IMPROVEMENTS IN REMOTE EQUIPMENT
TORQUING AND FASTENING

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TORQUING AND FASTENING
ABSTRACT

Remote torquing and fastening is a requirement of generic interest for application in an environment not readily accessible to man. The developments over the last 30 years in torque-controlled equipment above 200 Nm (150 ft/1b) have not been emphasized. The development of specialized subassemblies to torque and fasten equipment in a remotely controlled environment is an integral part of the Advanced Fuel Recycle Program at Oak Ridge National Laboratory. Commercially available subassemblies have been adapted into a system that would provide remote torquing and fastening in the range of 200 to 750 Nm (150 to 550 ft/1b).
INTRODUCTION

The objective of the Advanced Fuel Recycle Program (AFRP) is to develop the technology for reprocessing nuclear fuels. This technology will ultimately be demonstrated in the Hot Experimental Facility (HEF), which is now in the conceptual stage of development.

One problem associated with reprocessing technology is the development of in-cell torque tools to remotely disassemble equipment and piping joints. Historically, in-cell torque application has been accomplished with pneumatic and electrical impact wrenches. The Hanford and Savannah River production plants have used electric crane-held wrenches with no torque control (Fig. 1). In other facilities, torque wrenches were handled by master-slave (MS) manipulators. In the past, precise torque control has not been a strict requirement; therefore, it has only been used in a few cases, particularly those in a range of less than 200 Nm (150 ft/lb).

Hot Experimental Facility objectives necessitate the development of tooling that will be compatible with an in-cell environment and will provide adequate torque control for such items as connectors, component subassemblies, modular-drive attachments, or any other rotary-motion subassemblies.

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Fig. 1. Reprocessing impact wrench torque system.
The tool development program consists of selecting developed and proven components, providing a proper match between electrical and mechanical components, using modern controls, and testing the assembly under simulated conditions.

The subject of this paper encompasses the detail and system-level stages of the in-cell torque-tool development program. Program testing is being conducted initially in cell E of the Thorium-Uranium Recycle Facility (TURF) (Fig. 2), and final demonstration of equipment operational procedures will be carried out at the Remote Operational and Maintenance Demonstration facility (ROMD). Both the TURF and ROMD facilities are located at the Oak Ridge National Laboratory.

The state of the art for remote torquing and fastening has not progressed significantly beyond the 1940 era of the Hanford-Purex impact wrench. This system is based on an Ingersoll-Rand model No. 534 impact hammer subassembly coupled to a 3/4 hp, 440 V three-phases ac motor. Torque monitoring is accomplished by listening to the pitch of the hammer blows through an in-cell microphone system. Upon compression, a Teflon-retained ring-type seal allows metal-to-metal contact of connector parts. This contact produces a high-pitched sound, indicating that the seal is complete. However, the actual service life of this system is not well-documented.

DEVELOPMENT OBJECTIVES

The use of fasteners and torque tools can be separated into at least two categories: (1) in-situ cell usage and (2) usage in a
Fig. 2. TURF cell-E EMM & power manipulator.
decontamination-cell situation. The latter case requires greater than single-arm manipulation (i.e., special dedicated tooling or an MS manipulator). The torquing and fastening tool system discussed in this paper addresses the in-situ or in-cell situation only.

In order to focus on the generic value of this development activity, its application limitation will be discussed. It became necessary to adopt a design philosophy for the HEF equipment; however, that very philosophy imposed limitations on the application of the torque-tool system.

One of the limitations is that all in-cell equipment must be modular in design. Therefore, the two categories of equipment for an HEF facility must break down into modules. The first category includes equipment such as complex machines having multiple moving mechanical components, for example, an electric motor, bearings, ball-screw assemblies, and gear boxes. Such equipment will be used in the HEF dissolver system (Fig. 3). This equipment must be designed so it can be disassembled into modules. The second category is chemical process equipment fixed in racks that are 3.7 m x 3.7 m x 15 m (12 ft x 12 ft x 50 ft) in size. The equipment must be grouped in these racks so that it can be removed as modular assemblies.

In order to be maintained by an impact tool, modularized equipment must have the following characteristics.

- Fastening shall be used to resist the load generated during operation (equipment motion) or to provide seating force to seal an assembly (not in motion).
Fig. 3. Modularized process equipment.
• Torque range shall be 200 to 750 Nm (150 to 550 ft/1b).
• Nut or screw "running" shall be a maximum of 6 Nm (5 ft/1b).
• The fastener shall be so oriented in one of the orthogonal planes that is is easily accessible by an impact wrench.
• All sockets shall be uniform in size for the torque range and provision made for a wrench reaction point.
• Screw-type devices shall be used to engage out-of-orthogonal, plane-oriented equipment.
• Modules shall be self-guiding (minimal manipulation) by use of guide brackets or alignment pins.

DESIGN AND TESTING — PLANNED METHOD OF ACCOMPLISHMENT

The torque system design and testing activity has been phased and is discussed in the following order:

• commercial equipment characterization,
• design of the drive system,
• position indication subsystem development,
• impact hammer selection,
• design of integrated system, gearbox, and dc motor,
• characterization of new system, and
• manipulation demonstration.

Each one of the above items represents a step leading to the ultimate development of the torque system.

Commercial equipment characterization

For this type of operation, there are basically three types of torquing devices from which to choose: nut/bolt driver, nut runners, and impact wrenches.
Nut/bolt drivers and nut runners have high-torque motors which deliver torque directly to the fastener until power is shut off or a clutch disengages. Impact wrenches derive torque from a motor whose torque is magnified by a special type clutch. This clutch converts inertia into a series of rotary blows which incrementally build up torque in the fastener. The effect is similar to that obtained when a hand wrench is placed on a nut and the opposite end is struck with a hammer.

There are two major differences between these three tools. The first is that in the nut/bolt drivers and nut runners, 100% torque must be reacted by the operator or by a special fixture. The impact wrench requires only a small fraction (1 to 5%) of torque to react. The second difference is that the impact tool has a much greater torque capability.

In order to achieve optimum design of a remote torquing and fastening system, we consider it important to understand the fundamental operation of impact tools. During a typical impact wrench run, the tool operates in several different modes. The initial mode is "rundown," an action similar to that of a nut runner. The next mode is low torque (as seen in a nut/bolt driver), which is limited to the motor torque output. In the final mode or high torque stage, torque is applied incrementally (1- to 5-sec duration).

The transition from nut runner and nut/bolt driver to impact tool must be understood in order to select a hammer mechanism and motor drive. At impact tool startup, there is low resistance to turning, and the motor and hammering mechanism
(hammer and anvil) will rotate in unison. The hammer and anvil will remain engaged, and the tool is operating in direct drive, that is, as a nut/bolt driver or nut runner. The start-up stage of operation as nut/bolt driver or nut runner is relatively short. For example, a 2.5-cm "rundown" and torque application, equivalent to that of a nut/bolt driver, takes less than one second at 2000 to 7000 rpm. As the output torque increases (due to the resistance of the fastener), the hammering mechanism allows the hammer to release, and the hammering cycles begin.

The amount of torque the tool delivers with each impact pulse (hammer impacting an anvil) is a function of the torque resistance offered by the fastener. The length of time during which torque is delivered in the hammer cycle is determined primarily by how much kinetic energy has been stored in the hammer mechanism and then transferred to the fastener.

A study of the relationship of torque and rotation at the socket and drive motor of the tool is part of the development program. Detailed observations were made for both the pneumatic and electrical impact tools. A torsional load transducer and a continuous recorder were used (Figs. 4 and 5) for these observations. It was noted that the applied torque varies during the torque cycle.

For each hammer blow, the same amount of kinetic energy is being converted into work, but a varying amount is converted into torque. The fastener rotation during a hammering cycle determines the amount of torque input. Usually, a torque level greater than the previous pulse is reached because less energy is used to rotate the
Fig. 4. Torque transducer assembly.
Fig. 5. Development operator's console.
nut/bolt as it is being preloaded. As the final torque is reached, the pulse takes on a more narrow shape. The change in the shape of each hammering step reflects the rotation of nut and bolt; the area under the curve is constant. The step changes become sharper and sharper as the slack in the system is removed (see Appendix for further analysis).

Design of drive system

A survey of vendors' impact tool equipment in the range of 200 to 750 Nm revealed that all of the equipment is pneumatic. The type of air motor used for all of these impact tools is the vane type, which operates at a constant air pressure of $620 \times 10^3$ Pa (90 psig). The air motor has a zero horsepower at zero speed, increasing with increasing speed until the horsepower peaks at approximately 50% of free speed. The horsepower then decreases to zero again at free speed. In general, the air motors suffer a derating with decreasing operating pressure, due to the lack of flow or drop in pressure caused by line losses. This loss is approximately a 14% decrease in horsepower per $70 \times 10^3$ Pa (10 psig) pressure drop. Most applications in the HEF would require 61 to 91.5 m (200 to 300 ft) of hose; therefore, the line loss would be significant. The size of handling equipment for the hose would also become prohibitive. For these reasons we considered it important to use electrically operated tools.

Selection of a dc motor and control is the next step in drive system design. There are basically three different types of dc motors adaptable to velocity servomechanisms. (Servomechanism implies that the impact tool will have to be controlled by a
closed-loop system in order to better approach the characteristics of an air motor.) The three types are: (1) shunt motor with a three-phase silicon-controlled rectifier (SCR) controller [best torque range 1.4 to 10,800 Nm (1 to 8000 lb/ft)]; (2) permanent magnet motor, with transistor-switching or SCR phase controller [best torque range 0.01 to 1360 Nm (0.005 to 1000 lb/ft)]; and (3) moving coil motor, with transistor or SCR switching controller [best torque 0.01 to 13.5 Nm (0.05 to 10 lb/ft)]. The best type of motor for this application was judged to be the permanent magnet motor because of its size-to-power ratio, excellent response, and torque-to-speed characteristics similar to those of the air motor.

For development purposes, a dc permanent magnet servomotor with a tachometer was selected. This motor has a 2.45-hp rating and is continuously variable from 0 to 4000 rpm. Appropriate gearing was provided for use with impact tools requiring 6000 to 7000 rpm.

Position indication subsystem development

Present design studies for the HEF indicate that direct viewing through shielded windows will not always be possible. In-cell viewing will be by TV located on the bridges, the wall, and on an electromechanical manipulator support system. Experience with TV-viewing systems has shown that, even with up-to-date equipment, TV pictures lack fidelity; therefore, multiple images must be used to provide the operator with some depth perception. With the larger barn-type cell used in fuel reprocessing, it is
almost impossible to give the operator a field of view with good depth or three dimensions. In order to improve this situation, additional visual guidance will be provided to the operator (Fig. 5).

The problem of locating the wrench in a plane, which is normal to the bolt axis, is being addressed. In order to accomplish this positioning, two types of photosensors are being investigated. The first is a quadrant detector—a sensor containing four closely grouped photodetectors which are used in conjunction with a Fresnel target pattern. The outputs of the three sensors, such as quadrant detectors, could then be displayed simultaneously on an oscilloscope.

The second type of sensor being investigated is a 32 x 32 array of self-scanned photodiodes contained in a single, integrated circuit. This sensor, in conjunction with a limited amount of circuitry, can produce a low-resolution video image, which can be displayed on either an oscilloscope or a television monitor. This low-resolution image will provide visual guidance to the targets. The advantages of this type of sensor over a television camera are its small size, its simplicity, and its potential use for automated component identification.

Current efforts are directed toward a laboratory demonstration of these two types of sensor approaches. Included in the development effort is the selection of appropriate lenses, suitable packaging, and the distribution of electronic hardware associated with the sensors.
Impact hammer selection

The objectives that apply to impact hammer selection are as follows:

- Allow impact hammer to be used with an electromechanical manipulator.
- Provide torque control in the range 200 to 750 Nm.
- Optimize the hammer mechanism operating reliability.
- Minimize the torquing tool envelope.

In order to meet the above goals, it was decided that a commercially available impact hammer mechanism (clutch) would be used. A survey of available clutch-type tools revealed that there are basically two categories of hammer mechanism design: cam and centrifugal. Cam-type mechanisms, which are driven by a motor through a gear reducer, store energy (see Appendix), which is accomplished by compressing a spring and releasing it suddenly. These mechanisms operate in the 2000- to 3000-rpm range and deliver about 20 blows per second.

The other type of mechanism is the centrifugal hammer type, whose operation depends on storage of energy through centrifugal speed. The hammer is rotated at a motor speed of 3000 to 7000 rpm, depending on the manufacturer's design. The energy is stored in one revolution, and the impact rate is about 10 to 25 blows per second.

The basis for hammer selection depends on the ability of the mechanism to be controlled and driven by an electric drive.
As stated earlier, the drive motor is selected to match the pneumatic motor characteristics. This "matching" criterion is especially important in the centrifugal device since its successful operation depends on high initial torque.

The first step in hammer selection is to characterize the behavior of the pneumatically driven tools by using a test fixture that has the capability of continuously recording the impact wrench output. The pneumatic wrench is controlled through a panel that is capable of regulating pressure and time. Typical output tracers for a torque cycle are shown in Fig. 6.

The next step is to take the same hammer mechanism, match it to the dc servomotor driver, and observe its operating characteristics. The resulting data was plotted in a number of different ways in order to compare both the variables and the characteristics of the two systems.

It was found that the best method of control for hammer mechanisms is the blow per second vs torque technique. Analysis shows that these two variables are least affected by varying parameters, for example, changes in friction, time control, pressure, or current (Fig. 7.).

**Design of integrated system**

The design of the impact torque tool must meet all of the following objectives: allow impact wrench to be used with an electromechanical manipulator, provide torque control, triaxial position indication, decontamination capability, and optimization of hammer mechanism. Setting the goals for the above tools is achieved in two stages.
Fig. 6. Torque vs time.
Fig. 7. Impact hammer blows vs axial load.
The first stage is the design of a prototype tool that could be used as a test bed to demonstrate the numerous design features. Figure 8 shows the assembly of the prototype impact tool which is designed to test all the features needed to meet program objectives.

Previously, use of the impact wrench was limited to the horizontal and vertical position. If the impact wrench could be handled by an electromechanical manipulator (EMM), then additional in-cell tasks could be performed. Therefore, both designs have been made so that they can accommodate crane hook or EMM usage.

Earlier models of the impact wrench did not have the torque control necessary to make this type of tool more effective in the performance of tasks such as nut running, preloading of fasteners, engagement of connectors, and torque reversal for nut/bolt unfastening. The prototype torquing and fastening tool, however, has two modes of torque control.

One of the torque control systems explored is mechanical in nature and uses a torsion bar within the hammer mechanism that must be preset prior to in-cell use. A signal is transmitted to the power center when a preset torque limit is reached, thus automatically shutting off the impact wrench. This feature is available from the tool vendor but must be changed in-cell for each torque value. Torque reversal is not feasible with this system.
Another control system considered is a design with the capability of controlling torque over 200 to 750 Nm by regulating operating time, input pressure, current or pulse counting.

This latter method of torque control was chosen for the prototype tool. Presently, position indication is only at the subassembly level of development. The wrenches have provisions for mounting; however, because of the nature of this equipment, it was decided that the modularized sensor be mounted on the wrench so it can be easily replaced.

The second step of this design effort is to incorporate those features of both the pneumatic and electrical prototype tools that proved effective during testing. Numerous mechanical design changes were made to simplify the design (Fig. 9). The most obvious change is the one resulting from the decontamination requirement for a smooth outer shell. Decon requirements also meant enclosing the tool motor completely, leading to potential heat dissipation problems. Added design features should prevent this problem. The tool system has been provided with panel light indication at 71°C and automatic shutoff at 88°C for added component protection.

The optimization of the hammer mechanism is continuing. The lower revolutions-per-minute device is preferred (ball-cam hammer), but reliability will be the prime factor in the final decision. Both hammer mechanisms can be accommodated by using an adapter.
Fig. 9. Prototype Impact Log Book
COMMENTS AND STATUS

The impact tool system tested and developed to date in this program has been used in out-of-cell situations. The components for the torque system have been tested and integrated into an assembly. This assembly constitutes the prototype impact tool. There are two phases to be completed in the development plan: system characterization and manipulation demonstration. The status of the features of the prototype impact tool are as follows.

- The wrench can be used by either a manipulator or a crane hook, with selection of a specific handle.
- Torque control is achieved by presetting torque remotely and monitoring hammer blows electronically.
- Triaxial position indication at this time is still under development but shows promise of being an aid to the operator.
- The hammer mechanism has been characterized. The optimization of the mechanism is strictly a question of reliability, and, at this point, the lower rpm devices have been selected.
- The design envelope has been optimized by selecting the minimum size hammer mechanism.
- Decontamination capability is provided by design.

Design objectives have been evaluated and integrated based on performance. The characterization of this system is ongoing, and the final system will reflect a balance of these
objectives. The manipulation demonstration in the TURF and ROMD facilities will serve to demonstrate the use and reliability of this remote torquing system and its applicability in other facilities such as function reactors, light water reactors, or LMFBRs.

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REFERENCE

APPENDIX

ANALYSIS OF THE BEHAVIOR OF THE IMPACT TOOL HAMMER MECHANISM

We can summarize the behavior of an impact tool hammer assembly as follows. The energy \( E \) in the system is a balance of potential energy \( PE = \mu(x) \) and kinetic energy \( KE = V(\dot{x}) \)

or

\[
E = \mu(x) + V(\dot{x}),
\]

where

\[
V(\dot{x}) = 1/2 \, m \, (\dot{x})^2
\]

and

\[
\mu(x) = \int_{x_0}^{x} f(x) \, dx.
\]

The KE can further be categorized as:

\( V_c(\dot{x}) = \) input from dc servomotor, and also \( V_d(\dot{x}) = \) output to bolt.

The above equation can be used to represent total energy \( (TE) \) in the system. The kinetic energy, \( V_c(\dot{x}) \), is input into the impact hammer by a pneumatic or electric motor analogous to charging a capacitor. After the hammer is charged or stored \( \mu(x) \), the input energy is released again in the form of kinetic energy \( V_d(x) \). For the hammer mechanism, the potential energy can be expressed by \( f(x) = Fd \). If \( d \) is kept to a minimum (the travel distance), coulomb friction neglected, and impacts assumed to be perfectly elastic, the system behavior may then be expressed by

\[
V_c(\dot{x}) = \mu(x) = V_d(\dot{x}).
\]
We can readily see that $y(x)$ is dependent on the characteristic of the hammer mechanism, but the charge time is proportional to the rotation of travel of the hammer ($d$) or $(X-X_0)$. Therefore, the impact force will not increase as the rotational displacement is varied.

Since the area under the curve $E$ will remain the same for a given motor drive setting capable of providing a given KE input, it must further be seen that the torque characteristics can be further controlled by varying $\dot{x}$, the input variable velocity of the motor. We are at the mercy of $y(x)$, which is characteristic of the specific hammer mechanism. Figure A1 gives examples of changing hammer blows. Figure A1(a) shows the effects of a weak hammer blow when the fasteners have the greatest rotation. Travel distance "a" is the time spent overcoming friction, travel distance "d" is the time spent torquing, and travel distance "c" is residual effects of friction. Figure A1(b) demonstrates torque application when the fasteners have already been loaded, and Fig. A1(c) shows the idealized torque application with no fastener slippage.

The basis for this impact tool design was therefore:

1. Characterization of $y(x) = \int_{x_0}^{x} f(x) \, dx$ for each hammer mechanism,
2. Selection of drive system to maximize the range of $\dot{x}$ in the range of the hammer mechanism, and
3. Characterize torque from 200 to 750 Nm (150 to 550 ft/lb) as a function of $\dot{x}$ or the establishment of transfer function ($F$) for each hammer mechanism.

It must be noted that the spring rate of structure or component that is being torqued affects the ultimate results. This is
Fig. 1. A: Typical torque pulses.
readily understandable since a "soft" structure will introduce an additional "path" for transfer of $V_c(\lambda)$. Therefore, the design of subassemblies of modules must have a spring stiffness at least one order of magnitude greater than the applied torque.
FIGURE CAPTIONS

Fig. 1. Reprocessing impact wrench torque system.

Fig. 2. TURF cell-E EMM & power manipulator.

Fig. 3. Modularized process equipment.

Fig. 4. Torque transducer assembly.

Fig. 5. Development operator's console.

Fig. 6. Torque vs time.

Fig. 7. Impact hammer blows vs axial load.

Fig. 8. Prototype impact tool

Fig. 9. Prototype impact tool.

Fig. A1. Typical torque pulses.