valve *b* is provided on the overflow chamber to prevent any inflow of air to said chamber.

Fig. 61 is an external view of the injector illustrated at fig. 60.

Water Control Arrangements.—The water supply to the injector is regulated by sliding the steam inlet nozzle *a* away from or towards the combining nozzle *c* (see fig. 59) so as to vary the annular water inlet area *d* between said nozzles. In some cases the steam nozzle is fixed and the combining and discharge nozzles are moved relatively thereto (see fig. 60).

The water is admitted as a thin annular film in order to present as large an area as possible to the steam for condensation purposes.

Action of Exhaust Injector.—The velocity of the incoming exhaust steam is entirely dependent upon the degree of vacuum* at the mouth of the steam nozzle. To produce said vacuum, the presence of both the steam and water is necessary so that the former may be condensed by the latter. The exhaust steam from an engine is generally of sufficient pressure to flow to the injector of itself.

With an exhaust injector constructed as shown at figs. 59 and 60, the pressure during normal working at the mouth of the steam nozzle is generally about 5 lbs. to 6 lbs. per square inch absolute.

The velocity of saturated steam at a pressure of 16 lbs. per square inch absolute, containing, say, 20 per cent of moisture into a region at a pressure of 6 lbs. per square inch absolute, is 1,640 ft. per second. We will assume that 9 lbs. of cold feed water under a head of 2 ft. are mixed with each 1 lb. of steam. The velocity of the feed water into a region at a pressure of 6 lbs. absolute will be about 37 ft. per second. The momentum and velocity of the combined steam and water jet is readily obtained as follows:—

$$\begin{array}{ccccc} \text{Momentum of steam jet} & + & \text{Momentum of water jet} & = & \text{Momentum of combined jet} \\ 1 \times 1640 & + & 9 \times 37 & = & 1973 \text{ units.} \end{array}$$

* As the degree of vacuum at the mouth of the steam nozzle is dependent upon the temperature of the feed water by which the steam is condensed, exhaust injectors are unable to deal with hot feed water.

That is the velocity per unit mass (1 lb.) of the jet is 197·3 ft. per second. If a water jet were discharging from a boiler into a region at an absolute pressure of 6 lbs. per square inch, then to give said jet a velocity of 197·3 ft. per second, the boiler would require to be under a gauge pressure of about 254 lbs. per square inch. Thus if we neglect all consideration of losses, an exhaust injector working under the conditions just described should be able to feed a boiler under a pressure of nearly 254 lbs. per square inch. To illustrate the actual performance of an injector under conditions very similar to those described, the result of a test is here given:—

Exhaust steam pressure. Pounds per square inch absolute.	Feed water temperature.	Head of Feed water.	Delivery temperature.	Delivery pressure. Pounds per square inch absolute.
	Deg. Fah.		Deg. Fah.	
16	60	2 feet	162	95

The reasons for the low delivery pressure have already been fully discussed.

One of the drawbacks of exhaust injectors as made until quite recently has been the lowness of the maximum delivery pressure when exhaust steam alone is employed; 80 lbs. by gauge has been the highest obtainable under ordinary working conditions.

To vary the delivery pressure without altering the construction of the injector or the water head, two courses are open, namely:—

(1) To increase the water inlet area, so as to increase the rate of condensation of the steam and intensify the vacuum at the steam nozzle mouth.

(2) To reduce the water inlet area, so as to lessen the load which the steam has to carry.

Both of these courses can be adopted successfully within narrow limits, but eventually the disadvantages outweigh the advantages. Thus, if the water supply is increased, there comes a point when the rate of retardation of the velocity of the jet through the combining nozzle is greater than the rate of increase in velocity due to the increased vacuum. On the other hand, if the water supply is greatly reduced, the decrease in vacuum and consequent reduction in the velocity of the entering steam quite over-balances the advantage gained by the reduction of the load which the steam has to carry. If the water supply is insufficient, the jet will not be condensed within the limits of the combining nozzle, and the injector will not work.

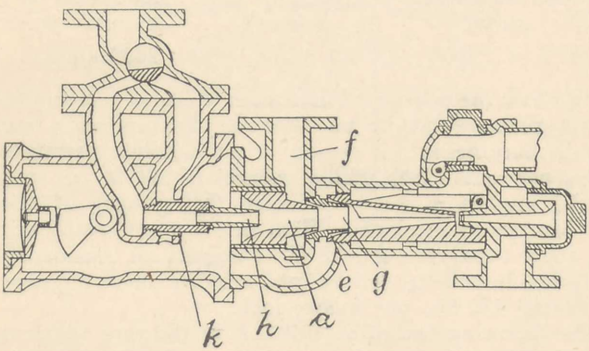


FIG. 62.

To improve the working of an exhaust injector it follows therefore that the vacuum at the mouth of the steam nozzle must be increased without increasing the ratio of water to steam.

In the injector illustrated at fig. 62 the object aforesaid has been attained by dividing the steam inlet area into two portions, one steam jet entering at *a* and the other at *e*. The whole of the water enters at *f*. As only a part of the steam meets the water in *g*, condensation is very rapid, and a vacuum of from 25 in. to 27 in. mercury is produced. The said vacuum is maintained at the supplementary steam inlet *e*. Fig. 63 is an external view of an exhaust injector of the type illustrated at fig. 62, but without the special live steam fittings there shown.

The theoretical velocity of steam (containing 20 per cent of moisture) at 16 lbs. per square inch pressure absolute issuing into a region at an absolute pressure of 2 lbs. per square inch is about 2,260 ft. per second. We will assume that with every pound of steam there is mixed 9 lbs. of water under a head of 2 ft. The velocity of flow of the water under a head of 2 ft. into a region at an absolute pressure of 2 lbs. per square inch is about 48 ft. per second. The momentum of the combined steam and water jet is obtained as follows:—

$$\begin{array}{ccccc} \text{Momentum of steam} & + & \text{Momentum of water} & = & \text{Momentum of combined jet} \\ 1 \times 2260 & + & 9 \times 48 & = & 2692 \text{ units.} \end{array}$$

from which the velocity of unit mass (1 lb.) of the combined jet is given as 269 ft. per second. To produce a water jet discharging into a region at an absolute pressure of 2 lbs. per square inch with a velocity of 269 ft. per second would require a water pressure (by gauge) of about 470 lbs. per square inch. Thus an injector working under the above conditions should, if we neglect all consideration of losses, be able to deliver into a boiler under a pressure of nearly 470 lbs. per square inch.

The following test of an injector, of the type illustrated at fig. 62, under conditions very similar to those just considered, will show what the actual performance is:—

Exhaust steam pressure. Pounds per square inch absolute.	Feed water head.	Feed water temperature.	Delivery temperature.	Delivery pressure. Pounds per square inch absolute.
	Feet.	Deg. Fah.	Deg. Fah.	
16	2	60	168	135

The important point is that whilst the proportion of water to steam passing through the injector in the test just dealt with is practically the same as in the previous test (as is evidenced by the uniformity of delivery temperatures, with the same steam pressures and feed temperatures), the actual delivery pressure is increased from 95 lbs. to 135 lbs. per square inch absolute.

Table VIII. shows comparative tests on two exhaust injectors constructed as shown in figs. 59 and 62 respectively.

TABLE VIII.—COMPARATIVE TESTS OF OLD AND NEW TYPE EXHAUST INJECTORS.

Head of water 2ft. in both sets of tests.

I.—OLD TYPE.

Exhaust steam pressure. Pounds per square inch absolute.	Feed water temperature.	Delivery temperature.	Delivery pressure. Pounds per square inch (gauge).	Delivery. Gallons per hour.
	Deg. Fah.	Deg. Fah.		
16	60	162	80	1,800
18	60	166	92	1,800
20	60	169	100	1,800
22	60	174	110	1,850
25	60	176	120	1,850
		II.—NEW TYPE.		
16	60	168	120	1,800
18	60	172	130	1,800
20	60	174	145	2,000
22	60	177	155	2,000
25	60	180	165	2,000

To increase the delivery pressure of the injector at fig. 62 so as to adapt it to work against any pressure, one live steam nozzle *h* only is employed. With earlier exhaust injectors, as at fig. 59, both a live steam nozzle as *h* and a supplementary live steam injector were necessary.

To work the exhaust injector when live steam is not available, as, for example, when the engine to which the injector is fitted is not working, a connection is provided through which live steam, throttled down to atmospheric pressure, can flow. In the injector illustrated at fig. 62 such throttled live steam inlet is at *k*.

To ensure the exhaust steam supply to an injector being as dry as possible, the injector steam pipe should be connected to the top of the engine exhaust pipe if the latter is horizontal, or to the side of same if it is vertical. The application of an exhaust injector to a range of boilers is shown in fig. 64.

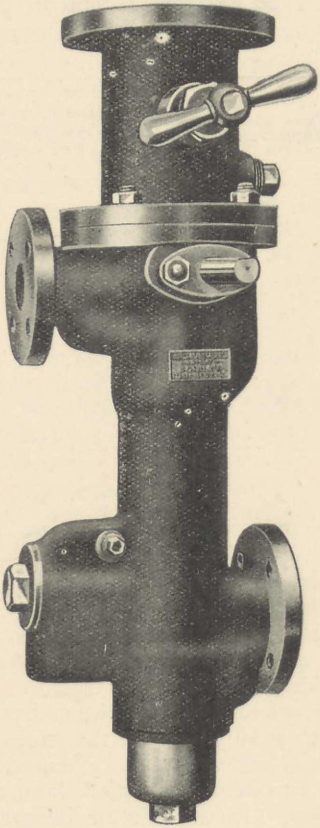


Fig. 63.

The correct size of injector for feeding any particular boiler is determined by the evaporative power of the boiler.

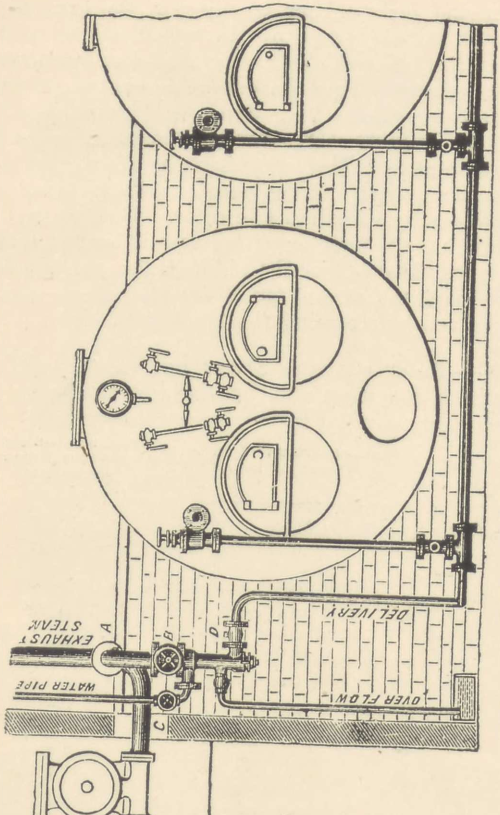


FIG. 64.

To determine the quantity of steam available per minute from an engine for working an injector, the area of the engine piston is multiplied by the speed of the piston, *i.e.*, area in feet × stroke in feet × 2 × number of revolutions per minute. Thus an engine having a piston of 15 in. diameter, stroke 2 ft. 6 in., and running at 80 revolutions per minute will provide about 491 cubic feet of steam per minute. The following table shows the quantity of steam required and the amount of water delivered by the various sizes of exhaust injector. If, however, the engine supplying the injector with steam is lightly worked, or cut-off takes place earlier than half stroke, the available quantity of exhaust steam found by the above method should be halved before using the table.

TABLE IX.—TABLE SHOWING STEAM REQUIRED AND WATER DELIVERED FOR EACH SIZE OF EXHAUST INJECTOR.

Size of injector.	Maximum amount of water delivered per hour.	Minimum quantity of exhaust steam required per minute.
Number.	Gallons.	Cubic feet.
2	60	26
3	150	60
4	270	100
5	420	156
6	600	225
7	830	306
8	1,080	400
9	1,370	506
10	1,700	625
11	2,050	756
12	2,450	900
13	2,870	1,056
14	3,330	1,225
15	3,820	1,406

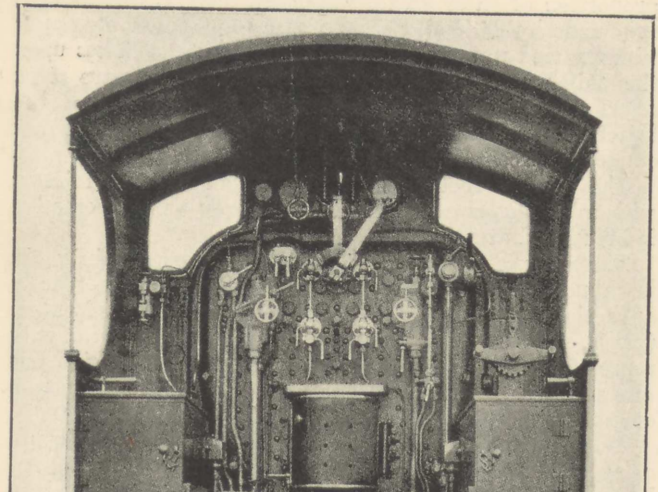


FIG. 65.

The exhaust steam from an engine contains a certain proportion of oil or grease suspended therein, and as it is very undesirable to allow any grease to pass into the boiler with the feed water on account of its injurious effects upon the boiler plates and tubes, grease separators are often fitted on the steam pipe between the engine and the injector. As a rule, however, grease separators are not employed except upon locomotives.

The general method of fitting up an exhaust injector with a supplementary live steam portion on a locomotive is illustrated at fig. 65.

The great saving effected by the use of exhaust injectors in connection with non-condensing engines will be readily appreciated by simple calculations according to formula (xxx.). The exhaust steam utilised in the injector not only represents the saving of a great number of heat units, but also the saving of a large quantity of water—both points of great importance.

As an example of the economy to be effected by the use of an exhaust injector, suppose the steam boiler being fed by the latter to be working at 120 lbs. per square inch absolute, whilst the feed water supplied to the injector is at 60 deg. Fah, and suppose that the latter water is pumped directly into the boiler at a temperature of 60 deg. Fah., then the total heat units above 32 deg. Fah. required from the boiler furnace to evaporate 1 lb. of the water at a pressure of 120 lbs. absolute are 1,158. If now the feed water be passed through the injector and be heated to 168 deg. Fah., then the boiler furnace will have to give only 1,050 heat units to evaporate 1 lb. of the water at 120 lbs. absolute pressure. This represents a gain of 108 units of heat or 9·3 per cent. A saving of 10 per cent in coal consumption is generally reckoned as the result of the employment of an exhaust injector. Less cold water will, of course, be drawn from the feed tank, as the exhaust steam provides about one-ninth of the total water delivered into the boiler.

CHAPTER X.

Compound Injectors.

In our discussion of the steam nozzle we referred to that arrangement of lifting and forcing steam nozzles in which the incoming water jet is caused to move with a high velocity before it is acted on by the forcing steam jet.

In compound injectors, two steam nozzles—termed lifting and forcing nozzles—are employed. The water jet is given a high velocity by the lifting nozzle, such velocity being then converted into pressure in a diverging nozzle, and again reconverted into velocity at the entrance to the combining nozzle of the forcing portion. A compound injector is simply two injectors, the one—termed the forcer—receiving the delivery of the other termed the lifter.

A compound injector is illustrated at fig. 66. The lifting portion comprises a steam nozzle *a*, a water inlet *b*, and a combining delivery nozzle *c*. The jet issuing from the latter nozzle passes to the inlet end of the combining delivery nozzle *d* of the forcing portion, where it is acted on by the steam jet from the forcing steam nozzle *e*. The steam valves *g* and *h* of the steam nozzles are operated by a cross bar *i* from a rod *j* connected to the hand lever *k*, which also operates the overflow valve *m* by means of a connecting link and eccentric. To start the injector the hand lever *k* is pulled towards the left in the illustrated example, and first opens the overflow valve *m* and at the same time moves the bar *i* to open the steam valves. As, however, the area of the forcing steam valve *h* is larger than that of the lifter valve *g*, the steam holds the former valve upon its seat and the latter valve is opened. When water appears at the overflow, the lever *k* is given a further movement to the left, which closes the overflow valve *m* and opens the forcing steam valve *h*. Delivery is then effected into the boiler being fed.

The lifting portion of the compound injector is designed on the lines of an ejector; that is to say, the effective (or throat) area of the steam inlet nozzle is less than the effective (or throat) area of the delivery nozzle, so as to ensure a very strong suction effect in the water supply pipe. The forcing portion of the appliance is designed solely as an injector; that is, with the effective steam inlet area greater than the effective delivery area, so as to give a jet of maximum velocity.

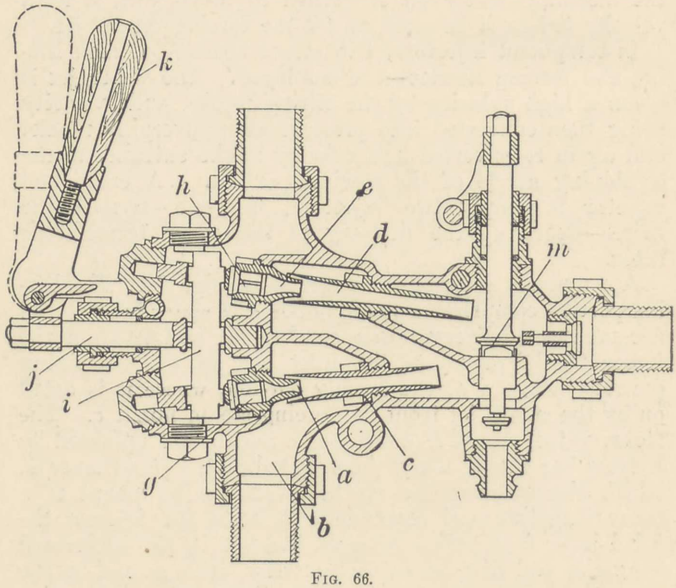


FIG. 66.

The quantity of water passed by the lifting portion of the injector to the forcing portion varies with variations in the pressure of the steam entering the injector; hand regulation of the steam supply to suit the steam pressure is therefore unnecessary.

To vary the quantity of water delivered by a compound injector when the steam pressure remains constant, the flow of steam to the lifting steam nozzle is controlled.

In the arrangement illustrated at fig. 67, the first movement of the handle *a* opens the pilot valve *b* and allows steam to pass through the chamber *c* to the valve *d* controlling the entrance to the lifting steam nozzle. By manipulating the valve *d* the steam supply of the lifter and therefore the quantity of water drawn into the injector are varied, but the steam supply to the forcer remains unaltered. In other cases the water supply is throttled in the feed supply pipe.

Some idea of the capabilities of the compound injector will be given by the following table:—

TABLE SHOWING CAPABILITIES OF A COMPOUND LIVE-STEAM INJECTOR.

Steam pressure, pounds per square inch..	30	45	60-120	135-150	165-180
Lift in feet with cold feed water	6½	16½	20	16½	13
Feed water temperature permissible with water under head, deg. Fah.	130	140	150	145	145
Maximum feed water temperature with 6 ft. lift, deg. Fah.	120	135	140	135	130

Compound injectors should give at least an over-pressure or excess of delivery pressure over steam pressure of 12 lbs. to 15 lbs. with steam pressures up to 60 lbs., 20 lbs. with steam pressures from 60 lbs. to 120 lbs., and about 30 lbs. over-pressure for higher steam pressures.

The temperature of the water delivered from a compound injector is on an average 80 deg. Fah. higher than the temperature of the injector supply water. Thus if the injector receives its feed water at a temperature of 140 deg. Fah., the delivery temperature will be about 220 deg. Fah., or above boiling point. It will, of course, be understood that the actual increase in the temperature of the water delivered by, over that of the water supplied to, the injector depends upon the pressure of the steam. The amount of the increase is less for lower pressures and somewhat higher with high pressures. Fig. 61 is a general view of a compound live steam injector.

When discussing the exhaust injector, we referred to the difficulty experienced for many years in making this class of appliance deliver against pressures greater than from

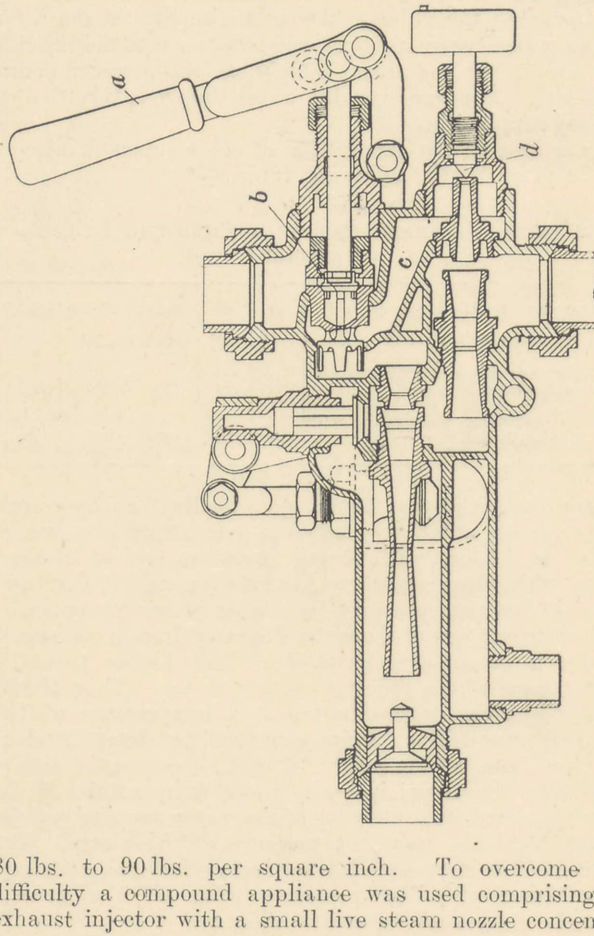


Fig. 67.

80 lbs. to 90 lbs. per square inch. To overcome this difficulty a compound appliance was used comprising an exhaust injector with a small live steam nozzle concentric

with the exhaust steam nozzle, and a supplementary live steam injector receiving the delivery from the exhaust injector and forcing it into the boiler being fed.

A typical example of this class of injector is illustrated at fig. 69. In this, *a* indicates the exhaust steam inlet nozzle, *b* the small live steam nozzle of the exhaust portion, *c* the water inlet, and *d* the live steam nozzle of the supplementary live steam portion. The delivery from the exhaust portion passes by way of the pipe *e* to the supple-

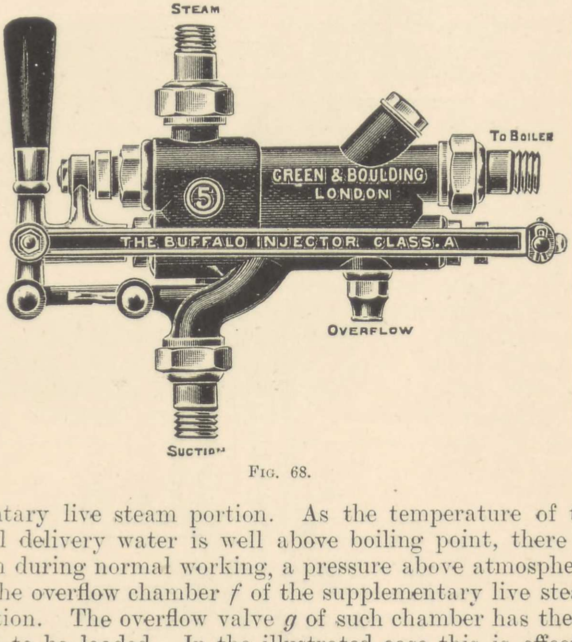


FIG. 68.

mentary live steam portion. As the temperature of the final delivery water is well above boiling point, there is, even during normal working, a pressure above atmospheric in the overflow chamber *f* of the supplementary live steam portion. The overflow valve *g* of such chamber has therefore to be loaded. In the illustrated case this is effected by the pressure in the delivery chamber of the supplementary injector through the plunger *h* and lever *i*. When exhaust steam is not available for working the exhaust portion of the injector, live steam throttled down to atmospheric pressure is supplied to the nozzle *a*.

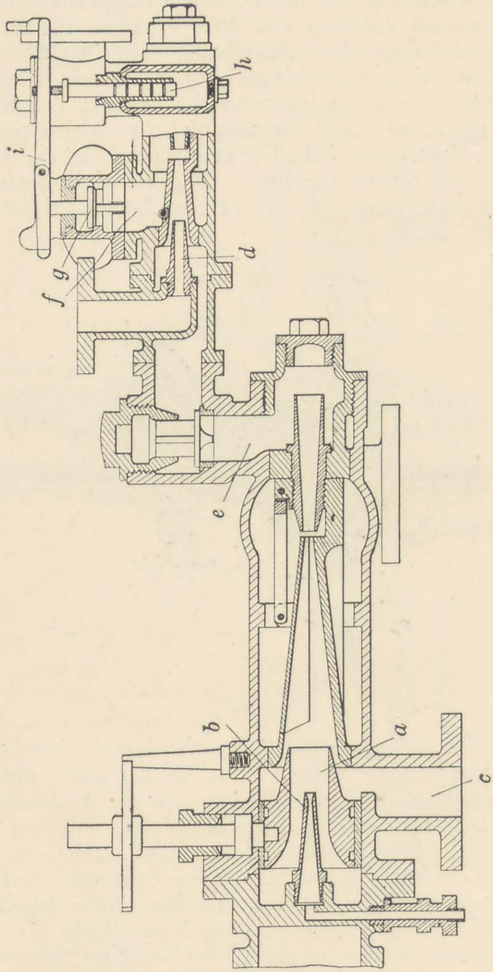


Fig. 69.

The operation of the compound exhaust injector illustrated at fig. 69 may be briefly described as follows: The exhaust steam at a temperature of 212 deg. Fah. coming from the engine or other source of supply passes first through a grease separator, if such is required, and then enters the injector, meeting the feed water, which is under a head, and which is at a temperature of, say, 50 deg. Fah. Condensation of the steam immediately takes place and a vacuum is formed. The small live steam nozzle *b*, known as the inducer, serves to increase the flow of exhaust steam to the injector. The combined jet of condensed steam and water rushes through the combining nozzle into the delivery nozzle of the exhaust injector. In the latter nozzle the kinetic energy of the jet is converted into pressure energy, and the jet leaves the exhaust injector at a temperature of about 180 deg. Fah.; the temperature of the feed water has been raised 130 deg. Fah. in the exhaust portion of the injector. The delivery from the latter now passes to the supplementary live steam portion. Here it is acted on by a jet of live steam, which raises the temperature of the water from 180 deg. Fah. to about 280 deg. Fah., and gives it sufficient pressure to overcome that in the boiler which is being fed. The very high final deliverery temperature will be noted. If a live steam injector only had been feeding the boiler under the same conditions as the compound exhaust injector, the delivery temperature would have been about 160 deg. Fah.

Questions.

(1) Define an injector, and describe the essential parts of same, giving particulars of their functions.

(2) The pressure of the steam entering the steam nozzle of a live-steam injector is 200 lbs. per square inch absolute, and the pressure at the delivery end of said nozzle is 5 lbs. per square inch absolute (vacuum of 20 in.). Describe and illustrate with rough diagrams the nature of the pressure drop in and beyond the nozzle if the latter is of (*a*) convergent and (*b*) divergent form. What shape do you recommend for the inlet to the nozzle throat, and what taper for the outlet from said throat? What is the effect of having the degree of divergence of the outlet too great?

(3) Derive a formula for calculating the ratio of areas of a steam nozzle throat and mouth to provide for any desired pressure drop between said points, and calculate the said ratio for steam of a boiler pressure of 150 lbs. per square inch absolute discharging into a vacuum of 28 in. (1 lb. per square inch absolute pressure). Can you give any simple approximate formula for obtaining said ratio of areas?

(4) What form of nozzle do you recommend to obtain the greatest velocity of discharge of steam? What is the effect of (*a*) under and (*b*) over expanding the steam upon its velocity of discharge? At what point of a steam nozzle does the greatest wear take place?

(5) Describe and illustrate by a sketch an arrangement of separate concentric lifting and forcing steam nozzles for an injector. What special overflow arrangement is necessary for this arrangement of nozzles? Describe and illustrate a form of valve for controlling both the lifter and forcer in an effective manner.

(6) Describe the function of an injector combining nozzle, and set out the chief factors which enter into the determination of the design or shape and length of the nozzle. What is considered a suitable length of combining nozzle (compared with the diameter of the delivery nozzle throat) for a hot-water injector?

(7) Where does the greatest wear take place in an injector combining nozzle and what are the causes of such wear? Describe and illustrate a form of combining nozzle which is particularly advantageous for use when wear is excessive.

(8) How do you recommend that the jet delivered from the combining nozzle be treated to obtain the greatest delivery pressure? Give a formula showing the effect of any treatment you propose upon the velocity of said jet.

(9) Draw an injector delivery nozzle of conical form of 3 in. length having its smallest diameter one-half the largest, and give velocity and pressure curves for said nozzle assuming the initial velocity to be 100 ft. per second and the initial pressure head of the jet to be zero.

(10) Design an injector delivery nozzle to give a pressure curve such that the difference between the pressures at any two equidistant sections of the nozzle is the same.

(11) What is the effect of employing a delivery nozzle with too rapid a taper? Where does the greatest wear of the delivery nozzle take place? Describe a method of minimising the effect of such wear upon the throat area of the nozzle.

(12) The velocity of the jet at the throat of the delivery nozzle of an injector is 150 ft. per second, the diameter of the throat is ·8 in., and the density of the jet at that point 59 lbs. per cubic foot. How much water does the injector deliver per hour? If the water entering the injector were cooled so that the density of the delivery jet was increased to 60 lbs. per cubic foot and the velocity to 160 ft. per second, what would be the increase in the amount of water delivered per hour?

(13) What provision would you make in an injector to make it a lifting appliance? What is meant by an automatic appliance? Describe three forms of automatic arrangements.

(14) Discuss the variations of pressure in the combining nozzle at various points in its length, and describe a satisfactory arrangement of overflow apertures or the like which provide for automatic working of the injector.

(15) What is the difference in the requirements as to overflow arrangements between an injector working with cold feed water and one working with hot feed water and high-pressure steam? Describe an arrangement of overflow apparatus suitable for the latter injector.

(16) Why is the pressure in the delivery chamber of an injector not entirely satisfactory for loading the overflow valves? Describe a method of neutralising the disadvantages of the use of such pressure for such purpose.

(17) Discuss the requirements of injector overflow arrangements.

(18) What is the effect of varying the pressure of the steam entering an injector whilst maintaining the supply of water and the water temperature constant? Discuss the matter in all its bearings.

(19) Describe a means for automatically varying the water supply to an injector to suit requirements, and set out the principle on which it is based.

(20) What is an adjustable injector? Describe and sketch one form of same.

(21) What is a "one-movement" injector. Discuss the advantages or disadvantages of controlling the water supply of an injector by (*a*) moving the steam nozzle, (*b*) moving the combining nozzle, (*c*) water valves external to the injector.

(22) How is the velocity of discharge of steam affected by the presence of moisture in the steam? Give a velocity formula, and make one or more calculations to show the effect you set out.

(23) Give a general formula, based upon the pressure and volume of a fluid before and after discharge and upon a coefficient, for calculating the velocity of discharge of a fluid. What is the value of the coefficient for (*a*) dry saturated steam, (*b*) superheated steam, and (*c*) air, all expanding adiabatically.

(24) What is meant by the expression "maximum weight flow of steam"? Fully describe the phenomenon, and discuss the conditions necessary to obtain same.

(25) Calculate the weight of steam discharged per

minute from a boiler under a pressure of 219 lbs. per square inch absolute into the atmosphere through an aperture of ·006 square foot area. An approximate formula may be used for this calculation. What would be the difference in the discharge if the exhaust pressure, instead of being atmospheric, were (*a*) 131 lbs. per square inch absolute, and (*b*) 160 lbs. per square inch absolute?

(26) Compare the velocity, mass, momentum, and kinetic energies of two fluids of different densities issuing through orifices of equal cross-sectional area from a vessel under pressure. What is the water velocity equivalent of 23 lbs. pressure per square inch absolute, taking the density of water as 60 lbs. per cubic foot?

(27) An injector under test gives the following result:—

Steam pressure, lbs. per sq. inch absolute.	Feed-water temperature, deg. Fah.	Height of lift.	Pressure at point where steam and water unite.	Delivery temperature, deg. Fah.	Delivery pressure, lbs. per sq. inch absolute.
100	60	3 ft.	5 lbs. per sq. in. abs	140	120

(*a*) Calculate the ratio of water to steam passing through the appliance

(*b*) Calculate the theoretical velocity of inflow of the steam and water.

(*c*) Calculate, neglecting all consideration of losses, what the delivery pressure and temperature of said injector should have been under ideal conditions.

(*d*) Calculate the heat units (contained in each 1 lb. of steam used) lost from the injector by radiation, the heat units actually used to produce the delivery jet, and give the number of heat units utilised in heating the feed water.

(*e*) Calculate the efficiency of the injector as a pump.

(28) What are the quantities or values which affect the ratio of water to steam passing through an injector?

(29) Describe an exhaust injector, pointing out its limitations. What is the chief difference between an exhaust and a "live" steam injector? Can an exhaust injector lift its feed water? If so, under what conditions?

(30) What methods are practicable for increasing the delivery pressure of an injector without altering its constructional details? What are the limitations to such methods?

(31) Describe the reasons for and the precautions necessary in fitting up and using an exhaust steam injector.

(32) Estimate the economy of steam and water obtained

(33) Describe a compound or double injector, and discuss

by the use of an exhaust injector.

its merits or demerits as compared with a simple injector having concentric lifting and forcing steam nozzles.

(34) Describe a form of compound injector in which exhaust steam is employed along with live steam, giving particulars of the approximate gradations of pressure and temperature of the delivery jet as it passes through the appliance.

(35) An injector using steam of 120 lbs. per square inch absolute pressure, water at 100 deg. Fah., and having the overflow pressure atmospheric, the delivery pressure 145 lbs. per square inch absolute, and the delivery temperature 180 deg. Fah., draws 14,400 lbs. of water per hour from the hot well. Ascertain approximately the throat area of the steam nozzle and the throat area of the delivery nozzle.

(36) What is meant by the maximum velocity of efflux of steam from a vessel? Give a formula to determine this maximum. Is there such a thing as a maximum velocity of discharge (independent of the exact exhaust pressure conditions) into an exhaust space?

JOHN HEYWOOD LTD., Excelsior Printing and Bookbinding Works, Manchester.

CATALOGUE
OF
STANDARD
TECHNICAL BOOKS

For Reference and for Help
in Systematic Study of Scientific
and Engineering Subjects

FOR

ENGINEERS,
MECHANICS,
DRAUGHTSMEN, and
SCIENCE STUDENTS.

Any Book herein mentioned can be sent to any address at home or abroad upon receipt of published price and postage.

LONDON, ENGLAND:

THE TECHNICAL PUBLISHING CO., Ltd.,

55-56, CHANCERY LANE, W.C.

LIST OF STANDARD

TECHNICAL BOOKS

FOR

Engineers, Mechanics, Science Students, &c.

WORKSHOP COSTS (for Engineers and Manufacturers). By SINCLAIR PEARN and FRANK PEARN (Directors of Messrs. Frank Pearn and Co., Ltd., West Gorton Ironworks, Manchester, with a Preface by G. Croydon Marks, M.P., Assoc. Memb. Inst. C.E., M.I.M.E., &c. Second Edition. Post folio, bound in roan with numerous Specimen Workshop Cost Forms. Price 12s. 6d. net. post free 13s. net, abroad 13s. 6d. net. 1.

"It is not often that such an elaborate work, and one of which the utility is undoubted, is placed at the disposal of the public interested in such questions at a price which certainly is insufficient to repay the cost of publication."—*The Times Engineering Supplement.*

"To managers and those interested in the control of engineering works the book affords a striking object lesson as to what can be done in the way of cost-keeping."—*Textile Recorder.*

"After a careful perusal of the notes and examination of the examples given, we are able fully to endorse the authors' claims as to the reliability of the system."—*British Trade Journal.*

"This is a book which ought to find a place in the offices of every workshop."—*Page's Magazine.*

"Manufacturers and engineers whose work lies in this department will find the book invaluable."—*The Steamship.*

THE MANAGEMENT OF SMALL ENGINEERING WORKSHOPS. By ARTHUR H. BARKER, B.A., B.Sc., Wh.Sc., &c., Author of "Graphical Calculus," "Graphic Methods of Engine Design," &c. Crown 8vo. Cloth, price 7s. 6d. net, post free 8s. net, abroad 8s. 6d. net. 2.

"We may congratulate Mr. Barker upon this book. . . . The treatment of the subject is extremely satisfactory."—*Textile Recorder.*

"There is much sound advice in this work, and the proprietors of small shops, as well as their managers, may derive much benefit from a perusal of its pages."—*English Mechanic.*

"May be conscientiously recommended as containing much sound advice on a matter of the highest importance in connection with manufacturing operations."—*Journal of Gas Lighting.*

"As a generally suggestive scheme for the efficient control of small engineering workshops, we would certainly commend Mr. Barker's book, while those in charge of larger concerns may profit not a little by perusing it."—*Mechanical World*

ENGINEERING ESTIMATES AND COST ACCOUNTS. By F. G. BURTON, formerly Manager of the Milford Haven Shipbuilding and Engineering Company, Ltd. Second Edition. Crown 8vo. Cloth, price 3s. net, post free 3s. 3d. net, abroad 3s. 6d. net. 3.

"Contains many useful hints and valuable information."—*The Builder.*

"To all who have to do with engineering estimates and cost accounts, this treatise should prove eminently useful. It is to be heartily commended to the notice of managers, foremen, and draughtsmen."—*Steamship.*

"The author has undoubtedly a thorough grasp of his subject. . We have formed a high opinion of the work. It is, perhaps, the best book of its kind and price in the market, and has our hearty commendation."—*Science and Art of Mining.*

THE ELEMENTS OF ENGINEERING ESTIMATING. A Practical Book for Managers, Engineers, and the Works Office. By A. SUGGATE, A.M.I.C.E. Crown 8vo. 135 pages. Price 3s. 6d. net, post free 3s. 9d. net, abroad 4s. net. 4.

THE PROPORTIONS AND MOVEMENT OF SLIDE VALVES. By W. D. WANSBROUGH, author of "The A B C of the Differential Calculus," &c. Price 4s. 6d. net, post free 5s. net, abroad 5s. 6d. net. 5.

"For those who desire an intimate knowledge of the subject, the book is one of the best."—*Power.*

"The book should prove a useful contribution to the literature of slide valve problems."—*Marine Engineering.*

"This publication will come as a boon to all who are in touch with, or controlling, engines of any class."—*Irish Engineering Review.*

"Students, draughtsmen, and engineers generally, who are in search of a complete investigation of slide valve problems, should find in Mr. Wansbrough's work a most useful aid."—*Page's Magazine.*

GOVERNORS AND GOVERNING MECHANISM. Second and Enlarged Edition. Crown 8vo, 120 illustrations and folding plates. By R. H. HALL, Mech. Engineer. Price 4s. net, post free 4s. 6d. net, abroad 5s. net. 6.

"A clever and comprehensive treatise."—*Page's Magazine.*

"This is an excellent book, and covers the field in a fairly comprehensive manner."—*Power.*

"May be heartily recommended to our readers as an excellent epitome of the subject."—*Model Engineer.*

"This useful little book will, no doubt, attract many readers amongst practical engineers."—*The E ectrician.*

"A credit to the author, and to the press from which it is issued, and extremely useful to all engaged in engineering."—*Irish Engineering Review.*

STEAM ENGINE DESIGN. By H. E. KING. For the use of Engineers, Draughtsmen, and Students. [*Ready Shortly.* 54.

SPECIFICATION FOR A LANCASHIRE BOILER (Working Pressure 200 lb.) By "INSPECTOR," M.I.M.E. Cloth, 2s. 6d. net, post free 2s. 9d. net, abroad 3s. net. 13.

"A book which we can well recommend to our readers' favourable notice."—*The Contractor.*

"In the hands of a prospective buyer of boilers this little brochure would help to solve many questions of a practical nature."—*The Ironmonger.*

"This work is one that all who deal with Lancashire boilers may take up with advantage if they desire information upon the boiler or its seating."—*Surveyor.*

WORKING OF STEAM BOILERS, being Instructions respecting the Working, Treatment and Attendance of Steam Boilers. By EDWARD G. HILLER, Chief Engineer, the National Boiler and General Insurance Co., Limited. Price, paper covers 1s., cloth 1s. 6d. net, post free 1s. 9d. net, abroad 2s. net. B 45.

NOTES ON THE MATERIAL, CONSTRUCTION, AND DESIGN OF LAND BOILERS. By EDWARD G. HILLER, M.I.C.E., M.I. Mech. E., Chief Engineer, National Boiler and General Insurance Co., Limited. Price, paper covers, 1s. net, post free 1s. 3d. net, abroad 1s. 6d. net; cloth, 1s. 6d. net, post free 1s. 9d. net, abroad 2s. net; leather, 2s. 6d. net, post free 2s. 9d. net, abroad 3s. net. B 31.

NOTES ON BOILER TESTING. By FREDERICK GROVER, M.I. Mech. E., A.M. Inst. C.E., &c. Price 1s., post free 1s. 3d. net, abroad 1s. 6d. net. 14.

PRACTICAL NOTES ON THE CONSTRUCTION OF CRANES AND LIFTING MACHINERY. By EDWARD C. R. MARKS, Assoc. M. Inst. C.E., M.I. Mech. E. New and Enlarged Edition. Crown 8vo. Cloth, price 3s. 6d. net, post free 4s. net, abroad 4s. 3d. net. 15.

"We can heartily recommend this book to all enquirers and students interested in the subject."—*Model Engineer.*

"The book is to be recommended as the best on the subject, and will be found of the greatest service to all who have to use or design lifting machinery.—*The Steamship.*

"The author of this book has had splendid facilities for becoming practically acquainted with the subject upon which he writes. . . . The book is admirably illustrated throughout, and is a most useful addition to the literature on this subject."—*Engineering Times.*

AIR COMPRESSORS AND BLOWING ENGINES. Specially Adapted for Engineers. By CHAS. H. INNES, M.A., Lecturer on Engineering at Rutherford College, Newcastle-on-Tyne. Crown 8vo. Price 4s. 6d. net, post free 5s. net, abroad 5s. 3d. net. 16.

"One of the most useful handbooks on the subject we have yet seen."—*Mechanical World.*

"Is by no means the least valuable of this thoroughly practical series."—*The Ironmonger.*

"To all who wish to study and know about air compressors this book is invaluable."—*The Steamship.*

"This book deals as thoroughly with the subject of air compressors as any work with which we are acquainted."—*Engineering Times.*

"For those who can readily handle mathematics, the theoretical treatment of the subject is good."—*The Engineer.*

THE FAN: INCLUDING THE THEORY AND PRACTICE OF CENTRIFUGAL AND AXIAL FANS. By CHARLES H. INNES, M.A., Lecturer on Engineering at Rutherford College, Newcastle-on-Tyne, author of "Centrifugal Pumps, Turbines, and Water Motors," and "Problems in Machine Design." Crown 8vo. Cloth, price 4s. net, post free 4s. 6d. net, abroad 4s. 9d. net. 17.

"This book is one of a series of admirable text-books issued by the same publishers."—*Mining Journal.*

"This book constitutes another valuable addition to the admirable series of text-books on applied science."—*British Trade Journal.*

"The book probably contains all that can be usefully said of the fan, and is therefore to be warmly commended to all interested in the design and construction of such apparatus as from under the title adopted."—*Mechanical World.*

"This work is a complete mathematical treatment of the subject . . . . and is one of the best works on this particular subject we have yet reviewed."—*Power.*

"To the mining engineer and to others well versed in mathematics this book will appeal with a special interest."—*Iron and Coal Trades Review.*

VENTILATION AND DUST REMOVAL IN THE CARD AND GASSING ROOMS OF COTTON MILLS. By MARKS and CLERK. Over 60 illustrations. Gives a clear insight into the latest systems adopted. Crown 8vo. Cloth, price 2s. 6d. net, post free 2s. 9d. net, abroad 3s. net. B 49.

"Anyone who is directly interested in the subject will find the book a handy one."—*Textile Recorder.*

MACHINE DRAWING AND SKETCHING FOR BEGINNERS. By J. H. ROBSON, A.M.I.M.E., Lecturer on Engineering at the L.C.C. School of Engineering and Navigation. Illustrated by 350 illustrations, mostly plates. Nearly 200 pages. Specially adapted for class work and students generally. A book of exceptional value. Large 8vo. Cloth, price 2s. 6d. net, post free 3s. net, abroad 3s. 3d. net. 18.

THE APPLICATION OF GRAPHIC METHODS TO THE DESIGN OF STRUCTURES. Profusely illustrated. A Treatment by Graphic Methods of the Forces and Principles necessary for consideration in the Design of Engineering Structures, Roofs, Bridges, Trusses, Framed Structures, Walls, Dams, Chimneys, and Masonry Structures. By W. W. F. PULLEN, Wh.Sc., A.M. Inst. C.E., M.I. Mech. E. Second Edition. Crown 8vo. Cloth, price 5s. net, post free 5s. 6d. net, abroad 5s. 9d. net. 19.

"Engineers generally will find this work of much value."—*Steamship.*

"It has established its reputation as a handbook of value for engineers."—*Daily Telegraph.*

"Written by an expert . . . for experts, this manual should prove of much usefulness to all classes of engineers."—*The Scotsman.*

"The fact that a second edition of this work has been called for is evidence of its increasing popularity. . . . As a work exclusively devoted to graphic statics, this treatise is the best of its class. . Our advice is, get it."—*Canadian Engineer.*

GRAPHIC METHODS OF ENGINE DESIGN. By A. H. Barker, B.A., Wh.Sc., author of "Graphical Calculus," &c. Second Edition. Crown 8vo. Cloth, price 3s. 6d. net, post free 4s. net, abroad 4s. 3d. net. 20.

"Students of mechanics will find this book exceptionally interesting and useful."—*The Steamship.*

"This work bears evidence of great care in its preparation, and should prove useful to many young engineers."—*Engineers' Gazette.*

"The fact that this concise little work has entered upon its second edition should be sufficient to indicate the value of the volume to the young mechanic."—*Page's Magazine.*

"This is a useful little work. . . . The mechanism of the steam engine affords a very complete series of illustrations. . . . The book gives a clear exposition of their application to practical work."—*Iron and Coal Trades Review.*

PROBLEMS IN MACHINE DESIGN. For the Use of Students, Draughtsmen and others. By C. H. Innes, M.A., Lecturer on Engineering at the Rutherford College, Newcastle-on-Tyne. Second Edition. Crown 8vo. Cloth, price 4s. 6d. net, post free 5s. net, abroad 5s. 3d. net. 21.

"Of much value to draughtsmen and engineers, to whom we cordially recommend it."—*Mechanical World.*

"That the book will be valuable to many there can be no doubt, and it is to be cordially recommended to the notice of engineering students."—*Steamship.*

LOCI IN MECHANICAL DRAWING. PART III. Piston Acceleration. By Alexr. MacLay, B.Sc., C.E., M.I.M.E., &c. Demy 8vo. Cloth, with numerous illustrations, price 2s. 6d. net, post free 2s. 9d. net, abroad 3s. net. 22.

"The comprehensive and clear treatment of this subject in the present work should at once make the book a standard of reference on this topic, and it should form a most acceptable addition to the working library of every draughtsman and designing engineer."—*Marine Engineering.*

MODERN GAS AND OIL ENGINES. Profusely illustrated. A full and exhaustive Treatment of the Design, Construction and Working of Gas and Oil Engines up to date. By Frederick Grover, Assoc. M. Inst. C.E. Fifth New and Enlarged Edition. Crown 8vo. Cloth, price 5s. net, post free 5s. 6d. net, abroad 5s. 9d. net. 23.

"It is clear that those interested in gas and oil engines have recognised the usefulness of this work."—*The Ironmonger.*

"This fresh matter comes as a welcome addition to a work whose value is already appreciated by practical engineers."—*The Builder.*

' In its revised form it should meet with the increased appreciation of draughtsmen and others interested in the design and operation of gas and oil engines."—*Mechanical World.*

"'Modern Gas and Oil Engines' is an almost ideal text-book on the important subjects which occupy its pages. It is no reflection upon other eminent men who have written, and written with great ability, to say that Mr. Grover's work surpasses them all in its concise, lucid, and thorough practical line of exposition." —*Science and Art of Mining.*

HEAT AND HEAT ENGINES. A Treatise on Thermodynamics, as Practically Applied to Heat Motors. Specially written for Engineers. By W. C. Popplewell, M.Sc. Crown 8vo. Cloth, price 6s. net, post free 6s. 6d. net, abroad 6s. 9d. net. 24.

"The volume is one which we have pleasure in commending to the attention of those engineers who wish to gain a clear conception of the elementary laws of the heat engine."—*The Engineer.*

THE THETA-PHI DIAGRAM. Practically Applied to Steam, Gas, Oil and Air Engines. By HENRY A. GOLDING, Assoc. M. Inst. M.E. 3s. net, post free 3s. 4d. net, abroad 3s. 6d. net. 25.

"The most exhaustive and complete treatment upon the subject of entropy that we have seen."—*Machinery.*

"The work evidently supplies a void, as hitherto the information it gives was scattered about in the "Proceedings" of technical societies. . . . The work is well timed, and should prove useful to a large circle of readers."—*Engineer's Gazette.*

"There can be no question of the importance to designers of steam engines of a careful study of the Theta-Phi Diagram, and it must be conceded that Mr. Golding provides a work which renders this knowledge readily accessible."—*British Trade Journal.*

MODERN POWER GAS PRODUCER PRACTICE AND APPLICATIONS. By HORACE ALLEN. Price 6s. net, post free 6s. 6d. net, abroad 6s. 9d. net. 26.

Should be of great practical value to those who are installing producer plant." —*Page's Weekly.*

"Will be a useful addition to an engineer's library."—*Times.*

"The book should be heartily welcomed by every one interested in the subject.' —*Power.*

"It is a useful and up-to-date treatise on a subject of growing importance."—*Mechanical World.*

EXPERIMENTS ON THE PRESSURE AND EXPLOSIVE EFFICIENCY OF MIXTURES OF ACETYLENE AND AIR. By FREDERICK GROVER, A.M. Inst. C E., M.I. Mech. E., Author of "Modern Gas Engines" &c. Price 1s., post free 1s. 1d. net, abroad 1s. 3d. net. B 19.

PRINCIPLES OF FITTING. By J. G. HORNER. Price 5s. net, post free 5s. 3d. net, abroad 5s. 6d. net. B 52.

TURNING LATHES. A Manual for Technical Schools and Apprentices. A Guide to Turning, Screw-cutting, Metal Spinning, &c. &c. With 194 Illustrations. By J. LUKIN, B.A. Demy 8vo. Cloth, price 3s. net, post free 3s. 4d. net, abroad 3s. 6d. net. B 46.

THE RESISTANCE AND POWER OF STEAMSHIPS. By W. H. ATHERTON, M.Sc., and A. L. MELLANBY, M.Sc. Price 5s. net, post free 5s. 4d. net, abroad 5s. 6d. net. 27.

"The book will appeal to marine engineers and shipbuilders, and, in fact, to all who are interested in watching the development of steamships."—*Scientific American.*

"The book merits a place as a standard aid to the marine engineer and shipbuilder."—*The Steamship.*

"One of the admirable series of text-books published by the Technical Publishing Co., Ltd. . . . The authors are to be congratulated."—*British Trade Journal.*

MARINE ENGINEERS: THEIR QUALIFICATIONS AND DUTIES. With Notes on the Care and Management of Marine Engines, Boilers, Machinery, &c. By E. CONSTANTINE, Assoc. M. Inst. C.E., M.I. Mech. E. Second Edition. Crown 8vo. Cloth, price 5s. net, post free 5s. 4d. net, abroad 5s. 6d. net. 28.

"It is a masterly book of instruction, and the author can be congratulated upon having produced a really useful and serviceable manual."—*Western Mail.*

"We record with pleasure the advent of the second edition of this both useful and interesting book."—*The Model Engineer.*

"We recommend this book to all who desire to become sea-going engineers."—*The Steamship.*

THE NAVAL ENGINEER AND THE COMMAND OF THE SEA. By Francis G. Burton, author of "Engineering Estimates and Cost Accounts," &c. Crown 8vo. Cloth, price 2s. 6d. net, post free 2s. 9d. net, abroad 3s. net. 29.

"The story is written by one who has obviously carefully studied the subject and knows what he is writing about."—*Naval and Military Record.*

TRUTH ABOUT THE BELLEVILLE BOILER IN THE BRITISH NAVY. By Engineer Wilcke. Price 1s. 6d. net, post free 1s. 9d. net, abroad 2s. net. 30.

CONTINUOUS CURRENT DYNAMOS AND MOTORS AND THEIR CONTROL. By W. R. Kelsey, B.Sc., A.I.E.E., F.Ph.S., Electrical Inspector under the Board of Trade to the Taunton Corporation. New and Re-written Second Edition. Crown 8vo. Cloth, price 7s. 6d. net, post free 8s. net, abroad 8s. 3d. net. 31.

"The book may be safely recommended to all who take a practical interest in continuous current machines."—*Model Engineer.*

"The book has many features which the practical designer and builder of dynamos and motors will appreciate."—*Mechanical World.*

"We can confidently recommend this book as a simply-written and thoroughly practical introduction to the study of continuous current machinery."—*Electrical Magazine.*

ELECTRIC LIGHTING FOR MOTOR CARS. By Marks and Clerk. Primarily intended for Designers and Manufacturers of Electric Lighting Apparatus for Motor Vehicles. Contains special chapters on "The Theory of Motor Car Headlights," by A. C. W. Aldis, Esq., M.A. (Cantab), and "Photometric Observations," by G. A. Shakespeare, Esq., M.A., B.Sc., Lecturer at University of Birmingham. Nearly 50 Illustrations. Crown 8vo. Cloth, price 3s. 6d. net, post free 3s. 9d. net, abroad 4s. net. B 15.

"The book is not likely to stop at the first edition."—*The Electrician.*

ELECTRICAL TERMS AND PHRASES. In Shorthand (Pitman's). Especially valuable to the Shorthand Writer in Engineering Offices. By E. J. Rutland. Price 1s. net, post free 1s. 3d. net, abroad 1s. 6d. net. B 21.

ELECTRICAL MEASURING INSTRUMENTS AND SUPPLY METERS. By H. W. Moreton. Fully illustrated. Crown 8vo. Cloth. [*Ready Shortly.* 55.

THE A B C OF THE DIFFERENTIAL CALCULUS. By W. D. Wansbrough, author of "Portable Engines," "Proportions and Movement of Slide Valves," &c. Crown 8vo. Cloth, price 3s. net, post free 3s. 3d. net, abroad 3s. 6d. net. 32.

"If any man can grasp the Calculus, the author of this little book should enable him to do so, for he has put the matter as clearly as it is possible to do."—*Cassier's Magazine.*

"The book is the clearest exposition by far of the fundamental theories of the Calculus we have seen. . . Mr. Wansbrough's book is really entertaining; one smiles at its simplicity."—*The Engineer.*

"Having in view the character of the subject, the fact of a second edition being required is evidence that the attempt has been successful, and that the 'painless treatment for differential calculus' has at last been effected."—*Mechanical World.*

"We should recommend every student of mathematics who desires to thoroughly grasp the essential principles of the Calculus to make use of this well-named A B C. . . We were beguiled into reading the work right through . . . and are bound to admit that the difficulties of the Calculus seemed to disappear one after another."—*Page's Magazine.*

MACHINES AND TOOLS EMPLOYED IN THE WORKING OF SHEET METALS. By R. B. HODGSON, A.M.I. Mech. E., author of "Emery Grinding Machinery." Crown 8vo. Cloth, price 4s. 6d. net, post free 5s. net, abroad 5s. 3d. net. 33.

"There is a scarcity of technical literature on Presses, Press Tools, &c. Mr. Hodgson's book will be welcomed by engineers and students desirous of gaining a better knowledge of theoretical and practical sheet metal working. . . . Those of our readers who are interested in the working of sheet metals will find Mr. Hodgson's book very interesting and useful."—*Iron and Steel Trades Journal.*

"A most valuable and helpful handbook on the subject. It covers a very wide range. The explanations are singularly clear and concise."—*Birmingham News.*

"The book gives a comprehensive and clearly written account of its subject, fully illustrated by plans and diagrams, and full of useful instruction."—*The Scotsman.*

"A book which should prove advantageous to both employers and workmen engaged in iron and engineering trades. . . . In consequence of its admirable treatment the book is one that will convey to the ordinary working man that which it would be exceedingly difficult for him to grasp in any other form."—*Wolverhampton Express and Star.*

THE MANUFACTURE OF IRON AND STEEL TUBES. By EDWARD C. R. MARKS, Assoc. M. Inst. C.E., &c. &c. Second and Enlarged Edition. Crown 8vo. Price 5s. net, post free 5s. 6d. net, abroad 5s. 9d. net. 34.

"A very serviceable little guide to a most important industry."—*The Surveyor.*

"The book should prove a useful handbook for manufacturers and others interested in the subject."—*Scientific American.*

SCREWS AND SCREW MAKING. Including an account of the origin of Whitworth's, Swiss, British Association, and American Threads, &c., with 95 Illustrations. Demy 8vo. Cloth, price 3s., post free 3s. 4d. net, abroad 3s. 6d. net. B 41.

TIN-PLATE WORKING. By R. H. CLARKE. An Up-to-date Practical Handbook for the Tinsmith, Student and Amateur. Specially compiled for the Manual Training Teachers' City and Guilds Examination. Price 1s. 6d. net, post free 1s. 9d. net, abroad 2s. net. 35.

"A very practical handbook."—*Electrical Review.*

"Altogether a well thought out little manual."—*Ironmonger.*

"This well-illustrated and well-turned-out book deserves and has our hearty recommendation."—*Manual Training.*

SIMPLE STRUCTURAL WOODWORK. By GEORGE F. RHEAD (National Silver Medallist). The Popular Technical Manuals, No. 2. This book comprises a series of practical examples for schools, handicraft classes, and the home, planned to meet requirements of the City and Guilds Institute examinations. Crown 8vo. 120 pages. Price 2s. net, post free 2s. 3d. net, abroad 2s. 6d. net. 36.

"Manual training teachers and scholars will be glad to avail themselves of this little book."—*Work.*

"The purpose is well fulfilled, the text being lucid and practical and the illustrations excellent."—*Surveyor Municipal and County Engineer.*

SPLICING WIRE ROPES. By W. N. BRUNTON. Price 6d., post free 7d. net, abroad 9d. net. B 43.

THE EVOLUTION OF MODERN SMALL ARMS AND AMMUNITION. By E. C. R. MARKS. Price 5s. net, post free 5s. 6d. net, abroad 5s. 9d. net. B 44.

THE CHEMISTRY OF MATERIALS OF ENGINEERING. By A. HUMBOLDT SEXTON. Professor of Metallurgy, Glasgow and West of Scotland Technical College. Third Edition. Crown 8vo. Price 5s. net, post free 5s. 6d. net, abroad 5s. 9d. net. 37.

"An interesting and very useful text-book."—*Mechanical World.*

"An excellent book, we congratulate the author."—*Science and Art of Mining.*

"Not only a useful book for the beginner, but a convenient book of reference for the engineer."—*Railroad Gazette.*

"The book is one in which students of engineering will find much matter that is both interesting and useful."—*Glasgow Herald.*

MECHANICAL ENGINEERING MATERIALS. By ED. C. R. MARKS, Assoc. M. Inst. C.E., M.I. Mech. E., &c. &c. Second Edition. Price 2s. 6d. net, post free 2s. 9d. net, abroad 3s. net. 38.

"Mr. Marks is a practised author upon engineering subjects, and what he has to say he relates in a clear and concise fashion."—*Engineering Times.*

"This is undoubtedly the best book on this subject, and we have no hesitation in recommending it to all engineers and to those who are interested in this important subject."—*The Steam hip.*

"Amongst the many treatises on the subject of metallurgy the small work before us is entitled to a prominent place."—*Irish Engineering Review.*

SAND AND CLAY: THEIR ANALYSIS AND PHYSICAL PROPERTIES. By A. C. PASSMORE. Price 2s. net, post free 2s. 3d. net, abroad 2s. 6d. net. 39.

"The work serves its purpose admirably, and in many cases it will prove a handy book of reference."—*E tat's Gazette.*

"It offers a plain and practical statement of the analysis and physical properties of sand and clay in their relation to building work."—*Building World.*

METALLURGICAL MANUAL OF IRON AND STEEL: Their Structure, Constitution, and Production. A Practical Treatise for the Use of Manufacturers, Engineers, and Students. By HORACE ALLEN, Author of "Modern Power Gas Producer Practice and Applications," &c. &c. Nearly 400 pages. Crown 8vo. Price 6s. net, post free 6s. 6d. net, abroad 6s. 9d. net. 40.

MODERN IRONFOUNDRY PRACTICE. Vol. I.: Foundry Equipment, Materials Used, and Processes Followed. By GEO. R. BALE, Assoc. M. Inst. C.E. Crown 8vo. 400 pages. Price 5s. net, post free 5s. 6d. net, abroad 5s. 9d. net. 41.

"To the practical man 'Modern Ironfoundry Practice' should prove a decided acquisition."—*Science and Art of Mining.*

"To the engineering student, draughtsman, young moulder and patternmaker, the volume would appear to be of inestimable value."—*Model Engineer.*

"A useful and up-to-date work on foundry practice. . . . The book is well balanced and will be found a useful guide to modern practice in ironfounding."—*Mechanical World.*

MODERN IRONFOUNDRY PRACTICE. Vol. II.: Machine Moulding and Moulding Machines, Physical Tests of Cast Iron, Methods of Cleaning Castings, &c. &c. By GEO. R. BALE, Assoc. M. Inst. C.E. Crown 8vo. Price 3s. 6d. net, post free 4s. net, abroad 4s. 6d. net. 42.

"The machines illustrated and described are typical ones, besides which the reader will find a very exhaustive account of the physical tests of cast iron, which has now to be very thoroughly tested before being put into use."—*Scientific American.*

"This is the second and apparently concluding volume of an up-to-date and practical work on ironfounding, which we can recommend with confidence. . . . The chapter on testings is very complete, and taken altogether the complete work is a most satisfactory exposition of the science and art of ironfounding according to present-day practice."—*Mechanical World.*

THE PRINCIPLES OF IRONFOUNDING (AND FOUNDRY METALLOGRAPHY). By A. HUMBOLDT SEXTON, F.I.C., F.C.S., late Professor of Metallurgy at Glasgow and West of Scotland Technical College, Author of "Chemistry of Engineering Materials," &c, and JOHN S. G. PRIMROSE, A.G.T.C., A.I.M.M., Lecturer on Metallurgy, Glasgow Technical College, Member of the Iron and Steel Institute, Institute of Metals, &c. Price 6s. net, post free 6s. 6d. net, abroad 6s. 9d. net. 43.

HYDRAULIC POWER ENGINEERING. A Practical Manual on the Concentration and Transmission of Power by Hydraulic Machinery. By G. CROYDON MARKS, M.P., Assoc. M. Inst. C.E., M.I.M.E., Fellow of the Chartered Institute of Patent Agents. Second Edition, Revised and Enlarged. Price 10s. 6d. net, post free 11s. net, abroad 11s. 6d. net. B 27.

"Can be unhesitatingly recommended as a useful and up-to-date manual on hydraulic transmission and utilisation of power."—*Mechanical World.*

NOTES ON THE CONSTRUCTION AND WORKING OF PUMPS. By EDWARD C. R. MARKS, Assoc. M. Inst. C.E., M.I.M.E., &c. Second Edition. Crown 8vo. 260 pages. Price 4s. 6d., post free 5s. net, abroad 5s. 3d. net. 44.

"A convenient and useful book of reference."—*The Builder.*

"Should enable engineers and manufacturers to select the apparatus best suited to their requirements."—*Mechanical World.*

"The book is one which may be recommended to all who are interested in the field of engineering work."—*Marine Engineering.*

"Purchasers of pumps will find a collection of information in this volume which may save them much trouble."—*Page's Magazine.*

CENTRIFUGAL PUMPS, TURBINES, AND WATER MOTORS: including the Theory and Practice of Hydraulics (specially adapted for Engineers). By CHAS. H. INNES, M.A., Lecturer on Engineering at the Rutherford College, Newcastle-on-Tyne. Fourth Edition, enlarged. Crown 8vo. Price 4s. 6d. net, post free 5s. net, abroad 5s. 3d. net. 45.

"This extremely useful work, in its fourth edition, has received further additions, notably an extra chapter on centrifugal pumps and fans for high lifts and pressures."—*Model Engineer.*

"The fourth and enlarged edition of an admirably-written book on the centrifugal pump, &c."—*British Trade Journal.*

"Those who have wanted a simple treatment of the theory and design of centrifugal pumps and turbines will welcome this book, and we can recommend it."—*The Engineer.*

"To most of our readers this book must be favourably known. We have on many occasions recommended it to inquirers as the best book on the subject in English, at a reasonable price."—*American Machinist.*

HOW TO USE WATER POWER. By HERBERT CHATLEY, B.Sc. Price 2s. 6d., post free 2s. 9d. net, abroad 3s. net. 46.

"The author of this little book gives a succinct account of the principles of hydraulic engineering in an easily comprehendible form."—*Mechanical World.*

"For brevity, simplicity, and clearness this is one of the best little books we have seen, and we recommend it to readers interested in the application of water power."—*Mining Journal.*

INVENTIONS, PATENTS, AND DESIGNS. With notes and the full text of the new British Patents and Designs Acts of 1907 and 1908. By Sir G. CROYDON MARKS, M.P. Crown 8vo. 128 pages. Second and Popular Edition (revised and enlarged), bound in flush cloth 1s. net, post free 1s. 3d. net, abroad 1s. 6d. net. In full cloth, gilt lettering, price 2s. net, post free 2s. 3d. net, abroad 2s. 6d. net. 47.

". . . The author gives a short but clear summary of the nature of Letters Patent, and an outline of the simpler legal points with which the inexperienced inventor shou'd be acquainted before applying for a patent. . . . The advantages of provisional protection are well brought out, and the chapter on licensing inventions, which shows careful selection, is especially worth the attention of an intending lessor."—*The Times.*

"The information given throughout the book is most useful to persons unfamiliar with matters concerning patents."—*The Engineer.*

NOTES AND JUDGMENTS ON THE "WORKING" OF BRITISH PATENTS. By Sir G. CROYDON MARKS, M.P., A.M. Inst. C.E. These notes have been prepared for the guidance of those interested in the "working" of British Patents, and have reference mainly to Section 27 of the 1907 Act. Crown 8vo. Cloth. Price 2s. 6d. net, post free 2s. 9d. net, abroad 3s. net. B 48.

RAMIE (RHEA) CHINA GRASS, THE NEW TEXTILE FIBRE. All about it. A book for Planters, Manufacturers, and Merchants. By HERBERT A. CARTER. This book is the outcome of a demand for a text-book on the new textile fibre, owing to the great increase which has arisen in the spinning and weaving of it in Europe and the United States. Large crown 8vo. 140 pages, with numerous Illustrations. Cloth, price 5s. net, post free 5s. 6d. net, abroad 5s. 9d. net. 48.

"Mr. Carter has succeeded in placing in concise form a great deal of information on this subject."—*Textile World Record.*

". . . forms a useful text-book. . . of considerable service."—*Bulletin of the Imperial Institute.*

FLYING MACHINES: PRACTICE AND DESIGN, THEIR PRINCIPLES AND WORKING. By RANKIN KENNEDY, author of "Modern Engines and Power Generators," "Electrical Installations," "Marine Propellers and Internal Combustion," &c. &c. A comprehensive work, with over 60 Illustrations, popularly treated, but upon a strictly scientific basis, with new formulæ, rules, measurements, and principles. Large 8vo. Cloth, price 5s. net, post free 5s. 6d. net, abroad 5s. 9d. net. 49.

"The author has given a general review of the flying machine problems. . . . the idea being to enable the designer to proceed in a methodical manner."—*Royal Automobile Club Journal.*

"It is a genuine attempt to help the art forward."—*Cassier's Magazine.*

"A useful treatise. . . . the author going well into the subject of the design and working of aeroplanes."—*Mechanical World.*

OPENING BRIDGES. By GEO. WILSON, M.Sc. Price 2s. net, post free 2s. 3d. net, abroad 2s. 6d. net. 50.

PRACTICAL GYROSCOPIC BALANCING. By HERBERT CHATLEY, B.Sc. Eng. (Lond.), Author of "How to Use Water Power," "The Force of the Wind," "The Problem of Flight." Crown 8vo. Illustrated. Price 2s. net, post free 2s. 3d. net, abroad 2s. 6d. net. 52.

THE INDICATOR AND ITS DIAGRAMS: WITH CHAPTERS ON ENGINE AND BOILER TESTING. By CHARLES DAY, Wh.Sc. Including a Table of Piston Constants. This book is of a thoroughly practical character, and contains numerous diagrams from actual practice. Third Edition. Crown 8vo. Cloth, price 4s. 6d. net, post free 5s. net, abroad 5s. 6d. net. 7.

"The book is one of the most useful we have met with."—*Mechanical World.*

"A really excellent book, which we have pleasure in recommending."—*Electricity.*

"The book is notable for its lucid explanations. There is scarcely a point left untouched."—*Textile Recorder.*

"We have not the slightest doubt this useful little work will be thoroughly appreciated by all who peruse it."—*Machinery.*

"The second portion of the book contains a vast amount of valuable information, which should secure a large sale for the book. . . . It is well worth a place on the shelves of all progressive engineers, and scientific men generally."—*Age of Steel.*

THE MOTOR CAR RED BOOK. By W. C. BERSEY. A List of Motor Cars Manufactured or Sold in the United Kingdom, 1906–1912. Price 5s. net, post free 5s. 3d. net, abroad 5s. 5d. net. 53.

"For the concentrated character of the information, we know of no more useful reference book to the industry."—*The Motor Trader.*

THE MOTOR CAR IN THE FIRST DECADE OF THE TWENTIETH CENTURY. By W. EDEN HOOPER. A Souvenir and a Historical Survey of Mechanical Road Locomotion in England from Early Times to 1908. Amongst the unique Coloured and other Plates are the following reproductions:—Rare coloured French plate about 1830; Gurney's Steam Carriage, 1829; Hancock's Steam Omnibus, 1833; A Cartoon of 1829; First Road Steam Locomotive (Cugnot's), 1769; James Steam Carriage, 1829; A Sketch of Goldsworthy Gurney; Gurney's First Steam Carriage, 1827; Gurney's Steam Carriage going through Highgate Tunnel, 1828; Gurney's Steam Coach at Hyde Park Corner, 1829; Cartoon of 1840 by Leech, "The Horse Guards on Steam Mounts"; and many others equally historical and interesting. Price 42s. net, post free 43s. net, abroad 45s. net. B 63.

HOW TO INVENT AND HOW TO DEAL WITH INVENTIONS. By VABUSE. The aim of the author has been to place in the hands of inventors such information which will enable them to undertake the task of satisfactorily judging their productions. Price 6d. net, post free 7d. net, abroad 9d. net. 56.

MARINE ENGINEERING ESTIMATES AND COSTS. For those engaged in the Technical and Commercial Sides of Engineering and Shipbuilding. By C. R. BRUCE, M.I.N.A., M.I.E.S. Price 4s. 6d. net, post free 4s. 9d., abroad 5s. net. S 1.

IRON AND STEEL ANALYSIS. By A. CAMPION, F.I.C., F.G.S., Professor of Metallurgy in the Glasgow and West of Scotland Technical College. 80 pages, Foolscap 8vo. Strongly bound cloth. Price 2s. 6d. net, post free 2s. 9d., abroad 3s. net. Interleaved for notes, 6d. extra. S 2.

EVERY WEEK BUY

THE PRACTICAL ENGINEER

THE MACHINE TOOL ENGINEER, AND ENGINEERS' GAZETTE.

Price 2D. Weekly.

Brightly written.

Interesting and Instructive.

A Journal for the Scientist, the Practical Worker and the Student in Modern Engineering.

Full of Crisp News and Useful Comment with Valuable Formulæ and Workshop Receipts of Utility to all Branches of the Profession.

SUBSCRIPTION.

United Kingdom	12s. 6d. post free.
Colonies and Abroad ..	15s. post free.

Order from your Newsagent, or through all of Messrs. W H. Smith and Son's Bookstalls and Bookshops, or send **Two Stamps** for Current Number to

THE PRACTICAL ENGINEER

THE MACHINE TOOL ENGINEER,

And ENGINEERS' GAZETTE,

55 & 56, Chancery Lane, London, W.C., England.

10000 22/4/12