EFFECTS OF A BEAM MOUNTED GAS COMPRESSOR ON A CONVENTIONAL PUMPING UNIT

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This paper analyzes the effects on reducer torque and structural loading of a beam mounted gas compressor on a Conventional pumping unit.

Introduction

When a column of gas forms in the casing above the annular fluid level, casing pressure is developed. This pressure can restrict the producing formation and thus limit the amount of oil and gas the well can produce.

To relieve this pressure and increase production, beam mounted gas compressors (BMGC) are used. See Figure 1. These compressors draw gas from the casing, compress it and then release it into the flow line. On a conventional pumping unit they are attached to the pumping unit walking beam between either the center and equalizer bearings, or the center bearing and horsehead, and are either single or double-acting. The single-acting compressor draws gas from the casing on the upstroke of the pumping unit, then compresses and releases it on the downstroke. The double-acting has two reservoirs that enable it to draw gas in one, and compress and release gas in the other. This cycle takes place on both the upstroke and the downstroke of the pumping unit, thus moving twice the volume of gas as the single-acting.

With the pumping unit supplying the necessary energy to operate the BMGC, the walking beam is inevitable subjected to additional loading. The effects of these loads are additional reducer torque and structural bearing loads, which depending on the conditions, may or may not be adverse. This paper analyzes these loads and the effects they have by reviewing how the loads are determined and then analyzing an actual case.

Since the effects of a single-acting compressor are equivalent to the downstroke effects of the double-acting compressor, the remainder of this paper only considers a double-acting compressor, mounted between the equalizer and center bearing.

Beam Mounted Gas Compressors

Though stroke lengths, piston diameters, and discharge pressures may vary, some typical sizes are 8, 10, and 12 in. diameters with maximum discharge pressures of 200, 150, and 100 P.S.I., respectively. All of which are available with a 24, 36, or 48 in. stroke. With the respective sizes and maximum pressures, the beam compressor is capable of creating somewhat large forces that act on the pumping unit's walking beam. The peak forces created from the pressure in the cylinder can very well exceed 10,000 lbs., with the peaks generally occurring near the end of the compressor and pumping unit stroke, which in most cases keep the additional torque load to a minimum.

To better understand how the loads originate and also the operation of the compressor, a BMGC's cycle can be categorized into four basic steps as can be seen in Figure 2. Note that the double-acting compressor shown has two reservoirs with a suction and discharge line running to each.

The first step which occurs during the upstroke of the pumping unit, has one reservoir drawing in gas from the casing while the other is compressing gas. Step II is at or near the top of the stroke when the gas being compressed reaches a pressure greater than the line pressure thus allowing it to be released into a discharge line. At the same time the other reservoir has completed filling. Step III, during the downstroke of the pumping unit, has the piston reversing direction and as a result the reservoir that just completed compressing and releasing its gas, begins refilling while the reservoir that has filled begins compressing gas. Finally, step IV occurs at the bottom of the stroke with one reservoir releasing and the other filling to complete the cycle. Throughout each pumping unit stroke this cycle is repeated.

The result of the varying pressures in the compressor is a varying force on the walking beam at the point where the BMGC is attached. To determine the pressure gradient either a pressure transducer can be installed and the pressure recorded during one cycle or it can be calculated.

In order to calculate it, a mathematical relationship is needed. By assuming the gas from the casing behaves as an ideal gas, the following ideal gas relationship can be applied to the BMGC's system:

$$P_N(V_N)^K = \text{constant} = C \ldots\ldots\ldots\ldots\ldots (1)$$

Where for any position "N" during the compressor stroke, the value of the pressure multiplied by the volume raised to the K power is equal to a constant value. "K" is the specific heat ratio. To determine this constant for a particular unit, the suction pressure, discharge pressure, compressor stroke length, and at which point in the stroke the gas is released must be known. The suction and discharge pressure can be found from pressure gages located at the casing and at the flowline into which the gas is released. The stroke length can be found by finding the difference in the compressor lengths from when it is fully extended to when it is completely retracted. To determine where along the stroke the gas is released, pressure gages must be monitored at the discharge line to see when the flowline pressure is reached, because at this point is when the check valve is opened and the gas released, also it must be noted where in the stroke this happens. Typically it is 50 to 100% into the stroke. With this information the following equation can be used to determine the "PV" value.

$$C = P_D P_S \left(\frac{S \times D^2 \times \pi \times .0025}{P_D^{1/K} - P_S^{1/K}}\right)^K$$

C = Constant

Where Pd = Discharge/Flowline Pressure, PSI

Ps = Suction/Casing Pressure, PSI
S = Compressor Stroke, In.
% = Percent into the stroke the gas is released
D = Piston Diameter, In.
K = 1.26 For Methane

From C it is possible to determine the compressor volumes at the beginning and end of the compression cycle from the following:

$$V_S = \left(\frac{C}{P_S}\right)^{1/K} \quad \ldots\ldots(4)$$

$$V_D = \left(\frac{C}{P_D}\right)^{1/K} \quad \ldots\ldots(3)$$

Where Vd = Volume at discharge, In.^{3}
Vs = Volume at beginning of cycle, In.^{3}

For an analysis of the pressure variation at points between the beginning and end the following equation can be used during the upstroke:

$$P_N = \frac{C}{(V_S - (PRP_N \times \pi \times D^2 \times S \times .25))^K} \quad \ldots\ldots(5)$$

During the downstroke the following equation is used:

$$P_N = \frac{C}{(V_S - ((1 - PRP) \times \pi \times D^2 \times S \times .25))^K} \quad \ldots\ldots(6)$$

Where P_N = Pressure in cylinder, PSI

PRP_N = Polished rod position

Figure 3 is a plot of the pressure during one cycle. Whenever P_N is larger than the discharge pressure, it should be used and not the calculated value.

Since the BMGC being analyzed is double-acting, the piston has pressure on its backside opposite the side compressing the gas. From Figure 5 it can be seen that the net reaction on the piston due to the pressure is the difference in the compression reservoir pressure and the suction reservoir pressure. The resulting force is:

$$R = P_{NET} \times \pi \times D^2 \times .25 \quad \ldots\ldots(7)$$

Where R = Force on walking beam, Lbs.

P_{NET} = Net pressure on piston, PSI

To be able to relate this force to more practical terms it can be converted to an equivalent polished rod load by using the following equation:

$$E = \frac{R \times Q}{A} \quad \ldots\ldots(8)$$

Where E = Equivalent Polished rod load, Lbs.

R = Force on walking beam due to compression, Lbs.

Q = Perpendicular distance from force (R) to center bearing

A = Front working center of pumping unit

See Figure 5.

If the equivalent polished rod loads were plotted versus the position, they would increase proportionally as they did in the plot of the pressure in Figure 3. This leads to the analysis of the effects these forces have on the pumping unit.

Pumping Unit Loading

Since a double-acting BMGC is doing work on both the upstroke and downstroke there is always a load on the walking beam. With the mounting between the center and equalizer bearings, the force, as a result of the net pressure on the piston, during the upstroke is upward tending to push on the beam. The effect is a moment on the walking beam about the center bearing in the same direction as a moment applied by a well load.

During the downstroke of the pumping unit, the force is in the direction opposite that of the upstroke. The tendency is for the compressor to pull down on the walking beam creating a moment in the direction opposite the moment created by the well load.

In effect, the upstroke loading appears as an increased well load to the reducer and on the downstroke a decreased well load. With the equivalent polished rod loads for incremental positions of the pumping unit cycle, the next step is to relate these loads to reducer torque.

Polished rod load is typically converted to reducer torque by using the following equation.

$$\text{Torque } (T_N) = (W - SU) \times TF \text{...............} (9)$$

T_N = Torque on reducer, In-Lbs.

W = Polished rod load, Lbs.

SU = Structural unbalance of pumping unit, Lbs.

TF = Torque factor of unit at the position being calculated.

Generally, well loads are taken from dynamometer cards at 15° increments, and with the use of the torque equation are converted to torque. However, in order to account for the torque caused by the BMGC, the equivalent polished rod loads from the compressor must be added to the dynamometer card loads before the torque equation can be used. It is a misconception that a dynamometer card alone is adequate for the torque analysis, because the card doesn't include the effects of the BMGC but only contains well loads.

Then to find the net reducer torque the counterbalance torque must be accounted for by using the following equation:

$$T_{NET} = T_N - CBT_{90} * \mathrm{SIN}\ (\theta) \quad \ldots\ldots\ldots\ldots\ldots (10)$$

T_{NET} = Net reducer torque, In-Lbs.

CBT_{90} = Counterbalance torque at 90°, In-Lbs.

θ = Crank angle

To determine whether or not the reducer is overloaded, the torque should be calculated at 15° crank angle increments and the peak compared to the reducer rating.

The loading on the structure, mainly the center bearing, tends to decrease on the upstroke of the unit and increase on the downstroke. This is a result of the upward force applied by the compressor partially negating the pitman and well loads on the upstroke and adding to these loads on the downstroke.

Unit Analysis

A case was analyzed where a beam mounted gas compressor was installed on a C-456D-256-120 pumping unit. The compressor was double-acting with an 8 in. cylinder and was mounted on the walking beam between the center and equalizer bearings.

The gas was being drawn from the casing at a pressure of 114.92 PSI and released from the compressor into a flowline of 140 PSI. This discharge pressure was reached at approximately 90% of the compressor stroke. The measured length of the compressor from top connection to bottom at the top and bottom of the stroke was 104.2" and 162.2" respectively, resulting in net compressor stroke of 58".

From the pressure and volume information a PV^K constant was calculated using equation (2) to be 382786 and with the use of equations (3) and (4) the volume at discharge is 313 in² and at the beginning of the cycle 2937 in².

With the use of equations (5), (6), (7), and (8), and the aid of a computer the pressures, loads created by the pressures, and equivalent polished rod loads were calculated and tabulated in Table 1. As noted from the table, the most severe loading occurs near the end of both the upstroke and downstroke from 135° to 180° and from 315° to 360°. In relationship to a dynamometer card, Figure 7 is a card taken from this installation, while Figure 8 depicts a card that takes into account the equivalent polished rod loads from the compressor. As one can see, the changes are primarily in the upper right and lower left portions of the card. Table 2 contains the results of a torque analysis of both cards with the unit properly balanced in both instances. In this case the increase in torque is virtually negligible with only an increase of 5% from 326457 in-lbs to 342977 in-lbs. The amount of counterbalance torque actually decreased from 493200 in-lbs to 482600 in-lbs.

By taking the analysis one step further and analyzing the required motor horsepower, we find that the requirements actually decrease. Without the BMGC, 22.5 horsepower is required and with the compressor 18.0 is required. This happens as a result of a decrease in the cyclic load factor which is directly proportional to the horsepower.

By again using a computer, the structural bearing loads were calculated and tabulated in Table 3. The loads on the upstroke are decreased slightly whereas the downstroke loads increased. However, in this particular case the loading differences are negligible.

Summary

Whether or not the loading from the installation of a BMGC is detrimental to the pumping unit, can only be determined by a load and torque analysis of each unit. By doing so, variables that dictate the loads such as compressor size, where it is mounted, and the suction and discharge pressures can be accounted for and a thorough analysis made. As one can conclude, the loads tend to follow similar patterns. They become large near the end of the pumping unit stroke, which at this point very little torque can be transmitted to the reducer. Also, the loads on the walking beam are in a direction that make them negligible to the center bearing, and in the case study actually decreased the peak load.

Assuming the pumping unit is not fully loaded prior to the BMGC installation, it is very unlikely the additional loads due to the installation will produce any damaging effects; however, this should be verified by a complete analysis since each case is different.

Table 1
Loads Created by the BMGC

CRANK ANGLE (degrees)	POLISHED ROD POSITION	CYLINDER PRESSURE (psi)	* FORCE ON WALKING BEAM (lbs)	* EQUIVALENT POLISHED ROD LOAD (lbs)
0	0.000	140	-6287	-2946
15	0.017	17	89	42
30	0.079	18	160	76
45	0.181	21	303	146
60	0.306	26	551	270
75	0.440	34	964	476
90	0.569	48	1657	814
105	0.686	72	2861	1384
120	0.785	117	5117	2421
135	0.867	140	6287	2898
150	0.930	140	6287	2822
165	0.975	140	6287	2761
180	0.998	140	6287	2726
195	0.996	16	-76	-33
210	0.965	17	-106	-47
225	0.904	18	-173	-79
240	0.814	21	-296	-139
255	0.701	25	-507	-244
270	0.573	32	-865	-425
285	0.440	45	-1503	-742
300	0.310	70	-2748	-1349
315	0.193	126	-5581	-2706
330	0.096	140	-6287	-3003
345	0.029	140	-6287	-2964
360	0.000	140	-6287	-2946

* A positive number indicates a reaction equivalent to a well load.
A negative (-) number indicates a reaction opposite the direction of a well load.

Table 2
Reducer Torque Comparison

CRANK ANGLE (degrees)	POLISHED ROD POSITION	REDUCER TORQUE WITHOUT BMGC (in-lbs)	REDUCER TORQUE WITH BMGC (in-lbs)
0	0.000	-21649	-13287
15	0.017	40918	44701
30	0.079	154130	162996
45	0.181	326457	342977
60	0.306	173445	201115
75	0.440	176626	219166
90	0.569	131931	193246
105	0.686	-3825	80983
120	0.785	-74835	32599
135	0.867	-60858	42549
150	0.930	-36967	38947
165	0.975	-9471	38689
180	0.998	42419	57426
195	0.996	77866	75172
210	0.965	97158	92519
225	0.904	175864	170788
240	0.814	291984	289226
255	0.701	160470	164273
270	0.573	326246	342684
285	0.440	228229	266202
300	0.310	93264	167293
315	0.193	108329	246514
330	0.096	24592	143309
345	0.029	-30133	34330
360	0.000	-21649	-13287

Table 3
Center Bearing Loading Comparison

CRANK ANGLE (degrees)	POLISHED ROD POSITION	CENTER BEARING LOADS WITHOUT BMGC (lbs)	CENTER BEARING LOADS WITH BMGC (lbs)
0	0.000	13891	20873
15	0.017	22544	22518
30	0.079	25526	25485
45	0.181	30644	30567
60	0.306	23983	23845
75	0.440	25505	25252
90	0.569	26052	25584
105	0.686	22603	21733
120	0.785	19772	18129
135	0.867	20348	18242
150	0.930	20360	18223
165	0.975	18927	16845
180	0.998	20139	18204
195	0.996	20249	20269
210	0.965	18788	18805
225	0.904	11852	11851
240	0.814	5917	5850
255	0.701	7143	6853
270	0.573	2119	1357
285	0.440	2308	1307
300	0.310	5826	6007
315	0.193	7532	10263
330	0.096	11537	13945
345	0.029	16400	18454
360	0.000	18891	20873

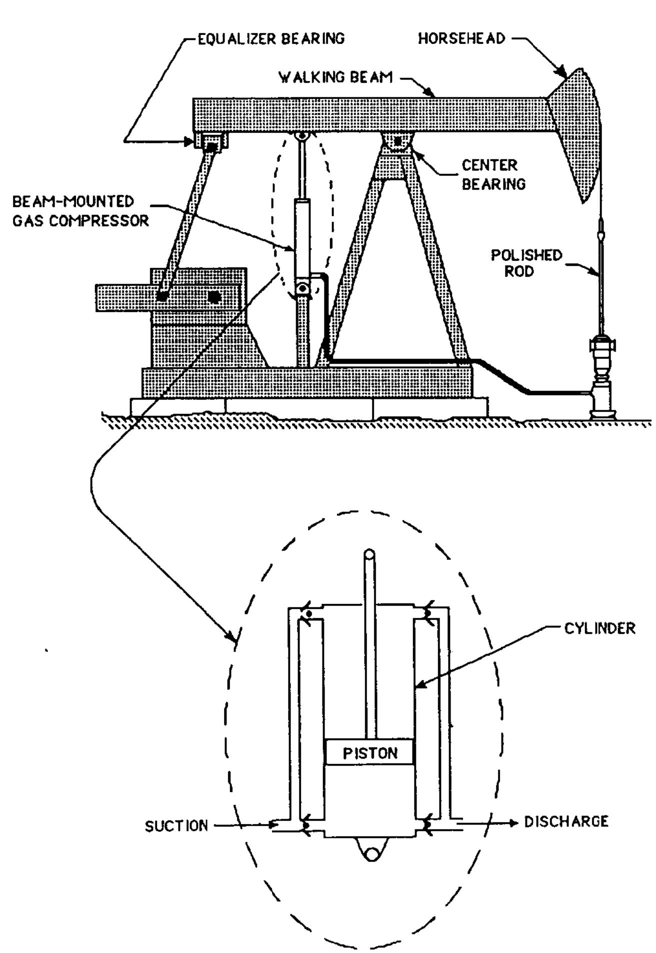

Figure 1 - BMGC mounted on a conventional pumping unit

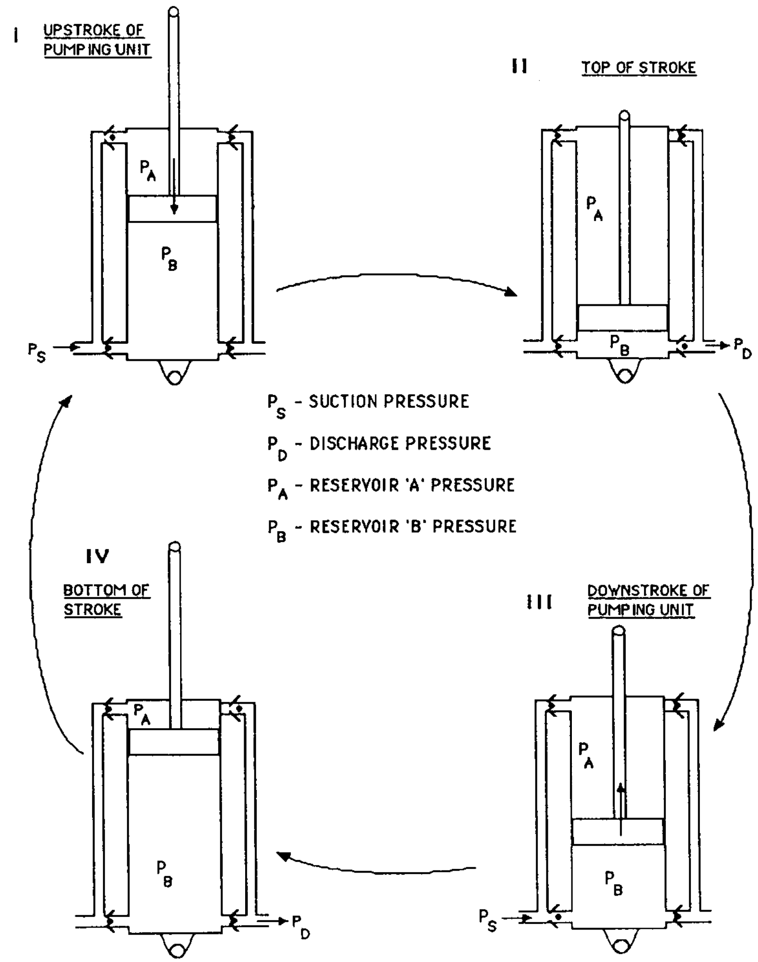

Figure 2 - BMGC cycle

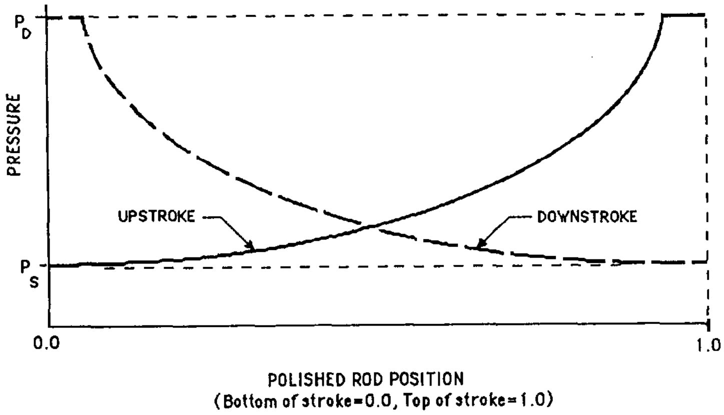

Figure 3 - BMGC pressure vs polished rod position

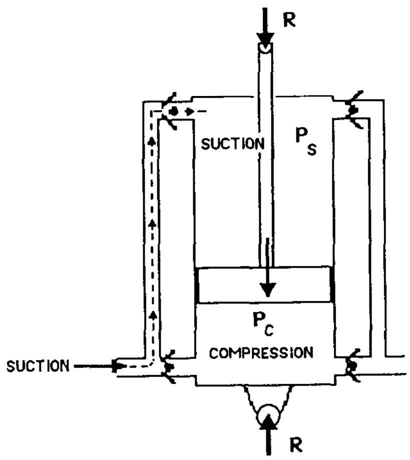

Figure 4 - Loading created by the BMGC

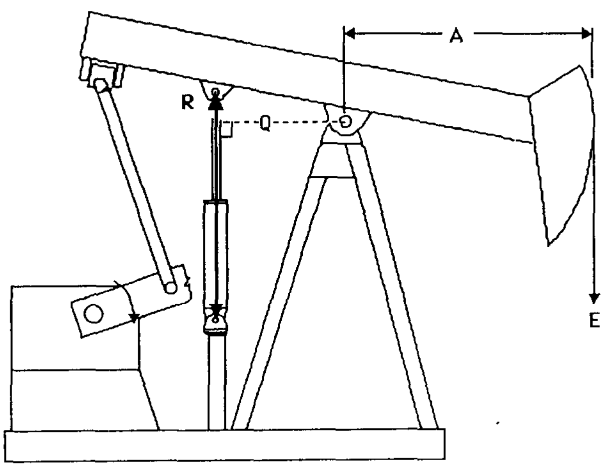

Figure 5 - Pumping unit loading

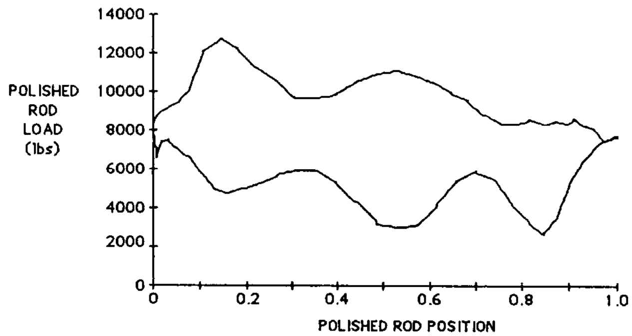

Figure 6 - Dynamometer card without consideration of BMGC

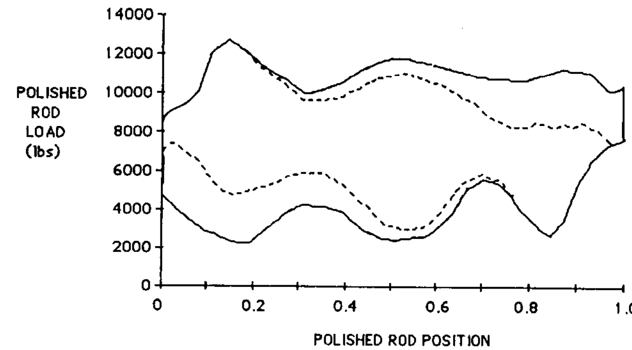

Figure 7 - Dynamometer card containing BMGC loads

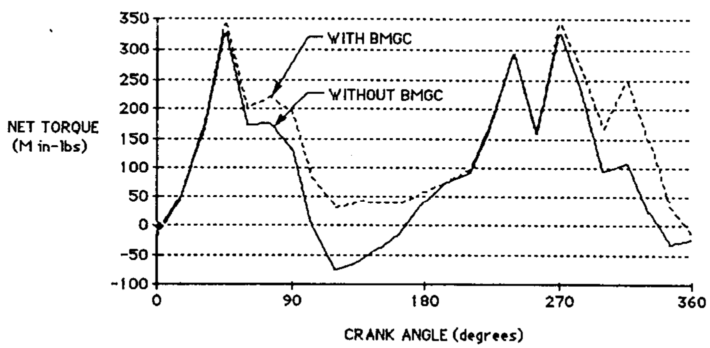

Figure 8 - Net reducer torque with and without the BMGC

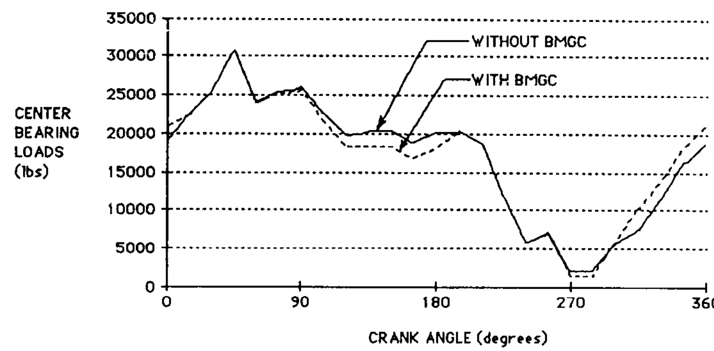

Figure 9 - Center bearing loading with and without the BMGC