STEAM INJECTORS:

THEIR THEORY

AND

USE.

TRANSLATED FROM THE FRENCH OF

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PREFACE.

The following brief treatise was translated without abridgment from *Nouvelle Mecanique Industrielle:* M. LEON POCHET. It first appeared in its present form in *Van Nostrand's Magazine* for March and April, 1877.

The literature bearing on the subject of the Injector is not abundant, and to many engineers the philosophy of the action of the apparatus is obscure. In view of the fact that uses for the Injector have been devised other than the feeding of boilers, is judged to be a sufficient reason for the preparation of this little work.

STEAM INJECTORS.

GENERAL THEORY OF STEAM INJECTORS.

IT is some years since M. Giffard introduced the injector apparatus which bears his name and which filled the scientific world with profound astonishment.

This ingenious apparatus, in which a jet of steam heading out of a boiler enters into the same boiler bringing with it a quantity of additional water, seems to proceed in accordance with a philosophical law contradictory to the ordinary laws of physics. It was in appearance a sort of perpetual motion. If the mechanical properties of heat had been better known, nothing would have appeared more simple.

M. Reech published, in 1858 (in the Memorial du Genie Maritime), a theory founded upon laws of the old philosophy, which gives a good account of the functions of the Giffard injector.

The new theory which we proceed to give—of the Giffard injector in particular, and of steam injectors in general—rests upon the mechanical theory of heat. It seems the more important inasmuch as injectors are now applied to such a great variety of purposes.

We will describe first the action of an injector. A tube terminating in a conical pipe A, Fig. 1, discharges a jet of steam which comes from the boiler. This tube opens into a chamber B B, which communicates with a reservoir of cold water RR by a vertical tube CC. It is this water which is introduced into the boiler.

Under the action of the jet of steam the air in the chamber B is rarefied and the rarefaction results in the atmospheric pressure raising the water of the reservoir in the tube C. As soon as the cold water comes in contact with the steam in the chamber B a portion of the steam

is condensed and the apparatus is so regulated that this condensation is complete. Then we have, merely, a jet of liquid to pass on through the contracted

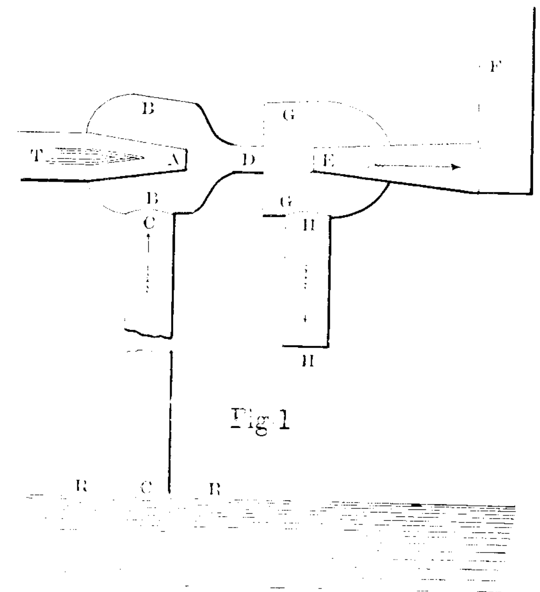


Fig 1

section D and be introduced into the conical diverging tube which communicates with the boiler. The tube EE is nothing more than a Venturi tube.

In such a tube, water introduced with

a certain velocity V would be able to overcome the pressure due to the height $\frac{V^2}{2g}$, provided that the liquid column be uninterrupted and the widening be progressive; or if we call ω the section of the tube at its origin, and ω' the section of the same tube at its entrance into the boiler, **H** the height of water corresponding to the effective pressure of the boiler, that is to say its absolute pressure diminished by one atmosphere, we shall have

$$\frac{V^2}{2\,g}\left(1-\frac{\omega^2}{\omega'^2}\right)=H.$$

In the Giffard injector we make

$$\frac{\omega}{\omega'}=0,16,$$

whence

$$\frac{\omega^2}{w'^2}=0,0256,$$

consequently

$$\frac{V^2}{2g}\times 0,9744=H.$$

But we should be able to give to the

ratio $\frac{\omega}{\omega'}$ greater value, and we can always assume approximately:

$$\frac{V^2}{2g}=H.$$

The condition that the liquid jet, through the tube E E, is uninterrupted, assumes that the jet is at too low a temperature to be transformed into steam at the atmospheric pressure in its passage through the chamber G G. Hence its temperature should be below 100°.

Should the temperature exceed 100° the working of the apparatus is imperfect. We are warned by a production of steam which fills up the chamber G G and which escapes by the discharge pipe H H.

Then we should remove the pipe A of the contracted orifice D in order to increase the useful section of this orifice, or, we should lessen the quantity of steam of the jet by diminishing the orifice A by the conical rod T. Thus the conditions of the proper working of the apparatus are:

1st. That all of the jet of steam A be condensed by affluent water;

2d. That the temperature of the mixture be lower than the temperature of corresponding saturation at the mean pressure of G.

In case that the chamber G communicates with the atmosphere, the temperature of corresponding saturation is 100°. But it might happen that there be any pressure in the chamber G. It may be higher or lower than the atmospheric pressure. We will make no special hypothesis upon its value.

Call t_0 the temperature of the boiler;

t the temperature of the jet of steam at the moment that it fills the chamber BB. This temperature will be the temperature of saturation corresponding to the mean pressure. It is easy to see that this pressure equals the atmospheric pressure diminished by the height of the column of water CC, if the feeding reservoir RR is lower, and augmented by this height if it is higher.

τ, the temperature of the liquid jet at

the moment that it enters the divergent tube EE. This temperature is lower than 100° when the chamber GG communicates with the atmosphere, which is the case in the Giffard injectors;

θ, the temperature of the water in the reservoir RR.

Suppose one kilogramme of saturated steam issuing from the pipe A and containing a proportion x of steam.

Let y be the weight of water which the kilogramme of steam raises and conveys from the reservoir RR.

Let w be the velocity of the steam jet in A;

V the velocity of the mixture at the entrance of the tube E. To solve the problem we will write two equations.

1st. We assume that since there is no heat lost or gained by the material of the apparatus, the sum of internal heat augmented by the calorific equivalent of the living force has not changed during the phenomena; for this heat should always appear under the form of heat, or under the form of living force. This is in accordance with the theorem of living force developed by the theory of heat.

2d The theorem of rational mechanics relating to momentum is here applicable, as it always is, whatever the exchange of heat may be, for in the equation of this theorem the interior forces disappear. The amount of internal heat above 0° in a kilogramme of steam of the jet **A** is:

$$\int_0^{t} ldt + (r - Apu)x.*$$

* The internal heat of any humid vapor, that is, part liquid and part vapor, is determined as follows:

Let x=the weight of the vapor in a kilogramme of the mixture.

Then $1-x$ the weight of the water in a kilogramme of the mixture.

Then the excess of internal heat of the vapor at $t°$ compared with water at 0° is,

$$(r + L - Apu)x.$$

In this expression,

L is the quantity of heat required to raise a kilogramme of the liquid from 0° to $t°$ when in contact with its vapor.

The calorific equivalent of the living force has for value $\frac{Aw^2}{2g}$.

The total quantity of heat is

$$\frac{Aw^2}{2g} + \int_0^{t} ldt + (r - Apu)x.$$

The internal heat of the affluent water is

$$y\int_0^{\theta} ldt.$$

A is the reciprocal of the Mechanical Equivalent of heat, or $\frac{1}{424}$.

p is the pressure.

r is the latent heat of the vapor.

u is the excess of volume of the vapor over the liquid which yielded it.

pu is therefore our expression for work;

And Apu is its heat equivalent.

Now, as the latent heat liquid in the vapor is $L(1-x)$, the total internal heat,

$Q = L(1-x) + (r + L - Apu)x$

$= L + (r - Apu)x.$

Or if l is the specific heat of the liquid at the temperature $t°$, then

$$L = \int_0^{t} ldt$$

$$Q = \int_0^{t} ldt + (r - Apu)x.$$

The living force is almost nothing and may be neglected; the internal heat augmented by the calorific equivalent of the living force before the mixture, will be then

$$\frac{Aw^2}{2g} + \int_0^t ldt + (r - Apu)x + y\int_0^\theta ldt.$$

The mixture forms, the steam is completely condensed, there is a diminution of volume and production of negative mechanical work. The total internal heat is augmented by the calorific equivalent of the work corresponding to the condensation, and which is

$$Apux.$$

When the mixture passes through the contracted section D, the sum of the living force and of the internal heat is

$$\frac{Aw^2}{2g} + \int_0^t ldt + rx + y\int_0^\theta ldt \quad . \quad . \quad (A)$$

The introduction commences in the tube EE, the total weight of the mixture is $(1+y)$, its internal heat, since it is entirely in a liquid state, is expressed by

$$(1 + y\int_0^t ldt,$$

And its living force is,

$$(1+y)\frac{AV^2}{2g},$$

since V is the common velocity.

The sum of internal heat above O, and of the calorific equivalent of the living force at the inlet of the tube EE, is therefore

$$(1+y)\left(\int_0^\tau ldt + \frac{AV^2}{2g}\right)$$

This new expression of the total heat should be equal to the expression in (A). We have the equation

$$\frac{Aw^2}{2g} + \int_0^t ldt + rx + y\int_0^\theta ldt = (1+y)\left(\int_0^\tau ldt + \frac{AV^2}{2g}\right) \quad . \quad . \quad (B)$$

Now in consequence of the fundamental formula for the cooling of vapors, we have

$$\frac{Aw^2}{2g} = \int_t^{t} ldt + r_0x_0 - rx.^*$$

Substituting this value in the preced-

* r_0 and r designating the quantities of heat of vaporization at the temperatures t_0 and t.

ing equations it gives, after some transformations,

$$\int_{\theta}^{t} ldt + r_{o}x_{o} = (1+y)\left(\int_{\theta}^{\tau} ldt + \frac{AV^2}{2g}\right) \quad (C)$$

This is the equation of living force. We will now establish the equations of momentum. We shall have them due to the contracted section D,

$$\omega V,$$

for this section is necessarily equal to that of the inlet of the tube EE and the weight of water has for its value

$$1000\omega V.$$

Its mass is

$$\frac{1000\omega V}{g}.$$

In short, the expression for momentum is

$$\frac{1000\omega V}{g} V = \frac{1000\omega V^2}{g}$$

The amount of work of the affluent water from the reservoir RR may be neglected. As for the steam jet A, the amount delivered per second is evidently equal to that of the final mixture divided by $(1+y)$, and its velocity is w and its

quantity of work is therefore, before its passage into section D

$$\frac{1000\omega}{g} V \cdot \frac{w}{1+y}.$$

and we find also for the increase of the momentum during a second from one side to the other of the contracted section D

$$\frac{1000\omega}{g} V\left(V - \frac{w}{1+y}\right)$$

Let us call π the pressure per square meter in the chamber GG, which is equal to the atmospheric pressure in the Giffard injectors, and P the pressure in the chamber BB which is equal to the atmospheric pressure diminished or increased by the column of water CC according as the supply reservoir RR is below or above the apparatus.

The contracted section D separates the jet into two parts, that above being subjected to the pressure P, and that below to the pressure π. The impulse of exterior forces during the exchange of velocities will be, per second

$$(P-\pi)\omega,$$

We have then, in accordance with the principle of equality of moments

$$\frac{1000\omega}{g}V\left(V-\frac{w}{1+y}\right)=(P-\pi)\omega,$$

whence

$$V\left(V-\frac{w}{(1+y)}\right)=\frac{(P-\pi)g}{1000}\quad . \quad . \quad . \quad . \quad (D)$$

The two equations (C) and (D) include the theory of steam injectors. I copy here:

$$\int_{\theta}^{t} ldt+r_0x_0=(1+y)\left(\int_{\theta}^{\tau} ldt+\frac{AV^2}{2g}\right), \quad (C)$$

$$V\left(V-\frac{w}{1+y}\right)=\frac{(P-\pi)g}{1000}. \quad . \quad . \quad . \quad . \quad (D)$$

GIFFARD INJECTORS FOR FEEDING BOILERS.

In the Giffard injector (Fig. 1) the reservoir R R is ordinarily near the chamber B B. the pressure P is nearly equal to the atmospheric pressure, and as the chamber G G is in communication with the atmosphere,—we can take

$$P=\pi,$$

and we ought to have

$$\tau<100^\circ.$$

Equation (D) of the moments reduces to

$$V-\frac{w}{1+y}=0,$$

whence

$$V=\frac{w}{1+y}.$$

Take now the equation (C). In the second member of that equation we may neglect the term $\frac{AV^2}{2g}$.

In effect the range of temperature τ and θ is always large enough. The following table demonstrates that V varies almost precisely as the difference $(\tau-\theta)$:

The equation (C) gives then approximately,

$$1+y=\frac{\int_{\theta}^{t_0} ldt+r_0x_0-\theta}{\tau-\theta} \quad . \quad (E)$$

Under this form we observe that y diminishes as τ increases. The mini-

Weight of Water Raised and Velocity of Mixture in Giffard Injector for Feeding Boilers.

		Pressure of the Boiler.								
		5 Atmospheres. (152°,22), $w = 714^m$.				3 Atmospheres. (133°,91), $w = 596^m$.				
Value of θ	Temperature of the Mixture.	Weight of Water raised per kilogr'm of steam suppos'd to be dry y	Velocity of the Mixture per second. V	Height to which the jet is raised. $\frac{V^2}{2g}$	Mechanical Work produced. $y\frac{V^2}{2g}$	Weight of Water raised per kilogr'm of steam suppos'd to be dry y	Velocity of the Mixture per second. V	Height to which the jet is raised. $\frac{V^2}{2g}$	Mechanical Work produced. $y\frac{V^2}{2g}$
		k.	m.	m.	km.	k.	m.	m.	km.
$\theta = 13°$	100	6,35	97,29	482,3	3063	6,29	81,73	340,5	2141
	80	8,55	74,87	285,7	2443	8,47	62,94	201,9	1710
	60	12,61	52,54	140,7	1774	12,50	44,15	99,4	1242
	40	22,70	30,17	46,4	1053	22,50	25,36	32,8	737
	20	90,42	7,82	3,12	282	89,63	6,57	2,21	197
	13	∞	0	0	0	∞	0	0	0
$\theta = 50°$	100	11,06	59,29	179,2	1981	10,95	49,88	126,8	1389
	80	19,10	35,57	64,5	1232	18,92	29,95	45,6	864
	60	59,50	11,85	7,17	425	58,74	9,97	5,07	298
	50	∞	0	0	0	∞	0	0	0

mum of y corresponds then with the maximum of τ, that is to say, at 100°.

We see also that y increases when θ increases: that is to say, in proportion to the heat of the feed water.

y increases in the same proportion that the temperatures τ and θ approach each other in value. Consequently when $\theta = 100°$ y is infinite.

Experience demonstrates that the action of the injector ceases before reaching this limit, at about 70°.

We are able then to state the following propositions.

1st. The proportion of conveyed water increases, and, consequently the velocity of the mixture diminishes when the temperature of the water in the supply reservoir is raised.

2. The proportion of conveyed water increases when the temperature of the mixture diminishes. It is a minimum when the temperature of the mixture is of 100°. The velocity then attains its maximum.

3d. It diminishes, on the contrary,

when the steam is not dry and the proportion of water which it contains increases.

On pages 20 and 21 is a table of the values of y and of V for the boiler pressures of five and three atmospheres and the temperatures of 13° and 50° of the feeding water, the steam being dry.

The proportion of water raised increases rapidly as the temperature τ of the mixture diminishes.

The velocity diminishes in nearly the same ratio.

The quantity of water raised corresponding to the boiler pressures five atmospheres and three atmospheres are nearly the same, but the velocities are widely different. If we would have the velocity, and. consequently the composition of the mixture corresponding to the pressure of the boiler, we must make

$$\frac{V^2}{2g} = \begin{cases} 41{,}32 \text{ for five atmospheres,} \\ 20{,}66 \text{ for three atmospheres.} \end{cases}$$

whence

$$V = \begin{cases} 28^m{,}50 \text{ for five atmospheres,} \\ 20^m{,}15 \text{ for three atmospheres.} \end{cases}$$

These numbers correspond to

$$y = \begin{cases} 24{,}06 \text{ for five atmospheres,} \\ 31{,}02 \text{ for three atmospheres.} \end{cases}$$

The temperature of the mixture is nearly 40° if the temperature of the feed water is 13°.

ACTION OF THE GIFFARD INJECTOR.

There are two ways of considering the action of the Giffard injector. We measure the mechanical work performed without taking into account the heat carried away by the mixture; or, with taking this heat into account.

Following the last mode of operation, which is the only rational one, when employed in feeding the boiler the injector performs good service. It is clear that, since there is no loss of heat outside, and that the final living force is nothing, all the heat carried away by the steam jet is restored in the mixture, excepting that corresponding to the mechanical work accomplished, $\frac{AV^2}{2g}y$. Equation (C) is the mathematical expression of this fact.

It is not necessary to take into account the friction in the tubes, if we consider them impervious to heat, for the friction produces heat which is not lost by external radiation, but is found in the internal heat of the mixture.

The quantity of heat augmented by the living force in the mixture at the moment of its entrance into the convergent tube counted above, the temperature θ (which is the exterior temperature, and which serves as the starting point for the temperature in the equation (C) is

$$(1+y)\left(\int_0^{\tau} l dt + \frac{AV^2}{2g}\right).$$

The portion $(1+y)\frac{AV^2}{2g}$ represents the heat equivalent of the living force of the mixture. This living force in part disappears in the work of introduction into the boiler. If things are so regulated that the height $\frac{V^2}{2g}$ precisely correspond to the relative pressure of the boiler, the living force $(1+y)\frac{V^2}{2g}$ is effectually destroyed by the back pressure of the boiler; the introduction into the boiler will have no velocity, and the quantity of heat introduced will be definitely

$$(1+y)\int_{\theta}^{\tau} l dt,$$

or about

$$(1+y)\ (\tau-\theta).$$

If the velocity V of the mixture is greater than that which corresponds to the relative pressure H, of the boiler (measured by a column of water), the mixture will possess a certain living force

$$\frac{V^2}{2g} - H,$$

at its entrance into the boiler, but this living force becomes heat, in the motions which it occasions, so that no more heat can disappear than the quantity

$$AH\ (1+y).$$

Thus we know whatever the velocity V of the mixture, there will disappear only the quantity of heat corresponding to the work

$$(1+y)\ H,$$

of the introduction into the boiler. This work comprises two terms:

$$y\ H \text{ and } 1 \times H.$$

The first represents the useful work in feeding; the second corresponds in reality to a quantity of heat which ought to be found in the heat of the mixture; this is a loss which is balanced by a previous gain made by the steam at the moment of exit from the boiler.

Suppose, then, that the velocity V of the mixture be precisely that which gives

$$\frac{V^2}{2g} = H,$$

so that the introduction of the mixture into the boiler be made without velocity: determine, by experiment, the quantity of water raised y, its temperature τ', and calculate the same temperature by the formula (E). The heat which should be brought back to the boiler is, theoretically,

$$(1+y)\ (\tau-\theta).$$

The heat brought back is, in reality,

$$(1+y)\ (\tau'-\theta).$$

There is then, practically, a loss of heat

$$(1+y)(\tau-\tau').$$

Add to this loss the loss of heat resulting from work accomplished,

$$\frac{AV^2}{2g}y,$$

we have for total loss

$$(1+y)(\tau-\tau')+\frac{AV^2}{2g}y.$$

This waste of heat applied to the introduction of y kilogrammes of water into the boiler, gives for waste of heat per kilogramme:

$$i=\frac{1+y}{y}(\tau-\tau')+\frac{AV^2}{2g} \quad . \quad . \quad (F)$$

Here is an experiment given by M. Reech. M. Giffard succeeded in feeding a boiler at five atmospheres, by conveying a weight of water equal to fifteen times that of the steam. The temperature of the mixture was 48°, the affluent water being 13°. We have then, in this experiment:

$$\theta=13^\circ$$
$$y=15.$$

The formula (E) gives

$$\tau=53^\circ,01.$$

Now, then, the experiment indicates

$$\tau=48^\circ.$$

Consequently,

$$\tau-\tau'=5^\circ,01.$$

Now the waste of heat per kilogramme of feed water is, (equation F):

$$i=\frac{16}{15}+5{,}01+\frac{41{,}32}{424}$$

$$=5{,}34+0{,}097=5.44 \text{ heat units.}$$

This number represents $\frac{155}{100}$ of the difference of temperature $(\tau-\theta)$.

This quantity of heat really disappears and is not found again. All other quantities of heat carried out of the boiler by the jet of steam y have been returned by the mixture.

Now, consider an ordinary supply pump, which introduces water into the boiler at the temperature of 13°. This pump is operated by the engine, and produces useful work,

AH

(H=height of water corresponding to the excess of the boiler-pressure over that of the condenser), per kilogramme of feeding water, but it utilizes only a fraction k of force by reason of the friction or resistances of various kinds.

Now this force is only that of the steam; and we know that ordinary non-condensing steam engines use scarcely three parts in 100 of the heat transmitted to the boiler: then the introduction of a kilogramme of water into the boiler, by means of a feed-pump, requires an expense of heat of

$$\frac{1}{0,03}\ \frac{AH}{k}.$$

The co-efficient k of the ordinary pump is 0,50, consequently this quantity of heat has for its value

$$66,6\ AH.$$

Observe that the water introduced is not of the temperature θ of the supply reservoir. To raise the temperature of this water from θ to τ it will be necessary to consume a quantity of heat

$(\tau-\theta)$, but as the heating apparatus never uses but $\frac{2}{3}$ of the heat developed by the combustion there will be $\frac{1}{3}$ of the heat lost. The total consumption of heat will be then,

$$j=66,6\ AH+\tfrac{1}{3}(\tau-\theta).$$

By feeding by means of an injector, we had found the loss of heat to be,

$$i=0,155\ (\tau-\theta).$$

It is easy to see that the first is the more important. Suppose, for example, a pressure of five atmospheres, we will have:

$$H=41.32,$$
$$\tau=48^\circ,$$
$$\theta=13^\circ,$$

$j=18.20$ heat units, $i=5.44$ heat units.

In this case, the cost of feeding by means of an injector is less than one-third of that of feeding by means of a pump.

These results would be somewhat modified if we had the use of a good condensing engine. Here the work of the engine is 10-100; that of the pump, if well established, 60 100, introducing water from the condenser at a temperature of 50°. The injector should work equally with the water of the condenser. We should always have $y=15$.

The value of τ resulting from equation (E) would be

$$\tau=87^{\circ},70.$$

If we admit that the loss of heat by the injector will be more than 15,5-100 of the difference $\tau-\theta$ we will have for this loss

$$i=0{,}155\times 37{,}7=5 \text{ heat units, } 84.$$

The loss of heat from feeding by means of a pump, and of heating the water from 50° to 87°,70, will have for value

$$j=\frac{1}{0.10}\,\frac{AH}{0{,}60}+\frac{1}{3}\,(87.70-50)=14.12 \text{ heat units.}$$

This is more than double the number 5.84 heat units. The economic advantage is always on the side of the Giffard injector; and this apparatus possesses also a great simplicity in adjustment and working, and an important economy of heat. The injector gains from nine to

thirteen *calories* per kilogramme (heat units) of water introduced into the boiler.

THE GIFFARD INJECTOR EMPLOYED AS A PUMP.

The Giffard injector is a suction and forcing pump, but its mechanical performance is weak, because the greater part of the heat is employed in raising the temperature of the water. The expression of the mechanical rendering per kilogramme of wasted steam is,

$$y\,\frac{V^2}{2g},$$

and its calorific equivalent

$$y\,\frac{AV}{2g}.$$

We have inserted in the table the values of the product $y\,\frac{V^2}{2g}$ for the different cases (pages 20, 21).

We see, by an inspection of the table, that the velocity of the mixture varies from 0 to 97^m, 29, that it corresponds to the height of water from 0 to 482 metres. Consequently we are able to introduce water into a reservoir against a pressure of,

$$\frac{482}{10{,}33} = \text{about 46 atmospheres.}$$

The mechanical work $y\frac{V^2}{2g}$ augments with the velocity. At the temperature 100° of the mixture, there are 3,063 kilogrammetres per kilogramme of wasted steam. If we compare this number with those of the preceding table, we see that it is only one-eighth of the theoretical work of steam in a non-condensing engine. The mechanical work would be one-third of that which would be performed by a pump placed in the same conditions for the same purpose.

If we take the numbers corresponding to the temperature of 40° of the mixture temperature for which $y = 22{,}70$, $V = 30{,}17$, the mechanical work produced is no more than 1,053 kilogrammetres.

It is reduced to the third of that which it was for $\tau = 100°$.

Considering, in a general manner, the Giffard injector as an exhausting pump, the problem ought to stand thus :

The height to which it is necessary to raise the water being given, what will be the mechanical performance of the injector ?

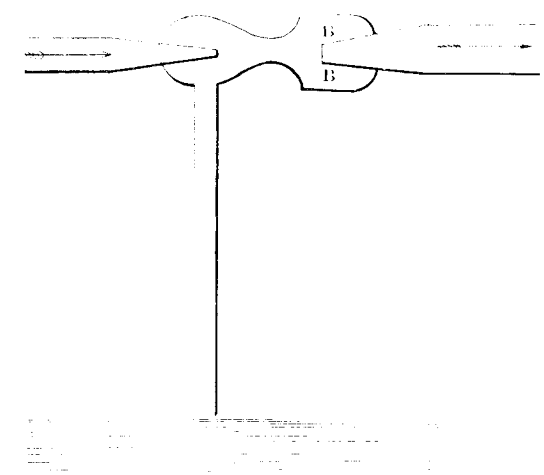


Fig 2

At first sight it is clear that we wish to place the apparatus at the greatest possible height above the reservoir to be drained. In doing this we diminish the pressure in the chamber BB (Fig. 2). Consequently, we lower the temperature t of the steam jet, which is nothing more than the temperature of saturation corresponding to the pressure in the reservoir BB. The raising of the water into the chamber BB will be the same as raising it in a suction pump. We should be able then to raise it to the ordinary practical limits, that is to say, eight metres. Under these conditions, there will be the greatest possible fall of the temperature of the steam from the boiler to the orifice of the injector. We shall have the maximum of mechanical effect, and there will be, between an injector working thus, and an ordinary feeding injector, when the steam is at about 100°, the same difference as between a condensing, and a non-condensing steam engine.

The height being eight metres, the pressure in the chamber BB will be

$$10{,}33 - 8^{m} = 2{,}33 \text{ of water}$$

or

$$176^{mm} \text{ of mercury.}$$

The temperature of corresponding saturation is about 63°. Going back to equations (D) (E), we will have for determining the minimum of the velocity of the mixture the equation

$$\frac{V^2}{2g} = h.$$

Equation (D) will give us

$$V\left(V - \frac{W}{(1+y)}\right) = -8g;$$

from these two equations we deduce

$$1 + y = \frac{Vw}{V^2 + 8g} = \frac{\sqrt{hw}\,\sqrt{2g}}{h+4}. \qquad \text{(G)}$$

The weight of water raised will be the greater as w will be greater. Now, the initial and final temperatures of the steam, during its flow, being determined, w depends only upon the moisture of the steam. The quantity of water raised will be greater in proportion as the steam is dry.

The mechanical work produced by the apparatus will be

$$\tau = y\,(h+8). \qquad \text{(H)}$$

Here is a table of quantities of water raised by an injector pump placed at eight metres above the exhausting reservoir, and fed by dry steam at five atmospheres:

Total Height of Elevation.	Injection of Dry Steam at 152°,82. Velocity, $w=930^m$.		
	Weight of Water raised per kilogram. of Steam used.	Mechanical Work obtained.	Temperature of Water raised.
$h+8$	y	W	τ
		km.	
1+8=9	41,0	369	"
10+8=18	46,37	835	"
50+8=58	26,47	1535	"
100+8=108	19,18	2071	"
200+8=208	13,51	2810	"
500+8=508	8,30	4216	"
800+8=808	6,89	5163	100°

This table shows that the mechanical performance of the injector pump increases

rapidly with great heights, otherwise it is inferior to ordinary steam pumps. It is then for great elevations that the injector gives most satisfactory results. It would be little economy to employ it in common use, but it would be very useful for draining exceptional leaks in mines, when the ordinary pumps are not sufficient. The extreme simplicity of this kind of apparatus will often give it the preference over others that do more perfect work.

We ought again to remark the importance of placing the injector at the height of eight metres above the reservoir. Whereas, in the first table we have found a performance of 3,063 kilogrammetres for an elevation of $482^m,3$ the injector being at the level of the reservoir; the table above gives us a performance of 4,216 kilogrammetres for a height nearly equal to the former—that is to say, 508 metres.

DRAINING SMALL DEPTHS WITH THE INJECTORS.

Steam injectors are employed at the side of vessels for draining leaks in the hold. Under these circumstances the total height of lifting varies from five to ten metres, and each kilogramme of expended steam furnishes about 900 kilogrammetres. Now, in a ship's engine, one kilogramme of steam produces about 17,000 kilogrammetres, according to M. Freminville's treatise on Marine Engines. A pump in the hold would perform only one-fourth of this work, according to that, one kilogramme of steam would produce a useful work of 4,250 kilogrammetres.

Compare the number 900 kilogrammetres with this last, and we shall find the ratio of the work of the steam injector to that of the pump,

$$\frac{900}{4250} = 0,211.$$

According to M. Freminville, this number should, in practice, be diminished to 0,16.

MEANS OF AUGMENTING THE PERFORMANCE OF THE INJECTOR PUMP. — WATER INJECTORS.

We should consider two reservoirs of water R and R′ (Fig. 3) placed at the heights, respectively H and h, above a

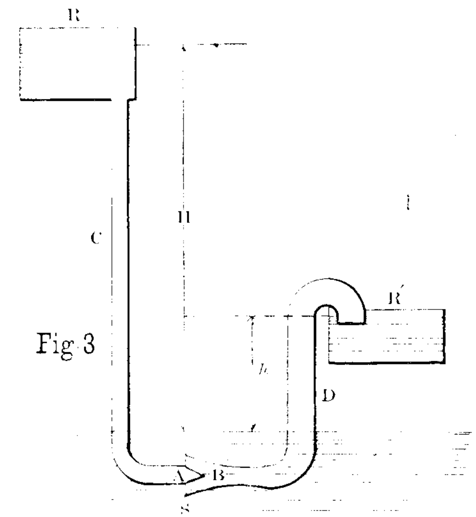


Fig 3

reservoir S. R serves to raise the water from S, and discharge it into R′. Let us place below the reservoir R a vertical tube C, terminating in a nozzle A, entering into a funnel-shaped opening of a second vertical tube D, leading to the reservoir R′. If we open the stop-cock of the tube C, the water will run out by the nozzle A with great velocity, carrying a part of the surrounding liquid, and if the apparatus is well regulated it will be able to raise the water to the height of the tube D, and it would flow into the reservoir R′. The fluid vein proceeding from the upper reservoir R will, by communicating its motion to the water of the reservoir, carry it on to R′. We will consider the condition of the working of the apparatus.

Let P be the weight of water delivered at the nozzle **A**;

P′ the weight of water carried per second;

v the velocity of the water leaving the nozzle;

u the velocity of the water at its entrance into the tube B.

Suppose the velocity of the affluent water about the tube B to be neglected,

the interchange of quantities of motion will give us the equation

$$Pv=(P+P')u \quad . \quad . \quad . \quad (I)$$

Suppose that the water falls into the reservoir R′ without appreciable velocity, then all loss of living force will be avoided, consequently the velocity u will be determined by the condition

$$\frac{u^2}{2g}=h,$$

whence

$$u=\sqrt{2gh} \quad . \quad . \quad . \quad . \quad (J)$$

but

$$v=\sqrt{2gH} \quad . \quad . \quad . \quad . \quad (K)$$

To be sure we neglect the friction. Substitute these values of u and v in equation I, and we have

$$P\sqrt{2gH}=(P+P')\ \sqrt{2gh},$$

whence

$$\frac{P'}{P}=\frac{\sqrt{H}-\sqrt{h}}{\sqrt{h}} \quad . \quad . \quad . \quad (L)$$

To ascertain the modulus of such an engine, it will be necessary to divide the useful work

$$W_u=P'h,$$

by the motor work or propelling force,

$$W_m = P\,(H-h).$$

This will give:

$$\rho = \frac{P'}{P} \cdot \frac{h}{H-h} = \frac{\sqrt{H}-\sqrt{h}}{\sqrt{h}} \cdot \frac{h}{H-h}$$

$$= \frac{\sqrt{h}}{\sqrt{H}+\sqrt{h}}$$

or again,

$$\rho = \frac{1}{1+\sqrt{\frac{H}{h}}} \quad . \quad . \quad . \quad (M)$$

The elevation H is necessarily higher than that of h. The minimum of ratio $\frac{H}{h}$ is 1, hence the useful mechanical work will always be less than $\frac{1}{2}$

We have supposed that the conveyed water comes to the nozzle A with no velocity. This is not so; and to make it more easily understood, we will take a very large funnel **MN** (Fig. 4).

If the suction of the tube at M differs but little from that at R, it is clear that the velocity of the water carried will be quite sensible.

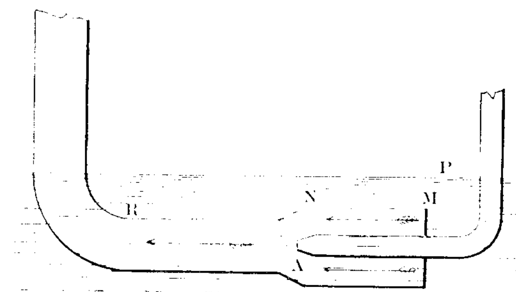


Fig 4

Call this velocity v'. The equation (I) should be written in the following manner:

$$Pv + P'v' = (P+P')u \quad . \quad . \quad (N)$$

Since the apparatus is placed nearly at the surface of the water in the discharging reservoirs, the flow of water between the two points **M** and **N** can be produced only by difference of pressure. At **M** we have atmospheric pressure augmented by the slight height of water MP. At N we should have atmospheric pressure diminished by a small amount. If we call x this difference or *depression* expressed in head of water, we shall have

$$v'^2 = 2gx \quad . \quad . \quad . \quad . \quad (O)$$

The velocity v of discharge at the nozzle A will be given by the equation

$$v^2 = 2g\,(h+x) \quad . \quad . \quad . \quad (P)$$

and the common velocity of the water mixed in the tube R should satisfy the relation

$$u^2 = 2g\,(h+x) \quad . \quad . \quad . \quad (Q)$$

In short, if we call ω, Ω, O, the sections of the pipe and of the tube at M and at R, we shall have the relations:

$$\left.\begin{aligned} P &= 1000\,\omega\,v \\ P' &= 1000\,\Omega v' \\ P+P' &= 1000\,O\,u \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad (R)$$

which gives the following:

$$O\,u = \omega\;v + \Omega v'.$$

The heights H, h and the weight P being given, the equations (N), (O), (P), (Q), (R), will allow us to calculate seven unknown quantities:

$$P',\ v,\ v',\ u,\ \omega,\ \Omega,\ O,\ x.$$

Here are eight unknown quantities. One of these is to be determined before

solving the problem. It is the value x of the difference of pressure, mentioned above, and which varies with the adjustment of the apparatus.

The formula expressing the work indicates in what way this value varies.

The useful work is

$$W_u = P'h,$$

and the motor work or propelling force

$$W_m = P\,(H-h).$$

The rendering of the system is then

$$\rho = \frac{W_u}{W_m} = \frac{P'}{P}\;\frac{h}{H-h}.$$

Now, from equations (N), (O), (P), (Q), we deduce:

$$\frac{P'}{P} = \frac{v-u}{u-v'} = \frac{\sqrt{H+x} - \sqrt{h+x}}{\sqrt{h+x} - \sqrt{x}} \quad . \quad . \quad (S)$$

Substitute in the value of ρ, it becomes:

$$\rho = \frac{\sqrt{H+x} - \sqrt{h+x}}{\sqrt{h+x} - \sqrt{x}}\;\frac{h}{H-h}$$

$$= \frac{\sqrt{h+x} + \sqrt{x}}{\sqrt{H+x} + \sqrt{h+x}},$$

or again,

$$\rho = \frac{1+\sqrt{\frac{x}{h+x}}}{1+\sqrt{\frac{H+x}{h+x}}} \quad . \quad . \quad . \quad (T)$$

If we compare this formula with (M), we discover that they are the same, when we make $x=0$. This supposes the velocity $v'=0$ It is easy to prove that the performance (T) increases with the difference x, and that it is always less than 1. We shall have the theoretical limit of its value in making $x=10^{m},33$. This difference may be artificially made by raising the injector above the reser-

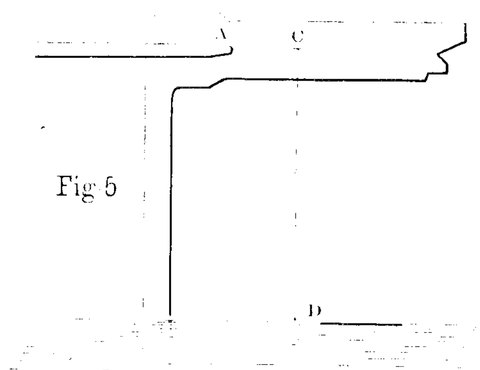


Fig 5

voir (Fig. 5), which will force the water to rise to the height C D.

Suppose

$$P=1, \quad H=500^{m}, \quad h=5^{m}.$$

If, at first, we suppose the depression to be nothing, the equations (L) and (M) give us:

$$P' = \frac{\sqrt{500}-\sqrt{5}}{\sqrt{5}} = 9,$$

$$\rho = \frac{1}{1+\sqrt{100}} = 0,091.$$

To arrange the apparatus in a way to realize a depression of five metres, we make $x=\rho$ in equations (S), (T), and we have:

$$P' = \frac{\sqrt{505}-\sqrt{10}}{\sqrt{10}-\sqrt{5}} = 20,85,$$

$$\rho = \frac{1+\sqrt{\frac{5}{10}}}{1+\sqrt{\frac{505}{10}}} = 0,211.$$

By this hypothesis the performance will have more than doubled. The arrangement of the apparatus has then a great influence upon its action. It should be so that the water raised arrives with considerable velocity at the injector. For this object we direct currents of water by means of several successive funnels (Fig. 6).

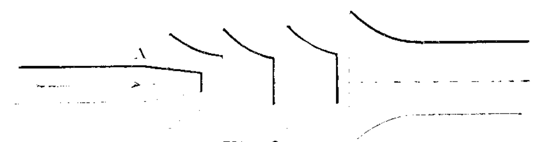


Fig. 6

This arrangement is found in several machines of English origin.

We readily understand that the jet of a steam injector may be used as a water injector.

A steam injector conveys, for example, fifteen kilogrammes of water per kilogramme of steam discharged, and communicates a velocity capable of surmounting a pressure of five atmospheres. In other words, the height to which the jet will be able to rise is 41^m.32.

If we make this liquid jet pass through the nozzle A in Figs. (3), (4), (5), (6),

we will be able to carry a new quantity of water, which, it is true, is not raised so high. But if it is not necessary to raise the water above five meters to reach the 41,^{m}32, it is clearly to our interest to adopt this arrangement.

The same considerations which we have recommended to place the injector above the supply reservoir are applicable here.

We will now give an account of the theoretical performance of the apparatus in (Fig. 7).

The proportion of water carried by the steam jet is reckoned from the height at which the injector is placed above the discharging reservoir, from the pressure of the boiler and from the amount of water practically raised. Equations (D) and (E) will furnish the proportion of water carried and the velocity of the mixture. The reservoirs B and C being at the same pressure, that is to say, the atmospheric pressure diminished by the height A R, the equation (B) will give:

$$v = \frac{u}{1+y}.$$

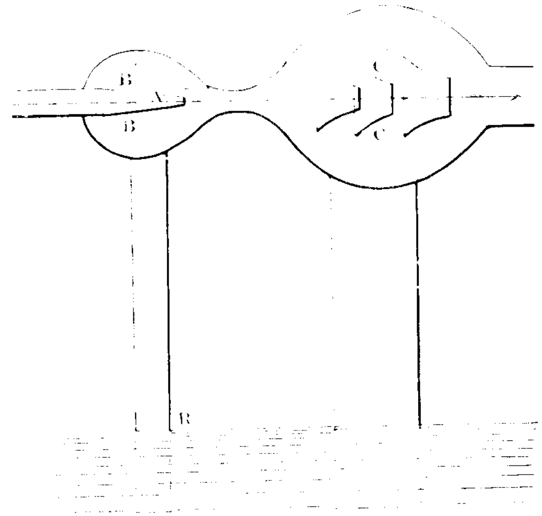


Fig 7

Knowing the velocity v of the liquid jet at its entrance into the chamber C, we calculate the weight of water carried by the jet by means of formula (S), in giving a start to the depression x, and in calculating the dimensions of the suction apparatus by means of equation (R). We will remember in equation (O) that the velocity v' represents the projection upon the axis of the jet, of the real

velocity of the raised liquid streams. It will be necessary, then, to take into account the angle at which the funnels taper.

It is practice alone which can determine the best arrangement of this apparatus, whose theoretical principles only we have established.

We will show by an example how these theoretical calculations may be made.

Example.—To raise the water from the hold of a vessel, knowing the height of suction to be five meters; and the fall of the discharge, five meters. The pressure of the boiler is three atmospheres.

The table (3) gives us for the velocity of the flow of dry steam of three atmospheres at $\frac{1}{2}$ atmosphere final pressure, 739 meters. It will be $w=739$. We will then have equation (D):

$$v=\frac{w}{1+y}.$$

If we make $y=20$, we will find:

$$v=\frac{739}{21}=35^{m},20.$$

Velocity of Flow of Dry Steam under Different Initial and Final Pressures.

The velocity of a mixture of steam and water, containing x of steam and $1-x$ of water may be obtained by multiplying the numbers of the table below by $\sqrt{x}$

Pressures and Final Temperatures.		Pressures and Initial Temperatures.							
		8 at. 170°,81	7 at. 165°,54	6 at. 159°,22	5 at 152°,22	4 at. 144°,00	3 at. 133°,91	2 at. 120°,60	1½ at. 111°,74
Press-ures.	Temper-ature.	485,79	489,71	494,11	499,13	545,02	512,22	521,70	527,99
atmos.	degrees	meters.	meters.	meters.	meters.	meters.	meters.	meters.	meters.
7	165,34	223	..	..	..	..	..	..	..
6	159,22	325	238	..	..	..	..	..	..
5	152,22	411	349	258	..	..	.	..	..
4	144,00	494	445	381	283	.	..	..	..
3	133,91	580	540	490	421	349	..	..	..
2,4	126,46	636	601	558	501	420	279	..	..
2	120,60	676	645	606	555	485	372	..	..
1,8	117,30	698	668	632	584	519	417	190	..
1,6	113,69	721	694	658	613	553	460	276	..
1,4	109,68	746	719	686	645	588	504	347	..
1,2	105,17	773	748	717	678	625	549	463	274
1,00	100,00	803	779	750	714	666	596	477	366
0,90	97,08	819	797	769	734	688	621	498	409
0,80	93,88	837	815	788	755	710	647	543	451
0,70	90,32	856	835	809	777	735	676	578	494
0,60	86,32	877	857	833	802	763	706	615	539
0,50	81,71	900	882	858	830	792	739	655	585
0,40	76,25	928	910	888	861	826	770	699	636
0,30	69,49	960	944	924	899	866	821	751	694
0,20	60,45	1,002	988	969	947	918	877	814	765
0,10	46,21	1,065	1,052	1,037	1.017	993	958	906	865

The height designated by H is here:

$$\frac{v^2}{2g} = H = 63^m,18.$$

Since the pressure which exists in chamber C about the funnels is only ½ atmosphere, we will be able to realize at the axis of the jet only a slight depression; we will suppose two meters. It will be, then, $x=2$, and we will have equation (S):

$$\frac{P'}{P} = \frac{\sqrt{63,18+2} - \sqrt{5+2}}{\sqrt{5+2} - \sqrt{2}} = 4,41.$$

The weight of water positively raised then will have been:

$$4,41 \times 20 = 88^k,20,$$

per kilogramme of discharged steam. The mechanical work produced has for its value:

$$88^k,20 \times 10^m = 882 \text{ kilogrammeters},$$

per kilogramme of steam.

The table (1), that for a weight of water raised to $89^k,63$, which is nearly equal to 88,20; the Giffard injector produces only 197 kilogrammeters. If we

keep the injector at five meters above the surface of the water to be raised and make use of this apparatus without the intervening injector of water, we should calculate the weight of water carried in the following manner. We should have for the necessary velocity to cause the jet to attain to five meters of height of discharge:

$$v = \sqrt{2g \times (5+5)} = 14 \text{ meters},$$

$$1 + y = \frac{739}{14} = 52,80,$$

whence

$$y = 51.80$$

The weight of water raised would be only $51^k,80$, in place of $88^k,20$, which we have found in employing the water injector.

The steam injector arranged with a water injector to serve as a pump has then a notable advantage over one used solely for steam. Several machines founded on these principles are used in England for draining mines. The preceding considerations demonstrate that the employment of these machines is not very convenient. Their use is justified only when it is necessary to accomplish rapid drainage with simple apparatus.

EJECTOR CONDENSERS.

Let us conceive that we put an injector on the escape pipe of a condensing engine. The apparatus will work as usual, that is to say, according as the escaped steam arrives it becomes condensed by the contact with the cold water furnished by the reservoir, will maintain a vacuum, and the mixture of water with the condensed steam imparting a great velocity will be capable of surmounting the excess of exterior atmospheric pressure over the pressure of escaping steam.

We shall thus be able, by the simple interposition of the injector apparatus, to supplant the air pump and all the accessories, and we shall economize the work absorbed by this pump, often very considerable.

Such is the principle of ejector con-

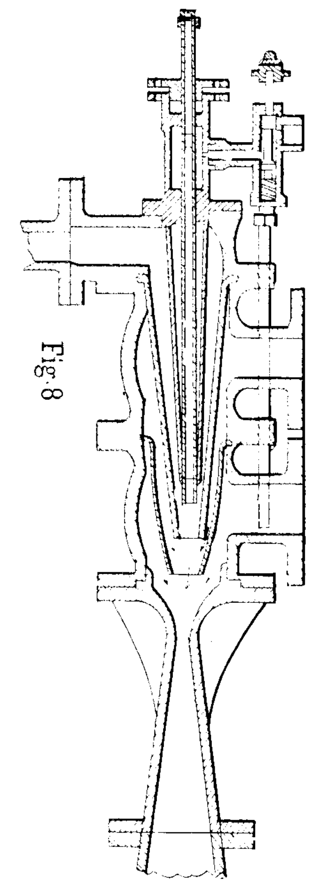


Fig. 8

densers, the employment of which tends, probably, to become general.

Professor Rankine has reported the experiments made in 1868, upon a condenser ejector of the Morton system. The apparatus (Fig. 8.) differs from ordinary steam injectors in that the cold water is drawn by the central tube; the escape of steam is distributed about the central jet by the very long and concentric funnels. In this way the living force of the cold water suffers no loss.

This living force is considerable, since the pressure which exists about the cold water, at the moment in which it mixes with the jet of steam, is necessarily less than the pressure at the escape, without which the steam would not flow out, so that the water possesses the velocity due to the excess of the atmospheric pressure over the pressure of the escape.

In the above named experiments Prof. Rankine has found the following results:

	Per sq. centimeter.
Absolute pressure at the boiler...	3^k,427
Absolute pressure at commencement of escape.............	0^k,756
Mean pressure maintained behind the pistons by the condenser ejector.............	0^k,299
Pressure kept near the funnels.	0^k,210

	Centigrade.
Temperature of cold water....	8°,4
Temperature of water of condensation..................	30°,3
Weight of cold water employed per kilogramme of steam....	28^k,40

These results are, as we see, very satisfactory. They are not widely different from those which were obtained by the air pumps, but these latter require a notable expenditure of moving force.

In the machine experimented, Rankine valued the effective force at twenty-four horses, and the economy realized by the replacing one horse power air-pump at four per cent.

The theory of the condenser ejector does not differ from that of ordinary injectors, only there has been no account taken of the velocity at which the cold water arrives, that here has considerable value, which is neglected in our general equation (C). To take account of it, we should add to the first member of this equation a term

$$\frac{AU^2}{2g} y,$$

representing the living force of the weight of water y.

In reality we will be able to neglect the term $\frac{V^2}{2g}$.

The determination of the velocity and the calculation of the dimensions of the apparatus are made, according to the method heretofore explained. This is a problem which presents no difficulties.

The Morton apparatus has one peculiarity which we ought to describe. To put the apparatus in motion we allow a priming of steam from the boiler to pass through a central tube. It may happen that the pressure falls below the proper limits for the working of the apparatus. Under these circumstances, the cold water flows into the escape-pipes, and

then into the cylinders. Every time that this inconvenience threatens the central steam jet is automatically opened by a spring piston, and its power communicates to the cold water jet a sufficient impulse to prevent its deviation towards the cylinders, and re-establishes the normal working of the machine.

THE INJECTOR EMPLOYED IN A HYDRAULIC PRESS.

Suppose that we place an injector at the foot of a cylinder of a hydraulic press (Fig. 9). The jet of hot water may be introduced into the cylinder so that the pressure will be lower than that which corresponds to the velocity of flow. We should be able then to work a hydraulic press with a pressure

$$\frac{V^2}{2g}.$$

Suppose the pressure at the boiler be five atmospheres, the jet, according to table, pages 20 and 21, could rise to the height of 482 meters; this corresponds to

482,000 kilogrammeters per sq. meter, or to

50 atmospheres.

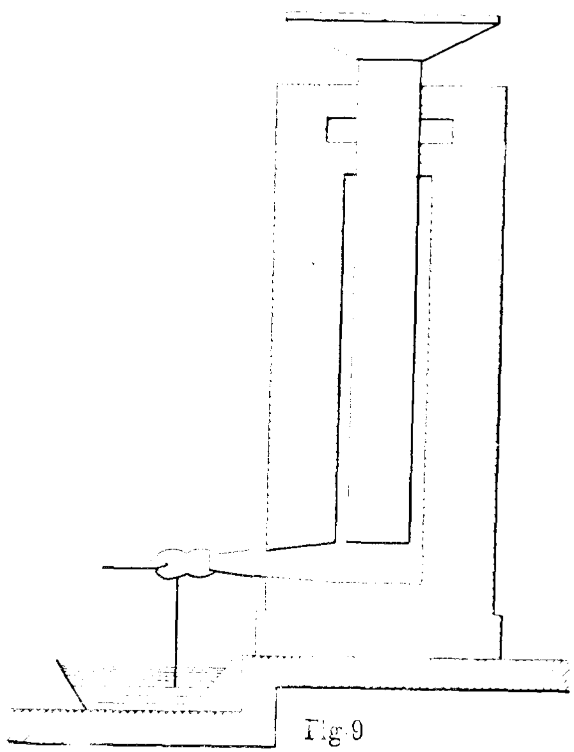


Fig 9

The same table shows that the mechanical work realized in these conditions is only

3063 kilogrammeters,

per kilogramme of steam used.

The work diminishes in the same proportion as the pressure diminishes. Consequently, it would be better to work with high pressure and so diminish the diameter of the hydraulic press.

This is a novel application of the injector, and may prove of service in situations where a hydraulic press is necessary, and where a pump of sufficient power is wanting.

Such an application has not yet been made, at least to our knowledge, and needs preliminary experiments.

PUMPING GAS BY STEAM — EXPLANATION OF THE FEEBLE WORKING OF FEEDING OR DRAINING INJECTORS.

That which causes the weakness of the performance of a steam injector employed as a draining pump is the disproportion between the height to which a liquid mixture may be raised, which is several hundred meters, and the height to which we in reality raise them. The apparatus does not give its maximum of performance excepting for very great heights. The disproportion which exists between the specific gravity of the body raised, which is here water, and that which is carried along with it, that is to say, steam, is another source of loss.

In effect, the relative velocity of the water raised at the moment when it mixes with the condensed steam is so

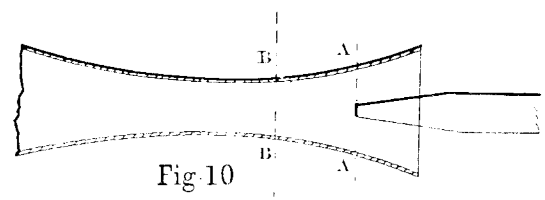


Fig 10

slight that is not worth taking into account. Let us suppose, on the contrary, that the steam escape in a gaseous medium which it will drag along by a sort of lateral friction (see Fig. 10), and as in a water injector. It may happen that the velocity of affluent air be considerable, consequently cannot be neglected.

Now this circumstance will augment the performance, just as in the water injector.

Let W be the velocity of the steam, w that of affluent air, v that of the supposed mixture in the contracted section B B, y the weight of air drawn per kilogramme of steam.

The momentum of the steam will be

$$\frac{W}{g},$$

that of affluent air,

$$\frac{w}{g}y,$$

that of the mixture,

$$(1+y)\,V.$$

Neglecting the difference of pressure in the sections A and B, we will have the following equation:

$$W+wy=(1+y)V,$$

whence we find

$$V=\frac{W+wy}{1+y}.$$

The living force of the steam was

$$\frac{W^2}{2g},$$

that of affluent air,

$$\frac{w^2}{2g}y,$$

that of air after the mixture,

$$\frac{V^2}{2g}y.$$

The performance of the apparatus will be then:

$$\rho = \frac{y\frac{V^2}{2g}}{\frac{W^2}{2g}} = \frac{y\left(1 + \frac{w}{W}y\right)^2}{(1+y)^2}$$

When we suppose the velocity of the material drawn to be nothing, as that in the Giffard injector for feeding boilers, $w=0$, and the formula of the performance is reduced to

$$\rho' = \frac{y}{(1+y)^2}.$$

We see at once how feeble this action is when y is not very small.

When $y=1$ $\rho' = \frac{1}{4} = 0,25$

When $y=15$ $\rho = 0,058.$

If, on the contrary, the ratio $\frac{w}{W}$ be slightly increased, the term $\left(1 + \frac{w}{W}y\right)^2$ will increase very sensibly the value of the mechanical performance.

For example.—If we make $y=15$ and $\frac{w}{W}=\frac{1}{6}$ the ratio realizable in practice, the calculations give $\rho=0,72$ in place of 0,058, which we have found in making $w=0$.

This simple discovery suffices to make us understand that steam injectors are more likely to give good results when employed as gas pumps than when used as water pumps.

LOCOMOTIVE EXHAUST.

The exhaust of a locomotive is only an application of preceding considerations. Only in these machines the steam jet is intermittent, notably augmenting the results which we should obtain with a continuous jet. People do not know that the draught of the locomotive hearth is, if I may say so, due only to the escaping steam; this is, truly, the fundamental principle of the construction of these powerful machines.

Prof. Zeuner has demonstrated by calculation, and by experiment, that the weight of air drawn into the chimney of a locomotive is proportional to the weight of steam expended. So the combustion is more active when the engine works fastest.

It has been long known, in a general way, that the velocity of a locomotive engine ought to be pushed to its utmost limits when required to perform important work, as when ascending a slope.

Generally the proportion of the weight of air drawn to the weight of steam employed is between 2 and 3 to 1.

M. Péclet reports the experiments made by M. Glépin upon the draught produced by the continuous steam jets opening into cylindrical tubes. The results are quite various, according to the diameter and the length of the tubes.

We have usually found that the performance increases with the length of the tubes. Its value has been a maximum with the tubes from $0^m,50$ to $0^m,55$ in diameter, and from 3 meters to $3^m,50$ in height. It was then raised to 0.1145. We here call performance the proportion of the living force of air drawn to the living force of the steam jet.

It seems demonstrated that the intermittent jets produce superior action, and according to MM. Flachat and Petiet the work produced by the intermittent injections of steam in the chimneys of locomotives varies from 0.5 to 0.16 of the work which the steam is able to produce.

Usually in this kind of apparatus it is necessary to multiply the surfaces of the contact of steam and air. Then annular jets are better adapted for such work than the compact cylindrical jets.

STEAM BLOWERS.

Several important processes are founded upon the conveying of air by a jet of steam. This method has been applied to the ventilation of mines. It was also employed to ventilate the great machine gallery at the Paris Exposition, in 1867.

To conclude, M. Siemens has made a new application in the manufacture of steel. In his apparatus, the air is drawn by a double tube through an annular jet and a central conical jet. The air is introduced by an annular central orifice, and by orifices through the outside partition of the apparatus. The contacts of the fluid entering and the fluid escaping are thus greatly multiplied, and this circumstance is eminently favorable to the action.

The mixture of steam and air is introduced into a reservoir containing bits of coke crushed and washed by a current of cold water. The steam is condensed and the air escapes mostly free from steam.

As a means of forcing currents of air for purposes of ventilation this method of M. Siemens is worthy of consideration.

KÜRTING'S LOCOMOTIVE INJECTOR.

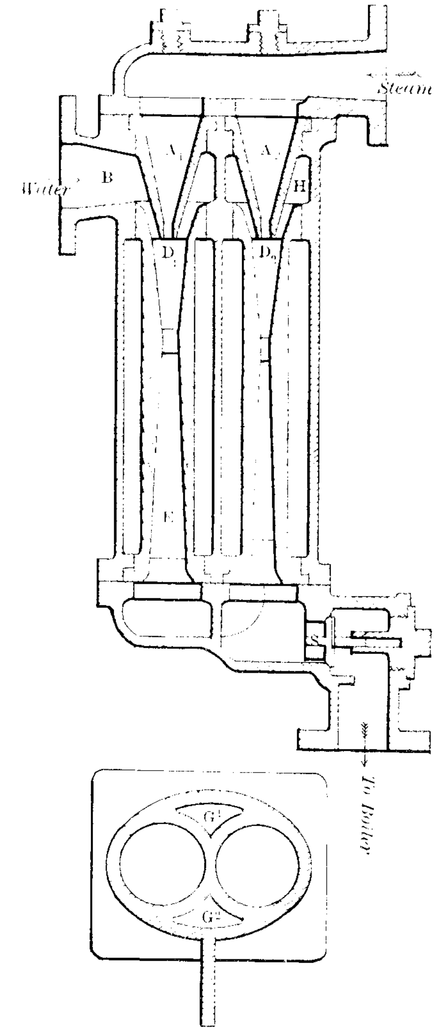
Locomotive injectors as hitherto constructed labor under the disadvantage of feeding with cold water only, and they can hardly be relied upon if the temperature of the latter exceeds 104 deg. Fahr. Even then they require the most careful adjustment of the water supply. The reasons for this defect may be traced to the principles upon which the injectors are constructed.

With an injector of correct proportions the certainty of action depends upon the velocity with which the water enters the space where the steam and water combine. In locomotive injectors to which the water can flow with only a very small pressure, this velocity depends mainly upon the vacuum produced in the condensing nozzle. This vacuum must be kept as high as possible. With *constant* steam pressure and temperature of water, the vacuum obtained is lower when the condensing nozzle is fed with too much or too little water; in the first case because the jet of steam has not sufficient power to impel the water which gives a back pressure; in the second case because the temperature of the mixture is not low enough, and consequently the vacuum is lessened. For these reasons the water supply requires to be very carefully regulated. With *variable* steam pressures and temperatures of the feed water, the vacuum becomes lower with increasing temperature of water and also with increasing steam pressure, as in both cases the temperature in the condensing space is raised, the maximum of which can be only 212 deg. Fahr. But at this point the certainty of action is *nil;* generally speaking, this temperature should not exceed 194 deg. Fahr. As the increase of temperature with high pressure steam is about 90 deg. Fahr., it follows that the feed water should not be hotter than 104 deg. Fahr. On this account many rail-

ways will not allow their drivers to warm the feed water in the tenders, as the reliability of the injectors increases with the coldness of the water, and certainty is of the first importance in railway management. This defect, is almost entirely done away with in Körting's universal injector, which works with equal certainty at all pressures. This apparatus consists of two steam jet pumps combined. The second pump or real injector which forces the water into the boiler receives it from the primary or assistant injector under pressure, so that the second pump has only to overcome the difference in pressure existing between that of the boiler and that already overcome by the primary injector.

The required quantity of steam is therefore divided, and only a small portion of it used in the first part of the apparatus. Consequently the increase of temperature is much less than in ordinary injectors; the water entering it may therefore be much warmer without bringing the temperature in the condensing space above 194 deg. Fahr., which is the maximum here as in ordinary injectors. The temperature of the feed water may safely be as high as 158 deg. Fahr. A special feature of this primary injector is that, with increased steam pressure, it delivers, without regulation, more water at increased pressure to the second part of the apparatus.

The second pump delivers into the boiler the water forced into it by the primary injector. The certainty of action of this second part of the apparatus depends upon the pressure with which it is fed by the assistant injector, and not upon any vacuum. As with increasing steam pressure the velocity of the water entering the second pump is also increased, it follows that with the same temperature of feed water, the reliability of this apparatus remains the same under all steam pressures, while with ordinary injectors it decreases as the steam pressure increases. On this account no water regulation is necessary. The tem-



perature in the condensing space does not come in question with the second part of the apparatus; it may, if required, exceed 212 deg. Fahr., and in fact does exceed it, for with feed water of 158 deg. Fahr., and 120 lbs. boiler pressure, the water fed into the boiler is actually 257 deg. Fahr. The apparatus therefore must not be provided with an overflow communicating with the atmosphere, as otherwise the high temperature would cause the formation of steam and an escape of water. The apparatus is started by opening a small cock behind the injector, similar to that with which other injectors are provided for letting the water out of the pressure pipe.

The foregoing illustration shows the Körting universal injector in longitudinal and cross sections. The working steam simultaneously enters the two steam nozzles A_1 and A_2 in the injector. The jet of steam from A_1 draws the requisite water through the pipe B, and forces it through the cone D_1 with corresponding velocity. This velocity is

transformed into pressure in the diverging tube E which communicates by means of the chambers G_1 and G_2 (see cross section) with the space H of the second pump. From here the water enters under pressure the condensing space D_2, whence it is forced by the steam issuing from the nozzle A_2 into the boiler through the back pressure valve S. While starting the injector a cock communicating with space E_2 is opened till water escapes from it, after which it is slowly closed.