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# Carbon Foot Print Reduction Techniques With Rotating Machinery

**Vasanth M Bhat**
Integrity Manager
Singapore Refining Company Singapore
Total of 25 years of work experience in various industries, mainly chemical, petrochemical & refining.
Has a Bachelors Degree in Mechanical and certified Category IV Vibration Specialist from Vibration Institute USA. Currently pursuing a Masters in Rotating Machinery from University Of Zaragoza

**Quek Ser King Aaron**
Energy & Loss Control Manager
Singapore Refining Company
Singapore
Graduated from National University of Singapore with a Bachelors Degree in Chemical Engineering in 2005.
Obtained a Masters in Chemical & Biomolecular Engineering from NUS & MIT in 2006.Total of 9 years working experience in Singapore Refining Company, including the 3 years as Energy and Loss Control Manager.
Awarded the Outstanding Energy Manager of the year in 2015 by National Environment Agency

## ABSTRACT

Global Environment concerns and need for more optimized production costs is driving all industries including Oil & Gas to look at more ways to reduce energy consumption without compromising on plant throughputs. This is specially challenging for older units to carry out modifications on existing assets and justify these with good paybacks for the capital investment needed. The article looks at the possible solutions available in this regard relating to rotating equipment and are illustrated with case studies to back these recommendations in most cases.


## INTRODUCTION

The article highlights the challenges faced in the refining business in particular with regards to energy consumption .It focusses on ways to get step reduction in energy consumption by use of various simple to complex techniques to see the reduction in energy demand.

Each possible strategy is explained with actual experiences to show the effectiveness of this approach.. Various techniques discussed are:

1) Stepless Capacity control in reciprocating compressors
2) Conversion of condensing turbine drive to Variable speed motor drives for compressors.
3) Centrifugal compressor coatings and seals which could result in lower energy and better performance.
4) Reducing wastage of steam by changing seal types in steam turbines and eliminating slow roll.
5) Converting large pumps & fans from spill back or pressure control valves to Variable speed drives.
6) Changing air cooled exchanger blades to more efficient FRP (Fibre reinforced Plastic) blades & better belt drives.
7) Looking at more efficient motors
8) Exploring HPRT( Hydraulic Power Recovery Turbine) options.

## ENERGY OVERVIEW

Other than crude, catalyst and chemicals, one of the big segments of the cost for an operating refinery is energy. The pie chart (figure 1) below shows the typical energy spend in a typical mid- size refinery

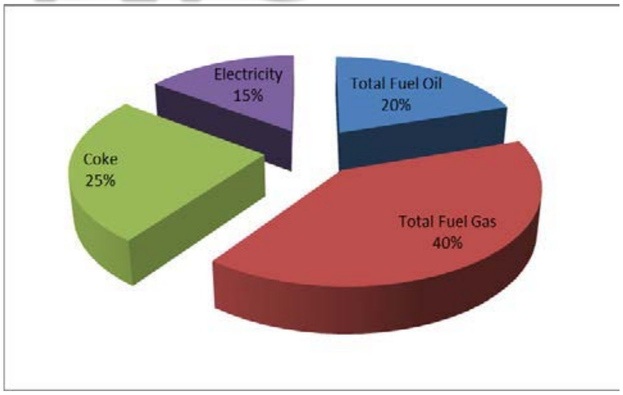


*Figure 1: Energy Usage MMBtu/hr distribution in a typical refinery*

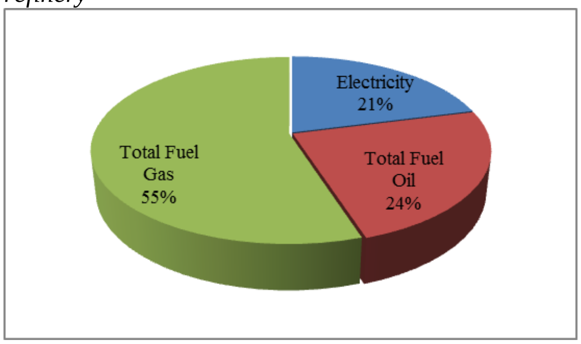


*Figure 2: Energy cost split in a typical refinery*

The chart above (figure 2) shows the approximate energy cost split for an Asia pacific refinery.
Approximately 20 - 25% of the refinery's energy cost is used to run prime movers namely compressors, pumps fans etc. This is made up of electricity which is related to running motors and fuels which are indirectly related as they are used to generate steam to run the steam turbines. This means that any energy efficiency improvements in rotating equipment provide a good opportunity in terms of reducing overall refinery operating costs.
As with any process the biggest opportunities lie with the big machines which are typically the centrifugal compressors, reciprocating compressors and feed pumps of the various units. Of course the base assumption is machines are operating as close to BEP (Best Efficiency Point) in most cases and the suggested schemes can possibly improve the performance from that position.

**EQUIPMENT TYPE & CASE STUDIES:**

**Centrifugal Compressors:**

Centrifugal compressors offer some great unique opportunity for energy saving. Some of the typical saving opportunities lie in:

1) Anti-fouling coating resulting in improved performance over longer period of time
2) Using non-metallic labyrinth seals which can produce lower clearance and also deform slightly.
3) Changing of seals to dry gas seals from wet seals.
4) Performance enhancement using water or liquid washing systems.

Option 1: **Anti fouling coating**
There could be much more potential upgrades in impeller flow passage improvements etc. but these are normally expensive upgrades and may not achieve substantial gains to justify them in most cases.

Case Study 1:
The wet Gas compressor in FCC unit was proving to be a limiting as at the end of a four year run the compressor internal would be fouled and the extent of fouling would be so high that the steam turbine driver was not able to attain maximum speed conditions.
So when a capacity creep was initiated it was considered if it was necessary to upgrade the compressor to handle higher flow with an incremental capacity increase in the steam turbine capability. The compressor OEM suggested that a more cost effective approach could be to coat the compressor rotor with anti-foulant coating. (refer photo 1)

*Photo 1: Wet Gas Compressor rotor with coating*

With a minor upgrade to the turbine the machine was put into to operation and this time around the compressor managed to operate at max operating speed at the end of the run without limiting plant through put.
The coating managed to withstand the four and half years of continuous operation.( refer photo 2)

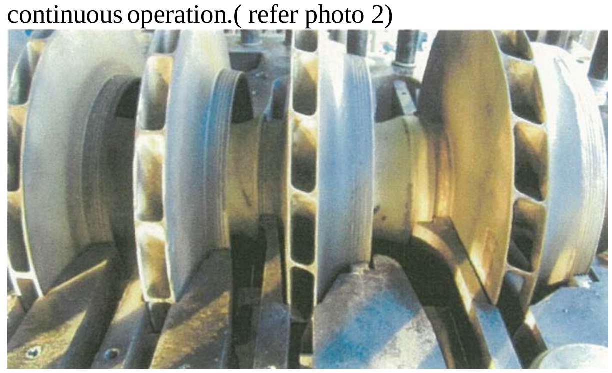

*Photo 2: Same Wet Gas compressor rotor condition after close to five years of service.*

Lesson learnt:

a) Initially it was assumed there would be no change required to the rotor or other parts but on assembly it was observed that the labyrinth seals rubbed against the rotor when the horizontal split casing was tightened. It was realized the coating thickness ate up the clearance resulting in jamming of the compressor rotor.
b) The fouling in the compressor rotor reduced but the foulant continued to foul the as flow passages of the stationary diaphragms. We are considering providing a coating to the stationary parts as well so that the performance of the machine could be maintained.

Option 2: **Change to non-metallic labyrinth seal** has been attempted but has not been very successful due to poor gas conditions resulting in rapid wear. The refrigeration compressor (refer photo 3) have these type of seals and there is no direct comparison data to show the difference in performance with a normal seal.
Generally it has been accepted in the industry that it would reduce the internal leakage loss which could be in most cases be around 2%, so depending on the size of the machine the benefits could outweigh the costs. In terms of costs there could be a fivefold increase in costs between conventional labyrinth seals and use of non-metallic material.

*Photo 3: Compressor with non-metallic seals at impeller mouth rings.( appear in white colour)*

Option 3: **Change from wet seals to dry gas seals:**

The wet gas compressor in the FCC (Fluid Catalyst Cracking ) unit had wet seals which had its own wet seal system. This seal system requires seal oil pumps, exchangers and the wet seal consumes some power loss.
Due to reliability issues with the wet seal the compressor was retrofitted with dry gas seals and thus eliminated the need for a seal oil system and its associated losses.
In many installation similar to this seal oil reclamation units are attached which also need power for pumps, heaters etc. The annual operation costs for seal oil system just based on energy costs was in the tune of US$150,000 per annum in this case. It could be higher or lower depending on the size of the prime mover for the seal oil systems. A typical loss in wet contact seals compared to dry gas seals would be in the range of 10 to 15 KW per casing, again a small number but when seen as a total system can prove to be significant.
Lessons Learnt:
1) As with any typical retrofit project keep some contingency in terms of schedule and costs to take care of surprise findings.
2) Keep an option for full reversal as in some rare cases Dry Gas Seals may not end up as reliable solution.
3) Depending on familiarity of the site personnel to dry gas systems we need to have a robust training program to ensure operators understand the dry gas seal panel which is more complicated in terms of looks compared to seal oil systems.

Option 4: **On line washing systems**
Running a fouled compressor always results in higher specific power consumption. On line wash offers a quick remedy to performance degradation but has its own challenges. Water or naphtha has been used to wash but the effectiveness of the wash would depend on the location of the nozzles with respect to the fouling.
With washing there are 2 major risks, namely machine could end up going to high vibration if the distribution of the fouling and the cleaning effectiveness varies with location on the rotor, secondly with dry gas seals it gets further complicated with possible risk of contamination reaching the dry gas seals and potentially resulting in its failure.
Due to these risks industry has used it more effectively for gas turbines, main air blower and to much lesser extent in wet gas compressor and recycle gas compressors. Washing of steam turbine rotors has also been done but maintaining a good steam quality would be the area of focus rather than depending on water washing as a solution.

**Steam Turbines**:

The next potential saving in energy comes in terms of large drives used for operating the centrifugal compressors. In refineries in earlier times steam turbine was the choice as power supply was not as reliable (from the Grid) as it is today. Also most compressors need variable speed option and hence steam turbine offered the best option for drives. The limited availability of large motors and reliable cost effective variable speed drives forced a less efficient steam turbine as driver of choice. As we know steam turbines come in two major categories namely condensing steam turbines and back pressure type. The low overall efficiency for condensing steam turbines makes them ideal candidates for retrofits. The typical options considered for this category of machines are:

1) Motorization with variable speed drives
2) On line washing of steam turbine
3) Installing better seals to lower and eliminate leakage.
4) Eliminating slow roll of

turbines. Option 1: **Motorization**

Motorization of compressors with more efficient motors with variable seed drives are one of the common ways in which significant energy efficiency gains have been witnessed . There are two categories in this. One being going for a synchronous motor directed connected to the compressor. The challenges have been in getting a match in terms of power and speed for high speed motors. This option is viable only when we have large motor capacities beyond 5 MW range. For smaller size 2 pole or 4 pole motor with Variable speed drive (VSD) through a gear box or using a mechanical speed variator. Both systems have their pros and cons, each site preferences and project economics being the key factors which impact the final choice.

Case Study:

A recycle gas compressor in the Catalyst Reformer unit is driven by a condensing steam turbine. As the machine was built in early 80’s the compliance to latest requirements of applicable API standards was low. The turbine has been very reliable with minimal major issues. The low efficiency, lack of compliance to new API requirements in terms of over speed protection and vibration protection made this machine an ideal candidate for motorization.
Due to the relatively smaller motor requirement (less than 2 MW) a standard motor gearbox configuration was selected. (refer figure 3 for layout )The mechanical VSD option was considered but due to space and lob oil system complications it was considered not economical for this application.
There are few big challenges to consider:

1) Space & capability on the existing compressor foundation to accommodate a motor and gearbox in the same space occupied by the turbine.
2) Due to additional lateral critical and torsional critical speeds the operating range could be limited compared to the steam turbine drive.
3) This type of activity takes much longer duration compared to standard turn around (TAR) windows. If the machine retrofit adds days into the TAR window the associated production loss would greatly impact the payback for such projects.
4) The steam turbine as in this case was very reliable comparatively VSD drives could offer challenges in matching the availability numbers. VSD drives also offer an opportunity in terms of providing back up option of driving the motor direct on line (DOL) without the VSD. This will come with its own complications on how to effectively manage this condition in terms of unit operation. It is important to note generally that steam turbines have better MTBF (Mean time between failure) compared to electrical drives but the MTTR (Mean time to repair) is higher comparatively which makes it critical that all critical spares in terms of power cards etc. are available so that repairs could be fixed with trained competent crew in the shortest possible time.

The payback for such projects has been typically over 2 years and execution costs would determine if it would be in the below 3 years range.

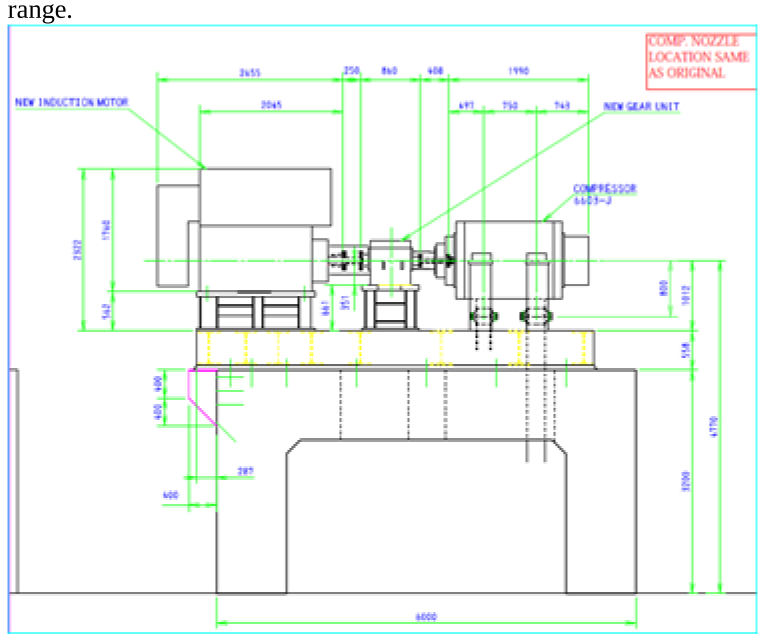


Figure 3 *Shows the layout of the compressor with motor and gearbox replacing the condensing turbine. Note the changes in the foundation (highlighted in pink) to accommodate the longer length of the skid.*

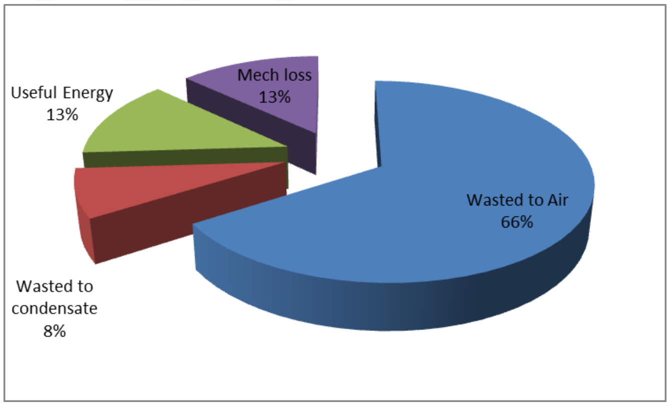


*Fig4 : Shows the amount of energy wasted in the turbine which was considered for motorization*

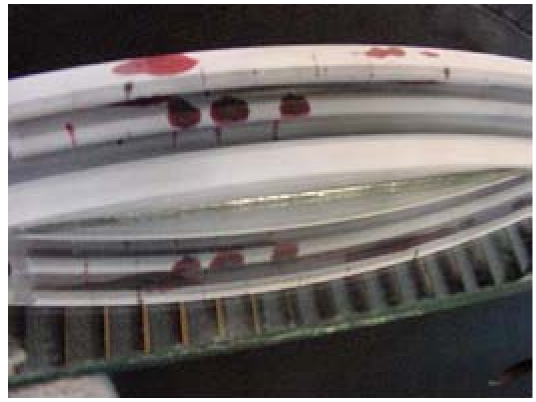

*Photo4: Shows the Steam turbine rotor with cracks observed at the dovetails of the second row blades suspected due to SCC and additional stress imposed due to frequent water wash*

Option 2: **Online washing of steam turbine**:

Online washing has been adopted for steam turbines especially condensing turbines due to build-up of salt and the silica deposits on the blade passage resulting in reduced output from the turbine.
Two types of washing techniques have been used; one involves shutting the turbine and filling the casing with demineralized water and carrying out slow roll of the rotor. The water quality is measured and based on turbidity level the washing is stopped. Elaborate procedures need to be in place to ensure this is done without hiccups. The other technique involves direct injection of hot treated water in the steam inlet with machine operating at close to normal operating condition. This option is most preferred as it involves typically no loss in production but the risks for this process could also be high. The few major issues are:

1) Ensuring the water injection is just right as too much water will cause blade tips erosion.
2) Also the effectiveness of the wash compared to the offline wash is lower which means you need to have a series of washes to achieve reasonable performance.
3) Some of the rotors may not be in a position to withstand the shock loading; there was a case in which there was a crack observed on the wheel disc which was attributed due to high stress from water wash.

The best option is maintaining the right boiler water quality through regular monitoring and blowdown ensuring good steam quality.
Water washing during turnarounds and removing salt deposits in between dovetails is another good practice to ensure avoidance of SCC cracks at blade dovetails.

Option 3: **Installing better seals to reduce leakage loss.**

The most common seals for steam turbines are the conventional labyrinth seals. For general purpose turbines the typical seal involves carbon segment seal. This has been the seal of choice for most of the seal vendors. They will wear out resulting in steam leaking to the atmosphere and also resulting in bearing oil contamination resulting in turbine reliability issues.
There are few options now ranging from involving installing of brush seals to use of steam seals from mechanical seal vendors. Mechanical seal vendors have developed special design seals which use a combination of high temperature seals and non- contact grooved face design which almost eliminates any leak from the seal. ( refer fig 5)

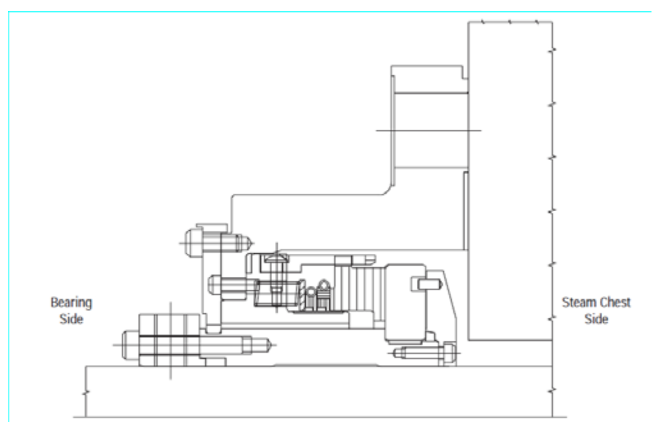


*Fig 5: Cross sectional drawing of a mechanical seal installed on a steam turbine to replace its carbon segmental seal into almost zero leak seal based on face grooves similar to dry gas seals in compressors ( Image Courtesy : John Crane )*

A badly leaking seal could result in steam loss to the tune of US$ 50,000 to 100,000 which can easily payback such upgrades. But few challenges exist:

1) The retrofit of existing seals of turbines to much seals would

require major modifications and normally it is irreversible, so performance assurance of the new seals is a must before deciding on a change.

2) The steam quality has a direct impact on the seal performance, if the steam is wet and machine has regular starts and stops the seal life is likely to be short.
3) Compared to carbon ring seals the cost of seals even in terms of repair can be quiet high so a minimum life of 3 to 4 years would be required to justify their installation.

Option 4: **Slow roll of turbine**

Many occasions general purpose steam turbines which drive pumps are left on slow roll so that they can quickly come to speed when the pump is to be operated. Rotor bow is not a major concern with short span rotors observed in general purpose steam turbines. There is some confusion in operators mind that all steam turbine rotors need slow roll. An complete audit for all turbines to be done, to see the need for slow roll. We have observed this could be required in very rare cases such as slurry bottoms or tower bottom service where there is a risk for solidification and settling of solids.
When such an exercise was conducted in the refinery it was observed about close to 15 turbines were on slow roll when they were not intended to be. Stopping them from slow roll was estimated as a potential saving of half a million dollars a year.

**Reciprocating Compressors:**
After the centrifugal compressors the next biggest power consuming machines are reciprocating compressors, the typical services they would be found are overhead compressors , nett gas compressors , recycle gas compressor and most of the make-up compressors compressing hydrogen in Hydroprocessing units.
There are some options for energy saving which include:

1) Stepless capacity control
2) Improved pocket valve design
3) Lower power ,wear& tear with converting non lube to lube compressors

Option 1: **Stepless Capacity Control**
The typical capacity control involves unloaders which can be 2 to 5 steps depending on the cylinder configuration and no of stages. In recent time’s stepless control provided by few vendors have gained popularity and are now being considered as standard even when we procure new machines .The reader is advised to look at references available from vendors on the principle of operation of stepless capacity controls.

It is not common to see reciprocating compressors at 100% loading. In most cases there would be multiple units running with some on 50 or 75% loading and further control is achieved through a spill back control valve. The spill back flow and the reverse flow in the compressor when the unloader engages both result in power loss which could be substantial in many cases.

With a stepless control system installed the valve opening can be controlled to the level required eliminating the opening of the spill back valve and reverse flow only for the percentage of the stroke that is required to achieve the capacity control . ( refer to fig 6 for the impact on the PV diagram & photo 5 shows a typical installation)

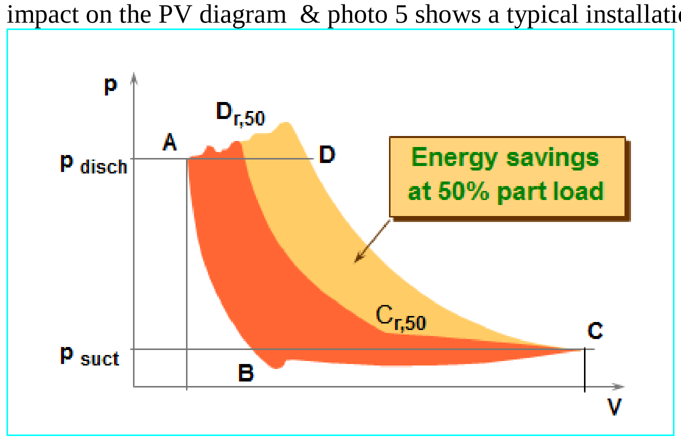


*Fig 6 : Shows the energy saving achieved by using stepless control ( Image courtesy : Hoerbiger )*

*Photo 5: Actuator mounted on a suction valve of typical reciprocating compressor cylinder for achieving stepless control ( Image courtesy : Hoerbiger )*

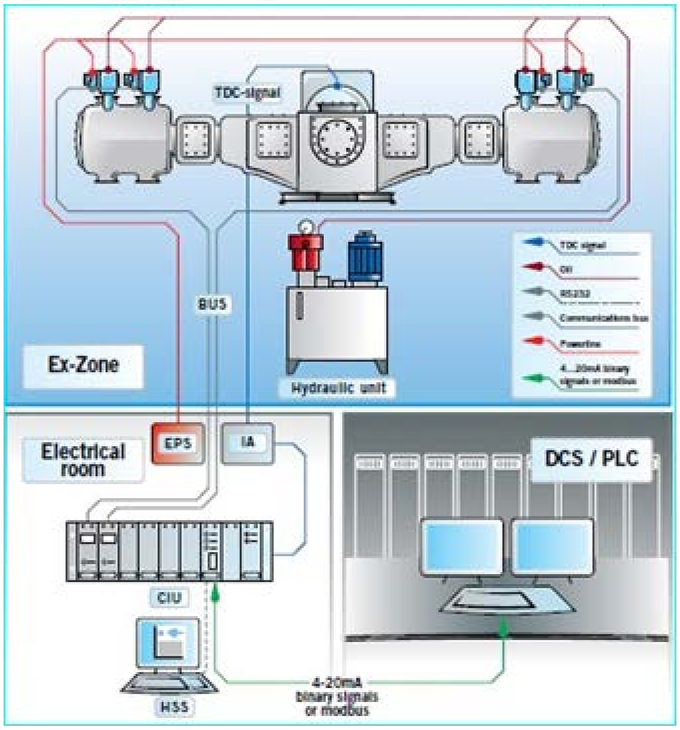


*Fig 7 : Typical installation of a stepless control system on a reciprocating compressor ( Image Courtesy : Hoerbiger)*

Case Study:

We had 3 make up compressors in the Hydrocracker unit which were operating in 2x100% and one on 75% loading with the spill back in open position to ensure the required capacity of make-up hydrogen to the reactor loop.
A stepless control system was installed on one of the compressors and close to 15% energy saving was achieved post the installation with close to quarter of a million dollars savings in terms of energy costs .

| | Stepless Control | Spillback |
|---|---|---|
| Power requirement (kW) | 1257.31 | 1475.92 |
| Compressor Flow Rate | 1430.82 | 1800.02 |
| Spillback Valve Opening (%) | 0 | 17.42 |

Estimated power savings, kW
= 1475.92 – 1266.39
= 209.53 kW

*Fig 8: Energy saving per hour achieved post installation of a stepless control system.*

Some of the key lessons learnt and best practices adopted:

Ensure a pulsation analysis is carried out to ensure there would be no surprises post the installation.
b) In most cases the compressor valves would need a replacement as the actuator acts only on certain types of valves best. Valve covers would need replacement as actuators would require new covers and could potentially new cover bolt tightening torques.
c) Look at fail safe option more carefully as you will have to configure the machine to trip or load when there is a failure in the hydraulic unit or the control system.
d) Review the rod reversal calculations to see the impact of the change, this would be provided with compressor OEM or these system providers.

We have installed this stepless control on majority of our compressors and have been operating successfully with some problems during initial commissioning, post which they have operated with no issues. We have noticed improved life of valves compared to similar compressors with similar valves; this could be attributed to the more gradual unloading of the valves.

Option 2: **Efficient Valves**

Many of the old compressors have valves which are metal plate type with non optimized valve lifts. Valve losses account for 2 to 5% and can be improved by optimizing on valve lift.
Increased valve lift results in improved performance but could result in lower life. So a balance is required. Today with improved valve analysis programs popular valve vendors can provide more efficient and reliable valves using optimized lift and use of nonmetallic rings.

Option 3: **Conversion from Non Lube to Lube type**

Conversion of non-lube to lube type compressors offers another opportunity to see some power loss reduction. The typical rule of thumb for mechanical losses in a compressor is around 5%. This includes the losses in the bearings and a significant part could be from the friction losses in the piston rider rings, sealing rings and the rod packing’s.

. In applications which are sensitive to oil contamination non lube type compressors were selected. The carbon filled Teflon grades are almost the same for both applications but their performance in lube condition is generally better.
This conversion has higher motivation towards increased reliability compared to energy saving but in a recent case in our Nett Gas compressor we observed about 10 amps reduction in motor current for similar condition pre and post the conversion
.
The economics of this project will have an element of power saving but more so in terms of LPO (Loss profit opportunity) losses due to outages related to equipment repair and overhauls.

**Centrifugal pumps**

The biggest population in terms of rotating equipment in atypical refinery is centrifugal pumps. So they can range from huge multistage feed pumps to the fractional horse power dosing pumps. In terms of energy saving potential they are more on the large motor or power requirement pumps and pumps which operate off BEP or with spill back valves open.

The various options available are:

1) Conversion to variable speed drives from fixed speed motors
2) Upgrade of impeller geometry to more energy efficient impeller designs
3) Use of nonmetallic wear rings and bushes
4) To a lesser extent with more energy efficient mechanical seal flushing plans.
5) HPRT ( Hydraulic Power Recovery Turbine)

Option 1: **Variable Speed Drive**
The best selection of pump is one which operates close to its BEP , but unfortunately over years pump duty points could be changed based on new operating conditions . This results on many occasions for pressure control valves to open or sometimes need dual pump operation with both pumps operating far away from BEP. This results in not only a low efficiency operation but also an bad actor pump with high bearing and seal failures. So these pumps provide an opportunity for improvements both in terms of energy saving and reliability improvements.

Some of the technologies available are:
a) Variable speed motor drives
b) Mechanical drives uses variators
c) Magnetic couplings/ fluid couplings

Options b and c would offer challenges in terms of space in the existing skid, option a) would be limited by space in substation to house the drives.

Two other options exist but are generally not considered efficient and have its own issues namely using steam turbine drives or trimming down impellers based on new duty point. By trimming down you will loose the flexibility to have increased flow if that is a desire in the near future, in this case you will need to keep one full size impeller as spare and though a low cost option in some cases depending on the cost of holding the spare and maintenance cost to remove and trim the impeller. The Fig 8 shows the change in performance with trimming in a tower reflux pump.

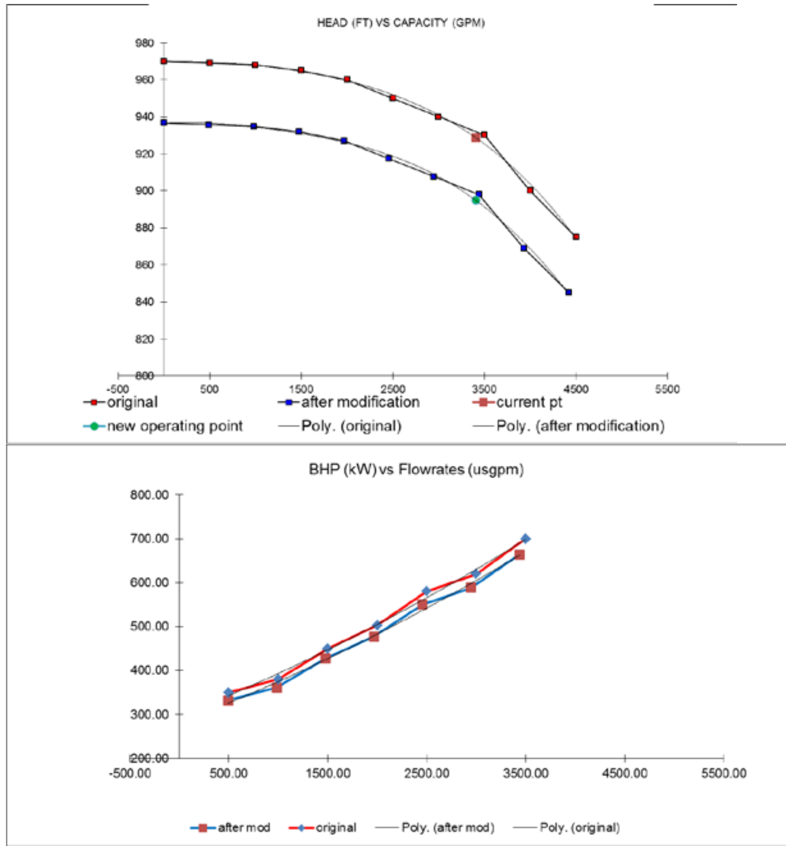


*Fig 8 : Pump curves showing difference in performance before and after impeller trimming for a tower reflux pump*

The second example (figure 9) is for a tower bottoms pump which was operating at 2965 rpm with the spill back valve open at 15% minimum. A VSD drive was installed for the motor and machine speed was reduced to about 2600 rpm. There was a power saving of over 100 KW resulting in a daily saving of close to US$500/day.

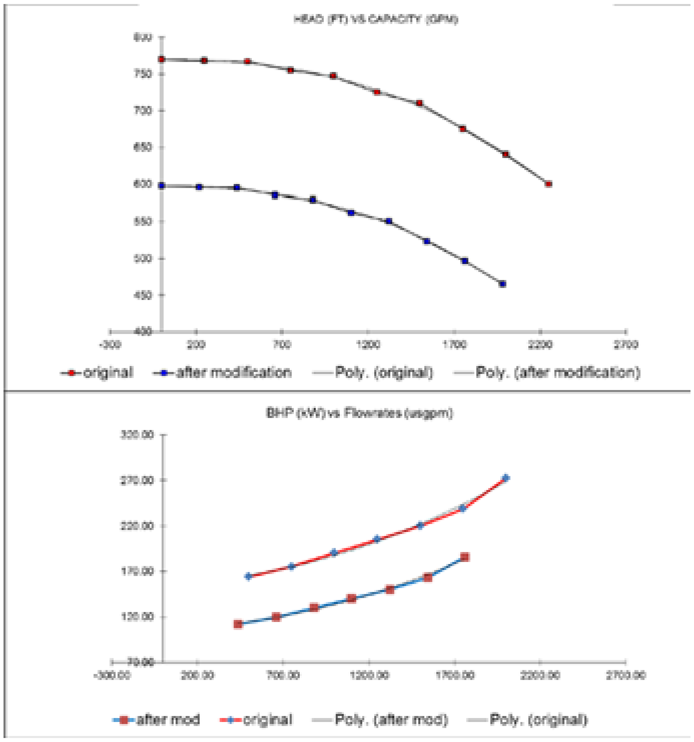


*Fig9 : Pump curves showing difference in performance before and after VSD installation for a tower bottoms pump*

Photo 6: Pictures show a pump before and after upgrade

The application of VSD is not only for process pumps but also for pumps in oil movement areas which typically could have varying flow conditions making it a challenge to select a pump with the best efficiency. In a case, VSD when applied to fuel oil pump of just 55 KW rating could save as much as US$50,000 per annum for a pair of pumps. In our case there was a significant reduction in speed from 2965 rpm to 1900 rpm.

Considerations prior to installation of VSD:
1) Minimum operating speed to ensure no overheating. Operating speed below normal operating speeds could end up in high vibration especially when one of the natural frequencies of pump casing, structural members and piping coincide. Impact tests could be conducted to identify potential resonances and look at speeds at which the unit may not be operated on long term.
2) Review pump seal drawings especially when we have plan 23 and 52 & 53 which have pumping rings or pumping features would be impacted by lower operating speeds.
3) Motor bearings could face electrical fluting due to stray currents. If this is observed consider installing grounding brush or insulated bearings.

Option 2: **Upgrade of impeller designs**

There lies an opportunity for energy saving especially if there are pumps which were supplied in the early 70 or 80s. These could be explored with the pump OEM as well as other OEMS who can rerate the pump and upgrade to latest API 610 requirements. This again results in not only a more energy efficient pump but also a more reliable pump.
The challenge with this like in most cases the justification in terms of cost of implementation vs payback. Needless to say a pump with lower efficiency and also bad actor in terms of failures would offer as an ideal candidate for this change.
The refinery has done some power frame upgrades and has explored proposals on performance enhancements but has not managed to find a cost effective solution to implement one yet. The loss in writing off existing assets also hinders the justification for these kinds of enhancements.

Option 3: **Non metallic Wear Rings**

Use of non- metallic wear rings with PEEK ( Poly Ether Ether Ketone ) or similar nature composites materials has been in use for a close to a decade now . Many new pump specs ask that as material of choice if temperature of pumping fluid permit its selection.
The anti-galling properties make them a great choice avoiding jamming in pumps. But the energy saving is achieved due to the lower clearances possible with these materials compared to conventional materials resulting in lower leakage losses at the impeller wear rings and also at pump throat bushes. The throat bush clearance controls the quantity of seal flush going to the stuffing box. In case of seal flush coming from pumps own discharge the reduced flow could be significant in some cases. Pump manufacture's & material suppliers will offer the most relevant information on which material to use and how to install them.

Case study: A 14 stage amine pump had major issues and rotor failures due to seizure at pump center bush and at pump wear rings. Material of the wear rings, center bush and throat bushes change to metal vs PEEK. (refer to photo 7&8)

*Photo 7 : Pictures show the multistage pump impeller wear ring and stationary wear ring made of PEEK material and insert.*

*Photo 8 : Pictures shows an14 stage pump stationary wear ring made of PEEK material inserts( assembly in progress)*

The typical complications are if the PEEK should be on stationary wear ring or on the rotating impeller. Our experience has been in case of end section pumps it is easier to install the nonmetallic ring on the impeller as it is easier to carry out by just removing the impeller. In case we intend to do that on the stationary part we would need to remove the casing which could complicate the installation as we would need to disconnect piping and could end up have piping stress issues .
But in case of split casing pump we realized it easier to install the PEEK on the stationary part as it would be easy to install and replace them.

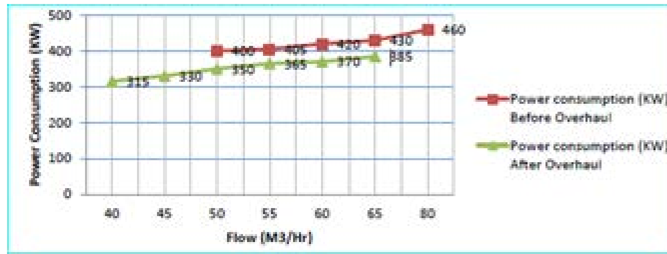


*Fig 10 : Shows the multistage pump performance pre and post overhaul.*

The only complication lies in machining the split ring to the right clearances which needs a closer evaluation.

In this case a 50 KW saving was observed on a average but some of it could be due to improved performance of the pump after an overhaul .In any case a 50% saving of the above number would also justify this change.

We have done similar changes on end suction pumps in which case we observed a marginal reduction is power and improvement in performance but the reliability improvement would add value to this change.

Option 4: **Energy Efficient Flush Plans**

Plan 21 seal flush plan is commonly used for boiler feed water pump and some hot HC services. This involves liquid from pump discharge flow through a typical 3mm orifice and into a water cooled exchanger before entering the seal chamber. A much more efficient method is using pumping ring in the seal assembly. The pumping ring recirculates the liquid in the seal chamber through a cooler and the cooled liquid from the seal cooler is moved back into the seal chamber. The enclosed system combined with the thermo-syphon effect of the cooler results in much lower heat duty to the cooler . Studies done have shown an API Plan 21Configuration in a heat transfer oil application consumes 42Kilowatts (57 horsepower), compared to 5.8 kilowatts (7.8Horsepower) for Plan 23 in the same application.

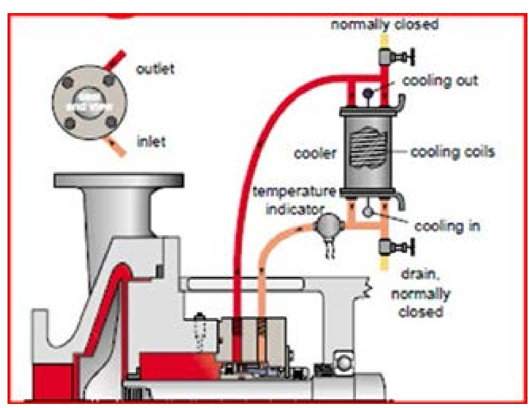


*Fig 11: A typical mechanical seal flush plan 23 layout.* (Image courtesy: Flowserve Corporation)

Option 5: **High pressure recovery turbines (HPRT)**

High pressure recovery turbines ( HPRT ) have been used in applications where we have let down from high pressure to low pressure systems typically seen in Hydroprocessing units which have a HP circuit at close to 180 barg pressure to low pressure systems at close to atmospheric conditions . Instead of using a control valve to let down the pressure it would be more energy efficient to let down using a HPRT which is a pump running in the reverse direction as a turbine.
The issues with these upgrades are in many cases of plot suitable to install these long trains. A HPRT drive consists of pump driven by motor through a gearbox on one side and a HPRT on the other end of the motor connected through a clutch. Further we would need a lubricating oil system to ensure lubrication of the entire train.

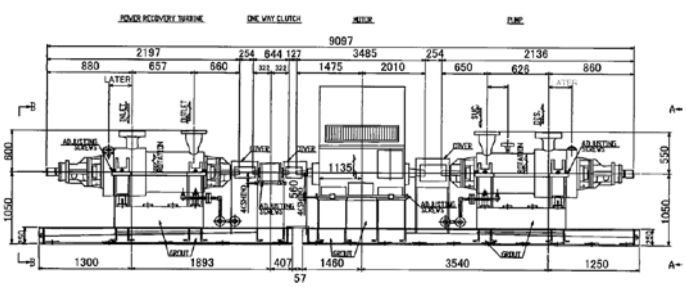

*Fig 12 : Atypical layout of HPRT coupled to a motor driven feed pump in a typical hydro processing unit.*
A feed study was done to install a HPRT system in one our hydrocracker units. Due to complications involved in the installation in terms of piping and cost of the installation of new system resulted in a payback which was not considered favourable to execute the project. The most attractive offer resulted in close to 500 KW saving but the payback of over 4 years made the justification difficult.

**Fans:**

The application of fans in refinery is in various forms and types. The centrifugal fans are found in forced draft and induced draft applications in heaters and incinerators. Cooling tower fan and fin fan are typically axial type slow speed applications.

In most cases the fans are operated at slow speeds typically using four pole motors or 8 pole motors. Capacity control is normally achieved through dampers but a more efficient energy method would be is to use variable speed drives. We have observed in many applications the fan speed could be reduced from rated speed of 1450 to 800 rpm. Operating at these speeds shows a typical energy saving of 20% and also improves reliability of these fans.

Some of the lessons learnt are to have the option to have direct start provision for the motor as in the case of Forced draft fan drive failure the whole unit trips and the LPO (lost profit opportunity) could be quite substantial. So in case issues with VSD which need longer time to fix it would be better to connect the motor for direct on line so that the LPO are reduced. Secondly as any system change operator training in critical to ensure smooth start up and stable operation.

In case of fin fans various energy saving upgrades are possible. Some could achieve higher flow for the same power consumption. This typically involves installing better profile blades with specialized honey comb tips (refer photo 9). This set up due to better aerodynamic profile perform better with lower leakage losses.

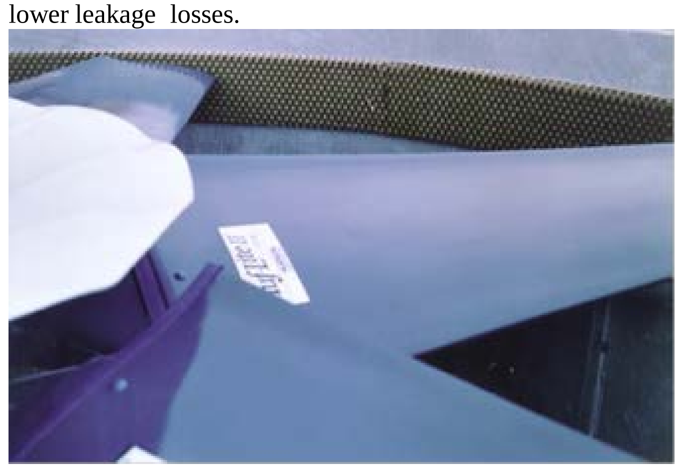

*Photo 9: A fin fan blade upgrade with tip seal installation to improve efficiency*
In a recent upgrade for a large steam turbine overhead condenser the savings documented is as below:

| | HPS Consumption vs Ambient Air Temperature & Airflow |
|---|---|
| HPS Savings (Cleaning), Ton/hr. | 1.30 |
| HPS Savings (AFC change out), Ton/hr. | 0.20 |
| HPS Savings (AFC change out), USD/yr. | 86999 |

*Fig 13: Summary of the savings achieved through fin fan blade upgrade.*
Few other alternatives for energy saving in fans consist of installing the fan directly on motor shaft and operating the motor on a VSD drive. This eliminates gearbox or belt and pulley system in addition to energy saving. This has been gaining popularity for cooling tower fan applications and too much lower extent for fin fan applications due to safety ratings issues with the motor. But things are being done to get the

approvals. Easier to justify for new installations but could be a challenge for retrofitting existing facilities as it may need to relook at the fan support structure for the motor support and installation.

## CONCLUSIONS

The tutorial highlights the potential to reduce energy consumption in various types of rotating machinery found in a typical refinery and what has been the experience and lessons learnt from these changes. Most of these require an energy saving culture in the organization to make it successful. A company desire to have a lower carbon foot print is one of the key elements for an energy saving program. The ones highlighted here are the typical applications known to the author at this point in time and there are surely many more options available which every rotating equipment engineer should explore throughout the Life Cycle of rotating equipment. New materials and technologies are being tested to further enhance the capabilities of machinery making them more efficient and reliable.

## REFERENCES

a) Pump Energy Conversation techniques by J C Cone Texas A&M Pump Symposium 1983.
b) Seal support System by Fluid Sealing Association member Jason Ferris published in pump zone magazine Aug 2013.
c) Energy Savings for Centrifugal Compressors by Doug Fisher Elliott Group.
d) Hydrocom Brochure from Hoerbiger
e) Hudson Tip Seal Manual

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