Deliberations on Gearbox Sizing: A Case Study

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Abstract

In Oil & Gas, Refinery centrifugal compressor is commonly used to compress the gas for different application like for chemical industry, refinery or to export the gas. To achieve the high discharge pressure & flow, higher speed machines are mostly selected which is achieved by speed increasing gear box with electrical motor or with steam/gas turbine as driver, hence normally compressor train is conglomeration of equipment required for specific duty. If any of equipment with in train is not properly sized, designed or got changed during production then it may impact the performance of complete train and finally end up the loss of production in plant. The present paper is mainly focused on case study of sizing of gear box due to over-specifying in specification and its impact if a change on equipment size is carried out at late stage of design.

Keywords

Double helical gear box for speed increaser, American Petroleum Institute API 613, AGMA (American gear manufacturing association), Helical Gear Mesh, Torque, centrifugal compressor train

I. Introduction

The detailed sizing and selection of centrifugal compressors and its train mostly driven by Compressor designer/manufacturer based on their most proven software and experience. Once the boundary process conditions and required flow rates are established by End user and/or Purchaser, they are given to Compressor designer/manufacturer for sizing and selection of complete train. Each equipment within train is then be sized by Compressor designer/manufacturer based on the requirement and under guideline of international standard specified by End user and/or Purchaser.

For this case study, different configurations were developed by joint effort of Compressor designer/manufacturer & End user and/or Purchaser to address varying process requirements and applications. Benefits and limitations of each configuration was evaluated and compressor train configuration was selected (refer below fig. 1).

Refer below figure (Fig.1) for train configuration which was selected for given centrifugal compressor train:

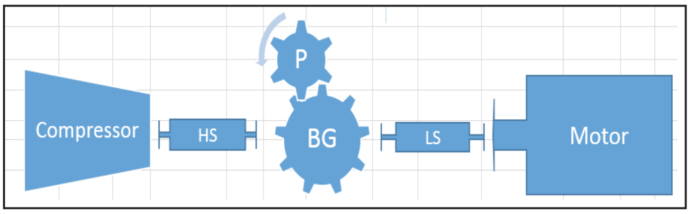

Fig. 1: Train Configuration

Refer below equipment general detail:

- Centrifugal compressor
- Double helical gear box for speed increaser (fig. 2),
- Synchronous main motor,
- Low speed flexible element coupling (LS) and high speed flexible element coupling (HS).

Refer below cross sectional view of double helical gear box:

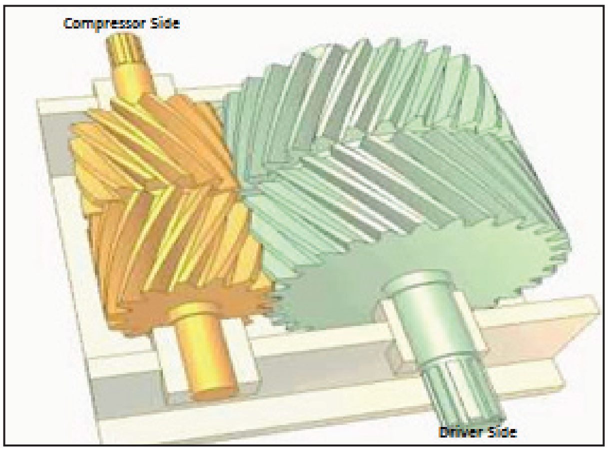

Fig. 2: Typical Double Helical Speed Increaser

II. Methodology for Train Rating

Normally, based on the compressor maximum power requirement, gear box and main motor along with couplings rating are selected under guideline specified in specifications. Also, during this selection, main motor short circuit torque (based on available torque at shaft end derived based on motor air gap torque) and/or start up torque (whichever is higher) must be considered in the coupling rating selection to avoid torsional pulsation. This is validated by torsional analysis of the complete train. Up on finalization of major train component, other auxiliaries (like lube oil system, seals & seal system, controls etc.) are then be sized.

Refer below typical flow chart (fig. 3) followed for most of train sizing:

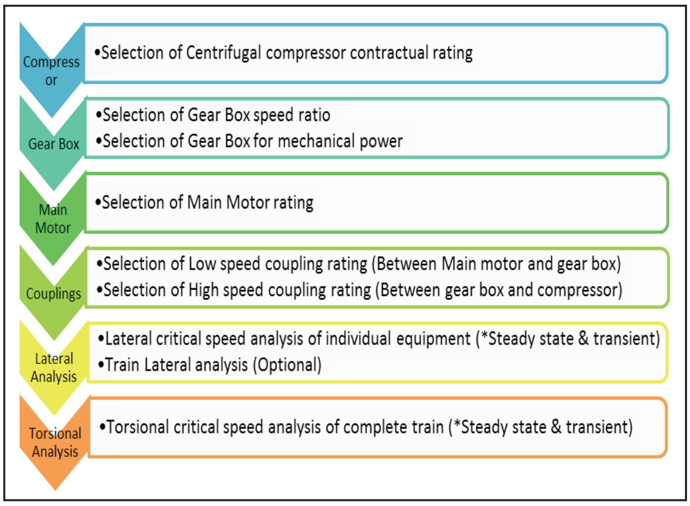

Fig. 3: Typical Flow Chart For Most of Train Sizing
(*Transient condition is startup & shut down)

III. Case Study

A. Initial Design Consideration

A centrifugal compressor complete train was ordered to Compressor designer/manufacturer, during initial design phase, Compressor designer/manufacturer had selected the gear box mechanical rating as 8000 kW (with service factor mandated by API 613) based

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on the compressor maximum power requirement of 8000 kW. However Main motor kW rating was selected for 10000 kW based on the additional (10%) motor margin, (7%) API margin & (2.5%) aging losses as per End user requirement.
Based on 8000 kW rating of gear box, the gear box inertia 1080 Kg-m2 was considered in complete torsional train analysis. Low speed and High speed coupling were also selected based on the 8000 kW rating of gear box.
Main motor starting study was conducted by Compressor designer/manufacturer based on the gear box inertia of 1080 Kg-m2 and cold/hot starting sequences was established which validate the low speed coupling selection.

B. Specification Design Requirement

In most of Oil & Gas and Refineries have had used to follow American Petroleum Institute API 613 for gear box. Allowable bending stress is fairly higher when gear box is designed based on AGMA (American gear manufacturing association) compare to API 613 [3], hence it is fairly said that if we specify the gear box as per API 613 then gear box size is fairly bigger than AGMA 2001 or ISO standard gear box although main design criterion remains same.

C. Forces in a Helical Gear Mesh

The helical gear's transmission force, Fn which is normal to the tooth surface, can be resolved into a tangential component, F1, and a radial component, Fr. See Fig. 4. [6]

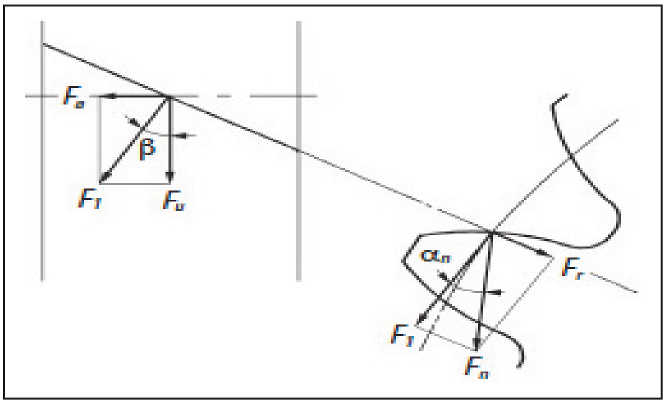

Fig. 4: Forces Action on Helical Gear.

$F_1 = F_n \cos\alpha_n$ & $F_r = F_n \sin\alpha_n$
The tangential component, F_1, can be further resolved into circular subcomponent, F_u, and axial thrust subcomponent, F_a.
$F_u = F_1 \cos\beta$ & $F_a = F_1 \sin\beta$

Substituting and manipulating the above equations result in:

$F_a = F_u \tan\beta$ & $Fr = \mathrm{Fu}\dfrac{\tan\alpha n}{\mathrm{Cos}\beta}$

The directions of forces acting on a helical gear mesh are shown in fig. 5. [4-5]

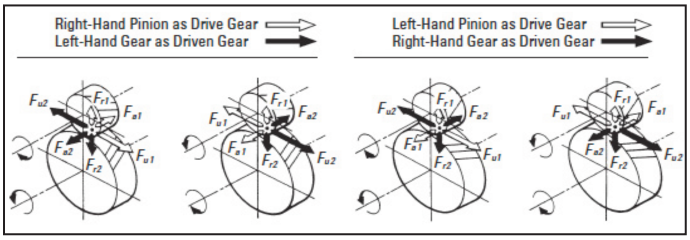

Fig. 5: Forces Action on Helical Gear Mesh

The axial thrust sub-component from drive gear, Fa1, equals the driven gears, Fa2, but their directions are opposite. Again, this case is the same as tangential components Fu1, Fu2 and radial components F_{r1}, F_{r2}.
Refer below sample example of bending stress calculated based on API 613. The same was further checked by AGMA based tools. [2]

Bending stress as per API 613 (Calculated for example)

$$Sp = Wt * \frac{Pnd}{Ft} * SF * \frac{1.8CosY}{J} = 233 \text{ MPa}$$

Where,
W_t = 18723 Pound Tangential force,
P_{nd} = 3.629 Inch. Normal diametral Pitch,
F_t = 10.78
J = 0.548 AGMA Geometry factor [2]
SF = Service factor
Cos Y = 0.99

As per API 613 and its modified specifications were part of the contract, gear box mechanical rating should be same as main motor name plate rating (10000 kW). When the configuration was finalized and about to start getting manufactured, a serious flaw was surfaced. The flaw was not meeting the mechanical rating of gear box as per API 613 (clause 2.2.1, *"For electrical motor drivers, the gear unit rated power shall be the motor nameplate rating multiples by the motor service factor)* [1] and its modification captured by end user specification. This requirement was overlooked by the Compressor designer/manufacturer and not considered in the gear box sizing at initial stage and subsequently not considered in the complete train sizing. However, this is still believed that design was not detrimental for practical purpose.
Specification requirement was as below:

Gear box mechanical rating	10000 kW (same as main motor name plate rating)
Coupling (Low speed & High speed)	Suitable for 10000 kW of gear box and 10000 kW main motor

D. Impact on the Train Sizing Due to Specification Requirement

When realized, it was agreed by Compressor designer/manufacturer to correct rating (10000 kW) of gear box. As the consequence, the gear box bull gear & pinion size is increased, gear box rotor inertia also increased (1980 Kg-m²) which was not considered in coupling design in a view of torsional vibrations during start up with asynchronous motor. As gear box size has increased which impact the complete train footprint.
Refer below summary of changes in the train: Fig. 6

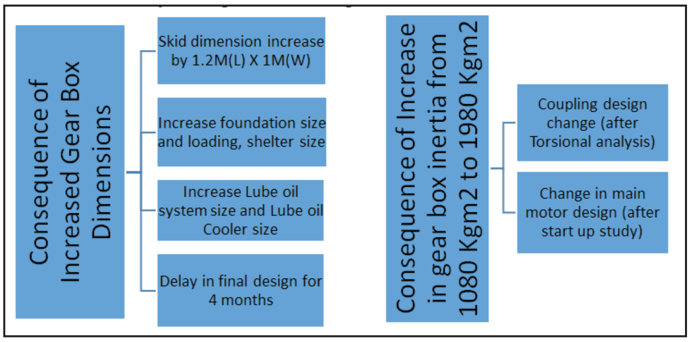

Fig. 6: Summary of Changes

E. Change in Low Speed Coupling Design

Based on new gear box inertia, following options were evaluated to accommodate the revised gear box inertia (1980 Kg-m²) to avoid torsional vibration.

Option 1: Clutch type coupling

Option 2: Elastomer coupling (Hole set type Damper coupling)

Option 3: 1100 Distance between shafts ends (DBSE) spacer with higher rigidity VS mass ratio (Titanium or equivalent material)

Option 4: 1300 Distance between shaft ends (DBSE) spacer material high alloy steel

Option 4 was finally selected based on the end user experience and least impact on the complete train foot print.

End user didn't have experience/confidence on the usage of Slip clutch and hole set damper, so these two options were discarded. Spacer material of titanium was last priority in consideration of high spare cost of titanium material.

Refer below figure (fig. 7) for offered options :

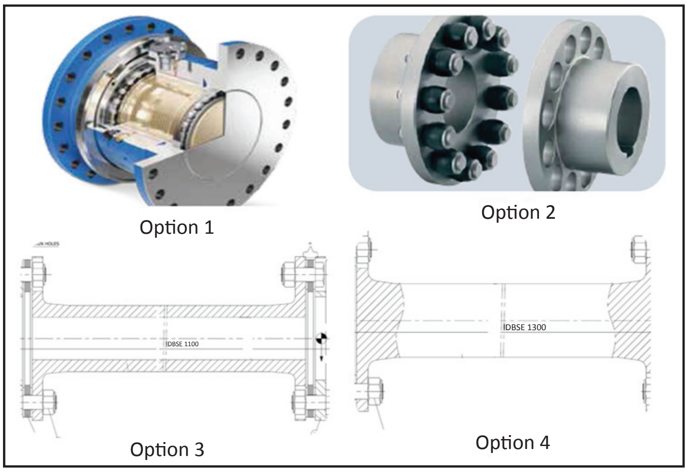

Fig. 7: For Coupling Options [7-8]

F. Change in Main Electric Motor design

During evaluation and finalization of the coupling options, it was found out that due to higher inertia of the gear box, motor is not suitable for subsequent start up as mandated by end user specification (3 consecutive cold starts and 2 consecutive hot starts with interval of 30 minutes).

Since gear box moment of inertia increased from 1080 Kg-m² to 1980 kg-m², heat loss during main motor start up is increased, also to overcome higher inertia motor had to draw more current which is directly proposal to heat losses. As motor has more heat loss during start up, next consecutive start-up required much more cooling time which is not in line with the starting requirement of end user specification requirement.

Refer below equation normally used by motor manufacturer:

$$E_2 = \underbrace{J(2\Pi n_1)^2 \int_{SStart}^{Send} S * ds}_{} + \underbrace{2\Pi n_1 \int_{t=0}^{t=a} S * Mm * dt}_{}$$

1). Heat loss proportional to the inertia (J) of driven equipment (Gear box + Compressor). 2). Heat loss is proportional to maximum speed equipment is accelerated to.	1). Heat loss is proportional to time needed to start up driven equipment to rated speed (Depends on counter torque for given motor torque).

Where,
J = Train moment of inertia
S = Speed of equipment/train
E2 = Heat loss

Refer below figure (Fig.8) showing over heating of main motor – rotor shoe pole (Bluish Mark noticed)

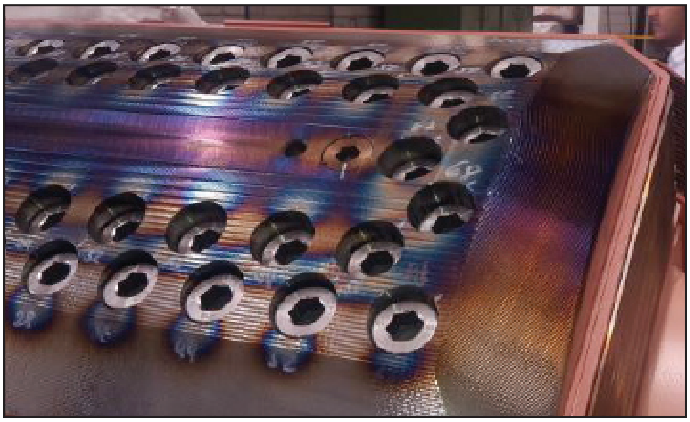

Fig. 8: Generic view of Motor – Rotor Shoe Pole Overheating

Main (synchronous) motor rotor has 4 poles mounted on rotor with bolting. Pole has rotor windings.

Due to overheat despite during start up, the shoe pole gets over heated then allowable temperature.

Refer figure (fig. 9) for specification starting requirement:

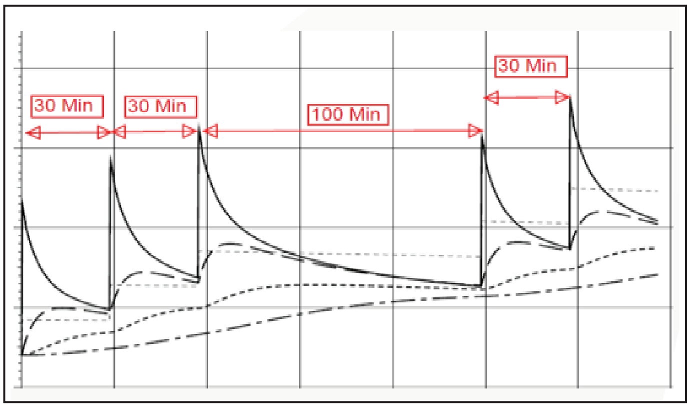

Fig. 9: Motor Starting Sequence [9]

Existing motor was not meeting the consecutive starting requirement as stated in above fig. 9, motor required more cooling time due to excessive heat loss. Existing motor needs approx. 8 hours cooling time between second and third cold start and similar approx. 8 hours for third and fourth hot start. More cooling time lead to delay in starting the compressor train and finally loss of production which is directly a loss of revenue.

Main motor was also had to be redesigned to meet the starting requirement. Complete motor rotor was rebuilt with higher cooling capacity to accommodate the additional heat loss in main motor. The major impact was redesign the complete train, cost and schedule.

Although it was agreed jointly by compressor design engineer and purchaser, it was major flaw in current train but still it must be pounder upon trail what was the read tangible benefit drawn with such a large change which is summarized below:

1. Overall skid size increased by 1.2M(L) X 1M(W) meter & shelter size change, increase foundation loading

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2. Scrap the existing motor and start design and manufacturing of new motor
3. Scrap existing gear box and start design, manufacturing of new gear box
4. Considerable delay in engineering
5. Overall delay of machine by 6-8 months

IV. Conclusion

It was well understood process gas used for compression was a clean and hence no chance of fouling means no extra power required or no detoriation of efficiency in future operation. This indicates that gear box with existing rating (8000 kW) with lesser service factor/design margin would have served required operation.

This case study shows that there is no impact on gear/pinion design on above real scenario (if gear box rating kept as 8000 kW). This paper emphasis to use deliberation and consensual joint judgment while designing the train in present Oil & Gas for a quick return of investment (ROI) point of view. Practical approach should also be considered while changing the major equipment which is under advance stage of manufacturing and while doing so, it is to be ensured that there is no compromise in safety and quality.

References

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