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一般社団法人日本機械学会
状態監視振動診断技術者コミュニティ　第10回ミーティング

プロセス用遠心圧縮機で経験した振動事例紹介

株式会社 日立製作所
インダストリアルプロダクツビジネスユニット
機械システム事業部 気体機システム部
髙橋　直彦

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1．VSD駆動機で発生した曲げねじり振動

出典:
Torsional-Lateral Coupled Vibration of Centrifugal Compressor System at Interharmonic Frequencies Related to Control Loop Frequencies in Voltage Source Inverter, 38th Turbomachinery Symposium, Case Study 14, 2009

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1-1　対象機械

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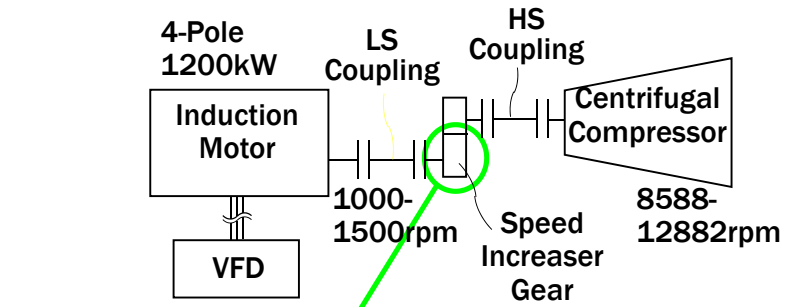
Train Data

VFD Induction Motor + Gear + Centrifugal Compressor



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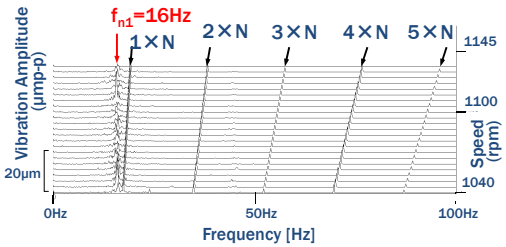
Lateral Vibration on Low Speed Gear Shaft



Cascade Plot of LS Gear Shaft Vibration

Only max. 10μm (p-p) vibration with 16Hz was detected on HS gear shaft.

50μm (p-p) vibration was detected on LS gear shaft around 1040-1140rpm (17.3-19.0Hz), 570kW.

Dominant Frequency Component: ca. 16Hz

Close to calculated 1st torsional natural freq. f_{n1} (15.71Hz)

No significant vibration on compressor shaft or motor shaft was found.

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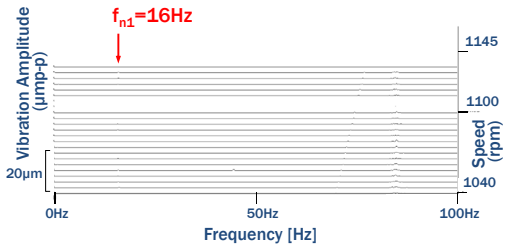
Lateral Vibration on High Speed Gear (Pinion) Shaft



Cascade Plot of HS Gear Shaft Vibration

Concern: What is the Excitation Source?

Max. 10μm (p-p) Vibration with 16Hz Detected on HS Gear Shaft

Lower Vibration on HS Gear Shaft

Stiffer Pinion Bearings Than Bull Gear Bearings at Part-Load Due to Uploading Bull Gear and Downloading Pinion

+

Pinion Lighter Than Bull Gear

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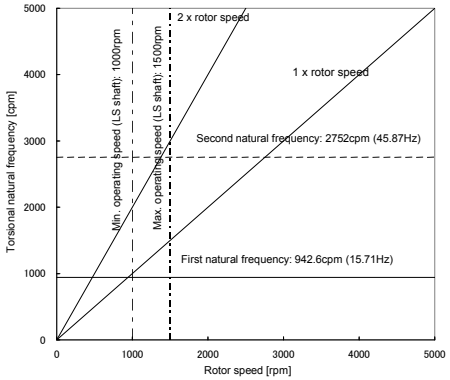
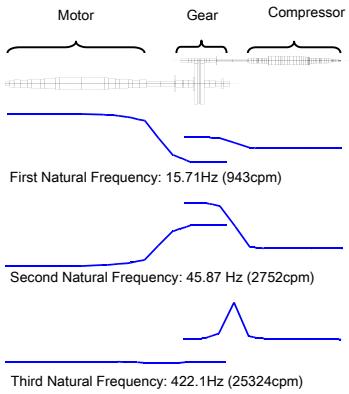
Analysis for Torsional Natural Frequencies (FEM)



Campbell Diagram for Torsional Vibration



Torsional Vibration Mode Shapes

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Lateral & Torsional Vibration Calculation

Calculated Critical Speeds [rpm]

		First	Second	Operating Speed
Lateral	Compressor	5220	17700	8588-12882
	Pinion (HS Shaft)	21200-25600 Dependent on Load	-	8588-12882
	Bull Gear (LS Shaft)	3380-9450 Dependent on Load	-	1000-1500
	Motor	2078	-	1000-1500
Torsional		943(=15.71Hz)	2752	

Shaft synchronous resonance is excluded. Only Possibility = VFD?

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1-5 原因推定

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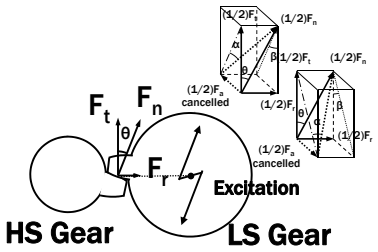
Estimation of Shaft Torque Causing 50μm Shaft Vibration

Shaft torque is deduced by one-way lateral-torsional analysis.

Torque vibration excites shaft vibration via force on teeth.



Force Applied on Gear Teeth (Double Helical)

Transverse Pressure Angle θ

$\theta = \tan^{-1}(\tan\alpha/\cos\beta)$

α= Normal Pressure Angle

β= Helix Angle

$T_t = (PCD/2) \times F_t$

$= (PCD/2) \times F_n \times \cos\theta$

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1-6 軸トルク，モータトルクの推定

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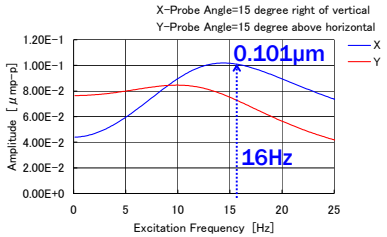
Lateral Vibration Analysis of LS Gear Shaft



Frequency Response to Unit Excitation Force of Gear Shaft

(Bearing Stiffness and Damping Calculated @ 1140rpm, 570kW)

Unit Excitation on Tooth Surface

$F_{n_p.u.} = 9.8 \cdot \sin(2\pi f_{n1} t)$ [N]

LS Gear Shaft Vibration Amplitude at Probe @ 16Hz

=0.101μm (p-p) (Calculated)

$F_n = 9.8 \times (50/0.101) = 4851$ [N]

$T_t = (PCD/2) \times F_n \times \cos\theta$

$= (0.806/2) \times 4851 \times \cos 21.9^\circ$

= 1815 [N・m]

→ 21.3% Rated Torque

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1-7 軸トルク，モータトルクの推定

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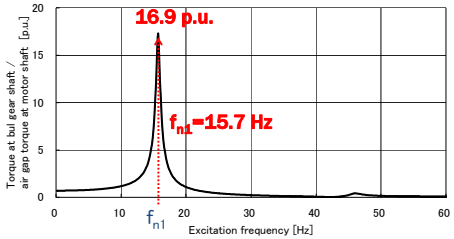
Estimation of Excitation Torque at Motor Air Gap

Amplification Factor for Torsional System



Frequency Response at LS Gear Shaft Assuming ζ=0.02

T_{AG} = 1815 / 16.9

= 107 [N・m]

1.3%rated torque at motor air gap is suspected.

Close to maximum measured data among interharmonic frequencies during motor factory test (1.5%rated torque)

Merely 1.3% of air gap torque fluctuation can cause 50 μm p-p lateral vibration!

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1-8 原因と結果

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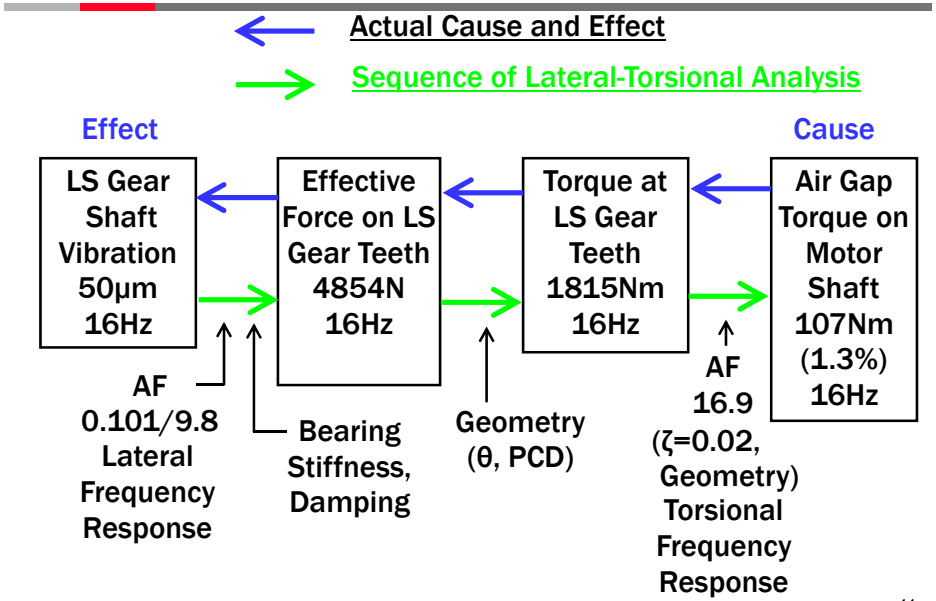

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1-9 強度評価

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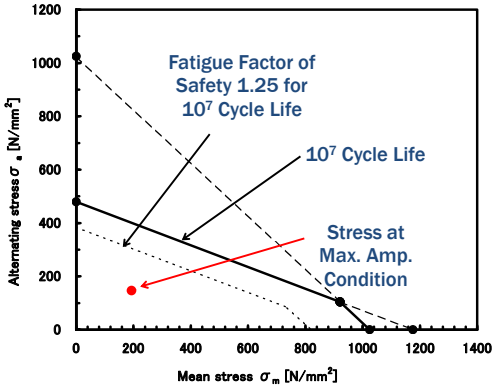
High Cycle Fatigue Evaluation



Mechanical strength was verified.

No modification was made on machinery or VFD.

Modified Goodman Diagram (Low Speed Coupling)

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1-10 加振トルク源の分析

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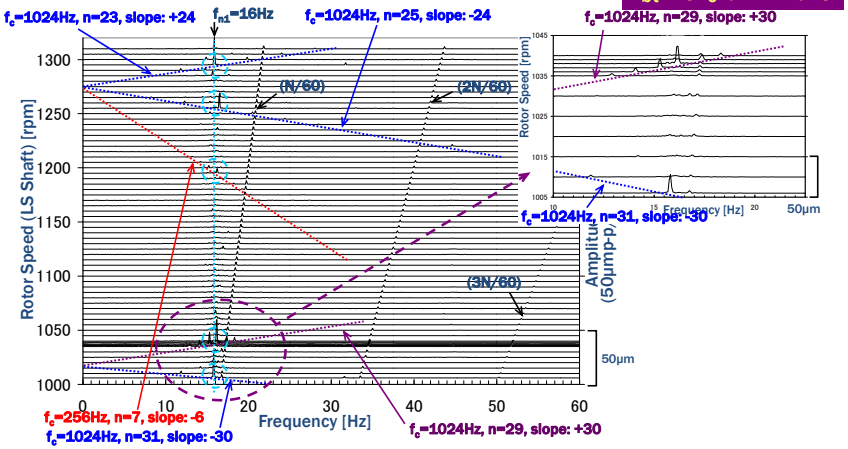
Detailed Measurement (LS Gear Shaft Vibration)

$f_{bt} = |f_c-(n\pm1)f|$



Cascade Plot of LS Gear Shaft Vibration

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1-10 加振トルク源の分析

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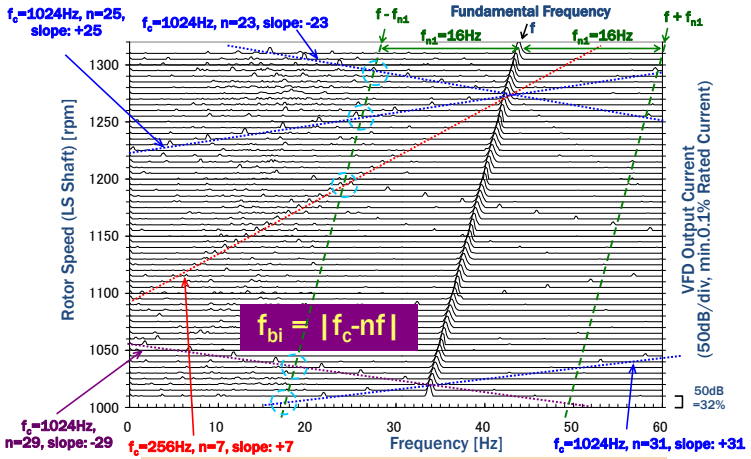
Detailed Measurement (VFD Output Current)



Cascade Plot of VFD Output Current

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1-11 Interharmonicsの周波数成分パターン

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Inclined Streaks of Interharmonic Frequencies in Shaft Vibration Frequencies and VFD Output Frequencies

- Inclined streaks in LS shaft vibration frequency have:
  Difference of Harmonics of Multiples of 6 and Sampling Frequencies of VFD (1024Hz, 256Hz)
- Inclined streaks in VFD Output Current Frequency have:
  Difference of Harmonics of Odd Numbers Other Than Multiples of 3 and Sampling Frequencies of VFD
  Inclination is opposite to that of shaft vibration.

⬇

Firm correlation between shaft vibration and VFD output current is suspected.

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1-11 Interharmonicsの周波数成分パターン

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Relation of Between Shaft Vibration Frequencies and VFD Output Frequencies

VFD Output Freq. [Hz]:
$f_{bi} = |f_c - nf|$ ← Sideband Frequencies

Shaft Vibration Freq. [Hz]:
$f_{bt} = |f_c - (n \pm 1)f|$

f：VFD Fundamental Freq. [Hz]
f_c：Arbitrarily Existing Constant Freq. [Hz]
n：Positive Odd Integer Other Than 3

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1-12 VFDにおけるサンプリング

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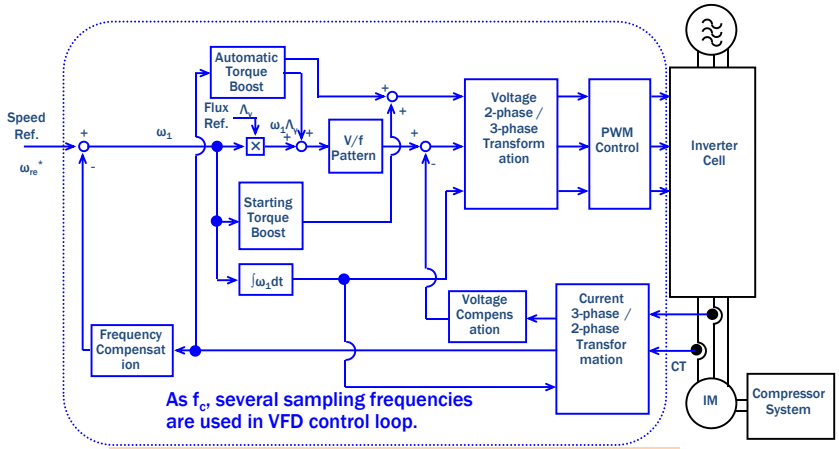
VFD Control Loop



Block Diagram of VFD Control System

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1-13 側帯波の原因推定

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Assumed Cause of Sideband in VFD Output Current

Harmonics were contained in a feedback control signal.

↓

More than half the sampling frequency folded back into lower frequencies.

↓

Sideband frequencies were raised by modulation between harmonic frequencies and sampling frequencies occurred in VFD control loop.

→ Harmonics enhance sidebands.

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1-15 まとめ

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- Measurement of torque & current in factory test is important in case VFD characteristics are unknown.
- Strength evaluation by lateral-torsional analysis is essential to determine mechanical soundness.
- Information of any possible frequencies used in VFD control and the resulting amplitude of torque pulsation should be disclosed in advance by VFD vendor.
- Reduction of amplitude of fundamental frequency harmonics would decrease amplitude of sideband frequencies.

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2. VSD駆動機で発生したカップリング破損

出典:
VSD駆動モータに起因したトラブル事例と設計上の留意点, ターボ機械, 45(1), 2017, pp.23-28

Importance of Control Engineering to Minimize Torsional Vibration in Variable Speed Drive Systems, PCIC Europe (BER-66), 2016

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2-1 概要

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対象機械	遠心圧縮機 （天然ｶﾞｽ昇圧用）	
対象機械仕様	回転速度[min^{-1}]	ﾓｰﾀ出力[kW]
	低速軸: 約1,000～1,500	約 6,000 誘導電動機＋ VSD(可変速)装置
	高速軸: 約7,150～10,725	
固有振動数	ｷﾞﾔ低速軸ﾗﾃﾗﾙ1次固有振動数:1500min^{-1}以上 ｷﾞﾔ高速軸ﾗﾃﾗﾙ1次固有振動数:11000min^{-1}以上 ﾄﾚｲﾝねじり固有振動数:1次1077min^{-1}(17.95Hz), 2次3391min^{-1}(56.52Hz)	
発生した現象	圧縮機試運転開始し(運転回転速度範囲1000～1500min^{-1})約30時間後にギヤ低速軸の軸振動が200μm(p-p)となり，運転を停止した。その後の調査で低速軸ｶｯﾌﾟﾘﾝｸﾞｽﾍﾟｰｻ亀裂を発見した(1回目)。ｶｯﾌﾟﾘﾝｸﾞｽﾍﾟｰｻ交換後，運転回転速度範囲を1215～1500min^{-1}に変更して運転開始約30時間後，低速軸の軸振動が増大し再度運転を停止した。調査後，低速軸ｶｯﾌﾟﾘﾝｸﾞｽﾍﾟｰｻ亀裂を発見した(2回目)。	

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2-2 対象機器

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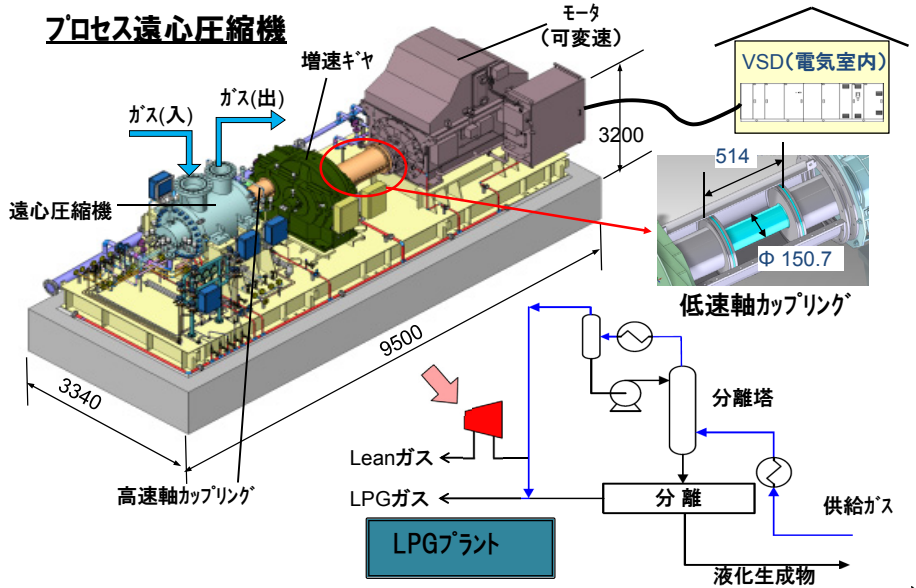
プロセス遠心圧縮機



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2-2 対象機器

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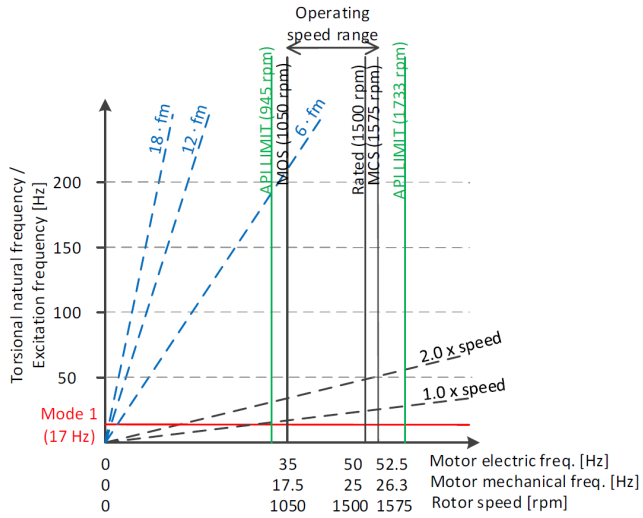
ねじり固有振動数計算予測と運転回転速度範囲，加振周波数



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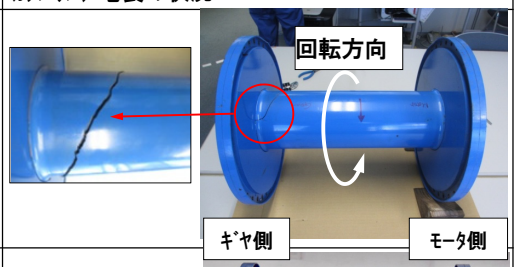
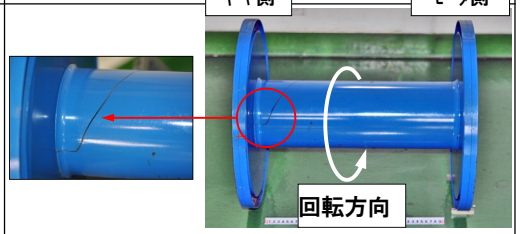
カップリングの破損

	現象	ｶｯﾌﾟﾘﾝｸﾞ亀裂の状況
1回目	ｶｯﾌﾟﾘﾝｸﾞｽﾍﾟｰｻに亀裂が発生し、ｷﾞﾔ・ﾓｰﾀの軸振動が過大となり、圧縮機を手動停止。	
2回目	ｶｯﾌﾟﾘﾝｸﾞを交換して運転再開。 1回目と同じ位置に亀裂が発生し、ｷﾞﾔ・ﾓｰﾀの軸振動が過大となり、圧縮機が自動停止。	

亀裂の特徴：駆動機回転方向と逆に45° の亀裂

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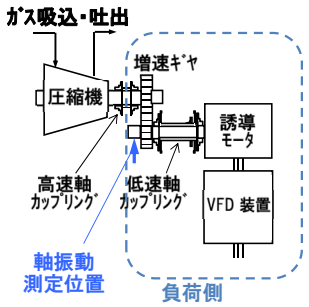
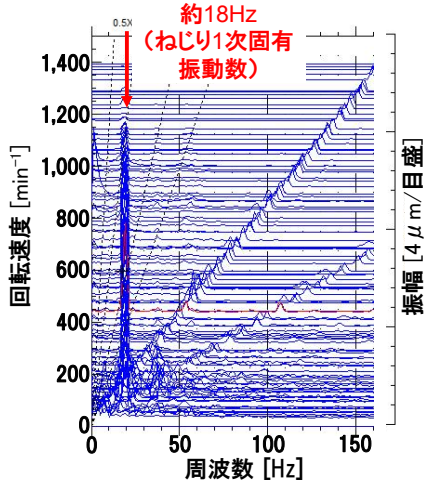
増速ギヤ軸振動の増大





ｷﾞﾔ低速軸軸振動振幅の周波数分析

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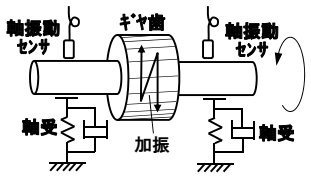
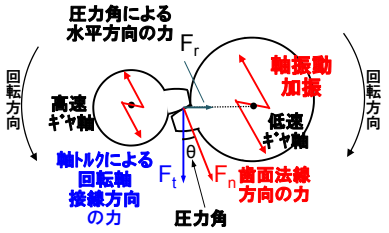
ギヤ軸のラテラルねじり連成振動の原理



歯面ﾄﾙｸ伝達力による軸ﾗﾃﾗﾙ強制振動



ｷﾞﾔ歯面に働く力

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VFD調整不足、異常についての検討

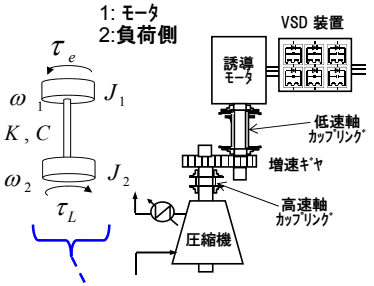
$$\frac{\omega_1}{\tau_e} = \frac{1}{(J_1+J_2)s} + \frac{J_2}{J_1(J_1+J_2)\left(\omega_n^2+2\zeta\omega_n s+s^2\right)}$$

剛体回転

$$= \frac{1}{(J_1+J_2)s} \cdot \frac{1+2\zeta s/\omega_n + s^2/\omega_0^2}{\left(1+2\zeta s/\omega_n + s^2/\omega_n^2\right)}$$

1次ねじり振動

添字
1: モータ
2:負荷側



共振時の応答倍率(AF)

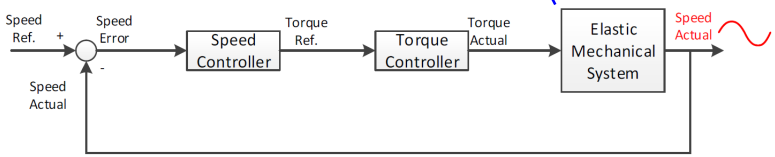
$$AF = \frac{J_2}{J_1} \cdot \frac{1}{2\zeta}$$

ねじり固有ﾓｰﾄﾞ減衰比

速度制御のPIﾊﾟﾗﾒｰﾀ

$$K_P\left(1+\frac{1}{T_I s}\right)$$

$$= K_a \cdot \frac{C}{(J_1+J_2)}\left(1+\frac{1}{T_I s}\right)$$



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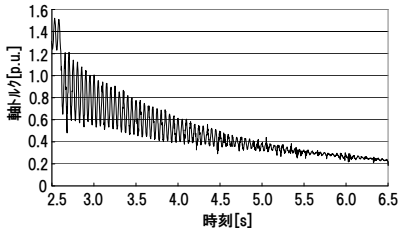
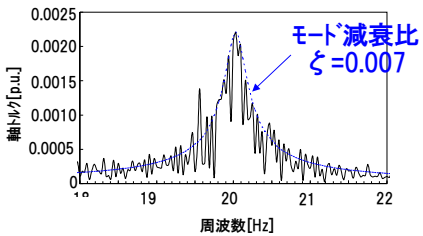
圧縮機装置の振り振動の機械的減衰の測定例



コーストダウン時の測定軸トルクの
減衰時刻歴波形例



コーストダウン時の測定軸トルクの
周波数分析例

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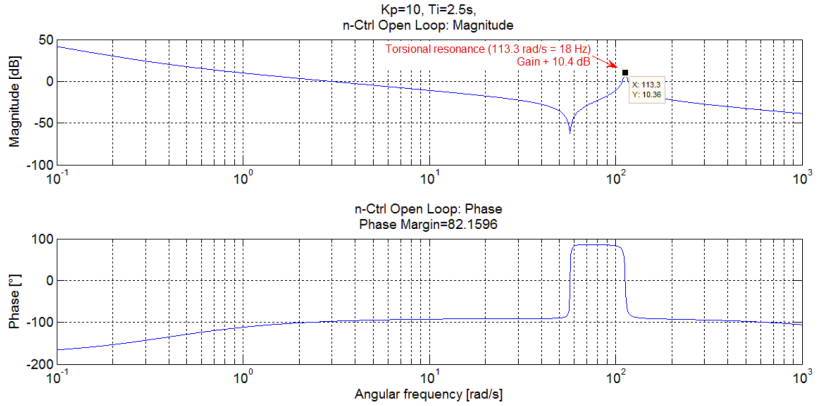
VSDの速度制御パラメータ調整

速度制御のPIパラメータ： High controller gain (対策前)



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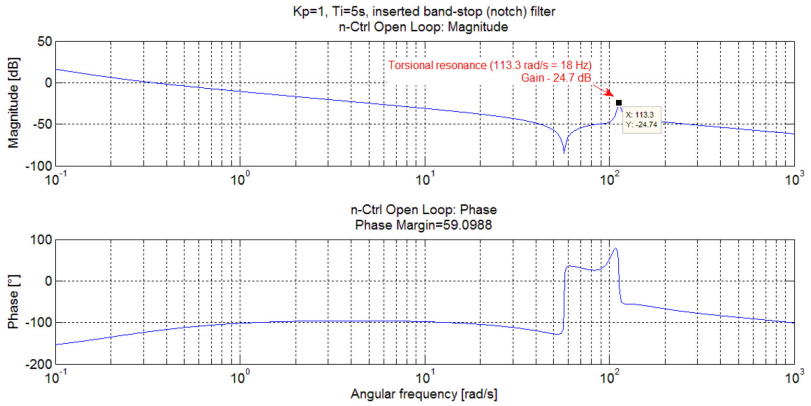
Inspire the Next

VSDの速度制御パラメータ調整

速度制御のPIパラメータ： Low controller gain with notch filter (対策後)



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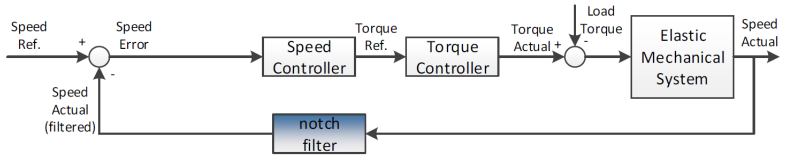
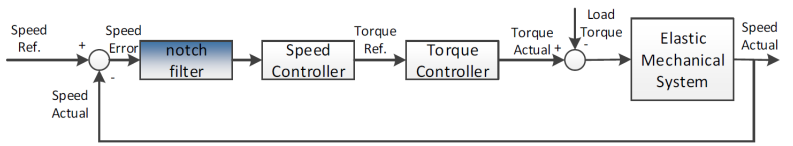
VSDの速度制御パラメータ調整

速度制御のPIパラメータ： Low controller gain with notch filter (対策後)



a) Notch filter in feed-back loop.



b) Notch filter on speed error.

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2-5 対策・結果

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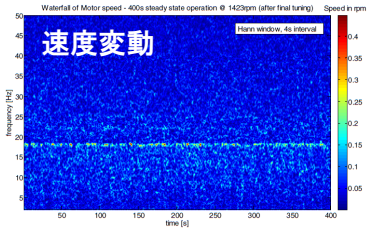
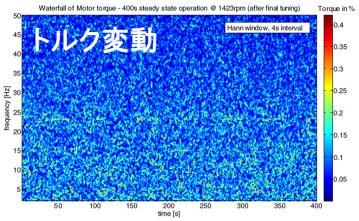
対策後の測定結果

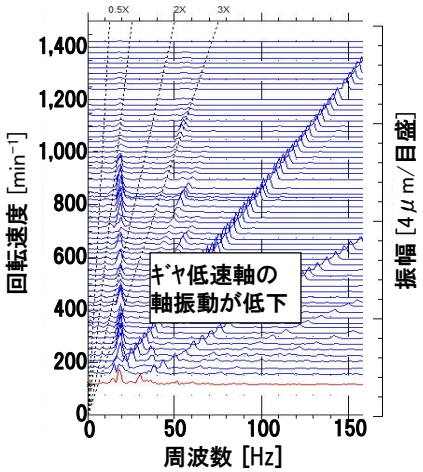


a) Actual speed – remanent small component at about 17.5 Hz visible.



c) Motor torque – no distinct component at 17.5 Hz.

インバータ内モニタリング信号



ギヤ低速軸振動

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2-6 教訓

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- 機械のねじり固有モード減衰比は推定するしかないため，VSD装置の速度制御パラメータはできる限り安定の余裕が取れるように与える。
- 汎用のVSD装置の速度制御は加速などの速度追従の応答性を重要視しており，直接的にねじり振動を抑制する制御は入れていないため，もし固有振動数の成分がフィードバック信号に大きく含まれる場合はこの成分を低減させるフィルターを付加することを要求する。

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3. メカニカルシールに起因する軸振動

出典：
多目的蒸気源試験設備用蒸気圧縮機の完成，ターボ機械，41(2), 2013, pp.65-71

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3-1 対象機器

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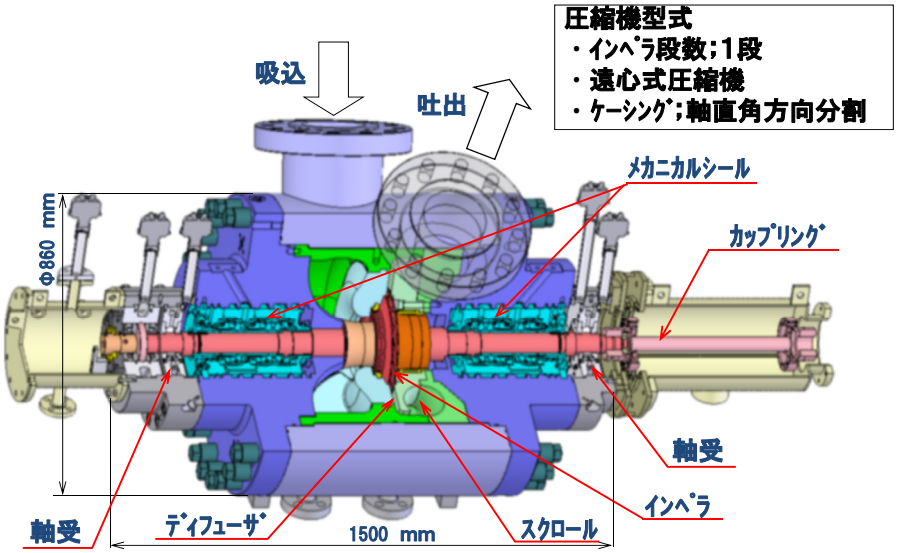

蒸気圧縮機断面図

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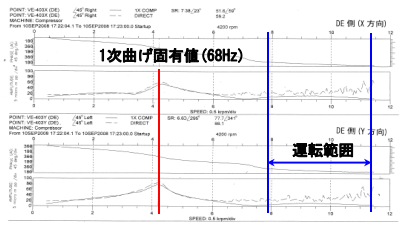
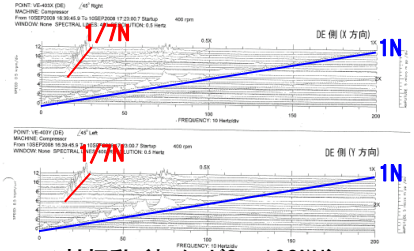
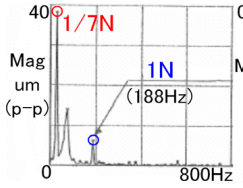
3-2 発生した現象



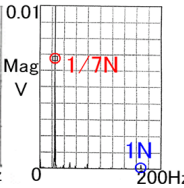
軸振動（ﾎﾞｰﾄﾞ線図0～100%N）



軸振動（ｶｽｹｰﾄﾞ0～100%N）



軸振動(FFT)



2次ｼｰﾙ室圧力脈動(FFT)

試験時の軸振動状況

過大軸振動の特徴

1.回転速度(1N)の約1/7の周波数振動が卓越した分数調波振動である。

2.ﾒｶﾆｶﾙｼｰﾙ2次ｼｰﾙ室で1/7N周波数の脈動が観察された。

発生源はﾒｶﾆｶﾙｼｰﾙと推定

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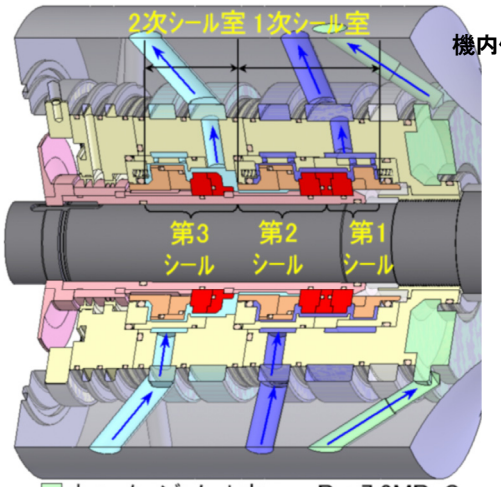
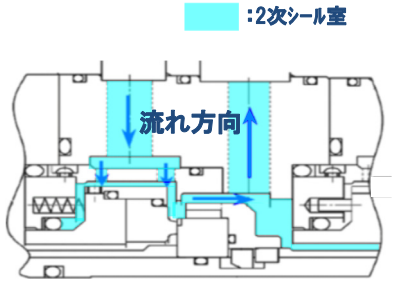
3-3 メカニカルシール構造



:ｳｫｰﾀｰｼﾞｬｹｯﾄ水　Ps=7.2MPaG

:1次シール水（高圧Pa）　Pa=7.7MPaG

:2次シール水（低圧Pb）　Pb=3.5MPaG



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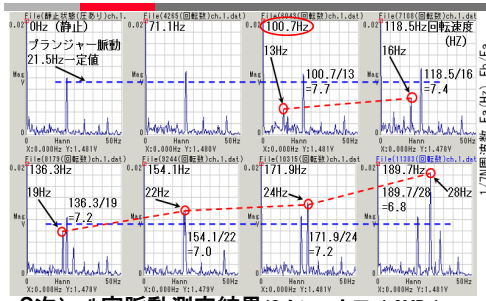
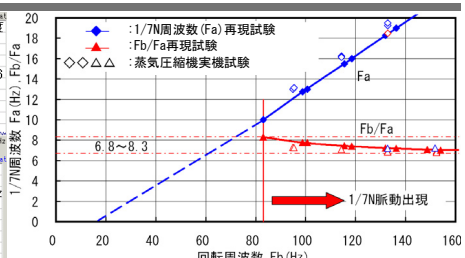
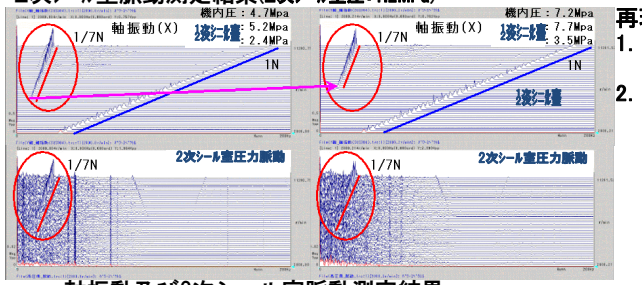
3-4 振動再現試験(シール単体試験)



2次ｼｰﾙ室脈動測定結果(2次ｼｰﾙ室圧:1.2MPa)



回転数と1/7N振動周波数比較



軸振動及び2次ｼｰﾙ室脈動測定結果

再現試験結果

1. ﾒｶﾆｶﾙｼｰﾙ単体で1/7N軸振動を再現した。
2. 2次ｼｰﾙ室で同じ周波数(1/7N)の圧力脈動を確認し、1/7N振動の発生源は2次ｼｰﾙ室内の脈動であることを確認した。

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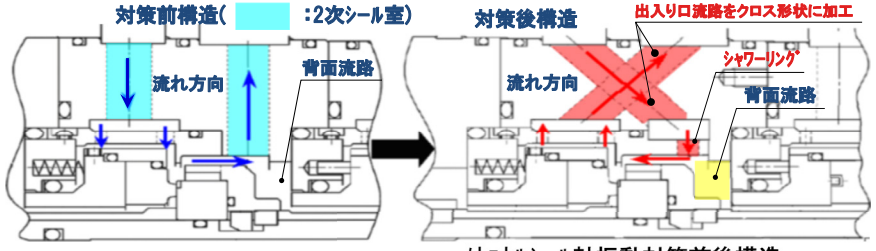
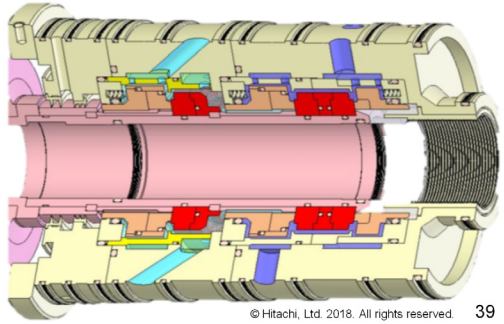
3-5 対策



ﾒｶﾆｶﾙｼｰﾙ軸振動対策前後構造

回転環の外側を流れるシール水が1/7倍速の旋回成分を有していることが，CFD解析の結果わかった。

シール水の旋回を抑制するため，流れ方向を逆転し，ｼｬﾜｰﾘﾝｸﾞを設けた。



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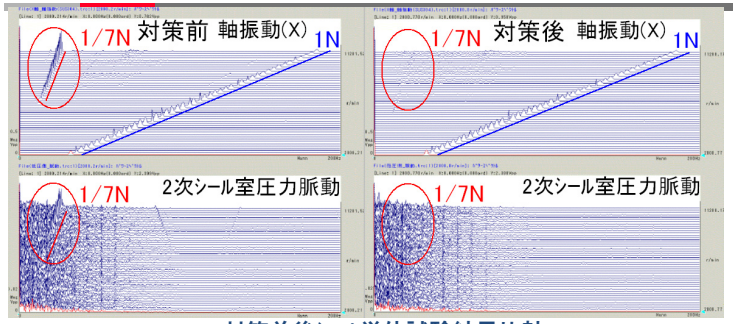
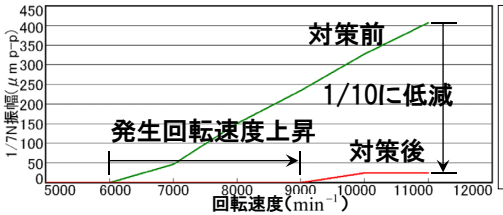
3-6 対策結果



試験条件
機内圧:7.2MPaG
1次シール室:7.7MPaG
2次シール室:3.5MPaG

対策前後シール単体試験結果比較



対策効果

1. **2次シール室の1/7N脈動は消滅**
2. **1/7N振動振幅は約1/10まで低減し、発生回転速度は高回転速度側へ移動**

対策前後構造振動比較

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3-7 教訓

- **間隙部を流れる流体が旋回するとホワールの原因となる。**
- **シール単体試験で現象が再現できたため，対策効果の把握が容易であった。**

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4. 動静翼干渉による羽根車の共振

出典:
遠心圧縮機用羽根車に及ぼす変動応力の実験的検証,
ターボ機械, 35(12), 2007, pp.745-752

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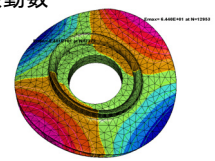
4-1 動静翼干渉

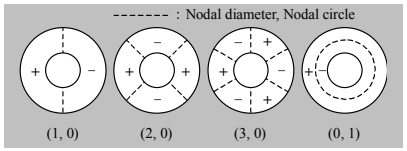
羽根車共振条件

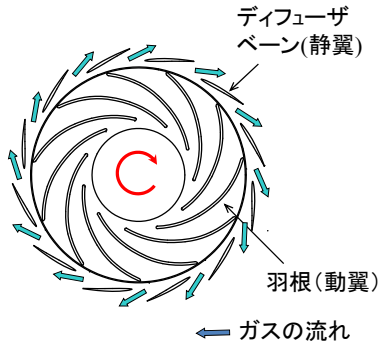
$h Z_s \pm n = m Z_r$

$h f \times Z_s = f_n$

Z_s : ディフーザベーン数
Z_r: インペラ羽根枚数
h: 高調波次数
n: 直径節の数
m: 任意の整数
f_n:インペラの固有振動数
f: 回転周波数







動翼と静翼が繰り返しすれ違い，
周期的な圧力変動が起こる

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4-2 試験目的と条件

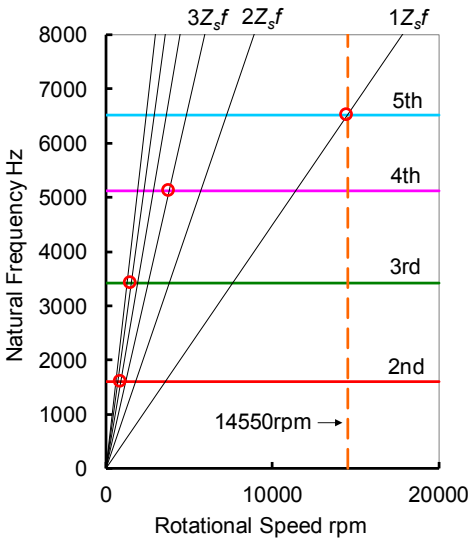

Fig. Campbell Diagram *Zs*:27, *Zr*:11

○試験目的

①共振時の羽根車応力の測定
②ディフューザ径比の影響

○試験条件

圧縮機	単段試験機
羽根車	Φ300
ガス	Air
圧力	吸込大気圧
備考	動翼枚数:11 静翼枚数:27

高回転数で
共振が起こるように
静翼の羽根枚数を決定

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4-3 試験結果①

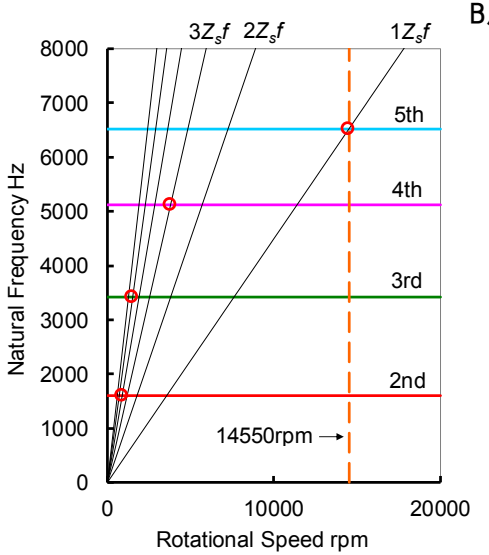
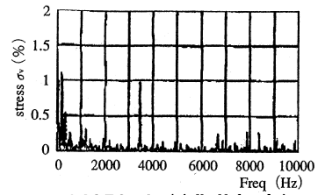
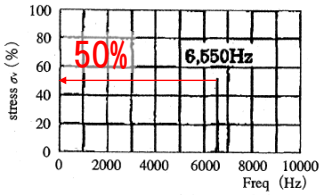


Fig. Campbell Diagram *Zs*:27, *Zr*:11

B点近傍の計測結果(羽根付根応力)



14850min^{-1}(非共振点)



14550min^{-1}(共振点)

※縦軸:定格19100min^{-1}における遠心応力が100%

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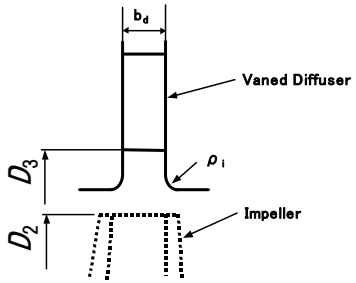
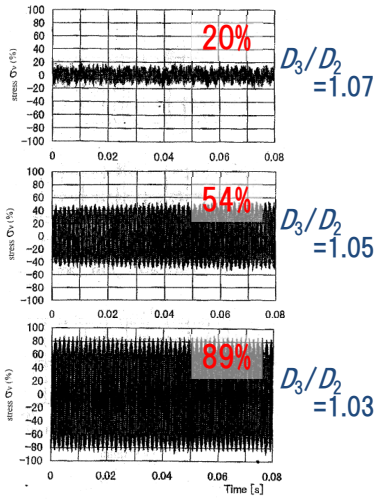
4-4 試験結果②





ディフューザ径比:D_3/D_2

D_3/D_2が小さいほど・・・
・圧縮機効率up
・旋回失速防止性能の向上

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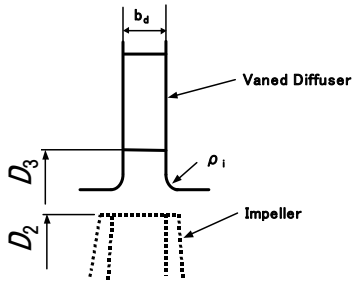
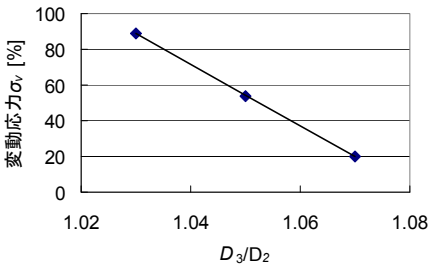
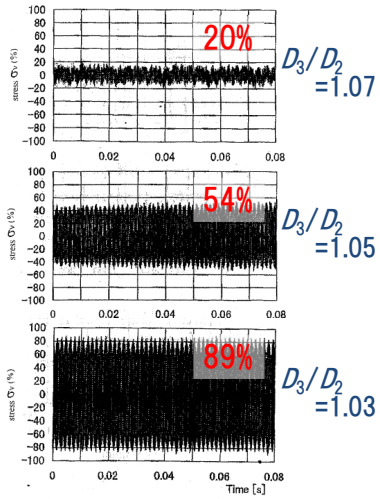
4-4 試験結果②







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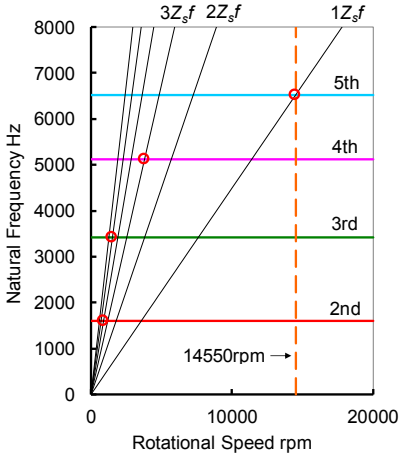

Fig. Campbell Diagram *Zs*:27, *Zr*:11

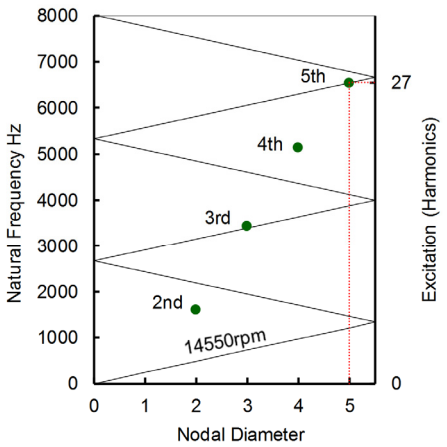

Fig. Interference Diagram Zs:27, Zr:11

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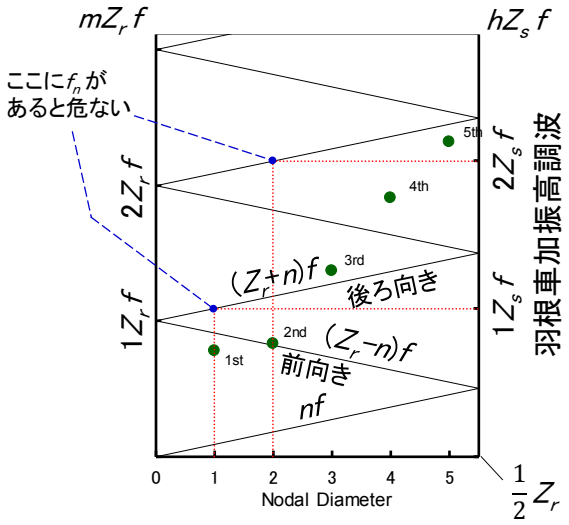
SAFE DIAGRAMの作り方



羽根車共振条件

$h Z_s \pm n = m Z_r$

$h f \times Z_s = f_n$

Z_s：静止翼数
Z_r：動翼数
h：高調波次数
n：直径節の数
m：任意の整数
f_n：羽根車の固有振動数
f：回転周波数

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- **羽根車の羽根枚数Z_rとディフューザの羽根枚数Z_sを組合せを決める際は，運転範囲で共振条件が成り立っていないことを確認する。ただし，共振条件が成り立ってなくても，共振する場合はあるので，基本的にhZ_sfが羽根車の固有振動数に一致しないようにする。**
- **ディフューザ径比D_3/D_2を大きくすることで，共振時の応力を低減することができる。**
- **羽根つきディフューザは圧縮機の性能を向上させるが，羽根車周囲に変動応力を発生させるため，羽根車の節直径共振に注意が必要である。**

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END

プロセス用遠心圧縮機で経験した振動事例紹介

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株式会社 日立製作所
インダストリアルプロダクツビジネスユニット
機械システム事業部 気体機システム部
髙橋　直彦

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