SELECTION OF THERMAL ISOALTOR FOR A PROPYLENE TRANSFER PUMP – A CASE STUDY

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Abstract –
In a Petrochemical Complex, liquid Propylene as a product is transferred by a transfer pump from storage tanks for ship loading / other required facilities. Liquid Propylene has a boiling point of minus 47.7 degree C, hence operating fluid must be kept below minus 47.7 degree C during complete pumping operation and storage.

To protect concrete from such a cold temperature exposure a thermal isolator was selected. In addition, isolator was supposed to tolerate pump dead weight, provide proper anchorage during dynamic motion, shall not fail even at high stresses generated during a trip due high vibration or a short circuit. Most importantly, it was supposed to isolate high vibration and dynamic forces thus ensuring smooth operation of other pumps of similar type. The case study provides outline how such objectives were achieved by in-house detailed engineering and calculations.

Key words - API VS6 pump, unbalance force, transmissibility, isolation, cold transfer

Introduction – In a Petrochemical Complex, Propylene as a product, is transferred by a transfer pump from storage tanks. Liquid Propylene has a boiling point of minus 47.7 deg C, hence operating fluid must be kept below minus 47.7 deg C.

A VS6 (a double casing diffuser type API 610 nomenclature) Single suction pump is normally selected for the purpose. VS6 type pump can create its own NPSHa using its can length. VS6 type pumps are quite different from other pumps because they have more flexible motor and pump discharge casings than comparable horizontal pumps and a more flexible attachment of these casings to the foundation. This is in addition to having very long line-shafting that connects the motor to the belowground liquid-end pump bowl assembly. Because of this unique construction, vertical turbine pumps (VS6) are more prone to structural and shaft vibration problems This type of pumps have a major usage in hydrocarbon industries.

In such a low temperature operation, “cold flow “from pump base plate to concrete can cause cracks and damage to foundation. Use of Epoxy grout between base frame and concrete foundation becomes a borderline case. The reason was high ambient temperature and very low operating fluid temperature. Subject pump which was transfer liquid propylene was to be fixed on concrete foundation, wet portion of pump in pit below the ground level. Pit wall opening was increased to take care of insulation requirement of can to prevent cold flow / increasing flowing fluid temperature. Refer fig 1

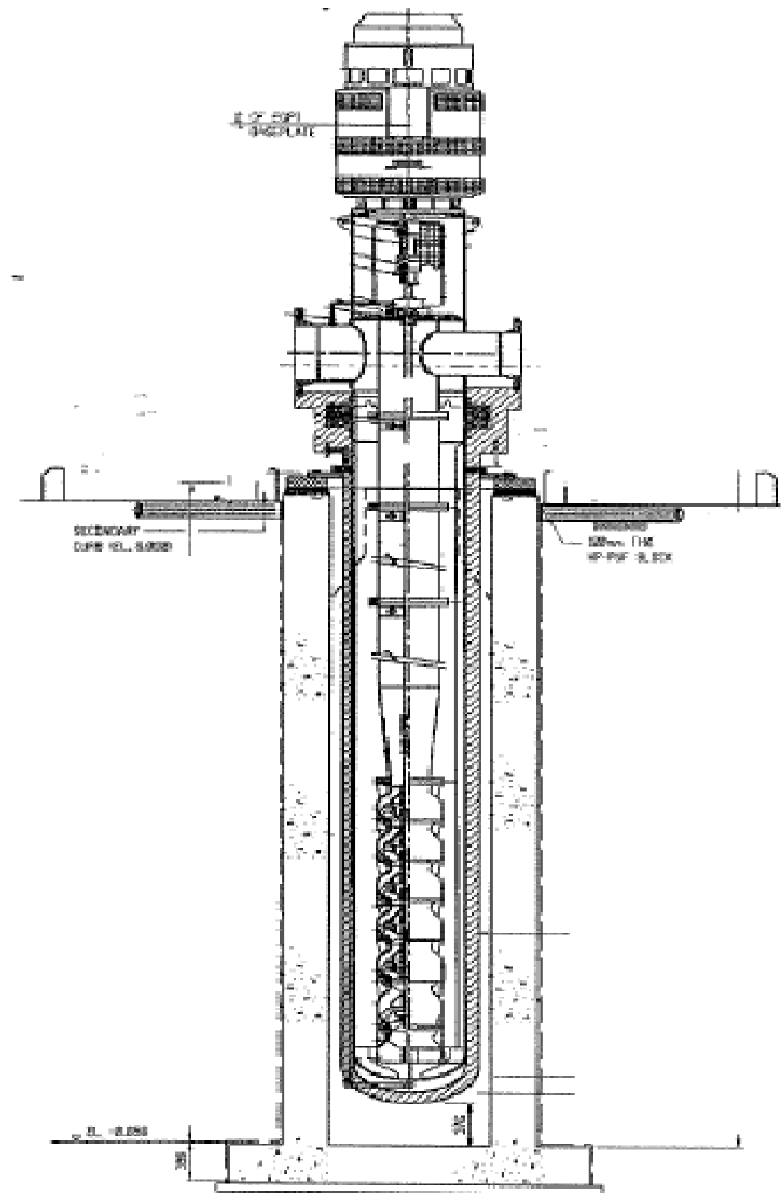


Fig 1 -Installation arrangement of pump under case study

Since the existing base plate of pump was undersized with respect to new pit wall square opening, an additional sole plate of 1¾ inch thick of SS 316L was considered. Sole plates were stiffened at 8 places for better stability in case of swaying of pump during operation. Number of Anchor bolts increased from four to eight to ensure better dynamic stability utilizing added sole plate area of coverage on foundation. Refer Fig 2 for sole plate details.

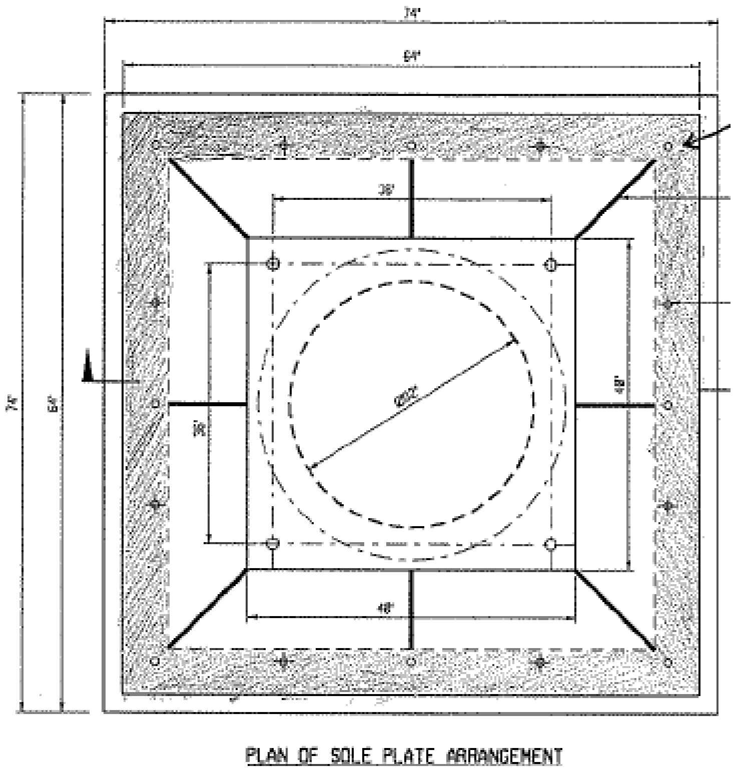


Fig 2 –Sole plate [shaded area of sole plate shown where Isolator was to be placed]

The challenge was to select a thermal isolator between metallic base frame of pump and concrete. The isolator was to withstand all dynamic forces acting during pump operation even considering worst operating scenarios as well which means it shall act as vibration isolator as well to protect concrete and nearby pump installation. An isolator is a resilient support which decouples an object from steady state or forced vibration. To reduce the transmitted vibration, isolators in the form of springs are used. Common springs used are pneumatic, steel coil, rubber (elastomeric) and other pad materials.

To avoid major modification in already laid piping at site, the target of vibration isolator thickness was kept 2 inches or below. The target vibration isolation was kept more than 80%. The anchor bolts were also to be isolated with sleeves made of same material to eliminate any metal to metal contact.

Subject Pump data - The subject VS6 type pump operating details are as follows –

Rated Flow – 3965 US GPM, Pump RPM- 1800, Power -500HP, Minimum Continuous Flow – 2500 USGPM. Impeller dia – 13.4inch No of stages- 6, Discharge pressure Pd =153.6 psig , suction pressure – Ps – 3.7 psig ,motor and pump directly coupled and total wt of system- 20000 lb. Pumping fluid temperature – minus 54 deg F. Balance quality of machine - Gr 2.5 ISO 1940, Concrete- C 40 grade concrete . Vibration acceptance criteria – ISO 10816-3

The scheme was to arrange the thermal isolator in such a way that there is no cold transfer from metal parts to concrete. To have this arrangement from drawing board to site execution various vendors were contacted and inhouse assessment of the proven features and design were carried out. Then the chosen type of thermal isolator was to be reviewed in terms of vibration isolation also.

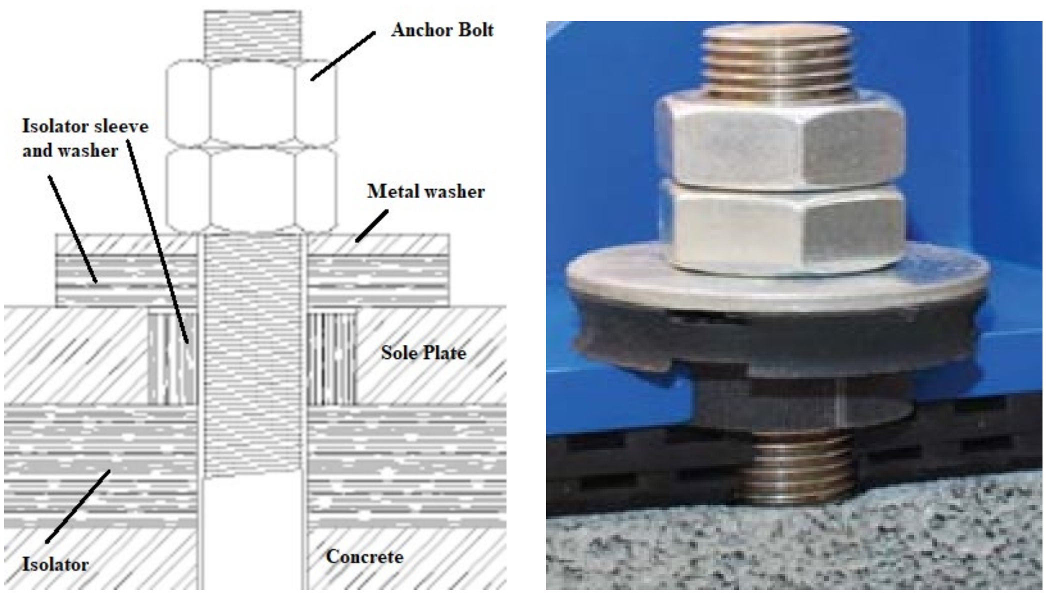


Fig 3 Cross sectional arrangement showing Isolator Pad, Isolator sleeve and washer

First objective was to calculate major forces those shall act on machine foundation and second objective was to select a suitable thermal isolator which could withstand pump dead weight, provide proper anchorage during dynamic motion, shall not fail even at high stresses generated during a trip due high vibration or a short circuit. Most importantly, it was supposed to isolate vibration and dynamic forces and cold flow to concrete thus ensuring smooth operation of other pumps of close vicinity.

A. Downward Axial thrust & Dead Load –
Downward axial thrust was calculated by vector summation of downward forces due to discharge pressure and upward forces due to suction pressure as shown below with 6 number in line impeller arrangement.

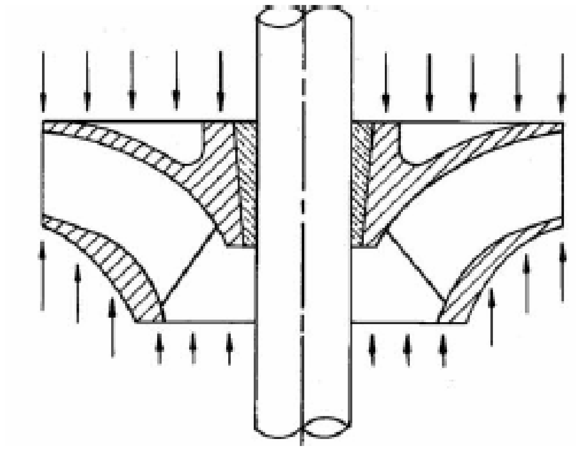


Fig -4 – Direction of forces acting on individual impeller in VS6 type pump.

Note - Since all data supplied by Isolator Supplier was in Imperial Units, all results are shown in Imperial units.

1a. Hydraulic Axial thrust (downward) calculated using discharge pressure, shroud area, suction pressure and intake area and number of stages as 114400 lbf
1b. Hydraulic Axial thrust (downward) calculated using Darfur and Nelson equations as **8190** lbf

Higher among 1a and 1b was considered -
2. Dead wt of pump & motor including piping volume was calculated as **24300** lbf

Note -Shaft thrust (upward) & sleeve thrust (upward) considered as ZERO effect on compressing force on isolator.

Hence Total Thrust was calculated as @ 114400+24300 = **138700-pound force**

Axial stress on bolts = [(Maximum bowl pressure *discharge flow area) + (bearing load) + bolt preload * no. of bolts] / (no. of bolts *effective cross-sectional area of bolt in perpendicular plane of snub tightening)

B. Forces generated when vibration reaches trip value.

An unbalance will cause vibration in rotor bearing system which shall be transferred by antifriction bearing to pump casing and subsequently to foundation bolt and isolator pads. To simulate worst case scenario, damping is not considered at all.
The radial component 1x vibration will act as force of shear to foundation bolts and isolator. Hence isolator as well should strong enough to avoid damage by such forces.
Total line shaft weight -231 lb.
Impeller assly weight – 272+85 = 357 lbs.
Coupling weight – 100 lbs.
Motor rotor weight – 7787/3 = 2629 lbs.

1/3 rd of Motor wt. is taken because coupled with a flexible coupling motor unbalance does not really affect the pump
Total rotor wt – **3317** lbs
RPM – 1800, 30 Hz
Two different scenarios were selected for disturbing forces –
Forces caused by unbalance in rotating parts –
F = 22.5*6.015*(Balance grade as per ISO) * Rotor weight / RPM. Normal balance grade is 6.3 however Balance grade 16 was selected as conservative approach.
With balance grade 16, F= 3600 lbf
Forces generated at the event of trip -
A higher than trip value (10 mm/ sec RMS) of vibration was set as. 15mm/ sec RMS (0.6 inch / sec. RMS) to analyze the integrity of thermal isolator. This was because rotor system becomes less stiff due to introduction of isolator and thus reacts more to unbalance. We assume that there is only one peak in complete spectrum and that is at 1X. (Note -For simplicity of calculation, true sine waves are considered.)
Referring the conversion table of Vibration Severity Chart as per ISO 10816-3, the calculated peak to peak displacement in inches i.e.
D = (19100 / 1800) * 0.6*2√2 = 9.55 mils pk-pk as per chart below -

CONVERSION FORMULAS FOR VIBRATION UNITS

$$D = \left(\frac{19100}{F}\right) V$$

$$D = \left(\frac{8383}{F}\right)^2 A$$

$$V = \left(\frac{F}{19100}\right) D$$

$$V = \left(\frac{3687}{F}\right) A$$

$$A = \left(\frac{F}{8383}\right)^2 D$$

$$A = \left(\frac{F}{3687}\right) V$$

$$G\text{'s} = 0.0142 \left(\frac{F}{1000}\right)^2$$

$$OZ\text{-}IN = D\left(\frac{W}{125}\right)$$

$$OZ\text{-}IN = G\,\frac{(0.563W)}{\left(\frac{RPM}{1000}\right)^2}$$

$$\#FORCE = 1.79\ OZ\text{-}IN \left(\frac{RPM}{1000}\right)^2$$

$$C.F. = (W)(.00034\,F^2R)$$

D = PEAK TO PEAK DISPLACEMENT - (mils)
1 mil = 0.001 inch
V = PEAK VELOCITY -inches/second
A = PEAK ACCELERATION -G's
1 G = ACCELERATION DUE TO GRAVITY = 386.087 inches/second
F = FREQUENCY - CYCLES PER MINUTE

OZ-IN = THE UNBALANCE IN OUNCE-INCHES
(1 OZ AT 1 IN RADIUS = 1 OZ-IN
1 OZ AT 10 IN RADIUS = 10 OZ-IN)
W = WEIGHT OF ROTATING ELEMENT IN POUNDS
C.F. = CENTRIFUGAL FORCE
R = RADIUS IN FEET

Fig -5 Vibration Conversion Chart

Equivalent oz-inch = 9.55 * (3317 / 125) =253.41 oz-inch
Centrifugal Force due to Rotation= 3317 * 0.00034*1.1 *30*30 = 1030.002 lbf
Unbalance Force = 1.79*253.41*1.8*1.8 = **1469.67** lbf
Total force = @ **2500 lbf**

ISOLATOR PAD SELECTION BASIS

Type of Isolator was selected keeping very important things in mind –

1. Selected Isolator dynamic natural frequency was to be less than half of lowest disturbing frequency.
2. It shall withstand all dynamic and static forces encountered during pump operation for a prolonged duration as calculated above.
3. It should not de rate by weathering and cold. It maintains its property of isolation in all ranges of ambient temperature required in Pump data sheet.
4. It should comply to ASTM2000 D-04

Size selected for isolator pad which shall be placed underneath sole plate – 64”x50”x2 “thick as shown in shaded area in Fig.2

Selection Procedure – To start with, all information was passed on to prospective suppliers of Vibration Isolators with a drawing (as shown in fig 1 and 2). Response were solicited with all back up calculations with temp.-stiffness curve, dynamic spring rate, transmissibility curve, mech/ physical properties.

Vibration Isolation analysis -

Total wt of unit including piping load, liquid wt etc. = 24249 lbf
RPM – 1800, Lowest Forcing / disturbing frequency Ff= 1800 / 60 = 30 Hz (1800/60)
The dynamic natural frequency of selected type of isolator pads Fn= 9 Hz.
$Ff / Fn = \sqrt{(1 + T) / T}$
T is Transmissibility.
If the simple system is subjected to Forced Vibration at frequency f, and sinusoidal foundation motion at amplitude x, the absolute value of the mass response amplitude y expressed as a ratio $|y/x|$, also known as Transmissibility T. In simple words, Transmissibility is a measurement used in the classification of materials for vibration management characteristics. It is a ratio of the vibrational force being measured in a system to the vibrational force entering a system. If a vibration isolator pad has a transmissibility of 11.64%, it means that 11.64% of the vibrating force is being transmitted through the pad.
Isolation = 100- Transmissibility. considering the system as linear system.
Isolation = 89.36 % from transmissibility chart which served the purpose of selection.

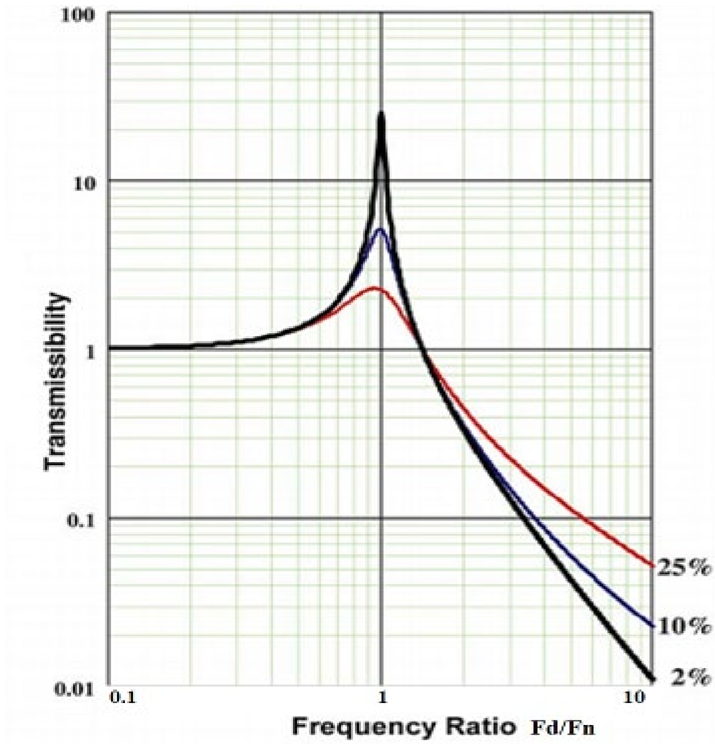


Fig 6-Transmissibility curve

Compressive stress on isolator pad (having full coverage as shown in Fig
Total Static Load / Isolator Coverage area = 138649 lbs. / 1392 Sq inches
= @99 psi < design value of 120 psi.
The isolator performance was also cross checked with Load Deflection graph which was available in Isolator pad Technical Information sheet. The relation between natural frequency and static deflection of a linear, single-degree-of freedom system is shown below.

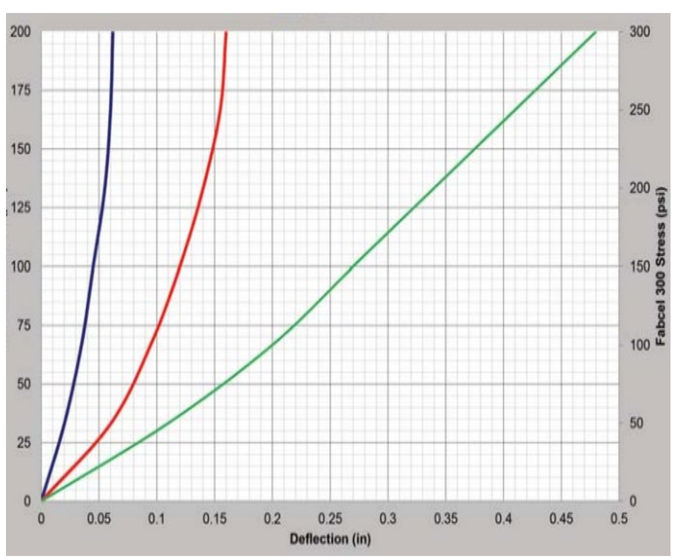


Fig 7- Load deflection diagram

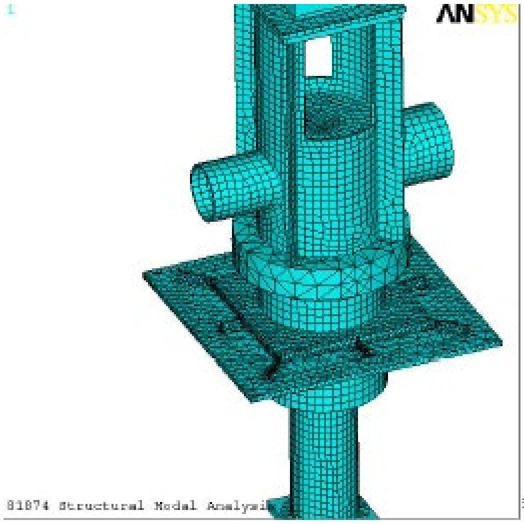
Thermal isolation
Heat transfer coefficient of Isolator material k = 0.274 W / m deg C
= 0.23 kcal / m deg C. Note -The thermal conductivity calculation done in house excel sheet using SI units.
Based on heat transfer coefficient of Isolator and pump constructional details, a cold flow calculation was performed. Results showed that most of the cold shall pass towards open face of sole plate. The "Cold "transfer calculation showed that a mere 1.2 mm thick isolator shall start resisting cold flow.

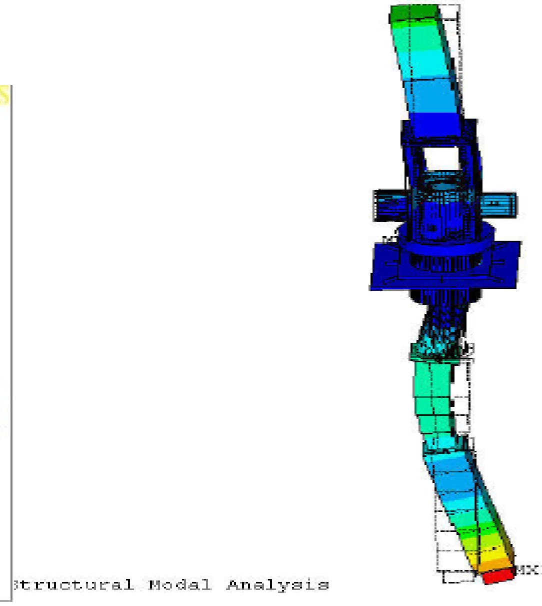
Heat transfer calculation table

Minimum allowable temperature of concrete	0 deg C
Pumping temperature	- 47 dig
Can inside radius r1	400 mm
Can outside radius r2	406.4 mm
Can insulated radius r3	440 mm
K conductivity of can W/m deg C	19
Total Cold Flow	-353479 W.
Actual band width of pump CAN transferring the Cold	0.035 m
Actual cold flow	-12371.74 W
KW	-12.371
Kcal / hr.	-10639.89

A – area of sole plate in contact with isolation pad -1.029m2.
Q/A = T1-T2 (t/k), t = @1.025 mm. The actual pad thickness was 50 mm so it will only permit 195kcal / hr. of coldness into concrete.

Forces generated due to high Vibration
When high vibration is occurring, the radial forces is transmitted to pump body and base plate which in turn transmitted to foundation bolts in a particular side.
This acts as Shear force applied in isolator sleeve / washer.
Shear stress on these bolts as calculated was far less than design value suggested in supplier catalog.
FEA analysis by Pump Supplier for Wet structural natural frequency –
Some iteration was to be done in finalizing the level of constraints for isolators in FE modeling of system.
The analysis conducted by Pump Supplier showed first six structural natural frequencies are well separated for 1x synchronous speed excitation. FEA Model and sample Operational Defection shape shown in fig.





FEA Model **3rd Below Grade Mode 23 Hz.**

Fig -8 – Structural Natural Frequency analysis by FE software.

Besides all above studies for isolator selection, utmost care was taken to install the pump with Isolator pads in presence of Pump installation expert, Isolator application engineer and E&C Rotating equipment engineer.

The pump was commissioned and was put to operation in year 2007. Since then, it has been in operation without any reported trouble.

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