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21 September 1990

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To: University of Houston
Civil and Environmental Engineering Department

Attn: Dr. Michael O'Neill, Chairman and Professor
Mr. Reda Moulai, Research Assistant

Ref: *Corps of Engineers Project for Vibratory Wave Equation Analysis*

It was a pleasure speaking with you over the telephone today.

First, below is the basic data for our major vibratory hammers

Size	M	m	e	w	k	f	N
1150	2950	328.6	3.5	2200	2400	1600	142.6
2300	4300	657.2	3.5	2200	2400	1600	239.5
2300L	4000	657.2	3.5	2900	2400	1600	239.5
4600	6400	1314.4	3.5	5600	4800	1600	479.1
1400	2600	159.1	3.84	2000	2400	2400	127.6
2800	4050	318.2	3.84	2900	2400	2400	203.6

For the units for these figures:
M, the mass of the case, is in lb_m. *This is the case weight only;* the vibrating mass of the hammer will vary with the type of clamp that is attached to same. There are two basic types of clamps that are used most of the time in vibratory driving: sheeting clamps (which also clamp H-Beams) and caisson beams. For the 1150, 2300, 2300L, 1400, and 2800, the sheeting clamp weighs 1350 lbs.; for the 4600, the sheeting clamp weights 4000 lbs. The "Seven (7) Foot" beam we manufacture, which refers of the maximum diameter of pipe or caisson it drives, weighs 3700 lbs, including clamps. Theoretically speaking, this could be mounted on any of the hammers above; in reality, one would normally not use it on small hammers, such as the 1150 and 1400.
m, the mass of the eccentrics, is in lb_m.
e, the eccentric arm length, is in inches.
w. the weight of the bias masses, is in lb_m. Keep in mind that all of these hammers, except for the 1400, can have additional bias weight mounted onto the suspension.
k, the spring constant of the suspension, is in lbs/in. This is a theoretical value from very old data; for Vulcan and other manufacturers (such as ICE and Casteel) who use the dock fender type of springs, things are presently in a state of flux as far as the stiffness of the springs is concerned. We plan to run some tests on these and we can share the results with you when they are complete.
f, the frequency of rotations, is in RPM.
N, the theoretical hydraulic power at the rated frequency and maximum hydraulic pressure.

Now we can address some of the other matters and issues that are relevant to the present project.

As far as power curves go, as I.F. Goncharevich and K.V. Frolov point out in their book *Theory of Vibratory Technology*:

> The power which is required to operate the vibratory machine in the given regime and the power which can be transmitted by a vibrator of the specific type are determined by a whole complex of factors: vibrator parameters, machine characteristics, and the acting loads in the machine. It is not possible to impart additional power to the vibratory machine by simply increasing motor output. Each vibratory machine consumes strictly determined power, whose value is

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dependent on a whole set of factors acting in the vibratory machine-vibration exciter-load system.

The power that a vibratory hammer puts out will thus vary with the conditions of the machine; however, it can only put out at a given frequency what the hydraulic motor can offer. Most manufacturers (and we are no exception) who use hydraulic motors use fixed displacement ones; their power output is a function of a) the volumetric displacement of the motor, and b)the pressure of the hydraulic oil. Since (a) is fixed, and all hydraulic systems are designed with a maximum pressure, it follows that the theoretical maximum power possible with any vibratory system is linearly proportional with the frequency, ranging from zero at rest to the maximum power described in the table above.

All of this, however, is complicated by the engine-pump system. If the power curve of the engine, adjusted in speed to correspond with that of the exciter motor, fall below that of the motor, then the motor will probably slow down until there is sufficient power to keep things turning. Given the large number of combinations of power packs and hammers in the field, this could make matters very complex. We generally attempt to keep the engine power curve above that of the motor to avoid problems such as this.

One overlooked aspect of the rotation of eccentrics is the effects of variation of the rotational speed during each rotation of the eccentrics. Theoretically speaking, the only way for an eccentric to have a perfectly constant rotational speed with the continuously varying torque of the eccentrics is for the eccentrics and any connecting parts to have infinite rotational inertia, at which point no motor could turn them from rest! Studies have shown that, in pure vibratory pile driving equipment, rotational speed will not vary more than 5%, while with impact-vibrational equipment this can vary up to 40%. I believe, however, that the rotational inertia of the eccentrics and other elements in the drive train is far more important than the eccentric moment length, and deserves more consideration than it has been given.

Then there is the matter of friction. Our studies into this field show that a) there is a great deal of information available on the energy losses of the various components of the system, such as the bearings, gears, motors, plumbing, etc., b) much of this information is difficult to apply to our actual situation, and c) the correlation between the theoretical correlations and the reality of the systems is very uncertain. While we can supply you with some of what we have for the various components of the system, this is a field that cries for some meaningful research. We can discuss this further when you have had a chance to digest what is before you.

Finally, we hope that the result of your research is more user-friendly than WEAP. For our part, we have spent much of the last three years developing and commercializing user friendly programs for both impact and vibratory driving. We presently have code that can analyze both ECH impact hammers and vibratory hammers using a wave equation type of analysis; however, our efforts have been stalled by a) lack of time, b) lack of resources, c) prejudice against our efforts because we additionally manufacture the equipment, and d) lack of a reasonable model of the soil under vibratory action (we feel we have impact action as well as anyone). At this stage we would be interested in working out a plan to share the results of our work on a mutually beneficial basis. Please let us know if you and interested in such discussions. We can furnish you with sample programs for your review.

I hope that all of this information proves helpful to you. If you have any further questions, please let me know.

Yours very truly,

VULCAN IRON WORKS INC.

Don C. Warrington
President

cc: Mrs. V.S. Warrington
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