Comparative vacuum performance calculations of mechanical booster pumps

Mechanical dry vacuum pumps are increasingly replacing traditional steam ejectors in vacuum secondary steelmaking plant applications because of their ‘greener’ credentials. It is extremely important that accurate prediction of mechanical booster capability is made, otherwise the vacuum system, when delivered, may not meet the user’s requirements, forcing changes to process parameters or subsequent additions to equipment to compensate for the shortfall.

Some manufacturers use simplified prediction models which are likely to predict a performance better than can be achieved in practice. The magnitude of the potential errors can be very significant, and grows with increasing number of mechanical booster stages. Edwards Ltd, a major supplier of vacuum equipment, has an advanced simulation capability and provides a more accurate prediction of the system performance and hence better definition of equipment requirements.

**Author:** Dr Neil Turner
**Edwards Ltd**

Despite current world economic fluctuations and their effect on steel prices, underlying demand for steel generally, and high quality steel in particular, remains significantly positive. To meet this demand for quality steels, significant attention is focused on expanding secondary steel processing, including vacuum degassing (VD) and vacuum oxygen decarburisation (VOD) processes (including Ruhrstahl Heraeus (RH)-type systems). In addition, lowering energy consumption and reducing greenhouse gas emissions are becoming ever more important in modern steel plants. This is why mechanical dry vacuum pumps are increasingly replacing traditional steam ejectors in these applications. This technology is today commonly used with VD plants as it has proven reliable in many installations worldwide, and is increasingly used for VOD and even RH plants with huge suction speed requirements.

A typical installation for steel degassing is shown in Figure 1. To meet the performance requirements under vacuum of <1mbar, this requires installation of multiple stages of mechanical boosters, combined with appropriate backing pumps. The design of these vacuum systems demands profound understanding of mechanical booster performance in different pressure regimes.

The compression ratio vs. backing pressure with different pumping speed efficiency characterises the performance of a mechanical booster, and this information is often provided by a manufacturer to assist in vacuum system design.

**Fig 1 Example of a mechanical vacuum system**
The calculation of these characteristics needs to be done using a sophisticated model of vacuum performance based on a full geometrical description of the mechanical booster. This takes into account all profile and clearance dimensions as well as dynamic effects.

It is straightforward to characterise the steady-state performance of a vacuum pumping system when all its elements are known and it discharges to atmospheric pressure. A simple speed (or throughput) curve versus inlet pressure suffices, and the tiny variations in performance when the discharge pressure changes as a result of changing weather conditions are generally miniscule and can be ignored.

The backing pump for a mechanical booster is not always known in advance, so the manufacturer cannot provide a pumping speed characteristic for their machine. There needs to be some means of calculating the performance of a combination. If the backing pump can be characterised with a speed curve (or equivalent), then some manufacturers can use proprietary methods based on the detailed design of their product to reasonably accurately predict the performance of the combination.

Most manufacturers would choose not to make the proprietary methods available to people outside their own organisation, viewing them as their intellectual property. There is a method, based on compression characteristics, that has been widely used to estimate the performance of a booster in combination with an arbitrary backing pump. In the absence of sophisticated methods, that method is still used by some people (manufacturers without any other model and end-users) to predict the performance of a mechanical booster with a given backing pump-set. It will be shown that this method of calculation is very likely to predict a performance better than can be achieved in practice.

**CALCULATING BOOSTER PERFORMANCE**

*Figure 2* resembles the information provided by some manufacturers of mechanical boosters. Each of the (almost) parallel curves in the chart is a compression ratio curve shown as a function of outlet pressure; the compression ratio is normalised to the peak compression. Each curve represents a different, constant pumping speed $S$ for the booster, denoted by $K_{\eta v}$, where $\eta v = S / S_0$ is the proportion of the physical displacement $S_0$ of the mechanical booster realised as pumping speed. Lines of constant inlet pressure appear on the chart as parallel lines at 45° to the (logarithmic) axes.

A method for predicting the vacuum performance can be illustrated with reference to the data in *Figure 2*. For example, suppose the booster in this chart has a physical displacement of 47,500 m$^3$/hr, with peak compression $K_{max} = 85$. What does the backing pump speed need to be to achieve a pumping speed of 33,250 m$^3$/hr at 0.67 mbar inlet pressure, at the inlet of the mechanical booster? To find the answer find the intersection of the 0.67 mbar inlet pressure line with the compression curve for $\eta v = 33,250 / 47,500 = 0.7$. This happens at an outlet pressure of 14 mbar, which, therefore, requires 1,600 m$^3$/hr backing speed. The same information can be interrogated to answer different questions, as desired, for example, to calculate the compression for a given throughput with a given outlet pressure, or to predict the pressure that could be achieved at the mechanical booster inlet.

It is common to present the allowed region of operation on the same chart as the performance characteristics. For example, in Figures 3 and 4, the bold pink lines divide the space into the area to the left or below the line where operation is allowed, and the area above or to the right of the line where it is not.

**Fig 2** Normalised compression characteristics for a mechanical booster pump

**Fig 3** Compression characteristics with thermal limit line. A high compression booster with severe thermal limits
of the line, where operation is restricted. Normally the indication is of some kind of thermal limitation. The two figures are plotted on the same scales, but have quite different mechanical booster characteristics. The data in Figure 3 describe a booster pump likely to have fairly tight clearances which is capable of producing relatively high compression ratios at low flow, but has a large area of thermal restriction. The data in Figure 4 describe a booster which is thermally more capable, probably because it has larger internal clearances, but which does not produce such high compression at low flow. (The booster described in Figure 4 is an Edwards booster, which also happens to have a slightly lower displacement than the one in Figure 3, which is from another manufacturer, though you cannot tell this directly from the characteristics presented.)

It might be considered at first sight that the second design is inefficient and less likely to be suitable for a given application, but it could in fact be more suitable than the apparently more efficient design. For example, consider an application that requires pumping a throughput of 23kg/h at an inlet pressure of 0.67mbar or below. The continuous black line in each chart connects the points on each booster characteristic which correspond to pumping 23kg/h, and the broken grey line corresponds to an inlet pressure of 0.67mbar (as before). In Figure 3 those lines intersect at an outlet pressure of around 12mbar, whereas in Figure 4 the indicated outlet pressure is around 6.5mbar. However, only for the second booster is the point of intersection within the allowed operating envelope of the machine, thus the first booster is unable to take advantage of its superior compression ratio. In fact, the first point at which the 23kg/h contour emerges into the region of permitted operation for the first booster is when the backing pressure is around 4.5mbar. So, although the first booster produces more compression at low flow than the second, it will need a 40% larger backing pump-set, than the smaller displacement booster with the lower compression characteristic.

If all the characteristics on the chart are measured or calculated accurately, then methods like those described above should give an accurate estimate of the performance in a given configuration. However, it is common for only the zero-flow compression curve to be measured, and the other characteristics to be derived from it. (In fact, in Figures 2-4, only the zero-flow compression characteristic is estimated accurately and the characteristics for higher values of $\eta V$ are calculated using the simple method described next.)

**THE SIMPLE CALCULATION METHOD**

The throughput $Q$ from inlet to outlet of a booster is given by the difference between the displaced throughput $Q_d$ and the back-leakage $Q_l$.

$$Q = Q_d - Q_l$$

\[Q = Q_0 - Q_l\]
The leakage is assumed to be characterised by a conductance, which multiplies the pressure difference between inlet and outlet.

\[ Q_L = C(p_{in} - p_{out}) \]  \hspace{1cm} (2)

Note that \( C \), the leakage conductance, is not a constant, and is generally unknown. Substituting this expression into equation 1 and dividing by \( p_{in} \), the inlet pressure, gives the following:

\[ S = S_0 - C(K - 1) \]  \hspace{1cm} (3)

Here, \( K = p_{out}/p_{in} \) is the prevailing compression ratio. In the next step, we assume that the leakage conductance can be derived from the zero-flow compression characteristic, by setting \( S = 0 \) in the preceding equation.

\[ C = \frac{S_0}{K_0 - 1} \]  \hspace{1cm} (4)

It is not obvious how to use these conductance values in the case where the inlet and outlet pressures are different from the zero-flow case. The actual conductance will be a complicated function of the inlet and outlet pressures, and is dependent on the geometrical parameters of the mechanical booster. Generally, in these simple calculations, it appears to be assumed that the conductance depends only on the outlet pressure.

Making this substitution yields equation 5 for the pumping speed:

\[ S = S_0 \frac{K_0 - K}{K_0 - 1} \]  \hspace{1cm} (5)

This can be rearranged to give an equation for \( K \) in terms of \( K_0 \) and \( p_{in} \):

\[ K = K_0 \left(1 - \frac{p_{in}}{p_{out}}\right) \]  \hspace{1cm} (6)

In both of these expressions, the values of \( K \) and \( K_0 \) should be taken to be at the same outlet pressure. This last equation is the one plotted in Figures 1-3.

Where the backing speed \( S_0 \) is known, then the assumption of continuity \( S = S_0 K \) can be used to estimate the compression.

\[ K = \frac{K_0}{1 + \frac{S_0}{S_0 - 1}(K_0 - 1)} \]  \hspace{1cm} (7)

If any of the assumptions made during the calculation of the flow performance of the mechanical booster from the zero-flow compression characteristic are invalid then it is likely that calculation method will be inaccurate.

In fact, the assumptions used in the estimation of the leakage conductance in the simple calculation are only actually valid in the molecular flow regime, when the gas behaves as a collection of independent molecules, a situation only likely to occur at the very lowest outlet pressures <0.05-0.1 mbar.

When the outlet pressure of the booster pump is relatively high >5-10 mbar, the leakage flow through the clearances will be in the continuum regime, where the gas in the channels behaves as continuous fluid. Then the leakage flow can be characterised by a conductance multiplying the pressure difference, but that conductance is far from depending only on the upstream pressure (the pressure at the outlet of the mechanical booster). The conductance of the leak paths will increase significantly with downstream pressure (at the inlet of the mechanical booster). It is well known, for example, that the conductance of a duct in the continuum laminar flow regime, where the gas flow depends primarily on the viscosity of the gas, is approximately proportional to the average of upstream and downstream pressures. Any estimate for the leakage conductance calculated with very low downstream pressure, will always be an underestimate of the actual conductance when the downstream pressure is higher.

It is possible/likely that the compression at zero-flow would be sufficiently high to choke the flow through the clearance. Under those conditions, the leakage flow is not proportional to the pressure difference, but depends only on the upstream pressure. Again, an estimate for the leakage conductance calculated from the choked flow through the pump clearance, will always be an underestimate for the actual conductance when the downstream pressure is higher.

Both failures, using a choked flow condition to estimate the continuum conductance and neglecting the increase in conductance with increasing downstream pressure, lead to the leakage flow being underestimated when the pressure difference reduces, ie, the performance of the mechanical booster will be overestimated to an increasing degree as the pumping speed increases.

In most conditions of interest the flow regime in the leakage clearances would be described as transitional. Here, the flow is not entirely described by the viscosity of the gas, but neither is it a collection of independent molecules. In this regime the effective leakage conductance will also increase with increasing inlet pressure, in a similar way to the continuum regime so the simple calculations will be similarly inaccurate.

When the pressure at the outlet of the booster pump is low, the dominant back-flow is not through the clearances at all, but is caused by so-called ‘root trapping’ of gas caught between the rotor profiles and transferred back to the inlet by the rotation of the machine. This element of back-flow also does not depend on the pressure difference between the inlet and outlet of the mechanical booster, but depends only on the pressure at the outlet. By failing to account for these effects, the simple calculation will again underestimate the back-flow, leading to overestimation of the pumping performance as the pumping speed increases.

It is clear that the simple calculation is very likely to overestimate performance across the entire range of outlet pressure.
MORE SOPHISTICATED CALCULATION METHODS

To accurately capture the performance of the booster pump requires a physical model of all the back-flow processes within the machine. The Edwards models include an accurate geometrical description of the entire mechanism, which enables the calculation of root-trapping effects, and leakage models for the clearances that work across the entire pressure range. Booster performance is measured regularly on laboratory test benches, providing data for model refinement (see Figure 5).

COMPARISON OF CALCULATION METHODS

Figure 6 compares the compression curves (lines) derived from the zero-flow compression curve, using the method described earlier, with compression curves (symbols) calculated at the same pumping speeds using the Edwards performance model.

In all cases the performance according to the sophisticated calculation is inferior to the simple calculation. As expected, the simple method consistently understates the leakage flow.

The two calculation methods agree most closely when the pumping speed is low (light blue) when the inlet and outlet pressures most closely resemble the zero-flow case. The deviation gets progressively worse as the pumping speed increases, as the inlet pressure at any given point will be much higher than that on the zero-flow curve.

The deviation is greater at low pressures. In the worst case depicted on the chart ($\eta_v=0.8$, $p_{out}\approx0.2\text{mbar}$), the compression ratio predicted by the simple calculation is more than a factor of ten greater than that from the sophisticated calculation.

COMPARISON WITH MEASURED CHARACTERISTICS

Generally, real-life performance measurements agree much better with the sophisticated calculation than they do with the simple calculation.

The average deviation between the measured compression ratio in Table 1 and the sophisticated calculation is approximately 15%, whereas the average deviation between the measured values and the simple calculation is approximately 75%.

This is the outcome anticipated above and shows the magnitude of the discrepancy between real life performance and predictions that users of the simple calculation might discover. Note that the data and calculations in Table 1 are for a different machine from those referred to in earlier figures. The equality of the measured values of the normalised compression ratio at the two points where $\eta_v=0.7$, is entirely a coincidental peculiarity of the particular system under test, as is the equality of the two values calculated using the

<table>
<thead>
<tr>
<th>$\eta_v$</th>
<th>Outlet pressure mbar</th>
<th>Normalised compression ratio</th>
<th>Measured</th>
<th>Simple calculation</th>
<th>Sophisticated calculation</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.7</td>
<td>0.496</td>
<td>0.124</td>
<td>0.234</td>
<td>0.144</td>
<td></td>
</tr>
<tr>
<td>0.7</td>
<td>2.82</td>
<td>0.124</td>
<td>0.234</td>
<td>0.159</td>
<td></td>
</tr>
<tr>
<td>0.5</td>
<td>0.059</td>
<td>0.120</td>
<td>0.184</td>
<td>0.122</td>
<td></td>
</tr>
</tbody>
</table>

Table 1: Comparison of compression calculations with speed-curve measurements

**Fig 6** Calculated compression characteristics of a mechanical booster generated using two different methods

**Fig 7** Comparison of calculated pumping speed with measured performance
simple method. Overestimating the compression for a given backing pressure leads to overestimation of the pumping speed available at a given inlet pressure. Figure 7 shows predictions of pumping speed using both the simple and sophisticated methods against measured performance. The simple calculation overestimates the pumping speed available by approximately 20%, whereas the sophisticated calculation is a very good match for the measured performance. The simple calculation could lead the designer to propose a vacuum system with four pump sets to meet a user requirement, when in fact five sets would be required. If the calculation of performance involves calculating multiple stages of mechanical booster, then errors of this magnitude will be compounded at each step.

**IMPLICATIONS FOR END-USERS**

Difficulties arise for end-users in comparing performance modelling from different sources. It is important that the user understands the methods used and appreciates the likely deficiencies. Where an end-user approaches a vacuum systems manufacturer like Edwards with an advanced simulation capability, it is likely they will receive an accurate prediction of the system performance, and the system specified by the manufacturer is very likely to meet the user’s requirements. When the system specification is based upon overly-simple calculations, it is very likely that the system performance is being overestimated. The vacuum system, when delivered, may not meet the user’s requirements, forcing changes to process parameters or subsequent additions to the vacuum pumps and supporting infrastructure to compensate for the shortfall. The magnitude of the potential errors can be very significant, and grows with increasing number of mechanical booster stages. **MS**

Dr Neil Turner is Senior Technologist with Edwards Ltd, part of Atlas Copco Vacuum Technique, Global Technology Centre, Burgess Hill, UK.

**CONTACT:** Anke.Teeuwsen@edwardsvacuum.com

---

**DEDICATED TO STEEL DEGASSING**

Edwards is the smart choice if you’re looking for a complete module with proven performance, or if you want to replace any mechanical booster. The new Edwards GMB40K leads the pack in the large booster class thanks to:
- An extremely compact design
- Unrivalled performance that results in faster pump down
- A small carbon footprint because of its low energy consumption
- Worry-free maintenance with spare parts that are readily available