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The Theory of Steam Jet Pumps

TKI 25

Jet Pumps serve to convey or compress gases, vapours, liquids or solids by using the kinetic energy of a gaseous or liquid motive medium. This medium is brought to a high velocity by expansion, and accelerates the medium to be compressed or conveyed.

For Steam Jet Pumps, steam is used as motive medium. These pumps are used to a great extent for the production of vacuum.

1 Head
2 Motive Nozzle
3 Converging Cone
4 Throat
5 Diverging Cone
} Diffusor (Mixing Nozzle) [3–5]

p_1 = Motive steam pressure
p_0 = Suction pressure
p = Counter pressure

$\dot{M}_1$ = Motive steam flow
$\dot{M}_0$ = Suction flow
$\dot{M}$ = Mixed vapours flow

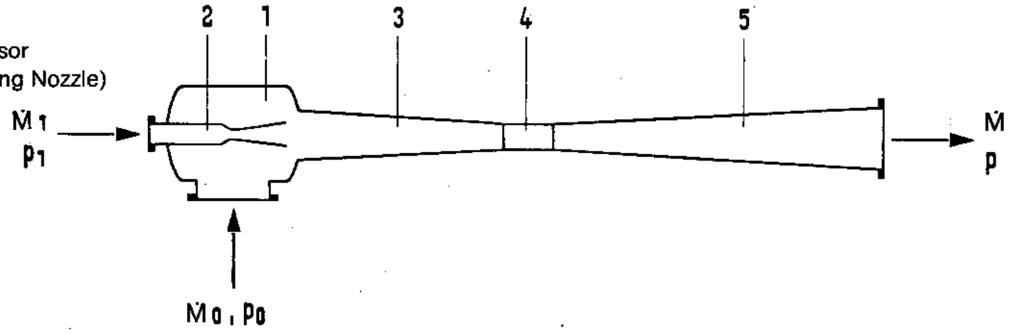


Fig. 1 Cross-section of a Steam Jet Pump.

The steam is expanded in the motive nozzle. At an assumed free of loss expansion, the process is represented by a vertical line in the h, s.-diagram by Mollier. If expansion from p_1 to p_0 takes place, saturated steam being used as motive medium (point 1 in fig. 2), the diagram is as follows:

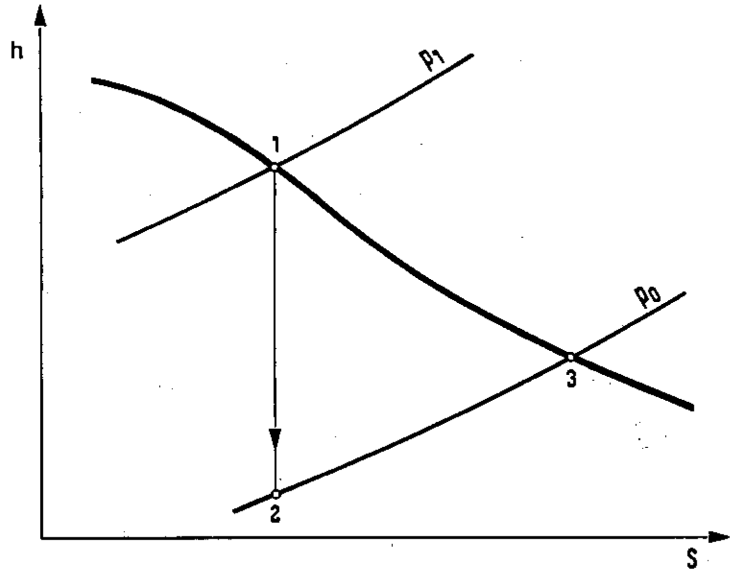


Fig. 2

Figure 2 shows that the end point of the expansion (point 2) lies in the wet steam range. Therefore, water droplets are present in the motive jet. If p_0 is < 6 mbar, the temperature of the water droplets is under 0°C and ice will begin to form.

The converging cone of such jet pumps must be heated to prevent the build up of ice on the walls as this would reduce the cross-sectional area of the flow. In the expansion a part of the internal energy of the motive steam is transformed into kinetic energy and velocity is obtained, which can be calculated by the well-known formula:

$$w = 44{,}72\sqrt{\Delta h}\ \text{[m/s], when } \Delta h \text{ in kJ/kg} \quad \text{or} \quad w = 91{,}5\sqrt{\Delta h}\ \text{[m/s], when } \Delta h \text{ in kcal/kg resp.}$$

At an expansion ratio $p_1/p_0 = 1{,}73$ ($p_0/p_1 = 0{,}83$) – this is called the critical pressure ratio – the speed of sound is reached and at $p_1/p_0 > 1{,}73$ supersonic velocity is reached. In such a case the motive nozzle has the shape of a Laval cone in having a converging section and a diverging section. In the narrowest cross-section the velocity of sound is reached at a pressure $p_s = p_1/1{,}73$.

The supersonic velocity is produced in the divergent section of the motive nozzle, situated after the narrowest cross-section, by expansion from p_s to p_0.

The steam flow in kg/h through such a nozzle is proportional to the flow cross-section at the narrowest point as well as the velocity, and inversely proportional to the specific volume of the steam.

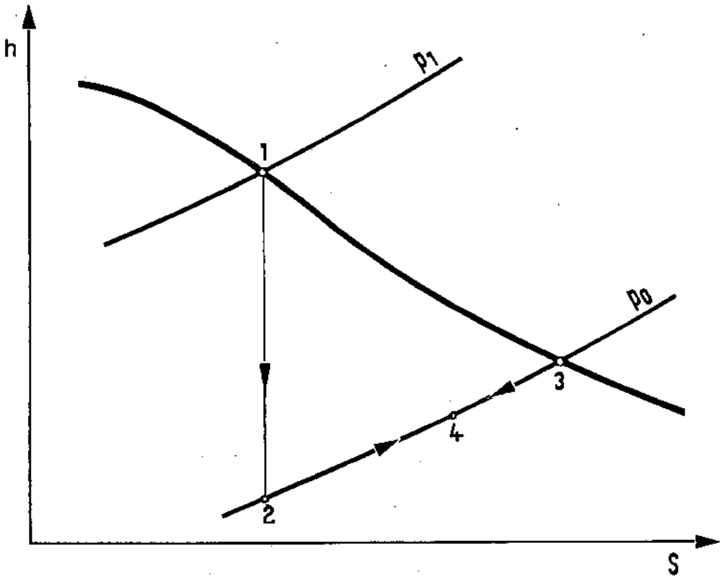
$$\dot{M}_1 = \text{prop.} \frac{A_e \cdot w_e}{v_e} \text{ kg/h}$$

As A_e is proportional to the diameter d_e^2 and v_e in the first approximation is only inversely proportional to the p_s, if one accepts that the steam temperature is constant and given that $p_s = p_1/1{,}73$

$$\dot{M}_1 = \text{prop. } d_e^2 \cdot p_1 \text{ kg/h}$$

Therefore, in a nozzle, in which steam is expanded by a pressure drop $p_1/p_0 > 1{,}73$, the motive flow in essence is only dependent upon the narrowest diameter d_e and upon the steam pressure p_1 before the nozzle. It is independent of the pressure after the nozzle, therefore, independent of the suction pressure. One sees that for a nozzle of a given diameter d_e, only the calculable steam through flow is to be expected when the calculation pressure before the nozzle predominates. If, for example, a motive nozzle is designed for a through flow of $\dot{M} = 50$ kg/h at a motive pressure of $p_1 = 4$ bar (= Absolute Pressure), at a pressure $p_1 = 8$ bar the through flow will be 100 kg/h appx.

In the converging part of the diffusor, at a constant pressure p_0, the mixing of the motive and the suction flow takes place. If saturated steam is to be drawn from pressure p_0, one can see this mixing process in the h, s-diagram in Fig. 3.



The expanded motive steam from position 2 mixes with the drawn off steam from position 3.

The mixing point 4 lies on the curve 2–3 proportional to the quantities.

Fig. 3

Here by impulse exchange, the drawn in vapour will be accelerated and the motive steam slowed down.

The pressure ratio p/p_0 is a measure of the compression and is called the compression ratio. One must distinguish between subcritical, critical and super critical compression ratios according to whether p/p_0 is less, equal or greater than 1,73. By the same token one has in the throat of the diffusor subsonic, sonic or supersonic velocity. Most jet pumps have super critical compression ratios and in the throat supersonic velocities.

According to the Law of Continuity $\frac{w \cdot f_H}{v} = \frac{w \cdot f_H \cdot p}{R \cdot T}$ must be constant.

Whereas $\frac{f_H}{R}$ is constant, w, p and T are variable.

In passing the throat the flow velocity drops from supersonic to subsonic. Associated with this is a sudden rise in pressure. One speaks of a pressure jump, or of a compression shock. At the same time there is a change in the temperature, it rises with equal suddenness, as one may physically feel on the outside of the jet pump.

Where this point of change lies depends upon the operation of the jet pump. It can lie in the smallest cross-section, before it or beyond it. In the diverging cone, due to the increasing cross-sectional area, the velocity is still further reduced and thereby the pressure continues to rise.

This process is shown in the simplified drawing Fig. 4 and 5.

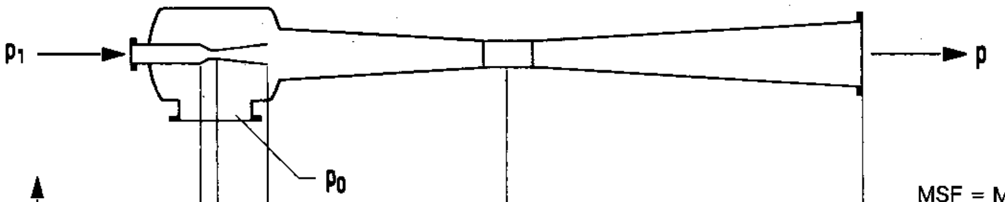
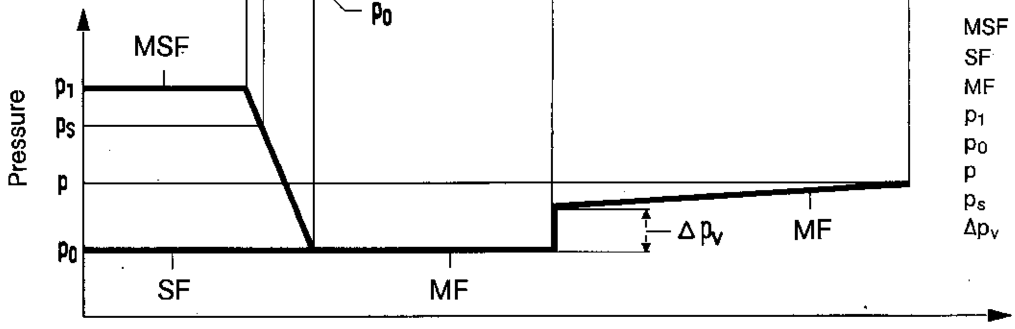


Fig. 4



MSF = Motive steam flow
SF = Suction flow
MF = Mixed flow
p_1 = Motive steam pressure
p_0 = Suction pressure
p = Counter pressure
p_s = $p_1/1{,}73$
Δp_v = Compression shock

Fig. 5 Direction of Flow

If one considers the process as "without loss" then one can see it represented in the h, s-diagram as shown in Fig. 6.

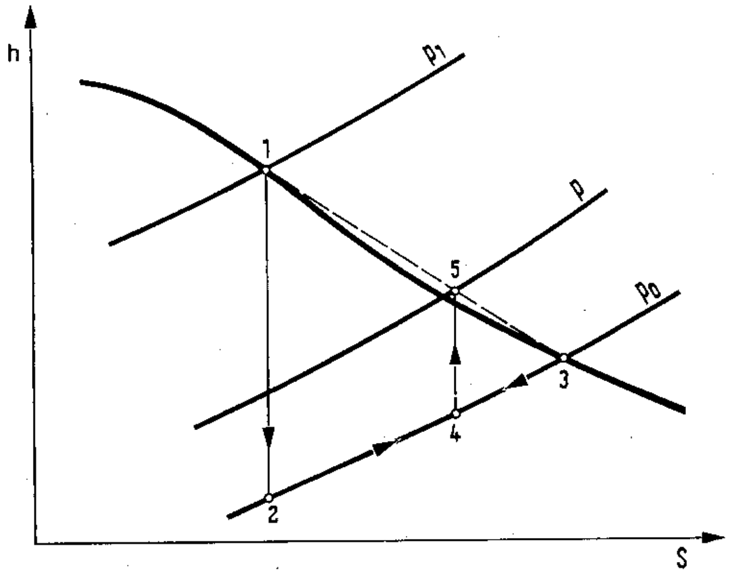


Fig. 6

In reality losses do occur, associated with the rise in entropy, so that the change in the conditions of expansion 1–2' and the compression 4'–5' result in the curved lines which can be seen in the h,s-diagram shown in Fig. 7.

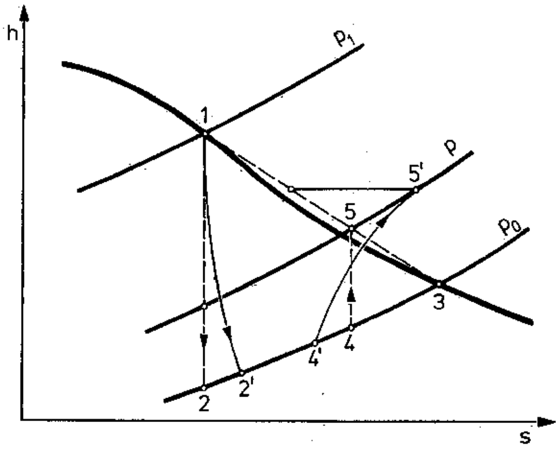


Fig. 7

1 –2	Free of loss expansion of the motive steam.
1 –2'	Actual expansion of the steam with loss.
4 –5	Free of loss compression of the mixed flow.
4'–5'	Actual compression of the mixed flow with loss.

The question is often asked, "what is the temperature of the mixed flow leaving the outlet of the Jet Pump", therefore, the temperature at condition 5'. This one may find easily, with the help of the h,s-diagram, as shown in Fig. 8.

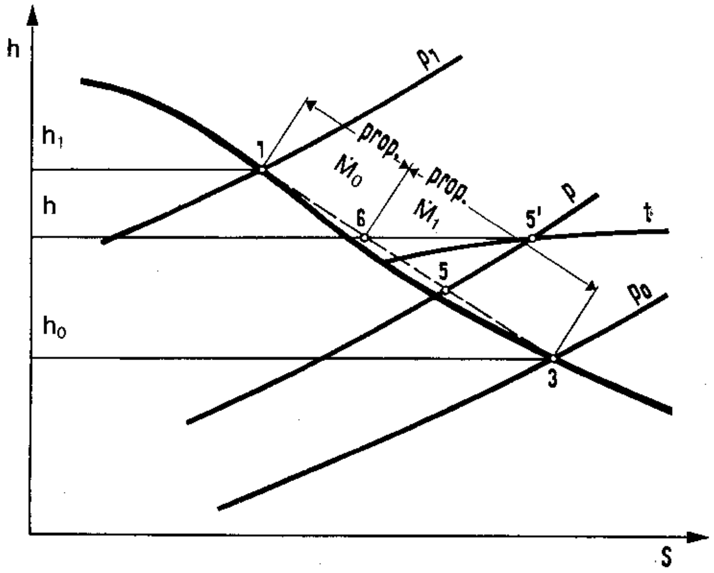


Fig. 8

Thereby the heat content of the mixed flow at the outlet is calculated according to the following formula:

$$h = \frac{\dot{M}_1 \cdot h_1 + \dot{M}_0 \cdot h_0}{\dot{M}_1 + \dot{M}_0} \cdot \quad \text{kJ/kg}$$

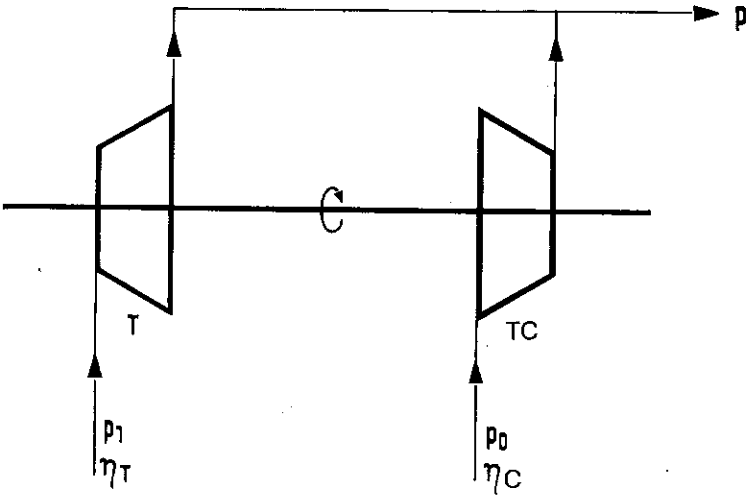
The position 5' one finds at the point of intersection of the lines of constant enthalpy with the Isobar p. The sought after temperature one reads off the Isotherm which intersects position 5'.

In general when air is the suction medium the outlet temperature is lower.

Considering the efficiency of Jet Pumps

It is often the opinion that the efficiency of Jet Pumps is poor. However, before one can make such a judgement, one must first establish the type of efficiency of which one is speaking. There are various definable efficiencies for a Jet Pump that should not need to be examined here. There are suitable efficiencies that do allow the comparison of the various Jet Pumps with one another and are also suitable for comparison with other compressors.

However, in practice, it is useful to compare a Steam Jet Compressor with a Turbo-Compressor. Because a Steam Jet Pump is driven by steam it should be compared with a Turbo-Compressor also driven by steam, which is, of course, via a turbine. In Fig. 9 such a plant is shown schematically.



T = Turbine
TC = Turbo-Compressor

Fig. 9

Here the motive steam is expanded from p_1 to p, whilst the suction flow is compressed from p_0 to p. These pressures should correspond to those for the Steam Jet Compressors that are to be compared.

In general the efficiency of a machine is considered to be the relationship of the work gained to the work put in. In the comparison model the turbine has an efficiency η_T and the compressor an efficiency η_C Both are approximately 65 %. The total efficiency of the comparison model is, therefore:

$$\eta = \eta_T \cdot \eta_C\text{, thus } \eta = 0{,}65 \cdot 0{,}65 = 0{,}42.$$

Naturally, one can make the comparison with a turbine driven principally by a generator. With the generated current an electric motor is powered, which then supplies the driving capacity to the Turbo-Compressor. Such a system would, in reality, be best. The efficiency is then, naturally, lowered further.

For a Steam Jet Compressor one finds the corresponding efficiency from the h, s-diagram as follows:

$$\eta = \frac{\dot{M}_0 \cdot \Delta h_2}{\dot{M}_1 \cdot \Delta h_1^*}$$

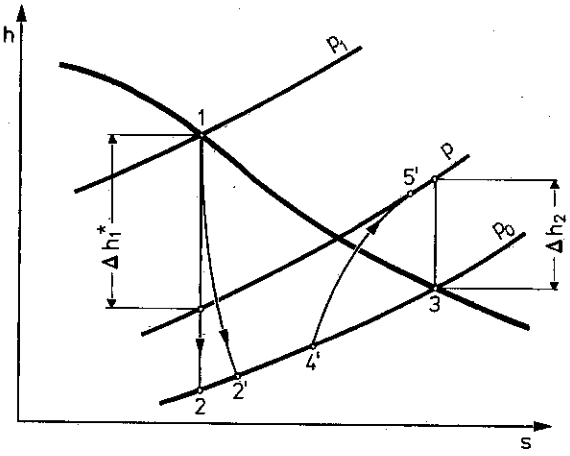
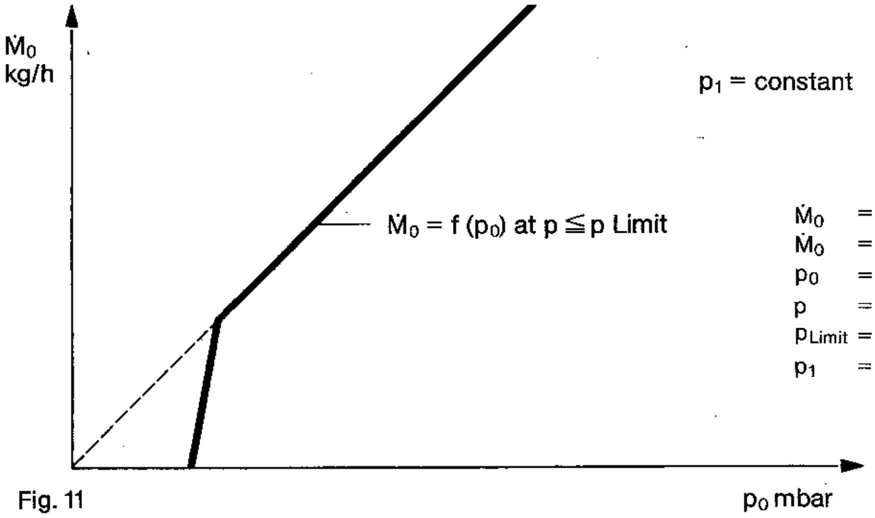


Fig. 10

The, as defined, “Machine Efficiency” of a Jet Pump lies between 20 and 40 % and is, therefore, somewhat comparable with the mechanical model.

Operating characteristics of Steam Jet Pumps

The characteristic line shown in Fig. 11 explains the operation of a Steam Jet Pump. This is the correlation of suction flow $\dot{M}_0$ and suction pressure p_0, also called the Suction Curve: under certain conditions the correlation looks somewhat as follows:



$\dot{M}_0$ = Suction flow in kg/h
$\dot{M}_0$ = f (p_0) = Suction curve
p_0 = Counter pressure in mbar
p = Suction pressure in mbar
p_{Limit} = Limit counter pressure in mbar
p_1 = Motive pressure in bar abs.

Fig. 11

Above the break off point one sees that the suction flow $\dot{M}_0$ is directly proportional to the suction pressure. Thus $\dot{M}_0$ = prop. p_0.

As the specific volume of the suction medium is inversely proportional to the suction pressure (V_0 = prop. $\frac{1}{p_0}$), one finds $\dot{M}_0$ = prop. $\frac{1}{\dot{V}_0}$ or $\dot{M}_0 \cdot \dot{V}_0$ = constant.

$\dot{M}_0 \cdot v_0 = \dot{V}$ = Volume flow or suction capacity in m³/h.

A Steam Jet Pump, therefore, conveys over a wide range with an almost constant volume flow.

If one tests the behaviour of a Jet Pump against varying counter pressures then one finds a dependency as shown in Fig. 12.

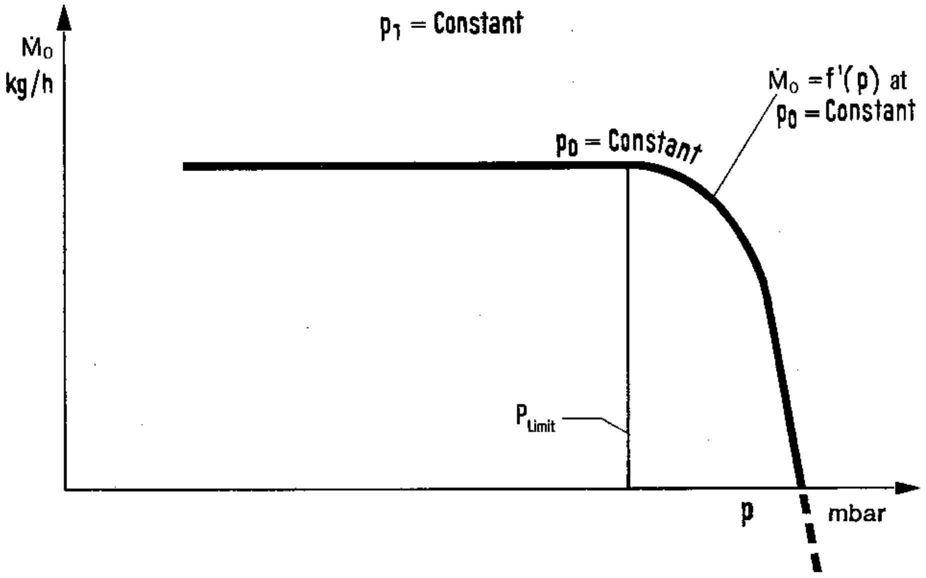


Fig. 12

Fig. 12

The Jet Pump compresses a constant suction flow $\dot{M}_0$ from the suction pressure p_0 to the low counter pressure, as seen in Fig. 12. If, however, the counter pressure rises over the limit counter pressure p_{Limit}, the Jet Pump can only maintain the suction pressure p_0 if the suction flow is reduced corresponding to the falling part of the curve. It may even become negative as is indicated by the dashed part of the curve in Fig. 12. Negative suction flow means a reversed flow direction therefore, steam will be exhausted from the suction connection. One may explain this by the following:

If the pressure at the outlet connection rises over a given value, the energy of the motive jet is no longer sufficient to maintain the pressure difference between p and p_0, then backstreaming from the high pressure side p to the low pressure side p_0 occurs.

For a Jet Pump of given dimensions a whole host of curves $\dot{M}_0 \mp f'(p)$ are ascertainable using p_0 as a parameter.

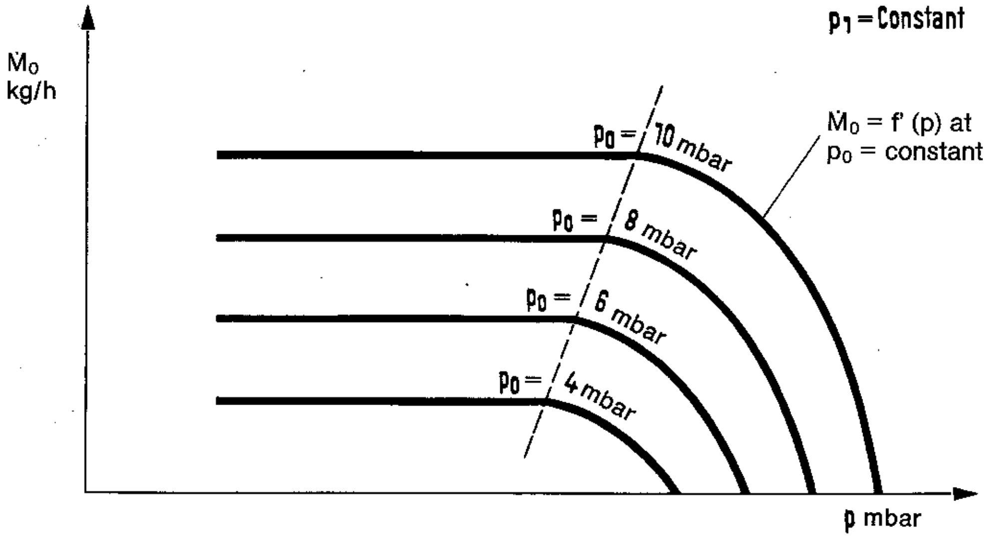


Fig. 13

The broken line joins the limit counter pressure, p_{limit}.

If one draws diagrams Fig. 11 and Fig. 13 on one graph, so one gets a representation on which the whole operating characteristics of a given Jet Pump, at a constant motive steam pressure, become visible. Here then, on the abscissa is shown the suction pressure as well as the counter pressure.

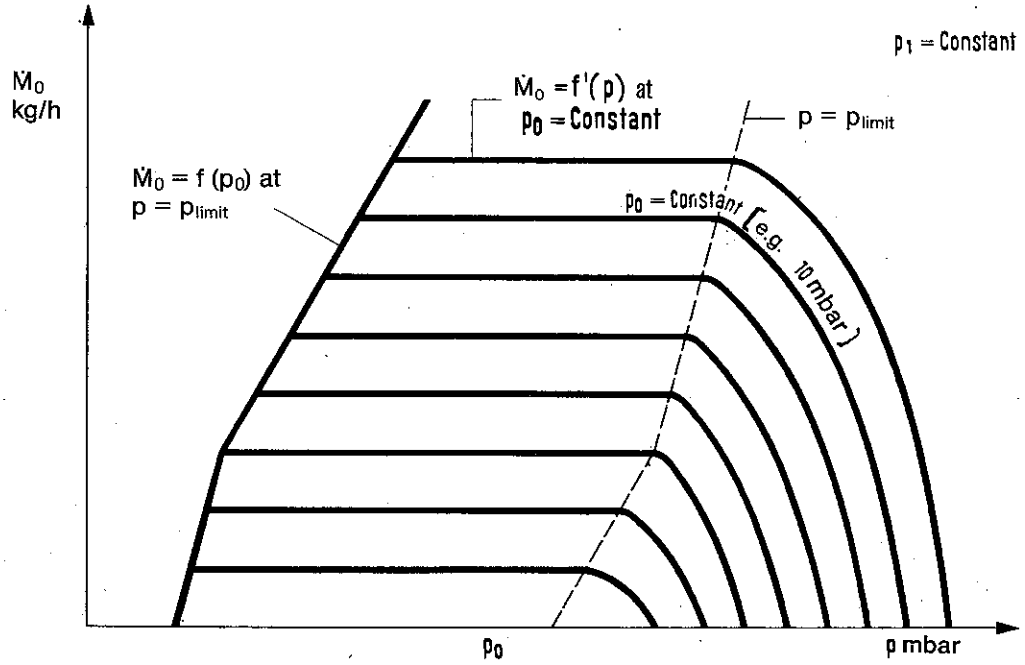


Fig. 14

One can in this diagram also construct curves $\dot{M}_0 = f''(p_0)$ at constant counter pressure, i. e. p = constant.

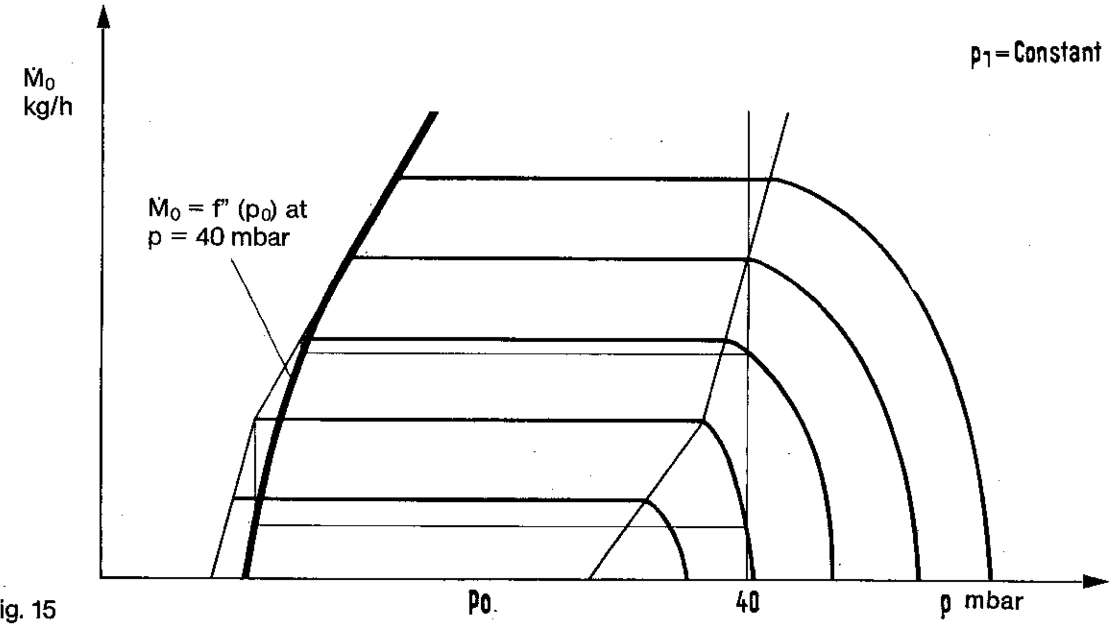


Fig. 15

Up to now it has been assumed, in all considerations of the operating characteristics, that the motive steam pressure p_1 was constant. In Fig. 16 are the characteristic curves of 2 differing motive steam pressures.

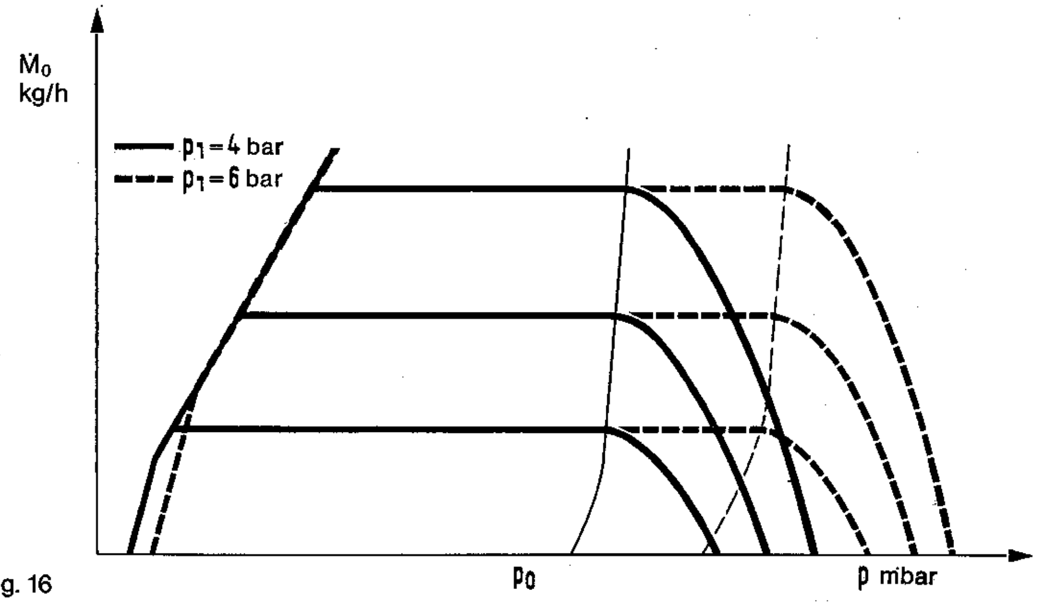


Fig. 16

One sees that higher motive steam pressure principally effects the counter pressure, above the breakoff point the suction curve will, practically, remain unchanged.

This means, therefore, the following rules can be noted:

1. Changing the motive vapour flow (by changing the motive steam pressure or by another motive nozzle bore) has, as a result, only an effect on the reachable counter pressure (limit counter pressure) and no fundamental change of the suction curve.
2. A fundamental change in the suction curve is only possible by changing the diffusor (mixing nozzle) and thereby the whole Jet Pump.

This means that the capacity of a Jet Pump ($\dot{M}_0 \cdot p_0$ = volume flow) is determined by the dimensions of the mixing nozzle, whereas the dimensions of the motive nozzle or the motive steam pressure decide the reachable counter pressure (limit counter pressure).

The capacity of a Jet Pump cannot, then, be increased by the replacement of the motive nozzle. Greater capacities, therefore, larger volume flows, can only be achieved with Jet Pumps with larger mixing nozzle dimensions.