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Rotor dynamic response of a centrifugal compressor due to liquid carry over: A case study

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ABSTRACT

This paper presents a case study in which a high speed gas lift centrifugal compressor experienced significant vibration problems associated with start-up and operation. The two-stage 8-impeller unit was used to compressor natural gas associated with oil production. During operation the unit experienced significant levels of vibration that made its continued operation impossible. A rotor-dynamic analysis of the compressor unit confirmed the presence of significant unbalance mass, a direct result of liquid within the process gas ('liquid carry over'). Results of the rotor dynamic analysis was validated with field data. In addition, the paper shows the difference in rotor vibration synchronous response for varying molecular gas weights, particularly at critical speed.

1. Introduction

Oil field development is directly related to the economic and technical challenges required to ensure financial viability. Environmental legislation as well as the extreme working conditions (high oil viscosity reservoirs) requires unconventional technical solutions for current and future turbomachinery employed within the oil and gas industry. The above-mentioned factors, as well as other operational parameters such as compressor operability range, make these machines prone to rotor-dynamic problems. This is compounded by the fact that the compressed gas maybe of an aggressive nature such as Hydrogen Sulphide (H_2S). Recent research [1,2] explored the effect of wet gas on the rotordynamics characteristic of a high-speed centrifugal compressor. 'Wet gas' is defined as the inclusion of liquid in the gas (two phase medium), the ratio of which is dependent on several factors. Ransom et al. [2] concluded that the presence of wet gas had a direct effect on the thermodynamic performance whilst it had no effect on the compressor's lateral vibration.

The cost associated with the unplanned downtime of gas lift/re-injection centrifugal compressors is extremely high and causes of such downtime include high rotor dynamic vibration levels. Forced response and self-induced vibration are the two main sources of rotor vibration within centrifugal compressors. The former source is driven by the residual unbalance and process fouling while the latter is driven by the cross coupling forces generated by the impellers high process pressure and labyrinth seals. Although unbalance is an inevitable source of vibration, it can still be reduced through precise low and high speed balancing. On the other hand fouling is a function of the process gas cleanness, which in oil and gas operations depends heavily on the reservoir formation, and the performance of the knock-out vessels and scrubbers used to clean the processed gas prior to entering the compressor. While destabilising cross coupling forces generated by the impellers

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and seals are a function of rotor speed, differential density and pressure of the gas within the impellers and across the seals span [3,4]. Irrespective of the cause of vibration the implications of down time cannot be underestimated.

Operational specifications of turbomachinery consider both rotor response analysis (forced response) and self-induced vibration (instability) as a vital checkpoint for any rotor operation success. For response analysis the amplitude of rotor displacement as a function of speed is investigated in addition to identification of critical speeds. Traditionally rotor dynamic analysis explores the effect of unbalance masses at varying locations along the rotor in order to assess the sensitivity of the rotor response, providing an insight to how reliable the rotor is in terms of tolerating the inevitable source of unbalance, which in turn ensures maintaining optimal seal clearances for improved efficiency. Similarly, a rotor dynamic instability analysis provides insight into the damped natural frequency as well as instability frequency [5] of the compressor.

This paper presents a case study in which a high-speed gas lift centrifugal compressor experienced significant vibration problems associated with the start and operation of the unit. The two-stage 8-impeller unit was used to compressor natural gas associated with oil production. During operation the unit experienced significant levels of vibration that made its continued operation impossible. A rotor-dynamic analysis of the compressor unit, as well as field data, are presented which confirmed the presence of significant unbalance mass, a direct result of 'liquid carry over' caused by the process gas. In addition, the paper shows the difference in rotor vibration synchronous response for varying molecular gas weights, particularly at critical speed.

2. Case study background

The compressor in this investigation is a two-stage compressor having 4-impellers in each stage arranged in series with an inter-stage cooler. The compressor train consists of a single casing low pressure compressor driven by a 9 MW Electric Motor via an increaser single stage double helical gearbox through flexible couplings (see Fig. 1) The compressor is rated 7.8 MW with a rated maximum continuous operating speed of 6387 RPM. The compressor suction pressure is 1.5 Bar while its discharge pressure has a maximum of 24 Bar. The compressor utilises Honey Comb Pocket Damper Seal (HCPDS) at the balance piston. In addition, the compressor utilises labyrinth seals for its inter-stage sealing system (impeller eye and shaft Seal). A 5-Pad tilting type journal bearing having Load on Pivot (LOP) configuration is used with 50% pivot offset, whilst 11 pad tilting type thrust bearing is used for the rotor thrust.

The compressor was commissioned in January 2012 however the unit continually experienced unacceptable high levels of rotor vibration while traversing its first critical speed. Field data showed that the level of vibration varied from one start-up to another and also depended on the gas type under compression ('Sweet' or 'Sour gas'). Sweet gas has a molecular weight of 18 kg/kmol while sour gas has molecular weight of 42 kg/kmol. Fig. 2 shows a bode plot of the compressor NDE bearing which shows high vibration amplitude (150 μm) at the compressor first critical speed (2926 RPM). The amplitude of 150 μm corresponds to a particular start-up that uses a sour gas. In addition, Fig. 2 shows superimposed the rotor vibration for a shutdown sequence; comparatively it was observed that the rotor vibration amplitude at start-up was three times higher than that observed during shutdown. This difference was noted on numerous occasions during the operation of the compressor train.

To overcome the high vibration amplitude whilst traversing the first critical speed a new start-up sequence was implemented temporary. The new start-up procedure involved the use of sweet gas whenever the compressor had to be started from a cold condition. During this new start-up sequence the compressor rotor vibration was analysed to explore any similarities or differences relative to the previous compressor rotor dynamics behaviour under sour gas conditions. Fig. 3 shows a trend plot of the compressor NDE proximity probes which contains both start-up and shutdown data when 'sweet gas was the working fluid. From Fig. 3 it can be seen that the vibration amplitude at both events were significantly less than that noted under conditions that employed 'sour' gas, for a comparison see Table 1.

3. Compressor failure

After few months of operation that included numerous starts and stops, the compressor tripped (shut down) due to high vibration amplitude reaching up to 320 μm (Fig. 4) and an axial displacement of 0.28 mm just after traversing its first critical speed of 2826 rpm. Also noted at the tripped condition was extremely high thrust bearing temperature (517 °C). Field data

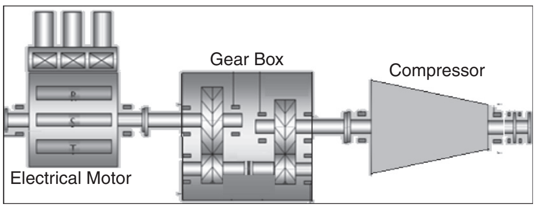

Fig. 1. Gas lift compressor train layout.