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Reciprocating Compressor Diagnostics, Detecting Abnormal Conditions from Measured Indicator Cards

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Abstract: Modern compressor analyzers enable plant operators to monitor compressors by acquiring and storing pressure changes measured inside the working cylinder, in suction and discharge chambers, furthermore ultrasonic signals and temperatures. The paper focuses on pressure changes inside the cylinder and the possibilities of detecting abnormal operating conditions, in particular valve leakage, by analyzing deviations of recorded pressure volume cards from ideal diagrams.

1.- Introduction.

The indicator diagram, introduced by James Watt some 200 years ago, is still a valuable tool to assess what is going on inside a working cylinder. However, although modern equipment is available, many errors are still possible and even small errors can seriously affect the outcome of an analysis. This assessment of a compressor is normally done by recording cylinder pressure, pressures in suction and discharge chambers, a number of vibration signals in frequency ranges from 6 to 45 kHz and finally crank shaft rotation over time together with the position of the top dead centre. Furthermore, it is helpful to measure mean values of suction and discharge temperatures in the suction and discharge chambers, as close to the valves as possible. The composition of the gas sample has to be known so that real gas behaviour can be predicted by means of some equation of state. This enables the analyst to establish the diagrams that would be obtained if the same gas were compressed in an ideal compressor. It is hoped that the comparison of the measured, real indicator diagrams with these ideal diagrams provides valuable information as to the state of wear of the compressor.

The present paper limits its scope to the information that can be collected from the different pressure recordings, the conclusions that can be drawn and the type of errors that have to be avoided and how they can be recognized. In this context, it is helpful to distinguish between three kinds of faults or errors:

- Real compressor faults due to wear of sealing elements, only leaking valves will be considered.
- Errors in adjusting the measuring set up.
- Errors in evaluating the recorded data.

In order to produce all kinds of faults under identical conditions, the diagrams presented in this paper were not measured but calculated by the method described in [1].

2.- Real compressor faults due to wear of sealing elements.

Ideally, the compression and expansion events are processes of constant entropy, provided the compression chamber is perfectly gas-tight and heat-insulated. With P as the absolute pressure, V as the volume and m_v as the exponent of the volume isentrope which is assumed to be constant, the corresponding process will follow the law $P.V^{m_v} = c$ or $\log P = \log c - m_v . \log V$. This is a straight line relationship

$Y = C + m_c.X$ with intercept $C = \log c$ and slope $-m_v$. Plotting a measured *P-V*-relationship in logarithmic scales should therefore produce a straight line. Any deviation from the straight line is easy to be detected and can sometimes be seen even with the naked eye. It may be caused by one or more of the following facts:

(i) Leakage of sealing elements like valves, piston rings and packings. The leakage area may be constant as will be the case with a worn or even broken valve plate or valve ring or a missing poppet. Typical diagrams with constant leakage are shown in Figures 1 through 6, "leakage factor 1" signifying "normal valve leakage", just at the limit of the quality acceptance test of a major valve manufacturer, "leakage factor 10" would mean "10 times as much" and so on. The leakage may also vary with pressure differential across the valve. This has to be expected with a valve plate not perfectly plane, or a shaped valve ring not perfectly round or whose mean diameter differs from the one of the seat groove due to thermal expansion or production tolerances. In all cases, as long as the pressure difference pressing the sealing element against its seat is small, the sealing element will not be deformed noticeably and the valve will be leaking. As this pressure difference builds up, the sealing element will deform, match its seat and the valve will become tight. Besides the typical leakage distortion of a *P-V*-diagram or log*P*-log*V*-diagram, noticeable only with heavily leaking valves, a leakage may also be detected by ultrasonic signals [5], or by the rise in discharge temperature and the reduction in compressor capacity both quantified in [1].

(ii) Heat exchange between the gas and the cylinder walls will make that, by definition, the process will not be isentropic. However, this is not a fault but an inherent property of that compressor cylinder. In most cases, the influence of heat exchange on the compression cycle is small and can be neglected.

(iii) When a compressor cylinder operates with an open pocket valve whose passage area is not infinitely large, a pressure drop will be produced by the flow of gas into or out of the pocket. This pressure drop causes an increase in specific entropy of the gas. Hence and by definition, the process inside the cylinder plus pocket will no more be isentropic.

(iv) Real gas properties: Even if a process is perfectly isentropic, the gas compressed is always a real gas whose **exponent *m* of the volume isentrope is not necessarily constant** in the range from P_1, T_1 to $P_2, T_{2,is}$ in which the compressor cylinder operates. Here T_1 is the nominal absolute suction temperature and $T_{2,is}$ the temperature where the isentrope through P_1, T_1 reaches the nominal discharge pressure P_2. For example, in the case of propane, Figures 1, 2 and 7 through 11, the exponent of the volume isentrope m_v varies from 1.397 to 1.427 (corresponding to inclinations of 54.4 and 55.0 degrees) when pressure increases along an isentrope starting from 323.15 *[K]* and 10 *[barabs]* up to 23.5 *[barabs]*. Consequently, even when compression is really isentropic, its logarithmic plot cannot be a perfect straight line.

(v) Standing acoustic waves may develop inside the working chamber with lowest frequency along the longest dimension which usually is the cylinder diameter, i.e. across the cylinder bore which acts as a tube closed at both ends, resulting in a diagram distortion as simulated in Figure 12. The wave or disturbance starts when a gas velocity is suddenly changed, here during closure of the discharge valve. The frequency $f[1/s]$ is given by $f = c/\lambda$ where c is the velocity of sound $[m/s]$. The wave length in a tube closed at both ends $\lambda = 2.(D+e)[m]$ is twice the cylinder bore D plus an end correction $e \approx 0$. Such standing waves were reported in [3] where two pressure sensors were mounted flush with the cylinder liner and opposite to each other. The signal of each of these two sensors was modulated as shown on the expansion line in Figure 12. When both signals were added, the resulting pressure curve was very smooth, signifying that both modulations were out of phase by exactly half a period. Since in addition, the modulating frequency was very close (error ±3%) to the one given by the above formula, there is much evidence that this phenomenon often observed is due to standing waves and hence inoffensive.

(vi) Errors in crank angle and clearance volume will be discussed later.

3.- Errors in adjusting the measuring set up.

(i) Standing acoustic waves can also occur inside the measuring channel, when pressure sensors are not flush mounted with the cylinder wall but communicate with the cylinder through a channel. In this case, the error is two-fold: First there is a modulation due to channel impedance of the incoming signal as to its value, thus rendering further analysis more difficult and less precise [2]. Second there is a certain time lag between a pressure change inside the cylinder and the instant this change is recorded by the sensor. In case the first natural frequency of the channel is far from the dominant harmonic of cylinder pressure variation, this time lag corresponds to the time the pressure wave takes to travel along the channel, it can be found from static considerations and *may simply be treated like a crank angle error*. In the case of Figure 7, this error is of the order of magnitude of about 0.75 degrees for a channel length of 100 mm or 4 inches. At high rotational speeds, the influence of channel impedance increases and may render measured indicator diagrams almost illegible. This influence of channel impedance can be minimized by the use of a *channel resonance correction software* [4] available on some analyzers. Another method [5] eliminates these high frequency modulations by displaying the *Fourier*-spectrum of measured cylinder pressure variation on the analyzer monitor. If harmonics much higher than the first or the second one appear to be dominant, these can be marked and will be skipped during further analysis.

(ii) Real crank angle decoder errors: With analyzers, cylinder volumes are not measured directly, but are calculated from recorded crank angles. These are sensed by a crank angle decoder fixed to the crank shaft, giving, according to type and model used, from 1 to 360 tics per crankshaft revolution, the more, the smaller the impact of fluctuations of angular crank shaft velocity. Its zero point should correspond to piston top or bottom dead centre position. However, total clearances in main journals, crank pin journals and connecting rod journals may add up to 0.015 inch, while, with a typical crank radius of 8 inches, a crank shaft rotation from dead center by 1 degree will move the piston by a theoretical 0.0015 inch only. It is therefore not astonishing that this zero point adjustment is "the field engineers most frustrating task" ([6], section 3, page 26), and crank angle errors are rather common. Maybe proximity probes near the cross head show a way to overcome this problem.

In the case of a crank angle error, the table $P=f(V)$ as recorded in the analyzer will not correspond to the real process in the compressor, but will have its column of crank angles shifted by a constant amount. Diagrams plotted with volumes calculated from shifted crank angle tables are shown in Figures 7, 8, 10 and 11. As a result, at least two consequential errors may occur: First, even if expansion or compression processes are real isentropes with constant exponents, they will appear as curved lines, their curvature may thus be mistaken as a symptom of a leaking valve. Secondly the area of the P-V-card will change thus producing erroneous power estimates: In the case of Figure 7, the *error in planimetering the indicator diagram in order to find power consumption is of the order of 1.6% per degree of crank angle error.* Volumetric efficiencies taken from the indicator card will also be in error.

It is interesting to note that the plot of a perfectly isentropic process in the log$P$-log$V$-diagram with a crank angle decoder error produces a line curved in a typical way, particularly in the vicinity of the inner dead centre, where cylinder volume is smallest and where small changes will be most "exaggerated" by a logarithmic scale: As can be seen from Figure 8, a positive crank angle error produces a curvature towards the inside of the diagram which is difficult to be explained by valve leakage. A negative crank angle error produces a barrel-like curvature towards the outside, thus at first sight similar to what may be caused by leaking valves. In both cases of crank angle errors, maximum curvature occurs near the inner dead centre, while in the case of a valve leakage (suction valve, discharge valve or even both) these curvatures take another form, see Figures 2, 4 and 6. With some experience, it should therefore be possible to differentiate between these different causes of isentrope curvature. Positive crank angle errors move both, expansion and compression lines, towards the inside of the diagram, making it narrower, and vice versa, as can be seen in Figures 7 and 8.

4.- Errors in evaluating the recorded data.

(i) Assumption of a wrong clearance volume: When plotting log(P)-log(V)-charts, knowledge of clearance volume is needed in order to calculate cylinder volume from crank angles. In many cases, actual clearance volume may not be available and has to be estimated. If this estimate is wrong and the clearance volume used in this calculation is different from the one available to the gas during the compression cycle, another type of error occurs. Its impact can be seen from Figure 9. This time and contrarily to what is produced by a crank angle error or a leakage error, the starting point of the expansion line tends to go away from the real inner dead centre point with increasing clearance errors.

5.- Conclusion.

The three types of errors investigated above, valve leakage, crank angle error and clearance error, produce different distortions of the lines of the indicator diagram, especially when plotted in logarithmic scales. Non-parallelity of logarithmic expansion and compression lines is produced by all the three types of errors and is not typical, more typical for each of them is the kind of distortion and the shape of curvature of the deformed straight line. It requires some experience to allocate the observed shape of a distorted indicator diagram, preferrably plotted in logarithmic scales, to a certain type of error. Small valve leakages as they may occur in a normally worn compressor valve after several thousand hours of operation produce only small deviations of the indicator diagram that may be not be noticeable because covered by other errors, even when the gas compressed is as light as hydrogen. It seems to be easier to detect such leakages by observing the variations of ultrasonic signals. Analyzer software either allowing to easily adjust and correct crank angles and clearance volumes, or even carrying out some corrections automatically in order to find the best fit of parallel straight lines to a double logarithmic plot of measured expansion and compression lines would be helpful. A similar analysis is necessary to include the impact of other leakages as in piston rings and packings.

6.- References:

[1] E.H.Machu, *How leakages in valves can influence the volumetric and isentropic efficiencies of reciprocating compressors,* 1990 International Compressor Engineering Conference At Purdue, proceedings.

[2] H. Sockel, F.Ottitsch, Vienna Technical University, *Numerical and experimental investigation of a pressure measuring system with a restrictor,* Journal of Wind Engineering and Industrial Aerodynamics, 42-44 (1992), pages 975-985.

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[4] R.E.Harris, A.J.Smalley, *Computer based diagnostic tools for compressor performance evaluation,* South-West-Research Institute, San Antonio, Texas, and Energy-Sources Technology Conference, New Orleans, Jan 14-18, 1990.

[5] Beta Monitors International Inc., Calgary, Canada, *Reference Manual,* 1995.

[6] 1990 Pipeline and Compressor Research Council Report No. 84-10a, *Field Measurement Guidelines, compressor cylinder performance summary,* May 1984, third revision February 1990

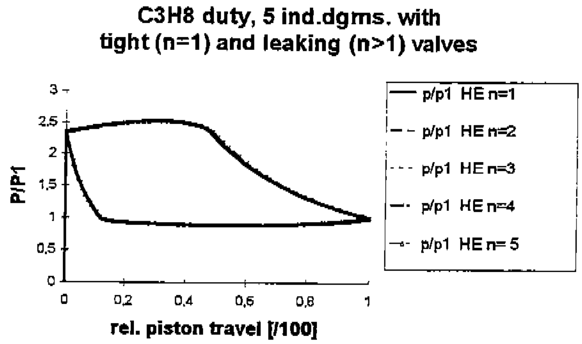

Figure 1: Indicator diagrams with different valve leakages drawn one over the other, with leakages (leakage factors for SV/DV):
n=1: 0/0, n=2: 10/0, n=3: 20/0,
n=4: 0/20, n=5: 20/20 (U3P96_29)

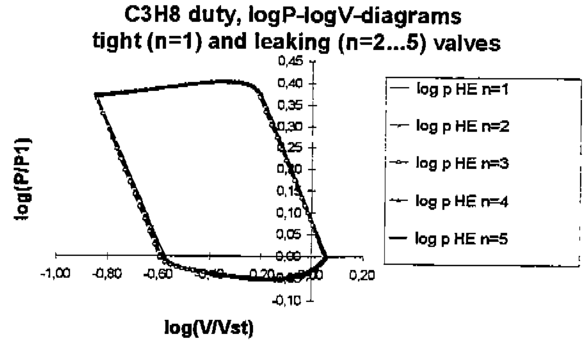

Figure 2: Same parameters as for Figure 1, logarithmic scales.

(U3P96_30)

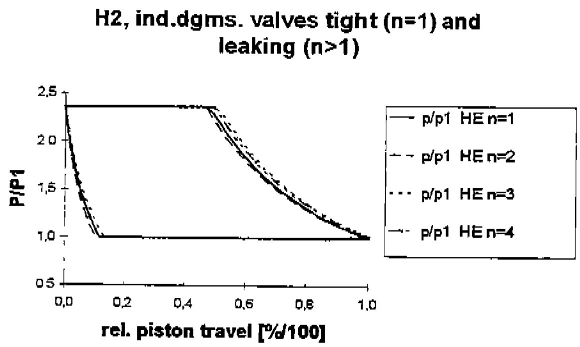

Figure 3: Light gas (H2), 4 indicator diagrams with leakage factors for SV/DV:
n=1: 0/0, n=2: 10/0, n=3: 0/10,
n=4: 10/10 (U1P96_15)

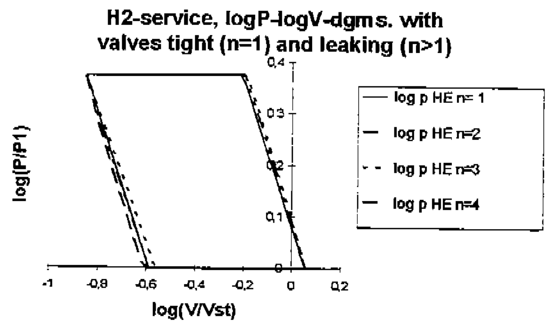

Figure 4: Same parameters as for Figure 3, logarithmic scales

(U1P96_16)

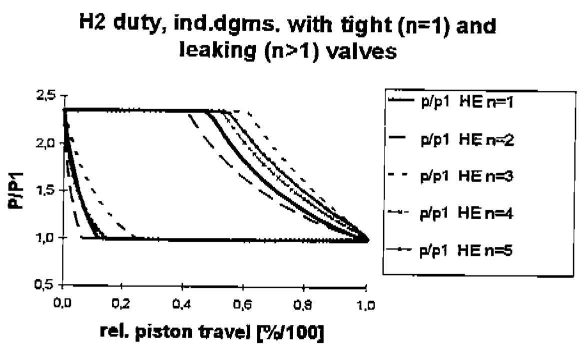

Figure 5: Light gas (H2), 5 indicator diagrams with heavily leaking valves, leakage factors for SV/DV:
n=1: 0/0, n=2: 50/0, n=3: 0/50,
n=4: 30/30, n=5: 50/50 (U5P96_31)

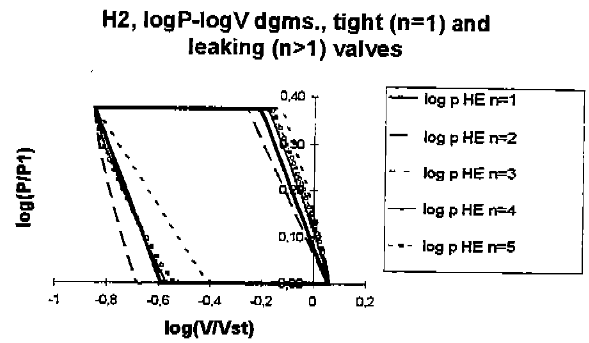

Figure 6: Same parameters as for Figure 6, logarithmic scales.

(U5P96_32)

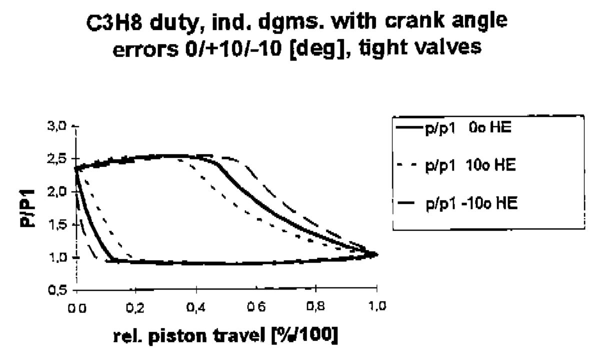

Figure 7: Crank angle decoder errors $\Delta X = 0/+10/-10[\deg]$, tight valves (U3P96_3)

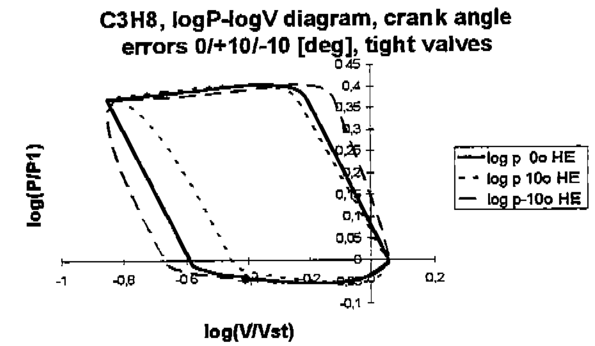

Figure 8: Log-log-diagram, same parameters as in Figure 7 (U3P96_4)

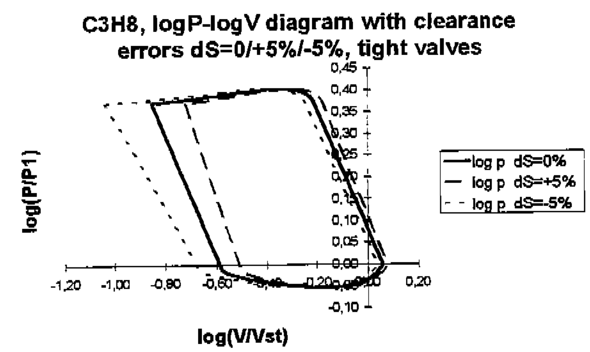

Figure 9: Log-log-diagram with clearance volume errors $\Delta S = 0/+5\%/-5\%$, tight valves (U3P96_5)

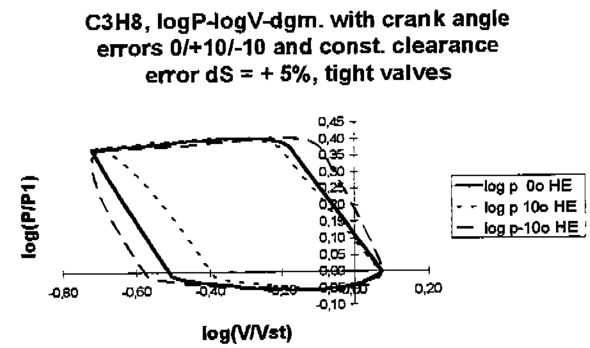

Figure 10: Same diagram as in Figure 8, in addition to crank angle errors there is a constant clearance volume error $\Delta S = +5\%$ (U3P96_6)

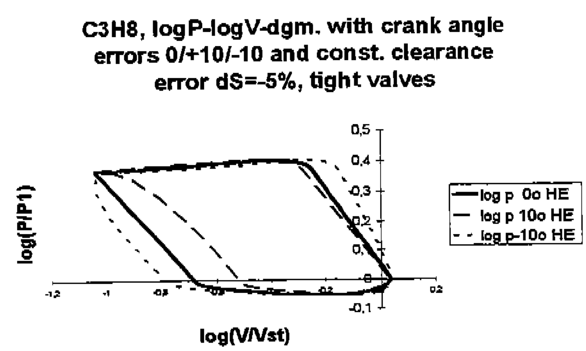

Figure 11: Same Diagram as in Figure 8, in addition to crank angle errors there is a constant clearance volume error $\Delta S = -5\%$ (U3P96_7)

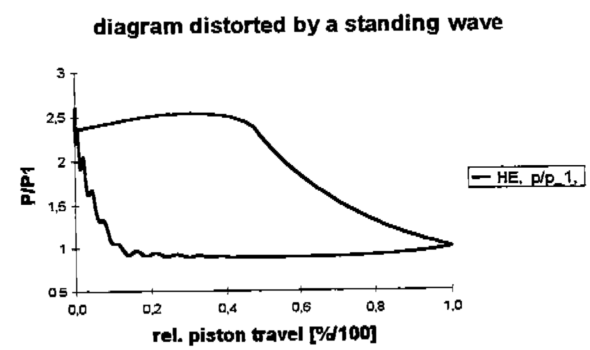

Figure 12: Diagram with a standing wave starting at the end of the discharge event superimposed. (U6P96_1)