Towards the single pump solution: Recent development in high speed machines for dry vacuum pumping

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The goal of a single (dry) pump capable of reaching high vacuum and itself exhausting to atmospheric pressure has long existed in the vacuum field. The development of a single-shaft, high-speed pump EPX (second generation) mechanism has gone some considerable way to achieving this goal. The pump exploits molecular drag and a fluid dynamic mechanism and is discussed in terms of the theory of operation, internal flow mechanisms and the technology innovations enabling the mechanism to be produced in large volume quantities. Further development and application examples will be discussed to illustrate the system-simplification that the pump can provide. © *2005 American Vacuum Society.* [DOI: 10.1116/1.1978889]

I. INTRODUCTION

A full generation of the use of dry pumping mechanisms in the vacuum field has been developed and applied. Dry pumps can significantly simplify and improve vacuum systems[1] and are often the first generic choices of primary pumping mechanism employed. Different types of dry pump are widely used in for example, the semiconductor and chemical process industries. Typically the medium vacuum is attained and used as the process medium. Recent advancements in technology have allowed the development of the IPX and EPX (second-generation) vacuum pump mechanisms. IPX is the BOC Edwards designated name for their manufactured pump described here, EPX is the second-generation version (lower power and footprint). These are single pumps achieving high vacuum and exhausting to atmospheric pressure. They are especially suited to load-lock and low to medium harsh duty applications.

II. PRINCIPLE OF OPERATION

EPX pumps achieve performance by means of a combination of molecular drag and fluid dynamic mechanisms. They consist of inlet molecular drag stages of the Holweck[2] type with an "in-series" multistage regenerative compressor as the backing stage, both of which are mounted on the same shaft. The simplest pump configuration is illustrated in Fig. 1. The Holweck stage (Fig. 2), comprises a stator in which are machined a set of parallel helical grooves forming a set of parallel pumping channels. The rotor is simply a plane cylinder. Alternatively the stator may be a plane cylinder and the helical grooves machined in the rotor. Both types are used in the pumps to be described. A particular feature of the machines described is that the helical grooves occupy less than one turn so that the pressure distribution is the same in each channel, see Fig. 1. Since there are no relatively high pressure sections adjacent to lower pressure sections, leakage between channels is minimized and the mechanism is relatively efficient both in terms of compression ratio achievable (per unit length of channel) and in terms of pumping speed.

This design feature enables the development of an axially compact machine but its limitation on compression ratio does lie in the fact that the pumping channels are relatively short so that, for the same peripheral speed, the compression ratios achievable are less than those for a multiturn helix. The compression ratios are maximized by effectively isolating the higher pressure sections. This also minimizes back-leakage and keeps speed efficiency high. Additionally this means that, for practical dimensions and rotational frequencies, the exhaust pressure cannot be more than a few mbar. Even so, useful compression ratios (of the order of 10^7) can be achieved for rotational frequencies of $\sim$300 Hz and the machine can develop an exhaust pressure that is sufficiently high for the fluid dynamic or regenerative compressor stage to operate.

A simple theory of the Holweck stage can be developed along similar lines to that described by Helmer[3] for the case of a Gaede mechanism. In the case discussed here, the theory is adapted for the Holweck case and gas transport is described by the Knudsen transitional flow expression. Equating the volumetric flow per unit length of channel to the back leakage flow driven by the developed pressure difference gives

$$\frac{uA}{L} = \left\{ C_m + C_v \left(\frac{1 + k_1 P}{1 + k_2 P} \right) \right\} \frac{1}{P} \frac{dP}{dx}, \tag{1}$$

where k_1 and k_2 are the constants in the Knudsen transitional flow formula. C_m is the molecular channel conductance and C_v is the viscous laminar channel conductance (at the pressure P and linear distance along the channel x). The conductances here include the leakage conductance from adjacent channels, which is equivalent to an additional component of backflow. u is the component of peripheral velocity resolved along the channel and weighted in proportion to the moving area of channel. A is the cross-sectional area and L the total length of a channel.

This solves to yield

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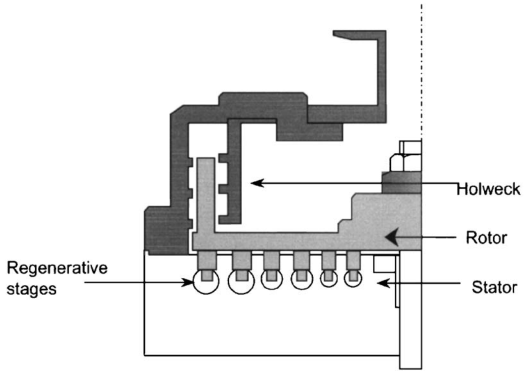

FIG. 1. Pumping mechanism detail.

$$\frac{uA}{C_m} = \frac{C_v}{C_m}(K-1) + \ln K + \left(\frac{k_1}{k_2} - 1\right)\ln\left(\frac{1 + Kk_2P_d}{1 + k_2P_d}\right), \qquad (2)$$

where K is the compression ratio (channel exhaust to inlet) and P_d is the inlet pressure. Compression ratio K is a complicated function of channel area $A=d.b.$ (where a is the channel height and b the width) and exhaust pressure P_e,

$$K \propto (Ap_e)^{-1}. \qquad (3)$$

K reduces with increasing P_e as the channel exit moves into continuum flow. Equation (3) gives relative performance for real pump designs—optimizing of channel section geometry for maximized K.

Note that, if leakage is ignored and flow is molecular, then K reduces with

$$K \propto \exp\left(\frac{uA}{C_m}\right) \propto \exp\left(\frac{uL}{d}\right) \qquad (4)$$

increasing pressure as the channel exit moves into continuum flow. Equation (4) gives good guidance on relative performance for real pump designs. The theory is however, too oversimplified to give accurate and absolute predictions, especially for varying cross sections.

Equations (3) and (4) give pressure ratios correlating well with experimental values for constant section channels. In particular it shows that compression ratio is very sensitive to the channel area (which falls dramatically as the channel area is increased), and also shows that the compression ratio falls rapidly with increasing exhaust pressure as the channel exit moves into continuum flow. Practically however, it is most useful in comparing the effects of differing geometrical parameters for real pump designs in which the channel section is reduced toward the exhaust in order to improve compression ratio.

The inlet stage of the 500 m^3/h capacity EPX machine, for example, is primarily designed for volumetric capacity and modest compression ratio; effectively acting as a booster for later stages of the machine. The equations have been used to guide the initial choice of channel dimensions and the final design refined by Monte Carlo and phenomenological modelling. It is interesting to note that the measured peak volumetric speed of this machine is within 5% of the theoretical volumetric speed (μA)—indicating that interchannel leakage is relatively small. This confirms relatively small interchannel leakage of the Holweck stages. The leakage has greater effect on compression ratio due to the exponential dependence.

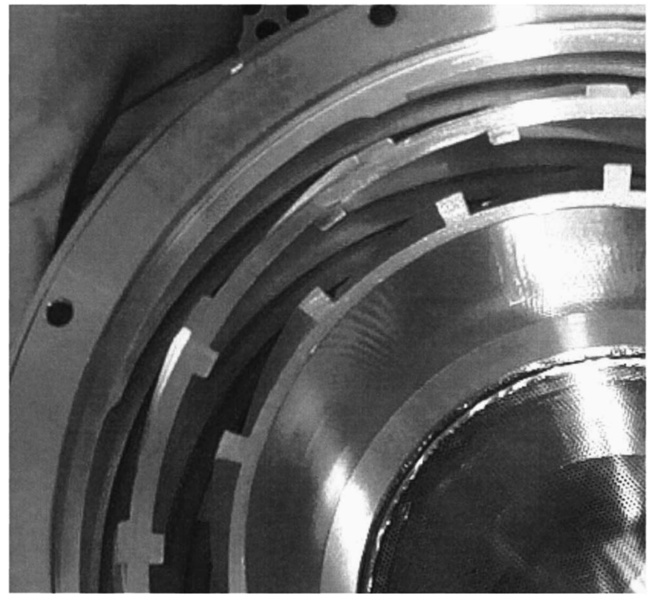

FIG. 2. View of Holweck channels.

Note for the rotor channel the resolved velocity is

$$u = u_\theta \frac{2a+b}{2(a+b)} \qquad (5)$$

and for the stator channel

$$u = u_\theta \frac{b}{2(a+b)}. \qquad (6)$$

Equations (5) and (6) apply for high-speed inlet stage (500 m^3/h): Holweck channel in rotor (at 300 Hz, $u \sim 100$ m/s). Later stages are designed for compression with relatively shallow channels. These stages have Holweck channels in the rotor, mainly due to logistics of machining and assembly.

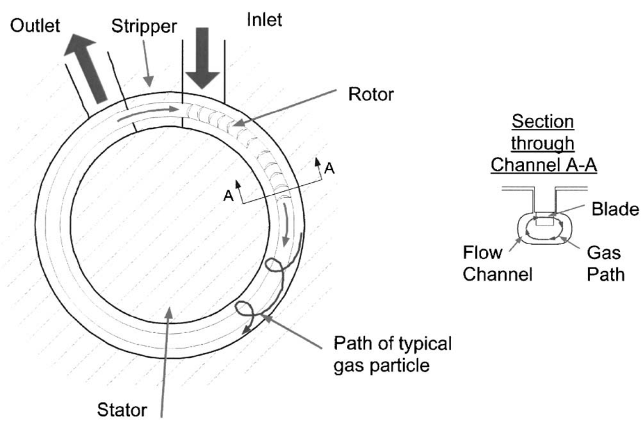

FIG. 3. Regenerative pumping stage.

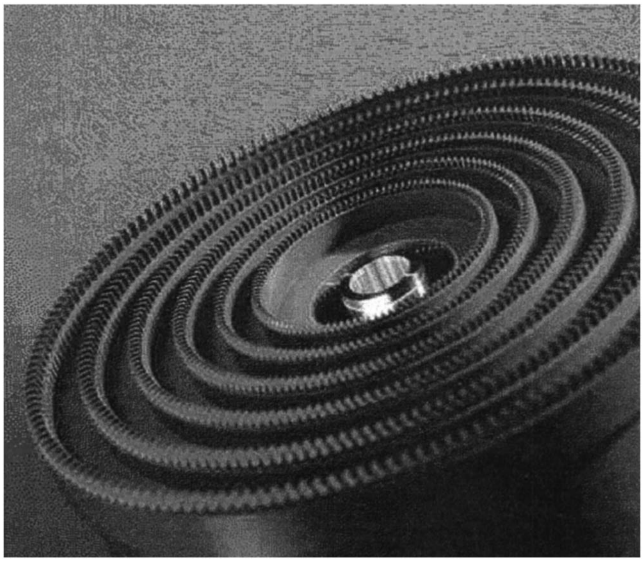

FIG. 4. Circumferentially arranged a set of axially oriented vanes.

To maximize speed of the 500 m^3/h EPX machine the Holweck channels are relatively deep and, in order to maximize compression ratio, the channels are machined in the rotor so that a relatively large area of the channel is in motion. However, since the Holweck stages near the exhaust have channels of relatively small depth (in order to maximize compression ratio) there is no great advantage in constructing the channels in a rotor. Also helical rotors are much more difficult to manufacture so the higher-pressure stages are designed with the channels in the stator.

The *regenerative* stage of the pump, sometimes called a vortex or fluid dynamic pump, is illustrated in Fig. 3. The rotor comprises a disk on which a set of axially oriented

FIG. 5. Stripper sections.

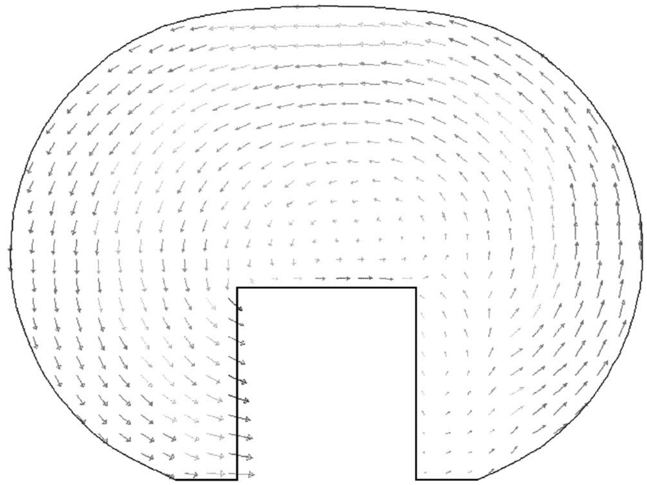

FIG. 6. Circulation pattern in regenerative stage at high pressure.

vanes are circumferentially arranged, Fig. 4. As the disk spins, the vanes move at high speed through a toroidal channel stator. Through most of its length the toroidal channel cross section is circular and significantly larger than the vanes. The inlet and outlet ports are separated by a portion of the channel, referred to as the "stripper," see Fig. 5. Here the channel cross section changes to a shape similar to and slightly larger than the vane cross section.

The motion of the vanes entrains, and induces a rotation in, the gas so that the gas follows a helical path though the toroidal stator, passing through the rotor vanes and re-

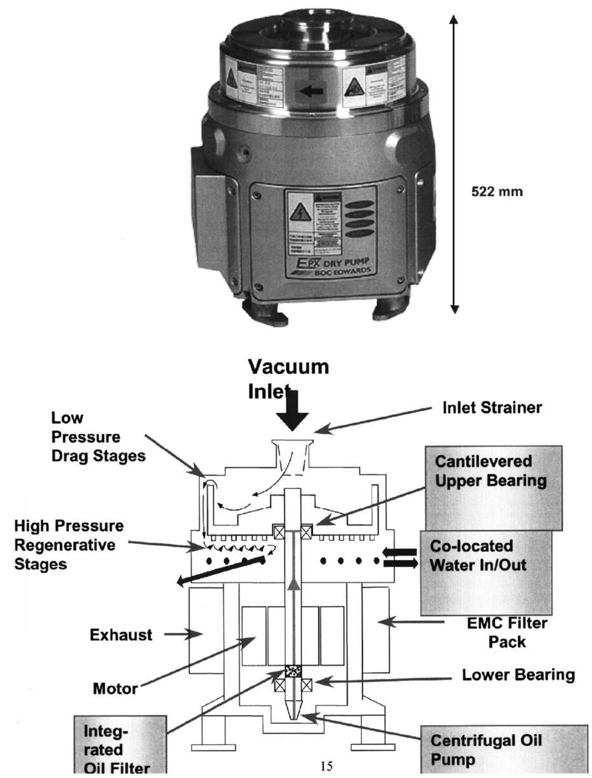

FIG. 7. Pump external and internal.

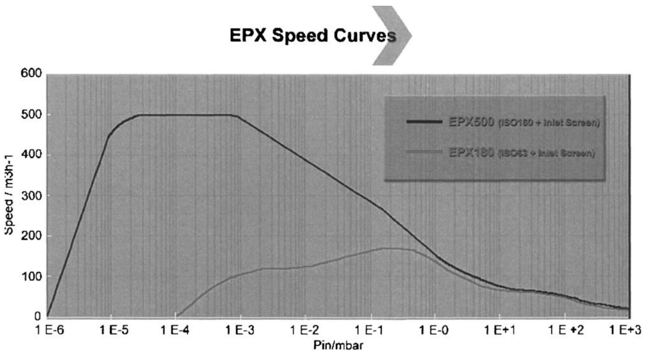

FIG. 8. EPX range pumping speed curves.

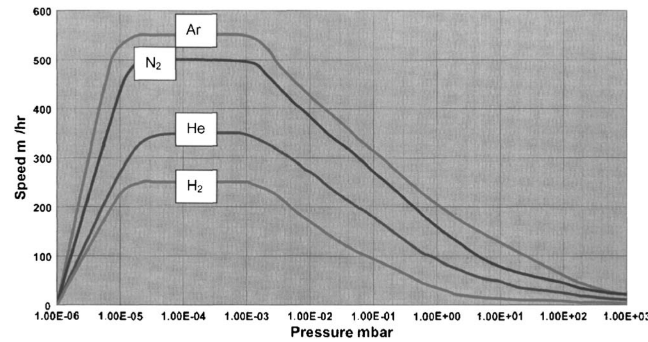

FIG. 9. EPX500 pumping speed curves for a range of gases.

entering them several times during one rotation as in Fig. 3. The circumferential rotation of the gas is thus regenerated a number of times during each rotation—hence the reference to the device as *a regenerative compressor*. As the vanes pass through the stripper section, gas between the vanes is carried back to the pump inlet. This action constitutes the main source of leakage. The circumferential gas rotation is thus *regenerated*. N.B. the regenerative mechanism tends to maintain the circumferential velocity of gas although the mean velocity is less than the blade velocity.

A number of authors have attempted to develop theoretical models of the pumping action. Iverson[4] considered the vanes as a very rough surface while Wilson *et al.*[5] and Sixsmith[6] considered the vortex action. The latter analyzed the two components of velocity v_t (tangential) and v_e (meridional plane perependicular to v_t) at each point. The tangential or "through" flow Q through the open channel area A is a function of A, ω (rotational frequency) and radial dimensions. The latter also determine losses and back leakage. The principle applies to viscous flow regime but there is a small molecular drag component. Both these analyses require experimentally determined empirical factors. Conclusions from the extensive empirical evidence indicate that a vortex action does occur. Experimentally the compression ratio has found to be determined almost entirely by carryover and leakage though the stripper.

Figure 6 shows the result of a Computational Fluid Dynamic (CFD) analysis of the flow at relatively high pressure (near atmosphere)—the circulation through the blades is clearly evident. This is further illustrated in Fig. 6. The complex nature of the flow renders theoretical analysis extremely difficult so that as discussed both types of theoretical model developed incorporate a number of empirical factors which have to be determined experimentally. The article's authors' approach has relied on CFD and experimental analysis. Here flow is "matched" at the Holweck/regenerative mechanisms' interface. Since the action of the regenerative pumping stage requires hydrodynamic flow, the pump cannot produce significant speeds or compression ratios at inlet pressures less than a few mbar. Also, because of the relatively large carryover, the regenerative pumping stage does not generate large compression ratios. Thus, in order to achieve inlet pressures in the range in which the drag stage can operate, the pump is designed with a number of regenerative stages in series. These stages are formed at progressively smaller diameters on the same rotor and the average K is 3 to 4 for a stripper section. The circumferential gas rotation is *regenerated*.

III. MECHANISM

In different machines, greater speeds and lower ultimates are achieved by "stacking" more Holweck stages while retaining the same regenerative backing stages. The Holweck stages are of smallest cross section nearest to the regenerative stage and maximizes compression for the regenerative interface. These stages are arranged radially so that the axial length of the machine is not significantly increased as more stages are added. The Holweck stages nearest to the regenerative stage are designed with relatively small channel cross sections. The reduction in the characteristic dimension (which reduces C_m and C_v) enables greater compression ratios to be achieved, as indicated by Eq. (1). This is particularly important at higher pressures (in transitional flow and approaching viscous flow) so that the Holweck stages can generate a sufficiently high pressure for the regenerative stages to become effective.

The simplicity of the two pumping mechanisms are exploited so both can be on the same rotational shaft, hence allowing a single shaft operation. This allows no or noncontacting, seal mechanism to be used where the flow from the first Holweck stages are matched and optimized in the high-pressure fluid dynamic mechanisms. The pump is an inverter driver and is shown in Fig. 7. The literature documents optional ways of employing the regenerative principle: periph-

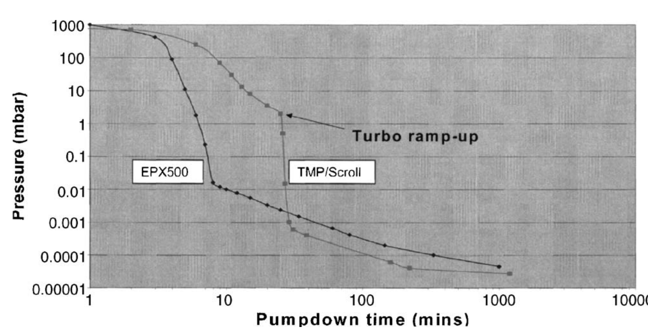

FIG. 10. Typical pump down performance.

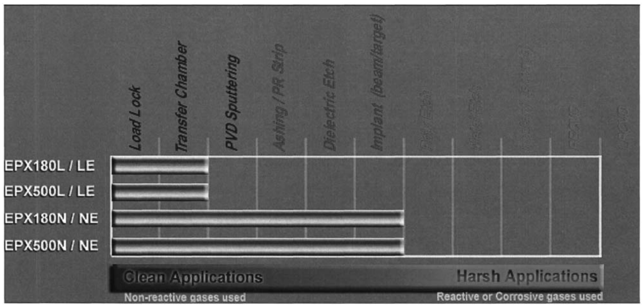

FIG. 11. Application range of EPX.

eral, side channel, and vortex. The models discussed above utilize CFD and with empirical elements are used for analysis and optimization of pump parameters including speed, power, temperature distribution, stress, tolerance, and lubrication.

IV. PERFORMANCE

Figure 8 shows the speed curves of the series of EPX (second generation) machines. The common platform of the regenerative step gives common performance at the higher pressures (>1 mbar). This is compared in Fig. 10, with the pump down by a 500 l/s turbomolecular and 30 m^3/h scroll pump combination. Figure 9 shows a range of speed curves for different gases. The low pressure speed curves (in molecular flow) follow the classical molar mass $M^{1/2}$ dependence. An application example is shown in Fig. 10, where a 1000 l laser cavity was evacuated by a 500 m^3/h machine. Figure 11 gives an indication of semiconductor related light LE and medium (requiring gas module) NE nonreactive gas duty applications on which the EPX are used.

V. SUMMARY

The theory and mechanism of operation of an EPX (second generation) machine has been described and is exploited to enable high vacuum pumping in a single pump mechanism. The pump high rotational shaft frequency allows a compact device which typically sees application in load-lock, medium duty and laboratory applications. The ultimate pressure of the pump is in the 10^{-7} mbar range with a peak pumping speed of 500 m^3/h. The inverter drive allows atmospheric cycling and rapid load-lock evacuation, typically in less than 10 s. The second generation EPX versions have been progressed to enable medium semiconductor duty application. The pumps have found wide-ranging use elsewhere, for example in research applications where cleanliness and robustness are paramount.

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