SOLVING SYNTHESIS GAS HIGH PRESSURE COMPRESSOR VIBRATION INSTABILITY

By Samy Abou Rayan

Qatar Fertilizer Company (QAFCO)
Mesaieed, State of Qatar

INTRODUCTION

- Qatar fertilizer company operates a huge fertilizer complex that manufactures Ammonia and Urea with a total capacity of 2 & 3 million tons each, being the world's largest single site producer of Urea. The facility has four trains commissioned in stages since 1973.
- The synthesis gas compressor installed in Ammonia 1 & 2 plants consists of two trains ; the low & intermediate pressure casings are driven by two back pressure and condensing steam turbines , the high pressure casing is driven by Gas turbine (fig.1).

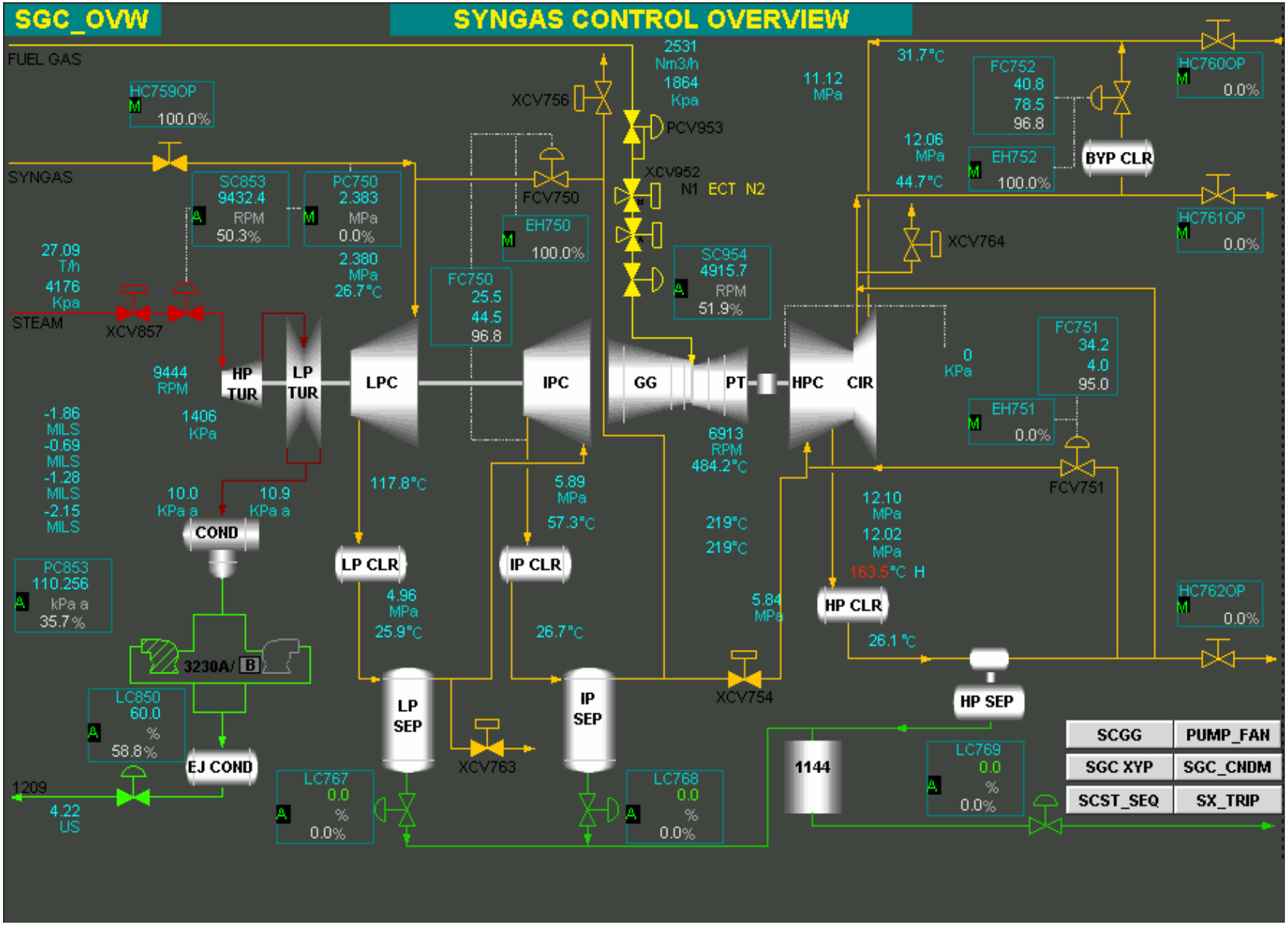

SGC_OVW
SYNGAS CONTROL OVERVIEW
FUEL GAS
HC759OP
M
100.0%
SYNGAS
SC853
9432.4
RPM
50.3%
PC750
2.383
MPa
0.0%
2.380
MPa
26.7°C
27.09
T/h
4176
Kpa
STEAM
XCV857
9444
RPM
HP TUR
LP TUR
1406
KPa
-1.86
MILS
-0.69
MILS
-1.28
MILS
-2.15
MILS
10.0
KPa a
10.9
KPa a
COND
PC853
110.256
kPa a
35.7%
3230A/ B
LC850
60.0
%
58.8%
EJ COND
1209
4.22
US
LC767
0.0
%
0.0%
LP SEP
XCV763
LC768
0.0
%
0.0%
IP SEP
XCV754
1144
LC769
0.0
%
0.0%
HP SEP
HP CLR
26.1 °C
5.84
MPa
12.10
MPa
12.02
MPa
163.5°C H
LP CLR
4.96
MPa
25.9°C
IP CLR
26.7°C
117.8°C
5.89
MPa
57.3°C
219°C
219°C
6913
RPM
484.2°C
LPC
IPC
GG
PT
HPC
CIR
FC750
25.5
44.5
96.8
EH750
M
100.0%
FCV750
XCV756
2531
Nm3/h
1864
Kpa
PCV953
XCV952
N1 ECT N2
SC954
4915.7
RPM
51.9%
11.12
MPa
31.7°C
FC752
40.8
78.5
96.8
EH752
M
100.0%
12.06
MPa
44.7°C
BYP CLR
HC760OP
M
0.0%
HC761OP
M
0.0%
XCV764
FC751
34.2
4.0
95.0
0
KPa
EH751
M
0.0%
FCV751
HC762OP
M
0.0%
SCGG
PUMP_FAN
SGC XYP
SGC_CNDM
SCST_SEQ
SX_TRIP

Fig 1

Introduction (continue)

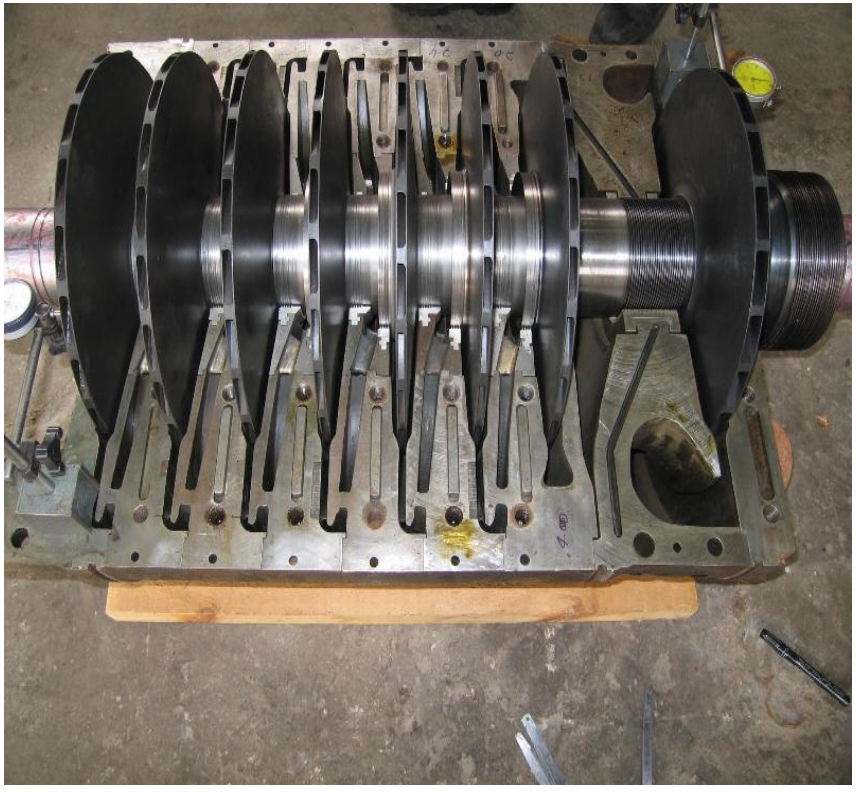
- The compressors are in operation since 35 & 28 years, respectively, both plants suffered heavy production losses due to the weak design of the high pressure compressor .

- The rotors vibration were always high more than 3 mils p/p(75 microns p/p), with frequent rotors replacement due to heavy rubbing at impeller eyes and balance piston honey comb seal.

THE HIGH PRESSURE COMPRESSOR DESIGN

- Compressor was designed on 1970 where Rotor Dynamic instability was not considered.
- The rotor have experienced several sub synchronous vibrations since its first run

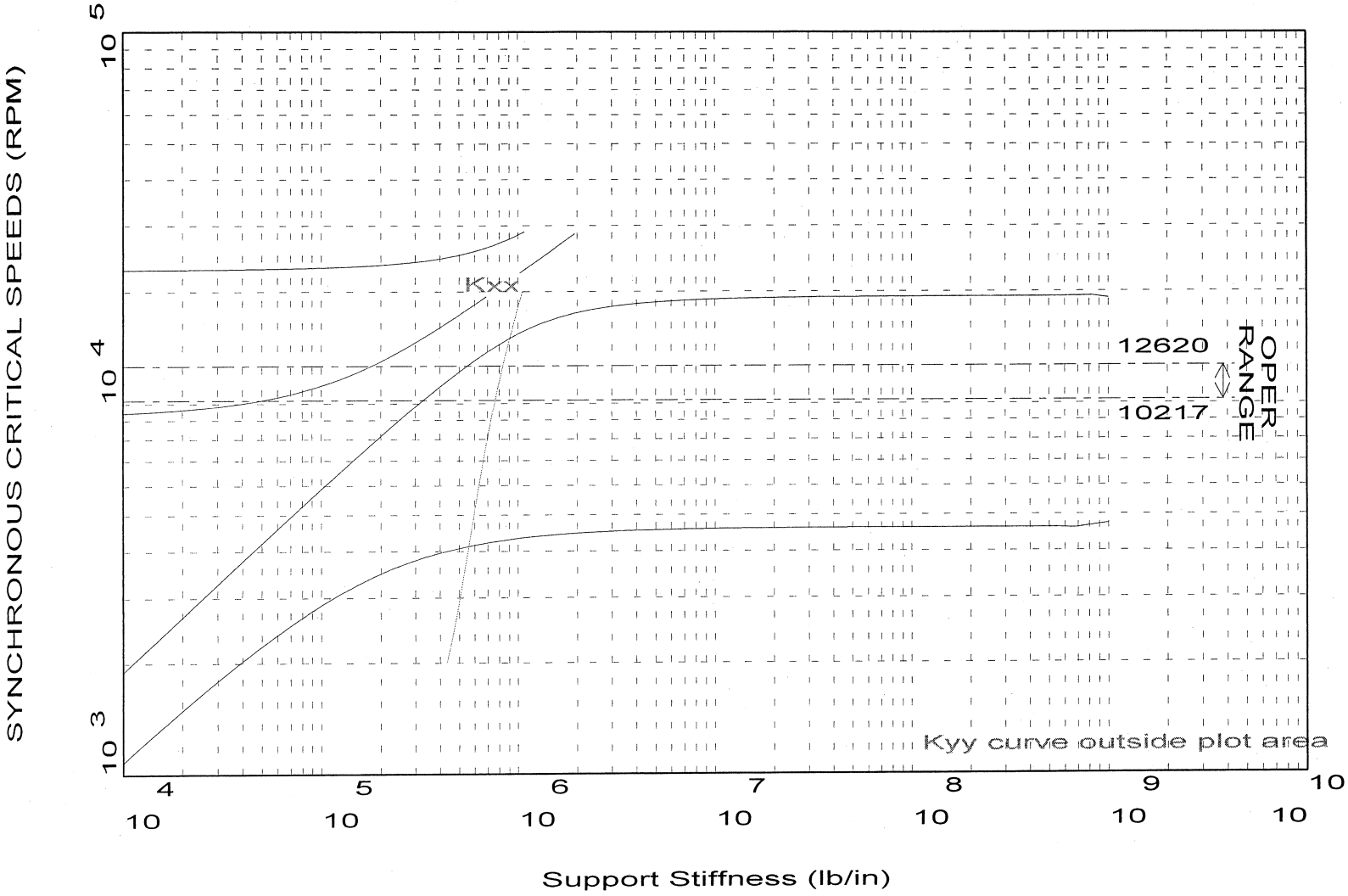
at 12620 rpm MCS with low natural frequency of 4,606 rpm ,with flexibility ratio > 2.74 which is outside Fulton diagram .



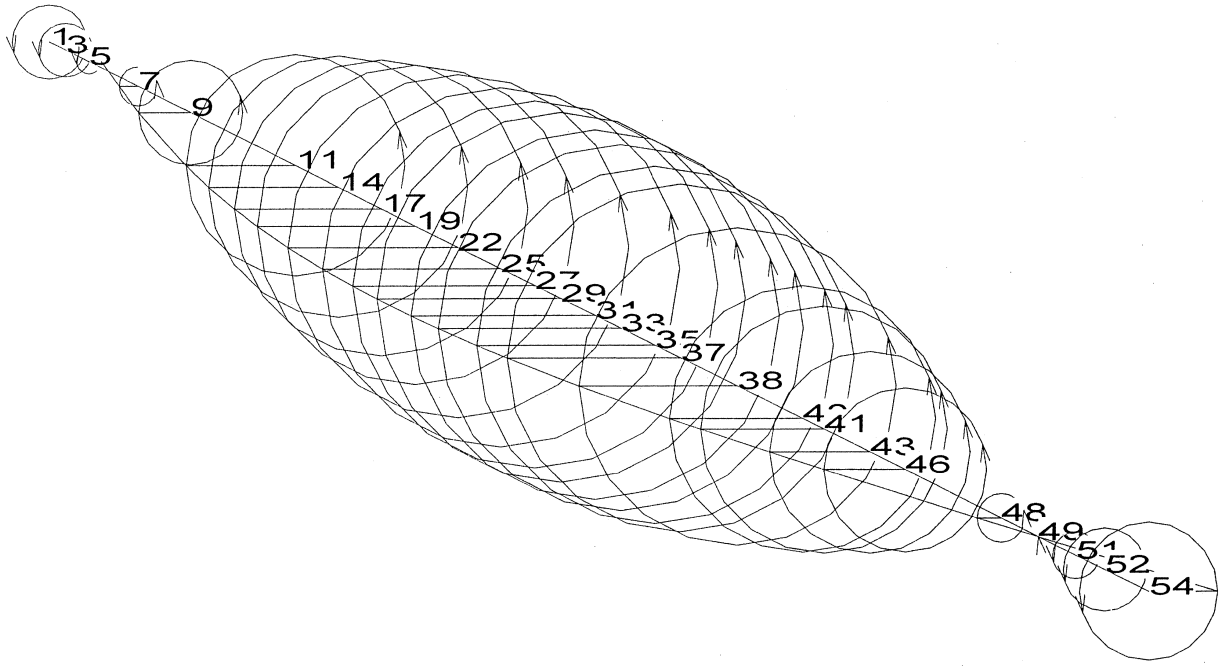
UNDAMPED CRITICAL SPEED PLOT

QAFCO Existing Equipment

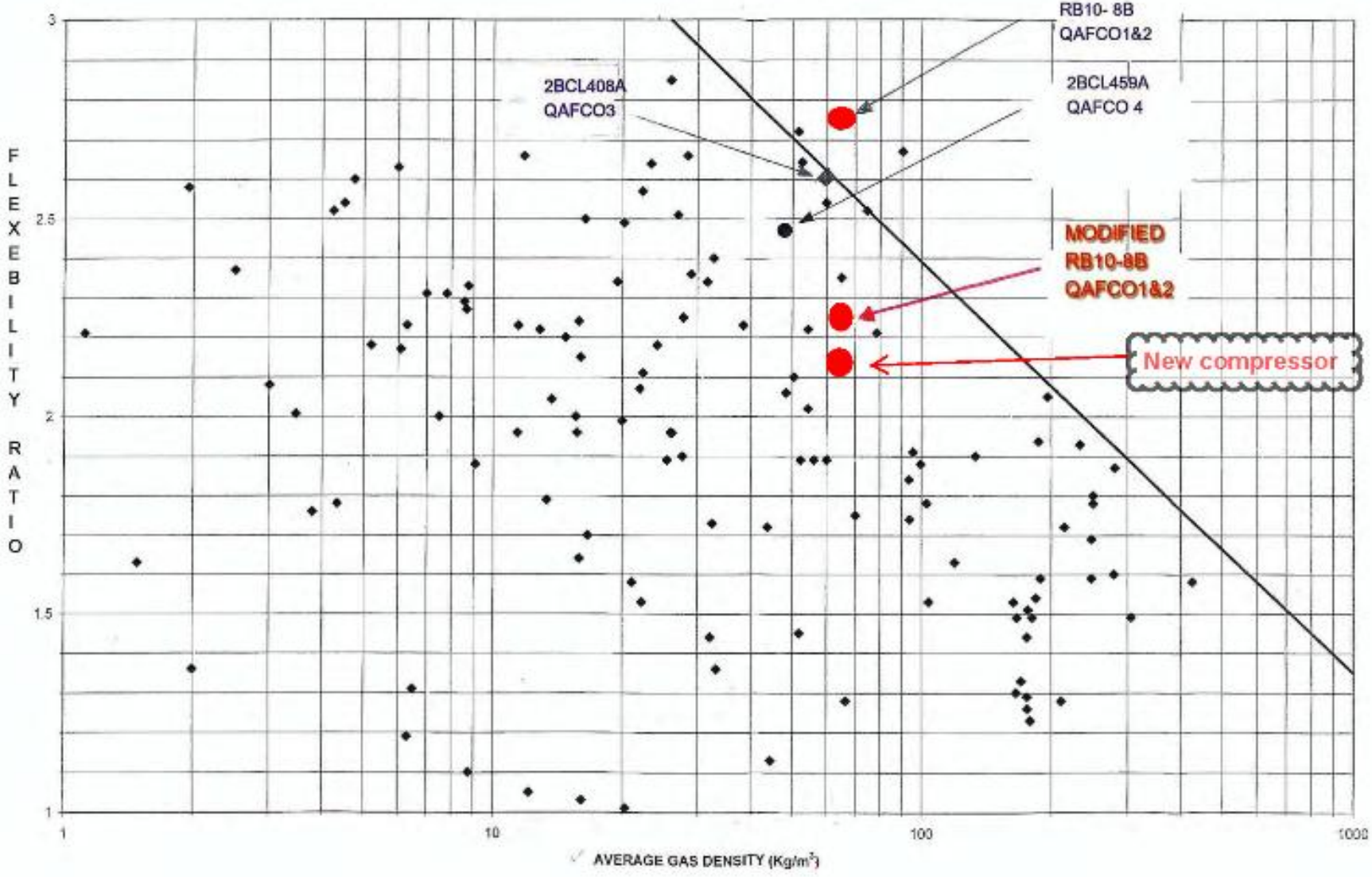
RB10-8B Compressor SN-709/1057



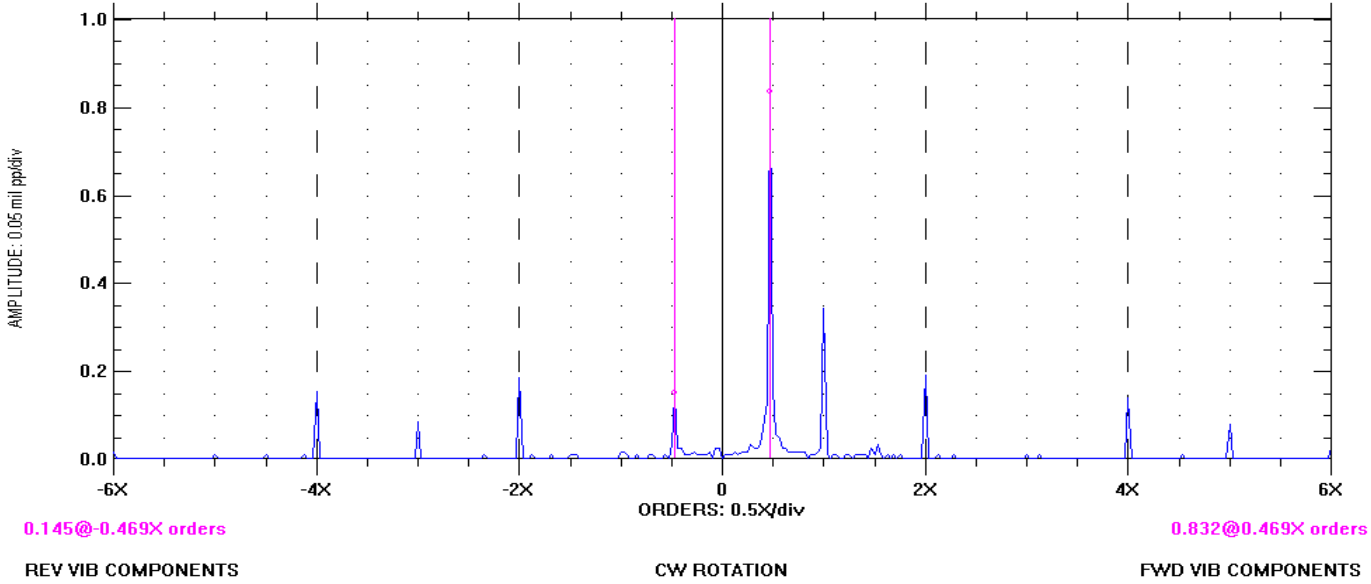
ROTORDYNAMIC MODE SHAPE PLOT
QAFCO Existing Equipment
RB10-8B Compressor SN-709/1057
ANALYSIS POINT: Support Stiffness= 100000000 (lb/in)
CRITICAL SPEED = 4606 rpm
STATION 29 ORBIT FORWARD PRECESSION



FULTON DIAGRAM

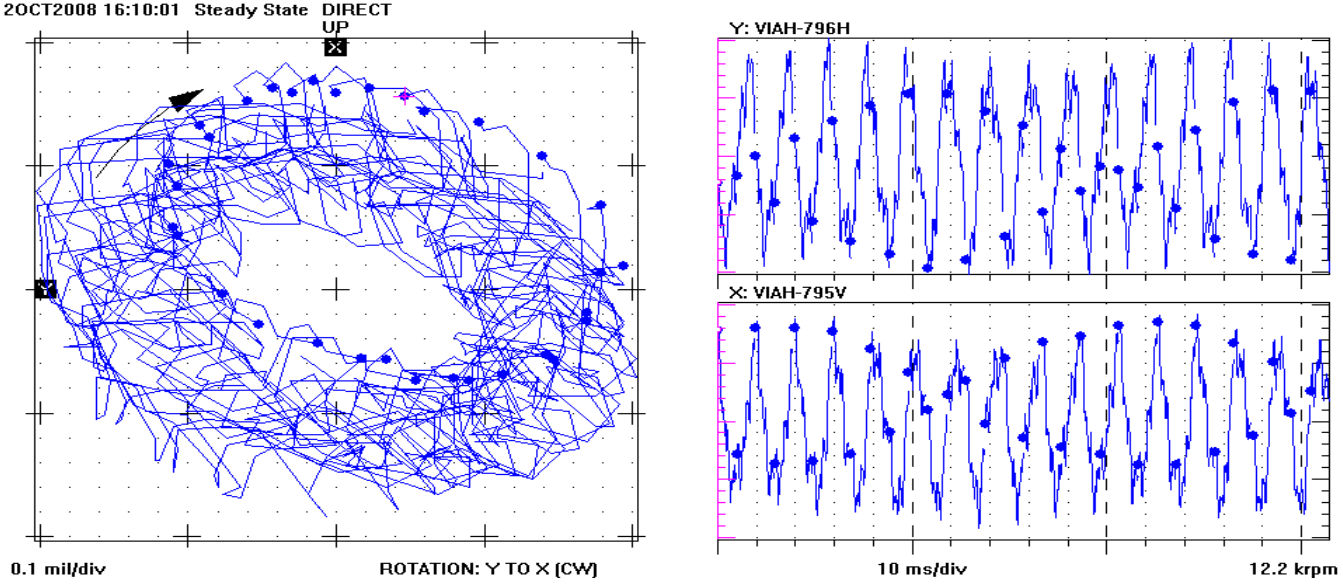


POINT: VIAH-796H ∠90° Left Waveform Pk to Pk: 1.69 mil pp
POINT: VIAH-795V ∠0° Waveform Pk to Pk: 1.53 mil pp
MACHINE SPEED: 12.2 krpm
27 OCT 2008 08:00:06 Steady State
WINDOW: None SPECTRAL LINES: 512



Y: VIAH-796H ∠90° Left Waveform Pk to Pk: 2.04 mil pp
X: VIAH-795V ∠0° Waveform Pk to Pk: 1.85 mil pp

12OCT2008 16:10:01 Steady State DIRECT



High pressure compressor design (continue)

- In order to eliminate the rotor instability ,Qafco requested the OEM to perform a rotor dynamic study.

- The rotor dynamic study results (shown in table 2), indicates that the rotor minimum logarithmic decrement (the natural logarithmic of one amplitude peak divided by the subsequent one which is a measure of how fast the free vibration experienced by the rotor system decay), found to be -2.4 (API required min.+0.1), and Qafco based on experience aims for a minimum log dec of +0.2 to ensure system stability .

Synthesis Gas High pressure compressor specification [Table 2]

Process parameters	
Gas handles	Synthesis Gases
Capacity [make up / recycle] M3/H	1360 / 4162
Maximum continuous speed rpm	12620
Suction pressures [make up/recycle] bara	120 /210
Discharge pressures [makeup/recycle] bara	220 /230
Rotor dynamic parameters	
Minimum forward precession Log Dec	-2.4
First lateral critical speed (rigid) rpm	4606
Rotor stifness ratio (RSR)	0.36
Flexibility ratio (N_{mcs} / $N_{cr\text{-}rigid}$)	2.74
whirl frequency ratio(1st – 8th eye seals)	0.7
whirl frequency ratio (balance piston seal)	0.62
whirl frequency ratio (center seal)	1.13

High pressure Compressor design (continue)

- The main reasons of rotor instability is due to weak design the rotor is light running above its second critical speed, which make it very sensitive to any source of instability like; oil whirl/whip caused by seals lock up, internal friction caused by loose rotor components shrink fits and aerodynamic cross coupling forces , Caused by impeller , shaft labyrinths and balance piston rotating strips.

- The rotor stiffness ratio ($RSR=N_{cr1\text{-}rigid}/N_{mcs}=0.36$) determines whether rotor is stable ,i.e. resistant to sub synchronous vibrations or not.

High pressure Compressor design (continue)

- The other parameter which influence the rotor stability is:

  Whirl frequency ratio = $k/D\Omega$

  k = cross coupling spring constant

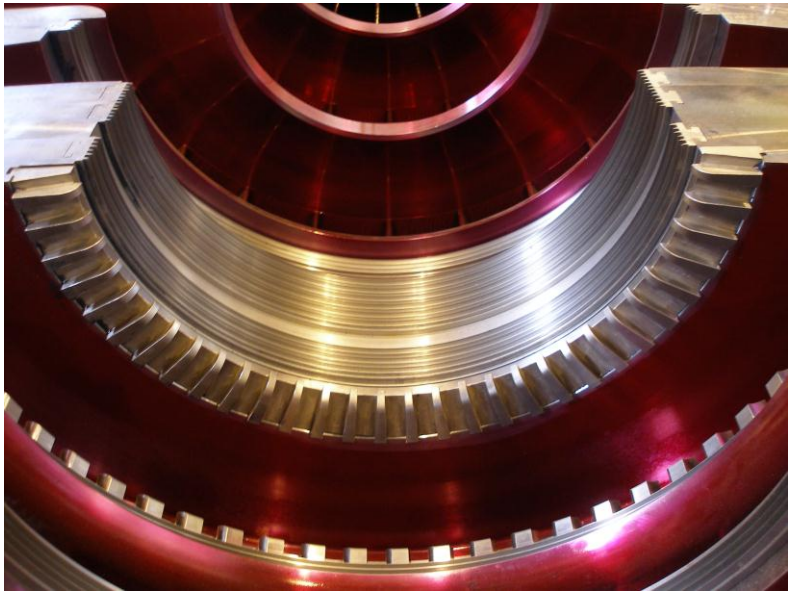
  D = Damping coefficient

  Ω = Rotor angular velocity)

  If the whirl frequency ratio is greater than rotor stiffness ratio , the cross coupling force in the seal will dominate over the damping force, which acts in the opposite direction, and the rotor will destabilize, leading to unstable behavior that increases the vibration amplitude .

Design modifications options

- If shaft stiffening is not possible, the whirl frequency ratio has to be reduced by deswirling the flow i.e. decreasing the flow swirl before it enters the seal. This can be accomplished by swirl brakes ,as shown on the above photo for the new compressor design.



- In order to improve the compressor operation and to avoid heavy production losses , two upgrade options were offered :
- Four modifications to the compressor offered by OEM , where the combined effect will increase the minimum Log Dec from -2.4 to +0.21, leading to a stable rotor during operation. These Modifications are listed in (Table3).

Design modifications options(continue) [Table 3]

step	Description	Minim. Log Dec Improvement
1	Replace impellers eye seal with TAM seal	- 2.12
2	Replace center & BP laby.with hole pattern seal	- 0.44
3	Redesign bearings	- 0.11
4	Increase shaft Dia. at impellers bores	**+0.21**

2. New foot print compressor with dry gas seals from a reputed supplier to replace the existing unstable compressor.

Rotor dynamic comparison between both options is listed in table 4 .

Rotor dynamic design comparison [Table 4]

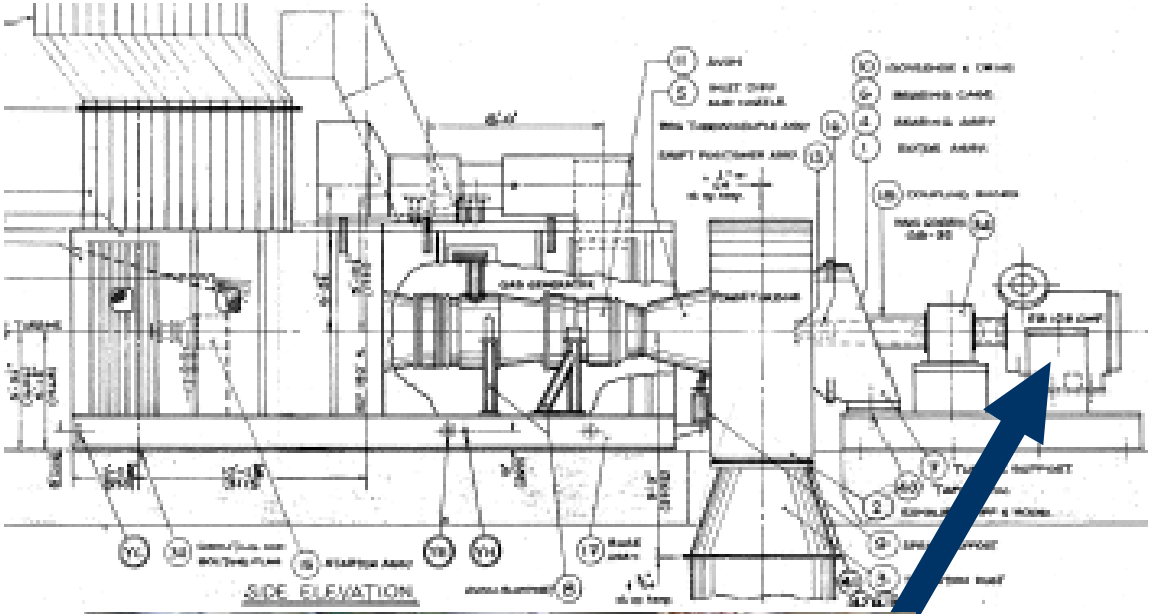
Design parameters	**Modified rotor**	**New compressor**
Actual rated capacity (m3/hr)	1360/3600	1512 /4162
Rotor mass (KG)	290	674
Shaft length & bearing dia. (mm)	1630/ 100	2134/125
Bearing span (mm)	1384	1676
Impeller bore (mm)	127	210
Minimum forward Log Dec	+0.21	+0.283
First lateral critical rigid (rpm)	5661	6012
Rotor stiffness ratio (RSR)	0.4485	0.4995
Flexibilty ratio	2.23	2.10

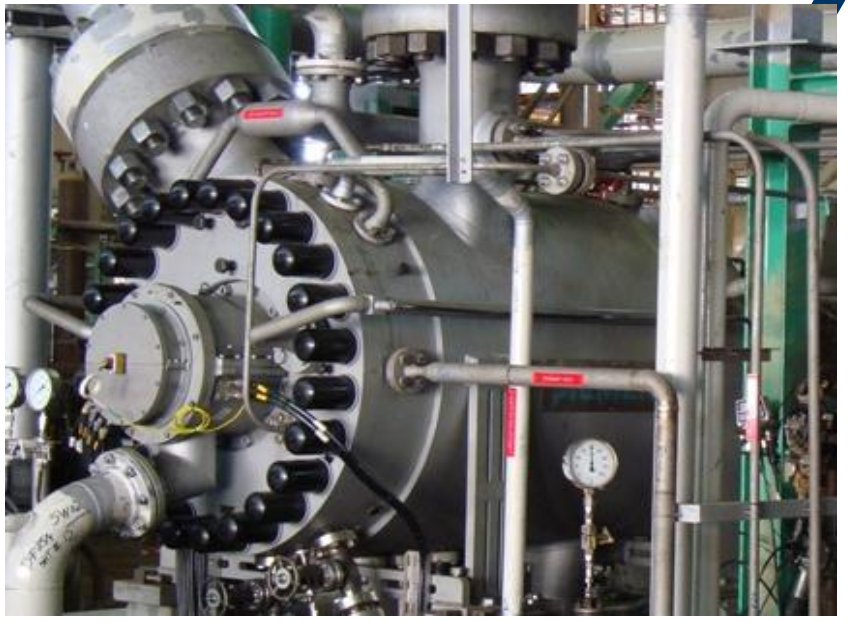
COMPRESSOR REPLACEMENT

- The foot print compressor option 2 was selected due to the following advantages:

1. Higher in capacity by 10% for make up stage and 15% for circulator stage .
2. Rotor dynamic design is better ,with higher Logarithmic decrement and higher 1st lateral critical speed i.e. lower rotor flexibility ratio and higher stiffness ratio.
3. Compressor is offered with dry gas seals ,which eliminates the wet seal and its auxiliaries .
4. The Internal rate of return (IRR), is higher with shorter payback of 13.83 months compared to option 1 of 19.6 months.

NEW FOOT PRINT COMPRESSOR


SIDE ELEVATION



NEW REPLACED COMPRESSOR OPERATION

- Compressor Started November 2007 in A2 plant and December 2008 in A1 plant, with low vibration levels and excellent performance .
- Performance comparison with the replaced compressor shows an increased capacity of 10 % approximately.
- No production losses or unforeseen stops due to the new Synthesis Gas HP compressor since start-up at both A1 & A2 plants .

THANK YOU