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**Vacuum Control
When Using
Steam Jet Vacuum Pumps**

TKI 21

If the vacuum in a plant has to be controlled at a constant level, the system must be regarded as a whole at the initial design stage, i.e. all the individual parts of the vacuum plant such as the vacuum pump, heat exchangers, pipework, etc., must be taken into consideration. In doing so the following question arises: –

Which parts of the plant influence the vacuum?

In answering this question it becomes obvious which part of the plant has the greatest influence on the vacuum and it is to this part that the controls should be fitted.

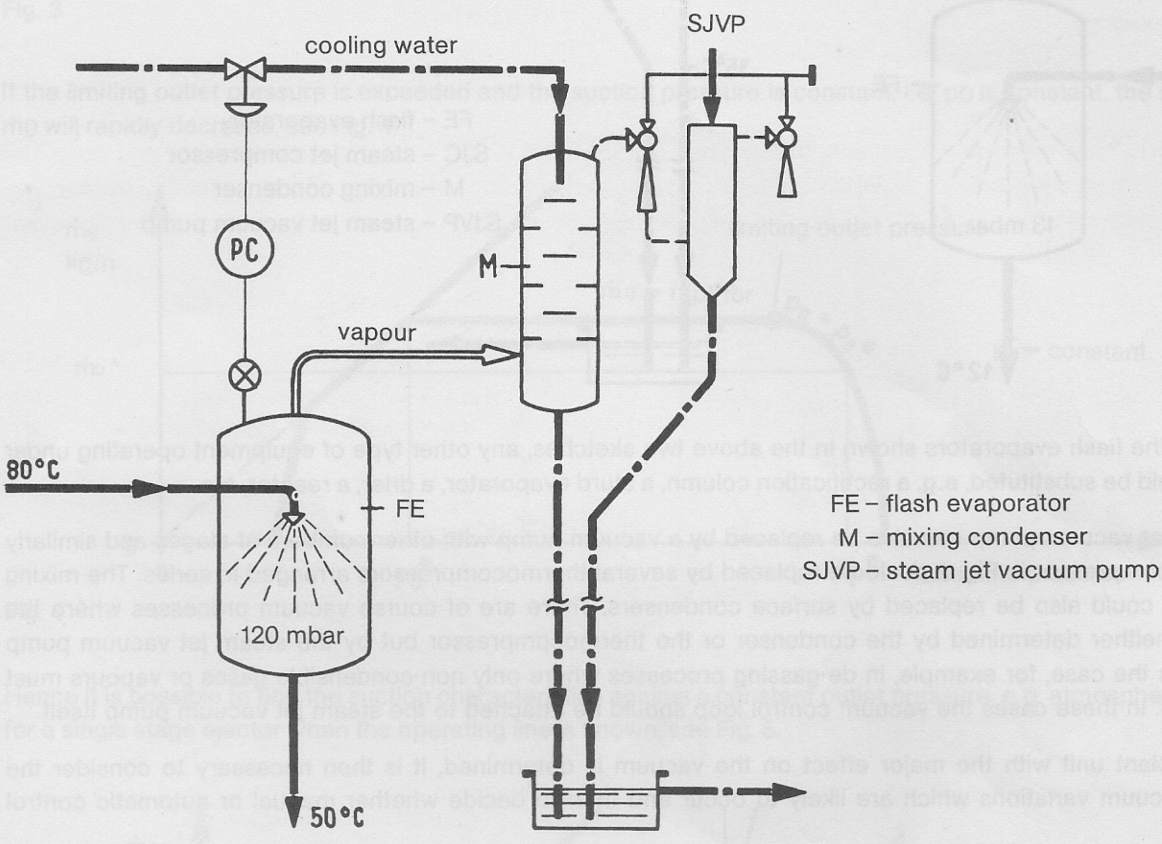

Fig. 1. Condensation plant with the **vacuum controlled by cooling water flow control.**

In the plant as shown in Fig. 1 the cooling water flow through the mixing condenser has the major effect on the vacuum. The steam jet vacuum pump itself is necessary to maintain the vacuum in that it removes the non-condensible gases but it cannot essentially control the vacuum.

For example, if the plant is completely airtight and no gases enter the plant with the product feed stream which is to be cooled, the steam jet vacuum pump can be shut off after the required vacuum has been reached. It therefore has no further influence on the vacuum. It would be incorrect in this case to try to maintain a constant vacuum by controlling the steam jet vacuum pump.

In the plant shown in Fig. 2 the thermocompressor is designed to take the flash vapour at 13 mbar from the flash vessel and to compress this to 60 mbar so that it can condense in the mixing condenser. The steam jet vacuum pump removes only the non condensible gases. In this case the thermocompressor has the major effect on the vacuum and the control loop should be fitted to this item to control the vacuum. It would also be wrong in this case to control the operation of the steam jet vacuum pump in order to maintain a constant vacuum in the flash evaporator.

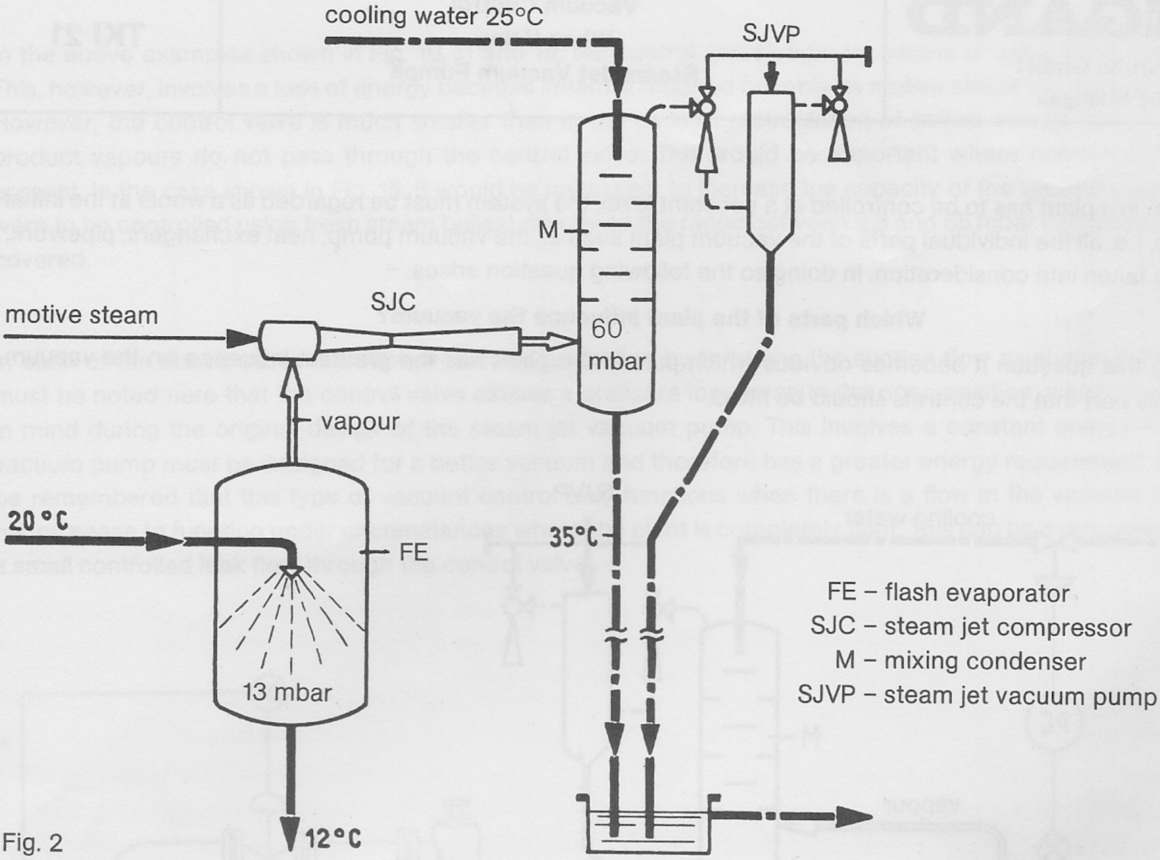

Fig. 2

In place of the flash evaporators shown in the above two sketches, any other type of equipment operating under vacuum could be substituted, e.g. a rectification column, a sturd evaporator, a drier, a reactor, etc.

The steam jet vacuum pump can also be replaced by a vacuum pump with other numbers of stages and similarly the thermocompressor in Fig. 2 could be replaced by several thermocompressors arranged in series. The mixing condensers could also be replaced by surface condensers. There are of course vacuum processes where the vacuum is neither determined by the condenser or the thermocompressor but by the steam jet vacuum pump itself. This is the case, for example, in de-gassing processes where only non-condensible gases or vapours must be removed. In these cases the vacuum control loop should be attached to the steam jet vacuum pump itself.

When the plant unit with the major effect on the vacuum is determined, it is then necessary to consider the potential vacuum variations which are likely to occur and then to decide whether manual or automatic control is required.

There are many plants which are manually operated and controlled which can operate for days without requiring attention. This is because there are either no disturbance factors or that these are removed before they can affect the vacuum.

It is sometimes more effective and less expensive to provide a constant cooling water pressure rather than installing a large scale vacuum control unit. This would be the case where variations in the cooling water pressure are the only cause of variations of the vacuum.

There are a number of possible methods which can be used to control a vacuum at the steam jet vacuum pump itself or at a thermocompressor. In order to understand these possibilities it is necessary to describe the method of operation of the steam ejectors.

The suction flow $\dot{m}_0$ in kg/hr of a single jet ejector is directly proportional to the suction pressure p_0 (absolute pressure) over a large range, assuming that the outlet pressure does not exceed a certain limit value, see Fig. 3. This shows the suction pressure and the outlet pressure both on the abscissa.

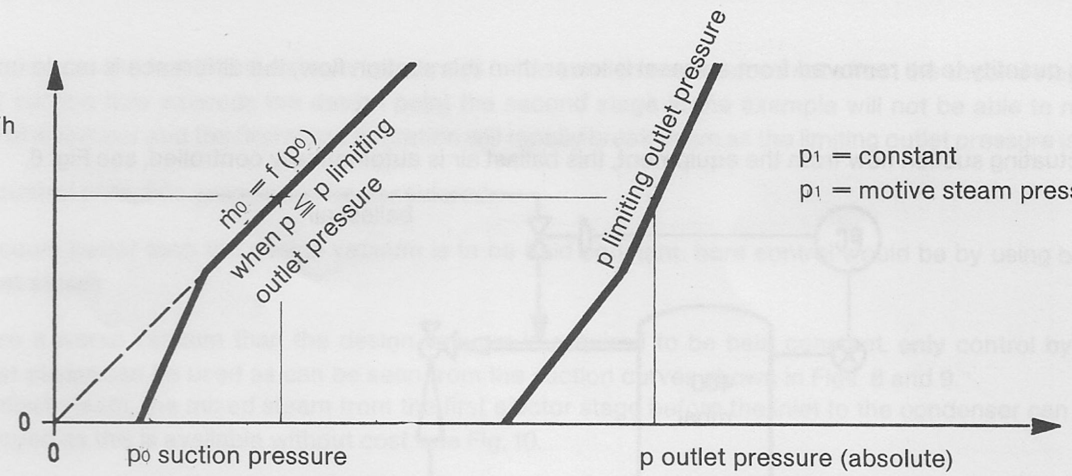

Fig. 3

If the limiting outlet pressure is exceeded and the suction pressure is constant, i.e. p_0 is constant, the suction flow $\dot{m}_0$ will rapidly decrease, see Fig. 4.

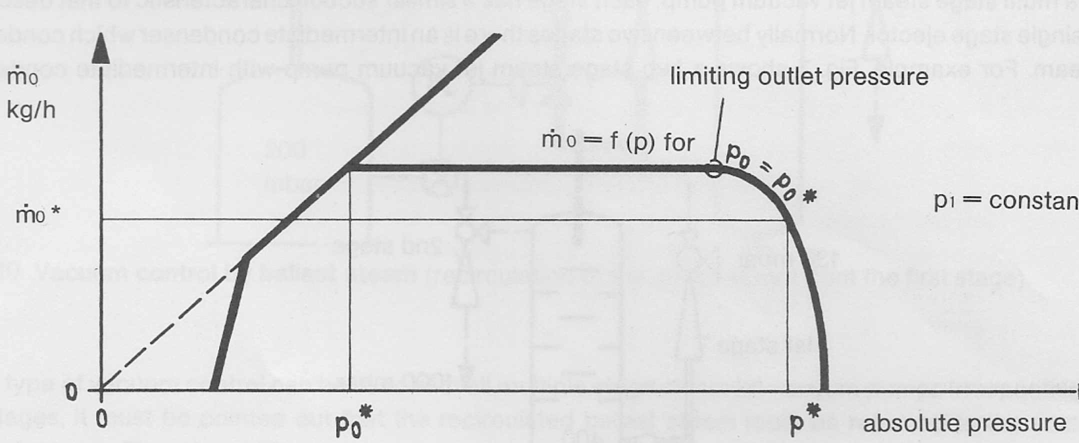

Fig. 4

Hence it is possible to find the suction characteristics against a constant outlet pressure, e.g. atmospheric pressure for a single stage ejector when the operating line is known, see Fig. 5.

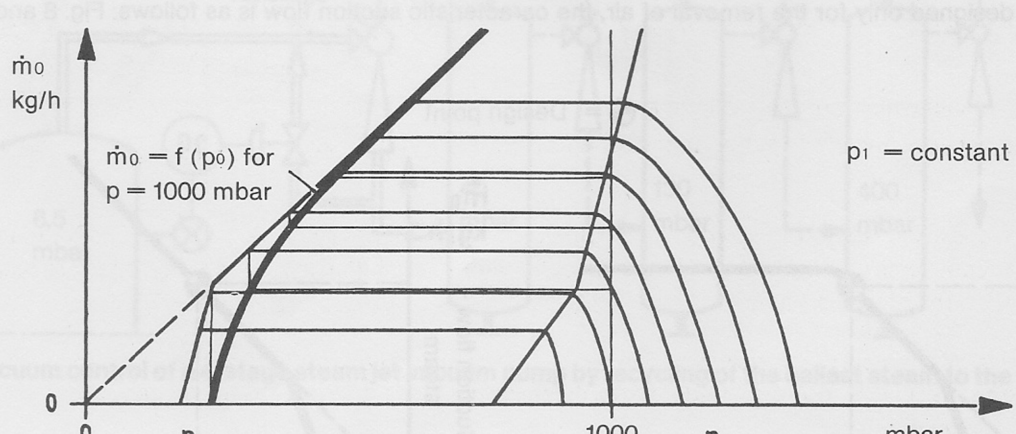

Fig. 5

The caracteristic curve of the suction flow which depends on the suction pressure assuming a constant outlet pressure, is thus steady (linear). To control the vacuum produced by such a single stage vacuum ejector, i.e. the suction pressure, it is only necessary to ensure that the suction flow is constant. This flow is given from the characteristic curve and the required suction pressure.

Where the quantity to be removed from a vessel is lower than this suction flow, the difference is made up by using ballast air.

With a fluctuating suction flow from the equipment, this ballast air is automatically controlled, see Fig. 6.

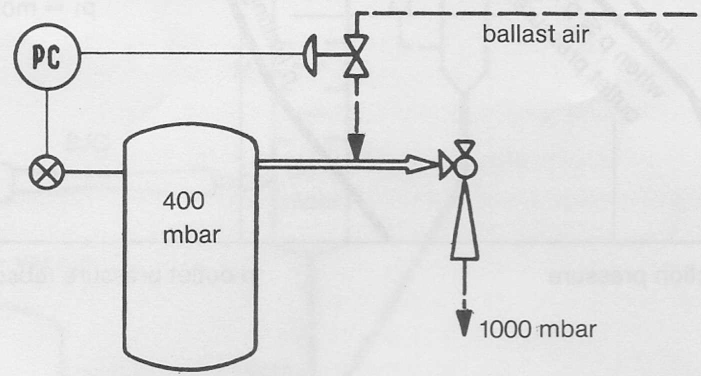

Fig. 6

Vacuum control by ballast air

In the case of a multi stage steam jet vacuum pump, each stage has a similar suction characteristic to that described above for the single stage ejector. Normally between two stages there is an intermediate condenser which condenses the motive steam. For example, Fig. 7 shows a two stage steam jet vacuum pump with intermediate condenser.

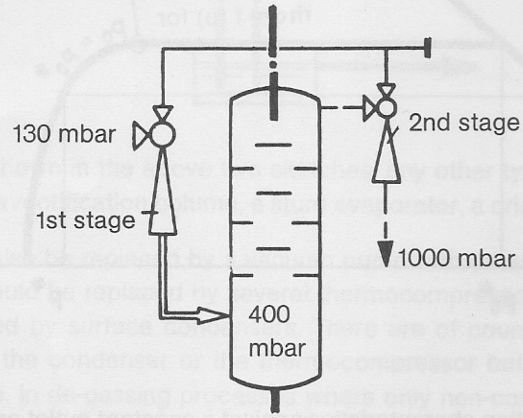

Fig. 7

If this pump is designed only for the removal of air, the caracteristic suction flow is as follows: Fig. 8 and Fig. 9.

= Design point

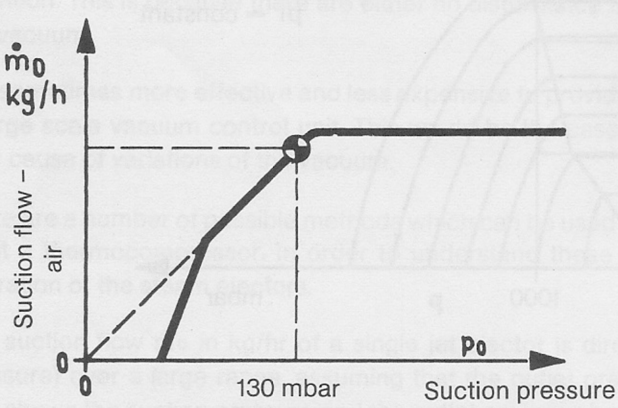

Fig. 8

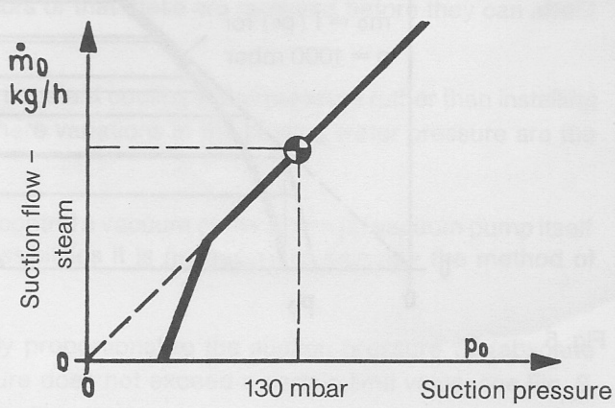

Fig. 9

The suction characteristics for air and steam differ in that the suction curve for air is not linear above the design point whereas the suction curve for steam remains linear above this point.

This follows from the fact that the steam condenses in the condenser and does not affect the second stage, whereas if the air suction flow exceeds the design point the second stage in the example will not be able to maintain the vacuum of 400 mbar and the first stage operation will rapidly break down as the limiting outlet pressure is exceeded.

For the control here, two cases must be considered.

a) A vacuum better than the design vacuum is to be held constant: here control would be by using **ballast air or ballast steam**

b) Where a worse vacuum than the design vacuum is required to be held constant, **only** control by the use of **ballast steam** can be used as can be seen from the suction curves shown in Figs. 8 and 9.
As ballast steam, the mixed steam from the first ejector stage before the inlet to the condenser can be usefully employed as this is available without cost, see Fig. 10.

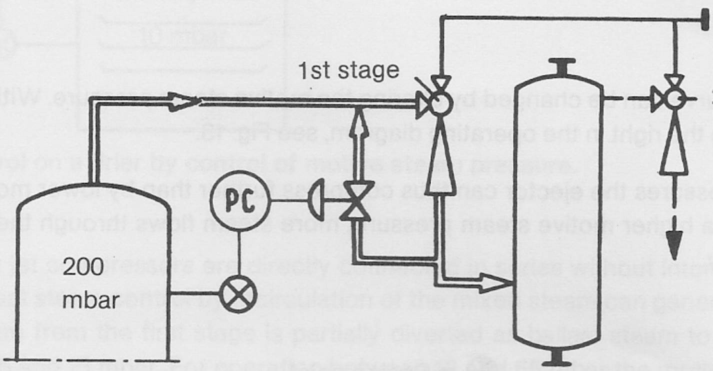

Fig. 10 **Vacuum control by ballast steam** (recirculation of the mixed steam from the first stage).

This type of vacuum control can be used with all multiple stage steam jet vacuum pumps irrespective of the number of stages. It must be pointed out that the recirculated ballast steam must be returned to the suction side of the first stage, see Figs. 11 and 15.

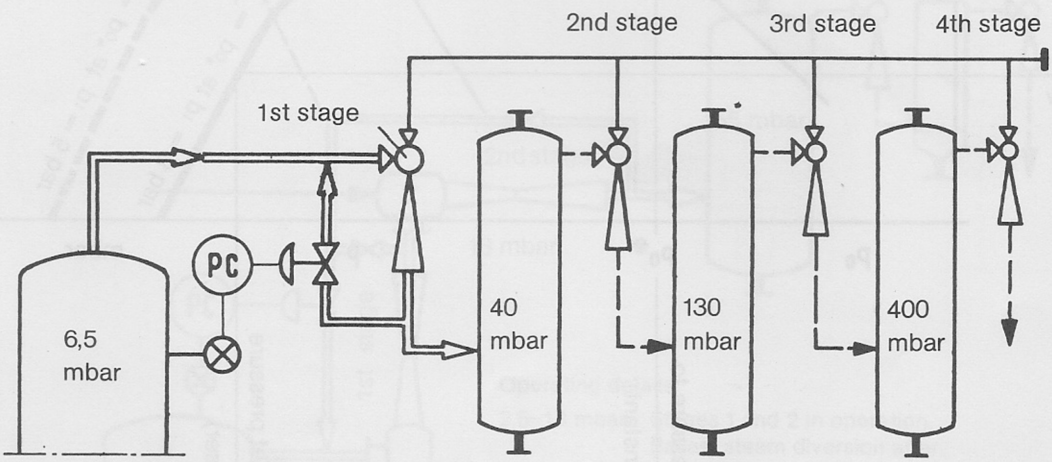

Fig. 11 **Vacuum control of a 4-stage steam jet vacuum pump by recycling of the ballast steam to the first stage inlet.**

The same method can be used for the vacuum control of thermocompressors as shown in Fig. 2, page 2. Under certain conditions, the suction pressure of a thermocompressor can also be controlled by varying the motive steam pressure. This is, however, only possible when the operating line $m_0 = f(p)$ for p_0 = constant, does not fall steeply when the limiting outlet pressure is exceeded, but falls slowly to zero, see Fig. 12.

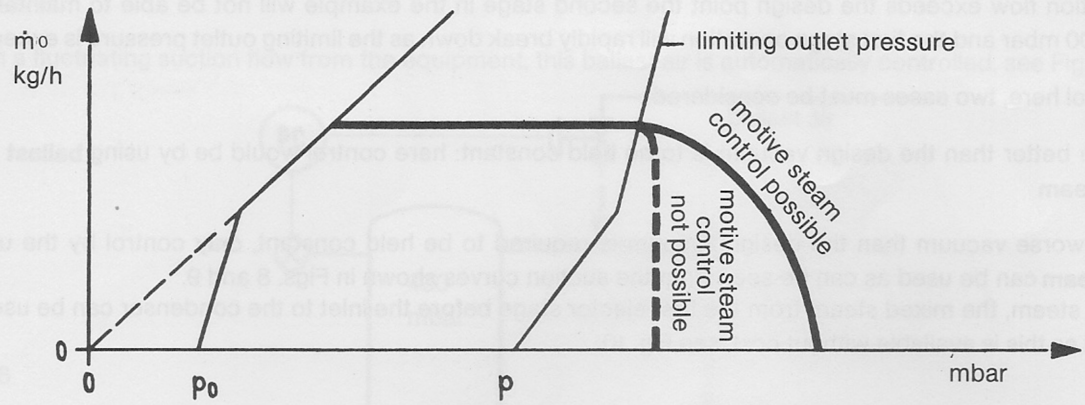

Fig. 12

The limiting outlet pressure curve can be changed by altering the motive steam pressure. With higher motive steam pressures the curve moves to the right in the operating diagram, see Fig. 13.

With higher motive steam pressures the ejector can thus compress further than by lower motive steam pressures; this is obvious because with a higher motive steam pressure, more steam flows through the motive steam nozzle providing more motive energy.

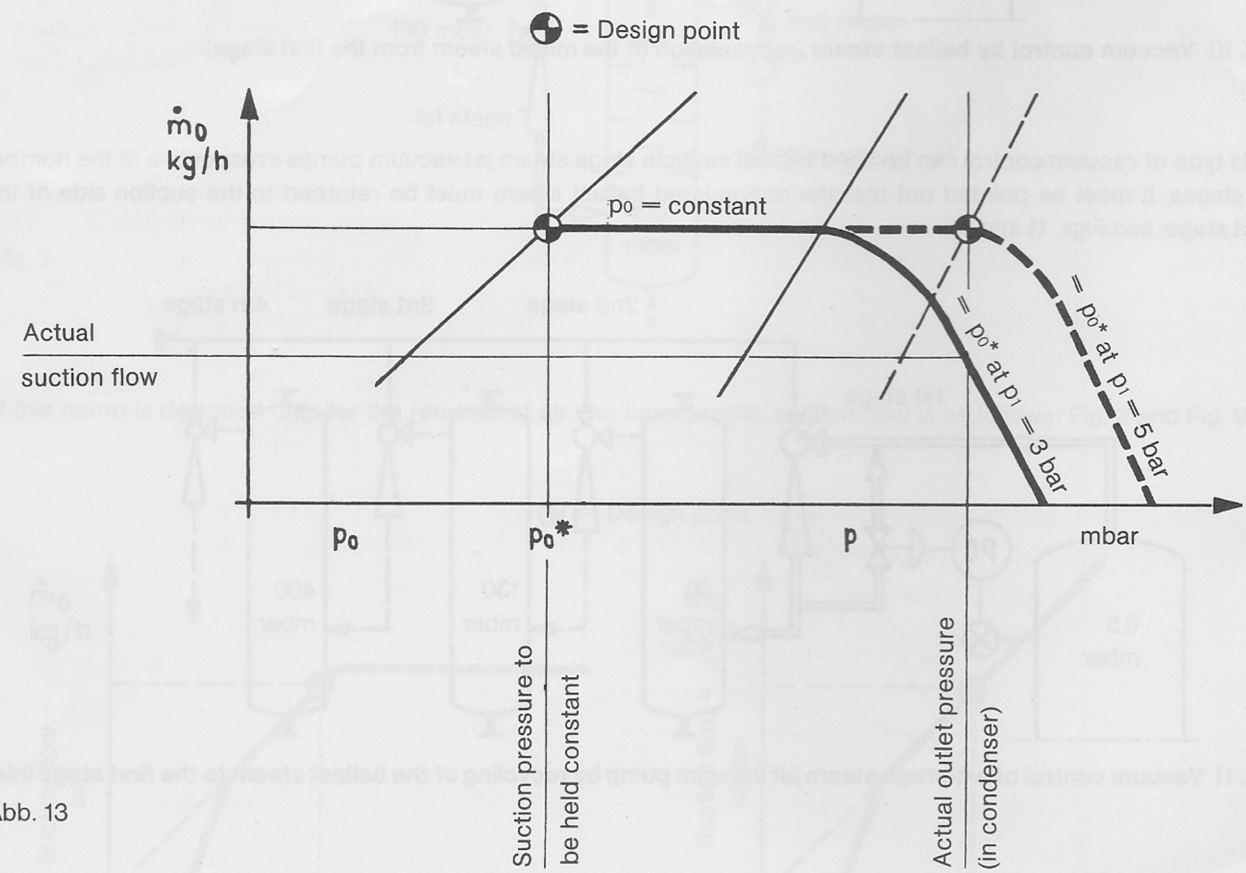

Abb. 13

With the reduced suction flow, as shown in Fig. 13, a vacuum control loop which controls the motive steam pressure will give a motive steam pressure of $p_1 = 3$ bar. This is because the suction pressure (vacuum) p_0^* required, corresponds to the suction flow point on the operating line, when the motive steam pressure is 3 atmospheres, under the prevalent outlet pressure in the condenser. The appropriate control equipment is shown in Fig. 14.

This method of control is simple and also saves steam and water when a smaller quantity is to be removed.

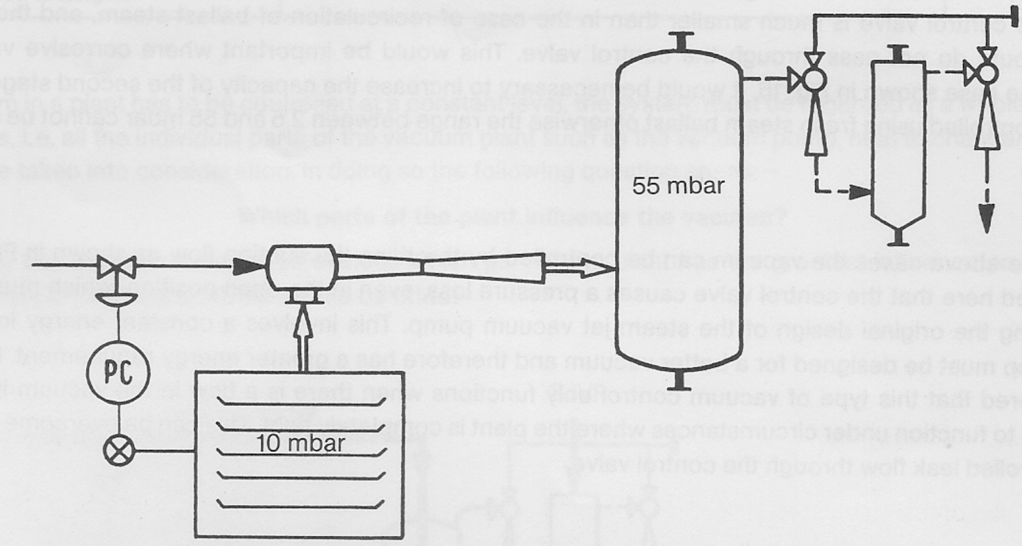

Fig. 14 **Vacuum control** on a drier **by control of motive steam pressure.**

Where several steam jet compressors are directly connected in series without intermediate condensers, only the above described ballast steam control by recirculation of the mixed steam can generally be used, see Fig. 15. Here the mixed outlet steam from the first stage is partially diverted as ballast steam to the inlet to the first stage for operation between 2.5 and 13 mbar. For operation between 13 and 55 mbar the motive steam to the first stage and the recycle line from the first stage are shut off and the ballast steam is recycled from the outlet of the second stage to the inlet of the first stage.

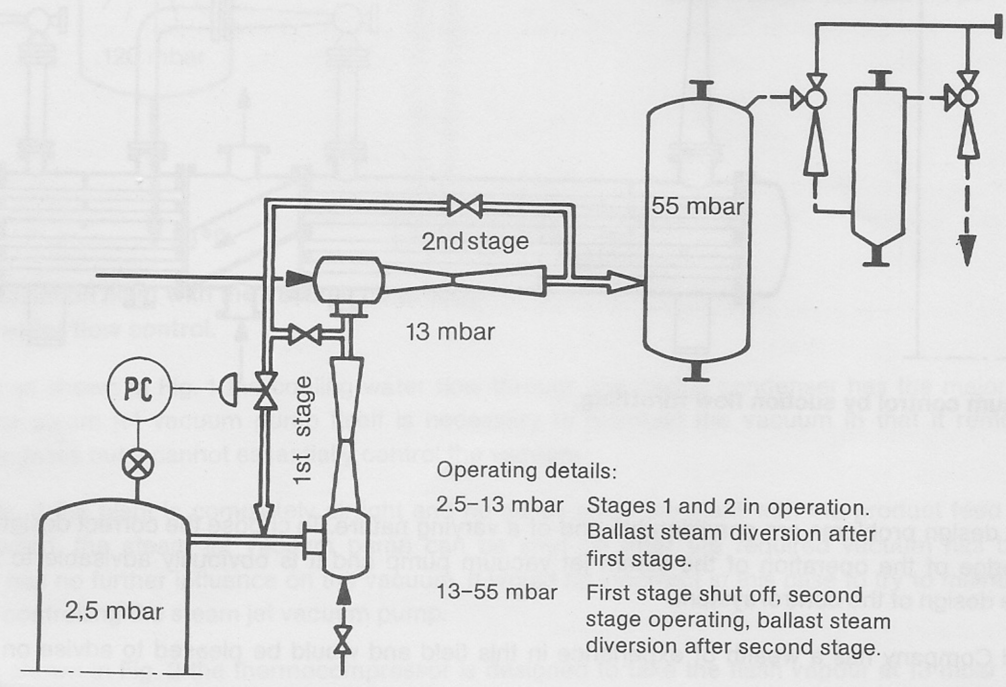

Fig. 15 **Vacuum control by ballast steam** of two steam jet compressors arranged directly in series.

In the above examples shown in Fig. 10, 11 and 15, the control can also be by means of using fresh ballast steam. This, however, involves a loss of energy because steam is required not only as motive steam but also for the control. However, the control valve is much smaller than in the case of recirculation of ballast steam, and the extracted product vapours do not pass through the control valve. This would be important where corrosive vapours are present. In the case shown in Fig. 15, it would be necessary to increase the capacity of the second stage if the unit were to be controlled using fresh steam ballast otherwise the range between 2.5 and 55 mbar cannot be completely covered.

In each of the above cases the vacuum can be controlled by throttling the suction flow as shown in Fig. 16, but it must be noted here that the control valve causes a pressure loss, even in the open position, which must be borne in mind during the original design of the steam jet vacuum pump. This involves a constant energy loss and the vacuum pump must be designed for a better vacuum and therefore has a greater energy requirement. It must also be remembered that this type of vacuum control only functions when there is a flow in the vacuum line and will hence cease to function under circumstances where the plant is completely tight. This can be overcome by allowing a small controlled leak flow through the control valve.

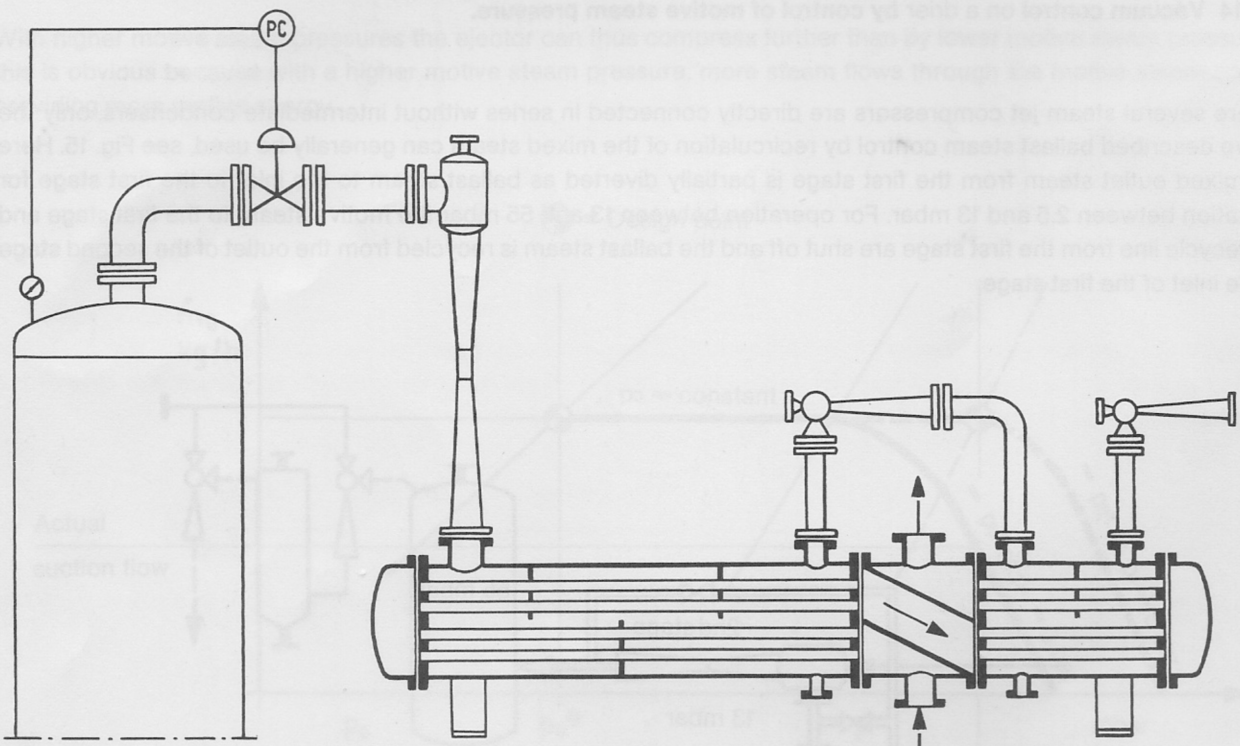

Fig. 16 **Vacuum control by suction flow throttling.**

The potential design problems are considerable and of a varying nature. To choose the correct design requires an exact knowledge of the operation of the steam jet vacuum pump and it is obviously advisable to have expert advice for the design of the control system.

The Wiegand Company has a wealth of experience in this field and would be pleased to advise on this type of application.