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CRITICAL FACTORS OF HIGH-PRESSURE HYDROGEN COMPRESSION IN HYDROGEN TECHNOLOGIES

KRITICKÉ FAKTORY VYSOKOTLAKÉ KOMPRESE VODÍKU U VODÍKOVÝCH TECHNOLOGIÍ

Abstract

Hydrogen technology is connected with many problems occurring during storage, liquefaction, distribution and utilization of hydrogen. The economy of all these processes is affected by medium-pressure and high-pressure compressors efficiency, which is with regard to thermodynamic properties of hydrogen function namely of their working space leakage. An experimental study of these phenomena was used to proposition of mathematical model describing the effect of reflux from working space on compressor efficiency.

Abstrakt

Vodíková technologie je spjata s řadou problémů vyskytujících se při skladování, zkapalňování, distribuci i využívání vodíku. Hospodárnost všech těchto procesů je ovlivňována účinnosti středotlakých i vysokotlakých kompresorů, která je s ohledem na termodynamické vlastnosti vodíku funkcí zejména netěsnosti jejich pracovních prostorů. Experimentální zkoumání těchto jevů bylo využito k navržení matematického modelu popisujícího vliv zpětného proudění z pracovního prostoru na účinnost kompresorů.

1 INTRODUCTION

Even if hydrogen is only the power carrier, not the power source, it has nowadays valuable property for the energy utilization, due to its capability to be stored. Hydrogen can be stored in gaseous state, liquid state or in metallic or chemical hybrids. Purposeful and now also realistic utilization of hydrogen power industry presents itself for storage of electricity from wind farms whose direct connection to synchronic transmission network endangers the balance between the electrical power system production and consumption.

Interesting technologies regarding the storage of oversupply from wind power plant production were published in [3]. General diagram of hybrid power plant with electricity reversible connection to transmission network and electrolysis through inspection and control units enabling solution of wind power station capacity fluctuation is displayed in Fig. 1. A medium-pressure rotary compressor is used for delivery of produced hydrogen to distributing network; high-pressure piston compressor is used for hydrogen compression to accumulator and liquefier. A fuel cell is used for inverse transformation. The hydrogen is stored in accumulator or distributed for further utilization. Even at dual transformation (wind power to hydrogen and later hydrogen power to electricity), the total efficiency of wind energy utilization can reach 25 %.

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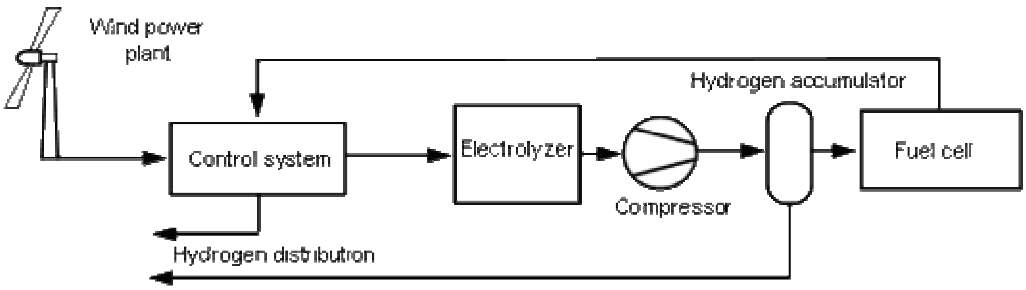

Fig. 1 Storage of oversupply from wind power plant

The hydrogen power supply exploitation with reasonable economy is therefore necessary joined with efficiency of the hydrogen compressors working in middle-pressure mode during hydrogen transport in networks (screw compressors and turbo compressors). However, for its liquefying only the high-pressure piston compressors are suitable.

2 HYDROGEN COMPRESSORS

The compressors processing the hydrogen and its mixtures are exposed to particularly hard conditions. The hydrogen diffuses quickly to materials as the small size of hydrogen atoms enable his penetration to metallic lattice, especially to locations with disturbed texture under the surface even at normal temperature. Atom aggregation into H_2 molecules and its accumulation in the cavities prevents back diffusion even at pressure increase in these areas. Then cracking of material without apparent changes on the surface occurs, leading - due to hydrogen explosiveness - unpleasant disruptions.

Besides embrittlement, decarburization at higher pressure takes place, which leads to reduction of steel strength and ductility, thus the usage of alloyed steels preventing grain boundary corrosion, to which the plain steels are subjected, is necessary. Liquid hydrogen enhances this risk. Hydrogen corrosion namely in places of local pulsating pressure strain is the most serious hydrogen impact on metallic structures.

Light gas compression is connected with the effort to achieve the acceptable operation economy, which in a large extent depends on optimum compressor choice. In this competition, the turbo compressors have disadvantageous position, because compressor wheel material strength limits the permissible circumferential velocity and thus maximum pressure ratio of the stage. This amounts due to hydrogen low mass only value 1,08; so the turbo compressors can not be used for achieving higher pressure ratios.

The advantage of non-lubricated screw compressors is the cleanness of the compressed gas. However, fix adjusted pressure ratio accompanied by inadaptability to changing service conditions does not allow their usage for high-pressure accumulator filling. Moreover, the efficiency course demonstrates its low values in the area of light gases (see Fig. 8). Achieving higher efficiencies is here limited by the permissible rotor circumferential velocity.

The only possible option for high pressures achieving are the vertical piston compressors or balanced-opposed compressors, which thanks to valve operating gear obtain all desired pressure ratios.

The construction and disposition of the balanced-opposed compressors enables namely easy adjustment of their main parameters. The operation price and economy are comparable to screw compressors, the noise level is lower.

3 HIGH-PRESSURE PISTON COMPRESSORS

The diagram of high-pressure, five-stage, balanced-opposed compressor for hydrogen compressing (Fig. 2) on 20 MPa pressure shows reverse circulating flows $\dot{m}_c$ (Fig. 3) through leak valves and also around pistons and external losses $\dot{m}_o$ due to leakage into surroundings. Compressor input P_K is used not only for supplied gas amount $\dot{m}_d$ compressing, but also for compressing of total gas amount in cylinders $\dot{m}_S$:

$$P_K = \frac{\dot{m}_S \cdot a_{pol}}{\eta_K} \tag{1}$$

Whereas

$$\dot{m}_s = \dot{m}_d + \dot{m}_o + \dot{m}_c \tag{2}$$

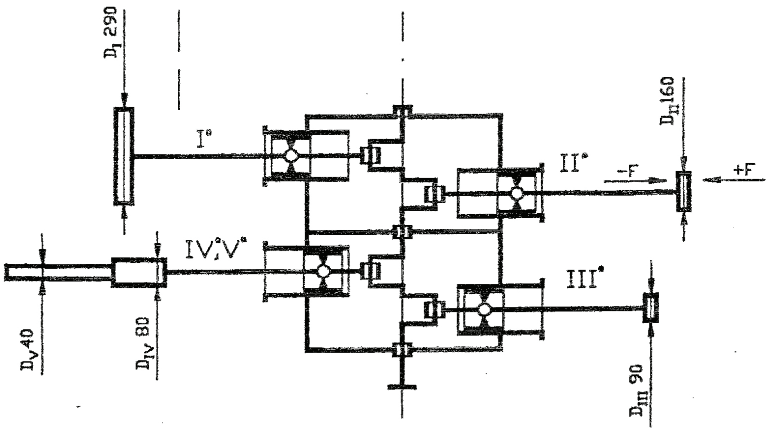

Fig. 2 High-pressure compressor for hydrogen

From the flow diagram in Fig. 3, leakage coefficient λ_N can be determined, which affects substantially the machine input, especially for gases with high gas constant r.

$$\lambda_N = \frac{\dot{m}_d}{\dot{m}_S} \tag{3}$$

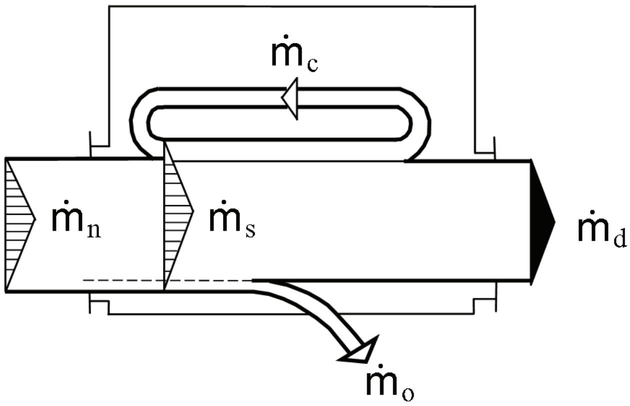

Fig. 3 Mass flow diagram

Valve leakage is from this point of view crucial, because for repaired, long time not-revised valves the primary leakage under the valve plate can reach even more then 20 %. Imperfect mounting of valve into valve chest seat and namely late valve plate fitting contribute to this phenomenon.

Generally it is possible any leakage of working space like leakage of valves, piston rings, seals to consider as gas flow through the conduit. This fact was utilized to mathematical modeling of absolute working space leakages ($\dot{m}_o + \dot{m}_c$).

Models were subsequently reviewed experimentally by analysis of gas flow in conduits of various relative lengths.

4 GAS FLOW THROUGH THE CONDUIT

Typical example of reflux from the compressors working space is the mass flow through the conduit between the piston machine seal and plunger. The actual, alternative and equivalent cross-section of the conduit between the seal and plunger is demonstrated in Fig. 4.

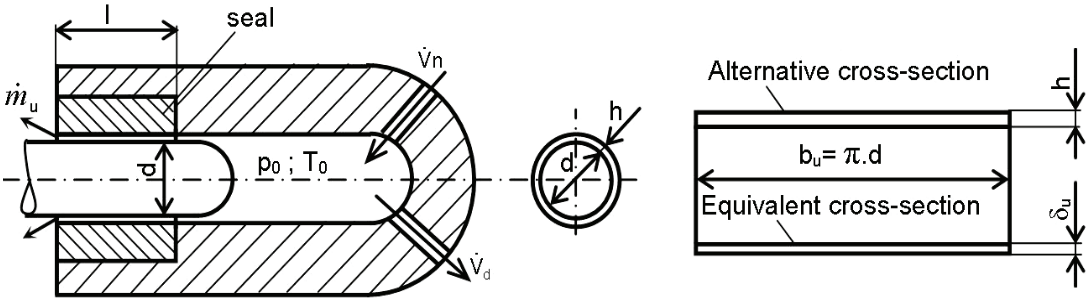

Fig. 4 Cross-sections of the conduit

The calculation of gas mass flow through the conduit whose height h is changing irregularly is difficult. The conduit height change in the compressor working space is caused by e.g. eccentric position of piston rod in seal, radial grooves in the valve plate seat, axial grooves in the cylinder or variable oil film thickness.

For simplification of mass flow calculation, equivalent cross-section of conduit is used.

$$S_{ek} = b \cdot \delta \tag{4}$$

The conduit length is determined by the seal length. The characteristics of flow through the alternative cross-section originated after actual cross-section rollout does not change with respect to very small conduit height *h* against plunger diameter *d*, it is dependent upon reduced conduit length

$$\bar{x} = \frac{l}{h} \tag{5}$$

4.1 Mass flow through the conduit

The volume flowing through the inlet conduit cross-section is

$$\dot{m} = S \cdot w_1 \cdot \rho_1 = S \cdot w_1 \cdot \rho_0 \cdot \frac{\rho_1}{\rho_0} \tag{6}$$

It is possible to express the velocity w_1 with using the reduced velocity μ_1 [1], [5] as

$$w_1 = \mu_1 \cdot w_k = \mu_1 \cdot \sqrt{(\kappa \cdot r \cdot T_k)} = \mu_1 \sqrt{\left(2 \cdot r \cdot T_0 \cdot \frac{\kappa}{\kappa + 1}\right)} \tag{7}$$

Further after substitution for ρ_1 / ρ_0 is

$$\dot{m} = S \cdot \sqrt{\left(\frac{2}{r \cdot T_0}\right)} \cdot p_0 \cdot \mu_1 \cdot \sqrt{\left(\frac{\kappa}{\kappa+1}\right)} \cdot \left(1 - \frac{\kappa-1}{\kappa+1} \cdot \mu_1^2\right)^{1/(\kappa-1)} \tag{8}$$

Since the term

$$\psi' = \mu_1 \cdot \sqrt{\left(\frac{\kappa}{\kappa+1}\right)} \left(1 - \frac{\kappa-1}{\kappa+1} \cdot \mu_1^2\right)^{1/(\kappa-1)} \tag{9}$$

is the discharge coefficient of the conduit, is the mass flow

$$\dot{m} = S \cdot p_0 \cdot \psi' \cdot \sqrt{\left(\frac{2}{r \cdot T_0}\right)} \tag{10}$$

For low conduits, where $\bar{x}$ > 1000 and h < 0,1 mm, Haken-Poiseuille law is also used for the mass flow calculation

$$\dot{m} = \frac{b \cdot h^3}{24 \cdot \eta \cdot r \cdot T_0 \cdot l}\left(p_0^2 - p^2\right) \tag{11}$$

This parabolic function deals only with concave part of non-linear characteristics (see Fig. 6) between the start and the critical point K.

To perform experimental verification of the model, we have made a conduit (Fig. 5) of two armoured plates denoted 1 and 2 mutually adjusted with milled out inlet and outlet chambers. The paper sealing defining the conduit dimensions and shape was inserted between the two plates. Several holes for pressure connection were drilled out along the conduit.

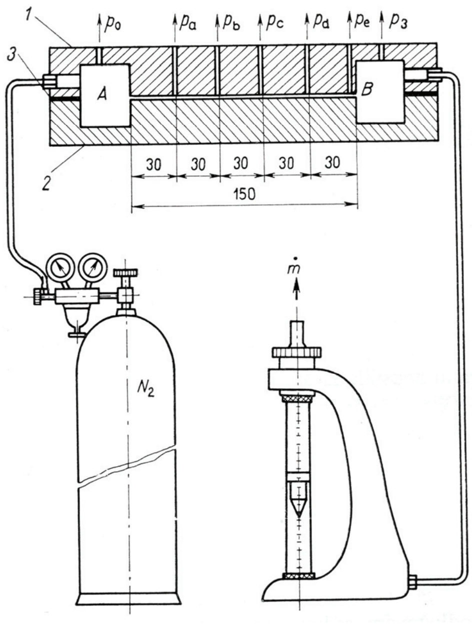

Fig. 5 Experimental stand

The tests were carried out with a gradually rising pressure p_0 controlled by a reduction valve fitted on a nitrogen cylinder. The nitrogen flow was measured by a float-type flow meter. The compliance with the analytical solution according to equation (10) was obvious to critical pressure ratio only. The characteristics of gas flow through the conduit (Fig. 6) represent graphical representation of observed functions $\dot{m} = f(p_0)$. Set of characteristics has two limit curves namely characteristic for an orifice plate ($\bar{x} = 0$) and characteristic for a conduit with infinite length ($\bar{x} \to \infty$) and with zero mass flow rate.

Points K mark the position of critical pressure ratio; points G mark the laminar-to-turbulent flow transition [1].

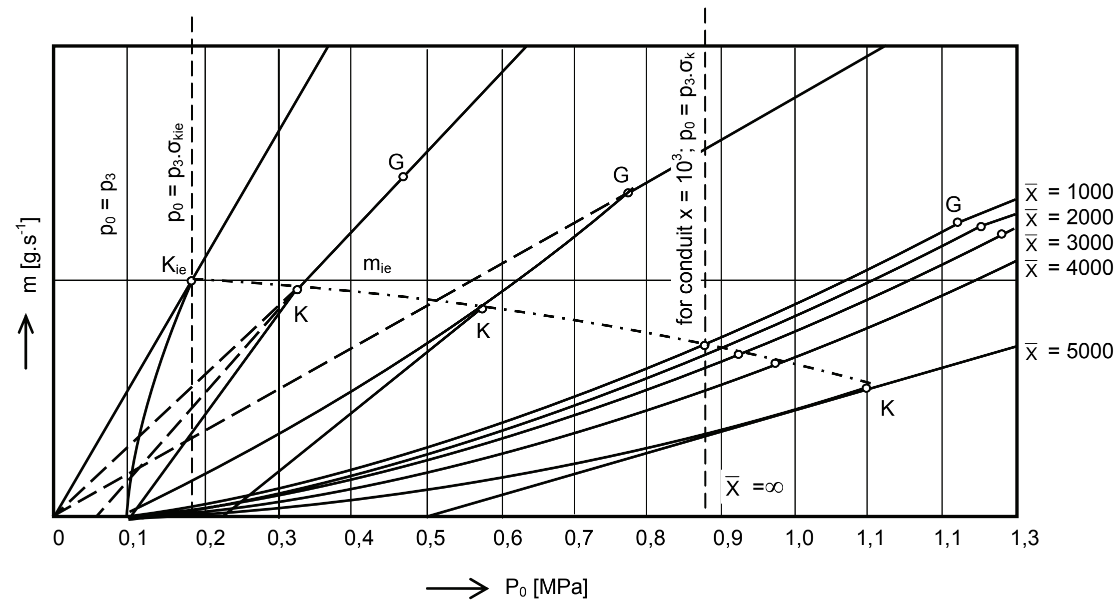

Fig. 6 Characteristics of gas flow through the conduit

4.2 Level of conduit leakage

Providing that flow discharges through the equivalent cross-section with critical speed w_k (coefficient of discharge ψ is maximum) and without contraction $(\alpha = 1)$, it applies:

$$\dot{m} = S_{ek} \cdot p_0 \cdot \psi_{\max} \cdot \sqrt{\left(\frac{2}{r \cdot T_0}\right)} = S \cdot p_0 \cdot \psi' \cdot \sqrt{\left(\frac{2}{r \cdot T_0}\right)} \tag{12}$$

After substituting for area *S (=b.h)* and S_{ek} equivalent cross-section height can be determined.

$$\delta = h \cdot \frac{\psi'}{\psi_{\max}} \tag{13}$$

Equivalent cross-section δ expressing at the same time influence of conduit height h, length l and roughness ξ on mass flow through the conduit can be considered to be the level of conduit leakage. It is evaluated from measurement of compressor working space leakage using the relation:

$$\delta = \frac{\dot{m}}{p_0} \cdot \frac{\sqrt{(r \cdot T_0)}}{\sqrt{2} \cdot b \cdot \psi_{\max}} = \frac{\dot{m}}{p_0} \cdot X \quad \text{, whereas } X = \frac{\sqrt{(r \cdot T_0)}}{\sqrt{2} \cdot b \cdot \psi_{\max}}$$

If we know the level of leakage δ, we can easily calculate without difficult determination of conduit discharge coefficient ψ', the mass flow through the conduit based on the relation valid for orifice plates:

$$\dot{m} = p_0 \cdot b \cdot \delta \cdot \psi_{\max} \cdot \sqrt{\left(\frac{2}{r \cdot T_0}\right)} \tag{14}$$

4.3 Unsteady flow through the conduit

The biggest loss of compressed gas through the working space leakages appears in the moment when the pressure in cylinder p_2 reaches its maximum, which is usually at the end of compression. In this moment, following amount escapes through the working space leakages.

$$\dot{m}_{\max} = p_2 \cdot S_{ek} \cdot \psi_{\max} \cdot \sqrt{\left(\frac{2}{r \cdot T_2}\right)} \tag{15}$$

Resulting working space leakage is specified by mean value of mass flow $\dot{m}_{stř}$ whose value can be determined by direct measurement or more hardly by calculation. The ratio

$$\gamma = \frac{\dot{m}_{stř}}{\dot{m}_{\max}} \tag{16}$$

is denoted as dynamic coefficient. Its evaluation from indicator diagram is described in Fig. 7 (TDC ~ top dead centre, BDC ~ bottom dead centre). The mean value of mass loss $\dot{m}_{stř}$ is demonstrated by the height of shaded rectangle whose area is the same as the area under the curve of function $\dot{m} = f(\tau)$.
After dynamic coefficient determination it is possible to describe the mean value of reflux using the relation:

$$\dot{m}_{stř} = \dot{m}_{\max} \cdot \gamma = p_2 \cdot b \cdot \delta \cdot \psi_{\max} \cdot \gamma \cdot \sqrt{\left(\frac{2}{r \cdot T_2}\right)} \tag{17}$$

The mean values of mass outflow of valve, seals and piston leakages were evaluated through direct measurement, see [1].

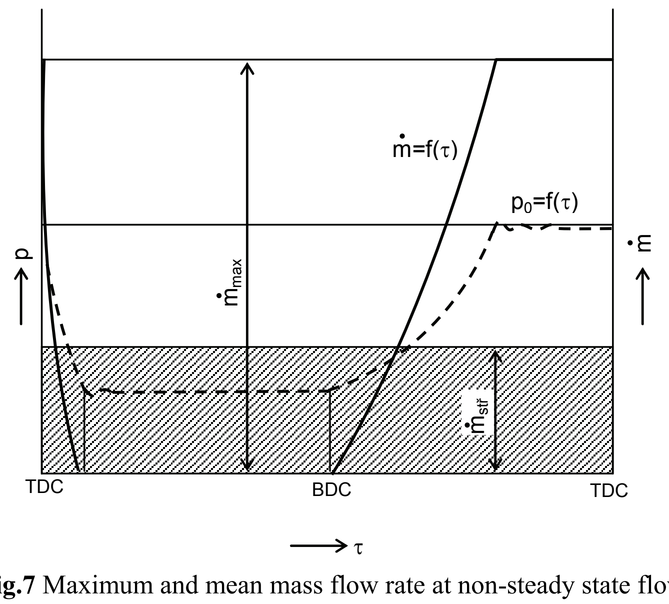

Fig.7 Maximum and mean mass flow rate at non-steady state flow

5 RELATIVE LEAKAGE OF WORKING SPACE

The absolute value of mean mass leakage loss does not provide sufficiently objective information on compressor working space leakage. Only a comparison of the mean value $\dot{m}_{stř}$ and first-stage filling of the machine $\dot{m}_{sI}$ can determine how big part of the air entering the working space circulates or escapes to the environment due to leakage.

$$v = \frac{\dot{m}_{stř}}{\dot{m}_{sI}} \tag{18}$$

Working space filling (see Fig. 3)

$$\dot{m}_{sI} = S_I \cdot L_I \cdot n \cdot \lambda_s \cdot \rho_{sI} \tag{19}$$

where

$\rho_{sI} = \dfrac{p_{s1}}{r \cdot T_{s1}}$ is the intake gas density,

λ_S is the coefficient of working space filling,

S_I is the area of all pistons at the compressor first stage.

Relative leakage is, using the equation (17)

$$v = \frac{r \cdot T_{sI}}{p_{sI}} \frac{p_{2z} \cdot b \cdot \delta \cdot \psi_{\max} \cdot \gamma}{S_I \cdot L \cdot n \cdot \lambda_{sI}} \sqrt{\left(\frac{2}{r \cdot T_{2z}}\right)} \tag{20}$$

Pressure p_{2z} and temperature T_{2z} determinate gas state at the end of compression at the stage whose relative leakage is being calculated.

If the term on the equation (20) right side is modified as a dimensionless factors product, there will be

$$v = \frac{\psi}{\lambda_{sI}} \cdot \sqrt{\left(\frac{2 \cdot T_{sI}}{T_{2z}}\right)} \cdot \frac{\frac{p_{sII}}{p_{sI}} \cdot \sqrt{(r \cdot T_{sI})}}{L \cdot n} \cdot \frac{S_{ek}}{S_I} \cdot \frac{p_{2z}}{p_{sII}} \cdot \gamma \tag{21}$$

Into this relation, pressure p_{sII} was introduced in the second-stage suction and thus also the external pressure ratio of the first stage $\sigma' = p_{sII} / p_{sI}$.

In principle, for all machines the product

$$\frac{\psi}{\lambda_{sI}} \cdot \sqrt{\frac{2 \cdot T_{sI}}{T_{2z}}}$$

has approximately equal numerical value, it is 0,8. If the term

$$K = \frac{\sigma_1 \cdot \sqrt{(r \cdot T_{sI})}}{L \cdot n} \tag{22}$$

is designated as the factor of machine main parameters, as it comprises their influence on leakage, the final result for relative leakage of working space calculation is simplified to

$$v = 0{,}8 \cdot K \cdot \frac{S_{ek}}{S_I} \cdot \frac{p_{2z}}{p_{sII}} \cdot \gamma \tag{23}$$

Factor K, level of conduit leakage at gas reflux and gas pressure ratio p_{2z} / p_{sII} and dynamic coefficient γ , are the critical factors for high-pressure compression that cannot be omitted in optimization of hydrogen compressors design.

5 CONCLUSION

The hydrogen technology economy depends on the efficiency of used compressors. The critical factors of compression point out to the way of connection with hydrogen thermodynamic properties and compressor main parameters.

The numerical value of factor K varies in wide ranges from 100 to 5000. Compressors transferring light gases, small machines, low speed machines and machines with high pressure ratio at first stage have high values of K factor. The most advantageous (i.e. the lowest) values can be found with large high-speed balanced-opposed compressors compressing heavy gases.

For the screw compressors, the efficiency is affected namely by rotor circumferential velocity u (m.s^{-1}) closely associated with internal leakages and thus also with working space utilization. The main dimension factor is here

$$K = \frac{\sigma_I \cdot \sqrt{(r \cdot T_s)}}{u} \tag{24}$$

The circumferential velocity affects also working space overfill due to dynamical action of intake gas, friction, restriction and also gas heating during the suction. Since the speed u_{max} = 150 m.s^{-1} was determined [4] as maximum speed with regard to machine structure and properties of material used, it is difficult (see Fig. 8) to achieve in light gases with high gas constant r (e.g. helium and hydrogen) advantageous values of machine efficiency.

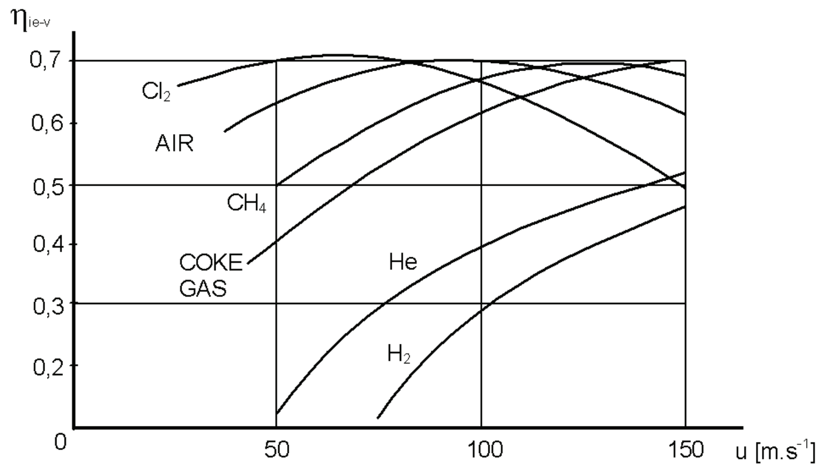

Fig. 8 Dependence of internal isentropic efficiency on rotor circumferential velocity

Primarily, the internal leakages are affected by rotor dimensional size, which determines the height e of gaps between rotors and between rotors and stator. For smaller cross-section rotors the ratio e/D is bigger and the working space utilization is smaller. The same thing appears with higher pressure ratio of stage.

The main parameters factor is of considerable importance at deciding, if it is economic to utilize the given compressor without any modifications for different gas compression. The magnitude of working space relative leakage changes in this case only due to gas constant r change. The leakage increase results only from comparison of these values. If the air compressor (r = 287 J.kg^{-1}.K^{-1}) is used for hydrogen compressing ($r = 4124$ J.kg^{-1}.K^{-1}), they will increase in the rate of $(4124/287)^{1/2}$, i.e. 3,8 times.

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